COBEM 2021 26th International Congress of Mechanical Engineering
ABCM Associação Brasileira de Engenharia e Ciências Mecânicas
November 22-26, 2021, Virtual Congress, Brazil

COB-2021-0191
ENERGY AND EXERGO-ECOLOGICAL ANALYSIS OF A REFRIGERATION SYSTEM USING R290 AND R1234YF AS A REPLACEMENT FOR R134A

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Abstract. *This paper presents a mathematical model of a vapor compression refrigeration system operating under steady-state conditions. This model was used to design an energy-efficient system, with geometrically compact heat exchangers, that operates with the most appropriate ecological refrigerant in terms of environmental impact, focusing the refrigerants R290 and R1234yf. The ecological analysis was performed based on TEWI (total equivalent warming impact) and ECOP (ecological coefficient), while the energy and exergy analysis were performed based on COP and Exergy Efficiency. The energy and exergo-ecological analysis indicated that the system operating with R290 is the only system among those evaluated with higher energy, exergy and ecological performance than the system operating with R134a. Therefore, among the evaluated systems, the system with R290 is the only one suitable to replace the system with R134a. Finally, the optimization procedure shows that all systems achieved higher performance for an evaporation temperature of -3°C and a condensation temperature of 50°C. Furthermore, in these systems, the heat exchangers reached an optimal size for the refrigerant diameter of 6 mm and water diameter of 12mm.*

Keywords: *Energy and exergo-ecological analysis, Ecological refrigerants, Steady-state model, Optimization method, vapor compression refrigeration system.*

1. INTRODUCTION

From the 1850s to the 1920s, natural refrigerants were predominantly used in vapor compression refrigeration systems, but the use of these refrigerants was drastically reduced from the 1930s due to the advent of synthetic refrigerants (CFCs and HCFCs). From the 1930s to the mid-1980s, CFCs and HCFCs were used extensively in refrigeration and air conditioning systems. In the late 1980s, the international community established the Montreal protocol to eliminate these synthetic refrigerants due to their high ozone depletion potential (ODP). The high ODP of these refrigerants is related to the presence of the chlorine molecule. Thus, HFCs refrigerants were developed in the early 1990s, with R134a being widely used in domestic and commercial refrigeration systems, according to de Paula (2021).

Over the years, it was observed that the systems operating with these refrigerants have high energy consumption and produce a relevant environmental impact due to high GWP of these refrigerants used (Gill et al.,2019). Thus, in the late 1990s, the international community established the Kyoto protocol to control and phase out HFCs over the years.

In recent years, the international community has established the 21^{st} United Nations Climate Change Conference and Kigali amendment to accelerate the elimination of these non-ecological refrigerants (high GWP). In this way, the current scenario is marked by the search for ecological refrigerants (low GWP) to replace non-ecological refrigerants. In view of this need, several researches have carried out works oriented to develop energy-efficient refrigeration systems that operate with ecological refrigerants (low GWP), such as R290 and R1234yf. Among these works, we can highlight the following papers: de Paula et al. (2019) developed a steady-state model of a refrigeration system to compare the environmental and energy performance of the R1234yf with R134a. Garcia et al. (2018) developed a dynamic model to evaluate the possibility of the R1234yf to be a drop-in replacement for a pre-designed VCRS with R134a. Sánchez et al. (2017) developed a steady-state model and performed an experimental analysis to evaluate the energy performance of R1234yf, R1234ze(E), R600a, R290, R152a as drop-in replacement for R134a. The evaluated parameters were COP, mass flow rate, power consumption, and discharge temperature. In most of these works, mathematical models were used to improve the performance of the systems and to evaluate which is the most appropriate ecological refrigerant to replace the old systems.

The main goal of this paper is to propose a steady-state model to design a vapor compression refrigeration system that simultaneously produces chilled water (5°C) for an indirect expansion air-conditioning system and hot water (50°C) for a restaurant to clean the cutlery, plates and pans. The proposed model uses volumetric and global efficiency curve obtained from commercial compressors available in the market to describe the compression process more realistically and an optimization method (Nelder-Mead Simplex) to design a system with a compact geometric structure and determine the thermodynamic condition in which it operates with the best performance. The ecological analysis was performed based on TEWI (total equivalent warming impact) and ECOP (ecological coefficient), while the energy and exergy analysis were performed based on COP and Exergy Efficiency.

Finally, the energy, exergy and ecological performance of the system operating with R290, R1234yf and R134a was compared to determine the most appropriate ecological refrigerant to replace R134a.

2. MATHEMATICAL MODELING

A mathematical model of a vapor compression refrigeration system (VCRS) operating under steady-state conditions was developed using the Engineering Equation Solver (EES) software. The input variables of the VCRS model are shown on the left, and the output variables on the right, according to Fig. 1. Figure 2 shows the refrigeration plant layout under study, as proposed by de Paula (2021).

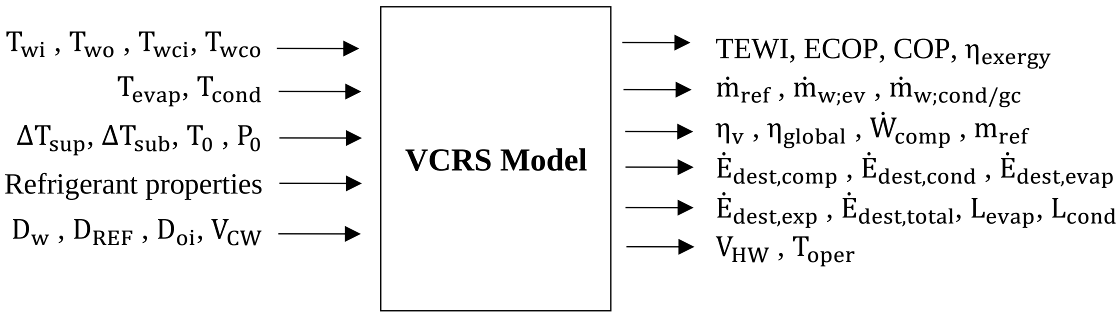

Figure 1. Scheme of the input and output variables of the model.

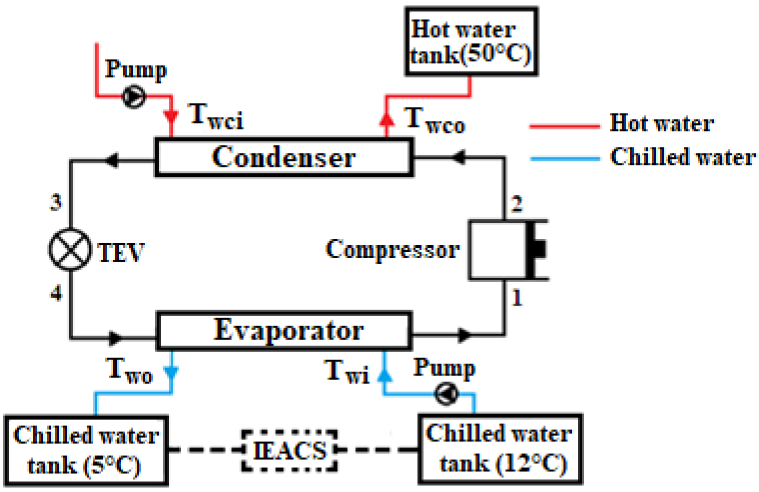

Figure 2. VCRS layout under study.

During the development of the mathematical model, the following aspects were assumed: (1) the pressure drop on the refrigerant side in the evaporator and condenser was not considered; (2) the pressure drop on the waterside in the evaporator and condenser/gas cooler was considered; (3) the contamination of the refrigerant by the compressor oil was not considered; (4) the heat loss of heat exchangers to the environmental was not considered; (5) the expansion device chosen was a thermostatic expansion valve (TEV), and this device was modeled as adiabatic; (6) the pipes between components were considered two meters long each.

This system was designed to produce and store 1200 liters of chilled water (5°C) for an indirect expansion air-conditioning system (IEACS) because the system operates in a period with a lower electricity tariff. Thus, reducing the energy cost, as proposed by de Paula et. al (2020a).

Finally, this system was also designed to simultaneously produce and store at least 600 liters of hot water (50°C) for a restaurant to clean the cutlery, plates and pans, focusing on energy cost reduction and taking advantage of the heat rejected by the condenser.

2.1 Evaporator model

The evaporator used was a concentric tube type, with the refrigerant flowing through the inner tube and the water counterflowing through the annular space. The cooling capacity ($\dot{Q}_{evap}$) and water mass flow rate at the evaporator $(\dot{m}_{w;evap})$ were obtained by the energy balance for steady-state condition and are given by Eq. 1.

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

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], $\dot{m}_{w;evap}$ 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 evaporator length (L_{evap}) was calculated using the logarithmic mean temperature difference method $(\Delta T_{ml;evap})$, Eq. 2, according to (Bergman et al., 2011).

$$\dot{Q}_{evap} = U_{evap} \cdot A_{evap} \cdot \Delta T_{ml;evap} \tag{2}$$

$$U_{evap} = \left(\frac{1}{\bar{h}_{ref;evap}} + \frac{1}{\bar{h}_{w;evap}}\right)^{-1} \tag{3}$$

$$A_{evap} = \pi \cdot D_{REF} \cdot L_{evap} \tag{4}$$

$$\Delta T_{ml;evap} = \frac{[(T_{wi} - T_1) - (T_{wo} - T_4)]}{\ln((T_{wi} - T_1)/(T_{wo} - T_4))} \tag{5}$$

In these equations T_1 is refrigerant temperature at the evaporator outlet [°C], T_4 is refrigerant temperature at the evaporator inlet [°C], U_{evap} is the overall heat transfer coefficient of the evaporator, A_{evap} is the surface area of the evaporator, D_{REF} is the inner diameter of the inner tube (diameter for the refrigerant side), $\bar{h}_{ref;evap}$ is refrigerant average convective coefficient in the evaporator [W/m²K], $\bar{h}_{w;evap}$ is water average convective coefficient in the evaporator [W/m²K] and L_{evap} is evaporator length [m]. Finally, the thermal resistance of conduction(wall) was not considered because this term is small in relation to the thermal resistances of convection.

2.2 Condenser model

The condenser was also a concentric tube type, with the refrigerant flowing through the inner tube and water counterflowing through the annular space. The heat transfer rate at the condenser $(\dot{Q}_{cond})$ and water mass flow rate at the condenser $(\dot{m}_{w;cond})$ were obtained by the energy balance for steady-state condition and are given by Eq. 6.

$$Q_{cond} = \dot{m}_{ref} \cdot (i_2 - i_3) = \dot{m}_{w;cond} \cdot Cp_w \cdot (T_{wco} - T_{wci}) \text{ [kW]} \tag{6}$$

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 condenser length (L_{cond}) is calculated using the logarithmic mean temperature difference method, Eq. 7.

$$Q_{cond} = U_{cond} \cdot A_{cond} \cdot \Delta T_{ml;cond} \tag{7}$$

$$U_{cond} = \left(\frac{1}{\bar{h}_{ref;cond}} + \frac{1}{\bar{h}_{w;cond}}\right)^{-1} \tag{8}$$

$$A_{cond} = \pi \cdot D_{REF} \cdot L_{cond} \quad (9)$$

$$\Delta T_{ml;cond} = \frac{[(T_2 - T_{wco}) - (T_3 - T_{wci})]}{\ln((T_2 - T_{wco})/(T_3 - T_{wci}))} \quad (10)$$

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 $\bar{h}_{ref;cond}$ is refrigerant average convective coefficient in the condenser [W/m²K].

Where $(\bar{h}_{w;evap})$ and $(\bar{h}_{w;cond})$ are determined by correlation of constant heat flow because water has laminar flow. However, $(\bar{h}_{ref;evap})$ and $(\bar{h}_{ref;cond})$ are determined by discretizing the heat exchangers and adopting a constant enthalpy step. Finally, these coefficients are the arithmetic mean of the obtained local convective heat transfer coefficients (h_{local}) in the domain evaluated in each component. At each point, (h_{local}) is obtained from the correlations proposed by Gnielinski (1976) in single-phase region, Shah (2016) and Shah (2017) in two-phase region, respectively, for condensation and for boiling. In addition, the thermal resistance of conduction (wall) was also not considered for the same reason mentioned in subsection 2.1.

2.3 Compressor model

The refrigerant mass flow rate in the compressor $(\dot{m}_{ref})$ is given by Eq. (11).

$$\dot{m}_{ref} = \rho_1 \cdot \forall_{cil} \cdot N \cdot \eta_v \quad (11)$$

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. (12), according to Da Riva (2011).

$$\dot{W}_{comp} = \frac{\dot{m}_{ref} \cdot (i_2 - i_1)}{\eta_{global}} \quad (12)$$

The volumetric (η_v) and global (η_{global}) efficiency curves of a compressor operating with R290, R1234yf and R134a were represents 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, as proposed by de Paula et. al (2020a):

I. For a reference cooling capacity of 1.2 kW.

II. Based on the reference 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. All selected commercial compressors have a frequency of 50 Hz because manufacturers only supply this component with this frequency.

Table 1. Selected Compressors.

Refrigerant	Model	Manufacturer	Displacement (cm³)	Rotation (rpm)
R134a	NT6217ZV	Embraco	20.4	2900
R1234yf	CAJ4492N-FZ	Tecumseh	25.95	2900
R290	NEK6217U	Embraco	14.28	2900

The volumetric and global efficiency curves were obtained utilizing polynomial regression as proposed by Minetto (2011). According to the manufacturer, the experimental data has an uncertainty of 5%. The equations for volumetric and global efficiency are presented in Tab. 2.

Table 2. Global and volumetric efficiency curves.

Refrigerant	R134a	R1234yf	R290
Volumetric efficiency	$\eta_v = 0.8243 - 0.0205 r_p$	$\eta_v = 0.9320 - 0.0484 r_p$	$\eta_v = 0.9038 - 0.0288 r_p$
R^2 of η_v	74.72%	98.33%	93.01%
Global efficiency	$\eta_{global} = 0.1537 + 0.1704 r_p - 0.0269 {r_p}^2 + 0.0015 {r_p}^3$	$\eta_{global} = -0.0083 + 0.2690 r_p - 0.0481 {r_p}^2 + 0.0026 {r_p}^3$	$\eta_{global} = -0.0564 + 0.3750 r_p - 0.0784 {r_p}^2 + 0.0054 {r_p}^3$
R^2 of η_{global}	96.21%	80.98%	95.73%

3. ENERGY ANALYSIS

The coefficient of performance (COP) of the refrigeration system is given by Eq. (13).

$$\mathrm{COP} = \frac{\dot{\mathrm{Q}}_{\mathrm{evap}}}{\dot{\mathrm{W}}_{\mathrm{comp}}+\dot{\mathrm{W}}_{\mathrm{pump;evap}}+\dot{\mathrm{W}}_{\mathrm{pump;cond}}} \tag{13}$$

Where $(\dot{\mathrm{Q}}_{\mathrm{evap}})$, $(\dot{\mathrm{W}}_{\mathrm{pump;cond}})$ and $(\dot{\mathrm{W}}_{\mathrm{pump;evap}})$ are respectively the cooling capacity and electrical power consumption by the pump in the condenser and evaporator. These parameters are calculated by Eq. (14) and Eq. (15).

$$\dot{\mathrm{W}}_{\mathrm{pump;cond}} = \frac{\dot{\mathrm{m}}_{\mathrm{w;cond}} \cdot \Delta \mathrm{P}_{\mathrm{cond}}}{\rho_{\mathrm{w}} \cdot \eta_{\mathrm{pump}}} \tag{14}$$

$$\dot{\mathrm{W}}_{\mathrm{pump;evap}} = \frac{\dot{\mathrm{m}}_{\mathrm{w;evap}} \cdot \Delta \mathrm{P}_{\mathrm{evap}}}{\rho_{\mathrm{w}} \cdot \eta_{\mathrm{pump}}} \tag{15}$$

In these equations, η_{pump}, $\Delta\mathrm{P}_{\mathrm{cond}}$ and $\Delta\mathrm{P}_{\mathrm{evap}}$ are respectively the overall pump efficiency and the pressure drop on the water side in the condenser and evaporator. These parameters are calculated by Eq. (16) and Eq. (17).

$$\Delta \mathrm{P}_{\mathrm{cond}} = \frac{8 \cdot f \cdot L_{total} \cdot \dot{\mathrm{m}}_{\mathrm{w;cond}}^2}{\pi^2 (D_W - D_{REF})^5 \rho_w} \tag{16}$$

$$\Delta \mathrm{P}_{\mathrm{evap}} = \frac{8 \cdot f \cdot L_{total} \cdot \dot{\mathrm{m}}_{\mathrm{w;evap}}^2}{\pi^2 (D_W - D_{REF})^5 \rho_w} \tag{17}$$

Where f is the Darcy friction factor calculated by Shah and London (2014) correlation for laminar flow or Li, Seem and Li (2011) correlation for turbulent flow and $\mathrm{L}_{\mathrm{total}}$ is the total length of the pipe. In addition, $\mathrm{L}_{\mathrm{total}}$ is the sum of the heat exchanger length with an equivalent length. Mathematically, this equivalent length is the sum of the straight lengths of the tubes that connect the tanks (chilled water or hot water) to the respective heat exchanger (evaporator or condenser), with the corresponding straight length related to the curves and connections. In this paper, the equivalent length was considered seven meters long and η_{pump} equal to 0.5.

4. EXERGY ANALYSIS

The exergy efficiency is defined by Eq. (18), according to Shikalgar and Sapali (2019) and Roy and Mandal (2019).

$$\eta_{\mathrm{exergy}} = 1 - \frac{\dot{\mathrm{E}}_{\mathrm{dest,total}}}{\dot{\mathrm{W}}_{\mathrm{comp}}} \tag{18}$$

In this equation $(\dot{\mathrm{E}}_{\mathrm{dest,total}})$ is the total exergy destruction from the system. This is the sum of exergy destruction of the compressor $(\dot{\mathrm{E}}_{\mathrm{dest,comp}})$, condenser $(\dot{\mathrm{E}}_{\mathrm{dest,cond}})$, evaporator $(\dot{\mathrm{E}}_{\mathrm{dest,evap}})$ and expansion valve $(\dot{\mathrm{E}}_{\mathrm{dest,exp}})$.

$$\dot{\mathrm{E}}_{\mathrm{dest,comp}} = \dot{\mathrm{E}}\mathrm{x}_1 - \dot{\mathrm{E}}\mathrm{x}_2 + \dot{\mathrm{W}}_{\mathrm{comp}} \tag{19}$$

$$\dot{\mathrm{E}}_{\mathrm{dest,cond}} = \dot{\mathrm{E}}\mathrm{x}_2 - \dot{\mathrm{E}}\mathrm{x}_3 - \dot{\mathrm{Q}}_{\mathrm{cond}} \cdot \left(1 - \frac{\mathrm{T}_0}{\mathrm{T}_{\mathrm{cond}}}\right) \tag{20}$$

$$\dot{\mathrm{E}}_{\mathrm{dest,evap}} = \dot{\mathrm{E}}\mathrm{x}_4 - \dot{\mathrm{E}}\mathrm{x}_1 + \dot{\mathrm{Q}}_{\mathrm{evap}} \cdot \left(1 - \frac{\mathrm{T}_0}{\mathrm{T}_{\mathrm{evap}}}\right) \tag{21}$$

$$\dot{\mathrm{E}}_{\mathrm{dest,exp}} = \dot{\mathrm{E}}\mathrm{x}_3 - \dot{\mathrm{E}}\mathrm{x}_4 \tag{22}$$

Exergy of a refrigerant circulating in the VCRS is calculated by Eq. (23).

$$\dot{\mathrm{E}}\mathrm{x} = \dot{\mathrm{m}}_{\mathrm{ref}} \cdot [(\mathrm{h} - \mathrm{h}_0) - \mathrm{T}_0(\mathrm{s} - \mathrm{s}_0)] \tag{23}$$

In which (h_0) and (s_0) are the enthalpy and entropy values of the dead state at the pressure (P_0) and temperature (T_0).

5. ECOLOGICAL ANALYSIS

The ecological performance of the system was calculated using two different environmental parameters: ECOP (Ecological coefficient of performance) and TEWI (Total Equivalent Warming Impact). ECOP is calculated by Eq. (24), as suggested by Karakurt et al. (2016).

$$\mathrm{ECOP} = \frac{\dot{Q}_{evap}}{\dot{E}_{dest,total}} \tag{24}$$

The second parameter is the TEWI, this parameter takes into account both direct (related to refrigerant leakage during the lifespan of the system) and indirect (related to the electrical power consumption of the compressor and pumps over the lifespan of the system) emissions and it is calculated by Eqs. (25-27), according to Antunes and Bandarra Filho (2016).

$$\mathrm{TEWI} = \mathrm{TEWI}_{Direct} + \mathrm{TEWI}_{INDirect} \tag{25}$$

$$\mathrm{TEWI}_{Direct} = \mathrm{GWP} \cdot m_{ref;total} \cdot L_{rate} \cdot L_{time} + \mathrm{GWP} \cdot m_{ref} \cdot (1 - \alpha_{recup}) \tag{26}$$

$$\mathrm{TEWI}_{INDirect} = 365 \cdot T_{oper} \cdot (\dot{Q}_{evap}/\mathrm{COP}) \cdot \beta \cdot L_{time} \tag{27}$$

In which $(m_{ref;total})$ is the refrigerant charge of the system, (L_{rate}) is the annual rate of refrigerant emitted (replacement and leaks), (L_{time}) is the useful life of the refrigeration, (α_{recup}) is the refrigerant life recovery rate and (β) is the CO_2 emission factor in producing electricity. Finally, (T_{oper}) is the operating time and it represents the required time for the system to produce 1200 liters of chilled water (5°C) and at least 600 liters of hot water (50°C). The $(m_{ref;total})$ is sum of the refrigerant charge inside the pipe between components $(m_{ref;pipe})$ calculates by Eq. (28) and the refrigerant charge inside the heat exchanger (evaporator and condenser) calculates by Eq. (29).

$$m_{ref;pipe} = \rho_{ref} \cdot \left(\frac{\pi \cdot D_{REF}^2}{4}\right) \cdot L_{pipe} \tag{28}$$

$$m_{ref;HE} = \bar{\rho}_{ref} \cdot \left(\frac{\pi \cdot D_{REF}^2}{4}\right) \cdot L_{HE} \tag{29}$$

Where (L_{pipe}), (L_{HE}), (ρ_{ref}) and $(\bar{\rho}_{ref})$ are the pipe length between components, the heat exchanger length, the density of the refrigerant inside the pipe and the average density of the refrigerant inside the heat exchanger (evaporator or condenser), respectively. As can be seen, L_{HE} refers to L_{evap} for the evaporator and L_{cond} for the condenser. The average density of the refrigerant is obtained by discretizing the heat exchanger following the same procedure described in subsection 2.1. Finally, the average density of the refrigerant $(\bar{\rho}_{ref})$ is the arithmetic mean of the obtained local densities (ρ_{local}) of the refrigerant in the domain evaluated in each component. If the point is in the single-phase region, (ρ_{local}) is determined by the pressure acting in the component and local enthalpy of the point. However, the point is in the two-phase region and (ρ_{local}) is calculated by Eq. (30).

$$\rho_{local} = [\alpha_{local} \cdot \rho_V + (1 - \alpha_{local}) \cdot \rho_L] \tag{30}$$

Where (α_{local}), (ρ_L) and (ρ_V) are the void fraction, the density of the saturated liquid and the density of the saturated vapor, respectively. The void fraction is determined by the correlation Hughmark (1962).

6. OPTIMIZATION METHODOLOGY

The Nelder Mead Simplex method was used to design the heat exchangers with reduced geometric structure and an efficient system from the energy point of view. This algorithm is described in detail in Press et al. (1990), and it was used to determine the evaporator and condenser optimal length based on a range of values for the refrigerant diameter ($\mathbf{D_{REF}}$), evaporation temperature and condensation temperature. Where, D_{REF} varies from 6 mm to 7 mm, the evaporation temperature varies from -5°C to -3°C and condenser temperature varies from 50°C to 55°C. In this paper, it is assumed that the water diameter ($\mathbf{D_W}$) is twice the refrigerant diameter. Finally, this range of values was applied for both evaporator and condenser.

7. SIMULATION PARAMETERS

The thermodynamic considerations adopted were based on literature review and they are presented in Tab. 3. In addition, the main considerations for calculating of the TEWI are presented in Tab. 4.

Table 3. Thermodynamic considerations.

Variable	Value considered
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/gas cooler inlet ($\mathbf{T_{wci}}$)	25 °C
Water temperature in the condenser/gas cooler outlet ($\mathbf{T_{wco}}$)	50 °C
Dead state temperature ($\mathbf{T_0}$) and pressure ($\mathbf{P_0}$)	25°C, 101.3 kPa

Table 4. Considerations for calculating of the TEWI parameter.

Variable	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), de Paula et al. (2020a), de Paula et al. (2020b)
$\beta = 0.082[kgCO_2/kWh]$	Reference value for Brazil	Rees (2016)
$L_{taxa} = 12.5\%$	Centralized system, normal operation, catastrophic losses during service and maintenance	Airah (2012), de Paula et al. (2020a), de Paula et al. (2020b)

8. RESULTS AND DISCUSSION

Based on the adopted simulation parameters, the energy, exergy and ecological performance of the systems were evaluated. These simulations were performed to determine the most suitable ecological refrigerant to replace the R134a and the thermodynamic condition in which the systems operate with the highest energy and exergo-ecological performance.

The results presented in Tab. 5 clearly show that the system operating with R290 has the highest energy and exergo-ecological performance among the analyzed systems due to its higher COP value, higher η_{exergy} value, higher ECOP value and lower TEWI value. All these values were achieved for the following optimal thermodynamic condition: $T_{evap} = -3°C$ and $T_{cond} = 50°C$. Moreover, the optimal lengths of the heat exchangers occurred for D_{REF} = 6mm and D_W = 12mm.

Table 5. Energy, exergy, ecological and geometric characteristics of the optimized systems.

Refrigerant	$\dot{W}_{comp}$ (kW)	$\dot{W}_{total}$ (kW)	$\dot{Q}_{evap}$ (kW)	L_{cond} (m)	L_{evap} (m)	COP (-)	η_{exergy} (%)	ECOP (-)	TEWI (tons of CO_2)	$\dot{E}_{dest,total}$ (kW)	T_{oper} (h/day)	$\dot{m}_{ref}$ (kg/h)
R134a	0.603	0.605	1.24	13.75	22.97	2.05	40.75	3.48	2.97	0.357	7.86	32.13
R1234yf	0.821	0.823	1.46	28.30	6.49	1.78	34.31	2.71	2.47	0.539	6.67	49.49
R290	0.627	0.628	1.30	14.70	7.05	2.07	41.17	3.53	2.12	0.369	7.49	17.93

In general, the COP of the system with R290 is higher due to its lower electrical power consumption in the compressor because this system has the lowest mass flow rate and one of the highest global efficiency. The mass flow rate of the R290 is the lowest due to its lower density and lower compressor displacement volume. However, the reason for the COP of the system with R290 is higher than the COP of the system with R134a is because the system with R290 has greater cooling capacity. This fact can be explained by the greater variation of enthalpy in the evaporator of the system with R290.

The system with R290 has the highest exergy performance. Most of the time, the exergy efficiency of the system with R290 is greater than the exergy efficiency of the other systems due to its lower total exergy destruction. However, the reason that the η_{exergy} of the system with R290 is greater than the η_{exergy} of the system with R134a is due to the effect caused by $\dot{W}_{comp}$ on η_{exergy} is greater than the effect caused by $\dot{E}_{dest,total}$ on η_{exergy}.

The system with R290 also has the highest ecological performance due to its higher ECOP and lower TEWI. However, the reason that the ECOP of the system with R290 is greater than the ECOP of the system with R134a is due to the effect caused by $\dot{Q}_{evap}$ on ECOP is greater than the effect caused by $\dot{E}_{dest,total}$ on ECOP.

To obtain more information about the TEWI values presented in Tab. 5, the $TEWI_{Direct}$ and $TEWI_{INDirect}$ values are shown in Tab. 6 for each system described above. In addition, the refrigerant charge inside of each system is also presented in Tab. 6.

Table 6. TEWI values related to direct and indirect emissions.

$\dot{Q}_{evap.ref} = 1.2$ kW, $T_{evap} = -3$°C, $T_{cond} = 50$°C				
Refrigerant	**$m_{ref;total}$ [g]**	**TEWI [kgCO$_2$]**	**$TEWI_{Direct}$ [kgCO$_2$]**	**$TEWI_{INDirect}$ [kgCO$_2$]**
R134a	280.2	2969	834.8	2134.2
R1234yf	367.8	2466	3.2	2462.8
R290	110.9	2119	4.8	2114.2

As noted in Tab. 6, the environmental impact due to indirect emissions ($TEWI_{INDirect}$) is the most significant contribution, and it corresponds to 71.88% of TEWI value for the system with R134a and more than 99% of TEWI value for other systems with ecological refrigerant. Therefore, to reduce the environmental impact of a refrigeration system, it is not enough to choose a refrigerant based only on GWP, because this parameter does not evaluate the environmental impact related to indirect emissions.

9. CONCLUSIONS

The main conclusions of this paper are summarized as follows:

- Analyzing the COP and η_{exergy}, it is observed that the system operating with R290 is the most efficient from an energy point of view.
- Evaluating the ECOP and TEWI, it is observed that the system operating with R290 produces the lowest environmental impact among the evaluated systems.
- The simulation results show that all systems achieved the highest energy performance for the following thermodynamic condition: T_{evap} = -3°C and T_{cond} = 50°C. In this context, all systems obtained the optimal length of the evaporator and condenser for D_{REF} = 6mm and D_W =12mm.
- According to TEWI results, the environmental impact due to indirect emissions is the most significant produced by a refrigeration system.

Therefore, among the systems evaluated, the results show that the system with R290 is the only system suitable to replace the system with R134a.

10. ACKNOWLEDGMENTS

The authors appreciate the support of the following Brazilian research financing institutions: CAPES, CNPq, and FAPEMIG.

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