Thermo-fluid performance evaluation of a split-winglet supported elliptical tube type Fin-and-tube heat transfer surface

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ABSTRACT

Vortex generator supported Fin-tube heat transfer surface having inline elliptical tubes is investigated by performing numerical simulations governed by finite volume method. The present investigation is aimed at minimizing the air-side thermal resistance by utilizing the rectangular winglet pairs. The thermo-fluid performance of the baseline model is investigated for the circular and elliptical tube configurations. It is found that the former resulted in higher heat transfer whereas the latter reduces the pressure drop. The effect of split winglet pair’s span-wise and stream-wise separation, and attack angle on heat transfer and pressure drop performance is examined in detail and the results were presented in terms of $Nu$, $f$ and $\eta$. Optimum stream wise and span wise locations for the front and rear winglet pair were identified based on highest enhancement factor. It is also found that the optimum attack angle of the front and rear winglet pair is different for maximum enhancement factor.

Keywords: Fin-and-tube heat exchanger; Split winglet pair; Rectangular winglets; Elliptical tubes; Stream wise location; span wise location; angle of attack

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1. Introduction

Fin-and-tube heat exchangers (FTHX) are the gas-liquid type of heat exchanger and are used in air-conditioning and refrigeration systems, in automobile and petrochemical industries, and for cooling of electronic chips. Usually, the liquid side thermal resistance and the fin side conductive resistance in FTHX are very low due to the high heat transfer coefficient of the liquid flowing inside the tube and the high thermal conductivity of the fin material respectively. However, the heat transfer coefficient of air is very low by virtue of its thermo-physical properties and therefore the air-side thermal resistance has greater dominance in the overall thermal resistance. Moreover, in these heat exchangers, the thermal mixing is poor in the wake region behind the tubes. Thus, to improve the heat exchanger performance against these thermal and geometrical constraints several heat transfer enhancement techniques are used.

Passive heat transfer enhancement methods are more common in practice since the only power required during their operation is for pumping of the fluid across the heat exchanger to compensate against the pressure drop. The winglet type vortex generator (VG) is a passive heat transfer enhancement tool that has the potential to introduce secondary flow vortices which intensifies the hot and cold fluid mixing near the wake region. They also delay the flow separation and modify the thermal boundary layer. Both, the heat transfer and the pressure drop performance of the heat exchanger depend on the winglet configuration and location.

To enhance the heat transfer contribution of the air-side, many researchers have opted for modifications in the fin geometry and installation of VGs over the fin surface. However, the changes in fin configuration have resulted in higher pressure drop and slight improvement in heat transfer [1]. The installation of winglets as VGs over the fin surface intensifies the heat transfer coefficient and thereby results in significant heat transfer enhancement. Further,
many researchers in the open literature have opted for two winglet orientations. For instance, Biswas et al. [2], Torii et al. [3], and Jain et al. [4] used the “common-flow-up” winglet orientations which effectively compresses the wake region and accelerates the flow near the wake region. Vasudevan et al. [5], Sohankar and Davidson [6], and Kwak et al. [7] used the “common-flow-down” winglet orientation, wherein, the high momentum fluid is introduced in the tube wake region to improve thermal mixing [8].

A few authors have also investigated the inline and staggered orientation of winglets. For instance, Joardar and Jacobi [9, 10] studied the influence of inline (alternate tubes) and staggered (consecutive tubes) placement of delta winglet VGs. They found that for the same heat transfer enhancement, the staggered winglet configuration incurs relatively lower pressure drop. Similarly, He et al. [11], based on their numerical study reported that, compared to inline arrangement, the staggered arrangement of VGs resulted in almost same heat transfer effect but with reduced pressured drop.

Many authors have also considered optimization techniques in the open literature to solve multi-objective problems. For instance, Nascimento et al. [12] used computational fluid dynamics (CFD) techniques and genetic algorithms for a plate-fin heat exchanger with offset strip-fins to improve the design capabilities. The authors reported a reduction of pressure drop of 55.4% and 72.3%, respectively at the hot and cold sides. Strategic selection of the population size can help a multi-objective genetic algorithm to balance the convergence time and the accuracy of the results. Galuzio et al. [13] proposed a python based new software-‘Bayesian optimization software’ that was able to determine better quality Pareto front approximations with minimum evaluations of the objectives. The owl optimization algorithm [14] applied to the shell and tube heat exchanger resulted in significant performance enhancement for both single and multi-objective optimization. Lemouedda et al. [15] used optimization techniques based on Pareto optimal strategy and proposed optimum sets of
attack angle for Reynolds number ranging 200-1200. They compared the inline and staggered positioning of tube rows and reported that the latter gave better heat transfer effect over the considered range of $Re$. Similarly, Salviano et al. [16] performed a SIMPLEX method based optimization study for FTHX and confirmed the heat transfer enhancement advantage of staggered tube rows over inline tubes. For a given design space, the SIMPLEX method provides the simplest possible polytope and is used for the optimization studies for non-linear problems.

Venegas et al. [17] presented a critical review of the desiccant coated heat exchangers and discussed possible developments for enhancement in energy and dehumidification performance of the system. Wajs et al. [18] performed simulations for a mini-channel heat exchanger to realize flow rate and pressure field distribution along with the temperature distribution in the solid and fluid zones. Tian et al. [19] compared the performance of delta and rectangular type VGs and found the former with better overall performance, whereas, the later delivered better heat transfer effect.

Saha et al. [20] also performed a synergy based study for comparison of rectangular and delta VGs arranged in “common-flow-up” and “common-flow-down” orientation. They reported higher heat transfer enhancement with rectangular winglet pairs. Sinha et al. [21] examined various possible winglet arrangements with “common-flow-up” and “common-flow-down” orientations. They reported that the “common-flow-up” orientation in series resulted in greater heat transfer performance. Gentry and Jacobi [22, 23] performed experiments using the naphthalene sublimation technique to study the influence of delta winglet VG in a flat-plate flow.

In recent years many investigators have focused their research on determination of the optimum winglet locations in FTHX. For instance, Pesteei et al. [24] examined experimentally five different locations of delta winglet pair around a single tube and
presented optimum locations with respect to the centre of the tube. Arora et al. [25] performed a similar type of investigation with three rows of inline tubes and twenty possible locations of VGs. Sarangi and Mishra [26] and Sarangi et al. [27] carried out numerical simulations with rectangular VGs in a FTHX and determined the optimum winglet number, winglet attack angle, and winglet locations (stream wise and span wise). They also examined the influence of wavy nature of the rectangular winglets in FTHX [28]. Naik and Tiwari [29] considered “common-flow-down” oriented VGs and examined the influence of attack angle, $Re$, and stream-wise and span-wise locations of the VGs.

The review of the existing literatures shows that the VGs have the potential to enhance the heat transfer performance effectively using mechanisms such as delaying flow separation, modification of the boundary layer, and vortex generation. The rectangular winglet pairs are more proficient in heat transfer augmentation than delta winglet pairs [19, 20], however, they also suffer relatively higher pressure drop. The present study attempts to minimize the pressure drop associated with the use of rectangular winglet pairs. To achieve this, the VGs are positioned with a heretofore-unused combination- ‘split winglet pairs over the elliptical tubes’ for enhancement in heat transfer with lower pressure drop penalty. The full-length winglet used in the previous literature [27] is split into equal parts in the present study and their optimum locations have been investigated to provide greater control of thermal mixing behind the tubes.

In comparison to the circular tubes, the elliptical tube shape incurs relatively lesser pressure drop due to its aerodynamic shape [30]. Thus, the present analysis makes use of this advantage of the elliptical tubes and its overall performance in comparison to the conventional circular tube FTHX has been examined under same surface area condition. It is expected that the split winglet pairs together with the elliptical tubes can significantly enhance the thermo-fluid performance of the FTHX.
2. Mathematical Formulation

2.1. Physical and computational model

The split winglet supported FTHX has been shown in Figure 1 having five rows of tubes along the stream and four rows of tubes transverse to the stream. The basic geometry (baseline model) of the considered FTHX has been referred from Joardar and Jacobi [9, 10]. The split winglet pairs are symmetrically positioned around the elliptical tubes as shown in Figure 2. The total winglet length \((L_1+L_2)\) is 10.67 mm which is the same as the circular tube diameter. The fin spacing is 3.63 mm and the channel length along the stream is 127 mm. The pitch for the tube is 25.4 mm and the VGs with “common-flow-up” configuration are placed at suitable attack angles. The surface area of the elliptical tube \((a = 4 \text{ mm}, b = 6.52 \text{ mm})\) is kept same as that of the circular tube. Due to the symmetric orientation of the tubes and winglets the region shown within the indicated region in Figure 1 is chosen as the computational domain.

2.2. Governing Equations

For simplifying the modelling of the heat transfer process for the FTHX, some assumptions have been made. The flow has been considered as steady, laminar and incompressible. The working fluid is air and its density varies with temperature governed by the ideal gas equation. The following Navier-Stokes equations are applicable for the considered flow model:

Continuity equation: \[ \frac{\partial}{\partial x_i} (\rho u_i) = 0 \]  \hspace{1cm} (1)

Momentum equation: \[ \frac{\partial}{\partial x_i} (\rho u_i u_k) = \frac{\partial}{\partial x_i} \left( \mu \frac{\partial u_k}{\partial x_i} \right) - \frac{\partial p}{\partial x_k} \]  \hspace{1cm} (2)

For the operating temperature range, the dynamic viscosity \(\mu\) is assumed constant.
Energy equation: 
\[ \frac{\partial}{\partial x_i} (\rho u_i T) = \frac{\partial}{\partial x_i} \left( \frac{k_a}{C_p} \frac{\partial T}{\partial x_i} \right) \] (3)

Conduction equation is solved to obtain the temperature distribution on winglets and fin surfaces

Conduction equation: 
\[ \frac{\partial^2 T}{\partial x_i^2} = 0 \] (4)

2.3. Boundary Conditions

The boundary condition applied over the surfaces of the computational domain has been shown in Figure 3(a) & (b). The inlet boundary is imposed with velocity inlet condition \((U_{in} = 2–3.3 \text{ m/s corresponding to } Re_H = 500-819)\) with a uniform temperature of 310.6 K. Symmetry condition is prescribed to the upper, lower, and side boundaries. Pressure outlet condition is applied to the outlet surface. The cold tube walls are applied with no-slip and constant temperature condition \((T_w = 291.77 \text{ K})\) with the assumption that the tube wall material is Aluminium bearing high thermal conductivity and the liquid inside the tubes has very high convective heat transfer coefficient.

2.4. Computation of Performance Parameters

The performance parameters are defined as follows [30]:

Overall heat transfer, 
\[ Q = \dot{m}c_p(\bar{T}_o - \bar{T}_{in}) \] (5)

LMTD, \( \Delta T = \frac{(T_w - \bar{T}_{in}) - (T_w - \bar{T}_o)}{\ln[(T_w - \bar{T}_{in})/(T_w - \bar{T}_o)]} \) (6)

Reynolds number, \( Re_H = \frac{\rho u_{in} H}{\mu} \) (7)

Nusselt number, \( Nu = \frac{hH}{k_a} \) (8)
Heat transfer coefficient, 

\[ h = \frac{Q}{A_r \Delta T} \]  

(9)

Friction factor as defined in Nascimento et al. [31]:

\[ f = \frac{\Delta p}{\rho u_{in}^2 L} \frac{2}{H} \]  

(10)

Enhancement factor [27, 32]:

\[ \eta = \left( \frac{Nu}{Nu_0} \right) \left( \frac{f}{f_0} \right)^{-1/3} \]  

(11)

2.5. Numerical Method

Finite volume method based three-dimensional numerical simulations were performed with Fluent 17- a multi-grid solver for thermo-fluid analysis of the considered FTHX. SIMPLE (semi-implicit method for pressure linked equation) algorithm was employed for coupling of velocity and pressure. The entire flow domain consisting of solid (Fin, tubes, and winglets) and fluid zones were meshed with structured hexahedral elements. The generated grid system is shown in Figure 4.

The convective terms in the governing equations are discretized by the second-order upwind scheme which provided an improvement of about 0.5% over the first-order upwind scheme. The tolerance for convergence was set to 10^{-6} for energy, and that for continuity and momentum were set to 10^{-3}.

3. RESULTS AND DISCUSSION

3.1. Grid test and validation of numerical results

A grid assessment test prior to performing numerical solutions ensures best mesh size for computational accuracy. The grid test shown in Table 1 has been conducted for both circular and elliptical tube type FTHX.
The grid test performed, has covered a wide range of grid sizes, starting from a coarse grid number of about 700,000 to a very fine grid size of about 1,900,000. The Nusselt number obtained against each grid number for the circular and elliptical tube configuration increases continuously as the mesh is refined to a finer size. However, the change in Nusselt number beyond the grid number of about 1,800,000 is almost negligible. This clearly indicates the attainment of grid independence of the solution. Therefore, the cell size corresponding to a grid number of about 1,800,000 has been considered in the present numerical study for generation of grids. Similar grid tests were also performed for VG supported FTHX models.

The results obtained from CFD for the baseline model have been validated with experimental results [10, 33] to ensure reliability and accuracy of the numerical model. The details of the experimental models [10, 33] has been shown in Table 2. It can be visualized from Figure 5(a) and 5(b) for $h$ and $\Delta p$ respectively, the present simulated results agree well with the experimental results of Joardar and Jacobi [10] with maximum deviation being less than 10%. The uncertainty reported in the experimental results varied ±4% and ±15% for $h$ and $\Delta p$ respectively. Similarly, Figure 5(c) shows a comparison for the $Nu$ obtained using present CFD and the results obtained through experiments of Kays and London [33]. The maximum deviation of the CFD results from the experiment lies within 7% with a reported uncertainty of ±5%. To summarize, the numerical model shows good agreement with the experimental results [10, 33], and thus is reliable for predicting the performance of the FTHX.

3.2. Heat transfer and pressure drop assessment of plain FTHX with circular and elliptical tube geometry

In this section, the performance of smooth FTHX has been investigated by considering circular and elliptical tube geometries separately. For ideal comparison of the performance of
the two tube geometries, their surface area has been taken same. Since the two tube geometries result in different hydraulic diameters, the fin spacing has been taken as the standard characteristic dimension for the calculation of $Re_H$.

In Figure 6, the variation of $Nu$ for circular and elliptical tube configurations against $Re_H$ ranging 500-800 is shown. The plot shows the heat transfer performance marginally high for the circular tube at low $Re_H$ and increases substantially when the $Re_H$ increases beyond 700. Figure 7 shows that the circular tube has a larger frontal area than the elliptical tube and with rise in $Re_H$ heat transfer rate enhances due to higher local convective heat transfer coefficient. However, compared to the elliptical tube, larger frontal area in case of circular tube results in substantial increase in form drag and the pressure drop. The elliptical tube geometry owing to its aerodynamic shape, suffers relatively lower pressure drop and friction factor (Figure 6). An enhancement factor based on circular tube geometry ($\eta_c$) has also been presented for elliptical tube geometry in Figure 6. Due to low pressure drop of the elliptical tube, its thermo-fluid performance is higher compare to circular tube type FTHX.

3.3. Effect of span wise variation of split winglets pairs.

In the present sections effect of span wise variation of split winglet pairs on the performance of FTHX has been examined by performing numerical simulations. The two winglet pairs having a length of 5.335 mm each were placed symmetrically around the elliptical tube with a constant attack angle of 10°. The stream wise centre distance of first and second winglets are maintained $\Delta X_1 = -2.67$ mm and $\Delta X_2 = +2.67$ mm respectively. The negative sign indicates the upstream placement of the VG relative to the tube centre whereas a positive sign shows a downstream position of the VG pair. The first winglet pair is moved away from the tube whereas the second winglet pair was brought close to the tube, both with an interval of 0.5 mm.

In Figure 8, the Nusselt number plot shows the variation in heat transfer performance of
the FTHX with span wise distance of VGs. The first and second winglet pairs were initially placed at $ΔY_1 = 8.5$ mm and $ΔY_2 = 7.5$ mm respectively. For this location, the split winglet pair practically acts like a full length winglet (Figure 9a), which drives the fluid near to the wake region and enhances the local heat transfer coefficient. When the $ΔY_1 = 9$ mm and $ΔY_2 = 7$ mm (Figure 9b), a small gap is created between the split winglets and the majority of the fluid is driven towards adjacent tube by rear winglet pair of split winglets and that compresses the thermal isolated zone which increases both heat transfer and pressure drop. As $ΔY_1$ is advanced further to 9.5 mm, more fluid is driven towards the downstream tubes. However, with $ΔY_2$ is brought close to 6.5 mm, the gap between split winglets is increased further and the majority of the fluid is passed through the gap without coming close to tube wake region (Figure 9c). This results in a slight decrease in heat transfer performance. However, as more fluid is constrained to move within the gap between the split winglet pairs the pressure drop rises continuously as shown in the plot of friction factor in Figure 8.

For enhanced heat transfer, the intermixing between low temperature fluid behind the tubes and relatively high temperature fluid in the mainstream flow is desirable. From the vorticity contours in Figure 10, it can be clearly observed that the vorticity magnitude is greater near to the wake region of the adjacent tube for $ΔY_1 = 9$ mm and $ΔY_2 = 7$ mm. The high vorticity magnitude suggests stronger secondary flow intensity which promotes the effective exchange of heat between cold fluid behind the tubes and the mainstream flow and thus facilitates greater heat transfer enhancement.

Enhancement factor in Figure 8 concludes the overall effect of the variation in span wise placement of the split winglet pair. The enhancement factor reaches a maximum value at locations $ΔY_1 = 9$ mm and $ΔY_2 = 7$ mm and then falls continuously. The increase in Nusselt number is not significant beyond $ΔY_1 = 9$ mm and $ΔY_2 = 7$ mm, however, the pressure drop penalty increases substantially. This shows that beyond these span wise locations, the
enhancement in pressure drop is larger than the heat transfer enhancement and thus the enhancement factor drops continuously.

3.4. Effect of stream wise variation of split winglets pairs.

The effect of split winglet pair’s stream wise locations on the performance of the FTHX has been examined by performing numerical simulations. The two VG pairs were placed symmetrically over the elliptical tubes with a constant attack angle of 10°. The first and second winglets were placed at their optimum span wise locations $\Delta Y_1 = 9$ mm, and $\Delta Y_2 = 7$ mm respectively. The first winglet pair and the second winglet pair were advanced simultaneously with constant step size of 0.5 mm towards the upstream and downstream of the tube respectively. The negative sign indicates distance measured upstream of the tube whereas the positive sign is for the distance measured downstream of the tube.

The variation in $Nu$ with stream wise distance of split winglet pairs is shown in Figure 11(a). Initially, the first and second winglet pairs were placed at stream wise locations corresponding to $\Delta X_1 = -2.67$ mm and $\Delta X_2 = +2.67$ mm respectively. With the increase in stream wise eccentric separation ($\Delta X$) between the split winglet pairs, the heat transfer performance falls continuously.

On comparing the temperature contours from Figure 12(a) and 12(b) it can be clearly seen that the temperature in the wake zone is higher when there is no stream wise eccentric separation ($\Delta X_1 = -2.67$ mm, $\Delta X_2 = +2.67$ mm) between the two split winglet pairs and the gap between the winglets is only due to span wise eccentric separation. As the gap is narrow, very less amount of fluid is constrained to move within the gap and the majority of the fluid is driven towards the wake region. As a result, both $Nu$ and $f$ remain high for the present configuration.

When the second winglet pair advances downstream, it does not form the converging
region with the tube surface. This causes the flow to impinge with a relatively low velocity with majority of the fluid passed within the gap between the split winglet pairs. This results in relatively low thermal mixing and heat transfer performance (Figure 11a) with less pressure drop (Figure 11b). On Comparing the velocity contours in Figure 12(c) & (d), it can be clearly visualized that greater compression of the winglet supported tube as well as the adjacent tube is achieved when there is no stream wise eccentric separation between the winglets \((\Delta X = 0 \text{ mm})\). Thus, winglet location corresponding to \(\Delta X_1 = -2.67 \text{ mm}, \Delta X_2 = +2.67 \text{ mm}\) offers maximum heat transfer performance.

An enhancement factor has been presented in Figure 11(c) to summarize the overall effect of stream wise location of the split winglet pairs on the performance of the FTHX. The enhancement factor is maximum at stream wise locations: \(\Delta X_1 = -2.67 \text{ mm}, \Delta X_2 = +2.67 \text{ mm}\) \((\Delta X = 0 \text{ mm})\), and gradually decreases as the stream wise eccentric separation is increased \((\Delta X = 1 \text{ mm}; 2 \text{ mm}; 3\text{mm})\) between the two winglet pairs. For all the other considered stream wise locations of the split winglet pairs both \(Nu\) and \(f\) reduces continuously. However, the continuous reduction in enhancement factor suggests that the drop in heat transfer is more dominant than reduction in pressure drop for increased stream wise eccentric separation \((\Delta X = 1 \text{ mm}; 2 \text{ mm}; 3\text{mm})\).

3.5. Effect of variation in attack angle of the split winglet pair.

The influence of attack angle of the split winglet pair’s on the performance of the FTHX has been examined by performing numerical simulations. The attack angle of each winglet pair was varied from 5°-20° with a step size of 5°. A total of 16 possible combinations of attack angle for the two winglet pairs \((\beta_1 & \beta_2)\) were examined and presented in Table 3. The length of the first and the second winglet pair is maintained as 5.335 mm. The split winglet pairs were placed at their optimum stream wise and span wise locations corresponding to: \(\Delta X_1 = -2.67 \text{ mm}, \Delta Y_1 = 9 \text{ mm}, \Delta X_2 = +2.67 \text{ mm}, \text{ and } \Delta Y_2 = 7 \text{ mm}.)
Table 3 shows the variation of $Nu$, $f$, and $\eta$ as the functions of first attack angle ($\beta_1$) and second attack angle ($\beta_2$) of split winglet pairs for both $Re_H = 611$ and $Re_H = 819$. It is observed from the table that $Nu$ is always higher when the attack angle for the first winglet pair ($\beta_1$) is more compare to the second winglet pair ($\beta_2$). The two winglet pairs contribute differently to the overall heat transfer performance. The first winglet pair guides the incoming flow towards the constricted section created between the second winglet pair and the tube and also towards the downstream side through the span wise eccentric gap created between the two winglet pairs. The second winglet pair placed relatively closer to the tube, however, is responsible only for minimization of the wake zone of the VG supported tube which is evident from Figures 13(a) & 13(b).

When the attack angle of first winglet pair is kept larger compare to the second winglet pair, it is visualized from Figure 13(c) the wake regions of VG supported tube and the adjacent tubes are effectively compressed compare to the earlier cases (Figure 13a-b). This results in enhanced thermal mixing and heat transfer performance. Table 3 shows that increasing the attack angle of either the first or the second VG pair increases the exposed area of the winglet and thereby offers more resistance to the incoming flow. Thus, the pressure drop increases substantially across the flow domain.

Table 3 also lists the enhancement factor to assess overall effect of $Nu$ and $f$. For $\beta_1 = 15^\circ$, and $\beta_2 = 5^\circ$, significant enhancement in heat transfer performance is obtained with a moderate pressure drop. Thus, the enhancement factor is maximum for these angular positions of the two winglet pairs for both $Re_H = 611$ & 819.

Table 4 provides a comparison of the maximum overall thermo-fluid performance obtained in the present study with some of the recent contributions from different researchers. The Table shows that the split winglet pairs supported to the elliptical tube combination enhances the heat transfer performance with a moderate pressure drop penalty. Thus, a greater value of $\eta$
has been obtained with the proposed winglet and tube combination. The Table also shows a higher value of $\eta$ when VGs with holes have been used [38]. It is expected that the thermo-fluid performance of the proposed split winglet configuration can be further improved by providing holes on the winglet surface.

4. CONCLUSIONS

The present investigation analyses the thermo-fluid performance of a FTHX with the rectangular winglets placed with a heretofore-unused combination- split winglet pairs over the elliptical tubes. The numerical investigation revealed that the split winglet pairs placed at optimum attack angles compresses the wake region of the winglet supported tube as well as the adjacent tube and resulted in significant heat transfer enhancement. Influences of split winglet pair’s stream wise and span wise location were also investigated. Following are some major conclusions drawn from the present study:

- The FTHX with elliptical tube configuration results in greater overall thermo-fluid performance by virtue of its significantly low pressure drop compared to FTHX with circular tube configuration.
- Based on the highest enhancement factor optimum stream wise and span wise locations were obtained at $\Delta X_1 = -2.67$ mm, $\Delta X_2 = +2.67$ mm and $\Delta Y_1 = 9$ mm, $\Delta Y_2 = 7$ mm respectively for split winglet pairs where the enhancement factor was increased to 13-15%.
- It is observed that the attack angle of the front and rear winglet pair has a great influence on the overall thermo-fluid performance of the FTHX. For both $Re_H = 611 \& 819$, the enhancement factor was found maximum (about 15-16% higher than the baseline model) when the front winglet is set at an attack angle of 15° and the rear winglet with an attack angle of 5°.
The thermo-fluid performance of the circular tube geometry is expected to be higher than the elliptical tube configurations at higher Reynolds number \((Re_H > 900)\). Therefore the results presented in the present numerical investigations for the elliptical tubes with split winglet pairs are limited for applications involving flows with low velocity \((Re_H < 900)\). Moreover, in the present work, computations have been performed considering in-line arrangement of tubes. Similar computations can be performed for the staggered arrangement of tubes to have a comparison on the basis of the cooling effect produced and the pressure drop penalty.

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**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>(A_{min})</td>
<td>Minimum free flow area ((m^2))</td>
<td></td>
</tr>
<tr>
<td>(A_T)</td>
<td>Total heat transfer surface area ((m^2))</td>
<td></td>
</tr>
<tr>
<td>(a)</td>
<td>Length of semi-minor axis ((m))</td>
<td></td>
</tr>
<tr>
<td>(b)</td>
<td>Length of semi-major axis ((m))</td>
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<td>(C)</td>
<td>Specific heat of Aluminum ((J/kg K))</td>
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<td>(C_P)</td>
<td>Specific heat of air ((J/kg K))</td>
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<tr>
<td>(D)</td>
<td>Tube diameter ((m))</td>
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</tr>
<tr>
<td>(f)</td>
<td>Friction factor ((m))</td>
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<tr>
<td>(F_t)</td>
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<td>(FTHX)</td>
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<td>(h)</td>
<td>Heat transfer coefficient ((W/m^2 K))</td>
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<tr>
<td>(H)</td>
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<tr>
<td>(k)</td>
<td>Thermal conductivity ((W/m K))</td>
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</table>
$L$ Flow length (m)
$L_1$ Front winglet length (m)
$L_2$ Rear winglet length (m)
$m$ Mass flow rate (kg/s)
$n$ Number of tube rows
$Nu$ Nusselt Number
$P$ Pressure (Pa)
$P_a$ Atmospheric pressure (Pa)
$P_l$ Longitudinal tube pitch (m)
$P_t$ Transverse tube pitch (m)
$\Delta P$ Pressure drop (Pa)
$\overline{p}$ Total average pressure (Pa)
$Q$ Heat transfer capacity (W)
$Re$ Reynolds number
$Re_D$ Reynolds number based on tube diameter
$Re_{Dh}$ Reynolds number based on hydraulic diameter
$Re_H$ Reynolds number based on channel height
$T$ Temperature (K)
$\Delta T$ Log mean temperature difference (K)
$\overline{T}$ Total average temperature (K)
$u$ Velocity in x-direction (m/s)
$v$ Velocity in y-direction (m/s)
$VG$ Vortex Generator
$w$ Velocity in z-direction (m/s)
$\Delta X$ Stream wise eccentric separation (m)
\( \Delta X_1 \) Stream wise position for front winglet (m)

\( \Delta X_2 \) Stream wise position for rear winglet (m)

\( \Delta Y \) stream wise eccentric separation (m)

\( \Delta Y_1 \) Span wise position for front winglet (m)

\( \Delta Y_2 \) Span wise position for rear winglet (m)

**Subscripts**

- \( al \) Aluminum
- \( a \) Air
- \( c \) Circular tube configuration
- \( down \) Bottom surface of the domain
- \( e \) Elliptical tube configuration
- \( in \) Inlet parameter
- \( max \) Maximum
- \( o \) Outlet parameter
- \( 0 \) Smooth channel
- \( up \) Top surface of the domain
- \( w \) Wall

**Greek symbols**

- \( \alpha \) Thermal diffusivity \( (m^2/s) \)
- \( \beta_1 \) Angle of attack for front winglet (degree)
- \( \beta_2 \) Angle of attack for rear winglet (degree)
- \( \eta \) Enhancement factor
- \( \mu \) Dynamic viscosity \( (Pa \cdot s) \)
- \( \rho \) Density of air \( (kg/m^3) \)
REFERENCES


**Figure Captions**

**Figure 1.** Top view of the winglet supported Fin-and-tube heat transfer surface.

**Figure 2.** Top view of split winglet orientation around elliptical tubes.

**Figure 3.** (a) Top view, and (b) side view (shown for third and fourth tubes only) of the computational domain with applied boundary conditions.

**Figure 4.** Cut section view of meshes around the split winglet pair and elliptical tube.
Figure 5. Comparison of CFD results with baseline model of: (a) & (b) experimental results for baseline case of Joardar and Jacobi [10], and (c) baseline model experimental results of Kays and London [33].

Figure 6. Heat transfer and fluid flow parameters for circular and elliptical tube geometries.

Figure 7. Top view of the considered tube geometries: (a) Circular Tube, and (b) elliptical tube.

Figure 8. Nusselt number, Friction factor and Enhancement factor as a function of winglet pair’s span wise position.

Figure 9. Temperature Contours (unit: K) over the fin surface for: (a) $\Delta Y_1 = 8.5$ mm, $\Delta Y_2 = 7.5$ mm (b) $\Delta Y_1 = 9$ mm, $\Delta X_2 = 7$ mm, and (c) $\Delta Y_1 = 9.5$ mm, $\Delta Y_2 = 6.5$ mm.

Figure 10. Vorticity magnitude (unit: s$^{-1}$) contours over the fin surface for: (b) $\Delta Y_1 = 9$ mm, $\Delta X_2 = 7$ mm, and (c) $\Delta Y_1 = 10$ mm, $\Delta Y_2 = 6$ mm.

Figure 11. Thermo-fluid performance parameters as a function of stream wise locations of split winglet pairs.

Figure 12. Temperature contours (unit: K) over the fin surface for: (a) $\Delta X_1 = -2.67$ mm, $\Delta X_2 = +2.67$ mm (b) $\Delta X_1 = -4.17$ mm, $\Delta X_2 = +4.17$ mm. Velocity contours over the fin surface for: (c) $\Delta X_1 = -2.67$ mm, $\Delta X_2 = +2.67$ mm (d) $\Delta X_1 = -4.17$ mm, $\Delta X_2 = +4.17$ mm.

Figure 13. Velocity contours (unit: m/s) over the fin surface for: (a) $\beta_1 = 5^\circ$, $\beta_2 = 15^\circ$, (b) $\beta_1 = 10^\circ$, $\beta_2 = 20^\circ$, and (c) $\beta_1 = 15^\circ$, $\beta_2 = 5^\circ$.

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Compression of the wake region of winglet supported tube due to higher attack angle of the second split winglet pair.

Compression of the wake region of winglet supported tube and adjacent tube due to higher attack angle of the first winglet pair.
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<table>
<thead>
<tr>
<th>Grid number</th>
<th>Nu&lt;sub&gt;c&lt;/sub&gt;</th>
<th>Grid number</th>
<th>Nu&lt;sub&gt;e&lt;/sub&gt;</th>
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Table 2. Details of the FTHX model used for validation of the numerical model.

<table>
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<th>Kays and London [33]</th>
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<td>n</td>
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<td>H</td>
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Table 3. Performance parameters variation with attack angle of the split winglet pairs.

<table>
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<th>$Re_H = 611$</th>
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<th></th>
<th>$Re_H = 819$</th>
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<td></td>
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<td>$f$</td>
<td>$\eta$</td>
<td>$Nu$</td>
<td>$f$</td>
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</table>
Table 4. Comparison of maximum overall enhancement obtained with proposed split winglet configuration with recent contributions from different researchers.

<table>
<thead>
<tr>
<th>Researches</th>
<th>Surface modification</th>
<th>Flow condition</th>
<th>$\eta_{max}$</th>
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<tbody>
<tr>
<td>Present Work</td>
<td>Split Rectangular winglet pairs</td>
<td>$Re_H = 500-819$</td>
<td>16.2%</td>
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<tr>
<td>Naik and Tiwari [29]</td>
<td>Rectangular Winglets</td>
<td>$Re_D = 2000-4000$</td>
<td>10%</td>
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<tr>
<td>Gholami et al. [34]</td>
<td>Wavy Rectangular winglets</td>
<td>$Re_{Dh} = 400-800$</td>
<td>7.4%</td>
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<tr>
<td>Gholami et al. [35]</td>
<td>Corrugated Fins</td>
<td>$Re_{Dh} = 200-900$</td>
<td>16%</td>
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<tr>
<td>Qian et al. [36]</td>
<td>Rectangular Winglets</td>
<td>$Re_{Dh} = 1300-2000$</td>
<td>11%</td>
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<tr>
<td>Naik and Tiwari [37]</td>
<td>Rectangular winglets</td>
<td>$Re_D = 2000-10000$</td>
<td>10%</td>
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<tr>
<td>Modi et al. [38]</td>
<td>Rectangular winglets with holes</td>
<td>$Re_{Dh} = 400-2000$</td>
<td>16.5%</td>
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<tr>
<td>Khan and Li [39]</td>
<td>Rectangular and delta winglet</td>
<td>$Re_{Dh} = 380$ to $1140$</td>
<td>14.4%</td>
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