

Article

Impact of Evaporator Operating Mode Switching on the Performance of CO₂ Commercial Refrigeration Systems

Ionuț Dumitriu , Costel Ungureanu  and Ion V. Ion 

Faculty of Naval Architecture, “Dunărea de Jos” University of Galați, Domnească St., No. 111, 800008 Galați, Romania; secretariat.fan@ugal.ro

* Correspondence: ionut.dumitriu@ugal.ro (I.D.); costel.ungureanu@ugal.ro (C.U.); Tel.: +40-745-479-605 (I.D.); +40-747-864-866 (C.U.)

Abstract

Commercial refrigeration systems represent some of the largest energy consumers in supermarkets, and therefore particular attention needs to be paid to increasing energy efficiency to reduce overall energy consumption and meet climate goals by 2030. This study investigates the performance of a CO₂ (R744) commercial refrigeration system with evaporators operating alternately in dry and flooded modes. This operation is possible due to a particular adjustment using liquid sensors installed in the middle of both low-temperature (LT) and medium-temperature (MT) liquid separators, which transmit information to the controllers that regulate the compressor rack and evaporators, to switch from dry to flooded operation when the liquid level rises and vice versa when the level drops. The results show that the correct regulation of the system of 50% with 6K superheat operation and 50% with 3K superheat operation on MT evaporators, respectively, and 50% with 6K superheat operation and 50% with 4K superheat operation on LT evaporators leads to a reduction of energy consumption compared to 100% operation of all evaporators with 6K superheat by 6.9% per year for the compressor rack.

Keywords: energy consumption; commercial refrigeration; carbon dioxide; dry/flooded evaporators

1. Introduction

Approximately 30% of global final energy consumption and 26% of global energy-related emissions represent the operations of buildings, of which 18% are indirect emissions from the production of the electricity and heat used in buildings, and 8% are direct emissions in buildings [1]. Around 75% of all man-made greenhouse gas emissions in the EU are CO₂ emissions from energy use and thus represent a primary contributor to global warming [2]. Romania planned to cut its greenhouse gas emissions by 85% by 2030 compared to 1990, aiming to achieve 34 MtCO₂eq. This will be possible due to massive reductions in the energy sector (−87%), industry (−77%), and agriculture (−44%) [3].

The frozen food global market size was valued at USD 503.75 billion in 2024, and it is estimated to grow to USD 712.76 billion by 2030. The Compound Annual Growth Rate (CAGR) of 6.0% is predicted to increase from 2025 to 2030. In the next years, the increasing popularity of self-stable foods among consumers worldwide is expected to boost industry growth [4]. The estimated value for the dehydrated meat products market was USD 862.3 million in 2023. The expectation is still that it will grow strong, at a CAGR of



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around 4.7% during the forecast period (2024–2032), leading to an increase in demand for self-stable food and for convenient products globally [5].

Supermarket refrigeration systems are an important part of the cold chain and represent a critical role in distribution, food storage, and retailing. Refrigeration is responsible for most of the electricity consumption of retail food stores. For hypermarkets, its consumption represents between 25% and 30% and for food-dominant convenience stores, more than 60% [6]. Refrigeration systems are followed by HVAC systems (26%), lighting (8%), and electrical equipment (6%). Particularly for an HVAC system, 8% of energy consumption is due to cooling, 6% to heating, and 11% to the fans [7].

Nowadays, CO₂ (R744) is the preferable natural working fluid for the entire commercial refrigeration industry, because of its characteristics. It has high safety features (A1 ASHRAE classification, non-toxic, and non-flammable) and low Global Warming Potential (GWP = 1). Supermarket refrigeration installations need large amounts of refrigerant and so, have a large direct environmental impact regarding refrigerant leakages [8]. In accordance with the prediction, there will be between 4000 and 6000 new stores annually with CO₂ refrigeration installations between 2020 and 2030, and a total of 65,000–85,000 markets using this technology in Europe by 2030 [9]. At the conclusion of the last century, it was known that the thermo-physical properties of CO₂ were substantially corrected. In the last years, the deep investigations of CO₂ refrigeration systems and heat transfer got a prosper development, being able to promote the practical applications and improve energy efficiency [10]. The standard transcritical CO₂ refrigeration system has significant throttling losses which reduces its energy efficiency, especially when operating at higher ambient temperatures [11]. Many methods including mechanical subcooling, parallel compression, and ejectors have been tested in the field to achieve superior system performance and improved energy efficiency. When the ambient temperature rises, the CO₂ at the exit of a gas cooler (GC) is throttled into a flash tank, resulting in a higher production of vapor flash affecting the system's performance [12]. In the case of parallel compression, the optimum high pressure is directly dependent on the GC outlet temperature. It is higher when the GC outlet temperature rises, and it practically does not depend on the evaporation temperature. In contrast with the optimum high pressure, the optimum intermediate pressure depends on both the GC outlet temperature and the evaporation temperature. It is higher when the evaporation level and the GC outlet temperature are higher. It was possible to state two general expressions from the field data to establish the optimum GC and intermediate pressures for CO₂ single stage transcritical refrigeration installations with parallel compression, with these being dependent only on the evaporation level and the GC outlet temperature [13]. In the case of ejector technology, simulations were conducted to examine and optimize the refrigeration system for different ambient temperatures and compared with a traditional booster system. The results indicated that the compressor work of the new booster system with an ejector can be reduced by more than 10% compared to the standard booster refrigeration system at an ambient temperature of 35 °C, resulting in a coefficient of performance (COP) increase of more than 11% [14]. Refrigeration systems with ejectors have become a convincing alternative to the standard refrigeration cycle, as these systems can reclaim partially the exergy losses that take place during expansion [15]. Uniquely, the integrated mechanical subcooling system provides an improvement to the refrigeration cycle. This united cycle (standard plus mechanical subcooling) can be designed three different ways: extracting CO₂ from the GC outlet, from the subcooler outlet, or from the liquid receiver [16]. The optimal energy performance of three systems (standard, with mechanical subcooling, with parallel compression) were compared for three heat rejection levels: 25.0 °C, 30.4 °C, and 35.1 °C. The field tests indicated increments in COP of 4.1% at 25.0 °C, 7.2% at 30.4 °C, and 9.5% at 35.1 °C, because of the utilization of integrated me-

chanical subcooling, and of 7.8%, 13.7% and 17.5% specifically when dedicated mechanical subcooling was used [17]. Thermoelectric subcooling cycles are improving the COP of the system between 20 to 25.6% but integrated with energy recovery systems (thermoelectric generators or expanders) lift these data up to 37.8% [18].

Field activity revealed that a refrigeration system could encourage evaporator over-feeding, both in on/off and continuous operations, consequently resulting in optimal heat transfer area utilization [19]. The utilization of overfed evaporators was proven to conduct a potential decrease of total irreversibility, down to 22% lower than the standard basic unit and the improved basic system, and 5% lower in comparison with parallel compression [20]. Because of incorrect sizing of the pipe route on the suction side, the additional drop in evaporating pressure leads to higher energy consumption at the compressor rack for systems that use CO₂ as a refrigerant. The entire system consumes around 1.5% more energy for every one-degree increase in temperature loss due to an additional pressure drop [21]. TESCO, the leading U.K.-based grocery retailer already installed CO₂ transcritical refrigeration systems in around 1000 of its stores, which means around one-third of its food outlets. The retailer plans to be HFC-free chain until 2035 [22]. ALDI U.S., a leader already in the U.S. for grocery users of CO₂ transcritical systems, has communicated that all its stores will use natural refrigerants by 2035. It is the first engagement of this kind for a major U.S. supermarket chain. From the company's more than 2300 stores in the U.S., more than 600 already use CO₂ and/or propane [23]. CARREFOUR Belgium has reached an important milestone: 77% of its possessed stores and 30% of its franchised sites are equipped now with CO₂ transcritical refrigeration systems. The Belgian supermarket chain, with 318 convenience stores, 349 supermarkets, and 40 hypermarkets, has the ambition to remodel all its owned stores to CO₂ technology by 2030 [24]. In northern Italy, a supermarket has diminished its energy bills, by a forecasted EUR 18,000 (USD 19,500) per year, only by using a CO₂-based rack that incorporates the store's refrigeration, HVAC, and hot water production demands into a single system [25]. More than 50% of METRO stores are now using only natural refrigerators in refrigeration installations. The international cash & carry retailer is committed to phasing out all its old HFC and HCFC systems by 2040 under its own F-gas Exit Program [26]. European retailer AHOLD DELHAIZE, a store owner in the U.S., Europe, and Indonesia, passed to natural refrigerants, as part of its official climate plan. For its European businesses, all refrigeration installations will use natural refrigerants by 2040 [27]. Transcritical CO₂ refrigeration is now the specification for all new stores in California for EMERYVILLE, a California-based discount retailer Grocery Outlet aiming to expand this technology to other regions in the U.S. [28]. The big Australian supermarket chains concentrated on transitioning to the new refrigeration technology and have chosen to remodel stores directly with new transcritical CO₂ racks, rather than perform retrofits and extend the life of existing installations with HFO blends [29]. The first nation in the Gulf Cooperation Council to approve the international agreement to phase down HFCs is the United Arab Emirates [30].

Official prices for energy in Europe, including non-recoverable taxes, have increased in recent years [31]. Rising energy prices become particularly important for assessing the economic impact of operating costs, especially when it becomes clear that prices cannot be kept under control.

Energy efficiency in general, including for CO₂ transcritical refrigeration systems which are the main energy consumers in supermarkets, has become an important topic in developing new technologies with the purpose of consuming less energy compared to the previously installed systems.

This study quantifies the net energy impact of flooded operation by accounting for compressor and auxiliary consumption, offering a system-level evaluation rarely reported

for commercial CO₂ installations. Unlike previous studies that focused on steady dry or fully flooded operation, this work evaluates the energy performance of a CO₂ commercial refrigeration system under alternating dry/flooded evaporator operation, reflecting real-world control strategies, which has not been studied before and therefore is not found in the existing literature. Based on the assessed energy performance, this study outlines the operating conditions under which switching between dry and flooded modes is energetically justified. The assessment is based on in-site measurements from a commercial installation, providing operational insight beyond laboratory-scale studies. The system is also analysed from an economic and environmental point of view.

2. CO₂ Commercial Refrigeration System with Evaporators Operating Alternately in Dry and Flooded Modes

2.1. Description of the System

The refrigeration installation serving a hypermarket with a total sales area of 3748 m² in a city of the middle western part of Romania was analysed. The transcritical CO₂ system was designed for a freezing capacity of 56.7 kW at a consumer's evaporating temperature of −31 °C, and a refrigerating capacity of 110 kW at a consumer's evaporating temperature of −5 °C. The system was designed for a maximum outdoor temperature of +39 °C, and inside the store, for a maximum temperature of +25 °C and relative humidity of 60%.

The cooling system is a “booster” type, and it is equipped with a liquid separator on both suction sides: on the circuit of the freezing compressors and on the circuit of the refrigeration compressors. The components of the CO₂ refrigeration system are shown in Figure 1, the principle of operation is in Figure 2, and the refrigeration cycle is in Figure 3. The components of the refrigeration system that are shown in Figure 1 are as follows.

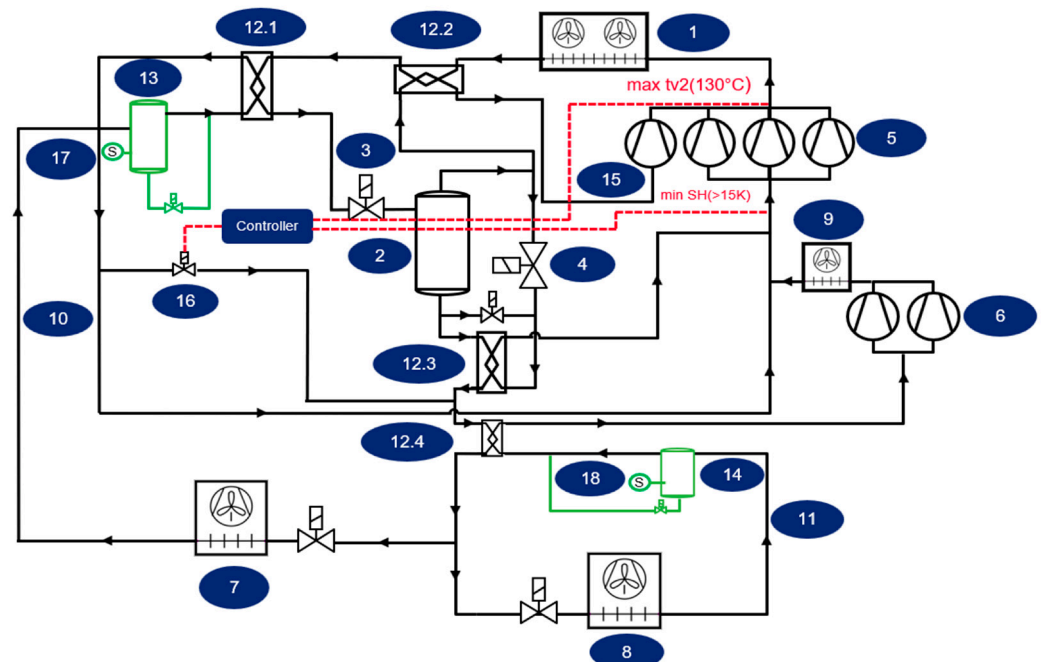


Figure 1. Components of refrigeration system with evaporators operating alternately in dry and flooded modes: 1: Gas Cooler; 2: Medium Pressure Receiver; 3: HP Control Valve; 4: MP Control Valve; 5: MT Compressors; 6: LT Compressors; 7: MT Evaporators; 8: LT Evaporators; 9: LT Desuperheater; 10: MT Suction Line; 11: LT Suction Line; 12.1: HT Plate Heat Exchanger; 12.2: ECO Plate Heat Exchanger; 12.3: MT Plate Heat Exchanger; 12.4: LT Plate Heat Exchanger; 13: MT Liquid Separator; 14: LT Liquid Separator; 15: Eco Compressor; 16: Liquid Injection; 17: MT Liquid Level Sensor; and 18: LT Liquid Level Sensor.

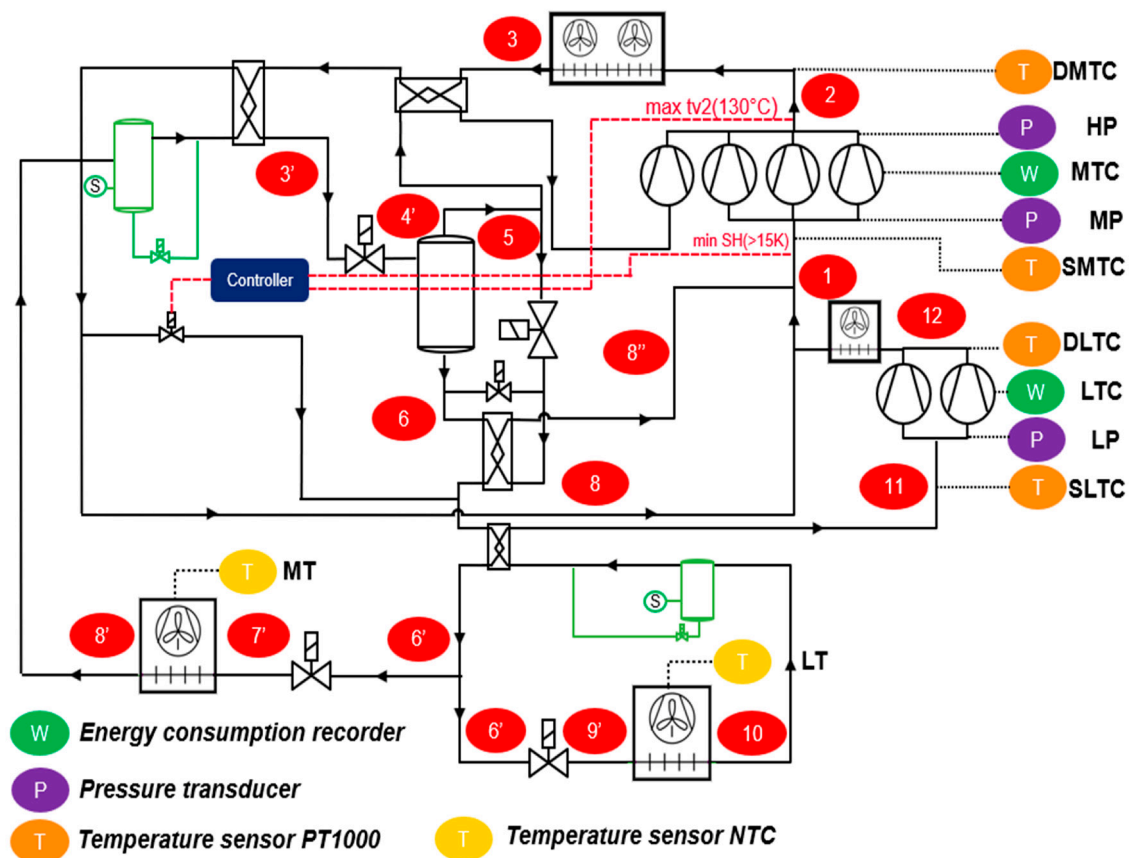


Figure 2. Operating principle of the refrigeration system with evaporators operating alternately in dry and flooded modes.

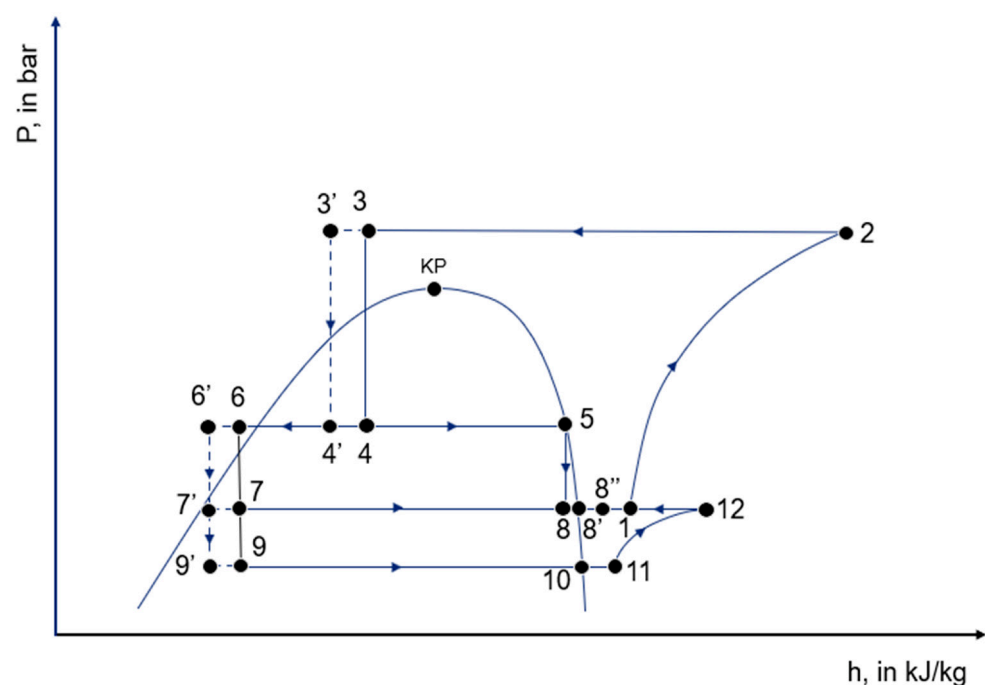


Figure 3. Log P-h diagram of the CO₂ refrigeration cycle.

The refrigeration cycle consists of the following transformations presented in Table 1.

Table 1. Transformations of the refrigeration cycle.

Transformation Stage	Description of the Transformation
1-2	Compression of superheated CO ₂ gas from medium pressure to high pressure in the MT compressors.
2-3	Subcooling of CO ₂ gas in GC by the surrounding air. Since the cycle is above the transcritical point of CO ₂ , no condensation takes place.
3-3'	Additional cooling of CO ₂ gas during circulation through the two heat exchangers, HT and ECO.
3'-4'	Expansion of CO ₂ gas passing through the HP control valve to the receiver.
4'-5	Separation of CO ₂ gas into the receiver
5-8	Expansion of CO ₂ gas MP control valve, where the pressure is reduced to the medium value.
8-8''	Heating of the biphasic CO ₂ mixture passing through the MT heat exchanger.
6-6'	Subcooling of the liquid CO ₂ refrigerant in the medium- and low-temperature heat exchangers.
6'-7'	Expansion of liquid CO ₂ refrigerant to the medium-temperature evaporators.
7'-8'	Evaporation of liquid CO ₂ refrigerant extracting heat from the medium-temperature consumers (refrigerated display cabinets and cold rooms), which results in cooling of the products.
6'-9'	Expansion of liquid CO ₂ refrigerant to the lower-temperature evaporators.
9'-10	Evaporation of biphasic CO ₂ refrigerant extracting heat from the low-temperature consumers (freezing cabinets and freezing cold rooms).
8'-1	Superheating of CO ₂ gas (separated in the MT liquid separator) in the MT heat exchanger. The MT liquid separator and the MT heat exchanger on the suction line of the compressors allows the refrigeration evaporators to operate in alternating dry/flooded mode.
10-11	Superheating of CO ₂ gas (separated in the LT liquid separator) in the LT heat exchanger. The LT liquid separator and the LT heat exchanger on the suction line of the compressors allows the freezing evaporators to operate in alternating dry/flooded mode.
11-12	Compression of the superheated CO ₂ gas from LP to MP.
12-1	Desuperheating of the CO ₂ refrigerant gas between compressions.

In usual operations, the evaporators work with dry expansion (at the outlet of the evaporator, the refrigerant is completely evaporated, so only gas is left). The gas temperature is 6K above the evaporation temperature and therefore completely superheated. But when the refrigerant in a gaseous state is superheated, the performance of the evaporator is reduced in this part of the coil. To improve the performance of the coil, the superheat is reduced by 3K at the refrigeration consumers and by 2K at the freezing consumers during 50% of the operation, regulated by the liquid level sensors. The improved performance is used to increase the evaporation temperature of the entire MT circuit by 3K, and the entire LT circuit by 2K, thus benefiting from a reduction of energy consumption at the compressor rack by 3% for every 1K gained.

Figure 4a,b presents the principles of operation of evaporators in the dry system and in the flooding system, for MT evaporators.

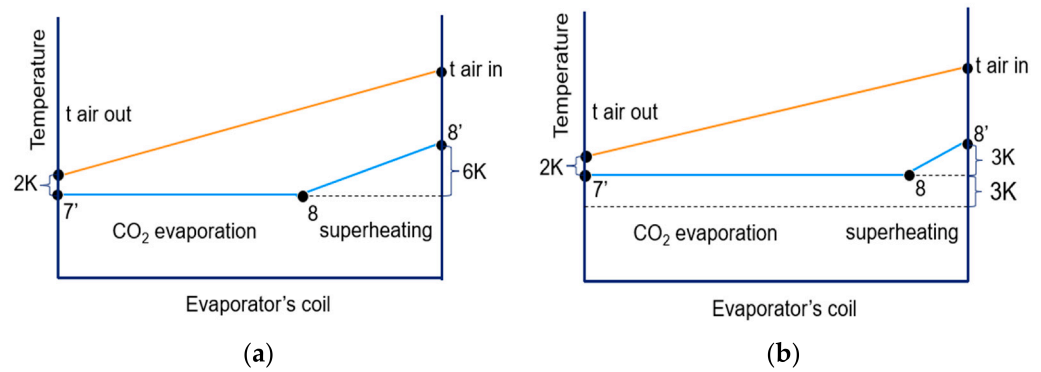


Figure 4. (a) Dry evaporation MT. (b) Flooding evaporation MT.

The compressor rack consists of seven semi-hermetic piston compressors, all manufactured by Bitzer. Table 2 shows how the compressors were arranged according to the refrigeration system configuration.

Table 2. Compressor rack configuration.

Circuit	Model	No. of Compressors	Inverter
LT ₁	2DSL-5KB	1	Schneider
LT ₂	2DSL-5KB	2	Without
MT ₁	4FTU-20LKC	1	Schneider
MT ₂	4GTE-30KC	2	Without
ECO	4JTEU-15KLC	1	Schneider

Table 3 shows the characteristics of the GC in both the transcritical (above critical point of 31 °C) and subcritical regimes, on both the air and refrigerant (CO₂) sides. The values are those from the manufacturer's data sheet (Guntner).

Table 3. GC parameters.

Transcritical Operation			
$\dot{Q}$ [kW]	231	$T_{in\ CO_2}$ [°C]	115
$A_{reserve}$ [%]	34.4	$T_{out\ CO_2}$ [°C]	41
$\dot{V}_{air}$ [m ³ /h]	49.736	Δp_{CO_2} [Pa]	1.39
v_{air} [m/s]	1.4	$\dot{V}_{CO_2}$ [m ³ /h]	24
$T_{air\ in}$ [°C]	39	$\dot{m}_{CO_2}$ [kg/s]	4065
Subcritical Operation			
$\dot{Q}$ [kW]	162	$HG\ T_{in}$ [°C]	56
$A_{reserve}$ [%]	4.7	T_{cond} [°C]	14
$\dot{V}_{air}$ [m ³ /h]	49.736	$t_{out\ CO_2}$ [°C]	12.1
v_{air} [m/s]	1.4	Δp_{CO_2} [Pa]	1.06
$T_{air\ in}$ [°C]	5	$\dot{V}_{CO_2}$ [m ³ /h]	16.68
		$\dot{m}_{CO_2}$ [kg/s]	2257

Table 4 shows the characteristics of the desuperheater, on both the air and refrigerant sides. The values are those from the manufacturer's data sheet (Guntner).

Table 4. Desuperheater parameters.

Low Temp	Compressors	Discharge	
$\dot{Q}$ [kW]	3.5	$t_{in\ CO2}$ [°C]	54
$A_{reserve}$ [%]	98	$t_{out\ CO2}$ [°C]	41
$\dot{V}_{air}$ [m ³ /h]	6905	Δp_{CO2} [Pa]	0.25
v_{air} [m/s]	1.2	$\dot{V}_{CO2}$ [m ³ /h]	17.65
$T_{air\ in}$ [°C]	39	$\dot{m}_{CO2}$ [kg/s]	906

2.2. Experimental Procedure

The data acquisition system was conveniently configured using a PC with Eckermann software. The temperature, pressure, and energy consumption measuring devices were made with industrial-type sensors installed to continuously monitor the operation of the system. The tolerance of these sensors is acceptable for industrial systems, with a maximum $\pm 1\%$. Table 5 lists the measuring equipment and its accuracy.

Table 5. List of measuring equipment and its accuracy.

Type of Sensors	Manufacturer and Model	Accuracy
Temperature sensors	Eckelmann NTC type −50 °C...+50 °C	$\pm 0.2\%$ of reading
Temperature sensors	Eckelmann PT1000 type −50 °C...+200 °C	$\pm 0.3\%$ of reading
Pressure transducers LP	Huba Control (4–20) mA; (1–26) bar	$\pm 0.35\%$ of control set
Pressure transducers MP	Huba Control (4–20) mA; (1–59) bar	$\pm 0.35\%$ of control set
Pressure transducers HP	Huba Control (4–20) mA; (1–161) bar	$\pm 0.35\%$ of control set
Energy consumption recorders	Schneider iEM3255	$\pm 1\%$ of reading

The system operation was monitored for one year for the thermodynamics conditions described below. To model the thermodynamic properties of the CO₂ agent and air, a Microsoft Excel spreadsheet program, as an interface with the REFPROP version 9.1 library [32], was used.

The electrical power of each compressor is determined by correlating total consumption data with the theoretical performance of each unit. The power absorbed by each compressor is calculated using a third degree polynomial equation, based on the saturation temperature at the suction side and the pressure and saturation temperature at the discharge of each compressor [33]:

$$\dot{W}_{compr,50Hz,10K} = C_1 + C_2 \cdot A + C_3 \cdot B + C_4 \cdot A^2 + C_5 \cdot A \cdot B + C_6 \cdot B^2 + C_7 \cdot A^3 + C_8 \cdot A^2 \cdot B + C_9 \cdot A \cdot B^2 + C_{10} \cdot B^3 \quad (1)$$

$$\dot{m}_{CO2,50Hz,10K} = C_1 + C_2 \cdot A + C_3 \cdot B + C_4 \cdot A^2 + C_5 \cdot A \cdot B + C_6 \cdot B^2 + C_7 \cdot A^3 + C_8 \cdot A^2 \cdot B + C_9 \cdot A \cdot B^2 + C_{10} \cdot B^3 \quad (2)$$

where C_1 to C_{10} are coefficients determined experimentally (Table 6); A is the saturation temperature [°C] at the suction side of the compressor, and B is the pressure [bar] at the discharge for MT compressors and the saturation temperature [°C] at the discharge for LT compressors, respectively. The values for the polynomial expressions, both for the power absorbed and mass flow, have been cross-checked using the compressor manufacturer's software [34].

Table 6. Coefficient used in Equations (1) and (2).

	2DSL	-5KB	4FTU	-20LKC	4GTE	-30KC	4JTEU	-15KLC
	$\dot{W}$ [W]	$\dot{m}$ [kg/h]	$\dot{W}$ [W]	$\dot{m}$ [kg/h]	$\dot{W}$ [W]	$\dot{m}$ [kg/h]	$\dot{W}$ [W]	$\dot{m}$ [kg/h]
C ₁	5.84×10^2	7.98×10^2	-1.70×10^4	1.52×10^3	-1.13×10^4	1.41×10^3	-9.12×10^3	8.05×10^2
C ₂	-2.81×10^2	2.44×10	-7.72×10^2	4.11×10	-5.52×10^2	4.46×10	-4.14×10^2	2.17×10
C ₃	2.95×10^2	-3.62	6.98×10^2	-4.89×10^{-1}	5.09×10^2	-2.99	374×10^2	-2.58×10^{-1}
C ₄	-5.95	2.60×10^{-1}	-1.06×10	613×10^{-1}	-6.93	6.35×10^{-1}	-5.69	3.23×10^{-1}
C ₅	7.29	-4.99×10^{-2}	1.08×10	1.31×10^{-1}	7.40	-2.64×10^{-2}	5.81	6.91×10^{-2}
C ₆	-8.77×10^{-1}	-3.12×10^{-2}	-4.05	-3.28×10^{-2}	-2.55	-6.06×10^{-3}	-2.17	-1.73×10^{-2}
C ₇	-3.18×10^{-2}	9.96×10^{-4}	-5.98×10^{-2}	7.13×10^{-3}	-3.05×10^{-2}	4.60×10^{-3}	-3.20×10^{-2}	3.76×10^{-3}
C ₈	4.76×10^{-2}	-3.14×10^{-5}	384×10^{-2}	7.29×10^{-4}	1.61×10^{-2}	341×10^{-5}	2.06×10^{-2}	3.84×10^{-4}
C ₉	-1.61×10^{-2}	-7.19×10^{-4}	-2.23×10^{-2}	-8.15×10^{-4}	-1.22×10^{-2}	-1.77×10^{-4}	-1.19×10^{-2}	-4.30×10^{-4}
C ₁₀	-1.18×10^{-3}	1.91×10^{-4}	9.88×10^{-3}	1.08×10^{-4}	5.92×10^{-3}	3.90×10^{-5}	5.30×10^{-3}	5.72×10^{-5}

The GC outlet temperature is calculated via the approach temperature method since it indicates how closely the heat exchanger can perform compared to its limit, and the high pressure (HP) is optimized for every temperature of the air entering the GC. Correlations for the calculation of the GC outlet temperature and the system HP are given in Table 7 [35]. The system operates in subcritical conditions at ambient temperatures lower than 30 °C, while it operates in transcritical conditions at ambient temperatures equal to or higher than 30 °C. The subcooling degree of the condenser varies at ambient temperatures between 14 and 30 °C. The GC outlet temperature was taken as 2 °C higher than the ambient temperature at transcritical operation [36].

Table 7. Correlations for the calculation of the GC outlet temperature and the system's high pressure for the CO₂ cases.

Air GC Inlet Temperature Range	GC Outlet Temperature $T_{gc,out}$ [°C]	GC Outlet Pressure HP [bar]
$T_{gc,air,in} \leq 5$ °C	12.1 °C	Saturated pressure of 14 °C
5 °C < $T_{gc,air,in} \leq 14$ °C	$T_{gc,air,in} + 7.1$ °C	Saturated pressure of $T_{cond} = T_{gc,air,in} + 9$ °C
14 °C < $T_{gc,air,in} \leq 30$ °C	$0.7692 \cdot T_{gc,air,in} + 9.23$	$1.397 \cdot T_{gc,out} + 32.09$
$T_{gc,air,in} > 30$ °C	$T_{gc,air,in} + 2$ °C	Optimized

When the ambient temperature diverges from the designing point, the air temperature and relative humidity inside the market also change; what causes variation of refrigeration load from the design load is determined by the market refrigeration schedule. The following equation is used to quantify the deviation [37]:

$$Load_{factor} = 1 - (1 - L_{min}) \frac{85 - T_{amb}}{85 - 40} \quad (3)$$

where T_{amb} is the ambient dry-bulb temperature (°C), and L_{min} is the minimum designed load (0.66 for medium temperature and 0.8 for low temperature) [37].

The cooling loads are at minimum values at ambient temperatures lower than 18 °C, and at design values when ambient temperatures equal to or higher than 25 °C. Load fractions were considered for the evaporators as shown in Table 8. By contrast to the method of degree day, which uses the daily mean temperature, the bin method uses bin weather data as the source of temperature data for energy analysis with the hypothesis that the exterior loads are a linear function of the ambient temperature [38]. In this study, the meteorological data available in the online catalogue on the website of the National Meteorological Administration of Romania were used [39]. The dataset was reduced to 8760 parameters using the five-year average temperatures of each hour. The number of hours in each temperature interval was calculated from −20 to +40 °C. Table 9 shows the values of the number of operating hours, as a function of outdoor temperature variation.

Table 8. Load fractions were considered for evaporators as a function of ambient temperature.

T_{amb} [°C]	$\dot{Q}_{MT}$ [kW]	$\dot{Q}_{LT}$ [kW]
$T_{amb} < 18$ °C	$0.66 \cdot \dot{Q}_{MT,design}$	$0.80 \cdot \dot{Q}_{LT,design}$
18 °C $< T_{amb} \leq 25$ °C	$\left[1 - (1 - 0.66) \frac{25 - T_{amb}}{25 - 18}\right] \dot{Q}_{MT,design}$	$\left[1 - (1 - 0.80) \frac{25 - T_{amb}}{25 - 18}\right] \dot{Q}_{LT,design}$
$T_{amb} > 25$ °C	$\dot{Q}_{MT,design}$	$\dot{Q}_{LT,design}$

Table 9. Number of operating hours.

Exterior Temp.	Operating Hours
+40 °C	0
+35 °C	77
+30 °C	399
+25 °C	1018
+20 °C	1425
+15 °C	1420
+10 °C	1461
+5 °C	1272
0 °C	1077
−5 °C	471
−10 °C	126
−15 °C	14
−20 °C	0

3. Energy and Exergy Evaluation

To determine the performance of the refrigeration system several assumptions are made:

- Pressure drops are neglected, both in the piping network, as well as, in the GC and evaporators.
- Throttling processes are considered isenthalpic in the expansion valves.
- At the exit of the evaporators, the superheat is 6K both for MT and LT consumers in 50% of the operating time; while in the other 50%, the superheat is 3K for MT consumers and 4K for LT consumers.
- A total of 100% separation of the biphasic mixture in the MP receiver.
- The outlet temperature of the CO₂ refrigerant is 2K higher compared with the ambient temperature, for both the GC and desuperheater.
- Kinetic energy inside the path flow of the refrigerant is neglected.

3.1. Energy Model

The energy model of the CO₂ transcritical refrigeration installation consists of the equations presented below [40]:

$$\dot{W}_{CM\ LT} = \dot{m}_{LT}(h_{12} - h_{11}) \quad (4)$$

$$\dot{W}_{CM\ MT} = \dot{m}_{MT}(h_2 - h_1) \quad (5)$$

$$\dot{W}_{EM\ LT} = \dot{W}_{CM\ LT} / \eta_{EM\ LT} \quad (6)$$

$$\dot{W}_{EM\ MT} = \dot{W}_{CM\ MT} / \eta_{EM\ MT} \quad (7)$$

$$\dot{Q}_{GC} = \dot{m}_t(h_2 - h_3) \quad (8)$$

$$\dot{m}_t = \dot{m}_{LT} + \dot{m}_{MT} \quad (9)$$

$$h_6 = h_7 \quad (10)$$

$$h_6 = h_9 \quad (11)$$

$$\dot{Q}_{EV_LT} = \dot{m}(h_9 - h_{10}) \quad (12)$$

$$\dot{Q}_{EV_MT} = \dot{m}(h_7 - h_8) \quad (13)$$

$$\dot{Q}_{12.1+12.2} = \dot{m}_t(h_3 - h_{3'}) \quad (14)$$

$$h_{6'} = h_7 \quad (15)$$

$$h_{6'} = h_9 \quad (16)$$

$$\dot{Q}_{EV_LT} = \dot{m}(h_{9'} - h_{10}) \quad (17)$$

$$\dot{Q}_{EV_MT} = \dot{m}(h_{7'} - h_{8'}) \quad (18)$$

The CO₂ refrigeration plant can be thermodynamically optimized by minimizing the power supplied to the compressor ($\dot{W}_{CM} \rightarrow \min$) and by keeping $\dot{Q}_{EV} = \text{const}$, i.e., by maximizing the COP value [40].

$$COP = \frac{\dot{Q}_{EV}}{\dot{W}_{CM}} \quad (19)$$

Table 10 shows the evaporation temperatures at the consumers requiring the lowest value, i.e., cold rooms and display cabinets storing/exposing fish and meat products on the refrigeration circuit, and cold rooms and display cabinets storing/exposing ice cream on the freezing circuit. These evaporation temperatures determine both the sizing of the refrigeration system and the flooded operation of the evaporators, which allows the evaporation temperature of the whole system to be increased and the COP to be improved.

Table 10. Interior and evaporation temperatures depending on the exposed products.

Products Exposed	Interior Temperature Air	Evaporation Temperature of CO ₂
Fish cabinets	−1/+1 °C	−5 °C
Meat cabinets	−1/+1 °C	−5 °C
Fish cold rooms	−1/+1 °C	−5 °C
Meat cold rooms	−1/+1 °C	−5 °C
Ice cream cabinets	−22/−24 °C	−31 °C
Ice cream cold rooms	−22/−24 °C	−31 °C

The COP in an alternating dry/flooded operation for both MT and LT circuits can be written as:

$$COP_{alt_tot} = COP_{alt_MT} + COP_{alt_LT} \quad (20)$$

Assuming 50% operation in a dry system with 6K superheat and 50% operation in a flooded system with 3K superheat on a refrigeration circuit, and 4K superheat on a freezing circuit, the alternative COP calculation becomes:

$$COP_{alt_MT} = \frac{(COP_{dryMT} + COP_{floodedMT})}{2} \quad (21)$$

where

$$COP_{dryMT} = \frac{\dot{Q}_{EV-MT}}{\dot{W}_{CM-6^\circ C}} \quad (22)$$

$$COP_{floodedMT} = \frac{\dot{Q}_{EV-MT}}{\dot{W}_{CM-3^\circ C}} \quad (23)$$

$$COP_{alt_LT} = \frac{(COP_{dryLT} + COP_{floodedLT})}{2} \quad (24)$$

where

$$COP_{dryLT} = \frac{\dot{Q}_{EV-LT}}{\dot{W}_{CM-32^{\circ}C}} \quad (25)$$

$$COP_{floodedLT} = \frac{\dot{Q}_{EV-LT}}{\dot{W}_{CM-30^{\circ}C}} \quad (26)$$

3.2. Exergy Model

The exergy of a thermodynamic system is the highest theoretical useful work (electrical work or shaft work) achievable, as the system is placed into complete thermodynamic equilibrium with the thermodynamic environment, when the system interacts with this environment only. The total exergy of a system carries physical exergy, chemical exergy, kinetic exergy, and potential exergy. The physical exergy carries thermal exergy (associated with the system temperature) and mechanical exergy (associated with the system pressure). For a particular thermodynamic state, at a temperature T and pressure p , the thermal exergy can be calculated along the isobaric line at p (from state $[T, p]$ to state $[T_0, p]$), while the mechanical exergy can be calculated along the isothermal line at T_0 (from state $[T_0, p]$ to state $[T_0, p_0]$) [41].

$$E_{sys} = E_{sys}^{PH} + E^{KN} + E^{PT} + E^{CH} \quad (27)$$

$$e_{sys} = e_{sys}^{PH} + e^{KN} + e^{PT} + e^{CH} \quad (27a)$$

All chemical, kinetic, and potential exergy has been neglected.

$$E_{sys}^{PH} = (U - U_0) + p_0(V - V_0) - T_0(S - S_0) \quad (28)$$

$$e_{sys}^{PH} = (u - u_0) + p_0(v - v_0) - T_0(s - s_0) \quad (28a)$$

The exergy analysis is established on exergy balances formulated for the overall system [40],

$$\dot{E}_{F,tot} = \dot{E}_{P,tot} + \sum_k \dot{E}_{D,k} + \dot{E}_{L,tot} \quad (29)$$

and for each k -th component can be resolved through an exergy balance for this component:

$$\dot{E}_{F,k} = \dot{E}_{P,k} + \dot{E}_{L,k} \quad (30)$$

For the simple CO₂ transcritical refrigeration plant, we have the following [42]:

$$\dot{E}_{F,tot} = \dot{W}_{EM} \quad (31)$$

$$\dot{E}_{P,tot} = \dot{E}_{P,EV} \quad (32)$$

$$\dot{E}_{L,tot} = \dot{E}_{P,GC} \quad (33)$$

$$\dot{W}_{EM} = \dot{E}_{P,EV} + \sum_K \dot{E}_{D,k} + \dot{E}_{P,GC} \quad (34)$$

For the analysed installation, we have:

$$\dot{W}_{EM,alt} = \dot{W}_{EM_MT,alt} + \dot{W}_{EM_LT,alt} \quad (35)$$

$$\dot{W}_{EM_MT,alt} = \dot{W}_{EM,MT,dry-6^{\circ}C} \times 50\% + \dot{W}_{EM,MT,flooded-3^{\circ}C} \times 50\% \quad (36)$$

$$\dot{W}_{EM_LT,alt} = \dot{W}_{EM,LT,dry-32^{\circ}C} \times 50\% + \dot{W}_{EM,LT,flooded-30^{\circ}C} \times 50\% \quad (37)$$

Splitting the exergy destruction of the refrigeration installation leads to the following equations [43]:

$$\dot{E}_{D,CM,LT} = \dot{W}_{CM,LT} - (\dot{E}_{12} - \dot{E}_{11}) \quad (38)$$

$$\dot{E}_{D,CM,MT} = \dot{W}_{CM,MT} - (\dot{E}_2 - \dot{E}_1) \quad (39)$$

$$\dot{E}_{D,GC} = (\dot{E}_2 - \dot{E}_3) - (\dot{E}_{air,out} - \dot{E}_{air,in}) \quad (40)$$

$$\dot{E}_{D,EV,LT,alt} = (\dot{E}_{10} - \dot{E}_{9'}) - (\dot{E}_{air,out} - \dot{E}_{air,in}) \quad (41)$$

$$\dot{E}_{D,EV,MT,alt} = (\dot{E}_{8'} - \dot{E}_{7'}) - (\dot{E}_{air,out} - \dot{E}_{air,in}) \quad (42)$$

$$\dot{E}_{D,TV,LT,alt} = (\dot{E}_{6'}^M - \dot{E}_{9'}^M) - (\dot{E}_{9'}^T - \dot{E}_{6'}^T) \quad (43)$$

$$\dot{E}_{D,TV,MT,alt} = (\dot{E}_{6'}^M - \dot{E}_{7'}^M) - (\dot{E}_{7'}^T - \dot{E}_{6'}^T) \quad (44)$$

To examine the exergy for each component, Equations (47)–(50) are used. Air inlet and exit temperatures were presumed to be T_{amb} and $T_{amb} + 5^\circ\text{C}$, respectively [36] and the source temperature was taken as $T_{amb} + 275.5\text{ K}$. The MT evaporator's air inlet and exit temperatures were presumed to be 2 and 0°C , respectively, and its source temperature was taken as 274 K, while the LT evaporator's air inlet and exit temperatures were presumed to be -24°C and -25°C , respectively, and its source temperature was taken as 250 K [44].

Exergy rate of the mass flow is:

$$\dot{E}_{mass} = \dot{m} [h - h_0 - T_0(s - s_0)] \quad (45)$$

Exergy transfer rates from the GC and evaporators:

$$\dot{E}_{GC} = \dot{Q}_{GC} \left(1 - \frac{T_0}{T_{H1,GC}} \right) \quad (46)$$

$$\dot{E}_{MT,alt} = \dot{Q}_{MT} \left(\frac{T_0}{T_{L1,MT,alt}} - 1 \right) \quad (47)$$

$$\dot{E}_{LT,alt} = \dot{Q}_{LT} \left(\frac{T_0}{T_{L1,LT,alt}} - 1 \right) \quad (48)$$

The exergy efficiency of the entire system is [40]:

$$\varepsilon_{tot} = \frac{\dot{E}_{MT,tot,alt} + \dot{E}_{LT,tot,alt}}{\dot{W}_{tot,alt}} \quad (49)$$

4. Economic Impact Evaluation

To evaluate the economic impact of a CO_2 commercial refrigeration system with evaporators operating alternately in dry and flooded modes, the energy saving should be calculated using the following:

$$ES = \frac{\Delta E}{E_{DX}} \times 100 \quad (50)$$

$$\Delta E = E_{DX} - E_{alt} \quad (51)$$

where the annual energy consumption for each system is

$$E_{DX} = P_{DX} \cdot t_{DX} \quad (52)$$

$$E_{alt} = P_{dry} \cdot t_{dry} + P_{flood} \cdot t_{flood} \quad (53)$$

P_{DX} = power of the dry system, kW;

P_{dry} = power of the alternating system operating in dry mode, kW;

P_{flood} = power of the alternating system operating in flood mode, kW;

t_{DX} = operating time of the dry system, h/year;

t_{dry} = operating time of the alternating system in dry mode, h/year;

t_{flood} = operating time of the hybrid system in flood mode, h/year;

with

$$t_{dry} + t_{flood} = 8760 \text{ h} \quad (54)$$

$$t_{dry} = t_{flood} \quad (55)$$

The annual operating cost is:

$$C_{energy} = E_{annual} \cdot C_{el} \quad (56)$$

where C_{el} is the electricity price, EUR/kWh.

The annual cost saving is:

$$\Delta C_{op} = \Delta E \cdot C_{el} \quad (57)$$

The baseline capital cost of dry-expansion CO₂ evaporators was derived from standard supermarket refrigeration installations [45–47]. Flooded and overflooded evaporator configurations require additional components such as liquid separators, enhanced heat exchanger surfaces, and control systems, leading to a 25–50% increase in evaporator-side CAPEX, consistent with the reported supermarket case studies and DOE analyses [48,49].

$$\Delta CAPEX = CAPEX_{alt} - CAPEX_{DX} \quad (58)$$

The Simple Payback Period (SPP) is:

$$SPP = \frac{\Delta CAPEX}{\Delta C_{op}} \quad (59)$$

5. Environmental Impact Calculation

The indirect CO₂ emissions due to electricity use can be calculated as follows:

$$CO_{2,indirect} = E_{annual} \cdot EF_{grid} \quad (60)$$

where EF_{grid} is the emission factor (kg CO₂/kWh). The average CO₂ emission factor for electricity supplied from the grid in the EU is approximately 0.35 kg CO₂/kWh [50].

The emission reduction is:

$$\Delta CO_2 = \Delta E \cdot EF_{grid} \quad (61)$$

6. Results and Discussion

Figure 5a,b shows the cooling capacity variation of the system in dry operation as a function of ambient temperature without and with the heat taken from the suction pipe, respectively. Figure 6 shows the variation of the cooling capacity of the system with the heat taken from the suction pipe in alternating operation as a function of the ambient temperature. Figure 7a,b shows the variation of the COP for dry and alternating system operations, respectively.

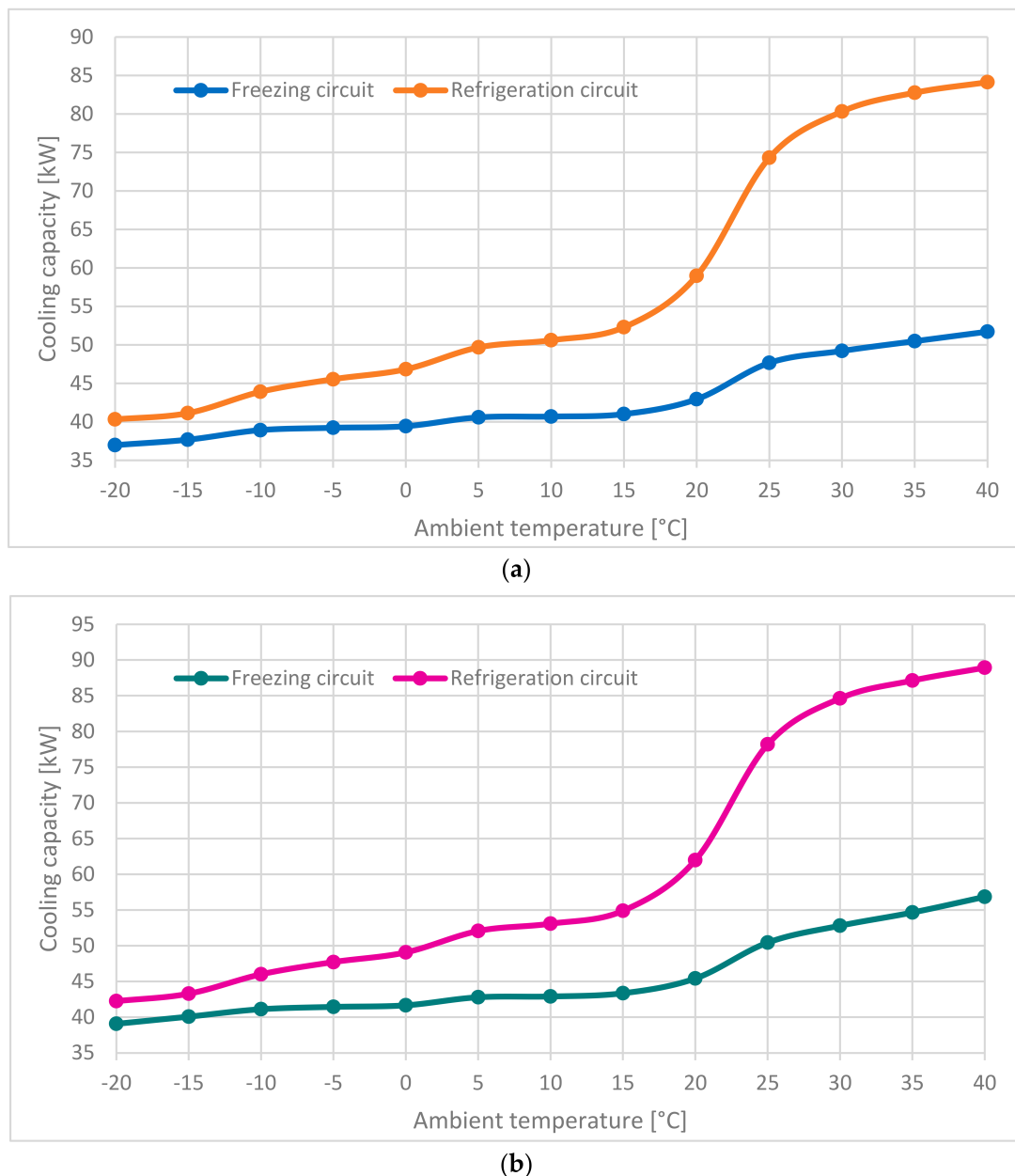


Figure 5. (a) Cooling capacity of the system in dry operation without heat taken from the suction pipe as a function of ambient temperature. (b) Cooling capacity of the system in dry operation with heat taken from the suction pipe as a function of ambient temperature.

The cooling capacity for LT consumers (cold rooms and display cabinets) varies from 36.99 kW at a $-20\text{ }^{\circ}\text{C}$ ambient temperature to 51.72 kW at a $+40\text{ }^{\circ}\text{C}$ exterior temperature, and for MT consumers from 40.32 kW at a $-20\text{ }^{\circ}\text{C}$ ambient temperature to 84.13 kW at a $+40\text{ }^{\circ}\text{C}$ exterior temperature. A very similar variation in cooling capacity can be observed for the two circuits: refrigeration and freezing, because the influencing factors inside the store, like day/night operation and store temperature, are the same. For temperatures higher than $20\text{ }^{\circ}\text{C}$, the cooling capacity for the refrigeration circuit has a steeper increase. Below $20\text{ }^{\circ}\text{C}$, the cooling capacity for MT consumers drops from 48 to 37%, while for LT consumers, it drops from 72 to 65%. Above $20\text{ }^{\circ}\text{C}$, the cooling capacity for MT consumers increases from 54 to 74%, while for LT consumers, it increases from 76 to 87%.

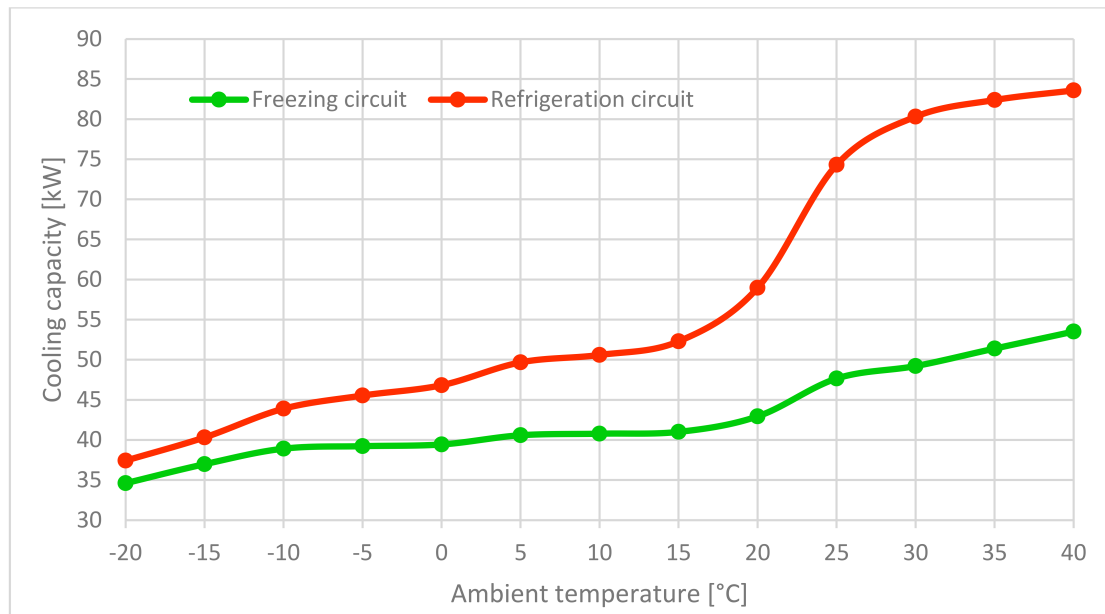


Figure 6. Cooling capacity together with heat taken from the suction pipe of the system in alternating operation. In case of alternating operation, the values for the cooling capacity considering also the extra capacity needed for the heat taken from the suction pipes are smaller for the refrigeration circuit, as well as for the freezing circuit, compared to dry operation. This is the first indication that alternating operation has an increased efficiency compared to dry operation. For LT consumers, the cooling capacity varies from 34.62 kW at a $-20\text{ }^{\circ}\text{C}$ ambient temperature to 53.51 kW at a $+40\text{ }^{\circ}\text{C}$ exterior temperature, and for MT consumers, from 37.43 kW at a $-20\text{ }^{\circ}\text{C}$ ambient temperature to 83.59 kW at a $+40\text{ }^{\circ}\text{C}$ exterior temperature.

In case it is considered for both circuits as well, the heat taken from the suction pipes and the variation of the cooling capacity is similar with the previous situation, only the values are higher corresponding to the extra capacity needed to offset the extra heat. More precisely, for LT consumers, cooling capacity fluctuates between 39.07 kW at a $-20\text{ }^{\circ}\text{C}$ ambient temperature and 56.85 kW at a $+40\text{ }^{\circ}\text{C}$ exterior temperature, and for MT consumers from 42.25 kW at an ambient temperature to 88.93 kW at a $+40\text{ }^{\circ}\text{C}$ exterior temperature.

In dry operation mode, the variation of the COP of the freezing plant is smooth as a function of ambient temperature; while for the refrigeration installation, it is steep from the $5\text{ }^{\circ}\text{C}$ outside temperature, from when the MT compressors start to consume more electricity for the needed cooling capacity. This happens due the combination of the compressors for this plant, described in Table 1, having a smaller compressor for step 1 when working with the inverter, and two bigger compressors working in ON/OFF mode.

In alternating operation, the variation of the COP is also smooth in the case of a freezing compressor rack, with slightly smaller values compared to the dry operation due to the fact that the condensing temperature is $3\text{ }^{\circ}\text{C}$ higher, in 50% of the time; because of the lift of the evaporating temperature for the MT compressors, even if the evaporating temperature is increased by $2\text{ }^{\circ}\text{C}$ for the LT compressors, considerably higher values will be found in the case of a refrigeration compressor rack. The variation of the COP is also steep, due to the same combination of the compressors.

Figure 8 indicates the exergy efficiency variation of the dry and hybrid systems.

The values for the alternating operation mode are relatively higher due to dramatically reduced exergy destruction, because of a smaller temperature difference between the air and refrigerant and reduced entropy generation in the heat transfer. Above $0\text{ }^{\circ}\text{C}$, the exergy efficiency drops because the energy consumed by the compressors to produce the needed useful work at the evaporators is higher.

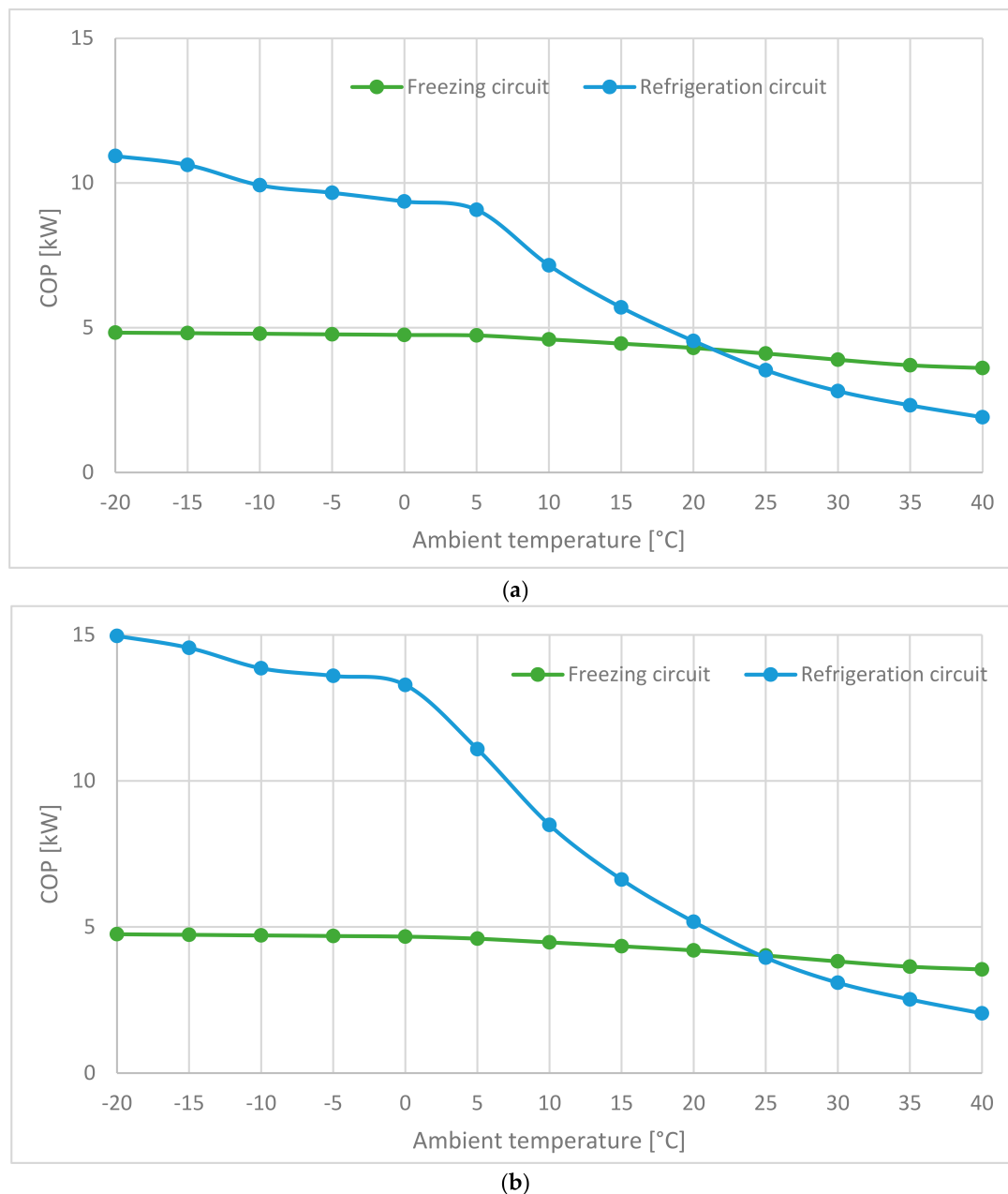


Figure 7. (a) COP of system in dry operation. (b) COP of system in alternating operation.

Figure 9a shows the annual energy consumption of the compressor rack, separately for freezing and refrigeration, in both dry and flooded operation modes and Figure 9b shows the annual energy consumption for compressor rack cumulative freezing and refrigeration.

Being a booster type refrigeration system—the LT compressors discharge into the suction line of the MT compressors—the higher condensing temperature, by 3 °C for the LT compressors, in 50% of time, due to the lift of the evaporating temperature of the MT compressors by 3 °C, is influencing the energy consumption of the LT compressor rack, which is somehow higher in the case of an alternating operation compared to the dry operation, even if the evaporating temperature is also increased by 2 °C. However, the energy consumption of the MT compressor rack is considerably smaller in alternating operation, and so, overall, the entire system consumes less energy.

Both values of energy consumption gathered for the LT and MT compressor racks indicate a 6.9% reduction if the system operates fully in an alternating flooded system instead of fully in dry operation.

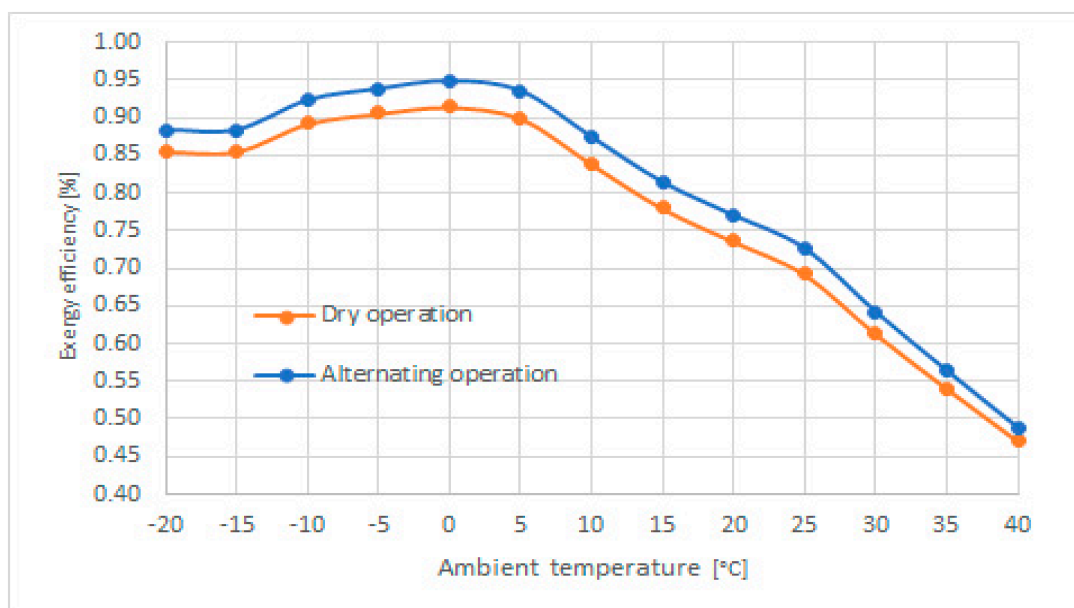
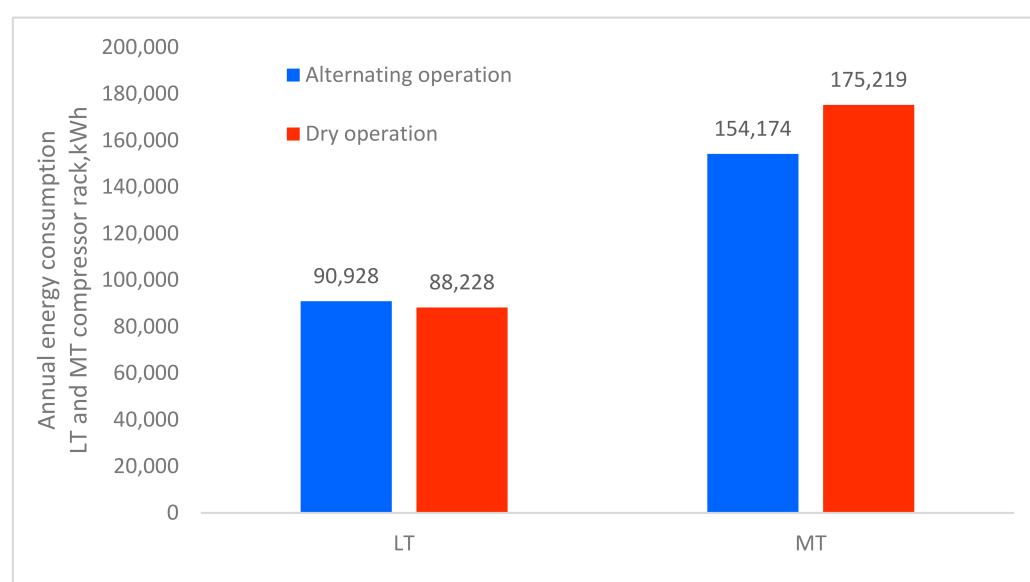


Figure 8. Exergy efficiency of system in dry and alternating operation modes.

Considering the total annual energy consumption of a dry system (1,161,091 kWh) and of an alternating operation system (1,142,746 kWh), the electricity cost of 0.2872 EUR/kWh, with a grid factor of 0.35 kg CO₂/kWh and the additional CAPEX of EUR 10,000 [48,51], resulted energy savings of 18,345 kWh/year (6.9%), cost savings of 5268.68 EUR/year, and a payback period of 1.89 years. The price of energy and the cost of the additional equipment installed, compared to the standard installation, are important for determining the exact profitability of the additional investment. In practice, an additional investment is considered acceptable if it can be recovered in less than two years.

The refrigeration system with evaporators operating in alternating dry and flooded modes achieves a CO₂ reduction of 6420.75 kg CO₂/kWh.



(a)

Figure 9. Cont.

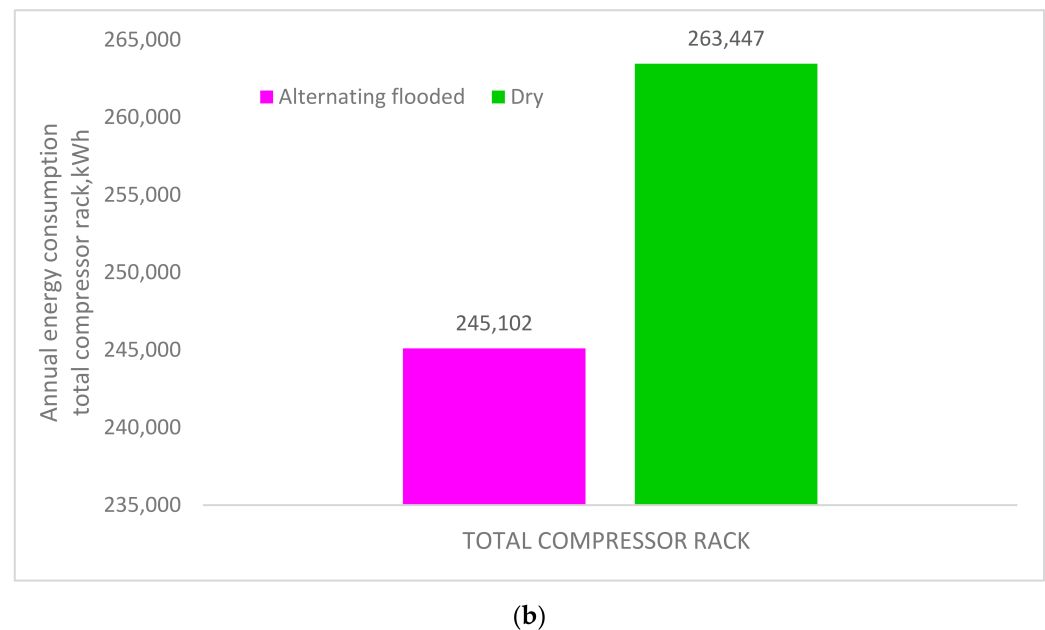


Figure 9. (a) Annual energy consumption of LT and MT compressor racks. (b) Annual energy consumption of compressor rack.

7. Conclusions

Evaporators' alternating operation of a CO₂ commercial refrigeration system was investigated to find out the impact on system performance. The evaporators operate alternatively in dry and flooding mode, with 50% for each mode from the total operation time. The analysed system operates with a CO₂ refrigerant on a transcritical cycle, without an ejector, and is equipped with a liquid separator on both the freezing and refrigeration circuits to allow the alternating operation of the evaporators. This particular adjustment of the operation is possible due to liquid sensors installed in the middle of both the LT and MT liquid separators that transmit information to the controllers to switch from flooded operation (3K superheat for MT and 4K for LT) to dry operation (6K superheat for both MT and LT) when the liquid level drops and vice versa when the level rises, exceeding the level of regulation used so far by the refrigeration industry, with only fully dry operation or with only fully flooded operation. In this way, because of eliminating the risk of liquid droplets in the compressor crankcase, it is possible to safely increase the evaporating temperature by 3K for the MT compressors and by 2K for the LT compressors compared with the previously used regulation methods, in 50% of the total operating time, and therefore reduce the energy consumption by operating at a higher evaporating temperature for the compressor racks. The operation of the installation was simulated by using REFPROP version 9.1. The pressure at the GC was optimized considering the conditions inside the store (that can influence the refrigerating capacity of the evaporators) and the weather conditions appropriate to the region where the system is located. The energy and exergy analysis of the system was carried out, and the results show that the alternating operation leads to an energy reduction of 6.9% for the combined compressor rack. This energy assessment of a CO₂ (R744) commercial refrigeration system with evaporators operating alternately in dry and flooded modes leads to control strategy improvements rather than hardware changes. Exergy efficiency is higher for alternating operation compared to dry operation because of smaller temperature difference between a CO₂ refrigerant and the air, reducing entropy generation in heat transfer processes. A payback period of 1.89 years is reasonable considering the small additional investment needed only for two liquid separators, liquid sensors, and electrical cables for connection with the controllers of the system. This special

control is possible by using the alternative settings of the controllers and so, extra costs for automatization are not needed. Also very important is the reduction of CO₂ emissions of 6420.75 kg CO₂/kWh achieved with this new operating alternating flooded mode. All these benefits make this new operating system attractive for the refrigeration industry.

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Abbreviations

The following abbreviations are used in this manuscript:

ASHRAE	American society of heating, refrigerating, and air-conditioning engineers
CAGR	Compound Annual Growth Rate
CO ₂	Carbon dioxide
COP	Coefficient of Performance
DMTCs	Discharge Medium Temperature Compressors
DLTCs	Discharge Low Temperature Compressors
ECO	Eco Compressor
EU	European Union
ERS	Ejector Refrigeration System
GC	Gas cooler
GWP	Global Warming Potential
HVAC	Heating, Ventilation, Air Conditioning
HEX	Heat Exchanger
HFC	Hydrofluorocarbon
HFO	Hydrofluoro-olefins
HG	Hot gas
HP	High Pressure
HT	High Temperature
MP	Medium Pressure
MT	Medium Temperature
MTC	Medium Temperature Compressors
LP	Low Pressure
LS	Liquid Separator
LT	Low Temperature
LTC	Low Temperature Compressors
SMTC	Suction Medium Temperature Compressors
SLTC	Suction Low Temperature Compressors

Symbols

A	Surface in m ²
$\dot{E}$	Exergy rate in W
E	Exergy kJ
e	Specific exergy in kJ/kg
h	Enthalpy in kJ/kg
$\dot{m}$	Mass flow rate in kg/s
p	Pressure in Pa
S	Entropy kJ/kg
s	Specific entropy kJ/kg·K
T	Temperature in °C
U	Internal energy in kJ
u	Specific internal energy in kJ/kg
V	Volume in m ³
v	Specific volume m ³ /kg
v	Velocity m/s
$\dot{W}$	Electrical power in W
$\dot{V}$	Volumic flow rate in m ³ /h
$\dot{Q}$	Heat flow rate in kW

Greek symbols

Δ	Difference
η	Isentropic efficiency
ε	Exergy efficiency

Subscripts

air	Air
amb	Ambient
alt	Alternative flooding
cond	Condensing
CD	Condensator
CH	Chemical
CM	Compressor
D	Exergy destruction
EM	Electrical Motor
EV	Evaporator
F	Fuel Exergy
H ₁	High Temp Source
in	Inlet
KN	Kinetic
L	Exergy loss
L ₁	Low Temp Source
M	Mechanical
out	Outlet
P	Product exergy
PH	Physical
PT	Potential
sys	System
tot	Total
T	Thermal
TV	Throttling valve

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