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EXPERIMENTAL EVALUATION OF PERFORMANCE OF R32 AS A SUBSTITUTE FOR R410A IN COOLING SYSTEMS

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Abstract. *The following paper presents an experimental evaluation of a cooling system using R410A, verifying if the refrigerant R32 is a possible substitute due to its considerably lower GWP. This research was motivated by the current necessity to find alternative refrigerants that have the lowest impact possible to the environment. To make this work possible, an experimental cooling bench was setup, with nominal capacity of 5 TR (~15,6 kW) so that it was possible to obtain operational parameters such as: cooling capacity, compressor power consumption, coefficient of performance and compressor discharge temperature. The environmental impact was calculated by using the TEWI and LCCP methodologies. Tests were performed varying the evaporation temperature between -7 and 7 °C, while the condensing temperature was maintained in 41 °C. Besides the evaporation temperature, the compressor operation frequency was varied from 45 to 60 Hz. The results obtained showed that compressor power consumption using R32 increased from 3 to 5% when compared to R410A. The cooling capacity presented itself 3 to 7% higher when using R32. The Coefficient of Performance was 2 to 8 % better when using R32. The compressor discharge temperature was 20 to 30 °C lower when using R410A. To all evaluated test conditions, the Coefficient of Performance and cooling capacity were higher for R32, therefore concluding that R32 performance was superior than R410A performance. Finally, calculating the TEWI and LCCP to both analysed refrigerants, it was noticed that direct impacts were reduced using R32, mostly due to its lower refrigerant mass in the system and its considerably lower GWP. On the other hand, due to higher power consumption, indirect impacts were higher for R32.*

Keywords: R410A, R32, cooling system, TEWI, LCCP

1. INTRODUCTION

The HVAC industry has evolved constantly since the XIX century, when the refrigeration cycle by gas compression was discovered. As new systems were studied and developed, as well as new materials and fluids, cooling systems have become more efficient, economic and, therefore, widely used by society. Initially, natural gases were used for cooling purposes, however, low energetic efficiency and problems due to toxicity and flammability became a big concern. As a solution, scientists developed synthetic fluids, such as chlorofluorocarbons (CFC) and hydrochlorofluorocarbons (HCFC). An example of such fluids is R22, which is still used in cooling systems, especially in developing countries.

During the XX century, scientists started to concern about the situation of the environment. In 1974, Rowland and Molina demonstrated that synthetic fluids used in the HVAC industry were highly contributing for the ozone layer depletion. Consequently, a search for new refrigerants and alternative solutions began.

As an incitement for the development of new sustainable technologies, environmental laws were created, limiting the use of refrigerants that negatively affect the ozone layer. Those laws were established through the Montreal Protocol (1987) and they determined goals that would eliminate the use of refrigerants with big environmental effects. Since 1987, the Montreal Protocol has been revised eight times.

Resulting from the search for alternative refrigerants, scientists developed the hydrofluorocarbons (HFC). These fluids do not have chlorine in their composition and, therefore, do not react with the ozone layer, avoiding its degradation. On the other hand, the HFCs were discovered to increase the greenhouse effect, and hence increase in global warming. Kyoto Protocol (1997) established a reformulation on environmental laws to minimize the effect on global warming.

The last revision of the Montreal Protocol established the Kigali Amendment (2016), which includes the production and consumption control of HFCs. A schedule was determined, dividing the countries in three categories, stipulating different goals for each group. Industrialised countries, which had already stopped the production of HFCs, will have a reduction of 85% from baseline consumption until 2036. Developing countries, such as Brazil, China and South Africa, will stop producing HFCs until 2024 and reduce 80% from baseline consumption until 2045. A third group including

India, Iraq and Pakistan, will stop producing HFCs between 2024 and 2026 and reduce its consumption in 85% from baseline until 2047.

In 1991, looking for alternatives to substitute R22 for HVAC purposes, Allied Signal produced the refrigerant R410A, an azeotropic mixture of R32 (difluoromethane) and R125 (pentafluoroethane). This mixture have zero Ozone Depletion Potential (ODP). Due to its high operating pressure, R410A cannot be used as a drop-in substitute from R22. The design of new systems were required to use this refrigerant and be in agreement with the Montreal Protocol (Schultz, 2014). Nowadays, R410A is the main refrigerant used to substitute R22 for residential applications in industrialised countries (Chen, 2008; Han *et al.*, 2012; Padalkar *et al.*, 2014).

Although R410A does not affect the ozone layer, according to the Intergovernmental Panel on Climate Change (IPCC), R410A has a GWP (Global Warming Potential) of 2088, which is considered highly elevated. Following the goals established in the Kyoto Protocol, it is anticipated that R410A will no longer be used in the future. For this reason, scientists started to search for new alternative refrigerants with lower GWP as substitutes for R410A.

While searching for alternative refrigerants, tests with pure R32 started to take place as substitute for R410A (Dalkilic and Wongwises, 2010; Xu *et al.*, 2013). R32 has a GWP of 675 (Forster *et al.*, 2007), which is, approximately three times lower than the value for R410A. R32 belongs to a class of refrigerants with low flammability and has a higher volumetric capacity in comparison with R410A, what makes systems with R32 have, potentially, a higher COP (Barve *et al.*, 2012).

Leck (2010) reported that, as compressor discharge temperatures using R32 are commonly higher, systems using this refrigerant show a lower reliability due to fatigue of its valves and thermal tension of the lubricant. To minimize this effect, it is suggested that R32 is used in a mixture, instead of completely pure.

Taira *et al.* (2011) observed that due to its higher volumetric capacity when compared to R410A, the amount of R32 used into the system could be reduced by 50%, using the same system. As the quantity of refrigerant is lower, the direct impact on greenhouse effect might be reduced. In several countries, especially in Asia, R32 has already been proposed as a substitute to R410A. On the other hand, due to its flammable properties, there is still hesitation to operate with R32, specially for residential applications.

2. METHODOLOGY

2.1 EXPERIMENTAL SETUP

The bench used to collect experimental data was developed and operated at the Energy, Thermal Systems and Nanotechnology Laboratory of the Federal University of Uberlândia. The bench consists of a cooling system with cooling capacity similar to an air conditioner with residential applications.

The mechanism can be represented as single refrigeration cycle, formed by a scroll type compressor with variable rotation, a condensing unit cooled by water, an electronic expansion valve and an evaporating unit. In this cycle, the refrigerant (in this case R410A or R32) leaves the compressor and is transported directly to the condensing unit, where it exchanges heat with water provided from a cooling tower. As the refrigerant leaves the condensing unit, it passes through a Coriolis type flow meter and expands in the expansion valve. Afterwards, the refrigerant enters the evaporating unit, where it is heated until a pre-set superheating point and then returns to the compressor. Fig. 1 illustrates the experimental bench.



Figure 1. Cooling unit used to collect experimental data

Due to high pressures and temperatures of the refrigerant during operations, the bench was built to maintain the safety of the laboratory objects and the people working on it. The tubes were selected considering proper material and thickness enough to withstand the operating conditions. The compressor is a hermetic scroll type, manufactured by Emerson, designed to operate between 50 and 60 Hz, with volumetric flow rate between 7,01 and 9,55 m³/h and nominal capacity

of 5 TR (~15,6 kW). This compressor was chosen based on the operating conditions and acknowledging that it is ideal for R410A applications. The tubes were insulated to minimize heat exchange with the outside environment.

The thermal load was controlled using three resistances, monitoring the evaporation temperature. These resistances are responsible to simulate the thermal load inside the insulated reservoir. The water is thus responsible for exchanging heat with the refrigerant in the evaporating unit. The increase of evaporation temperature influences in the energetic balance of the system. As a response, the condensation temperature elevates, as the other system conditions remain unchanged. To monitor the condensation temperature, the water flow coming from the cooling tower was changed.

The cooling tower was designed to enable the water used at the condensation unit to arrive as cold as possible. After leaving the condensation unit, the water is warm due to heat exchange with the superheated refrigerant from the compressor. This water is transported through a no-insulated pipeline, facilitating it to exchange heat with the external environment. Inside of the cooling tower, there is a fan responsible for generating an air flow to cool the hot water. As the water becomes cold, it returns to the condensation unit. To control the water flow returning to the condensation unit, a valve was installed. Fig. 2 presents a sketch of the experimental setup.

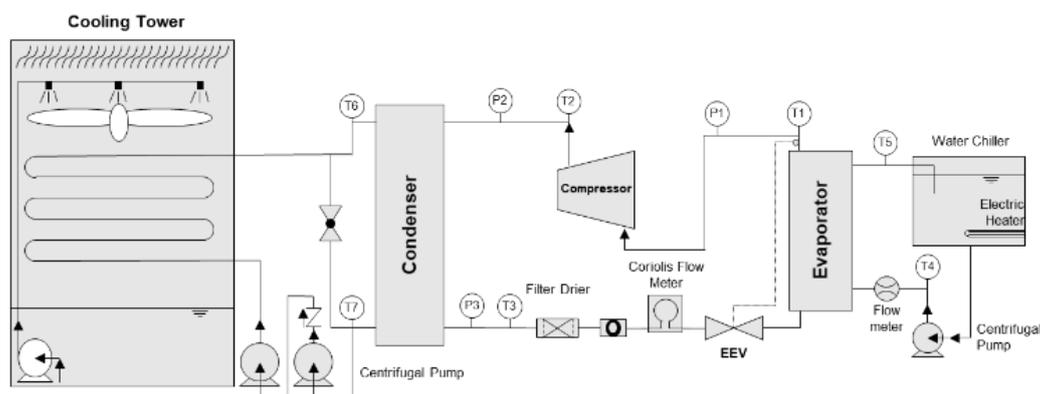


Figure 2. Scheme of the experimental setup

The cooling tower is extremely influenced by ambient temperature, as it uses air to remove heat from the water. During colder days, the water can become too cold, lowering the condensation temperature. On the other hand, during warmer days, the water might not exchange enough heat, increasing the condensation temperature. Due to this dependency, a bypass was installed, connecting the output of the condensation unit with its input. Therefore, the hot water leaving the condensation unit would mix with the cold water coming from the cooling tower, allowing a more effective temperature control and, thus, favouring the system stabilization.

To evaluate the performance of R410A and R32, tests in different conditions were conducted, maintaining some variables fixed and varying others. Superheating was fixed at 10 °C, controlled by the electronic expansion valve. It was possible to control the opening of the valve, which controls the output pressure of the evaporating unit and, consequently, the degree of superheating. The condensation temperature was fixed at 41 °C, controlled by the water flow coming from the cooling tower entering the condensation unit. The two varied parameters during tests were the evaporation temperature and the operation frequency of the compressor. To perform the tests, the compressor was run in 45, 50, 55 and 60 Hz, varied through a frequency inverter. To each of this frequencies, the evaporation temperature was evaluated at the following values: 7 °C, 5 °C, 3 °C, 1 °C, -1 °C, -3 °C, -5 °C and -7 °C. The evaporation temperature was controlled through the thermal load, as mentioned before.

The system instrumentation was able to obtain all the essential data to calculate the energetic performance and the environmental effects. Pressure and temperature sensors were installed in the following system spots: condensation unit entrance and exit of refrigerant; evaporating unit refrigerant and water inlet and outlet, evaporating unit refrigerant outlet and water inlet and outlet. To obtain temperature data, RTD (Resistance Temperature Detector) sensors PT-100 type were used, inserted directly in the refrigerant and water lines. The pressure was measured using piezo resistive sensors, also installed directly in the fluid lines. The range of the sensors installed in the condensation unit was 0-25 bar, while in the evaporating unit was 0-10 bar.

2.2 TEWI

To verify if a refrigerant is suitable to substitute another one already in use, it is necessary to analyse the environmental effects of each refrigerant. To calculate these effects, the TEWI (Total Equivalent Warming Impact) and the LCCP methods were adopted. The impact of a refrigerant can be divided in two parts. The first part is classified as the direct impact. That means this is the part related to the amount of refrigerant released to the external atmosphere. This impact is strongly influenced by the refrigerant GWP, by the type of the system and the maintenance of components and operations. The second part is classified as the indirect impact, related to the amount of electricity required by the system

and the pollution caused generating electric energy. The local energy matrix of the system is the main factor that contributes to this impact. Eq. (1) shows how the TEWI method is calculated.

$$TEWI = CO_2equ_{DIRECT} + CO_2equ_{INDIRECT} \quad (1)$$

Where:

CO_2equ_{DIRECT} : Direct impact

$CO_2equ_{INDIRECT}$: Indirect impact

The direct impact part are calculated according to Eq. (2)

$$CO_2equ_{DIRECT} = (GWPmL_{annual}n) + m(1 - \alpha)GWP \quad (2)$$

Where:

GWP: Global Warming Impact [-];

m: Refrigerant load [kg];

L_{annual} : Leaking rate per year [%];

n: System operating life [years];

α : Recovery/recycling factor from 0 to 100 [%].

The indirect impact part are calculated according to Eq. (3)

$$CO_2equ_{INDIRECT} = \beta E_{annual}n \quad (3)$$

Where:

β : Indirect emission factor (kgCO₂ per kWh);

E_{annual} : Energy consumption per year (kWh per year).

2.3 LCCP

Similar to the TEWI method, the LCCP also evaluates the environmental effect of refrigerants. It also divides the effect in direct and indirect impacts. The difference between TEWI and LCCP is that LCCP takes some factors for impact calculation that are ignored when using the TEWI method. Therefore, TEWI can be used when a quick comparison is intended. For a more detailed evaluation, the LCCP method should be used (Harmonization of Life Cycle Climate Performance Methodology, 2016).

Equation (4) shows how to calculate environmental effects using the LCCP method.

$$LCCP = \text{Direct Emissions} + \text{Indirect Emissions} \quad (4)$$

The direct emissions are calculated according to Eq. (5)

$$\text{Direct Emissions} = m(nL_{annual} + EOL)(GWP + Adp.GWP) \quad (5)$$

Where:

GWP: Global Warming Impact [-];

m: Refrigerant load [kg];

L_{annual} : Leaking rate per year [%];

n: System operating life [years];

EOL: End of life refrigerant leakage [%].

Adp.GWP: GWP of atmospheric degradation product of the refrigerant [kgCO₂ per kg]

The indirect emissions are calculated according to Eq. (6)

$$\text{Indirect Emissions} = nE_{annual}\beta \sum (m_u MM) + \sum (m_r RM) + m(1 + nL_{annual})RFM + m(1 - EOL)RFD \quad (6)$$

Where:

m: Refrigerant charge [kg];

L_{annual} : Leaking rate per year [%];

n: System operating life [years];

EOL: End of life refrigerant leakage [%].

β : Indirect emission factor (kgCO₂ per kWh);
 E_{annual} : Energy consumption per year (kWh per year);
 m_u : Mass of unit [kg];
 MM : CO₂ produced per material [kgCO₂ per kg];
 m_r : Mass of recycled material [kg];
 RM : CO₂ produced per recycled material [kgCO₂ per kg];
 RFM : Refrigerant manufacturing emissions [kgCO₂ per kg];
 RFD : Refrigerant disposal emissions [kgCO₂ per kg].

3. PRELIMINARY RESULTS

3.1 INDIVIDUAL ANALYSIS

The first part of the results consist in an individual analysis of each refrigerant (R410A and R32) operating using the same cooling system. To each fluid, a total of thirty two tests were conducted, varying evaporation temperature and compressor operation frequency.

3.1.1 COOLING CAPACITY

Fig. 3a presents the cooling capacity values in kW for each evaporation temperature using R410A. It can be observed that for higher evaporation temperatures, the cooling capacity is also higher. This result is expected due to the increase of the difference between condensation and evaporation temperatures. This bigger difference entails the refrigerant entering the evaporating unit with bigger steam quality, causing a decrease in the difference between inlet and outlet enthalpies. Consequently, the cooling capacity decreases. Higher frequencies show higher cooling capacities. This can be explained by the First Law of Thermodynamics. Fig. 3b is similar to Fig. 3a, showing results for R32. The values obtained the same tendencies as using R410A, being in concordance with the theory. From Fig. 3, it can be observed that R32 shows higher cooling capacities. The highest cooling capacities using R410A were approximately 15 kW, while these values using R32 were approximately 17 kW

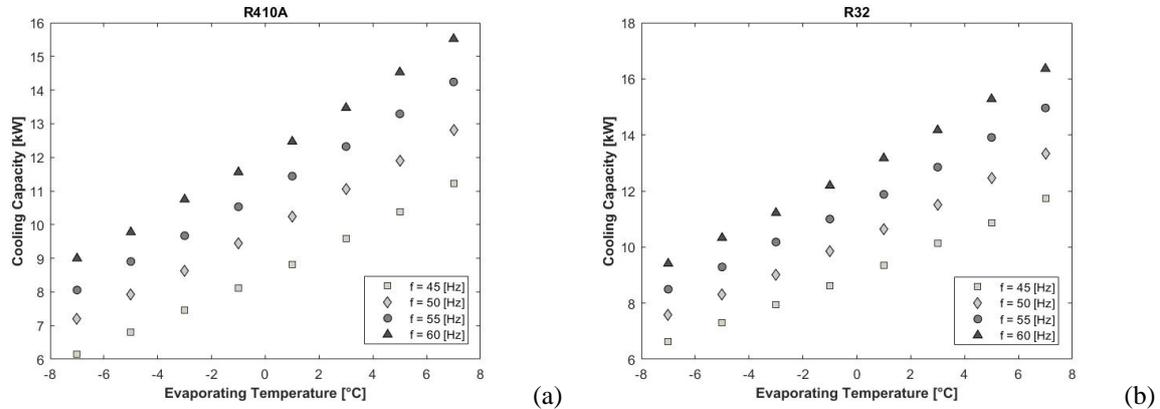


Figure 3. Cooling capacity for different evaporation temperatures and different operation frequencies, using R410 (a) e R32 (b).

3.1.2 INPUT POWER

Fig. 4a shows the compressor input power values, in W, to each evaporation temperature using R410A. It can be observed that lower evaporation temperatures presented higher input powers. Higher cycle evaporation temperatures, maintaining a constant condensation temperature, cause less pressure variation during compression phase. Therefore, it is expected a lower power consumption. As input power is directly proportional to operation frequency, it is also expected higher values for bigger frequencies. Fig. 4b shows input power data for R32. For R410A, the highest input power values were approximately 3570 W, while the lowest values were approximately 2750 W. As for R32, the highest input power values were approximately 3700 W, while the lowest values were approximately 2900 W.

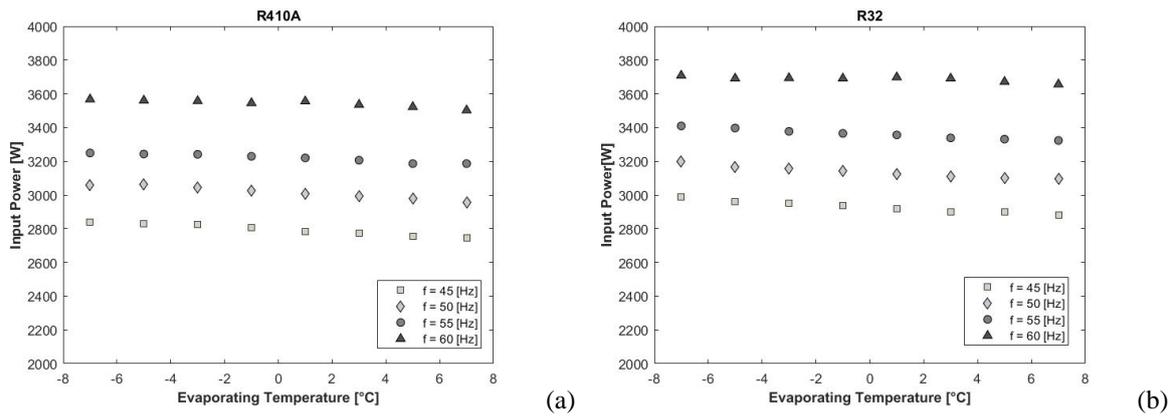


Figure 4. Input power for different evaporation temperatures and different operation frequencies, using R410 (a) e R32 (b).

3.1.3 COP

Knowing the values for cooling capacity and input power for each test condition, it is possible to calculate the system coefficient of performance for the different conditions for each refrigerant. The values can be seen in Fig. 5. It can be observed that higher evaporation temperatures lead to higher COP value. From Fig. 5 it is also possible to observe that COP values are higher for R32 than for R410A in the same conditions. COP varied from 5 to 2,5 for both refrigerants

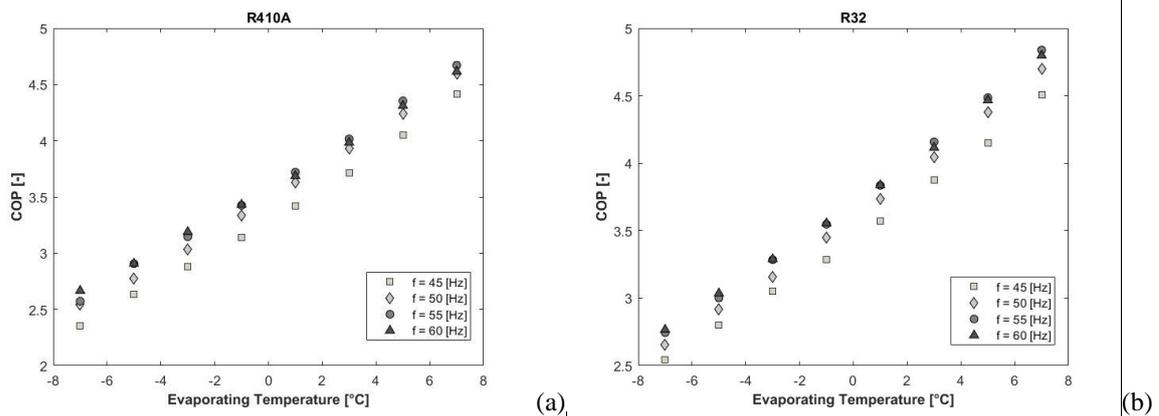


Figure 5. COP for different evaporation temperatures and different operation frequencies, using R410 (a) e R32 (b).

3.1.4 DISCHARGE TEMPERATURE

Fig. 6 presents values for the compressor discharge temperature, for the different test conditions. It can be observed that for lower evaporation temperatures, the discharge temperature increases, as expected. As previously said, lower evaporation temperatures leads to higher pressure differences during compression phase if the condensation temperature is maintained the same. In a thermodynamic cycle, pressure and temperature are correlated, which means, higher pressure differences lead to higher temperature differences. For the 45 Hz values, the discharge temperature was considerably higher. The increase of internal gas leakage during the compression phase causes this effect (Chloe and Kim, 2000). For R410A, the temperature increased approximately 25 °C. For R32, the temperature increased approximately 35 °C.

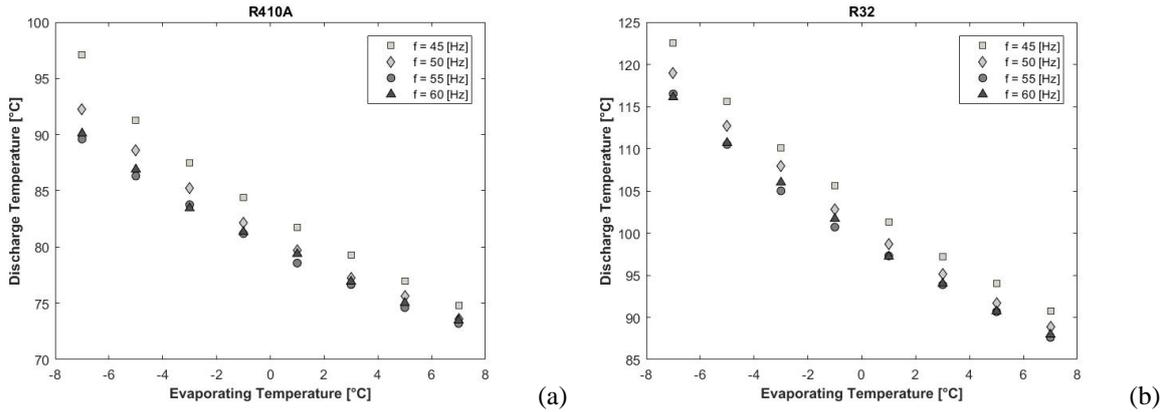


Figure 6. Compressor discharge temperature for different evaporation temperatures and different operation frequencies, using R410 (a) e R32 (b).

3.2 COMPARATIVE ANALYSIS

The second part of results consisted in evaluate the performance of both refrigerants, comparing the results obtained in both cases.

3.2.1 COOLING CAPACITY

Figure 7 shows the relation between cooling capacities using R32 and R410A for all test conditions. It can be observed that for all cases, the cooling capacity was higher for R32. This is expected due to bigger enthalpy differences for R32, when compared with R410A in the same evaporation temperature. The obtained values suggest that cooling capacity using R32 in 3,8 to 7,8% higher than using R410A.

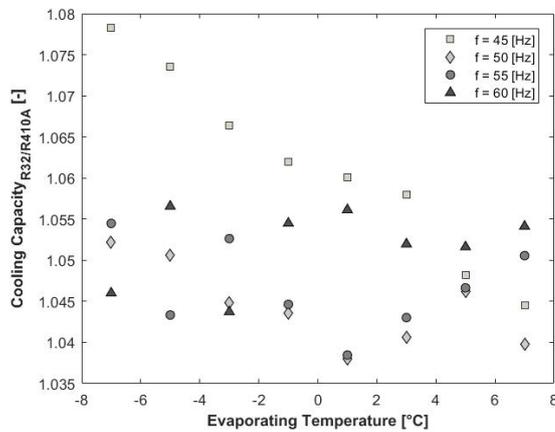


Figure 7. Ratio between cooling capacities for R32 and R410A, respectively, for all test conditions.

3.2.2 INPUT POWER

Values comparing the compressor input power using R32 and R410A for all test conditions are presented in Fig. 8. It can be observed that for all test conditions, the power consumption using R32 was from 3,3 to 5,3% higher than the power consumption using R410A. This can be explained applying the First Law of Thermodynamics in the compressor. Even though R32 has a bigger specific volume and, therefore, has lower mass flow, the enthalpy differences between entrance and exit of the compressor are wider for R32. Looking at Fig. 9, it can be seen that the isentropic lines for R32 are more inclined at the superheated steam region, explaining the bigger difference of enthalpy.

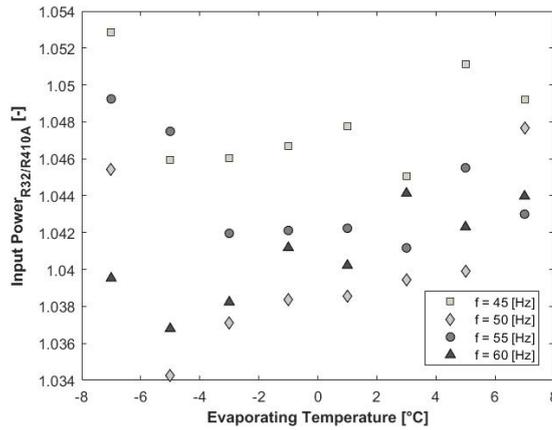


Figure 8. Ratio between input power for R32 and R410A, respectively, for all test conditions.

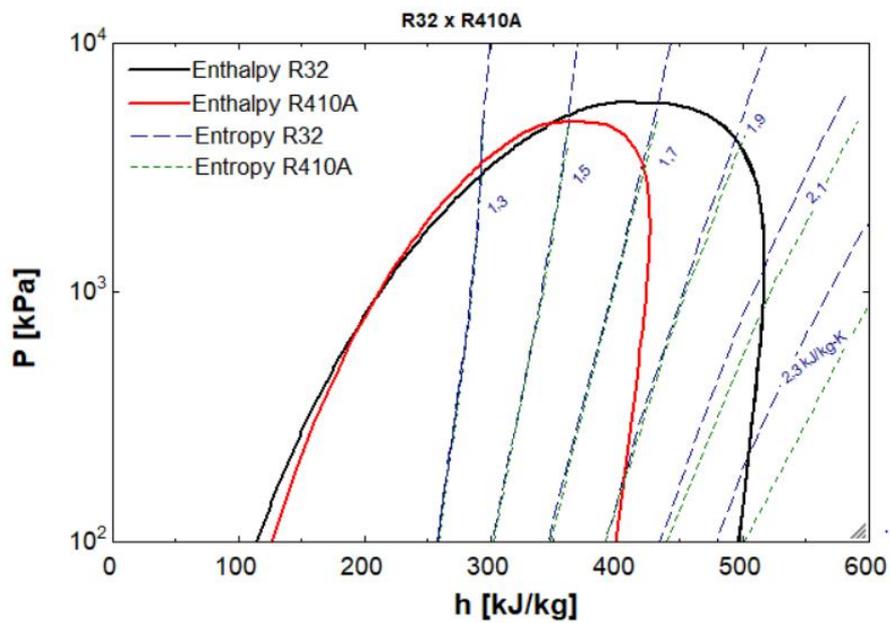


Figure 9. Enthalpy and entropy values for R32 and R410A. Figure generated using Engineering Equation Solver [EES].

3.2.3 COP

Values comparing the coefficient of performance of R32 and R410A are represented in Fig. 10. It can be observed that for all test conditions, COP was 2 to 8,1% higher for R32. The results are according to expected. Looking back at Fig.7 and Fig. 8, it is noticeable that increases for cooling capacity were more significant than increases for input power. Therefore, COP will increase for all test conditions.

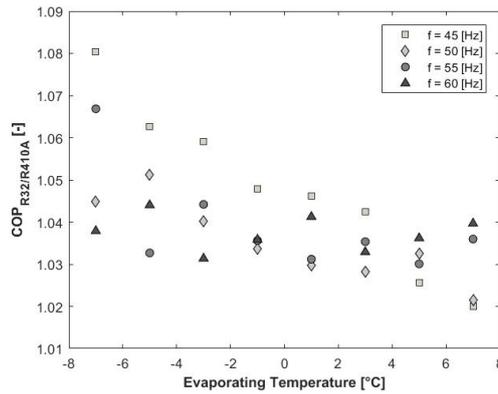


Figure 10. Ratio between COP for R32 and R410A, respectively, for all test conditions.

3.2.4 DISCHARGE TEMPERATURE

Fig. 11 presents values comparing the compressor discharge temperature using R32 and R410A for all test conditions. It can be noticed that an increase of 20 to 30% was reported when substituting R410A for R32. Looking back at Fig. 9, the isentropic lines for R32 are more inclined, which causes a higher increase of temperature.

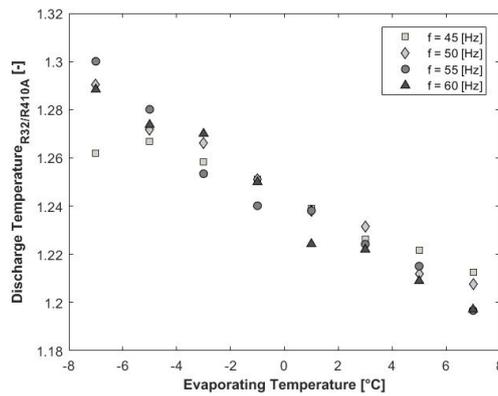


Figure 11. Ratio between discharge temperature for R32 and R410A, respectively, for all test conditions.

3.2.5 OVERALL PERFORMANCE

To evaluate which refrigerant had a better performance, an overall comparison was made, analysing both COP and cooling capacity for all test conditions. Fig. 12 shows values of cooling capacity and COP for all conditions. It is noticeable that all values are located on the upper right side of the figure. This means that for all analysed conditions, there was an increase for COP and cooling capacity values. Therefore, it can be concluded that R32 showed a better performance when compared to R410A.

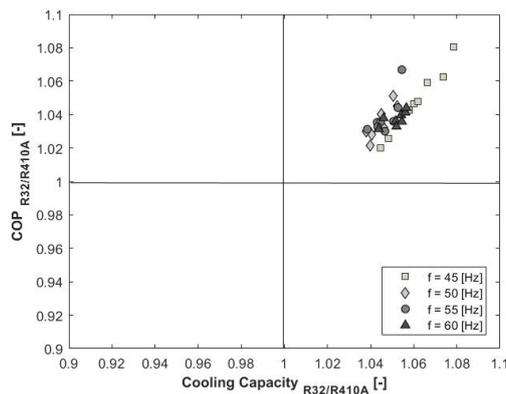


Figure 12. Values of COP and cooling capacity for R32 and R410A for all test conditions.

3.3 TEWI

To calculate the TEWI, the annual refrigerant leakage was considered 7% for both R32 and R410A. This value was used because the system is a chiller, with typical cooling application (Methods of calculating Total Equivalent Warming Impact (TEWI), 2012). The GWP for R32 is 675, while the GWP for R410A is 2088, as previously stated. The refrigerant charge for R410A was 2,045 kg, while for R32 was 1,5 kg. The total equipment operating time is ten years, with 70% of the refrigerant recovered by the end of the equipment life. To calculate the annual electric energy consumed by the equipment, it was taken input power of two different test conditions: 60 Hz and 7 °C; 60 Hz and -7 °C. The input power was multiplied by the amount of operating hours per year. It was considered 3600 hours of operation. To calculate the CO₂ emissions, it was used the Brazilian energy matrix (0,073 kgCO₂/kWh).

For the R410A, the TEWI was approximately 13,47 tonCO₂. For the R32, the TEWI was approximately 10,60 tonCO₂. Therefore, even though the power consumption using R32 is higher, increasing its indirect impact, it still has lower total environmental effect, due to its lower direct impact. The direct impact for R32 is lower due to its lower refrigerant charge and lower GWP.

3.4 LCCP

To calculate the LCCP, refrigerant charge, annual refrigerant leakage, GWP, and system operating time and the amount of CO₂ produced per kWh was maintained the same as for the TEWI method. The other parameters were taken from the International Institute of Refrigeration. The end of life refrigerant leakage was considered 15%. The GWP of atmospheric degradation product of refrigerant was considered zero to both refrigerants. The mass of unit was 50 kg. The CO₂ produced per material was 12,6 kgCO₂/kg. It was considered that 67% of the material was recycled and the CO₂ produced per recycled material was 0,63 kgCO₂/kg. The refrigerant manufacturing emissions were 10,7 kgCO₂/kg for R410A and 7,2 kgCO₂/kg for R32. The refrigerant disposal emissions were 0,07 kgCO₂/kg for both refrigerants.

For the R410A, the LCCP was approximately 13,02 tonCO₂. For the R32, the LCCP was approximately 11,25 tonCO₂. Once again, even though indirect impact for R32 is higher, the overall impact is lower.

4. CONCLUSIONS

After experimental data was collected and calculations on environmental impact were made, it can be concluded that R32 could be a possible drop-in substitute for R410A in cooling systems applications.

R32 showed higher cooling capacities and COP when compared to R410A. Therefore, the overall performance of R32 was better than the overall performance of R410A, even though R32 offered higher compressor power consumption and higher discharge temperatures.

TEWI and LCCP analysis exposed that the environmental impacts using R32 are lower than using R410A. This is mostly due to its lower GWP and lower refrigerant charge, reducing direct impact. On the other hand, indirect impact is higher due to higher power consumption.

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