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Low-GWP Refrigerant blends as Replacements of R410A for Domestic Heat Pumps

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Abstract: Domestic heat pump is regarded as the most important contributor to greenhouse gas (GHG) emissions from buildings. R410A has been one of the most popular refrigerants in the heat pump system because of its excellent thermodynamic performance with near azeotropic nature. However, R410A greatly impacts climate with its global warming potential (GWP) which is evaluated as 2088 CO_2-equivalent. Thus, this research purpose is to investigate refrigerant blends with a low GWP to replace R410A in domestic refrigeration applications. A screening of blends is carried out among the list of 5 low-GWP refrigerants, including R451A, R454A, R454C, R455A, and R459B, which are mainly combined by R32, R1234yf, R1234ze(E), and Carbon Dioxide. The selection is based on the target GWPs (lower than 300), and the coefficient of performances (COPs) are analyzed on a single-stage heat pump cycle model by Python. The results present that the volumetric capacities of low-GWP refrigerants are all lower than R410A, but their COPs are higher except for R455A. R454A is considered the most accessible alternative to R410A while its COP is 2% higher than R410A with the largest volumetric capacity among low-GWP refrigerants. The climate impacts of these low-GWP refrigerants are also studied by the Life Cycle Climate Performance, which shows 22%~25% lower than the emissions of R410A. This research might supply a clue for the development of these new refrigerants to replace R410A.

Keywords: Low-GWP refrigerants, R451A, R454A, R454C, R455A, and R459B

1. Introduction

Global warming has been an international problem. The heat pump system is evaluated as one of the reasonable sources of GHG emissions from buildings and industry. Therefore, companies and scholars have invented considerable resources to decrease the climate impact of domestic heat pump systems. For example, reusing the waste heat for heat transfer[1,2], improving the compressor efficiency[3], or developing refrigerants. Because of its excellent coefficient of performance(COP) and noninfluence of ozone depletion, R410 has been the dominant refrigerant in the HVAC&R market, especially in domestic heat pumps based on its high performance[4]. However, the global warming potential (GWP) value of R410A, i.e., 2088[5], presents the extraordinary climate impact of this binary blend fluid. The substitute for R410A has become the most popular topic for the development of next-generation refrigerants.

Among pure refrigerants, R32 seems the most competitor for R410A, and it has been applied in all the Japanese Refrigeration markets[6]. From Xu et al.'s[7] report, R32 improves 9% of the COP compared with R410A, with a lower GWP of 675. Nevertheless, this GWP value still cannot meet the target of several international regulations, such as European F-Gas directives[8], the Kigali Amendment[9], and Japan's F-Gas control policy[10]. The ways for developing refrigerants and heat pump systems were outlined in the article by Miyara et al.[11], including natural fluids, the low GWP refrigerants, and refrigerant mixtures.

Natural refrigerant such as CO_2 has become attractive for the reachers again[12], while the low GWP refrigerants are a popular topic for refrigerant development. From the report of A. Pal et al.[13], the advance of low GWP refrigerant on the environmental impact is confirmed. However, due to the limitation of thermophysical properties, the low GWP refrigerants, e.g., Hydrofluoroolefins (HFOs), mainly were applied as blend components. Pham and Rajendran11) noted that R32 and

HFOs blends offer potential solutions for R410A replacement in the heat pump system. R. Akasaka[14] studied the experimental performance of R1234ze(E) and its mixture with R32. A detailed analysis of this binary blend in horizontal micro-fin tubes was also carried out by the team of Prof. Koyama[15]. Creamaschi et al.[16] studied the performance of compressor operation with R32 and R1234yf as a drop-in replacement of R410A. Wang et al.[17] investigated mixtures R32/R1234yf on condenser heat transfer experimentally and theoretically. A comparison of R32/R1234yf and R32/R1234ze(E) on heat transfer of evaporator was also discussed by Nakamura et al.[18].

CO_2 was an attractive fluid in-vehicle heat pump applications decades ago. However, CO_2 was once abandoned because of its high critical pressure and low thermal capacity. With the increasing impact of global warming, CO_2 caused the attention of the heat pump industry again. To improve the performance and drop the critical pressure, CO_2 was tried to mix with low GWP refrigerants[19]. Dai et al.[20] assessed the thermodynamic properties of CO_2 blends with R32 and R41. Raabe[21] simulated a molecular study with a mixture of R1234yf, R1234ze(E), and CO_2 on the vapor-liquid phase equilibrium. A comparative assessment of irreversible losses of mixtures CO_2/R32/R1234yf and CO_2/R32/R1234ze(E) on the heat pump system was shown by Fukuda et al.[22].

The database of REFPROP has updated the thermal properties of many mixtures combined with low-GWP refrigerants. Five refrigerants, including R451A, R454A, R454C, R455A, and R459B mainly combined with R32, R1234yf, R1234ze(E), and CO2 with a target GWP value lower than 300, are chosen for this research. Bolaji[23] made a theoretical assessment of five low-GWP refrigerants including R451A compared with R134a and carried out that the COP of R451A was 5% higher than that of R134a, while its specific power consumption was 4% lower. Sjoholm et al[24]. analyzed the compressor performances between R454A and R454C. Vedat and Atilla[25] studied R454A and R454C experimentally in a refrigeration system with R404A. They also published results of R455A for replacing R404A recently[26]. The study of R454C and R455A operating in commercial refrigeration systems for R404A replacement was carried out by Makhnatch et al.[27]. And Mota-Babiloni et al.[28] investigated the same refrigerant by experiment. For the R404A alternative, four low-GWP refrigerants were evaluated by Llopis et al.[29] experimentally and optimized. The results outlined the energy consumption reductions of R454C, R459B, R457A, and R455A compared with R401A were 2.45%, 11.55%, 10.69%, and 2.9%, respectively.

However, the researches on these five new low-GWP refrigerants are insufficient, and there is no study to compare their performance with R410A. Therefore, the object of this article is to provide an evaluation of these mixtures on domestic heat pumps by cycle simulation, while the performance of the heat pump cycle with R410A is held as the reference baseline.

2. Approach

2.1 Testing refrigerants

Five refrigerants, including R451A, R454A, R454C, R455A, and R459B, are evaluated on the domestic heat pump cycle. Their COPs and volumetric capacities are investigated and compared with those of R410A. Table 1 shows thermodynamic properties, the safeties, and the GWPs of testing refrigerants. The thermodynamic characteristics of testing refrigerants are calculated with the data from REFPROP 10.

2.2 Heat pump cycle

A theoretical heat pump cycle is presented for the simulated model with four basic components: vapor compressor, evaporator, condenser, and expansion valve. Two water loops are used for heat transfer. The conditions of operation are displayed in Table 2. Seven states are outlined in the cycle corresponding to critical locations in the system, as shown in Fig 1.

To simplify the modeling processes, several hypotheses are supposed in the simulation:

- The temperature drop of refrigerant blends in the two-phase region is neglected.
- The irreversible losses in the compressor and connected pipes are ignored.
- The binary blends are evenly mixed in tubes.

Based on the solution method shown in Fig 2, the heat pump cycle iterates temperatures in the evaporator and condenser. The average temperature difference agrees with the designed value within a prescribed tolerance, which is less than 0.0001.

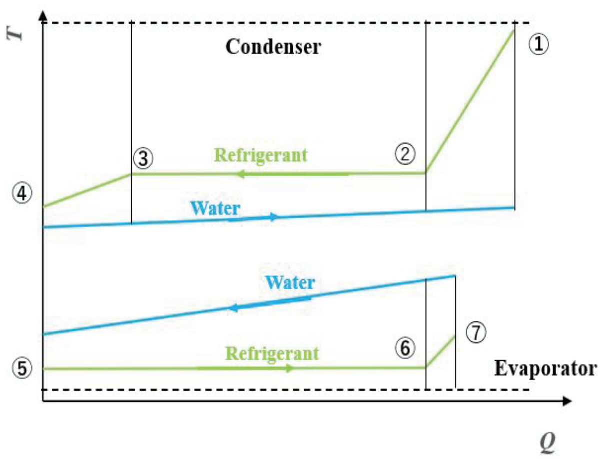


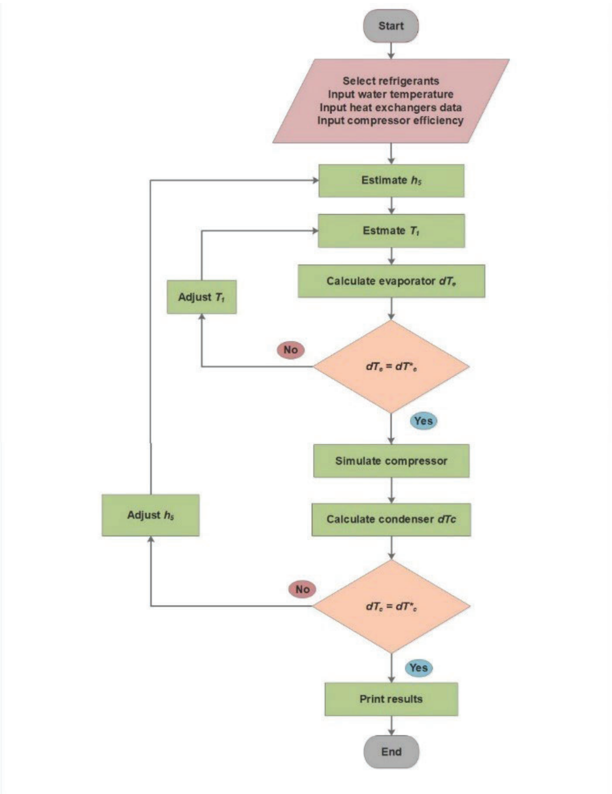
Fig. 1: The heat balance in the heat exchangers.

Table 1. The thermal properties of refrigerants.

	R410A	R451A	R454A	R454C	R455A	R459B
Composition	R32/ R125	R1234yf/ R134a	R32/ R1234yf	R32/ R1234yf	CO2/R32/ R1234yf	R32/R1234yf/ R1234ze(E)
Mass fraction	50%/ 50%	89.8%/ 10.2%	35%/ 65%	21.5%/ 78.5%	3%/21.5%/ 75.5%	21%/69%/ 10%
Critical temperature [℃]	71.3	94.4	81.7	85.7	85.6	87.5
Critical pressure [MPa]	4.90	3.44	4.63	4.32	4.65	4.36
Critical density [kg m^{-3}]	459.53	477.5	456.4	461.6	454.9	460.4
Molecular weight [g mol^{-1}]	72.6	112.7	80.5	90.8	87.4	91.2
ODP	0	0	0	0	0	0
GWP	2088	133.5	237	146	146	142.5
ASHRAE classifcation	A1	A2L	A2L	A2L	A2L	A2L

Table 2. The condition of operation.

Conditions	Water temperature (℃)		Load (kW)	Superheat (K)	Subcooling (K)
	Condenser	Evaporator			
Cooling	31.7 →34.7	24.3 → 15.3	11.8	5	5



* Denotes input value

Fig. 2: Solution logic of the simulation.

2.2.1 Compressor

The compressing process in the model is calculated based on the compressor isentropic efficiency. The suction pressure P_{suc} is decided by the evaporator outlet temperature T_1 and superheat temperature T_{sh}, while the discharge pressure P_{dis} is reposed to the condenser outlet enthalpy h_5, sub-cooling temperature T_{sc} and the pressure ratio r. Eq(1) and Eq(2) show the foundational calculation of the compressor model.

$$r = P_{dis}/P_{suc} \tag{1}$$

$$\eta_{isen} = \frac{h_{2s}-h_1}{h_2-h_1} \tag{2}$$

Here h_{2s} and h_2 present the isentropic and realistic enthalpy of compressor discharge, which are calculated into kJ/kg. h_1 is the compressor suction enthalpy, kJ/kg.

A loop calculation is applied inside the compressor model to evaluate the discharge pressure P_{dis}. The pressure ratio r is initialed as 2.5. Once the discharge pressure is assumed, the state point of the condenser inlet and outlet could be calculated correspondingly. Then the results are compared with the data obtained from the last simulation process in the evaporator to meet the specified value with the tolerance.

2.2.2 Volumetric capacity

Volumetric capacity is considered as the unit mass of the circulating mixture divided by the specific volume of the refrigerant vapor before the input of the compressor, which is equivalently calculated as Eq(3) and Eq(4). In a comparative analysis of various refrigerants, volumetric capacity is regarded as the indicator of the relative capacities in a similar system. The simulation model computes volumetric capacity for an ideal compressor with zero clearance volume without valve losses and gas leakage, which deems volumetric efficiency equal to 1 simultaneously.

$$Q_{v,h} = D_1 * (h_2 - h_5) \tag{3}$$

$$Q_{v,c} = D_1 * (h_1 - h_6) \tag{4}$$

Here $Q_{v,h}$ and $Q_{v,c}$ present the heating and cooling volumetric capacity respectively in kJ/m^3. D_1 denotes the density of mixture at the compressor inlet, in kg/m^3. h_5 is the enthalpy of condenser outlet in kJ/kg, when h_6 means the enthalpy of evaporator inlet, in kJ/kg.

2.2.3 Heat exchanger

The performance of heat exchangers is specified according to the average temperature difference. Moreover, the input data of heat exchangers include the temperature of the water inlet and outlet, the superheat temperature on the evaporator, and the sub-cooling temperature on the condenser.

The heat exchanger average temperature difference is presented as Eq(5)[30].

$$\Delta T_{hx} = \frac{Q_{hx}}{UA_{hx}} \tag{5}$$

ΔT_{hx} is computed by considering different flow regimes in the individual heat exchanger. For example, a condenser might have three regimes: superheat, two-phase, and sub-cooling. Assuming it as a constant value for overall heat transfer coefficient U, and considering $Q_{hx} = \sum Q_i$ and $Q_i = UA_i\Delta T_i$ (where A_i and ΔT_i present heat transfer area and temperature difference in different regimes), Eq(5) can be transformed into Eq(6):

$$\Delta T_{hx} = \frac{A_1}{A_{hx}}\Delta T_1 + \frac{A_2}{A_{hx}}\Delta T_2 + \frac{A_3}{A_{hx}}\Delta T_3 = \frac{\sum A_i \Delta T_i}{A_{hx}} \tag{6}$$

Noting an alternative relation that $A_i = Q_i / \sum \Delta T_i$, Eq(6) is expressed as a harmonic mean weighted with the fraction of heat transfer in different regimes:

$$\frac{1}{\Delta T_{hx}} = \frac{Q_1}{Q_{hx}\Delta T_1} + \frac{Q_2}{Q_{hx}\Delta T_2}\frac{Q_3}{Q_{hx}\Delta T_3} = \frac{1}{Q_{hx}}\sum\frac{Q_i}{\Delta T_i} \tag{7}$$

3. Results and discussion

3.1 Input data

A simulation is performed under the testing conditions shown as follows, which are similar to the operating conditions on a domestic heat pump system:

- Cooling capacity: 11.8 kW
- Water inlet and outlet temperature of evaporator: 24.3℃ / 15.3℃
- Water inlet and outlet temperature of condenser: 31.7℃ / 34.7℃
- The average temperature difference between evaporator and condenser: 9.0℃ / 7.3℃
- Superheat temperature: 5.0℃
- Sub-cooling temperature: 5.0℃
- Compressor isentropic efficiency: 0.8

3.2 Superheat and sub-cooling

The cooling COPs of low-GWP refrigerants are displayed in Fig. 3 compared with R410A. Most of the testing refrigerants perform better than R410A except R455A. R451A and R454A have the best efficiencies, which are 2% higher than R410A. The COPs of R454C and R459B are 1.4% and 1.6% higher, respectively. On the opposite, the COP of R455A is about 3% lower than R410A.

The volumetric capacities of testing refrigerants are all lower than R410A, which are shown in Fig. 4. The result of R451A is the lowest which is 56.4% lower than R410A. Compared to that of R410A, the reduction of R454A, R454C, R455A, and R459B are about 22.3%, 32.8%, 26.8%, and 34.1%, respectively.

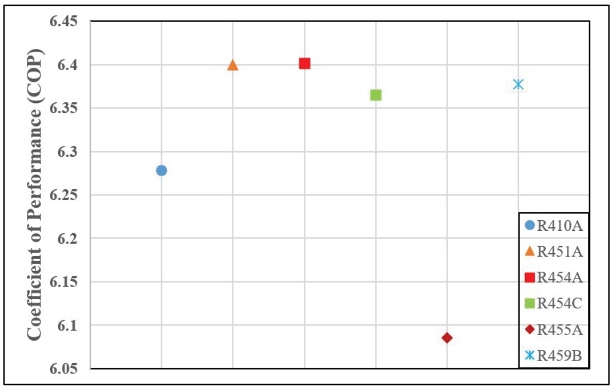


Fig. 3: The COPs of low-GWP refrigerants compared to R410A.

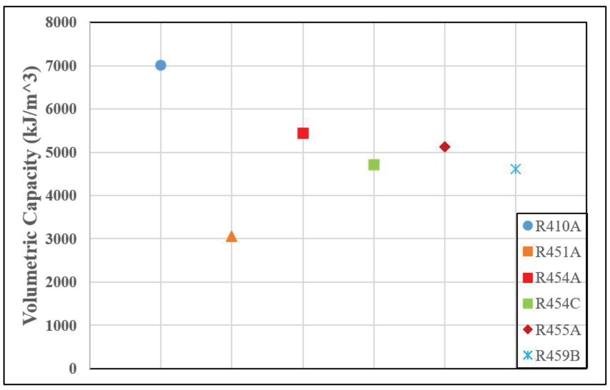


Fig. 4: The volumetric capacity of low-GWP refrigerants compared to R410A.

In summary, most of the testing refrigerants are the accessible alternatives for R410A with better efficiencies in the domestic heat pump system except R455A. But the results of volumetric capacity carry out that the low-GWP can not drop in R410A replacement directly. The compressor needs to be optimized specially to apply the abilities of these new refrigerants.

3.3 Cycle performance

In order to analyze the behaviors of low-GWP refrigerants in the heat pump cycle, Figure 5 shows the p-h diagrams of testing fluids.

It is obvious that R410A has the highest suction and discharge pressure. Thus the consumption of the compressor would be increased even if the heat transfer area is large. The pressure ratio between discharge and suction of R451A is the lowest, which means it can satisfy the cooling load with less work and obtain higher efficiency. The saturation line of R454C, R455A, and R459B are very similar. But the pressure of R455A is much higher than R454C and R459B, which causes the higher compressor work and leads to a worse COP. On the contrary, the pressure of R454A is lower than R410A while its saturation area is close to R410A. Hence R454A can obtain a better performance.

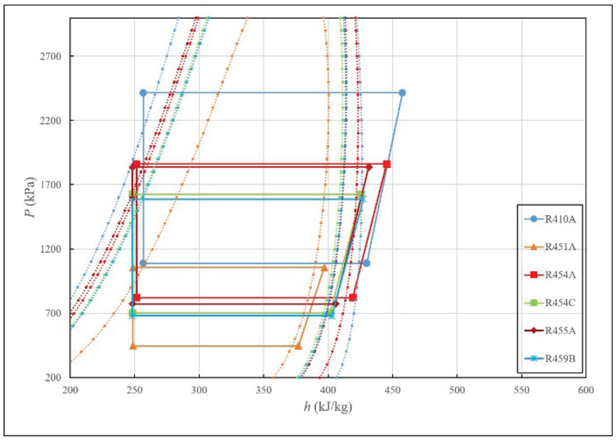


Fig. 5: The p-h diagrams of testing fluids.

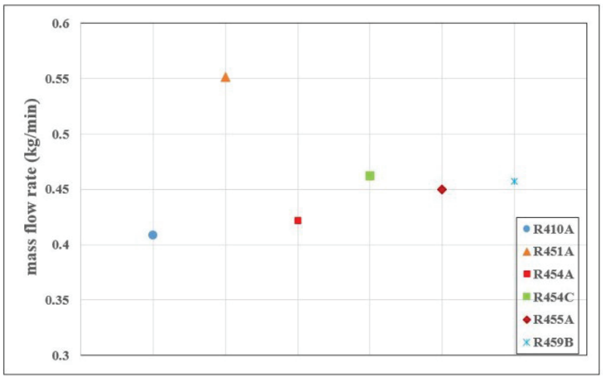


Fig. 6: The mass flow of testing fluids.

Figure 6 shows the mass flow of low-GWP refrigerants compared to R410A. The results display an opposite tendency to the results of volumetric capacity. All the new refrigerants have higher mass flow rates, which means that they can obtain the cooling demand with lower pressure and less heat transfer.

3.4 LCCP analysis

The life cycle climate performance (LCCP) is applied to analyze the climate impact of these low-GWP refrigerants for providing the environmental advantages to replace R410A. The calculating methods are referred from the study by Changru[31]. The results of the LCCP analysis are presented in Fig. 7. The COPs and GWP values are set as the only variants in the LCCP calculation in order to analyze the influence of the performance and climate impact of refrigerants, while other factors are considered constant.

The indirect emissions, which are mainly affected by energy consumption, are similar since the COPs of testing refrigerants are very close. Therefore, The GWP values become a significant contributor. Because the GWP value of R410A is far higher than the low-GWP refrigerants, the total $CO_{2\text{-}e}$ emissions of these new blends are about 22% ~ 25% lower than that of R410A. The results provide evidence for the low-GWP refrigerants to replace the R410A application.

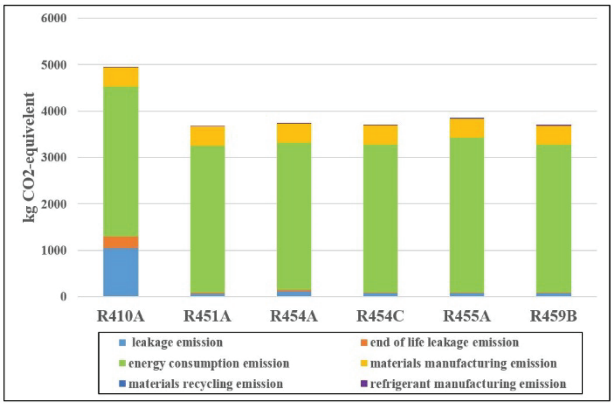


Fig. 7: The LCCP analysis of testing fluids.

4. Conclusion

To alleviate the global warming effect and reduce the emission of greenhouse gases from the domestic heat pump industry, five low-GWP blend refrigerants, which are selected to meet the target GWP value which is lower than 300, are studied for the alternative of R410A. A simulation model of the heat pump cycle is computed and the COP and the volumetric capacity of the binary blends are evaluated by the heat pump model using R410A as a reference. The main results are shown as follows:

a) R454A has the best performance of COP and volumetric capacity in all low-GWP refrigerants, while its volumetric capacity is lower than R410A.

b) R451A has a similar COP to R454A but its volumetric capacity is the worst.
c) The data of mass flow rate verify the results of volumetric capacity.
d) The total emissions of low-GWP refrigerants on LCCP analysis are about 22% ~ 25% lower than that of R410A.

In summary, R454A, a blend of R32 and R1234yf, is regarded as the most accessible refrigerant to replace the application of R410A in the domestic heat pump system. However, the others low-GWP refrigerants still have considerable potential for R410A alternatives. Even though it is not enough to decide on the substitute of R410A only using the COP and the volumetric capacity by simulation. Considering the studies analyzed on these new refrigerants are insufficient, this article may provide clues and contributions for choosing and developing these next-generation refrigerants.

Nomenclature

h	enthalpy (kJ/kg)
A	Heat transfer area (m^2)
D	Density (kg/m^3)
P	pressure (MPa)
Q	Heat transfer (kW)
T	Temperature (°C)
U	Heat transfer coefficient ($kW/(m^2*K)$)
r	Compressor ratio (–)

Greek symbols

η	efficiency (–)

Subscripts

c	Cooling
dis	Compressor discharge
h	Heating
hx	Heat exchanger
isen	Isentropic
suc	Compressor suction
v	Volumetric

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