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Next generation of ejector-supported R744 booster systems for commercial refrigeration at all climates

Nouvelle génération de systèmes booster R744 supportés par éjecteur pour la réfrigération commerciale pour tous les climats

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A B S T R A C T

The pernicious effects of synthetic refrigerants on different environmental aspects leave natural refrigerants as the only alternative for vapour compressions systems. Among natural refrigerants, CO_2 (R744) has become the preferred choice for commercial refrigeration at almost any location and climate. However, efficient R744 refrigeration systems for warm climates have a great level of complexity, implementing technologies such as mechanical subcooling or ejector that increase the investment costs. The goal of the novel hybrid configuration presented in this work is to simplify ejector-supported R744 commercial refrigeration systems while maintaining all the benefits of the ejector implementation in transcritical operation mode. Moreover, utilization of a low-pressure accumulator layout in the subcritical mode removes all the practical challenges related to the booster layout operating at low ambient temperatures. This solution is based on: (i) MT and LT compressor suction groups, (ii) non-superheated MT evaporation with increased evaporation temperature, and (iii) ejector utilization throughout the year. The ejector is actively operated as a high-pressure-control device at elevated ambient temperatures ('summer mode'), while it is passive and acts as a check-valve at lower ambient temperatures ('winter mode'). The experimental campaign performed proved that this novel system configuration is more energy efficient than booster systems with parallel compressors at any condition, improving the COP by around 40 % at the most extreme gas cooler outlet temperatures tested, i.e., 40 °C (summer mode) and 10 °C (winter mode).

1. Introduction

International agreements such as Montreal Protocol and its amendments (UNIDO 2016) or the EU F-Gas Regulation 517/2014 (European Commission 2014) have had a crucial impact on the progressive phase-down of many traditionally used synthetic refrigerants, and on the higher implementation of low Global Warming Potential (GWP) fluids without Ozone Depletion Potential (ODP) in refrigeration and heat pumps systems. Ammonia (GWP = 0, ODP = 0) has been ubiquitous in industrial applications throughout refrigeration history, but for smaller units there is an ongoing competition between synthetic and natural refrigerants. Many manufacturers utilize unsaturated hydrofluorocarbons (HFCs) in their units, also known as hydrofluoroolefins (HFOs) for several reasons. They can be used efficiently in vapour compression systems, while being (supposedly) non-toxic, less flammable than hydrocarbons, operating at relatively low pressure if compared to CO_2, and with low GWPs reported (on a 100 year basis). However, there is an increasing concern about their potential to pollute in other ways, such as due to emissions during their manufacturing, or due to the degradations products, all this contributing to their lifecycle emissions (ecos 2021). Other argument against HFOs is that it degrades in the atmosphere into trifluoroacetic acid (TFA), which then accumulates in freshwater sources. It is still unclear if TFA is a potential hazard to humans or other organisms in the concentrations existing nowadays, but further research is needed (Frank, 2021). Consequently, it can be agreed with Ciconkov (Ciconkov, 2018) that natural refrigerants are the best long-term alternative, and there is an increasing number of OEMs

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Nomenclature

Roman symbols

c_p	spec. heat cap. J kg^{-1} K^{-1}
h	specific enthalpy J kg^{-1}
$\dot{m}$	mass flow rate kg s^{-1}
p	pressure Pa
P	power consumption W
$\dot{Q}$	cooling load W
T	temperature K, °C
$\dot{V}$	volumetric flow rate m^3 s^{-1}

Greek symbols

η	efficiency -
Φ_m	mass entrainment ratio -
ρ	density kg m^{-3}

Acronyms

BS	booster system
CapEx	capital expenditures
COP	coefficient of performance
EEV	electronic expansion valve
FGV	flash gas bypass valve
GWP	global warming potential
HFC	hydrofluorocarbon
HFO	hydrofluoroolefin
HP	high pressure
HPV	high-pressure valve
LT	low temperature
MT	medium temperature
ODP	ozone depletion potential
OpEx	operating expenses
PC	parallel compression-supported
SEB	simplified ejector-supported booster
SM	summer mode
TFA	trifluoroacetic-acid
WM	winter mode

Subscripts

comp	compressor(s)
disc	discharge ejector
ej	ejector
evap	evaporator
in	inlet
LT	low temperature
mot	motive
MT	medium temperature
out	outlet
ref	refrigerant (R744)
s	entropy
suct	suction ejector

implementing them.

Carbon dioxide (CO_2, R744) is becoming the preferred choice for commercial refrigeration and supermarket applications in Europe, and it is projected that more than 65 000 stores will utilize R744 transcritical systems in this continent by 2030 (Zolcer Skačanová and Battesti, 2019). The main reason for this success is that R744 guarantees long-term legal and environmental safety, while it is becoming a competitive refrigerant in terms of cost and efficiency of the refrigeration systems at almost any climatic condition (Gullo et al., 2018, Karampour and Sawalha, 2018). To reach an acceptable performance under warm environmental temperature conditions, the standard R744 booster system should be enhanced by one or several of the methods/technologies developed in the last decades.

- Parallel compressors are the basic improvement for units operating in warm climates. It is agreed in the literature that parallel compressors contribute to reduce the overall power consumption of the R744 booster systems (Bell, 2004, Gullo et al., 2016). However, its implementation in booster systems is justified depending on the system location and number of operating hours at relatively high ambient temperatures (Javerschek et al., 2015).
- Overfed or non-superheated evaporators as opposed to evaporators with superheating control to increase the evaporation temperature are especially beneficial due to the $\Delta p/\Delta T$ ratio of CO_2. Energy savings around 3 % per 1 K evaporation temperature increase can be achieved in any location. However, an accumulator is needed downstream of the non-superheated evaporators to protect the compressors, combined with a device, either a pump (Gullo et al., 2016) or an ejector (Minetto et al., 2014) to return the liquid from the accumulator to the liquid receiver.
- Subcooling the high-pressure stream downstream of the air-cooled gas cooler under warm or very warm ambient temperatures. This measure contributes also to increasing the specific cooling capacity of booster systems and reducing the discharge pressure, thus enhancing the system efficiency. A complete review on this topic is provided in reference (Llopis et al., 2018), which analyses studies on different alternatives such the utilization of internal heat exchangers, integrated subcooling methods, and dedicated subcooling methods. According to this review, COP improvements can be well above 10 %. Adiabatic cooling or water cooled gas coolers contribute also in the direction of subcooling the CO_2, but have limited potential in very humid climates, where there is water scarcity or hard waters (corrosion and scaling), or where flexibility is needed (Girotto, 2017).
- Ejectors have also been proposed to reduce the energy consumption of R744 booster systems. These devices substitute the high-pressure control valves and can recover part of the expansion work to precompress vapour at lower pressure and thereby reduce the work of compression by elevating the suction pressure level. COP improvements are mostly achieved at warm environmental conditions, and different percentages are reported in different publications (Elbel and Lawrence, 2016).

All these alternatives can contribute to significant improvements of the performance and energy efficiency of R744 booster systems. However, the associated increase in complexity of the system leads to higher investment costs for the system owner, which might be hindering the further spread of climate-friendly CO_2 commercial refrigeration systems to emerging economies or where smaller units are needed. Already some manufacturers of CO_2 compressor packs are looking into this direction of simplifying their designs. The novel configuration presented in this work is aimed at (i) efficient operation in a very wide range of ambient temperature conditions, including cold and hot climates or locations with large variations between seasons or within the day, and (ii) simplifying the layout of ejector-supported R744 booster systems to reduce CapEx (Capital Expenditures) and OpEx (Operating Expenses). These goals are fulfilled through this innovative system configuration that enables switching between transcritical and subcritical operation modes as a function of the ambient temperature with a reduced number of system components. This article describes the concept, and then features the experimental methodology utilized to analyse its feasibility and benchmark to a conventional R744 booster system and a parallel compressor-supported booster system. The results determined from the measured data are then discussed and conclusions are drawn on the potential of this system layout in the future.

2. Optimized ejector-supported R744 booster system

The system concept suggested in this article is an adaptation of an ejector-supported booster system, focusing on simplifying state-of-the-art solutions, which accommodate ejectors, parallel compression, overfed evaporation, etc., while keeping the efficiency enhancement inherent to these technologies. For simplicity, from now on it will be referred as Simplified Ejector-supported Booster (SEB). Fig. 1 shows a diagram of proposed system layout, which main characteristic is that it consists of only two compressor groups with ejector implementation, which is rather unusual.

The SEB is conceived for multitemperature centralized units, with medium temperature (MT) and low temperature (LT) refrigeration loads. Each of the refrigeration temperature levels has a compressor group associated, i.e., MT compressors and LT compressors, respectively. The LT compressors operate as usual in R744 booster systems, while the MT compressors compress the combined flow from the liquid receiver and the discharge stream from the LT compressors. In the proposed configuration, the ejector is always part of the system, either active or passive:

- Active ejector. Under moderate to elevated ambient temperatures (summer mode, Fig. 1 left), the system operates as a classic ejector cycle, with the ejector acting as the high-pressure control device according to a control algorithm for optimal high-side pressure depending on the outlet temperature of the gas cooler. In this case, the ejector lifts the entire CO_2 stream from the MT evaporators to the pressure level of the liquid receiver. The achievable pressure lift by the ejector depends on the motive/primary stream (gas cooler outlet) conditions and needs to be at a sufficient level to secure the supply of liquid refrigerant from the liquid receiver to the individual expansion valves upstream of the MT evaporators. As a rule of thumb, the authors estimate that a pressure lift close to 3 bar would be enough to operate in summer mode, but in the end it will depend on the installation (pressure drop in piping and evaporators) and on the expansion valves utilized.
- Passive ejector. At lower ambient temperatures (winter mode, Fig. 1 left), the SEB is a subcritical CO_2 system with a low-pressure receiver (suction accumulator). The ejector functions as a passage device between the MT evaporators (suction port) and the liquid receiver (discharge port), while the motive side of the ejector is closed. A separate check valve upstream of the ejector suction port may be needed in case the ejector is not equipped with an internal check valve. The liquid refrigerant is supplied to the individual MT and LT evaporators directly from the gas cooler outlet (in condenser mode). The high-pressure is then a consequence of the operation of the expansion valves upstream of the evaporators and the gas cooler / condenser outlet conditions. The ejector motive stream would only be enabled as a safety function and during a limited period of time to compensate for any drastic increase of the high pressure with rapidly changing refrigeration load conditions.

Independently of the mode, the liquid receiver is always in between the MT evaporators and the MT compressors (with or without ejector pressure lift), involving that the SEB layout is especially suited for non-superheated MT evaporation and elevated evaporation temperature without risk of liquid hammering of the MT compressors, without the necessity to install an additional separation vessel upstream of the compressors.

There are several requirements that an ejector should fulfil to be suited for this system layout, as listed below:

- High pressure adjustment at different conditions. This could be achieved either with variable-geometry ejectors (modulating control), or with several fixed-geometry ejectors arranged in parallel (stepwise control).
- Load adaptation. The ejector should be able to operate with a reasonable efficiency for different cooling load demands.
- Entrainment ratio. Since all the vapor from the MT evaporators must be entrained by the ejector, the ejector for such a system layout must be designed for relatively high entrainment ratio (ratio of suction mass flow rate to motive mass flow rate).
- Check valves. It is an advantage if the ejector has internal check valves included in its design, avoiding external check valves. Either way, the pressure drop due to these check valves should be low when the ejector operates passive (winter mode).
- Internal pressure drop. Linked to the previous bullet point, the pressure drop through the passage from the suction port to the discharge port should be minimal in winter mode, when there is no driving force from the stream flowing through the motive nozzle.

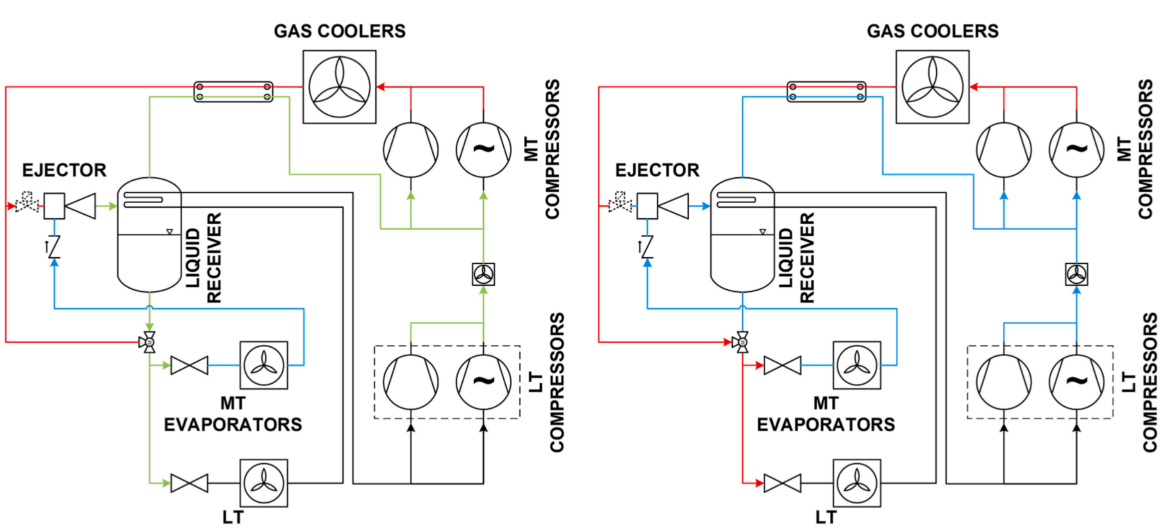

Fig. 1. Simplified ejector-supported R744 booster system. Left, operation in summer mode. Right, operation in winter mode. Pressure levels according to colours: red- high pressure, green – intermediate pressure, blue – MT pressure, black – LT pressure.

3. Methods

This section describes the experimental methodology (experimental setup, test conditions and data analysis and validation) followed in this study to compare the novel configuration of R744 booster system with other well-established system layouts. In subsequent sections, the following abbreviations are established: Booster System (BS), Parallel Compression-supported booster system (PC), and Simplified Ejector-supported booster system in Summer Mode (SEB-SM) and Winter Mode (SEB-WM).

3.1. Experimental setup

Fig. 2 shows a simplified diagram of the experimental setup used, which is located at NTNU/SINTEF Varmeteknisk laboratory in Trondheim (Norway). The setup consists of the CO_2 refrigeration system and several auxiliary circuits, it is versatile and allows experiments to be performed with different system configurations (booster, parallel compression, ejector supported parallel compression, etc.) and for a wide range of operating conditions. A more detailed explanation of the unit can be found in reference (Pardiñas et al., 2018).

The CO_2 compressor rack consists of eight semi-hermetic piston compressors (Table 1), all of them manufactured by Bitzer. Table 1 indicates how the compressors were arranged as a function of the system configuration tested.

The test facility comprises three gas coolers (plate heat exchangers) arranged in series to reject heat at different temperature levels, and utilizing glycol, ice water and an auxiliary R744 refrigeration system, respectively. Two or three of these gas coolers were utilized depending on the gas cooler outlet temperature needed for each test. WMV_2 is a three-way valve located downstream of the gas coolers that, in combination with the closing of WMV_3, allowed changing between summer mode and winter mode in SEB configuration. Concerning the high-pressure control devices, two of the alternatives were utilized during this test campaign: the high-pressure valve (HPV) in BS and PC configurations, and the low-pressure lift Multi Ejector (EJ_{AC}) in the SEB configuration. EJ_{AC} is an LP 1935 from Danfoss, i.e., with six cartridges in parallel, but only four of these cartridges were needed during the test campaign due to the test conditions considered, which made it effectively a LP 935. More information about the Multi Ejector can be found in reference (Kalinski, 2019), and according to this reference and to previous experience, the low-pressure lift Multi Ejector should fulfil all the requirements listed in section 2.

The liquid receiver is located downstream of the high-pressure

Table 1
Characteristics of the compressors utilised in the setup. System configurations nomenclature: Simplified Ejector-supported Booster (SEB), Booster System (BS) or Parallel Compression-supported booster system (PC).

Model	Name	Swept volume [m^3 h^{-1}] at 50 Hz	Inverter driven?	Configurations
2JME-3K	LT_1	3.5	Yes	All
2GME-4K	LT_2	5.0	No	None
4MTC-10K-40S	MT_1	6.5	Yes	BS, PC
	MT_2		No	BS, PC
2KTE-7K-40S	PAR_1	4.8	Yes	SEB, PC
	PAR_2		No	SEB
4JTC-15K-40P	MT_3	9.2	No	SEB, BS
	PAR_3		No	None

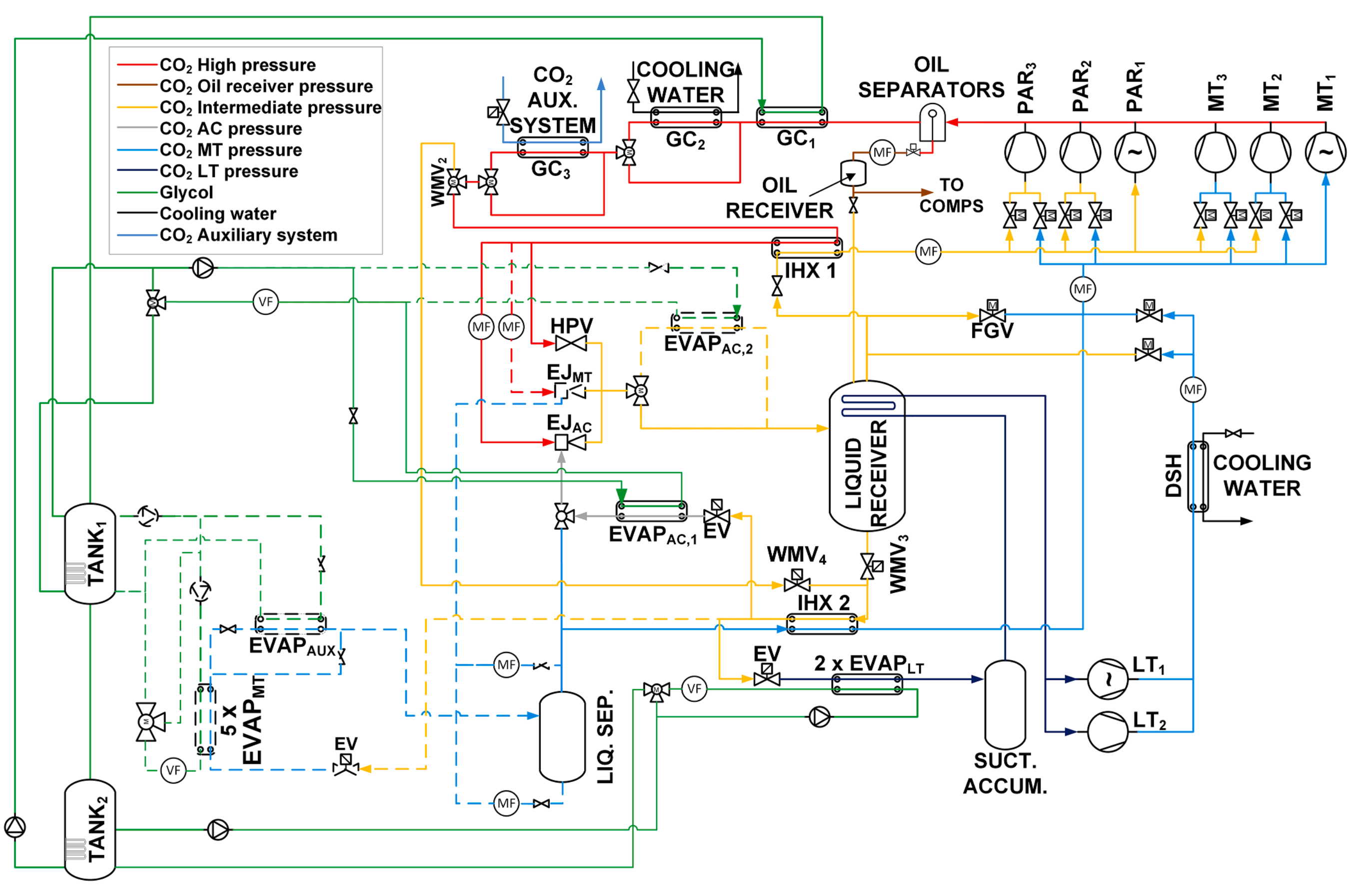

Fig. 2. Simplified diagram of the refrigeration system and secondary loops in NTNU/SINTEF laboratory (Trondheim, Norway). Dashed lines and components were not in use during this test campaign. MF represents a Coriolis mass flow meter (R744), while VF an electromagnetic volumetric flow meter (glycol side).

control devices. Liquid CO_2 was fed from the bottom of the liquid receiver to the MT and LT evaporators under all configurations except for the SEB in winter mode, when the liquid is directly supplied from the gas coolers as mentioned above. The control of the conditions (pressure) in the receiver was different depending on the system configuration tested:

- In BS, applying the flash-gas bypass valve (FGV) to throttle vapour to the MT compressor suction ports.
- In PC, utilizing the capacity control device (inverter) of the parallel compressor.
- In the SEB configuration, the liquid receiver pressure was a consequence of the control of MT evaporation pressure, which was achieved adjusting the MT compressor suction pressure (inverter + number of compressors in operation) in accordance to the achieved ejector pressure lift at each condition.

The MT evaporator chosen for this test campaign was $EVAP_{AC,1}$ (brazed plate heat exchanger SWEP B185 with 140 plates) for two reasons. First, because it was already connected to the low-pressure lift ejector EJ_{AC} needed in the SEB configuration. Second, because it allowed bypassing the liquid separator (LIQ. SEP. in Fig. 2) in SEB tests. For the sake of fairness in the comparison, also $EVAP_{AC,1}$ was utilized in the BS and PC by shifting the position of the three-way valve downstream of the evaporator, disconnecting the evaporator from the ejector. Concerning the LT load, one of the LT evaporators available in the system (coaxial heat exchanger Klimal VCI 38 WT5.5-G) was sufficient under all conditions and configurations. Both the MT and LT evaporators utilized glycol to emulate the cooling demands. The conditions of the glycol (inlet temperature and flow rate) were automatically controlled for each test by the dedicated pump and three-way valve available for each evaporator.

The CO_2 stream from the LT evaporators was superheated by an internal heat exchanger within the liquid receiver, compressed by the LT compressors, and desuperheated in a desuperheater (DSH in Fig. 2) to a CO_2 temperature between 20 °C and 30 °C. After, the refrigerant flowed either to the MT compressors in the BS and PC system configurations or joined the vapour from the liquid receiver in the SEB configuration.

The data acquisition system is based on the synchronization of data from two solutions for data processing. The first is based on Danfoss hardware and software (sampling rate of 5 s), logging data from industrial quality sensors while controlling the refrigeration system components according to these measurements and the different setpoints. The second is based on National Instruments hardware and LabVIEW software (sampling rate of 1 s), retrieving data from more precise measurement equipment and adjusting the conditions of the secondary loops (glycol, water, and auxiliary CO_2 loop) to establish the boundary conditions for the main R744 compressor pack (in terms of cooling and freezing loads or operating temperatures). Since the analysis shown below was based on data from LabVIEW sensors, only these are listed in Table 2.

Table 2
List of measurement equipment and accuracy.

Type	Manufacturer and model	Accuracy
Mass flow meters	Rheonik RHM[a]	±0.2 % of reading
Pressure transducers	Endress+Hausser PMP21	±0.3 % of set span[b]
Differential pressure transducers	Endress+Hausser PMD75	±0.035 % of set span[c]
Temperature sensors	Pt 100 Class B DIN 1/3 on tube	±1/3(0.3 K + 0.005*T[d])
Volumetric flow meter	Endress+Hausser Picomag	±(0.8 % of reading + 0.2 % of set span[e])
Active power meter (compressors)	Schneider Electric A9MEM3150	±1 % of reading

[a] Different sizes are used depending on the position in the loop.
[b] The set spans of the sensors at the individual pressure levels are: high-pressure = 150 bar, liquid receiver pressure and MT evaporators = 80 bar, LT evaporators = 40 bar.
[c] The set span for the differential pressure transducer measuring the lift of EJAC is 16 bar.
[d] T is the temperature measured in °C.
[e] Set span is 100 L/min in the MT evaporator and 50 L/min in the LT evaporator.

Table 3
Test conditions for the Simplified Ejector-supported Booster configuration

Parameter	Range	Observations
MT loads [kW]	18 / 24 / 30	Share MT to LT load equal to 3, 4 and 5.
MT evaporation temperature [°C]	-2, -5	Expansion valve control: superheat @ 5 K setpoint.
LT loads [kW]	6	-
LT evaporation temperature [°C]	-25	Expansion valve control: Superheat @ 8 K setpoint.
Gas cooler outlet temperature [°C]	10 / 15 / 20 / 25 / 30 / 35 / 40	Covers cold and warm climate regions.
Gas cooler pressure [bar] SEB-SM	- / - / 59 / 65 / 75 / 85 / 96	Shown according to the gas cooler temperatures above.
Gas cooler pressure [bar] SEB-WM	45 / 51 / 57 / - / - / - / -	
Gas cooler pressure [bar] BS/PC	51 / 55 / 61 / 66 / 76 / 86 / 98	
Desuperheater temperature [°C]	20 / 20 / 20 / 25 / 30 / 30 / 30	Shown according to the gas cooler temperatures above.

3.2. Test conditions

This article aims to prove experimentally the feasibility and enhanced performance of the Simplified Ejector-supported Booster system. Its successful operation must be proven in both summer and winter modes, with relevant MT and LT loads and under different operational conditions. The selected test conditions for the experimental campaign are as shown in Table 3. The cooling and freezing loads considered would correspond to those in small-sized supermarket, while the temperature range analysed should be enough to show the potential of this system layout almost anywhere in the world, with design gas cooler outlet temperature as high as 40 °C. This temperature is not currently a technical limit for the operation of a SEB unit but is typically utilized for the designs in Southern Europe. Moreover, even if the MT expansion valve operated according to superheat (Danfoss DAQ) and superheat was determined also with the LabVIEW-based DAQ (below 5 K), the evaporator was performing very close to saturated vapour at its outlet. This was observed at the sight glass located downstream of the MT evaporator, with alternation of pure vapour and vapour stream with liquid droplets entrained. It is worth pointing out that the temperature sensors in the experimental setup are located on the tubes, and even though thermal paste and insulation was used, they could not detect properly the presence of liquid in the core of the stream leaving the evaporator.

Concerning the enhanced-performance assessment mentioned in the previous paragraph, the SEB system was compared with a conventional R744 booster system (BS) and parallel compressor-supported system (PC) by testing these configurations considering the same conditions as in Table 3, except for i) the MT load, restricted to 30 kW, ii) the MT evaporator, operated as direct expansion, with 10 K superheat and evaporation temperature -8 °C, and iii) the liquid receiver pressure setpoint at 41 bar. Increasing the MT evaporation temperature from -8 °C with substantial superheat (direct expansion) as high as to -2 °C with (almost) overfed conditions is something that has been successfully proved in field tests (Hafner et al., 2014).

3.3. Data analysis and validation

For each test condition, data were collected during 15 minutes at steady state conditions, and they were averaged during that period and analysed as described in this section.

The loads, $\dot{Q}$, at the different evaporators in operation were calculated for both the CO_2 side and the glycol side, following Eq. (1) and Eq. (2), respectively.

$$\dot{Q}_{\mathrm{ref,evap}} = \dot{m}_{\mathrm{ref,evap}} \cdot \left(h_{\mathrm{ref,out,evap}} - h_{\mathrm{ref,in,evap}}\right) \tag{1}$$

$$\dot{Q}_{\mathrm{glycol,evap}} = \dot{V}_{\mathrm{glycol,evap}} \cdot \rho_{\mathrm{glycol}} \cdot c_{p_{\mathrm{glycol}}} \cdot \left(T_{\mathrm{glycol,in,evap}} - T_{\mathrm{glycol,out,evap}}\right) \tag{2}$$

For the refrigerant side:

- Mass flow rates, $\dot{m}$.
  - LT evaporators, directly measured with mass flow meter downstream of the LT compressors.
  - MT evaporators, determined by subtracting the LT evaporator mass flow rate from the liquid line mass flow rate.
- Refrigerant enthalpies, h. Determined using REFPROP 10 database (Lemmon et al., 2018).
  - Inlet to both evaporators, assuming saturated liquid at the pressure measured at the liquid receiver. The only exception would be with SEB-WM, for which the pressure sensor considered was located at the gas cooler outlet.
  - Evaporator outlets, as the maximum between the saturated vapour enthalpy, evaluated with the pressure sensors located at the outlet of each evaporator, and the enthalpy determined with pressure and temperature sensors at the outlet of each evaporator.

For the glycol side:

- Volumetric flow rates, $\dot{V}$, were measured with the electromagnetic flow meters connected to each of the evaporators.
- Glycol properties. The fluid used is a propylene glycol-water solution at 30 vol%, and its properties were taken from the ASHRAE Handbook Fundamentals (2009) (ASHRAE 2009). Specific heat capacity, c_p, was evaluated at average temperature between the temperatures measured between inlet and outlet for the evaporator. For the density, ρ, only the outlet temperature was utilized.
- Temperatures were measured with sensors located right upstream and downstream of each of the evaporators.

For validation, the loads calculated according to Eqs. (1) and (2) were compared for the configurations taken as reference, i.e., BS and PC, since in these two cases the liquid line stream was subcooled, guaranteeing reliable measurements of the CO_2 mass flow meter at that line. As observed in Fig. 3, the agreement between CO_2 and glycol was good (within 5 %), and consequently, the loads obtained by calculation from the glycol parameters are used for the rest of the study.

The Coefficient of Performance (COP) was chosen as the system performance indicator to compare the different system configurations. The COP is the quotient of the cooling output produced in each test at MT and LT levels divided by the power consumption of the compressors, as presented in Eq. (3). P_{comp} is the total power consumption of the compressors in the system, obtained by adding up the data from the active power meters for each individual compressor. A higher COP represents a more effective system i.e., providing the required cooling and requiring less power to the compressors.

$$\mathrm{COP} = \left(\dot{Q}_{\mathrm{glycol,MT}} + \dot{Q}_{\mathrm{glycol,LT}}\right) / P_{\mathrm{comp}} \tag{3}$$

The ejector performance with the SEB configuration was also evaluated as shown Eq. (4), which is the ejector efficiency definition from Elbel and Hrnjak (2008) (Elbel and Hrnjak, 2008). ϕ_m is the mass entrainment ratio of the ejector calculated as the ratio of the mass flow rate on the suction side of the ejector (secondary flow) to the mass flow rate at the motive nozzle (primary flow). h_{suct} is the enthalpy at the ejector suction port, which was assumed as equal to the enthalpy at the MT evaporator outlet. $h_{s(\mathrm{suct}),p(\mathrm{disc})}$ is the enthalpy of an isentropic compression from the ejector suction conditions to the ejector discharge pressure. h_{mot} corresponds to the enthalpy at the ejector motive port, determined according to pressure and temperature measurements at that location. Finally, $h_{s(\mathrm{mot}),p(\mathrm{disc})}$ is the enthalpy of the isentropic expansion from the ejector motive conditions to the ejector discharge pressure.

$$\eta_{\mathrm{ej}} = \phi_m \cdot \frac{h_{s(\mathrm{suct}),p(\mathrm{disc})} - h_{\mathrm{suct}}}{h_{\mathrm{mot}} - h_{s(\mathrm{mot}),p(\mathrm{disc})}} \tag{4}$$

3.4. Uncertainty determination

The uncertainty analysis in this study was performed based on the

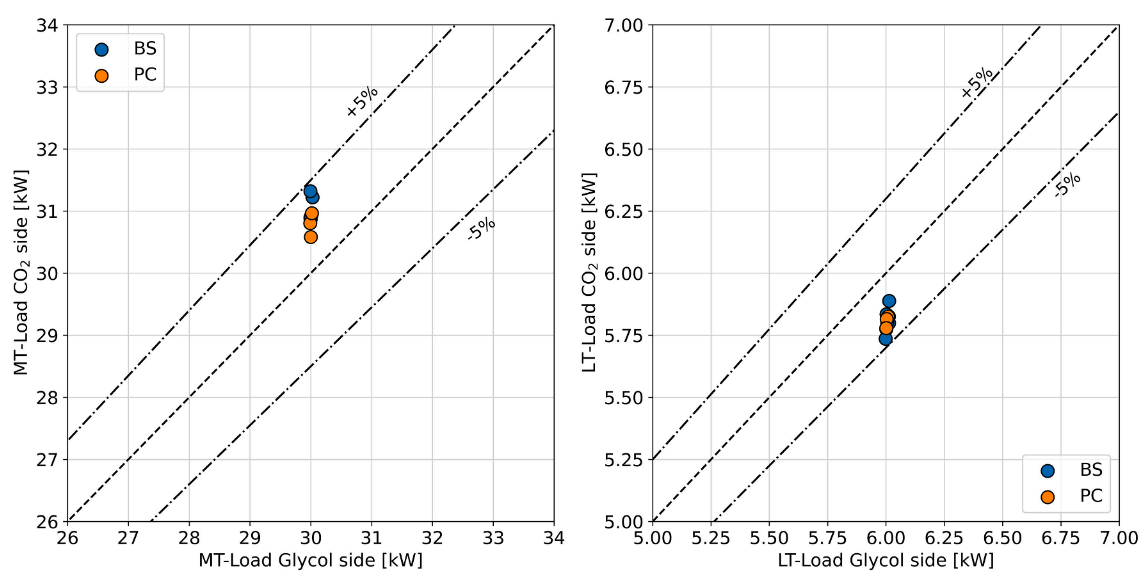

Fig. 3. Load calculated in the glycol side vs. load calculated in the CO2 side in the MT (left) and LT (right) evaporators. BS = Booster System, PC = Parallel Compression supported booster system.

Guide for Uncertainty of Measurement (JCGM 2008). This Guide classifies uncertainties in type A and type B.

- Type A uncertainties basically correspond to standard deviations of the parameter during the test.
- Type B uncertainties are due to the different sensors and databases utilized to determine thermodynamic properties. The information about the sensors was taken from the different datasheets by the manufacturers and is included in Table 2.
  - It is worth pointing out that, due to the lack of precise information from these sensors, it was assumed that the accuracy indicated corresponds to the worse case, i.e., with a coverage factor equal to 1. On the other hand and as suggested in the Guide (JCGM 2008), it was assumed uniform distribution of the measurements within the range indicated by the manufacturer. Thus, it was possible to divide these accuracies from the datasheets by the square root of three.
  - As aforementioned, thermodynamic properties for CO_2 and the glycol solution were determined according to REFPROP 10 (Lemmon et al., 2018) and ASHRAE Handbook Fundamentals (2009) (ASHRAE 2009), respectively. For the uncertainty of these thermodynamic properties, an approach as that described in reference (Aprea et al., 1997) was followed, combining the deviations of the thermodynamic property due to the uncertainty of each of the parameters used for its determination.
- Propagation of uncertainties as defined in the Guide (JCGM 2008) was used to determine the uncertainties corresponding to the different parameters/indicators.

For the sake of clarity of the figures, uncertainties are not represented for each data point. However, a summary of average relative uncertainties is included in Table 4. Relative uncertainty is shown in percentage and represents the ratio of the propagated uncertainty for the parameter to the value taken by the given parameter at the test point. A coverage factor equal to 2 was considered for the propagated uncertainty, indicating that the results have a level of confidence around 95 %.

4. Results and discussion

4.1. Ejector performance in the novel system layout

The two critical factors to prove the successful operation of the novel system concept are the obtained pressure lift from the ejectors during summer mode (Fig. 4 left) and the pressure drop over the ejector during winter mode as a result of being operated as a check valve (Fig. 4 right). The ejector pressure lift in summer mode (Fig. 4 left) followed a near-linear trend of increasing pressure lift of the ejector with increasing gas cooler outlet temperature, ranging from around 1.5 bar at 20 °C up to about 6 bar at 40 °C. The value recorded at the lowest temperature evaluated, 20 °C, could be insufficient to circulate all the mass flow through the MT evaporators, even if this would depend on the sizing of the lines and expansion valves. At all the other conditions, the pressure lifts attained were satisfactory, provided the relatively high entrainment ratios required in this system configuration. Concerning winter operation (Fig. 4 right), the check valves integrated in the design functioned as they should and caused pressure drop from the suction port to the discharge port of the ejector below 0.1 bar for almost all the test conditions, which is acceptable.

Fig. 5 depicts the results for the other two criteria that should be met by the ejector in this innovative configuration: on the left hand side the entrainment ratio and on the right hand side the ejector efficiency. The entrainment ratio was imposed by the mass balances existing in the system at the different conditions and was relatively high as would be expected due to the system nature. The entrainment ratio decreased nearly linearly from above 0.6 at 20 °C to below 0.4 at 40 °C, in parallel with the increase of pressure lift at the same range of gas cooler outlet temperatures. Thus, the ejector was able to lift the entire suction stream with the given entrainment ratio. How efficiently this ejection process occurred is represented by the ejector efficiency in Fig. 5 (right). Here it can be observed that the ejector is not as efficient at the lowest and highest gas cooler outlet temperatures, and ejector efficiency peaks at 25 °C. Even if the ejector efficiency values are slightly higher with the MT evaporation temperature of -2 °C when compared to -5 °C, the difference is not important. Moreover, the ejector efficiency was independent of the ratio between MT and LT loads, given all the other conditions are equal, which indicates that the ejector would perform indistinctly with very different operating conditions (part load).

In conclusion, the commercial ejector selected for this innovative system layout fulfilled all the requirements listed in section 2, i.e. it controlled the high side pressure satisfactorily depending on the conditions, it operated efficiently with very different loads, achieved relatively high pressure lifts with the imposed entrainment ratio (active ejector), and confirmed a low pressure drop through the integrated check valves (passive ejector).

4.2. Performance of the novel system layout

Fig. 6 depicts the COP, as calculated with Eq. (3), for the novel SEB system layout. The graph on the left hand side indicates the values determined with data at -2 °C MT evaporation temperature, and the one on the right hand side for the -5 °C condition. As had been indicated in Table 3, summer mode operation (circles in the figure) was exclusive at gas cooler outlet temperatures from 25 °C to 40 °C, winter mode (squares) was exclusive at 10 °C and 15 °C, and both options were tested and evaluated at 20 °C. COP increased as gas cooler temperature decreased, and for most of the data the system performance was best with a higher share of MT load to LT load. The effect of the ratio MT load to LT load on the COP was higher in winter mode than in summer mode, being the reason that in summer mode the fraction of the mass flow rate expanded by the ejector reaching the receiver as vapour was directly compressed by the MT compressors. Only liquid was delivered to the evaporators and the inlet vapour fraction was lower. On the other hand, in winter mode the expansion occurred directly from the high pressure, meaning that the inlet vapour fraction at the evaporators was higher. The lower the MT to LT ratio, the higher the share of useless vapour mass flow rate that had to go through a double compression, i.e., LT compressors and MT compressors. If compared both operation modes at 20 °C, Fig. 6 shows that summer mode would be better from an efficiency (COP) point of view. However, there is a technical aspect to clarify why there should be a transition from summer mode to winter mode above this temperature: the achievable pressure lift with the ejector. As shown in Fig. 4, the pressure lift at 20 °C was around 1.5 bar, which could be insufficient to overcome pressure drops through the

Table 4
Average relative uncertainties (in percentage) of some of the parameters determined during the experimental campaigns, for the different configurations.

Configurations	$Q_{glycol,MT}$	$Q_{glycol,LT}$	$Q_{glycol,MT+LT}$	P_{MT}	P_{LT}	P_{PAR}	P_{comp}	COP
BS	4.90 %	9.50 %	4.44 %	2.27 %	4.19 %	-	2.09 %	4.99 %
PC	5.33 %	13.06 %	4.99 %	1.62 %	5.55 %	2.56 %	1.32 %	5.17 %
SEB @-2 °C	7.76 %	9.29 %	6.51 %	3.01 %	5.14 %	-	2.69 %	7.14 %
SEB @-5 °C	6.83 %	9.50 %	5.79 %	2.77 %	5.18 %	-	2.44 %	6.45 %

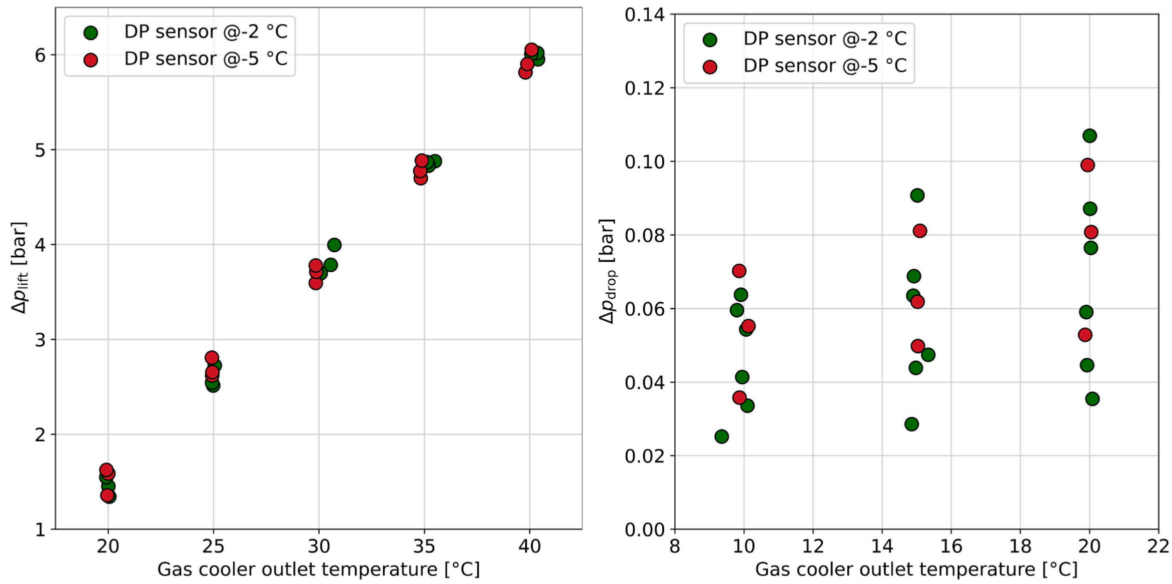

Fig. 4. Pressure lift of the ejector during operation in summer mode (left) and pressure drop over ejector during operation in winter mode (right), as a function of the gas cooler outlet temperature.

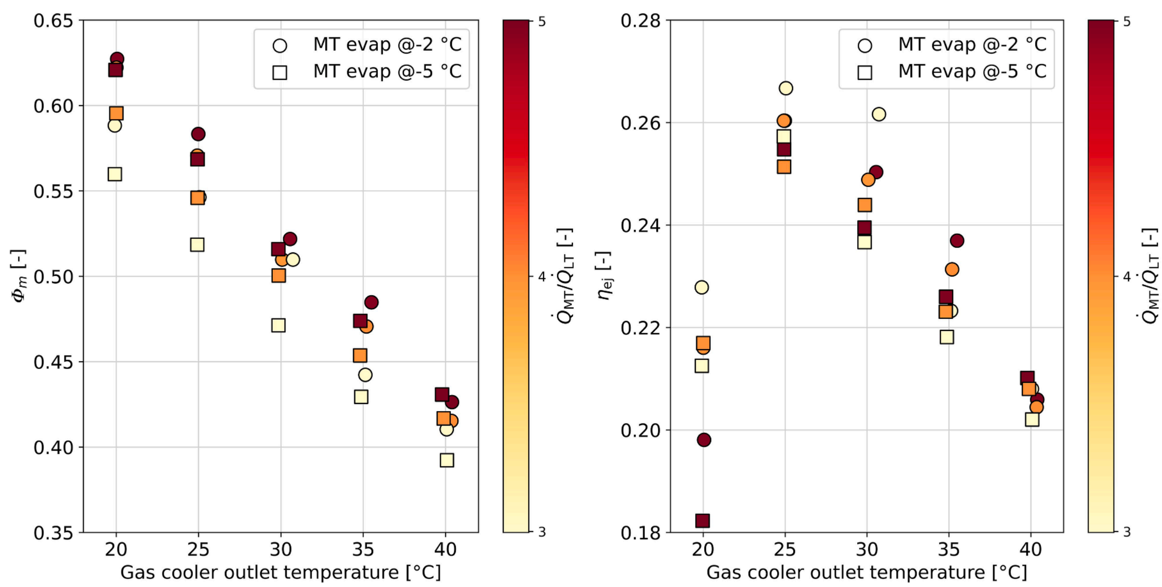

Fig. 5. Ejector performance (left, entrainment ratio; right, ejector efficiency) in the SEB system as a function of the gas cooler outlet temperature, at different MT evaporation temperatures, i.e., -2 °C and -5 °C. The filling colour of the circles indicates the ratio between MT and LT loads.

supply and return lines, and through the MT evaporators and their expansion valves.

The initial control strategy for the MT evaporator expansion valve in SEB winter mode was through superheat at evaporator outlet, with a very low superheating setpoint so that conditions close to saturated vapour should be achieved. By doing so, the expectation was that the gas cooler pressure (condensation temperature in this case) would be directly related to the gas cooler outlet temperature, achieving saturated liquid at the gas cooler outlet. A more thorough analysis of the data gathered in winter mode showed very good relation between gas cooler outlet temperature and condensation temperature (evaluated with a pressure sensor at the gas cooler outlet). However, for some of the tests the CO_2 mass flow rates measured through the LT evaporators were higher (in the range of 10 % and 15 %) than those that would be determined utilizing the cooling load obtained from the secondary loop, according to Eqs. (1) and (2) and assuming energy balance in the evaporator. A potential explanation to this would be that the conditions upstream of the expansion valve to the evaporators would be different to saturated liquid, due to pressure drop through valves and pipes, due to heat ingression from the laboratory, due to incomplete condensation from the gas cooler (vapour quality slightly above 0 on average during the test) or due to a combination of all of them. Thus, an alternative control strategy was suggested and evaluated, which would control high pressure with the opening degree of the MT evaporator expansion valve to attain around 1 K or 2 K subcooling degree at the gas cooler outlet. Both control strategies were compared as shown in Fig. 7, where the squares represent results determined with the initial strategy (superheat SH control of the electronic expansion valve EEV), while circles depict those with the alternative strategy (EEV control according to high pressure HP). Except for a couple of test points, the system performed best (higher COP) when the expansion valve was controlled according to high pressure. The reason was that the fact of having a subcooled CO_2

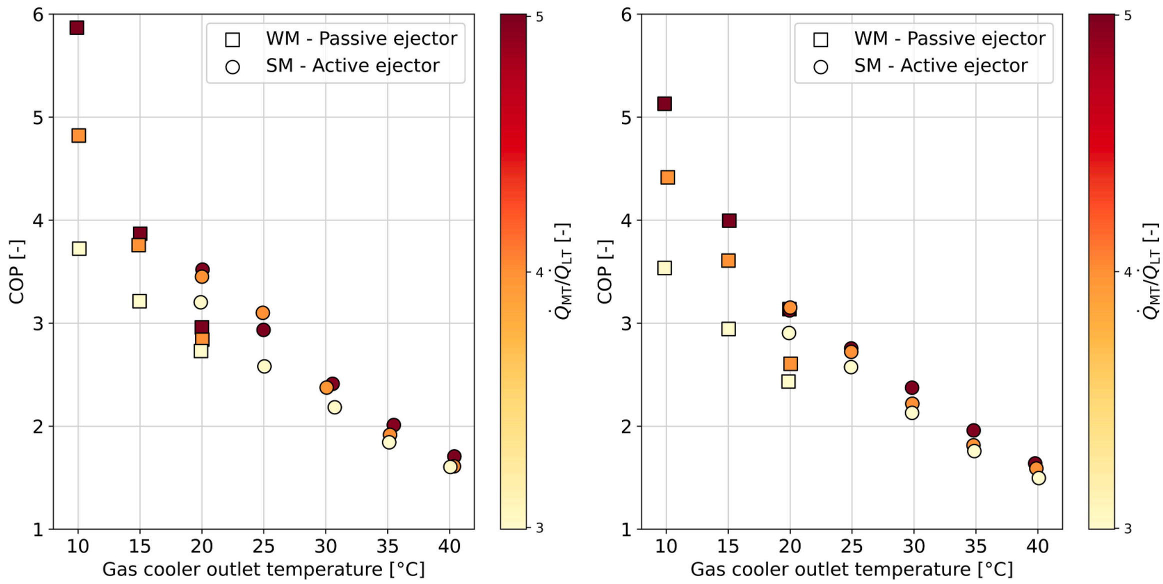

Fig. 6. COP of the SEB system layout as a function of the gas cooler outlet temperature and of the operating mode (winter mode or summer mode), at different MT evaporation temperatures, i.e., -2 °C (left) and -5 °C (right). The filling colour of the circles indicates the ratio between MT and LT loads.

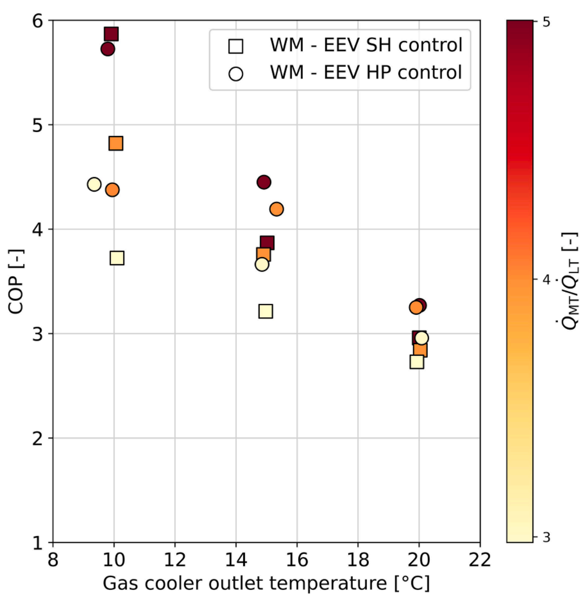

Fig. 7. Comparison of control strategies for winter mode utilizing the COP as a function of the gas cooler outlet temperature and at MT evaporation temperature equal to -2 °C. The filling colour of the circles indicates the ratio between MT and LT loads.

stream at the gas cooler outlet caused that lower CO_2 mass flow rates were needed at the same conditions and to attain the same cooling loads. This alternative control strategy would require reliable communication between the main system (compressor pack) and individual expansion valves for the evaporators.

4.3. Benchmark of novel system layout to conventional booster systems

Simplified Ejector-supported Booster (SEB) system layout was benchmarked against more conventional CO_2 booster configurations, namely booster system (BS) and parallel compression-supported (PC) to analyse its potential to enhance energy performance, and the results of this comparison are shown in Fig. 8. For all the configurations the MT cooling load was 30 kW. In the case of the SEB layout, winter mode data were considered at 20 °C gas cooler outlet temperature, disregarding the results for summer mode for the reasons indicated in previous paragraphs of relatively low pressure lift. Fig. 8 left includes the COP determined for each test, while Fig. 8 right shows the percentage of COP improvement due to utilizing the SEB system layout compared to BS or PC configuration at equal operating conditions. The most important findings are the following:

- SEB system layout, independently of the MT evaporation temperature considered, outperformed a BS throughout the gas cooler outlet temperature range considered. This indicates the great potential that this innovative configuration has for all climates due to the ability to switch from a transcritical (summer mode) to a subcritical (winter mode) operation. In percentage, the largest benefits were achieved at the lowest and highest temperatures tested. If considered the case with MT evaporation temperature equal to -2 °C, the enhancement was approximately 38 % at 10 °C and 42 % at 40 °C. For MT evaporation temperature equal to -5 °C, these percentages decreased to 25 % and 36 %, respectively. At 20 °C, the benefit of utilizing this novel configuration would be between 5 % and 10 %, and mostly linked to the increased MT evaporation temperature.
  - The positive effect in the low (gas cooler outlet) temperature end would come from the reduced high pressure and increased MT evaporation temperature. This would compensate the negative impact of the higher inlet vapour fraction into the evaporators, particularly important in the case of the LT evaporator. However, it would be important to keep in mind that the current operating envelopes of CO_2 reciprocating compressors do not recommend relatively low pressure ratios, as would be the case in SEB with winter mode. Manufacturers would be encouraged to work on their equipment (at reliability and efficiency levels), since the number of hours at which commercial refrigeration systems would benefit of this winter mode operation is not negligible.
  - The benefit at higher gas cooler outlet temperatures would be due to the combination of the ejector and increased evaporation temperature.
- PC configuration was only possible in the experimental setup from 25 °C gas cooler outlet temperature and above, due to the loads considered and the size (displacement) of the parallel compressor capacity installed. Even at 25 °C, the difference in performance between PC and BS configurations was negligible, but it became

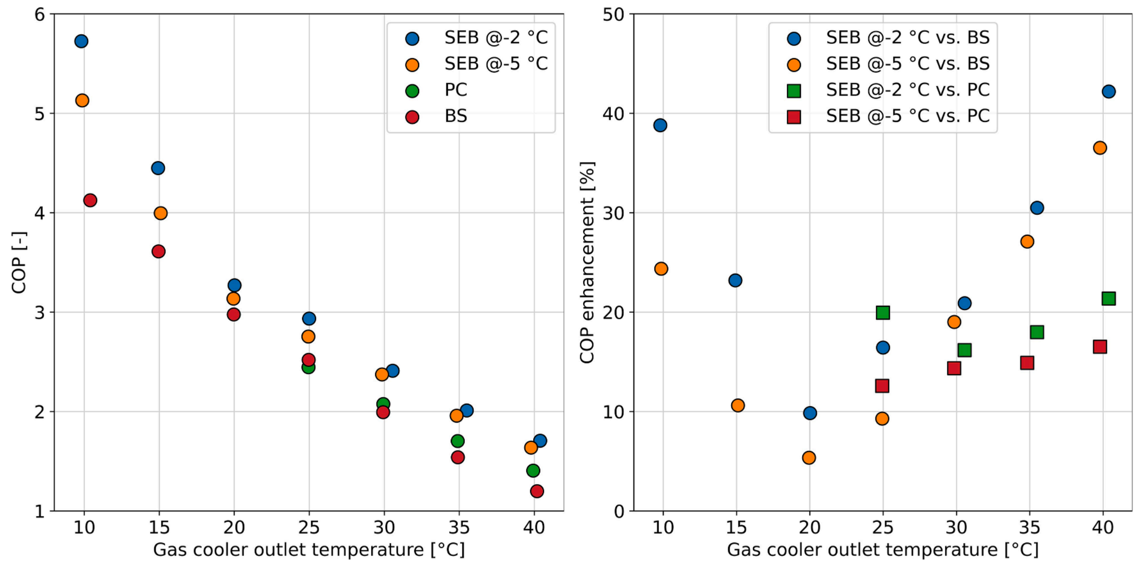

Fig. 8. Comparison of the Simplified Ejector-supported Booster (SEB) system layout compared to a traditional Booster System (BS) and Parallel Compression-supported booster system (PC). Left, COP for the different system layouts as a function of gas cooler outlet temperature. Right, COP enhancement in percentage due to using SEB layout compared to the benchmark configurations BS and PC. For the SEB, two MT evaporation temperatures considered, namely -2 °C and -5 °C.

increasingly important as the gas cooler outlet temperature increased.

- If compared SEB and PC system configurations, the results show that the enhancement was much more constant as a function of gas cooler outlet temperature. The benefit of utilizing the novel system configuration was mostly in the range of 10 % to 20 %, being slightly higher in the case of MT evaporation temperature equal to -2 °C.

5. Conclusions

This article has introduced and proven the potential of a novel CO_2 booster system with ejector for commercial refrigeration. In contrast with other ejector-supported system layouts, the ejector-supported booster system presented focused on simplicity (reduced number of components) but still efficient operation at a very wide range of conditions. The concept relies on increased MT evaporation temperature due to non-superheated evaporation and appropriate switching between two working modes as a function of ambient conditions, namely, summer mode, with active ejector, and winter mode, with passive ejector. The system layout was tested in the laboratory and benchmarked against more conventional CO_2 booster refrigeration units, and the main findings of this study are listed below.

- The ejector suited for this system layout should accommodate high entrainment ratios with sufficient pressure lifts in summer mode, while it should enable an active control of the high side pressure and should perform efficiently at different conditions (various heat rejection temperatures, load scenarios, etc.). Moreover, in winter mode the (passive) ejector should invoke relatively low pressure drop between its suction and discharge ports. The market is prepared to supply ejectors that meet all these requirements, as shown in the experimental campaign using a state-of-the-art ejector.
- Expansion valves had a significant impact on the high-pressure level in winter mode, when the ejector motive port was closed, and liquid was supplied to the cabinets directly from the gas cooler. A conventional control of expansion valve according to superheating degree at the evaporator outlet could lead to incomplete condensation in the gas cooler. A better regulation of high pressure in winter mode was achieved by controlling the expansion valve according to the subcooling degree at the gas cooler outlet.
- The novel configuration proposed in this study outperformed a conventional booster system at all ambient conditions, with enhancements of the COP (first law efficiency) between 5 % and 42 %. The improvements were larger at the higher end of the temperature range tested due to the combined effect of ejector and slightly increased MT evaporation temperature, and at the low end due to reduced pressure ratio for the MT compressors.
- If compared to a more advanced booster system, such as the parallel compression-supported booster system, the benefits in terms of efficiency were still as high as 22 %.

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.

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Supplementary materials

Supplementary material associated with this article can be found, in the online version, at doi:10.1016/j.ijrefrig.2022.10.027.

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