Performance Characterization of Two Selected Refrigerants in a Flat-Plate Micro-Tube Condenser

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Abstract

This work presents a detailed characterization study of two-phase condensing flows of two refrigerants, R134a and R-245fa, in a single water-cooled micro-channel with a 0.4 mm x 2.8 mm cross-section (0.7 mm hydraulic diameter and 7:1 aspect ratio) and length of 190 mm. A single micro-channel was chosen for the study to eliminate flow mal-distribution issues commonly encountered in micro-channel banks. The experiments included a parametric study of the effects on average heat transfer coefficient and overall pressure drop of variations across the micro-channel condenser with mass fluxes between 50 and 500 kg/m^{2}s, saturation temperatures between 30ºC and 70ºC, and inlet super heats between 0ºC and 20ºC. The inlet state was kept either at 100% quality or superheat and the outlet condition at 0% quality; these calculations were based on water-side heat rejection. Careful design and instrumentation of the test setup resulted in energy balance uncertainty within +/-11% and uncertainty of average heat transfer coefficient within +/-12%. It was determined that inlet superheat has little effect on heat transfer coefficient and overall pressure drop, but that the corresponding effects of saturation temperature and mass flux are significant.

1. Introduction

Major improvements in compact thermal devices have led to their application to a wide variety of technologies in the automotive, HVAC, oil/gas and aerospace industries. This rapid development of compact thermal devices has, in turn, increased research interest in developing the fundamental science and engineering of small-scale fluid flow and energy transport. The behavior of micro-devices is quite different from that of macro-scale devices. One of the major advantages of micro-devices over macro-devices is the increase in surface-to-volume ratio, which results in a proportional increase in the rate at which heat is rejected. Another effect of the diminished scale is the increased importance of surface forces and the diminished importance of body forces vs. surface tension for length-scales of less than 1 mm. In fact, friction is a major problem in micro-systems and is independent of the device's mass but proportional to its surface area.

Compared with traditional heat exchangers with channels of conventional size, micro-channels exhibit superb heat transfer characteristics. Since micro-channels have an increased heat transfer surface area and a large surface-to-volume ratio, they provide a much higher heat transfer capacity for a given weight/volume of heat exchanger. This heat transfer capacity will allow for the development of highly compact heat exchangers, which will, in turn, lead to system-level miniaturization. The reduced size at the system level can potentially offer substantial savings in the costs of system fabrication, transportation of the units, and building space use. Furthermore, micro-channels can support high heat fluxes with small temperature gradients and thus minimize failure risks due to thermal stress. This is especially important for cooling high-flux electronics and the next generation of microprocessors.

Considerable research has already been conducted on micro-scale single-phase flow phenomena, but most of this work has been driven by the needs of specific applications, from the original work of Tuckermann *et al.* [1] on cooling VLSI devices, to the work of Choi *et al.* [2] on using sub-cooled liquid N_2 to cool optical elements of a Giga-eV X-Ray radiation source. Results are available for only a limited range of flow characteristics and in many cases for relatively large flow channels, with dimensions on the order of millimeters rather than micrometers.

In recent years researchers have also investigated two-phase flow; however, most of these investigations have focused on only a few parameters. This work has resulted in a few general micro-channel condenser design correlations. But while this work has shed light on boiling heat transfer, little of it has focused on condensation, even though condensation in micro-channels is as important as boiling. A fundamental understanding of flow and heat transfer behavior of condensation at the micro-scale is essential for efficient design and analysis of systems and devices utilizing micro-scale structures. This is particularly the case for condensation in micro-channels, where little or no previous work is available. Previous work on condensation is summarized in the paragraphs below.

In a paper by Webb *et al.* [3] the two-phase flow was treated as an all-liquid flow with an equivalent Reynolds number, which was then used in a single-phase turbulent flow equation to predict condensation heat-transfer coefficient. The data used for the model verification were obtained for tubes with internal diameters ranging from 3.14 mm to 20 mm. However, the model was found to under-predict the experimental Nusselt number.

Garimella and Coleman [4] presented a study of the effect of tube diameter and shape on the flow regime transitions for a co-current flow of air-water mixture in tubes with small hydraulic diameters ranging from 5.5 mm to 1.3 mm and velocities ranging from 0.1 to 100 m/s and 0.01 to 10.0 m/s. Although the studies were limited to adiabatic flow and are not directly applicable to phase-change situations, it was found that for tube diameters less than 10 mm, the combined effect of the tube diameter and surface tension suppresses the stratified regime and increases the intermittent flow regime. These effects are expected to be compounded in rectangular channels of similar hydraulic diameters, and more so for high aspect-ratio channels.

Cavallini *et al.* [5] have presented results of two-phase flow frictional pressure drop data and condensation heat transfer coefficients using RI34a and R410a inside a flat multi-port, mini-channel tube. The results were for a tube with 1.4 mm hydraulic diameter and a length of 25.4 mm.

Rose [6] presented a theoretical model to predict film condensation heat transfer from a vapor flowing horizontally in square and equilateral triangular sections of mini-channels or micro-channels. His model was based on fundamental analysis that assumed laminar condensate flow on the channel walls and took into account surface tension, interfacial shear stress, and gravity. Channel sizes ranged from 0.5 to 5 mm, withR134a, R22, and R410a as the working fluids. The channel wall temperature was considered uniform and the vapor was considered saturated at the inlet.

The majority of the available studies on condensation in micro-channels are focused on circular or low aspect-ratio rectangular micro-channels. However two-phase flow patterns in the commonly used high aspect-ratio micro-channels are different from the low aspect-ratio micro-channels of the same hydraulic diameter, due to the ease of forming capillary bridges and cross-channel flow in the channels. The flow patterns significantly affect heat transfer and pressure drops, particularly in micro-channels where the effect of surface forces is significantly higher. The present work uses two different refrigerants to study the effect of saturation temperature, inlet superheat and mass flux on heat transfer coefficient and pressure drop in a rectangular, single micro-channel with hydraulic diameter of 0.7 mm.

2. Experimental Apparatus

The experimental set-up was designed and built such that heat losses were minimal. A schematic diagram of the test facility is shown in Figure 1. The testing setup consisted of two loops: (1) the refrigerant loop and (2) the water-chilling loop. The refrigerant loop consisted of a high precision Coriolis flow meter, variable-speed gear pump, pre-heater, heater, test section, reservoir, and sub-cooler. The gear pump circulated the refrigerant at a precise flow rate; then, the refrigerant passed through a flow meter where the refrigerant flow rate was measured. The use of a gear pump instead of a compressor ensured that pure refrigerant was tested, rather than refrigerant contaminated with oil leakage from the compressor. The gear pump allowed us to accurately quantify the heat transfer coefficient and pressure drop in the micro-condenser. It also allowed us to conveniently set and adjust small mass flow rates of the refrigerant across a broad range.

To control the operation pressure of the test loop, two electrical heaters were attached to the surface of the liquid reservoir. The heaters were connected to a Watlow® PID controller as a means of pressure stabilization. The controller received a 0-5 Volt output signal from the absolute pressure transducer, determined the corresponding system pressure, and controlled the power input to the heaters on the reservoir, thereby controlling the system pressure by increasing and adjusting the reservoir temperature.

To minimize heat loss, Dewar cylinder shells with a high vacuum between their walls were installed to insulate the test loops. The glass Dewar cylinders had diameters of 1 and 2 inches, respectively. The condenser and evaporator Dewars spanned the length of the condenser channel and evaporator tubes, respectively, as shown in Figure 4. In addition, the inner cylinder of the condenser Dewar acted as a water jacket. Figure 5 shows the details of the Dewar cylinders. In addition, a copper elbow connected the Dewar cylinders, as shown in Figure 4; the elbow had Kapton® heater patches controlled by a PID temperature controller, which kept the surface temperature of the elbow equal to that of the inlet refrigerant and thus minimized heat loss from the inlet T-connection, which acted as an adiabatic thermal shield.

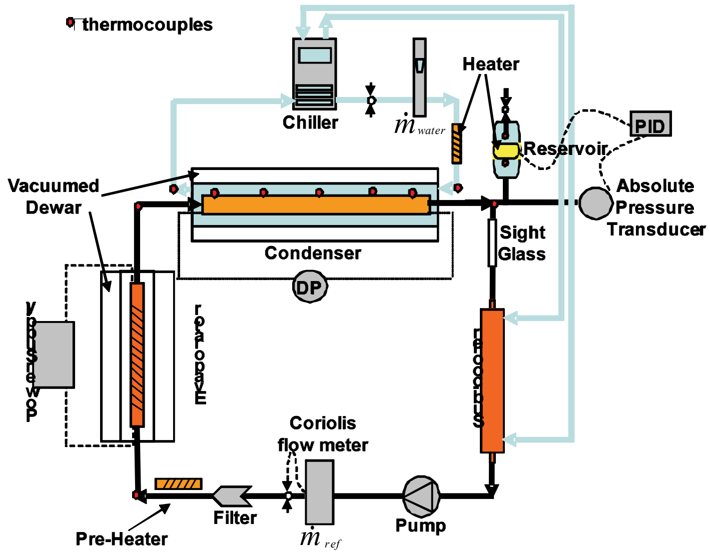

Figure 12. Schematic of experimental test setup.

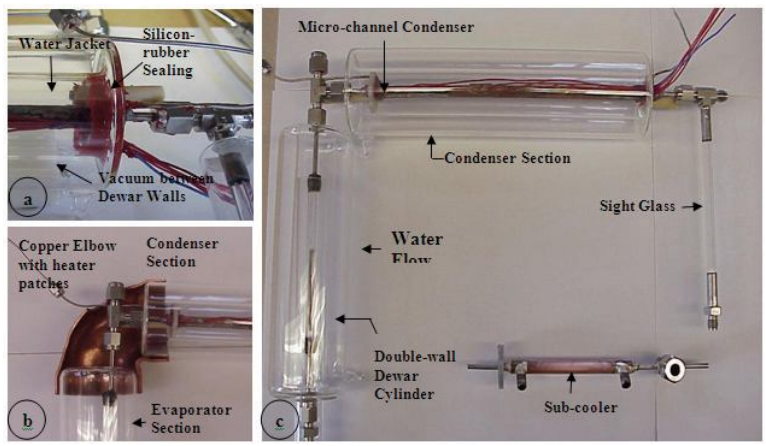

Figure 13. Details of test section, (a) details of the joint between the condenser section and sight-glass ,(b) details of the joint between the evaporator Dewar and condenser Dewar, and (c) view of the arrangement of various parts in the refrigeration loop.

3. Micro-channel Condenser Fabrication

The micro-channel was fabricated from a long OFHC copper block. A slitting saw in a milling machine was used to construct the micro-channel. For the inlet and outlet of the condenser, two 1/8 inch holes were drilled at each end, and two stainless steel tubes were soldered in the holes. In addition, two 1/16 inch holes were drilled, each at the ends of the channel, for pressure drop measurements. Then, the channel was filled with jeweler's wax, and the top surface was scraped flat and chemically cleaned in preparation for electroplating the channel. The channel was submerged into a copper-sulfate solution to be plated to the desired thickness on the remaining open surface of the channel. After electroplating, the wax in the channel was melted by heating the channel and then drained. Next, the channel was chemically cleaned to remove traces of wax or silver paint. On the outer surface of the channel, five single constantan wires and one copper wire were soldered to act as T-type thermocouples. The fabricated copper micro-channel is shown Figures 3 and 4.

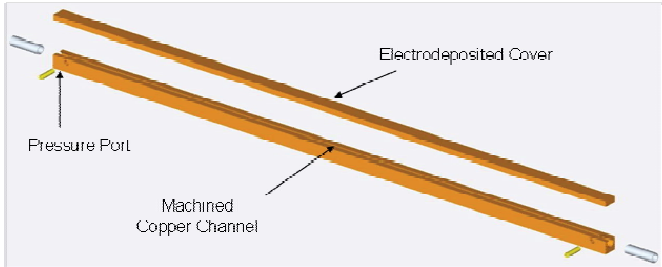

Figure 14. A drawing of the fabricated micro-channel.

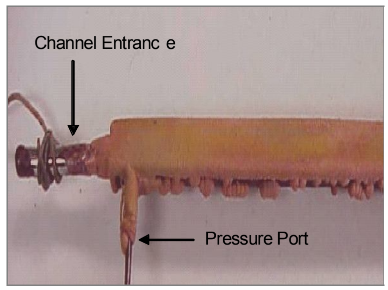

Figure 15. Electroplated micro-channel.

4. Test Fluids

Two refrigerants, R134a and R245fa, were used in this investigation to broaden the range in values for the various parameters studied in this investigation. Table 1 shows a comparison of the refrigerants: R245fa has a slightly better latent heat, a different density, and a different surface tension than R134a.

5. Results

Experimental tests were conducted to study the effect of three differen4t parameters on average heat transfer coefficient and pressure drop for the two refrigerants selected in this study. The parameters are presented in Table 2.

Table 3. Comparison of refrigerant properties.

	T_{sat} (°C)	P_{sat} (kPa)	Conductivity (mW/m.K)		Viscosity (μPa.s)		Density (kg/m³)		Enthalpy (kJ/kg)		Surface Tension (mN/m)
			liquid	vapor	liquid	vapor	liquid	vapor	liquid	vapor	
R134a	30	770	79.0	14.33	185.8	12.04	1187.5	37.53	241.7	414.8	7.42
	70	2116	61.7	20.45	106.4	14.65	996.2	115.6	304.3	428.7	2.61
R245fa	30	179	79.2	12.52	376.4	10.51	1325.1	10.11	239.6	427.5	13.41
	70	610	70.2	14.78	226.8	12.0	1204.7	33.43	295.7	456.6	8.35

Table 4. Range of testing parameters.

Testing Parameters	Range
Mass Flux (kg/m²s)	50 – 500
Saturation Temperature (ºC)	30 – 70
Super Heat (ºC)	0 - 15

6. Effect of Inlet Superheat

6.1 Average Heat Transfer Coefficient

It is worth mentioning here that saturation temperature is used to calculate the average heat transfer coefficient. Since in this study only the average heat transfer coefficient was measured for the entire channel length, it was not feasible to find the heat transfer in the superheated region. This is because local heat transfer coefficient is required to calculate the length of the superheated region, from which heat transfer in that region can be estimated. Figure **16** and Figure 6 illustrate that as inlet superheat increases, the average heat transfer modestly increases. This could be because we used the saturation temperature as the liquid temperature in calculations of heat transfer coefficient in the superheated region or because the length of the annular flow region increased, which would cause more heat to be transferred than the heat transferred at the lower inlet superheat. The heat transfer difference between 0 and 15ºC superheat was insignificant (10-15% of total heat transfer in the channel); and the local heat transfer coefficient at the inlet was significantly high; hence, the "extra" heat due to inlet superheat was quickly absorbed in the initial part of the channel. Another possibility is that if the heat transfer coefficient was significantly high at the inlet, a condensation film would be readily formed next to the walls while the fluid at the channel core was still at the superheated state. That is, a thermally non-equilibrium condition at each channel cross section would be formed. It is also obvious from the figures that the results of both refrigerants show similar patterns, which shows consistency in the data collection. In addition, R245fa has a higher heat transfer coefficient than that of R134a, which is mainly due to the higher latent heat of R245fa than that of R134a.

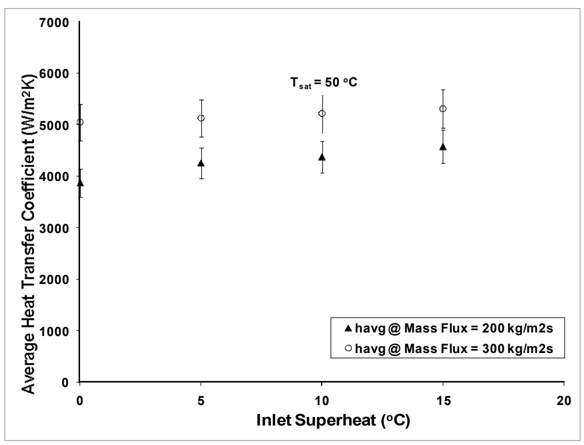

Figure 16. Effect of inlet superheat on average heat transfer coefficient at T_{sat} = 50ºC for R134a.

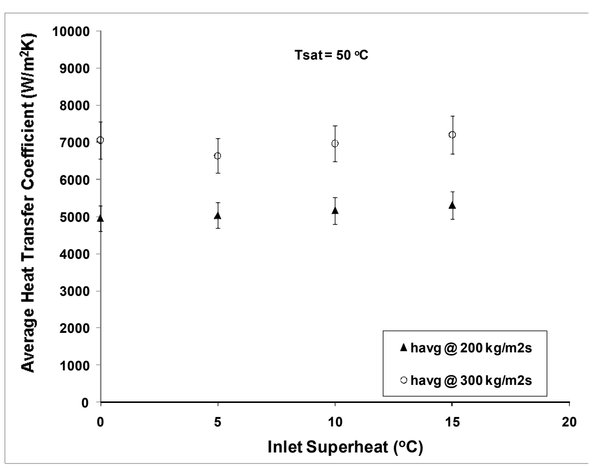

Figure 17. Effect inlet superheat on average heat transfer coefficient at T_{sat} = 50ºC for R245fa.

6.2 Pressure Drop

As a function of degree superheat at a given mass flux (200 & 300 kg/m^{2}s) and at 50ºC saturation temperature, the pressure drop is almost constant at different superheats, as shown in Figure **18** and Figure 8. It is known that pressure drop of two-phase flows differs from one flow regime to another, which means that pressure drop is dependent on flow regime. Thus, it is believed that the flow pattern in the significant length of the channel (after the annular flow zone) is more or less the same as in the zero-superheat case. Since all the tests were conducted under the condition of 100% quality entering the condenser and 0% quality leaving the condenser, the overall flow pattern is not much different at various superheats and can be expected to be almost the same except at the inlet, where the annular flow is expected to be a bit longer in higher superheats.

7. Effect of Mass Flux

7.1 Average Heat Transfer Coefficient

Figure 20 and Figure 10 show the effect of mass flux on average heat transfer for the refrigerants R134a and R245fa,respectively, at a fixed saturation temperature of 50ºC. This saturation temperature is chosen as the midpoint of the experimental range (30 – 70ºC). As seen in the figures, the heat transfer coefficient is positively affected by increasing the mass flux in the range of 50 – 500 kg/m^{2}s. This is because the velocity is increased, enhancing the convection effects. The heat transfer coefficient is dominated by the thin-film condensation process, and when the velocity is increased, the annular flow regime becomes longer, therefore contributing to the increase in heat transfer.

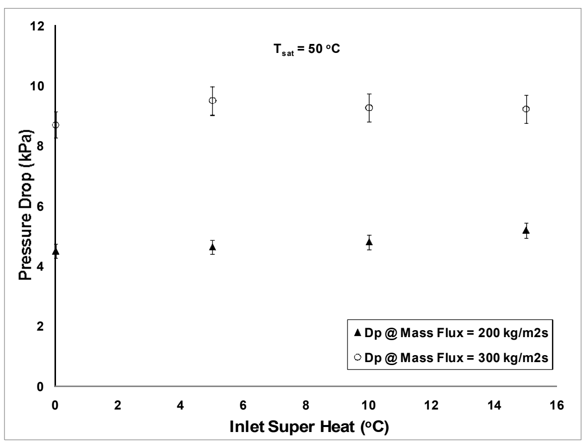

Figure 18. Effect of inlet superheat on pressure drop at T_{sat} = 50ºC for R134a.

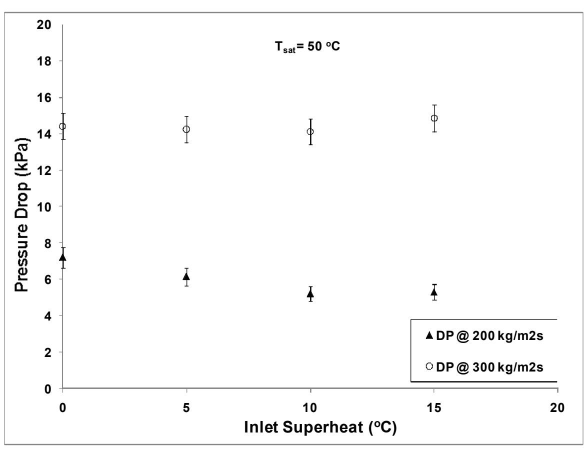

Figure 19. Effect of inlet superheat on pressure drop at T_{sat} = 50ºC for R245fa.

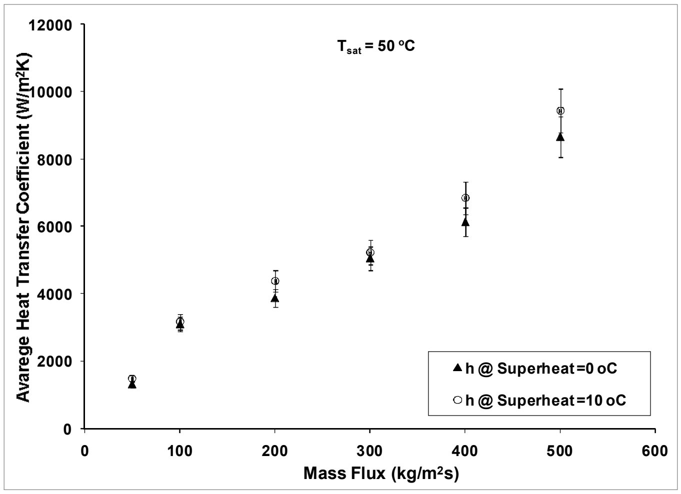

Figure 20. Effect of mass flux on average heat transfer coefficient for R134a at T_{sat} = 50ºC.

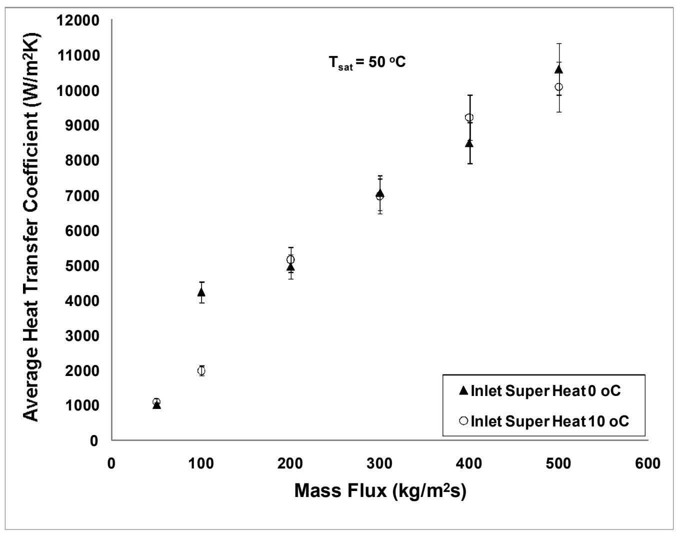

Figure 21. Effect of mass flux on average heat transfer coefficient R245fa at T_{sat} = 50ºC.

When the performances of the two refrigerants in terms of average heat transfer coefficients are compared, it is clear that R245fa has on average 15% better heat transfer than that of R134a. For example, at mass flux of 300 kg/m^{2}s the average heat transfer coefficient of R134a is 5051 W/m^{2}K, whereas for R245fa it is 6018 W/m^{2}K.

7.2 Pressure Drop

It is known that pressure drop has a squared power relation with the velocity and hence is significantly affected by increase in the mass flux, as shown in Figure **22** and Figure 12. It is noteworthy that R245fa has almost three times higher pressure drop as R134a does. This is mainly due to the higher surface tension, liquid viscosity and vapor density values of R245fa. However, from these properties vapor density contributes the most to the increase in pressure drop because vapor is dominant in the channel. The vapor density of R245fa is more than three times lower than that of R134a (Table **3**); therefore, the magnitude of pressure drop R245fa compared with R134a at the same mass flux is three times higher.

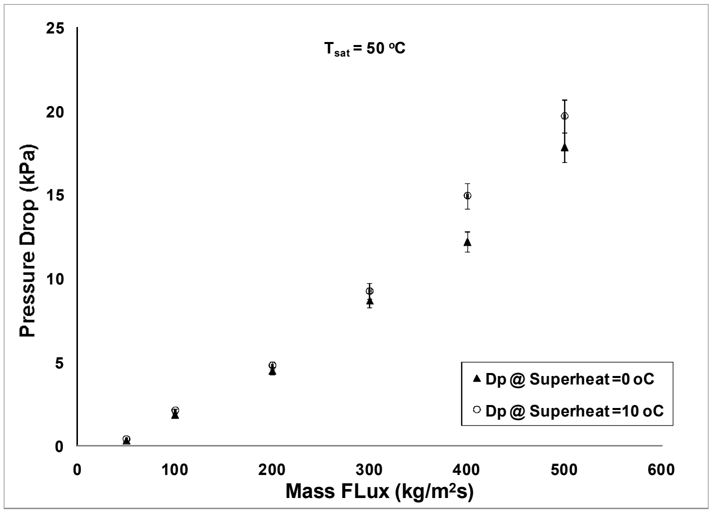

Figure 22. Effect of mass flux on pressure drop R134a at T_{sat} = 50ºC.

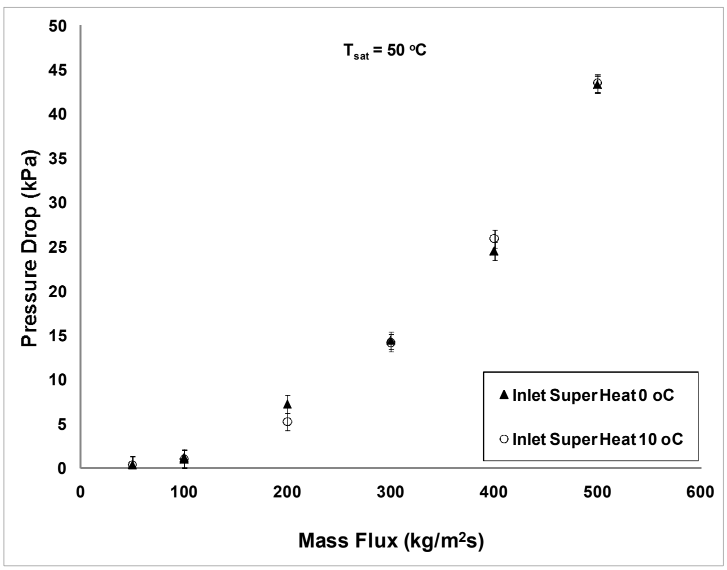

Figure 23. Effect of mass flux on pressure drop R245fa at T_{sat} = 50ºC.

8. Effect of Saturation Temperature

8.1 Average Heat Transfer Coefficient

Average heat transfer coefficient shows a descending trend with increasing saturation temperature, as shown in Figure 24 and Figure 14. This is mainly due to a reduction in the latent heat, h_{fg}, as the saturation temperature increases from 30ºC to 70ºC. In addition, as system pressure increases with increased saturation temperature, the specific volume decreases, and as a result, the vapor velocity decreases, which reduces the heat transfer. Therefore, the heat transfer is higher at lower saturation temperatures.

8.2 Pressure Drop

Increasing the saturation temperature significantly decreases the pressure drop, as seen in Figure **26** and 16. This is due to the sensitivity of pressure drop to liquid viscosity; hence, the pressure drop is higher at 30ºC than at 70ºC because the liquid viscosity is higher at 30ºC. Another factor that contributes to the decrease in pressure drop is saturation temperature: as saturation temperature increases, the volumetric flow rate decreases, thus lowering velocity and decreasing pressure drop.

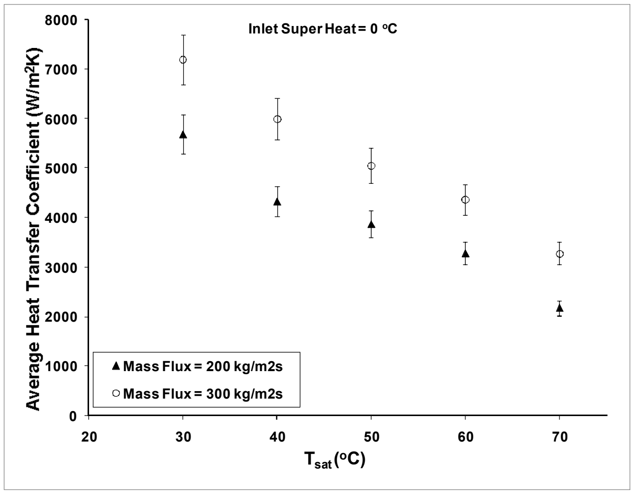

Figure 24. Effect of saturation temperature on average heat transfer coefficient for R134a.

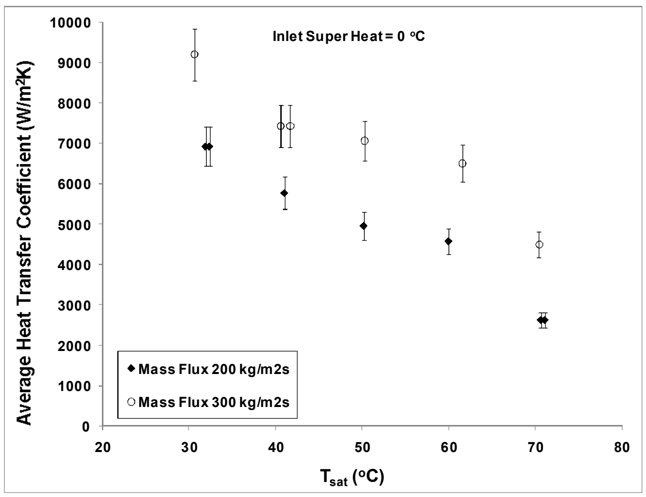

Figure 25. Effect of saturation temperature on average heat transfer coefficient for R245fa.

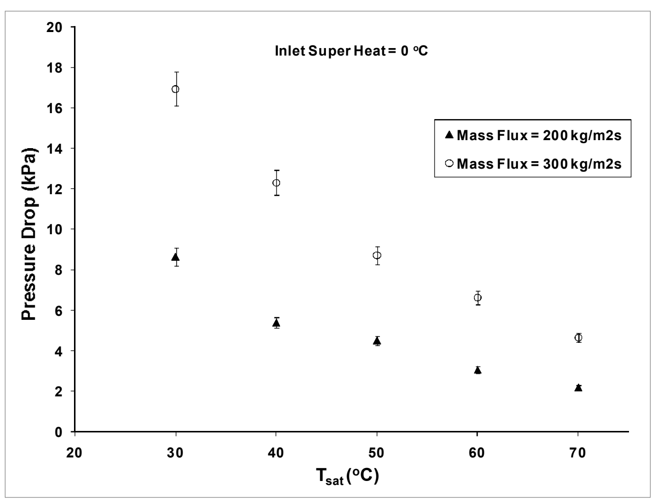

Figure 26. Effect of saturation temperature on pressure drop for R134a.

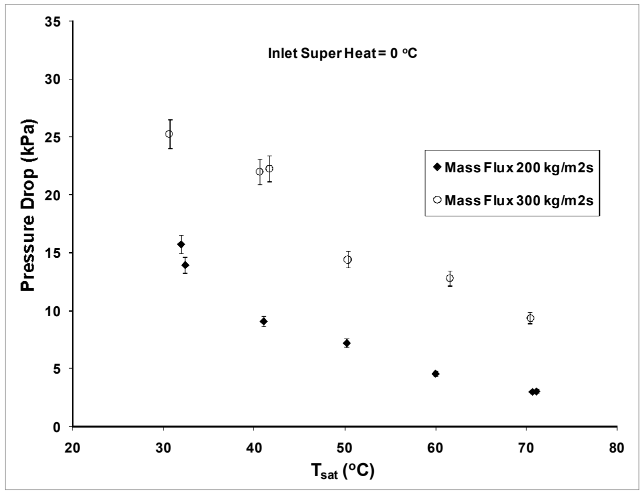

Figure 27. Effect of saturation temperature on pressure drop for R245fa.

9. Conclusions

To characterize condensation heat transfer and pressure drop coefficients in micro-channels with small hydraulic diameter and high aspect ratio, a rectangular micro-channel made of copper, with dimensions of 190 mm × 2.8 mm × 0.4 mm, was fabricated. Parametric tests as a function of various controlling parameters were conducted for condensation of refrigerants R134a and R245fa. The study revealed that for both refrigerants the heat transfer and pressure drop coefficients decreased with an increase in the saturation temperature. The results also show that the average heat transfer coefficients were only slightly affected by changes in the inlet superheat; the corresponding effect on pressure drops was insignificant as well. As expected, an increase of both heat transfer coefficient and pressure drop was observed as mass flux increased. A comparison of the two refrigerants demonstrated that refrigerant R245fa on average has 15%-better heat transfer than that of R134a. However, the pressure drop for refrigerant R245fa was almost three times higher than that of R134a for the same operating conditions.

10. References

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Dr. Amir Shooshtari received his Ph.D. degree in mechanical engineering from the University of Maryland, College Park, in 2004. Since 2005, he has been a member of the research faculty at the University of Maryland. His research interests include electronic cooling, two-phase flow, fluid flow in porous media, and modeling of electrohydrodynamics. Dr. Shooshtari is the author or co-author of over 15 publications in international journals and conferences. He is a member of ASME and ASHRAE.

Dr. Michael Ohadi is the Provost and Interim President of the Petroleum Institute. He received his Ph.D. from the University of Minnesota in 1986. He is internationally recognized for his work in enhanced heat and mass transfer in heat exchangers and energy systems. He has conducted many research projects for both industry and government. He has published over 140 refereed technical papers and is the author or co-author of nine U.S. patents. He is fellow of both ASME and ASHRAE, and has won numerous awards from both societies. He was recently featured in the technology section of the Washington Post for technology transfer efforts.