THERMO-ECONOMIC ANALYSIS FOCUS ON REFRIGERANT SELECTION IN CO_2 SECONDARY CASCADE REFRIGERATION SYSTEM

Hugo José Abreu de Souza, João Victor Guimarães Bastos, Ralney Nogueira de Faria, Tiago de Freitas Paulino, Paulo Eduardo Lopes Barbieri[a*]

Federal Center of Technological Education of Minas Gerais (CEFET-MG), Av. Amazonas, 7675, Nova Gameleira, 30510-000, Belo Horizonte (MG), Brazil
[a] barbieri@cefetmg.br

ARTICLE INFO

Keywords: *Cascade refrigeration, thermo-economic analysis, refrigerant selection, R744.*

Received: Mar 08, 2025
Reviewed: Mar 18, 2025
Accepted: Mar 23, 2025

ABSTRACT

Cascade refrigeration systems are widely used in commercial refrigeration because of their compactness and their ability to provide two different cooling demands at medium and low temperature with high values of COP compared with single-stage vapor compression cycles. Thus, for low-temperature cycle, CO_2 (R744) is commonly used due to its low-cost, environmental advantages and the high working pressures at low temperatures. For the high temperature cycle, R134a and R404A, as well as its corresponding drop-ins, are the most used solutions due to its feasibility, availability, and good performance. Because of environmental issues related to global warming effects, research on alternative refrigerant has attracted interest. Despite the large number of works, there are no study that investigated environment friendly refrigerant alternative for high-temperature circuit in a combined CO_2 Secondary Cascade System. This paper presents a comparative study with different refrigerants to replace R134a, it considers R32, R450A, R454C, R513A, R1234yf, R290, R600a and R1270 as options in high temperature cycle. Evaluation and optimization of the thermodynamic and economic performance were conducted by EES (Engineering Equation Solver). From the simulations results is demonstrated that the use of R1270 is the best option for a high temperature cycle.

NOMENCLATURE

c_p specific heat at constant pressure [$J.kg^{-1}.K^{-1}$]
D diameter [m]
(dP/dZ)fr refrigerant pressure loss by friction [Pa]
g acceleration of gravity [$m.s^{-2}$]
G mass flux [$kg.s^{-1}.m^{-2}$]
h specific enthalpy [$kJ.kg^{-1}$]
H_f heat transfer coefficient between the wall and the refrigerant [$W.m^{-2}.K^{-1}$]
L coil length [m]
$\dot{m}$ mass flow rate [$kg.s^{-1}$]
N rotation speed [rps]
p perimeter [m]
P pressure [Pa]
S solar radiation [$W.m^{-2}$]
T temperature [K]
U_L combined coefficient involving radiation and convection between the absorber / coil and the environment [$W.m^{-2}.K^{-1}$].
v specific volume [$m^3.kg^{-1}$]
V volumetric displacement [$m^3.rev^{-1}$]
x quality
y dependent variable
W distance between the centers of two adjacent tubes

Greek symbols

α void fraction
η efficiency
θ inclination of the absorber relative to the horizontal
ρ density [$kg.m^{-3}$]

* Corresponding author: Paulo Eduardo Lopes Barbieri. Federal Center of Technological Education of Minas Gerais (CEFET-MG), Av. Amazonas, 7675, Nova Gameleira, 30510-000, Belo Horizonte (MG), Brazil. barbieri@cefetmg.br.

Superscripts

0 initial value

1. INTRODUCTION

Energy saving and environment protection has been an important issue all over the world. Supermarkets have massive cooling load and traditionally employ synthetic refrigerants. Thus, they contribute significantly towards direct and indirect global warming (Purohit et al, 2017). Approximately half of the energy consumption in a supermarket is associated with the refrigeration system (Mota-Babiloni et al, 2015). In addition fifteen percent of the electricity consumed worldwide is used for refrigeration and the cold-chain accounts for approximately 1% of CO_2 emission in the world (Tassou et al., 2011).

Since the Montreal (1987) and Kyoto (1997) protocols, many efforts were performed to evaluate replacements of the CFC, HCFC and HFC (Bolaji and Huan, 2013, Sharma et al., 2014, Antunes and Bandarra Filho, 2016, Mota-Babiloni et al, 2017, Panato et al, 2017, Makhnatch et al, 2019, Heredia-Aricapa et al, 2020).

Carbon dioxide has received considerable attention as an alternative to the commonly used synthetic refrigerants in supermarket refrigeration systems, in an effort to develop systems with lower environmental impact (Sharma et al., 2014). Although CO_2 has a high critical pressure (7.38 MPa) and a low critical temperature (30.97 °C), its high operating pressure leads to a high vapor density and thus a high volumetric refrigeration capacity. The volumetric refrigerating capacity of CO_2 (22,545 kJ/m³ at 0 °C) is 3-10 times larger than CFC, HCFC, HFC and HC refrigerants (Kim et al., 2004). In addition, carbon dioxide has no Ozone Depletion Potential (ODP); a Global Warming Potential (GWP) of one; it is nontoxic, nonflammable and inexpensive (Sharma et al., 2014). Carbon dioxide as a refrigerant has been reported to be used in indirect, cascade and transcritical cycles (Gupta, D.K., Dasgupta, 2014).

A cascade system is comprised of separate high-temperature and low-temperature circuits, coupled through a heat exchanger called the cascade condenser or cascade heat exchanger. The cascade condenser functions as an evaporator for the high temperature circuit and a condenser for the low temperature circuit. Generally, the high-temperature circuit is a single-stage direct expansion system, but the low temperature circuit can either be a direct expansion system or a secondary loop system. These configurations of CO_2 system are presented in detail by Sharma *et.al* (2014) and Bellos and Tzivanidis (2019).

Cascade systems and secondary loop systems using CO_2 as a refrigerant can be used to reduce the direct impact on the environment due to their lower HFC refrigerant charge (Sharma et al., 2014) and this system seems to be more efficient than the others, especial for the warm climates (Catalán-Gil et al 2018). The use of CO_2 in the low stage solves the problem of the low critical point which leads to the transcritical operation and makes the system to operate with lower pressure levels (Bellos and Tzivanidis, 2019). In that case an environment friendly refrigerant, CO_2, is used in low-temperature circuit. As well as an environment friendly alternative can be chosen for high-temperature circuit.

In the literature, there are many studies that use CO_2 direct expansion cascade (Figure 1a) as the configuration of the system. In that case CO_2 is the refrigerant in low temperature cycle and other refrigerant take place in high-temperature circuit. In order to select an environment friendly refrigerant to high-temperature circuit different studies were carried out such as Bellos and Tzivanidis (2019), Massuchetto *et. al* (2019) and De Paula *et. al* (2021). They reported the use of several refrigerants in high-temperature circuit, R290, R1234yf, R152a, R717 (De Paula *et. al*, 2021), R600, R600a, R1270 (Bellos and Tzivanidis, 2019), RE170 (Massuchetto *et al.*, 2019), R134a (Cabello *et al.*, 2017), R1234ze (Llopis *et al.*, 2015).

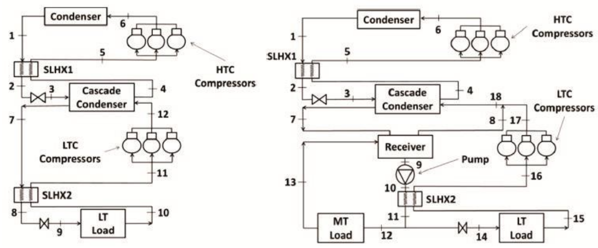


Figure 1. Schematic of CO_2 secondary/cascade refrigeration systems: (a) CO_2 direct expansion cascade system; (b) Combined CO_2 secondary cascade system. (Sharma *et al.*, 2014).

For a supermarket, if the chosen configuration is the CO_2 direct expansion cascade, two parallel systems need to take place. One to produce cold in low temperature and other to produce cold in medium temperature. One alternative to produce cold a low and medium temperature with the same system is used the combined CO_2 secondary cascade system (Figure 1b).

Considering the studies previously presented, there are works focus in presented an environment friendly refrigerant to high-temperature circuit for a CO_2 direct expansion cascade system. However, to the best of the author's knowledge, there are no study that investigated environment friendly refrigerant option for high-temperature circuit in a combined CO_2 Secondary Cascade System. In that way, this paper presents a comparative study with different refrigerants alternatives to replace R134a in an R134a/R744 combined secondary cascade system all of them with low GWP than R134a. It considered R32, R450A, R454C, R513A, R1234yf, R290, R600a and R1270 as options in high temperature cycle. Evaluation and optimization of the thermodynamic (energy and exergy) and economic performance were conducted. The major contribution of this paper is to quantify the benefits in energy and economic perspective for the various low-GWP refrigerants in high-temperature circuit of combined CO_2 secondary cascade system.

The information of the refrigerants selected to replace R134a in the high temperature cycle are presented in Table 1.

- The pressure drop and heat loss/gain in the lines were ignored;
- The pump isentropic efficiency ($\eta_{s,pump}$) is assumed to be 65%;
- Work input by the pump is assumed 1% of work input of compressor low-temperature, $\dot{W}_{pump} = 0.01\dot{W}_{comp,LT}$ (Sharma et al. 2014);
- The kinetic and potential energies are not considered;
- Evaporator and Condenser Fans are not considered;
- Saturated refrigerant occurs at the exit of evaporator LT, cascade condenser, condenser.

Table 1. Information of the refrigerants selected by the high temperature cycle [CoolProp; Mota-Babiloni et al., 2017; Dai et al., 2015; Gasservei; Climalife].

Refrigerant	R134a	R32	R450A	R513A	R454C	R1234yf	R290	R600a	R1270
Normal boiling point [°C]	-26	-51.7	-23.1	-29.2	-45.6	-29.4	-42	-12	-47.7
Critical temperature [°C]	101	94.7	104.4	97.7	85.7	94.7	96.68	134.7	91.06
Critical pressure [kPa]	4059	3382	3820	3855	4319	3382	4247	3640	4555
GWP 100 years	1370	675	604	573	148	4	20	20	1,8
Atmospheric lifetime [years]	13.4	4.9	-	-	-	0.029	0.041	0.016	0.001
ASHRAE 34 – Safety code	A1	A2	A1	A1	A2L	A2L	A3	A3	A3

3. Mathematical model

In this paper a combined CO_2 secondary/cascade refrigeration system proposed by Sharma et al. (2014) was modeled. The schematic diagram of the modeled system is shown in Figure 2.

In this refrigeration system CO_2 is pumped around the medium temperature load. It is volatile, so unlike a conventional secondary fluid such as glycol it does not remain as a liquid. Instead, it partially evaporates, providing a significantly greater cooling capacity. This reduces the pump power required, and the temperature difference needed at the heat exchanger.

3.1 Thermodynamic Modeling

Thermodynamic modeling includes mass, energy and entropy balance for all 122 components of the cycle along with the reasonable assumption as follows:

- The system operates under steady state;
- Only saturated liquid and saturated vapor exit the receiver;
- The expansion processes were considered isenthalpic;

The following sets of governing equations are applied to all components of cycle while considering it as a control volume.

I. Mass Balance:

$$\sum \dot{m}_{in} = \sum \dot{m}_{out} \tag{1}$$

II. Energy Balance:

$$\dot{Q} + \sum_{in} \dot{m}_{in} h_{in} = \sum_{out} \dot{m}_{out} h_{out} + \dot{W} \tag{2}$$

III. Entropy Generation:

$$\dot{S}_{ger} = \sum_{out} \dot{m}_{out} s_{out} - \sum_{in} \dot{m}_{in} s_{in} - \sum_{k} \frac{Q_k}{T_k} \geq 0 \tag{3}$$

IV. Exergy Destruction (Gouy–Stodola equation):

$$\dot{E}x_D = T_o \dot{S}_{ger} \tag{4}$$

where $\dot{m}$ is the mass flow rate, $\dot{Q}$ is the heat transfer, h is the specific enthalpy, $\dot{W}$ is the compressor or pump power, s is the specific entropy, T is the absolute temperature, $\dot{S}_{gen}$ is the entropy generation and $\dot{E}x_D$ is the exergy destruction. Governing equations used for the thermodynamic modeling of individual components of systems are given in Table 1; wherein the state points are referred to Figure 2.

Table 2. Governing equations for different components of system.

Compressor LT	**Compressor HT**
$\dot{W}_{comp,LT} = \dot{m}_{16}(h_{17} - h_{16})$ (5)	$\dot{W}_{comp,HT} = \dot{m}_5(h_6 - h_5)$ (8)
$\dot{E}x_{D,comp,LT} = T_o\dot{m}_{16}(s_{17} - s_{16})$ (6)	$\dot{E}x_{D,comp,HT} = T_o\dot{m}_5(s_6 - s_5)$ (9)
$\eta_{s,comp,LT} = \frac{(h_{17s} - h_{16})}{(h_{17} - h_{16})}$ (7)	$\eta_{s,comp,HT} = \frac{(h_{6S} - h_5)}{(h_6 - h_5)}$ (10)
Pump	**Evaporator LT**
$\dot{W}_{pump} = \frac{(\dot{m}_9 v_9 \Delta P_{pump})}{\eta_{s,pump}}$ (11)	$\dot{Q}_{evap,LT} = \dot{m}_{14}(h_{14} - h_{15})$ (14)
$\dot{E}x_{D,pump} = T_o\dot{m}_9(s_{10} - s_9)$ (12)	$\dot{E}x_{D,evap,LT} = T_o\left(\dot{m}_{14}(s_{15} - s_{14}) - \frac{\dot{Q}_{evap,LT}}{T_{camara,LT}}\right)$ (15)
$\eta_{s,pump} = \frac{(h_{10s} - h_9)}{(h_{10} - h_9)}$ (13)	
Evaporator MT	**Condenser**
$\dot{Q}_{evap,MT} = \dot{m}_{12}(h_{13} - h_{12})$ (16)	$\dot{Q}_{cond} = \dot{m}_6(h_6 - h_1)$ (18)
$\dot{E}x_{D,evap,MT} = T_o\left(\dot{m}_{12}(s_{13} - s_{12}) - \frac{\dot{Q}_{evap,MT}}{T_{camara,MT}}\right)$ (17)	$\dot{E}x_{D,cond} = T_o\left(\dot{m}_6(s_1 - s_6) + \frac{\dot{Q}_{cond}}{T_{amb}}\right)$ (19)
Expansion Valve 01	**Expansion Valve 02**
$h_2 = h_3$ (20)	$\dot{m}_{11} - \dot{m}_{12} = \dot{m}_{14}$ (22)
$\dot{E}x_{D,EV1} = T_o\dot{m}_2(s_3 - s_2)$ (21)	$h_{11} = h_{14}$ (23)
	$\dot{E}x_{D,EV2} = T_o\dot{m}_{14}(s_{14} - s_{11})$ (24)
SHX 1	**SHX 2**
$\dot{Q}_{SHX1} = \dot{m}_1(h_1 - h_2) = \dot{m}_4(h_5 - h_4)$ (25)	$\dot{Q}_{SHX2} == \dot{m}_{10}(h_{10} - h_{11}) = \dot{m}_{15}(h_{16} - h_{15})$ (28)
$\dot{E}x_{D,SHX1} = T_o(\dot{m}_1(s_2 - s_1) + \dot{m}_4(s_5 - s_4))$ (26)	$\dot{E}x_{D,SHX2} = T_o(\dot{m}_{10}(s_{11} - s_{10}) + \dot{m}_{15}(s_{16} - s_{15}))$ (29)
$\varepsilon_{SHX1} = \frac{\dot{m}_1(h_1 - h_2)}{\dot{m}_4 cp_4(T_1 - T_4)} = \frac{\dot{m}_4(h_5 - h_4)}{\dot{m}_4 cp_4(T_1 - T_4)}$ (27)	$\varepsilon_{SHX2} = \frac{\dot{m}_{10}(h_{10} - h_{11})}{\dot{m}_{15}cp_{15}(T_{10} - T_{15})} = \frac{\dot{m}_{15}(h_{16} - h_{15})}{\dot{m}_{15}cp_{15}(T_{10} - T_{15})}$ (30)
Cascade-Condenser	**Receiver**
$\dot{m}_{18} = \dot{m}_8 + \dot{m}_{17}$ (31)	$\dot{m}_7 + \dot{m}_{13} = \dot{m}_9 + \dot{m}_{18}$ (34)
$\dot{Q}_{CC} = \dot{m}_3(h_4 - h_3) = \dot{m}_7(h_{18} - h_7)$ (32)	$\dot{m}_7 h_7 + \dot{m}_{13} h_{13} = \dot{m}_8 h_8 + \dot{m}_9 h_9$ (35)
$\dot{E}x_{D,CC} = T_o(\dot{m}_3(s_4 - s_3) + \dot{m}_7(s_7 - s_{18}))$ (33)	$\dot{E}x_{D,REC} = T_o((\dot{m}_8 s_8 + \dot{m}_9 s_9) - (\dot{m}_7 s_7 + \dot{m}_{13} s_{13}))$ (36)
$COP = \frac{\dot{Q}_{evap,MT} + \dot{Q}_{evap,LT}}{\dot{W}_{comp,LT} + \dot{W}_{comp,HT} + \dot{W}_{pump}}$ (37)	$\dot{E}x_{D,TOTAL} = \sum_{Components} \dot{E}x_D$ (38)
	$\eta_{II} = 1 - \left(\frac{\dot{E}x_{D,TOTAL}}{\dot{W}_{comp,LT} + \dot{W}_{comp,HT} + \dot{W}_{pump}}\right)$ (39)

In Equations 5 to 39: η_s is the isentropic efficiency, ε is the effectiveness of the heat exchanger, c_p is the specific heat, v is the specific volume, COP is the coefficient of performance and η_{II} is the exergetic efficiency.

In Eq. (7), the isentropic efficiency of compressor LT ($\eta_{s.com,LT}$) depends on the pressure ratio (rpcomp,LT) and it is given by Jain et al (2015a):

$$\eta_{s,comp,LT} = 0.00476\, rp^2_{comp,LT} - 0.09238\, rp_{comp,LT} + 0.89810 \qquad (40)$$

In Eq. (10), the isentropic efficiency of compressor HT ($\eta_{s.com,HT}$) depends on the pressure ratio (rpcomp,HT) and it is given by Jain et al (2015a):

$$\eta_{s,comp,HT} = -0.00097\, rp^2_{comp,HT} - 0.01026\, rp_{comp,HT} + 0.83955 \qquad (41)$$

3.2. Economic analysis

In the present study, the investment cost (C) of all the heat exchangers, compressor, pump and expansion valve is considered whereas the cost of refrigerant and connecting pipes is neglected. Sanaye and Malekmohammadi (2004) reported that the sum of the costs of other items such as valves, refrigerant, connecting pipes and the structure of the system contributes 0.84% of the total investment cost. In this research, the reference component costs of a particular type and size reported by Rezayan e Behbahaninia (2011), Mosaffa et.al (2016), Jain et.al (2016) and Jain et.al (2015b) are applied for the components of systems are given in Table 2.

Table 3. Investment cost of each component of the system.

Compressor	**Pump**
$C_{comp,LT} = \frac{573\,\dot{m}_{16}}{0.8996 - \eta_{s,com,LT}} rp_{comp,LT} \ln(rp_{comp,LT})$ (49)	$C_{pump} = 705.48\dot{W}^{0.71}_{pump}\left(1 + \frac{0.2}{1 - \eta_{s,pump}}\right)$ (51)
$C_{comp,HT} = \frac{573\,\dot{m}_5}{0.8996 - \eta_{s,comp,HT}} rp_{comp,HT} \ln(rp_{comp,HT})$ (50)	
Condenser	**Receiver**
$C_{cond} = 1397\, A^{0.89}_{cond}$ (52)	$C_{REC} = 280.3(\dot{m}_7 + \dot{m}_{13})^{0.67}$ (53)
Evaporator	**Cascade-Condenser**
$C_{evap,MT} = 1397\, A^{0.89}_{evap,MT}$ (54)	$C_{CC} = 385.5\, A^{0.65}_{CC}$ (56)
$C_{evap,LT} = 1397\, A^{0.89}_{evap,LT}$ (55)	
Expansion Valve	**SHX**
$C_{EV1} = 114.5\, \dot{m}_1$ (57)	$C_{SHX1} = 516.621\, A_{SHX1} + 268.45$ (59)
$C_{EV2} = 114.5\, \dot{m}_{14}$ (58)	$C_{SHX2} = 516.621\, A_{SHX2} + 268.45$ (60)

The investment cost of the heat exchangers can be expressed as a function of heat transfer area (A) of each component, which can be obtained by Eq. (42) (Bejan and Kraus, 2003).

$$\dot{Q} = UA\Delta T_{ml} \qquad (42)$$

where A is the heat transfer area of heat exchangers, U is the overall heat transfer coefficient and ΔT_{lm} is the logarithmic mean temperature difference. In the present study the overall heat transfer coefficient (U) is provided by Silva (2019).

Total cost rate (total annual cost, C_{total}) of the overall system is given by Eq. (43), including investment and maintenance cost rate ($C_{inv,main}$), operational cost rate (C_{op}), and penalty cost rate of CO_2 emission (C_{env}).

$$C_{total} = C_{inv\ main,} + C_{op} + C_{env} \qquad (43)$$

Thus, investment and maintenance cost rate of the overall system can be estimated by Eq. (44).

$$C_{inv,main} = (CRF \cdot MF)\sum C_k \qquad (44)$$

where C_k is the investment cost of each component (Table 2), CRF is capital recovery factor, depends on interest rate (i) and system lifetime (n), as described by Eq. (45).

Maintenance factor (MF) is taken as 1.06 (Jain et al, 2016).

$$CRF = \frac{i(1+i)^n}{(1+i)^n - 1} \qquad (45)$$

Operational cost (C_{op}) of the cycles is mainly due to the electricity consumed by compressors and pump, as described by Eq. (46).

$$C_{op} = t_{op}\, C_{ele}[\dot{W}_{comp.LT} + \dot{W}_{comp.HT} + \dot{W}_{pump}] \qquad (46)$$

where t_{op} is the annual operating hours and C_{ele} is the electricity price.

Due to the increasing environmental concerns and specifically global warming issues, considering the environmental impacts is becoming essential in modeling of the thermal systems. The penalty cost for CO_2 emission (C_{env}) is calculated by Eq. (47) (Aminyavari et al, 2014).

$$C_{env} = m_{CO2} C_{CO2} \tag{47}$$

In Eq. (47), C_{CO2} is the cost of unit carbon dioxide emission. m_{CO2} is the CO_2 emission mass, which can be determined using emission conversion factor is calculated by Eq. (48) (Sanaye and Shirazi, 2013).

$$m_{CO2} = t_{op}\,\lambda\,[\dot{W}_{comp.LT} + \dot{W}_{comp.HT} + \dot{W}_{pump}] \tag{48}$$

where λ is the emission conversion factor.

4. Methodology

4.1. Simulation parameters

The thermo-economic analyses and optimization of the combined CO_2 secondary/cascade refrigeration system are conducted based on the first and the second laws of thermodynamics. The main thermodynamic parameters for the simulation of the desired cycle system are listed in Table 3. The reference state in exergy destruction of this study is expressed as: $T_o=T_{amb}$ and P_o=101.3 kPa.

Thermo-economic modeling of the system has been conducted based on simulation code in EES (Engineering Equation Solver) (Klein, 2020). Ambient temperature ranging between 20°C and 40°C (20°C, 25°C, 30°C, 35°C e 40°C) was used to determine the performance of the system and nine fluids were considered R134a, R32, R450A, R454C, R513A, R1234yf, R290, R600a and R1270 as options in high temperature cycle. The condensation temperature of the cascade-condenser heat exchanger was set equal to the evaporation temperature of the evaporator MT.

Table 4. Input data for combined CO_2 secondary/cascade refrigeration system (Sharma et.al, 2014)

Parameter	Value
Medium-temperature space[°C]	-5
Low-temperature space [°C]	-30
Medium cooling load [kW]	120
Low cooling load [kW]	65

The overall heat transfer coefficient, U, is an important parameter for calculating the investment cost of heat exchangers. The reference values of overall heat transfer coefficients for all heat exchangers are listed in Table 4.

Table 4 - Reference values of overall heat transfer coefficients (Silva, 2019).

Heat Exchanger	*U [W/m² °C]*
Condenser	98.7
Cascade-condenser	220.5
Evaporator LT	95.6
Evaporator MT	95.6
SHX 1	12665
SHX 2	3595

For the considered combined CO_2 secondary/cascade refrigeration system the parameters for the economic analysis are given in Table 5.

Table 5. Parameters for the economic analysis (Aminyavari et al, 2014).

Parameter	Value
Maintenance factor (*MF*)	1.06
Annual operating hours (t_{op}),	7000 h
System life time (*n*)	15 years
Interest rate (*i*)	14%
Electricity price (C_{ele}),	0.06 US$/kWh
Cost of unit carbon dioxide emission (C_{CO2})	90 US$/ ton_{CO2}
Emission conversion factor (*λ*)	0.11 kg/kWh (MCTIC,2018)

4.2. System optimization

The present optimization process is devoted to investigate and optimize the combined CO_2 secondary/cascade refrigeration system for typical operational conditions for each working fluid. The evaporator temperature difference LT ($\Delta T_{evap,LT}$), evaporator temperature difference MT ($\Delta T_{evap,MT}$), cascade-condenser temperature difference ($\Delta T_{cascade}$), condenser temperature difference (ΔT_{cond}), superheat SHX 1 ($\Delta T_{sup,SHX1}$), superheat SHX 2 ($\Delta T_{sup,SHX2}$) and pump circulation ratio (RC_{pump}) are selected as the continuous decision variables and the examined range are given in Table 6. In this study, the considered objective function is the exergy destruction, Eq. (38) of the whole cascade system which should be minimized. The Direct Optimization Algorithm is used for minimization of exergy destruction.

5. Results and Discussion

5.1. Model validation

To validate the model of this work, the data calculated by the current model is compared to those of Sharma *et.al* (2014). As shown in Figure 3, data of the present work matches very well to the ones reported by Sharma et.al, (2014), in other words, the largest deviation is 0.0172%.

Table 6. Considered design parameters or system optimization and their corresponding range of variation.

Parameter	Range of variation
$\Delta T_{cascade}$, *[°C]*	3 - 15
ΔT_{cond}, *[°C]*	5 - 20
$\Delta T_{evap,LT}$, *[°C]*	5 - 15
$\Delta T_{evap,MT}$ *[°C]*	5 - 15
$\Delta T_{sup,SHX1}$, *[°C]*	5 - 15
$\Delta T_{sup2,SHX2}$, *[°C]*	5 - 15
RC_{pump} *[-]*	1.25 - 3

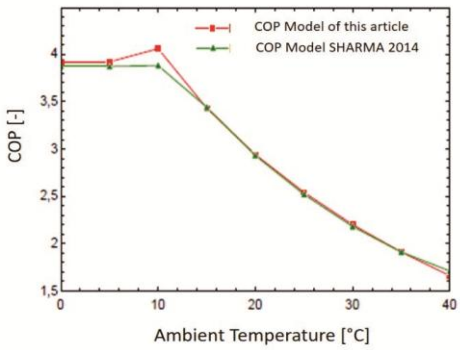


Figure 3. Model validation for different ambient temperatures

5.2. Model results

The optimal differences of temperature, superheat and pump circulation ratio for combined CO_2 secondary/cascade refrigeration system are shown in Table 7. Note that the superheat SHX 1(ΔTsup,SHX1) is the same for all fluids evaluated except for the R32.

The COP and Exergy Destruction of combined CO_2 secondary/cascade refrigeration system at different ambient temperature is shown in Figure 4.

It is observed in Figure 4(a) that the COP increases with the reduction of the ambient temperature, Figure 4(b) shown that the exergy destruction rises with the increase of the ambient temperature for all the fluids evaluated. In Figure 4 it is also noted that there is a small variation in the values of COP and Exergy Destruction between the refrigerants evaluated in which R1270 and R600a presented a slightly better behavior.

Table 7. Optimal differences of temperature, superheat and pump circulation ratio for combined CO_2 secondary/cascade refrigeration system.

Refrigerant	$\Delta T_{cascade}$ *[°C]*	ΔT_{cond} *[°C]*	$\Delta T_{evap,LT}$ *[°C]*	$\Delta T_{evap,MT}$ *[°C]*	$\Delta T_{sup,SHX1}$ *[°C]*	$\Delta T_{sup,SHX2}$ *[°C]*	RC_{pump} *[-]*
R134a	3.02	5.03	5.02	5.02	15.00	5.02	1.84
R32	3.02	5.03	5.02	5.02	5.02	5.02	1.84
R450A	3.02	5.03	5.02	5.02	15.00	5.02	1.84
R454C	3.02	5.03	5.02	5.02	15.00	5.02	1.84
R513A	3.02	5.03	5.02	5.02	15.00	5.02	1.84
R1234yf	3.02	5.03	5.02	5.02	15.00	5.02	1.84
R290	3.02	5.03	5.02	5.02	15.00	5.02	1.84
R600a	3.02	5.03	5.02	5.02	15.00	5.02	1.84
R1270	3.02	5.03	5.02	5.02	15.00	5.02	1.84

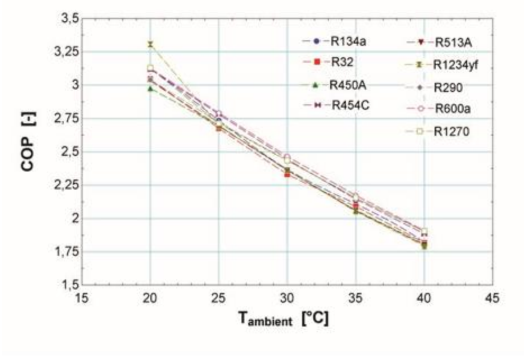
The volumetric flow rate in the suction of the compressor HT and the discharge temperature in compressor HT of combined CO_2 secondary/cascade refrigeration system at different ambient temperature is shown in Figure 5.

The volumetric flow rate in the suction of the compressor is a parameter that influences the dimensions of the compressor, thus, higher volumetric flow rates are associated with a higher capacity compressor. In Figure 5(a), it is observed that R32 has the lowest volumetric flow rate in the suction. On the other hand, R600a is the one with the highest volumetric flow rate in the suction. It is also observed that the ambient temperature has little influence on the volumetric flow rate in the compressor suction.

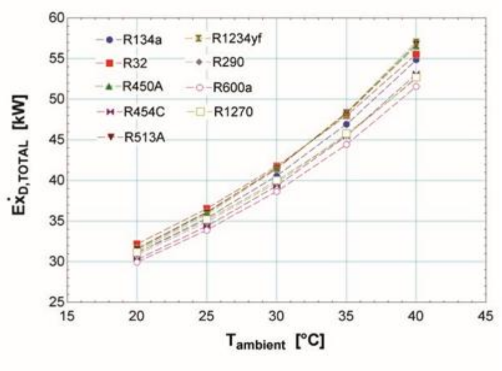
In Figure 5(b), it is observed that R32 has the highest compressor discharge temperature in relation to the other refrigerants evaluated. The compressor discharge temperature is another parameter that must be evaluated in order to avoid premature decomposition of the lubricating oil.

Considering the results presented in Figure 4(a), it can be observed that, regardless of the refrigerant fluid, the variation in the ambient temperature had a great influence on COP. Thus, in order to assess the behavior of the system in a more critical condition (lower COP), Table 8 presents the results obtained by the thermo-economic model of the system for an ambient temperature of 40°C.

From the analysis of the data in Table 8, it is possible to notice that all refrigerants presented compressor discharge temperature lower than 86°C except R32 (129°C). R600a presents the biggest volumetric flow rate and R32 has the smallest one. The smaller exergy efficiency is 37.59 (R1234yf) and the biggest is 40.07 (R600a). In terms of COP, it varies between 1.79 and 1.9. The best performance is achieved by hydrocarbon fluids (R600a, R290 and R1270) both with COP equal to 1.90. The synthetic fluid that reached better COP is R454C. The lower total cost is presented by hydrocarbon fluids, among its, R1270 presents the lower one. In addition, the synthetic fluid that came closest to hydrocarbons was R454C, it has a total cost about 3% higher.

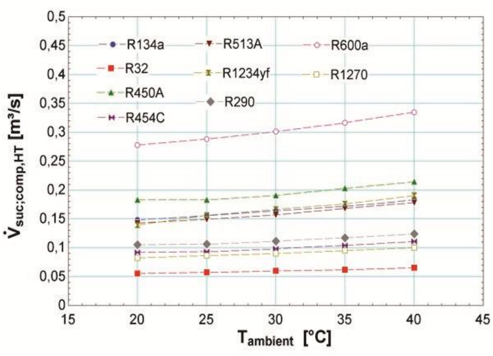


(a) COP

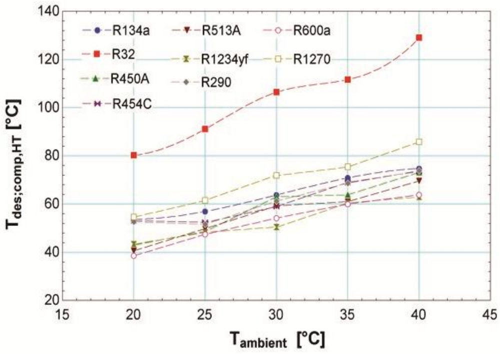


(b) Exergy Destruction

Figure 4. Comparison of COP and Exergy Destruction of combined CO_2 secondary/cascade refrigeration system at different ambient temperatures.



(a) Volumetric flow



(b) Discharge temperature

Figure 5. Comparison of volumetric flow rate in the suction of the compressor HT and compressor HT discharge temperature of combined CO_2 secondary/cascade refrigeration system at different ambient temperature.

Table 8. Results obtained for refrigeration system for an ambient temperature of 40°C.

Parameter	R134a	R32	R450A	R454C	R513A
COP [-]	1.82	1.81	1.80	1.88	1.80
η_{II} [%]	38.24	38.14	37.94	39.56	37.83
$\dot{E}x_{D,total}[kW]$	54.87	55.55	56.54	53.12	56.78
$\dot{W}_{comp,HT}[kW]$	88.65	88.47	90.00	86.02	90.28
$\dot{W}_{TOTAL}$ $[kW]$	101.48	101.73	102.29	98.09	102.57
$\dot{m}_{R744,LT}$ $[kg/s]$	0.247	0.245	0.247	0.247	0.247
$\dot{m}_{R744,MT}$ $[kg/s]$	0.634	0.627	0.894	0.900	0.894
$\dot{m}_{comp\,HT}$ $[kg/s]$	1.517	0.846	1.610	1.568	1.759
$\dot{V}_{suc,comp\,HT}$ $[m^3/s]$	0.183	0.065	0.214	0.111	0.178
P_{cond} *[kPa]*	1170	2815	1038	1949	1229
$T_{des,comp\,HT}$ [°C]	74.73	129.00	73.14	73.65	69.57
$C_{inv,main}$ $[US\$/yr]$	160,359	154,819	163,238	160,072	163,788
C_{op} $[US\$/yr]$	42672	42785	43014	41248	43134
C_{env} $[US\$/yr]$	7,041	7,060	7,097	6,806	7,117
C_{TOTAL} $[US\$/yr]$	210,144	204,733	213,422	208,198	214,113

Parameter	R1234yf	R290	R600a	R1270
COP [-]	1.79	1.90	1.90	1.90
η_{II} [%]	37.59	39.89	40.07	40.00
$\dot{E}x_{D,total}[kW]$	57.04	52.55	51.57	52.69
$\dot{W}_{comp,HT}[kW]$	90.93	85.00	84.77	84.56
$\dot{W}_{TOTAL}$ $[kW]$	103.22	97.26	96.84	97.00
$\dot{m}_{R744,LT}$ $[kg/s]$	0.247	0.246	0.247	0.246
$\dot{m}_{R744,MT}$ $[kg/s]$	0.894	0.631	0.900	0.890
$\dot{m}_{comp\,HT}$ $[kg/s]$	1.950	0.788	0.829	0.751
$\dot{V}_{suc,comp\,HT}$ $[m^3/s]$	0.190	0.124	0.335	0.100
P_{cond} *[kPa]*	1162	1545	608.7	1859
$T_{des,comp\,HT}$ [°C]	62.97	73.85	63.84	85.80
$C_{inv,main}$ $[US\$/yr]$	164,956	155,924	155,730	154,285
C_{op} $[US\$/yr]$	43,404	40,901	40,724	40,792
C_{env} $[US\$/yr]$	7,162	6,749	6,720	6,731
C_{TOTAL} $[US\$/yr]$	215,596	203,643	203,244	201,877

6. Conclusion

In this paper, a thermo-economic model is used to evaluate alternatives to replace R134a in an R134a/R744 combined secondary cascade system. From the results it is concluded that between the fluids evaluated, the hydrocarbon family (R290, R600a and R1270) are the ones that present the best ratio between total cost and performance. Among these fluids, the R1270 presented the lowest total cost and it fits as the best option for the high temperature cycle.

7. REFERENCES

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, n° 1, pp. 119-127. Bejan, A.; Kraus, A. D., 2003, Heat Transfer Handbook. Wiley.

Bellos, E.; Tzivanidis, C., 2019, "A Theoretical Comparative Study of CO2 Cascade Refrigeration Systems". Appl. Sci., vol. 9, p.790. https://doi.org/10.3390/app9040790

Bolaji, B.O., Huan, Z., 2013. "Ozone depletion and global warming: case for the use of natural refrigerant – a review". Renew.Sustain. Energy Rev. vol. 18, pp.49–54.

Cabello, R. et al. Energy evaluation of R152a as drop in replacement for R134a in cascade refrigeration plants. Applied Thermal Engineering, v. 110, p. 972-984, 2017.

Catalán-Gil, J.; Sánchez, D.; Llopis, R.; Nebot-Andrés, L.; Cabello, R., 2018, "Energy Evaluation of Multiple Stage Commercial Refrigeration Architectures Adapted to F-Gas Regulation". Energies, vol.11, pp.1915.

Climalife Technical Data R454C. Available in: https://www.climalife.co.uk/454c

Dai, B., Dang, C., Li, M., Tian, H., & Ma, Y. (2015). Thermodynamic performance assessment of carbon dioxide blends with low-global warming potential (GWP) working fluids for a heat pump water heater. International Journal of Refrigeration, 56, 1-14.

De Paula, Cleison Henrique et al. Energetic, Exergetic, Environmental, and Economic Assessment of a Cascade Refrigeration System Operating with Four Different Ecological Refrigerant Pairs. International Journal of Air-Conditioning and Refrigeration, v. 29, n. 03, p. 2150025, 2021.

Dokandari, D.A.; Hagh, A.S.; Mahmoudi, S.M.S., 2014, "Thermodynamic investigation and optimization of novel ejector-expansion CO2/NH3 cascade refrigeration cycles (novel CO2/NH3 cycle)". International Journal of Refrigeration, vol. 46, pp. 26–36.

Gasservei. Technical Data Sheet R-450A. Available in: https://gas-servei.com/shop/docs/technical-data-sheet-r-450a-gas-servei.pdf

Gasservei. Technical Data Sheet R-513A. Available in: https://gas-servei.com/shop/docs/dados-tecnicos-r-513a-gas-servei.pdf

Gupta, D.K., Dasgupta, M.S., 2014. "Simulation and performance optimization of finned tube gas cooler for trans-critical CO2 refrigeration system in Indian context". Int. J. Refrigeration, vol. 38, pp. 153–167.

Heredia-Aricapa, Y., Belman-Flores. J. M., Mota-Babiloni, A., Serrano-Arellano, J., García-Pabón, J. J., 2020, "Overview of low GWP mixtures for the replacement of HFC refrigerants: R134a, R404A and R410A", International Journal of Refrigeration, vol. 111, pp. 113-123.

Jain, V.; Sachdeva, G; Kachhwaha, S. S., 2015a, "Thermodynamic modeling and parametric study of a low temperature vapor compression-absorption system based on modified Gouy-Stodola equation", Energy, vol. 79, pp. 407-418.

Jain, V.; Sachdeva, G.; Kachhwaha, S. S., 2015b, "Energy, exergy, economic and environmental (4E) analyses based comparative performance study and optimization of vapor compression-absorption integrated refrigeration system". Energy, vol. 91, pp. 816832. ISSN 0360-5442

Jain V, Sachdeva G, Kachhwaha SS, Patel B., 2016, "Thermo-economic and environmental analyses based multi-objective optimization of vapor compressionabsorption cascaded refrigeration system using NSGA-II technique". Energy Convers Manage; vol.113. pp.230–342.

Kim, M.H., Pettersen, J., Bullard, C.W., 2004. "Fundamental process and system design issues in CO2 vapor compression systems". Prog. Energy Combust. Sci., vol. 30, pp.119174.

Klein, S.A., 2020, "EES – Engineering Equation Solver", Version 10.836, F-Chart Software, http://fchartsoftware.com

Llopis, Rodrigo et al. Energy and environmental comparison of two-stage solutions for commercial refrigeration at low temperature: Fluids and systems. Applied Energy, v. 138, p. 133-142, 2015.

Makhnatch, P., Mota-Babiloni, A., López-Belchí, A., Khodabandeh, R., 2019, "R450A and R513A as lower GWP mixtures for high ambient temperature countries: Experimental comparison with R134a". Energy, vol. 166, pp. 223.

MCTIC – Ministério da Ciência, Tecnologia, Inovações e Comunicações. Fator Médio – Inventários corporativos (tCO2/MWh), 2018. Disponível em: <

https://www.mctic.gov.br/mctic/opencms/ciencia/SEPED/clima/textogeral/emissao_corporativos.html>.

Mosaffa,A.H.; Farshi, L.G.; Ferreira, C.A.I.; Rosen, M.A., 2016, "Exergoeconomic and environmental analyses of CO2/NH3 cascade refrigeration systems equipped with different types of flash tank intercoolers". Energy Conversion and Management. vol. 117, pp. 442 – 453.

Mota-Babiloni, A., Navarro-Esbrí, J., Barragán-Cervera, A., Molés, F., Peris, B. and Verdú, G, 2015, "Commercial refrigeration e An overview of current status". International Journal of Refrigeration, vol. 57, n° 1, pp. 186-196.

Mota-Babiloni, A., Makhnatch, P., Khodabandeh, R., & Navarro-Esbri, J. (2017). Experimental assessment of R134a and its lower GWP alternative R513A. International Journal of Refrigeration, 74, 682-688.

Massuchetto, Luiz Henrique Parolin et al. Thermodynamic performance evaluation of a cascade refrigeration system with mixed refrigerants: R744/R1270, R744/R717 and R744/RE170. International Journal of Refrigeration, v. 106, p. 201-212, 2019.

Panato, V. H., Porto, M. P., Bandarra Filho, E. P., 2017, "Experimental performance of an R-22-based refrigeration system for use with R-1270, R-438A, R-404A and R-134a". Int. Journal of Refrigeration, vol. 83, pp. 108-117.

Purohit N., Gupta, D. K., Dasgupta, M. S., 2017, "Energetic and economic analysis of trans-critical CO2 booster system for refrigeration in warm climatic condition". International Journal of Refrigeration, vol. 80, n° 1, pp. 182-196.

Rezayan, O.; Behbahaninia, A., 2011, "Thermoeconomic optimization and exergy analysis of CO2/NH3 cascade refrigeration systems". Energy , vol. 36, n° 1, pp. 888–895.

Sanaye S, Malekmohammadi H., 2004, "Thermal and economical optimization of air conditioning units with vapor compression refrigeration system". Appl Therm Eng; vol. 24, n° 13: pp. 1807–1825.

Sanaye S, Shirazi A., 2013, "Four E analysis and multi-objective optimization of an ice thermal energy storage for air conditioning applications". Int J Refrig ,vol.36, n° 3, pp.828–841.

Sharma, V., Fricke, B., Bansal, P., 2014. "Comparative analysis of various CO2 configurations in supermarket refrigeration systems". International Journal of Refrigeration, vol. 46, pp. 86–99.

Silva, T.,P., 2019, "Simulação termoeconômica de sistemas de refrigeração de ciclo transcrítico e subcrítico utilizando CO2" (in Portuguese). Master's Thesis, Graduate Program in Engineering of Energy, Federal Center of Technologic Education of Minas Gerais, Belo Horizonte, Brazil.

Tassou, S.A., Ge, Y.T., Hadawey, A., Marriott, D., 2011. "Energy consumption and conservation in food retailing". Appl. Therm.Eng. vol. 31, pp. 147-156.