THERMODYNAMIC ANALYSIS OF A RESIDENTIAL AIR CONDITIONING 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 chosing a refrigerant that allows the proper operation of the equipment. In the last forty years, the focus became environmental problems caused by refrigerants when released into 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 operating 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 indicate that ammonia presents the best performance among the selected refrigerants for the conditions assumed here, although not suitable for residential applications due to its toxicity. Moreover, R290 and R1234yf showed to be thermodynamically suitable refrigerants to domestic air conditioning equipment.

Keywords: air conditioning, thermodynamics analysis, environmentally friendly refrigerants

NOMENCLATURE

A	Area, m²
Bo	Boiling number
C	Thermal capacity, W
Co	Convection number
c_p	Specific heat at constant pressure, J/kg
D	Tube diameter, m
f	Friction factor
F	Enhancement factor
Fr	Froude number
G	Mass velocity, m/s
g	Gravity, m/s²
h	Convection coefficient, W/(m².K)
i	Enthalpy, kJ/kg
k	Thermal conductivity, W/(m.K)
K_{FR}	Correction factor
L	Tube length, m
$\dot{m}$	Mass flow, kg/s
N	Compressor rotation speed, s^{-1}
Nu	Nusselt number
p	Pressure, Pa
p_r	Reduced pressure
Pr	Prandtl number
$\dot{Q}$	Heat rate, W
$\dot{Q}_{evap}$	Cooling capacity, W
r_p	Pressure ratio
Re	Reynolds number
T	Temperature, ºC
u	Velocity, m/s
U	Global heat transfer coefficient, W/(m².K)
$\dot{W}_{comp}$	Compressor power, W
x	Vapor fraction
Z	Factor
$\forall_{cil}$	Compressor displacement volume, m³

Greek symbols

ε	Effectiveness
μ	Dynamic viscosity, Pa.s
η_e	Fin efficiency
η_{global}	Global efficiency
η_v	Volumetric efficiency
ρ	Density, kg/m³

Subscripts

1	Evaporator outlet
2	Compressor outlet
cb	Convection boiling
cnb	Convection nucleated boiling
cond	Condensing properties
e	External properties

evap	Evaporating properties
g	Vapor
i	Internal properties
l	Liquid
nb	Nucleated boiling
ref	Refrigerant
sat	Saturated
TP	Two phases
v	Volumetric
∞	Ambient air properties

INTRODUCTION

At the beginning of human industrialization, the effects of industrial activities over the environmental was unknown. 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 restrictions, 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 designed for the new refrigerant (Ciconkov, 2018).

Due to environmental concerns, new ODP free and low GWP refrigerants are being studied, and some promising fluids can be identified (Ciconkov, 2018). 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 objectives of this work are to develop a thermodynamic analysis and to compare the efficiency of residential air conditioning equipment operating with the most promising refrigerants, in order to identify the suitable refrigerant for domestic air conditioning applications. The mathematical modeling is developed in Python and the open library Coolprop (Bell et al., 2014) is adopted to evaluate fluid properties. The performance of the air conditioning equipment operating with different compressors designed for R134a, R290 and R1234yf is analyzed. The system operating with R717, R600a, R290, R1234yf, R1234ze(E), R22, and R134a are also analyzed.

METHODOLOGY

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

Vapor Compression Refrigeration Cycle

The vapor compression refrigeration cycle aims to remove heat from a determined space. It works consuming power to compress vapor and exchanging heat with cold and hot reservoirs. The standard vapor compression refrigeration cycle operates with a compressor, a condenser, an expansion device and an evaporator. The arrangement of each equipment in the cycle for an air conditioning can be seen schematically in Fig. 1.

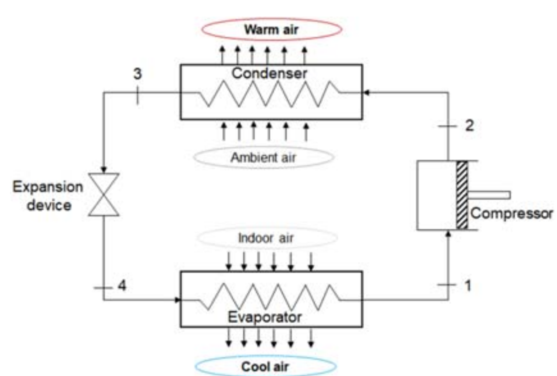

Figure 1. Vapor compression refrigeration cycle for an air conditioning.

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 considered 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 here.

Compressor

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

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

where $\dot{m}_{ref}$, ρ_1, $\forall_{cil}$, N and η_v are the mass flow, refrigerant density at point one, compressor displacement volume, compressor rotation speed 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 $\dot{W}_{comp}$, i_2, i_1 and η_{global} are the compressor power, enthalpy at points 2 and 1, and global efficiency, respectively.

The volumetric and global efficiencies are input parameters, which are 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.

Condenser

The condenser is responsible for exchanging heat from the cycle to the hot reservoir, which is assumed here as the external ambient 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.

In order to obtain the refrigerant state at the outlet of the heat exchanger, initially, it is required to evaluate the properties of the external ambient air. The airflow is crossed with tubular bundles, but it is simplified as a cross-flow cylinder because of the difficulty to get the needed parameters.

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

$$h_{air} = \frac{Nu_e \, k_{air}}{D_e} \tag{4}$$

where the Nusselt number, Nu_e, is given as,

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

and the external Reynolds Number, Re_e, is

$$Re_e = \frac{\rho \, u_\infty \, D_e}{\mu} \tag{6}$$

In Eqs. (4), (5) and (6), k_{air}, D_e, Pr, ρ, u_∞ and μ are, respectively, air thermal conductivity, external diameter of the condenser tube, Prandtl number, air density, air velocity and air dynamic viscosity. It must be noticed that Eq. (5) is valid for the values Re Pr > 0.2 (Bergman et al., 2011).

Superheated vapor

This first segment of the condenser is responsible for cooling superheated vapor until it became 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 evaluate the internal Nusselt number, Nu_i. For laminar flows ($Re_i < 2300$), Nu_i assumes a value of 4.36 (Bergman et al., 2011), and Re_i follows Eq. (6), but with refrigerant velocity and tube inner diameter being used. For turbulent flow ($Re_i > 3000$) internal Nusselt number is given as (Bergman et al., 2011),

$$Nu_i = \frac{(f/8)(Re_i - 1\,000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \tag{7}$$

where the friction factor for plain tube, f, is,

$$f = [0.790 \ln(Re_i) - 1.64]^{-2} \tag{8}$$

The correlation given by Eq. (7) is valid for $3000 < Re_i < 5 \times 10^6$ and $0.5 < Pr < 2000$ (Bergman et al., 2011). 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),

$$U_i = \frac{1}{\frac{1}{h_{ref}} + \frac{A_i \ \ln\left(\frac{D_e}{D_i}\right)}{2\,\pi\, k_{tube}\, L_{tube}} + \frac{A_i}{\eta_e\, h_{air}\, A_e}} \quad (9)$$

Finally, length tube required is,

$$L_{needed} = \frac{A_{needed}}{\pi\, D_i} \quad (10)$$

In Eq. (10), the area required, A_{needed}, is given as,

$$A_{needed} = -\ln\left(\frac{T_\infty - T_{cond}}{T_\infty - T_2}\right) \frac{\dot{m}_{ref}\, c_p}{U_i} \quad (11)$$

In Eqs. (9), (10) and (11), A_i and D_i, are the inner area and diameter, A_e and D_e are outer area and diameter. Furthermore, η_e, k_{tube}, L_{tube}, T_∞, T_{cond}, T_2 and c_p are fin efficiency, tube thermal conductivity, total tube length, ambient air temperature, condensing temperature, temperature of point 2 and refrigerant specific heat at constant pressure, respectively.

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, h_{TP}, is determined by Shah correlation (Kakaç and Liu, 2002) as,

$$h_{TP} = h_l\left(1 + \frac{3.8}{Z^{0.95}}\right) \quad (12)$$

where factor Z is calculated by,

$$Z = \left(\frac{1-x}{x}\right)^{0.8} p_r^{0.4} \quad (13)$$

and liquid convection coefficient, h_l, is given as,

$$h_l = 0.023\left[\frac{G\,(1-x)D_i}{\mu_l}\right]^{0.8} \frac{Pr_l^{0.4} k_l}{D_i} \quad (14)$$

In Eqs. (13) and (14), x, p_r, μ_l, Pr_l and k_l are, respectively, vapor fraction, reduced pressure, liquid dynamic viscosity, liquid Prandtl number and liquid thermal conductivity. For an average value of h_{TP}, the vapor fraction used was 0.5.

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

$$\varepsilon = 1 - \exp(-NTU) \quad (15)$$

where NTU is,

$$NTU = \frac{U_i A_i}{C_{min}} \quad (16)$$

and minimal thermal capacity is considered as,

$$C_{min} = \dot{m}_{air} \cdot c_{p_{air}} \quad (17)$$

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

$$L_{cond,total} = \dot{Q}_{cond,total} \frac{L_{tube}}{\rho_{ref} \cdot c_{p,ref} \cdot (T_{cond} - T_\infty)\dot{V}} \quad (18)$$

where the total heat rate required for condensation, $\dot{Q}_{cond,total}$, is,

$$\dot{Q}_{cond,total} = \dot{m}_{ref}\left(i_{g,sat} - i_{l,sat}\right) \quad (19)$$

where $i_{g,sat}$, $i_{l,sat}$, and $\dot{V}$ are, respectively, saturated vapor enthalpy, saturated liquid enthalpy and air volumetric flow.

If the required tube length is higher than the available tube length, Eq. (18) is used to find the total heat transferred and evaluate the refrigerant enthalpy on the condenser outlet. Otherwise, the refrigerant is subcooled.

Subcooled liquid

This section of the heat exchanger is very similar to the first one. The only difference is in Eq. (11), where T_{cond} replaces T_2 and T_{cond} is replaced by T_3, which must be determined by the available tube length.

Expansion Device

This equipment is responsible only for reducing the pressure of the refrigerant from the condensing pressure, P_{cond}, to evaporating pressure, P_{evap}. The expansion process is assumed as an isenthalpic process.

Evaporator

The evaporator is the heat exchanger used to cool the air of a determined space. The mathematical model for the evaporator 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, h_{LO}, is given as,

$$h_{LO} = \frac{0.023\, Re_D^{0.8}\, Pr_l^{0.4}\, k_l}{D_i} \quad (20)$$

The convection number, Co, is evaluated as,

$$\mathrm{Co} = \left(\frac{1-x}{x}\right)^{0.8} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} K_{FR} \qquad (21)$$

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

$$K_{FR} = (25\ \mathrm{Fr})^{-0.3} \qquad (22)$$

The Froude number, Fr, is given as,

$$\mathrm{Fr} = \frac{G^2}{\rho_l^2\ g\ D_i} \qquad (23)$$

where g is gravity.

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

$$\mathrm{Bo} = \frac{q"}{\dot{m}_{ref}\ \ i_{lg}} \qquad (24)$$

where i_{lg} is the enthalpy variation from point 3 to saturated vapor. If Bo < 1.9 $\times10^{-5}$ there is no nucleation and the enhancement factor is for pure convection boiling, given as,

$$F_{cb} = 1.0\ \mathrm{Co}^{-0.8}, \text{ for Co} < 1.0 \qquad (25)$$

$$F_{cb} = 1.0 + 0.8 \exp(1 - \mathrm{Co}^{0.5}), \text{ for Co} > 1.0 \qquad (26)$$

Otherwise, if Bo > 1.9 $\times10^{-5}$ there is nucleation and the enhancement factor for nucleated boiling is obtained as,

$$F_{nb} = 231\ \mathrm{Bo}^{0.5}, \text{ for Co} > 1.0 \qquad (27)$$

$$F_{cnb} = F_{nb}\ (0.77 + 0.13\ F_{cb}), \text{ for } 0.02 < \mathrm{Co} < 1.0 \qquad (28)$$

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

$$h_{TP} = F(1-x)h_{LO} \qquad (24)$$

where F is the corresponding coefficient from Eq. (25) to Eq. (28).

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

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.

RESULTS AND DISCUTION

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 Tab. 1. Furthermore, the compressor for different refrigerants followed the work of de Paula et al. (2020), shown in Tab. 2. For the simulations, the refrigerated room temperature, T_{room}, is 22 °C and the ambient temperature, T_∞, 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	**R134a**
Model	NT627ZV
Manufacturer	Embraco
Displacement [cm³]	20.4
Speed [RPM]	2900
Refrigerant	**R1234yf**
Model	CAJ4492N-FZ
Manufacturer	Tecumseh
Displacement [cm³]	25.95
Speed [RPM]	2900

Refrigerant	R290
Model	NEK6217U
Manufacturer	Embraco
Displacement [cm^3]	14.28
Speed [RPM]	2900

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 compression power and isentropic efficiency for all fluids given by 1.0 kW and 70%, respectively. Tables 3, 4 and 5 present the refrigerant mass flow rate, $\dot{m}_{ref}$, the cooling capacity, $\dot{Q}_{evap}$, the COP, the condensing pressure, p_{cond}, and the evaporating pressure, p_{evap}, for the best operating condition for each refrigerant.

Table 3. Simulation results for T_∞ = 30°C.

Refrigerant	$\dot{m}_{ref}$ [kg/s]	$\dot{Q}_{evap}$ [kW]	COP	p_{cond} [kPa]	p_{evap} [kPa]
R717	0.0036	4.07	4.07	1705	476
R600a	0.0139	3.96	3.96	608	171
R290	0.0133	3.97	3.97	1521	523
R1234yf	0.0309	3.92	3.92	1177	353
R1234ze(E)	0.0271	3.98	3.98	895	242
R22	0.0233	3.99	3.99	1714	547
R134a	0.0252	3.98	3.98	1154	326

Table 4. Simulation results for T_∞ = 35°C.

Refrigerant	$\dot{m}_{ref}$ [kg/s]	$\dot{Q}_{evap}$ [kW]	COP	p_{cond} [kPa]	p_{evap} [kPa]
R717	0.0034	3.75	3.75	1946	497
R600a	0.0130	3.68	3.68	698	180
R290	0.0126	3.65	3.65	1671	543
R1234yf	0.0296	3.60	3.60	1294	368
R1234ze(E)	0.0257	3.66	3.66	1004	253
R22	0.0221	3.69	3.69	1888	570
R134a	0,0238	3.67	3.67	1301	340

Table 5. Simulation results for T_∞ = 40°C.

Refrigerant	$\dot{m}_{ref}$ [kg/s]	$\dot{Q}_{evap}$ [kW]	COP	p_{cond} [kPa]	p_{evap} [kPa]
R717	0.0032	3.48	3.48	2124	517
R600a	0.0130	3.42	3.42	746	193
R290	0.0119	3.35	3.35	1857	563
R1234yf	0.0282	3.29	3.29	1442	383
R1234ze(E)	0.0247	3.37	3.37	1109	264
R22	0.0211	3.39	3.39	2078	592
R134a	0.0226	3.37	3.37	1454	355

In Tabs. 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 Tabs. 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 condensing 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.

Also from Tabs. 3, 4 and 5, as expected, COP decreases as the ambient temperature rises. Furthermore, COP for all fluids investigated here decreases with the same intensity. The greatest difference in performance between refrigerants is less than 6%.

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 (Da Riva and Del Col, 2011), the data selection is made by selecting six values above and below the best COP in the acceptable pressure ratio range for each refrigerant.

One notices in Figs. 2, 3 and 4 that the COP change with mass flow occurs due to different refrigerant charges in the air conditioning equipment. It is shown an optimal operation point, highlighting the importance of correctly specifying the gas charge. Still, R290 had the best COP among the analyzed refrigerants.

Table 6 presents the best operating conditions for different ambient temperatures and compressors. Results in Tab. 6 reveal that the system working with R290 and R134a produces similar cooling effect. However, the system working with R290 has a smaller power consumption and, consequently, a higher COP. Then, the results from Tab. 6 indicate a best operation of the system with R290 over R134a. Furthermore, it is also noticed in Tab. 6 that the system working with R1234yf gives the highest cooling effect compared to the system working with R290 and R134a. Nevertheless, the R1234yf results in a higher power consumption, leading to the smaller COP between the refrigerants investigated.

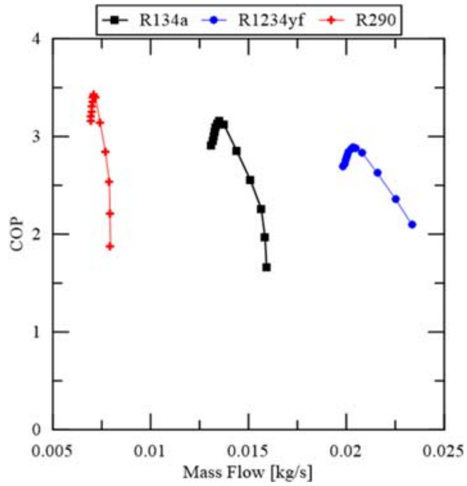

Figure 2. Simulation results for T_∞ = 30 °C.

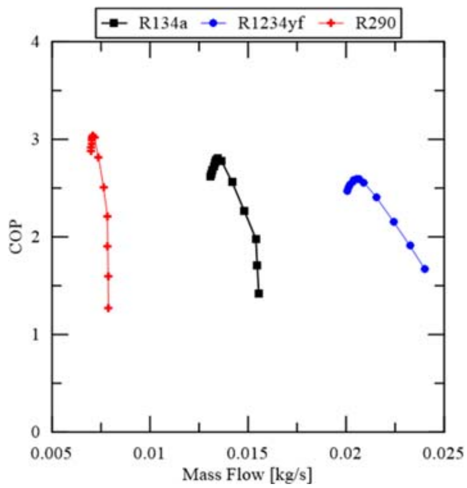

Figure 3. Simulation results for T_∞ = 35 °C.

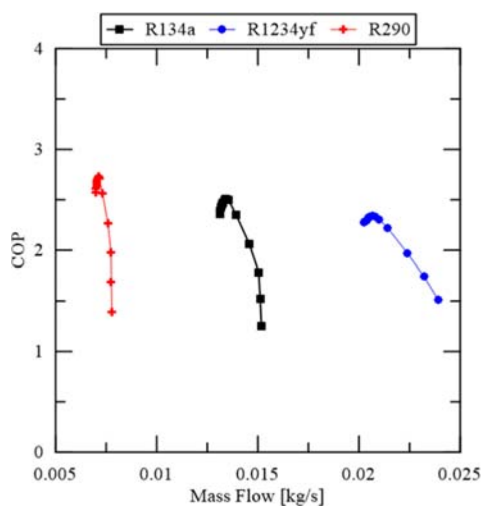

Figure 4. Simulation results for T_∞ = 40 °C.

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

Refrigerant	COP	$\dot{Q}_{evap}$ [kW]	$\dot{W}_{comp}$ [kW]
	T_{amb} = 30 °C		
R290	3.43	2.22	0.65
R134a	3.16	2.22	0.70
R1234yf	2.87	2.67	0.93
	T_{amb} = 35 °C		
R290	3.04	2.13	0.70
R134a	2.80	2.14	0.76
R1234yf	2.59	2.55	0.98
	T_{amb} = 40 °C		
R290	2.73	2.05	0.75
R134a	2.51	2.04	0.81
R1234yf	2.34	2.44	1.04

CONCLUSIONS

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 system design.

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

In conclusion, R717 performed slightly better in heat exchangers when retrofitting analysis was performed, 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.

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