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ENERGY AND EXERGY ANALYSIS FOR A VRF SYSTEM FOR A RESIDENTIAL BUILDING

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Abstract. Energy efficiency can be the solution for minimizing emissions that negatively impact the environment and minimizing natural resource consumption. VRF (Variable Refrigerant Flow) are systems that can perform at a high capacity with several indoor units interconnected with few outdoor units. Therefore, this work aims to the analyze an air conditioning system with variable refrigerant volume (VRF) for the installations of a residential building in Teresina, Brazil. An energetic, an exergetic analysis and an economic analysis are performed. The selection of equipment for the system through catalogues and software from manufacturers was carried out. Exergetic analysis is applied to identify which component of the system most destroys exergy and, thus, make the relevant changes in that component to conserve more energy. An economic analysis was carried out considering the operation costs. It was observed that the condenser has the greatest destroyed exergy and the compressor the best exergy efficiency.

Keywords: energy efficiency, HVAC systems, cooling.

1. INTRODUCTION

The world is focused on reducing global emissions to control climate change. The energy sector has a huge responsibility to make this possible. It is estimated that the residential sector demands 20% of global energy (IEA, 2021). In countries with a tropical climate, such as Brazil, air conditioning is a necessity for thermal comfort. HVAC (Heating, Ventilation, Air Conditioning) systems have a great contribution of energy consumption compared to other equipment in a house. Therefore, the energy efficiency of these systems is essential for reduce the energy demand.

Variable Refrigerant Flow (VRF) system are a direct expansion system which has several indoor units connected to one outdoor unit with a high energy efficiency (Ozahi *et al.*, 2017). These are systems that work with variable speed compressors and electronic expansion valves that control the flow of refrigerant to the evaporator units (ASHRAE, 2010). In Brazil, these systems have been gaining more space in commercial buildings, but they are not very commonly used in residential buildings.

Duarte *et al.* (2020) carried out a study comparing the performance of a VRF system and a VAV for a commercial building in Florianópolis, Brazil. The authors found that, despite the VAV system having a higher COP, the energy consumption of the VRF is lower, mainly due to the behavior at partial loads. Qian *et al.* (2021) analyzed the VRF usage profile for different building types in 5 different regions of China. The energy consumption, the purpose of the equipment, operating hours and the behavior of the ICOP were verified. The authors noted that in most regions, VRF was used for air conditioning. Furthermore, with this study, the authors were able to identify the pattern of use of these equipments.

Devecioglu and Oruç (2020) made an investigation to use R466A, a low GWP fluid, in the place of R410A in a VRF system. The study considered three evaporation temperatures and three condensation temperatures. The authors verified that the system with R466A presented a better energy performance than the R410A system both for cooling and heating. Recently, Hernandez and Fumo (2019) carried out a review of VRF systems for residential applications. The study approached aspects such as capacities, types of refrigerant fluid, energy performance and thermal comfort. The authors showed that there are still modifications to be made in the VRF system, in piping, in the electronic valves, for example, so that these systems are more feasible in homes.

Saab *et al.* (2018) studied the VRF systems for humid environment with many refrigerants. The authors applied not only energy analysis but also exergetic analysis. The most efficient fluids were R410A and ammonia. And the component with the greatest destroyed exergy was the compressor. Santos (2018) presented an exergy analysis for the VRF system through the exergetic efficiency for the whole system using differents refrigerants. To substitute R410A the HFOs showed promising results.

Mayolino (2015) performed an exergy analysis for window-type air conditioning equipment for residential use. The author observed which factors could reduce the destruction of exergy from the evaporator, a component with greater exergy destroyed and identified that the air flow has a great influence on this parameter. Caliskan *et al.* (2011) analyzed

the exergy results for three conventional systems varying the dead state temperature. The authors noticed the difference in specific exergies and in exergies destroyed due to this variation.

The recent studies show a variation in the energy performance of VRF systems according to the climate and the use profile of the building. Therefore, it is important to carry out a study of a VRF system in a city with a hot and dry climate, such as Teresina, Brazil. Exergy analysis is also an important tool, because through the destroyed exergies it is possible to perceive which equipment and which conditions need to be optimized. This work aims to analyze the energy and exergy performance of a VRF-type air conditioner for a residential building in Teresina, Brazil.

2. SYSTEM ANALYSIS

The residential building has eight apartments and twelve floors. It was considered that each apartment has four rooms to be air conditioned, three bedrooms and a living room. The thermal load has already been calculated. As VRF systems can have many evaporator units connected to a single condenser unit, figure 1, the system modeling was done for only one typical floor of the building, which can be replicated to the other floors.
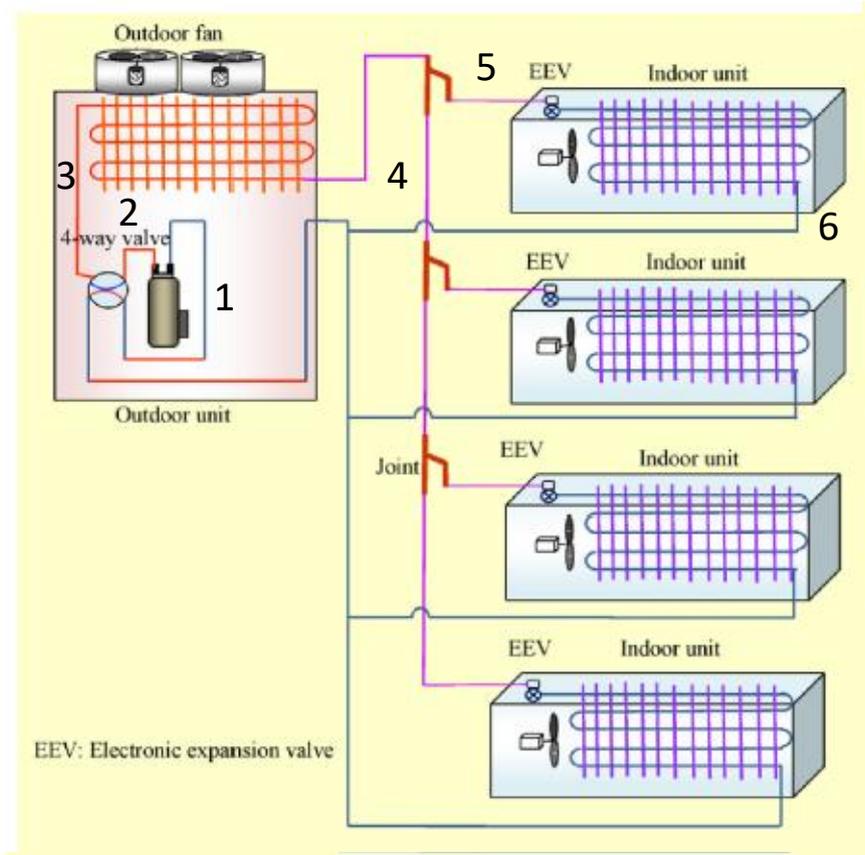

Figure 1 - VRF equipment operation scheme (Adapted from Devecioglu and Oruç, 2020).

The calculation and selection of VRF equipment must be performed using software provided by the manufacturers. For this study, the manufacturer Trane was chosen.

With the equipment defined and based on the data provided by the selection software, the first law of thermodynamics for control volume is applied to determine the states of the thermodynamic cycle, Eq. (1), Eq. (2) e Eq. (3).

$$\dot{Q}_{evap} = \dot{m}_{evap}(h_5 - h_6) \tag{1}$$

$$\dot{Q}_{cond} = \dot{m}_{cond}(h_3 - h_4) \tag{2}$$

$$\dot{W}_{comp} = \dot{m}_{cond}(h_2 - h_1) \tag{3}$$

Where $\dot{m}$ is the mass flow rate, h is the enthalpy, $\dot{Q}$ the heat load and $\dot{W}$ the power. The COP and the EER is defined by the manufacturer. It is important to note that the mass flow from the evaporator is not the same as for the condenser.

The exergy is a property that can be defined as the maximum work that could be obtained from a system at a specified state (Moran *et al.*, 2016). For this property to be calculated, it is necessary to define the thermodynamic state and a dead state, that is, a reference state with pattern conditions for the work fluid. In this study, only the physical exergy, Eq. 4, was considered.

$$ex = (h - h_0) - T_0(s - s_0) \tag{4}$$

s is the entropy. The subscript 0 refers to the dead state. The dead state for the R140A fluid was considered atmospheric pressure (101.3 kPa) and a temperature of 0ºC. For air, the reference temperature was established as 20°C. Exergy, unlike energy, can be destroyed. For the main components of the cycle, this value must be determined (Eq.5, Eq. 6, Eq. 7). The exergy efficiency, Eq. 8, Eq. 9, Eq. 10, can be determined as well.

$$Ex_{dest,evap} = \left(1 - \frac{T_{0,air}}{T_a}\right) * \dot{Q}_{evap} + \dot{m}_{evap} * (ex_5 - ex_6) \tag{5}$$

$$Ex_{dest,cond} = -\left(1 - \frac{T_{0,air}}{T_a}\right) * \dot{Q}_{cond} + \dot{m}_{cond} * (ex_3 - ex_4) \tag{6}$$

$$Ex_{dest,comp} = \dot{W} + \dot{m}_{cond} * (ex_1 - ex_2) \tag{7}$$

$$\eta_{comp} = \frac{\dot{m}_{cond} * (ex_1 - ex_2)}{\dot{W}} \tag{8}$$

$$\eta_{cond} = \frac{\left(1 - \frac{T_{0,air}}{T_a}\right) * \dot{Q}_{cond}}{\dot{m}_{cond} * (ex_3 - ex_4)} \tag{9}$$

$$\eta_{evap} = \frac{\dot{m}_{evap} * (ex_5 - ex_6)}{\left(1 - \frac{T_{0,air}}{T_a}\right) * \dot{Q}_{evap}} \tag{10}$$

Where T_a is the ambient temperature at which the heat transfer occurs. To calculate the operating cost, the hours of operation of the equipment were estimated and their nominal power was considered. For this, a constant load profile was considered for the hottest month of the year in the city (October). The operating cost can be calculated by Eq. 11.

$$C_{op} = hr * \sum P * T \tag{11}$$

C_{op} is the operating cost for a month, *hr* the number of operating hours, *P* the power of the equipment and *T* the electricity tariff. There are some parameters that had to be assumed for the beginning of the analysis. For this, the software Coolpack had been used. For the analysis, the EES software has be chosen.

3. RESULTS AND DISCUSSION

In this work, with the value of the thermal load already calculated, the capacity of the evaporator units for each room of the eight apartments was selected using the VRF software from the manufacturer Trane. Table 1 shows the value of the capacities of the indoor units, their power, consumption and cost. Note that there are three capacity values given by the manufacturer. First, the nominal capacity value, which is basically for a commercial framework. Second, the maximum capacity value for which the evaporators are designed. And finally, the actual effective heat exchange capacity.

For some cases, especially for the living rooms, it was noticed that, although the nominal capacity is above, the effective capacity is below the requested thermal load. However, at this study was considered the hottest month of the year in the city of Teresina. So, dry bulb temperature for design is well above the annual average. To verify whether this equipment would achieve thermal comfort throughout the year, a detailed study of the thermal load should be carried out. This data was important to be analyzed, as it was noticed that the nominal capacity of the equipment does not always achieve the requested thermal load.

To calculate the monthly consumption, the nominal electrical power given by the manufacturer's software was considered. The number of hours of operation per day was considered 4 hours for room equipment and 10 hours for bedroom equipment. It was considered a month with 30 days. The electricity tariff was defined based on the local concessionaire, established at the value of 0.58182 R$/kWh.

Table 1 – Thermal Load and Equipment data of the indoor units for a floor of the residential building in study.

APT	ROOM	Total Thermal Load (kW)	Nominal Capacity (kW)	Capacity (kW)	Effective Capacity (kW)	Electrical Power (W)	Hours per month	Consumption of the month (kWh)	Monthly Cost (R$)
01	Living room	7.00	8.0	7.05	6.84	53	120	6.36	3.70
	Bedroom 1	2.06	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 2	2.06	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 3	3.88	4.5	3.95	3.80	19	300	5.70	3.32
02	Living room	7.00	8.0	7.05	6.84	53	120	6.36	3.70
	Bedroom 1	2.06	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 2	2.06	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 3	2.76	3.6	3.15	3.03	19	300	5.70	3.32
03	Living room	7.00	8.0	7.05	6.84	53	120	6.36	3.70
	Bedroom 1	2.06	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 2	2.06	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 3	2.76	3.6	3.15	3.03	19	300	5.70	3.32
04	Living room	7.00	8.0	7.05	6.84	53	120	6.36	3.70
	Bedroom 1	2.06	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 2	2.06	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 3	3.58	4.5	3.95	3.80	19	300	5.70	3.32
05	Living room	7.23	8.0	7.05	6.84	53	120	6.36	3.70
	Bedroom 1	2.16	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 2	2.16	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 3	3.66	4.5	3.95	3.80	19	300	5.70	3.32
06	Living room	7.23	8.0	7.05	6.84	53	120	6.36	3.70
	Bedroom 1	2.16	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 2	2.16	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 3	2.83	3.6	3.15	3.03	19	300	5.70	3.32
07	Living room	7.23	8.0	7.05	6.84	53	120	6.36	3.70
	Bedroom 1	2.16	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 2	2.16	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 3	2.83	3.6	3.15	3.03	19	300	5.70	3.32
08	Living room	7.23	8.0	7.05	6.84	53	120	6.36	3.70
	Bedroom 1	2.16	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 2	2.16	2.8	2.45	2.35	9	300	2.70	1.57
	Bedroom 3	3.98	4.5	3.95	3.80	19	300	5.70	3.32
Total per floor -		116.94	141.2	124.00	119.64	720	-	139.68	81.27

For outdoor units, the manufacturer determines the power and EER, equivalent to the COP for cooling. From this, also estimating 10 hours of operation per day, the monthly electricity consumption and its cost were calculated. It was observed that VRF systems have a very high efficiency. They produce a high cooling effect, in the case of the study

121 kW of effective capacity, using only 17.74 kW. For each floor, 32 indoor units were connected to a single outdoor unit.

Table 2 - Characteristics of outdoor units for a floor of the building in study.

	Electrical Power for cooling	EER	Hours per month	Consumption of the month (kWh)	Monthly Cost (R$)
Outdoor Unit per floor	17.74	6.92	300	5322	3096.45

With information from the manufacturer and with the help of the Coolpack software, it was possible to model a thermodynamic cycle for this system in the EES. Compressor inlet temperature (T1) was taken based on manufacturer data. The title of the mixture entering the evaporator after expansion occurred in the valve was estimated to be 0.34 based on the Coolpack. The thermodynamic properties can be seen in Table 3. The working fluid of this system is R-410A.

Table 3 - Properties of thermodynamic states of the refrigeration cycle.

State	Pressure (kPa)	Temperature (ºC)	Enthalpy (kJ/kg)	Entropy (kJ/kg.K)	Exergy (kJ/kg)
1	719	1.7	425.7	1.836	57.58
2	2726	79.3	474.6	1.868	98.02
3	2726	79.3	474.6	1.868	98.02
4	2726	43.0	271.7	1.237	67.29
5	719	-3.3	271.7	1.266	59.29
6	719	1.7	425.7	1.836	57.58

With the thermodynamic properties, the second thermodynamic law was applied and the destroyed exergies and exergetic efficiencies were determined. There are 6 control volumes: the compressor, the condenser and four evaporators, with the effective capacities of 2.35 kW, one with 3.03 kW, another with 3.80 kW and finally the 6.84 kW. These results are shown in Table 4. The component with the highest destroyed exergy and the lowest exergetic efficiency was the condenser. This means that optimization efforts must be concentrated on this equipment. Another important aspect that was observed is that evaporators with different capacities have the same exergetic efficiency. This is because the properties remain the same. The fluid's mass flow is what is change in the equipment. Therefore, the exergy destroyed is different for each equipment, increasing as the mass flow increases.

Table 4. Exergy analysis results of the proposed system.

Component	Destroyed Exergy (kW)	Exergy Efficiency
Condenser	7.5520	0.3218
Compressor	3.0870	0.8260
Evaporator 2.35 kW	0.0808	0.4745
Evaporator 3.03 kW	0.1042	0.4745
Evaporator 3.8 kW	0.1307	0.4745
Evaporator 6.84 kW	0.2353	0.4745
System	14.34	0.1574

Exergy destroyed from equipment comes at a cost. Therefore, the lower this value, the lower the expense with this equipment. Because of this, a parametric analysis is performed to assess how the destroyed exergy behaves in relation to some parameters of the thermodynamic cycle. For this study, two temperatures were chosen to assess this sensitivity of the destroyed exergy: the compressor inlet temperature and the condensation temperature. As the most relevant destroyed exergies for the system were from the compressor and the condenser, these values were analyzed.

In the ideal thermodynamic refrigeration cycle, the compressor inlet temperature is the same as the fluid evaporation temperature. However, in real cycles, the fluid is superheated, it has its temperature increased to ensure that there is no liquid in the compressor. In this study, a superheat degree of 5ºC was adopted, which is an average value for refrigeration cycles. The compressor inlet temperature will depend on the evaporation temperature and the degree of superheat. In

figure 5, it was observed that the higher this temperature, the greater the destroyed exergy, both for the compressor and for the condenser. Being for the compressor a greater variation than for the condenser.
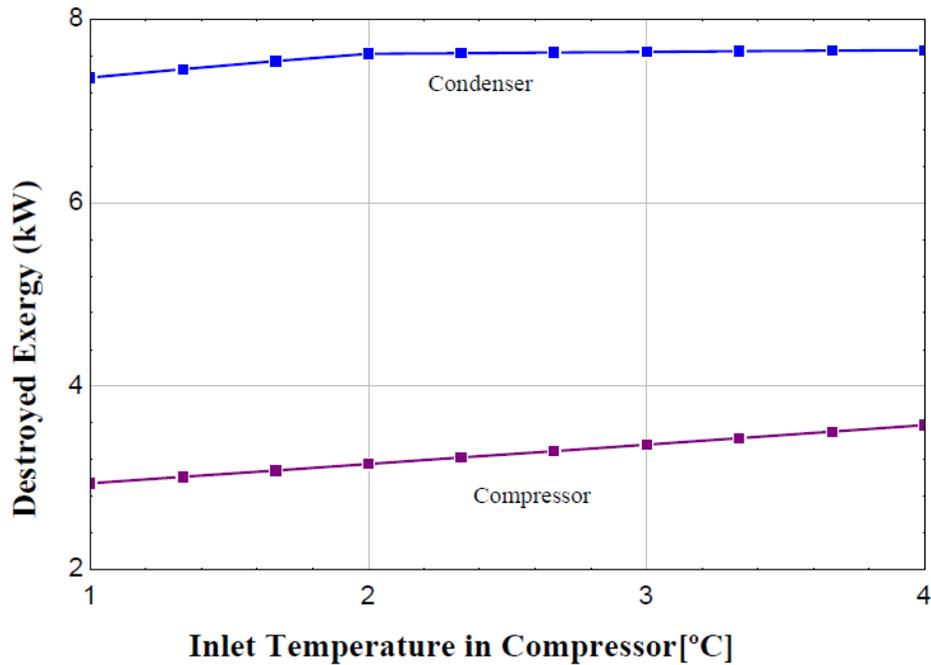

Figure 2 - Parametric Analysis among the inlet temperature in compressor and the destroyed exergy of the compressor and the condenser.

The condensation temperature depends on the temperature of the outdoor environment where the outdoor unit is located. The catalog data adopts the external dry bulb temperature as 35ºC. However, this temperature, in hot and dry places like Teresina, can reach 38ºC, which causes an increase in the system's condensation temperature. In figure 3, it can be seen that the higher the temperature, the lower the exergy destroyed by the compressor. On the other hand, the exergy destroyed from the condenser increases.
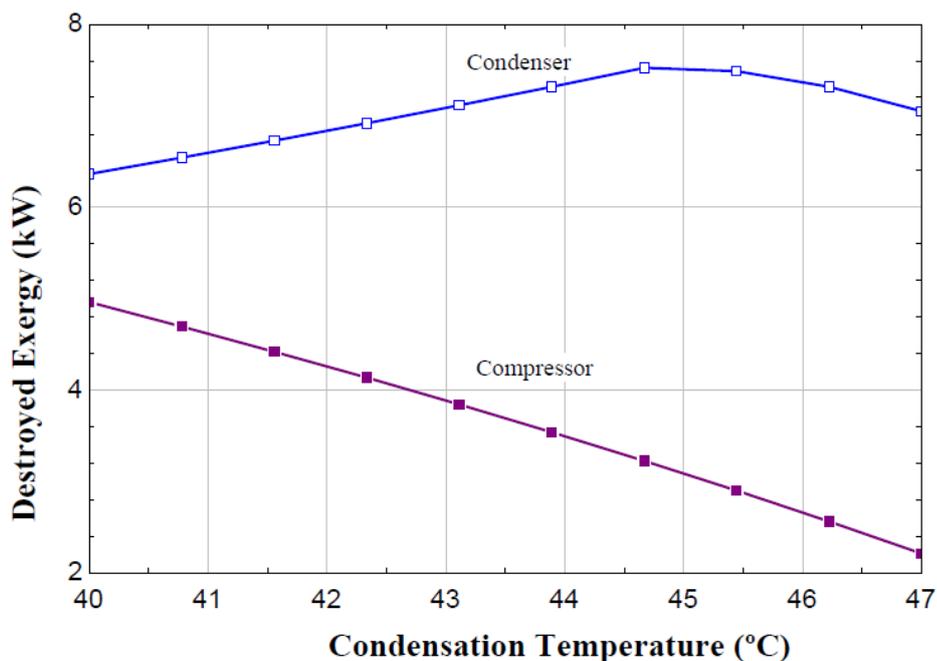

Figure 3 - Relation among the condensation temperature and the destroyed exergy for the condenser and the compressor.

Note that the curve of the destroyed exergy of the condenser increases up to a temperature of 44.6 °C and then decreases. This fact is due to the variation in the entropy of the fluid that reaches the maximum at this temperature and pressure.

Therefore, it is important to find the optimum condensing temperature point, which was estimated to be 10°C higher than the external dry bulb temperature. This control can be done by installing the condenser unit correctly, in shaded places, away from the incidence of direct solar radiation.

4. CONCLUSIONS

In this study, indoor and outdoor units of a VRF air conditioning system were selected for a residential building with eight apartments per floor in Teresina, Brazil. The exergetic analysis for the main components was carried out. The software from the manufacturer Trane was used to dimension the VRF system with 32 evaporator units connected to a condenser for a typical floor of the building. The total thermal load available for the floor was 121 kW. The operating cost of the condenser unit in a month of high thermal demand was estimated at R$ 3096.15 per floor.

The equipment with the greatest exergetic efficiency was the compressor. The smallest exergy destroyed was in the evaporators. This is due to the this individual devices have a small energy exchange compared to others. The condenser was the equipment with the greatest exergy destroyed. A parametric analysis was performed to verify the impact of compressor inlet temperature and condensation temperature on the value of exergies destroyed by the compressor and the condenser. As the compressor inlet temperature increased, both destroyed exergies increased. However, with the condensation temperature, there was an opposite behavior, wherein the exergy destroyed of the compressor decreased and the exergy destroyed of the condenser increased.

For future work, we suggest an exergoeconomic and exergoenvironmental analysis of this system and a comparison of the VRF system with other HVAC systems.

5. ACKNOWLEDGEMENTS

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