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An Investigation on the Utilization of R470A for Air-Conditioning Systems Towards 2025

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ABSTRACT

The refrigerants with GWP > 750 will not be used in air conditioners according to the application of European Union restrictions that will become valid as of 2025. In this context, this study aims to investigate theoretically the utilization of the non-flammable and low-GWP refrigerant of R470A in the air conditioning system. In the analysis, R470A and R32 are compared for the evaporation temperatures of 5, 10, and 12°C while the condenser temperatures of 40 and 50°C. Although the COP of R470A is determined to be lower than R32 for a given situation of evaporation and condenser temperature, it can be safely used in systems requiring a higher amount of refrigerant charge due to its non-flammable property.

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1. Introduction

The European Union (EU) aims to reduce the use of refrigerants in air conditioners and other systems to decrease the environmental impact of refrigerants. However, HFCs remain essential in many applications due to their energy efficiency, safety, and economic benefits. The use of HFCs is not banned but must be reduced gradually to a level required for the sustainable growth of the air conditioning, heat pump, and refrigeration industry. The gradual reduction is aimed at reducing its use, preventing leaks, and enhancing the reuse of HFCs. HFOs are proposed as promising refrigerants with low global warming potential (GWP) that can be used in pure or mixture form [1]. According to the ASHRAE safety classification, HFOs are mildly flammable A2L-class refrigerants. Therefore, it may be appropriate to mix HFC and HFO by providing an optimization between flammability and GWP. Flammability, economy, low GWP, stability, critical temperature, and toxicity criteria are considered for determining suitable refrigerants [2].

The use of refrigerants with a GWP greater than 150 in mobile air-conditioning systems has been prohibited since 2017. It is banned to use refrigerators and freezers for commercial applications that contain HFCs with GWP>150 as of 2022. Single-split air-conditioning systems containing less than 3 kg of fluorinated greenhouse gases, that contain fluorinated greenhouse gases with GWP >750 will be banned as of 2025 [3]. In the EU, phase-down is measured by the amount of HFC supplied to the market from January 01, 2015. As shown in Fig. (**1**), total market utilization will be reduced from 100% to 21% in 2030. A reduction of more than 50% is estimated until 2023.

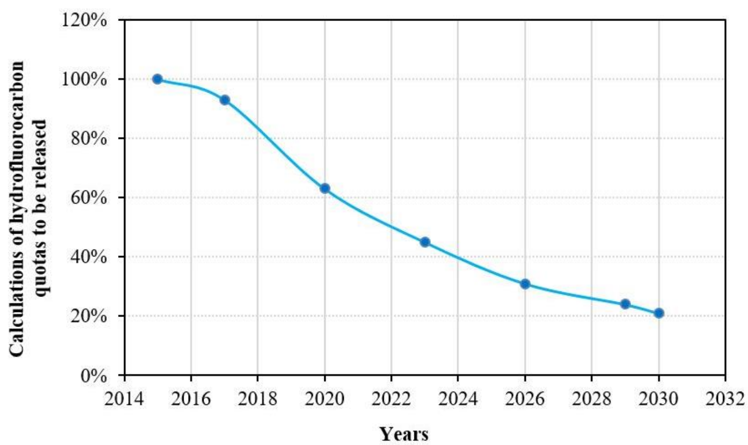

Figure 1: HFC reduction plan as defined in EU regulation.

Europe, North America, and Japan state the GWP value as 750 with legal regulations. However, the Kigali Amendment, an international agreement designed to phase out the use of HFCs with high GWP, provides a quota assignment scheme as a guide rather than clearly defining fixed GWP values. In addition, device manufacturers have not reached a consensus on the determination of the refrigerant. Previously, the idea of using R410A and R32 instead of R22 was widely accepted. The increase in alternative refrigerants and the change of regulations are effective in this situation. From this point of view, why the GWP value should be 750 is open to discussion. According to the manufacturers, R32 has become widespread because it is pure, has high energy performance, and is cheap as well as has a GWP value slightly lower than 750. However, A2L class flammability limits its use in large amounts. GWP is one of the criteria to be considered for selecting refrigerant type. A major influence on the product life cycle of the air conditioning system is due to the carbon dioxide emission when fossil fuels are burned to produce the electrical energy consumed during the operation of the system. Switching to a low GWP refrigerant, which reduces system efficiency and consumes more electricity, would be inappropriate in terms of lifecycle operating costs and total equivalent warming impact (TEWI). Furthermore, minimizing the charge of refrigerant reduces the risk of leakage. Hence, it will be beneficial in terms of both reducing greenhouse gas emissions and safety (i.e., flammability).

Meanwhile, it has been essential to investigate the alternative refrigerants which have low-GWP and flammability class of A1. One of these alternatives is R470A which is a direct drop-in replacement for R410A. Note that R470A is in the safety class of A1 and is a blend of R1234ze(E), R125, R32, R744, R134a and R227ea. Also, it has energy performance and discharge pressure similar to R410A. In addition, it can be used in air conditioning and refrigeration systems.

Literature survey shows that R466A, R452B, R454B, and R32 as alternatives to R410A were compared in terms of exergy and energy performance for two-speed heat pump models [4].

The utilization of L41b in air source heat pump operating with R410A was studied [5]. The reduction in coefficient of performance (COP) values was calculated to be 2.3–10.1% in comparison with R410A. Furthermore, R134a and its low-GWP alternatives of R1234yf, R513A and R516A were compared in heating and refrigerating systems [6]. The highest and the lowest heating capacity amounts were found to develop in heating mode for R513A and R516A cases, respectively. In another study, R466A with low GWP was used as a substitute for R410A in a variable refrigerant flow (VRF) system [7]. The results indicated that COP values of R466A were found to be higher than R410A about by 4% and 5–15% in heating mode and cooling mode, respectively. The best replacements for commonly used refrigerants and their mixtures for air-conditioning systems were studied [8]. The mixture of R152a and R1234ze(E) with 70%/30% was recommended as the best refrigerant for air-conditioning systems because of the high COP value as well as economic cost. Moreover, R410A was compared with R446A, R32, L41a and DR5 in a heat pump [9]. It was found that the energy efficiency ratio values of R32 and DR5 were higher than R410A.

R32 based 22 different alternative blends were investigated as a replacement for R410A [10]. The amount of cooling capacity of 22 mixtures was slightly greater than R410A, while their COP value was computed to be about 10% lower than R410A. Although both the cooling capacity and COP values of R32 are higher than R410A, however, it is inflammable. Moreover, the GWP of the R32 value is not very low which can be regarded as a disadvantage situation. The thermodynamic performance of VRF systems was theoretically studied using R32, and R410A [11]. It was observed from the analysis that the COP of R32 was 6% higher in cooling mode and 5% higher in heating mode in comparison with R410A. In another paper, drop-in tests of low-GWP alternative refrigerants (R32, D2Y60, and L41a) for R410A in a 10.55 kW capacity split heat pump unit were achieved [12]. The results indicated that R32 and L41a were suitable replacement candidates for R410A. The performance of the air conditioner with R32 and four different compositions of R32/CO_2 blend was investigated experimentally and numerically [13]. It was found that the rate of increase in the cooling capacity is lower than the rate of increase in energy consumption of the air conditioner as the percentage of CO_2 in the mixture increases. As a result, it is observed that the COP of the system is reduced by 9% for R32/CO_2 (90%/10%) mixture compared to R32 system.

The implementation of R32, R452B, and R454B as replacements for R410A was studied [14]. The capacity improved between 4.9% and 13% for cooling cases using R32 while COP did not increase in all cases. For R452B and R454B, the capacity nearly remained the same, but COP was enhanced between 1.6% and 8.0%. The research regarding the use of R32 and its mixtures as refrigerants for residential AC systems is summarized [15]. The performance of R410A is compared with R32 [16]. It was seen that R32 capacity is higher compared to R410A in both cooling and heating modes in all the simulated operation conditions. The discharge temperatures of R32 are always greater compared to R410A. Furthermore, the refrigerant hold-up of R32 inside the evaporator and condenser is always determined to be lower R410A.

The performance difference was investigated using R410A and R32 in a vapor-injected heat pump system [17]. It was found experimentally that the capacity and COP improvements using R32 can be obtained as 10% and 9%, respectively compared to the R410A case. Further investigations on the alternative refrigerant utilization to R32 can be found in the available literature [18-23].

Since refrigerants with GWP > 750 will not be used in air conditioners according to the application of EU restriction which will be valid as of 2025, it has been aimed to investigate the suitable refrigerants for the available systems until the near future. This investigation, it is aimed to theoretically compare the energy performances of R470A (safety class of A1 and GWP=909) with R32 in air-conditioning systems for different evaporation and condenser temperatures. The utilization of non-flammable refrigerant will be beneficial in terms of using it in the air conditioning system especially requiring a high amount charge of the refrigerant.

2. The Studied Refrigerants

The GWP values and safety classification of the refrigerants used for air conditioning systems are presented in Table **1**. The GWP of R32 is lower than R470A by 26%. Obviously, GWP of R470A is 21% higher than the limit value of 750. Furthermore, R470A is non-flammable but R32 is classified as mildly flammable by ASHRAE.

Table 1: Composition, GWP, and safety classification of the investigated refrigerants.

Refrigerants	Composition	Mass Fraction (%)	Safety Classification	GWP (AR5)
R32	R32	100	A2L	675
R470A	R1234ze(E)/R125/R32/R744/R134a/R227ea	44/19/17/10/7/3	A1	909

The pressure-enthalpy (*P*-*h*) behavior of the studied refrigerants is shown in Fig. (**2**). As it is seen the critical temperature of R32 is greater than R470A.

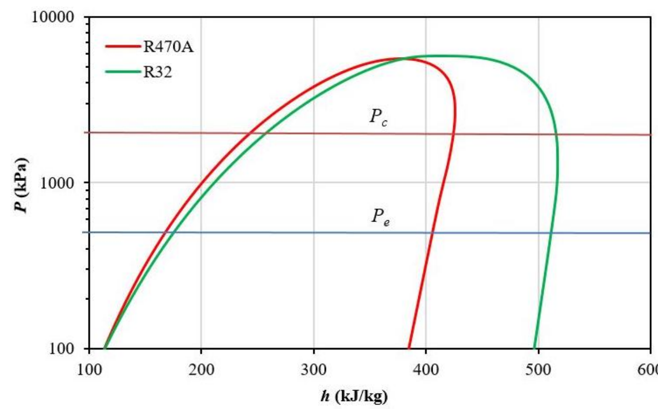

Figure 2: Pressure-enthalpy diagram of the compared refrigerants.

The critical temperature and pressure values for the investigated refrigerants are given in Table **2**. When the critical temperature of the refrigerant is high, the vapor pressure should be low; hence the volumetric cooling capacity will be lower for the refrigerant. Since the properties of R470A and R32 are similar (Table **2**) refrigerant charge, mass flow rate, and pipe size parameters should also be similar. However, the pipe size for the R470A system would be somewhat greater than the R32 system. Furthermore, as it is known the amount of refrigerant charged to the system will be dependent on the size of system components (compressor, condenser, and evaporator), length of piping, and refrigerant type. An air conditioner operating with R32 requires a refrigerant charge of 1000 g. Due to the similar properties of R32 and R470A, it can be estimated that the refrigerant charge amount of about 1100-1300 g may be suitable for the system working with R470A.

Table 2: Some properties of the studied refrigerants [24].

Property	R470A	R32
Boiling point (1 atm) (°C)	-62.5	-51.6
Critical temperature (°C)	88.7	78.1
Critical pressure (kPa)	5591.0	5782
Liquid density (kg m^{-3})	1088.0	961.01
c_p liquid (kJ kg^{-1}K^{-1})	0.854	1.9
k liquid (mW m^{-1} K^{-1})	82.8	125.9
Liquid viscosity (μPa s)	135.0	113.7
Latent heat (kJ kg^{-1})	267.3	382.0

The distribution of thermal conductivity coefficient (k) of the refrigerants with temperature (T) for liquid and vapor phases is plotted in Fig. (**3**). In fact, higher k of the liquid phase enables a smaller surface area for the heat exchanger (i.e., evaporator). Therefore, the initial investment cost is reduced. Obviously, k values of the refrigerant for the vapor phase are close to each other.

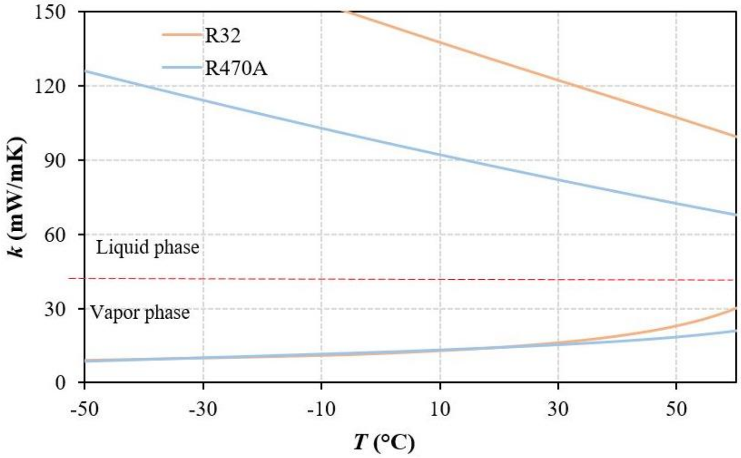

Figure 3: Variation of thermal conductivity coefficient with temperature for the refrigerants.

It is useful to note here that during the evaporation process of a refrigerant, the temperature at which a liquid refrigerant begins to vaporize is known as the saturated liquid temperature. Moreover, the temperature at which the last drop of liquid refrigerant vaporizes is known as the saturated vapor temperature. Then, the difference between saturated vapor temperature and saturated liquid temperature at constant pressure is known as the "temperature glide" of the refrigerant. For the studied evaporation temperatures, temperature glide values between 4-8 K would not be a problem.

3. Theoretical Analysis

The theoretical study is performed for a basic vapor compression refrigeration cycle in the investigation. The basic components of the system as well as its pressure-enthalpy diagram are schematically depicted in Fig. (**4**).

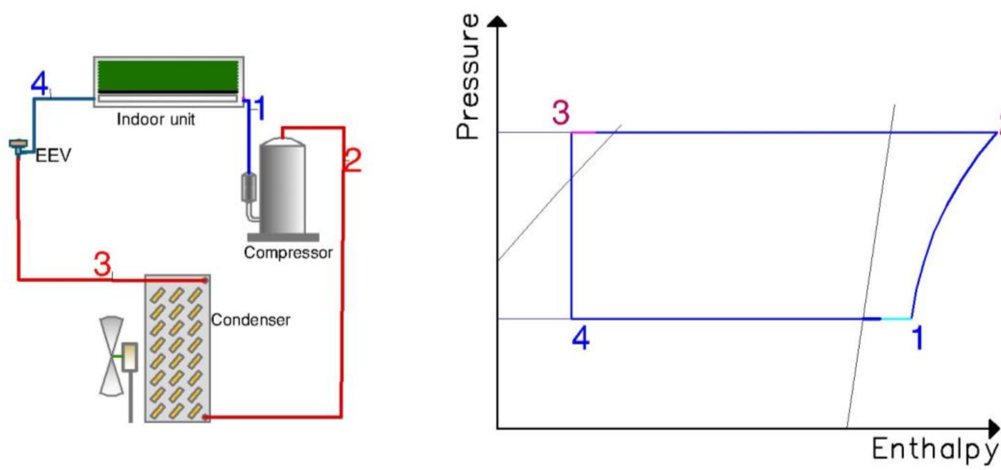

Figure 4: A schematic sketch of the vapor compression refrigeration cycle for the analysis.

The system parameters considered for the theoretical analysis are given in Table **3**. Also, it is assumed that the system operates in a steady-state regime; the pressure and heat loss in the system components and pipes are neglected; the evaporator and condenser don't consume energy; the kinetic and potential energy changes are ignored.

Table 3: Assumed system parameters in the theoretical analysis.

Cooling capacity (W)	5000
Evaporation temperature, T_e (°C)	5/ 10/ 12
Condensing temperature, T_c (°C)	40/ 50
Superheat (K)	7
Sub-cooling (K)	4
Isentropic efficiency (%)	90

It is remarked that similar evaporation and condenser temperatures (Table **3**) were considered in the previous studies [25]. The mass flow rate ($\dot{m}$) of refrigerant flowing in the system in kg/s is defined as

$$\dot{m} = \frac{Q_e}{(h_1 - h_4)} \tag{1}$$

where Q_e is the cooling capacity in kW, *and h* the enthalpy values in kJ/kg. The subscript numbers indicate the states of the cycle as demonstrated in Fig. (**4**). The power consumption of the compressor (W_{el}) in kW is computed as

$$W_{\mathrm{el}} = \dot{m}(h_2 - h_1) \tag{2}$$

In addition, the isentropic efficiency of the system (η_s) can be calculated as

$$\eta_s = \frac{(h_2' - h_1)}{(h_2 - h_1)} \tag{3}$$

where h' corresponds to the enthalpy for the case that entropy remains constant through the inlet and exit states of the compressor. The energy performance of any air-conditioning or refrigeration system is given through the coefficient of performance (COP) which is found by the ratio of cooling capacity to the power consumption of the compressor [25]:

$$\mathrm{COP} = \frac{Q_e}{W_{\mathrm{el}}} \tag{4}$$

4. Results and Discussions

The distribution of mass flow rate ($\dot{m}$) for the studied evaporation temperature (T_e) and condenser temperature (T_c) is depicted in Fig (**5**). Firstly, $\dot{m}$ increases as values of T_e and T_c decrease and increase, respectively for both refrigerants. The increase in $\dot{m}$ due to higher T_c for R470A is greater than that for R32 at a specific T_e. For a given case, $\dot{m}$ values of R32 are found as lower than R470A about by 45%.

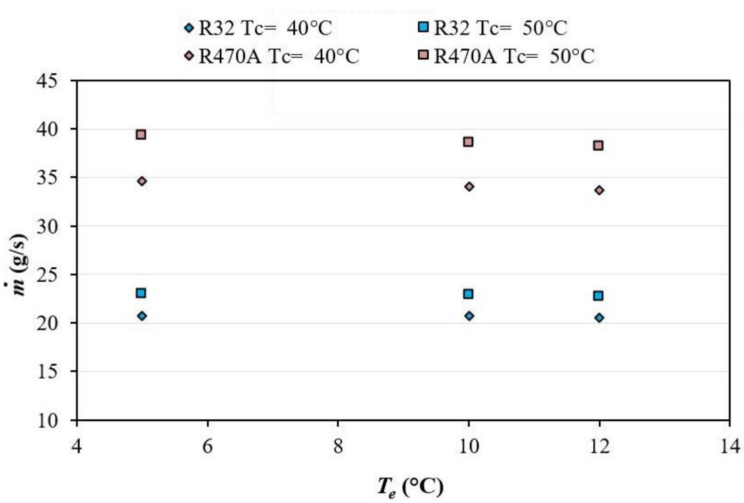

Figure 5: The relationship between mass flow rate and evaporation temperature.

The variation for the power consumption of compressor (W_{el}) with T_e is given in Fig. (**6**). Obviously, W_{el} is reduced as T_e increases for both refrigerants such that the amounts of W_{el} for the case of R32 at T_c =40°C are 788 W and 600 W at T_e values of 5°C and 12°C, respectively. On the other hand, W_{el} increases as T_e takes greater values, for example considering the case of R470A at T_e =10°C points out that W_{el} takes 1195 W and 1550 W for T_c values of 40°C and 50°C, respectively. Moreover, R470A consumes higher power compared to R32 for a given case of T_e and T_c.

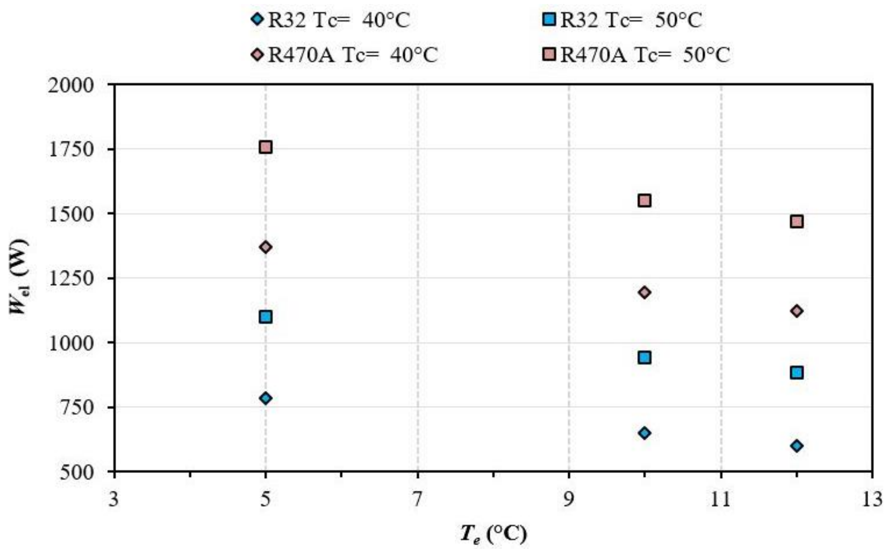

Figure 6: Variation of compressor's power consumption with evaporation temperature.

The behavior of COP for the covered cases in the present investigation can be seen in Fig. (**7**). It is obvious that COP of R32 is greater than R470A for a given case. It can be stated that the COP of R470A is lower than R32 nearly by 40% for a fixed situation of T_e and T_c. Note that this is an expected situation because COP increases as W_{el} is reduced according to Eq. (4). Hence, the distribution of W_{el} in Fig. (**6**) exactly supports the results of COP observed in Fig. (**7**). Furthermore, COP is enhanced as T_e increases for a given refrigerant at a constant T_c, but it is reduced as T_c increases from 40 to 50°C at a constant T_e.

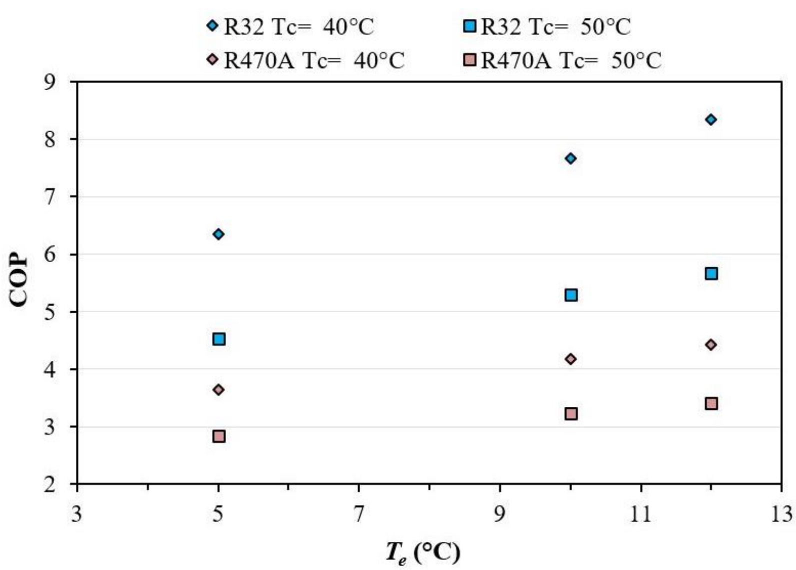

Figure 7: The variation of COP with evaporation temperature for investigated cases.

The distribution for the discharge temperature of the compressor (T_d) with the covered combinations of T_e and T_c is demonstrated in Fig. (**8**). It is clear that T_d is reduced when T_e increases, however, T_d increases as T_c has greater values. More importantly, T_d of R470A is higher than R32 for the covered cases.

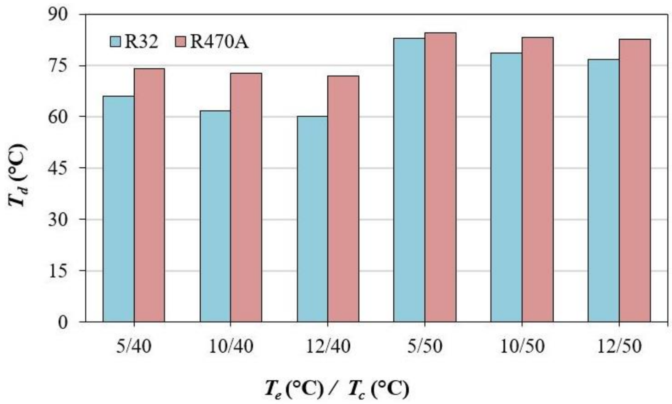

Figure 8: The distribution of compressor discharge temperature for the covered cases.

5. Conclusion

Comparisons of R32 and R410A have been made frequently in the past. R410A has a high GWP value while R32 is flammable, therefore the amount of refrigerant charge used for the equipment is limited for indoor spaces. Consequently, the refrigerants should be investigated considering the parameters of non-flammability, economy, and thermodynamic performance such as cooling capacity and power consumption. Hence, R32 is compared theoretically with R470A for the air-conditioning systems in this study. Although the energy performance of R470A is determined to be worse than R32, it can be suggested that R470A may be used in air-conditioning systems because of its suitable GWP as well as non-flammable properties. The basic results can be summarized as follows:

- W_{el} of R470A is higher than R32
- COP of R32 is greater than R470A
- T_d of R32 is always smaller than R470A for a given condition of T_e and T_c
- GWP of R470A is bigger than R32, however the former is non-flammable, therefore it can be utilized safely for systems requiring a higher amount of refrigerant charge without violating the restrictions.

Conflict of Interest

All authors declare that they have no conflicts of interest.

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Nomenclature

COP	coefficient of performance
GWP	global warming potential
h	enthalpy, kJ/kg
k	thermal conductivity coefficient, W/mK
$\dot{m}$	mass flow rate, kg/s
P	pressure, kPa
Q_e	cooling capacity, kW
T	temperature, °C

T_d	discharge temperature of compressor, °C
T_c	condensing temperature, °C
T_e	evaporation temperature, °C
W_{el}	power consumption of compressor, kW
η_s	isentropic efficiency

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