

Comparative Exergetic Analysis of Synthetic Refrigerants R-22 and Natural Fluid R-290 on a Test Bench with Residential Chiller System

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Abstract

Currently, in the world, there is a real race, involving several manufacturers, in the search for more efficient refrigerant fluids, with the need to develop new refrigerant fluids, to replace the fluids that are being used in air conditioning and refrigeration systems, aiming to maintain or improve the performance of such systems, reducing their electrical energy consumption. Therefore, this work intends to be another contribution to this scenario and aims to carry out an exergy analysis comparing the operating characteristics of a residential Chiller system, in which the natural fluid Propane (R-290) was used, operating at 60 Hz (normal frequency), and at 70 Hz (with inverter controller). The refrigerant R-290 was compared with the HCFC fluid R-22 in the same indirect expansion refrigeration system. For comparison purposes, the main operating parameters of a refrigeration or air conditioning system were observed, such as pressures and temperatures, electrical energy consumption and exergy of the system components. Analysis of the Coefficient of Performance (COP) and system exergy showed that the high frequency (70 Hz) caused total energy waste to increase by 30% compared to R-22 refrigerant. When working with both fluids at 60 Hz, the only negative point for the natural fluid was the higher suction pressure, indicating that the chilled water temperatures were lower with R-22. Under normal conditions, the results expected in this work were obtained, in which the natural fluid showed satisfactory behaviour, with values higher than those obtained for the synthetic fluid, for local psychrometric conditions, being another alternative for replacing R-22 in systems air conditioning and refrigeration.

Keywords: *exergy analysis; refrigeration; HVAC-R; natural refrigerant; air conditioning.*

1. Introduction

This study was produced and developed at the Climatization and Thermal Comfort Laboratories – ClimatConT and Refrigeration of the Federal University of Pará – UFPA, and aims to analyse types of refrigerant fluids used in air conditioning and refrigeration systems, replacing the current fluids used commercially, aiming to maintain the safety conditions of the systems, with the lowest environmental impact and evaluating their performance and thermal efficiency indexes.

In a refrigeration or air conditioning system, a set of refrigerant fluids are used, many of which are already discontinued or have ceased production, as is the case of hydrochlorofluorocarbons – HCFCs, such as the R-22 refrigerant fluid, but there is still a large quantity of these fluids in stock and that continue to be used by the systems. This work proposes to provide support for

decision-making on system modernization, taking as a reference the energy and exergy analyses of the system components.

Due to global economic development and, aiming to prevent substances harmful to the environment from continuing to be used by countries, mainly in air conditioning and refrigeration systems, given the significant increase in economies and, consequently, in the use of refrigerant fluids, both at a residential, commercial and industrial level. In the last 60 years, countless meetings and meetings have been held in different countries around the world. Starting with the Montreal Protocol (1987), in Canada, which restricted the use of chlorofluorocarbon refrigerants (CFC, such as R-11), reaching COP 26, in Scotland in 2021, which discontinued the use of carbon dioxide (CO₂) in production systems.

Long before the Montreal Protocol, several studies were carried out, and in them, the destruction of the ozone layer and the increase in global warming were proven, creating indicators such as the Global Warming Potential (GWP) and the Ozone Depletion Potential (ODP), for constant monitoring of these parameters. Since then, several fluids have been used, analysed, and those that have been proven to present some type of damage, or that contribute to global warming, have been discontinued, had their production terminated, or were banned from use in air conditioning and heating systems. refrigeration.

Even so, nowadays, the indiscriminate use of some fluids such as HCFCs (R-22) still continues to harm the environment, as when these substances are released into the atmosphere, during maintenance processes, for example, there is an increase in of global warming, or simply, they destroy the earth's protective ozone layer. Thus, starting from the contextualization of the use of refrigerant fluids in refrigeration systems, where new substitutes were sought for the fluids currently used, this work was designed and carried out with the objective of observing the behaviour and evaluating whether it would be possible to use natural fluids, with a low degree of contribution to global warming, in a water and air cooling system, simulating a commercial air conditioning system, which was set up on a Chiller test bench in the laboratory, where comparative exergy results were obtained between two types of refrigerant fluids, in order to serve as input in the selection of refrigerants for air conditioning and refrigeration systems.

In this work, what is expected is to present a set of comparative characteristic curves of the parameters used in the energy and exergy analyses of the synthetic fluids R-22 and the natural fluid R-290, operating in a refrigeration system at frequencies of 60 and 70 Hz.

2. Contextualizing the use of Refrigerant Fluids

Starting in the 1950s, when researchers from several international entities realized that the use and release of refrigerant fluids into the local atmosphere had a direct impact on our planet, causing global warming and destruction of the ozone layer, a global monitoring of these fluids was created annually. effects. Since then, with the popularization, initially of refrigeration systems and, later, of air conditioning, the massive use of these systems has worsened the conditions of the Earth's protection systems (Ozone layer) [2].

Numerous world events were held and, at each event, solutions were sought for this scenario, which is getting worse every day, with the sharp increase in residential, commercial and industrial refrigeration, and with the air conditioning of environments: residential, commercial, industrial and automotive, the use of refrigerant fluids has grown significantly, but without effective actions to control the maintenance of these systems.

According to the International Energy Agency – IEA, with the pollution generated, as societies grow technologically, in recent years, there has been an increase in global emissions of carbon dioxide – CO₂

related to energy, which are around 33 Gigatons, and the concentration of CO₂ in the atmosphere is greater than 420 parts per million – ppm. [1]

According to [2], in Brazil, in the areas of air conditioning and refrigeration, it is common to obtain, from the responsible installation technicians, information on the modernization of systems or on the replacement of refrigerant fluids used in refrigeration and air conditioning equipment, but without informing the criteria or parameters used in the process.

Companies in the air conditioning and refrigeration sectors, which produce refrigerant fluids, are looking for fluids to replace hydrochlorofluorocarbons – HCFCs, such as R-22 and Hydrofluorocarbons – HFC, such as R-410A, considering that production and the commercialization of HCFCs has already been interrupted in Brazil, and HFCs are being discontinued from 2024 onwards, even so, there are still large stocks of fluids produced and stored for use. In this sense, it is necessary to identify or prepare new fluids, in order to use parameters to maintain the energy and exergy efficiency of the systems.

These are CO₂ emissions where it is generated directly as a product of a reaction, however, other chemical compounds used in industry and commerce are also as polluting as CO₂ and, with pollution factors hundreds to thousands of times more intense than carbon dioxide. This factor was called Global Warming Potential – GWP, to account for its pollution potential (global warming) in relation to CO₂, which has GWP equal to 1 [2].

Among these chemical compounds are the refrigerant fluids used in heating, refrigeration and air conditioning systems, such as Trichlorofluoromethane, known as refrigerant R-11 and Dichlorofluoromethane, or R-12, these fluids were mass produced interrupted in Brazil in the 1990s [3].

According to ABRAVA (2023), the refrigeration and air conditioning markets account for approximately 82% of the consumption of refrigerant fluids, which will have less availability as deadlines for discontinuation or production stoppage approach, as is the case with R410a fluid. Therefore, refrigerant changes during a retrofit or new installation are certain, and everyone interested on the planet needs to understand the paths that the market is proposing.

Thus, all these developments described above, in summary, imply transformations for the HVAC-R market as a whole. Designers, installers and suppliers, as well as customers, will need to conduct updates (or even component replacements) on their already installed systems.

However, there are already several reports of serious accidents involving technicians who believed they were working with a type of fluid, such as HFC R-134A, whose ASHRAE flammability and toxicity classification is A1, that is, a non-flammable and non-toxic refrigerant, in a refrigeration system, when in fact some other technician had already replaced the system fluid with R-290, whose flammability and toxicity classification is A3, that is, this is a flammable and non-toxic refrigerant fluid [2 and 3].

These refrigerants have, respectively, GWP values equal to 4,750 and 10,910, which means that releasing 1.0 kg of R-11 into the atmosphere represents pollution caused by the release of approximately 5,000 kg of CO₂. For R-12, this value represents almost 11,000 kg of CO₂ released into the atmosphere for each kg of this refrigerant [2 and 3].

Table 1 shows the types of refrigerant fluids most used in air conditioning systems, regarding their composition, ODP, GWP and ASHAE Safety Classification [2], [7] and [14].

Table 1. Types of refrigerant fluids used in air conditioning systems.

Number	ODP	GWP	ASHAE Safety Classification
CFCs			
R – 11	1	4750	A1
R – 12	1	10900	A1
HCFCs			
R – 22	0.055	1810	A1
HFCs			
R – 32	0.0	650	A1
R – 134a	0.0	1430	A1
R – 410a	0.0	2090	A2L
HFOs			
R – 1234yf	0.0	4	A2L
R – 1234ze	0.0	6	A2L
R – 449a	0.0	1396	A1
Natural Refrigerants (NRs)			
R – 290	0.0	~5	A3
R – 600a0	0.0	~5	A3

As can be seen in Table 1, many of the refrigerant fluids used in air conditioning systems seek to present themselves as ecological fluids, which do not harm the environment where they are used, however fluids such as R-22, R-134a and R-410a and its likely replacement R-449a, have a strong contribution to global warming, not living up to what is disclosed by resellers and manufacturers of these products [2].

A viable solution to this scenario would be the collection of these discontinued or banned refrigerant fluids, by resellers, technicians and engineers who work directly in the air conditioning and refrigeration areas, through appropriate collection and storage procedures, so that recycling companies, in each State in Brazil, can give a new destination to the current fluids used.

This is a slow solution process, and many technicians, due to lack of financial resources, knowledge and technical training, still today many professionals release fluids from air conditioning and refrigeration systems into nature, without worrying about their consequences, this being a common practice. And, even Brazil, having signed the Kigali amendment, in 2024, there are still no concrete actions for control, storage and reuse or disposal of these types of products in the various Brazilian states.

Precisely because of these two factors mentioned above, the refrigerant fluid research and development industry

has been seeking to improve systems that use refrigerant fluids, whether for heating, refrigeration and air conditioning, with a focus on reducing electricity consumption, combined with the use of fluids that are less aggressive to the environment. Some of these fluids are Hydrocarbons – HC, among them the most used are Propane (R-290) and Isobutane (R-600a), with the main advantage being the low GWP value which is equal to 3.0 for both fluids. [4], [5], [6]

In addition to this advantage of having a low level of pollution (global warming), according to [3], several studies indicate that mechanical vapor compression systems that operate with these HCs present better energy performance when compared to the synthetic fluids currently used.

3. Working Methodology

For the development of the experiment, a test bench was built with a Chiller system for air conditioning environments on the premises of the UFPA Refrigeration Laboratory to carry out applied research, consisting of a condenser unit and an evaporator unit, with a set of temperature sensors and pressure, for measuring all local psychrometric parameters and during tests.

3.1. Test Bench

For the experiment, an Elgin condensing unit with 1.5 hp of power, with a capacity of 1.6 kW of refrigeration for the R-22 fluid, was used to cool an insulated container with a volume of water equal to 40 liters, as shown in Figure 1, a water pump with a flow rate of approximately 6.0 l/min and a forced air heat exchanger for circulating water at low temperatures to simulate thermal load.



Figure 1. Evaporator unit of the Chiller system test bench.

The condensing unit used was from the Elgin brand, as shown in Figure 2, model TUM-2053 E, for medium evaporation temperatures, voltage of 220 V, single-phase, with a maximum capacity of 2,325 W of refrigeration and electrical consumption of 1,400 W at 0 °C evaporation.



Figure 2. Condenser unit of the Chiller system test bench.

The water cooler evaporator has the equivalent length of 15 meters of 5/8" internal diameter copper piping, without fins, as shown in Figure 3.



Figure 3. Cooling system of water-cooled evaporator.

The circulation system shown in Figure 4 for the secondary evaporator consists of a 1/3 cv water pump with a fixed flow rate of 6.0 l/min.



Figure 4. Water circulation system of the water-cooled evaporator.

To exchange the heat of the secondary fluid with the ambient air, a split-type evaporator unit was used, initially sized for refrigerant fluid and with a nominal capacity of 1.6 kW, with an air flow of 350 m³/h.

3.2. Refrigeration Cycle

In the water-cooling system, it is carried out by the mechanical vapor compression refrigeration system, shown in Figure 5, the evaporator coil is submerged in the water reservoir, in which fluid circulates at low

pressure and temperature in a mixed state, absorbing the heat from the liquid while the liquid is circulated by a centrifugal pump to a heat exchanger between air and water to obtain an air-conditioned environment with lower temperature and humidity. The first part of the experiment was testing the R-22 fluid in the simulator; no adaptation to the condensing unit was necessary as the equipment was designed to work with this fluid.

All fluid that was initially in the condensing unit was collected and treated, after which the main and secondary fluid evaporator was installed, pressurizing the entire system with dry nitrogen for 24 hours at the end of the procedure to ensure that there were no leaks.

The test bench set up for the experiment can be represented by the refrigeration cycle diagram shown in Figure 5, for the ambient temperature and pressure, of the place to be air-conditioned, that is, T_0 and p_0 :

With the verification completed, the system evacuation process began to remove moisture and other non-condensable gases that could cause damage to the system and consequent loss of performance.

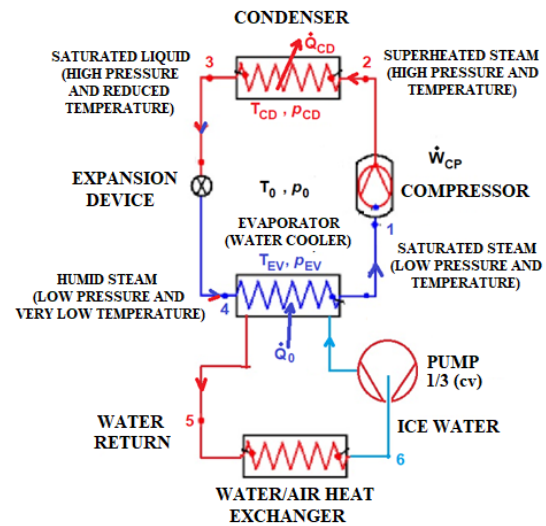


Figure 5. Schematic of the water-cooling system.

For the cooling test, the R-22 load used was 1.2 kg, which was the load necessary until the system's liquid sight glass showed that only fluid in the liquid phase existed at the inlet of the expansion device.

After collecting the synthetic fluid, the system was completely cleaned and 0.680 kg of R-290 was loaded. The procedure for operating the system with R-290 fluid was the same as R-22 [7].

3.3. System Data Collection

Data collection on the constructed test bench was carried out at intervals of 10 minutes, for a total period equivalent to 150 minutes, where the following information was collected:

- Heat exchanger air inlet and outlet temperatures,
- Electric current and voltage during operation.
- Suction, discharge, expansion device inlet and outlet pressures and evaporator coil inlet and outlet pressures,

- Temperatures of suction, discharge, compressor body, inlet and outlet air passing through the condenser,
- Temperatures of the chilled water reservoir and exchanger return,

From these data, the following were calculated: the mass flow; the real power consumed by the system; overheating; subcooling; the cooling capacity of the system; the cooling effect; the COP; the work of the compressor; the capacity of the evaporator; the capacity of the condenser; exergies and exergy efficiency.

3.4. Thermodynamic Analysis of the Basic Refrigeration Cycle

Applying steady-state conditions to the energy equations for each stage of the cycle, shown in Figure 5, the following parameters were obtained: [7]

$$W_{CP} = \dot{m}_f \cdot (h_1 - h_2) \quad (1)$$

Where Equation (1) represents the work performed by the compressor, in kW, in an isentropic process, during its operation, with \dot{m}_f being the mass flow of refrigerant fluid, in (kg/s) and h_1 is the value of the enthalpy at the inlet of the compressor, in (kJ/kg) and h_2 is the enthalpy at the compressor outlet, in (kJ/kg).

Equation (2) represents the rate of amount of heat rejected in the condenser – \dot{Q}_{CD} , in kW, where h_2 is the value of the enthalpy at the condenser inlet, in (kJ/kg) and h_3 is the enthalpy at the condenser outlet, in (kJ/kg), the unit of the amount of heat [2 and 7].

For [1] the mass flow rate – \dot{m}_f of refrigerant fluid, in kg/s, can be calculated by Equation (3):

$$\dot{Q}_{CD} = \dot{m}_f \cdot (h_2 - h_3) \quad (2)$$

Where: $\dot{Q}_0 = \dot{Q}_{EV}$ represents the refrigeration capacity of the cycle, in (kW), and this value can be calculated by Equation (4):

$$\dot{m}_f = \frac{\dot{Q}_0}{(h_1 - h_4)} \quad (3)$$

Equation (4) represents the rate of quantity of heat absorbed in the evaporator, h_4 is the enthalpy value at the evaporator inlet, h_1 is the enthalpy value at the evaporator outlet, \dot{m}_L mass flow of water from the cooler, in kg/s, h_5 is the heat exchanger inlet enthalpy and h_6 is the heat exchanger outlet enthalpy.

$$\dot{Q}_{41} = \dot{Q}_0 = \dot{Q}_{EV} = \dot{m}_f \cdot (h_1 - h_4) = \dot{m}_L \cdot (h_6 - h_5) \quad (4)$$

The energy of the expansion device is obtained by an isenthalpic process, that is, $h_3 = h_4$.

Equation (5) represents the efficiency of the ideal cycle, that is, efficiency of the First Law of Thermodynamics [2].

$$COP_{1^\circ Law} = \frac{\dot{Q}_0}{\dot{W}_{CP}} = \frac{\dot{m}_L \cdot (h_5 - h_6)}{\dot{m}_f \cdot (h_2 - h_1)} \quad (5)$$

However, the real system is different from the ideal, so the theoretical system was idealized, starting from an adaptation of the Carnot cycle to real conditions, namely:

- Modification of the compressor entry point, which changed from admitting the fluid still in the mixing phase to a more advanced point where the fluid is already completely saturated vapor, due to the constructive characteristic of the compressors used since the liquid phase is practically incompressible;
- During compression, even if it occurs isentropically, there is an increase in the temperature of the fluid, thus the efficiency of the cycle is also lower than that observed in the Carnot cycle, and
- The isentropic expansion process is ideal, but is not used in reality, due to the high cost of components. These are just some modifications that make the theoretical cycle different from the Carnot cycle and justify its lower efficiency.

3.5. Exergetic Analysis of the Basic Refrigeration Cycle

For [8] energy analysis evaluates how energy is used in the system, based on the first law of thermodynamics, however, there is no accounting for spontaneous processes and irreversibilities that may occur during the operation of the system, such as the balance of energy does not consider the way in which energy is degraded. At this point, exergy analysis is considered, which complements the system's energy information by using the second law of thermodynamics as a basis, that is, it takes into account the entropy in each process, allowing the calculation of the real efficiency of the equipment, informing the amount of work that can be performed by the system.

Exergy efficiency in one of its many definitions is the following: “is the ability to produce the desired effect without loss or with minimal waste of the resource used”. Being calculated by Equation (6): [8]

$$\psi = \frac{\text{Generated exergy}}{\text{Exergy supplied}} \quad (6)$$

Where ψ is the exergy generated and the value in the analyzed component and the exergy supplied is the total value of exergy that the system obtained for operation.

The exergy analysis carried out by [9] applied to a refrigeration system is calculated for each component separately. The specific exergy – e_x , in kJ/kg, of the system components was obtained using Equation (7):

$$e_x = (h_x - h_0) - T_0 \cdot (s_x - s_0) \quad (7)$$

Where: h_x and s_x are, respectively, the enthalpy and entropy at the observed point, h_0 and s_0 are, respectively, the enthalpy and entropy in relation to the ambient

temperature and T_0 is the temperature of the refrigerated environment [2 and 7].

According to [1] for the evaporator, exergy relates the entry and exit of fluids in it, that is, $\dot{m}_L = \dot{m}_5 = \dot{m}_6$ and $\dot{m}_f = \dot{m}_1 = \dot{m}_4$, following the numerical sequence in Figure 5, we arrive at Equations (8) and (9):

$$E_{EV,Ent} = \dot{m}_f \cdot (e_1 - e_4) - \dot{Q}_0 \cdot \left(1 - \frac{T_0}{T_{EV}}\right) \quad (8)$$

or

$$E_{EV,Q} = \dot{Q}_0 \cdot \left(1 - \frac{T_0}{T_{EV}}\right) - \dot{m}_L \cdot (e_5 - e_6) \quad (9)$$

Where, $\dot{m}_L \cdot (e_5 - e_6)$ represents the water vapor exergy, T_0 is the ambient temperature and T_{EV} is the evaporation temperature, both in (K).

The irreversibility of the evaporator – I_{EV} , in kW, can be calculated by Equation (10):

$$I_{EV} = T_0 \cdot \left[(\dot{m}_L \cdot s_5 - \dot{m}_f \cdot s_1) - (\dot{m}_f \cdot s_4 - \dot{m}_L \cdot s_6) \right] \quad (10)$$

Next, the component evaluated was the compressor, with the total exergy of the compressor being defined – E_{CP} , in kW, is obtained by Equation (11):

$$E_{CP} = \dot{m}_f \cdot (e_2 - e_1) - W_{CP} \quad (11)$$

Where, $e_2 - e_1$ is the difference between the specific input and output exergies of the compressor, W_{CP} is the electrical power consumed by the compressor, in kW.

For the refrigeration cycle condenser, the exergy equation – E_{CD} is obtained by Equation (12):

$$E_{CD} = \dot{m}_f \cdot (e_2 - e_3) + \dot{Q}_{CD} \cdot \left(1 - \frac{T_0}{T_{CD}}\right) \quad (12)$$

Where: \dot{Q}_{CD} is the thermal power rejected in the condenser, in (kW) and T_{CD} is the condensation temperature, in (K).

And for the expansion device, since there is no heat generation or power consumption, we have Equation (13) for total exergy of the expansion device – E_{DE} :

$$E_{DE} = \dot{m}_f \cdot (e_4 - e_3) \quad (13)$$

Therefore, using Equation (14) the exergy efficiency can be calculated – η_{Ex} applied to a refrigeration system can be calculated as: [2, 7 and 8].

$$\eta_{Ex} = \eta_{2^\circ Law} = COP_{1^\circ Law} \cdot \left(1 - \frac{T_0}{T_{CD}}\right) = \frac{\dot{m}_L \cdot (e_5 - e_6)}{\dot{m}_f \cdot (e_2 - e_1)} \quad (14)$$

In this way, by applying the data obtained during the tests to these equations, information related mainly to the effective performance of the equipment can be obtained.

3.6. Results

Two tests were carried out on the refrigeration system with the R-22 fluid, the first at a frequency of 60 Hz, two tests with the R-290 fluid at a frequency of 60 Hz and two tests with the R-290 fluid at a frequency of 70 Hz, all lasting 150 minutes in order to observe how the system behaves according to ambient conditions, the operation before and after the chilled water circulation pump was compared.

Figure 6 shows the variation in suction pressure for the two fluids and for the frequencies evaluated.

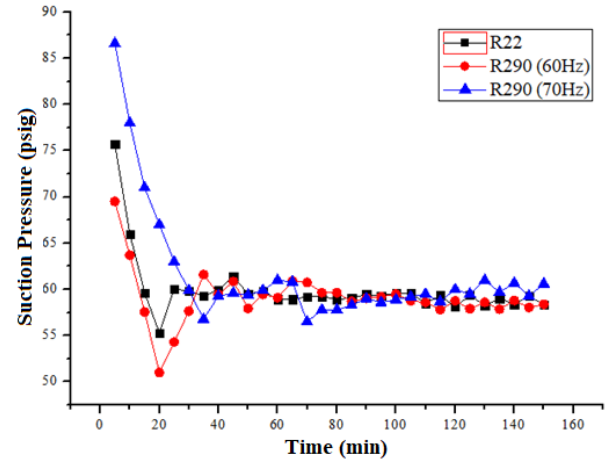


Figure 6. Refrigerant fluid suction pressures over time.

For both the 60 Hz and 70 Hz frequencies, in Figure 6, there was a variation between pressures of 57 to 61 psi, and for R-22, the variation with temperature stabilization was between pressures of 58.5 to 59.5.

The main cause of this event is linked to the fact that the thermostatic expansion valve works with a fluid in the bulb calibrated to the R-22 response, and because R-290 absorbs more energy per mass, the device was unable to maintain a continuous flow, which ended up harming the temperature reduction, since the evaporation temperature suffered directly from this fluctuation in values.

Figure 7 shows the graph of variation of compressor discharge pressure over time.

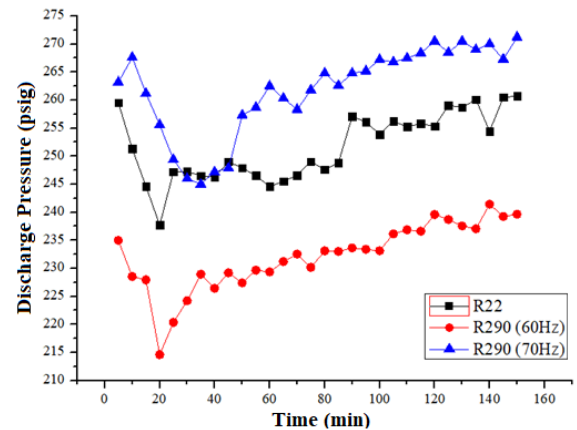


Figure 7. Refrigerant fluid discharge pressures over time.

It can be seen in Figure 7 that for the frequency of 60 Hz the lowest pressure of R-290, the average pressure value was 237 psig, which shows an advantage over R-22 which had an average pressure of 255 psig, indicating a reduction of the compression ratio, since the suction pressure values were similar, with the reduction in discharge pressure, the mechanical efforts on the compressor are reduced in addition to the reduction in electrical energy consumption.

For the frequency of 70 Hz, the value of the average discharge pressure of the refrigerant R-290 was 265 psig, exceeding that of R-22, this fact is associated with the compressor operating outside the working speed, which is fixed at 60 Hz, which caused an increase in temperature in general and thus an increase in condensation pressure.

The graph in Figure 8 shows how the mixture saturation temperature varied during the tests.

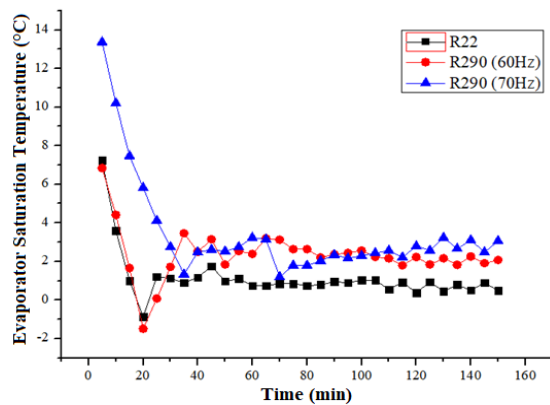


Figure 8. Saturation temperature in the evaporator.

As seen in the graph in Figure 8, the Chiller system running with R-22 fluid, with the compressor for R-22 and the expansion device calibrated for R-22, the lowest evaporation temperatures were reached, with values varying minimally around 0.5 °C, while the system, after changing the R-22 fluid to R-290, was unable to reduce the evaporation temperature below 2 °C, even with the increase in volumetric displacement caused by the operating frequency of 70 Hz.

Figure 9 shows the saturation temperature curve of the fluids analyzed in the condenser.

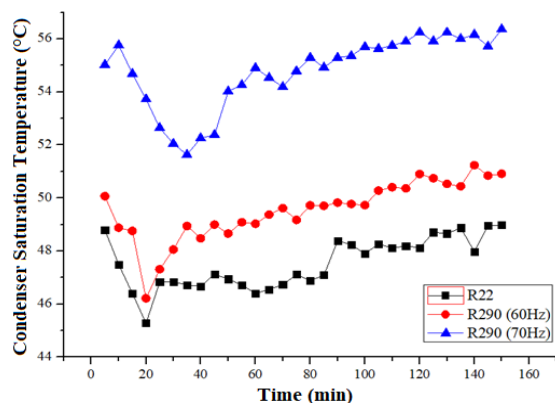


Figure 9. Saturation temperature in the condenser.

It can be seen in Figure 9 that for the frequency of 60 Hz, in the same way that the saturation temperature in evaporation was high compared to R-22, the fact is repeated for the natural fluid now in condensation, with temperatures of saturated fluid varying at values close to 50 °C, for R-22, the temperature was close to 48 °C even though the pressure at this point was above the values collected for R-290.

Figure 10 lists the compressor power consumption for the two types of refrigerant fluids and at different frequencies.

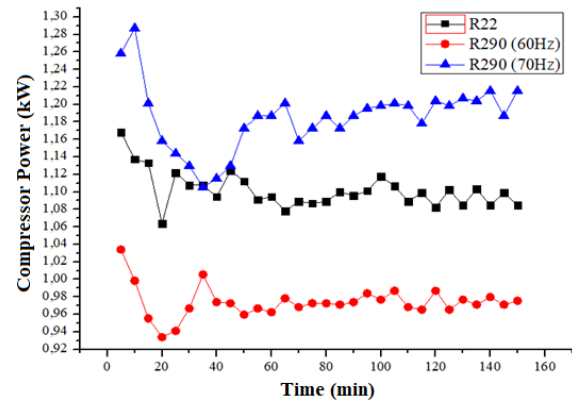


Figure 10. Electrical power consumed by the compressor.

The highest saturation temperature depends only on the chemical composition of the fluid. For the 70 Hz frequency, the temperature values are close to 55 °C, once again indicating non-standard compressor operation, confirmed by the increase in electrical energy consumption. However, the higher condensation temperature indicated the greater temperature difference between the R-290 fluid and the external air in the heat exchange in the condenser, facilitating the conversion of the fluid from a superheated vapor state to liquid at high pressure at the condenser exit. [10]

The consumption recorded for R-22 served as a reference for the evaluation of R-290, which for the 60 Hz frequency was lower, however, as explained in Figures 8 and 9, the saturation temperature during evaporation was higher than that of R-22, so consumption was lower due to a lower capacity to absorb energy while the system was running on natural fluid.

For the frequency of 70 Hz, the consumption is well above the reference value for the R-22 fluid, which agrees with the fact that the single-phase electric motor, when forced to work outside the nominal frequency, causes excessive energy losses in the form of heat, observed in the increase in electricity consumption of the refrigeration system.

Figure 11 shows the calculated refrigerant mass flow rate of the refrigeration cycle for different fluids and operating frequencies.

It can be seen that the values calculated for the R-22 fluid match the absorbed thermal load, compared to R-290, the average specific mass of the synthetic refrigerant was 21.61 kg/m³, while the specific mass of the average natural fluid was 11.02 kg/m³ for the 60 Hz frequency

and 11.17 kg/m^3 for the 70 Hz frequency, which explains the values close to 0.0115 kg/s for R-22 circulating in the system and 0.0057 kg/s for the natural fluid. These values present a significant difference in terms of the amount of refrigerant fluid to be used within the refrigeration system. [11]

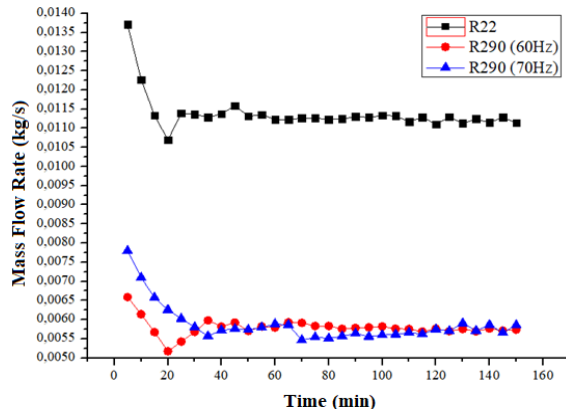


Figure 11. Refrigerant mass flow rate.

The main point of energy analysis is the calculation of the COP, which is the amount of power absorbed in the evaporator over the electrical power consumed by the equipment during operation. For the evaluated system, the COPs were calculated for the three tests and the data were plotted on the graph in Figure 12.

In the steady state, the calculated COP for the R-22 was 3.491, for the R-290, with a frequency of 60 Hz it was 3.570, an increase of 2.26% and 3.344 for the frequency of 70 Hz, a reduction of 4.21%, respectively, when compared to the synthetic fluid. Again indicating that the increase in frequency made the compressor less efficient, even with the increase in volumetric displacement.

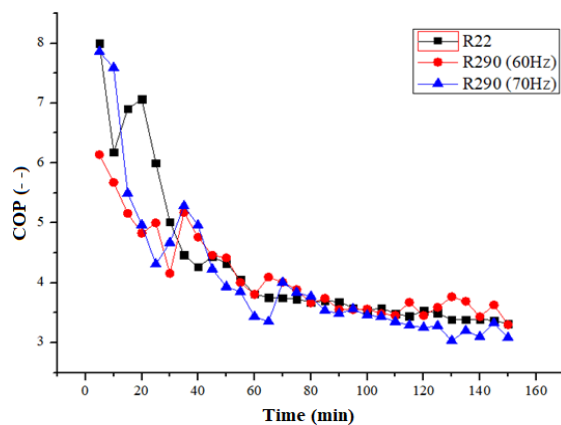


Figure 12. System performance coefficient during tests.

It is important to remember that even though R-290 at 60 Hz has a COP value greater than that of R-22, this difference was due to the fact that the evaporation temperature of R-290 was higher, where the average of R-290 was 2.2°C and R-22 averaged 0.8°C .

In addition to the energy analysis, the exergy factors and irreversibilities for the system were calculated as a way of complementing the explanations of the system's

characteristics and mainly the negative points that occurred when increasing the operating frequency from 60 Hz to 70 Hz.

It is known that some authors such as [5] and [12], mention that the difference in viscosity, being much smaller in R-290 compared to R-22, can cause a more efficient heat exchange in the evaporator, which causes a reduction in the entropy of the system and reduces the final exergy value.

Figure 13 shows the results of the exergy analyzes calculated for the refrigeration system evaporator.

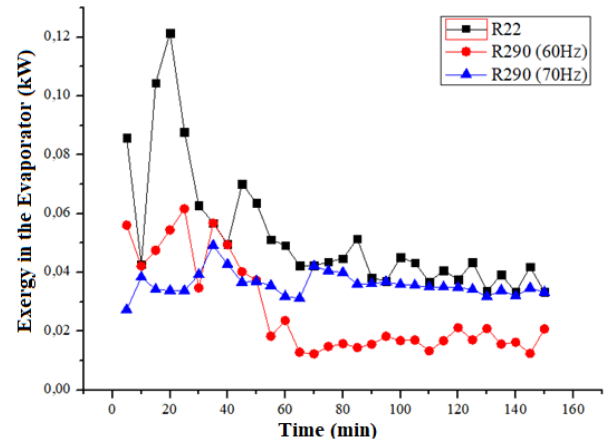


Figure 13. Calculated exergy for the evaporator.

For the evaporator, the calculated values showed little difference between R-22 and R-290 at 70 Hz, being in the range of 0.04 kW , however the values for R-290 at 60 Hz were unexpected, since the value was almost 4 times lower, as shown in the graph in Figure 13.

Figure 14 shows the results of the exergy analyzes calculated for the refrigeration system condenser.

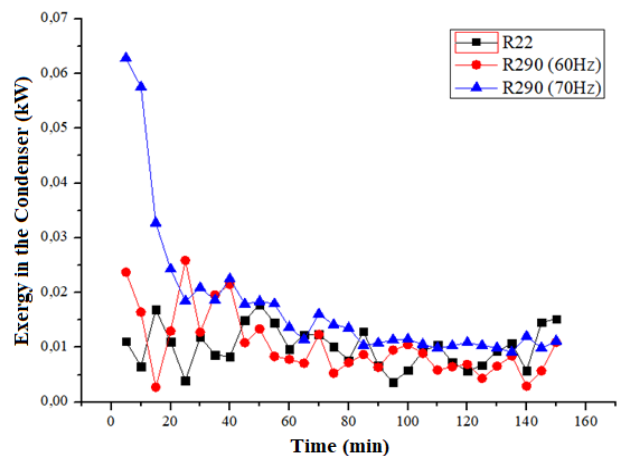


Figure 14. Calculated exergy for the condenser.

For the condenser, the average calculated exergy values for both R-22 and R-290 at 60 Hz were close to 0.007 kW , with little difference for the exergy value for R-290 at 70 Hz that increased to 0.013 kW .

As the condenser is a component in which the variation in availability generated a low irreversibility value, that is, it was not a piece of equipment that contributed to the

increase in electrical energy consumption during the tests.

One of the advantages of exergy analysis over energy analysis is exactly the consideration of the variation in entropy in each component, as shown in Figure 15, which presents the variation in exergy calculated for the component with the highest energy consumption of the entire system, the compressor.

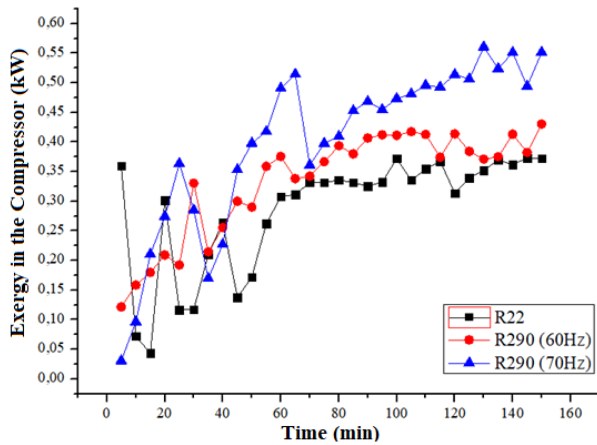


Figure 15. Calculated compressor exergy.

In addition to the energy analysis, Figure 15, the exergy factors and irreversibilities for the system were calculated, as a way of complementing the explanations of the system's characteristics and mainly the negative points that occurred when increasing the frequency operating from 60 Hz to 70 Hz.

The exergy analysis also provides the system efficiency factor, making a relationship between the availability in the evaporator and the total electricity consumption of the equipment, as shown in the graph in Figure 16. [13]

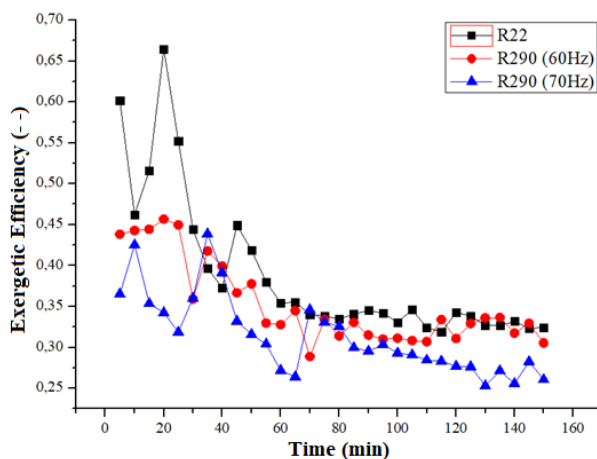


Figure 16. Exergetic efficiency of the system.

In the steady state, as shown in Figure 16, the exergy efficiency values were 34% for R-22 and 32% for R-290 at 60 Hz, while the efficiency for R-290 at 70 Hz was 27%.

Table 2 summarizes the main properties related to the use of energy by the system.

Table 2 Summary of exergy properties.

Thermodynamic Properties	Refrigerant Fluids		
	R-290 (60 Hz)	R-290 (70 Hz)	R-22
Exergy in the compressor (kW)	0,37	0,52	0,33
Exergy in the evaporator (kW)	0,02	0,04	0,04
Exergy in the expansion device (kW)	0,10	0,13	0,16
Exergy in the condenser (kW)	0,007	0,013	0,007
COP	3,570	3,344	3,491
Exergetic Efficiency (%)	32	27	34

The observation focuses on the following detail, the pressure and temperature values for R-290 at 60 and 70 Hz had a difference of 2%, so the temperature of the chilled water was the same for both tests, which contributes to the idea that there was a drop in the electrical efficiency of the engine being driven by the frequency inverter, confirmed by the increase in condensing pressure and power consumption.

4. Conclusions

Taking into account the psychrometric conditions of the UFPA Refrigeration Laboratory, it is possible to state that the tests to obtain data and the development of energy and exergy analyzes were satisfactory, as they presented the differences and similarities between the two types of refrigerant fluids analyzed, providing subsidies and conditions for replacing R-22 fluid with R-290.

The statement made above meets the needs listed by the national and international market, as the refrigerant fluid R-22 was already discontinued in 2015, even so, in Brazil, this is the fluid most used in air conditioning systems and, its As a substitute, the R-410a fluid, as is known, will be discontinued in 2024, that is, it will be necessary to use a refrigerant fluid to replace the fluids currently used.

Even so, after completing the experiments and obtaining the calculations and graphs of the refrigeration system components, some characteristics can be highlighted:

- The use of flammable fluid, when operated by responsible and instructed technicians, did not generate the slightest concern during operation.
- The use of natural refrigerant in a system originally designed for use with synthetic refrigerant resulted in a reduction in evaporator power of 2%, however, it increased COP by 3%.
- R-290 at a frequency of 60 Hz obtained an exergy efficiency only 2% lower than R-22 operating at 60 Hz.

As it is a flammable fluid, there are risks associated with the uncontrolled use of R-290, which are minimized mainly by qualifying refrigeration technicians to respect

the rules necessary to carry out maintenance and monitoring of this type of system.

In this way, this work adds to several others carried out, agreeing that the use of the natural fluid R-290 is a refrigerant with excellent thermodynamic characteristics for application in mechanical vapor compression refrigeration systems (mainly at low condensation saturation pressure).), high enthalpy (which helps to reduce the amount of mass required in the system), compatibility with mineral lubricating fluid (eliminating the use of highly hygroscopic synthetic lubricating fluids) and, as made clear in this work, the reduction of electrical energy consumption.

According to the thermodynamic analysis developed for hydrocarbon fluids, it can be observed:

- That natural fluids reduce the pressure levels developed in the condenser and evaporator;
- The use of R-290 fluid and mixtures involving hydrocarbons provides a tripling of the latent heat of vaporization in relation to R-22. This factor leads to a reduction of around 50% in the need for mass refrigerant charge in the refrigeration system for the same equipment capacity.
- Hydrocarbons have lower compressor discharge temperatures, providing longer useful life for these components.
- The coefficient of performance of the system with hydrocarbons and mixtures showed an increase of around 5% in relation to R-22. Consequently, less compression work is required for hydrocarbons in relation to R-22 due to their thermophysical properties related to density in the liquid and vapor phases.

Analyzing the results of the exergy analysis carried out in the residential Chiller system, it is important to realize that the R-290 fluid, operating at a frequency of 60 Hz, presents itself as being the best condition to be used as a replacement fluid for R-22, in refrigeration and air conditioning systems, making it necessary to observe a reduction in the quantity of fluid, in mass, to be placed in the systems, in addition to the temperature and pressure conditions of evaporation and condensation, including the subcooling and superheating processes during installation of systems.

Thus, it is possible to use the existing compressor platform for synthetic refrigerant fluids with very few modifications for operation with natural fluid, reducing the cost of research and development, in addition to reducing the use of oils with a complex chemical composition that is quite aggressive to the environment and mainly the focus on reducing electricity consumption, which brings together the objectives of always benefiting the Environment, Sustainability and the rational use of electricity seeking to reduce pollutant emissions.

5. Acknowledgment

The authors would like to thank the Faculty of Mechanical Engineering – FEM, the Institute of Technology – ITEC and the Postgraduate Program – PPGEM at the Federal University of Pará – UFPA.

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