HEFAT2008
6th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
30 June to 2 July 2008
Pretoria, South Africa
Paper number: CZ1

HEAT TRANSFER ANALYSIS OF FLAT AND LOUVERED FIN-AND-TUBE HEAT EXCHANGERS USING CFD

Čarija Z.* and Franković B.
*Author for correspondence
Faculty of Engineering,
University of Rijeka,
Rijeka,
Croatia,
E-mail: zcarija@rijeka.riteh.hr

ABSTRACT

This paper analyzes the fluid flow and heat exchange on the air side of a multi-row fin-and-tube heat exchanger. A comparison is given between fin-and-tube heat exchanger characteristics with flat and louvered fins in a wider range of operating conditions defined by Reynolds number (based on fin spacing and air frontal velocities). The detailed representation of calculated data for the louvered heat exchanger shows significantly better heat transfer characteristics and a slightly higher pressure drop. The CFD procedure was validated by comparing the numerical simulation results with the experimental results showing minimal average Nusselt number deviation and almost perfectly corresponding pressure drop.

INTRODUCTION

The usage of fin-and-tube heat exchangers in water chillers and heat pumps is increasing. Due to the relatively small heat transfer coefficient on the air side, the evaporators and condensers of these devices are relatively large in size. Decreasing their dimensions requires improvements in the heat exchange on the air side which can be achieved by redesigning the fin shape in the heat exchanger. Although many automotive companies and heat exchangers manufacturers have performed a lot of experimental research in fin heat exchangers, very little of the experimental data is publicly available due to its commercial value. However some extensive experimental data for louvered fins was reported by Devenport, Achaichia, Chang and Wang [1,2,3,4,5]. Expensive and long-term experimental research allows examination of only a restricted number of geometrical shapes. In addition to the experimental study with the increasing processing power of modern computers, numerical fluid flow simulations have recently been widely used in order to improve airside performance of fin heat exchangers [6,7,8,9].

Many papers report an improvement in airside performance by changing the fin geometry. One of the very popular enhancements is the interrupted surface [6, 8].

NOMENCLATURE

D	[mm]	Tube outside diameter
E	[W/kg/s]	Total energy
H	[mm]	Fin pitch
k	[W/mK]	Thermal conductivity
L	[mm]	Louver length
L_p	[mm]	Louver pitch
N_l	[-]	Louver number on fin surface
Nu	[-]	Nusselt number
p	[Pa]	Fluid pressure
P_l	[mm]	Longitudinal tube pitch
P_t	[mm]	Transverse tube pitch
Q	[W]	Heat flux
R	[J/molK]	Specific gas constant for air
Re_H	[-]	Reynolds number based on fin pitch
t	[mm]	Fin thickness
T	[°C]	Temperature
$T_{a,i}$	[°C]	Air inlet temperature
$T_{a,o}$	[°C]	Air outlet temperature
T_b	[°C]	Fluid bulk temperature
T_w	[°C]	Wall surface temperature
$\vec{u}$	[m/s]	Velocity vector

Greek symbols

Δp	[Pa]	Pressure drop
μ	[Pas]	Dynamic viscosity
θ	[°]	Louver angle
ρ	[kg/m^3]	Fluid density

This is because the interrupted surfaces can provide higher average heat transfer coefficients owing to periodical renewal of the boundary layer development. The most common interrupted surfaces are louvered fins whose performance is analyzed in this paper. Performance can also be improved by intervening on the tube side [10] or changing its shape from circular to oval [6,11] and arrangements [11].

Fluid flow and heat transfer simulation using CFD [12,13] tools allow performing cost-effective virtual "numerical experiments". Virtual experiments enable examination a large number of different shapes and evaluate the actual performance or improve it through new design, all within a short timeframe. In fin-and-tube heat exchangers, the fin shape has an important role in the heat exchange.

This research began with the premise of isothermal condition on the whole louver fin surface in which case the thermal conductivity of the fin is infinite. This paper is intended to examine the influence of main geometric parameters of the louvered fins on heat exchange and pressure drop in order to improve the overall performance of the louvered fin-and-tube heat exchanger. Calculated data from fluid flow simulations was compared to experimental ones in order to validate the procedure. It is safe to assume that the exact thermal conductivity of the fin will increase result accuracy, the proof of which will be the aim of the next step in this research.

GEOMETRY

The periodic geometry along the tube and the symmetry along the air flow direction allow simplification of the model geometry. The green dashed line in Figure 1 designates the computational domain. The plain view representation of the computational domain and the section cut are shown in figure with annotations for variable parameters that influence the performance of the fined heat exchanger which was analyzed in this paper. A 3D representation of the unit cell used in the case of louvered fins is shown on Figure 2. The computational domain in the flat fin case is identical except for the slats on the fin. Figure 2 shows the louvered fins solid body in red while the shaded volume represents the fluid zone.

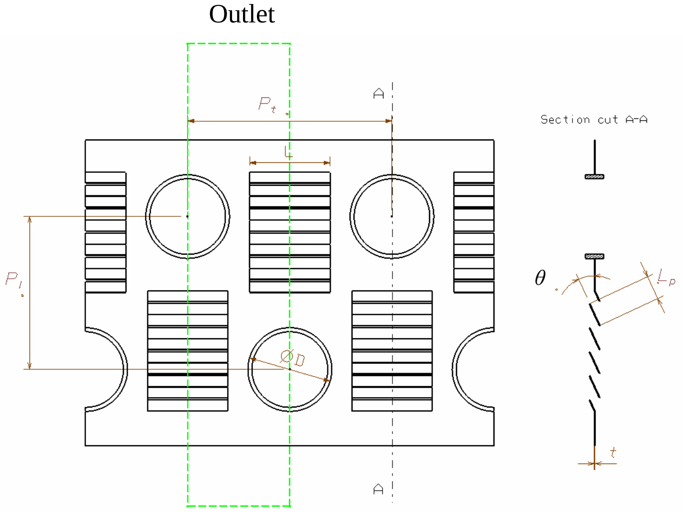

Figure 1 Plane and side views of louvered fins and definitions of geometric parameters

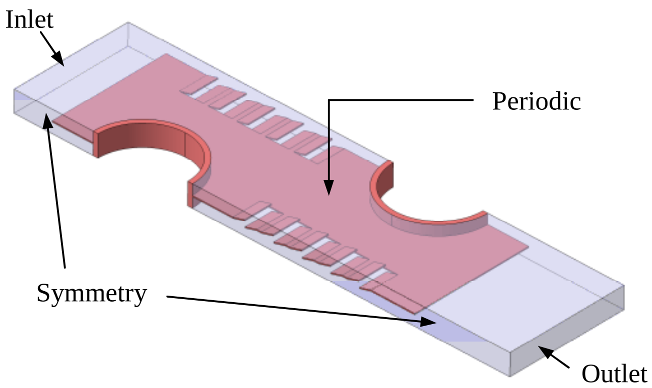

Figure 2 Fluid zone (shaded) around the louvered fin (red)

The CAD tool used allows simplicity in adjusting variable design parameters (L, L_p, N_l, θ). The values for main design parameters are listed in Table 1 with variable parameters, analyzed in this paper, shown as ranges.

Tube outside diameter	10.42 mm
The number of tube row	2
Longitudinal tube pitch (P_l)	19.05 mm
Transverse tube pitch (P_t)	25.4 mm
Fin thickness (t)	0.115 mm
Fin pitch (H)	2.06 mm
Number of louvers (N_l)	4 - 5
Louver angle (θ)	5 - 25°
Louver pitch (L_p)	15mm/N_l
Louver length (L)	4 - 12 mm

Table 1 The detailed geometry of the examined heat exchangers

COMPUTATIONAL MODELS

Conservation equations

The flow over the louvers is assumed to be laminar and steady-state. The equations for representing the conservation of mass, momentum and energy for the three dimensional models are as follows:

$$\frac{\partial \rho}{\partial t} + \nabla(\rho \vec{u}) = 0 \tag{1}$$

$$\frac{\partial}{\partial t}(\rho \vec{u}) + \nabla(\rho \vec{u}\vec{u}) = -\nabla p + \nabla\left[\mu\left(\nabla \vec{u} + \nabla \vec{u}^{T}\right)\right] \tag{2}$$

$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot \left(\vec{u}(\rho E + p)\right) = \nabla \cdot (k \cdot \nabla T) \tag{3}$$

In order to complete this set of equations, additional relations are required to link thermodynamic and transport properties of air. Air properties can be assumed to change according to ideal gas law.

$$\frac{p}{\rho} = RT \tag{4}$$

The dynamic viscosity of the air is a function of temperature and is obtained from Sutherland's law.

$$\mu = 1.45 \cdot 10^{-6} \cdot \frac{T^{1.5}}{T + 110} \tag{5}$$

Due to a small change in air temperature over the fin, specific heat can be assumed constant and is evaluated at the mean air temperature $(T_{a,i}+T_{a,o})/2$. The buoyancy and radiation effects have been neglected.

Solution algorithm

Fluid flow simulations were performed using commercial fluid flow solver Star-CCM+ [14] where the above set of equations is solved using standard finite volume techniques. Equations are integrated over the individual computational cells and over a finite time increment in case of unsteady simulations. The second-order upwind scheme was used to calculate convective flux on the boundary surfaces of control volumes. This 2nd order scheme is the least sensitive to mesh structure. The SIMPLE algorithm, which uses a relationship between velocity and pressure corrections, was used to enforce mass conservation and to obtain the pressure field.

Boundary conditions

The analyzed fluid zone consists of an inlet, an outlet, the periodic top and bottom boundaries and the symmetric boundaries on the sides (Figure 2). The center part of the domain is either a flat or louvered fin, depending on the analyzed case. The fluid flow along the upper and lower surfaces is considered periodic and the one along the sides is considered symmetrical. Uniform dry air flow with constant velocity ranging from 0.5-2.5 m/s and constant temperature $T_{a,i}$=30°C is assumed on the upstream boundary. The streamwise gradient for all variables is set to zero at the downstream end of the computational domain which is located at twice the tube diameter length from the last downstream tube row. No-slip conditions for velocity and constant wall temperature (T_w=60°C) are specified on all solid surfaces. The normal velocity components and normal gradients on the plane of symmetry are set to zero as well as the heat flux.

Numerical mesh

The influence of cell shape and number was examined in a test case prior to this one. Computational performance of meshes consisting of tetrahedral, polyhedral and trimmed volume cells, was analyzed separately. The secondary polyhedral meshes are made up of primary tetrahedral meshes of equal cell sizes. The trimmed cell mesher provides a high quality grid for complex problems which is a predominantly hexahedral mesh with minimal cell skewness. Trimmed cells with one or more corners and/or edges cut off can be seen at the domain boundaries.

Four layers of prismatic cells with a total thickness of 0.16 mm were created on all wall surfaces in order to increase accuracy. The prismatic cells of the tetrahedral mesh consist of six nodes while the arbitrary polyhedral trimmed mesh has thin cells of irregular shape near the wall. Preliminary analyses have shown that the trimmed cell meshes were the most appropriate for this case. These meshes allow the usage of a minimal number of cells while maintaining the same level of accuracy, with four times less volume elements than in tetrahedral meshes. Mesh independent solution was achieved using a trimmed volume mesh with 286433 cells, 850969 faces and 346426 vertexes.

Convergence criteria

Fluid flow simulations in all examined cases were performed iterating towards a final stable solution. It was observed that 300 iterations were sufficient to obtain the solution in all examined cases. The convergence criteria used were residuals (a measure of discretization equation solution inaccuracy; a perfectly converged case would approach a measure equal to the computer round-off error) and negligible change through further iteration in the mass averaged total pressure difference between the problem inlet and outlet section. A typical total pressure change during computation is shown in Figure 4.

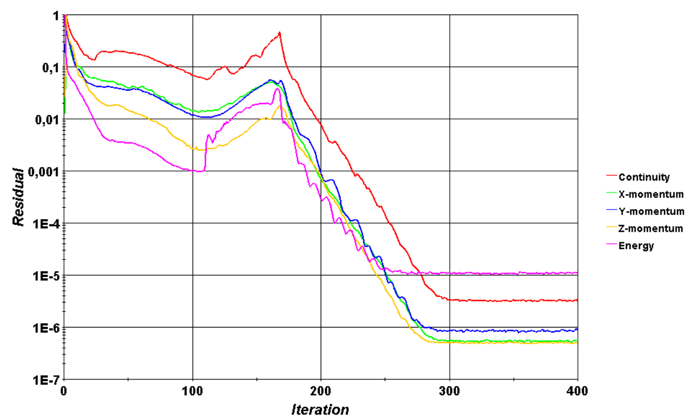

Figure 3 Residuals history during calculation

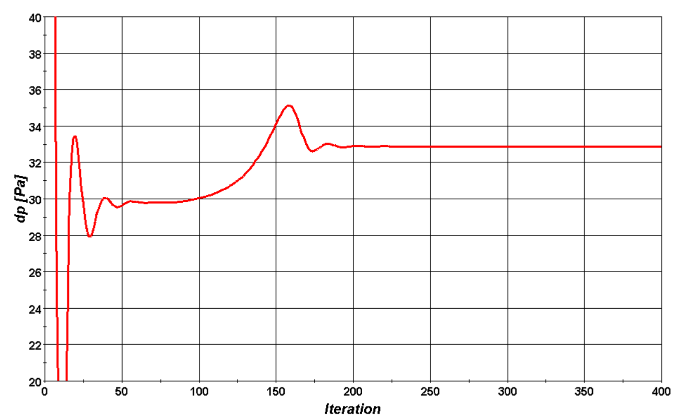

Figure 4 Pressure drop history during calculation

RESULTS

All results are presented as a function of Reynolds number Re_H with the length scale expressed through fin pitch.

Before analyzing the influence of individual parameters of the fin louvers, a comparison of calculated and experimental results [5] was performed in order to validate the procedure. The louvered fin heat exchanger was analyzed with fin parameters set as shown in Table 1 and with the following louver parameters $L/2$=6.25mm, L_p=3.75mm and θ=14°. The averaged Nusselt number based on fin spacing (Nu) and pressure drop (dp) between the inlet and outlet heat exchanger sections were used for comparison. The Nusselt number, which quantifies convective heat transfer from a surface, expresses the ratio of convective and conductive heat transfer between a solid boundary and a moving fluid. The area averaged temperature on the periodic surface of the model was used as the fluid bulk temperature (T_b) to determine the heat transfer coefficient, the most difficult parameter to assess.

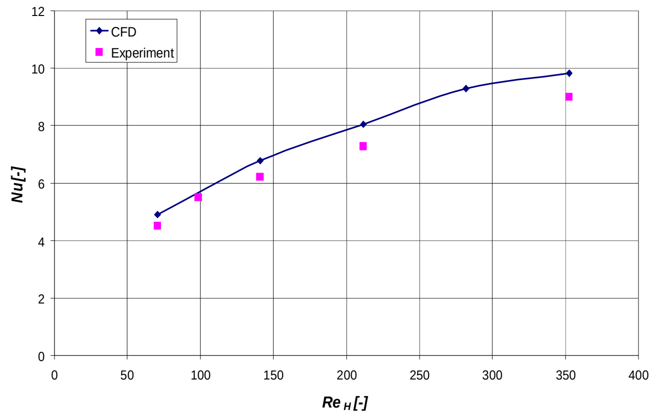

Figure 5 Comparison of numerically and experimentally obtained averaged Nusselt number for the louvered fin

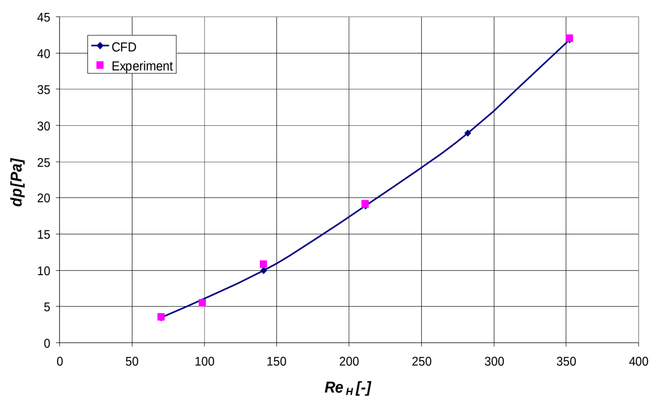

Figure 6 Comparison of numerically and experimentally obtained averaged pressure drop for the louvered fin

The numerically predicted pressure drop corresponds exceptionally well with the experimental results, as seen in the Figure 12. Calculated averaged Nusselt numbers are very similar to the experimentally obtained values, but overestimated by approximately 10% over the whole range of examined Re_H numbers. Such results are very similar to the previously published ones [6] but deviate less from experimental values. Minimal deviations assure the acceptable accuracy for the follow-up analysis of louvered fin exchangers.

The following examples analyze the performance of the louvered fin heat exchanger with a flat fin heat exchanger by comparing the averaged Nusselt number and pressure drop. Fluid bulk temperature was set to a constant value of T_b=43°C for all considered cases. The influences of louver pitch (L_p) and louver length (L) as variable parameters of the louvered fin are analyzed with a constant inlet air speed of 2 m/s. Values of averaged Nusselt numbers at varying louver lengths and a constant louver angle of 15° are shown on Figure 7 while the influence of varying louver angles using a constant louver length of L=8 mm is shown on Figure 8.

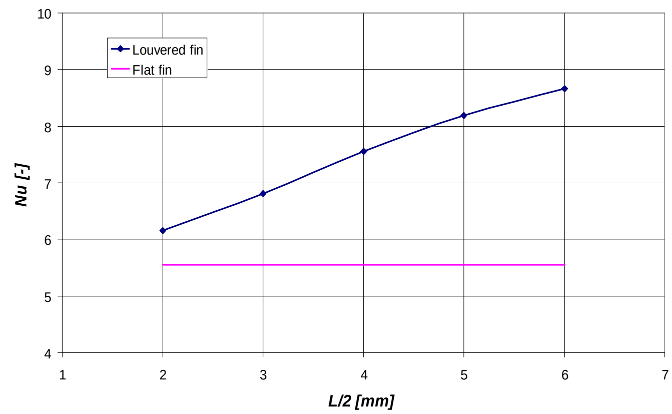

Figure 7 Louver length influence on heat transfer performance; Re_H=282

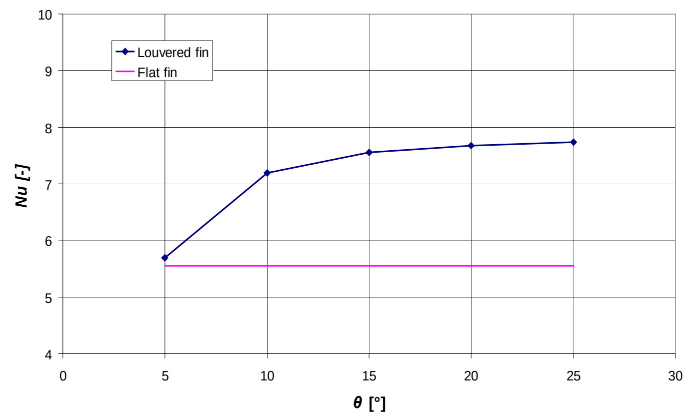

Figure 8 Louver angle influence on heat transfer performance; Re_H=282

These figures show that the heat transfer performance expressed through averaged Nusselt number has an almost linear increase with increased louver length. The averaged Nusselt number is 11% greater at minimum louver length and 56% greater at maximum analyzed louver length, when compared to flat fins. As the louver length decreases the performance gets closer to that of a flat finned heat exchanger (shown in magenta) which is actually the extreme case (L=0).
Averaged convective heat transfer increases with louver pitch up to the saturation point near 15°. Increasing the louver angle further doesn't guarantee an increase in performance but has a significant impact on pressure drop (Figure 12). When using an

optimal louver angle (15° in the examined case) the Nusselt number is 36% greater than when using flat fins. When the louver pitch is 5° or less, the convective heat transfer almost matches that of the flat fin case.

The number of louvers on a fin also influences the convective heat transfer from the fin surface. It should be noted that increasing the number of louvers decreases louver pitch L_p and the pressure drop while maintaining the same louver angle. Increasing the number of louvers from N_l=4 to N_l=5 also increases the Nusselt number, but not significantly (Figure 10).

The results using the constant air speed of 2 m/s presented so far lead to a conclusion that louvered fins can improve heat exchanger performance significantly. The influence of louvered fin variable parameters on convective heat transfer was therefore analyzed over the examined range of inlet air velocities (0.5 - 2.5 m/s) and is presented hereafter.

Louvered fins represented in the following figures are labeled by their parameters as L_L / θ / N_l displaying the length, angle and number of louvers.

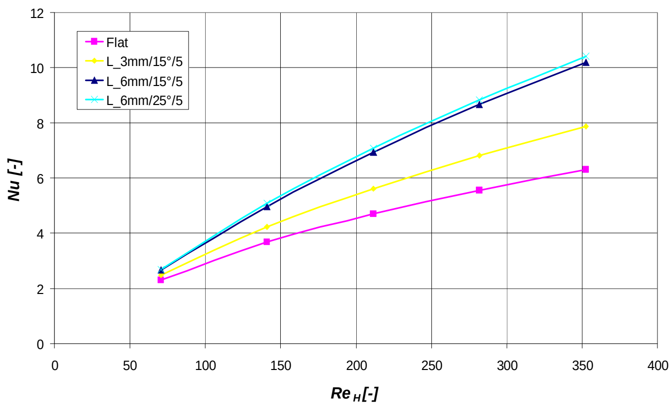

Figure 9 The influence of louver length and angle on heat transfer performance

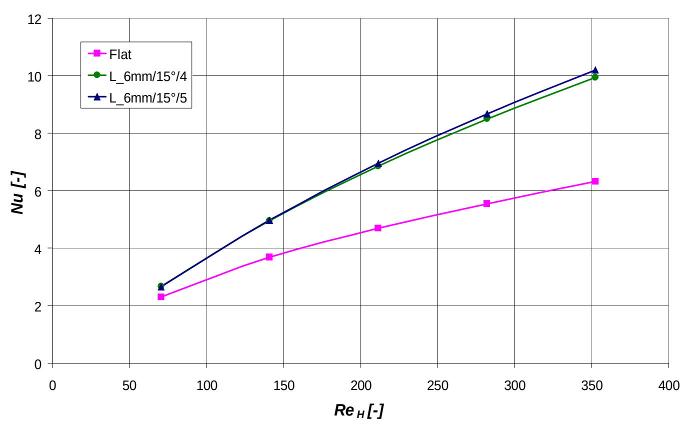

Figure 10 The influence of number of louvers on heat exchange

The results show that the louver length has the greatest influence on improving heat exchange. However, it is limited by transverse tube pitch (P_t). Heat transfer can be additionally improved by increasing louvered angle up to 15°. Further increases in the louver angle and the number of louvers do not enhance heat exchanger performance significantly (Figure 9, Figure 10).

Increasing the Re_H number leads to diminishing returns with all of the monitored values in all presented cases. This corresponds to trends observed in experimentally obtained results.

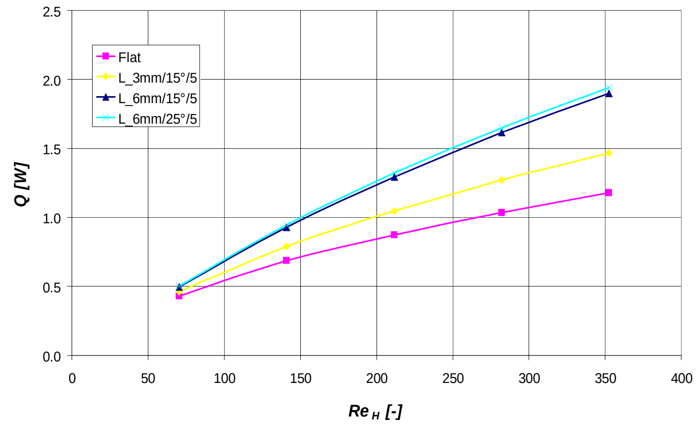

Figure 11 Heat transfer comparison between exchangers with louvered and flat fins

The actual louvered fin with 5 slats improved heat transfer at Re_H=350 by 58% compared to a flat fin. Pressure drop, unlike heat transfer, increased constantly with either louver length or louver angle. An increase in the abovementioned louver parameters reduced the flow section area between fins which in turn negatively affected the pressure drop between the inlet and the heat exchanger sections (Figure 12).

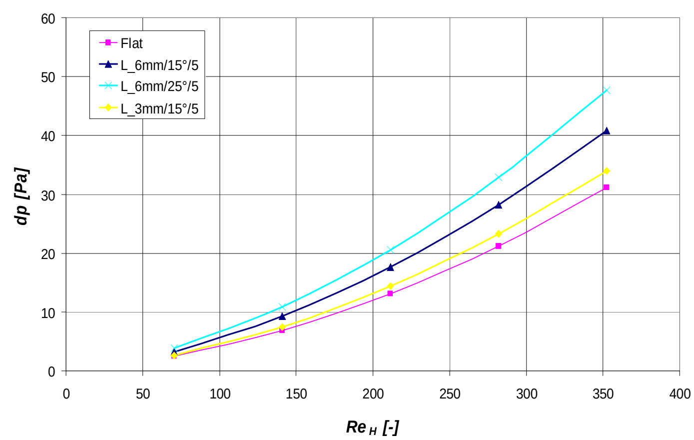

Figure 12 Pressure drop comparison in heat exchangers with louvered and flat fins

CFD simulations give us more than just a means to compute averaged integrals - they present a detailed insight into complex structure and interactions within fluid flow. The following figure shows a performance increase in heat exchange at Re_H=280, expressed through Nusselt number, using louvered instead flat fins. The comparison shows that the presence of louvers causes a sharp increase in local Nusselt number on the leading edge of each louver. A similar effect is also present on the flat fin but only on the leading edge of the fin itself.

The results show that the heat transfer improves with increased leading edge size which can be achieved by increasing the length of the louver itself (Figure 9) or increasing the number of louvers (Figure 10). This corresponds to the conclusions stated in the analysis of heat transfer performance influenced by varying individual fin louver parameters.

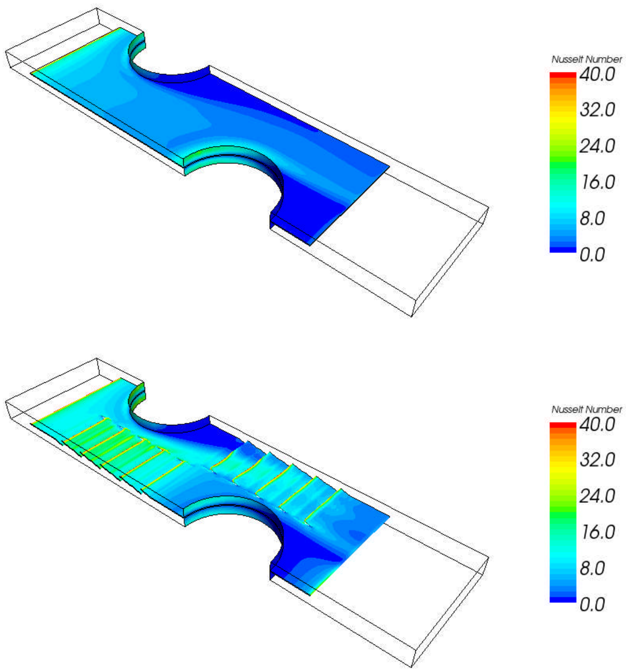

Figure 13 Local Nusselt number comparison between a flat and a louvered fin

In both flat and louvered fins, the zones of lessened heat transfer are situated behind the tubes. Additional improvements in fin geometry, such as vortex generators, could reduce these zones and improve the total boundary heat transfer from the fin surface.

CONCLUSION

A fluid flow solver was used to compare the performance between heat exchangers with flat and louvered fins. Prior to that, numerical simulations and experimental results have been compared in order to validate the procedure. The average value of calculated Nusselt number values appear overestimated by about 10% in the whole examined range of Re_H numbers. The numerical results should approach experimental ones more closely when final conduction is taken into account, which is the next step in this research.

Analyses with Re_H=280 determined that an increase in louver length has the greatest and almost linear influence on heat exchange improvement but is at the same time limited by transverse tube pitch (P_t). It is possible to further enhance performance by increasing the number of louvers and their angle. Increasing the louver angle over 15° serves no propose as it provides minimal improvements in performance at the expense of significantly increased pressure drop.

The comparison of heat exchanger performance with flat and louvered fins was performed over the range of Re_H numbers within 70-350. The greatest increase in heat transfer performance of 58% was obtained at the Re_H=350 when using louvered instead of flat fins. The interrupted surface can then provide a higher average heat transfer coefficient due to the periodical renewal of the boundary layer development when the heat transfer from a solid boundary to the moving fluid reaches its peak intensity. Care should be taken when choosing an optimal shape for louvered fins as the louvers noticeably increase the pressure drop.

REFERENCES

[1] Davenport C.J., Heat transfer and flow friction characteristics of louvered heat exchanger surfaces, In: J. Taborek, G.F. Hewitt and N. Afgan, Editors, Heat Exchangers, Theory and Practic, Hamisphdre, 1983, pp 397-412.

[2] Achaichia A., Cowell T.A., Heat transfere and flow friction characteristics of flat tube and louvered plate fin surfaces, Exp. Thermal Fluid Sci. 1, 1988, pp. 147-157.

[3] Chang Y.J., Wang C.C., A generalized heat transfer correlation for louver fin geometry, International Journal of Heat and Mass Transfer, Volume 40, Number 3, 1997., pp. 533-544.

[4] Wang C.C., Lee C.J., Chang C.T., Lin S.P., Heat transfer and friction correlation for compact louvered fin-and-tube heat exchangers, International Journal of Heat and Mass Transfer, Volume 42, 1998. , pp. 1945-1956.

[5] Wang, C.C., Chang, Y.P., Chi, K.Y., Chang, Y.J., A study of non-redirection louver fin-and-tube heat exchangers, Proceedings of the I MECH E Part C Journal of Mechanical Engineering Science, Volume 212, 1998. , pp. 1-14.

[6] Leu J.S., Liu M.S., Liaw J.S., Wang C.C., A numerical investigation of louvered fin-and-tube heat exchangers having circular and oval tube configurations, International Journal of Heat and Mass Transfer, Volume 44, Number 22, 2001 , pp. 4235-4243.

[7] Gilmour J.D., Whitby J.A., Turner G., Atkinsona K.N., Drakulic R., Heikal M.R., Cowell T.A., Two- and three-dimensional numerical models of flow and heat transfer over louvred fin arrays in compact heat exchangers, International Journal of Heat and Mass Transfer, Volume 41, Number 24, 1 1998 , pp. 4063-4080.

[8] Tiwari S., Maurya D., Eswaran V., Heat transfer enhancement in cross-flow heat exchangers using oval tubes and multiple delta winglets, International Journal of Heat and Mass Transfer, Volume 46, Issue 15, July 2003, pp. 2841-2856.

[9] Sahin, H.; Dal, A.R.; Baysal, E., 3-D Numerical study on the correlation between variable inclined fin angles and thermal behavior in plate fin-tube heat exchanger, Applied Thermal Engineering, v 27, 2007, pp. 1806-1816.

[10] Mi Sandar M., Gross U., Numerical study of fin-spacing effects in annular-finned tube heat exchangers, International journal of heat and mass transfer, 2004, vol. 47, no8-9, pp. 1953-1964.

[11] Taler D., Prediction of heat transfer correlations for compact heat exchangers, Forschung im Ingenieurwesen, Volume 69, Springer_Verlag 2005, pp. 137-150.

[12] Versteeg, H.K., Malalasekera, W., An Introduction To CFD The Finite Volume Method, ISBN 0-582-21884-5, Longman Gr., 1995.

[13] Ferziger J.H., Peric M., Computational methods for fluid dynamics, Third Edition, Springer, New York, 2002.

[14] Star-CCM+ User Guide, CD-Adapco, 2007.