EXPERIMENTAL AND NUMERICAL STUDY OF HEAT TRANSFER OVER A FINNED ELLIPTICAL FLAT TUBE FITTED WITH LONGITUDINAL VORTEX GENERATORS ON THE RECTANGULAR FIN SURFACE

Li Li,1,∗ Xiaoze Du,1 Yuwen Zhang,2 Chao Xu,1 Lijun Yang,1 & Yongping Yang1

1MOE Key Laboratory of Condition Monitoring and Control for Power Plant Equipment, North China Electric Power University, Beijing 102206, China
2Department of Mechanical and Aerospace Engineering, University of Missouri, Columbia, Missouri 65211, USA

∗Address all correspondence to Li Li E-mail: doubleli@ncepu.edu.cn

An experimental study of heat transfer over a finned elliptical flat tube fitted with and without longitudinal vortex generators (LVGs) mounted on the rectangular fins was carried out in a wind channel. Numerical simulation was performed to compare the experimental results and to verify the turbulence model. The test models were magnified four times greater than the original react model and correspondingly simplified according to the principle of similitude. Several fins were designed to construct the periodic boundary condition. Thermo couples were used to measure the temperatures on the fins and tube of the flow and surfaces. The experimental results showed that in the low Reynolds number regime, the heat transfer decreased continually from the entrance along the stream wise direction. However, in the high Reynolds number regime, the heat transfer had a maximum value in the middle points of the fins along the stream wise direction. It was demonstrated through the numerical simulation and experimental results that the configuration of the fin and tube with the LVGs on the fins could attain enhanced heat transfer without too much pressure drop.

KEY WORDS: extended surface, displaced enhancement device, compound enhancement, single-phase flow, vortex generators

1. INTRODUCTION

Air-cooled condensers are widely applied in power plants in regions where the water source is limited because of their significant advantage of saving water. Elliptical flat tubes with plains or continuous plain fins are typically used for single-row fin-and-tube bundle of air-cooled condensers because of their simple configuration and relatively low production cost in comparison to the fin-and-tube configuration with complicated enhanced fins (Du et al., 2013; Li et al., 2013b). To effectively reduce the dominant air-side thermal resistance an enhanced surface of plain fins needs to be developed. As a heat transfer enhancement method, longitudinal vortex generators (LVGs), punched out or mounted on the fin surface, are drawing increasing attention by researchers (Jacobi and Shah, 1995; He and Zhang, 2012). The heat transfer performance and flow characteristic are important to the optimization and design of heat exchangers.

There have been many experimental research studies that have concentrated on the enhancement of heat transfer by applying LVGs in a channel. Lau (1995) obtained the average Reynolds stresses and the turbulent kinetic energy for a channel with pairs of rectangular winglets periodically mounted along both the stream wise and span wise directions on one of the channel walls by using an...
X-wire and a quadruple hot-wire probe. Subsequently, experiments were conducted by Lau et al. (1999) with quadruple hot-wire probes to investigate the flow characteristic by using rectangular winglets to verify the turbulence models. Gentry and Jacobi (1997) performed experiments to investigate enhancement of heat transfer for flow over a flat plate using delta-wing vortex generators (VGs) at low Reynolds numbers through the naphthalene sublimation and flow visualization methods. Storey and Jacobi (1999) conducted heat transfer and flow visualization experiments to study the influence of stream wise vortices on frost growth in a rectangular channel.

Sommers and Jacobi (2005) experimentally studied the effects of LVGs on the fin surface of an air-to-refrigerant heat exchanger by employing large fin spacing in a frosting environment in a closed-loop wind tunnel. Type-T thermo couples were used to measure the inlet and outlet temperatures of the air. The convection heat transfer coefficient was shown to increase by 60–93% with the addition of the delta wings. Joardar and Jacobi (2008) conducted experiments in a closed-loop wind tunnel to evaluate the heat transfer enhancement of winglet VGs in a compact plain-fin-and-tube heat exchanger. They reported that the air-side heat transfer coefficient was increased by 16.5–44% for a single-row VG pair and 30–68.8% for a three-row VG array arrangement at Reynolds numbers in the range of 220–960. Kim and Yang (2002) performed experiments on a pair of delta wings with different angles of attack in common flow-up and common flow-down cases in a rectangular channel by using thermo-chromatic liquid crystal. The results demonstrated that the common flow-down cases showed better heat transfer characteristics than the common flow-up cases.

Yang et al. (2007) experimentally determined the effects of the angles of attack in VGs on the flow and heat transfer by using a five-hole probe and thermo-chromatic liquid crystal. It was shown that the case of a 45° angle of attack has the highest heat transfer enhancement of all cases. Wang et al. (2007) conducted experiments on the heat transfer and pressure drop in horizontal narrow rectangular channels with mounted LVGs. Water was chosen as the working fluid. Thermo couples were used to mea-

<table>
<thead>
<tr>
<th>NOMENCLATURE</th>
<th>Greek Symbols</th>
</tr>
</thead>
<tbody>
<tr>
<td>(A)</td>
<td>area ((m^2))</td>
</tr>
<tr>
<td>(C_f)</td>
<td>pressure drop coefficient</td>
</tr>
<tr>
<td>(c_p)</td>
<td>heat rate capacity ([kJ/(kg \cdot K)])</td>
</tr>
<tr>
<td>(D)</td>
<td>hydraulic diameter ((m))</td>
</tr>
<tr>
<td>(d)</td>
<td>diameter of base tube ((m))</td>
</tr>
<tr>
<td>(H)</td>
<td>height of fin or longitudinal vortex generator ((m))</td>
</tr>
<tr>
<td>(h)</td>
<td>heat transfer coefficient ([W/(m^2 \cdot K)])</td>
</tr>
<tr>
<td>(I)</td>
<td>current ((A))</td>
</tr>
<tr>
<td>(L)</td>
<td>length ((m))</td>
</tr>
<tr>
<td>(m)</td>
<td>mass flow rate ((kg/s))</td>
</tr>
<tr>
<td>(Nu)</td>
<td>Nusselt number based on the hydraulic diameter</td>
</tr>
<tr>
<td>(P)</td>
<td>pressure ((N/m^2))</td>
</tr>
<tr>
<td>(Re)</td>
<td>Reynolds number based on the hydraulic diameter</td>
</tr>
<tr>
<td>(s)</td>
<td>distance between fins ((m))</td>
</tr>
<tr>
<td>(T)</td>
<td>temperature ((K))</td>
</tr>
<tr>
<td>(u, v, w)</td>
<td>velocity ((m/s))</td>
</tr>
<tr>
<td>(U)</td>
<td>voltage ((V))</td>
</tr>
<tr>
<td>(V)</td>
<td>velocity vector ((m/s))</td>
</tr>
<tr>
<td>(W)</td>
<td>width of fin or base tube ((m))</td>
</tr>
<tr>
<td>(x, y, z)</td>
<td>Cartesian coordinates</td>
</tr>
</tbody>
</table>

Subscripts

- \(av\) average
- \(f\) fin
- \(in\) quantity evaluated at inlet
- \(loss\) heat transfer loss
- \(lower\) lower surface of the fluid domain
- \(max\) quantity evaluated at maximum
- \(min\) quantity evaluated at minimum
- \(out\) quantity evaluated at the outlet
- \(t\) flat tube
- \(upper\) upper surface of the fluid domain
- \(w\) wall
ensure the temperatures of the flow and walls. Wang et al.
(2007) concluded that LVGs could greatly improve the
heat transfer rate by 10–45% in the range of Reynolds
number from 3000 to 20,000. Allison and Dally (2007)
performed flow visualization and heat transfer experi-
ments to analyze the effects of delta-winglet VGs on the
performance of a fin-and-tube radiator. It was found that
the winglet surface had 87% of the heat transfer capac-
it but only 53% of the pressure drop of the louver fin
surface.

Zhang et al. (2008) compared the heat transfer of heat
exchangers with VGs punched out or mounted on the fins
trough naphthalene sublimation heat/mass analogy ex-
periments. Chen and Shu (2004) experimentally investigat-
gated the effects of a delta-wing VG on the flow and heat
transfer characteristics in fan and uniform flows. Only
one delta wing with a 40° angle of attack as a turbula-
tor was placed before an aluminum plate. Laser Doppler
velocimetry (LDV) was used to obtain the flow structures
and near-wall flow parameters. Their results showed that
the VG in fan flows has an insignificant overall effect
on heat transfer compared with uniform flow. Torii et al.
(2002) performed experiments to obtain the heat transfer
and pressure loss of a fin-and-tube heat exchanger using
in-line or staggered tube banks with delta-winglet VGs.
The results showed that delta-winglet-type VGs could
guarantee heat transfer and reduce pressure loss in a fin-
tube heat exchanger with circular tubes at relatively low
Reynolds number flow. Akbari et al. (2009) conducted ex-
periments to study the effects in a single passage of a two-
row fin-and-tube heat exchanger with two different con-
figurations of delta-winglet pair VGs. Wu and Tao (2012)
experimentally studied the heat transfer phenomenon in
rectangular channels with a pair of LVGs with different
angles of attack. Only one pair of LVGs in the channel and
thermo couples was used to measure the temperatures on
surfaces. Li et al. (2013a) performed experiments to in-
vestigate the thermal-fluid characteristics of a flat-fin heat
sink with a pair of VGs in a cross-flow channel. An in-
frared thermography camera was used to obtain the tem-
perature of the wall surface.

Besides the experimental investigations, manynumer-
sical simulations (Zhu et al., 1995; Chen et al., 1998a,b;
Wu and Tao, 2007; Bilir et al., 2010; He et al., 2012;
2013) have been conducted to demonstrate the enhance-
ment of LVGs on heat transfer in different configurations.
Although there has been much work carried out in order
to understand the heat transfer in fin-and-tube configura-
tions with or without LVGs, most experiments and numer-
ic simulations on fin-and-tube configurations are relat-
ed to circular base tubes. Few reports have focused on the
application of LVGs in a fin-and-tube configuration with
an elliptical flat tube in direct air-cooled condensers of a
power plant. Furthermore, most of the experiments in the
literature relating to VGs have focused on one channel
rather than multiple channels, which might bring discrep-
ancies with implementations of periodic and symmetric
boundary conditions. In this study, the heat transfer and
pressure drop on the finned elliptical-flat tube with and
without delta winglet VGs are studied experimentally and
numerically. This research can supply basic data for the
design of VGs on extended surfaces, which can be coop-
atively applied in some passive techniques (such as dis-
placed enhancement device technology) and compound
techniques (Webb and Kim, 2005; Bergles and Manglik,
2013) to improve energy transport.

2. EXPERIMENTAL APPARATUS AND
PROCEDURE

2.1 Experimental Setup

The experiments were conducted in a wind tunnel com-
prised of inlet, stabilization, convergent, test (where the
section area is 300 mm × 200 mm), and divergent sec-
tions, which are shown in Fig. 1. The air driven by the
blower flowed through the test model. Insulation cotton
was used to wrap the outside of the whole test section
to maximally reduce heat loss to the environment. The
frequency of the blower was controlled by a frequency
changer to obtain the desired velocity of the inlet flow.
An electric heating sheet stuck on the inner surface of
a flat base tube was used to supply constant heat flux.
A voltmeter and ammeter with 0.5% measure precision
were employed to measure the voltage and current of
the electric heating sheet to calculate the heating rate. The
inlet and outlet temperatures of the air were measured us-
ing type-T thermo couples placed on the upstream and
downstream areas of the test model. Temperature mea-
surements on the walls were carried out using thermo cou-
ples welded on the surfaces. All of the temperature data
were collected using an Agilent 34970A (Agilent Tech-
nologies, Inc., Santa Clara, CA, USA) data acquisition
system. These thermo couples were calibrated by a high-
resolution platinum resistance thermometer (at 0.01°C)
with a constant temperature water tank. The calibrated
uncertainties of the thermo couples were within ±0.5°C.
The air velocity was measured by a hot-wire anemome-
ter with 2% precision, inserted in the inlet or outlet of the
test section. Micropressure meter was used to measure
the pressure difference between the inlet and outlet of the test section. The velocities of the inlet flow in the present tests varied from 1 to 8 m/s.

The system was considered to be at steady state when all of the temperature fluctuations were within 1% for at least 5 min. This system was allowed to run until the steady state was eventually achieved, and then the data were recorded. The coolant flow rate was then changed in order to take another measurement at steady state. After obtaining the data of the finned elliptical-flat tube heat exchanger without VGs, the VGs were mounted on the fins. The experiments were conducted in the same way for the enhanced heat exchanger.

2.2 Geometry

A four-time magnified model was designed in the present experiment according to the similarity principle, taking into account the difficulty resulting from the small dimensions between the fins of the actual fin-and-tube in direct air-cooled condensers. The heat transfer characteristics of the two test models were measured in order to introduce the effect of the LVGs. One was the finned elliptical-flat tube without VGs as the baseline model, and the other was the same finned elliptical-flat tube with the LVGs on plain fins. Figure 2 depicts schematic diagram of the finned elliptical-flat tube with or without LVGs and the location of the thermo couple son the walls. The coordinate system is presented in Fig. 2, where $x$ is along the stream wise direction, $y$ is along the span wise direction, and $z$ denotes the fin pitch direction. The elliptical flat tube and fin were made from carbon steel and aluminum, respectively. The fins were welded with the tube using the nitrogen protection brazing method. For convenience in the manufacture and measurement, the tube was divided into two parts from the center. The electric heating sheet was stuck on the inner surface of one-half of the base tube. Insulation cotton with low thermal conductivity was placed in the base tube to reduce the heat flux from the electric sheet to the environment. Two parts with 18 fins welded on each half-base tube were connected by bolts and sealant. The left and right parts ($y$-direction) were used to construct the symmetry boundary. Up and down fins ($z$-direction) constructed the periodic boundary. The length and width of the fin was 500 m/min the $x$-direction and 76 mm in the $y$-direction. The fin thickness ($\delta_f$) was 0.5 mm with an 11-mm spatial arrangement in the $z$-direction. The thickness of the base tube was 1 mm in the $y$-direction.

For the enhanced configuration, four fins with five pairs of delta-winglet VGs along the streamwise direction were stuck on each fin by silicon glue with high thermal conductivity. For the delta winglet, the length was 20 mm and the height was 5 mm. The thickness of the delta winglet was equal to that of the fin. The angle of attack ($\beta$), with respect to the incoming flow direction, was equal to 30°. The winglets in each pair were located in the third and two-thirds locations from outside of the base tube along the $y$-direction. Six thermo couples were welded on the back surface of the heat sheet to monitor the temperature of heat sheet in order to avoid being destroyed by high heat flux. Similarly, six thermo couples were welded on the outside surface of the base tube. Furthermore, 12 thermo couples were welded on two fins to measure the temperature of the fins. Meanwhile, one ther-
FIG. 2: Test model and locations of the thermo couples.

mocouple was assembled at the inlet of the test section to obtain the inlet temperature of the fluid, and six thermo couples were assembled at the outlet of fin-and-tube to measure the temperature of flow after heat transfer.

2.3 Operation Procedure and Data Processing

The Reynolds numbers were based on the average velocities and the hydraulic diameter at the average flow cross-section area. The heat transfer performances were evaluated at the average bulk temperature of the fluid. The Reynolds and Nusselt numbers are defined as follows:

\[ \text{Re} = \frac{\rho u_{\text{av}} D_{\text{av}}}{\mu} \]  
\[ \text{Nu}_{\text{av}} = \frac{h_{\text{av}} D_{\text{av}}}{\lambda} \]  

where \( h \) is the heat transfer coefficient, which can be calculated using the following relationship:

\[ h_{\text{av}} = \frac{Q - Q_{\text{loss}}}{\eta_0 A_0 \Delta T} \]  

where \( Q \) is the heat transfer rate expressed as

\[ Q = UI \]  

where \( Q_{\text{loss}} \) is the heat transfer loss to the surroundings, which was always found to be within 10–15% in this experiment. Here, \( \eta_0 \) is the overall fin surface efficiency:

\[ \eta_0 = 1 - \frac{A_f}{A_0} (1 - \eta_f) \]  

where \( A_f \) is the total area of the fin surfaces \( A_0 \); and \( A_0 \) is the total area of the heat transfer, which is the sum of
the areas of fin and tube surfaces. In Eq. (5), \( \eta_f \) denotes the fin efficiency, which is calculated from the following relationship:

\[
\eta_f = \frac{T_{f_{av}} - T_{av}}{T_{f_{av}} - T_{t_{av}}} \quad (6)
\]

where \( T_{f_{av}} \) is the average temperature of the fin surfaces; \( T_{t_{av}} \) is the average temperature of the tube surface calculated from the values of the simulated results; and \( T_{av} \) is the average temperature of flow based on the inlet and outlet, i.e.,

\[
T_{av} = \frac{T_{in} + T_{out}}{2} \quad (7)
\]

The logarithm temperature difference is defined as

\[
\Delta T = \frac{(T_w - T_{in}) - (T_w - T_{out})}{\ln(T_w - T_{in}/T_w - T_{out})} \quad (8)
\]

The pressure drop coefficient is defined as

\[
C_f = \frac{\Delta P^*}{(1/2) \rho u^2_{av}} \quad (9)
\]

where \( \Delta P^* \) is the pressure difference between the inlet and outlet.

The uncertainty for function \( \phi = f(x_1, x_2, x_3, \ldots, x_n) \) is estimated by using the following function introduced in the literature (Moffat, 1988):

\[
\delta \phi = \sqrt{\left( \frac{\partial \phi}{\partial x_1} \right)^2 (\partial x_1)^2 + \left( \frac{\partial \phi}{\partial x_2} \right)^2 (\partial x_2)^2 + \cdots + \left( \frac{\partial \phi}{\partial x_n} \right)^2 (\partial x_n)^2} \quad (10)
\]

From the uncertainty in the measured data previously mentioned, the estimated uncertainties of the Reynolds and Nusselt numbers were calculated as 3.2 and 7.1%, respectively.

### 3. NUMERICAL SIMULATION

#### 3.1 Physical Model

A schematic diagram of the finned elliptical-flat tube model used in the simulation is shown in Fig. 3. Five pairs of delta winglets were mounted on the fin surface. Due to the periodic and symmetric characteristics of the fin-and-tube configuration, one piece of the fin in the \( z \)-direction and half of the domain in the \( y \)-direction of the structure were selected as the computational domain. The geometric parameters of the fin-and-tube configuration were the same as in the experimental model. The domain was extended by 30 mm for the entrance section to ensure inlet uniformity and was extended by 500 mm for the exit section to avoid the recirculation effect.

#### 3.2 Governing Equations and Boundary Conditions

The flow in the computational domain was considered to be incompressible with constant thermo physical properties. It was assumed that the flow is three-dimensional, turbulent, and has no viscous dissipation. The governing equations are the following:

- **Continuity equation**
  \[
  \nabla \cdot \mathbf{V} = 0 \quad (11)
  \]

- **Momentum equation**
  \[
  \rho (\mathbf{V} \cdot \nabla) \mathbf{V} = -\nabla p + \mu \nabla^2 \mathbf{V} \quad (12)
  \]

- **Energy equation**
  \[
  \rho c_p = (\mathbf{V} \cdot \nabla T) = \lambda \nabla^2 T \quad (13)
  \]

**FIG. 3:** Computational domain.
or the inlet boundary condition, the velocity and temperature of the air were assumed to be uniform, i.e., \( u = u_{in}, \ v = v = 0, \) and \( T = T_{in}. \) For the outlet boundary condition, \( \partial u/\partial x = \partial v/\partial x = \partial w/\partial x = \partial T/\partial x = 0. \) The periodic boundary conditions were set along the normal direction of the upper and lower surfaces of the domain, \( \phi_{upper} = \phi_{lower}. \) Constant heat transfer rate \( q \) was set through the inner surface of the base tube. The symmetric boundary conditions were set at the sidewalls of the fluid and solid domain, i.e., \( v = 0, \ \partial u/\partial y = \partial w/\partial y = 0, \) and \( \partial T/\partial y = 0. \)

### 3.3 Solution Method and Grid Independence

The governing equations with corresponding boundary conditions were solved by using the finite-volume method. The semi-implicit method for pressure linked equations (SIMPLE) algorithm was employed to deal with the problem of velocity and pressure coupling. Once the maximum residual of all variables—such as the pressure, velocity components, and temperature—were less than \( 10^{-5} \), the solution was believed to be converged. Several mesh schemes of the enhanced configuration were conducted to examine the dependence of mesh size on the accuracy of the analysis results. The total numbers of the mesh schemes were 2,479,000, 3,190,000, and 3,970,000. The typical face velocity of flow (3 m/s) was chosen as the test case, and the \( k - \omega \)-turbulence model was applied. The grid independence results are shown in Fig. 4. The second grid scheme was chosen as the mesh for all of the simulation models because the difference between the second and third cases was smaller than 1.5%.

### 4. RESULTS AND DISCUSSION

#### 4.1 Average and Local Heat Transfers in the Experimental Results

The thermocouple has the disadvantage of hardly getting the entire field of temperature through a limited number of thermo couples. Using more thermo couples allowed obtaining more local heat transfer. However, too many thermo couples can occupy the space and severely influence the flow field. The locations of thermo couples play a very important role in the precision of the average characteristic. From Fig. 2, it can be seen that the locations of the thermo couples on the fins are close to the outer side of the fin, where the temperature is relatively low in comparison to the location near the base tube. The temperature of the fins averaged by these thermo couples, which is used to calculate fin efficiency, would be lower than the true bulk value. At the same time, the locations of the thermo couples on the base tube were close to the outer side of the base tube, where the temperature is lower because of the edge effect. The temperature of the base tube averaged by these thermo couples, which is used to calculate the logarithmic mean temperature difference, might be lower than the true bulk temperature. For the purpose of finding the most appropriate way of utilizing the limited data from the measurements, six kinds of data processing methods for calculating the temperature of the base tube and fin are given in Table 1, where TC denotes thermo couples and Av denotes average. For example, in Case I, the temperature of the base tube comes from the average of the measured temperature values of the six thermo couples on the base tube, whereas in Case IV the temperature of the

<table>
<thead>
<tr>
<th>Item</th>
<th>Case I</th>
<th>Case II</th>
<th>Case III</th>
<th>Case IV</th>
<th>Case V</th>
<th>Case VI</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>( T_{tube} )</td>
<td>( T_{c} )</td>
<td>( T_{heater} )</td>
<td>( T_{tube} )</td>
<td>( \text{Av} (T_{c}, T_{heater}) )</td>
<td>( \text{Av} (T_{c}, T_{heater}) )</td>
</tr>
<tr>
<td>( T_{fin} )</td>
<td>( T_{fin} )</td>
<td>( T_{fin} )</td>
<td>( \text{Av} (T_{c}, T_{fin}) )</td>
<td>( T_{fin} )</td>
<td>( \text{Av} (T_{c}, T_{fin}) )</td>
<td>( \text{Av} (T_{c}, T_{heater}) )</td>
</tr>
</tbody>
</table>
base tube comes from the measured temperature values of the six thermo couples on the base tube and the six thermo couples on the heater.

The fin efficiencies and logarithmic mean temperature differences obtained from the methods given in Table 1 show different heat transfer performances. The results of the heat transfer on the plain fin with and without LVGs are shown in Figs. 5(a) and 5(b). The values calculated from Case I have the maximum heat transfer, while the values calculated from Case VI have the minimum heat transfer. The correlation between the Nusselt and Reynolds numbers after data processing could be obtained through the fits of these data, and the parameters are listed in Table 2. The heat transfer of the plain fin without the LVG is similar to the forced convection in a rectangular smooth channel, which corresponds to the well-known Dittus–Boelter correlation. Although the Reynolds numbers in the experiment were less than the range limited in the classical empirical correlation, the flow could be considered as turbulent because of the fin-and-tube structure. Comparing the Dittus–Boelter correlation (Wang et al., 2007) with the fitted results in Table 2, Case II appears to be the most reasonable.

Figure 6 shows a comparison of the heat transfer on the finned elliptical-flat tube with and without LVGs computed from Case II. A difference between the average Nusselt numbers was not appreciably found in this experiment. On the one hand, the LVGs were not assembled on all of the fins but only on several fins in the middle. On the other hand, the thermo couples could not cover most regions but only several points, which resulted in the measured temperature possibly not representing the temperature of the overall field.

According to the recorded temperatures of thermo couples on the fins, the local heat transfer performance was calculated to present the heat transfer tendency along the stream wise direction, which is shown in Fig. 7. After removing the obvious error points, it is seen that the overall tendency of the heat transfer decreases along the stream wise direction. The rule in the graphs agrees with Deb et al. (1995) (where the characteristic of the rectangular channel was studied) and Biswas et al. (2012) (where an elliptical tube was used). The Nusselt number decreased monotonically in the lower Reynolds number cases. The extent of the drop was significant at the starting points,

### Table 2: Parameters of the correlations

<table>
<thead>
<tr>
<th></th>
<th>Case I</th>
<th>Case II</th>
<th>Case III</th>
<th>Case IV</th>
<th>Case V</th>
<th>Case VI</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain fin without LVGs</td>
<td>$C$=0.0018</td>
<td>0.00283</td>
<td>0.0096</td>
<td>0.023</td>
<td>0.0423</td>
<td>0.0967</td>
</tr>
<tr>
<td></td>
<td>$n$=1.143</td>
<td>0.7678</td>
<td>0.9085</td>
<td>0.8133</td>
<td>0.7128</td>
<td>0.5866</td>
</tr>
<tr>
<td>Plain fin with LVGs</td>
<td>$C$=0.0022</td>
<td>0.02</td>
<td>0.0109</td>
<td>0.0081</td>
<td>0.0221</td>
<td>0.0561</td>
</tr>
<tr>
<td></td>
<td>$n$=1.1083</td>
<td>0.8093</td>
<td>0.8812</td>
<td>0.9327</td>
<td>0.7824</td>
<td>0.6494</td>
</tr>
</tbody>
</table>
FIG. 6: Comparison between the plain fin with and without LVGs.

FIG. 7: Local heat transfer on fins.
and then tended to be gentle at the ending points after the velocity field was fully developed. The heat transfer was strongest at the entrance section of the fin channel because of the entrance effect. The boundary layer thickness at the entrance was thin and the local heat transfer coefficient was highest. At lower Reynolds numbers, the heat transfer decreased continually along the stream wise direction due to the increasing thickness of the boundary layer and weakening of the heat transfer ability of the coolant air. With the increasing Reynolds numbers, the Nusselt number along the stream wise direction had an increasing value in the middle points of the fin. This is because the transition from laminar to turbulent flow could be supported by the forced convection theory in Bergman et al. (2011, chapter 6). The Reynolds number of the point where the heat transfer begins to increase under plain fins with the L VGs case is earlier than that for the plain fins without L VGs, which are marked in Fig. 7. This means the turbulent extent of the flow in the test model with L VGs is more intense than in the model without L VGs. The phenomenon on Fin 1 is more obvious than on Fin 2. This can be explained by the experimental differences, including the manufacture and assembling of the test models.

4.2 Comparison between Experimental and Numerical Results

Four turbulence models [the $k-\varepsilon$ model, renormalization group (RNG) $k-\varepsilon$—model, shear-stress transport (SST) model, and $k-\omega$ model] have been used to study the validity of the solver method by comparing the experimental and numerical results. Figure 8 illustrates the comparisons of the Nusselt numbers obtained from the different turbulence models for the baseline case, which is the fin-and-tube configuration without L VGs, with that from the experiments. While there are minor differences between the experimental and simulated values, the tendencies are the same. The Nusselt numbers from the $k-\varepsilon$ and realizable RNG $k-\varepsilon$ model are lower than those from the other two turbulence models. The results of the $k-\omega$ model have good agreement with the experimental results of Case II. According to the analysis of the aforementioned experiment results, Case II has the most reasonable result. The pressure drop coefficient of the simulation results in comparison with the baseline case is shown in Fig. 9. The graph of the experimental results is close to the results of the SST and $k-\omega$ models. From the comparison of both the heat transfer and pressure drop between the experiments and simulations, the $k-\omega$ model is recommended to be chosen as the turbulence model in this study.

Figure 10 shows a comparison between the Nusselt numbers obtained from the different turbulence models for the enhanced model, which was fin-and-tube configuration with L VGs on the fins, with that from the experiments. The results from the numerical solution are slightly higher than those of the experimental results. As previously mentioned, the experimental results did not show that the enhancement of the advanced model could be explained by the problem of the design of the test model and the point measurement method, which did not exist in the numerical simulation. Hence, the differences between the experimental and simulated results in the case of the fin-and-tube configuration with L VGs are reasonable. Through the comparison between the simulated results with the verified experimental results using the Dittus–Boelter correlation in the baseline case, the numerical simulation could be proven as a valid method to obtain the characteristic of the heat transfer in the enhanced case.

4.3 Comparison of Numerical Results between the Baseline and the Enhanced Model

The enhancements of the heat transfer and the penalty of the pressure drop for the enhanced model in comparison with the baseline model are shown in Figs. 11(a) and 11(b), respectively. The heat transfer of the enhanced model is improved by a ratio of 12%, while the pressure drop pays 40% more penalty than the baseline model. The positive effect of L VGs was lower than the experimental results in Sommers and Jacobi (2005) and Joardar and Jacobi (2008) but agrees with the results in Wang et al. (2007) due to the destruction of the boundary layers and disturbance produced by the L VGs. As shown in Fig. 11(c), from the overall performance, considering the heat transfer and pressure drop ($\text{Nu}/C_f^{-1/3}/(\text{Nu}_0/C_f^{-1/3})$) (where $\text{Nu}_0/C_f^{-1/3}$ refers to the $\text{Nu}/C_f^{-1/3}$ of the baseline case), the enhanced model with L VGs on the fins has an advantage over the one without L VGs. With increasing Re numbers, the enhancement of the overall heat transfer performance brought by the L VGs becomes smaller, which can be explained by the intensified turbulent flow under greater Reynolds numbers. The results agree with those of Zeng et al. (2010). The disturbance of the secondary flow comes from the L VGs playing a significant role at small Reynolds numbers. The growing main flow with the increase of the inlet velocity leads to the gradual weakening of the enhanced effect of the secondary flow. Namely,
FIG. 8: Comparison of the heat transfer between the experimental and simulated results in the baseline model.
FIG. 9: Comparison of the pressure drop coefficient between the experimental and simulated results in baseline model.

FIG. 10: Comparison of the heat transfer between the experimental and simulated results in the enhanced model.
comparing the contribution to heat transfer for local turbulence induced by the LVGs under small Re numbers, the intensified turbulent flow plays a leading role in both cases with and without LVGs. Therefore, the two curves in Fig. 11(c) converge with the increase of the Re number.

5. CONCLUSIONS

The experiment of the heat transfer on a finned elliptical-flat tube was tested in an open wind tunnel. Thermo couples were used to measure the temperatures of the flow and wall surfaces. Numerical simulations were conducted to verify the experimental results. In the low Reynolds number region, the heat transfer dropped continually from the entrance along the stream wise direction. However, in the large Reynolds number region, the heat transfer had a peak value in the middle range of fins along the stream wise direction. On the basis of the verification between the experimental and simulation results of the baseline case, further simulation of the fin-and-tube configuration with LVGs was carried out. From the numerical simulation, the fin-and-tube configuration with LVGs on the fins could achieve enhanced heat transfer and did not cause too much pressure drop.

ACKNOWLEDGMENTS

The financial support for this research project from the National Basic Research Program of China (973 Program; Grant No. 2009CB219804) and the 111 Project (No. B12034) is gratefully acknowledged.
REFERENCES


