Simulation of an aluminum foam heat exchanger using the volume averaging technique

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Abstract

Open cell metal foams show a high potential for heat transfer applications. In this paper the thermal and hydraulic performance of a metal foam heat exchanger with two tube rows in a staggered layout is studied. Two-dimensional simulations are performed using a bulk model based on the volume averaging technique. A louvered fin heat exchanger with the same tube layout and overall volume is considered as reference case. Velocities range from 1.2 m/s to 3.2 m/s. An experimental validation illustrates the quality of the simulation results. For the same fan power, the 10 ppi and 20 ppi foams show a smaller heat transfer than the louvered fin heat exchanger. This is mainly due to the larger airside heat transfer surface area of the louvered fin heat exchanger (because of the small fin pitch) compared to the foams. 30 and 35 ppi foams, having a specific surface area above 1000 m²/m³, show a better thermal hydraulic performance than the louvered fin design.

1. Introduction

Heat exchanger manufacturers are continuously searching for new and better designs. A promising idea is the use of open cell aluminum foam as alternative for the conventional fins. Due to its high porosity (and thus low weight), high surface-to-volume ratio, excellent fluid mixing and high thermal conductivity, open cell aluminum foams show potential for heat transfer applications.

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The volume averaging technique (VAT) is frequently used for flow modeling in porous media as shown by Whitaker et al. (1999). As metal foam is considered as a porous medium, VAT based modeling shows potential for metal foam heat exchangers. The details of the original structure are replaced by their averaged counterparts while the local flow and temperature distributions can still be resolved. The governing VAT transport equations were previously developed from the Navier-Stokes and energy equations by Whitaker et al. (1999) and Travkin et al. (2001). These equations for the phase averaged variables can be solved much faster than the traditional transport equations for local variables, which require for instance direct numerical simulations (DNS). Hence, the VAT approach is extremely suitable for the simulation and optimization of full heat exchangers.

The major challenge in the VAT approach is the definition of the closure problem. Because the details of momentum and energy transfer between the fluid flow and solid structure are lost during the averaging, closure relations are required. They can be obtained from experiments or CFD simulations of a representative heat exchanger volume with a very fine mesh. In literature correlations can be found for the permeability, inertial loss factor and interstitial heat transfer coefficient. Also expressions which relate the microscopic geometrical parameters (e.g. cell and strut diameter) to the macroscopic geometrical parameters (e.g. porosity and specific surface area) are available. A detailed review on existing fluid and thermal transport models for open cell metal foams can be found in Mahjoob et al. (2008) and Zhao et al. (2012).

The closure relations are only valid for the specific geometry and flow conditions under consideration. De Jaeger et al. (2013) developed a thermal hydraulic VAT model for the bulk flow through open cell metal foams relevant for heat transfer applications (i.e. porosities > 88%). In this paper the VAT model is experimentally validated. Next, a commercially available louvered fin heat exchanger for HVAC applications is compared to a metal foam heat exchanger having the same tube layout and overall volume. Different foam types are analyzed.

2. Experimental

2.1. Computational domain

The geometry of the simulated heat exchanger is listed in Table I. This geometry is the same as the heat exchanger tested in a wind tunnel by De Schampheleire et al. (2013). The heat transfer and pressure drop measurements obtained during the experiments are used to validate the VAT results. The heat exchanger consists of two tube rows placed in a staggered tube layout. Each tube row counts 10 tubes. The metal foam block has a height equal to 256 mm, a width of 426 mm and a flow depth of 24 mm. In the experiments a 10 ppi aluminum foam (AL1050) was used (ppi = pores per linear inch), i.e. MF10.462 of Table II. The other aluminum foams which are simulated are listed in Table II. The data of Table II were obtained from μCT scan data for the 10 and 20 ppi foams (De Jaeger et al. (2011)) and manufacturer’s data for the 30 and 35 ppi foam. Each foam is labeled as MFxx.yyy with xx the ppi value and yyy the specific surface area.
Table 1. Geometrical details

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat exchanger height</td>
<td>H</td>
<td>mm</td>
<td>256</td>
</tr>
<tr>
<td>Heat exchanger width</td>
<td>W</td>
<td>mm</td>
<td>426</td>
</tr>
<tr>
<td>Height exchanger depth</td>
<td>F</td>
<td>mm</td>
<td>24</td>
</tr>
<tr>
<td>Tube outer diameter</td>
<td>D_o</td>
<td>mm</td>
<td>7.2</td>
</tr>
<tr>
<td>Tube wall thickness</td>
<td>t_w</td>
<td>mm</td>
<td>0.27</td>
</tr>
<tr>
<td>Transversal tube pitch</td>
<td>P_t</td>
<td>mm</td>
<td>21</td>
</tr>
<tr>
<td>Longitudinal tube pitch</td>
<td>P_l</td>
<td>mm</td>
<td>12</td>
</tr>
</tbody>
</table>

Table 2. Parameters of simulated metal foams

<table>
<thead>
<tr>
<th>ID</th>
<th>ppi</th>
<th>d_cell (mm)</th>
<th>A_o (mm²)</th>
<th>φ</th>
<th>α_o (m²/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>MF10.380</td>
<td>10</td>
<td>4.90</td>
<td>0.0615</td>
<td>0.951</td>
<td>380</td>
</tr>
<tr>
<td>MF10.462</td>
<td>10</td>
<td>4.81</td>
<td>0.0998</td>
<td>0.932</td>
<td>462</td>
</tr>
<tr>
<td>MF20.580</td>
<td>20</td>
<td>2.92</td>
<td>0.0126</td>
<td>0.967</td>
<td>580</td>
</tr>
<tr>
<td>MF20.720</td>
<td>20</td>
<td>3.17</td>
<td>0.0377</td>
<td>0.937</td>
<td>720</td>
</tr>
<tr>
<td>MF20.860</td>
<td>20</td>
<td>2.88</td>
<td>0.0463</td>
<td>0.913</td>
<td>860</td>
</tr>
<tr>
<td>MF30.1016</td>
<td>30</td>
<td>1.90</td>
<td>0.0099</td>
<td>0.949</td>
<td>1016</td>
</tr>
<tr>
<td>MF35.1175</td>
<td>35</td>
<td>1.65</td>
<td>0.0075</td>
<td>0.949</td>
<td>1175</td>
</tr>
</tbody>
</table>

The computational domain used in the VAT simulations is shown in Figure 1. Two-dimensional simulations are performed. Four zones were defined: the pre-extended region, the post-extended region, the porous zone and the tube walls. The first three are considered fluid, while the latter is a solid. The tube walls were also meshed to take the heat conduction into account. The quality of the mesh was carefully assessed during the meshing. All zones were meshed with quadrilateral elements. In the porous zone the mesh gets finer towards the tube surface: at the tube surface the cell size was 0.25 mm, while at the inlet and outlet of the porous zone the cell size was 0.5 mm. The grid independency was verified by reducing the mesh size.

2.2. Computational method

All The VAT model developed by De Jaeger et al. (2013) was used. This is a thermal non-equilibrium model which means that two energy equations are considered (one for the air and one for the aluminum). This assumption is made due to the large difference in thermal conductivity between air and metal (here aluminum). At the inlet a uniform velocity in the x-direction and a constant air inlet temperature were imposed. The velocity as well as the temperature were determined from the measurements. At the outlet the static pressure was set to 0 Pa. The walls of the flow channel are considered adiabatic (i.e. well insulated during the experiment). The meshed tube walls have a thickness of 0.27 mm. A convective heat transfer coefficient was applied on the inner wall of the tubes. This convective heat transfer coefficient was determined via the Gnielinski correlation during the experiments. No slip boundary conditions were applied on the tube outer walls. The contact resistance between the tubes and foam is modeled as a temperature jump, as explained in De Jaeger et al. (2012). The double precision segregated solver was used. The coupled algorithm was applied for the pressure-velocity coupling. The discretization of the convective terms in the governing equations is done via a second order upwind scheme, while a second order central differencing scheme is applied for the diffusive terms. The gradients are evaluated via the least squares cell based method. The pressure gradient in the momentum equations is treated via a second order discretization scheme. Convergence criteria were set to $10^{-8}$ for continuity and velocity components and $10^{-12}$ for 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 air specific heat was calculated via a polynomial fit, the effective air viscosity and thermal conductivity were calculated according to eqs. (1) and (2), respectively ($\phi$ = porosity).
The molecular viscosity \( \mu_f \) was determined via the Sutherland approximation (De Jaeger et al. (1962)). The molecular thermal conductivity \( k_f \) was determined based on the kinetic theory (De Jaeger et al. (1962)). The density and specific heat of the AL1050 aluminum foam materials were considered constant (\( \rho = 2719 \text{ kg/m}^3 \) and \( c_p = 871 \text{ J/kgK} \)), while the effective thermal conductivity was calculated according to eq. (3). The factor 0.35 is used to take the tortuosity effect into account (De Jaeger et al. (2013)). The copper tubes have constant material properties: \( \rho = 8978 \text{ kg/m}^3, \lambda = 387.6 \text{ W/mK} \) and \( c_p = 381 \text{ J/kgK} \).

\[
\begin{align*}
\mu_{fe} &= \mu_f / \phi \\
k_{fe} &= \phi k_f \\
k_{se} &= 0.35 (1 - \phi) k_s
\end{align*}
\]

The flow is assumed to be laminar and steady, which is reasonable for the considered Reynolds numbers (Re = 530-1400, based on the tube diameter and the inlet velocity). For each of the simulations the resulting heat balance closes within 0.2% and there were no noticeable differences in the mass balance.

2.3. Model validation

A detailed description of the wind tunnel experiment on the metal foam heat exchanger which is simulated here, is provided in De Schampheleire et al. (2013). 10 ppi aluminum foam is used with a porosity of 0.932 (see MF10.462 in Table II). The tubes were connected to the foam via a pressure fitting.
The heat exchanger was experimentally characterized for six different air mass flow rates, resulting in inlet velocities ranging from 1.2 m/s to 3.2 m/s. Comparison of the measurements with the VAT predictions allows evaluating the accurateness of the VAT model. Figure 2 plots the heat transfer and pressure drop results. The uncertainty bars on the experimental data are also indicated. The maximum deviation in heat transfer is 5%, while the maximum deviation in pressure drop is 10%. The larger deviation for the pressure drop results is probably due to the entrance and exit losses which were neglected here (as the VAT model is for a bulk flow). However, overall there is a good match between experiments and simulations, which validates the VAT model.

3. Results and discussion

The pressure drops for the 10 ppi and 20 ppi foams are plotted as function of the mass flow rate in Figure 3. As a reference case a louvered fin heat exchanger (fin pitch of 1.4 mm) with the same tube layout and overall volume is added. This heat exchanger was also tested by De Schampheleire et al. (2013) in a wind tunnel. The louvered fin data are thus experimental data. The corresponding uncertainty bars are indicated.

With exception of MF20.580.b, it can be concluded that the pressure drop will decrease with an increase in averaged cell diameter. MF20.580.b is an exception: even though the cell diameter is small ($d_{cell} = 2.92$ mm), the pressure drop is also small compared to the other metal foams. Thus might be due to the small strut cross sectional area $A_o$ which results in less flow blockage. The louvered fin heat exchanger, which has a specific fin surface area (thus without the tubes) of 1427 m²/m³, shows a similar pressure drop as a metal foam heat exchanger with a specific foam surface area of a bit more than 720 m²/m³.

Figure 4 shows the heat transfer results for the 10 and 20 ppi foams. The heat transfer rate is calculated from the simulation results by multiplying the 2D heat transfer rate with the heat exchanger width under the assumption of a constant freestream temperature in the tubes. The heat transfer rate increases with increasing ppi value and specific surface area. The louvered fin heat transfer rates (with corresponding experimental uncertainties) are also plotted. It is assumed that the effect of the temperature gradient, which existed inside the tubes of the louvered fin heat exchanger during the experiments, is negligible. The louvered fin heat exchanger outperforms the foamed heat exchangers. The available heat transfer surface area is also much larger (1427 m²/m³ for the louvered fin heat exchanger vs. 860 m²/m³ or less for the foamed heat exchangers). The tested louvered fin heat exchanger has a fin pitch equal to 1.4 mm, which is much smaller than the cell diameters of the 10 and 20 ppi metal foams (see Table II). This explains the large specific surface area of louvered fin designs. The metal foam should thus have a larger specific surface area than listed in Table II to obtain a similar thermal hydraulic performance as state-of-the-art compact fin designs. An increased specific surface area, usually results in smaller cells and thus an increase in the number of pores per linear inch (the so called ppi value of the open cell foam).

In Figure 5 the heat transfer rate is plotted as function of the fan power. The fan power is defined as the product of the pressure drop over the heat exchanger and the volumetric flow rate at the heat exchanger inlet. Besides the louvered fin heat exchanger and the five 10-20 ppi metal foam heat exchangers, two other metal foam heat exchangers with larger ppi values are added. Their foam parameters are also listed in Table II. Notice that their cell diameters are closer to the 1.4 mm fin pitch of the louvered fin heat exchanger. The 35 ppi foam has a strut diameter of 0.10 mm, which is a practical lower limit (thinner struts are too fragile). For the same fan power, the 10 ppi and 20 ppi foamed heat exchangers have a lower heat transfer than the louvered fin heat exchanger. The 30 and 35 ppi heat exchangers cause a larger pressure drop than the 10 and 20 ppi foams, but due to their large specific surface area the heat transfer rate is also larger. For small inlet velocities, both heat exchangers show a similar thermal performance as the louvered fin heat exchanger at the same fan power. At higher velocities, the 30 and 35 ppi heat exchangers clearly show a higher heat transfer at the same fan power. At a fan power of 33W, for instance, the 30 ppi heat exchanger has a 6.4% higher heat transfer and the 35 ppi heat exchanger has an 11.7% higher heat transfer than the louvered fin heat exchanger. This illustrates that in the considered Reynolds range the use of open cell metal foam can result in a heat exchanger with better thermal hydraulic performance than a state-of-the-art finned heat exchanger.
4. Conclusion

In this paper a VAT model for bulk flow through metal foams is used to simulate a heat exchanger with a staggered tube layout in a velocity range from 1.2 m/s to 3.2 m/s. This is a typical range for HVAC applications. The simulation results were experimentally validated indicating the good quality of the simulations. Different foam types were analyzed. A louvered fin heat exchanger with specific surface area of 1427 m²/m³ is used as reference. For the same fan power, the studied 10 and 20 ppi foams have a smaller heat transfer than the louvered fin heat exchanger. This is mainly due to the much smaller specific surface area. The 30 and 35 ppi foam, with a specific surface area above 1000 m²/m³, show a better thermal hydraulic performance than the louvered fin heat exchanger.

References