Experimental validation of critical heat flux (CHF) predictive methods for a new synthetic fluid with low environmental impact

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Abstract

This work presents experimental critical heat flux (CHF) values for a low environmental impact synthetic refrigerant and their comparison with well-known correlations from scientific literature. Tests were performed with HFO-1234yf in an aluminum heat sink made up of seven mini-channels, each of them 2 mm wide and 1 mm high. The heated length was 25 mm. Experiments have been obtained in a variety of thermodynamic condition: the R1234yf saturation temperatures $T_{sat}$ ranged from 25 up to 65 °C (corresponding to medium-to-high reduced pressures), whilst the mass fluxes $G$ had been fixed to 150 up to 300 kg/m$^2$ s. The tests have been carried out by increasing the heat dissipated by the boiling refrigerant until the thermal crisis occurred and the corresponding heat flux value was recorded as CHF. The experimental results were finally compared to the well-known correlations of Wojtan et al. [1], Kuan [2], Katto-Ohno [3], Zhang et al. [4] and Anwar et al. [5] to investigate their effectiveness with the present data. The latter correlation was found to better predict the experimental values.

Keywords: CHF; Multi-minichannels; Flow boiling; R1234yf; Dryout

1. Introduction

In the last decades, the attention in both research and industry fields has risen towards micro heat sinks working with boiling refrigerants, by exploiting their latent heat. In such systems, the highest heat flux that can be dissipated is called “critical heat flux” (CHF). Beyond this threshold, the liquid film thickness

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at the duct wall is replaced by the vapor, thus vastly decreasing the heat transfer efficiency. The direct consequence is an uncontrollable rise in the wall temperature and the possible failure of the electronic device. In the scientific literature, several correlations have been generated in order to predict the CHF in boiling flows and have been validated with water or ordinary halogenated fluids (HCFCs and HFCs), whereas few studies cope instead with low GWP refrigerants and they are all related to a single minichannel geometry. If this solution allows a fine control of local parameters such as the vapor quality and the wall superheat, it can not realistically be a system able to remove heat from square electronic devices. Moreover, most studies only examine saturation temperatures close to the ambient, whilst sizeable databases and prediction methods concerning medium-to-high reduced pressures are still insufficient.

The purpose of this paper is therefore to present experimental results of saturated CHF for a low environmental impact refrigerant (HFO-1234yf, GWP < 4), flowing into an aluminum multi-minichannel heat sink, in which seven rectangular channels with an equivalent diameter of 1.33 mm and a heated length of 25 mm were carved. All the experimental results are finally compared with five well-known correlations taken from the scientific literature, in order to investigate their effectiveness in geometries and saturation temperatures out of their range.

2. Experimental set-up

The experimental set-up is located in the Refrigeration Laboratory at the Dipartimento di Ingegneria Industriale of the Università degli Studi di Napoli “Federico II”. A schematic diagram of the test facility is shown in Fig. 1. All the experiments have been performed in a closed refrigerant loop where the inlet pressure and mass flux were independently controlled. A brief description of the main and cooling loops, as well as of the test section, are given in the following subsections.

2.1. Main and secondary loops

The refrigerant close loop is represented with the black line in Fig. 1. The subcooled liquid R1234yf is pushed towards a Coriolis mass flow meter by means of a magnetic gear pump. The fluid reaches the aluminum test section in which an absolute pressure transducer and a resistance thermometer are able to measure the inlet conditions. The magnetic pump may elaborate volumetric flows up to 2.5 dm$^3$/min, by changing the rotating speed from 1650 rpm up to 3400 rpm. Another resistance thermometer procures the refrigerant outlet temperature. Before the condenser, the R1234yf passes through a throttling valve that can be manually controlled in order to adjust the system pressure and the mass flow rate. A plate heat exchanger condenses the organic fluid thanks to the secondary cold fluid. Finally, the refrigerant flows into a liquid receiver and therefore into a double pipe heat exchanger to ensure sub-cooled conditions at the inlet of the electric pump. The refrigerant close loop is also equipped with a by-pass circuit, in which a certain fraction of the entire mass flow rate can be recirculated avoiding the test section, by means of a manually-controlled by-pass valve.

Demineralized water flows into the secondary loop, in which a thermostatic bath establishes the system saturation temperature. From the suction pipe, the cooling water flows directly into the double pipe heat exchanger and therefore into the plate heat exchanger. Two by-pass circuits may avoid the cooling flow into the sub-cooler and/or the condenser.

2.2. Test section

The test section is an aluminum multi-minichannel heat sink. Seven minichannels are carved into the main block. Fig. 2 (a) and (b) shows a photograph of the aluminum heat exchanger and its cross sectional sketch, respectively. Each minichannel is 2 mm wide, 1 mm high and 35 mm long.
The test section is electrically heated from the bottom with AC power supply given to a ceramic heater placed in a dedicated slot underneath the heat sink. The heating element is theoretically able to provide 697 W (at ambient temperature) over a base area of 2.50x2.50 cm². The AC voltage supply is given by a solid state relay which can handle up to 400 V. A digital wattmeter measures the imposed power to the ceramic heater and four cylindrical resistance thermometers Pt100 are equidistantly placed alongside the channels length for the wall temperature measurements.

The thermal insulation of the test section (and of the whole test rig) is guaranteed with an appropriate layer (2 cm) of synthetic rubber. This insulating material has a thermal conductivity of 0.042 W/m K in the whole range of operating conditions tested. The insulation effectiveness has been tested in Fig. 3, in which the heat losses are exposed as a function of the imposed heat. For high heat rates (>200 W), the losses may be considered negligible (<5 %) and the goodness of the test section insulation is confirmed.
3. Data reduction and experimental results

3.1. Data reduction

The refrigerant mass flux flowing into the test section has been deduced from the measured mass flow rate by taking into account the number of channels and their cross sectional area:

\[ G = \frac{m}{N w_{ch} h_{ch}} \text{ [kg/m}^2\text{s]} \]  \hspace{1cm} (1)

In the above equation, \( \dot{m} \) is the measured mass flow rate, while \( N \) is the number of minichannels and \( w_{ch} \) and \( h_{ch} \) are the minichannels width and height, respectively.

The wall heat flux from which the critical heat flux has been deduced has a simple expression that takes into account the real heat exchange area, by means of the equation below:

\[ \dot{q}_w = \frac{Q}{(2 h_{ch} \eta + w_{ch}) N L_h} \text{ [W/cm}^2\text{]} \]  \hspace{1cm} (2)

In the above equation, \( L_h \) represents the heated length (25 mm), \( \dot{Q} \) is the measured imposed heat and \( \eta \) is the fin efficiency, following the same approach of Park and Thome [6]. In this work, an average value for the fin efficiency (0.9) has been considered.

The real wall temperature for the wall superheat evaluation is not the value measured by the cylindrical RTDs placed alongside the test section. By assuming one-dimensional regime the wall temperature results to be:

\[ T_{wall} = T_{RTD} - \frac{q_w s_1}{\lambda_{al}} \text{ [°C]} \]  \hspace{1cm} (3)

Where \( T_{RTD} \) is the measured temperature and \( s_1 = 2.5 \text{ mm} \) is the distance between the RTDs placement and the channel wall (see Fig. 2 (b)). An average value of \( \lambda_{al} = 240 \text{ W/m K} \) has been considered for the aluminum test section thermal conductivity.
3.2. Experimental procedure

The inlet operating conditions in terms of saturation temperature and mass flux were kept constant at the beginning of each test. All parameters were remote-controlled via Labview software and Arduino One controller. Minor adjustments during tests were obtained thanks to the main loop by-pass valve and the throttling valve, thus having a greater or smaller free flow passage for the refrigerant.

At the beginning of each test, the heat coming from the ceramic element was applied in small increments by controlling the AC voltage supply via Arduino One. This procedure went on from the onset of the boiling mechanism up to the critical condition.

The system was considered stabilized when the maximum relative deviation of each value from the measured average in the latest 2 minutes was inferior to 3 %. The recorded time was set to 2 minutes, with a recording frequency of 1 Hz. Each test was therefore made up of 120 measurements and saved into a text file, from which the nominal value of each data point was assigned to the sample average value. MATLAB software was used for the data post processing, whereas the thermodynamic calculations were obtained with the software REFPROP 9, developed by NIST [7].

At the end of the test, the thermal crisis was deduced from the plotted boiling curve as a sudden decrease of its slope. However, especially for high mass fluxes, no abrupt rises in the wall temperature were detected, as was also observed by Park et al. [6] in their copper multi-minichannel heat sink. Following the same approach of Mauro et al. [8], in the present work the critical heat flux has been defined as the wall heat flux in which the boiling curve decreases its slope below a chosen threshold of 2.5 W/cm² K.

As an example Fig. 4 shows two boiling curves obtained for a saturation temperature of 45 °C and a mass flux equal to 150 and 300 kg/m² s, respectively. In case of low mass fluxes, the boiling curve exhibit a steep increases in the wall temperature as soon as the critical heat flux is reached. On the contrary, at G = 300 kg/m² s, a milder decrease of the boiling curves’ slope is detected.

Table 1 displays the conditions investigated in this work in terms of mass flux and saturation temperature and the experimental values of critical heat flux identified.

![Boiling curves at saturation temperature of 45 °C and: (a) G = 150 kg/m² s and (b) G = 300 kg/m² s](image)
Table 1 Experimental conditions and results

<table>
<thead>
<tr>
<th>Sat. temperature $T_{sat}$ [°C]</th>
<th>Mass flux $G$ [kg/m² s]</th>
<th>CHF [W/cm²]</th>
<th>Sat. temperature $T_{sat}$ [°C]</th>
<th>Mass flux $G$ [kg/m² s]</th>
<th>CHF [W/cm²]</th>
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</table>
\[ \dot{q}_{\text{CHF,Kuan}} = 0.2305 \cdot G \cdot \Delta h_{LV} \cdot \left( \frac{L}{D_r} \right)^{-0.9056} \]  

Zhang et al. [4] developed a correlation based on the Weber number evaluated with the channel diameter and the heated length-to-diameter ratio. This predictive method was obtained with a large database concerning flow boiling of water in tubes with different diameters and heated lengths. The investigated system pressures varied from 1.01 up to 190 bar and the mass fluxes ranged from 5.33 up to 134000 kg/m² s. Also the inlet vapor quality was taken into account:

\[ \dot{q}_{\text{CHF,Zhang}} = 0.0352 \cdot G \cdot \Delta h_{LV} \cdot \left[ We_{\theta} + 0.0119 \cdot \left( \frac{L}{D_r} \right)^{3.31} \cdot \left( \frac{\rho_v}{\rho_l} \right)^{0.3641} \cdot \left( \frac{L}{D_r} \right)^{-0.311} \cdot \left( 2.05 \cdot \left( \frac{\rho_v}{\rho_l} \right)^{0.17} - x_{\text{in}} \right) \right] \]  

Finally, Anwar et al. [5] in their recent study developed a new simple correlation for saturated CHF in a single minichannel. The effectiveness is guaranteed for equivalent internal diameters ranging from 0.64 to 1.70 mm and mass fluxes from 100 to 500 kg/m² s. Several refrigerants (R134a, R1234yf, R152a, R22, R245fa, R290 and R600) were tested.

\[ \dot{q}_{\text{CHF,Anwar}} = 0.27 \cdot G \cdot \Delta h_{LV} \cdot \left( \frac{L}{D_r} \right) \]

4.2. Statistical analysis with present data

In this section, a statistical comparison of the experimental data points against the abovementioned prediction methods is performed. This analysis is based on the parameters mean relative error MRD, mean absolute error MAD, percentage of data points within ±30 % (\(\delta\)) and 50 % (\(\xi\)) of the experimental data. The results are summarized in Table 2.

The correlation of Wojtan et al. underpredicts all the experimental results with a MAD and MRD equal to 27.1 % and -27.1 %, respectively. 65 % of the data points is confined within the range ±30 %.

Kuan prediction method slightly overestimates the experimental points, especially those obtained at higher mass fluxes, with a MAD of 17.4 %. The same trend is observed with the correlation of Anwar et al. In this case, however, the mean error is lower and even lower than that obtained by the authors themselves for their database.

Both Katto-Ohno and Zhang et al. correlations intensely underpredict all the data points. Particularly for the latter prediction method, only the 15 % of the experimental CHF falls into an error band of ±50 %.

Table 2 Statistical parameters for the comparison of four well-known prediction methods and the present database

<table>
<thead>
<tr>
<th>Correlation</th>
<th>MAD [%]</th>
<th>MRD [%]</th>
<th>(\delta) [%]</th>
<th>(\xi) [%]</th>
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</thead>
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<td>Kuan (2006)</td>
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<td>85</td>
<td>100</td>
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<tr>
<td>Zhang et al. (2006)</td>
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<td>-54,0</td>
<td>0</td>
<td>15</td>
</tr>
<tr>
<td>Anwar et al. (2015)</td>
<td>10,0</td>
<td>7,8</td>
<td>100</td>
<td>100</td>
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</tbody>
</table>
5. Conclusions

Several tests to investigate saturated critical heat flux (CHF) of refrigerant R1234yf in a multi-minichannel aluminum heat sink have been proposed in this work. The experimental values obtained with mass fluxes ranging from 150 up to 300 kg/m² s and with saturation temperatures ranging from 25 °C up to 65 °C have been compared to the correlations of Wojtan et al. [1], Kuan [2], Katto-Ohno [3], Zhang et al. [4] and Anwar et al. [5].

It has been seen that the correlation of Anwar et al. [5], explicitly developed for R1234yf among other fluids, best predicted the experimental CHF values, with a MAD of 10 % and a MRD of 7.8 %. Moreover, all the database fell into a band error of ±30 %. Similar trends were also observed with the correlation of Kuan [2], which exhibits a MAD of 17.4 %. It worth noting that the most deviating points from the expected values are those related to the highest mass fluxes, in which the experimental boiling curve keeps its gentle change in the slope.

The remaining correlations of Zhang et al. [4], Katto-Ohno [3] and Wojtan et al. [1] intensely underpredict the present database. Again, the large discrepancies are more evident for high mass fluxes and also for the highest saturation temperatures. As a matter of fact, almost all the predictive methods were constructed for saturation temperature quite close to the ambient.

Although the present database has given preliminary statistics about the possible extension of classical correlations in new operating conditions, it requires to be further upgraded. This would allow a more deep and accurate analysis of the predictive methods currently available in open literature.

References


