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Experimental Assessment of Metal Foam and Louvered Fins as Air-side Heat Transfer Enhancement Media for Miniaturized Condensers

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ABSTRACT

An experimental investigation was conducted to evaluate and compare the thermal-hydraulic performances of cross-flow microchannel condensers using louvered fins and metal foams as extended surfaces. Three copper foam surfaces with pore densities of 10 and 20 PPI and porosities of 89.3% and 94.7% were compared with three aluminum louvered fins with lengths of 27 and 32 mm (in the flow direction) and heights of 5 and 7.5 mm. The experiments were carried out in a closed loop wind-tunnel calorimeter equipped with a R-600a refrigeration loop. A condensing temperature of 45°C was used in all tests, with face velocities ranging from 2.1 to 7.7 m/s. A comparison based on the air-side thermal conductance and pumping power per unit volume showed that the louvered fin surfaces performed better than the metal foams for all conditions investigated.

1. INTRODUCTION

The development of small-scale refrigeration systems is restricted by the size of the compressor as well as by the compactness of the heat exchangers. Thus, the improvement of compact condensers is essential for overcoming the size, weight and pumping power limitations currently associated with small-scale cooling technologies.

Metal foams have a large surface area density, a high effective thermal conductivity and a large interstitial heat transfer coefficient (Boomsma et al., 2003). For similar reasons, louvered fins are widely employed in the refrigeration industry; their geometry intensifies the energy transfer between the fluid and the solid wall through the formation and destruction of boundary layers (Cowell et al., 1995; Webb and Kim, 2005).

Several authors have addressed the use of metal foams in compact heat exchangers and their thermal-hydraulic performance in comparison with more conventional geometries. Kim et al. (2000) concluded that aluminum metal foams (with pore densities of 10 and 20 PPI and porosity of 92%) were less competitive than louver fins because of their poorer relationship between thermal conductance and pumping power per unit volume. Mancin et al. (2011) compared the thermal-hydraulic performance of several aluminum metal foams (5-40 PPI, ε = 90-96%, β = 341-1721 m²/m³) with that of a finned heat sink (β = 147 m²/m³) using a modified area goodness factor. Metal foams showed a significant improvement in terms of mass and volume reduction. De Schampheleire (2011) compared two “identical” finned-tube heat exchangers; one with louvered fins and the other with a 10-PPI metal foam on the air side. Heat transfer and pressure drop measurements were performed for face velocities between 1 and 3 m/s. A theoretical analysis based on the behavior of an area goodness factor as a function of the air velocity indicated a better performance of the louvered fin
heat exchanger. Dai et al. (2012) carried out a numerical comparison of flat-tube serpentine louver fin and flat-tube metal foam heat exchangers. For identical air pumping power and heat transfer rate conditions, the metal foam heat exchanger was found to be much smaller in size and lighter in weight. Muley et al. (2012) compared the thermal-hydraulic performance of metal foam heat exchangers with that of a wavy plate-fin (baseline) heat exchanger. While metal foam heat exchangers presented values of thermal effectiveness of the order of 10% higher than those of the baseline heat exchanger, the air-side pressure drop of the metal foam samples reached values up to sixteen times higher than those of the wavy plate-fin device. Ribeiro et al. (2012) compared the performance of three microchannel condensers with metal foams of the air side (pore densities of 10 and 20 PPI and porosities of 89.3 and 94.7%) with that of a similar condenser with 0.5-mm thick plain copper fins on the air side. For a fixed pumping power, the overall thermal conductances of the metal foam condensers were lower than that of the plain fin condenser.

This paper extends the analysis of Ribeiro et al. (2012) by comparing the thermal-hydraulic performance of the copper foam condensers with that of louvered fin condensers with similar overall dimensions. Again, the envisaged application is one of miniature-scale refrigeration using R-600a as a refrigerant. A comparison of the thermal-hydraulic performance of the two enhanced surfaces was carried out based on their air-side thermal conductance and pumping power per unit volume. For the conditions investigated, the louvered fins performed significantly better than the metal foams.

2. MICROCHANNEL CONDENSERS

2.1 Metal Foam Condensers
The cross-flow metal foam condenser geometry is illustrated in Fig. 1. All condensers were manufactured from copper, with the basic characteristics described in Table 1. The metal foams were brazed at the top and bottom of a 3-mm thick copper plate with seventeen 1.5-mm ID circular parallel microchannels. The length (in the direction of the air flow), height and width of the metal foam samples brazed on the plate were 35, 7 and 48 mm, respectively.

![Figure 1: Illustration of a metal foam condenser.](image)

<table>
<thead>
<tr>
<th>Condenser</th>
<th>Number of pores per inch [PPI]</th>
<th>Porosity [%]</th>
</tr>
</thead>
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<tr>
<td>MF1</td>
<td>20</td>
<td>89.3</td>
</tr>
<tr>
<td>MF2</td>
<td>10</td>
<td>89.3</td>
</tr>
<tr>
<td>MF3</td>
<td>10</td>
<td>94.7</td>
</tr>
</tbody>
</table>

2.2 Louvered Fin Condensers
The cross-flow louvered fin condenser geometry is shown in Fig. 2. All condensers were manufactured from aluminum, with the external dimensions of the fin arrangements described in Table 2. The fin thickness was 0.1 mm, the louver pitch was 1 mm and the fin pitch was 1.28 mm. The lengths of the inlet and redirection louvers were 2 and 1 mm, respectively. The louver angle was unknown. The fins were brazed at the top and bottom of two 2-mm thick aluminum plates with five 0.8 × 0.95 mm rectangular microchannels each. The length of each plate (in the air flow direction) was 15 mm, and the distance between them was 3 mm. All condensers had a width of 50 mm.

3. EXPERIMENTAL FACILITY

3.1 Air Loop
The closed-loop wind tunnel calorimeter shown in Fig. 3 has been described in detail in a previous work (Ribeiro et al., 2012), so only its main features will be presented here. The internal duct cross-section area was 300 × 300 mm.
The air flow was supplied by two DC blowers, and a set of four calibrated nozzles and two differential pressure transducers were used to measure the air flow rate. While one transducer measured the pressure difference between the nozzle inlet and the atmospheric pressure, the other measured the pressure drop across the nozzle. A relative humidity transducer and two type-T thermocouples placed upstream of the nozzles were used to estimate the air physical properties upstream of the flow measurement station.

An identical two-transducer arrangement was used to measure the air-side condenser pressure drop. Five type-T thermocouples were installed upstream and downstream of the condenser and three thermocouples were placed outside of the air loop for room temperature control. A set of electric heaters and a cooling coil connected to a thermostatic bath were used for controlling the air inlet temperature in the condenser. The use of flow straightening devices and an idle fan ensured the uniformity of the air temperature and velocity upstream of the air temperature measurements.

The measurement uncertainties (95% confidence interval) for temperature and humidity were 0.2°C and 0.02%, respectively. The differential pressure transducers used for measuring the condenser pressure drop and the nozzle pressure drop had uncertainties of 0.38 Pa and 0.45 Pa, according to their manufacturer. The remaining differential pressure transducers had an uncertainty of 2.49 Pa. Error propagation of the air flow rate yielded a maximum combined uncertainty of 0.3% for this quantity.

![Diagram](image)

**Figure 2:** Illustration of a louvered fin condenser.

### Table 2: Louvered fin characteristics.

<table>
<thead>
<tr>
<th>Condenser</th>
<th>Height [mm]</th>
<th>Length [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>LF1</td>
<td>5</td>
<td>32</td>
</tr>
<tr>
<td>LF2</td>
<td>7.5</td>
<td>32</td>
</tr>
<tr>
<td>LF3</td>
<td>5</td>
<td>27</td>
</tr>
</tbody>
</table>

3.2 Refrigerant Loop

As shown in Fig. 4, a variable-speed compressor was used to pump iso-butane (R-600a) through the loop. A hand-operated needle valve was used as an expansion device. The mass flow rate was measured with a Coriolis-type mass flow meter placed between the accumulator and the needle valve. Resistance temperature detectors (RTD’s) were placed upstream of the compressor and the mass flow meter. An absolute pressure transducer was placed at the condenser inlet. Two immersion-type thermocouples were installed upstream and downstream of the condenser. The compressor inlet temperature, the mass flow meter inlet temperature and the condenser inlet temperature were controlled by electric heaters positioned around the connection tubing via PID microcontrollers. The condenser inlet pressure was controlled by an electric heater wrapped around the accumulator. In all tests, the vapor was superheated at the inlet of the mass flow meter and of the compressor. The uncertainties of the absolute pressure and temperature measurements were 1450 Pa and 0.2°C, respectively. The mass flow meter presented a maximum error of 0.2% (full-scale), according to its manufacturer.

3.3 Experimental Conditions

To evaluate the thermal performance of miniaturized condensers, a condensing temperature of 45°C was used. A superheating degree of 2°C was employed at the condenser inlet during all tests. The air inlet temperature was 25°C and the refrigerant mass flow rate was 1.1×10⁻⁴ kg/s. The air flow rate was set at three different values: 1.4×10⁻³ m³/s, 2.3×10⁻³ m³/s and 3.3×10⁻³ m³/s, which corresponded to face velocities ranging from 2.1 to 7.7 m/s. On the whole, three different conditions were evaluated for each metal foam condenser. Each test was performed twice, thus resulting in six tests for each heat exchanger.
4. DATA REDUCTION

The frictional component of the total pressure drop was calculated subtracting the contraction and expansion losses upstream and downstream the test section and the inertial components from the experimental pressure drop. Thus (Shah and Sekulic, 2003),

\[
\Delta P_f = \Delta P_{me} - \frac{1}{2}(1 - \sigma^2 + K_e)\frac{G^2}{\rho_{in} - G^2_{air}} \left( \frac{1}{\rho_{out}} - \frac{1}{\rho_{in}} \right) + \frac{1}{2}(1 - \sigma^2 - K_e)\frac{G^2_{air}}{\rho_{out}}
\]

where,

\[
G_{air} = \frac{\rho_{in}V_{air}}{A_{min}}
\]

The pressure drop coefficients associated with the inlet contraction, \(K_c\), and outlet expansion, \(K_e\), were assumed equal to 0.5 and 1.0 because the wind tunnel cross-section area was much larger than the condenser face area (\(A_{in} >> A_{face}\)). \(\sigma\) is
the ratio of the minimum air flow passage and the wind tunnel cross-section area. For the metal foam condensers, \( \sigma \) was calculated as follows,

\[
\sigma = \frac{\varepsilon A_{\text{face}}}{A_{\text{min}}}
\]

where,

\[
A_{\text{min}} = A_{\text{face}} \varepsilon
\]

where \( \varepsilon \) is the metal foam porosity. The air-side pumping power is given by,

\[
\dot{W} = \Delta P_{\text{me}} \dot{V}_{\text{air}}
\]

The air-side heat transfer rate is given by,

\[
\dot{Q}_{\text{air}} = \rho_{\text{in}} \dot{V}_{\text{air}} c_{\text{p,air}} (T_{\text{out,air}} - T_{\text{in,air}})
\]

The overall thermal conductance \( UA \) can be obtained from the following relationship involving the log-mean temperature difference,

\[
UA = \frac{\dot{Q}_{\text{air}}}{(T_{\text{out,air}} - T_{\text{in,air}})} \ln \left( \frac{T_{\text{cond}} - T_{\text{in,air}}}{T_{\text{cond}} - T_{\text{out,air}}} \right)
\]

The air-side thermal conductance is given by (neglecting the wall heat transfer resistance),

\[
\frac{1}{\eta \bar{h}_{\text{air}} A_{\text{sur}}} = \frac{1}{UA} - \frac{1}{\bar{h}_{\text{ch}} N_{\text{ch}} A_{\text{ch}}}
\]

where \( N_{\text{ch}} \) is the number of microchannels and \( A_{\text{ch}} \) is the heat transfer area of each channel (perimeter times the channel length). The average heat transfer coefficient on the refrigerant side has been calculated using the correlation of Koyama et al. (2003) for condensation in microchannels. A detailed justification for the choice of this correlation for the present system has been presented by Ribeiro et al. (2012).

For the metal foam condensers, the air-side heat transfer coefficient was calculated using the interstitial surface area density defined as,

\[
A_{\text{sur}} = V \beta
\]

where \( \beta \) is the surface area density calculated using the Dukhan and Patel (2008) correlation. The total heat transfer surface area of the louvered fin condensers was calculated using the relationship given by Shah and Sekulic (2003) for corrugated fins.

The modified Colburn \( j \)-factor, which incorporates the overall surface efficiency proposed by Kim et al. (2000), is defined as,
The Reynolds number was defined for each surface using the conventional approaches adopted in the literature for each type of enhanced surface (Kim et al., 2000; Webb and Kim, 2005). Thus, for metal foam condensers, the height of the porous media was considered as the characteristic length of the Reynolds number, whereas for louvered fin condensers the hydraulic diameter $D_h$ was applied. Therefore,

$$\text{Re}_H = \frac{G_{\text{air}}H}{\bar{\mu}_{\text{air}}} \quad (12)$$

$$\text{Re}_D = \frac{G_{\text{air}}D_h}{\bar{\mu}_{\text{air}}} \quad (13)$$

where $\bar{\mu}_{\text{air}}$ is average air viscosity. By the same token, the friction factors $f_H$ and $f_D$ were calculated for the metal foam and louvered fin condensers as follows,

$$f_H = \frac{H}{L} \left(2\bar{\rho}_{\text{air}}\Delta P_f \right) \frac{2}{G_{\text{air}}^2} \quad (14)$$

$$f_D = \frac{D_h}{L} \left(2\bar{\rho}_{\text{air}}\Delta P_f \right) \frac{2}{G_{\text{air}}^2} \quad (15)$$

where $\bar{\rho}_{\text{air}}$ is average air density and $L$ is the condenser length. Considering all measurements as random and uncorrelated, an uncertainty propagation analysis with a confidence level of 95% resulted in measurement uncertainties for the friction factor, air-side thermal conductance and $j$-factor of $\pm 0.51\%$, $\pm 6\%$ and $\pm 12\%$, respectively.

5. RESULTS

The behavior of the experimental friction factor for metal foam condensers, calculated according to Eq. (14), is presented in Figure 5. The data trend is consistent with those of previous works (Kim et al., 2000; Dukhan and Patel, 2008) in the sense that $f_H$ increases with decreasing porosity and is higher for the lower pore density. This is due to the presumably smaller surface area presented by the samples with pore density of 10 PPI. The effect of pore density appears to be more pronounced than that of porosity (Ribeiro et al., 2012).

Results for $f_D$ are shown in Fig. 6 for the louvered fin condensers. The friction factor decreases with increasing Reynolds number, and the condensers with the highest fin height present lower friction factors. At a fixed volume flow rate, the largest face area resulted in the smallest air velocities, which, in turn, yielded the lowest pressure drop. For the conditions investigated, the fin height seems more influential on the friction factor than the fin length due to its much greater impact on the surface area.

Figure 7 shows the air-side heat transfer data in dimensionless form for the metal foam condensers. As expected, the $j$-factor decreases with increasing Reynolds numbers as a result of the increasing thermal capacity of the fluid stream. In general, an increase in the pore density (and interstitial area) reduces the overall surface efficiency (Kim et al., 2000; Ghosh, 2009). This explains the largest $j$-factors for MF3, which has the smallest pore density of all samples. Although MF2 has the same pore density, its $j$-factors are smaller than those for MF3 because of the higher porosity (and larger
hydraulic diameter) of the latter, which results in higher average heat transfer coefficients for MF3 because of the comparatively longer thermal entrance region (Ribeiro et al., 2012).

**Figure 5:** Friction factor as a function of the Reynolds number for the metal foam condensers.

Like the friction factor, the $j$-factor factor for louvered fin condensers decreases with increasing Reynolds number (see Fig. 8). Again, the influence of the fin height is much more pronounced than that of its length. At a given Reynolds number, the $j$-factor values of LF2 are smaller because of the lower face velocities.

In order to compare the two extended surfaces employed in this study, the air-side core volume goodness factors, i.e., the thermal conductance per unit volume and the pumping power per unit volume (Shah and Sekulic, 2003), are shown in Fig. 9. For the metal foam condensers, the sample with the largest porosity (and smallest surface area) showed the poorest performance. Additionally, for the louvered fin condensers, at large values of pumping power, the influence of the air-side heat transfer coefficient seems to be higher than that of the surface area, since condensers LF1 and LF3 exhibited a better thermal conductance per unit volume.
Figure 7: Colburn $j$-factor as function of the Reynolds number for the metal foam condensers.

Figure 9 also shows that, for the entire range of pumping power per unit volume, louvered fins have much higher thermal conductance than metal foams. So, at least for the geometric constrains investigated here (small face area on the air side), the use of a louvered fin as an air-side heat transfer enhancement medium is more advantageous than metal foams. As pointed out in a previous study (Ribeiro et al., 2012), these findings are in agreement with other authors (Kim et al., 2000; De Schampaheire, 2011; Muley et al., 2012), who also evaluated the thermal-hydraulic performance of metal foams with other types of enhanced surfaces.

Figure 8: Colburn $j$-factor as function of the Reynolds number for louvered fin condensers.

6. CONCLUSIONS

In order to consider their feasibility as part of a small-scale refrigeration system, louvered fin and metal foam condensers were evaluated experimentally. The condensers operated with forced convection for heat rejection, using air as a coolant. Different porosities and pore densities were employed for each metal foam condenser, whereas the fin height and length were varied for the louvered fin condensers.
A refrigerant loop facility that operated with iso-butane (R-600a) and a wind tunnel facility operating with air were designed and constructed for the small-scale heat exchangers performance evaluation. The condensing temperature used in the experiments was 45°C. Three different air flow rates were tested with face velocities ranging from 2.1 to 7.5 m/s.

![Graph](image)

**Figure 9:** Air-side thermal conductance per unit volume as function of the pumping power per unit volume.

The main conclusions of this study are as follows:

- **Metal foam condensers.** Increasing the pore density and decreasing the porosity resulted in higher values of friction factor. The \( j \)-factor was highest for the sample with the lowest pore density and the highest porosity. This is due to the reduction of the surface efficiency with the pore density (and surface area per unit volume). Overall, the pore density seemed to be the most influential parameter in determining the air-side heat transfer characteristics of the metal foam condensers.

- **Louvered fin condensers.** The effect of fin height was more important than that of the fin length on both the friction factor and Colburn \( j \)-factor. This is because of the aspect ratio of the heat exchangers, which were long and had a small face area.

- For the conditions evaluated in this study, louvered fins presented a much better thermal-hydraulic performance than metal foams and, thus, can be considered as a more appropriate heat transfer enhancement medium for small scale cooling applications.

**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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</thead>
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<tr>
<td>( A )</td>
<td>area</td>
<td>( m^2 )</td>
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<tr>
<td>( c_p )</td>
<td>specific heat</td>
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<td>inlet</td>
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</table>
\[ \dot{Q} \] heat transfer rate (W) \[ \text{Im} \] logarithmic mean

\[ Re \] Reynolds number (–) \[ \text{me} \] measured

\[ T \] temperature (°C) \[ \text{min} \] minimum

\[ UA \] overall heat conductance (W/°C) \[ \text{sur} \] surface

\[ V \] volume (m\(^3\)) \[ \text{tun} \] tunnel

\[ \dot{V} \] air flow rate (m\(^3\)/s)

\[ \dot{W} \] pumping power (W)

\[ \beta \] surface area density (1/m)

\[ \varepsilon \] porosity (%)

\[ \mu \] viscosity (kg/sm)

\[ \rho \] density (kg/m\(^3\))

\[ \sigma \] area ratio (–)

\[ \eta \] fin efficiency (–)

\[ \Delta P \] pressure drop (Pa)

\[ \Delta T \] temperature difference (°C)

**REFERENCES**


**ACKNOWLEDGEMENTS**

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