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Research Paper

Influence of subcooling in R-449A supermarket refrigeration system and screening of refrigerant mixtures for its energetic and environmental improvement

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A R T I C L E I N F O

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A B S T R A C T

Current demands of the cooling sector are focused on using refrigerants with a low Global Warming Potential (GWP) and increasing energy efficiency in vapour compression installations. The main objectives of this paperwork arise from these needs, in which the use of subcooling in an R-449A commercial refrigeration system is evaluated. In addition, eco-friendly alternative refrigerants are proposed and studied. Firstly, the effect of an integrated type of subcooling is semi-empirically analysed with Engineering Equation Solver (EES). Then, the refrigeration system's operation is simulated using low-GWP alternative refrigerants, R-454C and R-290, which have been scarcely studied in commercial refrigeration systems due to their safety classification. Moreover, new refrigerant mixtures are searched from the binary and ternary combination of pure refrigerants using the software REFPROP. The best refrigerant with safety classification A1, A2L, and A2/A3, for both medium temperature (MT) and low temperature (LT) systems is obtained from this search. After their determination, the supermarket's operation is simulated with these refrigerants. Finally, an environmental assessment is performed. The highest energy efficiency is obtained with R-152a for the MT system and the R-290/R-1270 mixture for the LT system. However, the highest reduction of GHG emissions is achieved by R-290, with a reduction of 73.78 % with respect to the baseline system.

1. Introduction

Cold chain is essential for food conservation and freezing while keeping the maximum standards and avoiding the growth of bacteria. However, 12 % of food in 2017 (526 Mt) was lost due to an insufficient cold chain or refrigeration [1]. In the current cold chain, most carbon emissions (60 %) originate from refrigeration equipment's electricity consumption (261 $MtCO_{2\text{-eq}}$). An improved cold chain with significantly lower food losses would increase these emissions to 589 $MtCO_{2\text{-eq}}$. Therefore, highly energy-efficient equipment, as well as low-GWP refrigerants, are required for commercial refrigeration.

To increase the energy efficiency of a supermarket cooling system, the main strategy studied consists of varying the system components, which is known as changing its configuration. Many configuration studies consisting of the addition of components at different points of the cooling cycle have been found in the literature [2,3,4,5].

However, it is known that further cooling of the refrigerant at the liquid line can significantly reduce power consumption and improve the COP. This strategy is known as subcooling. It can be performed in various ways, being the most common the dedicated mechanical subcooling. In this configuration, an auxiliary vapour compression system subcools the refrigerant at the condenser outlet before entering the expansion valve. The auxiliary system operates with a reduced temperature lift between the cold source and hot sink, reaching high COP values. Thus, COP benefits have been found with this technology [6,7,8,9,10].

In supermarkets, where there are usually two separate refrigeration systems (medium-temperature, MT, and low-temperature, LT), another subcooling technology can be used. It consists of an additional heat exchanger, the subcooler, which connects an MT low-temperature point with the LT liquid line. Thus, the LT liquid line is subcooled by the MT system. Yang and Zhang [11,12] studied this subcooling technology in a direct expansion supermarket system, concluding that 27 % and 20 % of energy savings can be achieved with R-404A and R-134a, respectively. Nevertheless, this subcooling technology has not been studied in indirect refrigeration systems, in which the LT system can be subcooled by the secondary fluid of the MT system. As mentioned in the study of

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Nomenclature

E	Energy consumption (kWh)
h	Enthalpy (kJ kg^{-1})
S	Entropy (kJ kg^{-1} K^{-1})
F	Fluorines
$\dot{Q}$	Heat transfer (kW)
H	Hydrogens
n	Lifetime of a refrigeration installation (years)
L	Percentual refrigerant leakage (-)
$\dot{W}$	Power consumption (kW)
P	Pressure (MPa)
m	Refrigerant mass charge (kg)
$\dot{m}$	Refrigerant mass flow rate (kg s^{-1})
N	Rotational speed (rpm)
T	Temperature (°C)
V	Volume (m^3)

Greek symbols

η	Efficiency (-)
ρ	Density (kg m^{-3})
Π	Flammability index (-)
α	Percentual recycling refrigerant at the end of the lifetime (-)
β	Carbon emission factor (kgCO$_2$ kWh^{-1})

Subscripts

ad	Adiabatic
c	Compressor
k	Condenser
crit	Critical
disch	Discharge
em	Electromechanical
o	Evaporator
vexp	Expansion valve
G	Geometric
in	Inlet
is	Isentropic
norm	normalized
out	Outlet
annual	Per year
ref	Refrigerant
suc	Suction
vol	Volumetric

Abbreviations

BMA1	Best A1 mixture
BMA2/A3	Best A2 or A3 mixture
BMA2L	Best A2L mixture
COP	Coefficient of performance
GWP	Global warming potential
HFC	Hydrofluorocarbon
LT	Low temperature
MT	Medium temperature
NBP	Normal boiling point
SC	Subcooling
SH	Superheat
TEWI	Total equivalent warming impact
wt	weight

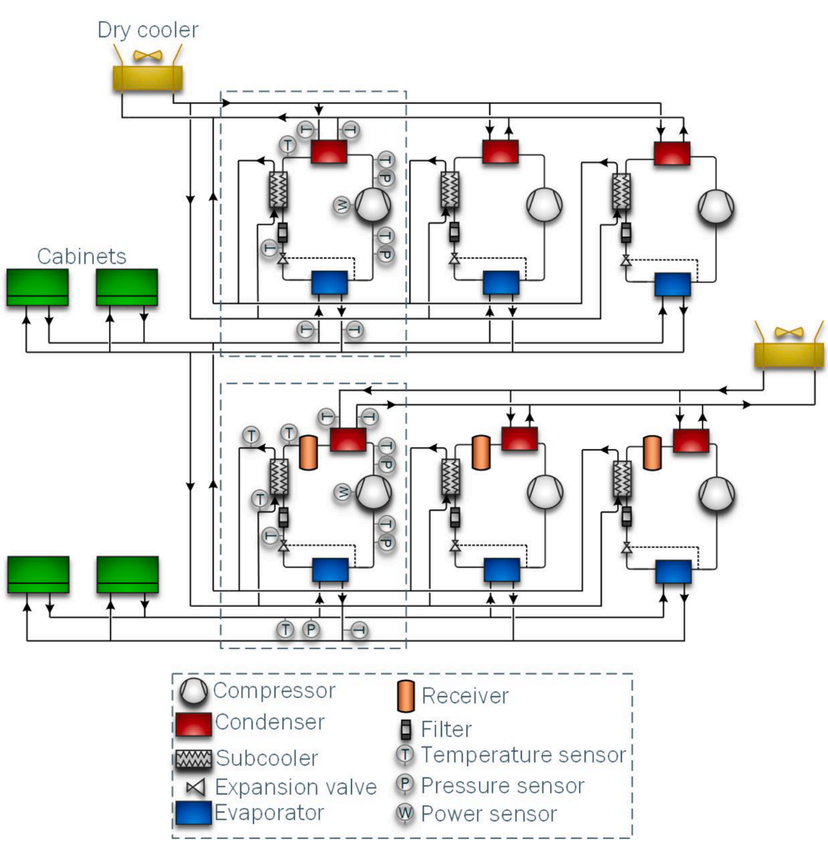
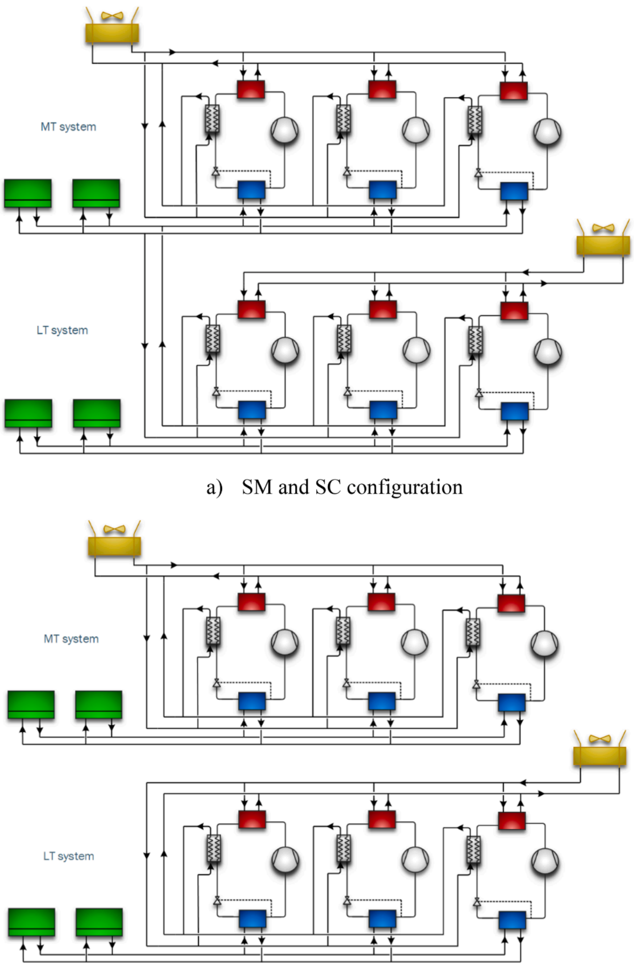


Fig. 1. Schematic of the baseline refrigeration system [2].



a) SM and SC configuration

b) No SC configuration

Fig. 2. Configurations of the studied cases.

Qureshi and Zubair [13], experimental work on residential, commercial, and industrial refrigeration equipment regarding this integrated subcooling needs to be done.

On the other hand, many refrigerants have been proposed regarding the low-GWP alternatives in supermarket refrigeration systems. Makhnatch et al. [14] performed a field test in an indirect supermarket refrigeration system using R-404A and R-449A. Results showed that R-449A is a suitable retrofit alternative to R-404A. However, this alternative has a GWP of 1282, so it can't be considered a long-term replacement. Citarella et al. [15] studied R-454C, R-449A, R-452A, R-455A, and R-290 as alternatives to R-404A in a 2.5 kW commercial refrigeration unit, showing that R-449A has the highest COP. However, R-454C can be a suitable alternative as a long-term replacement. Mota-Babiloni et al. [16] analysed R-454C and R-455A as alternatives to R-404A in a one-stage refrigeration system. R-454C and R-455A exhibited a slightly lower cooling capacity and a 10–15 % higher COP than R-404A. Oruç et al. [17] analysed R-454A and R-454C as R-404A alternatives in an experimental setup. The measured compressor power was reduced by 6 % and 15 % for R-454A and R-454C, respectively, leading to improvements in COP of 14 % and 10 %. R-290, R-1234yf, and R-600a were studied as alternatives to R-134a in a small-capacity chiller by De Paula et al. [18]. The analysis demonstrates that R-290 has the best energy, exergy, and environmental performance for the studied conditions. Mastrullo et al. [19] studied R-290 as a replacement for R-404A in a light commercial vertical freezer, obtaining a reduction of energy consumption up to 34 %. Although R-454C and R-290 appear to be the most promising alternatives to most commonly used high-GWP refrigerants, they have been scarcely studied in medium and high-capacity commercial refrigeration systems due to their safety classification (A2L and A3, respectively), which can limit the refrigerant charge. However, in indirect refrigeration systems where the access to the main cooling cycle room is restricted, refrigerant charge is not limited so that these refrigerants could be used.

This paper proposes a semi-empirical investigation of integrating subcooling units in an indirect supermarket refrigeration system operating with R-449A. Also, the implementation of low-GWP alternatives R-454C and R-290 is analysed. In addition, to see if new potential eco-friendly replacements exist, a screening of binary and ternary mixtures is carried out to obtain the best mixtures for both MT and LT systems.

2. Materials and methods

This section presents the configurations this article is based on and the strategy used, from the assumptions to the final model.

2.1. Baseline refrigeration system

The main focus of this study is to evaluate a supermarket's potential for energy efficiency improvement by analysing its subcooling and implementing different refrigerants. The baseline supermarket refrigeration system is located in Södertälje, in the south-central part of Sweden, with a latitude of 59.2°N. The supermarket has two refrigeration systems: medium temperature (MT), which covers a 222 kW thermal load, and low temperature (LT), which covers a 37.5 kW thermal load. These systems are connected in parallel, as shown in Fig. 1. An intermediate fluid is employed between the main refrigeration circuit and the cabinets to facilitate heat transfer. For the MT system, a mixture of water and propylene glycol (62 % water and 38 % propylene glycol by volume) is used, while the LT system uses pumped CO_2. Water serves as the heat transfer fluid between the main refrigeration circuit and the dry cooler. Unlike the MT system, the LT system incorporates a liquid receiver. Moreover, the subcooling of the LT system comes from the water/propylene glycol mixture of the MT system. So, the MT system produces more cooling power to provide the LT subcooling.

More specific information on the operation and performance of the MT and LT circuits in the baseline system is available in the studies conducted by Makhnatch et al. in 2017 [14] and 2018 [20], respectively. Additionally, a comprehensive report by Rogstam et al. in 2016 [21] provides extensive details about the system's characteristics, component specifications, operation, and measurements. In these works, the utilization of R-449A as a drop-in substitute for R-404A is analysed in the baseline system with some minor modifications, such as optimizing the charge and adjusting the expansion valve. Results showed a 10 % lower cooling capacity for R-449A and a similar COP with both refrigerants in the MT system. In the LT system, a 1.3 % lower cooling capacity and an 8.1 % higher COP were obtained with R-449A. Given that the rest of the parameters, such as discharge temperature, evaporating and condensing pressures, etc., were acceptable, it was concluded that R-449A is a suitable alternative to R-404A as a drop-in substitute.

2.2. Configurations

A total of 3 cases will be analysed in this work. The first one (SM) corresponds to the configuration existing in the supermarket, using the model of the compressors that this installation uses. These compressors are old (more than 15 years old), so their performance may not be optimal. Thus, the second case (SC) is that of the existing configuration in the supermarket, but using a modern compressor, which will presumably have better performance, thus improving the system's operation. Finally, the last case (no SC) corresponds to the cooling system with a modern compressor and without the subcooling from the MT cycle, so that the SC of the LT system, like the MT system, will be connected to the dry cooler. In addition to the best performance, the comparison using

Table 1
Studied refrigerants' properties.

Refrigerant	Composition (%wt)	NBP[a] (°C)	T_{crit}[a] (°C)	P_{crit}[a] (bar)	ASHRAE std 34	Molecular weight[a] (g/mol)	GWP_{100-yr}[b]
R-449A	R-134a/R-1234yf/R-125/R-32 26/25/25/24	−46.0	81.5	44.5	A1	87.2	1282
R-454C	R-32/R-1234yf 21.5/78.5	−45.9	82.4	43.2	A2L	90.8	146
R-290	R-290 100	−42.4	96.7	42.4	A3	44.1	0.02

[a] Values taken from REFPROP v10.0[22].
[b] Values taken from IPCC – AR6[23].

Table 2
Pure refrigerants used for determining the best mixtures.

Refrigerant	Chemical formula	NBP[a] (°C)	T_{crit}[a] (°C)	P_{crit}[a] (bar)	ASHRAE std 34	Molecular weight[a] (g/mol)	GWP_{100-yr}[b]
R-23	CHF3	−82.2	26.1	48.32	A1	70.01	14,600
R-32	CH2F2	−51.9	78.1	57.82	A2L	52.02	771
R-125	CHF2CF3	−48.3	66.0	36.18	A1	120.02	3740
R-134a	CF3CH2F	−26.3	101.0	40.59	A1	102.03	1530
R-143a	C2H3F3	−47.5	72.7	37.61	A2	84.04	5810
R-152a	CHF2CH3	−24.3	113.2	45.17	A2	66.05	164
R-227ea	CF3CHFCF3	−16.6	101.7	29.25	A1	170.02	3600
R-1234yf	CH2 = CFCF3	−29.7	94.7	33.82	A2L	114.04	0.501
R-1234ze(E)	CHF = CHCF3	−19.2	109.3	36.35	A2L	114.04	1.37
R-1234ze(Z)	CHF = CHCF3 (cis)	9.3	150.1	35.31	A2L	114.04	0.315
R-744	CO2	−88.0	30.9	73.77	A1	44.01	1
R-170	CH3CH3	−88.8	32.1	48.72	A3	30.07	0.437
R-290	CH3CH2CH3	−42.4	96.7	42.51	A3	44.09	0.02
R-1270	C3H6	−47.9	91.0	45.55	A3	42.08	1.8[a]
R-600	CH3-2(CH2)–CH3	−0.8	151.9	37.96	A3	58.12	0.006
R-600a	CH(CH3)3	−12.1	134.6	36.29	A3	58.12	3[a]
R-601	CH3-3(CH2)–CH3	35.6	196.5	33.67	A3	72.15	5[a]
R-601a	C5H12	27.4	187.2	33.78	A3	72.15	5[a]
R-218	C3F8	−37.1	71.8	26.4	A1	188.02	9290
R-41	CH3F	−78.5	44.1	58.97	A2	34.03	135
R-1243zf	C3F3H3	−25.7	103.7	35.18	A2	96.05	0.261

[a] Values taken from REFPROP v10.0[22].
[b] Unless indicated, values taken from IPCC – AR6[23].

modern compressors is due to the possibility of establishing a proper comparison. This is because, in the case without SC, compressors must be selected according to the corresponding refrigeration capacities. This means that as the MT system will have to deliver lower power, its compressors will also be smaller. Moreover, as the LT system will have to deliver higher power, its compressors must be larger. The representative schema of the different cases is shown in Fig. 2.

2.3. Refrigerants

This study assesses the performance of several refrigerants. First, the behaviour of R-449A is studied in the different cases mentioned above. Once the best case in energy terms has been determined, the performance of the refrigerants R-454C and R-290 is studied in the best-case configuration. These are two low-GWP refrigerants that could represent viable alternatives to R-404A and R-449A in the studied refrigeration system. The characteristics of these refrigerants are shown in Table 1.

In addition, the best mixture for the MT and the LT system is determined, and its operation in the best-case configuration is simulated. These mixtures are obtained from the combination of two and three pure refrigerants from the refrigerant list shown in Table 2.

The pure refrigerants chosen to determine the best mixtures, being a total of 21, have very varied properties. The selection of refrigerants has been made based on different parameters. On the one hand, several refrigerants with a safety classification A1 have been selected. All of them, except for R-744 (CO_2), have high GWP. However, combined with other refrigerants, they can reduce the flammability of the mixture. In addition, some refrigerants with A2L safety classification and low-GWP have been selected. These refrigerants generally perform better in refrigeration systems compared to A1 ones. Thus, they can improve the mixture's energy efficiency and environmental characteristics. Finally, refrigerants A2 and A3 have been selected, which also have low GWP and high energy efficiency in refrigeration systems.

2.4. Working conditions

As mentioned before, the supermarket refrigeration system operation has been studied by Makhnatch et al. [14,20] and Rogstam et al. [21], who monitored the system by placing temperature, pressure, and power sensors in different points of the cooling cycle. Data was obtained every 30 s, and more than 80,000 operational points (i.e., 80,000 groups of temperatures, pressures, and power) were collected. In the present work, conditions used to perform the simulations with the mathematical model have been extracted and, in some cases, adapted from the experimental data obtained in the mentioned studies.

2.4.1. Operating temperatures

The evaporating temperature of the MT supermarket refrigeration system ranges from − 21 to − 8 °C, while for the LT system it ranges from − 40 to − 30 °C. In order to take into account this variation in the simulations, all the operational points with which the system has worked are taken as inputs to the model. On the other hand, the condensing temperature for the MT system ranges from 22 to 44 °C, while for the LT system, it ranges from 14 to 44 °C. As with the evaporating temperature, all the operational points are taken as inputs to the model.

Regarding the superheating degree, it ranges from 1 to 15 K for the

Table 3
Operating temperatures according to actual supermarket operation.

Parameter	Considered value MT	LT
T_o (°C)	[−21; −8]	[−40; −30]
T_k (°C)	[22; 44]	[14; 44]
Superheating degree (K)	[1; 15]	[8; 13]
Condenser subcooling degree (K)	0	0
Subcooler subcooling degree (K)	[1; 6]	SC = f($\dot{Q}_{SC}$)

MT system and from 8 to 13 K for the LT system. Thus, as with the evaporating and condensing temperatures, all the operational points are taken as inputs to the model.

The subcooling degree for the MT system ranges from 1 to 6 K, so all the operational points are taken as inputs to the model. However, for the LT subcooling, a significant variation in the range for a particular refrigerant and a significant variation with different refrigerants is observed. Thus, a correlation obtained in previous work [2] for the analysed refrigeration system is used, Eq. (1), which correlates the transferred heat in the subcooler ($\dot{Q}_{SC}$) with the refrigerant temperature at the inlet of the subcooler ($T_{SC_{ref,in}}$) and the refrigerant mass flow rate ($\dot{m}_{ref}$).

$$\dot{Q}_{SC} = 3.7324 + 0.05169\ T_{SC_{ref,in}} - 78.6682\ m_{ref} - 2.0965\cdot 10^{-3}\ T^2_{SC_{ref,in}} + 245.011\ m^2_{ref} + 2.1259\ T_{SC_{ref,in}}\ \dot{m}_{ref} \quad (1)$$

A summary of the operating temperatures used for the simulations is shown in Table 3.

2.4.2. Compressor efficiencies

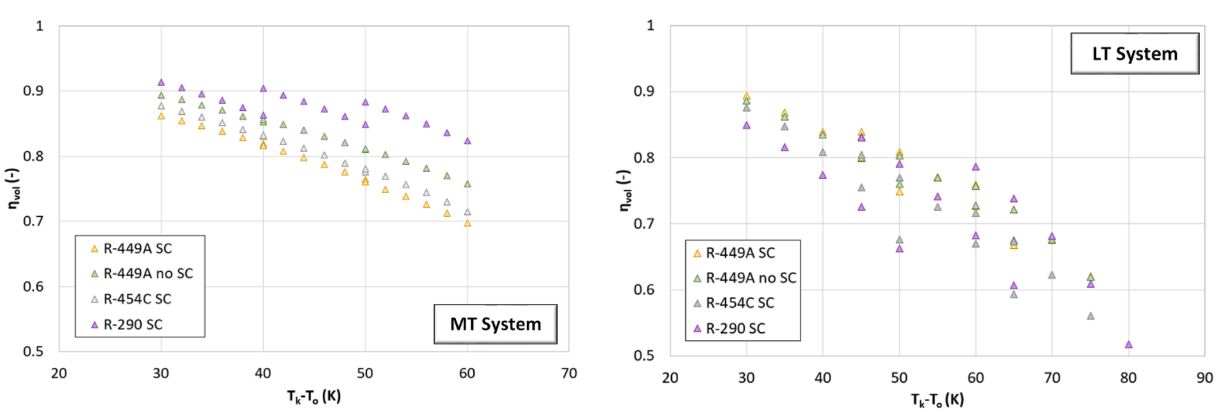
Semi-hermetic reciprocating compressors have been selected according to the maximum cooling capacity in SC and no SC cases to retrieve the necessary information about the efficiencies. Considering the conditions listed in Table 3, the compressors for R-449A SC are Bitzer 8GE-60Y (MT) and Bitzer 4GE-23Y (LT). For R-449A no SC, the chosen compressors are 6FE-44Y (MT) and 4FE-28Y (LT). As seen in the results section, the best case in energy terms is the SC one, so R-454C and R-290 will be studied with SC configuration. Then, R-454C SC chosen compressors are 6FE-44Y (MT) and 4FE-28Y (LT), and R-290 SC chosen compressors are 6FEP-44P (MT) and 6JEP-25P (LT) [24].

The mass flow rate for the R-449A SM case was not directly measured because flow meters would imply a modification of the original system. Therefore, for volumetric efficiency calculations of R-449A SM, the R-449A SC compressors are taken. The volumetric efficiency (η_{vol}) of each case is obtained by comparing the mass flow rate provided by the compressor manufacturer ($\dot{m}_{ref,manufacturer}$) with the theoretical mass flow rate calculated with Engineering Equation Solver (EES) ($\dot{m}_{ref,theoretical}$) [25], obtained from the geometric volume of the compressor (V_G), the rotational speed (N), the density at the compressor's suction point (ρ_{suc}), and taking unitary volumetric efficiency, Eq. (2) and (3). The compressor manufacturer data is shown in Fig. 3, and the resulting equations with the corresponding value of R^2 are shown in Table 4. For the best mixture calculation, since the volumetric efficiency is an input parameter, they can't be determined without previously establishing a volumetric efficiency. Therefore, R-449A SC equations are used for these calculations.

$$\eta_{vol} = \frac{\dot{m}_{ref,manufacturer}}{\dot{m}_{ref,theoretical}} \quad (2)$$

$$\dot{m}_{ref,theoretical} = V_G \frac{N}{60} \rho_{suc} \quad (3)$$

As can be seen, the highest volumetric efficiency in the MT system is achieved by R-290 for the whole range of considered temperature lifts. The worst efficiency is obtained with R-449A SC, being the maximum difference between this case and R-290 of about 12 %. In the LT system,



a) b)

Fig. 3. Compressors' volumetric efficiency for MT system (a) and LT system (b).

Table 4
Equations describing compressors' volumetric efficiency.

Studied case	System	Equation[a]	R^2
R-449A SM and SC	MT	$\eta_{vol,R-449A\ SM/SC,MT} = 1.0328 + 0.0053\,T_o - 0.0055\,T_k$	0.994
	LT	$\eta_{vol,R-449A\ SM/SC,LT} = 1.1827 + 0.0095\,T_o - 0.0059\,T_k$	0.935
R-449A no SC	MT	$\eta_{vol,R-449A\ no\ SC,MT} = 1.0339 + 0.0047\,T_o - 0.0044\,T_k$	0.995
	LT	$\eta_{vol,R-449A\ no\ SC,LT} = 1.1658 + 0.009\,T_o - 0.0059\,T_k$	0.921
R-454C SC	MT	$\eta_{vol,R-454C\ SC,MT} = 1.0454 + 0.0056\,T_o - 0.0053\,T_k$	0.995
	LT	$\eta_{vol,R-454C\ SC,LT} = 1.1752 + 0.0106\,T_o - 0.006\,T_k$	0.983
R-290 SC	MT	$\eta_{vol,R-290\ SC,MT} = 1.0085 + 0.0055\,T_o - 0.0017\,T_k$	0.984
	LT	$\eta_{vol,R-290\ SC,LT} = 1.1412 + 0.0113\,T_o - 0.0033\,T_k$	0.965

[a] T_o and T_k in °C.

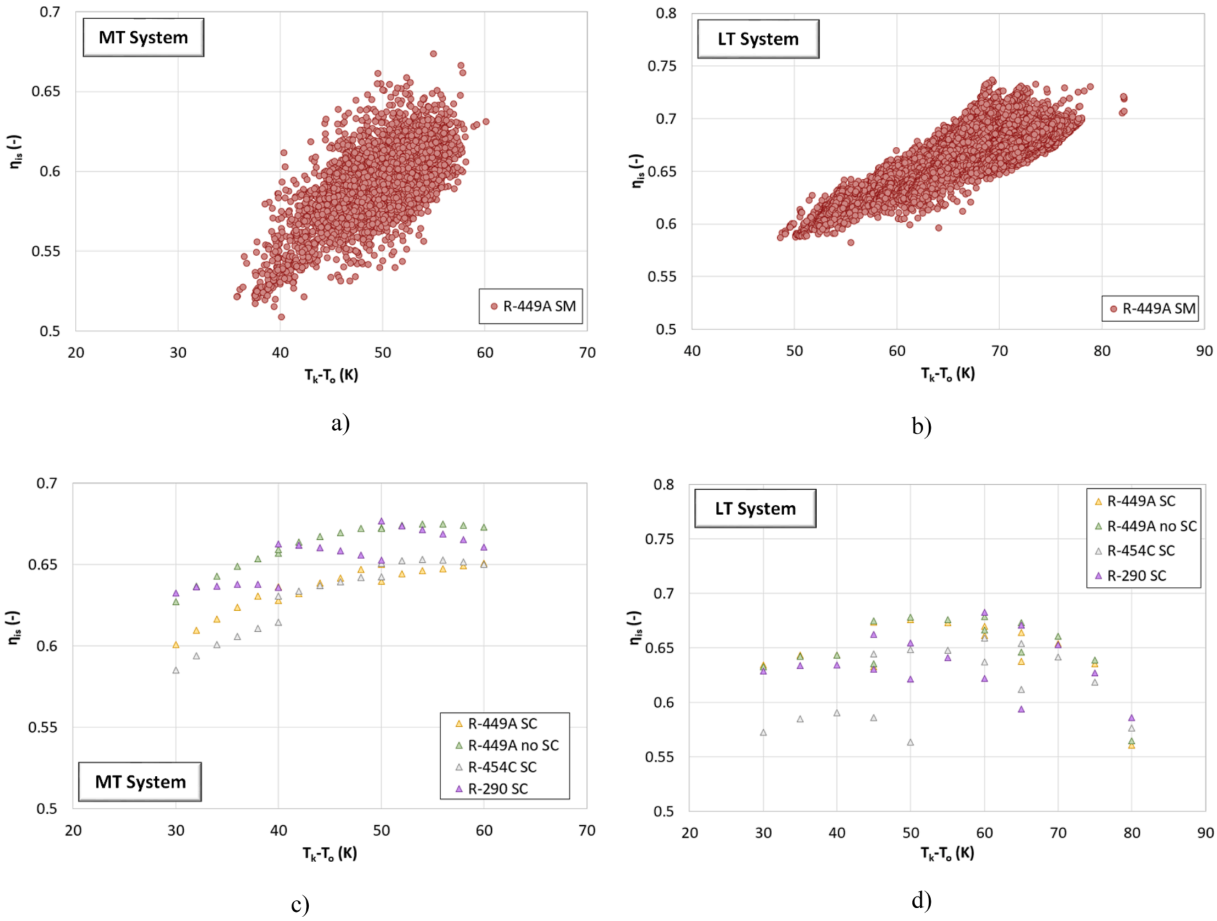


Fig. 4. Compressors' isentropic efficiency obtained from experimental data, (a) and (b), and from manufacturer data, (c) and (d).

differences between the cases are not so evident. R-449A SC achieves the best efficiency for low temperature lifts, while for high-temperature lifts, it is achieved by R-290. Here, the maximum difference in volumetric efficiency between the different cases is 0.9 %. Furthermore, the R^2 of the obtained equations is above 0.9 in all cases, so their accuracy is considered sufficient.

The isentropic efficiency of the R-449A SM case has been determined based on direct measurements of pressure and temperature at the suction and discharge points of the compressors. For the rest of the cases, values provided by the compressor manufacturer are used. The isentropic efficiency (η_{is}) has been calculated with Eq. (4), where $h_{disch,is}$ is the discharge enthalpy if the isentropic efficiency was unitary, h_{disch} is the real discharge enthalpy, and h_{suc} is the suction enthalpy. The compressor experimental efficiencies and the manufacturer data are shown in Fig. 4, and the resulting equations with the corresponding value of R^2 are shown in Table 5. As with volumetric efficiency, R-449A SC isentropic efficiency is considered for calculating the best mixture.

$$\eta_{is} = \frac{h_{disch,is} - h_{suc}}{h_{disch} - h_{suc}} \tag{4}$$

As is shown, the isentropic efficiencies obtained from the experimental measurements have a positive trend for the considered temperature lifts. That means that the compressors of the supermarket installation have been designed to work with higher temperature lifts (higher pressure lifts). This does not happen with the isentropic efficiencies obtained from the manufacturer data, where optimal compressors have been selected. The highest efficiency is achieved by R-449A no SC case for both MT and LT systems.

Since the electromechanical efficiency can't be obtained from the experimental measurements or the manufacturer data, the values corresponding to the minimum IE4 motor efficiencies are taken. These values depend on the produced power, being 0.94 for MT compressors and 0.92 for LT compressors [26].

2.5. Methodology

The methodology followed to obtain the results for each one of the configurations and refrigerants is based on the flow diagram shown in Fig. 5. It should be noted that the mathematical model is validated before comparing the configurations with the R-449A, comparing the simulated results with the experimental ones.

After validation, each configuration and the operating conditions and parameters previously determined are taken as inputs for the simulation. Thus, the interest parameters are obtained with the EES simulation, which will allow comparing the different analysed cases (SM, SC, and no SC) with R-449A. The best configuration, the one with a higher overall COP, is determined, and the operation of alternative refrigerants is studied with it. With these refrigerants, the same process is performed, that is, taking the operating conditions and parameters as inputs, modelling the behaviour, and obtaining the parameters of interest. Finally, the environmental analysis of all the proposed solutions is performed.

Moreover, the methodology for determining the best mixtures is based on the flow diagram shown in Fig. 6. A Python programme linked to REFPROP and Cantera has been developed to perform all the calculations. First, starting from the list of pure refrigerants shown above, all possible binary and ternary combinations between them are created, as well as all possible mass fractions with a 5 % step. Once this is done, the cooling cycle is calculated. The input parameters used are the

Table 5
Equations describing compressors' isentropic efficiency.

Studied case	System	Equation[a]	R^2
R-449A SM	MT	$\eta_{is,R-449A\ SM,MT} = 0.1192 - 0.0154\,T_o + 0.0145\,T_k + 0.00032\,T_o^2 - 0.000056\,T_k^2 - 0.00051\,T_oT_k$	0.626
	LT	$\eta_{is,R-449A\ SM,LT} = 1.3119 + 0.0499\,T_o + 0.0072\,T_k + 0.00076\,T_o^2 - 0.000075\,T_k^2 - 0.000016\,T_oT_k$	0.654
R-449A SC	MT	$\eta_{is,R-449A\ SC,MT} = 0.4353 - 0.0079\,T_o + 0.0072\,T_k - 0.000062\,T_o^2 - 0.000065\,T_k^2 + 0.00013\,T_oT_k$	0.999
	LT	$\eta_{is,R-449A\ SC,LT} = 0.3398 - 0.0162\,T_o + 0.0104\,T_k - 0.00026\,T_o^2 - 0.00013\,T_k^2 + 0.00013\,T_oT_k$	0.92
R-449A no SC	MT	$\eta_{is,R-449A\ no\ SC,MT} = 0.4363 - 0.0094\,T_o + 0.0086\,T_k - 0.00012\,T_o^2 - 0.000084\,T_k^2 + 0.00014\,T_oT_k$	0.999
	LT	$\eta_{is,R-449A\ no\ SC,LT} = 0.3281 - 0.0161\,T_o + 0.0114\,T_k - 0.00024\,T_o^2 - 0.00013\,T_k^2 + 0.00016\,T_oT_k$	0.921
R-454C SC	MT	$\eta_{is,R-454C\ SC,MT} = 0.3556 - 0.009\,T_o + 0.0112\,T_k - 0.000105\,T_o^2 - 0.000107\,T_k^2 + 0.00015\,T_oT_k$	0.999
	LT	$\eta_{is,R-454C\ SC,LT} = 0.2808 - 0.0145\,T_o + 0.0127\,T_k - 0.00022\,T_o^2 - 0.000147\,T_k^2 + 0.00012\,T_oT_k$	0.993
R-290 SC	MT	$\eta_{is,R-290\ SC,MT} = 0.5069 - 0.0046\,T_o + 0.0066\,T_k - 0.000085\,T_o^2 - 0.000059\,T_k^2 + 0.000093\,T_oT_k$	0.997
	LT	$\eta_{is,R-290\ SC,LT} = 0.4864 - 0.0091\,T_o + 0.0044\,T_k - 0.00014\,T_o^2 - 0.000009\,T_k^2 + 0.00015\,T_oT_k$	0.983

[a] T_o and T_k in °C.

configuration, which is the best configuration determined with the use of the R-449A; the refrigerant, which changes with each iteration; the operating conditions, which consist of the average of the conditions with which the supermarket installation has operated, thus the most representative conditions are taken; the compressor parameters, in which those obtained for the R-449A SC case are taken since these do not vary significantly with the refrigerant.

The parameters of interest are calculated from the REFPROP simulation, and filters are applied to discard unsuitable mixtures. Thus, mixtures whose normal boiling point (NBP) is higher than the minimum temperature with which the cooling system works are discarded. These minimum values are − 21.3 °C for the MT system and − 40.6 °C for the LT system. Also, mixtures whose discharge temperature (T_{disch}) exceeds 120 °C and mixtures with a glide greater than 7 K (R-449A glide value) are discarded. Then, the flammability of the suitable mixtures is calculated with Cantera software. Finally, mixtures are classified by their safety classification, and the best mixtures are those that, having a lower GWP and a higher COP than R-449A simultaneously, have the highest COP. The result of this procedure is the best mixture A1 (BMA1), the best mixture A2L (BMA2L), and the best mixture A2/A3 (BMA2/A3) for both MT and LT systems.

2.6. Equations

2.6.1. Vapor compression cycle

Being the evaporating and condensing temperatures defined, as well as the superheating (*SH*) and subcooling (*SC*) degrees, the compressor suction temperature (T_{suc}) can be obtained with Eq. (5), where $T_{o,dew}$ is the dew temperature at the evaporation pressure.

$$SH = T_{suc} - T_{o,dew} \tag{5}$$

Then, using the compressor isentropic efficiency, the discharge temperature is obtained, Eq. (6).

$$h_{disch} = \frac{h_{disch_{is}} - h_{suc}}{\eta_{is}} + h_{suc} \tag{6}$$

The expansion valve inlet temperature ($T_{vexp_{in}}$) is obtained knowing the subcooling degree in the MT system and the bubble temperature at the condensation pressure ($T_{k_{bubble}}$), Eq. (7). In the LT system, this temperature is obtained using the previously shown correlation for subcooling, which calculates transferred heat in the subcooler, and Eq. (8), where $h_{SC_{in}}$ is the enthalpy at the inlet of the subcooler.

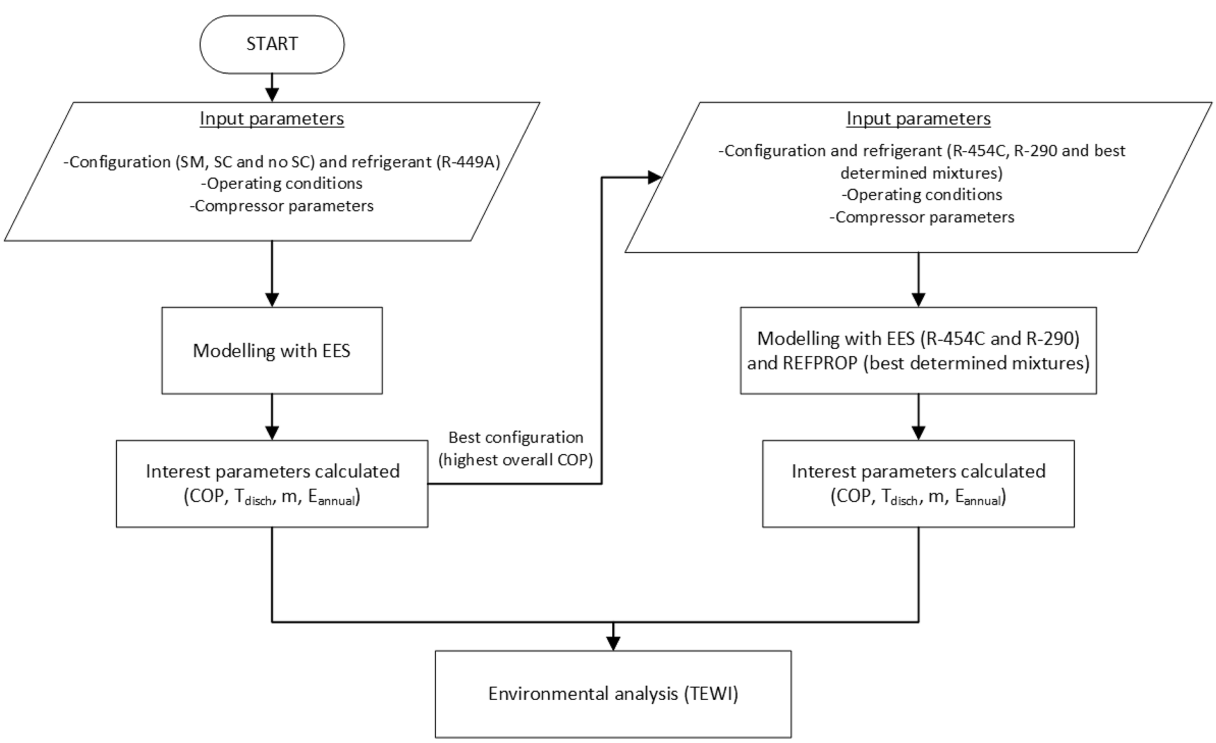


Fig. 5. Flow diagram of the general followed methodology.

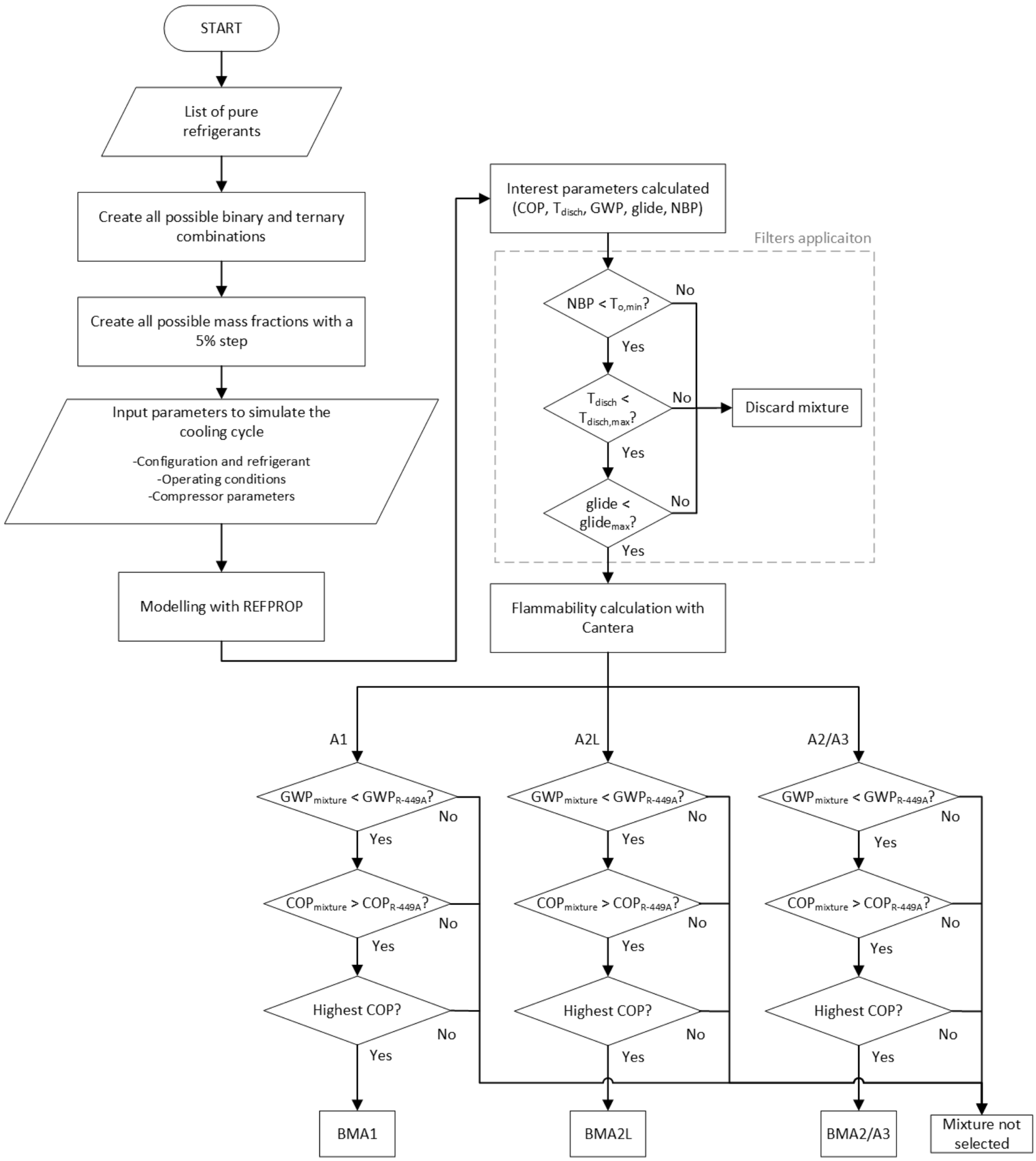


Fig. 6. Flow diagram of the followed methodology for the determination of the best mixtures.

$$SC = T_{k_{bubble}} - T_{vexp_{in}} \tag{7}$$

$$h_{vexp_{in}} = h_{SC_{in}} - \frac{\dot{Q}_{SC}}{\dot{m}_{ref}} \tag{8}$$

The mass flow rate is calculated with Eq. (9). An isentalpic expansion is considered at the expansion valve.

$$\dot{m}_{ref} = \rho_{suc}\ \omega\ V_G\ \eta_{vol} \tag{9}$$

The most important parameters for a refrigeration system are cooling capacity, power consumption, and COP, which are calculated with Eq. (10), (11), and (12), respectively, where $h_{o_{in}}$ and $h_{o_{out}}$ are the evaporator's inlet and outlet temperatures.

$$\dot{Q}_o = \dot{m}_{ref}\ \left(h_{o_{out}} - h_{o_{in}}\right) \tag{10}$$

$$\dot{W}_c = \frac{\dot{m}_{ref}}{\eta_{em}}\left(h_{disch} - h_{suc}\right) \tag{11}$$

$$COP = \frac{\dot{Q}_o}{\dot{W}_c} \tag{12}$$

Finally, global COP for analyzing the global impact of subcooling is calculated with Eq. (13).

$$COP_{global} = \frac{\dot{Q}_{o,MT} + \dot{Q}_{o,LT}}{\dot{W}_{c,MT} + \dot{W}_{c,LT}} \tag{13}$$

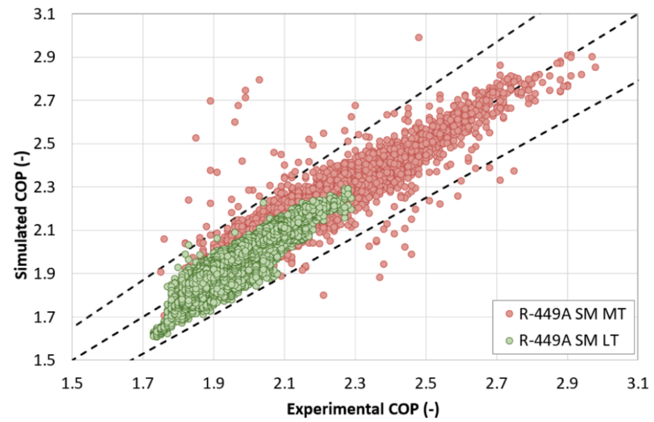


Fig. 7. COP comparison between measurements and the mathematical model.

2.6.2. Safety classification of obtained mixtures

All pure refrigerants considered to determine the best mixtures have a toxicity classification A. Therefore, it is assumed that the resulting mixtures will maintain this classification.

Pure refrigerants have different classifications regarding flammability level, so an estimation of the final mixture's flammability is needed. The estimation method used in this work is the one proposed by Linteris et al. [27], which is based on the mixture's molecular structure. This model uses the adiabatic flame temperature (T_{ad}) and the ratio of hydrogens (H) and fluorides (F) of the blend, Eq. (14) and (15). The adiabatic flame temperature is obtained with Cantera [28], a software for performing chemical thermo-kinetics and transport models. Using this method, $\Pi_{norm} = 0$ represents the 1/2L boundary, and $\Pi_{norm} = 40$ represents the 2L/3 boundary.

$$\Pi = arctan2\left(\frac{T_{ad} - 1600}{2500 - 1600}, \frac{F}{F + H}\right)\frac{180}{\pi} \tag{14}$$

$$\Pi_{norm} = \frac{\Pi - \Pi_{1,2L}}{90 - \Pi_{1,2L}} \tag{15}$$

2.6.3. TEWI

Since COP is insufficient to determine the environmental impact of a refrigeration system, another metric needs to be used. Here, the TEWI method is followed to consider the different factors involved in GHG emissions. This method calculates the $CO_{2,eq}$ emissions through Eq. (16). The total environmental impact is obtained with Eq. (17).

$$TEWI = GWP\ L_{annual}\ m\ n + GWP\ m\ (1 - \alpha) + E_{annual}\ n\ \beta \tag{16}$$

$$TEWI_{total} = TEWI_{MT} + TEWI_{LT} \tag{17}$$

GWP values of commercial mixtures are the ones shown in Table 1. GWP values will be shown in the next section for the best mixtures obtained. The annual leakage percentage (L_{annual}) is fixed at 12 %, and a lifetime (n) of 15 years is assumed. The end-of-life losses are 0.1 (α of 0.9). These values have been extracted from the International Institute of Refrigeration guidelines [29].

The refrigerant charge (m) initially used for R-449A was measured in one circuit. Thus, multiplying it by the number of circuits, MT and LT values are obtained for R-449A SM and SC cases. These values are 37.5 and 56.25 kg, respectively. The values are shown in the next section for the rest of the cases.

Regarding the annual energy consumption (E_{annual}), 700.968 MWh year^{-1} was obtained with R-449A SM, including all MT and LT circuits. For the other cases, the values will be extrapolated depending on the system's energy efficiency. Finally, Sweden's average carbon emission factor (β) is considered, having a value of 0.0088 kg $CO_{2,eq}$ kW/h [30].

3. Results and discussion

3.1. Model validation

To verify the validity and the accuracy of the proposed model, a comparison has been made between the results for the obtained COP in the analysis of the commercial installation and those obtained by using the model in the SM case, in both cases with the R-449A. This comparison is shown in Fig. 7. As can be seen, most points are between − 10 % and + 10 % of relative error, with 0.8 % of average error for MT and − 0.9 % of average error for LT. Thus, the mathematical model is considered to be validated.

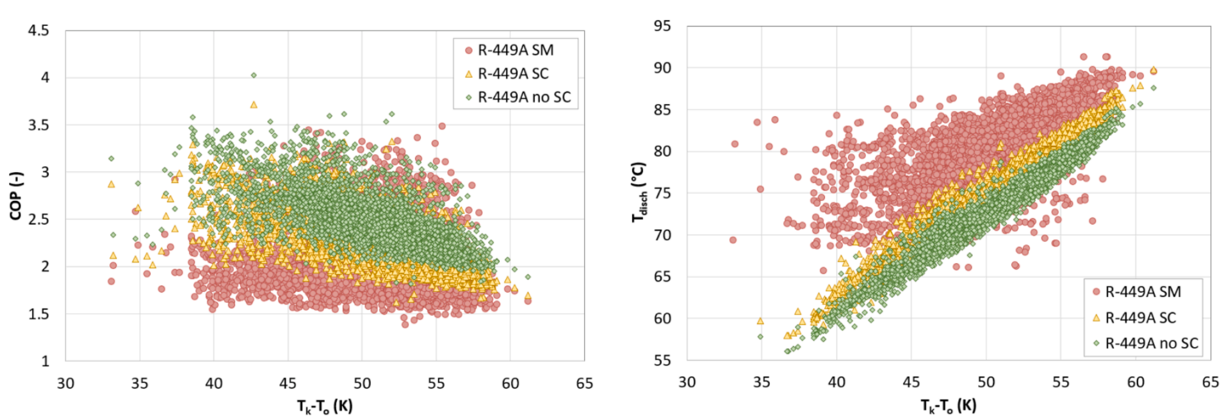
3.2. Subcooling analysis

This section presents the results obtained for the different studied cases using R-449A. The objective is to determine the best case in energy terms, analysing the subcooler's influence on the cooling system's operation. The parameters of interest shown in this section are the COP, which is an indicator of energy efficiency, and the discharge temperature, since too high temperatures, usually greater than 120 °C, can negatively affect the compressor's life.

3.2.1. MT system

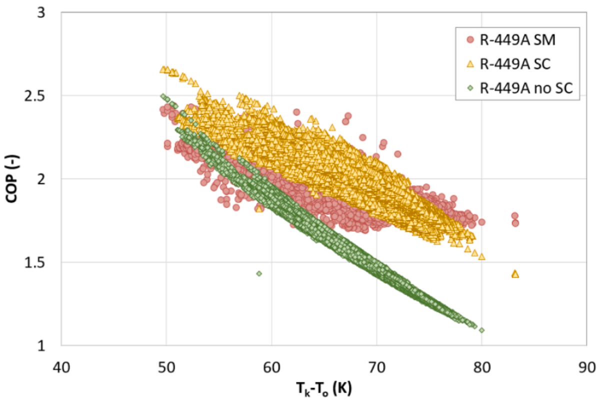
COP and discharge temperature results obtained for the three analysed cases in the MT system are shown in Fig. 8.

As can be seen, the SM case presents the lowest COP values, with an

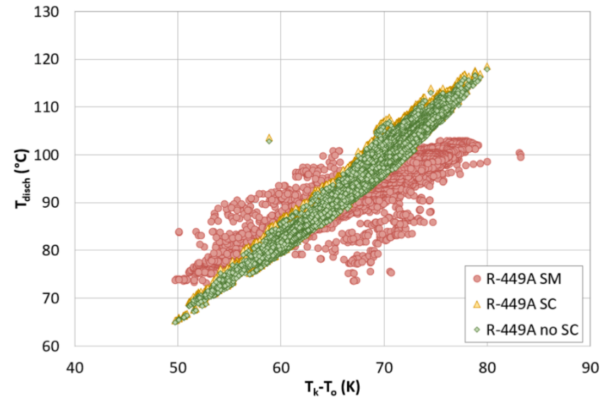


a) b)

Fig. 8. COP (a) and discharge temperature (b) for R-449A cases in the MT system.

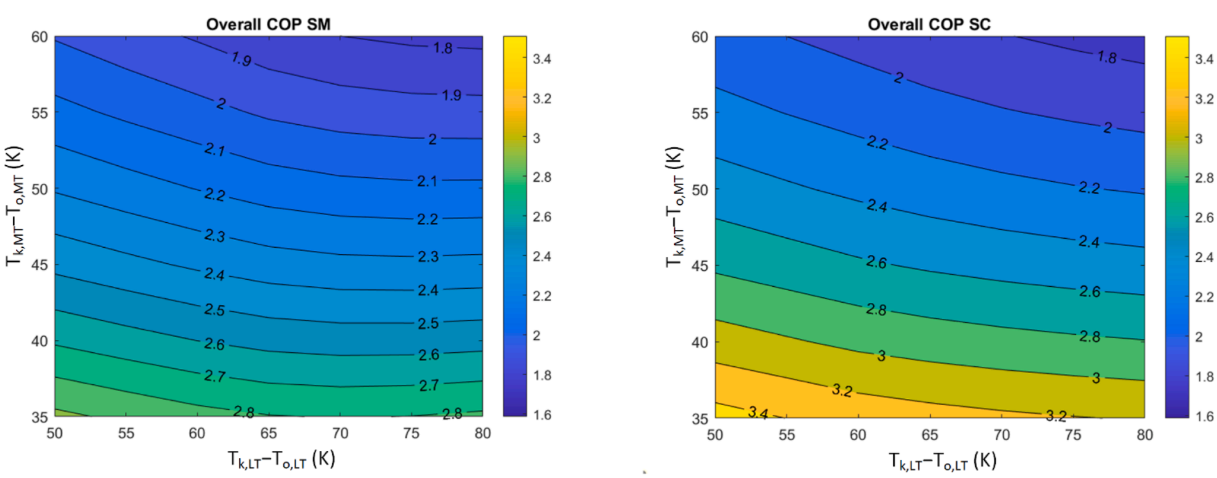


a)

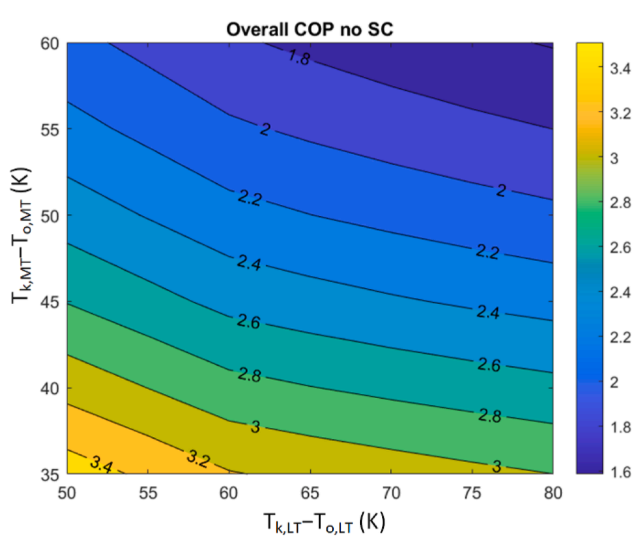


b)

Fig. 9. COP (a) and discharge temperature (b) for R-449A cases in the LT system.

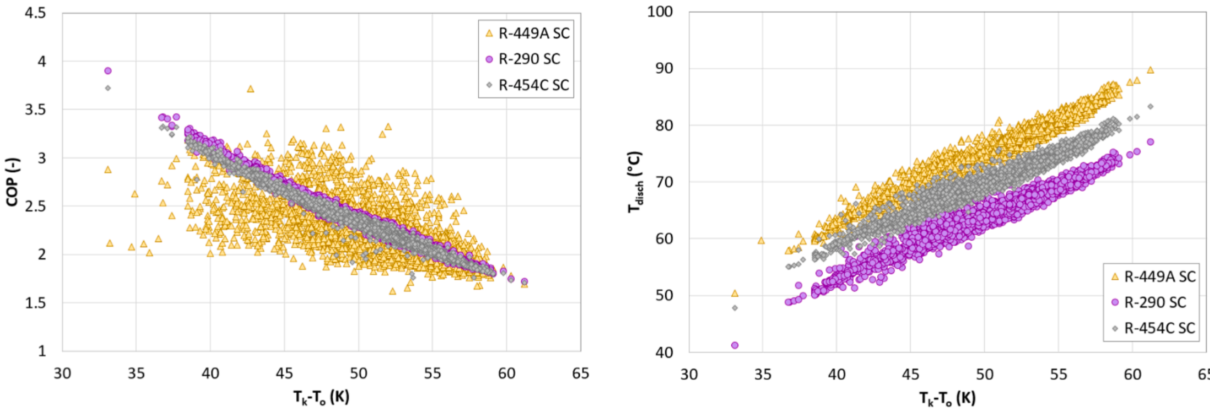


a) b)



c)

Fig. 10. Overall COP for R-449A cases.



a) b)

Fig. 11. COP (a) and discharge temperature (b) for R-449A SC, R-454C SC, and R-290 SC in the MT system.

average value of 2. The SC case has intermediate values between the two other cases, with an average of 2.25, which is 12.5 % higher than the SM case. On the other hand, the case without SC has the highest values, being 2.47 its average, 23.5 % higher than SM. These favorable differences with modern compressors are due to their higher isentropic efficiency since this reduces the work produced by the compressor and therefore, its electrical power consumed consumption. It is concluded that the most favorable case is the no SC one. This is because, in this case, all the cooling power is used by the evaporator to cool the food, while in the rest of the cases, there is a part of the produced cooling power that is diverted to subcool the LT system.

Regarding the discharge temperature, the highest average value is obtained with the SM case, which is 80.7 °C. The maximum reached temperature is 91.3 °C. In the SC case, the average temperature is 75.67 °C, and the maximum value is 89.76 °C. Finally, in the case without SC, the average is 73.45 °C, and the maximum value is 87.57 °C. The differences in discharge temperatures of different cases are mainly due to the variation in the isentropic efficiency of compressors. Despite the differences in discharge temperatures, it has been found that 120 °C is not reached, so the safety limits for the operation of compressors are not exceeded.

3.2.2. LT system

COP and discharge temperature results obtained for the three analysed cases in the LT system are shown in Fig. 9.

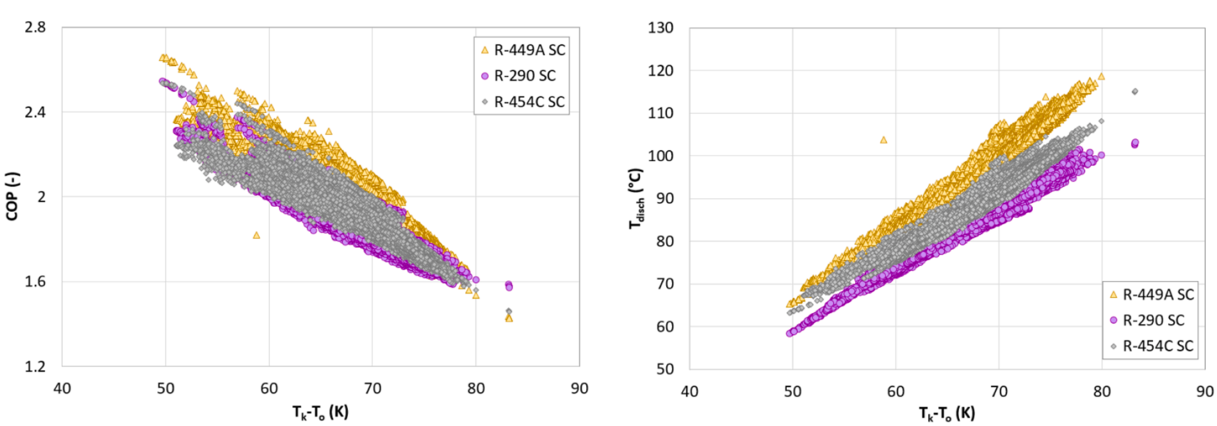
SM case has an average COP of 1.88, while SC has an average of 1.9, and no SC has an average of 1.46. This leads to a deviation of 1.06 % for SC with respect to SM and a deviation of − 22.34 % for no SC with respect to SM. Therefore, it is observed that SM and SC cases present a similar energy efficiency, while the no SC COP is significantly lower. This is because in the case without SC, the LT system does not receive the subcooling produced by the MT system, and therefore, to achieve the same cooling power as in the SM and SC cases, the compressor must produce a higher amount of work, thereby increasing its electricity consumption and decreasing its efficiency.

The discharge temperature of SM has an average value of 93.93 °C and a maximum value of 103 °C. On the other hand, SC has an average value of 99 °C and a maximum value of 118.6 °C. Finally, no SC has an average value of 98.3 °C and a maximum value of 117.8 °C. Moreover, as can be seen, the points obtained in the SC case are mostly hidden behind those obtained for the no SC case. This is due to the similarity of the isentropic efficiency. The maximum temperatures are close to 120 °C with SC and no SC. However, this limit is not reached. In addition, in these cases, 110 °C is exceeded less than 10 % of the total operating time. Therefore, the safety limit is not exceeded, so the compressor's life would not be affected.

3.2.3. Global system

Overall COP results for the different R-449A cases as a function of the MT and LT temperature lifts are shown in Fig. 10.

SM case is the one with the lowest energy efficiency. Its average overall COP is 2.128. On the other hand, SC average overall COP is 2.251, and no SC average overall COP is 2.144. These values represent improvements of 5.8 % for SC and 0.75 % for no SC with respect to SM. It



a) b)

Fig. 12. COP (a) and discharge temperature (b) for R-449A SC, R-454C SC, and R-290 SC in the LT system.

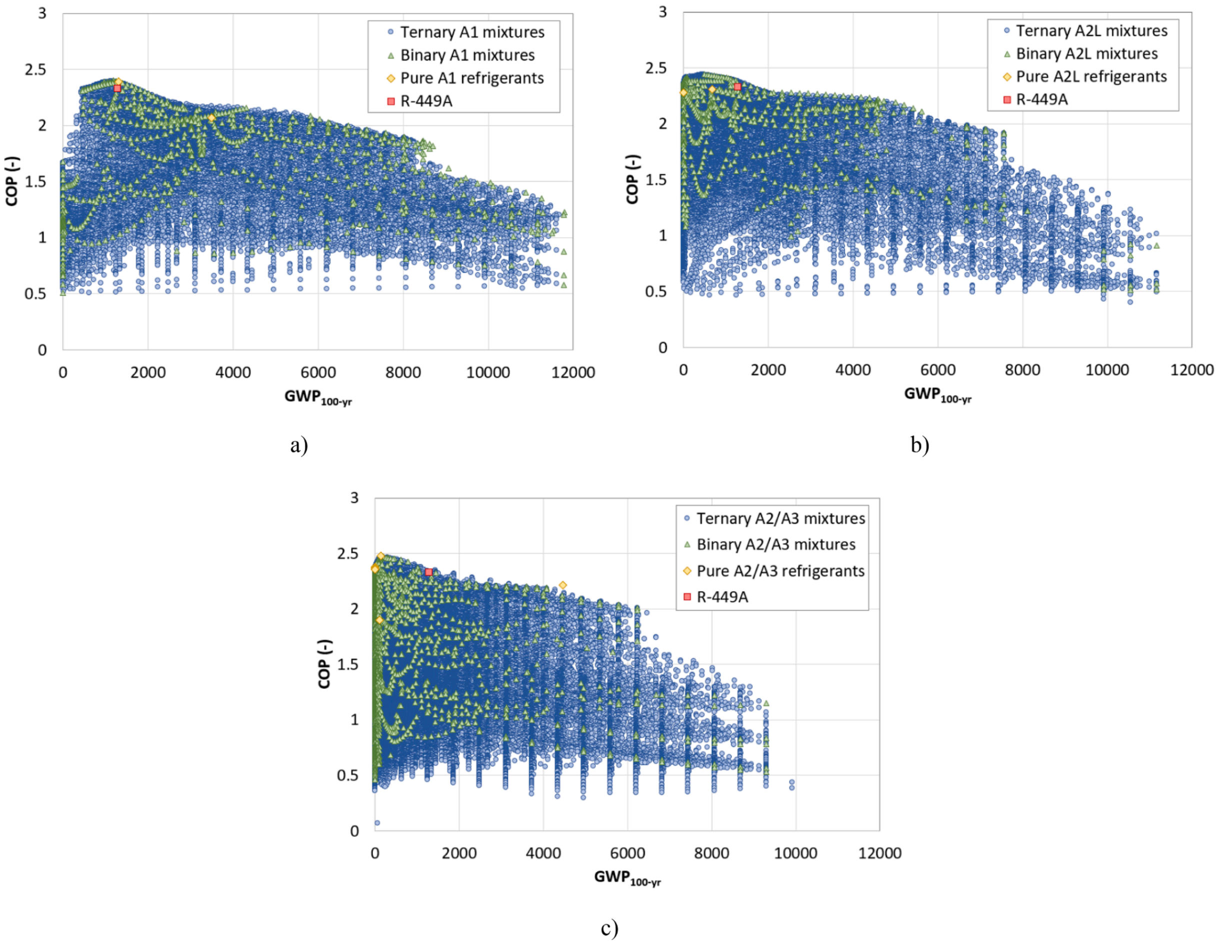


Fig. 13. Determined refrigerants for the MT system.

is therefore concluded that the most energy-efficient case is SC, that is, the configuration with the subcooling exchanger in the LT system with modern compressors. In addition, it can be seen that subcooling most benefits the system when the temperature lift is greater, that is, when the work produced by the LT compressors is greater.

3.3. *Commercial refrigerants implementation*

Low-GWP refrigerants R-454C and R-290 are here analysed, comparing them to R-449A with SC configuration.

3.3.1. *MT system*

Obtained COP and discharge temperature for R-454C and R-290 with SC configuration in the MT system are shown in Fig. 11.

The average COP for R-454C is 2.29, meaning an increase of 1.18 % with respect to R-449A SC and 14.5 % with respect to R-449A SM. On the other hand, the average COP of R-290 is 2.3, meaning an increase of 2.2 % with respect to R-449A SC and 15 % with respect to R-449A SM. These values show that both can be considered alternative refrigerants to R-449A in the MT system regarding their energy performance. Regarding the discharge temperature, the average for R-454C is 70.54 °C, while its maximum value is 83.3 °C. For R-290, the average is 64.68 °C, and its maximum is 77.06 °C. Therefore, there is no risk of damaging the compressor due to the high discharge temperatures.

3.3.2. *LT system*

Obtained COP and discharge temperature for R-454C and R-290 with SC configuration in the LT system are shown in Fig. 12.

The average COP for R-454C is 1.87, meaning a decrease of 1.5 % with respect to R-449A SC and 0.5 % with respect to R-449A SM. On the other hand, the average COP of R-290 is 1.83, meaning a decrease of 3.6 % with respect to R-449A SC and 2.1 % with respect to R-449A SM. Therefore, these refrigerants have a worse energy performance than R-449A. Regarding the discharge temperature, the average for R-454C is 91.58 °C, while its maximum value is 108.1 °C. For R-290, the average is 88.34 °C, and its maximum is 100.8 °C. Therefore, there is no risk of damaging the compressor due to the high discharge temperatures.

3.4. *Best mixtures determination*

As shown, a refrigeration system's energy performance depends on the chosen refrigerant. Following this statement, the results concerning the search for the best mixtures in energy terms are presented in this section.

3.4.1. *MT system*

Obtained mixtures for the MT system, separated according to their security classification, are shown in Fig. 13.

There are 2 pure A1 refrigerants suitable for the MT system. None of them have higher COP and lower GWP than R-449A. Moreover, 741 A1 binary mixtures suitable for the MT system have been found, 28 of which have a higher COP and a lower GWP than R-449A. Finally, 31,591 ternary A1 mixtures have been found for the MT system, 159 of which have higher COP and lower GWP compared to R-449A. This makes a total of 32,334 A1 refrigerants suitable for the MT system, 187 of which are better than R-449A in terms of energy efficiency and GWP. The mixture with the highest COP is composed of R-134a and R-1243zf in mass proportions of 0.75 and 0.25, respectively. Its COP is 2.3868, and

Table 6
Best A1, A2L, and A2/A3 mixtures for the MT system.

Composition	%wt	COP	GWP	Security classification	Glide (K)	T_{disch} (°C)	NBP (°C)
R-134a/R-1243zf	75/25	2.386	1147.5	A1	0.015	66.04	−25.84
R-134a/R-152a	30/70	2.442	573.8	A2L	0.120	78.93	−25.43
R-152a	1	2.476	164	A2	0.000	85.03	−24.32

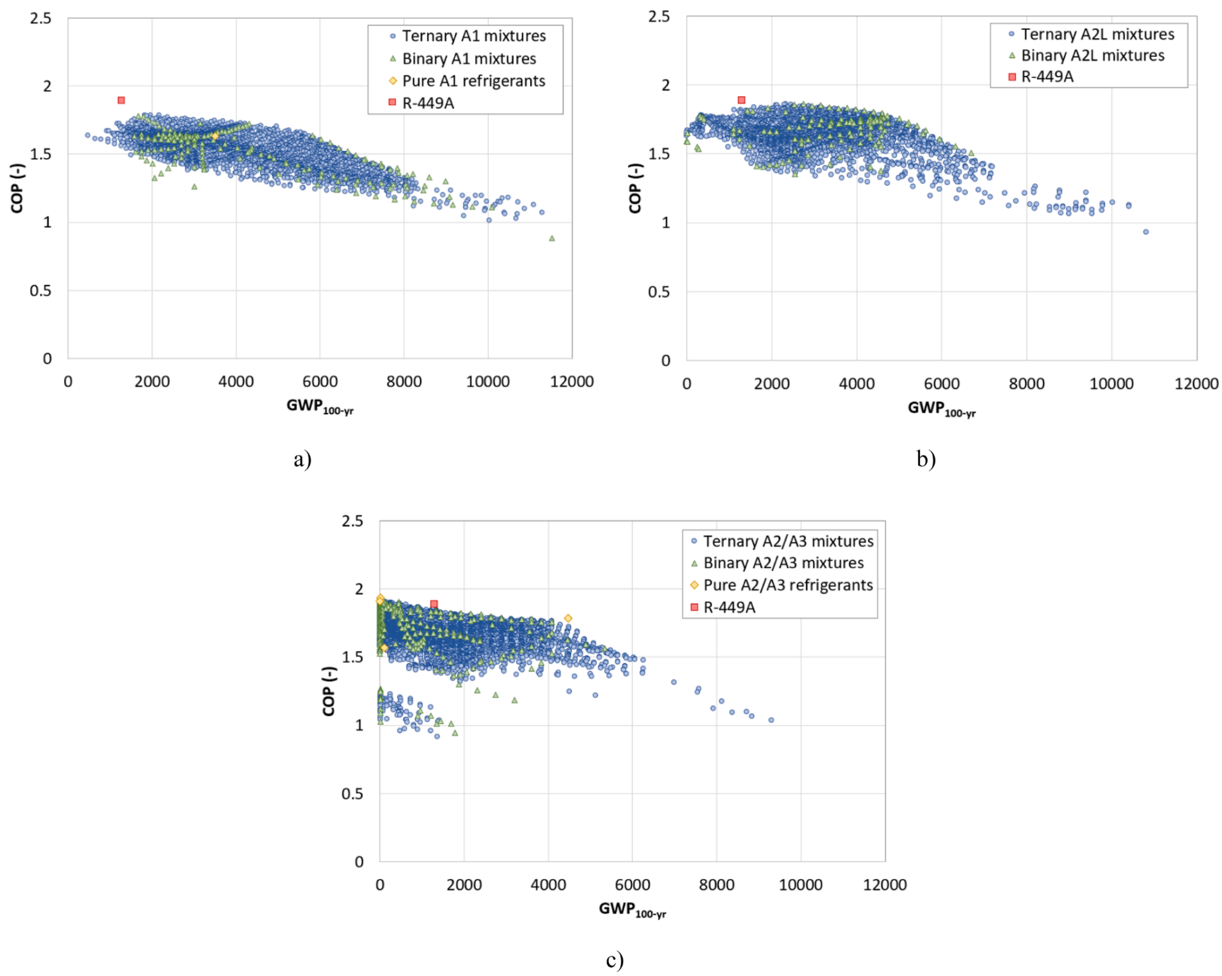


Fig. 14. Determined refrigerants for the LT system.

its GWP is 1147.5. Being the COP obtained with the R-449A 2.33 for the analysed conditions and its GWP of 1282, with the best A1 mixture, there is an increase in COP of 2.44 % and a reduction in GWP of 10.5 %.

Regarding A2L refrigerants, there are 2 pure A2L refrigerants suitable for the MT system. None of them have higher COP and lower GWP compared to R-449A. Moreover, 674 binary mixtures suitable for the MT system have been found, 77 of which have higher COP and lower GWP than R-449A. Finally, 40,083 ternary mixtures suitable for the MT system have been found, 1500 of which have higher COP and lower GWP than R-449A. This makes a total of 40,759 refrigerants suitable for the MT system, 1577 of which are better than R-449A in terms of energy efficiency and GWP. The highest COP is achieved by the mixture composed of R-134a and R-152a in mass proportions of 0.3 and 0.7, respectively. Its COP is 2.4426, and its GWP is 573.8, which means an improvement in energy efficiency of 5.28 % and a 55 % reduction in GWP compared to R-449A. As can be seen, the best A2L mixture has, as the best A1 mixture, R-134a in its composition. But in this case, the combination with R-152a gives it a higher COP and flammability rate.

Finally, regarding A2/A3 refrigerants, 6 pure A2/A3 refrigerants suitable for the MT system have been studied. 4 of them have higher COP and lower GWP than R-449A. On the other hand, 1637 A2/A3 binary mixtures suitable for the MT system have been found, 109 of which have higher COP and lower GWP than R-449A. Finally, 123,997 ternary A2/A3 mixtures suitable for the MT system have been found, 1436 of which have higher COP and lower GWP than R-449A. This makes a total of 125,640 A2/A3 refrigerants obtained for the MT system. 1549 of them are better than R-449A regarding energy efficiency and GWP. The one with the highest COP is the pure R-152a, with a value of 2.4768 and with a PCA of 164. This means an increase in COP of 6.76 % and a reduction in GWP of 87 %. Thus, the R-152a is present in two of the three best refrigerants obtained. The higher its content in the mixture with R-134a, the higher its COP, flammability, discharge temperature, and boiling point. Conversely, its GWP is lower.

Table 6 summarizes the results regarding the determination of the best mixtures for the MT system. These are the selected refrigerants for the refrigeration system simulation in the next section.

3.4.2. LT system

Obtained mixtures for the LT system, separated according to their security classification, are shown in Fig. 14.

Table 7
Best A1, A2L, and A2/A3 mixtures for the LT system.

Composition	%wt	COP	GWP	Security classification	Glide (K)	T_{disch} (°C)	NBP (°C)
R-449A	1	1.8929	1282	A1	5.6600	95.9	−46.0
R-290/R-1270	15/85	1.9357	1.533	A3	0.0018	95.3	−47.7

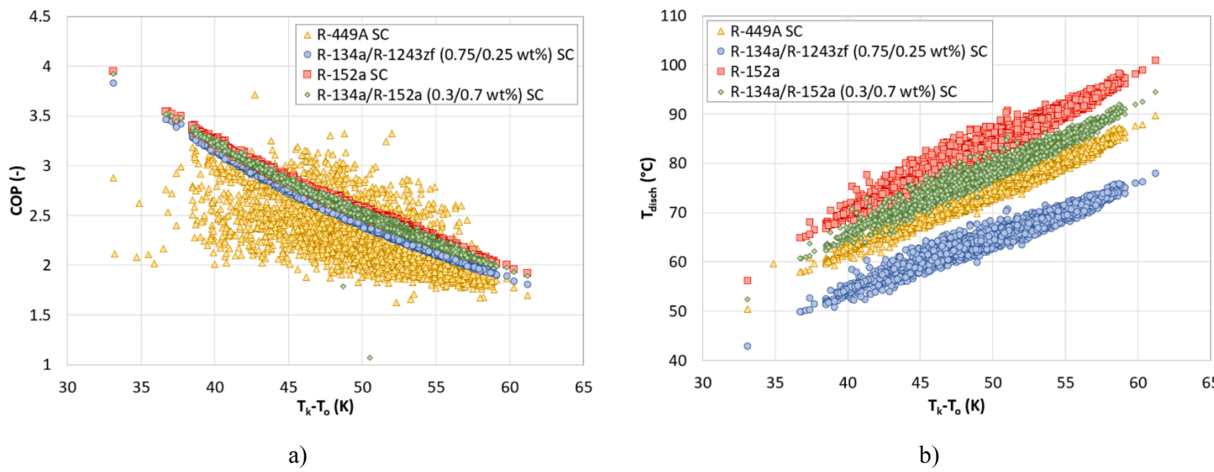


Fig. 15. COP (a) and discharge temperature (b) for the best MT mixtures.

There is only one pure A1 refrigerant suitable for the LT system, R-125. It has a lower COP and higher GWP compared to R-449A. Moreover, 173 A1 binary mixtures suitable for the LT system have been found, none with higher COP and lower GWP than R-449A. Finally, 4415 A1 ternary mixtures have been found for the LT system, none of which has higher COP and lower GWP compared to R-449A. All this makes a total of 4589 A1 refrigerants, all of them worse than R-449A in terms of energy efficiency and GWP. It is therefore concluded that R-449A is the most appropriate refrigerant for the LT system. Thus, this refrigerant will be maintained when studying the behaviour of the complete system with the best A1 mixtures.

Regarding A2L refrigerants, no pure refrigerant is suitable for the LT system. Moreover, 134 binary mixtures suitable for the LT system have been found, none of which have higher COP and lower GWP than R-449A. Finally, 4516 ternary mixtures suitable for the LT system have been found, none of which have higher COP and lower GWP than R-449A. This makes a total of 4650 refrigerants, all of them worse than R-449A in terms of energy efficiency and GWP. Therefore, as in the case of A1 mixtures, it is concluded that R-449A is the most appropriate refrigerant for the LT system, so this refrigerant will be maintained when studying the system's behaviour with the best A2L mixtures.

Finally, regarding A2/A3 refrigerants, 2 pure refrigerants suitable for LT system have been studied. Both R-290 and R-1270 have higher COP and lower GWP than R-449A for the studied conditions. On the other hand, 300 A2/A3 binary mixtures suitable for the LT system have been found, 34 of which have higher COP and lower GWP than R-449A. Finally, 9793 A2/A3 ternary mixtures have been found for the LT system, 116 of which have higher COP and lower GWP than R-449A. This makes a total of 10,095 A2/A3 refrigerants obtained for the LT system. 152 of these are better than R-449A regarding energy efficiency and GWP. The refrigerant with the highest COP is the mixture composed of 15 wt% of R-290 and 85 wt% of R-1270, with a COP of 1.9357 and a GWP of 1.533. This represents an increase in COP of 2.26 % and a reduction in GWP of 99.88 %.

Table 7 summarizes the results regarding the determination of the best mixtures for the LT system. These are the selected refrigerants for the refrigeration system simulation in the next section.

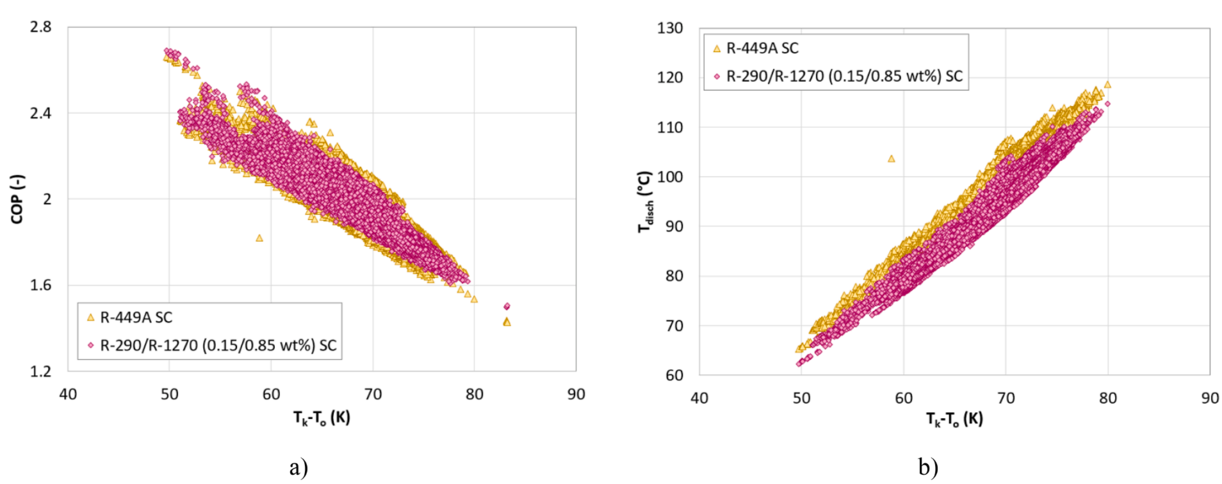


Fig. 16. COP (a) and discharge temperature (b) for the best LT mixtures.

Table 8
Summary of results for the different studied cases.

Studied case	System	Refrigerant	Average COP (-)	Average T_{disch} (°C)
R-449A SM	MT	R-449A	2.00	80.7
	LT	R-449A	1.88	93.9
R-449A SC	MT	R-449A	2.25	75.6
	LT	R-449A	1.90	99.0
R-449A no SC	MT	R-449A	2.47	73.4
	LT	R-449A	1.46	98.3
R-454C SC	MT	R-454C	2.29	70.5
	LT	R-454C	1.87	91.5
R-290 SC	MT	R-290	2.30	64.6
	LT	R-290	1.83	88.3
BMA1 SC	MT	R-134a/R-1243zf (75/25 wt%)	2.37	65.6
	LT	R-449A	1.90	99.0
BMA2L SC	MT	R-134a/R-152a (30/70 wt%)	2.39	79.5
	LT	R-449A	1.90	99.0
BMA2/A3 SC	MT	R-152a	2.42	85.0
	LT	R-290/R-1270 (15/85 wt%)	1.92	95.6

3.5. Best mixtures implementation

Once the best mixtures for MT and the LT systems have been determined, their implementation in the supermarket refrigeration system is analysed. For this purpose, the results of interest will be calculated with all the operating points with which this system has worked, as has been done previously with R-454C and R-290.

3.5.1. MT system

COP and discharge temperatures of the best mixtures in the MT system are shown in Fig. 15.

The average COP value for the best A1 mixture, consisting of 75 wt% of R-134a and 25 wt% of R-1243zf, is 2.37. This represents an increase in energy efficiency of 5.34 % with respect to R-449A SC, maintaining the safety classification A1 in this case. For the best A2L mixture, composed of 30 wt% of R-134a and 70 wt% of R-152a, the average COP value is 2.3933, meaning an increase of 6.37 % with respect to R-449A SC. Regarding the best A2/A3 refrigerant, R-152a, its average COP value is 2.42, meaning an increase of 7.5 % with respect to R-449A SC.

The discharge temperatures present a significant variation between the different refrigerants. The average value for the best A1 mixture is 65.6 °C, and the maximum is 78 °C. For the best A2L mixture, the average is 79.59 °C, and the maximum is 94.55 °C. Finally, for the best A2/A3 refrigerant, the average is 85 °C, and the maximum is 101 °C. Despite this variation, no case presents such a high discharge temperature that the compressor life is negatively affected.

3.5.2. LT system

COP and discharge temperatures of the best mixtures in the LT system are shown in Fig. 16. As previously said, there is no better A1 or A2L mixture than R-449A, so this refrigerant is kept in these cases for the system analysis.

COP and discharge temperature values for R-449A have been previously shown in the subcooling analysis. For the best A2/A3 mixture, the average COP is 1.92, meaning an increase of 1.05 % with respect to R-449A SC.

The average discharge temperature value for the best A2/A3 mixture is 95.6 °C, and the maximum is 114.7 °C. As 120 °C is not reached, the compressor life is not affected.

A summary of all the interest results regarding the system operation is presented in Table 8.

Table 10
Electrical consumption of different studied cases.

System	Total electrical consumption (MWh $year^{-1}$)
R-449A SM	700.968
R-449A SC	638.648
R-449A no SC	662.818
R-454C SC	636.976
R-290 SC	637.289
BMA1 SC	616.229
BMA2L SC	611.904
BMA2/A3 SC	604.863

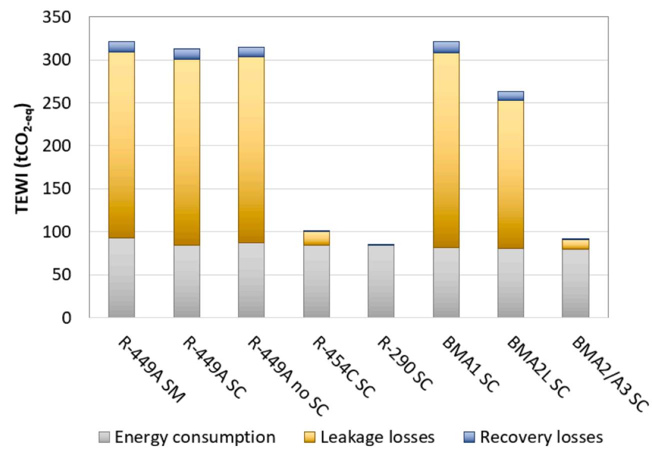


Fig. 17. TEWI results in Sweden.

3.6. TEWI

To calculate equivalent CO_2 emissions, the result of TEWI analysis, it is necessary to determine the refrigerant charge. To do so, a comparative method has been followed. The R-449A charge for SM and SC cases has been experimentally measured. Knowing the temperature and pressure at different points of the cooling cycle, the refrigerant density can be easily obtained. So, their charge can be estimated by calculating the alternative refrigerant's densities and comparing them to that of R-449A. The calculated values are shown in Table 9.

It is also necessary for the environmental analysis to calculate the electricity consumption produced by the operation of the cooling system. Thus, since the cooling power remains constant for all the cases, the variations in COP between them are translated into variations in electrical consumption. The values obtained for the different cases are shown in Table 10.

Once the necessary values have been defined, the parameters of interest are calculated. The results are shown in Fig. 17.

As it can be seen, the lowest environmental impact is obtained with the operation of R-290 in both MT and LT systems. This is because, due to its low GWP (0.02) and low refrigerant charge, the impact of leakage losses and recovery losses is practically zero. Thus, the amount of CO_2 equivalent emitted with this solution is 84.12 tons, representing a

Table 9
Refrigerant charge for different studied cases.

Charge $cycle^{-1}$ (kg)	R-449A SM	R-449A SC	R-449A no SC	R-454C SC	R-290 SC	BMA1 SC	BMA2L SC	BMA2/A3 SC
MT	12.50	12.50	12.50	3.05	6.78	15.67	13.76	12.42
LT	18.75	18.75	18.70	17.78	8.27	18.75	18.75	8.47

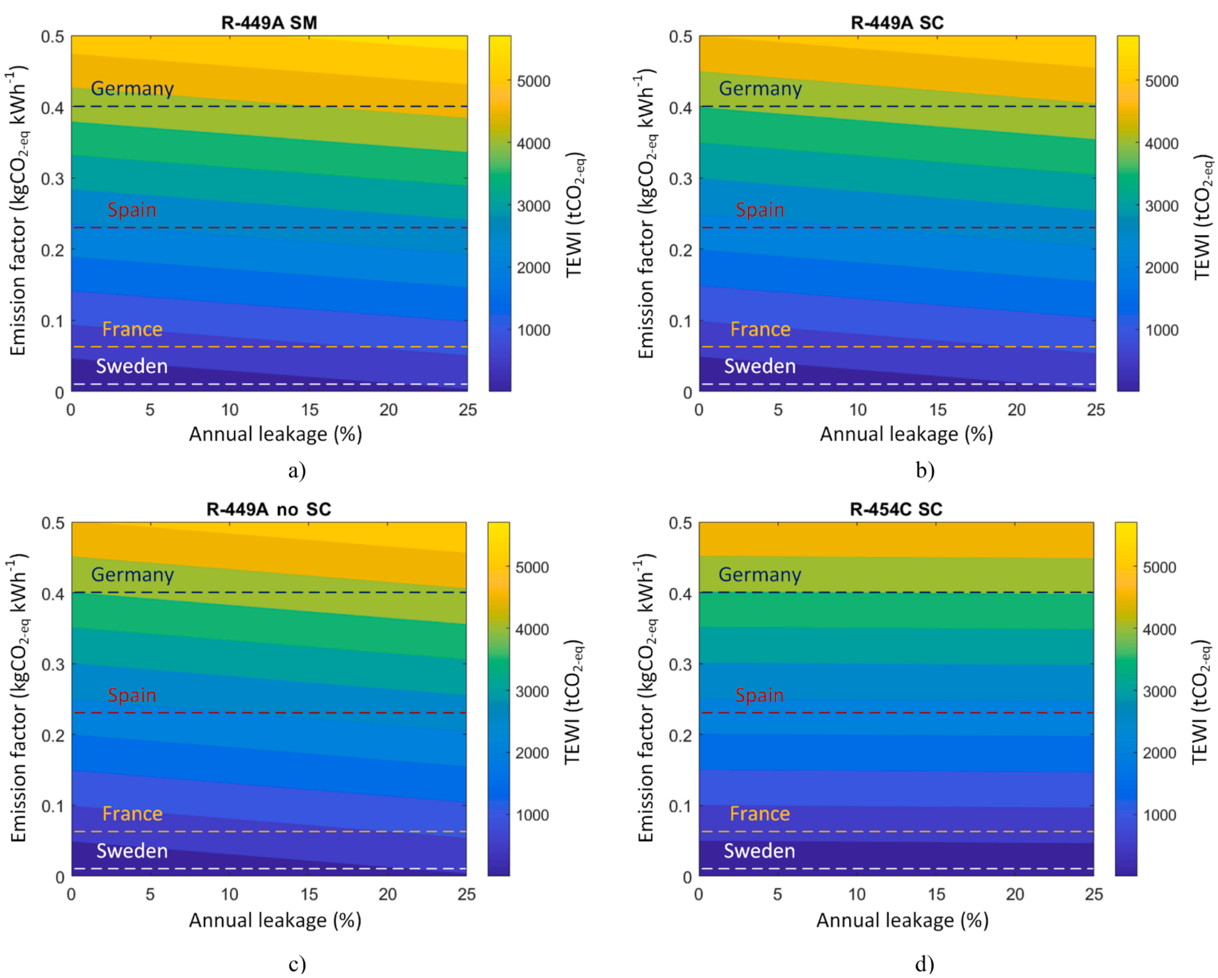


Fig. 18. TEWI results according to the emission factor and the % of leakage.

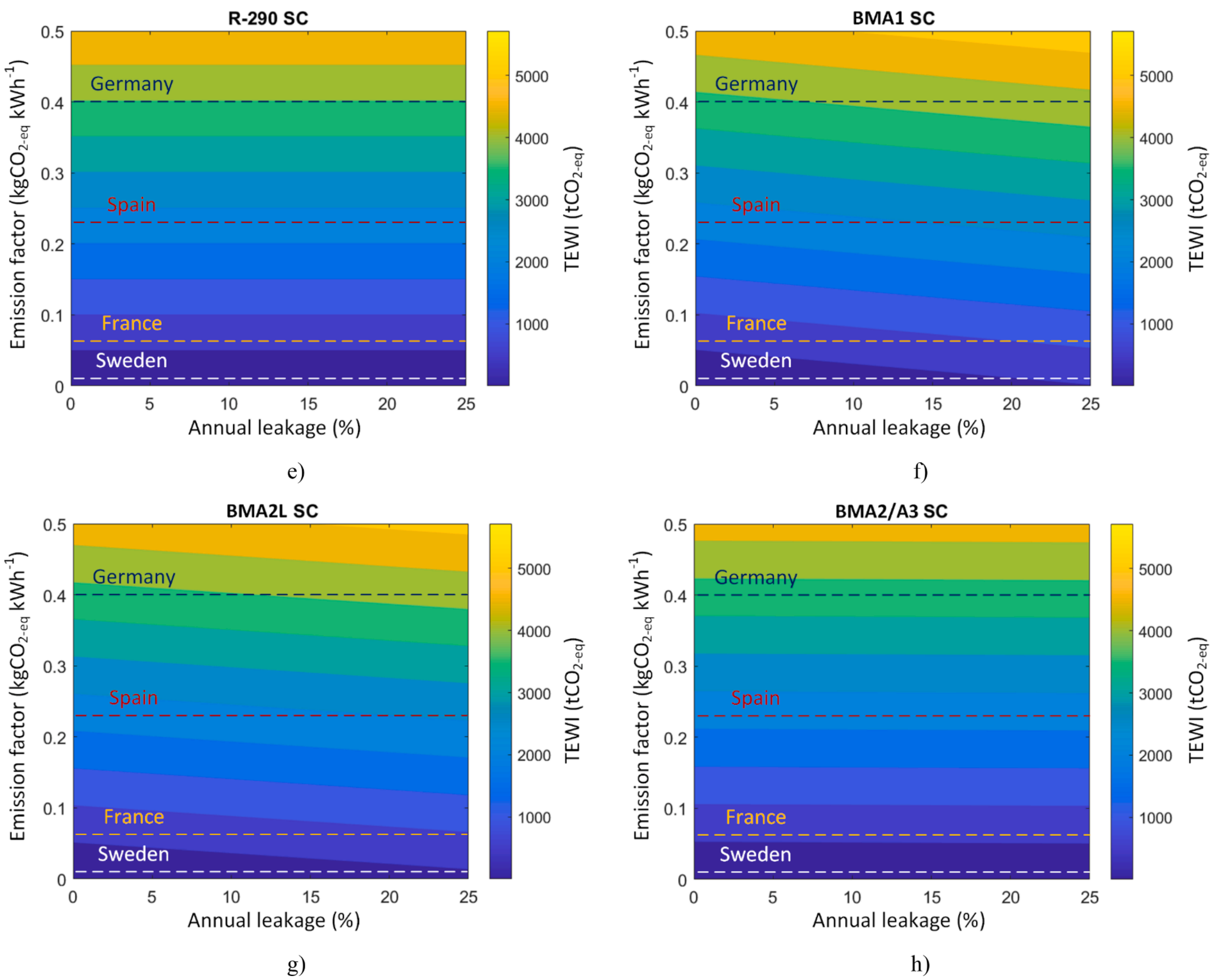


Fig. 18. (*continued*).

reduction of 73.78 % with respect to R-449A SM and 73.39 % with respect to R-449A SC. The second best result is obtained with the BMA2/A3 SC case, consisting of the use of the R-152a in the MT system and the mixture of R-290 and R-1270 in the LT system. Although in this case the impact due to energy consumption is lower than in case R-290 SC, the low emission factor in Sweden and the higher GWP make the impact due to leakage and recovery losses greater, resulting in a total impact of 91,53 tonnes of CO_2 equivalent, meaning a reduction of 71.48 % respect to R-449A SM and 71.05 % respect to R-449A SC. In addition, the use of the R-454C in both MT and LT systems also shows a significant reduction in environmental impact, with 101.42 tonnes of CO_2 equivalent. On the other hand, higher values of 320,86 and 262,8 tonnes, respectively, are obtained in the BMA1 SC and BMA2L SC cases due to the fact that the R-449A has been maintained in the LT system. In addition, in the BMA1 SC case, the density of the mixture used in the MT system requires a higher refrigerant charge, thus increasing the impact of recovery and leakage losses, resulting in a higher result than R-449A SM. Finally, with R-449A SC and R-449A no SC, reductions of 2.56 % and 2.53 %, respectively, are obtained with respect to R-449A SM, so the values obtained in both cases are very similar.

The environmental impact of each case strongly depends on the location of the installation since the emission factor depends on how energy is obtained in each country. Likewise, the result obtained also depends on the percentage of leaks selected since, with higher percentages, refrigerants with a higher GWP will cause a greater impact, and those with a reduced GWP will not be so affected by the increase in that percentage. To illustrate this variation of the results, Fig. 18 shows the results obtained for each case according to the percentage of leakage and the emission factor.

Taking as an example the operation of the studied installation located in Spain (with an emission factor of 0.232 kg CO_2 kWh^{-1}) with an annual leakage percentage of 12 %, the best environmental results are obtained with BMA2/A3 case, obtaining a reduction of 20.66 % respect to R-449A SM. With R-290 SC, the best resulting case in Sweden, a reduction of 16.87 % is obtained. Therefore, the environmental impact of a refrigeration system must be calculated with specific location parameters to obtain reliable results.

4. Conclusion

Commercial refrigeration is a worldwide sector,thanks to which it is possible to preserve food in an appropriate state for consumption. However, this sector is responsible for a large amount of GHG emissions to the atmosphere, contributing to global warming and climate change. Thus, this work responds to the main objective initially set, which is none other than the search for solutions with a lower environmental impact.

For this purpose, the measurements taken in a commercial refrigeration installation with the use of R-404A and R-449A have been analysed, concluding that R-449A is a viable alternative to R-404A. Once this has been determined, a model of the cooling system has been made using EES software to semi-empirically evaluate the possibility of removing the existing subcooling in the LT system, which comes from the secondary fluid of the MT system, so it produces a greater cooling power than necessary. Thus, after analysing the system's behaviour with and without subcooling, it has been concluded that the subcooling from the MT system is more beneficial in terms of energy since, in this case, the global COP of the installation is approximately 5 % higher than the case without subcooling.

R-449A is a refrigerant with a moderate GWP (1282), so it is considered a transition alternative until low-GWP refrigerants are implemented. Therefore, after obtaining the best configuration of the cooling system, the operation of R-454C (GWP = 146) and R-290 (GWP = 0.02) in the studied system has been analysed. The results show an improvement in the COP of the MT system of 1.18 % and 2.2 % compared to R-449A, respectively. However, in the LT system, a decrease in COP of 1.5 % and 3.6 % is obtained with respect to the R-449A SC, respectively. Even with worse results in the LT system, overall reductions in the electrical consumption of the installation are obtained, so both refrigerants can be considered low environmental impact alternatives to R-449A.

Then, in order to find new alternative refrigerants with higher COP and lower GWP than R-449A, refrigerant mixtures have been searched from a list of 21 pure refrigerants using a computer program written in the Python language. In the MT system, the best refrigerant A1 found consists of a mixture of 75 % by mass of R-134a and 25 % by mass of R-1243zf, and with it, an improvement in COP of 5.34 % and a reduction in GWP of 10.5 % compared to R-449A. As for the best refrigerant A2L found, it is a mixture of 30 % by mass of R-134a and 70 % by mass of R-152a, and with it, an improvement in COP of 6.37 % and a reduction in GWP of 55 %. Finally, the best refrigerant A2/A3 found for the medium temperature system is the pure R-152a, with an improvement in COP of 7.5 % and a reduction in GWP of 87 %. On the other hand, no A1 or A2L mixture was found to be better than the R-449A for the low temperature system, while the best A2/A3 refrigerant for this system is the 15 % mass mixture of R-290 and 85 % mass of R-1270, resulting in a COP improvement of 1.05 % and a GWP reduction of 99.88 %.

Once the energetic results of the different refrigerants have been obtained, an environmental analysis has been made, in which it has been determined that the best solution for the system under study is the operation of the R-290, both in the MT and LT systems, with the configuration including subcooling. This solution reduces emissions by 73.38 % compared to the current operation of the R-449A cooling system. This reduction is marked by the almost non-existent GWP of R-290, which means that the environmental impact of this refrigerant is caused exclusively by energy consumption.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

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