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Fin–tube junction effects on flow and heat transfer in flat tube multilouvered heat exchangers

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Abstract

Three-dimensional simulations of four louver–tube junction geometries are performed to investigate the effect on louver and tube friction and heat transfer characteristics. Three Reynolds numbers, 300, 600 and 1100, based on bulk velocity and louver pitch are calculated. Strong three-dimensionality exists in the flow structure in the region where the angled louver transitions to a flat landing adjoining the tube surface, whereas the flow on the angled louver far from the tube surface is nominally two-dimensional. Due to the small spatial extent of the transition region, its overall impact on louver heat transfer is limited, but the strong unsteady flow acceleration on the top louver surface augments the heat transfer coefficient on the tube surface by over 100%. In spite of the augmentation, the presence of the tube lowers the overall Nusselt number of the heat exchanger between 25% and 30%. Comparisons with correlations derived from experiments on full heat exchanger cores show that computational modeling of a small subsystem can be used reliably to extract performance data for the full heat exchanger.

1. Introduction

Flat tube corrugated multilouvered fins are used in many compact heat exchanger applications to enhance the air-side heat transfer performance. Louvers reduce the average thermal boundary-layer thickness by interrupting its growth and by enhancing mixing through large-scale instabilities, hence increasing the average heat transfer coefficient. Previous experimental and numerical studies have established that the heat transfer in multilouvered fins is influenced by three factors: (a) flow direction [1,2]; (b) thermal wake interference [3]; (c) flow instabilities and transport of coherent vorticity in the vicinity of the louver surface [4]. These three mechanisms have mostly been studied with a louver-centric view, i.e, heat transfer enhancement on a nominally two-dimensional louver, with the assumption that louvers contribute a significant portion to the overall heat transfer surface. For the most part this assumption is well justified. However, in exchangers with large fin pitches and small fin heights or tube pitch, the tube surface can contribute substantially to the total heat transfer. For example for a fin pitch of 1.5–2.0 times the louver pitch, and a tube pitch of 5 louver pitches, the tube surface area contributes between 20% and 30% of the total heat transfer area. This, coupled with the fact that the tube is the primary heat transfer surface with the largest potential for heat transfer, requires that attention be paid to the heat transfer from the tube surface.

Our specific geometry of interest is a flat tube multilouvered exchanger with corrugated rectangular channels. In order to gain some insight into what influences tube heat transfer, in this study we focus our attention on the region of the louver near the junction with the tube surface. In this region, along the height of the fin, the louver transitions from an angle θ to 0° into a flat landing adjoining the tube surface as shown in Fig. 1(d). [1] Cui

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[1] The corrugated fin curvature near the tube wall is neglected.

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Nomenclature

D_H^*	hydraulic diameter of equivalent duct
F_d^*	flow depth
F_p	non-dimensional fin pitch (F_p^*/L_p^*)
L_p^*	dimensional louver pitch (characteristic length scale)
Nu	Nusselt number, $\frac{L_p^* q''^* /(T_s^* - T_{ref}^*)}{\kappa}$
Nu_{D_h}	Nusselt number, $\frac{D_h^* q''^* /(T_s^* - T_{ref}^*)}{\kappa}$
Pr	Prandtl number
q''^*	specified heat flux
Re	Reynolds number ($u_b^* L_p^*/v$)
Re_{L_p}	Reynolds number ($V_c^* L_p^*/v$)
Re_{D_h}	Reynolds number ($V_c^* D_H^*/v$)
T_s^*	louver or tube surface temperature
T_{ref}^*	reference temperature, domain integrated mixed mean
t	non-dimensional time
u_b^*	mean bulk velocity
V_c^*	maximum mean flow velocity
x	streamwise coordinate, along louver pitch
y	cross-stream coordinate, along fin pitch
z	lateral coordinate, along fin height

Greek symbols

γ	mean temperature gradient in the streamwise direction
κ	thermal conductivity
v	kinematic viscosity
θ	non-dimensional modified temperature. Also louver angle

Superscript

$*$	dimensional quantities

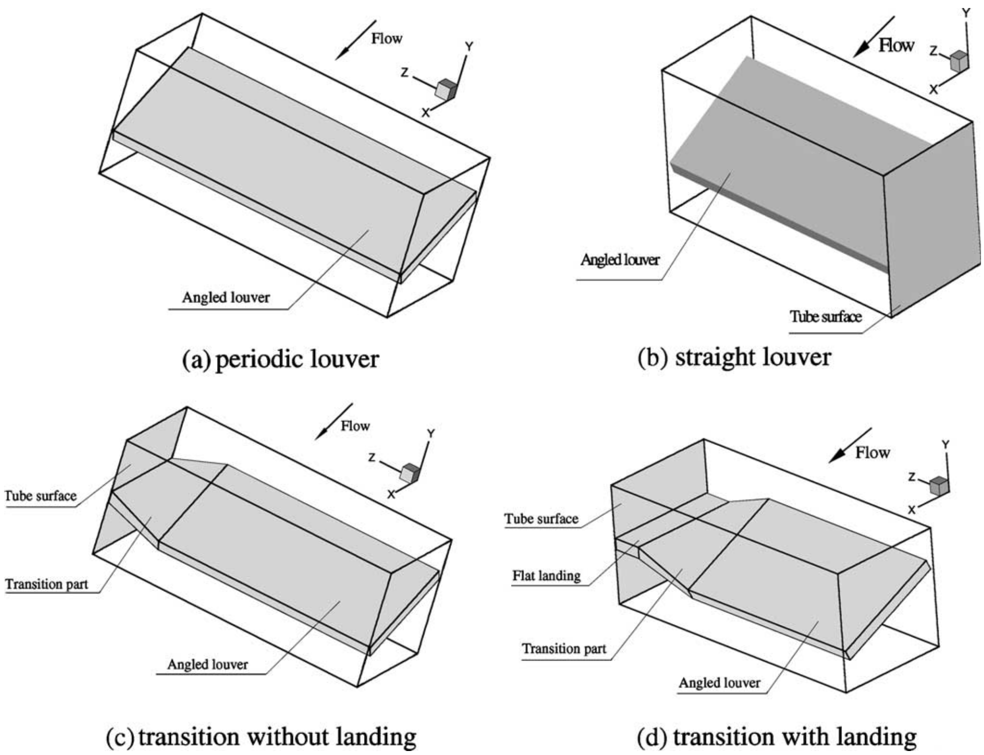


Fig. 1. Computational domain for four louver geometries: (a) periodic louver; (b) straight louver; (c) transition without landing; (d) transition with landing. Shaded areas are the louver and tube surface.

and Tafti [5] numerically investigated the geometry in Fig. 1(d) at a Reynolds number of 1100, based on louver pitch and bulk velocity. They found that although the flow on the angled portion of the louver was nominally two-dimensional with self-sustained flow oscillations characterized by spanwise vortices, the flow was strongly three-dimensional and unsteady in the transition region. An energetic unsteady vortex jet formed at the leading edge, which was drawn under the louver. The jet was complemented by a region of strong unsteady flow acceleration in the vicinity of the top louver surface. Evidence was presented that the temporal evolution of the two was correlated, which had a significant impact on local heat transfer coefficients. In spite of the high heat transfer in this region, the overall effect on mean louver heat transfer was found to be small because of the small spatial extent of the transition region. However, it was found that the strong acceleration near the junction with the flat landing had a significant effect on tube heat transfer.

Our research objective is to study the three-dimensional flow and temperature fields generated in compact heat exchangers and to determine whether these largely unknown characteristics can be used to further augment heat transfer by slight modifications to the base geometries. The objective of this paper is to extend the previous three-dimensional unsteady simulations to study three Reynolds numbers, 1100, 600 and 300. In addition to the Reynolds number effect, simulations are carried out on four variations of the transitional louver geometry to study the incremental effect of geometry at the junction with the tube. The heat transfer and friction results are presented separately for the louver and tube, and combined to estimate the overall effect. Comparisons are also made with existing louver-and-tube correlations in the literature to determine whether computational modeling of a subsystem can be used reliably to predict full heat exchanger core performance.

The paper is organized as follows: the numerical and computational method is presented briefly in the next section, followed by the description of the louver geometries. In the section on results, the general flow features, louver and tube friction and heat transfer characteristics are discussed. Finally comparisons are made with experimental correlations. This is followed by concluding remarks.

2. Numerical formulation

We solve the non-dimensional, time-dependent, incompressible Navier–Stokes and energy equations in conservative form in generalized curvilinear coordinates. The governing equations for momentum and energy are discretized with a conservative finite volume formulation using a second-order central difference scheme on a non-staggered mesh. The Cartesian velocities, pressure, and temperature are calculated and stored at the cell center, whereas contravariant fluxes are stored and calculated at the cell faces. A projection method [6] is used for the time integration of the discretized continuity and momentum equations.

The louvered fin geometry is approximated by an infinite array of louvers in both streamwise and cross-stream directions, which results in a simpler system with periodic repetition of the basic unit. Periodic boundary conditions for velocity, modified pressure and temperature are applied in the streamwise and cross-stream directions since the flow is assumed to be both hydrodynamically and thermally fully developed without any entrance or exit effects. No-slip, no-penetration boundary conditions for velocity and constant heat flux conditions are enforced on the louver and tube surface.

More details of the numerical algorithm, treatment of the boundary conditions, verification and validation of the computer program and strategies for parallel computing can be found in Tafti et al. [7–9] and Cui and Tafti [5].

3. Description of four louver geometries

Four louver geometries are considered in this paper (see Fig. 1): (1) periodic louver; the louver is assumed periodic in the spanwise direction with no tube. This simulation isolates any intrinsic three-dimensional effects brought about by secondary three-dimensional instabilities [10]; (2) straight louver; the angled louver extends all the way to the tube; this serves as a baseline case to study the effect of louver geometry transition; (3) louver with transition without landing; the angled louver directly transitions to the tube surface; (4) louver with transition and flat landing, which has been studied in detail by Cui and Tafti [5] at $Re = 1100$. Comparison of (3) and (4), highlights the role of the flat landing.

For all four geometries, the unit computational domain has a dimension of 1 (normalized by louver pitch L_p^*) in streamwise (x)-direction, fin pitch 1 (in this particular case, fin pitch F_p^* is same as L_p^*) in cross-stream (y)-direction, and 2.5 in spanwise (z)-direction along the fin height. Along the spanwise direction in geometry 4 (hereafter referred as *transition with landing*), the louver can be divided into three parts: angled louver (length, 1.75), transition part (length, 0.5), and flat landing (length, 0.25). A linear transition profile is prescribed between the angled louver and the flat landing with a small radii of curvature at the junction with the louver [11]. For geometry 3 (hereafter referred to as *transition without landing*), the angled louver part is extended to a length of 2.0, and the transition part is unchanged, but the flat landing between the transition and the tube surface is removed. Geometry 2, referred to as a *straight louver*, has a spanwise extent of 2.5. Finally, geometry 1 is referred to as a *periodic louver* and has a spanwise extent of 2.5.

In all cases, the thickness of the angled louver is 0.1 times the louver pitch with 25° louver angle. For the last three geometries, symmetry boundary conditions are imposed at a distance of 2.5 from the tube surface along the fin height, assuming that the flow is sufficiently removed from the extrinsic three-dimensional effects of the tube wall region and is nominally two-dimensional. This also assumes implicitly that the fin height is 5.0 louver pitches. For the periodic louver, periodic boundary conditions are implemented in the spanwise direction since the flow is homogeneous along this direction.

The computational domain surrounding each louver is resolved by $98 \times 98 \times 128$ computational cells in the x-, y- and z-directions, respectively for the transitioning geometries. For the periodic and straight louver, 96 computational cells are used in the z-direction along the fin height. A very fine, nearly orthogonal mesh, is used in the vicinity of the louver and tube surface, and in the transition region [5]. A posteriori extraction of the mean wall shear stress for $Re = 1100$ shows that the first grid point near the louver surface falls between 0.1 and 0.3 in local wall units based on the local shear stress. In the region with the largest shear stress (in the transition region), there are five grid points within 10 wall units normal to the surface, with the first at 0.3. Along the streamwise direction, the grid is nearly uniform with spacing of 5–7 wall units. Along the fin height or spanwise direction, the mesh is coarsest in the two-dimensional region of the geometry with the maximum spacing of 60 wall units and finest at the beginning and end of transition, and near the tube wall with spacing around 3 wall units. Spectral analyses show that the spatial and temporal resolution is fine enough to capture all the relevant scales in these calculations [5].

4. Results

In each of the calculations, a mean non-dimensional pressure gradient of unity is imposed in the streamwise direction to drive the flow. As the calculation proceeds, the flow rate, in response to the frictional and pressure drag losses in the calculation domain, adjusts to the mean pressure gradient and reaches a stationary (or steady state, in the case of low Reynolds number steady flow). Time signals of flow variables are recorded and a stationary flow is assumed when a near constant mean value or a quasi-periodic fluctuation in time is observed. Fig. 2 shows the temporal evolution of the spatially averaged Nusselt number for four louver geometries at a nominal Reynolds number of 1100. It is clear that all flows have adjusted to the mean pressure gradient and reached a statistically stationary state. Similar plots at nominal bulk Reynolds number of 600 and 300 also show that the flow has reached a stationary or steady state.

To characterize the heat transfer, we define a local instantaneous Nusselt number over the louver/tube surface based on the louver pitch as

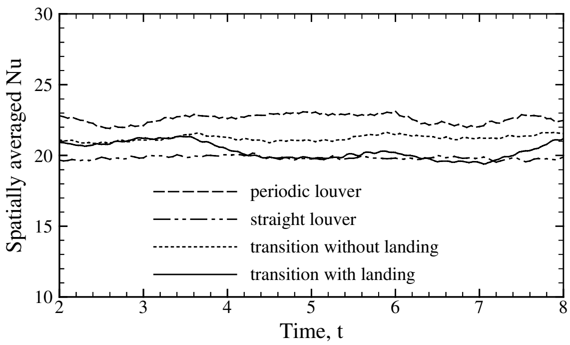


Fig. 2. Temporal evolution of the spatially averaged Nusselt number for four louver geometries at Reynolds number of 1100. All flows have adjusted to the mean pressure gradient and reached a statistically stationary state. Similar plots for louvers at Reynolds number of 600 and 300 also show that flow has reached a stationary state.

$$Nu = \frac{L_p^* q''^* / (T_s^* - T_{ref}^*)}{\kappa}$$

In terms of non-dimensional quantities the above can be re-written as

$$Nu = \frac{1}{\theta_s - \theta_{ref}}$$

where θ_s [2] is the modified non-dimensional surface temperature and θ_{ref} is the reference modified non-dimensional bulk temperature, which is defined as:

$$\theta_{ref} = \frac{\int\int |u|\theta \, dA_x}{\int\int |u| \, dA_x}$$

The surface-averaged Nusselt number is obtained by integration over the louver or tube surface as:

$$Nu = \frac{\int\int_\Omega d\Omega}{\int\int_\Omega (\theta_s - \theta_{ref}) \, d\Omega}$$

where Ω denotes the louver or tube surface. The Colburn j-factor as a measure of heat transfer is calculated as:

$$j = \frac{Nu}{RePr^{0.4}}$$

The Fanning friction coefficient is calculated as:

$$f = \frac{\Delta p^*}{\frac{1}{2}\rho V_c^{*2}} \frac{D_h^*}{4F_d^*} = \frac{D_h}{2} \frac{1}{V_c^2}$$

where D_h^* is the hydraulic diameter, $\Delta p^*/F_d^*$ is the prescribed pressure gradient across the calculation domain (unity non-dimensional value in present calculations), and V_c^* is the calculated maximum mean velocity.

[2] $T(x, y, z, t) = T_{in} + \gamma x + \theta(x, y, z, t)$, where γ is the mean temperature gradient.

4.1. General flow features

In the study of the louver with transition and flat landing [5], it is shown that flow on the angled louver portion is characterized by periodic spanwise vortex shedding at the Reynolds number of 1100. The spanwise vortices are nominally two-dimensional in nature with weak three-dimensionality across the fin height. The time signal at a location above the top louver surface exhibits a nearly periodic pattern, and the frequency spectrum shows a clear peak at 1.8 (non-dimensionalized by bulk velocity and louver pitch), which corresponds to the frequency of the spanwise vortex shedding. At this Reynolds number of 1100, all four louver geometries exhibit the same vortex shedding characteristic frequency. Although there is considerable geometry variation near the tube surface, its effects on the flow field on the louver away from the tube is minimal. Because of these similarities at the angled louver part, nearly identical flow and heat transfer behavior is expected for the four louver geometries. Any observable differences would come from the area near the tube surface.

At Reynolds number of 600, the flow unsteadiness becomes much weaker at the angled louver part. The time signals do not show a periodic pattern, and vortex shedding only occurs in an occasional manner, and there is no clear characteristic frequency. At Reynolds number of 300, the flow is completely steady and remains attached on the louver surface and there is no evidence of vortex shedding for all louver geometries. These results are in agreement with a previous two-dimensional investigation on the onset of instabilities for developing flow in a louver bank [4].

To facilitate our understanding of the unsteady nature of the flow and the associated vorticity dynamics, the $\nabla \boldsymbol{u}$ [12] vortex identification technique is used. This frame-invariant method identifies vortical structures as regions of large vorticity, where rotation dominates over strain to cause the rate-of-deformation tensor $\nabla \boldsymbol{u}$ (velocity gradient tensor) to have complex eigenvalues (one real and two conjugate complex eigenvalues). The complex eigenvalues imply that the local streamline pattern is closed or spiral, thus correctly eliminating near-wall shear layers. This methodology can also be separately applied in the x-, y-, or z-planes in order to identify streamwise, cross-flow, and spanwise vortices [10], respectively. The strength of the vortex is measured in terms of the imaginary part of the eigenvalue of the velocity gradient tensor and is denoted by λ_i. The strength of its three subsets, streamwise, cross-flow, and spanwise vortices is measured in terms of the imaginary part of the eigenvalue of the velocity gradient on the x-, y-, and z-planes, respectively, and is denoted by $\lambda_{i,x}$, $\lambda_{i,y}$, and $\lambda_{i,z}$, respectively.

Fig. 3(a)–(d) shows the volume-averaged vortical strength $\lambda_{i,x,y,z}$ distribution along the fin height at an arbitrary instant at Reynolds number of 1100. [3] Only the volumes with non-zero eigenvalues are included in the volume averaging. For the periodic case (Fig. 3(a)), the lines for streamwise ($\lambda_{i,x}$) and cross-flow ($\lambda_{i,y}$) vorticity are identically zero throughout the louver height. The only contribution to the total vorticity is from the spanwise vorticity ($\lambda_{i,z}$). Hence at $Re = 1100$, for the given louver geometry, the flow is strictly two-dimensional and intrinsic three-dimensional secondary instabilities have not developed. [4] For the straight louver (Fig. 3(b)), the spanwise vorticity dominates. However, there are small components of both streamwise ($\lambda_{i,x}$) and cross-stream ($\lambda_{i,y}$) vorticity present along the louver height. This implies that the three-dimensionality introduced by the presence of the tube wall permeates into the flow away from the wall and introduces weak three-dimensionality in a nominally two-dimensional flow. The spanwise vorticity ($\lambda_{i,z}$) is damped considerably by the viscous presence of the wall which is felt up to one louver pitch away from it, implying very thick boundary layers on the tube wall. Approaching the tube surface, there is a noticeable but slight increase for both streamwise ($\lambda_{i,x}$) and cross-flow ($\lambda_{i,y}$) vorticity as the spanwise and total vorticity decrease.

For the louver with transition and flat landing (Fig. 3(d)), on the angled louver, λ_i essentially maintains a constant value, with a dominant contribution from spanwise vorticity. However, in the transition region the flow is strongly three-dimensional. λ_i increases, with increasing contributions from streamwise and cross-stream vorticity, with a drop in contributions from spanwise vorticity. λ_i reaches a maximum in the center of the transition region and then decreases as the louver approaches the flat landing and the tube surface. The increase in the streamwise and cross-stream components of vorticity is related to the formation of an unsteady vortex jet under the bottom louver surface, which is described in detail in Cui and Tafti [5]. Not reflected in these plots, but related to the vortex jet, is the formation of a highly unsteady region of accelerated flow velocities on the top surface of the louver. For transition without landing (Fig. 3(c)), it is seen that the magnitude of coherent vorticity in the transition region is reduced. This is because, in the presence of the flat landing the fluid acceleration on the top surface and the vortex jet feed off the streamwise flow along the flat landing. In the absence of the flat landing, when the louver transitions directly to the tube surface, there is reduced access to

[3] To obtain the distribution, the volume averaging is performed in domains defined by decompositions used for parallel computation along the fin height.

[4] The nominally 2-D flow was perturbed by 3-D disturbances to seed any intrinsic three-dimensional secondary instabilities, but the perturbations were not self-sustaining.

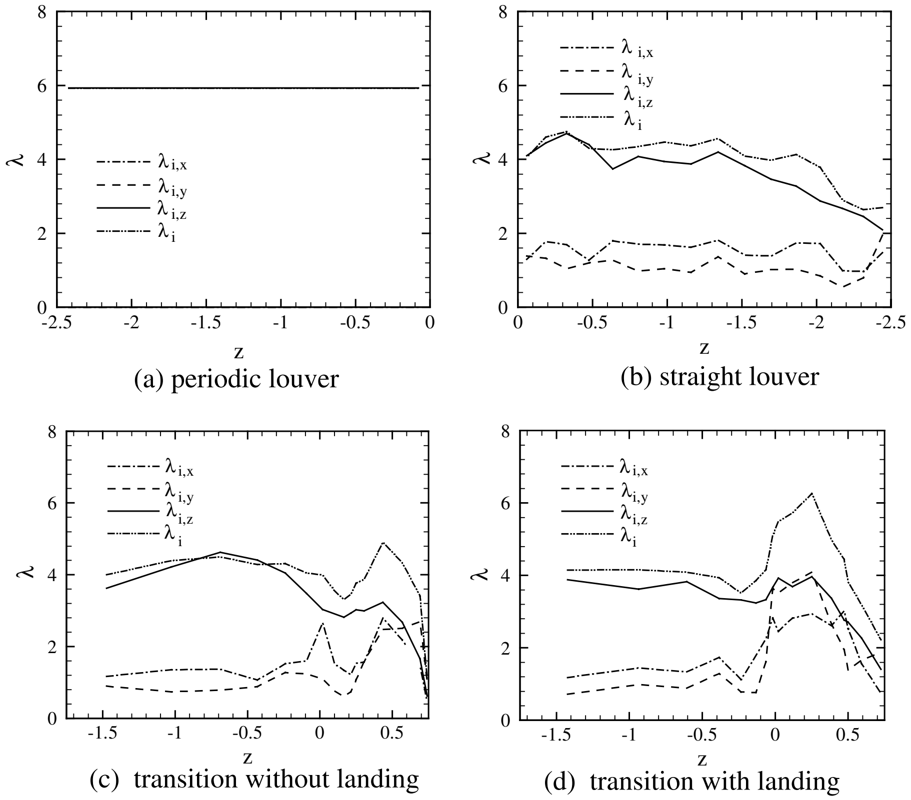


Fig. 3. Instantaneous volume-averaged vortical strength distribution along the fin height at Reynolds number of 1100 at an arbitrary instant for (a) periodic louver; (b) straight louver; (c) transition without landing; (d) transition with landing.

fluid mass, which results in the weakening of these flow structures.

4.2. Pressure and friction drag on louver and tube

Fig. 4(a)–(d) plots the *fractional* variation of mean form and friction drag per unit length along the fin height or spanwise direction at a nominal $Re = 1100$. [5] For all four geometries, at the angled louver portion, the form drag dominates the friction drag and is almost unchanged throughout the angled louver. This is best exemplified by the two-dimensional flow over the periodic louver in Fig. 4(a), in which the form drag contributes 80% to the overall pressure loss. For transition with landing (Fig. 4(d)), and transition without landing (Fig. 4(c)), the magnitude of pressure and friction drag is similar at the angled louver part. For the straight louver, although the form drag loss is four times the friction losses away from the tube surface, which is similar to other geometries, the contribution to total losses is dominated by the presence of the tube. Both frictional and form losses increase substantially in the vicinity of the tube surface because of viscous effects. As the flow approaches the tube, it slows down, and the flow angle reduces substantially, which leads to the increased contribution to form drag. For the transitioning geometries in Fig. 4(c) and (d), the trends are completely different. In the transition region, the form drag increases slightly and eventually vanishes at the flat landing. On the other hand, friction drag increases sharply in the transition region and reaches its largest value near the flat landing due to the accelerated high velocity boundary layer in that region before decreasing again on the flat landing. Similar, albeit weaker, distributions at the transition region are found for the transitioning geometry without the landing (Fig. 4(c)).

Fig. 5(a)–(c) plots the mean drag force distribution as a function of the fin height for the transition with landing geometry at three Reynolds numbers: 1100, 600, and 300. As the Reynolds number decreases, the contribution of pressure drag decreases while that of friction

[5] The form and friction drag are plotted as a fraction of the total losses. Since the mean pressure gradient is fixed at unity, the integrated area under the curves should add up to approximately (barring tube frictional losses) 2.5, the pressure loss expressed as a force on the computational domain.

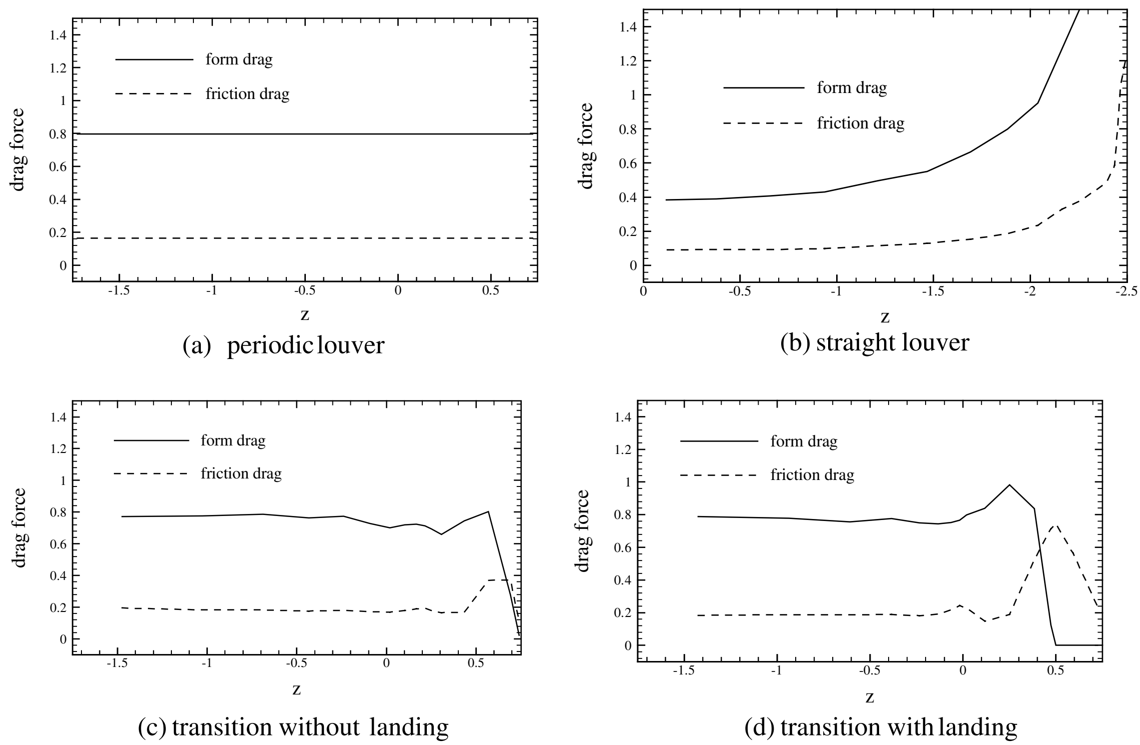


Fig. 4. Mean drag force distribution along the fin height as a fraction of the total losses: (a) periodic louver; (b) straight louver; (c) transition without landing; (d) transition with landing at Reynolds number of 1100.

drag increases at the angled louver part. At a nominal Reynolds number of 300, the two drag forces are nearly equal. The distribution at the transition region and flat landing follow the same trend as the Reynolds number decreases. Overall the changes in Reynolds number do not change the salient features of the drag distribution throughout the louver. This is also true for the other three louver geometries.

Fig. 6 plots the fractional contribution of friction losses on the tube surface to the total losses. For all three geometries, the contribution of the tube to overall losses is less than 8% of the total. The louvers with transition exhibit a higher contribution because of the increased shear stress on the tube surface as a result of the unsteady accelerating boundary layer in the vicinity.

4.3. Time-averaged heat transfer coefficient

Fig. 7(a)–(d) plots the time mean thermal field (modified temperature, θ) on the top surface of the louver. Because the heat flux is fixed on the louver and tube surface, a high surface temperature implies low heat transfer. In all cases, at a nominal $Re = 1100$, the shear layer at the leading edge of the louver separates and sheds vortices. Very near the leading edge, the heat transfer coefficients are high, but decrease in the recirculation zone which forms downstream of the leading edge. In the reattachment region, at half the louver length, the vorticity generated by the separated shear layer increases the heat transfer coefficient by increasing mixing. For the periodic geometry, in the absence of any extrinsic three-dimensionality, the surface temperature does not show any variations in the z-direction. For the straight louver, the thick thermal boundary layer on the tube surface dominates the temperature distribution on the top surface. For transition with landing, in the transition region, the low temperature/high heat transfer region on the top surface near the flat landing is a result of the unsteady accelerating boundary layer on the louver surface. Similar trends are observed for the transitioning louver with no landing.

Temperature contours on the lower surface are shown in Fig. 8(a)–(d). For the periodic louver the heat transfer coefficient is a maximum at the leading edge and decreases thereafter till near the trailing edge where it increases again. A high temperature/low heat transfer region formed in the transition region in Fig. 8(c) and (d) results from the presence of the vortex jet. The jet is detached from the louver surface and a stagnant recirculating region is formed underneath the jet. Similar to the top surface, a thick boundary layer near the tube surface exists for the straight louver on the bottom

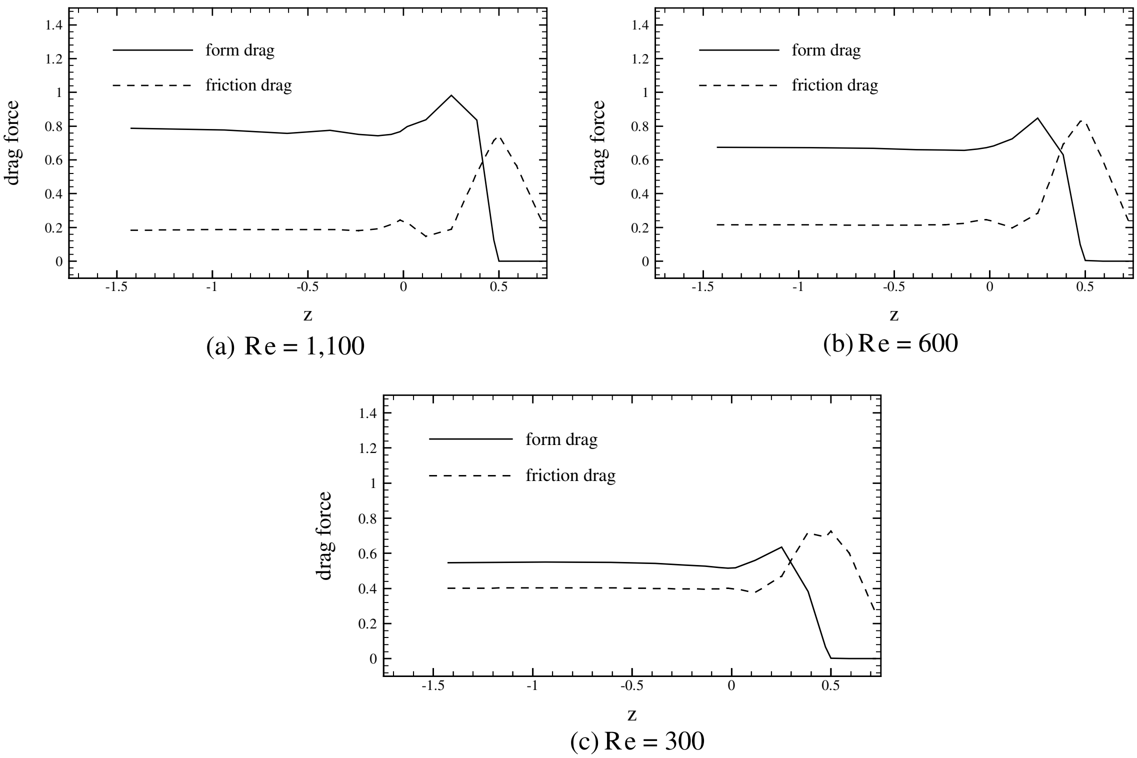


Fig. 5. Mean drag force distribution along the fin height as a fraction of the total losses for transition with landing at Reynolds number of (a) 1100; (b) 600; (c) 300.

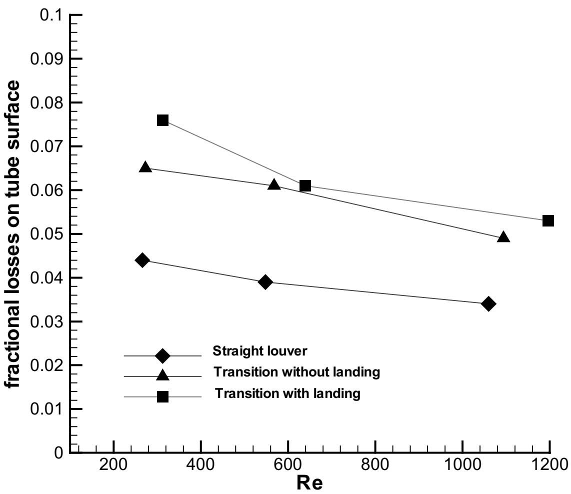


Fig. 6. Fractional contribution of friction on tube surface to overall pressure loss.

surface. Comparing temperature contours on the top and bottom louver surfaces for both the transitional geometries in the vicinity of the tube clearly shows the positive effect of the accelerating boundary layer on the top surface. Temperature contours have lower values in the immediate vicinity of the tube on the top louver surface than on the bottom surface.

In Fig. 9(a) and (b), the average (time and spatial) Nusselt number on the louver, and tube surface is plotted separately. In general, the first-order effect of the angled louver transitioning to 0° and a flat landing is to reduce the heat transfer coefficient. Also, the presence of the tube surface further reduces the heat transfer coefficient on the louver surface because of the presence of thick thermal boundary layers at the junction between fin and tube. These effects can either be countered or reinforced further by other non-linear effects as observed (unsteady boundary layer acceleration on louver top surface and vortex jet on bottom surface, separation) in the current study. The unsteady boundary layer acceleration on the top surface has a positive impact on louver heat transfer, whereas the formation of the vortex jet at the bottom has a neutral to negative impact. The results in Fig. 9(a) are consistent with these observations. The periodic louver exhibits the highest heat transfer coefficient, whereas the straight louver and the transitioning louver with a flat landing exhibit heat transfer coefficients which are between 15% and 25% lower. The transitioning louver without a landing lies between the two extremes and is between 6% and 15% lower. These results indicate that to maintain a high heat

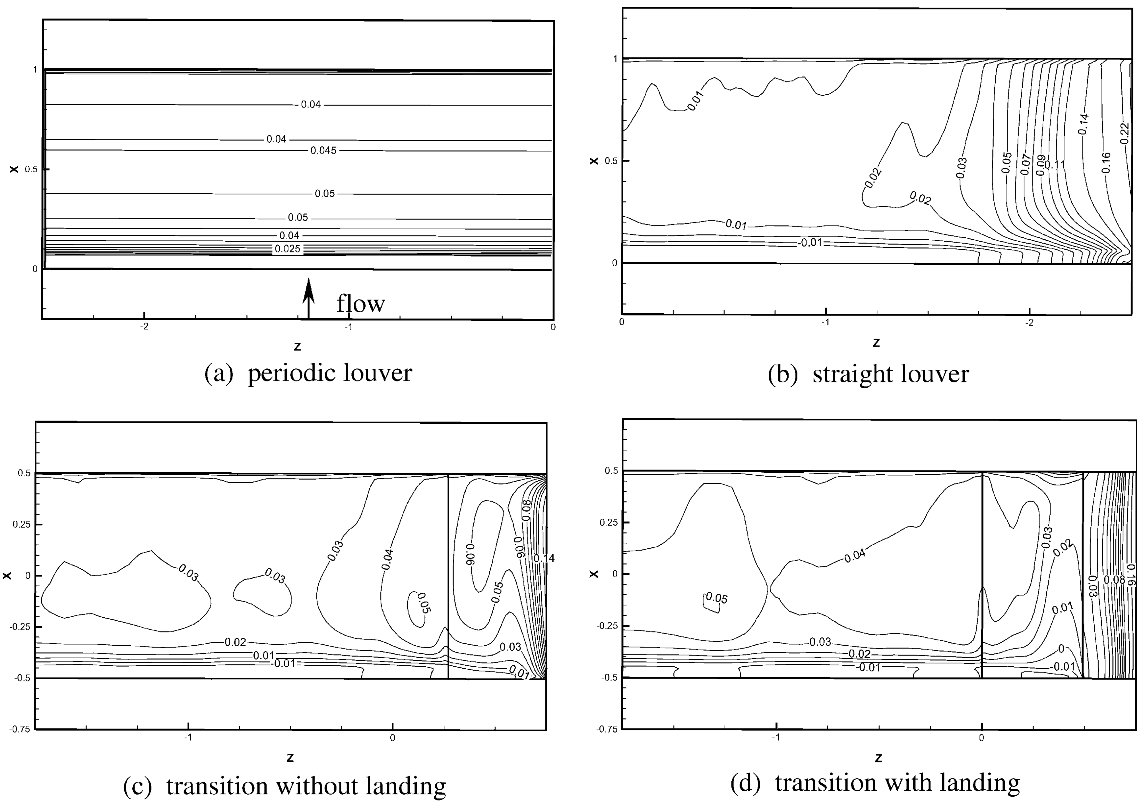


Fig. 7. Mean thermal field distribution on the louver top surface at Reynolds number of 1100 for (a) periodic; (b) straight louver; (c) transition without landing; (d) transition with landing.

transfer coefficient on a transitioning louver, the flat landing should be as small as physically possible.

On the other hand, the enhancement provided by the transitioning louver with a flat landing on the tube surface is quite strong. The tube Nusselt number is lowest for the straight louver because there is nothing that can break the thick thermal boundary layer that forms at the fin–tube junction. With the transitional louver, the unsteady boundary layer acceleration on the top surface and to some extent the vortex jet under the louver, help to perturb and thin the thermal boundary layer on the tube and increase the heat transfer coefficient. Without the flat landing, the unsteady nature of the flow is considerably weakened as noted in Fig. 3, and the augmentation on the tube surface is not as high. Transition with flat landing provides an augmentation of over a 100% over a straight louver, whereas with no landing, the augmentation is reduced to between 30% and 40%.

4.4. Overall friction and heat transfer coefficient for flat tube louvered heat exchanger

In this section, the overall heat transfer and friction factors for an equivalent duct of aspect ratio 5, bounded by louvered fins and the tube surface are presented. These are compared to theoretical flow results for fully developed laminar flow in ducts. Fig. 10(a) compares the calculated friction coefficient (f), and Fig. 10(b) plots the equivalent Nusselt number (Nu_{D_h}) versus Re_{D_h}.

The friction coefficient increases by a factor between 4 and 9 when compared to a fully developed laminar flow in a duct of aspect ratio 5. On the other hand the Nusselt number is augmented by factors varying from 2 to 3.5. The tube surface results in approximately a 25–30% reduction in the overall Nusselt number. Hence, for small tube pitches and large fin pitches, tube surface heat transfer becomes critical to the performance of the heat exchanger. In fact, in spite of the louver heat transfer being highest for the geometry without a landing, the overall Nusselt number is highest for the geometry with a flat landing because of a larger heat transfer coefficient on the tube surface. Between the three geometries, the louver with transition and flat landing exhibits the lowest friction coefficient, whereas the friction coefficient is highest for the straight louver. The result goes against Reynold's analogy, but is consistent with the fact that losses are dominated by louver form drag, which is reduced substantially in the transition region and vanishes

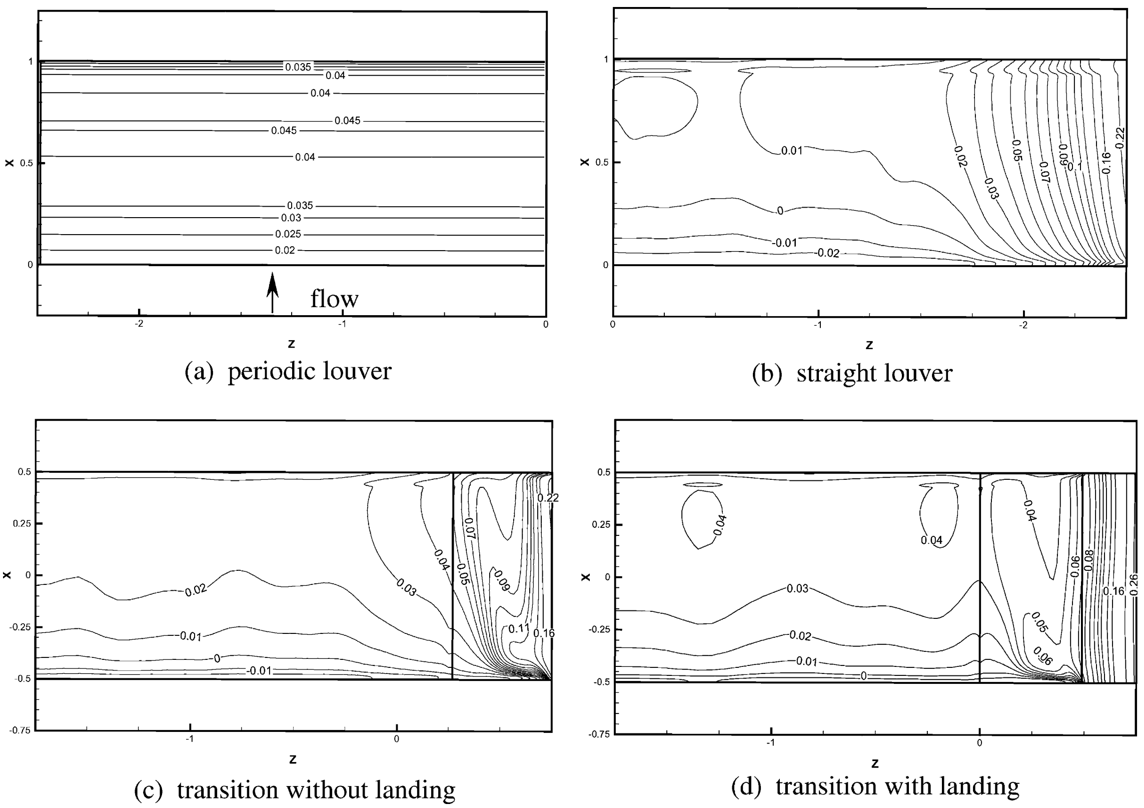


Fig. 8. Mean thermal field distribution on the louver bottom surface at Reynolds number of 1100 for (a) periodic; (b) straight louver; (c) transition without landing; (d) transition with landing.

at the flat landing. This, together with the heat transfer augmentation provided on the tube surface with a minor increase in skin friction, is responsible for the above result.

Finally we provide a comparison between the calculated results and previous experimental work. Both the calculated friction coefficient and Colburn j-factor are compared to relevant correlations available in the literature. This is provided to validate that in spite of the simplifying assumptions inherent in computer models, and the geometrical imperfections in real exchanger cores on which experiments are performed, physically consistent models of a subsystem of the full heat exchanger are capable of providing realistic performance measures of the full system. The friction coefficient is compared to the correlation of Chang et al. [13], (referred to as CHLW) and the j-factor to the correlation by Chang and Wang [14] (referred to as CH), and also to that of Sunden and Svantesson [15] (referred to as SS). The SS correlation is specific to flat tube arrangements with corrugated louvers in rectangular channels, whereas both the f- and j-correlations are more general in nature and include a wide range of multilouvered geometries [13]. The following geometrical values are used in the correlations: fin pitch $= 1$ (all lengths normalized by louver pitch), $\theta = 25°$, fin thickness $=$ 0.1, fin height $=$ tube pitch $= 5$, tube depth $= 15$, louver length $= 4.5$, major tube diameter $= 1$, [6] and louver height as $0.5 \sin\theta$.

Fig. 11(a) and (b) plot the f- and j-factor. Also plotted are upper and lower bounds of the experimental data from which the CHLW and CW correlations are derived. The calculated f-factor for all three cases falls within the upper bounds of the experimental data. We also note that the current calculations are relevant to the type C geometry in Chang and Wang [14], which generally exhibits a higher friction coefficient than the other types of multilouvered geometries. Similarly, the calculated j-factors for the transitioning louvers fall well within the experimental bounds of the CW correlation but are lower than the SS correlation. In both compar-

[6] Tube depth is used in the same context as flow depth. In our calculations, the flow depth is infinity. So a typical value of 15 is used. Similarly, the calculations do not simulate flow around the tube, so a value of 1.0 is assumed as the major tube diameter. In any case, for flat tubes, the contribution to pressure loss from the frontal area of the tube is negligible.

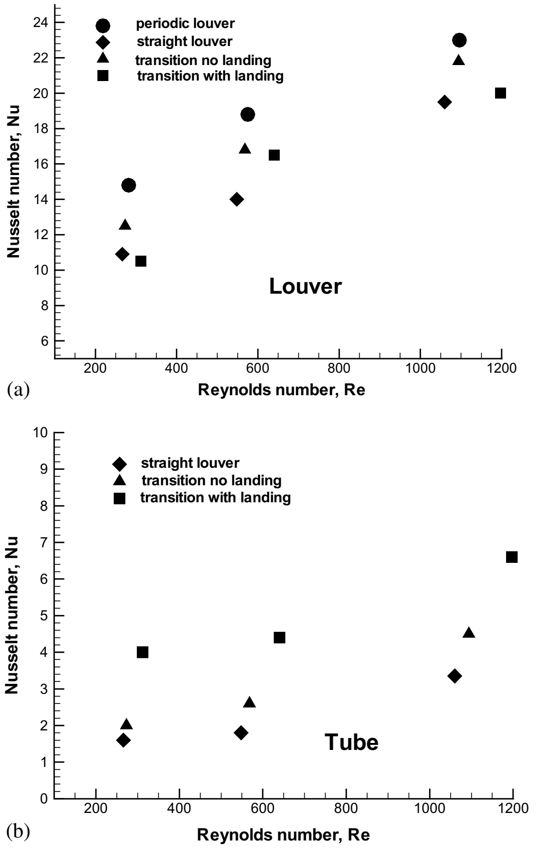


Fig. 9. Average Nusselt number versus the Reynolds number. (a) On the louver surface; (b) on the tube surface.

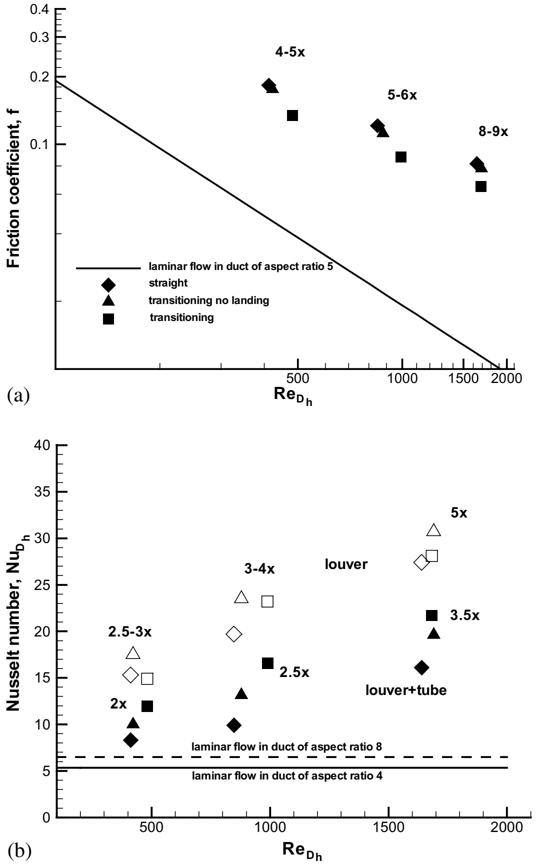


Fig. 10. (a) Friction coefficient, f for equivalent louvered duct; (b) Nusselt number for equivalent louvered duct. Diamond: straight louver; delta: transition no landing; square, transition with landing. Empty symbols: Nusselt number based on louver surface; filled symbols: Nusselt number based on louver and tube surface.

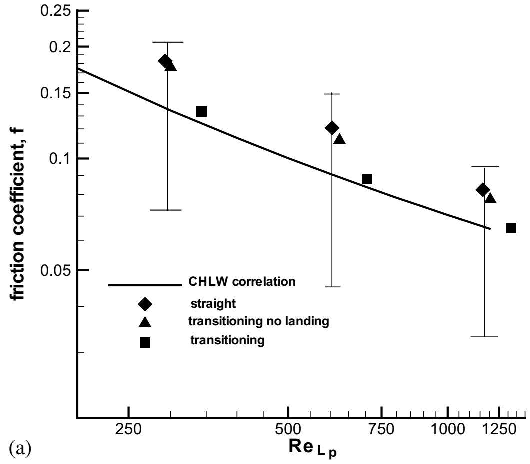
isons we find that the more realistic louver with a flat landing agrees best with the correlations.

5. Conclusions

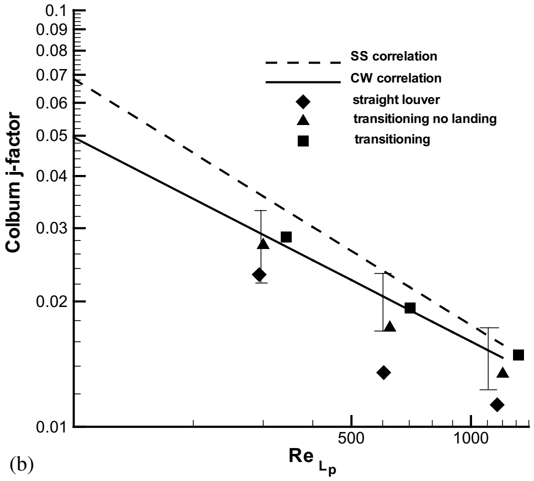
In this paper, we study the flow and heat transfer in four three-dimensional geometries (Fig. 1) of a flat tube corrugated multilouvered fins at three nominal Reynolds numbers: 1100, 600, and 300. The four geometries vary in the configuration of the fin at the junction with the tube face. They range from completely neglecting the effect of the tube surface to including the realistic transition of the angled louver into a flat landing adjoining the tube face. The objective is to study the impact of this region on louver as well as tube heat transfer coefficients and to determine whether modeling a small subsystem (a single louver) is representative of the performance of the full heat exchanger core.

The results show that away from the tube surface, the flow is nominally two-dimensional with weak three-dimensionality. For louvers that flatten out into a flat landing, conditions are created for highly three-dimensional and unsteady flow phenomena. Flow in the transition region is characterized by unsteady boundary layer acceleration on the louver top surface and a vortex jet under the louver bottom surface. The flow acceleration has a large impact on louver heat transfer locally. However, its impact is minimal on the averaged heat transfer coefficient over the whole louver. It is concluded that for best louver heat transfer performance, the transition and flat landing should be kept as small as possible. On the other hand, the boundary layer acceleration generated by the transitioning louver with a flat landing has a large impact on tube heat transfer and increases it by over 100% over a straight louver which does not transition to the tube surface.

It is found that the low heat transfer on the tube surface decreases the overall heat transfer capacity of the heat exchanger between 25% and 30%. Hence, augmenting heat transfer on the tube surface would have



(a)



(b)

Fig. 11. Comparison of calculated f- and j-factors with available correlations. Vertical lines establish limits of experimental data from which the CHLW and CH correlations are constructed.

large payoffs in small tube pitch, large fin pitch, multilouvered geometries. On the other hand, there is a minimal contribution (<8%) of tube frictional losses to total losses.

The agreement of calculated results with correlations derived from full core experiments validates that realistic three-dimensional computational modeling of a small subsystem is a viable and effective tool in generating performance data for heat exchangers.

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