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THERMODYNAMIC ANALYSIS OF A RESIDENTIAL AIR CONDITIONING EQUIPMENT OPERATING WITH ODP FREE AND LOW GWP REFRIGERANTS

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Abstract. *Throughout the evolution of air conditioning, the fluids commonly used have also undergone through changes. Firstly, the concern was to choose a refrigerant that allows the proper operation of the equipment. In the last forty years, the focus became environmental problems caused by the refrigerants when released to the atmosphere. The search for an eco-friendly refrigerant that can result in air conditioning applications with satisfactory thermodynamic efficiency is an actual and developing field. The present work performs a thermodynamic analysis of a residential air conditioning equipment with alternative refrigerants with ODP free and low GWP (less than 3). Computational modeling is developed in Python, focused on the thermodynamic analysis of the condenser and evaporator of the selected equipment, allowing a comparison between some promising fluids. The methodology developed allows selecting the best fluid to be used either in retrofitting old equipment to a new refrigerant or in developing a completely new refrigeration system. The performance and cooling capacity of the system working with ammonia (R717), isobutane (R600a), propane (R290), R1234yf, R1234ze(E), R22 and R134a for different operation conditions are studied. The results indicates that ammonia presents the best performance among the selected refrigerants for the conditions assumed here, although not recommended for residential applications. Moreover, R290 and R1234yf showed to be thermodynamically suitable refrigerants to domestic air conditioning equipment.*

Keywords: *air conditioning, thermodynamics, ODP, GWP, development, environment*

1. INTRODUCTION

At the beginning of human industrialization, the effects of industrial activities over the environmental was not known. As industrialization and scientific research advance, environmental preservation became a concern. Late in the XX century, two global environmental problems became focus of attention for the society: ozone depletion and global warming. In order to diminish the effects of the previously mentioned problems, the world community developed two international protocols. The ozone depletion concern resulted in the signature of the Montreal Protocol (1987), which affected significantly HVAC-R (Heating, Ventilation, Air Conditioning, and Refrigeration) industry. Until then, the refrigerants used in the HVAC-R systems were predominately fluids with high ODP (Ozone Depletion Potential) (Calm, 2008). Some years after the Montreal Protocol, the concerns about the global warming lead to the signature of the Kyoto Protocol (1997), which tries to diminish the production of GWP (Global Warming Potential) gases. As many refrigerants used then also presented high GWP, the Kyoto Protocol also affected the HVAC-R industry.

Because of these two protocols, the HVAC-R industry had to research alternative fluids that is under the ODP and GWP restriction, and satisfy efficiency, durability, security, and other desired parameters. Two main research fields can be pointed: retrofitting and development of new systems. In the first one, the refrigerant is changed and the equipment is the same. In the second one, every equipment in the system is sized for the new refrigerant (Ciconkov, 2018).

Due to concerns about the environment, new ODP free and low GWP refrigerants are being studied, and some promising fluids can be identified. Some of the most promising refrigerants are R290, R600a, CO_2, R1234yf, and R1234ze. However, many natural refrigerants, and low GWP HFCs (hydrofluorocarbons), HFEs (hydrofluoroethers), and HCs (hydrocarbons) are also being studied (Calm, 2008).

For domestic refrigerators and freezers, R600a is dominating the market in Asia and Europe. In some countries of Europe this refrigerant represents 95% of the overall residential refrigerators (Melo, 2009). In Brazil, most of the domestic air conditioning equipment operates with other fluids that will be phased out, such as R-22, R-134a, and R-404a (Melo, 2009).

The present work develops a thermodynamic study of a refrigeration cycle typically used in domestic air conditioning applications working with different refrigerants. In this sense, the objective of this work is to do a thermodynamic analysis and to compare the efficiency of residential air conditioning equipment operating with the most promising fluids, in order to identify the suitable refrigerant for domestic air conditioning applications. The performance of the air conditioning equipment operating with different compressors sized for R134a, R290 and R1234yf is analyzed. The system operating with R717, R600a, R290, R1234yf, R1234ze(E), R22, and R134a are also analysed.

2. METHODOLOGY

This section is divided into six parts. The first one is focused on describing refrigeration cycle by vapor compression. The next four parts cover the thermodynamics and energetic analysis of each component of the cycle. The last part shows the iterative process adopted in order to obtain the computational results.

2.1 Vapor Compression Cycle

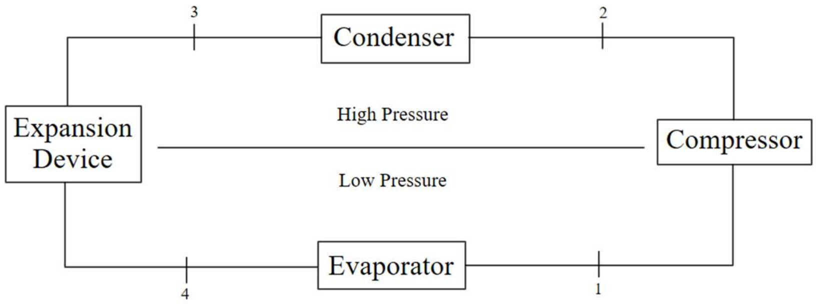

Figure 1. Diagram of a vapor compression cycle.

The vapor compression cycle generates the cooling effect by compressing vapor and increasing the pressure and temperature of the fluid and then exchange heat with cold and hot reservoirs. This system operates with a compressor, a condenser, an expansion device and an evaporator. The arrangement of each equipment in the cycle can be seen schematically in Figure 1.

Initially, saturated or superheated vapor at low pressure enters the compressor (point 1) and leaves as superheated vapor at high pressure, and then enters the condenser (point 2). At the condenser outlet, the refrigerant can be either vapor fraction or subcooled liquid (point 3). It must be stressed that, if the simulation returns vapor fraction at the condenser outlet, the heat exchanger is not suitable to be used for the refrigerant. In the expansion device, the fluid loses pressure at constant enthalpy and enters the evaporator as vapor fraction (point 4). The pressure drop in the condenser, evaporator and pipes are neglected.

2.2 Compressor

The compressor determines the mass flow and the power consumption. The first one is determined as,

$$\dot{m}_{ref} = \rho_1 \cdot \forall_{cil} \cdot N \cdot \eta_v \tag{1}$$

where $\dot{m}_{ref}$, ρ_1, $\forall_{cil}$, N and η_v are the mass flow, refrigerant density at point one, compressor displacement volume, rotations per minute and volumetric efficiency, respectively.

This equipment is also responsible for increasing the refrigerant pressure and, consequently, increasing temperature. For this process a global efficiency is attributed and, according to Da Riva and Del Col (2011), is obtained by,

$$\dot{W}_{comp} = \frac{\dot{m}_{ref}(i_2 - i_1)}{\eta_{global}} \tag{2}$$

where $\dot{W}_{comp}$, $\dot{m}_{ref}$, i_2, i_1 and η_{global} are the compressor power, mass flow, enthalpy at points 1 and 2 and global efficiency, respectively.

η_v and η_{global} are input parameters and functions of pressure ratio, given as,

$$r_p = \frac{p_2}{p_1} \tag{3}$$

where p_1 and p_2 are the pressure at points 1 and 2, respectively.

2.3 Condenser

This equipment is responsible for exchanging heat from the cycle to the hot reservoir, which is assumed here as the outside air. In the condenser, besides the phase change, superheated vapor and saturated liquid are cooled. This last step is desired but not mandatory. The modeling is done by dividing the condenser into three parts, one for each refrigerant physical state.

To determine the state of the refrigerant at the outlet of the heat exchanger, initially, is required to calculate the properties of the external air, which will absorb heat. The airflow is crossed with tubular bundles, but it is simplified as a cross-flow cylinder because of the difficulty to get the parameters needed.

Outside air convection coefficient, according to Bergman *et al.* (2011), is evaluated as,

$$h_{air} = \frac{Nu \cdot k_{air}}{D_{ext}} \tag{4}$$

where the Nusselt number, Nu, is given as,

$$Nu = 0.3 + \frac{0.62 \cdot Re_{ex}^{1/2} \cdot Pr^{1/3}}{[1 + (0.4/Pr)^{2/3}]^{1/4}} \cdot \left[1 + \left(\frac{Re_{ex}}{282\,000}\right)^{5/8}\right]^{4/5} \tag{5}$$

and the external Reynolds Number, Re_{ext}, is

$$Re_{ext} = \frac{\rho u_\infty D_{ext}}{\mu} \tag{6}$$

where $k_{air,ext}$, D_{ext}, Pr, ρ, u_∞ and μ are air thermal conductivity, external diameter of the condenser, Prandtl number, air density, air velocity and air dynamic viscosity, respectively. Equation (5) is valid for the values $RePr > 0.2$.

2.3.1 Superheated vapor

This first segment of the condenser is responsible for cooling superheated vapor into saturated vapor. It is assumed a constant thermal flux on the tube surface. In this step, the tube length required is determined. Firstly, it is needed to determine the internal Nusselt number. According to Bergman *et al.* (2011), for laminar flow ($Re_D < 2\,300$) Nu_D is given as,

$$Nu_D = 4.36 \tag{7}$$

and Re_D follows Eq. (6), but fluid velocity and tube inner diameter are used. For turbulent flow ($Re_D > 3\,000$) internal Nusselt number, according to Bergman *et al.* (2011), is given as,

$$Nu_D = \frac{(f/8)(Re_D - 1\,000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \tag{8}$$

where the friction factor for plain tube, f, is

$$f = [0.790\, ln(Re_D) - 1.64]^{-2} \tag{9}$$

This correlation is valid for $3\,000 < Re_D < 5 \times 10^6$ and $0.5 < Pr < 2\,000$. If the flow is transitional, the values are the weighted average between turbulent and laminar flows, based on Reynolds number. Then, the internal convection coefficient follows Eq. (4), but thermal conductivity is a refrigerant property and inner diameter is used.

Global heat transfer coefficient is evaluated by (Bergman *et al.,*2011),

$$U_i = \frac{1}{\dfrac{1}{h_{ref}} + \dfrac{A_i \cdot ln\left(\frac{D_e}{D_i}\right)}{2 \cdot \pi \cdot k_{tube} \cdot L_{tube}} + \dfrac{A_i}{\eta_e h_{air} A_e}} \tag{10}$$

Finally, length tube required is

$$L_{needed} = \frac{A_{needed}}{\pi \cdot D_i} \tag{11}$$

where the area required, A_{needed}, is given as,

$$A_{needed} = -\,ln\left(\frac{T_\infty - T_{cond}}{T_\infty - T_2}\right) \cdot \frac{\dot{m}_{ref} \cdot C_p}{U_i} \tag{12}$$

where A_i, D_i, A_e, D_e, η_e, k_{tube}, L_{tube}, T_∞, T_{cond}, T_2 and c_p are the inner area and diameter, outer area, diameter and fin efficiency, tube conductivity coefficient, total tube length, air temperature, condensing temperature, temperature of point 2 and specific heat at constant pressure, respectively.

2.3.2 Phase Change

The second part of the condenser is responsible for condensing the refrigerant. This step may require more tube length than is available and the refrigerant will leave as vapor fraction. Otherwise, the exit of this step is saturated liquid. The steps are similar to the previous condenser tube segment. The two-phase convection coefficient is determined by Shah correlation (Kakaç and Liu, 2002) as,

$$h_{TP} = h_l\left(1 + \frac{3.8}{Z^{0.95}}\right) \tag{13}$$

where factor Z is

$$Z = \left(\frac{1-x}{x}\right)^{0.8} p_r^{0.4} \tag{14}$$

and liquid convection coefficient, h_l, is given as,

$$h_l = 0.023\left[\frac{G(1-x)D_i}{\mu_l}\right]^{0.8} \frac{Pr_l^{0.4} k_l}{D_i} \tag{15}$$

where x, p_r, G, μ_l, Pr_l and k_l are vapor fraction, reduced pressure, mass velocity, liquid dynamic viscosity, liquid Prandtl number and liquid conduction coefficient. For an average value of h_{TP}, the vapor fraction used was 0.5.

Finally, effectiveness is obtained by (Bergman *et al.*, 2011),

$$\varepsilon = 1 - exp\,(-NUT) \tag{16}$$

where NUT is

$$NUT = U_i A_i / C_{min} \tag{17}$$

and minimal thermal capacity is given as,

$$C_{min} = \dot{m}_{air} \cdot c_{p\,air} \tag{18}$$

At last, tube length required for total condensation, $L_{cond,total}$, is obtained by,

$$L_{cond,total} = Q_{cond,total} \cdot \frac{L_{tube}}{\rho_{ref} \cdot c_{p,ref} \cdot\ (T_{cond} - T_\infty) \cdot \dot{V}} \tag{19}$$

where total heat required for condensation, $Q_{cond,total}$, is

$$Q_{cond,total} = \dot{m}_{ref}\left(h_{v,sat} - h_{l,sat}\right) \tag{20}$$

where $h_{v,sat}$, $h_{l,sat}$, $\dot{V}$ saturated vapor enthalpy, saturated liquid enthalpy and air volumetric flow.

If the required tube length is higher than the available tube length, Eq. (19) is used to find the total heat transferred and evaluate the refrigerant enthalpy on the condenser outlet. Otherwise, the refrigerant is subcooled.

2.3.3 Subcooled liquid

This section of the heat exchanger is very similar to the first one. The only difference is in Eq. (12), where T_2 is replaced by T_{cond} and T_{cond} is replaced by T_3, which must be determined by the available tube length.

2.4 Expansion Device

This equipment is responsible only for reducing the pressure of the refrigerant. The expansion process is assumed as an isenthalpic process.

2.5 Evaporator

This heat exchanger follows the same equations as the condenser. The refrigerant is a vapor fraction on the evaporator inlet. In this section, the only difference is the two-phase convection coefficient, which is given by Shah correlation (Kakaç and Liu, 2002).

The convection heat transfer coefficient of the liquid phase is given as,

$$h_{LO} = \frac{0.023 Re_D^{0.8} Pr_l^{0.4} k_l}{D_i} \tag{21}$$

The convection number, Co, is evaluated as,

$$Co = \left[\frac{1-x}{x}\right]^{0.8} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} K_{FR} \tag{22}$$

where the correction factor, K_{FR}, is,

$$K_{FR} = (25Fr)^{-0.3} \tag{23}$$

and the Froude number, Fr, is given as,

$$Fr = \frac{G^2}{\rho_l^2 \cdot g \cdot D_i} \tag{24}$$

where g is gravity.

The boiling number, Bo, is also required, and is evaluated by,

$$Bo = \frac{q"}{\dot{m}_{ref}\, i_{lg}} \tag{25}$$

where i_{lg} is the enthalpy variation from point 3 to saturated vapor. If $Bo < 1.9 \times 10^{-5}$ there is no nucleation and the enhancement factor is for pure convection boiling, given as,

$$F_{cb} = 1.0 Co^{-0.8} \tag{26}$$

$$F_{cb} = 1.0 + 0.8 exp(1 - Co^{0.5}) \tag{27}$$

for $Co < 1.0$ and $Co > 1.0$, respectively.

Otherwise, if $Bo > 1.9 \times 10^{-5}$ there is nucleation and the enhancement factor for nucleated boiling is obtained as,

$$F_{nb} = 231 Bo^{0.5} \tag{28}$$

$$F_{cnb} = F_{nb}(0.77 + 0.13 F_{cb}) \tag{29}$$

for $Co > 1.0$ and $0.02 < Co < 1.0$, respectively.

The two-phase convection heat transfer coefficient is evaluated as,

$$h_{TP} = F(1-x)h_{LO} \tag{30}$$

where F is the corresponding coefficient from Eq. (26) to Eq. (29).

Finally, the evaporator outlet is determined using the same formulation for subcooled liquid in the condenser. Thus, the entire vapor compression cycle is determined.

2.6 Iterative process

Two different iterative processes are performed here. The first iterative process is built in order to analyze the system working with a selected condenser and evaporator heat exchanger. It works with a double loop, in which the refrigerant mass flowrate and the evaporation temperature are varied until convergence is reached. The second iterative process is built in order to study the behavior of the system working with a selected compressor. It also works with a double loop, but instead of varying the refrigerant mass flowrate, the condensing temperature is varied together with the evaporation temperature until convergence is reached. In both iterative processes, the superheat degree is the input parameter. In each iteration the state of the refrigerant after the evaporator is found and compared with the state assumed previously. The convergence is assumed when the relative difference between the properties of the refrigerant before and after the calculations of the current iteration are smaller than 1%. Furthermore, it must be stressed that all simulations presented here consider a superheat degree of 10 K, while the subcooling degree is a result of the simulations.

3. RESULTS AND DISCUSSION

The air conditioning equipment considered in the present study it is the one described by Aguiar and Leal (2014), in which the geometric parameters of the condenser and evaporator are presented in Table 1. Furthermore, the compressor for different refrigerants followed the work of de Paula et al. (2020), shown in Table 2. For the simulations, the refrigerated room temperature is 22°C and the ambient temperature is 30°C, 35°C and 40°C. Condenser and evaporator inlet temperatures for the best operating condition varied from one refrigerant to another.

Table 1. Selected Heat Exchangers

Dimensions	**Condenser**	**Evaporator**
Inner Diameter [mm]	7.5	6.1
Outer Diameter [mm]	10	8.1
Number of Fins	445	277
Number of Tubes	44	30
Tube Length [mm]	595	370
Width [mm]	595	370
Height [mm]	370	370
Depth [mm]	60	60
Cross-Sectional Area [m²]	0.2202	0.1369
Airflow [m/s]	2.05	2.5

Table 2. Selected commercial compressors

Refrigerant	**Model**	**Manufacturer**	**Displacement [cm³]**	**Speed [RPM]**
R134a	NT627ZV	Embraco	20.4	2900
R1234yf	CAJ4492N-FZ	Tecumseh	25.95	2900
R290	NEK6217U	Embraco	14.28	2900

3.1 Thermodynamic analysis of heat exchangers.

In the present section, the analysis focus is the heat exchangers. For this reason, a hypothetical compressor is assumed with power compression and isentropic efficiency for all fluids given by 1.0 kW and 70%, respectively. Tables 3, 4 and 5 present the best operating condition for each refrigerant.

Table 3. Simulation result for ambient temperature equal to 30°C.

Refrigerant	Mass Flow [kg/s]	Cooling Capacity [kW]	COP	High Pressure [kPa]	Low Pressure [kPa]
R717	0.0036	4.07	4.07	1705.4	476.1
R600a	0.0139	3.96	3.96	608.0	171.3
R290	0.0133	3.97	3.97	1521.1	522.5
R1234yf	0.0309	3.92	3.92	1177.1	352.7
R1234ze(E)	0.0271	3.98	3.98	895.3	241.5
R22	0.0233	3.99	3.99	1713.7	546.7
R134a	0.0252	3.98	3.98	1153.5	326.0

Table 4. Simulation result for ambient temperature equal to 35°C.

Refrigerant	Mass Flow [kg/s]	Cooling Capacity [kW]	COP	High Pressure [kPa]	Low Pressure [kPa]
R717	0.0034	3.75	3.75	1946.1	497.3
R600a	0.0130	3.68	3.68	697.6	180.4
R290	0.0126	3.65	3.65	1670.7	543.1
R1234yf	0.0296	3.60	3.60	1294.3	368.1
R1234ze(E)	0.0257	3.66	3.66	1003.6	253.0
R22	0.0221	3.69	3.69	1887.8	569.6
R134a	0,0238	3.67	3.67	1300.9	340.0

Table 5. Simulation result for ambient temperature equal to 40°C.

Refrigerant	Mass Flow [kg/s]	Cooling Capacity [kW]	COP	High Pressure [kPa]	Low Pressure [kPa]
R717	0.0032	3.48	3.48	2124.3	517.4
R600a	0.0130	3.42	3.42	746.2	193.1
R290	0.0119	3.35	3.35	1857.4	562.5
R1234yf	0.0282	3.29	3.29	1442.2	382.7
R1234ze(E)	0.0247	3.37	3.37	1108.8	263.9
R22	0.0211	3.39	3.39	2078.2	591.5
R134a	0.0226	3.37	3.37	1453.8	354.5

In Tables 3, 4 and 5, one notices that the COP of all refrigerants studied does not vary significantly at the same ambient temperature, and the main difference is in the operating pressures. The air conditioning considered was designed to operate with R22. Results in Table 3, 4 and 5 show that R717 is the refrigerant that works with the pressure closest to the values obtained with R22. However, as other refrigerants operate with evaporating pressure lower than R22 and higher than atmospheric, this system can operate with any fluid analyzed.

For designing, R600a has the advantage of operating with the lowest pressure. This means that less material is required for pipes and heat exchangers, which can make the equipment cheaper. However, its flammability is a negative point, which requires greater care to avoid leaks. It must be commented that although the higher COP is obtained with R717 for the three ambient temperatures studied here, R717 cannot be used for domestic applications due to its high toxicity.

As expected, COP decreases as the ambient temperature rises. Furthermore, COP for all fluids investigated decreases with the same intensity. The greatest difference in performance between refrigerants is less than 6%.

3.2 Comparison of air conditioning equipment operating with different compressors

In this second analysis, different compressors operating with the same heat exchangers are compared. For this case, compressor power and global and volumetric efficiencies differ for each situation. For the best COP result, global efficiency is close to 75%, 80% and 83%, while volumetric efficiency is close to 43%, 46% and 47%, for R134a, R1234yf and R290, respectively. Because the global and volumetric efficiency equations are valid for pressure ratios between 1.5 and 4.0, the data selection is made by selecting six values above and below the best COP in the acceptable pressure ratio range for each refrigerant.

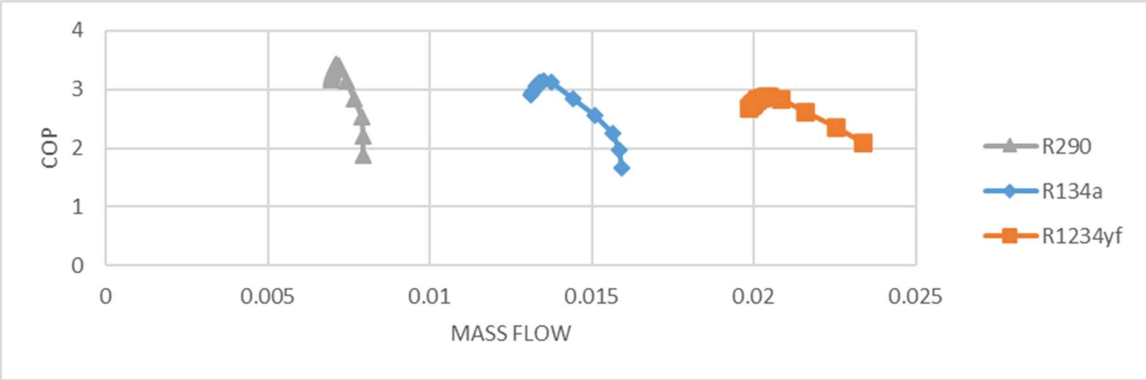

Figure 2. Simulation result for ambient temperature equal 30°C.

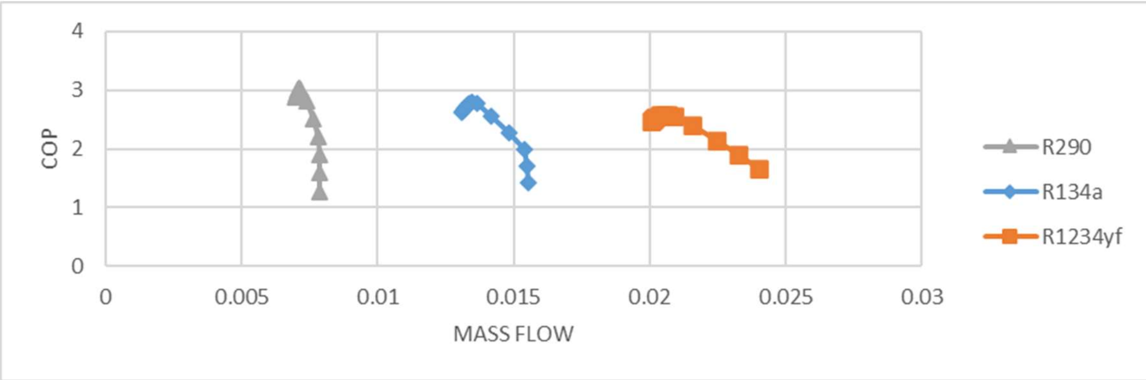

Figure 3. Simulation result for ambient temperature equal 35°C.

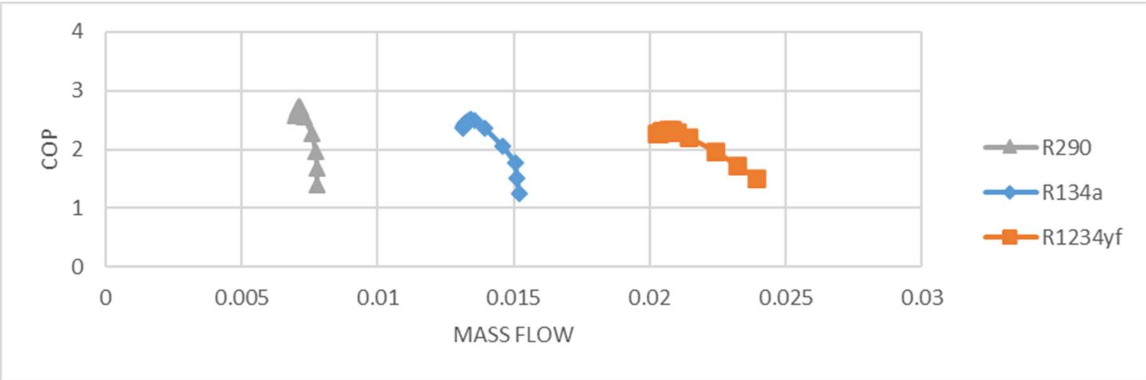

Figure 4. Simulation result for ambient temperature equal to 40°C.

Table 6. Best operating conditions for different ambient temperatures and different compressors.

Refrigerant	30°C			35°C			40°C		
	COP	Cooling Effect [kW]	Compressor Power [kW]	COP	Cooling Effect [kW]	Compressor Power [kW]	COP	Cooling Effect [kW]	Compressor Power [kW]
R290	3.43	2.22	0.65	3.04	2.13	0.70	2.73	2.05	0.75
R134a	3.16	2.22	0.70	2.80	2.14	0.76	2.51	2.04	0.81
R1234yf	2.87	2.67	0.93	2.59	2.55	0.98	2.34	2.44	1.04

One notices in Figures 2, 3 and 4 that the COP change with mass flow occurs due to different refrigerant charges in the air conditioning equipment, highlighting the importance of correctly specifying the gas charge. Still, R290 had the best coefficient of performance among the analyzed refrigerants.

Nevertheless, the results presented in Table 6 indicate the best operation of the system with R290 over R134a, as both manage to deliver the same cooling effect. Finally, the results in Table 6 also show that R1234yf gives the highest cooling effect.

4. CONCLUSION

Evaluating only the thermodynamic aspect of heat exchangers, R717 presented the best result in all ambient temperatures. Although, it is not possible to affirm that this refrigerant is the best option just with this analysis, because the compressor performance is not the same for different fluids, while COP difference to other fluids is less than 6%. Conversely, the considerably lower pressure of R600a can make difference in the cost of pipes and heat exchangers when considered for design.

Furthermore, when technical aspects of the compressor are considered, there are differences between the refrigerants. The main reason is that each pair of refrigerant/compressor results in a different mass flow, global and volumetric efficiency, which directly affects the behavior of system.

In conclusion, R717 performed slightly better in heat exchangers when retrofitting, but further study is required to determine compressor global and volumetric efficiencies for R717. However, R717 can not be used in residential applications because of its high toxicity. Therefore, for the different compressors compared here, R290 had a better operation than R1234yf to replace R134a, but both are thermodynamic suitable refrigerants to domestic air conditioning equipment.

5. ACKNOWLEDGEMENTS

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6. REFERENCES

Aguiar, R. A. Z. and Leal, V. A., *Reference Modeling for Window Air Conditioning Using Engineering Equation Solver Software* (in Portuguese). Graduation Thesis, Graduate Program in Mechanical Engineering, Federal University of Espírito Santo, Vitória, Brazil.

Bergman, T. L., Lavine, A. S., Incropera, F. P. and Dewitt, D. P., 2011. “Fundamentals of Heat and Mass Transfer”, 7th, ed. John Wiley & Sons Inc.

Calm J.M., 2008. “The next generation of refrigerants – Historical review, considerations, and outlook”. *International Journal of Refrigeration,* vol. 31, pp. 1123–1133.

Ciconkov, R., 2018. “Refrigerants: There is still no vision for sustainable solutions”. *International Journal of Refrigeration*, vol. 86, pp. 441–448.

Da Riva, E., Del Col, D., 2011. “Performance of a semi-hermetic reciprocating compressor with propane and mineral oil”. *International Journal of Refrigeration*, vol. 34, pp. 752–763.

Kakaç, S. and Liu, H., 2002. “Heat Exchangers – Selection, Rating and Thermal Design”, 2th, ed CRC Press.

Kyoto Protocol to the United Nations Framework Convention on Climate Change, 1997. United Nations (UN), New York, NY, USA.

Melo, C., 2009. *Use of alternative refrigerants in domestic refrigeration and compact commercial refrigeration equipment* (in Portuguese). Laboratórios de pesquisa em Refrigeração e Termofísica, http://www.protocolodemontreal.org.br/eficiente/repositorio/Apresentacoes/Seminario%20Difusao%20do%20Uso%20de%20Fluidos%20Refrigerantes/871.pdf. Accessed in 18 June 2021.

Montreal Protocol on Substances That Deplete the Ozone Layer, 1987. United Nations (UN), New York, NY, USA (1987 with subsequent amendments).

UNEP, 2007a. Decisions Adopted by the Nineteenth Meeting of the Parties to the Montreal Protocol on Substances that Deplete the Ozone Layer. United Nations Environment Programme (UNEP) Ozone Secretariat, Nairobi, Kenya (22.09.07).

7. RESPONSIBILITY NOTICE