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Improved correlations of the thermal-hydraulic performance of large size multi-louvered fin arrays for condensers of high power electronic component cooling by numerical simulation

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Abstract

In view of a good thermal performance of the multi-louvered fin arrays, it is intended to introduce the special fin structure into the design of condensers for high power electricity converter cooling on the CRH (China Railway High-speed) trains in order to improve the security of normal running for the high-speed trains. The geometrical size of the multi-louvered fin arrays needed for the high power condensers is relatively larger than that of the conventional louvered fin arrays and the flow state within the large size multi-louvered fin arrays is different as well, with the Reynolds numbers based on the louver pitch over the range of 2,850 to 11,000 in the present study. Flow and heat transfer characteristics of the large size multi-louvered fin arrays with various structural parameters are numerically simulated using Large Eddy Simulation (LES). More reasonable parameters correlated with the thermal-hydraulic performance of the large size multi-louvered fin arrays are identified compared to the conventional characteristic parameters employed in the published work. Finally, improved correlations for the Fanning friction $f$ factor and the Colburn-$j$ factor representing the flow and heat transfer characteristics, respectively, are put forward in terms of the identified parameters with special interval bounds for the condenser design of high power electronic cooling.
Keywords: Multi-louvered fin arrays; Condenser; High power electronic component cooling; Thermal-hydraulic performance; Large Eddy Simulation (LES); Correlation
List of symbols

Nomenclature

\( A_o \) \hspace{1cm} \text{air-side total heat transfer area of the fin arrays, m}^2

\( A_c \) \hspace{1cm} \text{minimum cross section area in the louver gap flow, m}^2

\( A_{fr} \) \hspace{1cm} \text{frontal surface area of the fin arrays, m}^2

\( c_p \) \hspace{1cm} \text{isobaric specific heat capacity, J/(kg K)}

\( c_v \) \hspace{1cm} \text{volumetric specific heat capacity, J/(kg K)}

\( D_h \) \hspace{1cm} \text{equivalent hydraulic diameter of pipe flow, mm}

\( f \) \hspace{1cm} \text{Fanning friction factor, –}

\( F_d \) \hspace{1cm} \text{flow length (or flow depth), mm}

\( F_h \) \hspace{1cm} \text{fin height, mm}

\( F_p \) \hspace{1cm} \text{fin pitch, mm}

\( F_{th} \) \hspace{1cm} \text{fin thickness, mm}

\( h_o \) \hspace{1cm} \text{air-side average heat transfer coefficient, W/(m}^2\text{ °C)}

\( j \) \hspace{1cm} \text{heat transfer Colburn factor, –}

\( L_a \) \hspace{1cm} \text{louver angle, °}

\( L_{fl} \) \hspace{1cm} \text{non-louvered flat-landing region width, mm}

\( L_h \) \hspace{1cm} \text{louver height, mm}

\( L_p \) \hspace{1cm} \text{louver pitch, mm}

\( L_{tr} \) \hspace{1cm} \text{louvered transitional region width, mm}

\( LFA \) \hspace{1cm} \text{a characteristic parameter correlating louver pitch, fin pitch and louver angle (Equation (22)), –}

\( p \) \hspace{1cm} \text{pressure, Pa}

\( \text{Pr} \) \hspace{1cm} \text{Prandtl number, –}

\( q \) \hspace{1cm} \text{heat flux, W}
Re_{dh}  
Reynolds number based on the equivalent hydraulic diameter, –

Re_{LP}  
Reynolds number based on the louver pitch, –

\bar{S}_{ij}  
resolved rate-of-strain tensor, –

T  
temperature, K

T_p  
flat tube pitch, mm

t  
time, s

Nu  
Nusselt number, –

u_{fr}  
frontal velocity, m/s

\bar{u}_i  
filtered velocity in LES, m/s

V_{\text{max}}  
characteristic velocity in the louver gap flow, m/s

x_i  
coordinate components, mm

x^+  
dimensionless distance of inlet effect region, –

Greek symbols

\rho  
density, kg/m^3

\nu_o  
kinematic viscosity of air, m^2/s

\mu  
dynamic or eddy viscosity, kg/(s m)

\sigma_{ij}  
shear-stress tensor, –

\lambda  
thermal conductivity, W/(m K)

\delta  
the effective spacing of the louver gap flow, mm

\Delta T  
temperature difference, K

\Delta p_{tot}  
total pressure drop, Pa

Subscript

\textit{air}  
flowing air

\textit{in}  
Inlet

\textit{out}  
Outlet
1 Introduction

Compact heat exchangers with multi-louvered fin arrays (also called parallel flow heat exchangers) are widely used for automotive and air-conditioning cooling due to their good thermal performances [1–5]. Investigations show that the high performance of the multi-louvered fin arrays compared to smooth fins are attributed to the continuous interruption of flow boundary layer by the extended louvered fins [2, 6].

In view of a good performance of the louvered fin arrays, it is intended to introduce the special fin structure into the design of condensers for high power electricity converter cooling on the CRH (China Railway High-speed) trains. The high power electricity converter as a power electronic component on the CRH trains, needs to be cooled down during operation in order to sustain normal running of the high-speed trains. Once the monitored core temperature in the converter achieves an upper limit temperature (usually 80-85 °C), the converter will be compulsorily stopped by an intelligent controller to protect the reliability and durability of the power electronic component. In this situation, a good design of the condenser for the high power electricity converter cooling is of significance to ensure the security of normal running of the high-speed trains.

Figure 1 shows the photo of a power electronic converter and its condenser for the component cooling on a CRH train. The air-side terminal heat radiator of the prototype condenser made up of alloy aluminum is constructed by straight fin arrays for forced air cooling. As the air-side thermal resistance of the condenser usually
accounts for 70–90% of the whole condenser thermal resistance [1–3], it does make sense to replace the original straight fins with multi-louvered fins to improve heat dissipation effect. However, the air-side geometrical size of the condenser for the power converter cooling shown in Figure 1 is relatively larger than that of the compact heat exchangers with multi-louvered fin arrays for automotive and air-conditioning cooling. Usually, most structural parameters of the multi-louvered fin arrays for the compact heat exchangers are small, with fin pitch and louver pitch on the magnitude order of several millimeters, as well as flow length in the range of 16–42 mm [1–5]. While the fin pitch and the louver pitch for the high power condenser are around fivefold to tenfold as those for the small size multi-louvered fins. Thus, it is necessary to carry out a further investigation to ascertain the thermal-hydraulic performance of the large-size multi-louvered fin arrays used for high power electronic cooling.

Figure 1 Photo of a high power electronic converter and its condenser for cooling on a CRH train

With regard to the state of the art of investigations on the multi-louvered fin arrays, the related subject matters mainly involve qualitative influence analysis of geometrical parameters [1, 2, 7–12], effect mechanism of flow and heat transfer [6, 13–21], generalized correlations of thermal-hydraulic performance [22–32], frost behaviors [33–37] and so on. Several types of typical multi-louvered fin arrays are
summarized in [29]. Specifically, the geometrical parameters comprise louver pitch 
\( L_p \), fin pitch \( F_p \), flow length \( F_d \), louver angle \( L_a \), fin thickness \( F_{th} \), fin 
height \( F_h \), louver height \( L_h \), non-louver region width \( L_{n}\), louvered transitional 
region width \( L_{nt} \), etc. In the aspect of a single geometrical parameter, it is reckoned 
that the pressure drop and the average heat transfer coefficient of the louvered fin 
arrays decrease as the increase of louver pitch \([1, 2]\). Tian \([2]\) argued that an optimal 
louver pitch to fin pitch ratio existed with a ratio range of 0.9–1.6. As for the flow 
length, most of the sizes in the publications are in the range of 16–42 mm \([1–5]\) and 
the heat transfer coefficient decreases meanwhile pressure drop increases as the flow 
length increases \([9]\). An optimal louver angle is also found in \([1, 9]\). The optimal angle 
is 27° for the case of a flow length of 24 mm. Kim and Bullard \([9]\) declared that the 
flow length is one of the main influence parameters of pressure drop and the effect of 
fin pitch on heat transfer is small. While Qi et al. \([11]\) argued that the flow length, the 
ratio of fin pitch to fin thickness and the number of louvers are the primary influence 
parameters, using Taguchi method to evaluate the influence extent of five factors on 
different control levels.

When it comes to the flow and heat transfer mechanism, it involves flow state, 
thermal wake effect, flow transition and unsteadiness, etc. The flow state represents 
the predominant flow direction, which can be delineated by flow efficiency \([13, 14]\). 
The flow state is generally divided into two categories, duct flow and louver gap flow 
\([15–18]\). Zhang and Tafti \([19]\) classified two types of thermal wake interferences that
occurred in multi-louvered fins, inter-fin interference occurring between adjacent
rows of louvers and intra-fin interference appearing on subsequent louvers of the
same row or fin, respectively dominated by louver gap flow at a high flow efficiency
and by duct flow at a lower flow efficiency. It was verified by them that the thermal
wake effects could be expressed as functions of the flow efficiency and the fin pitch to
louver pitch ratio. Besides, the flow transition analysis is aimed at disclosing the flow
mechanism of multi-louvered fin arrays from steady flow to unsteady flow in terms of
flow state [6, 21]. It is argued in [6] that most of the louvers exhibit unsteadiness by a
Reynolds number of 1300 except for the entrance louver and the first two louvers
following it.

Apart from the qualitative analysis aforementioned, the correlations of
thermal-hydraulic performance respectively for the flow and heat transfer
characteristics are highly concerned as it can help to provide design rules for real
engineering. Most of the generalized correlations are fitted in terms of the
dimensionless Fanning friction $f$ and Colburn-$j$ factor, except that a few ones adopted
Stanton or Nusselt number for heat transfer characterization [2, 4, 9, 16, 22–32].
Amongst all the correlations, it was worth mentioning that the generalized
correlations presented by Chang et al. [26, 27] was reckoned as the most widely
applicable formulas before 2005 because they collected the experimental data of 5
different types of louvered fin structures totally including 91 heat exchanger samples
from the literature. In 2006 they presented the amendment of their correlations with a
reduced error [29]. In 2009 Park and Jacobi [30] proposed more accurate, reliable and updated predictive correlations using a comprehensive experimental database consisting of 1030 heat-transfer and 1270 pressure-drop measurements from 9 independent laboratories for 126 sample heat exchangers. In recent years, relevant studies tend to focus on the subject matter of frost behaviors of louvered fins [33–37] and no obvious progress is observed for correlations.

Generally, the previous research work mainly considered the flow state with the Reynolds numbers (based on the louver pitch) lower than 2,000 for the small size multi-louvered fin arrays. The present study considered relatively large size multi-louvered fin arrays with the Reynolds numbers (based on the louver pitch) over the range of 2,850–11,000. To accurately capture the flow and temperature field information in the multi-louvered fin arrays with higher Reynolds numbers, Large Eddy Simulation (LES) was adopted to solve the 3-D transient flow and heat transfer characteristics using commercial software package FLUENT. Furthermore, it was noticed that the correlations [2, 4, 9, 16, 22–32] presented in the past usually utilized the ratios \( \frac{F_d}{L_p} \), \( \frac{F_p}{L_p} \), \( \frac{F_k}{L_p} \), \( \frac{F_{th}}{L_p} \), etc. as the correlated parameters. For the louver angle, usually \( L_a/90 \) was taken as corresponding correlated parameter. General mathematical form of the conventional correlations is shown in Equations (1) and (2). As a matter of fact, the dimensionless Fanning friction \( f \) and Colburn-\( j \) factor are not a monotonic function (or not a rigorous exponential function) of each correlated characteristic parameter, indicating that more reasonable parameters
correlated with the thermal-hydraulic performance of the large size multi-louvered fin arrays need to be identified for correlations. The present study will try to identify more reasonable characteristic parameters and present improved correlations for the multi-louvered fin arrays for the design of high power condenser unit.

\[ f = a_1 \text{Re}_{LP}^{b_1} \left( \frac{L_u}{90} \right)^{b_2} \left( \frac{F_p}{L_p} \right)^{b_3} \left( \frac{F_h}{L_p} \right)^{b_4} \left( \frac{F_d}{L_p} \right)^{b_5} \left( \frac{L_i}{L_p} \right)^{b_6} \]  

(1)

\[ j = a_2 \text{Re}_{LP}^{c_i} \left( \frac{L_u}{90} \right)^{c_1} \left( \frac{F_p}{L_p} \right)^{c_2} \left( \frac{F_h}{L_p} \right)^{c_3} \left( \frac{F_d}{L_p} \right)^{c_4} \left( \frac{L_i}{L_p} \right)^{c_5} \left( \frac{F_{in}}{L_p} \right)^{c_6} \]  

(2)

where \( a_i, b_i, \) and \( c_i \) (i - integer) are fitting constants.

2 Numerical model

2.1 Geometrical model

The structure of typical multi-louvered fin arrays with flat tube is illustrated in Figure 2 [38]. In view of the complexity of the multi-louvered fin arrays in a heat exchanger or condenser, it is efficient to make some assumptions for CFD (Computational Fluid Dynamics) calculations to save the computational efforts. The louvered fin surface is assumed to be flat. The heat source side at the flat tube wall is treated as equivalent heat source boundary conditions. Single computational unit for the multi-louvered fin arrays is considered for CFD calculations according to the symmetry and periodic feature, as shown in Figure 3. Extended regions before the inlet and after the outlet are considered to ensure the computational accuracy [6]. The inlet extended length is
chosen as $5 F_r$ as it is found that the inlet effect is not significant, while an extended length of $10–20 F_r$ is considered for the outlet extended region since the wake flow from the outlet has an evident effect on the flow characteristic. For a high frontal velocity condition, a longer outlet extended length is needed.

Figure 2 Schematic sketch of the flat tube and multi-louvered fins

Figure 3 Computational domain of the flat tube and multi-louvered fins as well as boundary conditions

2.2 Mathematical model

As the Reynolds numbers (based on the louver pitch) in the present study is over the range of 2,850 to 11,000 presumably with flow unsteadiness, the Large Eddy Simulation (LES) is adopted to solve the flow and heat transfer characteristics using FLUENT software package. The top-hat or box filter is used in finite volume implementation of LES in FLUENT [39]. Assuming an incompressible flow for the problem studied, the space-filtering equations of continuity, momentum and energy for the LES method are given in Equations (3–5) under Boussinesq hypothesis [40].

\[
\frac{\partial}{\partial x_i} (\rho \overline{u}_i) = 0 \tag{3}
\]

\[
\rho \frac{\partial}{\partial t} (\overline{u}_j) + \rho \frac{\partial}{\partial x_j} (\overline{u}_i \overline{u}_j) = -\frac{\partial p}{\partial x_j} + \frac{\partial \sigma_{ij}}{\partial x_j} - \frac{\partial \tau_{ij}}{\partial x_j} \tag{4}
\]
\[ \rho C_v \frac{\partial T}{\partial t} + \rho C_v \frac{\partial}{\partial x_j} (\bar{u}_i T) = -p \frac{\partial \bar{u}_i}{\partial x_j} + \sigma_{ij} \frac{\partial \bar{u}_i}{\partial x_j} - \frac{\partial q_j}{\partial x_j} \]  

(5)

where the shear-stress tensor \( \sigma_{ij} \) is defined as

\[ \sigma_{ij} = \mu(T) \left( 2S_{ij} - \frac{2}{3} \delta_{ij} S_{kk} \right) \]  

(6)

The effects of the unresolved subgrid scales of motion on the resolved flow variables are accounted for by the subgrid scale (SGS) terms in Equations (7) and (8).

\[ \tau_{ij} = \bar{\rho} \mu \bar{\tau} \delta_{ij} \]  

(7)

\[ q_j = \bar{T}u_j - \bar{T} \]  

(8)

The Smagorinsky-Lilly SGS model is used for the calculations and the local SGS stresses are taken to be proportional to the local rate of strain of the resolved flow \( \bar{S}_{ij} \), with the SGS viscosity evaluated as \([39, 40]\).

\[ \mu_{SGS} = \rho(C_{SGS} \Delta)^2 \sqrt{\frac{2\bar{S}_{ij} \bar{S}_{ij}}{S_{ij}}} \]  

(9)

where \( C_{SGS} \) is a constant; \( \Delta \) is the length scale; the resolved rate-of-strain tensor \( \bar{S}_{ij} \):

\[ \bar{S}_{ij} = \frac{1}{2} \left( \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \]  

(10)

In analogy to the SGS stresses, the subgrid heat flux \( q_j \) can be related to the filtered large scale temperature gradient by an eddy diffusivity \( \alpha_i \) as

\[ q_j = -\alpha_i \frac{\partial \bar{T}}{\partial x_j} \]  

(11)
For more information about the LES method, please refer to [39, 40].

2.3 Boundary conditions and solving method

As shown in Figure 3, in the computational domain only half of the louver height is considered due to symmetry and a symmetrical plane is justified at the half height cross section of the louvered fins. The upper and the lower surface of the domain is reckoned to be periodic boundary conditions due to the fact that there are multi-louvers in the vertical direction with a regular distribution. The inlet is set as the frontal face of the inlet extended region with an inlet temperature of 30 °C and a fixed frontal velocity dependent upon the specific cases (2–15 m/s for the large size), while the outflow condition is considered on the right side of the outlet extended region. The inlet turbulent intensity and the turbulent length scale are determined by an empirical formula referring to [40]. The wall surface of the side flat tube is set to be the convective heat transfer boundary conditions as done in [41]. The thermal conductivity of the alloy aluminum louvered fin is 110 W/(m K). Thermo-physical properties of the flowing air are considered as piece-wise linear [39].

With regard to the solving method based on the finite volume method, the second-order interpolation scheme is used for the pressure terms and the bounded central differencing scheme [39] is employed for the convective terms of the momentum equations. The second-order upwind differencing scheme is used for the
convective terms of the energy equations. To ensure the computational convergence, calculation results of the laminar flow are adopted as the initial fields of the LES calculations. The time step of computation is 0.03–0.05 s and the total calculating time interval is 15–20 s. The residual control accuracies of the continuity, momentum and energy equations for the convergent solution are $10^{-4}$, $10^{-4}$, $10^{-9}$, respectively. For some special cases with a high frontal velocity, it is hard to attain a convergent solution at some time steps and then the solution is reckoned to be convergent until the variations of the outlet temperature and the pressure drop remain to be within 0.5%.

### 2.4 Parameter definitions

The maximum velocity through the louver gap flow is taken as the characteristic velocity, as defined in Equation (12) [27, 38].

$$V_{\text{max}} = \frac{u_p T_p F_p}{F_h (F_p \cos L_a - F_m)} \quad (12)$$

The Reynolds numbers based on the louver pitch is defined in equation (13).

$$\text{Re}_{LP} = \frac{V_{\text{max}} L_p \cos L_a}{V_o} \quad (13)$$

Total pressure drop between the inlet and outlet

$$\Delta p_{\text{tot}} = p_{in} - p_{out} \quad (14)$$
The air-side average heat transfer coefficient \( (h_o) \) is calculated by

\[
h_o = \frac{q_{air}}{A_o \Delta T} = \frac{\rho_{air} c_{p,air} \mu_f \Delta T}{A_o \Delta T} \frac{A_f (T_{out} - T_{in})}{A_o \Delta T} \quad \text{(15)}
\]

Please note that the numerical heat transfer temperature difference \( (\Delta T) \) in Equation (15) is calculated by the temperature difference between the volume-averaged fin temperature and the volume-averaged air temperature obtained by the 3-D CFD calculations.

For the dimensionless processing of the pressure drop and heat transfer coefficient, the numerical Fanning friction factor \( f \) is calculated by [22]

\[
f = \frac{\Delta p_{air \cdot f}}{\frac{1}{2} \rho_{air \cdot t} V_{max}^2} \frac{A_c}{A_o} \quad \text{(16)}
\]

And the dimensionless heat transfer Colburn-j factor is given as [22, 26–30].

\[
j = \frac{Nu}{Re Pr^{1/3}} = \frac{h_o}{\rho_{air} c_{p,air} V_{max}} Pr^{1/3} \quad \text{(17)}
\]

where the Nusselt number is

\[
Nu = \frac{h_o \cdot L_p}{\lambda_{air \cdot f}} \quad \text{(18)}
\]

### 3 Model validation

The experimental data from Kim and Bullard [9] is chosen to carry out the CFD model validation. The specific structural parameters of the multi-louvered fin are as
follows: $F_d = 20 \text{ mm}$, $F_p = 1.4 \text{ mm}$, $L_p = 1.7 \text{ mm}$, $L_a = 23^\circ$, $T_p = 10.15 \text{ mm}$, $F_h = 8.15 \text{ mm}$, $L_h = 6.4 \text{ mm}$. The structure of the corresponding flat tube can be found in [38]. To begin with, it is necessary to conduct a grid independence testing to identify the effective baseline cell size from the viewpoint of saving computational efforts. The condition with a frontal velocity of 1 m/s is chosen to verify the grid independence, utilizing cell sizes of 0.1 mm, 0.09 mm, 0.85 mm, 0.08 mm, 0.075 mm, 0.07 mm for the fluid part (flowing air) and those of 0.08 mm, 0.07 mm, 0.06 mm, 0.05 mm, 0.05 mm for the solid part (alloy aluminum). Generated meshes for the cases are comprised of 339, 458 cells, 519,544 cells, 629,419 cells, 821,744 cells, 893,234 cells, 985,590 cells, respectively. Figure 4 gives the calculated results of the pressure drop and the average heat transfer coefficient, indicating that a baseline cell size of 0.08 mm is suitable for the calculations. For the large size multi-louvered fin arrays, similar grid independence is also carried out and a baseline cell size of 0.4–0.5 mm is taken for the calculations.

Furthermore, three types of geometrical model, simplified full louver height model [3, 42], louvered fins model with flat-landing region [43], louvered fins model with both flat-landing region and transitional region [44, 45] are compared for the model validation. Note that the flow height ($F_h = 8.15\text{ mm}$) and the louver height ($L_h = 6.4 \text{ mm}$) is aimed to the louvered fins model with flat-landing region. Concerning the
simplified full louver height model, there is no flat-landing region and the louver height should be treated the same as the fin height. As for the louvered fins model with both flat-landing and transitional regions, there is not specific size reference and thus the transitional region width is chosen as 0.5 mm, 0.8 mm, 1.0 mm while the flat-landing region width is set as 0.475 mm, 0.675 mm, 0.875 mm for different cases, respectively.

Figures 5 and 6 give the calculation results of the total pressure drop and the average heat transfer coefficient versus the frontal velocity for different calculation models, respectively. It should be mentioned that the logarithmic mean temperature difference is used here to calculate the average heat transfer coefficient [9] in order to make the comparison on the same benchmark. Comparing with the experimental data from Kim and Bullard [9], it suggests that the simplified full louver height model evidently overestimates the pressure drop and the heat transfer coefficient, as the flat-landing region is replaced by the louvered fin structure. Using mean deviation to evaluate the relative error, the absolute mean deviation of the simplified full louver height model is 11.0%, while that of the louvered fins model with flat-landing region is 1.39%. Regarding the louvered fins model with both flat-landing and transitional regions, it is found that the absolute mean deviation for the cases of transitional region widths 0.5 mm, 0.8 mm, 1.0 mm are 2.32%, 4.61% and 6.08%, respectively. When it comes to the heat transfer coefficient, a maximum absolute mean deviation of 23.1% is observed for the simplified full louver height model. The absolute mean deviation of
the heat transfer coefficient for the louvered fins model with flat-landing region is 11.8%, while those for the louvered fins model with both flat-landing and transitional regions with transitional region widths 0.5 mm, 0.8 mm, 1.0 mm are 14.1%, 10.8% and 8.8%, respectively. Combining the calculation errors of the pressure drop and the heat transfer coefficient, it is assumed the louvered fins model with both flat-landing and transitional regions can make a more accurate prediction, which will be used in the following calculations. For the large size louvered fin arrays, the transitional region width is chosen as 2.0 mm.

Figure 5 Calculated total pressure drop versus the frontal velocity for different calculation models compared to the experimental data; FLR model – Louvered fins model with the flat-landing region; FL&TR model – Louvered fins model with both flat-landing and transitional regions

Figure 6 Calculated heat transfer coefficient versus the frontal velocity for different calculation models compared to the experimental data; FLR model – Louvered fins model with the flat-landing region; FL&TR model – Louvered fins model with both flat-landing and transitional regions
4 Results and discussion

4.1 Effect of the louver angle

A wide range of louver angles (15–30°) are chosen to explore the effect of the louver angle on the flow and heat transfer characteristics of the large size multi-louvered fin arrays, with specific geometrical parameters of $F_{sh} = 0.5$ mm, $F_d = 240$ mm, $160$ mm, $L_p = 8$ mm, $F_p = 4.75$ mm, $F_h = 25$ mm, $L_f = 1.0$ mm, $L_v = 2.0$ mm. Figure 7 shows the total pressure drop and the average heat transfer coefficient versus various louver angles with a flow length of 160 mm. It suggests that the total pressure drop and the average heat transfer coefficient firstly decrease, followed with an increase region till arriving at a maximum value after which a decrease appears again, as the louver angle ($L_u$) increases from 15° to 30° for a fixed frontal velocity ($u_f$). The minimum and maximum pressure drops arise at $L_u = 17°$ and $L_u = 27°$, respectively. The minimum heat transfer effect appears at $L_u = 17°$, while the maximum heat transfer effect occurs around $L_u = 25°–27°$, implying that there is an optimal louver angle for a good thermal performance of the multi-louvered fin arrays. The optimal louver angle is close to the case of 24 mm flow depth in Ref. [9]. More than that, the whole variation trend indicates that the pressure drop or the heat transfer coefficient is not a monotonic function of the louver angle. It is therefore not accurate to correlate the thermal-hydraulic performance with a monotonic mathematical relationship in the form of equations (1) and (2) as employed in the conventional correlations [2, 4, 9, 26–32].
Figure 7 The pressure drop and the heat transfer coefficient versus various louver angles ($F_d = 160$ mm)

With a careful examination on the geometrical relationship of the louvered fin arrays, the effective spacing of the louver gap flow ($\delta$) is defined in Equation (19) to delineate the characteristic of the louver gap flow, as it is reckoned that the flow state of the louver gap flow is effective with a high flow efficiency [15–19]. In order to get a dimensionless effective spacing, the ratio of the effective spacing ($\delta$) to the louver spacing perpendicular to adjacent louvers ($F_p \cos(L_a)$) is considered in Equation (20) or (21). It is shown that $\left(L_p/F_p\right) \cdot \tan(L_a)$ is a characteristic parameter correlating the louver pitch ($L_p$), the fin pitch ($F_p$) and the louver angle ($L_a$). It suggests that the three parameters $L_p$, $F_p$, $L_a$ take an interplay for the louvered gap flow. A notation ‘LFA’ is taken to represent the correlated parameter, as shown in Equation (22).

\[
\delta = F_p \cdot \cos L_a - L_p \cdot \sin L_a \quad (19)
\]
\[
m = \delta / \left(F_p \cdot \cos L_a\right) \quad (20)
\]
\[
m = 1 - \left(L_p / F_p\right) \cdot \tan L_a \quad (21)
\]
\[
LFA = \left(L_p / F_p\right) \cdot \tan L_a \quad (22)
\]

Figures 8 and 9 shows the pressure drop and the average heat transfer coefficient versus LFA, respectively, for different frontal velocities ($u_{fr}$) with a flow length of...
The flow and heat transfer characteristics should be categorized by different interval bounds of LFA. When $0.5 \leq LFA \leq 0.858$, the pressure drop is an exponential function of the LFA. The maximum pressure drop appears at $\delta \approx F_{th}$. In the ranges of $0 < LFA \leq 0.5$ and $0.858 < LFA \leq 1$, the pressure drop decreases as the louver angle increases, so does the average heat transfer coefficient. The average heat transfer coefficient increases with the increase of LFA in the interval of $0.5 \leq LFA \leq 0.75$ and the optimal heat transfer effect appears in the interval of $0.75 \leq LFA \leq 0.858$. The maximum heat transfer tends to be where $\delta \approx 2F_{th}$.

4.2 Effect of louver pitch to fin pitch ratio

The visualization experiment done by Howard [46] characterized the flow structure in the louvered fin arrays as ‘effective flow’ and ‘low efficiency flow’ and declared that the transitional region from a low efficiency flow to an effective flow happened at $L_p/F_p = 0.7–0.8$ with a louver angle of $20^\circ$ in their study. When $L_p/F_p$ was higher than 0.8, the flow structure was reckoned as effective, while a value of $L_p/F_p$ less than 0.7 was assumed to be ineffective. A series of fin pitch values from 4.75 mm to 8.0 mm were chosen to ascertain the effect of $L_p/F_p$ over the range of 1.0–1.684. Figures 10 and 11 give the pressure drop and the average heat transfer coefficient
versus louver pitch to fin pitch ratio, respectively. It shows that the pressure drop or
the average heat transfer coefficient is an exponential function of $L_p/F_p$ when
$L_p/F_p > 1.3$. While the mathematical relationship between the pressure drop (or the
heat transfer coefficient) and $L_p/F_p$ isn’t a monotonic function when $L_p/F_p < 1.3$.

It indicates that the correlations for the thermal-hydraulic thermal performance of the
multi-louvered fin arrays should be fitted in different intervals of $L_p/F_p$ according
to the variation rules. Using the characteristic parameter $LFA$ identified in section
4.1 to show the variations of the Fanning friction factor $f$ and the heat transfer
Colburn-$j$ factor, as delineated in Figures 12 and 13, respectively. It is inferred that an
effective flow occurs when $LFA > 0.5$, corresponding to $L_p/F_p \approx 1.3$ and

$$ (L_p/F_p) \cdot \tan 23^\circ \approx 0.56. $$

Thus, for a good thermal performance of the multi-louvered fin arrays, $L_p/F_p \cdot \tan \alpha$ is recommended to be larger than 0.56.

Figure 10 The pressure drop versus the louver pitch to fin pitch ratio ($F_d = 240$ mm)

Figure 11 The average heat transfer coefficient versus the louver pitch to fin pitch
ratio ($F_d = 240$ mm)

Figure 12 The Fanning friction factor $f$ versus $LFA$ ($F_d = 240$ mm)

Figure 13 The heat transfer Colburn-$j$ factor versus the louver pitch to fin pitch ratio
($F_d = 240$ mm)

4.3 Effect of the fin thickness

To figure out the effect of fin thickness on the thermal-hydraulic performance of the
multi-louvered fin arrays, various fin thicknesses (0.15mm, 0.25mm, 0.3mm, 0.4mm, 0.5mm, 0.6mm, 0.75mm, 0.9mm and 1.0mm) are considered for a comparison with specific geometrical parameters $F_a = 240$ mm and $L_a = 25^\circ$. Figure 14 gives the pressure drop versus fin thickness for different frontal velocities ($u_{fr}$). It is further found that the pressure drop is a piecewise exponential function of the fin thickness ($F_{th}$) in the ranges of $0.15$ mm $\leq F_{th} \leq 0.3$ mm and $0.4$ mm $\leq F_{th} \leq 1.0$ mm, as Figure 14 can be drawn in a semi-log plot given by Figure 15. Regarding the average heat transfer coefficient versus fin thickness shown in Figure 16, no similar rule is found as that of the pressure drop even using a semi-log plot or a double-log plot. To further explore the rule of the average heat transfer coefficient versus fin thickness, the ratio $\delta/F_{th}$ is identified as the characteristic parameter, inspiring by the phenomenon that the louver gap flow is affected by the leading edge thickness of the louvers. Table 1 lists the ratio $\delta/F_{th}$ for different fin thicknesses. Figure 17 shows the variation trend of the average heat transfer coefficient in terms of the ratio $\delta/F_{th}$. It is found that the average heat transfer coefficient is a piecewise exponential function of the fin thickness ($F_{th}$) in the ranges of $0.924 \leq \delta/F_{th} \leq 1.232$ and $1.54 \leq \delta/F_{th} \leq 6.16$. The line of demarcation tends to be $\delta = 1.5 F_{th}$. The rate of change of the average heat transfer coefficient in the interval of $0.924 \leq \delta/F_{th} \leq 1.232$ is slower than that in the interval of $1.54 \leq \delta/F_{th} \leq 6.16$. It seems that the parameter $\delta/F_{th}$ is more suitable for characterizing the thermal-hydraulic performance considering a specific interval.
Table 1. The ratio of the effective spacing of the louver gap flow to the fin thickness

Figure 14 The pressure drop versus the fin thickness \((F_d = 240\, \text{mm})\)

Figure 15 The pressure drop versus the fin thickness in a semi-log plot

Figure 16 The average heat transfer coefficient versus the fin thickness \((F_d = 240\, \text{mm})\)

Figure 17 The average heat transfer coefficient versus the ratio of \(\delta / F_{th}\)

### 4.4 Effect of the flow length

As argued by \([10, 11]\) that the flow length \((F_d)\) is one of the primary influence parameters for pressure drop. Especially, for a longer flow length, the flow path within the multi-louvered fin arrays will be very long at a high flow efficiency, contributing to a higher pressure drop. Different flow lengths \((F_d = 240\, \text{mm}, 224\, \text{mm}, 208\, \text{mm}, 192\, \text{mm}, 176\, \text{mm}, 160\, \text{mm})\) are chosen by successively reducing 2 louvers \((2L_p = 16\, \text{mm})\) for each case followed to determine the effect of flow length.

Figure 18 (a) and (b) give the pressure drop and the average heat transfer coefficient versus flow length. It suggests that the pressure drop is an exponential function of flow length \((F_d)\), while the heat transfer coefficient tends to be a linear variation as the increase of flow length.

Figure 18 The pressure drop and the heat transfer coefficient versus the flow length
To effectively correlate the flow length with the thermal-hydraulic performance, the dimensionless distance $x^+$ is introduced in Equation (24) for the multi-louvered fin arrays, in analogy to the definition of the inlet region dimensionless length (reciprocal of the Graetz number) [47] in Equation (23). This is mainly because the flow boundary layers along the louver gap flow are frequently distributed by the leading edge of the louvers, representing a characteristic of the inlet length effect.

$$x^+ = \frac{L/D_p}{Re_{D_p} Pr} \quad (23)$$

$$x^+ = \frac{F_d/L_p}{Re_{L_p} Pr} \quad (24)$$

Figure 19 shows the Fanning friction factor $f$ versus the dimensionless distance $x^+$ for different flow lengths. The absolute mean deviation of the scattered data points is about 1.1% when a logarithmic relationship between the factor $f$ and $x^+$ is assumed. It suggests that it is more reasonable to take the dimensionless distance $x^+$ as the correlated parameter in relation to the flow length ($F_d$). Figure 20 provides the heat transfer Colburn-$j$ factor versus the dimensionless distance $x^+$. A logarithmic relationship between the factor $j$ and $x^+$ is also observed, verifying the suitability of the identified parameter in relation to the flow length ($F_d$) as well as the Reynolds number ($Re_{L_p}$) based on the presumption of inlet region effect.
4.5 Effect of louver height to fin height ratio

Different fin heights ($F_h = 25$ mm, $23$ mm, $21.2$ mm, $19.7$ mm, $18.3$ mm, $17.1$ mm) are chosen to get various ratios of louver height to fin height ($L_n/F_h = 0.760, 0.7391, 0.7170, 0.6954, 0.6721, 0.6491$), considering an identical condenser size with a change of the number of tube rows in the transversal direction. Figure 21 shows the pressure drop and the heat transfer coefficient versus $L_n/F_h$ in log-log plots, indicating there is an exponential relationship between the pressure drop (or the heat transfer coefficient) and $L_n/F_h$. It is therefore reasonable to take $L_n/F_h$ as the correlated parameter with regard to the louver height and fin height.

4.6 Correlations of the Fanning fraction $f$ and the Colburn-$j$ factor

According to the reasonable characteristic parameters identified for the thermal-hydraulic performance of the large size multi-louvered fin arrays, the correlations of the Fanning fraction $f$ and the Colburn-$j$ factor can be obtained through multiple regression method [2, 26–30]. In view that the mathematical relationships of the characteristic factors $j, f$ versus the louver angle are not monotonic, the
correlations in terms of the Fanning fraction $f$ and the Colburn-j factor are proposed in
the interval of $0.75 \leq LFA \leq 0.858$, in which cases the heat transfer performance of the
multi-louvered fins is relatively better than in the other intervals. Additionally, the
ratio of louver pitch to fin pitch is limited to the ‘effective flow’ region, meeting
$L_p / F_p > 0.56 \cdot (\tan L_a)^{-1}$. And the fin thickness is subject to the bound condition of $\delta$
$\geq 1.5 F_{th}$. Through multiple linear regression, the correlations of the Fanning fraction $f$
and Colburn-j factor are attained in Equations (25) and (26), respectively.

$$f = 0.07667 \left( \frac{F_d / L_p}{\text{Re}_{L_p} \text{Pr}} \right)^{0.3211} \left( \frac{L_p}{F_p} \right)^{-2.0217} (\tan L_a)^{-2.3501} \left( \frac{\delta}{F_{th}} \right)^{-2.5343} \left( \frac{L_h}{F_h} \right)^{-0.7862}$$

(25)

$$j = 0.65842 \left( \frac{F_d / L_p}{\text{Re}_{L_p} \text{Pr}} \right)^{0.6317} \left( \frac{L_p}{F_p} \right)^{-0.433} \left( \frac{F_d \tan L_a}{F_p} \right)^{-0.4825} \left( \frac{\delta}{F_{th}} \right)^{-1.1902} \left( \frac{L_h}{F_h} \right)^{-0.2074}$$

(26)

where the specific interval bounds are $0.75 \leq LFA \leq 0.858$; $L_p / F_p > 0.56 \cdot (\tan L_a)^{-1}$;
$\delta \geq 1.5 F_{th}$

Figure 22 shows the error limits of the fitting correlations for the Fanning friction $f$
and the Colburn-j factor. It indicates that 95% of the fitting data points are within the
$\pm 5\%$ confidence interval limits. The mean deviation errors for the factors $f$, $j$ are
$2.39\%$, $1.76\%$ respectively, which are very small compared to those in the literature
[26–30]. It is assumed that more reasonable characteristic parameters identified and
interval bounds give rise to a smaller error of the correlations, considering the specific
intervals of the correlated parameters for obtaining a good thermal-hydraulic performance of the multi-louvered fin arrays.

Figure 22 Error limits of the fitting correlations for the Fanning friction $f$ and the Colburn-$j$ factor

5 Conclusions

In order to enhance the heat dissipation effect of the condenser units for high power electricity converter cooling on the CRH trains, the flow and heat transfer characteristics of the relatively large size multi-louvered fin arrays are analyzed by large eddy simulation for the condenser units. It is verified that the correlated characteristic parameters of the thermal-hydraulic performance of the multi-louvered fin arrays adopted in the previous studies, such as $F_d/L_p$, $L_a/90$, $F_p/L_p$, $F_h/L_p$, $F_{th}/L_p$, etc., are not optimum due to the fact that the dimensionless Fanning friction $f$ and Colburn-$j$ factor are not a monotonic function (or not a rigorous exponential function) of each correlated parameter. Thus, more reasonable parameters correlated with the thermal-hydraulic performance of the large size multi-louvered fin arrays are identified compared to the conventional characteristic parameters employed. The correlated parameters are identified as $(L_p/F_p) \tan L_a$, $L_p/F_p$, $\delta/F_{th}$, $x' = (F_d/L_p)/(Re_Lp Pr)$ and $L_h/F_h$.

Moreover, specific interval bounds of the correlated parameters are ascertained for a
good thermal performance of the multi-louvered fin arrays. Improved correlations for the Fanning friction $f$ factor and the Colburn-$j$ factor representing the flow and heat transfer characteristics, respectively, are presented in terms of the identified characteristic parameters with the specific interval bounds, with a view to obtaining a good thermal-hydraulic performance of the multi-louvered fin arrays for high power condenser design. Comparing with the error ranges of the correlations in the published work, a smaller error of the correlations is obtained with 95% of the fitting data points within the ±5% confidence interval limits.

References


588 characterisation of compact heat exchangers for air heating and cooling in electric


593 characteristics in automotive louvered fins. Experimental Thermal and Fluid

595 [8] Liu M, Liaw J, Leu J, et al. 3-D simulation of thermal-hydraulic characteristics of
596 louvered fin-and-tube heat exchangers with oval tubes. ASHRAE Transactions

600 390–400.

601 [10] Hsieh CT, Jang JY. 3-D thermal-hydraulic analysis for louver fin heat
602 exchangers with variable louver angle. Applied Thermal Engineering 2006;

605 exchanger with corrugated louvered fins. Applied Thermal Engineering 2007; 27:
606 539–544.

heat exchangers with different fin configurations. Applied Thermal Engineering


[21] Tafti DK, Zhang X. Geometry effects on flow transition in multilouvered


[29] Chang YJ, Chang WJ, Li MC, Wang CC. An amendment of the generalized


Table Captions:

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Figure 2 Schematic sketch of the flat tube and multi-louvered fins

Figure 3 Computational domain of the flat tube and multi-louvered fins as well as boundary conditions

Figure 4 Testing of different grid systems

Figure 5 Calculated total pressure drop versus the frontal velocity for different calculation models compared to the experimental data; FLR model – Louvered fins model with the flat-landing region; FL&TR model – Louvered fins model with both flat-landing and transitional regions

Figure 6 Calculated heat transfer coefficient versus the frontal velocity for different calculation models compared to the experimental data; FLR model – Louvered fins model with the flat-landing region; FL&TR model – Louvered fins model with both flat-landing and transitional regions

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Figure 9 The average heat transfer coefficient versus $LFA$ ($F_d = 160$ mm)

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\[ \delta = F_p \cdot \cos L_a - L_p \cdot \sin L_a = 0.924 \text{ mm} \]

<table>
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<th>(F_{th}) (mm)</th>
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<th>0.30</th>
<th>0.40</th>
<th>0.50</th>
<th>0.60</th>
<th>0.75</th>
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<td>3.080</td>
<td>2.310</td>
<td>1.848</td>
<td>1.540</td>
<td>1.232</td>
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