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Numerical investigation of fin geometry on the air-side heat transfer and pressure drop characteristics of heat exchangers using in-line rectangular microchannels

B. Rogie\textsuperscript{a}, W. Brix Markussen\textsuperscript{b}, M. Ryhl Kærn\textsuperscript{c},

\textsuperscript{a} Technical University of Denmark, 2800 Kongens Lyngby, Denmark, brogie@mek.dtu.dk, CA
\textsuperscript{b} Technical University of Denmark, 2800 Kongens Lyngby, Denmark, wb@mek.dtu.dk
\textsuperscript{c} Technical University of Denmark, 2800 Kongens Lyngby, Denmark, pmak@mek.dtu.dk

Abstract:
The objectives of this research is to investigate the effect of different fin geometries on the performances of a microchannel heat exchanger, using CFD simulations. The fin profiles include straight fins, triangular fins, wavy fins and offset fins. For each fin geometry, air-side heat transfer and pressure drop correlations were developed and defined in terms of Colburn j-factor and Fanning f-factor. The microchannels are square 2.00 mm x 2.00 mm and placed in-line by 4.50 mm longitudinal tube pitch. The transverse tube pitch and the fin pitch were varied from 9.00 mm to 19.00 mm and 2.50 mm to 10.00 mm respectively, to investigate their influences on the heat exchanger performances. Furthermore, the complete thermal and hydraulic fully developed flow was covered using one CFD simulation, allowing to extract heat transfer and pressure drop characteristics for any number of channels. The results show that the offset fins present the best performances, in term of volume goodness factor, compared to all geometries. Using offset fins offers better heat transfer capability at a lower volume, which is beneficial in the need of reducing charge for refrigeration systems.

Keywords:
Microchannel, CFD, Refrigeration, Heat Transfer, Pressure Drop.

1. Introduction
The need of compactness for heat exchangers design in refrigeration systems led the industry to adopt microchannel heat exchangers. Compared to regular fin and tube heat exchangers commonly used in refrigeration and air-conditioning systems, they typically offer higher of heat transfer per unit of volume. However, their use is challenged by the water condensation retention, which becomes problematic for frosting conditions since the remaining water after a defrosting process will freeze again, therefore downgrading the performances of the heat exchanger.

A recently developed a new microchannel design, called Web-MPE [1], which significantly reduces the condensate retention together with a high compactness. The reduction in water retention can be up to 90\%, compared to traditional microchannels using louvered fins. The new profile includes drain paths in between the microchannels, to evacuate the melted water during defrosting. Indeed, the application of such microchannel heat exchangers is focused on low charge ammonia systems, where the microchannels heat exchangers are used as evaporators. The low temperature of the ammonia will induce frost formation inside the microchannels, affecting its performances. Therefore, there is a need of a good water drainage capability to avoid remaining liquid water inside the evaporator, which will freeze again after a defrosting cycle.

The goal of this study is to compare the performances of different common fin design, in term of heat transfer coefficient and pressure drop, with or without pathways for water drainage, using CFD simulations. Such ammonia evaporators generally employ fins-and-tubes evaporators with large fin pitches, tube pitches and tube diameters, which result in long frosting (and defrosting) time period (up to one day). However there is a need of charge minimization in ammonia refrigeration systems due to safety regulations. For example, nowadays the charge limit for Denmark is 5000kg. An exceed
of this limit will lead to higher cost in installation, maintenance, operation of the system due the increased safety precautions. The use of Computational Fluid Dynamics (CFD) is an interesting tool for investigating novel designs for low-charge ammonia evaporators. It allows to explore non-conventional designs by the use of optimization algorithms, such as the work Yildizeli and Cadirci [2] for microchannel heat sinks. CFD simulations are also a popular method to investigate the influence of different fin geometries on the performances of microchannel evaporators. The most common fin designs studied by the use of CFD simulations are straight fins [3], [4], louvered fins [5]–[8], offset fins [9]–[11], wavy fins [12]–[14], helically wound finned-tube bundles [15], [16], and pin fins [17], [18]. The results of the CFD simulations can be used as a substitute of experimental data used to correlate the thermo-hydraulic performances, generally in term of Colburn j-factor for the heat transfer, and Fanning friction f-factor for the pressure drop.

Bacellar et al. [19] developed j- and f- factors correlations for the air-side of a compact fin and tube heat exchanger. The tubes have a staggered arrangement but no fins are modelled. Chennu and Paturu [20] carried out CFD simulations to establish air-side correlations for offset fins. The correlations, also in term of j- and f- factors are distinctively for laminar and turbulent flow regimes. In a similar way, Ismail and Velraj [13] developed air-side correlations for offset fins and wavy fins. Deng [21] performed CFD simulations with a more advanced turbulence model, Large Eddy Simulations (LES), to improve the accuracy of the air-side correlations for flat tubes and louvered fins. Sadeghianjahromi et al. [22] focused on the louver angle of louvered fins, to establish air-side correlations depending on the angle level. All the above-mentioned studies use the effectiveness-NTU method or LMTD method with mass flow averaged temperature to calculate the Colburn j-factors. The simulations also include the entrance region in the calculation of the local hydraulic and thermal factors. If the entrance region is not of interest, or has a relative small length compared to the full geometry of the system, stream-wise periodic boundary conditions can be used. This method has been first proposed by Patankar et al. [23] to reduce the size of the computational domain and, therefore, reduce its complexity. Martinez-Espinosa et al. [24] used stream-wise periodic condition in heat exchanger with helically segmented finned tube, to establish fully developed air-side correlations. A recent review of the definition of the air-source correlations, but also fin design and definition of fin performances, can be found in the review of Qasem and Zubair [25]. They concluded that offset fins and pin fins offer best performances in term of j-factor, whereas plain-fins (or straight fins) are the most suitable to have a low friction f-factor.

This study focuses on the performances of a microchannel evaporator for four different kind of fin geometries: straight fins, wavy fins, offset fins and triangle fins. The performances of each fin design is defined in term of Colburn j-factor and Fanning friction f-factor. The influence of the number of channels is also investigated, as well as the effect of the entrance region for such compact geometries. The simulations do not take into account frost/defrosting conditions, even so some proposed fin geometries are designed to get a higher water drainage rate by the use of water pathways along the air-stream direction. This work should be considered as “a proof of concept” for improving performances of compact ammonia evaporators by the use of CFD simulations.

2. Microchannel Heat Exchanger using squared microchannels

2.1. Evaporator design

The microchannel heat exchanger consists of 2mm x 2mm squared microchannels, and four different fin geometries. An example of the heat exchanger design using straight fins is show in Figure 1.
Figure 1. 3D model of the microchannel evaporator with straight fins ($X_p = 13$ mm, $F_p = 2.5$ mm)

The fin pitch $F_p$ and the transverse tube pitch $X_p$ were varied in the CFD simulations to analyze the influence of the reduction of the frontal area on the characteristics of the evaporator. However, the geometry of the microchannels was fixed and is shown in the Figure 2. The geometry corresponds to the Web-MPE microchannels [1], where water drainage pathways are added for frosting/defrosting conditions.

![Figure 2](image)

(a) Side view

(b) Top view

Figure 2. Extruded aluminum profile before punching (a) and after punching (b).

The microchannels were extruded and punched aluminium profile. The longitudinal tube pitch $T_p$ was set to 4.5 mm. The profiles were punched to remove the linking material between the channels, thus allowing the melted water to drain during a defrost cycle. The remaining bridging, seen in Figure 2.b, were removed in the numerical model, assuming that their influence on the heat transfer and pressure drop is negligible. The fin thickness was set to 0.1625 mm for all fin designs.

2.2. Fin design

Four different fin design were modelled using CFD, in order to investigate their performances in term of heat transfer rate and pressure drop: Straight fins, wavy fins, offset fins and triangle fins. The straight fins and wavy fins do not offer extra pathways for the water to drain, unlike the other fins.
designs due to their discontinued geometry for the offset fins and inclination for the triangle fins. A portion of the evaporator is shown in the Figure 3, with the four different fin geometries.

Isometric view

Front view

Top view

Figure 3. Partial representation of the 3D models of the straight fins, wavy fins, offset fins and triangle fins ($X_p = 13$ mm, $X_p = 2.5$ mm)

The solder material to attach the fins on the microchannels has a height of 0.2 mm and was considered perfect (no void). The wavy fins have a period $p$ of 9 mm and an amplitude $a$ of 0.9 mm, as seen in Figure 3. The offset fins have a periodicity $o$ of 4.5 mm, meaning that the fins are discontinued every channel row. Finally, the fin angle $\phi$ for the triangle fins is kept below 45° to avoid bending during the welding process.

2.3. Length of the evaporator

For large fin pitches, considered in the investigation to accommodate long defrost period, the entrance region can be significantly large. It corresponds to the zone where the flow is in transition to its fully developed behaviour hydraulically and thermally (in the so-called fully developed region). To reduce the number of CFD simulations, the local heat transfer coefficient and pressure drop (per channel) have been extracted, in order to extend the global heat transfer coefficient and total pressure drop for any number of channels. The methodology is described in [26], for triangle fins.

The varying and fixed parameters used in the CFD model are summarized in the Table 1. In total, 62 CFD simulations were carried out.
Table 1. Heat exchanger characteristics for CFD

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Variation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin design</td>
<td>Straight, wavy, offset and triangle</td>
</tr>
<tr>
<td>Transverse tube pitch, $X_p$ (mm)</td>
<td>9.00, 13.00, 17.00, 21.00</td>
</tr>
<tr>
<td>Fin pitch, $F_p$ (mm)</td>
<td>2.50, 5.00, 7.50, 10.00</td>
</tr>
<tr>
<td>Longitudinal tube pitch, $T_p$ (mm)</td>
<td>4.50</td>
</tr>
<tr>
<td>Channel rows, $N$ (-)</td>
<td>35</td>
</tr>
<tr>
<td>Channel height x width (mm)</td>
<td>2.00 x 2.00</td>
</tr>
<tr>
<td>Fin thickness, $F_t$ (mm)</td>
<td>0.1625</td>
</tr>
<tr>
<td>Frontal air velocity, $U_{in}$ (m/s)</td>
<td>4.40</td>
</tr>
</tbody>
</table>

3. CFD Simulations

CFD is a powerful tool to understand and analyse the flow behaviour inside the microchannel evaporator, considering the non-uniform geometries of the fins. It allows extracting local data, which is difficult and subject to high uncertainty with an experimental setup. Moreover, CFD can drastically reduce the price of prototyping, by preliminary eliminates low performance fin designs, which is the aim of this study. All CFD simulations were carried out using the commercial software ANSYS 19.1 with the CFX solver.

3.1. Modelling

Only a portion of the whole evaporator was modelled in the CFD simulations, to keep a reasonable simulation time. The use of symmetry boundary conditions allowed such simplifications. An example of the CFD model with the boundary conditions is represented in Figure 4.

![3D model of the microchannels with straight fins](image)

The use of symmetries reduces the size of the domain to a quarter of the whole section. The temperature of the external channels’ and fins’ surfaces (in orange) were assumed constant at $T_{wall} = 6°C$, for the use the effectiveness NTU method to calculate the global heat transfer coefficient. Furthermore, wall temperature and inlet temperature were assumed independent of the heat exchanger effectiveness, due to the use of constant properties for the air. Only the air-side was modelled in the CFD simulations. It means that the heat conduction in the channels and the fins was not taken into account, leading to a fin efficiency of 1. Rogie et al. [26] showed that for similar compact geometries, and using aluminium microchannels and fins, the fin efficiency is closed to the unity. Finally, no-slip conditions were assumed for the wall. The air was calculated as an incompressible ideal gas due to low temperature change inside the evaporator ($\Delta T_{max} = 14°C$).
3.2. Meshing

For turbulent flows, the mesh size is dictated by the turbulence model chosen to resolve the flow boundary layer. It is even more sensitive for flow around obstacles where the laminar boundary layer regenerates at the tip of each channel, with a transitional flow in the wake due to vortex generation. For the in-line channel pattern, such as seen in Figure 4, the heat transfer coefficient is maximum at the leading edge of each channel and decreases along the channel in the flow direction. In the wake of the channels, recirculation zones are generated, with vortices with low velocities and low heat transfer rate. Nevertheless, the vortices help the mixing by transporting heat to the neighbourhood regions, resulting in higher heat transfer but also in a higher pressure loss.

Offset fins generate similar effects on the flow, with laminar boundary layers restarting at each fin tip and vortices generated in their wakes. Concerning the wavy fins, the sinusoidal shape lengthens the surface area of the fins, as well as increases the non-uniformity to the flow, having a positive effect on the heat transfer.

Sahiti et al. [18] showed that turbulent flows can appear below the transition Reynolds number for flow around obstacles. The k-ω SST turbulence model was selected, with a range of Reynolds number from 600 to 4000. Moreover, several studies reported that k-ω SST turbulence models perform better than k-ε turbulence models for offset fins [27], and that k-ω SST turbulence models show good agreement when compared with experimental data for flow around tubes [28].

An example of a mesh independence study is shown in Figure 5 for the triangle fin design, where the j- and f- factors are plotted in function of the mesh size. The $y^+$ value is kept under one to correctly resolve the flow at the boundary layer using the k-ω SST turbulence model.

![Figure 5. j- and f- factors as a function of mesh size for triangle fin (X_p = 13 mm, F_p = 5 mm)](image)

The mesh analysis was conducted for all fin designs, until the difference of j- and f- factors values between two consecutive meshes decreases below 1%.

3.3. Temperature profiles

The temperature profiles are shown in Figure 6, for all fin designs. The figure represents a partial view of the temperature field of the air-side of the evaporator at row 17th and 18th (centre of the evaporator), where the flow is fully developed.

For the straight fins and the triangle fins, the temperature is uniform along the flow direction. This means that the heat is carried away from the channels and the fins by natural convection in the transverse and vertical directions. It results in lower heat transfer capability compared to fin geometries where obstacles or swirl generators are implemented, such as the wavy fins and the offset fins. The presence of obstacles or corrugations, which locally increases velocity and thermal gradients, causes change of direction of the flow, as it can be seen in Figure 6 and Figure 7 and
therefore causes a change in the heat transfer coefficient. It is the reason why the average temperatures of the contour plots in Figure 6 are lower for wavy fins and offset fins, meaning that for the same conditions these fin designs offer a higher heat transfer rate. However, the enhancement of local heat transfer coefficients results in higher friction factor and pressure drop, which counterbalances the benefits of these fin designs. Large recirculation zones can be observed in the Figure 7 for the wavy fins, due to the corrugations. Recirculation can also be observed in the wake of the channels, due to the sudden change in cross section area, for all fin designs.

Figure 6. Temperature contours for all fin design at row 17th and 18th ($X_p = 13$ mm, $F_p = 5$ mm)

Figure 7. Velocity streamlines for wavy and offset fins at row 17th and 18th ($X_p = 13$ mm, $F_p = 5$ mm)
3. Fin performances

3.1. Heat transfer and pressure drop

The fin performances were evaluated by calculating the j- and f- factors for each fin design. The j-factor is given by Eq. (1), where the heat transfer coefficient is found using the effectiveness-NTU method.

\[
j = \frac{h \cdot Pr^{2/3}}{\rho \cdot c_p \cdot U_{\text{max}}} \quad (1)
\]

\[
h = NTU \cdot \frac{C_{\text{min}}}{A_{\text{tot}}} \quad (2)
\]

\[
NTU = -\ln \left(1 - \frac{T_o - T_i}{T_{\text{wall}} - T_i}\right) \quad (3)
\]

where \( h \) is the heat transfer coefficient, \( Pr \) the Prandlt number, \( \rho \) the density, \( c_p \) the specific heat capacity at constant pressure, \( U_{\text{max}} \) the maximum flow speed, \( C_{\text{min}} \) the minimum heat capacitance rate, \( A_{\text{tot}} \) the total fin & tube surface area and \( T_o, T_i, \) and \( T_{\text{wall}} \) are the outlet, inlet, and wall temperature, respectively.

Assuming that the density change between the inlet and the outlet is negligible for the air-side, the f-factor is defined by Eq. (4).

\[
f = \frac{A_c}{A_{\text{tot}}} \cdot \frac{2 \cdot \Delta P}{\rho \cdot U_{\text{max}}^2} \quad (4)
\]

where \( A_c \) is the minimum free flow area and \( \Delta P \) the pressure drop between the inlet and the outlet of the air-side.

The j- and f- factors were calculated for the global evaporator size with the 35 channels, but also locally along the longitudinal flow direction by using local values of \( h, \Delta P, U_{\text{max}} \) and accumulated value of the total area \( A_{\text{tot}} \). An example of the evolution of the j- and f- factors in function of the number of channels, for all fin designs, is found in Figure 8.

Figure 8. j- (left) and f- (right) factors in function of the number of rows, for all fin designs (\( X_p = 13 \) mm, \( F_p = 5 \) mm)

The offset fins offer the highest heat transfer capacity compared to all other fin designs. In term of friction factor, the wavy fins and the offset fins have the lowest performances, due to the presence of
secondary flow patterns. Straight fins and triangular fins have similar performances, for heat transfer rate and pressure drop, due to a similar wetted area and no features to improve the turbulence intensity of the flow.

For a low number of channels (N < 5), the f-factor is similar for all fin designs. For higher number of channels, the f-factor diverges into two sub-groups: continuous fins (straight and triangle) and discontinued and corrugated fins (wavy and offset).

3.2. Performance evaluation criteria for fin design

It exists several criteria to evaluate the performances of a fin design. One of the most used criteria is surface area goodness factor $j/f$, defined by London in 1964. Another criteria is the volume goodness factor $j/f^{2/3}$ which eliminates the Reynolds number. However, these criteria do not consider important parameters such as the compactness of heat exchangers, heat transfer area and fin efficiency, and involve having the same hydraulic diameter for all fin designs to have a fair comparison. To solve this issue, a more reliable core volume goodness factor was introduced by Shah and Sekulić [29]:

$$\eta_0 h_{std} \beta = \frac{c_p \cdot \mu}{Pr^{2/3}} \cdot \eta_0 \cdot \frac{4 \cdot \sigma}{D_h^2} \cdot j \cdot Re$$

$$E_{std} \beta = \frac{\mu^3}{2 \cdot g_c \cdot \rho^2} \cdot \frac{4 \cdot \sigma}{D_h^4} \cdot f \cdot Re^3$$

where $\eta_0$ is the overall heat transfer efficiency ($\eta_0 = 1 - A_f/A_{tot} \cdot (1 - \eta_f)$), $D_h$ the hydraulic diameter ($D_h = (4 \cdot A_c \cdot L)/A_{tot}$), $Re$ the Reynolds number and $g_c$ the proportionality constant in Newton’s second law of motion ($g_c = 1$). The term $\eta_0 h_{std} \beta$ represents the heat transfer power per unit temperature difference and unit core volume, whereas $E_{std} \beta$ represents the friction power expenditure per unit core volume. The volume goodness factors for the four fin designs and all geometries are shown in Figure 9.

![Figure 9. Volume goodness factor for the four different fin designs and for all geometries](image)

A higher value of $\eta_0 h_{std} \beta$ for constant $E_{std} \beta$ means that the fin design is capable to transfer more heat per unit volume at the same fluid flow power, and vice versa. Figure 9 shows that, overall, offset fins offer the best performances compared to all different fin designs. It should be pointed out that straight fins also offer good performances, with similar or even slightly better results compared to the offset fins, for some range of pressure drop. However, the offset fins provide higher heat transfer coefficients, which makes its adoption more relevant for low charge ammonia evaporators.
3.3. Entrance region length

The length of the entrance region varies in function of the fin design and the evaporator compactness. Figure 10 shows the channel’s row number where the flow becomes fully developed. The fluid is considered thermally fully developed when the change of local heat transfer coefficient between two consecutive rows becomes lower than 1%. The constant local heat transfer coefficient can be used to extrapolate the global heat transfer coefficient of the evaporator, for any number of rows. Concerning the local pressure drop, it should be noted that it reaches its asymptotic value sooner than for the local heat transfer coefficient.

![Figure 10. Row's number at the start of the fully developed region in function of Hydraulic diameter, for all fin designs](image)

The entrance region length increases for higher hydraulic diameters. Furthermore, the entrance region length decreases for fins with extended surfaces, due to a better mixing indicated by high heat transfer coefficients.

4. Discussion

The ability of a fin design to drain water was not taken into account to define their performances in this study. The performances of the different evaporator geometries are only for dry conditions, and should be used in designing prototypes in future experiments for frosting/defrosting conditions. However, the offset fins which present the best performances compared to all different fin designs, also offer drain pathways for the water due to their discontinues geometries. Overall, offset fins appear to be the perfect candidate to assure good performances in term of heat transfer rate and pressure, as well as for water drainage.

The characteristics of the wavy fins (periodicity $p$ and amplitude $a$) and the offset fins (periodicity $o$) have an influence on the heat transfer rate and pressure drop, as seen in Figure 6 and Figure 7. Additional investigation on these values is needed to optimise the evaporator performances. The use of optimisation algorithms, as well as CFD simulations, would be an interesting procedure to find the optimum fin characteristics.

The CFD simulations do not take into account any irregularities in the flow or the boundary conditions, which limit the results of this study where ideal conditions are met. Many singularities can be found in real evaporators, such as temperature non-uniformity, fluid flow inlet maldistribution, conduction in the fins, imperfect contact at the solder between channels and fins, manufacture uncertainties, and so on. An uncertainty of the results extracted by the CFD models should be anticipated, and further compared with experimental tests. A prototype of the evaporator using triangle fins is currently under investigation by the Danish Technological Institute, where experimental test will be carried out.
5. Conclusion

This study presented the performances of four different fin designs, straight fins, wavy fins, offset fins and triangle fins, for compact ammonia evaporators using microchannels. CFD simulations were carried out to extract the heat transfer coefficient and the pressure drop, for each fin design, to calculate the Colburn j-factor and the Fanning friction f-factor. The fin pitch, as well as the transverse row pitch were varied to analyse the influence of the compactness on the evaporator performances. The results show that the offset fins offer the best performances, by using the volume goodness factor criteria, compared to all other fin designs. The restart of the flow boundary layer at each fin tip increases the localised heat transfer coefficient without much penalty on the pressure drop, compared to wavy fins for example. Furthermore, the discontinuity of the offset fins allows a good drainage capability of such compact ammonia evaporator, for defrosting cycles.

References


