Numerical study of a round tube heat exchanger with louvered fins and delta winglets

H Huisseune¹, C T’Joen¹, P De Jaeger¹,², B Ameel¹ and M De Paepe¹

¹Ghent University, Department of Flow, Heat and Combustion Mechanics, Sint-Pietersnieuwstraat 41, 9000 Gent, Belgium
²NV Bekaert SA, Bekaertstraat 2, 8550 Zwevegem, Belgium

E-mail: Henk.Huisseune@UGent.be

Abstract. Louvered fin and round tube heat exchangers are widely used in air conditioning devices and heat pumps. In this study the effect of punching delta winglet vortex generators in the louvered fin surface is studied numerically. The delta winglets are located in a common-flow-down orientation behind each tube of the staggered tube layout. It is shown that the generated vortices significantly reduce the size of the tube wakes. Three important heat transfer enhancement mechanisms can be distinguished: a better flow mixing, boundary layer thinning and a delay in flow separation from the tube surface. The compound heat exchanger has a better thermal hydraulic performance then when only louvers or only delta winglets are used. Comparison to other enhanced fin designs clearly shows its potential, especially for low Reynolds number applications.

1. Introduction

Every day large amounts of heat are transferred in many industrial and domestic processes. Any energy savings in the heat transfer process have a significant impact on the fuel consumption and the greenhouse gas emissions. In this way, more energy efficient heat exchangers help to meet the 20-20-20 climate and energy targets of the European Union. In many applications air is one of the working fluids (e.g. coolers in compressed air systems, heat pumps, air conditioning devices, domestic heating, etc.). When heat is exchanged with air, the main thermal resistance is located at the air side of the heat exchanger. To increase the heat transfer rate, the heat transfer surface is enlarged by adding fins to the air side. When a high compactness is desired, complex interrupted fin patterns are used. A typical example is the louvered fin design, see Figure 1a. This design consists of an array of flat plates set at an angle to the incoming flow. The flow deflection depicted in Figure 1b is characteristic for louvered fin heat exchangers [1].

![Figure 1](image_url)

Figure 1. (a) Louvered fin design with main geometrical parameters; (b) flow deflection
The main disadvantage of the louvered fins is the high pressure drop. Delta winglets (DW) mounted on a heat transfer surface generate vortices (Figure 2a) which cause an intense mixing of the flow and thin the thermal boundary layers. In contrast to louvered fins, they enhance the heat transfer with a relatively low penalty in pressure drop [2]. An appropriate DW placement and orientation in fin-and-tube heat exchangers reduces the poor heat transfer region in the tube wakes. To this purpose, the common-flow-down configuration of Figure 2b is frequently used [3].

![Figure 2. (a) Delta winglet pair on a flat plate generating longitudinal vortices [3]; (b) Common-flow-down orientation of DWs on the fin of a round tube heat exchanger [4].](image)

The next generation of heat exchangers combines known enhancement techniques, resulting in so called compound designs [5]. The aim is that the compound design results in a higher performance than the individual techniques applied separately. To the authors’ knowledge, only a few studies on compound designs with louvered fins and vortex generators can be found in literature. Joardar and Jacobi [6] tested a louvered fin heat exchanger with flat tubes before and after adding leading edge delta wings on the heat exchanger face. By adding delta wings the average heat transfer enhancement was 21% under dry conditions and 23.4% under wet surface conditions for inlet velocities between 1 and 2 m/s. The associated pressure drop penalty was about 6%. Joardar and Jacobi [6] believe that further improvements are possible by optimizing the wing geometry and placement. Lawson and Thole [7] stamped delta winglets into the flat landings between the louvers and flat tube of a heat exchanger and they evaluated the tube wall heat transfer augmentation and associated pressure drop. They found an enhancement in tube wall heat transfer up to 47% with a corresponding pressure drop penalty of 19%.

As louvered fins are frequently used in many applications, a compound design of louvered fins and vortex generators might have a wide applicability. However, only a few studies on this kind of compound design were found in literature and they all focused on flat tube heat exchangers (typically for automotive applications). Hence, the objective of this work is to study the effect of delta winglets punched in the louvered fins of a round tube heat exchanger. A three-dimensional numerical study was performed using Computational Fluid Dynamics (CFD).

2. Computational domain and procedure
The three-dimensional computational domain is shown in Figure 3. Three tube rows in a staggered arrangement are considered. Delta winglet vortex generators are punched out of the louvered fin surface in a common-flow-down arrangement behind each tube. Each louver element between the tubes consists of an inlet louver, an exit louver and two louvers on either side of the turnaround louver. Each louver transitions from an angle $\theta$ into a flat landing adjoining the tube surface. Periodic conditions are applied on both sides of the domain as well as on the top and bottom. The height of the computational domain is equal to the fin pitch $F_p$ and the width is equal to transversal tube pitch $P_t$. The fin surface is located halfway the domain height. To take the inlet and exit effects into account, the domain is extended. The entrance length upstream of the fin equals 5 times the fin pitch $F_p$ and the domain extends 7 times the tube diameter $D_o$ downstream of the fin. The dimensions of the compound heat exchanger are listed in Table 1. They are selected based on a literature review of louvered fins and plain fins with delta winglets.
The mesh was generated using Gambit©. The air domain as well as the solid fin material were meshed to take the fin conduction into account. The quality of the mesh was carefully assessed during the meshing. The computational domain was divided into several subdomains. The fin material was meshed with quad elements. Most of the air subdomains were also meshed with quad elements. Only the subdomains with the transition zone between the angled louver and flat landing and the subdomains surrounding the delta winglets were meshed using unstructured tetrahedral elements. Boundary layers meshes were applied on the fin surface. The grid independency was confirmed by comparing the simulations results of the mesh used in this study (about 4,300,000 cells) to the simulations results of a finer mesh (about 14,000,000 cells). The commercial code ANSYS Fluent© is used for the simulations. The flow is assumed to be laminar, which is acceptable in the considered Reynolds range ($Re_{Lp} = 110 – 860$). At the inlet a uniform velocity parallel to the fin was imposed and the air temperature was set to 20°C. At the outlet the static pressure was set to 0 Pa (pressure outlet boundary condition). A constant tube wall temperature of 50°C was applied in the three tube rows.

The double precision segregated solver was used to solve the standard Navier-Stokes equations. The energy equation was turned on to compute heat transfer through the fin material and in the air. The SIMPLE algorithm was applied for the pressure-velocity coupling (Semi-Implicit Method for Pressure-Linked Equations [8]). The discretization of the convective terms in the governing equations was done via a second order upwind scheme, while a second order central differencing scheme was applied for the diffusive terms. The gradients were evaluated via the least squares cell based method. The pressure gradient in the momentum equations was treated via a second order discretization scheme. Convergence criteria were set to $10^{-8}$ for continuity, velocity components and energy. Setting smaller values for these criteria did not result in any notable differences in the flow field and heat transfer predictions. The air density was calculated as for an incompressible ideal gas, the specific heat and thermal conductivity were set to constant values ($c_p = 1006$ J/kgK; $\lambda = 0.02637$ W/mK ) and the dynamic viscosity was calculated with the Sutherland approximation. The fin has a thermal...
conductivity of 202.4 W/mK. Only for the smallest Reynolds numbers (Re_Dh < 200) steady simulations were found to converge. For higher Reynolds numbers unsteady simulations were performed. The time step varied between 1 µs and 1 ms (dependent on the Reynolds number). This allowed the residuals to decrease below 10^-8 in less than 50 iterations per time step. The mass-weighted average pressure drop and outlet temperature were monitored during the iterations to determine if the simulations had converged [9]. Local temperatures in the tube and louver wakes were also monitored. Unsteady data were averaged out over the time interval an air particle needs to travel three times the length of the computational domain. Averaging out over a longer time interval did not result in any notable differences in the simulation results. A validation experiment was performed which showed that there is an acceptable match between the simulation results and the experimental data [10].

3. Results and discussion

3.1. Effect of the delta winglet vortex generators

Figure 4 shows the temperature contours in planes parallel to the fin surface for the baseline geometry without delta winglets and the compound heat exchanger with delta winglets for an inlet velocity of 5.25 m/s. This inlet velocity corresponds to a Reynolds number Re_{Lp} = 857 / Re_{Dh} = 1216. The Reynolds numbers are based on the velocity V_c in the minimum cross sectional flow area and the louver pitch L_p or hydraulic diameter D_h, respectively:

\[ D_h = \frac{4A_cL}{A_o} \tag{1} \]

with A_c the minimum cross sectional flow area, L the flow depth and A_o the total heat transfer surface area. In Figures 4a and 4b the temperature field is plotted in a plane located at 15% of the fin spacing under the fin surface, while Figures 4c and 4d show the temperature field in a plane at 15% of the fin spacing above the fin surface. If no delta winglets are present (Figure 4a-c), the wake zones behind the tubes are very pronounced. The air temperature in these zones is very high (close to 50°C, i.e. the tube wall temperature) which indicates that these are regions of low heat transfer. By punching delta winglets in a common-flow-down configuration behind each tube, the size of the wake zones is significantly reduced (Figure 4b-d). The temperatures behind the tubes are lower and thus there is a better heat transfer. This effect is mainly seen in the first and second tube row. In the third tube row, however, the influence of the vortex generators on the local temperatures is much less because the generated vortices immediately leave the heat exchanger.

![Temperature contours in planes parallel to the fin surface at V_{in} = 5.25 m/s (Re_{Lp} = 857 / Re_{Dh} = 1216). Left column: without delta winglets; right column: with delta winglets](image-url)

Comparing Figures 4b and 4d shows that there is a clear difference in the shape of the wake zones underneath the fin surface and above the fin surface. In Figure 4b (closely under the fin surface) the flow narrows the wake and the wake seems more closed, while Figure 4d (closely above the fin surface) suggests that the hot air is removed from the wake towards the mainstream region. This is due to the vortex effect of the delta winglets on the tube wakes. To illustrate this, velocity vectors are
plotted in planes parallel to the inlet downstream of the delta winglets in the first tube row. Four different streamwise positions are considered, labeled (b) through (e) in Figure 5a. The velocity vectors in the first tube row as viewed from the outlet are shown in Figure 5b-e (inlet velocity $V_{in} = 5.25$ m/s ($Re_{Lp} = 857 / Re_{Dh} = 1216$)). Plane (b) is located immediately downstream of the punched holes in the fin surface. The longitudinal vortices generated by the delta winglet pair are indicated by $\textcircled{1}$. Figure 5b indicates that the rotation direction of the longitudinal vortices is such that the air in between the vortex cores flows down towards the fin surface. This downwash region (indicated by $\textcircled{2}$) explains the name “common-flow-down” of the delta winglet configuration. The tube wakes are zones of poor heat transfer. The swirling motion associated with the vortex generation causes hot air to be removed from the tube wake to the mainstream regions and vice versa. This enhanced mixing is an important mechanism of heat transfer enhancement. Another heat transfer enhancement mechanism is caused by the downwash flow on the fin with winglets. The induced wall-normal flow locally thins the boundary layer, which also improves heat transfer. On the other side of the fin (without winglets) an upwash region (indicated by $\textcircled{3}$) exists which yields a decrease in heat transfer. Joardar and Jacobi [3] explained that the heat transfer enhancement in the downwash region is greater than the reduction in the upwash region. Hence, a net heat transfer enhancement results. Figure 5b also shows recirculation in the wake regions behind the delta winglets, indicated by $\textcircled{4}$: a counter-rotating vortex is apparent. Three more downstream velocity planes are shown in Figure 5. The vortex strength rapidly reduces with downstream distance. Plane (c) corresponds to the entrance of the louver bank in the second tube row. Further downstream the longitudinal vortices can no longer be discerned. They are destroyed by the upward air flow which follows the louvers. The louvers are represented by the horizontal gray zones in Figures 5d and 5e. Thus, the vortices do not propagate far downstream due to the flow deflection in the downstream louver bank. This is in contrast to plain fins with vortex generators where the vortices persist for several wing spans downstream and enhance the heat transfer over large fin areas [3]. The flow field downstream of a DW pair of the second tube row is similar to the flow field downstream of a DW pair of the first tube row (not shown here due to space restrictions).

![Figure 5](image)

Figure 5. (a) Velocity planes parallel to the inlet downstream of the DWs in the first tube row; (b)-(e) velocity vector field at $V_{in} = 5.25$ m/s ($Re_{Lp} = 857 / Re_{Dh} = 1216$)

The vortex generators do not only affect the fin surface heat transfer, but also the tube surface heat transfer. The separation point on the tube surface is moved downstream which results in a reduced wake size. In Figure 6 the X component of the wall shear stress averaged over the tube height is plotted along the tube circumference for a tube in the first and last tube row. The point of separation is where the wall shear stress vanishes. Flow reversal is indicated here by negative values of the X component of the wall shear stress. The separation behavior in the second tube row is very similar to the first tube row. That is why it is not shown here. In the first tube row (Figure 6a) flow separation from the tube surface occurs at about 120° for the baseline geometry without delta winglets. When
delta winglets are added the separation is delayed and flow separation occurs at about 130°. Also note that at about 150° there is a reattachment of the flow to the tube surface (positive values of the X component of the wall shear stress), which is not the case for the baseline configuration. Compared to the first (and second) tube row, the flow separation angle in the third tube row is smaller. This is shown in Figure 6b. The separation angle is about 110° for the baseline geometry and 120° for the compound heat exchanger. The tube wake behavior in the third tube row tends towards the wake flow behind an infinite cylinder because the tubes in the last tube row are located near the fin trailing edge.

Figure 6. X component of wall shear stress averaged over tube height along the tube circumference for a tube in the first (a) and last (b) tube row (\(V_{in} = 2.69\) m/s (Re\(_{Lp} = 434\) and Re\(_{Dh} = 616\))

3.2. Performance evaluation of the compound heat exchanger

The performance of the compound heat exchanger studied in this work is compared to the performance of the corresponding heat exchangers in which the individual enhancement techniques are applied separately (thus only delta winglets or only louvers -- geometry as in Table 1). The LMTD method was used to simulate the heat transfer and pressure drop [9]. The performance of the heat exchangers is evaluated based on the Colburn j-factor and the friction factor f. They are plotted as function of the Reynolds number Re\(_{Dh}\) (Eq. (1)). To allow a comparison in terms of total heat transfer surface area or core volume, the performance evaluation criterion of Soland et al. [11] is used. They suggested evaluating the thermal hydraulic performance of heat exchangers by plotting the heat transfer performance factor J\(_n\) (Eq. (2)) as function of the pumping power factor F\(_n\) (Eq. (3)). The contraction ratio \(\sigma\) is defined as the ratio of the minimal cross sectional flow area to the frontal area. J\(_n\) is proportional to the heat transfer per unit volume and F\(_n\) is proportional to the pumping power per unit volume. J\(_n\) vs. F\(_n\) is thus a modification of the volume goodness factor proposed by London and Ferguson [12].

\[
J_n = \sigma \cdot \frac{j_{ReDh}}{D_h^2} \tag{2}
\]

\[
F_n = \sigma \cdot \frac{F_{ReDh}}{D_h^2} \tag{3}
\]

Figure 7. Comparison of the simulation data with louvered fins [13], slit fins [14] and plain fins [15] (fixed F\(_p\)): (a) Colburn j-factor, (b) friction factor and (c) modified volume goodness factor
The symbols in Figure 7 represent simulated data. The Colburn j-factors of the compound design are higher than when the individual enhancement techniques are applied separately (up to 16% higher compared to the louvered fin heat exchanger and up to 87% higher compared to the delta winglet heat exchanger). Also the associated friction factors are higher (up to 37% higher compared to the louvered fin heat exchanger and up to 156% higher compared to the delta winglet heat exchanger). If heat transfer as well as pressure drop are considered in terms of the volume goodness factor, then the compound design outperforms the simulated louvered fin heat exchanger and delta winglet heat exchanger. For a fixed pumping power per unit volume, the heat transfer per unit volume of the compound design is up to 14% higher compared to the louvered fin heat exchanger and up to 72% higher compared to the delta winglet heat exchanger. For the same thermal hydraulic performance, the compound heat exchanger can thus be made smaller in size. As a result, the material cost is lower.

The performance of the compound heat exchanger is also compared to the performance of slit fin and louvered fin designs. These are the most widely used interrupted fin surfaces. Also plain fins are considered for comparison. Their Colburn and friction characteristics are determined with correlations found in literature (louvered fins [13], slit fins [14] and plain fins [15]). The literature correlations can only be used within the geometry ranges of the experimental data they are based on. Outside these ranges no reliable predictions can be made. Hence, the geometries used to evaluate the literature correlations were chosen within their applicability ranges. To make a fair comparison heat exchangers with the same fin pitch \( F_p \) were studied. The fin density (i.e. the number of fins per unit length or thus \( 1/F_p \)) affects the pressure drop, the suppression of the vortex development, the flow development length and - in the case of louvered fins - the flow deflection. Consequently, an identical fin pitch means that all these effects are the same for the studied heat exchangers. The dimensions of the heat exchangers for the comparison study are listed in Table 2. The fin pitch equals 1.71 mm, identical to the fin pitch of the simulated geometry of Table 1. For all three geometries this value is within the applicability range of the corresponding correlations. The values of the other geometrical parameters are the dimensions of existing heat exchangers with a tube diameter as close as possible to the simulated \( D_o = 6.75 \) mm. They were selected from the database which was used to fit the correlations. These databases are published in the respective papers [13-15]. The louvered fin heat exchanger of the database of Wang et al. [13] has the same number of louvers per louver array as the simulated compound heat exchanger and louvered only heat exchanger.

| Parameter                      | Symbol | Louvers [13] | Slits [14] | Plain [15] |
|--------------------------------|--------|--------------|------------|------------|
| Outer tube diameter            | \( D_o \) (mm) | 6.7          | 7.3        | 6.7        |
| Transversal tube pitch         | \( P_t \) (mm) | 17.6         | 20         | 17.6       |
| Longitudinal tube pitch        | \( P_l \) (mm) | 13.6         | 17.32      | 13.6       |
| Fin pitch                      | \( F_p \) (mm) | 1.71         | 1.71       | 1.71       |
| Fin thickness                  | \( t_f \) (mm) | 0.115        | 0.11       | 0.115      |
| Number of tube rows            | \( N \) | 2            | 3          | 2          |
| slit height                    | \( S_h \) (mm) | 1.6          |            |            |
| slit breadth                   | \( S_s \) (mm) | 1           |            |            |
| number of slits                | \( S_n \) | 7           |            |            |
| louver height                  | \( L_h \) (mm) | 1.4          |            |            |
| louver angle                   | \( \theta \) (°) | 39          |            |            |

Table 2. Heat exchanger dimensions used in the comparison study: the fin pitch \( F_p \) is fixed and the other dimensions are selected from the respective databases [13-15]

The full lines in Figure 7 correspond to the literature correlations. The louvered fin and slit fin correlations consist of two parts depending on the Reynolds number. This explains the discontinuity in the curves. The plain fin heat exchanger shows the lowest Colburn j-factor, friction factor and volume goodness. This heat exchanger is only used if a low pressure drop is demanded and volume is not an issue. If a high compactness is needed, interrupted fin surfaces are preferred. The thermal performance of the compound design is better than the thermal performance of the louvered fin and slit fin heat
exchanger. The difference between the compound design and the louvered fin heat exchanger is mainly significant at low Reynolds numbers. At higher Reynolds numbers the gain is small. As explained by Huisseune [10], at low Reynolds numbers the delta winglets highly contribute to the improved Colburn j-factors, while at higher Reynolds numbers the performance is mainly determined by the louver geometry. The discontinuity in the louvered fin Colburn correlation appears for $Re_{Dc} = 1000$ (Reynolds number based on the collar diameter $D_c$ and the velocity in the minimum cross section flow area $V_c$). This corresponds for the given geometry with an inlet velocity of about 1.5 m/s. Thus for low Reynolds applications, such as HVAC&R, the combination of louvered fins and delta winglets clearly shows potential. For domestic air conditioning devices, for instance, the inlet velocity is typically 1.3 m/s [16]. When using a compound design instead of a louvered fin heat exchanger the heat exchanger can be smaller in volume for the same heat duty (see Figure 7c). The associated pumping power is then also lower. Alternatively, for the same heat exchanger volume, the compound design can work at lower velocities. This is interesting for indoor units of domestic air conditioning systems, because then low velocities are preferred: a high velocity air jet blown into the room feels uncomfortable for the room occupants and lower velocities also cause less noise.

4. Conclusions

Three-dimensional numerical simulations were performed of a round tube heat exchanger with louvered fins and delta winglet vortex generators. The generated vortices cause three important mechanisms of heat transfer enhancement: a better mixing, a reduction of the thermal boundary layer thickness and a delay of the flow separation from the tube surface. Further, it was found that the vortices do not extend far downstream as they are destroyed by the deflected flow in the downstream louver bank. The compound heat exchanger has a better thermal hydraulic performance than when only vortex generators or only louvers are used. The performance of the compound design is also compared to louvered, slit and plain fin heat exchangers. Especially for low Reynolds applications, the compound heat exchanger can be made smaller in size and thus more economical in cost.

Acknowledgements

The authors would like to express gratitude to the FWO for the financial support.

References

[1] Zhang X and Tafti DK 2003 *INT J HEAT MASS TRAN* 46 pp 1737–1750
[2] Tiggelbeck S, Mitra NK and Fiebig M 1994 *J HEAT TRANS-T ASME* 116 pp 880-885
[3] Joardar A and Jacobi AM 2007 *J HEAT TRANS-T ASME* 129 pp 1156-1167
[4] Pesteel SM, Subbarao PMV and Agarwal RS 2005 *APPL THERM ENG* 25 pp 1684-1696
[5] Bergles AE 2002 *EXP THERM FLUID SCI* 26 pp 335-344
[6] Joardar A and Jacobi AM 2005 *INT J HEAT MASS TRAN* 48 pp 1480-1493
[7] Lawson MJ and Thole KA 2008 *INT J HEAT MASS TRAN* 51 pp 2346-2360
[8] Sunden B 2007 *HEAT TRANSFER ENG* 28(11) pp 898-910
[9] Huisseune H, T’Joen C, De Jaeger P, Ameel B, De Schampheleire S and De Paepe M 2012 Performance enhancement of a louvered fin heat exchanger by using delta winglet vortex generators *INT J HEAT MASS TRAN* under review
[10] Huisseune H 2011 Performance Evaluation of Louvered Fin Compact Heat Exchangers with Vortex Generators *Ph.D. dissertation*, Ghent University, Belgium
[11] Soland JG, Mack JWM and Rohsenow WM 1978 *J HEAT TRANS-T ASME* 100 pp 514-519
[12] London AL and Ferguson CK 1949 *Transactions of the ASME* pp 17-26
[13] Wang CC, Lee CJ, Chang CT and Lin SP 1999 *INT J HEAT MASS TRAN* 42 pp 1945-1956
[14] Wang CC, Lee WS and Sheu WJ 2001 *INT J HEAT MASS TRAN* 44(18) pp 3565–3573
[15] Wang CC, Chi KY and Chang CJ 2000 *INT J HEAT MASS TRAN* 43(15) pp 2693-2700
[16] Fujino H 2009 *Proceedings of the 7th International Conference on Enhanced, Compact and Ultra-Compact Heat Exchangers* (Costa Rica) pp. 201-207