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THERMODYNAMIC MODELING OF A TWO-STAGE AIR-CONDITIONING SYSTEM WITH AN ECONOMIZER

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Abstract:
An extremely high external ambient temperature brings several issues for a proper operation of a vapor-compression refrigeration system. The decrease of the lubricating oil viscosity, reducing the compressor lifespan, and the reduction of the cooling capacity, and the increase of power consumption are key effects found in a refrigeration system when subjected to high ambient temperatures. With a focus on these aspects, this paper describes a thermodynamic modeling and comparative study between a single-stage refrigerating system and a two-stage refrigeration system with an economizer, operating at high ambient temperatures. The comparison of the coefficient of performance (COP) between these two cycles, changing external ambient temperatures and refrigerant fluids, are part of the analysis. As a baseline for this study, an air-conditioning unit for military purposes with a cooling capacity of 1.5 TR (tons of refrigeration), and using R-407c as refrigerant fluid, is applied. Moreover, other refrigerants are assessed, like R-32, R-1234yf, R-410a, and R-1234ze(E). The models consider the global thermal conductance of the heat exchangers and the total heat exchanger area is divided into corresponding fluid phases. The prediction of the latent heat transfer rate is performed using the Lewis analogy. A two-stage scroll compressor is modeled based on curve regression presented in the literature. Through the successive substitution method and the bisection method, a solution for the cycle operation is achieved under several internal and external conditions. With the analysis of the output data, a higher COP can be found for the two-stage system with economizer, regardless of the refrigerant fluid, due to the presence of the intermediate pressure, yielding a reduction of compressor power input, and an increasing of the COP and cooling capacity. These results are intended to be used as a tool for the design of efficient air conditioning systems under extreme high ambient temperatures, reducing the carbon footprint and the global warming impact of the air conditioning system.

Keywords: *Economizer, vapor-compression, air conditioning, refrigerant, high temperature.*

1. INTRODUCTION

The air-conditioning mechanism is commonly used by the military on many occasions, whether it is a war scenario, a mission in desert regions, or terrain reconnaissance. Often this cooling system is used in areas with temperatures far different from the nominal working temperatures, and implementing the cooling system in areas with extreme climatic conditions compromises its operation and hinders the development of the military's activities due to the discomfort that high temperatures cause, the president of ASBRAV (2019), Eduardo Hugo Müller states that an environment with a suitable temperature provides an increase in productivity and as a consequence improves the service provided.

Refrigeration systems that work in single-stage vapor compression cycles operating in extreme ambient temperature conditions have their efficiency drastically reduced due to the large difference between the evaporation and condensation temperature. These extreme conditions favor the increase of the compressor discharge temperature, which in turn causes serious damage to the equipment, because the increase of this temperature increases the viscosity of the lubricating oil, causing serious operating problems to the compressor moving parts, reducing its lifespan. In addition, the high ambient temperature conditions contribute to a drastic reduction in the cooling capacity and an increase in the work required by the compressor, resulting in a reduction in the system COP (Motta and Domanski, 2000).

In order to minimize these damages, one envisaged solution is the two-stage compression with an economizer, since this cycle with economizer according to the tests done by Mateu-Royo *et al.* (2020) guarantees better performance coefficients compared to the single-stage cycle, because the compressor works with a lower pressure difference when compressing from evaporation pressure to an intermediate pressure, where vapor injection occurs, and then compress again to condensation pressure.

The advantages of performing the two-stage compression with vapor injection are mainly in the increase of the cooling capacity provided by increasing the difference of enthalpies in the evaporator, and the reduction in the discharge temperature of the compressor compared to the single-stage (Tello-Oquendo *et al.*, 2016).

The present work consists in the development of a thermodynamic modeling of a single and two-stage vapor compression cycle working with the fluid R-407c, the model was validated by experimental data tabulated by Bahman and Groll (2020). The model output data will be compared on COP, cooling capacity, and discharge temperature for the two cycle models. Several simulation cases will be performed for different ambient temperatures, ranging from 35 ºC to 55 ºC, and finally, the same tests will be repeated and compared for other low-GWP (Global Warming Potencial) refrigerants.

2. MODEL PROCEDURE

The thermodynamic model consists of the basic mechanisms of an air conditioner, among them are the compressor, evaporator, condenser and the expansion device. For the two-stage cycle an expansion device and a counterflow heat exchanger (economizer) will be added. For the validation of the test model an air conditioner with cooling capacity of 5.3 kW in the nominal ambient temperature condition (35 ºC) was adopted, the working fluid used was R-407c.

The single and two-stage cycles are represented by Fig. 1 and Fig. 2, respectively, as follows

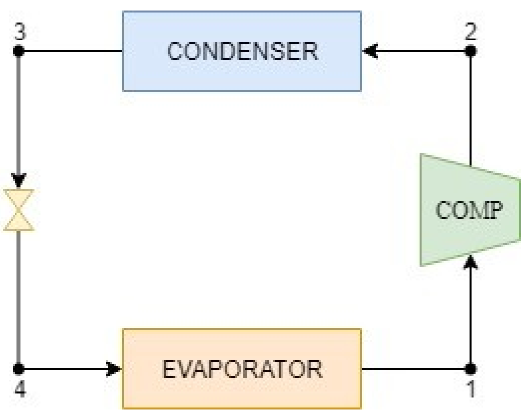

Figure 1. Single-stage cycle

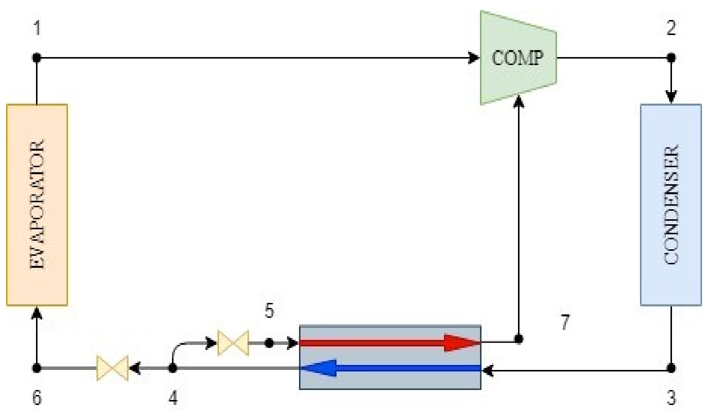

Figure 2. Two-stage cycle

2.1 Compressor

A compressor with a volumetric displacement of 17.1 m^3/h was considered for the model. The volumetric efficiency and the compressor efficiency, according to the diagram proposed by Tello-Oquendo *et al.* (2017), are 81.38% and 54.1%, respectively. For a better approximation of the real working condition, the heat loss of the compressor was considered, and the calculation of this heat loss is done according to Eq. (1) as follows

$$\dot{Q}_{loss} = UA_{comp}\left(T_{2A} - T_{amb}\right) \tag{1}$$

where $\dot{Q}_{loss}$ is the compressor heat loss, UA_{comp} is the compressor thermal conductance, T_{2A} is the adiabatic discharge temperature of the compressor, and T_{amb} is the ambient temperature. The real compressor power is calculated by Eq. (2) in the case of the single-stage cycle, and by Eq. (3) for the two-stage cycle, as follows

$$\dot{W}_{comp} = \frac{\mathring{m}_{overall}\left(h_2 - h_1\right)}{\eta_g} \tag{2}$$

$$\dot{W}_{comp} = \frac{\mathring{m}_{suc}\left(h_{2s} - h_1\right) + \mathring{m}_{inj}\left(h_{2se} - h_7\right)}{\eta_g} \tag{3}$$

where $\mathring{W}_{comp}$, $\dot{m}_{overall}$, $\dot{m}_{suc}$, $\dot{m}_{inj}$, h_2, h_{2se}, h_1, h_7 and η_g are the power done by the compressor, total mass flow rate, suction flow rate, injection flow rate, enthalpy at the compressor outlet by isentropic compression of the flow coming from the evaporator, enthalpy at compressor outlet by isentropic compression of the flow from the economizer, enthalpy at evaporator outlet, enthalpy of injection and compressor efficiency, respectively.

2.2 Condenser model

The condenser was calculated using the zone separation method, neglecting pressure losses. The scheme of distribution of the overall heat transfer coefficient is made by dividing the condenser area into three parts as we see in Fig. 3. The superheated discharge coming from the compressor enters the condenser and starts to exchange heat until it reaches the subcooled liquid condition. This separation by zones leads us to work with three different heat transfer coefficients present in the condenser, the equations that allow the calculation of these *UA's* are described below.

Figure 3. Condenser divided into thermodynamic phases.

$$\dot{Q}_{superheated} = \mathring{m}\left(h_3 - h_{vap}\right) = \mathring{m}\, cp_{vap}\left(T_3 - T_{amb}\right)\left(1 - e^{-\left[\frac{UA_{superheated}}{\mathring{m}\, cp_{vap}}\right]}\right) \tag{4}$$

$$\dot{Q}_{sat} = \mathring{m}\left(h_{vap} - h_{liq}\right) = UA_{sat}\left(T_{cond} - T_{amb}\right) \tag{5}$$

$$\dot{Q}_{subcooled} = \mathring{m}\left(h_{liq} - h_4\right) = \mathring{m}\, cp_{liq}\left(T_{cond} - T_{amb}\right)\left(1 - e^{-\left[\frac{UA_{subcooled}}{\mathring{m}\, cp_{liq}}\right]}\right) \tag{6}$$

where $\dot{Q}_{superheated}$ is the heat transfer rate exchanged in the superheat region, $\dot{Q}_{subcooled}$ is the heat transfer rate exchanged in the subcooling region, $\dot{Q}_{sat}$ is the heat transfer rate exchanged in the saturation region, $\dot{m}$ is the mass flow rate through the condenser, *cp* is the specific heat at constant pressure, UA_{sat}, $UA_{subcooled}$ and $UA_{superheated}$ are the overall thermal conductance for the saturation, subcooling and superheat region, respectively.

Also present in the model is the energy balance in the condenser on the refrigerant side, as displayed in Eq. (7). For these balance, $\dot{Q}_{cond}$ is the heat transfer rate in the condenser, $\dot{m}_{cond}$ is the mass flow rate passing through the condenser, h_{outlet} is the enthalpy leaving the condenser and h_{inlet} is the enthalpy entering the condenser.

$$\dot{Q}_{cond} = \dot{m}_{cond}\left(h_{outlet} - h_{inlet}\right) \tag{7}$$

2.3 Evaporator

As for the condenser, the evaporator was also divided into fluid phases, but for the calculation of this heat exchanger there is only the division into two areas, because the fluid already enters the evaporator saturated and exchanges heat until the superheating region when it is thrown into the suction line, as shown in Fig. 4.

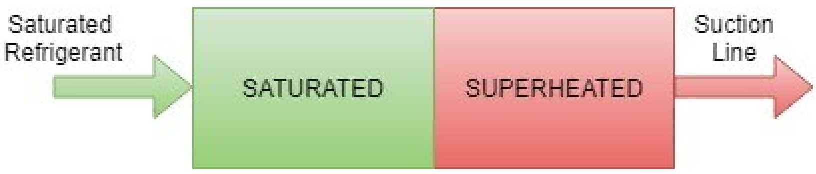

Figure 4. Evaporator divided into zones

The equations that define and solve this distribution of evaporator UA's are shown below.

$$\dot{Q}_{sat} = \dot{m}\left(h_{vap} - h_6\right) = UA_{sat}\left(Tbs - T_{evap}\right) \tag{8}$$

$$\dot{Q}_{superheated} = \dot{m}\left(h_7 - h_{vap}\right) = \dot{m}\, cp_{vap}\left(Tbs - T_{evap}\right)\left(1 - e^{-\left[\frac{UA_{superheated}}{\mathring{m} cp_{vap}}\right]}\right) \tag{9}$$

where $\dot{Q}_{superheated}$ is the heat transfer rate exchanged in the superheat region, $\dot{Q}_{sat}$ is the heat transfer rate exchanged in the saturation region, $\dot{m}$ is the mass flow rate through the evaporator, *cp* is the specific heat at constant pressure, UA_{sat} and $UA_{superheated}$ are the overall thermal conductances for saturation and superheat regions, respectively.

Moreover, we also show Eq. (10), describing the system cooling capacity by means of an energy balance in the evaporator. In this equation, $\dot{Q}_{evap}$ is the heat transfer rate of the evaporator, $\dot{m}_{evap}$ is the mass flow rate through the evaporator, h_{outlet} is the output enthalpy of the evaporator, and h_{inlet} is the input enthalpy.

$$\dot{Q}_{evap} = \dot{m}_{evap}\left(h_{outlet} - h_{inlet}\right) \tag{10}$$

2.4 Economizer

The heat exchanger operating at intermediate pressure acts to cool down the fluid leaving the condenser, thus, providing a decrease in the enthalpy entering the evaporator. Also, the heat transfer that the refrigerant loses in the flow direction 3 to 4 is passed to the flow 5 to 7 by a counter flow heat exchange, thus, ensuring an increase in the enthalpy of the intermediate injection into the compressor. This scheme is shown by Fig. 5 as follows.

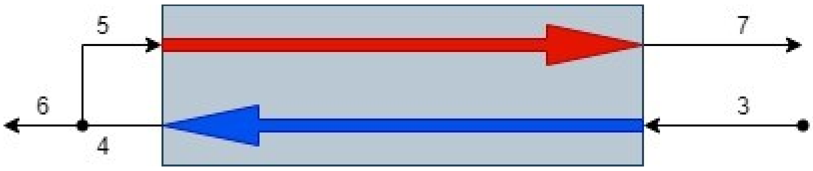

Figure 5. Economizer scheme

The energy balance in the economizer is represented by Eq. (11) as

$$\dot{m}_{total}\left(h_3 - h_4\right) = \dot{m}_{inj}\left(h_7 - h_5\right) \tag{11}$$

where $\dot{m}_{total}$, $\dot{m}_{inj}$, h_3, h_4, h_5 and h_7 are total system mass flow rate, injection mass flow rate, enthalpy at condenser outlet, enthalpy at economizer outlet, enthalpy at economizer upper flow inlet and injection enthalpy (compressor inlet), respectively.

2.5 Expansion Device

The expansion valve was considered adiabatic as represented by Eq. (12). The expansion performed by the valve occurs from high pressure (condensing) to low pressure (evaporating), as well as from high pressure to intermediate pressure in the case of the two-stage cycle with economizer.

$$h_{inlet} = h_{outlet} \tag{12}$$

3. METHODOLOGY

The following working fluids were used for the simulation cases: R-407c, R-410a, R-32, R-1234yf and R-1234ze(E). For each fluid, the volumetric displacement at the nominal test temperature was calculated, using Eq. (13). And then the volumetric displacement found for the other ambient temperatures was kept fixed, likewise the area of the heat exchangers was kept fixed for all fluids and test temperatures.

$$Dis_v = \frac{\dot{m}\, v_1}{\eta_v} \tag{13}$$

where Dis_v is the volumetric displacement, $\dot{m}$ is the mass flow rate, v_1 is the specific volume of the suction line and η_v is the volumetric efficiency of the compressor.

The mathematical model was implemented in the software EES (Engineering Equation Solver). The modeling flowchart is displayed in Fig. 6 for the single-stage cycle, and in Fig. 7 for the two-stage cycle. For the data convergence criteria, it was considered that the difference between the heat transfer rate exchanged on the air side and on the refrigerant side should be less than 1 W, and for the cycle with economizer, the injection point in the compressor should have a superheating of at least 5 ºC.

The adjustment of the heat exchanger temperatures was done using the bisection method, updating whenever the data did not converge and thus reducing the range more and more until the correct system temperature was found. The intermediate pressure was found by using an auxiliary program that performed successive substitutions of the pressure values until the point at the economizer outlet had a 5 ºC superheating.

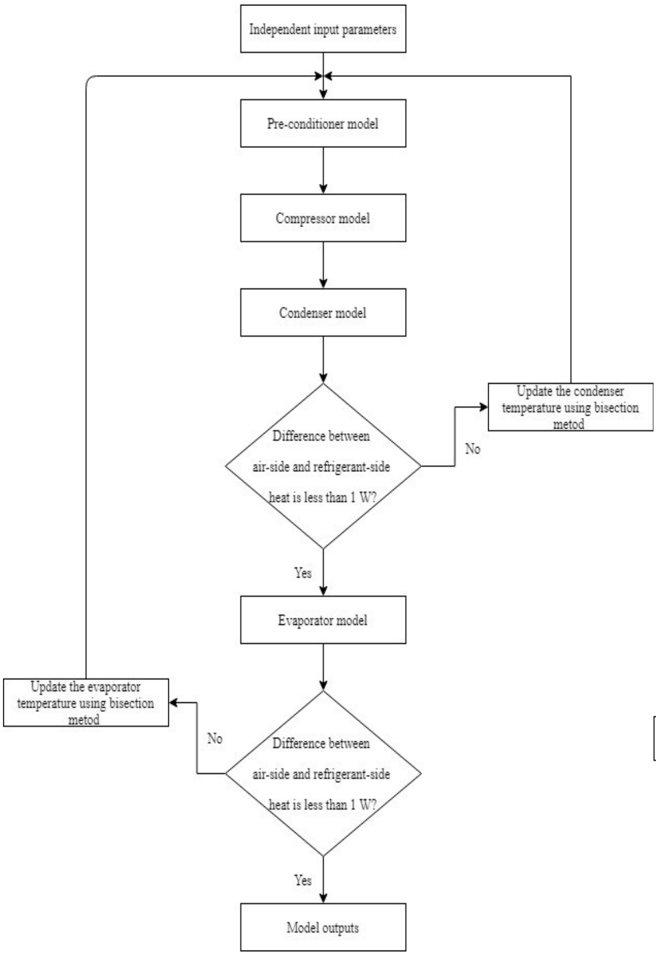

Figure 6. Single-stage flowchart

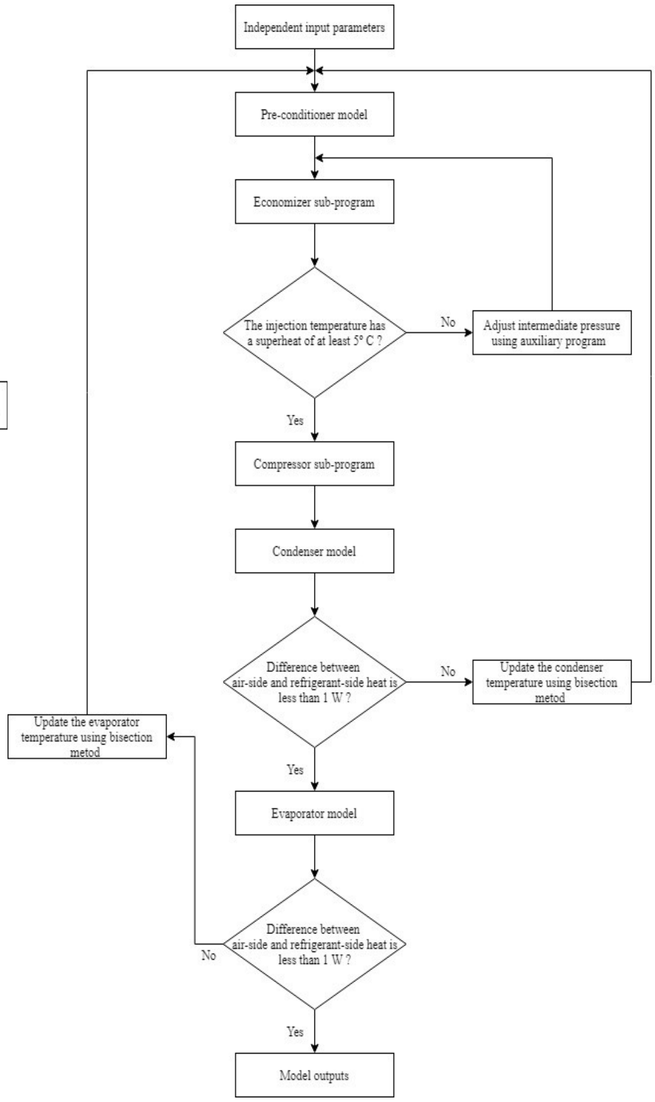

Figure 7. Two-stage flowchart

The temperatures at the heat exchanger outlet maintained a superheat of 5 °C, as well as a subcooling of 5 °C. This way, we ensured that there will be no liquid phase in the compressor suction line. The economizer has an effectiveness of 0.96 as suggested by Bahman *et al.* (2018). The conditions of ambient temperatures, TBS and TBU applied in the tests for all working fluids are represented in Tab. 1 and the input data are represented in Table 2.

Table 1. Test Conditions

Test	**1- (ºC)**	**2- (ºC)**	**3- (ºC)**	**4- (ºC)**	**5- (ºC)**	**6- (ºC)**	**7- (ºC)**	**8- (ºC)**	**9- (ºC)**	**10- (ºC)**	**11- (ºC)**
Tamb	35	37	39	41	43	45	47	49	51	53	55
TBS	27	27	27	27	27	27	27	27	27	27	27
TBU	19	19	19	19	19	19	19	19	19	19	19

Table 2. Input data

UA_{cond}(kW/C)	**$UA_{evapsens}$(kW/C)**	**UA_{comp}(kW/C)**	**Superheating(ºC)**	**Subcooling (ºC)**	**η_g**	**η_{vol}**
0.6186	0.1504	0.0026	5	5	0.54	0.81

4. RESULTS AND DISCUSSION

The compressor discharge temperature, as expected, increases as the ambient temperature increases, this occurs due to the increase of the condensing temperature that accompanies the increase of the ambient temperature. Fig. 8 shows the evolution of the discharge temperature for the single-stage cycle, and Fig. 9 plots the discharge temperature for the two-stage cycle.

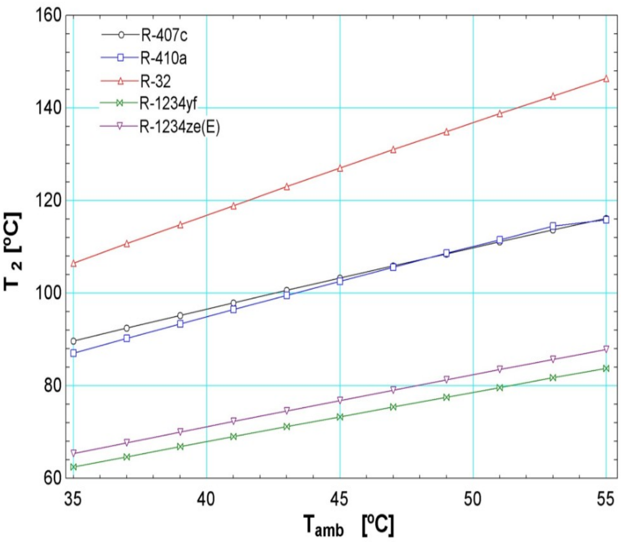

Figure 8. Discharge temperature for a single-stage cycle

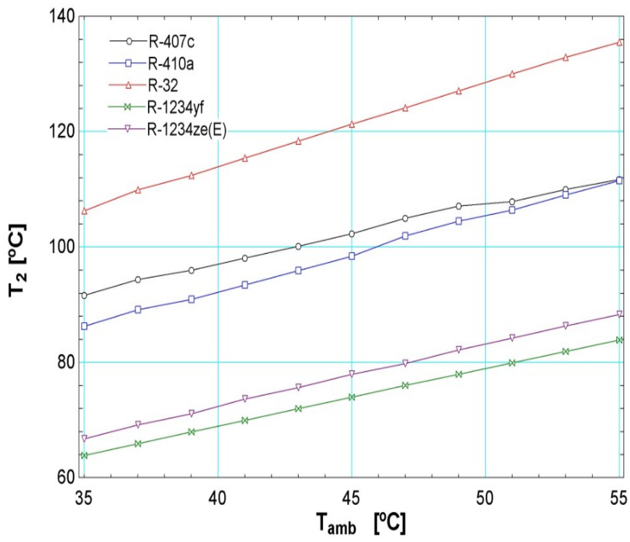

Figure 9. Discharge temperature for a two-stage cycle

As expected, the cycle with economizer reduces the temperature at the compressor outlet. This difference is notorious at the most extreme temperatures, with reductions of up to 7.45%. The economizer is able to reduce the discharge temperature, and by performing an injection at an intermediate pressure, the work done by the compressor is reduced, thus, decreasing the discharge enthalpy and consequently, decreasing the temperature at the compressor outlet.

When analyzing the isolated effects of each refrigerant we can notice that their behavior is similar because with the increase in ambient temperature they all have higher discharge temperatures, in addition the cycle with economizer presents a reduction in discharge temperature in all refrigerants at the most extreme temperatures. Despite the overall improvement we can still highlight the unique behavior of the fluids that as we can see for the same climatic conditions we have different discharge temperatures of the compressor, we note that the R-32 has the highest discharge temperature and the R-1234yf has the lowest temperature.

As seen in Fig. 8 and Fig. 9, the cooling capacity for the system with an economizer is much higher than the single-stage system, with gains ranging from 9% to 25%. This increase is possible due to the large reduction in the evaporator inlet quality provided by the heat exchange in the economizer, thus increasing the difference between the enthalpy inlet and outlet of the evaporator and, as a consequence, increasing the cooling capacity.

When analyzing the cooling capacity for each fluid, we can see that for the same climatic conditions in the two-stage cycle the R-407c has about 15.46% more capacity compared to R-410a, while in the single-stage cycle, the difference between the refrigerant fluids is less than 2%.

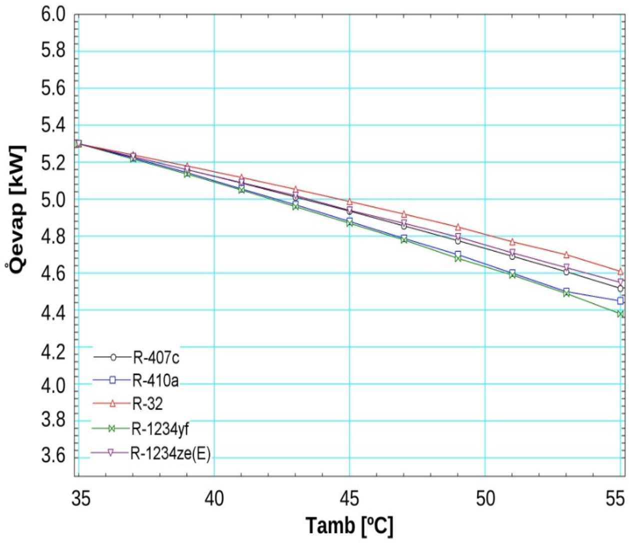

Figure 10. Cooling capacity for a single-stage cycle

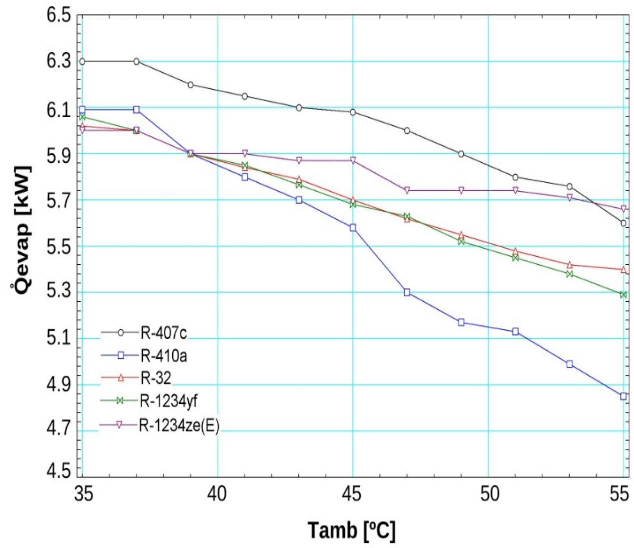

Figure 11. Cooling capacity for a two-stage cycle

Below we can see the behavior of the performance coefficient for different refrigerants tested in the basic air-conditioning model and in the model with intermediate injection with an economizer. By increasing of the ambient temperature, we observed a drop in system performance, as expected, since the more extreme climatic conditions require much more work from the compressor and provide lower cooling capacities.

Comparing the COP between the two refrigeration cycles, it is possible to observe that for temperatures closer to 35 ºC (nominal test), the single-stage cycle has a small advantage in performance. However, in the most extreme conditions, above 41 ºC, the two-stage model promoted a higher performance, with an increase of up to 15.16% in COP.

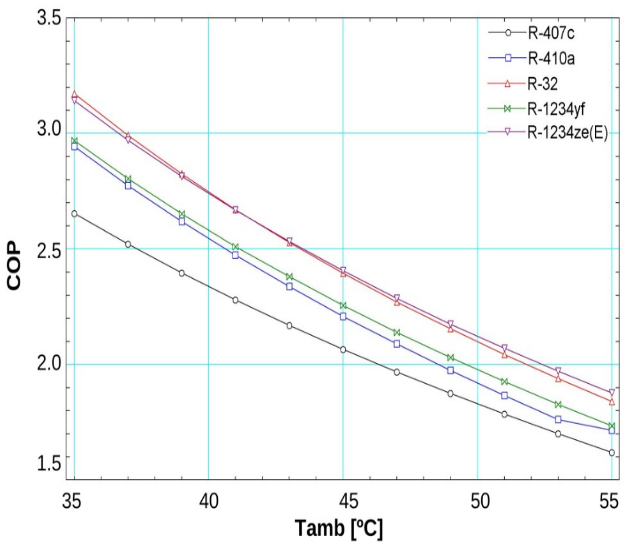

Figure 12. COP for a single-stage cycle

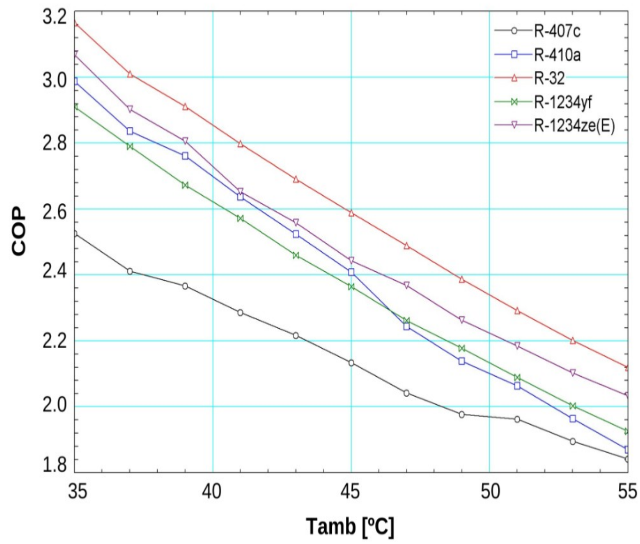

Figure 13. COP for a two-stage cycle

5. CONCLUSION

Putting together all the analyses made in terms of COP, discharge temperature and cooling capacity, we reached the conclusion that the refrigeration cycle by vapor compression with economizer is viable to implement, since in all cases it reduces the discharge temperature of the compressor, ensuring better operation of the system and longer machine lifespan. Furthermore, a perceptible gain in cooling capacity compared to the single-stage cycle was evidenced, with a better system performance, especially at higher ambient temperatures.

The work also brings comparisons between other refrigerants in the same test conditions, being possible to highlight the fluid that has better performance for specific climatic conditions. For lower compressor discharge temperatures, the refrigerant fluid R-1234yf is a viable option, at high ambient temperatures, the fluid maintains the discharge temperature at 83.8ºC, about 38.41% lower than R-32, which has the highest discharge temperatures. For higher cooling capacities, refrigerants R-1234ze(E) or R-407c can be chosen, which guarantee 14% more cooling capacity when compared to other fluids. Finally, we have the R-32 with best COP, however, considering that the R-32 has a COP of 10% higher than the R-1234yf for the extreme ambient temperature condition, and a discharge temperature of 61.59% higher in the same condition, the R-1234yf turns out to be a better choice.

6. ACKNOWLEDGEMENTS

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