Comparison of R404A Condensation Heat Transfer and Pressure Drop with Low Global Warming Potential Replacement Candidates R448A and R452A

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Abstract

The objective of this paper was to measure and compare the heat transfer coefficient and pressure drop of R404A and two potential replacement candidate refrigerants with lower global warming potential, R448A and R452A. Experiments were conducted inside a 4.7 mm horizontal tube at mass fluxes ranging from 100 to 800 kg m⁻² s⁻¹, and at three different saturation conditions (40, 50, and 60 °C). R448A and R452A are zeotropic refrigerant mixtures, with temperature glides of approximately 4 °C and 3 °C, respectively. The Cavallini et al. [1] correlation predicted the heat transfer data best for all three refrigerants, once mixture effects were accounted for using the equilibrium Silver [2], Bell and Ghaly [3] method. The Haraguchi et al. [4] correlation predicted the R484A frictional pressure drop data best (mean average percent error, MAPE = 15%), while the Friedel [5] correlation predicted the R404A (MAPE = 15%) and R452A data (MAPE = 14%) pressure drop the best. The heat transfer coefficients of R404A were slightly higher than those of R452A and slightly lower than those of R448A at equivalent conditions. At the same mass flux, (G =600 kg m⁻² s⁻¹), R448A and R452A pressure drops are on average 81% and 3.3% higher than that of R404A. For the same cooling capacity and condenser design, the pressure drop of R404A and R452A are approximately the same and the pressure drop of R448A is approximately 16% lower than that of R404A.

Keywords: low GWP, R448A, R452A, zeotropic, condensation, heat transfer coefficient, pressure drop

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Nomenclature

Symbols

c_p	specific heat capacity	$\rm J~kg^{-1}K^{-1}$
D	diameter	m
G	mass flux	kg m ⁻² s ⁻¹
h	specific enthalpy	kJ kg ⁻¹
h_{fv}	latent heat of vaporization	kJ kg ⁻¹
k	thermal conductivity	W m ⁻¹ K ⁻¹
l	length of a segment	m
ln	natural logarithm	-
\dot{m}	mass flow rate	kg s ⁻¹
Р	pressure	kPa
ΔP	differential pressure drop	kPa
\dot{Q}	heat duty	kW
$q^{''}$	heat flux	kW m^{-2}
T	temperature	$^{\circ}\mathrm{C}$
U	uncertainty	-
V	velocity	m s ⁻¹
x	thermodynamic quality	-
\tilde{x}	molar concentration in liquid Phase	-
y	bulk mass concentration	-
\tilde{y}	molar concentration in the vapor phase	-

Greek Letters

α	heat transfer coefficient	W m ⁻² K ⁻¹
α_c	convective heat transfer coefficient	W m ⁻² K ⁻¹
α_f	film heat transfer coefficient	W m ⁻² K ⁻¹
β	experimental data that were within \pm 20% of the predicted value	%
μ	dynamic viscosity	kg m ⁻¹ s ⁻¹
ρ	density	kg m ⁻³
σ	surface tension	N m ⁻¹
Subse	cripts	
bubble	bubble temperature	
avg	average	
cond	condensation	
decele	ration deceleration pressure gain	
dew	dew temperature	
f	liquid phase	
fric	frictional pressure drop	
i	test section segment $\#$	
in	inlet	
inner	inner diameter	
int	liquid/vapor interface	
measu	<i>ured</i> experimentally measured value	
Mix	mixture	
out	outlet	

outer outer diameter

rand random uncertainty

ref refrigerant

- s sensible heat duty
- sat evaluated at saturation conditions
- sys systematic uncertainty
- T total heat duty
- tot total uncertainty
- v vapor phase
- w water

wall wall

1. Introduction

Phase out of the hydrofluorocarbon (HFC) refrigerants due to their environmental impact has led to the emergence of hydrofluoroolefin (HFO) refrigerants as potential replacements for industrial and residential applications in the heating, ventilation, air conditioning and refrigeration (HVAC&R) industry. Compared to pure HFCs, HFOs have significantly lower global warming potential (GWP). However, previous investigations on the condensation heat transfer of pure HFO refrigerants, such as R1234ze (GWP <1) and R1234yf (GWP <1)[6], have reported lower heat transfer coefficients compared to their HFC predecessors [7, 8, 9, 10]. Furthermore, both R1234ze and R1234yf are mildly flammable A2L refrigerants (ASHRAE [6]). HFC/HFO refrigerant mixtures are another potential alternative to HFC refrigerants. They have the advantage of lower GWP than pure HFCs and lower flammability risk than some pure HFO while maintaining heat transfer performance closer to the original HFC refrigerants.

R448A (GWP = 1360) and R452A (GWP = 1950)[6] are HFO/HFC mixtures proposed as replacements for R404A (GWP = 3943) for commercial refrigeration applications such as supermarket and industrial refrigeration systems. Both the proposed replacements are compliant with the F-gas Regulation No 517/2014 [11], which

prohibits the use of refrigerants with GWP higher than 2500 for refrigeration equipment, beginning in the year 2020. R448A is a mixture of R32, R125, R134a, R1234ze and R1234yf (26/26/21/7/20% by mass). R452A is a ternary mixture of R1234yf, R32 and R125 (30/11/59% by mass). Both R448A and R452A are non-flammable A1 refrigerants [6]. Similar to R448A and R452A, R404A is also a refrigerant blend comprised of HFC components (R125, R143a, R134a 44/52/4 % by mass). However, R404A ($\Delta T_{glide} = 0.3$ °C) is a nearly azeotropic mixture, while both R448A ($\Delta T_{glide} = 4.2$ °C) and R452A ($\Delta T_{glide} = 2.9$ °C) are zeotropic mixtures.

Previous studies on phase change of nearly azeotropic refrigerant mixtures indicate that their performance is generally well predicted by correlations developed for pure fluids [12, 13, 14]. However, zeotropic condensation differs from the condensation of pure fluids and azeotropic mixtures due to a continuously changing composition of the liquid and vapor phases and corresponding saturation temperature as the mixture is condensed. The difference between the temperature at which condensation begins (T_{dew}) and ends (T_{bubble}) at a constant pressure is referred to as the temperature glide. Furthermore, as a zeotropic refrigerant undergoes condensation, preferential condensation of the less volatile components at the liquid-vapor interface occurs, resulting in concentration gradients in both phases, and the build up of the more volatile components near the interface. This causes a depression of the local saturation temperature (lower than the equilibrium saturation temperature), degrading heat transfer. The depression of interface temperature and additional mass transfer resistance during zeotropic condensation is associated with lower condensation rates compared to individual components of the mixture [15, 16]. The influence of these mixture effects becomes increasingly pronounced with an increase in temperature glide of the mixture [17, 18].

Table 1 lists the thermodynamic properties of refrigerants R404A, R448A, and R452A at $T_{dew} = 55$ °C. The enthalpy of vaporization of R452A is approximately equivalent to R404A, suggesting similar mass flow rates for the same refrigeration capacity. On the other hand, the enthalpy of vaporization of R448A is 38 percent greater. Thus, the two candidates show promise from a thermodynamic perspective. However, it is necessary to understand the heat and momentum transport phenomenon of the new refrigerants to design new equipment. Therefore, in this study, we measured the local condensation heat transfer coefficients of refrigerants R448A and R452A in a single horizontal tube and compared the results with R404A data. The data were also used to evaluate the predictive capability of well established condensation heat transfer and two phase frictional pressure drop correlations, with and without mixture corrections.

	ΔT_{glide}	P_{sat}	h_{fv}	$ ho_l$	$ ho_v$	$\mu_l \times 10^{-6}$	$\mu_v \times 10^{-6}$	$k_l \times 10^{-3}$	$\sigma \times 10^{-3}$
	$^{\circ}\mathrm{C}$	(kPa)	$(kJ \ kg^{-1})$	$(\mathrm{kg}\ \mathrm{m}^{-3})$	$(\mathrm{kg}~\mathrm{m}^{-3})$	$({\rm kg}~{\rm m}^{-1}{\rm s}^{-1})$	$({\rm kg}~{\rm m}^{-1}{\rm s}^{-1})$	$(W m^{-1} K^{-1})$	$(N m^{-1})$
R404A	0.3	2571	93.3	859.7	162.7	78.3	15.4	50.8	1.2
R448A	4.2	2417	129.5	937.0	117.9	90.7	14.9	67.7	2.2
R452A	2.9	2519	95.0	937.6	157.6	83.7	15.9	55.2	1.4

Table 1 Refrigerant properties at $T_{dew} = 55 \,^{\circ}\text{C}$, from Lemmon et al. [19]

2. Prior work

Mota-Babiloni et al. [20] conducted an experimental system level investigation of a vapor compression system designed for R404A using R448A as a drop-in replacement. Tests were performed for a range of condensing ($T_{sat} = 32, 42, 52 \,^{\circ}$ C) and evaporating ($T_{sat} = -33, -20, -8 \,^{\circ}$ C) conditions. They found both the cooling capacity and the power consumption of R448A to be lower and the COP of R448A to be higher than that of R404A. They concluded that R448A can be a suitable drop-in replacement candidate for R404A for medium temperature applications such as refrigeration. While there are no published system studies for R452A, its similar thermophyiscal properties to R404A suggest it would also be a good replacement.

At present, there are no experimental studies on condensation of R448A or R452A available in the open literature. However, the condensation of other HFO/HFC refrigerant mixtures has received increased attention in the quest to find the best low GWP fluids. Del Col et al. [21] investigated the heat transfer coefficients of binary mixtures of R32 and R1234ze at two different mass compositions (23/77% and 46/54% by mass). The temperature glides were 11 and 8 °C, respectively. Tests were conducted in a circular mini-channel with ID = 0.96 mm at saturation pressures equal to 1370 and 1770 kPa and mass fluxes ranging from 150 to 800 kg m⁻² s⁻¹. The heat transfer coefficients of the mixture containing higher concentration of R32 were higher, which can be attributed to the higher liquid thermal conductivity of R32 compared to R1234ze. The heat transfer coefficients of the mixtures were significantly lower than the heat transfer coefficients. This further reaffirms that the degradation in heat transfer is not solely due to variation in thermodynamic mixture properties, but also due to additional mass transfer resistance.

In our prior work [14], we measured local heat transfer coefficients during complete condensation of the binary mixture R450A (42% R134a and 58% R1234ze by mass) in a tube with ID = 4.7 mm. Experimental data were collected at a range of mass fluxes (100 to 550 kg m⁻² s⁻¹) and saturation temperatures (45 and 55 °C). R450A

is a nearly azeotropic refrigerant mixture, proposed to replace R134a, with $T_{glide} = 0.6$ °C and therefore degradation due to mixture effects was minimal. Heat transfer data were predicted well by the Cavallini et al. [1] correlation. The mean average percent error (MAPE) was equal to 8.1% and 8.9% with and without correction for mass transfer effects, respectively. Pressure drop values were best predicted by the correlation of Haraguchi et al. [4]. Using these correlations, the predicted heat transfer coefficients of R450A were on average 5% lower and the pressure drop on average was 8% higher than those for R134a at the same conditions.

Azzolin et al. [22] investigated the condensation of R455A (R32, R1234yf and R744 at 21.5/75.5/3.0% by mass) and R452B (R32, R1234yf and R125 at 67.0/26.0/7.0%by mass) inside a 0.96 mm and an 8 mm tube. The T_{qlide} for R455A and R452B are 9.8 and 1.1 °C, respectively. Experiments were conducted at an average saturation temperature equal to 40 °C, at mass fluxes ranging from 100 to 600 kg m⁻² s⁻¹ for an 8 mm tube, and 200 to 800 kg m⁻² s⁻¹ for 0.96 mm tube. The heat transfer coefficients for both refrigerants and diameters were predicted with reasonable accuracy (MAPE values lower than 20%) by the Cavallini et al. [1] with the Silver [2], Bell and Ghaly [3] approach used to correct for the additional heat and mass transfer resistances. Azzolin et al. [22] noted that the degradation in heat transfer due to mixture effects was more pronounced for the higher T_{glide} mixtures. This is inline with the conclusions of Kondou et al. [23] who reported condensation and boiling heat transfer coefficients of two binary mixtures of R_{32}/R_{123} and two ternary mixtures of R32/R1234yf/R744 in a microfin tube. Both Kondou et al. [23] and Azzolin et al. [22] found the pressure drop of refrigerant mixtures to be well predicted by correlations for pure fluids [24, 5].

The review by Fronk and Garimella [15] on zeotropic condensation indicates conventional condensation correlations for pure fluids cannot be accurately applied to mixtures. Past researchers have developed models based on non-equilibrium [25, 18] and equilibrium assumptions [3, 2, 1]. The non-equilibrium approach models the condensation process by determining the concentrations in the bulk vapor, bulk liquid and interface, and are therefore more realistic. However, the non-equilibrium framework relies on the accuracy of underlying heat and mass transfer correlations in the vapor and liquid phase and is more complex to implement. On the other hand, the equilibrium approach (also known as Silver, Bell and Ghaly or SBG approach) simplifies the calculation procedure by assuming that the liquid and vapor concentrations are in equilibrium and that the sensible heat from the bulk vapor is transferred to the interface via convection only without two phase enhancement [3]. Therefore, this approach neglects the mass transfer in the vapor phase, but compensates for it by underestimating the heat transfer in the vapor phase. The SBG approach has enjoyed wide spread use due to its simplicity and accuracy in predicting heat transfer coefficients for mixtures typically used in HVAC&R applications. However, caution must be exercised when implementing the SBG method as Fronk and Garimella [17] demonstrated that SBG temperature fails to accurately predict heat transfer coefficients for high glide mixtures.

An experimental investigation of condensation heat transfer of R448A and R452A is needed to understand and model their heat and mass transfer processes. This information is required to accurately design the heat exchangers in next generation, low GWP systems. While the research of Mota-Babiloni et al. [20] shows that the mixtures have potential as drop-in replacements for R404a, even greater performance improvements could be realized for equipment that is designed to exploit the temperature glide phenomenon using validated models and correlations.



3. Experimental approach

Figure 1: Condensation facility schematic



Figure 2: Rendering of test section with single segment shown

3.1. Test facility

The test facility used in this study is the same as the one used in our prior work [14], with minor modifications to the test section. Readers are referred to Jacob et al. [14] for a detailed description of the components in the facility and a discussion of single-phase validation testing. A brief description is included here for completeness.

Figure 1 shows the schematic of the facility which was used to measure heat transfer coefficients of the zeotropic refrigerants. The refrigerant circulates in the primary loop which has been labeled with state points 1 to 4 in the figure. At state point 1, the refrigerant enters the test section as a superheated vapor. In the test section, the refrigerant condenses completely and leaves as a subcooled liquid. The cooling in the test section is provided from the water loop which consists of a pump, a volumetric flow meter, a brazed plate heat exchanger and a reservoir open to the ambient. Sight glasses are installed at the inlet and outlet of the test section to ensure that the refrigerant entering and leaving the test section is completely vaporized and condensed, respectively. The refrigerant then flows through a post cooler. The cooling in the refrigerant and the water loop is provided using a Neslab Merlin M150 recirculating chiller. A magnetic drive gear pump (Micropump GC series with M25 gear set) is used to pump the refrigerant to the evaporator section where it is completely vaporized. Refrigerant mass flow rate is measured using a Micro Motion Coriolis flow meter (Part No. CMF010M). An inline static mixer is used to ensure that the refrigerant is fully mixed and no entrained liquid droplets are present in the vapor flow at the test section inlet.

3.2. Test section

The test section is a tube-in-tube heat exchanger. Refrigerant flows inside of a copper tube with an ID = 4.7 mm. It is cooled by counter-flowing water in the annulus. The outer tube of the test section is a PVC tube with ID = 15.8and OD = 21.3 mm. The total heat transfer length is 1.6 m. The test section is divided into seven instrumented segments to measure quasi-local condensation heat transfer coefficients of the refrigerant in each section, as condensation proceeds along the length of the tube. Figure 2 shows the inlet and outlet temperatures of the water in each section are measured using RTDs located in the middle of the coolant annulus. The copper outer surface temperatures are measured using self adhesive surface thermocouples. 3D printed mixers made of polylactic acid (PLA) are used to ensure that the water is fully mixed prior to the temperature measurements.

In the test section, two 25.4 cm long segments are situated towards the refrigerant inlet which are followed by five 17.8 cm long segments. The refrigerant undergoes desuperheating in the two longer segments at the beginning of the test section. The subsequent shorter sections are designed to measure condensation data at a higher resolution during condensation. The heat loss in the test section was minimized with polyethylene insulation (thickness = 1.9 cm).

4. Data reduction

During each experimental condition, the system was allowed to reach steady state. Temperature, pressure and flow rate measurements were then collected for a duration of 4 minutes at 1 Hz. Heat transfer coefficients, frictional pressure drop, and the corresponding uncertainties were calculated from the methods described below. Data processing was conducted in the Engineering Equation Solver [26] software using refrigerant property data from REFPROP 10 [19]. All saturation properties were calculated assuming the measured inlet pressure. For the data point with the highest pressure drop, the change in saturation temperature was less than 0.23 °C and the enthalpy of vaporization less than 0.4 kJ kg⁻¹ (0.3 %). Thus, the effect on pressure drop on calculated parameters was negligible.

4.1. Heat transfer coefficients

A heat loss analysis using a thermal resistance network was conducted for the test with the highest overall heat loss. The heat loss from the coolant was estimated by determining the resistances due to convection through water, conduction through the PVC tubing (thickness = 0.55 cm) and polyethylene insulation (thickness = 1.9 cm), and air convection over the insulation. A conservative $\pm 200\%$ uncertainty was

assigned to each of the resistances. The evaluated heat loss was less than 1% of the total heat duty for the segment with highest temperature difference between the ambient and the coolant. The differences in the calculated heat transfer coefficients from these heat duties was low (0.2%). Therefore, for simplicity, the heat loss was neglected in the calculation of heat transfer coefficient. For a given segment in the test section, the total heat lost by the refrigerant is assumed equal to the heat gained by the water. The heat gained by the water in each segment can then be evaluated from an energy balance (Equation 1).

$$\dot{Q}_i = \dot{m}_w \cdot (h_{w,out} - h_{w,in})_i \tag{1}$$

where $\dot{m_w}, h_w, \dot{Q}$ are the water mass flow rate, enthalpy, and heat duty gained in segment number *i*. Water properties are evaluated based on the measured temperature and pressure. On the refrigerant side, the heat duty evaluated using Equation 1 is used to determine the refrigerant enthalpy along the test section:

$$h_{ref,out,i} = h_{ref,in,i} - \frac{\dot{Q}_i}{\dot{m}_{ref}} \tag{2}$$

The enthalpy of the superheated refrigerant at the test section inlet is evaluated based on temperature, pressure and bulk concentrations of the refrigerant. The average quality (Equation 3) and equilibrium saturation temperature (Equation 4) in a segment where two phase flow occurs can be evaluated from REFPROP 10 [19].

$$x_{ref,avg,i} = f(P, h_{ref,avg,i}, y_1, y_2...)$$
 (3)

$$T_{ref,sat,avg,i} = f(P, h_{ref,avg,i}, y_1, y_2...)$$

$$\tag{4}$$

where x, P, y are the thermodynamic quality, pressure and bulk concentrations. The quasi-local heat transfer coefficient associated with the thermodynamic quality in Equation 3 is equal to:

$$\alpha_{cond,i} = \frac{q''}{(T_{int,i} - T_{wall,inner,i})} \tag{5}$$

where q'', T_{int}, T_{wall} are the heat flux normal to tube surface, liquid-vapor interface temperature and inner tube wall temperature. The inner wall temperature is evaluated by using a thermal resistance analogy and the measured outer wall surface temperature (Equation 6). The liquid-vapor interface temperature can be approximated as the equilibrium saturation temperature. For low temperature glide mixtures, the interface temperature and equilibrium saturation temperature are approximately the same [15, 17].

$$T_{wall,inner,i} = T_{wall,outer,i} + \frac{\dot{Q}_i ln \frac{D_{inner}}{D_{outer}}}{2\pi l_i k_c} \tag{6}$$

where D_{inner} , D_{outer} , k, l are the inner and outer diameter, thermal conductivity of copper and length of the segment. For heat flux and tube inner temperature calculations, the area occupied by the water mixers was assumed to be adiabatic. This is because the thermal conductivity of polylactic acid is very low (0.13 W m⁻¹ K⁻¹ , ASTM C518) compared to copper and, therefore, the heat transferred through the mixers is negligible compared to the total heat duty transferred.

4.2. Two-phase frictional pressure drop

A Rosemount 3051S MultiVariable pressure transducer is used to measure the absolute inlet pressure and the differential pressure across the test section. Equation 7 shows that the pressure drop measured by the sensor is the sum of the pressure drop due to two-phase frictional flow (ΔP_{fric}) and single-phase gas ($\Delta P_{v,tube}$) and liquid flow ($\Delta P_{v,tube}$). Additionally, there is a pressure gain due to momentum change of the flow from vapor to liquid. The details of the calculation procedure used to obtain the two-phase frictional pressure drop from the measured value are provided in Jacob et al. [14], and are therefore only summarized here.

$$\Delta P_{fric} = \Delta P_{measured} - \Delta P_{v,tube} - \Delta P_{f,tube} + \Delta P_{deceleration} \tag{7}$$

The pressure drop in the test section was modeled by dividing it into three different regions based on the flow occurring in them; single-phase vapor flow, two-phase flow and single-phase liquid flow (See Jacob et al. [14]). For each experiment, the bulk enthalpy and equilibrium mixture quality of the flow are calculated based on the approach described in the previous section (Equations 1 - 4). The location of the saturated condensation region along the length of the tube is then determined, which exists between the points where the bulk enthalpy equals the saturated vapor enthalpy (x = 1) and saturated liquid enthalpy (x = 0). The length from the start of the test section and the point at which thermodynamic quality, x = 1 occurred was the single-phase vapor flow region. Similarly, the length from the point x = 0 to the end of the test section was single-phase liquid flow region. The pressure drop due to single-phase liquid and vapor flow were evaluated using the Churchill [27] friction factor correlation. A conservative $\pm 25\%$ uncertainty was assigned to the evaluated single-phase friction factors. The pressure gain as a result of flow deceleration was evaluated by carrying out a momentum balance from the inlet to the outlet (Equation 8). The two-phase frictional pressure drop was then calculated using Equation 7.

$$\Delta P_{deceleration} = \rho_v V_v^2 - \rho_l V_l^2 \tag{8}$$

4.3. Uncertainty analysis

Table 2 lists the ranges and the values of systematic uncertainty, U_{sys} associated with the sensors used in this study. As shown in Equation 9, the total uncertainty in each of the measured variables is the root of the sum of squares of the systematic uncertainty and random uncertainty, U_{rand} .

$$U_{tot} = \sqrt{U_{sys}^2 + U_{rand}^2} \tag{9}$$

Systematic uncertainty of a sensor is dictated by the manufacturing process, physical traits and type of the sensor, repeatability, data acquisition capabilities, calibration process etc. Random uncertainties occur due to pressure and flow fluctuations in the measured values even when the system is at steady state. Random uncertainties were evaluated statistically by assuming a normal distribution.

$$U_{rand} = \frac{SD \ t_{95}}{\sqrt{N}} \tag{10}$$

where the SD, t_{95}, N are standard deviation in the data for each sensor collected over 4 minutes, t-value corresponding to the 95% confidence interval and number of total readings, respectively.

Uncertainty propagation in the calculated variables is evaluated using the procedure suggested by Kline and McClintock [28]. The uncertainty in the evaluated mass fluxes was due to the uncertainty in the refrigerant mass flow rate and tube diameter (± 0.01 mm). The average uncertainty in the evaluated mass fluxes was on average $\pm 0.4\%$. Similarly, the uncertainty in the calculated thermodynamic quality was low (on average equal to $\pm 1.7\%$).

For heat transfer calculations, the pressure in the test section was assumed to be equal to the absolute pressure measured at the test section inlet. The uncertainty due to this assumption was quantified by evaluating the root mean square of the systematic uncertainty of the pressure sensor (5.2 kPa) and the actual measured pressure drop for that data point. The resulting value was then used as the new total systematic uncertainty for that data point. In general, the contribution of pressure uncertainty to total uncertainty in heat transfer coefficients was low (on average 0.5%). A major source of uncertainty in the quality and heat transfer coefficients was from the measured water temperatures, which were used to evaluate the heat duty in each segment. The average uncertainties in the heat transfer coefficients for R404A, R448A and R452A were \pm 9.0%, \pm 7.6% and \pm 9.5%, respectively. The maximum uncertainties were \pm 21%, \pm 23% and \pm 26%, respectively. The highest measurement uncertainties were associated with lower thermodynamic qualities and low refrigerant mass fluxes. At these conditions, the heat duty in a given segment was comparatively lower, which resulted in a lower temperature change of the coolant and higher corresponding uncertainty.

	Table 2	Instrument	ranges	and	uncer	tain	ties
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Measurement	Range	Systematic Uncertainty
RTD (water temperature)	$-200 - 500 ^{\circ}\mathrm{C}$	$\pm 0.05^{\circ}\mathrm{C}$
Surface temperature	$-200 - 500 ^{\circ}\mathrm{C}$	$\pm 0.5 ^{\circ}\text{C}$
Type T thermocouple	$-200 - 500 ^{\circ}\mathrm{C}$	$\pm 0.5 ^{\circ}\text{C}$
Refrigerant mass flow meter	$0-30{ m g~s^{-1}}$	$\pm 0.1\%$
Water volumetric flow meter	$0.04 - 7.5 \mathrm{L} \mathrm{min}^{-1}$	$\pm \ 0.5\%$
Pressure	$0-10\mathrm{MPa}$	$\pm 5.68 \mathrm{kPa}$
Differential pressure	$0-62\mathrm{kPa}$	$\pm 0.05 \mathrm{kPa}$

5. Results and discussion

Table 3 shows the test matrix for the current study. Condensation data for R404A, R448A and R452A were collected for three different saturation conditions $(T_{sat,avg} = 40, 50 \text{ and } 60 \text{ }^{\circ}\text{C})$. For higher saturation conditions $(T_{sat,avg} = 50 \text{ and } 60 \text{ }^{\circ}\text{C})$, mass fluxes ranged from 100 kg m⁻² s⁻¹ to 800 kg m⁻² s⁻¹. For $T_{sat,avg} = 40$, data collection was limited to mass fluxes up to 500 kg m⁻² s⁻¹, due to the cooling capabilities of the chiller in the experimental loop. The average Δx in a segment was equal to 0.2. The maximum was 0.61 which corresponded to low refrigerant mass flux tests. The minimum was 0.07.

A constant heat flux was not applied in this study. Rather, a secondary coolant condensed the refrigerant, representative of a real system. Thus, the local flux (approximately 20 to 150 kW m-2) was a function of the refrigerant and coolant heat transfer coefficients, and the driving temperature difference. Recent work by Azzolin et al. [22] on in-tube condensation of refrigerants R455A and R452B showed a negligible dependence of measured condensation heat transfer coefficient on heat flux in the shear-dominated annular region. In their investigation on the effect of temperature difference between the refrigerant saturation temperature and the tube wall

 (ΔT) , they found that the influence of ΔT (and consequently, the heat flux) on heat transfer coefficients was most pronounced for gravity dominated regimes (associated with relatively low qualities and mass fluxes) and low ΔTs (<15°C). In the current study, 71.4% of the data points were shear dominated flows, as predicted by the Cavallini et al. [1] criterion. The water inlet temperature were set so that the driving temperature differences between the wall and refrigerant saturation were >30°C. Thus, the values and trends presented in Figure 4 are expected to be independent of the applied heat flux. More importantly, it was critical that the condensation resistance was always the limiting resistance. In our study, the refrigerant-side convective resistance was on average 4 times greater than the coolant convective resistance.

The test section was well insulated and heat loss to the ambient were assumed to be negligible during data analysis. On average, energy lost by the refrigerant was 3.2% (approximately 39 W) higher than the energy gained by coolant for all experiments. The highest and lowest heat loss measured in this study were 7.2% and the 0.26%, respectively. The largest heat loss occurred at the test section inlet where the refrigerant was in the superheated state by approximately 40 °C for each data point. To facilitate reproduction of this work, all measured and calculated variables for each refrigerant, as well as sample calculations for a representative case are publicly available in Jacob et al. [29].

Average saturation		Mass flux (kg m ^{-2} s ^{-1})						
temperature (°C)	100	200	300	400	500	600	700	800
40	Х	х	х	Х	х			
50	х	х	х	х	х	х	х	х
60	х	х	х	х	х	х	х	х

Table 3 Experimental test matrix

 $\mathbf{x} = \mathbf{data}$ collected

5.1. Frictional pressure drop

Due to the limited sensitivity and accuracy of the transducer at low differential pressures, the experiments with measured pressure drop values lower than 0.5 kPa are not reported in this study. Frictional pressure drops were on average 91% of the measured pressure drop values after accounting for the minor losses and acceleration gain due to momentum change of the flow. The average uncertainty in the measured frictional pressure drop values was 5.7% (≈ 0.23 kPa).

The resulting two-phase frictional pressure drop data were compared with the correlations of Friedel [5], Haraguchi et al. [4], Andresen et al. [30] and Cavallini

et al. [24]. The Friedel [5] correlation was developed based on adiabatic frictional pressure drop data of a wide range of fluids (water, refrigerants, air, oil, etc.) and tubes (12 mm $\langle D_h \langle 49 \text{ mm} \rangle$). The Haraguchi et al. [4] ($D_h = 8.4 \text{ mm}$) and Cavallini et al. [24] ($D_h = 0.5$ to 3.2 mm) correlations were developed to predict pressure drop during condensation of pure HFC refrigerants. Andresen et al. [30] proposed a twophase multiplier correlation for predicting pressure drop during condensation of HFC refrigerant mixtures at near critical pressures ($P_R > 0.8$).

To compare our data to pressure drop correlations, a second order polynomial fit with respect to length was performed on the qualities obtained from the energy balance calculations described in Section 4.1. This provided a function for thermodynamic quality with respect to length. The two-phase pressure drop correlations provide a relationship for dP/dL in term of thermodynamic quality, x. Therefore, the following integration was performed in the two-phase region to obtain the predicted pressure drop:

$$\Delta P_{predicted} = \int_{L_{x=o}}^{L_{x=1}} \frac{dP[x(L)]}{dL} dL \tag{11}$$

The calculation procedure used to evaluate the total two-phase frictional pressure drop from these correlations during complete condensation is detailed in Jacob et al. [14].

	R448A		$\mathbf{R452A}$		R404A		Overall	
Model	MAPE(%)	MPE(%)	MAPE(%)	MPE(%)	MAPE(%)	MPE(%)	MAPE(%)	MPE(%)
Friedel [5]	23	-16	14	-12	15	-6	18	-12
Haraguchi et al. [4]	15	11	18	17	34	34	20	18
Andresen et al. [30]	40	33	20	-16	52	52	35	18
Cavallini et al. [24]	29	-24	24	-24	19	-19	25	-23

Table 4 Comparison of the measured two-phase frictional pressure drop and the predicted pressured drop

The results are presented in Table 4 and Figure 3. The results from this study were compared against the predicted values from the literature by evaluating the Mean Percentage Error (MPE, Eq. 12)) and the Mean Absolute Percentage Error (MAPE, Eq. 13)). For R448A, the Haraguchi et al. [4] correlation predicted the experimental data best, with a MAPE value equal to 14.8 %.

For R452A, both the Haraguchi et al. [4] and the Friedel [5] correlations showed good agreement with the data, with MAPE equal to 17.8% and 14.2%, respectively.

Figure 3 shows that the Haraguchi et al. [4] correlation slightly over predicts the data (MPE = 13.9%), while the Friedel [5] correlation under predicts the data (MPE = -13.8%). The Cavallini et al. [24] correlation significantly under predicts the data for both the refrigerants. The Andresen et al. [30] correlation can be used to predict R452A pressure drop but not the pressure drop for R448A. Finally, the R404a data were best predicted the Friedel [5] correlation.

$$MPE = \frac{100\%}{n} \sum_{n=1}^{n} \frac{P_{predicted} - P_{measured}}{P_{measured}}$$
(12)

$$MAPE = \frac{100\%}{n} \sum_{n=1}^{n} \frac{|P_{predicted} - P_{measured}|}{P_{measured}}$$
(13)



Figure 3: Predicted versus measured two-phase frictional pressure drop with different correlations



Figure 4: Condensation heat transfer coefficient versus thermodynamic quality for R448A and R452A at mass fluxes from 100 - 800 kg m⁻² s⁻¹ and average saturation temperatures of 40, 50 and 60 °C

5.2. Heat transfer coefficients

Figure 4 shows the measured heat transfer coefficients of R448a and 452A versus the thermodynamic quality for different mass fluxes and saturation temperatures. The heat transfer coefficients increase with increasing quality and mass fluxes, and with decreasing saturation temperature. The experimental data were compared with several correlations from the literature. The highly cited empirical Shah [31] correlation was developed by analyzing condensation data in horizontal tubes for a wide range of operating conditions and fluids. Later, Shah [32] presented a revised version of the correlation which was validated against a database containing condensation data for 33 fluids in tubes with diameters ranging 0.1 to 49 mm, at reduced pressures ranging from 0.0008 to 0.946 and mass fluxes from 1.1 to 1400 kg m⁻² s⁻¹.

The semi-empirical Dobson and Chato [33] correlation was developed by considering the relative contribution of the shear and gravity effects during condensation and modeling the heat transfer mechanisms in each regime. The Thome et al. [34] multi-regime condensation model uses the flow regime map of El Hajal et al. [35] to identify the dominant heat transfer mechanisms occurring for a given flow. For annular, intermittent and mist flows, the heat transfer primarily occurs through the thin liquid film that develops between the vapor core and inner surface of the tube. For stratified and wavy-stratified flows, the heat transfer is a combination of the falling film heat transfer, α_f at the upper surface of the tube and the convection heat transfer coefficient, α_c through the liquid pool. The depth of the liquid pool as well as the areas associated with each mode of heat transfer can be determined by evaluating θ , the upper angle of the tube not wetted by the stratified liquid. The resulting two-phase heat transfer coefficient can be modeled as:

$$\alpha_{cond,thome} = \left[\frac{\theta\alpha_f + (2\pi - \theta)\alpha_c}{2\pi}\right] \tag{14}$$

Finally, Cavallini et al. [1] proposed a model that accounts for different flow regimes by categorizing them into either ΔT -dependent or ΔT -independent flows. ΔT is the temperature difference between the wall and the fluid saturation temperature. ΔT -independent flows include annular and intermittent flows where the flow structure is shear dominated and heat transfer is primarily due to convection. For ΔT -independent flows such as stratified and stratified-wavy flows, the role of gravitational force is more significant.

The correlations above were developed primarily for pure fluids. Therefore, the methodology developed by Silver [2], Bell and Ghaly [3] (SBG) may be used to obtain an estimate of the deterioration in the heat transfer due to mixture effects. The SBG approach assumes an equilibrium between liquid and vapor phases. The

additional resistance due to mass transfer effects is not explicitly considered but rather correlated to the sensible resistance in the vapor phase (Equation 15).

$$\alpha_{mix} = \left[\frac{1}{\alpha_{cond}} + R_{s,v}\right]^{-1} = \left[\frac{1}{\alpha_{cond}} + \frac{Q_{s,v}}{\dot{Q}_T}\frac{1}{\alpha_v}\right]^{-1}$$
(15)

where α_{cond} , $R_{s,v}$ are the condensate film resistance evaluated using correlations for pure fluids with mixture thermodynamic properties and the sensible resistance of the vapor, respectively. $\frac{\dot{Q}_{s,v}}{\dot{Q}_T}$ is the ratio between the sensible heat removed from the vapor and the total heat removed.

$$\frac{\dot{Q}_{s,v}}{\dot{Q}_T} \approx \frac{xc_{p,v}(T_{dew} - T_{bubble})}{h_{fv}} \tag{16}$$

where h_{lv} is the latent heat of vaporization. The vapor phase heat transfer coefficient, α_v , is evaluated using the Dittus and Boelter [36] equation, without considering enhancement due to two-phase effects. Silver [2], Bell and Ghaly [3] argued that this would result in a conservative estimate of the heat transfer which is desirable since mass transfer effects are neglected.

Past researchers have adopted and modified the SBG method to improve the predictions for various fluid mixtures and to account for heat transfer in various flow regimes. Notably, Del Col et al. [37] modified the model of Thome et al. [34] to predict heat transfer coefficient of zeotropic mixtures. Del Col et al.'s model applied the additional SBG resistance to both the convective and the film heat transfer coefficients in Equation 14. They also introduced a non-equilibrium friction factor to the falling film heat transfer to account for non-equilibrium effects in the stratified flow regimes.

The results from the comparison between the measured data and the correlations are summarized in Table 5 and visually illustrated in Figure 5. The term β represents the percentage of experimental data that were within $\pm 20\%$ of the predicted value. The measured data for all three refrigerants is best predicted by the Cavallini et al. [1] correlation with the SBG correction, with a MAPE value equal to 10% and β = 93%. The application of the SBG approach to the Cavallini et al. [1] correlation resulted in 9.9 and 7.8% reduction in heat transfer coefficient on average for R448A and R452A, respectively. Additionally, Figure 6 shows the differences in resulting MAPEs with and without the application of the SBG correction for the R448A and R452A data. In particular, for the Cavallini et al. [1] correlation, the application of the SBG correction resulted in the reduction of MAPE from 15.6% to 9.7%. The effect of the SBG correction was even more significant for the Shah [32] correlation, as the MAPE decreased from 27.4% to 15.3%. Conversely, the application of SBG correction to Thome et al. [34] correlation resulted in minimal differences on MAPE. This is because the magnitude of the heat transfer coefficients from Thome et al. [34] correlation were relatively smaller. It only slightly over predicts the mixture data (MPE = 5.6%) compared to Cavallini et al. [1] (MPE = 14.8%) and Shah [32] (MPE = 23.3%) correlations. Since R404A is a near azeotropic refrigerant, mixture effects were minimal and the application of SBG correction did not significantly improve the predictive capabilities of correlations.

The correlations by Thome et al. [34], Del Col et al. [37] and Shah [32] also predicted the data quite accurately, with MAPE values below 15%. The modification of the earlier correlation [31] to the Shah [32] correlation improved the predictability significantly from 36% MAPE value to 14%. Overall, the analysis conducted confirms that the existing correlations with the SBG correction are adequately able to predict the heat transfer coefficient of HFC/HFO refrigerant mixtures R448A and R452A for the range of conditions investigated here.

Table 5 Predictive capability of different condensation heat transfer correlations

	R448A		R452A		R404A		Overall	
Model	MAPE(%)	MPE(%)	MAPE(%)	MPE(%)	MAPE(%)	MPE(%)	MAPE(%)	MPE(%)
Shah [31] with SBG	34	32	38	36	41	40	37	35
Dobson and Chato [33] with SBG	23	18	25	17	30	15	25	17
Thome et al. [34] with SBG	11	0	13	-1	14	-1	12	0
Del Col et al. [37]	11	1	12	0	24	-13	12	1
Cavallini et al. [1] with SBG	10	6	10	5	9	4	10	5
Shah [32] with SBG	15	15	13	10	14	9	14	12



Figure 5: Predicted versus measured condensation heat transfer coefficients of R404A, R448A and R452A with different correlations



Figure 6: Predictive capability of different condensation correlations against experimental R448A and R452A data with and without Silver [2], Bell and Ghaly [3] (SBG) correction

6. Refrigerant comparison

6.1. Heat transfer coefficients

Since both R448A and R452A are proposed replacements for R404A, it is insightful to compare their performance with each other and to R404A. Figure 7 provides a comparison of the experimental heat transfer coefficients along with the predictions from Cavallini et al. [1] correlation for each refrigerant at two different mass fluxes (G = 200 and 600 kg m⁻² s⁻¹). The measured data are well predicted by the Cavallini et al. [1] correlation at the lower mass fluxes. However, the Cavallini et al. [1] correlation with the SBG correction slightly over predicts the data at higher mass fluxes for both R448A and R452A. This is consistent with the overall MPE value of 5.6% obtained after comparing all the experimental data of R448A and R452A. A positive MPE value indicates that the data is generally over predicted by the correlation.

The experimental data shows that the heat transfer performance of R448A and R452A is comparable to that of R404A; with the heat transfer coefficients of R404A being slightly higher than those of R452A and slightly lower than those of R448A. Among these fluids, R448A has the highest liquid phase thermal conductivity and

R404A has the lowest at similar operating conditions (See Table 1). Even though the liquid phase thermal conductivity of R448A and R452A is higher than R404A, the heat transfer coefficients are very similar, which may be attributed to the degradation due to mixture effects.



Figure 7: Condensation heat transfer coefficients of R448A and R452A at two different mass fluxes ($T_{sat,avg} = 60$ °C). The experimental data is compared with predictions from the Cavallini et al. [1] correlation for R404A, R448A and R452A



Figure 8: Predicted two-phase frictional pressure drop of R404A, R448A and R452A using Haraguchi et al. [4] correlation at two different mass fluxes (G = 600 kg m⁻² s⁻¹, $T_{sat,avg} = 60$ °C)

6.2. Pressure Drop

It is not possible to compare the experimental pressure drops from this study directly due to the differences in lengths associated with the condensation region (Eq. 7) during experiments. Therefore, the correlations that best predicted the data for each fluid were used to compare the two-phase frictional pressure gradients associated with condensation of each refrigerant. Figure 8 shows the frictional pressure gradient of R404A, R448A and R452A at two different mass fluxes (G = 200 & 600 kg m⁻² s⁻¹). R448A had the highest pressure drop, and R404A the lowest. For G = 600 kg m⁻² s⁻¹, the pressure drop of R448A is on average 81% higher than that of R404A. The pressure drop of R452A is on average 3.3% higher than that of R404A.

The transport and thermodynamic properties of these refrigerants can help explain the observed differences. R448A and R452A have a higher liquid phase viscosity compared to R404A (Table 1). During condensation, higher liquid phase viscosity results in a greater amount of shear between the liquid and the wall, as well as between the liquid and the vapor phases. Additionally, R448A and R452A have higher liquid phase densities and lower vapor phase densities than R404A. During the condensation process, this results in higher velocity vapor flows and lower velocity liquid flows compared to R404A at the same mass flow rate. The increased difference in velocities of the phases will result in a greater magnitude of interfacial shear between the phases. The combination of both of these factors result in higher pressure drops of R452A and R448A when compared to R404A at equal mass flows.

6.3. Comparison at equivalent cooling capacity

Table 6 Refrigerant flow rates and corresponding mass fluxes required to produce 1 kW of cooling at 0 $^{\circ}$ C evaporating temperature and the associated latent heats of vaporization from REFPROP[19]

Defigenent	Latent Heat of Vaporization	Mass Flow Rate	Mass flux in a tube with
Keirigeram	at 0 °C (kJ kg ⁻¹)	$(g \ s^{-1})$	$ID = 4.7 \text{ mm} (\text{kg m}^{-2} \text{ s}^{-1})$
R404A	166.3	6.0	343
R448A	206.1	4.9	277
R452A	164.5	6.1	347

The latent heat of vaporization for R448A is significantly higher than that of R452A and R404A. In practice, this would result in lower R448A mass flow rates compared to R404A in a vapor compression system utilizing these fluids. Table 6 shows the mass flow rates of each refrigerant required to produce 1 kW of cooling assuming an evaporating temperature equal to 0 °C. The resulting mass flow rate of R404A was approximately 24% higher than R448A.

The mass flow rates in Table 6 were then used to calculate the heat transfer coefficients and pressure drop gradients in a single condenser tube (with tube ID = 4.7 mm) operating at $T_{sat,avg} = 40$ °C, using the correlations that were found to predict the data best. Figure 9 shows the results. Heat transfer coefficients of all three refrigerants are approximately equal. The pressure drop of R404A and R452A is approximately equal, and the pressure drop of R408A is on average 16% lower than that of R404A.

Similar analysis was repeated to obtain the corresponding R448A and R452A mass fluxes required to produce equivalent cooling corresponding to R404A flowing at $G = 200, 400, 600, 800 \text{ kg m}^{-2} \text{ s}^{-1}$ in a single condenser tube with ID = 4.7 mm. The results are presented in Table 7. At mass fluxes lower than 600 kg m⁻² s⁻¹, the average R448A pressure drop is lower than that of R404A. The opposite trend is observed at higher R404A mass fluxes.



Figure 9: Heat transfer coefficients and pressure drop gradients for R404A, R448A and R452A at mass flow rates listed in Table 6 to produce 1 kW of cooling system in a condenser with ID = 4.7 mm and saturation temperature of 40 $^{\circ}C$

Therefore, even though R448A exhibits higher pressure drop when compared to R404A at the same mass flux, its may actually have a lower pressure drop when compared on an equal cooling capacity. The pressure drop of R452A is consistently similar to that of pressure drop of R404A. Overall, this analysis confirms that both R448A and R452A may be considered a viable replacements for R404A.

R404A mass flux	R448A mass flux	R448A pressure drop	R404A mass flux	R452A pressure drop
$({\rm kg} {\rm m}^{-2} {\rm s}^{-1})$	$({\rm kg} {\rm m}^{-2} {\rm s}^{-1})$	(% difference)	$({\rm kg}~{\rm m}^{-2}~{\rm s}^{-1})$	(% difference)
200	161	-34	202	1
400	323	-9	405	1
600	484	11	607	1
800	646	35	809	1

Table 7 Percent difference in frictional pressure gradient between low GWP zeotropic mixtures and R404A at different R404A mass flux for equivalent cooling capacity in a single 4.7 mm tube

7. Conclusion

An experimental investigation of condensation heat transfer coefficients and pressure drop values of low GWP refrigerants R448A and R452A and their predecessor R404A was conducted. Experiments were performed in a 4.7 mm tube at three different saturation conditions ($T_{sat,avg} = 40, 50, 60$ °C) and at mass fluxes ranging from 100 kg m⁻² s⁻¹ to 800 kg m⁻² s⁻¹. Based on the analysis conducted, the following conclusions can be drawn:

- The heat transfer coefficients of R448A are higher than those of R404A and the heat transfer coefficients of R452A are slightly lower than those of R404A at the same mass fluxes.
- Both R448A and R452A are zeotropic refrigerant mixtures. There are additional mass transfer resistances associated with their condensation process which result in degradation in heat transfer compared to their pure components. Therefore, predictions from correlations developed for pure fluids are corrected by adopting the equilibrium Silver [2], Bell and Ghaly [3] approach.
- Heat transfer coefficients of R404A, R448A and R452A are best predicted by Cavallini et al. [1] correlation with an overall MAPE = 10%.
- Two-phase frictional pressure drop of R448A is best predicted by the Haraguchi et al. [4] correlation, with a MAPE = 15%. Pressure drop of R404A (MAPE = 15%) and R452A (MAPE = 14%) are best predicted by the Friedel [5] correlation.
- At the same mass flux (G = 600 kg m⁻² s⁻¹), R448A and R452A pressure drops are predicted on average 81% and 3.3% higher than those of R404A.

• The latent heat of vaporization of R448A is approximately 24% higher than that of R452A and R404A at 0 °C. Therefore, a refrigeration system using these refrigerants will operate at different refrigerant flow rates. For a refrigeration system with 1 kW cooling capacity and a condenser with a single 4.7 mm tube, the heat transfer performance of all three refrigerants is similar. For the same case, the frictional pressure drop gradients of R404A and R452A are approximately equal, while the pressure gradient of R448A is on average 16% lower than that of R404A.

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