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EFFECT OF FIN THICKNESS ON FLOW AND HEAT TRANSFER IN MULTI-LOUVERED FINS

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ABSTRACT

High-resolution time-dependent numerical simulations are used to investigate the effect of thickness ratio on fluid flow and heat transfer performance in multi-louvered fins. Results for three fin thickness ratios, two louver angles, and a fin pitch to louver pitch ratio of one are reported for Reynolds number ranging from 50 to 1200. Thickness ratio is found to have a significant effect on flow efficiency, especially in geometries with small louver angles. For small louver angles, increasing thickness to louver pitch ratio from 0.05 to 0.15, decreases the flow efficiency by as much as 35-40%. As expected, increasing thickness ratio increases total pressure drop, most of which results from an increase in form drag. Heat transfer coefficient, on the other hand, is not influenced strongly by the thickness ratio. The increase in flow acceleration and local Reynolds number with increase in thickness ratio, on one hand, is offset by low flow efficiencies and recirculation zones on the other. As a consequence, some heat transfer degradation is found at low Reynolds numbers, however the degradation diminishes as the Reynolds number increases beyond 300. In general, larger thickness ratios lead to a lower ratio of *j/f*.

INTRODUCTION

Multi-louvered fins have been used in a variety of automotive, residential air-conditioning and refrigeration applications for the purpose of air-side heat transfer enhancement. In the past two decades, many experimental studies have attempted to understand the flow phenomena and heat transfer performance in multi-louvered fins. A variety of correlations have been developed from previous experimental studies for *j* and *f* factors as a function of Reynolds number and geometrical parameters (such as fin pitch, louver pitch and louver angle). However the main focus of earlier investigations [1-4] has been on the effect of louver angle, fin pitch and louver pitch. Very few studies in the literature have focused on the fin thickness. Chang et al. [5] included the thickness ratio effect when developing a general correlation for *j* factor. The correlation shows a weak dependency of *j* factor on the fin thickness ratio (δ_f / L_P) as shown in the following equation:

$$j = \mathrm{Re}_{L_p}^{-0.49} \left(\frac{\theta}{90}\right)^{0.27} \left(\frac{F_p}{L_p}\right)^{-0.14} \left(\frac{F_l}{L_p}\right)^{-0.29} \left(\frac{T_d}{L_p}\right)^{-0.23} \left(\frac{L_l}{L_p}\right)^{0.68} \left(\frac{T_p}{L_p}\right)^{-0.28} \left(\frac{\delta_f}{L_p}\right)^{-0.05}$$

They showed that their correlation represented 97.14% of all the experimental data in [1-5] with a deviation of 25%, and 87.76% of the data with 15% deviation. It is noticed that the *j* factor in the above correlation decreases very slightly with an increase of thickness ratio. As fin thickness ratio increases from 0.05 to 0.15, only a 6% difference is found in *j* factor. In their later work [6], a general correlation was given for friction factor. It shows a very strong dependence of *f* on fin thickness ratio. For the Reynolds number (based on louver pitch) less than 150, the only multiplier containing fin thickness in their correlation is given as:

$$f_1 = D_h^{-3.01} (\log_e ((b / F_P)^{0.48} + 0.9)))^{-1.435},$$

where f is expressed as:

$$f = F(\mathrm{Re}_{L_P}, L_P, F_P, D_m, T_P, T_h, \theta, L_l) f_1 .$$

In the Reynolds number range of 150 to 5000 the multiplier is:

$$f_2 = D_h^{-2.966} (\log_e ((b / F_P)^{0.5} + 0.9)))^{-0.527},$$

where f is expressed as:

$$f = F(\mathrm{Re}_{L_P}, L_P, F_P, D_m, T_P, T_h, \theta, L_l) f_2 .$$

Here, $b, L_P, F_P, T_P, D_m, \theta, L_l, D_h$ are fin thickness, louver pitch, fin pitch, tube pitch, major tube diameter, louver angle, louver length and hydraulic diameter, respectively, and $T_h = T_P - D_m$. In the above correlations, the friction factor f increases with an increase in thickness ratio at higher Reynolds numbers ($150 < \mathrm{Re} < 5000$). For a fin pitch ratio 1.0, as the thickness ratio increases from 0.05 to 0.15, friction factor f increases by nearly 50%. At low Reynolds numbers

($\mathrm{Re} < 150$), however, as thickness ratio increases from 0.05 to 0.15, f reduces by 20%.

Suga and Aoki [7] numerically studied the effect of thickness on heat transfer performance and pressure drop by using a laminar steady flow model. For multi-louvered fins with a fin pitch ratio of 0.73, a louver angle of 26 degree, and a Reynolds number of 192, they found that the Nusselt number increased slightly with increase of the thickness ratio from 0.04 to 0.08 whereas more than a 50% increase was found in pressure drop. Their findings were similar to the experimental correlation of Chang et al. [6].

Our objective in this paper is to study the effect of fin thickness on flow efficiency and subsequently on heat transfer and friction characteristics. We study three fin thickness ratios, for two louver angles and a fin pitch ratio of one.

NOMENCLATURE

b non-dimensional fin thickness (b^* / L_P^*),

D_h non-dimensional hydraulic diameter,

f friction factor, $f = \dfrac{2\Delta p_{fin} D_h}{4 F_d V_c^{\,2}}$,

F_p non-dimensional fin pitch (F_p^* / L_p^*),

F_d non-dimensional flow depth,

j Colburn factor, $j = \dfrac{< Nu^2 >_{fin}}{\mathrm{Re}_{L_P} P_r^{0.4}}$,

k thermal conductivity,

L_p^* dimensional louver pitch (characteristic length scale), $L_p = 1$,

Nu^1 non-dimensional heat flux, $Nu^1 = q = -\dfrac{\partial T}{\partial n}$,

Nu^2 non-dimensional heat transfer coefficient,

$$Nu^2 = \frac{h^* L_p^*}{k} = \frac{-\partial T / \partial n}{(1 - T_{ref})},$$

Pr Prandtl number,

Δp_{fin} non-dimensional total pressure drop,

q non-dimensional heat flux,

Re Reynolds number,

$$\mathrm{Re}_{in} = u_{in}^* L_p^* / \nu, \qquad \mathrm{Re}_{L_P} = V_c^* L_p^* / \nu,$$

T temperature,

u, v non-dimensional Cartesian velocity in x-, and y-direction, respectively,

u_{in}^* dimensional inlet velocity (characteristic velocity scale),

V_c Average velocity at minimum cross-sectional area

Greek symbols

θ degrees, louver angle,

η flow efficiency,

α degrees, flow angle,

ν kinematic viscosity

Superscripts

$*$ dimensional quantities,

Subscripts

fin based on fin,

in based on inlet

NUMERICAL METHOD AND COMPUTATIONAL GEOMETRY

The governing equations for momentum and energy conservation are solved in a general boundary conforming coordinate system. They are discretized with a conservative finite-volume formulation on a non-staggered mesh. For convenience, the governing equations are non-dimensionalized by a characteristic length given by the louver pitch L_P^*, a characteristic velocity scale given by the inlet velocity to the array (u_{in}^*) and a temperature scale given by ($T_{fin}^* - T_{in}^*$), where T_{fin}^* is the specified fin surface temperature. The non-dimensionalization results in a characteristic Reynolds number, $\mathrm{Re} = \mathrm{Re}_{in} = u_{in}^* L_P^* / \nu$, with Dirichlet boundary conditions $u_{in} = 1, T_{in} = 0$ at the entrance to the computational domain. Due to the recovering nature of the flow at the array exit, convective outflow boundary conditions are applied at this boundary. Periodic boundary conditions are applied in the transverse direction. The Prandtl number is fixed at 0.7 for air. At the fin surface, no slip, no penetration boundary conditions for the velocity field, and $T_{fin} = 1$ for the temperature field are applied. Details about the time-integration algorithm, treatment of boundary and louver surface conditions, and validation of the computer program can be found in Tafti et al. [8].

The configuration used in these calculations consists of an entrance and exit louver with four louvers on either side of the center or redirection louver. Figure 1 shows a louvered fin geometry and the corresponding computational domain which is resolved by 15 computational blocks, one for each louver, two each for the entrance, exit and redirection louver. The exit domain extends approximately 5.5 non-dimensional units downstream of the array. Each block is resolved by 96x96 finite-volume cells (a total of 138,240 cells). A grid independency study was performed at a resolution of 128x128 cells in each block (a total of 245,760 cells). As shown in Figure 2, both time-averaged non-dimensional heat flux and Nusselt number calculated on the 96x96 grid were within one percent of the fine grid calculation for $F_p = 1.5$ and $Re_{in} = 1000$. All results reported here are for a mesh resolution of 96x96 cells per block. For cases in which the flow is unsteady, time-averaged values are presented.

Table 1 summaries the geometrical parameters studied in this paper. Three thickness ratios (0.05, 0.1 and 0.15) are chosen, and for each thickness ratio, two louver angles (20 and 30 degrees) are investigated. The Reynolds number based on louver pitch varies from 50 to 1200.

THICKNESS EFFECTS ON FLOW EFFICIENCY

An important parameter used in describing the efficacy of louvers to dissipate or absorb heat, is the predominant flow direction. Beauvais [9] was the first to conduct flow visualization experiments on a louvered fin array. He demonstrated that louvers, rather than acting as surface roughness elements that enhanced heat transfer performance, acted to realign the airflow in the direction parallel to themselves. Davenport [10] performed flow visualization experiments identical to those of Beauvais and demonstrated two flow regimes, duct directed flow and louver directed flow. At low Reynolds number, the flow moves directly through the channel between two fins because of the larger hydraulic resistance brought about by thick boundary layers developing on louver surfaces. At higher Reynolds number, the flow shifted to a direction parallel to the louvers. Flow efficiency, which is a ratio of the flow angle to the louver angle, is used to describe the percentage of the fluid flowing along the louver direction. A 100% flow efficiency represents ideal louver directed flow while 0% represents complete duct directed flow. Generally, high flow efficiency is favorable for heat transfer performance.

Experimental measurements use dye injection at the inlet to the louver bank and flow efficiency is defined as the ratio of actual vertical distance (*N*) traveled by the dye to the ideal distance *(D*) based on louver angle as follows:

$$\eta_{\exp} = \frac{N}{D} = \frac{\tan(\alpha)}{\tan(\theta)} \approx \frac{\alpha}{\theta} .$$

Webb and Trauger [11] experimentally studied flow structure in multi-louvered fin geometries for six fin pitch ratios (0.7 to 1.5), one thickness ratio (0.0423), and two louver angles (20 and 30 degrees). Reynolds number (based on louver pitch) ranged from 400 to 4000. Their results showed that flow efficiency increased with an increase in Reynolds number up to a critical Reynolds number, which is only dependant on the louver angle and is given by

$$\mathrm{Re}_{w,c} = 828\left(\frac{\theta}{90}\right)^{-0.34} \tag{1}$$

For Reynolds numbers less than the critical value, flow efficiency is dependent on and increases with Reynolds number and louver angle, and decreases with fin pitch ratio.

$$\eta_w = 0.091\,\mathrm{Re}^{\;0.39}\left(\frac{L_p}{F_p}\right)^{0.44}\left(\frac{\theta}{90}\right)^{0.3} \tag{2}$$

Beyond the critical value, flow efficiency is only affected by fin pitch ratio as follows:

$$\eta'_w = 0.95\left(\frac{L_p}{F_p}\right)^{0.23} \tag{3}$$

Equations (2) and (3) are not consistent at the critical Reynolds number, hence equation (2) was modified by Sahnoun [12] to the form

$$\eta_S = \eta'_w - 3.717\times10^{-5}\left[828(\frac{2\theta}{\pi})^{-0.34} - \mathrm{Re}\right]^{1.1}\left(\frac{L_p}{F_p}\right)^{-1.35}\left(\frac{2\theta}{\pi}\right)^{-0.61}$$

Achaichia [2] used a numerical method to model the flow through a simplified two-dimensional louver array**.** In their numerical calculations, the flow efficiency is obtained by calculating the flow angle at the computational domain boundaries surrounding the louver. The louvers are assumed to be infinitesimally thin, and the flow to be fully developed. From their numerical simulation results, in his later paper [13], the following correlation for flow efficiency was given:

$$\eta_A = (0.936 - 243/\mathrm{Re} - 1.76\frac{F_p}{L_p} + 0.995\theta)/\theta \tag{4}$$

In 1996, Bellows [14] conducted experiments on the effect of fin pitch ratio and louver angle on flow efficiency. He modified Achaichia's correlation to account for developing flow effects and found that the absolute percent deviation was reduced from 57% to 7% based on his experimental results. His correlation is:

$$\eta_B = (-5 - 300/\mathrm{Re} - 10\frac{F_p}{L_p} + 1.34\theta)/\theta \tag{5}$$

The above experimental and numerical results have confirmed that, first, flow efficiency is a function of Reynolds number and geometry at low and intermediate Reynolds number. As Reynolds number increases, the flow undergoes a transition from duct directed flow (low efficiency) to louver directed flow (high efficiency). Second, there exists a critical Reynolds number beyond which the flow efficiency is a constant, which depends only on geometrical parameters. Third, flow efficiency is affected significantly by fin pitch ratio. Large fin pitch ratio leads to low flow efficiency.

It is also evident that none of the correlations have taken the effect of fin thickness into consideration. In fact, the fin thickness ratios are different in each study. The thickness ratio in Achaichia's numerical calculations was zero, in Bellows' experiments the ratio varied from 0.089 to 0.106, while in Webb's experiments it was fixed at 0.0423.

With that in mind, Figure 3 shows the effect of fin thickness ratios on flow efficiencies for two louver angles (20 and 30 degrees). The flow efficiency for the whole fin is calculated as α_{avg}/θ, where θ is the louver angle and α_{avg} is the average flow angle based on louvers 1-8 (the entrance, redirection, and exit louvers are not included). The flow angle for each louver is calculated based on $\alpha = \tan^{-1}\dfrac{\int v dx / L_P}{\int u dy / F_P}$.

The numerator is calculated on the top, and the denominator on the left face of the block surrounding each louver. The results, which include developing flow effects, show that thickness ratio has a large effect on the flow efficiency. Thicker louvers lead to lower flow efficiency for both louver angles. However, the effect of thickness ratio is much stronger for the smaller louver angle of 20 degrees. At low Reynolds number ($\mathrm{Re}_{in} = 50$), more than a 55% increment in flow efficiency is found when the thickness ratio is reduced from 0.15 to 0.05 for 20 degree louvered fins. The increase is 13% in the 30 degree geometry.

Two intrinsic length scales are thought to be very important in determining flow efficiency in louvered fins as shown in Figure 4. One is the distance between two adjacent louvers in the same fin defined as:

$$d_L = L_P \sin(\theta) - b\,. \tag{6}$$

A larger gap, d_L, translates into lower hydraulic resistance along the louver direction. This situation is favorable for the bulk of the fluid to flow in the louver direction (louvered directed flow), and as a result a higher flow efficiency is obtained. The other important length scale is the gap between two adjacent fins defined as:

$$d_F = F_P - L_P \sin(\theta) - b\cos(\theta) \tag{7}$$

Contrary to the louver gap, a larger fin gap promotes conditions favorable for fluid moving along the streamwise direction (duct directed flow). The ratio of the above two distances, $d = d_L / d_F = (L_P \sin(\theta) - b)/(F_P - L_P \sin(\theta) - b\cos(\theta))$, should be considered to be, at least partially if not fully, responsible for the mean flow direction. The above ratio shows a clear dependence of flow efficiency on the fin pitch ratio and louver angle, consistent with previous results. In addition it also has a dependence on fin thickness. The ratio, d, as well as flow efficiency is larger for a smaller fin pitch ratio and larger louver angle. The thickness effect on this ratio is not as obvious as that of fin pitch ratio and louver angle. As the thickness increases, both of the gaps decrease, however the rate at which the ratio d varies with thickness is dependent on the fin pitch and louver angle. The variation of the ratio d versus thickness is plotted in Figure 5 for 4 louver angles, 33, 30, 20 and 15 degrees. Besides showing a smaller ratio (lower flow efficiency) for a smaller louver angle, more importantly, it shows the thickness ratio effect and its dependency on louver angle. For large louver angles, the thickness ratio has a negligible effect on the flow efficiency, whereas geometries with small louver angles, exhibit a much higher sensitivity to thickness ratio. In general, the flow efficiency decreases with an increase in thickness ratio. The dependencies and trends shown in Figure 5 are consistent with the trends predicted by the numerical results.

In Fig. 3, we also find that the flow efficiency actually decreases at high Reynolds numbers for the thicker fins (b=0.15), and then recovers as the Reynolds increases further. The trend is much stronger for a louver angle of 20 degrees. The reason for this phenomenon is that when the fin thickness ratio increases, the gap between adjacent louvers is reduced. The percentage reduction of the gap is larger for the smaller louver angle geometry. As the Reynolds number increases, recirculation zones, which are induced by the thicker louvers, grow and block the flow path between two louvers. The blockage is more severe for small louver angles, and hence the flow efficiency drops. Figure 6 (a) and (b) show the velocities and streamlines for louver 4 at two Reynolds numbers, 500 and 1000 respectively, for a louver angle of 20 degrees. The large recirculation zones at Re_{in} =1000, reduce the flow angles (Fig. 6c) and hence the flow efficiency. Once the Reynolds number increases beyond its critical value when flow oscillations induce vortex shedding and increased mixing, the recirculation zones (in the mean) reduce in size and the flow efficiency starts a recovery.

Figure 7(a, b) plots the comparison of current results with previous correlations [12,13,14] for flow efficiency. The correlations do not explicitly include the effect of thickness. Bellows' correlation was based on his experiments in which the thickness ratio was varied from 0.89 to 1.09, while Sahnoun's correlation was based on his experiments with a fixed thickness of 0.043. The thickness ratio was zero in Achaichia's numerical simulations. As expected, with a zero fin thickness ratio, the predicted flow efficiency in Achaichia's results is higher than the experimental results. While Bellow's correlation shows the lowest flow efficiencies, the slope in Sahnoun's correlation is not consistent with other results. Our numerical results with a thickness ratio of 0.05 (the smallest thickness ratio in our simulations) are very close to, but less than Achaichia's correlation. At low Reynolds number, ($\mathrm{Re}_{L_P} = 50$), good agreement is found between the current results and the correlations of Achaichia and Sahnoun. At high Reynolds numbers ($\mathrm{Re}_{L_P} > 1200$), the asymptotic value of the current numerical results at b=0.05 is nearly the same as that in Sahnoun's correlation for a thickness ratio of 0.043. However, the predicted trend of flow efficiency from duct directed to louver directed flow is in much better agreement with the correlations of Achaichia and Bellows'. Also the critical Reynolds number in this study is found to be around 500 ($\mathrm{Re}_{w,c}$ in Eqn. 1).

EFFECT OF THICKNESDS ON FRICTION AND HEAT TRANSFER PERFORMANCE

Figure 8 shows the j factor and f factor versus Reynolds number for both louver angles. It can be seen that thickness ratio has a significant effect on the friction factor whereas little effect is found on the j factor except at low Reynolds numbers ($\mathrm{Re}_{in} < 300$). For the louvered fins of 30 degrees, j factor decreases around 16% when the thickness ratio increases from 0.05 to 0.15 at a Reynolds number of 50. The difference decreases as Reynolds number increases beyond 300. The 20 degree geometry follows a similar trend.

Increasing the thickness ratio has two counter effects on the heat transfer coefficient. On one hand, as thickness ratio increases the open flow area becomes smaller in both the streamwise and louver directions. Flow is accelerated and local Reynolds number is increased. This has a favorable effect on heat transfer. For example, the local Reynolds number (based on average flow velocity at minimum cross-sectional area) for a thickness ratio of 0.15 is 12% higher than that for a thickness ratio of 0.05. On the other hand, increasing thickness ratio leads to the reduction of flow efficiency accompanied by large recirculation zones on louver surfaces. Under this condition, heat transfer is inhibited and the heat transfer coefficient is degraded. For example, the flow efficiency is reduced by 35% when the thickness ratio increases from 0.05 to 0.15 at $\mathrm{Re}_{in} = 50$. Compared to the 12% increment of local Reynolds number, the large reduction in flow efficiency has a dominant effect on heat transfer performance. As a consequence, the j factor decreases at low Reynolds numbers. As the Reynolds number increases beyond 300, the reduction of flow efficiency is only 15% when thickness ratio is increased from 0.05 to 0.15. This reduction is comparable to the increment in local Reynolds number, as a result, no significant difference in j factor is found in this Reynolds number regime. The above scenario is shown in Figure 9 at a Reynolds number of 500. Louver directed flow with high flow efficiency is evident for the thin louver, whereas a recirculation zone with lower flow efficiency is found for the thick louver. The corresponding local heat flux on both louver surfaces is shown in Figure 9 (c) and (d). The heat flux for the thinner louver is nearly the same on both surfaces because of the higher flow efficiency. However, for the thicker louver, a high heat flux on the bottom surface is brought about by flow acceleration, whereas a lower heat flux on the top surface results from the recirculation zone and the lower flow efficiency. As a result, the total heat flux and the *j*-factor for both thickness ratios is almost the same.

As thickness ratio increases, contrary to *j* factor, *f* exhibits a different trend with Reynolds number. As Reynolds number increases, the increment in friction factor increases with thickness ratio. At low Reynolds number, $\mathrm{Re}_{in}=50$, little difference is found in the 20 degree case, whereas a 20% increment is found in the 30 degree case as the thickness ratio is increased from 0.05 to 0.15. At high Reynolds numbers, $\mathrm{Re}_{in}>500$, a 55% increment in *f* is found for both louver angles. Most of the contribution to the increase in pressure drop is a result of increasing form drag with an increase in thickness ratio. At high Reynolds numbers, the form drag increases substantially for thick louvers because of large recirculation zones. In the case with 20 degree louver angles, increasing the thickness ratio form 0.05 to 0.1 leads to a 30 % increase of form drag. A further increase in thickness ratio to 0.15 increases the form drag by an additional 20%. The effect of thickness ratio on increase in form drag is less for larger louver angles. For a louver angle of 30 degrees, the increments are 20% and 15%, respectively.

It is clear that as the thickness ratio increases, the enhancement in heat transfer performance is less significant than the corresponding pressure losses.

CONCLUSION

In this paper, high-resolution time-dependent numerical simulations are performed to investigate effect of thickness ratio on fluid flow and heat transfer performance in multi-louvered fins. Results for three fin thickness ratios, two louver angles, and a fin pitch ratio of one are reported for Reynolds number ranging from 50 to 1200. Thickness ratio is found to have significant effect on flow efficiency, especially in geometries with small louver angles. Existing correlations do not explicitly include the effect of thickness, and their use can lead to erroneous flow efficiencies. As expected, increasing thickness ratio increases total pressure drop, most of which comes from an increase in form drag. Heat transfer coefficient, on the other hand, is influenced by two counteracting effects, one being flow acceleration as thickness increases, and the other the reduction in flow efficiency and larger recirculation zones. Hence, some heat transfer reduction is found at low Reynolds numbers, however the reduction diminishes as the Reynolds number increases beyond 300. In general, larger thickness ratios lead to a lower ratio of *j/f*.

Although this paper has studied one fin pitch ratio, the results can be extended qualitatively to other fin pitch ratios and louver angles by using the ratio, *d*. To first order as *d* increases, flow efficiency increases. An increase in flow efficiency, usually results in an increase in heat transfer coefficient, till secondary effects like recirculation zones become dominant, which counteract the increase in heat transfer coefficient.

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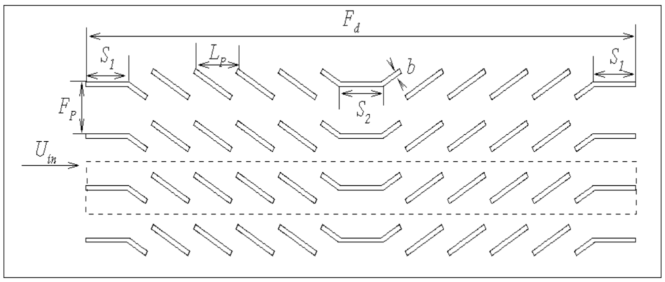
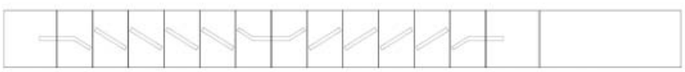
Table 1: Summary of non-dimensional geometrical parameters for the cases investigated

Case	F_p	θ	b	F_d
A-1	1.0	30	0.05	13
A-2			0.1	
A-3			0.15	
B-1		20	0.05	
B-2			0.1	
B-3			0.15	



(a)



(b)

Figure 1: Louvered fin geometry and multi-block computational domain.

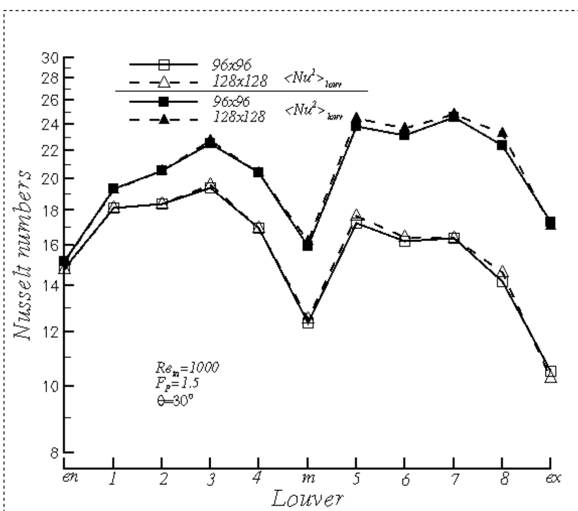


Figure 2: Grid independency study: comparison of louver by louver distribution of heat flux and heat transfer coefficient predicted on a 96x96 and 128x128 grid per computational block.

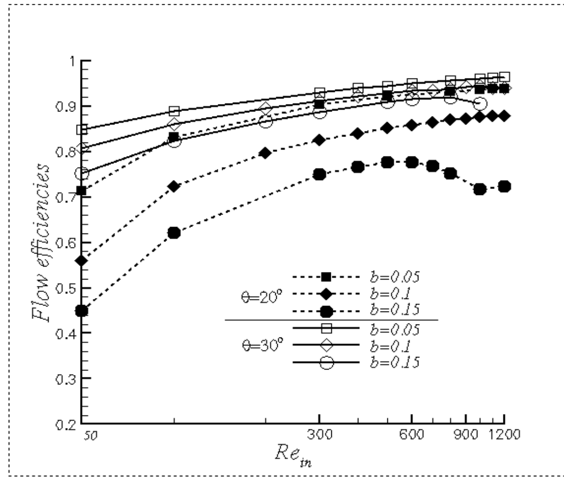


Figure 3: Computed Flow efficiency versus Reynolds number.

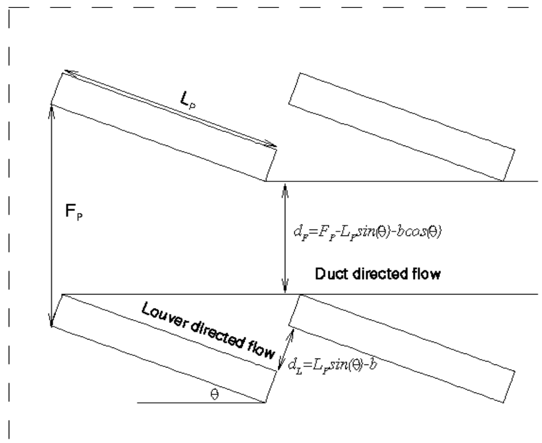


Figure 4: Louver gap and fin gap

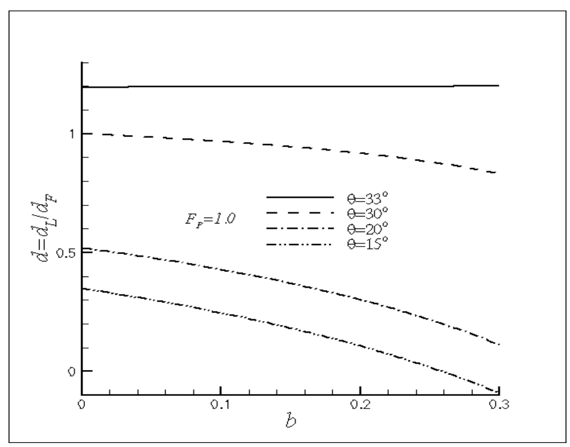
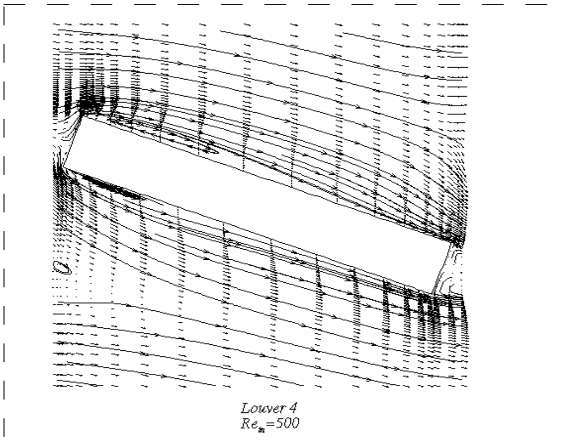
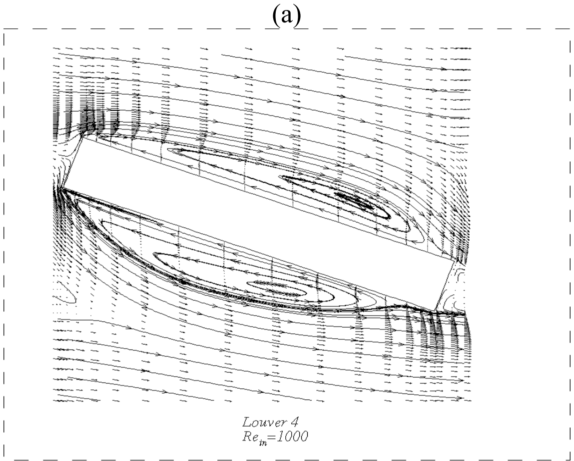


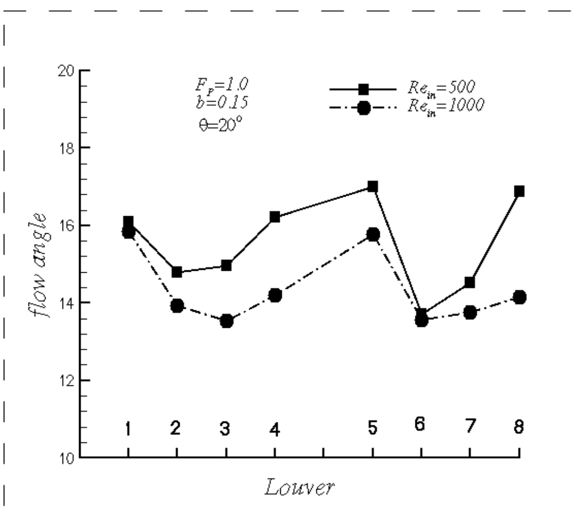
Figure 5: Gap ratio versus thickness ratio for different louver angles for a fin pitch ratio of 1.0.



(a)

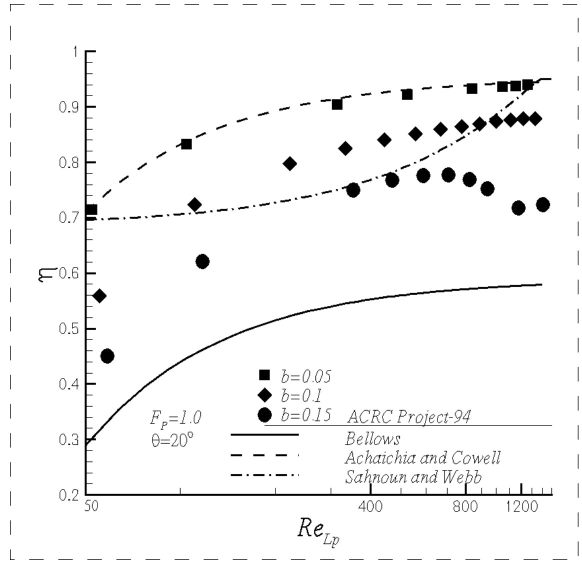


(b)

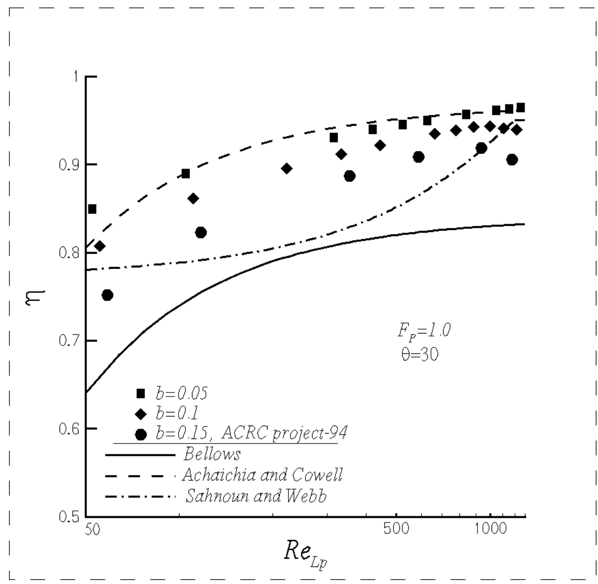


(c)

Figure 6: Effect of louver thickness on flow field and flow efficiency. Streamlines around louver 4 (a) Re_{in}=500, (b) Re_{in} =1000, and (c) local flow angles for both Reynolds numbers.

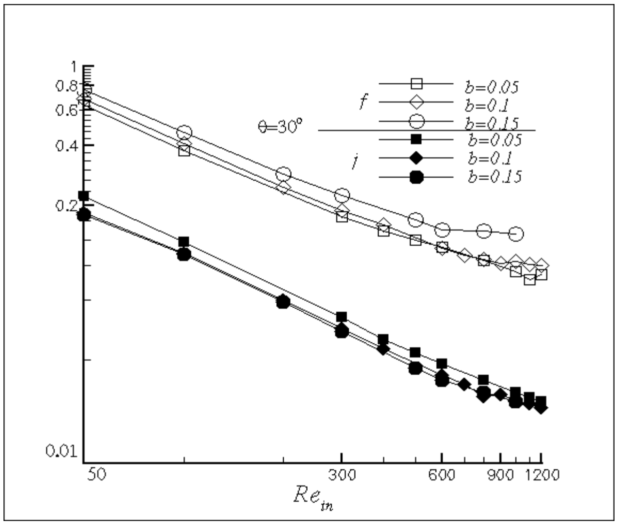


(a)

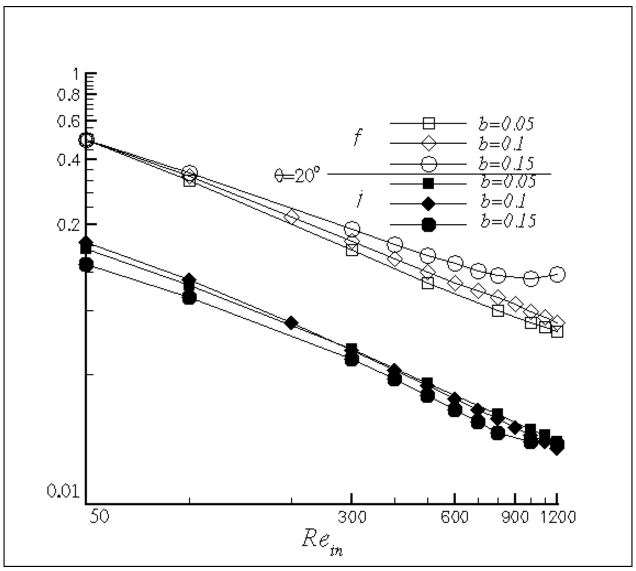


(b)

Figure 7: Comparison of predicted flow efficiencies with existing correlations.

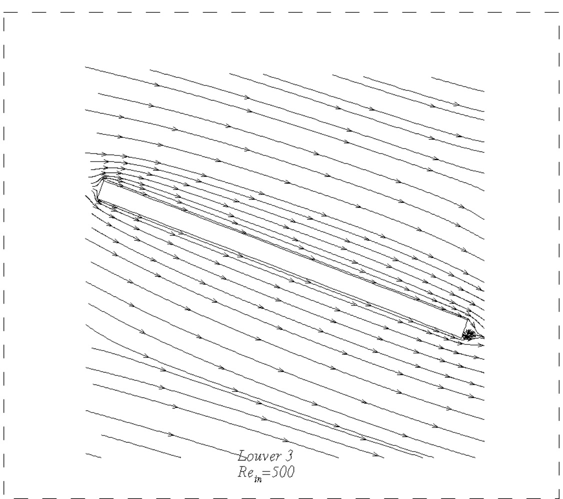


(a)

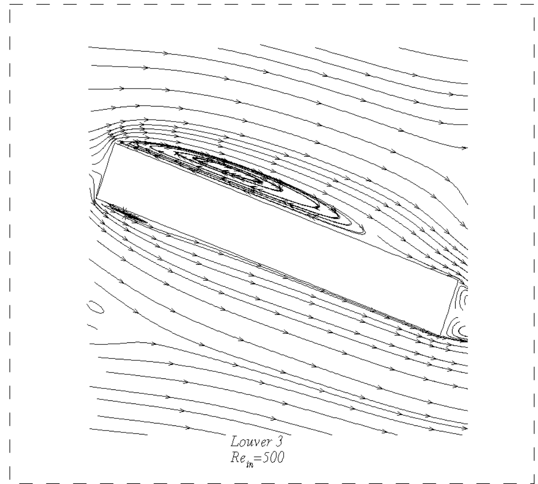


(b)

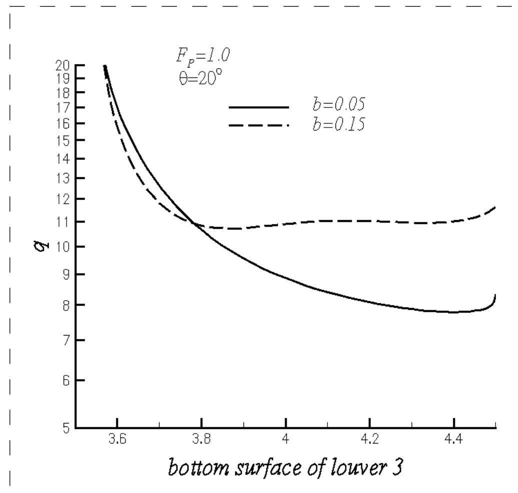
Figure 8: j and f versus Reynolds number for (a) 30 degree louvers; (b) 20 degree louvers.



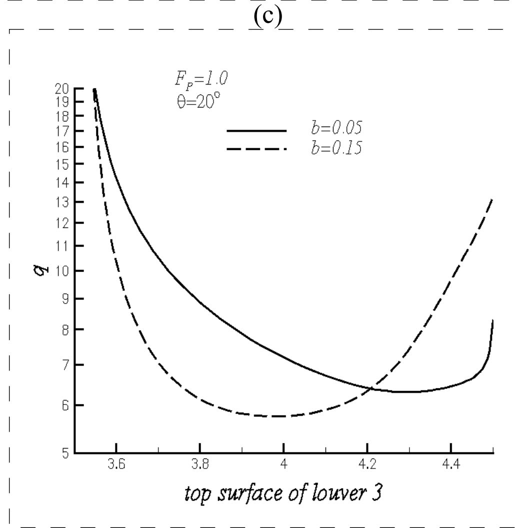
(a)



(b)



(c)



(d)

Figure 9: Streamline distribution around louver 3 at Reynolds number of 500 for two thickness ratios, (a) 0.05, and (b) 0.15. Non-dimensional heat flux distribution on (c) bottom, and (d) top surface of louver 3.