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Two-Phase Evaporation Pressure Drop Experimental Results for Low Refrigerant Mass Flux

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ABSTRACT

This study was conducted to evaluate pressure drop at extremely low velocity refrigerant flow through smooth circular evaporator tubes in household refrigeration systems. Experimental data was taken at mass fluxes under 70 kg/m\textsuperscript{2}s for two refrigerants, R134A and R600A, internal diameters of aluminum evaporators ranging from 0.186 inches (4.72 mm) to 0.317 inches (8.05 mm), U-bend internal radii ranging from 0.342 inches (8.69 mm) to 0.750 inches (19.05 mm) in horizontal and vertical orientations of evaporators. The geometry of the samples closely resembled a commonly used serpentine shape with multiple U-bends experiencing both, up- and down-flow of the refrigerant. Horizontal orientation test results with double the number of the U-bends suggest that horizontal U-bends (at least in tested geometries) at low mass fluxes less than 70 kg/m\textsuperscript{2}s do not play a significant role in a total frictional pressure drop in horizontal evaporators. A Curvature Ratio study showed no two-phase frictional pressure drop dependence on the curvature ratio of the U-bends. Main geometrical properties affecting two-phase frictional pressure drop in evaporators at low flows are internal diameter of the tube and a total length of evaporator tube. And main physical properties of the refrigerant flow affecting two-phase frictional pressure drop in evaporators at low flows are the type of refrigerant and its mass flow rate. Experimental databank of nearly 100 points collected within this study can be used to extend the application of the currently available two-phase pressure drop correlations or to develop a new one specifically for low mass fluxes.

1. INTRODUCTION

A 1\textdegree C saturation temperature drop as the result of the pressure drop through the evaporator reduces the COP of the common compressor by close to 1\%, which increases annual energy usage by approximately $7.68$ million, assuming 12 million household refrigerators in the US by 2017. Thus, the cost of energy required to compensate for the pressure loss through the system tubing needs to be understood as it is also a part of the overall performance of the appliance which must be competitive in the market.

Furthermore, evaporator tube geometry is optimized for material and manufacturing cost savings as well as energy and environmental considerations. Thus, lowering inner diameter of the tube to achieve material savings leads to pressure drop and higher energy usage concerns. Also in the desire to reduce space taken by the sealed system by tightening of the bend curvature ratio leads to experiments with evaporator shape and orientation. In addition, a reduction in internal diameters of the refrigerant system components allows decreasing the amount of refrigerant.
used, which should be recovered and recycled after the life of the refrigerator in order to decrease greenhouse effect. Deeper understanding of the diameter reduction tradeoffs will allow common usage of R600A refrigerant in the United States in the near future. This refrigerant provides 60-70 % higher cooling capacity compared to commonly used R134A, but it is being regulated to a maximum charge of 57 g per refrigerator due to its higher flammability.

At this time there are no available two-phase pressure drop correlations designed specifically for extremely low mass fluxes (less than 70 kg/m²s) due to the complexity of measurement of these low pressure drops. There is also no available experimental data that could be used to create a new correlation or extend the application of existing ones. Therefore, this study was conducted to investigate this low mass flux regime. Experimental data were taken at mass fluxes under 70 kg/m²s for two refrigerants, R134A and R600A, for internal diameters of aluminum evaporators ranging from 0.186 inches (4.72 mm) to 0.317 inches (8.05 mm), and with U-bend internal radii ranging from 0.342 inches (8.69 mm) to 0.750 inches (19.05 mm) in both horizontal and vertical orientations. The geometry of the samples closely resembled a commonly used serpentine shape with multiple U-bends experiencing both up- and down-flow of the refrigerant.

2. SCOPE OF WORK

The databank developed for this study is based on a full size evaporator with multiple U-bends in various orientations. The inlet and outlet refrigerant properties are controlled to reflect actual saturation properties commonly found in the modern bottom freezer refrigerator. Four internal diameters, four curvature ratios, two evaporator orientations and two refrigerants were tested. A detailed description of test samples is available in Table 1.

<table>
<thead>
<tr>
<th>Data set reference number</th>
<th>Refrigerant type</th>
<th>Tube internal diameter d, mm</th>
<th>Tube U-bend internal radius $R_{\text{internal}}$, mm</th>
<th>Total evaporator length L, m</th>
<th>Evaporator orientation</th>
<th>Total number of U-bends</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>R134A</td>
<td>8.00</td>
<td>14.61</td>
<td>10.10</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>2</td>
<td>R600A</td>
<td>8.00</td>
<td>14.61</td>
<td>10.10</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>3</td>
<td>R134A</td>
<td>6.78</td>
<td>13.46</td>
<td>10.06</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>4</td>
<td>R600A</td>
<td>6.78</td>
<td>13.46</td>
<td>10.06</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>5</td>
<td>R134A</td>
<td>6.17</td>
<td>13.46</td>
<td>10.01</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>6</td>
<td>R600A</td>
<td>6.17</td>
<td>13.46</td>
<td>10.01</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>7</td>
<td>R134A</td>
<td>4.72</td>
<td>13.46</td>
<td>9.95</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>8</td>
<td>R600A</td>
<td>4.72</td>
<td>13.46</td>
<td>9.95</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>9</td>
<td>R134A</td>
<td>8.00</td>
<td>19.05</td>
<td>10.31</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>10</td>
<td>R134A</td>
<td>8.00</td>
<td>14.61</td>
<td>10.10</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>11</td>
<td>R134A</td>
<td>6.78</td>
<td>19.05</td>
<td>10.27</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>12</td>
<td>R134A</td>
<td>6.78</td>
<td>13.46</td>
<td>10.06</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>13</td>
<td>R134A</td>
<td>6.78</td>
<td>8.69</td>
<td>9.78</td>
<td>Vertical</td>
<td>12V, 1H</td>
</tr>
<tr>
<td>14</td>
<td>R134A</td>
<td>6.78</td>
<td>13.46</td>
<td>10.06</td>
<td>Horizontal</td>
<td>13H</td>
</tr>
<tr>
<td>15</td>
<td>R600A</td>
<td>6.78</td>
<td>13.46</td>
<td>10.06</td>
<td>Horizontal</td>
<td>13H</td>
</tr>
<tr>
<td>16</td>
<td>R134A</td>
<td>6.78</td>
<td>13.46</td>
<td>9.76</td>
<td>Horizontal</td>
<td>26H</td>
</tr>
<tr>
<td>17</td>
<td>R600A</td>
<td>6.78</td>
<td>13.46</td>
<td>9.76</td>
<td>Horizontal</td>
<td>26H</td>
</tr>
</tbody>
</table>
Most of the data were taken for vertical orientation of the evaporator as it is the most common installation method in household refrigerators. This largest set of data was taken with a moderate U-bend radius, various tube internal diameters, and for both refrigerants (Data sets 1-8 in Table 1). Further, a few samples were remade with different U-bend sizes and tested vertically once again to capture the effects of the U-bend curvature ratio on the total pressure drop (Data sets 9-13). Next, some configurations were tested horizontally to understand the static pressure drop effect and to eliminate the vertical U-bends from the system (Data sets 14 and 15). And lastly, a double horizontal U-bend configuration was tested, where the same length of the evaporator was packed in the smaller space by doubling the number of horizontal U-bends (Data sets 16 and 17). Note that the total length L of the evaporator tube varies slightly from approximately 9.78 to 10.31 meters for different samples. This is due to intentional equality of the straight tubes lengths between adjacent U-bends (0.63 meters for all samples); while this dimension was kept the same, the U-bend length differed based on its radius. Another contribution to the total length difference was the length of the inlet/outlet ports of the evaporator. These ports were also participating in the heat exchange and total pressure drop through the evaporator, and local pressure drop through these was calculated using appropriate correlations.

Thus, this databank was created to understand most of the geometrical options considered during the design of the evaporator in a common refrigerator.

3. TEST FIXTURE

3.1 Test Fixture Description

The test fixture diagram is shown in Figure 1. It consists of a complete sealed system using the components of a household refrigerator. Figure 1 below shows the schematic of the test fixture used in this study.

![Figure 1: Experimental Fixture Diagram](image)

For all tests the fixture was located in a temperature controlled chamber maintained at 32.2° C. Temperature control is used to avoid small fluctuations in the heat flux and helped increase the effectiveness of the heater control. 32.2° C ambient temperature provides an additional heat load used to compensate for door openings in the actual refrigerator. Each evaporator being tested was located in an insulated enclosure consisting of thick foam panels covered with fiberglass sheets and vacuum panels. The enclosure is schematically shown in Figure 1 with a green rectangle. Heat transfer into the evaporator was achieved by evenly wrapping a flexible heater to the tube using...
aluminum tape around the full length of the evaporator. The insulation was needed to assure that all heat from the heater was effectively into the evaporator.

Two constant speed fans were installed inside of the insulation box to help mix the air for even heat flux distribution from the heater. The inlet and outlet refrigerant conditions were measured using Omega 4-wire RTDs and Setra absolute pressure gauges with 0-100 psia (0 to 689.5 kPa) ratings. Two Setra differential pressure gauges were installed in parallel to measure the pressure drop over the length of the evaporator. The operation ranges of these gauges, 0-1 psid (0-6.89 kPa) and 0-5 psid (0-34.47 kPa), were selected to maximize the accuracy of the measurements for the range of tested samples.

Upstream of the compressor an accumulator was used to prevent the liquid refrigerant entering the intake tube of the compressor. Two compressors in parallel were used to increase the flow range of the system. Low and high pressure charging ports were located on both sides of the main compressor to assist with servicing of the fixture. A secondary compressor was used in addition to the main one during testing with R600A. A backflow prevention valve was installed on the outlet of the secondary compressor to avoid potentially damaging backflow occurring during operation of the main compressor alone. Another Setra 0-5 psid (0-34.47 kPa) differential pressure gauge was installed across the condenser, and a mixing fan was used to enhance the heat rejection from the condenser. Upstream of the condenser, a site glass was used for visual control of the refrigerant subcooling. Another set of Omega 4-wire RTD and Setra absolute pressure gauge (0-300 psia, 0-2068.4 kPa) were located downstream from the condenser for quantification of subcooled refrigerant properties. A Coriolis flow meter was located further down the system for measurement of the all-liquid mass flow. A filter dryer was a part of the system downstream of the flowmeter and helped with the removal of the moisture from the system during operation.

3.2. Test Configurations

Figure 2 shows a sample of vertically tested evaporator with 12 vertical U-bends and a single horizontal U-bend. The direction of the flow was set up for mostly liquid to climb up one side of the evaporator through 6 vertical U-bends and 7 straight passes, cross over the horizontal U-bend at the top and flow down through 6 vertical U-bends and 7 straight passes to all vapor quality at the exit. This direction of the flow allowed higher differential pressure drops and improved accuracy of pressure drop measurements because the system operated closer to the midrange of the pressure transducers. All U-bends in a single evaporator were made with the same bend radius configured per Table 1. All straight passes of evaporators were oriented horizontally; portions of vertical straight tubes at the inlet and outlet of the evaporators were accounted for in calculations using corresponding straight tube pressure drop correlations including static pressure losses.

Figure 2: Twelve vertical and one horizontal U-bends in vertical evaporator orientation (samples 1-7 in Table 1)

Figure 3 represents all horizontal evaporator configurations with 13 horizontal U-bends. The horizontality and flatness of the evaporator set up was ensured with C-clamps and levels. As for the vertical configuration, the whole length of the evaporator tube between two differential pressure taps was accounted for in calculations.
Finally, Figure 4 represents the double U-bend horizontal configuration (26 U-bends). This configuration complicated the design of the insulating box due to its increased size in one direction. Thus, it was decided to fold this evaporator in two resulting in approximately 1.5 inches increase in elevation between the bottom and top layer of the 26 U-bends. This increase in elevation was assumed to be negligible. As for the 13 horizontal U-bend configurations, this set up was also clamped and leveled (not shown in Figure 4).

4. EXPERIMENTAL RESULTS

The experimental results are presented based on the orientations of the studies outlined in Table 1. For all graphical presentations of the data the plots created using R134A refrigerant are shown in solid lines and round markers, while plots created using R600A refrigerant are presented by dashed lines and diamond markers. Further, the data taken from the same evaporator are color coded the same for better visual analysis of the results. All pressure drop plots
have an additional point added at dP = 0 psid (kPa) and \( \dot{m} = 0 \text{ lb/hr (g/s)} \) assuming no pressure drop at no flow conditions. This point is used in determining a cubic trendline for each set of data which was shown to have the best graphical fit to the experimental data. The trendlines are purely for easier visualization of the results.

4.1 Vertical Orientation Study

Figure 5 shows measured two-phase pressure drop values through vertical evaporators with twelve vertical and one horizontal U-bend (data sets 1-8 in Table 1). This is the most complex part of the overall study conducted in this research.

Overall, this data set shows some similarities between two-phase pressure drop tendencies of R134A and R600A. Both refrigerants have minimal changes of the pressure drops for lower flows and larger tube internal diameters. As the flow increases pressure drop starts to climb and its gradient increases with decreasing internal tube diameter.

For both sets of data, for R134A and R600A, maximum tested flow is limited to a certain value for several reasons. For R600A testing maximum mass flow rate is governed by the abilities of the two parallel compressors to run with lower density refrigerant; density of R600A vapor is approximately 30% of the R134A vapor density. For R134A, the maximum tested mass flow is determined by the range of the largest differential pressure transducer used in the study (5 psid, (34.47 kPa)) and the goal to test with flows commonly used in the household refrigerators (normally 6 - 8 lb/hr, 0.76-1.00 g/s).

![Figure 5: Vertical evaporator orientation study with 12 Vertical and 1 Horizontal U-bends per data sets 1-12 from Table 1](image-url)
4.2 Vertical Curvature Ratio Study

Based on data sets 9-13 in Table 1, Figure 6 shows the results for several curvature ratios of the vertical evaporator configuration for R134A using tube internal diameters of 0.315 inches (8.00 mm) and 0.267 inches (6.81 mm). Four U-bend internal radii were tested: 0.750, 0.575, 0.530 and 0.342 inches (19.05, 14.61, 13.46 and 8.69 mm). The curvature ratio, D/d was calculated using equation below. An example calculation for the first data set with R134A, d = 0.315 inches (8 mm), R_{internal} = 0.750 inches (19.05 mm) is also shown below:

\[
\frac{D}{d} = \frac{2 \cdot R_{\text{internal}} + 2 \cdot t + d}{d} = \frac{2 \cdot 0.750\text{"} + 2 \cdot 0.030\text{"} + 0.315\text{"}}{0.315\text{"}} = 5.95
\]

Thus, a curvature ratio for this data set equals 5.95. In the above equation t stands for tube thickness.

Figure 6 shows that the pressure drop data points for same tube sizes are located in a very near proximity to each other seemingly independent from the curvature ratio of the samples. Thus for clarity of the plot only one trendline is plotted for each of the tube sizes internal diameter, since trendlines for other curvature ratios would be located in close proximity.

It is evident from the U-bend curvature ratio study that there is no clear defined relationship between the total two-phase pressure drop through the evaporator in the household refrigerator and the U-bend curvature ratio of the samples. However, a much stronger relationship is apparent for data sets with the same internal diameter of the tubes. This relationship is also evident in Figure 5 for data sets 1-12.

Figure 6: Vertical evaporator orientation study with 12 Vertical and 1 Horizontal U-bends per data sets 9-13 from Table 1.
4.3 Horizontal Orientation and Double U-Bends Study

Figure 7 represents the data for horizontal orientation of the evaporators in the 13 U-bend and 26 U-bend configurations. The data corresponds to data sets 14 to 17 in Table 1.

In the horizontal orientation of the U-bends the total pressure drop appears to be only slightly dependent on the number of the U-bends: the trendlines of the data points for 13 and 26 horizontal U-bends for the evaporators of very similar length are almost coincident. It appears that at low flows under 9.50 lb/m/hr (1.20 g/s) for R134A and 7.00 lb/m/hr (0.88 g/s) for R600A horizontal evaporators with U-bend radii at least as low as 0.5 inches (12.70 mm) have two-phase pressure drop that is independent of the number of the U-bends and can be treated as straight continuous horizontal tubes.

Furthermore, the close overlay of two curves, one with 13 U-bends and the other with 26 U-bends, for each of the refrigerants is evidence of the repeatability of the test process adopted for this research. Two different evaporators were used to collect this data and their manufacturing and installation, as well as test conditions, appeared to have a high accuracy and repeatability based on these results.

For future studies a simplifying assumption can be made based on this data that a single horizontal U-bend present in the geometry of the vertical evaporators at low flows can be treated as a part of the straight horizontal tube.

Figure 7: Horizontal evaporator orientation study with 13 Horizontal U-bends per data sets 14-15 and Double U-bend horizontal evaporator study with 26 Horizontal U-bends per data sets 16-17 in Table 1.
5. CONCLUSION AND FUTURE WORK

The goal of the work was to extend the current experimental data bank for refrigerant pressure drop during evaporation to conditions and geometries common for household refrigerators that could be used to extend the application of currently available two-phase frictional refrigerant correlations with a good accuracy or create a new correlation. Horizontal orientation tests with double the number of the U-bends showed that horizontal U-bends (at least in tested geometries) at low mass fluxes less than 70 kg/m²s do not play a significant role in a total frictional pressure drop in horizontal evaporators. A Curvature Ratio study showed no two-phase frictional pressure drop dependence on the curvature ratio of the U-bends. The main geometrical properties affecting total two-phase frictional pressure drop in evaporators at low flows are internal diameter of the tube and a total length of evaporator tube. And the main physical properties of the refrigerant flow affecting total two-phase frictional pressure drop in evaporators at low flows are the type of refrigerant and its mass flow rate. Refrigerant properties that might have some of the most significant effect are, perhaps, density and viscosity; however, the level of their influence needs to be further studied in the future work.

The next steps in studying two-phase frictional pressure drop at low flows is taking a more intricate approach towards adjusting one or more of the established two phase pressure drop models in order to extend their application to this range of flows. The data bank of nearly 100 data points created within this study can be used to check the prediction accuracy of two-phase frictional models such as Grönnerud (1979) and Müller-Steinhagen & Heck (1986) or one of the latest phenomenological models like Silva Lima & Thome (2012). Furthermore, a brand new empirical correlation could be potentially considered for this region of flows; however, it may require an increase in the data bank size for good accuracy.

Overall, the experimental data itself is very useful for future designs of evaporators in domestic refrigerators since the flows, geometries and orientations tested here are being commonly used today.

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