25th ABCM International Congress of Mechanical Engineering
October 20-25, 2019, Uberlândia, MG, Brazil

COBEM2019-0513
ENVIRONMENTAL AND ENERGY PERFORMANCE EVALUATION OF THE R1234YF AS AN ENVIRONMENTALLY FRIENDLY ALTERNATIVE TO R134A

Cleison Henrique de Paula
Post-Graduate Program in Mechanical Engineering, Federal University of Minas Gerais (UFMG), Antônio Carlos Avenue, 6627, Pampulha Campus, Belo Horizonte, MG, Brazil.
e-mail: cleissonhp@yahoo.com.br

Willian Moreira Duarte
University of Belo Horizonte (Unibh), Mário Werneck Avenue, 1685, Buritis Campus, Belo Horizonte, MG, Brazil.
e-mail: willian.duarte@prof.unibh.br

Thiago Torres Martins Rocha
Raphael Nunes de Oliveira
Antônio Augusto Torres Maia
Post-Graduate Program in Mechanical Engineering, Federal University of Minas Gerais (UFMG), Antônio Carlos Avenue, 6627, Pampulha Campus, Belo Horizonte, MG, Brazil.
e-mails: thiagotorres.rocha@yahoo.com.br, rphnunes@demec.ufmg.br, aamaia@demec.ufmg.br

Abstract. *In this paper, a static model of a vapor compression refrigeration system (VCRS) composed of the following components was developed: a reciprocating compressor, a thermostatic expansion valve, an evaporator and a condenser both of the coaxial type. This model was developed according to the following objectives: design a refrigeration system with an optimized geometric structure and compare the environmental and energy performance of the R1234yf with R134a. The proposed model used for each refrigerant volumetric and global efficiency curve obtained from the data supplied by commercial compressors. The environmental metric used to calculate the environmental performance was TEWI and the geometric structure was minimized by an optimization method. The COP analysis showed that the energy performance of the optimized system with R134a is 22.33% higher than the optimized system with R1234yf for an evaporation temperature of -5°C and condensation temperature of 50°C. The TEWI analysis also showed that the environmental performance of the optimized system with R1234yf is 32.2% lower for established thermodynamic condition. Therefore, R1234yf is not the most suitable refrigerant to replace R134a for this specific application and other environmentally friendly refrigerants should be evaluated.*

Keywords: *Static Model, Vapor Compression refrigeration System, Optimization method, COP, TEWI.*

1. INTRODUCTION

Analyzing the period between the 1970s and the present moment, there is a growing concern on the part of the countries with regard to the environmental impacts caused by vapor compression refrigeration systems due to the emission of environmentally harmful refrigerants. At the same time, a worldwide awareness campaign has been developed to the rational use of energy resources. This has boosted in recent years, the improvement of vapor compression systems. Thus, several researches have been carried out in order to improve the current designs of the vapor compression systems, so that they become more compact and efficient from an environmental and energetic point of view. From the environmental point of view is mandatory the use of a refrigerant with low GWP (global warming potential) and ODP (ozone depletion potential) equal to zero. For this, it is necessary study alternatives that can replace harmful fluids, especially R134a. In recent years, many studies have evaluated the possibility of R1234yf replacing R134a in vapor compression refrigeration systems.

Jarall (2012) carried out a theoretical and experimental study of a vapor compression refrigeration system with R1234yf. In this system, the evaporator and condenser are plate type heat exchangers, the compressor is hermetic and rotary type and its expansion device is a thermostatic expansion valve. First, the COP, the cooling capacity and the compressor efficiency of R1234yf were evaluated for two different condensing temperatures (40 °C and 45 °C) considering the evaporation temperature range of -8 °C up to 15.5 °C. The results indicated that the compressor efficiency ranged respectively from 44.53% up to 47.62% and 44.69% up to 50.09% for the condensing temperatures of 40 ° C and

45 °C. Subsequently, R1234yf and R134a were compared based on the same condensing temperatures and using the same system without any modification, but with the evaporation temperature ranging from -5°C up to 15 °C. The experimental results for the condensation temperatures of 40 °C and 45 °C indicated that R1234yf compared to R134a obtained lower values of cooling capacity, COP and compressor efficiency.

Navarro-Esbrí et al. (2013) performed an experimental analysis on a vapor compression refrigeration system in order to compare the energy performance of R134a and R1234yf. The system consists of a shell and tubes evaporator and condenser, an alternative type compressor, an internal heat exchanger (IHX) and a thermostatic expansion valve. The experimental tests were carried out varying the evaporation and condensation temperatures, superheating degree, rotation speed of the compressor and the use of the internal heat exchanger. The results showed that the cooling capacity of R1234yf is about 9% lower. In addition, the compressor working with R1234yf obtained a volumetric efficiency 5% lower compared to the value obtained by this same compressor working with R134a. Finally, the values obtained for the COP using R1234yf were 5% up to 30% lower than the values obtained with R134a.

Therefore, the main goal of this paper is to design a steady-state vapor compression refrigeration system operating with R1234yf and R134a and to compare the energy and environmental performance of these systems.

2. MATHEMATICAL MODEL OF THE VCRS

In order to design the vapor compression refrigeration system a steady state model was developed using the software Equation Engineering Solver (EES). In this model, the pressure drop was considered negligible in the heat exchangers and in the pipes. A thermostatic expansion valve was considered as isenthalpic expansion device. The heat loss and pressure drop in the pipes between components were considered negligible, the pipeline was considered two meters long. The input variables of the model are shown on the left and the output variables are shown on the right, as shown in Fig.1.

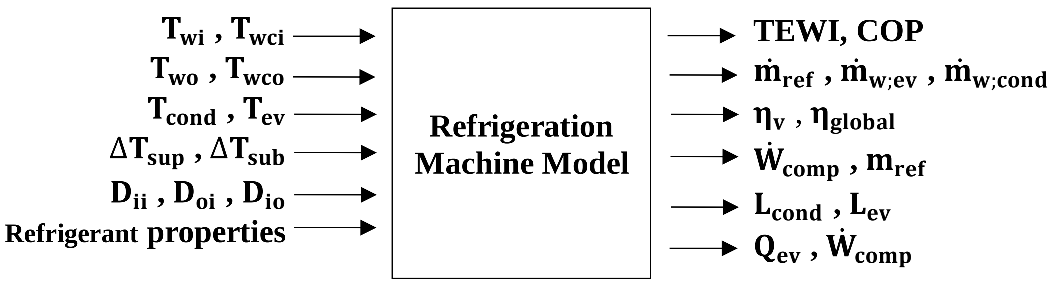

Figure 1. Scheme of the input and output variables of the model.

2.1 Coaxial evaporator

The evaporator is a heat exchanger of the concentric tubes type, with the refrigerant flows through the inner tube and water flows counterflow through the annular region. The cooling capacity (Q_{ev}) is obtained by the energy balance for steady state condition and is given by Eq. 1.

$$Q_{ev} = \dot{m}_{ref} \cdot (i_1 - i_4) = \dot{m}_{w;ev} \cdot Cp_w \cdot (T_{wi} - T_{wo}) \text{ [kW]} \tag{1}$$

In this equation $\dot{m}_{ref}$ is refrigerant mass flow rate [kg/s], i_1 is the refrigerant specific enthalpy at the evaporator outlet [kJ/kg] and i_4 is the refrigerant specific enthalpy at the evaporator inlet [kJ/kg], $\dot{m}_{w;ev}$ is water mass flow rate at the evaporator [kg/s], T_{wi} is water temperature at the evaporator inlet [°C] and T_{wo} is water temperature at the evaporator outlet [°C]. The cooling capacity is calculated using the logarithmic mean temperature difference method (ΔT_{ml}), Eq. 2, according to (Bergman et al., 2011).

$$Q_{ev} = UA_{ev} \cdot \Delta T_{ml;ev} \tag{2}$$

$$UA_{ev} = \left(\frac{1}{\bar{h}_{ref}\pi D_{ii} L_{ev}} + \frac{\ln(D_{oi}/D_{ii})}{2\pi k L_{ev}} + \frac{1}{\bar{h}_{w;ev}\pi D_{io} L_{ev}}\right)^{-1} \tag{3}$$

$$\Delta T_{ml;ev} = \frac{[(T_{wi} - T_1) - (T_{wo} - T_4)]}{\ln((T_{wi} - T_1)/(T_{wo} - T_4))} \tag{4}$$

In these equations T_1 is refrigerant temperature at the evaporator outlet [°C], T_4 is refrigerant temperature at the evaporator inlet [°C], $\bar{h}_{ref}$ is refrigerant average convective coefficient [W/m²K], $\bar{h}_{w;ev}$ is water average convective coefficient in the evaporator [W/m²K], D_{ii} is inner diameter of the inner tube (refrigerant diameter), D_{oi} is outer diameter of the inner tube, D_{io} is inner diameter of the outer tube (water diameter) and L_{ev} is evaporator length [m].

2.2 Coaxial condenser/Gas cooler

The condenser is of the concentric tubes type, with the refrigerant flows through the inner tube and water flows counterflow through the annular region. The heat transfer rate at the condenser (Q_{cond}) is obtained by the energy balance for steady state condition and is given by Eq. 5.

$$Q_{cond} = \dot{m}_{ref} \cdot (i_2 - i_3) = \dot{m}_{w;cond} \cdot Cp_w \cdot (T_{wco} - T_{wci}) \text{ [kW]} \quad (5)$$

In this equation i_2 is the refrigerant specific enthalpy at the condenser inlet and i_3 is the refrigerant specific enthalpy at the condenser outlet, $\dot{m}_{w;cond}$ is water mass flow rate at the condenser [kg/s], T_{wci} is water temperature at the condenser inlet and T_{wco} is water temperature at the condenser outlet. The heat transfer rate at the condenser is calculated using the logarithmic mean temperature difference method, Eq. 6.

$$Q_{cond} = UA_{cond} \cdot \Delta T_{ml;cond} \quad (6)$$

$$UA_{cond} = \left(\frac{1}{\bar{h}_{ref}\pi D_{ii} L_{cond}} + \frac{\ln(D_{oi}/D_{ii})}{2\pi k L_{cond}} + \frac{1}{\bar{h}_{w;cond}\pi D_{io} L_{cond}}\right)^{-1} \quad (7)$$

$$\Delta T_{ml;cond} = \frac{[(T_2 - T_{wco}) - (T_3 - T_{wci})]}{\ln((T_2 - T_{wco})/(T_3 - T_{wci}))} \quad (8)$$

In these equations T_2 is refrigerant temperature at the condenser inlet, T_3 is refrigerant temperature at the condenser outlet, $\bar{h}_{w;cond}$ is water average convective coefficient [W/m²K] at the condenser and L_{cond} is condenser length [m]. The refrigerant average convective coefficient is obtained from the correlations proposed by Gnielinski (1976) in single phase flow, Shah (2016) and Shah (2017) in two phase flow, respectively, for condensation and for boiling. Finally, the water average convective coefficient is obtained from the Dittus-Boelter correlation.

2.3 Reciprocating compressor

The refrigerant mass flow rate in the compressor ($\dot{m}_{ref}$) is given by Eq. 9.

$$\dot{m}_{ref} = \rho_1 \cdot \forall_{cil} \cdot N \cdot \eta_v \quad (9)$$

Where $\forall_{cil}$ is the compressor displacement volume [m³], N is the rotation speed of the compressor [Hz], ρ_1 is the refrigerant density in the compressor inlet [kg/m³] and η_v is volumetric efficiency. The electrical power consumption $\dot{W}_{comp}$ is given by Eq. 10, according to Da Riva (2011).

$$\dot{W}_{comp} = \frac{\dot{m}_{ref}(i_2 - i_1)}{\eta_{global}} \text{ [kW]} \quad (10)$$

The volumetric (η_v) and global (η_{global}) efficiency curves of a compressor operating with R134a and R1234yf were obtained by a polynomial regression in function of pressure ratio ($r_p = P_2/P_1$). This polynomial regression was performed on the efficiency data supplied by commercial compressors. The most suitable commercial compressor for each refrigerant was selected according to the following criteria:

I. Reference cooling capacity of 1.2 kW.

II. Based on the cooling capacity adopted, for a voltage of 220 V, frequency of 50 Hz, evaporation temperature equal to -5 ° C and condensation temperature equal to 50 °C, the commercial compressors were selected, as presented in the Tab. 1.

Table 1 – Selected Compressors.

Refrigerant	Model	Manufacturer	Displacement (cm³)	Rotation (rpm)
R134a	NT6217ZV	Embraco	20.4	2900
R1234yf	CAJ4492N-FZ	Tecumseh	25.95	2900

This regression was performed on the efficiency data supplied by the manufacturer and the order chosen for this polynomial was the one that best fit the regression curve to the data, considering according to the manufacturer an uncertainty of 5%, as shown in the Tab. 2.

Table 2 – Global and volumetric efficiency curves.

REFRIGERANT	R134a	R1234yf
Volumetric efficiency	$\eta_v = 1.0368 - 0.1517r_p + 0.0243{r_p}^2 - 0.0014{r_p}^3$	$\eta_v = 0.9041 - 0.0351r_p - 0.0013{r_p}^2$
R^2 of η_v	85.14 %	98.63 %
Global efficiency	$\eta_{global} = 0.2819 + 0.0766r_p - 0.0058{r_p}^2$	$\eta_{global} = -0.6924 + 1.1139r_p - 0.4256{r_p}^2 + 0.0801{r_p}^3 - 0.0074{r_p}^4 + 0.0003{r_p}^5$
R^2 of η_{global}	94.42 %	87.17 %

2.4 Energetic and environmental metrics

The coefficient of performance (COP) of the VCRS is given by Eq. 11, according to (Mendoza-Miranda et al., 2016).

$$COP = \frac{\eta_{global}(i_1 - i_4)}{i_2 - i_1} \tag{11}$$

The environmental performance of the VCRS was evaluated by Total Equivalent Warming Impact (TEWI), Eq.12, according to (Antunes, 2016). This parameter takes into account both direct emissions (due to refrigerant leakage during the life of the equipment) and indirect emissions (due to the compressor's electricity consumption over the life of the equipment).

$$TEWI = TEWI_{Direct} + TEWI_{INDirect} \tag{12}$$

$$TEWI_{Direct} = GWP \cdot m_{ref} \cdot L_{rate} \cdot L_{time} + GWP \cdot m_{ref} \cdot (1 - \alpha_{recup}) \tag{13}$$

$$TEWI_{INDirect} = 365 T_{oper} (Q_{ev}/COP) \cdot \beta \cdot L_{time} \tag{14}$$

In this equation L_{rate} is the annual rate of refrigerant emitted (replacement and leaks) given in [%], L_{time} is the life of the refrigeration system given in [Years], α_{recup} is the refrigerant life recovery rate given in [%], T_{oper} is the refrigeration system operating given in hours, β is the CO_2 emission factor for electricity generation given in $kgCO_2/kWh$. In this paper was considered T_{oper} = 12 [h/day]. The refrigerant charge is determined by the Equation 16 for single-phase flow and by the Equation 17 for two-phase flow, where α is the void fraction. The void fraction is determined by the Hughmark (1965) correlation.

$$m_{ref} = \bar{\rho}_{ref} \frac{\pi D_{REF}^2}{4} L \ \ [kg] \tag{15}$$

$$m_{ref} = [\alpha \rho_V + (1 - \alpha)\rho_L] \cdot V \ \ [kg] \tag{16}$$

In the Eq. 15, L is the heat exchanger length under study. Thus, L can be L_{ev} or L_{cond}. In addition, in the Eq. 16, V is heat exchanger volume corresponding to the two-phase flow.

The refrigerant charge inside the refrigeration system is the sum of the refrigerant charge inside the evaporator, condenser and pipes that connects the components. It was considered that each pipe has a length equal to two meters.

2.5 Optimization methodology used in the design

In order to design a refrigeration system with reduced geometric structure and efficient from the environmental point of view, the Nelder Mead Simplex method was used to perform the optimization task. This algorithm is available in the EES software subroutine. This method is used to determine the evaporator and condenser optimal length, as well as, the optimal refrigerant charge based on a range of values for the refrigerant and water diameters. The D_{ii} varying from 5 mm to 20 mm and the D_{io} varying from 10 mm to 40 mm.

2.6 Simulation parameters

The modeled vapor compression refrigeration system in this paper is constituted by the components shown in Fig. 2: a reciprocating compressor, a thermostatic expansion valve (TEV), an evaporator and a condenser both of the coaxial type.

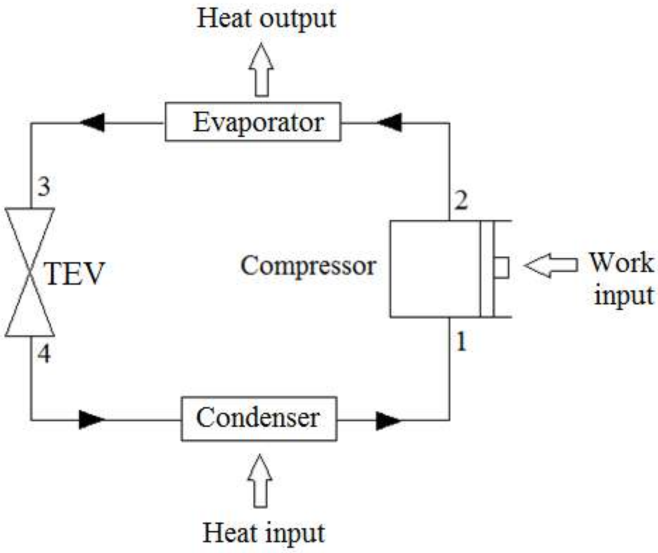

Figure 2. Basic components present in a VCRS.

The thermodynamic considerations adopted were based on literature review and they are presented in Tab. 3. Due to the established thermodynamic conditions, the evaluated refrigerants have a subcritical refrigeration cycle, as shown in Fig. 3.

Table 3 – Thermodynamic considerations adopted.

Evaporation temperature ($\mathbf{T_{ev}}$)	-5 °C
Condensation temperature ($\mathbf{T_{cond}}$)	50 °C
Superheating degree ($\Delta\mathbf{T_{sup}}$)	7 °C
Subcooling degree ($\Delta\mathbf{T_{sub}}$)	5 °C
Water temperature in the evaporator inlet ($\mathbf{T_{wi}}$)	12 °C
Water temperature in the evaporator outlet ($\mathbf{T_{wo}}$)	5 °C
Water temperature in the condenser inlet ($\mathbf{T_{wci}}$)	30 °C
Water temperature in the condenser outlet ($\mathbf{T_{wco}}$)	40 °C

Notice that the evaporator consists of a single-phase region (1'-1) and a two-phase region (4-1'). The condenser consists of two single-phase regions (2'-2 and 3-3') and a two-phase region (3'-2').

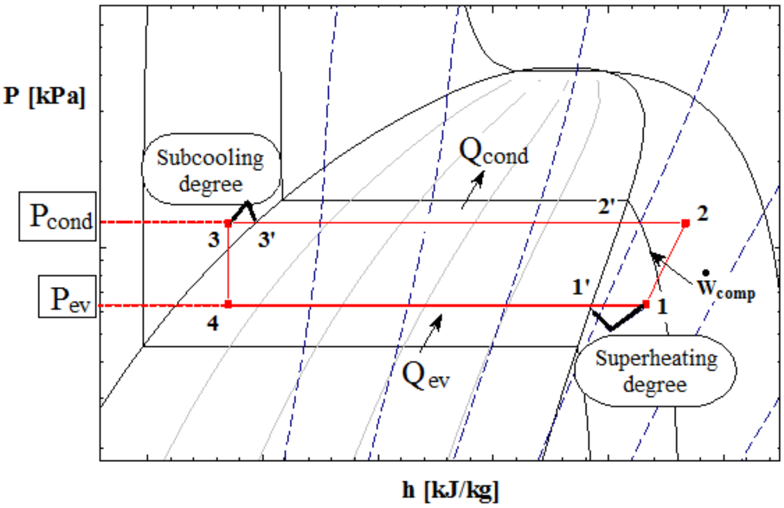

Figure 3. Vapor compression refrigeration cycle: R134a e R1234yf.

The main considerations for calculating of the TEWI are presented in Tab. 4. In this analysis, it was considered Brazil and two other countries with higher and lower CO_2 emission factors (β) in relation to the emission factor indicated for Brazil.

Table 4: Considerations for calculating of the TEWI parameter.

Parameter	Consideration	Reference
$L_{time} = 15$ [Years]	Equipment operating with economic useful life	Makhnatch and Khodabandeh (2014)
$\alpha_{recup} = 70\%.$	Refrigerant mass less than 100 kg	AIRAH (2012)
$\beta = 0.035[kgCO_2/kWh]$	Reference value for Sweden	Rees (2016)
$\beta = 0.082[kgCO_2/kWh]$	Reference value for Brazil	Rees (2016)
$\beta = 0.40[kgCO_2/kWh]$	Reference value for Luxembourg	Rees (2016)
$L_{taxa} = 12.5\%$	Centralized system, normal operation, catastrophic losses during service and maintenance	Airah (2012)

3. RESULTS AND DISCUSSION

Based on the adopted considerations and other information, the proposed static model obtained the following results presented in Tab. 5.

Table 5: Energy, geometric and environmental characteristics of optimized VCRS.

Optimized systems: Brazil.											
Refrigerant	$\mathbf{D_{ii}}$ (mm)	$\mathbf{D_{io}}$ (mm)	$\boldsymbol{\eta_{global}}$ (%)	$\boldsymbol{\eta_v}$ (%)	$\mathbf{L_{cond}}$ (m)	$\mathbf{L_{ev}}$ (m)	$\mathbf{m_{ref}}$ (kg)	**COP**	**TEWI** ($kgCO_2$)	$\mathbf{Q_{ev}}$ (kW)	$\mathbf{\dot{W}_{comp}}$ (kW)
R134a	**5**	**10**	**52.6**	**71.4**	**3.307**	**4.547**	**0.081**	**1.983**	**3300**	**1.126**	**0.568**
R1234yf	5	10	45.6	70.1	4.396	4.213	0.082	1.621	4385	1.319	0.814
Optimized systems: Sweden.											
R134a	**5**	**10**	**52.6**	**71.4**	**3.307**	**4.547**	**0.081**	**1.983**	**1546**	**1.126**	**0.568**
R1234yf	5	10	45.6	70.1	4.396	4.213	0.082	1.621	1872	1.319	0.814
Optimized systems: Luxembourg.											
R134a	**5**	**10**	**52.6**	**71.4**	**3.307**	**4.547**	**0.081**	**1.983**	**15162**	**1.126**	**0.568**
R1234yf	5	10	45.6	70.1	4.396	4.213	0.082	1.621	21389	1.319	0.814

The results presented in Table 4 show that the most efficient optimized refrigeration system from the environmental point of view for Brazil, Sweden and Luxembourg is the system with R134a, because it obtained the lowest TEWI value. In addition, the optimized system with R134a obtained the highest energy performance, however the refrigerant charge within the optimized system with R134a was approximately equal to the optimized system with the R1234yf. Therefore, other alternative refrigerants must be evaluated.

4. CONCLUSIONS

The main conclusions of this paper are summarized as follows:

- The COP_{real} analysis showed that the energy performance of the optimized system with R134a is 22.33% higher than the optimized system with R1234yf.
- The TEWI analysis showed that the environmental performance of the optimized system with R1234yf is 32.2% lower. Therefore, R1234yf cannot replace R134a for this specific application. Other environmentally friendly refrigerants should be evaluated such as R744, R290 and R600a.

5. ACKNOWLEDGEMENTS

The authors appreciate the support of the following Brazilian research financing institutions: CAPES, CNPq, and FAPEMIG. The authors would also like to express their heartfelt gratitude to the late Dr. Ricardo Koury, Professor at the Federal University of Minas Gerais and dear friend, who committed his exceptional vision and experience to the development of the area of refrigeration and heat pump.

6. REFERENCES

Ariah, 2012. The Australian Institute of Refrigeration, Air Conditioning and Heating Methods of calculating Total Equivalent Warning Impact (TEWI).

Antunes, A. H. P., Bandarra Filho, E. P., 2016. “Experimental investigation on the performance and global environmental impact of a refrigeration system retrofitted with alternative refrigerants”. International Journal of Refrigeration, vol. 70, p. 119-127.

Bergman, T. L., Incropera, F. P., Dewitt, D. P., Lavine, A. S. Fundamentals of heat and mass transfer (2011) John Wiley & Sons.

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, n. 3, p. 752-763.

Gnielinski, V., 1976. “New equations for heat and mass transfer in turbulent pipe and channel flow”. Int. Chem. Eng., vol. 16, n. 2, p. 359-368.

Hugmark, G., 1962. “Holdup in gas liquid flow”. Chemical Engineering Progress, n. 58, p. 62-65.

Jarall, Sad, 2012. “Study of refrigeration system with HFO-1234yf as a working fluid”. International Journal of Refrigeration, vol. 35, n. 6, p. 1668-1677.

Makhnatch, P., Khodabandeh, R, 2014. “Selection of low GWP refrigerant for heat pumps by assessing the life Cycle Climate Performance (LCCP)” In: 11TH International Energy Agency Heat Pump Conference. Montreal: International Energy Agency.

Mendoza-Miranda, J. M. et al., 2016. “Comparative evaluation of R1234yf, R1234ze (E) and R450A as alternatives to R134a in a variable speed reciprocating compressor”. Energy, vol. 114, p. 753-766.

Navarro-Esbrí, J. et al., 2013. “Experimental analysis of R1234yf as a drop-in replacement for R134a in a vapor compression system”. International Journal of Refrigeration, vol. 36, n. 3, p. 870-880.

Rees, S. J, 2016. An introduction to ground-source heat pump technology. In: Advances in Ground-Source Heat Pump Systems, p. 1-25.

Shah, M. M., 2016. “Comprehensive correlation for heat transfer during condensation in conventional and mini/micro channels in all orientations”. International Journal of Refrigeration, Elsevier, vol. 67, p. 22 - 41.

Shah, M. M., 2017. “Unified correlation for heat transfer during boiling in plain mini/micro and conventional channels”. International Journal of Refrigeration, Elsevier, vol. 74, p. 606-624.

7. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.