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3D CFD Simulation of Thermal Hydraulic Performances on Louvered Fin Automotive Heat Exchangers

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***Abstract*—**This study deals with Computational Fluid Dynamics (CFD) studies of the interactions between the air flow and louvered fins which equipped the automotive heat exchangers. 3D numerical simulation results are obtained by using the ANSYS Fluent 13.0 code and compared to experimental data. The paper studies the effect of louver angle and louver pitch geometrical parameters, on overall thermal hydraulic performances of louvered fins.

The comparison between CFD simulations and experimental data show that established 3-D CFD model gives a good agreement. The validation agrees, with about 7% of deviation respectively of friction and Colburn factors to experimental results. As first, it is found that the louver angle has a strong influence on the heat transfer rate. Then, louver angle and louver pitch variation of the louvers and their effects on thermal hydraulic performances are studied. In addition to this study, it is shown that the second half of the fin takes has a significant contribution on pressure drop increase without any increase in heat transfer.

***Keywords*—**CFD simulations, automotive heat exchanger, performances.

I. INTRODUCTION

IN compact heat exchangers, thermal resistance is generally dominant on the air-side and may account for 80% or more of the total thermal resistance. The air-side heat transfer surface area is 8 to 10 times larger than the water-side. Any improvement in the heat transfer on air-side therefore improves the overall performance of the heat exchanger.

Due to the high thermal resistance on the air-side, the optimization of such fins is essential to increase the performance of the heat exchangers which results in thermal systems enhancement. This helps to reduce CO_2 emissions through a reduction of mass and fuel consumption.

Optimization of louvered fin geometry in such heat exchangers is essential to increase the heat transfer performance and reduce weight, packaging, and cost requirements.

The aim of the present paper is to explore ways to approach the best compromise between thermal performance and pressure drop. Two distinct parameters, the louver pitch and the louver angle, are considered and 3-D simulations are conducted to predict the best performance.

II. GEOMETRICAL PARAMETERS OF LOUVERED FIN

Different views of fins of geometrical parameters for a louvered fin in this study are shown in Fig. 1.

All samples in this analysis have a 16mm tube width, a 1.4mm depth and the louver fin thickness is 0.1mm. Simulations are performed for different geometries with various fin pitch (Fp), louver pitch (Lp) and louver angle (α).

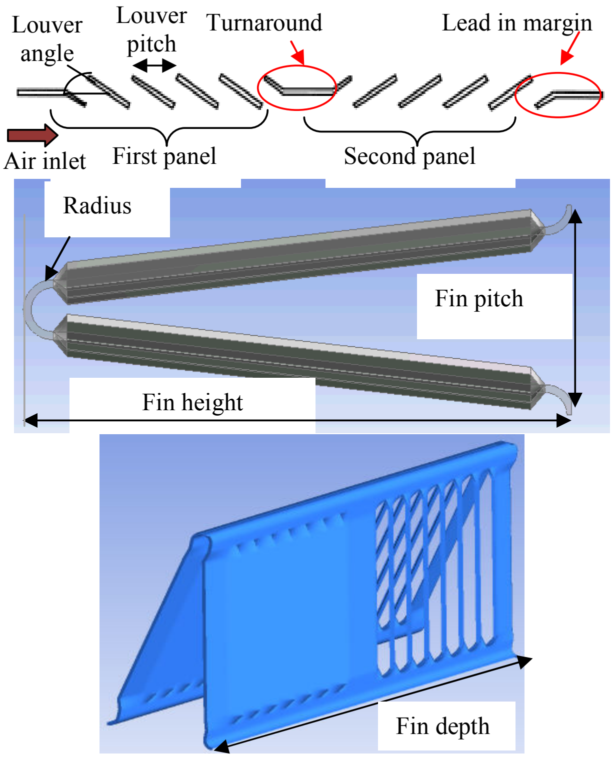

Fig. 1 Geometrical parameters of louvered fin

The values of these parameters are listed in Table I.

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TABLE I
GEOMETRICAL PARAMETERS AND OPERATING CONDITIONS OF THE LOUVERED FIN

Fin pitch (Fp)	1.15 mm
Fin depth (Fd)	16.0 mm
Fin thickness (δ)	0.1 mm
Louver pitch (Lp)	0.9 mm/1.0 mm/1.05 mm/1.2 mm/1.4 mm
Louver angle (θ)	20° - 40°

III. COMPUTATIONAL DOMAIN AND BOUNDARY CONDITIONS

The 3D computational domain is the representation of one convolution shown on Fig. 2. One convolution contains two fins of the actual air-side surface for different automotive heat exchangers like radiators, condensers... Two inlet and outlet rectangular channels are added to establish the flow before the entrance in the fins and to avoid backflow downstream of the convolution. Symmetry conditions are assumed on both sides of the domain. Periodic boundary conditions are applied at the left and the right of the domain. Tetrahedral mesh is used and the total number of cells for the 3D model is 10 millions (Fig. 3). Tube and fins have been meshed to take into account the material thermal resistance.

At the entrance, the velocity of the air is applied. For the fin, non slip boundary conditions were applied, as well as a constant temperature boundary condition.

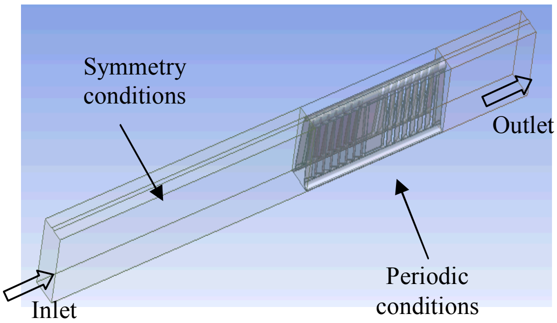

Fig. 2 Computational domain

Insensitivity of results to the mesh refinement is established and the comparisons between three different mesh sizes are presented on the Table II. This table shows that the difference with our mesh (10 millions of cells) never exceeds an average variation of 3.65% for friction and Coburn factors. 10 millions of cells are chosen for this parametric study in terms of results and time.

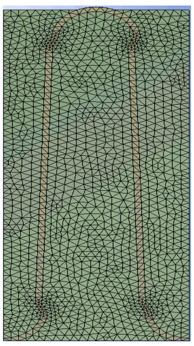

Fig. 3 Mesh

TABLE II
MESH SENSITIVITY TO FRICTION AND COLBURN FACTORS

Mesh size				*Deviation to the 10 million of cells*
	Frontal velocity (m/s)	*1.9*	*7.8*	
Friction factor	10 millions of cells	0.183	0.09	
	15 million of cells	0.178	0.086	3.65%
	20 million of cells	0.178	0.86	3.65%
Colburn factor	10 millions of cells	0.035	0.025	
	15 million of cells	0.035	0.025	0%
	20 million of cells	0.035	0.025	0%

IV. MODEL VALIDATION

Reference [1] found that the laminar flow region appears for Reynolds number (Re_{Lp}) less than 400. Reference [2] indicated that the flow remains laminar and steady for Reynolds number up to 1300. Laminar and different steady state turbulence models (k- ε standard, k- ε Enhanced wall treatment and k-w SST) are used to verify that the laminar model is more suitable on our range of Reynolds numbers.

Laminar steady, k-ω SST and k-ε-enhanced wall treatment models give the same friction and Colburn factors. k-ε standard is not adapted to our quality of mesh. We choose to use the laminar steady model.

Fig. 4 presents comparison of experimental data and simulation results for louvered fin.

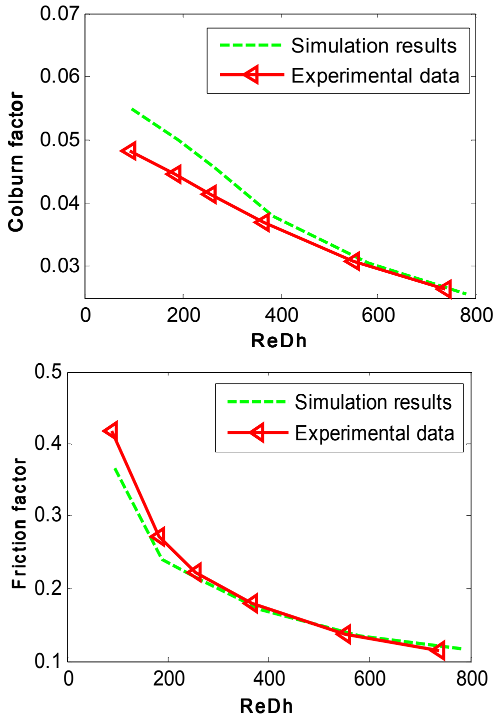

Fig. 4 Friction and Colburn factors comparisons between numerical and experimental results

The j and f factors calculation results give good agreement with less than 7% average deviation compared to test data. Close agreement is found between the computational results

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and experimental data at intermediate and high Reynolds number. However, at low Reynolds number there is some deviation. This can be explained by the importance of uncertainties of measurement at very low Reynolds numbers. Uncertainties in the experimental Stanton numbers and friction factors estimated by [3] have been reported to be 6.5% and 15%, respectively at low Reynolds number.

V. CFD Simulations and Discussions

The standard louvered fin has constant geometrical parameters (as the louver angle and louver pitch) along the fin which makes difficult to improve the thermal hydraulic performance. We propose to study fins with variable geometrical parameters and to identify which are the best combinations.

Also the effect of increasing or decreasing the louver pitch of the three first louvers of the fin is studied by analyzing the thermal hydraulic performances variation (Fig. 5)

Performances results are summarized on Fig. 6 by variation of friction factor and Nusselt numbers as function of Reynolds numbers based on the hydraulic diameter.

Reference fin: 6×6 louvers at 28°,
All louvers Lp 0.9 mm, leading margin 1.7 mm including ½ louver

Fin1: 2louvers Lp1.05mm, 6×6 louvers
Louvers 1&2. 13&14 Lp 1.05 mm.
Louvers 3-12 Lp 0.9 mm. leading marging1.4 mm

Fin2: 3louvers Lp1.2mm, 5×5 louvers.
Louvers 1-3. 12-14 Lp 1.2 mm.
Louvers 3-12 Lp 0.9 mm

Fin3: 3louvers Lp decrease, 5×5 louvers.
Louvers 1-3. 12-14 Lp 1.4mm - 1.2mm - 1 mm.
Louvers 3 -12 Lp 0.9 mm

Fin4: 3louvers Lp increase, 5×5 louvers.
Louvers 1-3. 12-14 Lp 1mm- 1.2mm - 1.4 mm.
Louvers 3-12 Lp 0.9 mm

Fig. 5 Louver fin geometries with varying louver pitch

It is shown on Fig. 6 that three geometries (fin2, fin3, fin4) decrease pressure up to 11% and heat transfer up to 6%.

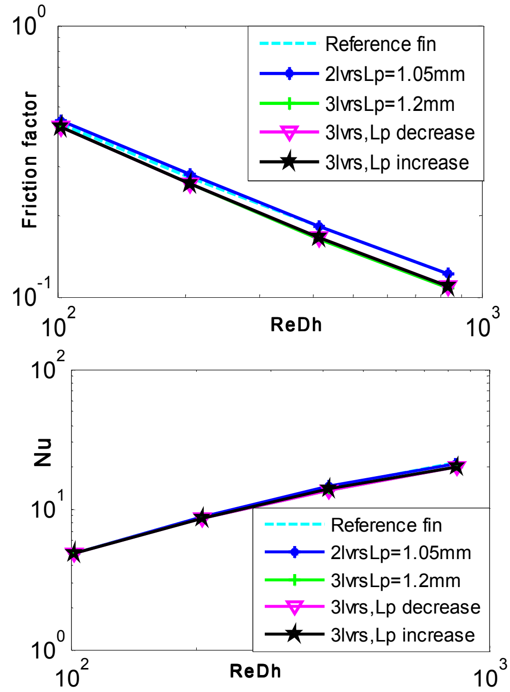


Fig. 6 Louver pitch variations effect on friction factor and Nusselt numbers

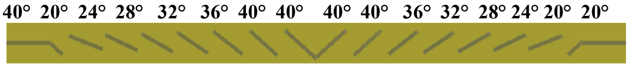
Thus, four different cases of successively increasing or decreasing louver angles (+2°,-2°, +4°, -4°) and constant angle (28°) are investigated (Fig. 7).

Reference fin: 6×6 louvers at 28°. Lp= 0.9 mm. leading margin 1.7 mm including ½ louver

Fin (-2°)
28° 28° 26° 24° 22° 20° 18° 18° 18° 18° 20° 22° 24° 26° 28° 28°

Fin (+2°)
20° 20° 22° 24° 26° 28° 30° 30° 30° 30° 28° 26° 24° 22° 20° 20°

Fin (+4°)
40° 20° 24° 28° 32° 36° 40° 40° 40° 40° 36° 32° 28° 24° 20° 20°



Fin (-4°)
40° 40° 36° 32° 28° 24° 30° 30° 30° 30° 28° 26° 24° 22° 20° 20°

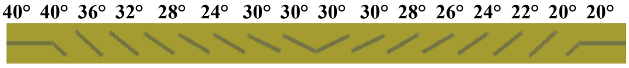


Fig. 7 Louver fin geometries with varying louver angle

Fig. 8 shows comparison of friction factor and Nusselt number for the different fins with varying louver angle listed before.

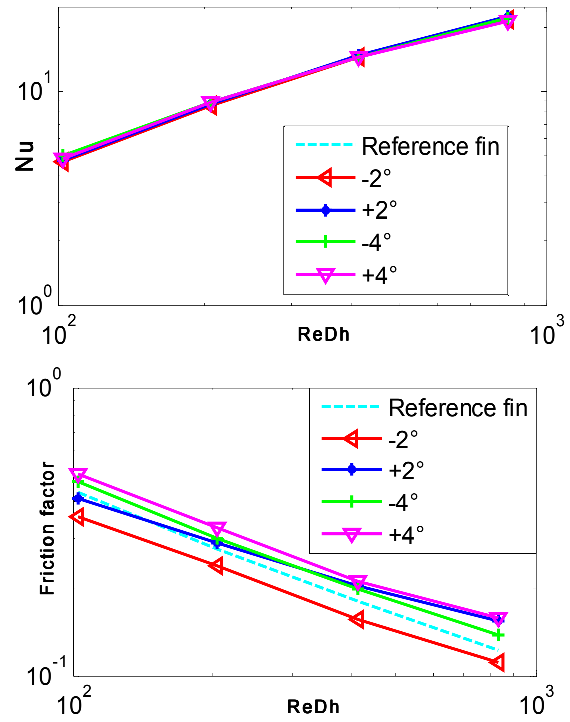


Fig. 8 Louver angle variations effect on friction factor and Nusselt numbers

It is shown that only the fin (-2°) decreases pressure drop and that all the fins decrease heat transfer performances up to 3%. The results for fin (-2°) are presented in Table III.

TABLE III
DEVIATIONS OF FIN (-2°) COMPARED TO REFERENCE FIN

Vair (m/s)	Fin (-2°)	
	% deviation friction factor	% deviation of Nusselt number
0.94	17.37	2.98
1.88	12.57	2.68
3.75	13.07	0.4
7.5	8.89	-0.21
Average	12.97	1.46

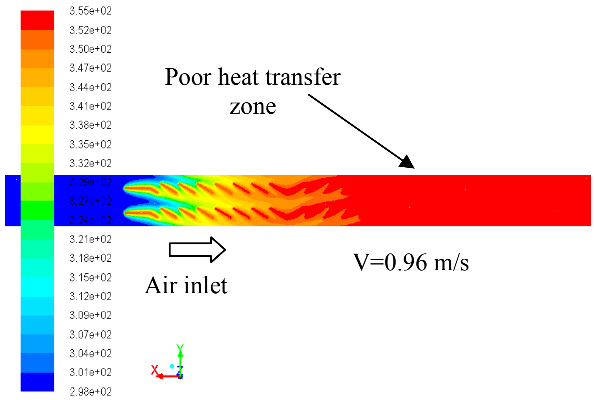
It is shown that for a -2° variation, the pressure drop decreases up to 17% at low velocity (V=0.98 m/s) and heat transfer to 3%.

The conclusion is that only the fin -2° is optimal due to the important decrease of pressure drop.

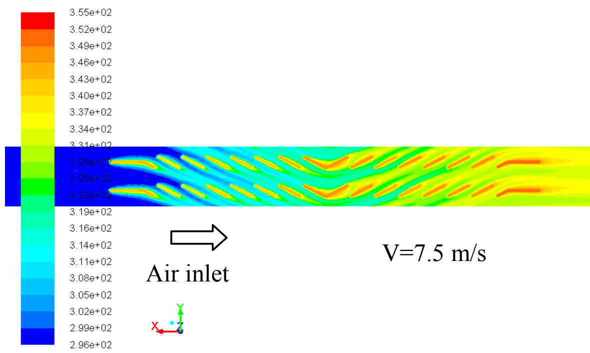
The study of [4] proves that the temperature gradient for successively variable louver angle is higher than that for the uniform louver angle. It is seen that both Nu for successively variable louver angles are higher than those for the uniform louver angle. For case A (+2°), case B (+4°), case C (-2°) and D (-4°), the maximum heat transfer improvement interpreted by j/j0 are 115%, 118%, 109% and 107%, and the corresponding friction factor ratio f/f0 are 116%, 119% 110% and 108%, respectively. Where j and f are friction factor ratio and Colburn factor ratio for variable louver angles configurations (case A-D) and f0 and j0 the values for uniform angle configuration (case E).

It is shown that the variations of louver angle have not the same effect compared to this study. It is probably due to the fact that the ranges of louver angles on this study are higher compared to those used by [5] and that the geometrical parameters are different.

In addition, since the temperature difference between the air and the fin (driving potential for heat transfer) decreases along the fin, the heat rate in the second panel is lower than the one in the first panel. This heat rate degradation is more pronounced as the air velocity decreases, as it is shown on the temperature fields for high and low velocities (Fig. 9).



(a)



(b)

Fig. 9 Temperature distribution (a) V=0.96 m/s (b) V=7.5 m/s

At low Reynolds number (Fig. 9 (a)) most of the air flows through the gap between the fins rather than through the louvers. This can be attributed to the high flow resistance presented by the louvers. At low Reynolds number, since the air has less kinetic energy, most of it passes through the path of least resistance, i.e., through the fin gaps, the flow is qualified as duct flow. The air temperature reaches the fin temperature in the second half of the louvered array, and as a result the heat transfer performance of the fin is poor. The temperature difference between the fin surface and the air is near zero. The result of heat transfer performance of the fin is poor.

At high Reynolds numbers, the boundary layers thickness around the louvers are very thick as the air flows through the gap between the fins rather than flowing into the gap between the louvers and the air flow is directed by the louvers. In this case air temperature increases less along the flow direction and a significant temperature difference is still observed between air and fin (Fig. 9 (b)). The result for heat transfer is thus higher at high Reynolds number.

This is congruent with the results of previous study of [5].

At low Reynolds number, second half of the louvers array only accounts for pressure losses without any significant heat transfer. This means that it is not necessary to keep a relatively high louver angle in the second panel to maintain a high heat transfer coefficient. So, it would be interesting to reduce the louver angle in this area to reduce the pressure drop.

VI. Conclusion

In this paper we describe flow and heat transfer characteristics in the array of louvered fins. Numerical results based on laminar steady model are compared to experimental results. Good agreement is shown between numerical results and experimental data with an average deviation of 7% in pressure drop and heat transfer. The effects of louver pitch and louver angle have been investigated by comparisons between friction and Colburn factors. The results shows that the louver angle and louver pitch have an important effect on thermal hydraulic performances. The fin (-2°) decreases very well pressure drop up to 17%. Then, the effect of the second row, at low and high velocities, in a louvered fin array is presented and flow pattern are described. The second panel provides important pressure losses without any increase in heat transfer. The second panel of the louvered fins should be modified to decrease pressure drop without any losses in heat transfer in heat exchangers.

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