On the comparison between compound louvered-vortex generator fins and X-shaped louvered fins

Bernd Ameel, Henk Huisseune, Joris Degroote, Michel De Paepe

Department of flow, heat and combustion mechanics, Ghent University-UGent

Bernd.Ameel@Ugent.be

Abstract. A recent evolution in heat exchanger design is the use of compound designs. One of the designs under study is a combination between a louvered fin and vortex generators. Several possible placements of the vortex generators are studied. These compound designs are compared with the X-shaped louvered fin, which maximizes the louvered area. It is shown that the X-shaped louvered fin exhibits the same heat transfer enhancement mechanism as the compound design, with respect to the rectangular louvered fin. The X-shaped louvered fin outperforms all of the compound designs.

1. Introduction

Tube-fine heat exchangers are commonly used for different applications, such as heat pump heat exchangers. Several different fin enhancements exist to obtain improved performance over the simple plain fin and tube heat exchanger. Louvered fins are a popular fin enhancement technique and have been investigated thoroughly. Both numerical and experimental investigations are available in literature. A logical next step would be the combination of several fin enhancement techniques, the so called compound designs. Lawson and Thole studied the combination of louvers and vortex generators for flat tube heat exchangers [1]. Of particular interest is the work of Huisseune et al. [2], where a compound design of a round tube-and-fin heat exchanger with louvered fins and vortex generators is studied. Rectangular louvers were studied and the vortex generators were placed in the unlouvered zones. The authors find that judicious placement of the vortex generators reduces the tube wake zone. This results in increased heat transfer and reduced form drag. This design is shown in Figure 1b.

Instead of placing vortex generators, another option would be to extend the louvered area around the tubes. This results in the so-called X-shaped design, illustrated in Figure 1a. Similar designs were experimentally studied by Wang et al. [3] and numerically by Hsieh and Jang [4]. In the X-shaped design there is no more space for large vortex generators. Smaller vortex generators could be placed in between louvers, but they would no longer reduce the tube wake. As has been shown by Huisseune et al., this is the most important effect by which the vortex generators improve performance.

In this work a comparison will be made between several compound louvered-vortex generator combinations, and the X-shaped louver.

2. The different geometries

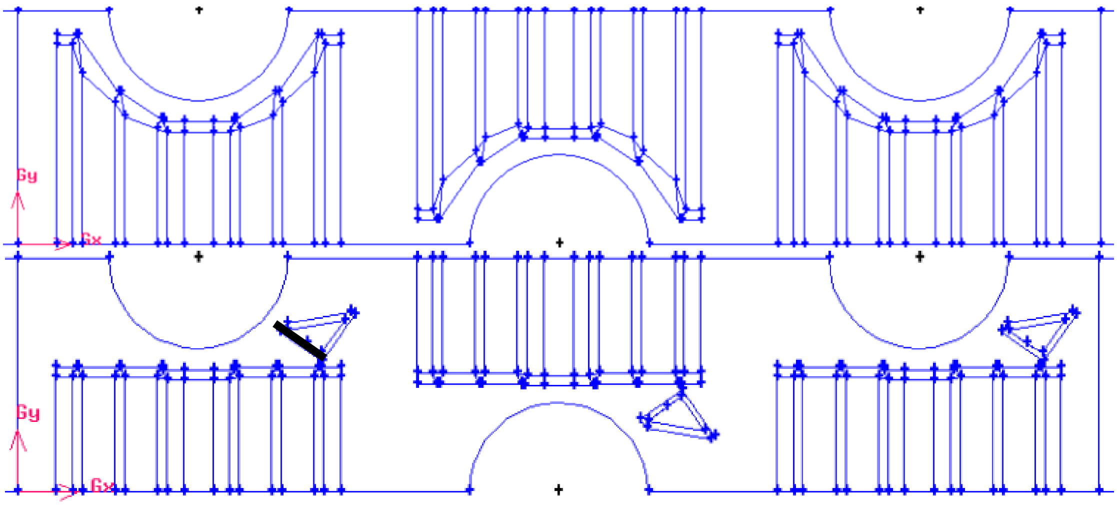

Figure 1 shows a) the X-shaped design (type 2) and the louvered and b) vortex generator compound design with common flow down delta winglet vortex generators.

Several designs for the louvered fin heat exchanger are compared, two of which are shown in Figure 1. The first design under study is a reference design with rectangular louvers. Panel a shows an X-shaped louvered fin design. A second variant is also studied (type 1), the main difference is the length of the flat landing. The louvers of the type 2 design are positioned at the same distance as the reference design, whereas in the type 1 design the louvers are placed as closely as possible to the tubes. The other designs are all compound designs, combining rectangular louvers with vortex generators. Three different positions are used for delta winglet vortex generators: common flow down, common flow up, and reversed common flow up [5]. These vortex generators are punched from the fin material. The requirement that the vortex generators are punched severely limits the possible placements for the vortex generators. For the reversed vortex generator, the upstanding edge is the leading edge, for the other two designs it is the trailing edge. The common flow down configuration is shown in panel b of Figure 1. Common geometrical parameters of the different designs are summarized in Table 1.

Table 1. Geometrical parameters

Parameter	Symbol	Range
Fin pitch	F_p	1.71 mm
Transversal tube pitch	P_t	17.6 mm
Longitudinal tube pitch	P_l	13.6 mm
Fin thickness (for material)	t_f	0.12 mm
Louver angle	θ	35°
Louver pitch	L_p	1.5 mm
Tube outer diameter	D_o	6.75mm
Delta winglet angle of attack	α_{VG}	35 °
Delta winglet height	h_{VG}	$0.9F_p$
Delta winglet base	b_{VG}	$0.9F_p$

The physical meaning of these parameters is illustrated in Figure 2, which shows a 3D representation of the heat exchanger part of the computational domain. The delta winglet vortex generators are positioned as closely to the tubes as possible, as shown on Figure 1b.

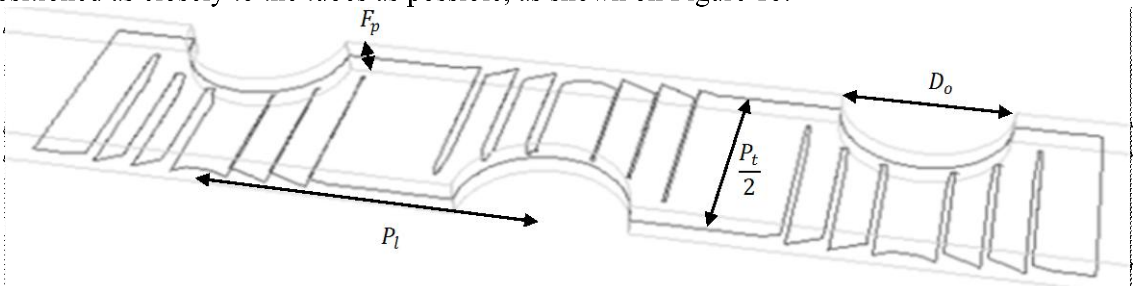

Figure 2. Heat exchanger geometry

3. Numerical method and data reduction

CFD simulations are performed for frontal speeds ranging from 0.6 m/s to 2.6 m/s, corresponding to a Reynolds number range of 100 to 515, defined on the frontal speed and on the hydraulic diameter. The hydraulic diameter is defined according to Wang et al. [3]. The calculations of all the geometries are performed using steady and laminar flow. For the Reynolds number range under consideration, the laminar and steady assumptions are reasonable for the flow in the heat exchanger core. However, at the heat exchanger exit, the flow is physically unsteady. Therefore the steady calculation is only an approximation of reality, which is expected to give reasonable results for quantities relating to the heat exchanger core. This is in fact the case, as shown by Leu et al. [6].

The thickness of the fin material is neglected for the flow domain. This greatly simplifies the meshing process. It has been shown by Hsieh and Jang [4] that there is no detectable influence of the fin thickness on the heat transfer characteristics. Therefore a value of zero is imposed on the fin thickness for the flow domain. The material heat transfer is calculated by a 2D material model (thin shell conduction), the boundaries are coupled to the fluid domain. This allows for taking fin efficiency effects into account. The temperature in the fin material is calculated by assuming 0.12 mm material thickness. A constant tube wall temperature is imposed, neglecting the conductive temperature drop across the tube wall, as well as the contact resistance between the fin and the tubes.

Symmetric boundary conditions are used for the transversal boundaries of the flow and material domain. For the top and bottom boundaries of the computational domain, periodic boundary conditions are imposed. The inlet is located 2 tube diameters upstream from the heat exchanger core. This allows the velocity profile to develop ahead of the heat exchanger, to accommodate the contraction in the core. At the inlet a uniform velocity profile is imposed, with a constant magnitude equal to the frontal velocity. To take the pressure recovery due to mixing into account and to avoid recirculation at the heat exchanger exit, the exit of the flow domain is situated 10 tube diameters behind the heat exchanger core. Simulations are performed using ANSYS Fluent.

Second order upwind discretization was used for convective terms in the momentum and energy equations. The diffusive terms are discretized with a second order central differencing scheme. The pressure interpolation is also formally second order accurate. The grid independency was checked by performing the calculations on two separate grids. The coarse grid has an average cell size of 130 µm and contains 4 million cells. The average cell size on the fine grid is 82µm and consists of 11 million cells. Over the investigated speed range, the maximum deviation of the f factor is 0.9% (for the lowest velocity). The maximum deviation on the Colburn j factor is 1.2% (for the highest velocity).

From the simulations the pressure drop and heat transfer rate are determined. These quantities are made dimensionless by using the modified Colburn j factor and the friction factor. The friction factor is determined according to the method used by Wang [8], incorporating entrance and exit losses in the core friction term.

$$j = St.Pr^{\frac{2}{3}}.\eta_o = \frac{U}{Cp.\rho.V}.Pr^{\frac{2}{3}} \qquad\qquad f = \frac{A_c}{A_o}\frac{\rho_m}{\rho_l}\left[\frac{2\Delta P \rho_l}{G_c^2} - (1+\sigma^2)\left(\frac{\rho_1}{\rho_2} - 1\right)\right]$$

4. Performance evaluation

In order to compare the performance of these different fin types, the method of Cowell [9] is used. For all fin designs under comparison, the tube and fin pitches are kept fixed. As a consequence, the hydraulic diameter and contraction factor are constant for all designs. By additionally fixing the mass flow rate and the number of transfer units, the heat transfer of the heat exchanger is also fixed. Under these conditions, it is shown by Cowell that the following equations hold for the relative volume V^* and the relative fan power P^*.

$$V^* = \frac{1}{Re.j} \qquad\qquad P^* = \frac{f.Re^2}{j}$$

In general, both the j factor and the f factor are a function of the fin geometry and the Reynolds number of the flow. These values are provided by the simulations, by calculating the different designs for several inlet velocities. Therefore, the quantities V^* and P^* are functions of the Reynolds number and the geometry. To compare the performance of the heat exchangers, the relative volume will be plotted as a function of the relative fan power, as a parametric equation of the Reynolds number. This is done in Figure 3. As a reference, the plain fin is also indicated on the figure. The plain fin was calculated with the same numerical method, boundary conditions and general geometry parameters as the other designs.

The X-shaped design version 2 corresponds to the X-shaped design where the louvers are separated from the tubes by the same distance as is done for the rectangular louvers. For the rectangular louvers this is necessary to provide enough space for the vortex generators, a constraint which is not applicable for the X-shaped design. For the X-shaped design, manufacturability is the only constraint.

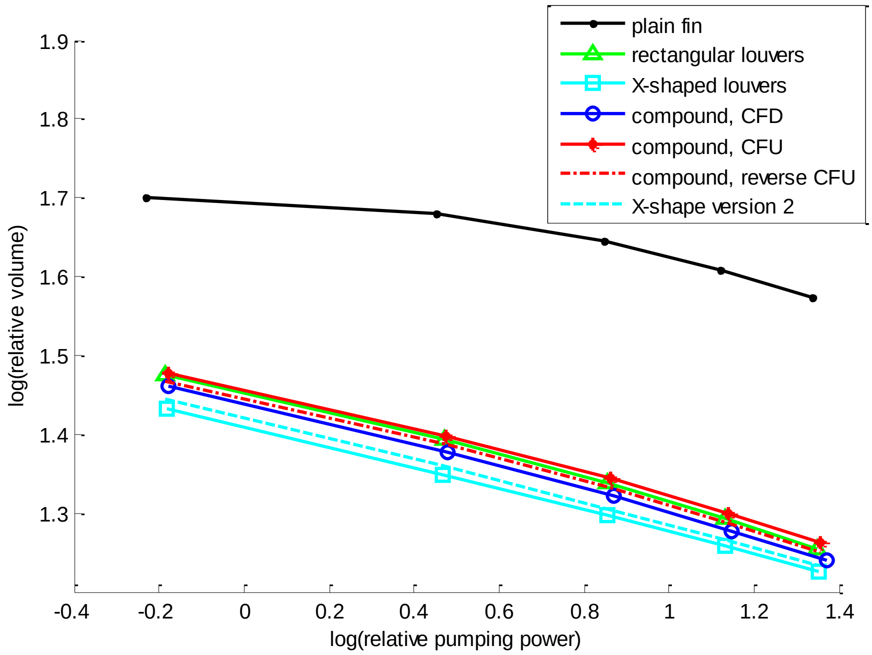

Figure 3 Performance evaluation of the different designs.

Figure 3 shows that the different designs are all vastly superior to the plain fin. The compound design with common flow up (CFU) vortex generators performs slightly worse than the simple rectangular louvered design. The reversed CFU vortex generators result in a small improvement. The common flow down (CFD) vortex generators outperform the reversed CFU vortex generator design for all Reynolds numbers under consideration.

Even though the compound design combining CFD vortex generators and rectangular louvered fins offer an improvement over the simple rectangular louvered fin, the X-shaped louvered fin outperforms this compound design. When the X-shaped louvers are placed closer to the tubes (version 1), the performance is improved even further. This is in agreement with the findings of Tafti and Cui [10], who investigated the fin-tube junction effects for louvered fin heat exchangers with flat tubes. They concluded that the distance between the louver transition zone and the tube (i.e. the flat landing) should be as small as possible.

Compared to the simple rectangular louvers, CFD vortex generators reduce the necessary volume by at least 3%, for the same pressure drop. The X-shaped louvers placed at the same distance from the tubes allow for a reduction of the volume by 6%. When the louvers are placed closer to the tube, the reduction of the heat exchanger volume is 7.5%. For lower speeds the improvement can increase to up to 10%.

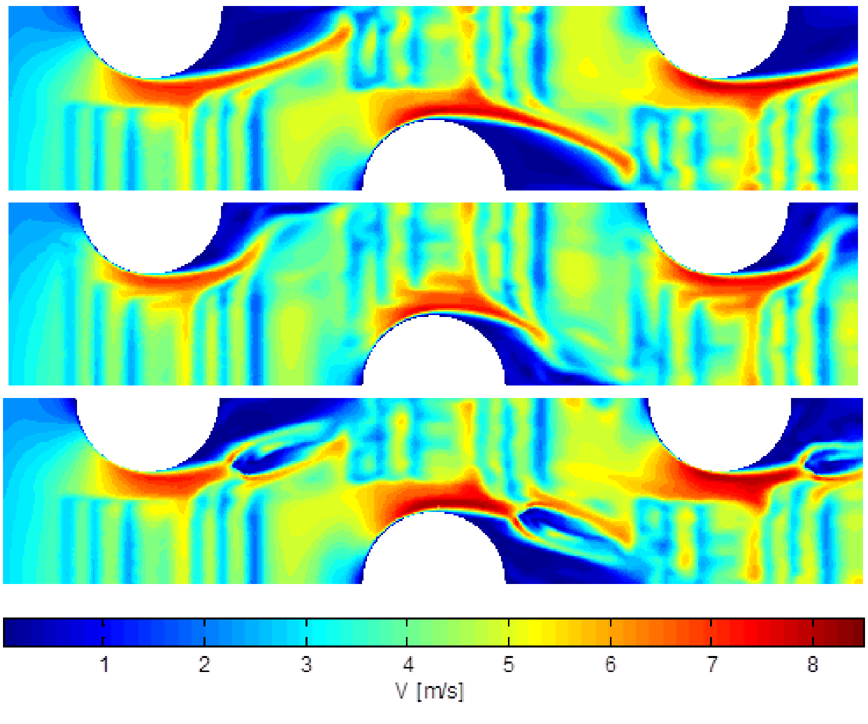

Figure 4 Velocity magnitude in the middle plane between two fins. a) rectangular louvers, b) X-shaped louvers, c) CFD compound design

Figure 4a shows that for the baseline design, there are large wake zones behind the tubes, where the velocity is low. This separation of the flow increases the pressure drop through the heat exchanger due to form drag. Because of the low velocity, heat transfer in these regions is also low due to the low local convection coefficient. A high velocity jet is present next to the tubes, which bypasses the louvers. This is caused by the increased flow resistance for flow which passes through the louvers, due to the frequent restarting of the boundary layer. The X-shaped louvers deflect the high velocity jet towards the tube wake. This is because of the shape of the bypass channel between the flat landings of the louvers and the tube. The tube wake zone is greatly reduced because of this impinging jet. As was already indicated by Huisseune, the vortex generators improve heat exchanger performance due to this tube wake reduction. For the CFD compound design, the high velocity jet impinges on the vortex generator, shedding vortices which promote flow mixing. This also reduces the size of the tube wake, but by a smaller amount. The X-shaped designs exhibit an even smaller tube wake and are therefore the design with the best performance. Placing the louvers closer to the tubes as is done for the type 1 x-shaped louvers reduces the bypass flow, which also results in increased performance.

5. Conclusions

Different enhancement techniques for louvered fin heat exchangers were studied. The successful techniques were shown to result in a smaller tube wake and a reduction in the tube-louver bypass flow. Common flow down vortex generators improve the heat exchanger performance compared to rectangular louvers. However, X-shaped louvered designs offer even larger improvements. All enhancement techniques offer greatly increased performance with respect to the plain fin. The X-shaped louvers with minimal flat landings can decrease the required heat exchanger volume by 7.5% for the same fan power, with respect to the baseline case of rectangular louvers.

References

[1] M. J. Lawson and K. A. Thole, "Heat transfer augmentation along the tube wall of a louvered fin heat exchanger using practical delta winglets," *International Journal of Heat and Mass Transfer,* vol. 51, pp. 2346-2360, May 2008.

[2] H. Huisseune, "Performance evaluation of louvered fin compact heat exchangers with vortex generators," Ghent University. Faculty of Engineering and Architecture, 2011.

[3] C. C. Wang, C. J. Lee, C. T. Chang, and S. P. Lin, "Heat transfer and friction correlation for compact louvered fin-and-tube heat exchangers," *International Journal of Heat and Mass Transfer,* vol. 42, pp. 1945-1956, Jun 1999.

[4] C.-T. Hsieh and J.-Y. Jang, "Parametric study and optimization of louver finned-tube heat exchangers by Taguchi method," *Applied Thermal Engineering*.

[5] A. M. Jacobi and R. K. Shah, "Heat-Transfer Surface Enhancement through the Use of Longitudinal Vortices - a Review of Recent Progress," *Experimental Thermal and Fluid Science,* vol. 11, pp. 295-309, Oct 1995.

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

[7] P. J. Roache, "Quantification of uncertainty in computational fluid dynamics," *Annual Review of Fluid Mechanics,* vol. 29, pp. 123-160, 1997.

[8] C. C. Wang, R. L. Webb, and K. Y. Chi, "Data reduction for air-side performance of fin-and-tube heat exchangers," *Experimental Thermal and Fluid Science,* vol. 21, pp. 218-226, Apr 2000.

[9] T. A. Cowell, "A General-Method for the Comparison of Compact Heat-Transfer Surfaces," *Journal of Heat Transfer-Transactions of the Asme,* vol. 112, pp. 288-294, May 1990.

[10] D. K. Tafti and J. Cui, "Fin–tube junction effects on flow and heat transfer in flat tube multilouvered heat exchangers," *International Journal of Heat and Mass Transfer,* vol. 46, pp. 2027-2038, 2003