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ABSTRACT

Using common refrigerants such as R410A in air conditioning systems leads to climate change due to their high global warming potential (GWP). Therefore, restrictive laws have been enacted to force manufacturers to use refrigerants with lower GWP than that of R410A. This simulation study explores the effects of substituting R410A with R468C on the performance of a fin-and-tube evaporator coil. A validated segment-by-segment heat exchanger model is used to find the optimum performance of using R468C as a working fluid by changing the heat exchanger circuitries. The results show that for identical superheat temperature (SHT) and saturated suction temperature (SST), R468C required a higher refrigerant flow rate, resulting in an increased cooling capacity (between 30-57%) and a lower coefficient of performance (about 10%) than R410A.

1. INTRODUCTION

In general, two significant harmful effects of using the residential and light commercial air conditioning and heat pump systems on the environment are the emission of greenhouse gases from burning fossil fuels to generate the needed electricity and the leakage of refrigerant. Unfortunately, conventional refrigerants such as R410A, which are widely used in the aforementioned systems, have a high global warming potential (GWP) and need to be replaced with less harmful, lower GWP refrigerants (UNEP (2016), Schulz & Kourkoulas (2014), etc.). The change to lower GWP refrigerants is a challenge for manufacturers, requiring substantial component redesign.

The literature about the effects of substituting traditional refrigerants with low GWP alternatives can be categorized into two distinct approaches, including pure heat exchanger performance analysis and system-level study through experimental measurements and numerical simulations.

Fin and tube heat exchangers (FTHXs) are primarily employed in the residential and light commercial air conditioning and heat pump systems. Several geometrical parameters, for example, circuitry, tube diameter and thickness, fin thickness and density, can be adjusted to enhance their heat transfer efficiency (e.g. Saleem et al. (2022), Chu et al. (2020), and Okbaz et al. (2020)). Reasor et al. (2010) evaluated the thermophysical properties of R1234yf as a drop-in replacement to R410A and R134a and presented the results of simulations using three refrigerants in microchannel and FTHX. In their study, they first compared the thermophysical properties of the refrigerants with each other. They then conducted simulations to determine the differences between the results obtained by using each refrigerant in an identical system. They found that although the thermophysical properties of R1234yf are very identical to R134a, they are not similar to R410A. Their simulation results showed similarities in outlet refrigerant temperature and heat load for R1234yf and R134a. However, around 40% variation was observed in the pressure drop, suggesting that the heat exchanger design can be further optimized.

Chu et al. (2020) focused on the experimental and numerical study of sinusoidal wavy FTHXs' airside performance with round and oval tube configurations. In terms of pressure drop, their results showed that the round configuration FTHXs had higher values with the fin pitch (F_p) of 3 mm, and the oval configuration conversely had higher values when the F_p was 1.8 mm. Moreover, by increasing the fin pitch from 1.8mm to 3.0mm, the heat transfer coefficient of FTHXs with round and oval configurations can be improved by about 10–20% and 9.3–25.2%, respectively, as the tube row in the heat exchanger increases to 6. Saleem et al. (2022) investigated how R1234ze(E) as a low-GWP fluid affects the performance of a fin-and-tube evaporator coil originally designed for R410A. They also presented an analysis of the effects of fin density and refrigerant circuitry on cooling capacity and refrigerant side pressure drop of an FTHX using R1234ze(E). They concluded that R1234ze(E) required a 26% lower flow rate than R410A for identical superheat and refrigerant SST, resulting in a 34% lower coil capacity, as well as a 15 times higher refrigerant pressure drop. Moreover, an increase in fin density of 33.3% increased the capacity by 5.1% and reduced the pressure drop by 4.5%.

Several studies investigated the drop-in performances of alternative refrigerants in a vapor compression systems. Kim et al. (2020) developed an air-to-air heat pump simulation model with low GWP refrigerants over the R410A heat pumps by optimizing the heat exchanger designs. Their results showed that the average performance of the R32 heat pump with optimized heat exchanger design was 6% which was more than DR5 and L41a with values of 3.4% and 2.4%, respectively. Li et al. (2021) presented a multi-objective optimization approach to optimize low GWP refrigerant mixture compositions and heat exchanger circuitries. They used the DOE/ORNL heat pump design model for the optimizations. Their results reached a 5.9% improvement in cycle efficiency and a 48.6% reduction in flammability with a refrigerant GWP of 268. A screening study was performed by Yu et al. (2021) to evaluate mixtures containing a maximum of five components with low GWP and flammability in air conditioning and heat pump systems that would sustain the energy efficiency and capacity of R410A for both heating and cooling applications. Although the screening results met the established criteria, the authors concluded that more experimental research was needed to confirm their findings.

In heat exchanger level analysis and design, parameters such as cooling capacity the pressure drop may be analyzed and optimized. In system-level performance evaluation, other metrics, for example, the coefficient of performance (COP) and energy efficiency ratio (EER), can be investigated. Research studies on the combination of both levels, especially by considering low GWP refrigerants, are rare in the reviewed literature. Therefore, in this paper, a numerical analysis is performed to investigate the effects of using R468C with a GWP of 284 (about 7.4 lower than R410A) not only in the heat exchanger but also in the system level analysis. In this study, R468C as a low GWP refrigerant is simulated in a fin and tube heat exchanger originally designed for R410A. Moreover, further studies have also been performed to investigate the effect of different circuitries on the performance of the fin and tube heat exchanger using R468C.

2. METHODOLOGY

2.1 Heat Exchanger Model and Validation

A validated segment-by-segment heat exchanger model called xFin (Sarfraz et al., 2019) was used in this study. The xFin model was based on finite difference modelling which divide the heat exchanger into N elements in the direction of refrigerant flow and apply a lumped analysis using the ϵ-NTU method, to determine the state of each elements and calculate refrigerant and air side heat transfer and pressure drop. Moreover, the xFin model calculates the refrigerant and moist air phase transitions.

The xFin model can be used to conduct parametric studies by modifying heat exchanger circuitry and geometry as was previously demonstrated by Saleem et al. (2022). One of the purposes of this study is that the xFin model was used to see how the heat exchanger performance can be affected by changing in the coil circuitry at the same inlet refrigerant and airside conditions. The xFin code has been validated by previous works by Sarfraz et al. (2019, 2020) and Saleem et al. (2021, 2022) and Table 1 shows the summary of the previous validation efforts.

Table 1: Validation of xFin model

Fluid	Reference	Number of evaporators tested	Range of capacities tested	MAPE between experimental and model-predicted capacities	Number of experiments
Water	Sarfraz et al. (2019)	1	15.0 – 20.0 kW	2.1%	3
R410A	Sarfraz et al. (2020)	1	6.3 – 7.8 kW	0.7%	3
R410A	Saleem et al. (2021)	3	2.8 – 18.0 kW	1.0%, 2.4%, and 0.9%	66
R1234ze	Saleem et al. (2022)	1	2.4 – 4.25 kW	1.4%	24

2.2 Computational Test Matrix and Simulation

Since the evaporator cooling capacity is the essential outcome, the values of the three parameters, including saturated suction temperature (SST), superheat temperature (ΔT_{SH}), and air inlet velocity ($V_{a,i}$), were fixed in this investigation. These values are shown in Table 2 as a computational test plan for R410A and R468C. All of the simulations in this study used a dry coil conditions, and the corresponding air inlet dry-bulb (DB) and wet-bulb (WB) temperatures are shown in Table 3. Since R468C is a zoetrope refrigerant, the values of SST in Table 2 for this refrigerant are the mean value between the evaporator inlet temperature and the saturated vapor temperature.

The value of SHT is fixed at 11.1 K (Table 2) as a means for comparing the performance of R468C with R410A. Therefore, an iteration loop was written to iterate on the refrigerant mass flow rate to find its value that satisfies 11.1 K while keeping refrigerant SST and inlet pressure constant.

The information in Tables 2-4 served as inputs to the xFin model to calculate coil capacity and pressure drop. Moreover, the correlations of air and refrigerant side that were used in the model and some properties of R410A and R468C were calculated at 10℃ using REFPROP 10 (Lemmon et al., 2018), are shown in Table 5 and 6, respectively. Additional information about the used refrigerants, such as composition, normal boiling point, etc., was calculated by REFPROP 10 and listed in Table 7. From Tables 6 and 7, it can be seen that not only thermodynamic but also transport properties of R468C are close to R410A ones. Therefore, R468C can be considered a refrigerant similar to R410A, except its GWP is 285, which is noticeably lower (about 7.4) than R410A.

Table 2: Computational test matrix

Refrigerant	Heat exchanger surface condition	Saturated suction temperature (SST)	Air inlet velocity ($V_{a,i}$)	Number of different heat exchangers	Test superheat	Number of tests
R410A (Baseline)	Dry	5℃ (41°F) 10℃ (50°F) 15℃ (59°F)	1 m/s (≈200 fpm)	HX1 HX2 HX3 HX4	11.1 K	12
R468C						12
					Total number of simulations:	24

Table 3: Air inlet temperature condition for simulations

Mode	Dry bulb temperature	Wet bulb temperature	Dew point temperature
Evaporator dry	26.67℃ (80°F)	14.44℃ (58°F)	4.44℃ (40°F)

Table 4: Geometrical parameters of the simulated heat exchanger coil (HX1, Baseline)

Number of circuits	4
Number of rows	3
Number of tubes per row	16
Number of tubes per circuit	12
Tube type	Smooth
Tube material	Copper
Tube wall thickness	0.51×10^{-3} m
Tube outer diameter	9.53×10^{-3} m
Tube longitudinal spacing	2.19×10^{-2} m
Tube traverse spacing	2.54×10^{-2} m
Tube length	0.495 m
Fin type	Wavy fins
Fin thickness	1.14×10^{-4} m
Fin spacing	1.57×10^{-3} m
Half wavelength of fin wave	5.51×10^{-3} m
Wave amplitude	2.1×10^{-3} m

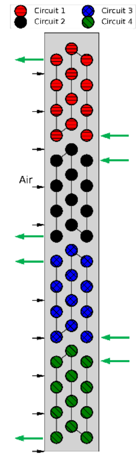

Figure 1: Schematic of the simulated heat exchanger coil with four circuitries (HX1, Baseline)

Table 5: Correlations on refrigerant and air side used in the model's simulation runs

Fluid (s)	**Correlation type**	**Model***
R410A & R468C (single-phase)	Heat transfer	Dittus Boelter equation
	Pressure drop	Blasius equation
R410A & R468C (two-phase)	Heat transfer	Shah
	Pressure drop	Lockhart and Martinelli
Air	Heat transfer & Pressure drop	Correlation for wavy fins

* Saleem et al. (2021, 2022) and Sarfraz et al. (2020) are references for the models used in this study.

Table 6: Key thermodynamic and transport properties of fluids at 10℃

Fluid	P_{sat} **(kPa)**	ρ_l **(kg/m^3)**	ρ_v **(kg/m^3)**	h_{fg} **(kJ/kg)**	μ_l **(μPas)**	μ_v **(μPas)**
R410A	1090.0	1128.2	41.2	213.2	142.7	12.7
R468C	1114.3	1046.2	38.6	215.9	147.3	11.9

Table 7: Some other important thermodynamic properties of using fluids

Fluid	**Composition**	**Normal boiling point**	**Critical temperature**	**Critical pressure (kPa)**	**Typical temperature glide (K)** $^{(2)}$
R410A	R32/R125 (50%/50%)$^{(1)}$	-51.4℃ (-60.5℉)	71.4℃ (160.5℉)	4901	0.1
R468C	R1132a/R32/R1234yf (6%/42%/52%)$^{(1)}$	-56.6℃ (-69.9℉)	77.0℃ (170.6℉)	4880	7 to 8

1: Mass percentages
2: The typical temperature glide was calculated for a temperature range from 5℃ to 15℃ (used in this study).

3. RESULTS

The results of this study are categorized into two groups. The first one shows the effects of substituting R468C with R410A on the performance of heat exchanger 1 (HX1, Fig. 1 and Table 4), which is originally designed for R410A. The second one consists of investigating alternate heat exchanger circuitries (HX 2-4) to see the effect of different circuitries on the performance of the heat exchanger while using R468C. Moreover, two sets of figures are used to show the results: one figure of cooling capacity vs. refrigerant mass flow rate and the other of cooling capacity vs. COP. To calculate the reported COP, the hypothetical compressor's isentropic efficiency was fixed at 75% for either of the refrigerants, e.g.

$$\eta_{is} = \frac{\dot{m}\left(h_{2,s} - h_1\right)}{\dot{W}_c} \tag{1}$$

where $\dot{m}$ is the refrigerant mass flow rate, $h_{2,s} - h_1$ is isentropic enthalpy change, and $\dot{W}_c$ is the compressor's power consumption.

3.1 The Effects of Substitution of R468C with R410A on HX1's Performance

Since HX1 has been initially designed for R410A, in this part, the investigation of the effect of using R468C as a low GWP refrigerant on HX1 performance is the main aim. Fig. 2 shows the refrigerant flow rate and cooling capacity variation for a ΔT_{SH} fixed at 11.1 K for three different SST. As can be seen, for 5℃ and 10℃ SST, the cooling capacity of R468C is higher than R410A. For 15℃ SST, the maximum achievable SHT was about 9 K for R468C (red box in Fig. 2). Due to this reason, the cooling capacity of R468C, only for this case, is lower than R410A with an SHT of 11 K. The variation of COP and cooling capacity for the same cases of Fig. 2 is shown in Fig. 3. It is also noticeable that the COP of a system with 75% of compressor isentropic efficiency while using R410A as working fluid is higher than the time R468C was used for all of the cases. The simulation results are summarized in Table 8.

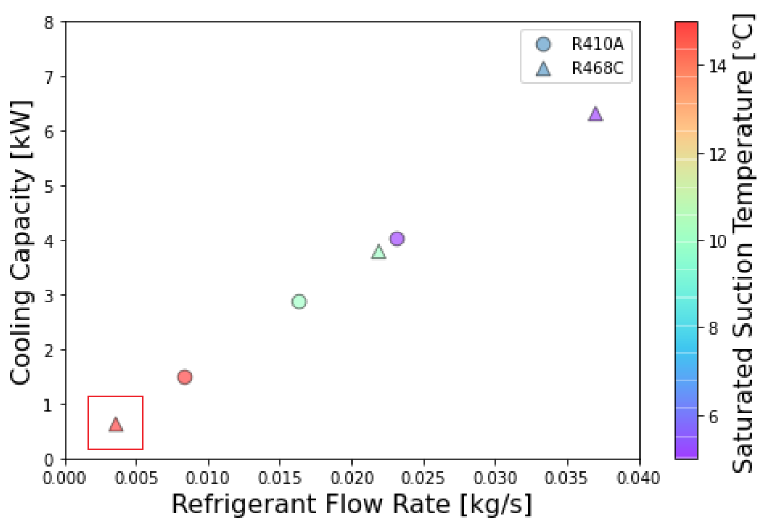

Figure 2: Variation of refrigerant flow rate and cooling capacity to achieve constant superheat at constant saturated suction temperature (5℃, 10℃, and 15℃) for HX 1

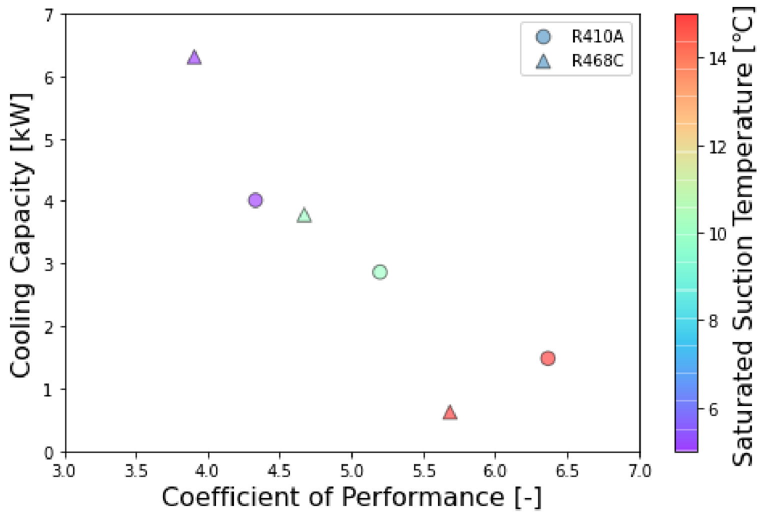

Figure 3: Variation of coefficient of performance and cooling capacity to achieve constant superheat at constant saturated suction temperature (5℃, 10℃, and 15℃) for HX 1

Table 8: The results obtained for HX1 (baseline) using R410A and R468C for three different SST

	SST = 5℃ (41°F)				**SST = 10℃ (50°F)**				**SST = 15℃ (59°F)**			
	ΔT_{SH}^{*} (K)	$\dot{Q}$ (kW)	ΔP (kPa)	$\dot{Q}/\Delta P$ (kW/kPa)	ΔT_{SH} (K)	$\dot{Q}$ (kW)	ΔP (kPa)	$\dot{Q}/\Delta P$ (kW/kPa)	ΔT_{SH} (K)	$\dot{Q}$ (kW)	ΔP (kPa)	$\dot{Q}/\Delta P$ (kW/kPa)
R410A	11.1	4.01	2.38	1.68	11.0	2.86	1.09	2.62	11.0	1.48	0.27	5.48
R468C	11.5	6.31	6.32	0.99	11.1	3.78	2.12	1.78	8.86	0.62	0.07	8.86

* SH: superheat

3.2 The Effects of Different Circuitries on the Heat Exchanger Performance Using R468C

One of the other purposes of this study is to determine the effects of different circuitries on the heat exchanger performance while using R468C. In this regard, the design of HX1 (baseline) was changed in terms of circuitries. It should be stated that all information provided for HX1 in Table 4 remain the same for the other three heat exchangers (i.e. HX 2 – 4), and only the circuitries were redesigned as shown in Fig. 4.

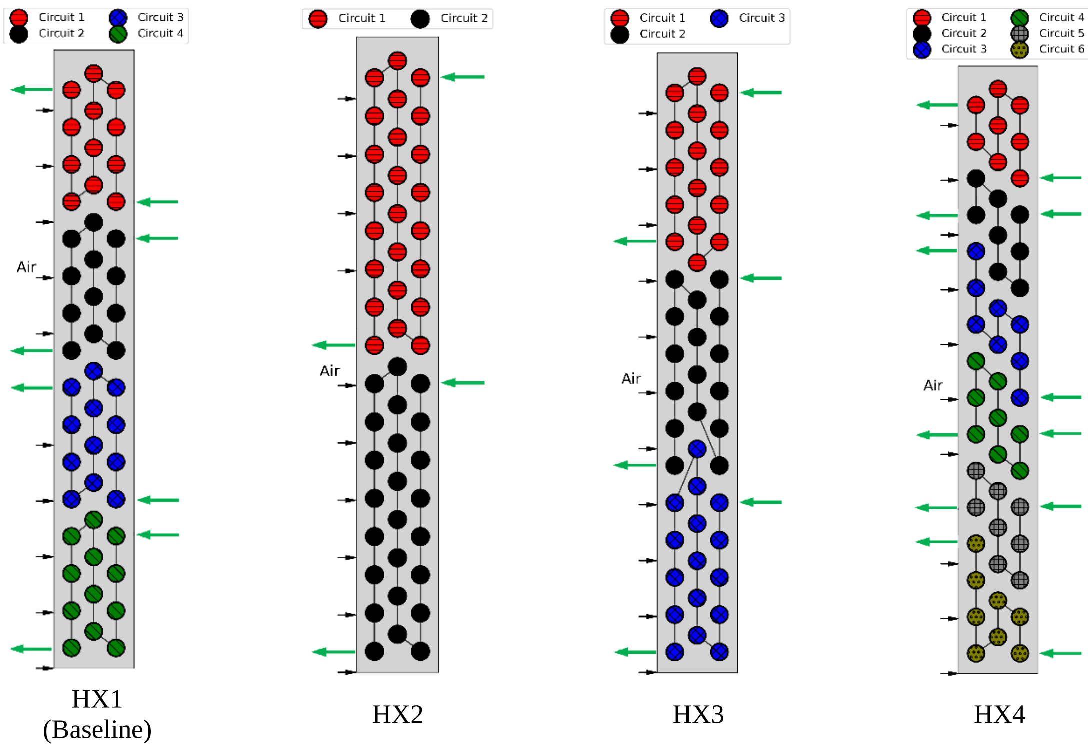

Figure 4: All of the heat exchangers considered for the simulations

Fig. 5 and 6 display the cooling capacity variation versus the refrigerant flow rate and the variation of COP of all heat exchangers for the same cases, respectively. For the other three HXs, the cooling capacity of R468C is higher, while the COP is lower compared to the case using R410A. Again, for 15℃ SST, the maximum achievable SHT was about 9 K for all cases using R468C (red box in Fig. 5). Because of that, the cooling capacity of R468C is lower than R410A, with an SHT of about 11 K.

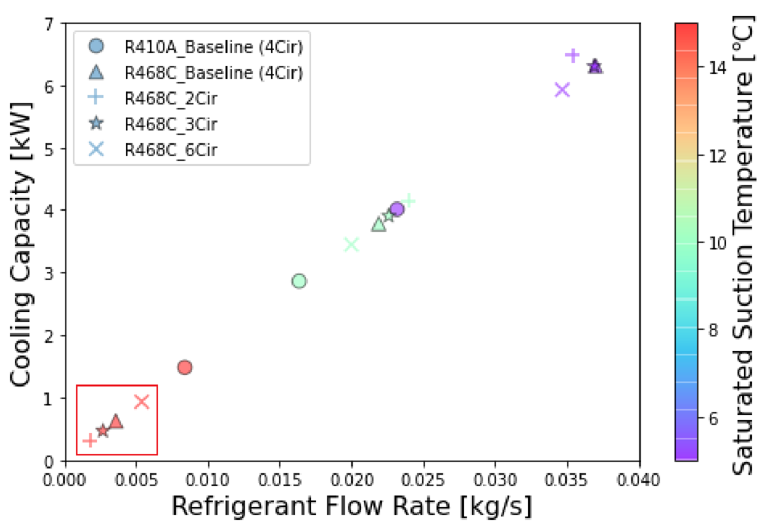

Figure 5: Variation of refrigerant flow rate and cooling capacity to achieve constant superheat at constant saturated suction temperature (5℃, 10℃, and 15℃) for all of the circuitries

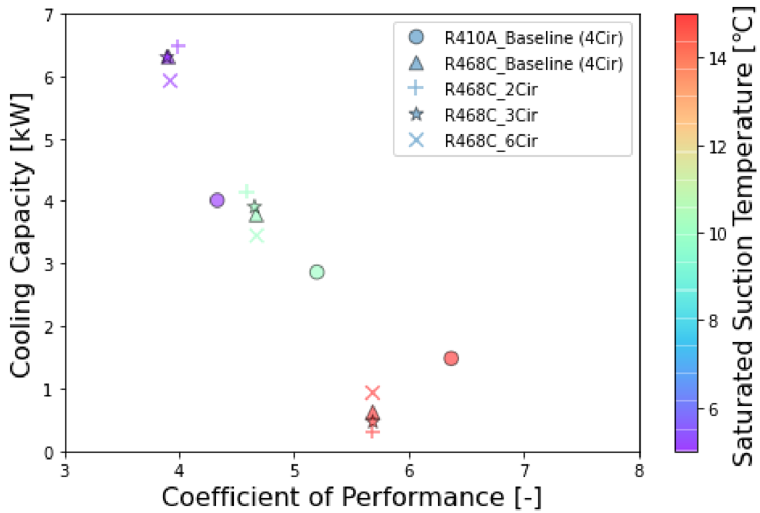

Figure 6: Variation of coefficient of performance and cooling capacity to achieve constant superheat at constant saturated suction temperature (5°C, 10°C, and 15°C) for all of the circuitries

4. CONCLUSION

This paper presented the influence of using R468C as a low GWP alternative refrigerant to cooling capacity and COP of a four circuits fin and tube heat exchanger originally designed for R410A. Moreover, the effects of different circuitries on performance when using R468C were included in the investigation. In total, 24 simulations were performed with the two working fluids (R410A and R468C), covering three different SST (5°C, 10°C, and 15°C) in a dry evaporator testing mode.

The major findings of the simulations can be summarized as:

- R468C required a more refrigerant flow rate (between 30-60%) for identical ΔT_{SH} and SST, resulting in a higher cooling capacity (between 30-57%) but a lower COP (about 10%) than R410A.
- Cooling capacity and COP values for R468C for each of the four different circuitries are close to each other.

This study was restricted to an evaporator coil under a dry testing mode, and refrigerant circuitry was the only parameter that changed. Future work will include modifying a range of other parameters, including tube length and diameter, fin's type and density, as well as a wider variety of coil sizes and refrigerants circuitries. Moreover, other low-GWP refrigerants such as R468B, R444A, etc. can be considered for the future simulations. Furthermore, it should be investigated if other parameters are more suitable for comparing heat exchanger performance with low GWP fluids, such as COP, with equal heat exchanger capacity achieved by adjusting evaporation temperature.

NOMENCLATURE

EER	Energy Efficiency Ratio	P	Pressure, kPa
CFD	Computational Fluid Dynamics	ΔP	Pressure drop, kPa
Cir	Circuitry	$\dot{Q}$	Heat transfer rate, kW
COP	Coefficient of Performance	V	Velocity, m/s
DB	Dry Bulb	$\dot{W}$	Power consumption, kW
FPI	Fins Per Inch		
FTHX	Fin and Tube Heat exchanger	**Greeks**	
GWP	Global Warming Potential	ρ	Density, kg/m^3
HX	Heat exchanger	μ	Viscosity, µPas
MAPE	Mean Absolute Percent Error	η	Efficiency
SH	Superheat		

SST	Saturated Suction Temperature, ℃	**Subscripts**	
WB	Wet Bulb	a,i	Air inlet
xFin	Cross-fin	c	Compressor
T	Temperature, ℃	is	Isentropic
ΔT	Temperature difference, ℃	l	Liquid
F	Fin	p	Pitch
h_{fg}	Latent heat of vaporization, kJ/kg	sat	Saturation
$\dot{m}$	Mass flow rate, kg/s	v	Vapor

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