View metadata, citation and similar papers at core.ac.uk

brought to you by CORE

provided by Purdue E-Pubs

Purdue University
Purdue e-Pubs

International Refrigeration and Air Conditioning Conference

School of Mechanical Engineering

2018

Comparison of mini-channel condensation heat transfer for R513A and R134a

Jordan Alexander Morrow
Kansas State University, United States of America, jordan17@ksu.edu

Jarrod Booth
Kansas State University, United States of America, jarrodab@ksu.edu

Melanie Derby
Kansas State University, United States of America, derbym@ksu.edu

Follow this and additional works at: https://docs.lib.purdue.edu/iracc

Morrow, Jordan Alexander; Booth, Jarrod; and Derby, Melanie, "Comparison of mini-channel condensation heat transfer for R513A and R134a" (2018). *International Refrigeration and Air Conditioning Conference.* Paper 1974.
https://docs.lib.purdue.edu/iracc/1974

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html

Comparison of mini-channel condensation heat transfer for R513A and R134a

Jordan MORROW[1], Jarrod BOOTH[1], Melanie DERBY[1]*

[1]Kansas State University, Mechanical and Nuclear Engineering,
Manhattan, Kansas, USA
jordan17@ksu.edu, jarrodab@ksu.edu, derbym@ksu.edu

* Corresponding Author

ABSTRACT

There is increased interest in emerging, low global warming potential (GWP) refrigerants due to the phase out of hydrofluorocarbon (HFC) refrigerants (e.g., R134a); there are many questions that still need to be answered about replacement refrigerants, including heat transfer performance. One promising alternative to R134a (GWP = 1300) is R513A (GWP = 573), which is a non-flammable mixture comprised of 44% R134a and 56% R1234yf by weight. While R513A is attractive due to its lower GWP and A1 flammability rating, heat transfer data are limited. Condensation experiments were conducted for R134a and R513A in a vapor compression loop using multiport (i.e., 9 parallel channels) extruded aluminum tubes; each channel's hydraulic diameter was 0.72 mm. Experiments covered a range of average qualities (e.g., $0.2 < x < 0.8$) and mass fluxes (e.g., $300 < G < 500$ kg/m^{2}s). Heat transfer coefficients and pressure drops are reported. The heat transfer coefficients of R513A were found to be very similar to R134a. Condensation data were compared to the condensation heat transfer coefficient correlations developed by Shah (2009) and Kim and Mudawar (2013). The MAE of the Shah correlation for R513A was 22% and 6.5% for the Kim and Mudawar (2013) correlation.

1. INTRODUCTION

The Montreal Protocol prompted the phase out of chlorofluorocarbon (CFCs) refrigerants due to their ozone depletion potential (ODP), and zero-ODP hydrofluorocarbon refrigerants (e.g., R134a and R410A) replaced CFCs in some applications (Molina (2009)). Due to an interest in further lowering global warming potential (GWP), new refrigerants are of interest (Calm (2008)). Possible replacements include nonflammable, mildly flammable, and flammable refrigerants, rated A1, A2L, and A3, respectively, by ASHRAE (Wang et al (2012)). Therefore, there is increased motivation to reduce overall system charge as one approach to mitigate flammability risks (Pham and Rajendran (2012)); reducing heat exchanger volumes through using mini-channels (e.g., $0.2 < D_h < 3$ mm, Kandlikar and Grande (2003)) would reduce overall system charge.

Previous R134a condensation research included heat transfer, pressure drops, and flow regimes. Researchers observed that condensation heat transfer in single mini-channels increased with decreasing diameter, increasing mass flux, and increasing quality (0.691 mm, Kim et al. (2003); 0.4–4.91 mm, Garimella (2004); 0.96 mm, Cavallini (2006); 0.577, 1, and 1.2 mm, Wang and Rose (2006); 0.506 and 1.524 mm, Bandhauer et al. (2006)). Garimella (2004) studied R134a condensation heat transfer in 0.4–4.91 mm tubes at mass fluxes between 150 and 750 kg/m^{2}s. Heat transfer coefficients ranged from 2000 W/m^{2}K at low qualities and mass fluxes and up to 20,000 W/m^{2}K at the highest qualities and mass fluxes. Similarly, Cavallini et al (2006) measured R134a condensation heat transfer in a 0.96-mm circular tube; at a mass flux of 278 kg/m^{2}s, the heat transfer coefficients were 1750–6000 W/m^{2}K over the quality range, and at a higher mass flux of 555 kg/m^{2}s, the heat transfer coefficients were 4000–10,000 W/m^{2}K.

Due to the complexity of measuring small heat transfer rates associated with single channels and prompted by industrial and automotive applications, researchers also invested condensation heat transfer in multiple, parallel mini-channels. Researchers found similar trends in heat transfer coefficients' dependence on diameter, mass flux, and quality (10 channels, D_h=1.46 mm, Wang et al. (2002); 8 channels, D_h=1.11 m and 19 channels, D_h=0.80 mm, Koyama et al. (2003); 13 channels, D_h=1.4 mm, Cavallini et al. (2005); 5 and 7 channels, D_h=1 mm, Derby et al. (2012)) Several researchers observed flow regimes in mini-channels (Cavallini (2003), Garimella (2004); 2.67 and 4.91 mm, Coleman and Garimella (2003)), as flow regimes impact heat transfer and pressure drops. For example, Garimella et al. (2002)

developed a pressure drop model for intermittent flow during condensation of R134a for horizontal, 0.5- and 4.91-mm circular channels.

Condensation heat transfer correlations for small diameter tubes and refrigerants include those developed by Shah (2009) for tubes 2–49 mm, Kim and Mudawar (2013) for tubes 0.424–6.22 mm, and Cavallini et al. (2002) for tubes 3.1–8.8 mm. The Shah (2009) correlation was developed using 22 fluids including water, halocarbon and hydrocarbon refrigerants including R134a, propane, and other organics. This correlation was developed for horizontal, vertical, and downward inclined tubes. Kim and Mudawar (2013) developed a correlation for horizontal and vertical, single and multi-tubes using 17 fluids, including R134a, CO_2, methane, and FC-72. The horizontal tube correlation created by Cavallini et al (2002) was developed using seven refrigerants including R22, R134a, R125, R32, R236ea, R404A, R407C, and R410A.

R513A is an interesting lower GWP alternative to R134a. R513A is a R134a/R1234yf mixture (44/56 mass %); one advantage that R513A has over some of the other alternatives is the non-flammability of R513A (i.e., rated A1) (Devecioglu and Oruc 2015). With R513A being a relatively new refrigerant, the thermodynamic properties of this blend are still being understood. Akasaka et al (2015) studied the thermodynamic properties of this new mixture including critical density, pressure, and temperature, subsequently developing a mixture model of R513A. The majority of researchers modeled or tested R513A performance in current systems as a replacement for R134a. Shapiro (2012) experimentally tested several R134a alternatives' in a commercial bottle freezer and noted that the performance of R513A was similar to that of R134a. Kontomaris et al (2012) studied R513A as an alternative to R134a in a 560-ton centrifugal chiller. They observed that the chemical stability and miscibility with POE oil were both similar to R134a. They also found that R513A had minimally (0.6%) higher energy consumption. At full capacity, the COP of R513A was 2.4% lower than R134a, but at 20% full capacity, the COP of R513A was 7.6% higher. The total equivalent warming impact (TEWI) of R513A was 0.8-5.3% lower in high electricity carbon intensity areas and 31-48% lower than R134a in low electricity carbon intensity areas. Schultz and Kujak (2013) investigated several R134a alternatives (i.e., R513A, R450A, N-13b, R1234ze, ARM-42a) in a 230-ton water-cooled screw compressor-based water chiller. It was found that the cooling capacity of R513A was similar to R134a, although R513A showed a 1.4–4.3% lower energy efficiency ratio (EER). The compressor performance was similar with R134a and R513A; R513A has lower discharge temperature between 3.33 and 6.11 ºC. R513A was a near drop in replacement in terms of cooling capacity and a 3-4% reduction in efficiency. Mota-Babiloni et al. (2015) modeled multiple emerging refrigerants in a single-stage cycle and noted that R513A had a similar energy efficiency and a lower COP and cooling capacity. Belman-Flores et al (2017) modeled R513A in a small refrigeration system; R513A had similar power consumption and the COP of R134a was 6% lower than R513A. Mota-Babiloni et al. (2017) looked at R513A in an experimental vapor compression cycle. It was found that R513A had slightly higher COP (1.8 compared to 1.6 at evaporating temperature of -15 ºC), higher cooling capacity, and lower discharge temperature than R134a. R513A has a lower liquid thermal conductivity, liquid viscosity, vapor denisty and latent heat than R134a. They suggested that a new TXV would be necessary to run the system with R513A compared to R134a. In the vapor compression test facility, R513A had a higher cooling capacity, higher COP and lower discharge temperature. Llopis et al (2017) studied R513A and R450A as alternatives to R134a and R507A in a commercial direct expansion refrigeration system for medium temperature applications. Theoretically, in an ideal single stage vapor compression cycle, R513A had higher volumetric cooling capacity (0.3-4.3%) and lower COP (-2.5%-5.4%) than R134a. In the experimental system, they ran 24 hour energy consumption tests using a single stage cycle and showed that R513A had similar energy consumption to R134a.

While the majority of the literature modeled and tested overall system performance of R513A, a few research groups measured heat transfer coefficients. Kontomaris et al (2013) found that the condenser overall heat transfer coefficients (UAs) were 10 to 20% lower with R513A compared to R134a. Schultz and Kujak (2013) found that R513A had slightly lower (e.g., 9%) overall heat exchanger condensation heat transfer coefficients and mildly lower (e.g., 14%) refrigerant condensation HTCs. There are very limited data on HTCs values for condensation and evaporation of R513A. Kedzierski and Park (2013), studied boiling heat transfer coefficients of R134a, R513A, and R1234ze(E) in a microfin tube (i.e., overall diameter 9.5 mm with 60 0.2-mm high fins). It was found that of the three fluids, R134a had the highest heat transfer performance and liquid thermal conductivity. R513A HTCs were within 5% lower of R134a and nearly identical performance was achieved when the quality was less than 0.3, while R1234ze(E) were around 14% lower than R134a. The authors attributed differences in liquid thermal conductivity and reduced pressure as the causes for lower HTCs.

Since published data regarding the fundamental heat transfer performance of R513A are lacking, the objectives of this research are to experimentally study condensation heat transfer of R513A and compare these to heat transfer coefficients for R134a. This research will also use previously developed mini-channel heat transfer correlations to predict condensation heat transfer of R513A and determine how well these correlations work with this alternative refrigerant.

2. EXPERIMENTAL APPARATUS

2.1 Vapor Compression Cycle

Experiments were conducted in a vapor compression cycle (Figure 1) in which refrigerant mass flux and condensation temperature and quality were controlled; a brief description of the system follows. The condenser was divided into three parts: a tube-in-tube heat exchanger (pre-condenser), the test section, and a tube-in-tube heat exchanger (post-condenser). The temperature and pressure were measured at the pre-condenser entrance to calculate refrigerant enthalpy, and an energy balance was conducted on the pre-condenser to determine test section inlet quality. Heat transfer coefficients and pressure drops were measured in a condensation test section comprised of nine parallel channels; each channel had a hydraulic diameter of 0.72 mm. The post condenser was used to completely subcool the refrigerant, which was verified using a thermocouple at the post-condenser exit to monitor the subcooling. Mass flow rates of the fully-condensed liquid refrigerant were measured by a Coriolis mass flow meter (Micro Motion 2700) placed before the metering valve. Subcooled refrigerant flowed through the metering valve and subsequently entered a tube-in-tube evaporator where the refrigerant was boiled and then superheated. This superheated refrigerant entered the compressor (Aspen 14-24-1101), and, following compression, an oil separator ensured the return of the polyolester (POE) oil back to the compressor.

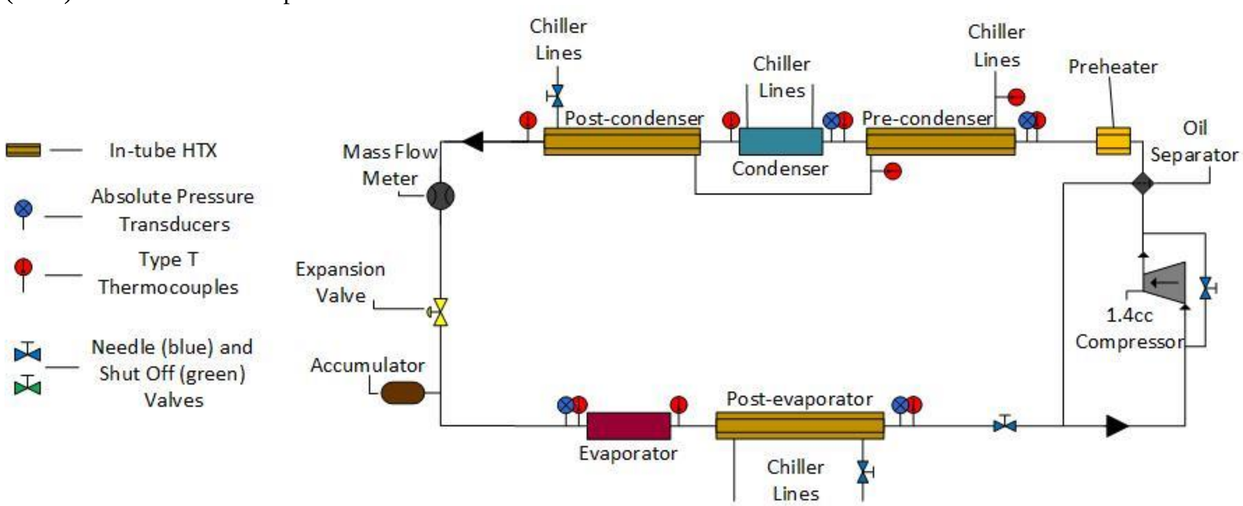


Figure 1: Experimental Vapor Compression Cycle Facility

2.2 Test Section

The condensation test section was designed to measure heat transfer coefficients and pressure drops. Nine parallel channels– comprised of an aluminum extruded tube with overall dimensions of 9.5 mm by 1.3 mm– were cooled by up to four thermoelectric coolers (TEC). Four tee-shaped copper blocks with three vertically aligned thermocouples in each block were used to measure condensing heat flux. The thermocouples were 1.59 mm in diameter and the holes were 6 mm from each other with the highest one being 1 mm from the top. The temperature at the top of the copper block was calculated by extrapolating from the closest thermocouple to the wall and the temperature gradient. From there, the condensing wall temperature was calculated using the thermal resistances in the indium layer and the test section lower wall. All TECS were connected to a cooling block used to reject heat and maintain TEC hot-side temperatures. On the refrigerant side, inlet and exit fluid temperatures were measured using thermocouple probes (Omega TQSS-116G-6). Absolute pressure (Omega PX309-300A5V) was measured at the test section inlet and the pressure drop across the test section was measured with a differential pressure transducer (PX409-050DWU5V). Fluid temperatures were obtained from the saturated temperature of the pressure measured at each test location using a linear pressure drop model.

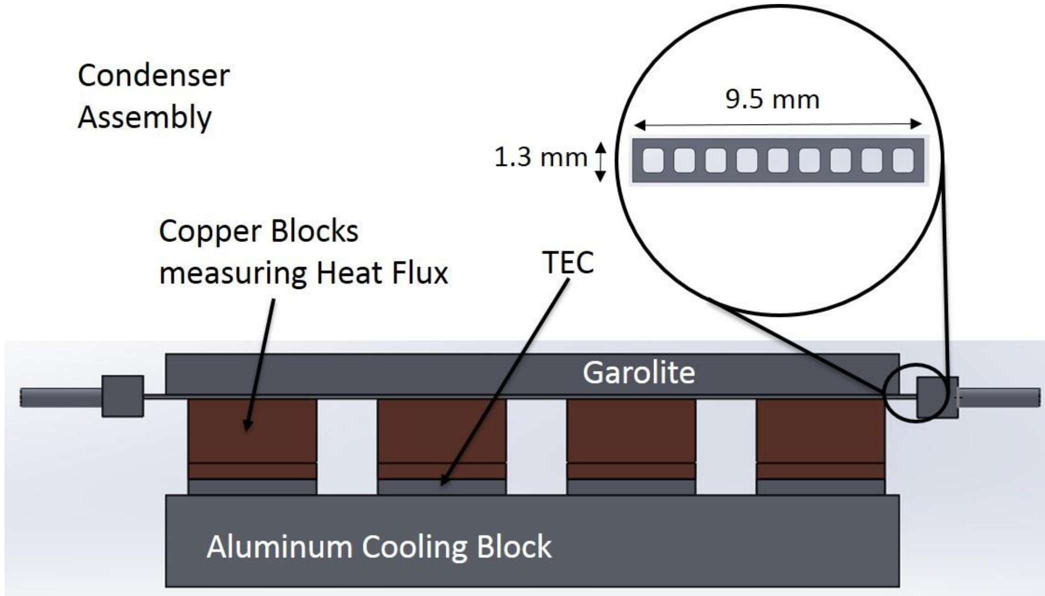


Figure 2: Test Section

2.3 Uncertainty Analysis

Effort was taken to reduce the uncertainties of the heat transfer measurements. T-type thermocouples were calibrated using 9 points within the range, including the boiling and ice points, with a reference thermometer (Fluke-1523). The reference thermometer had an uncertainty of ±0.03 °C and due to the DAQ, calibrated thermocouple uncertainties were estimated to be ±0.2 °C. The temperature gradient uncertainty in the copper blocks was calculated using the equation developed by Kedzierski and Worthington (1993),

$$\omega_{\frac{dT}{dy}} = \sqrt{\omega_{T_i}^2 + (\frac{q"D}{6k_{cu}})^2}\sqrt{\frac{1}{\sum_{i-1}^{N}(y_i-\bar{y})^2}} \tag{1}$$

where ω_T was the uncertainty of the calibrated thermocouples, y_i was the distance of the *i*th thermocouple from the cooling surface, and $\bar{y}$ was the average distance of the thermocouple from the condensation surface. The blocks were made of oxygen-free copper which helped reduce the thermal conductivity uncertainty. Pressure transducers were calibrated using a dead weight tester for a full-scale uncertainties of ±0.25% (absolute) and ±0.08% (differential); this corresponds to ±0.75 psi for absolute transducers and ±0.04 psi for differential transducers. The refrigerant mass flow rate was measured using a Coriolis flow meter with a measurement uncertainty of ±0.5%.

The entire system was well insulated to minimize heat loss. The inlet quality uncertainty was most affected by the pre-condenser, which was a tube-in-tube heat exchanger with the cooling fluid [i.e., water or ethylene glycol mixture (50%/50%)] entering at 10 to 20 °C. The inlet quality was calculated using an energy balance between the heat transfer in cooling the superheated refrigerant and the heat transfer into the cooling fluid. The biggest uncertainty in the pre-condenser was the cooling flow rate which was measured using a rotameter with a ±6% full scale uncertainty; therefore, the cooling fluid flow rate was kept above 40% full scale. The heat transfer coefficient uncertainty was calculated using the propagation of uncertainty analysis using the Kline and McKlintock (1953) approach,

$$\omega_{htc} = \sqrt{(\frac{\omega_{q"_{cond}}}{T_f-T_w})^2 + (\frac{\omega_{T_f}q''_{cond}}{(T_f-T_w)^2})^2 + (\frac{\omega_{T_w}q''_{cond}}{(T_f-T_w)^2})^2} \tag{2}$$

where q''_{cond} is the condensation heat flux, T_f is the fluid temperature, and T_w is the wall temperature.

3. RESULTS AND DISCUSSION

Validation of the experimental apparatus was conducted using liquid and results are presented in section 3.1. Sections 3.2 and 3.3 include experimental heat transfer coefficient and pressure drop data, respectively, including a comparison to condensation heat transfer coefficient correlations.

3.1 Single-Phase Validation

To show the validity of the apparatus, single-phase liquid R134a and R513A tests were conducted. With the lower heat load encountered in single-phase experiments, only the first TEC was used. Experiments included a laminar range of Reynolds numbers (900<*Re*<1600). Figure 1 shows the energy balance between the heat transfer measured in the four copper blocks and the energy balance of the cooled refrigerant. The plot includes both the R134a and R513A tests. Figure 2 shows the Reynolds number versus the Nusselt number for each test. The Nusselt number was then compared to the Shah and London (1978) correlation (when $x^* > 0.0015$) given by

$$Nu_x = 4.364 + 8.68(10^3 x^*)^{-0.506} e^{-41x^*}, \tag{3}$$

where x* is the dimensionless axial coordinate for the thermal entrance region and is given by

$$x^* = x'/D_h Pe, \tag{4}$$

in which *x'* is the distance from the entrance of the channel and the Peclet number, *Pe*, equals *Re Pr*. The correlation was developed to account for thermal entrance affects. The Shah and London (1978) correlation follows the trend well accounting for the uncertainties of the measurements. The increasing Nusselt number trend with respect to Reynolds number is explained by the entrance effects, which this correlation accounts for as well. With the good single-phase validation, the next step is two phase testing.

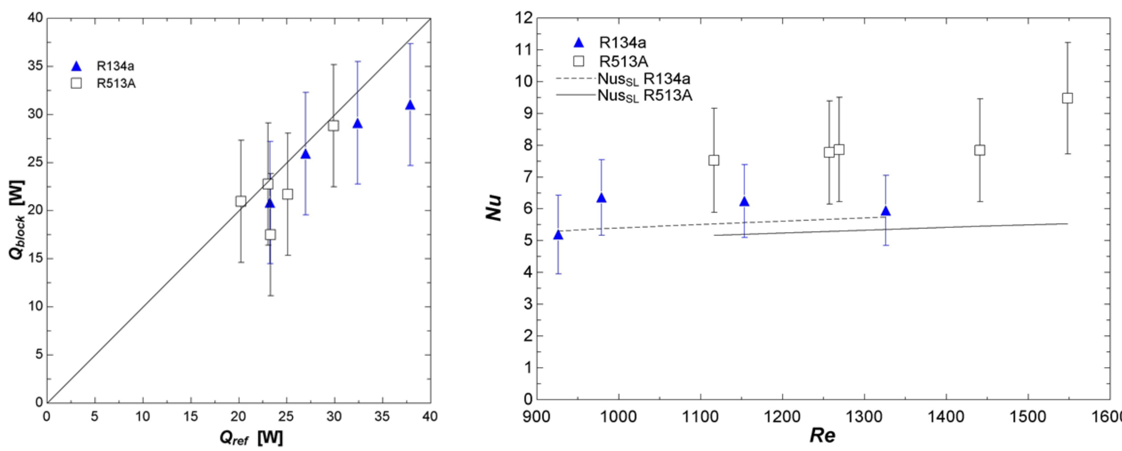
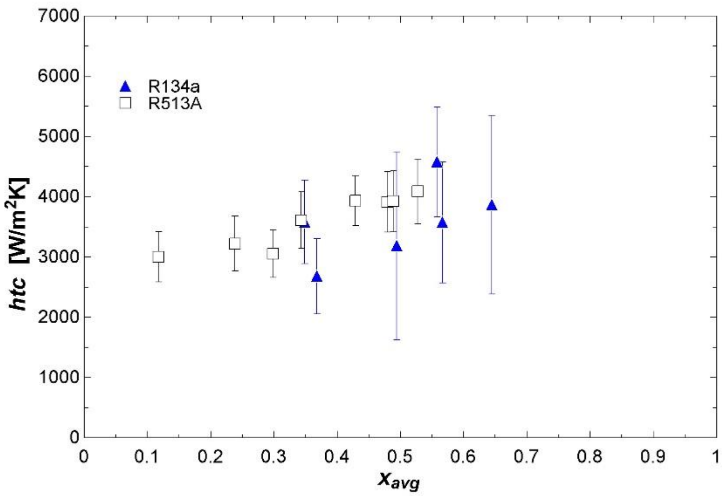


Figure 3. Single phase energy balance (left) and single phase Nusselt number (left)

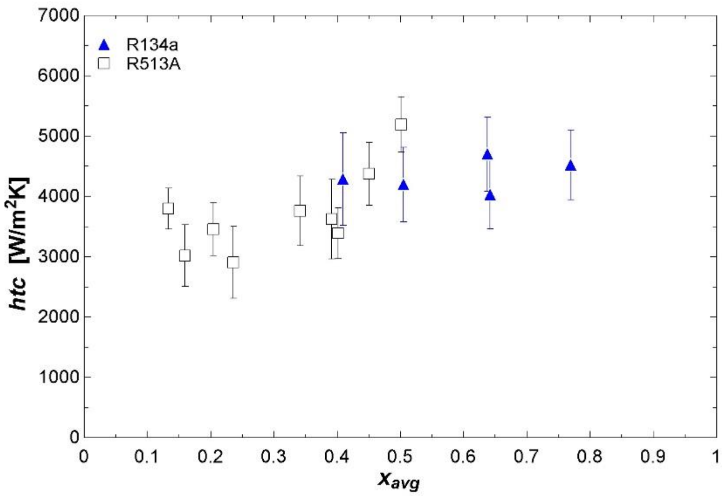
3.2 Condensation Heat Transfer Coefficients

Following single-phase validation, condensation heat transfer coefficients and pressure drops were measured. Baseline R134a data were collected first, followed by R513A for three mass fluxes: 300, 400, and 500 kg/m^{2}s. All data were recorded for saturation temperatures between 36 and 40 ^{o}C. Figure 1 shows the data collected for the mass flux of 300. The higher uncertainties of the R134a are due to very low heat fluxes (e.g., 25.7-34.9 kW/m^2) and small fluid-wall temperature differences (e.g., 2.6-2.9 ^{o}C). These uncertainties are reduced for higher mass fluxes of 400 kg/m^{2}s and 500 kg/m^{2}s as shown in Figures 2 and 3, respectively. Overall, condensation is a function of mass flux and quality, commensurate with findings in the literature (CITE), and condensation heat transfer coefficients of R513A are comparable to those of R134a. When compared to R134a, the HTCs for all mass fluxes fall within the range (5-14% lower) found in the literature (Shultz and Kujak (2013) and Kedzierski and Park (2013)) or even closer.

a)



b)



c)

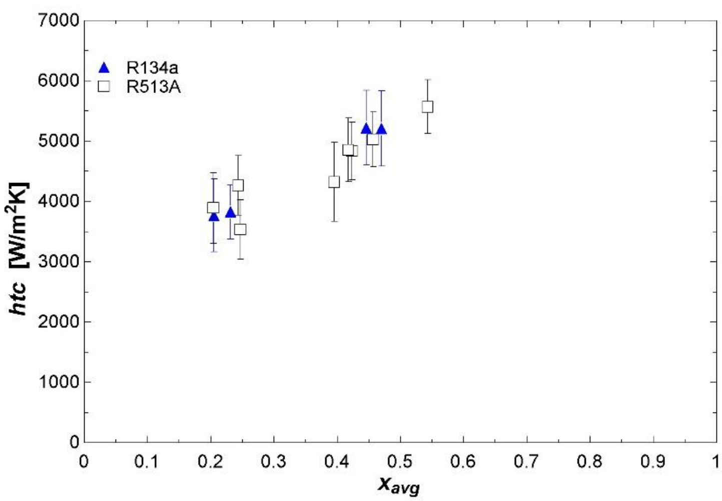


Figure 5. Heat transfer coefficients at mass fluxes of a)300, b) 400, and c) 500 kg/m^2

One of the objectives was to compare the measured R513A and R134a heat transfer coefficients to predicted heat transfer coefficients from condensation correlations, particularly since these correlations were not developed using emerging refrigerants. Two condensation correlations: Shah (2009) and Kim and Mudawar (2013). Both correlations were developed with R134a, but not with R1234yf, the second component of R513A. Figure 5 shows the experimental HTC values compared to the predicted values for the Shah (2009) correlation and Kim and Mudawar (2013) correlation in Figure 6 (left) and 6 (right), respectively. For these experiments, the Shah (2009) correlation generally overpredicts the R134a and R513A data. The Kim and Mudawar (2013) correlation, developed for condensation in mini-channels, reasonably predicts these data, with almost all of the R513A data being within 20%. The mean average error (MAE) for the Shah (2009) correlation for R513A was 22% and the MAE for the Kim and Mudawar (2013) correlation was 6.5% for R513A.

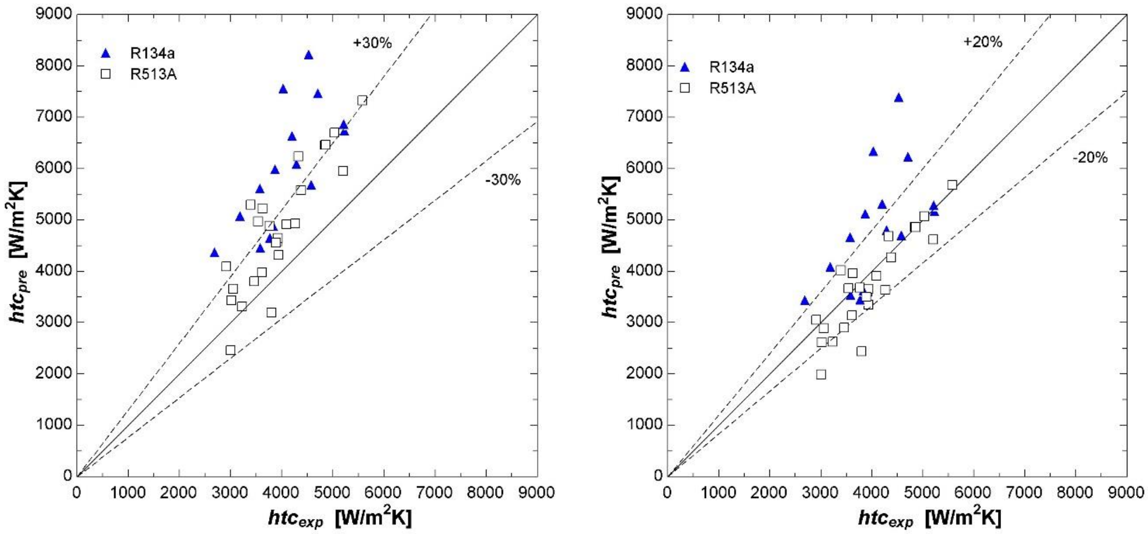


Figure 6. Data compared to the (left) Shah (2009) and (right) Kim and Mudawar (2013) correlations

3.3 Pressure Drops

The pressure drop was measured during tests using a differential pressure transducer. Figure 7 shows the measured pressure drops of R513A for all the tests. The pressure drop is reported for the average quality for the total test section. The horizontal bars show the total quality change over the drop in pressure. The higher mass fluxes have higher pressure drops, but it can be seen that the average quality and total quality change affect the pressure drop as well.

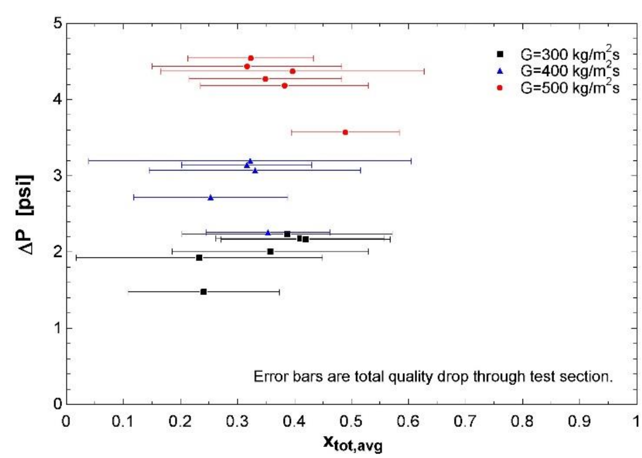


Figure 7. Test section pressure drop of R513A

4. CONCLUSIONS

Condensation heat transfer coefficients were experimentally measured and compared for R134a and lower GWP alternative R513A. The following conclusions may be drawn from this study:

- The heat transfer coefficients of R513A were similar to baseline R134a for all mass fluxes; heat transfer coefficients were a function of mass flux and quality.
- Experimental data was compared to two correlations; the MAE of the Shah (2009) correlation was 22% and 6.5% for the Kim and Mudawar (2013) correlation for R513A.
- Pressure drop increased with increasing mass flux and showed a dependence on the average quality across the test section and the total quality drop in the test section.
- More condensation data–as well as evaporation and COP data– are needed to fully understand the performance of R513A.

NOMENCLATURE

COP	coefficient of performance	(–)
D	diameter of thermocouple	(m)
D_h	hydraulic diameter	(m)
ΔP	pressure drop	(psi)
G	mass flux	(kg/m^2 s)
htc	heat transfer coefficient	(W/m^2 K)
k_{cu}	thermal conductivity of copper	(W/m K)
MAE	mean average error	(%)
Nu	Nusselt number	(–)
Nu_x	local Nusselt number	(–)
ω	uncertainty	
Pe	Péclet number	(–)
Pr	Prandtl number	(–)
q”	heat flux	(W/m^2)
Q	heat transfer rate	(W)
Re	Reynolds number	(–)
T	temperature	(C)
TEWI	total equivalent warming index	
x*	dimensionless axial coordinate for the thermal entrance region	
x’	distance from channel entrance	(m)
x	quality	(–)

Subscript

avg	local average
block	for the copper blocks
dT/dy	with respect to the temperature gradient
exp	experimental
f	fluid
htc	with respect to the heat transfer coefficient
pre	predicted
$q”_{cond}$	with respect to the condensation heat flux
ref	refrigerant
T_i	with respect to the temperature reading
T_f	with respect to the fluid temperature
T_w	with respect to the wall temperature
tot, avg	total average
w	wall

REFERENCES

Agostini, B., Watel, B., Bontemps, A., & Thonon, B. (2002). Friction factor and heat transfer coefficient of R134a liquid flow in mini-channels. *Applied Thermal Engineering, 22*(16), 1821-1834.

Akasaka, R., Higashi, Y., Yamada, Y., & Shibanuma, T. (2015). Thermodynamic properties of 1, 1, 1, 2-tetrafluoroethane (R-134a)+ 2, 3, 3, 3-tetrafluoropropene (R-1234yf) mixtures: Measurements of the critical parameters and a mixture model based on the multi-fluid approximation. *International Journal of Refrigeration, 58*, 146-153.

Bandhauer, T. M., Agarwal, A., & Garimella, S. (2006). Measurement and modeling of condensation heat transfer coefficients in circular microchannels. *Journal of Heat Transfer, 128*(10), 1050-1059.

Belman-Flores, J., Mota-Babiloni, A., Ledesma, S., & Makhnatch, P. (2017). Using ANNs to approach to the energy performance for a small refrigeration system working with R134a and two alternative lower GWP mixtures. *Applied Thermal Engineering, 127*, 996-1004.

Calm, J. M. (2008). The next generation of refrigerants–Historical review, considerations, and outlook. *International Journal of Refrigeration, 31*(7), 1123-1133.

Coleman, J. W., & Garimella, S. (2003). Two-phase flow regimes in round, square and rectangular tubes during condensation of refrigerant R134a. *International Journal of Refrigeration, 26*(1), 117-128.

Derby, M., Lee, H. J., Peles, Y., & Jensen, M. K. (2012). Condensation heat transfer in square, triangular, and semi-circular mini-channels. *International Journal of Heat and Mass Transfer, 55*(1-3), 187-197.

Devecioğlu, A. G., & Oruç, V. (2015). Characteristics of some new generation refrigerants with low GWP. *Energy Procedia, 75*, 1452-1457.

Garimella, S. (2003). *Condensation flow mechanisms in microchannels: basis for pressure drop and heat transfer models Proc. 1st Int.* Paper presented at the Conf. Microchannels and Minichannels (ASME, Rochester, NY) pp.

Garimella, S., Agarwal, A., & Killion, J. D. (2004). *Condensation pressure drops in circular microchannels.* Paper presented at the ASME 2004 2nd International Conference on Microchannels and Minichannels.

Garimella, S., Killion, J., & Coleman, J. (2002). An experimentally validated model for two-phase pressure drop in the intermittent flow regime for circular microchannels. *Journal of Fluids Engineering, 124*(1), 205-214.

Kedzierski, M., & Park, K.-J. (2013). Horizontal convective boiling of R134a, R1234yf/R134a, and R1234ze (E) within a micro-fin tube. *Journal of Enhanced Heat Transfer, 20*(4).

Kedzierski, M., & Worthington III, J. (1993). Design and machining of copper specimens with micro holes for accurate heat transfer measurements. *EXPERIMENTAL HEAT TRANSFER An International Journal, 6*(4), 329-344.

Kim, M., Shin, J., Huh, C., Kim, T., & Seo, K. (2003). *A study of condensation heat transfer in a single mini-tube and a review of Korean micro-and mini-channel studies.* Paper presented at the ASME 2003 1st International Conference on Microchannels and Minichannels.

Kim, S.-M., & Mudawar, I. (2013). Universal approach to predicting heat transfer coefficient for condensing mini/micro-channel flow. *International Journal of Heat and Mass Transfer, 56*(1-2), 238-250.

Kline, S. a., & McClintock, F. A., 1953, "Describing Uncertainties in Single-Sample Experiments," ASME Mech. *Eng, 75*, 3-8.

Kontomaris, K., Kauffman, J. P., & Kulankara, S. (2012). A Reduced GWP Replacement for HFC-134a in Centrifugal Chillers: XP10 Measured Performance and Projected Climate Impact.

Koyama, S., Kuwahara, K., Nakashita, K., & Yamamoto, K. (2003). An experimental study on condensation of refrigerant R134a in a multi-port extruded tube. *International Journal of Refrigeration, 26*(4), 425-432.

Llopis, R., Sánchez, D., Cabello, R., Catalán-Gil, J., & Nebot-Andrés, L. (2017). Experimental analysis of R-450A and R-513A as replacements of R-134a and R-507A in a medium temperature commercial refrigeration system. *International Journal of Refrigeration, 84*, 52-66.

Mota-Babiloni, A., Makhnatch, P., Khodabandeh, R., & Navarro-Esbri, J. (2017). Experimental assessment of R134a and its lower GWP alternative R513A. *International Journal of Refrigeration, 74*, 682-688.

Mota-Babiloni, A., Navarro-Esbrí, J., Barragán-Cervera, Á., Molés, F., & Peris, B. (2015). Analysis based on EU Regulation No 517/2014 of new HFC/HFO mixtures as alternatives of high GWP refrigerants in refrigeration and HVAC systems. *International Journal of Refrigeration, 52*, 21-31.

Owhaib, W., & Palm, B. (2004). Experimental investigation of single-phase convective heat transfer in circular microchannels. *Experimental Thermal and Fluid Science, 28*(2-3), 105-110.

Pham, H. M., & Rajendran, R. (2012). R32 and HFOs as low-GWP refrigerants for air conditioning.

Schultz, K., & Kujak, S. (2013). System drop-in tests of R134a alternative refrigerants (ARM-42a, N-13a, N-13b, R-1234ze (E), and OpteonTM XP10) in a 230-RT water-cooled water chiller. *Air-Conditioning, Heating, and*

Refrigeration Institute (AHRI) Low-GWP Alternative Refrigerants Evaluation Program (Low-GWP AREP),(Ed.).

Shapiro, D. (2012). Drop-in testing of next-generation R134a alternates in a commercial bottle cooler/freezer.

Wang, H., & Rose, J. W. (2005). *Film condensation in horizontal microchannels: effect of channel shape.* Paper presented at the ASME 3rd International Conference on Microchannels and Minichannels.

Wang, W.-W. W., Radcliff, T. D., & Christensen, R. N. (2002). A condensation heat transfer correlation for millimeter-scale tubing with flow regime transition. *Experimental Thermal and Fluid Science*, *26*(5), 473-485.

Wang, X., Amrane, K., & Johnson, P. (2012). Low global warming potential (GWP) alternative refrigerants evaluation program (Low-GWP AREP).