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Experimental evaluation of the performance of an ejector for a single compression multi-temperature CO_2 refrigeration unit

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ABSTRACT

A novel vapor-compression system concept employing carbon dioxide as the refrigerant is proposed to serve the needs of a typical medium-size refrigerated truck used for multi-temperature (MT and LT) goods delivery. The system design is based on the implementation of an ejector as the only component increasing the refrigerant pressure from the LT evaporation pressure to the MT evaporation pressure, thus allowing the realization of a unit providing cooling effect at two different temperature levels with only one stage of compression. The ejector was experimentally tested in order to evaluate its ability to effectively entrain mass flow rate from very low pressure conditions at the suction nozzle, corresponding to the LT evaporator outlet conditions. In addition, a simple preliminary thermodynamic evaluation of the performance of the refrigerating unit at -25°C LT saturation temperature and 35°C gas cooler outlet temperature is conducted.

Keywords: Refrigeration, Carbon Dioxide, Refrigerated transport, Multi-temperature transport, Ejector.

1. INTRODUCTION

Road transport of temperature-controlled goods plays a crucial role in the cold chain. It is reported that around 31% of the food supply chain includes refrigerated transportation (Bagheri et al., 2017). Traditionally, temperature-controlled logistics is organized in order to distribute goods separately for each product segment, with specific temperature requirements. However, in recent years the market is pushing more and more towards the use of trucks equipped with temperature-specific compartments, which allow the simultaneous transport of different product segments in separate chambers of the same truck (Frank et al., 2021), thus allowing more flexibility in the logistics and reducing the number of vehicles on the road, especially for the last-mile delivery of goods.

At the same time, the approval of the EU F-Gas Regulation 517/2014 (European Commission, 2014) and the consequent progressive ban of commonly used synthetic refrigerant which will occur in the near future exponentially increased the interest in employing natural refrigerants (in particular CO_2) in newly developed refrigeration units.

According to the literature (Gullo et al., 2018; Karampour and Sawalha, 2018), the commonly implemented cycle for multi-temperature CO_2 stationary units is given by a booster cycle with a double stage of compression. The baseline booster cycle can be further modified with the implementation of parallel compression and the use of an ejector providing the lift from the MT evaporation pressure to the liquid receiver intermediate pressure. In such a system, a subcritical compressor is needed to increase the CO_2 pressure from the LT evaporation pressure to the intermediate pressure of the liquid receiver. While these

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systems are generally employed for stationary applications, such as supermarkets, Fabris et al. (2021) numerically evaluated the performance and the operational optimization of such a system for refrigerated transport applications. However, the analysed unit still implemented two stages of compression.

In this study, a novel CO_2 cooling unit concept for multi-temperature refrigerated transport applications is proposed, based on the implementation of an ejector as the only component dedicated to the increase of the refrigerant pressure from the LT to the MT evaporating pressure, thus allowing the removal of the LT subcritical compressor from the system configuration. To verify the actual feasibility of such a unit arrangement, experimental tests were performed on an ejector to assess its performance in the desired low-temperature range (down to -25°C LT saturation temperature), far from the operational field in which CO_2 ejectors are traditionally employed. A numerical evaluation of two CO_2 refrigeration unit concepts employing an ejector to provide pressure lift after the LT evaporation has been carried out by Banasiak et al. (2019), but an experimental study on ejectors operating in such applications was still to be performed.

This study firstly explains in detail the novel refrigerating unit concept, then it describes the experimental setup and the operating conditions used for the assessment of the ejector performance. Finally, the ejector performance results are provided and discussed, and a simple preliminary thermodynamic evaluation of the performance of the refrigerating unit is conducted.

2. REFRIGERATION UNIT CONCEPT

The CO_2 refrigeration unit concept presented in this paper is intended to propose a novel and simple unit arrangement for the fulfilment of the refrigerating needs of a multi-temperature medium-size refrigerated van, employed for the short-range road delivery of chilled and frozen goods. The proposed system is designed to simultaneously supply 4-5 kW of Medium-Temperature (MT) refrigeration at an air temperature of 0°C and 1-2 kW of Low-Temperature (LT) refrigeration at an air temperature of -20°C by taking advantage of the pressure lift provided by an ejector, whose placement and implementation inside the unit represents the novel aspect described in this study.

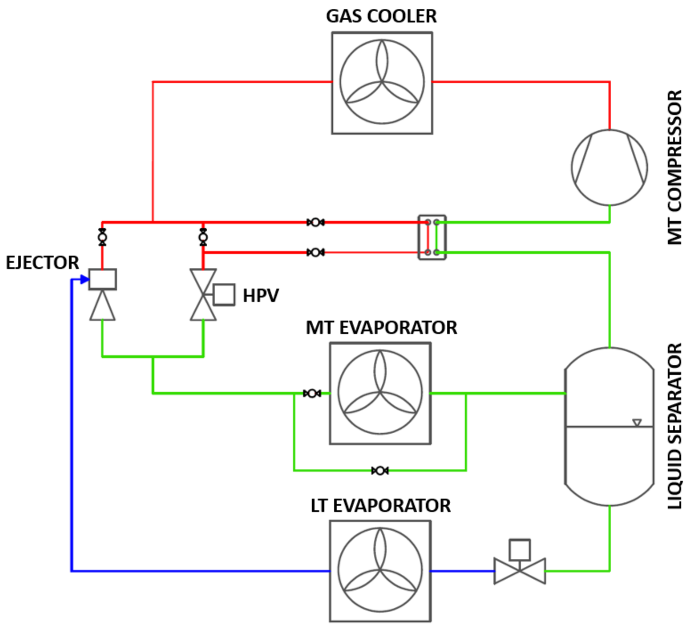

Figure 1: Simplified schematics of the refrigeration unit concept.

Differently from commonly developed multi-temperature CO_2 units, the ejector does not provide a pressure lift from the MT evaporation pressure to an intermediate pressure level in which a liquid receiver is placed, thus allowing compression from the intermediate pressure level instead of from the MT evaporation pressure level and a consequent reduction of the system compression work. In this kind of systems, an additional subcritical compressor is needed to bring the refrigerant from the LT evaporator outlet to the liquid receiver. Instead, in the proposed unit concept the ejector is employed to provide the pressure lift from the LT evaporation pressure to the MT evaporation pressure, thus allowing the complete removal of the additional subcritical compressor and enabling the realization of a multi-temperature cycle with only one compression stage, as the LT to MT pressure increase is provided by the ejector without external work requirement.

The simplified schematics of the refrigeration unit concept is presented in Figure 1. After the compression, the refrigerant rejects the heat to the external ambient in a gas cooler and then it is sent to the ejector motive

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nozzle to provide the energy required to entrain mass flow rate from the suction nozzle. Since a fixed-geometry ejector is suggested for this application, a high-pressure valve (HPV) is implemented in parallel to adjust the high-pressure according to environmental conditions and cooling load. The mixture at the outlet of the HPV-ejector stage is sent to the MT evaporator, where it provides the required chilled-storage cooling effect, and then to a liquid separator. The liquid separator works as a suction accumulator before the MT compressor, to deliver the refrigerant vapor phase (eventually superheated in an internal heat exchanger) to the compressor suction, while the liquid phase is expanded to the LT evaporator inlet and then entrained by the ejector.

The schematics presented in Figure 1 also allows the operation in case only one of the two refrigerating effects is needed: in case of LT-only load requirement, the MT evaporator can be bypassed; in case of MT-only load requirement, instead, the ejector is not in operation and the system works as a simple back-pressure cycle, with all the mass flow rate expanded in the HPV.

While the refrigerating unit concept presented in Figure 1 could potentially work, depending on the ejector performance assessment which will be discussed in the next sections, for different ranges of MT and LT cooling load requirements, the evaluations which will be carried out throughout this study are referred to the usual needs of a medium-size refrigerated van employed for short-range delivery of goods, consisting of a MT load in the range of 4-5 kW and a LT load in the range of 1-2 kW. The choice of the ejector size for the thermodynamic performance evaluation of the system which will be presented in the next sections will be done consistently with this resolution.

3. EXPERIMENTAL EVALUATION

In relation with the multi-temperature CO_2 refrigeration unit design presented and described in Section 2, to verify the actual feasibility of the application of an ejector to provide the required pressure lift from the LT load evaporation pressure (corresponding to a saturation temperature of down to -25°C) to the MT load evaporation pressure (to be evaluated depending on the available lift provided by the ejector), an experimental campaign has been carried out.

3.1. Experimental setup

The experimental evaluation of the performance of the ejector under low-temperature operating conditions has been performed in the SuperSmart-Rack test facility, located in NTNU/SINTEF laboratories in Trondheim (Norway). The SuperSmart-Rack test facility is a flexible and versatile experimental setup offering the implementation of various solutions and configurations to recreate completely the refrigerating needs of a supermarket for both chilled and frozen storage over a wide range of operating conditions. A detailed description of the system can be found in (Pardiñas, Hafner et al. 2018). A simplified schematics of the unit is provided in Figure 2, where the dashed lines and components were not used during this experimental campaign.

The schematics of this experimental setup differs from the refrigeration unit concept presented in Figure 1. However, the main purpose during the experimental tests was to evaluate the performance of the ejector at specified operating conditions, i.e., to accurately control the state at the motive, suction and discharge ports of the ejector.

The ejector installed in the SuperSmart-Rack system and considered for the tests is a Multi Ejector CTM Combi HP 1875 LE 600 from Danfoss and it is composed by four vapour ejector cartridges (with increasing capacity) and two liquid ejector cartridges (with increasing capacity) in parallel. However, the cooling unit concept presented in Section 2 is intended to be used for limited load requirements (4-5 kW of MT load and 1-2 kW of LT load) compared with the Multi Ejector capacity. For this reason, the experimental campaign was conducted engaging only the smallest cartridge of the Multi Ejector pack (VEJ1), which is characterized by a motive nozzle throat diameter of $1 \cdot 10^{-3}$ m. Further information about the Multi Ejector can be found in

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Kalinski (2019), while the main geometric parameters of the VEJ1 ejector are reported in Banasiak et al. (2015).

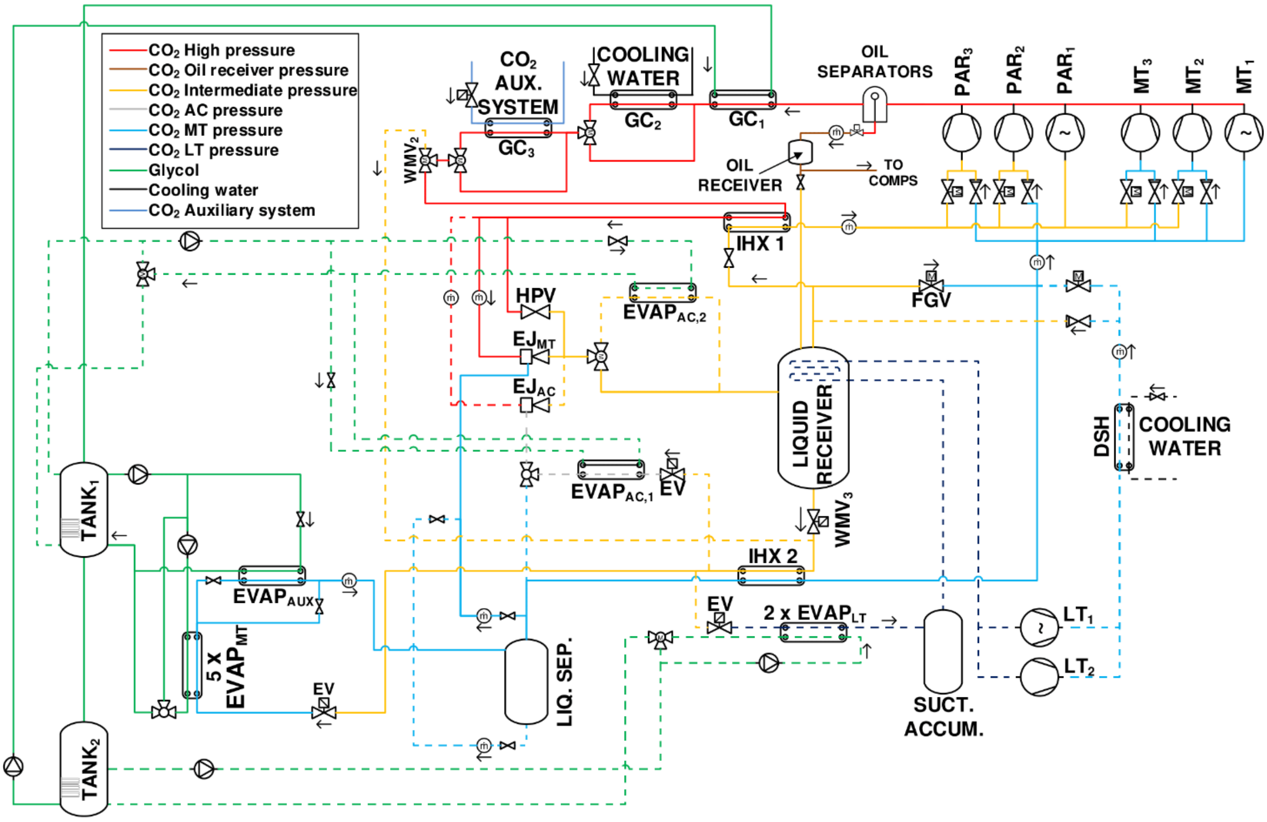

Figure 2:Simplified schematics of the experimental setup used for the ejector performance tests. Dashed lines and components were not in use during the experiments.

Referring to the schematics presented in Figure 2, the MT and intermediate (receiver) pressure levels, corresponding to the suction and discharge ejector ports, are adjusted by the capacity control of the MT and parallel compressors, respectively.

The heat is rejected to three gas coolers (plate heat exchangers), working in series at different temperature levels against glycol, cooling water and CO_2 coming from an auxiliary system, respectively. The individual gas coolers can be partially or completely bypassed depending on the desired temperature at the outlet of the gas cooler section.

The fluid at the desired gas cooler outlet conditions is sent to the ejector motive nozzle, while the exceeding mass flow rate is expanded in the HPV, granting at the same time the needed high pressure control. The CO_2 mix at the outlet of the HPV and at the ejector discharge nozzle is sent to the liquid receiver, whose pressure is controlled through the parallel compressors capacity (or through a flash gas valve FGV) and from which the liquid refrigerant is expanded and sent to the evaporators (helical coaxial tube-in-tube heat exchangers). The control of the MT evaporation pressure is achieved adjusting the MT compressor capacity, while the evaporators feeding valves operate to guarantee a desired superheat (in the range of 8 – 10 K) at the evaporators outlet. The difference between MT and liquid receiver pressure levels represents the pressure lift provided by the ejector.

The data acquisition system of the experimental setup consists of a hybrid combination of Danfoss controllers with industrial quality sensors (sampling rate of 5 s) and of National Instruments hardware (LabVIEW programming) with high-precision sensors (sampling rate of 1 s), whose data are then synchronized for data processing. However, the data analysis for this experimental campaign was based only on the high-quality

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data acquired by the LabVIEW sensors. The whole refrigerating system is equipped with sensors of various nature, but since the experimental campaign was focused exclusively on the ejector performance, only the sensors monitoring the ejector motive nozzle conditions (pressure, temperature, mass flow rate), the suction nozzle conditions (pressure, temperature, mass flow rate), the pressure lift (differential pressure) and the discharge nozzle conditions (pressure) are considered for data analysis. The list of the considered sensors and their accuracy are reported in Table 1.

Table 1: List of the equipment used for data acquisition and their accuracy.

Type	**Manufacturer and model**	**Accuracy**
Mass flow meters	Rheonik RHM	±0.2 % of reading
Pressure transducers	Endress+Hausser PMP21	±0.3 % of set span
Differential pressure transducers	Endress+Hausser PMD75	±0.035 % of set span
Temperature sensors	Pt 100 Class B DIN 1/3 on tube	±1/3(0.3 K + 0.005*temp(°C))

3.2. Test conditions

The objective of the experimental campaign was to verify the feasibility of operating the ejector at relatively low suction nozzle pressures and the consequent operating constraints and limiting parameters. The test conditions selected for the experimental campaign performed on the ejector are listed in Table 2.

Table 2: List of the test conditions of the experimental campaign.

Motive nozzle conditions	**Suction nozzle conditions**	**Pressure lift**
• T_{motive} = 35°C, p_{motive} = 90 bar • T_{motive} = 25°C, p_{motive} = 66 bar • T_{motive} = 15°C, p_{motive} = 54 bar	• $p_{suction}$ = 16.8 bar (T_{sat} = -25°C) • $p_{suction}$ = 19.7 bar (T_{sat} = -20°C) • $p_{suction}$ = 22.9 bar (T_{sat} = -15°C) • $p_{suction}$ = 26.5 bar (T_{sat} = -10°C) • $p_{suction}$ = 30.5 bar (T_{sat} = -5°C) • SH = 8 K for each condition	• $\Delta p_{lift,min}$ = 2 bar • $\Delta p_{lift,max}$ = maximum available lift • $\Delta p_{lift,step}$ = 1 bar

The ejector was tested considering three conditions at the motive nozzle representing operation over a wide range of external ambient temperatures, representative of the average European climatic conditions. The lowest suction nozzle pressure tested corresponds to a saturation temperature of -25°C, to verify the feasibility of providing an LT cooling effect to maintain an internal air temperature of -20°C in the truck compartment. Other suction conditions were tested to collect performance data to evaluate the performance also in transient operating conditions, such as during a pulldown to the nominal operating conditions. For each test condition, a pressure lift variable from a minimum value of 2 bar to the maximum available value for each specific condition was considered, thus allowing a complete evaluation of the ejector performance and the assessment of its optimal operating points.

4. RESULTS

4.1. Ejector performance

As first and crucial step, the ejector was experimentally tested in order to evaluate its potential to entrain refrigerant mass flow rate from low-pressure conditions and thus allow a refrigeration system design as the one presented in Section 2.

The performance of an ejector is determined by the primary stream mass flow rate at the motive nozzle ($\dot{m}_{motive}$) and by the secondary stream mass flow rate at the suction nozzle ($\dot{m}_{suction}$) or, alternatively, by the ejector mass entrainment ratio, defined as:

$$\phi_{ejector} = \frac{\dot{m}_{suction}}{\dot{m}_{motive}} \quad (1)$$

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Moreover, the ejector efficiency represents the ejector ability to recover expansion work with respect to the maximum possible expansion work rate recovery potential, as defined by Elbel and Hrnjak (2008):

$$\eta_{ejector} = \phi_{ejector} \frac{h(s_{suction}, p_{discharge}) - h_{suction}}{h_{motive} - h(s_{motive}, p_{discharge})} \tag{2}$$

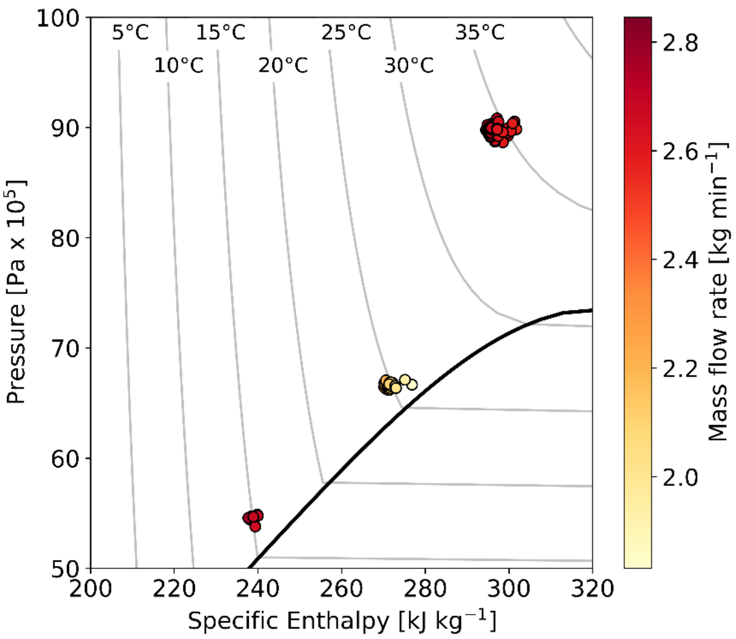

Figure 3: Motive nozzle inlet conditions and mass flow rate.

Due to the supersonic flow conditions at the motive nozzle outlet, for which neither suction pressure nor pressure lift can influence the motive nozzle mass flow rate (Banasiak et al., 2015), the motive nozzle experimental points, achieved according to the test matrix described in Table 2, are presented only as a function of the motive inlet conditions in Figure 3, which also reports the mass flow rate at the ejector motive nozzle. It can be observed that the three desired motive nozzle conditions are accurately achieved during the experimental tests and that, for a specific gas cooler outlet condition, the experimental points present very limited variations of the mass flow rate. The average motive mass flow rates are reported in Table 3 with the respective standard deviation σ.

Table 3: Motive nozzle average experimental mass flow rate.

T_{motive}=35°C, p_{motive}=90 bar	T_{motive}=25°C, p_{motive}=66 bar	T_{motive}=15°C, p_{motive}=54 bar
$\dot{m}_{motive}$=2.688 kg min^{-1} (σ=0.063 kg min^{-1})	$\dot{m}_{motive}$=2.173 kg min^{-1} (σ=0.111 kg min^{-1})	$\dot{m}_{motive}$=2.736 kg min^{-1} (σ=0.056 kg min^{-1})

Differently from the ejector motive mass flow rate, the suction nozzle mass flow rate is a function of more than two independent parameters, since it is strongly dependent on the expansion energy provided by the motive mass flow rate, on the suction nozzle inlet conditions and on the discharge pressure level (directly linked to the pressure lift requirement to be provided by the ejector). As it was described in Section 3.1, the superheat at the ejector suction nozzle was maintained by the evaporators feeding valves in the range of 8-10 K. According to Banasiak et al. (2015), within a limited range (from 0 K to 10 K) the influence of superheating on the ejector performance is barely measurable and, since all of the experimental points fall under these conditions, the effect of superheating at the suction nozzle will not be considered in this study. The ejector mass entrainment ratio and the ejector efficiency are therefore presented in Figures 4 to 6 as a function of the motive nozzle conditions (grouped as the three desired motive conditions), of the suction pressure and of the required pressure lift.

As it can be observed from Figure 4, for a gas cooler outlet temperature of 35°C, the experimental results show a wide range of possible operating points for the ejector. Firstly, it can be observed that the ejector is able to entrain mass flow rate even from the lowest suction pressure condition, corresponding to a saturation temperature of -25°C. This means that, considering the unit concept presented in Figure 1, the LT evaporation can be performed at a pressure level low enough to grant the preservation of an air temperature inside the LT truck compartment of around -20°C. Operation in transient conditions, i.e. at the system start or during a pulldown, is ensured as well by the results at other suction pressure conditions. The ejector mass entrainment ratio decreases as the pressure lift requirement increases, with the exception of the experimental points at 16.8 bar and 19.7 bar, which present a slight increase for low pressure lifts and, after

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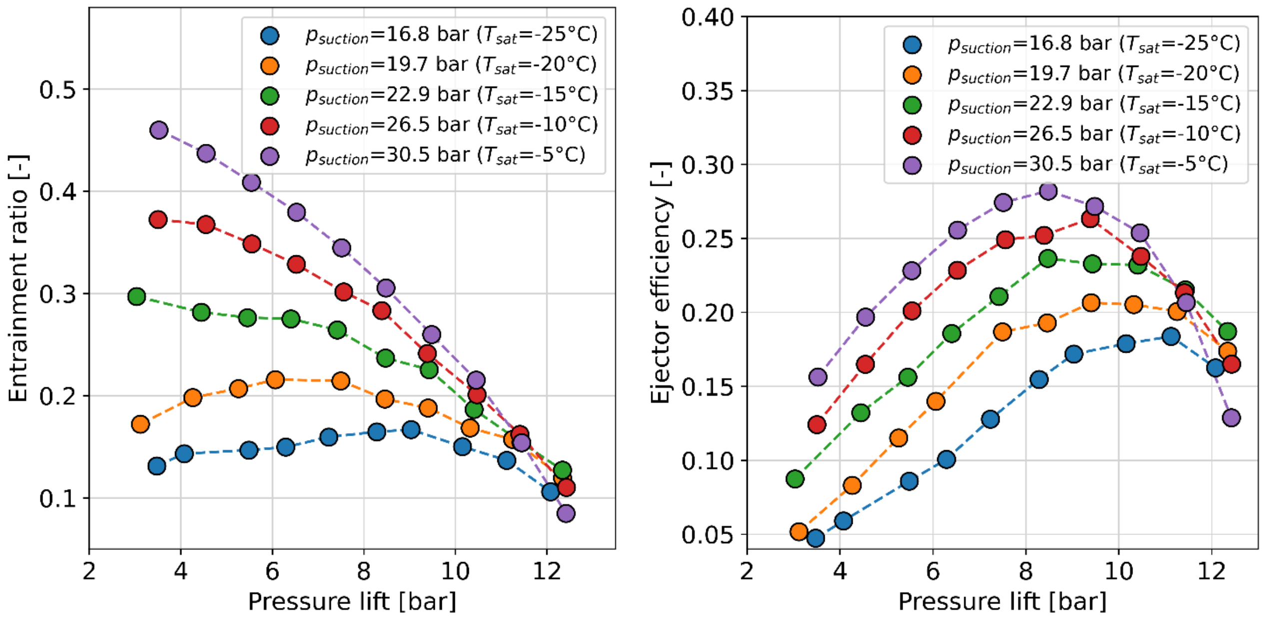
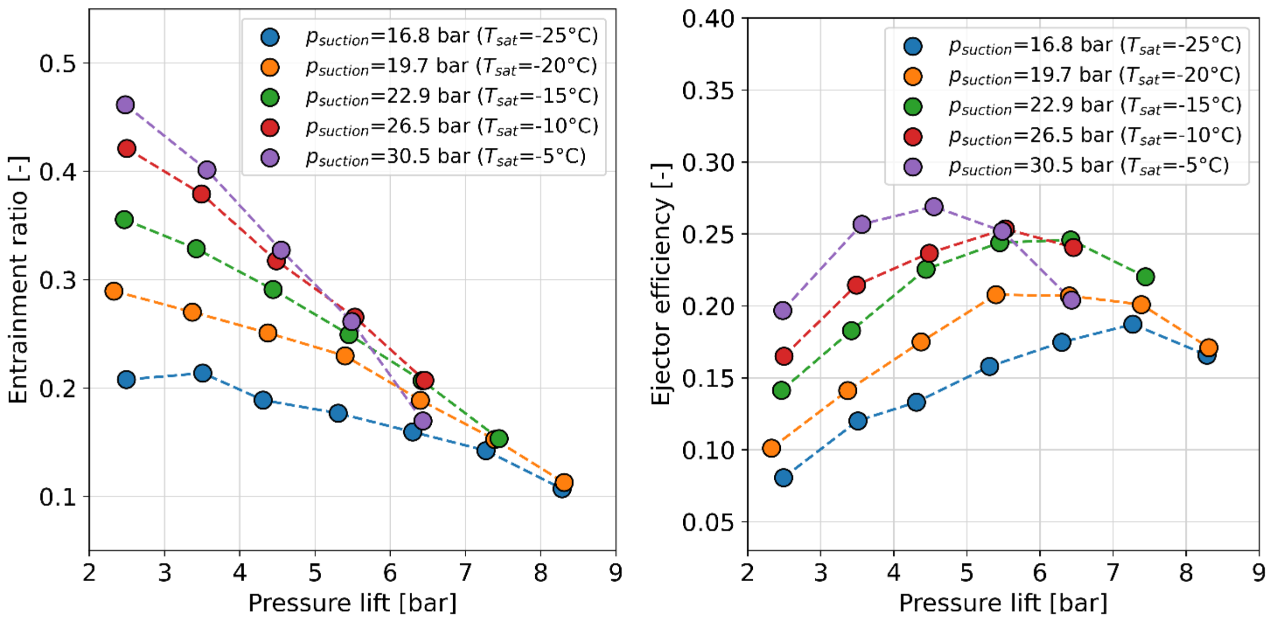
reaching a maximum, start decreasing for high pressure lift requirements. On the contrary, the ejector efficiency shows a clear parabolic trend for each of the suction pressure conditions.



(a) (b)

Figure 4: Performance of the ejector with motive conditions equal to T_{motive} = 35°C, p_{motive} = 90 bar: (a) Ejector entrainment ratio; (b) Ejector efficiency.

Figure 5 presents the entrainment ratio and the ejector efficiency for a gas cooler outlet temperature of 25°C. It can be observed that the reduced expansion energy available at the ejector motive nozzle is reflected in a reduction of the maximum achievable pressure lift. However, the trend for both the entrainment ratio and the efficiency are the same compared with the trend obtained for the 35°C gas cooler outlet conditions, where in this case also the points at 16.8 bar and 19.7 present an almost monotonic decrease in the entrainment ratio with increasing lift requirements.



(a) (b)

Figure 5: Performance of the ejector with motive conditions equal to T_{motive} = 25°C, p_{motive} = 66 bar: (a) Ejector entrainment ratio; (b) Ejector efficiency.

Figure 6 reports the ejector performance for a gas cooler outlet temperature of 15°C. In such conditions, the available expansion energy at the motive nozzle is so low that the ejector is not able to entrain mass flow

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rate from a suction pressure of 16.8 bar and, even for higher suction pressure conditions, the maximum pressure lift that can be provided by the ejector is very limited and never exceeds a value of approximately 4.5 bar.

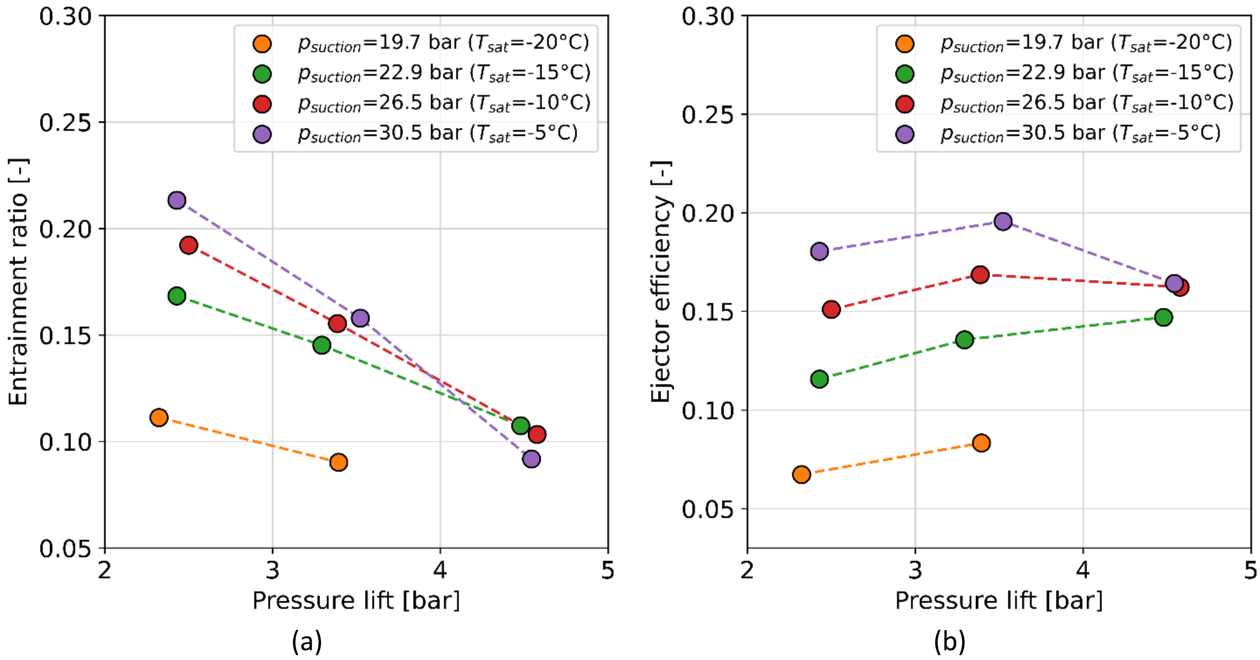

Figure 6: Performance of the ejector with motive conditions equal to T_{motive} = 15°C, p_{motive} = 54 bar: (a) Ejector entrainment ratio; (b) Ejector efficiency.

A few interesting considerations on the ejector performance can be made by looking at the experimental results. Firstly, as it can be expected in normal operating conditions of the ejector, the general behavior is that the performance is higher, both in terms of entrainment ratio and efficiency, for higher suction pressure conditions. However, it can be observed consistently between each of the three motive conditions experimentally tested that, for high values of pressure lift requirement, close to the maximum achievable lift, the general trend can be not respected, and the higher suction pressure experimental points can present performance parameters which are comparable and, in some cases, worse than lower suction pressure points. The ejector performance of higher suction pressure points is significantly degraded towards the maximum pressure lifts. In addition, the maximum achievable lift is reduced for higher suction pressure conditions, as it can be observed in Figure 5, where the maximum lift at 16.8 bar and 19.7 bar suction conditions is around 8.5 bar, 7.5 bar at 22.9 bar suction pressure and 6.5 bar for 26.5 bar and 30.5 bar suction pressure. In conclusion, the experimental results here presented clearly show the importance of a careful choice of the ejector operating point, since its performance is significantly dependent on the pressure lift requirement.

It must be pointed out that the specific ejector considered in this experimental campaign was designed and optimized for very different operating conditions than the ones considered in the application described in this study. Therefore, an even better performance could be reached with a specific ejector design matching the needs of the proposed application.

4.2. Refrigeration unit performance evaluation

After the experimental evaluation of the operation of the ejector in low-pressure suction conditions, a preliminary thermodynamic evaluation of the performance of the system presented in Figure 1 has been conducted. The ejector experimental data at 35°C gas cooler outlet presented in Figure 5 have been used for the assessment of the unit cooling effect production and of its energy consumption under warm ambient conditions. The thermodynamic cycle has been therefore defined according to the experimental values of pressure, temperature and mass flow rate recorded during the tests at a suction pressure of 16.8 bar

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(saturation temperature equal to -25°C), at different lift requirements and with the assumption that the unit compressor, equipped with an inverter, would be able to deliver to the high-pressure side of the system exactly the mass flow rate measured at the ejector motive nozzle during the experiments (no excess mass flow rate expands in the HPV). The compressor isentropic efficiency, required to evaluate the thermodynamic point after the compression, has been evaluated as a function of the compressor pressure ratio from the database of the Dorin CD series (suitable for this kind of applications) supplied by the manufacturer. The conditions at the outlet of the liquid separator are assumed to be saturated vapor (to the compressor) and saturated liquid (to the expansion valve performing an isenthalpic expansion to the LT evaporation pressure). The MT and LT cooling effects are reported in Figure 7 as a function of the experimental pressure lift together with the system COP, defined as:

$$COP = \frac{Q_{MT} + Q_{LT}}{P_{comp}} \tag{3}$$

Based on the ejector experimental data, the system is able to provide up to 4.9 kW of MT cooling and up to 2.0 kW of LT cooling, depending on the desired ejector lift. The overall system COP is maximized (reaching a value of 1.92) at the maximum available lift provided by the ejector, since higher lifts lead to a reduced power draw in the compression section to bring the refrigerant from the MT evaporation pressure level (which increases with the available lift) to the gas cooler high-pressure value. It has to be noticed that the MT cooling effect could be increased without affecting the LT cooling effect production with the selection of an adequate compressor, able to deliver to the high-pressure side (and therefore, through the HPV, to the MT evaporator) more mass flow rate than the one elaborated by the ejector motive nozzle.

As a conclusion, the system design point (i.e. the pressure lift requirement to be provided by the ejector and the compressor size and rotational speed) has to be carefully chosen according to the ejector performance, to the system thermodynamic COP and to the expected MT and LT refrigeration needs for the specific application.

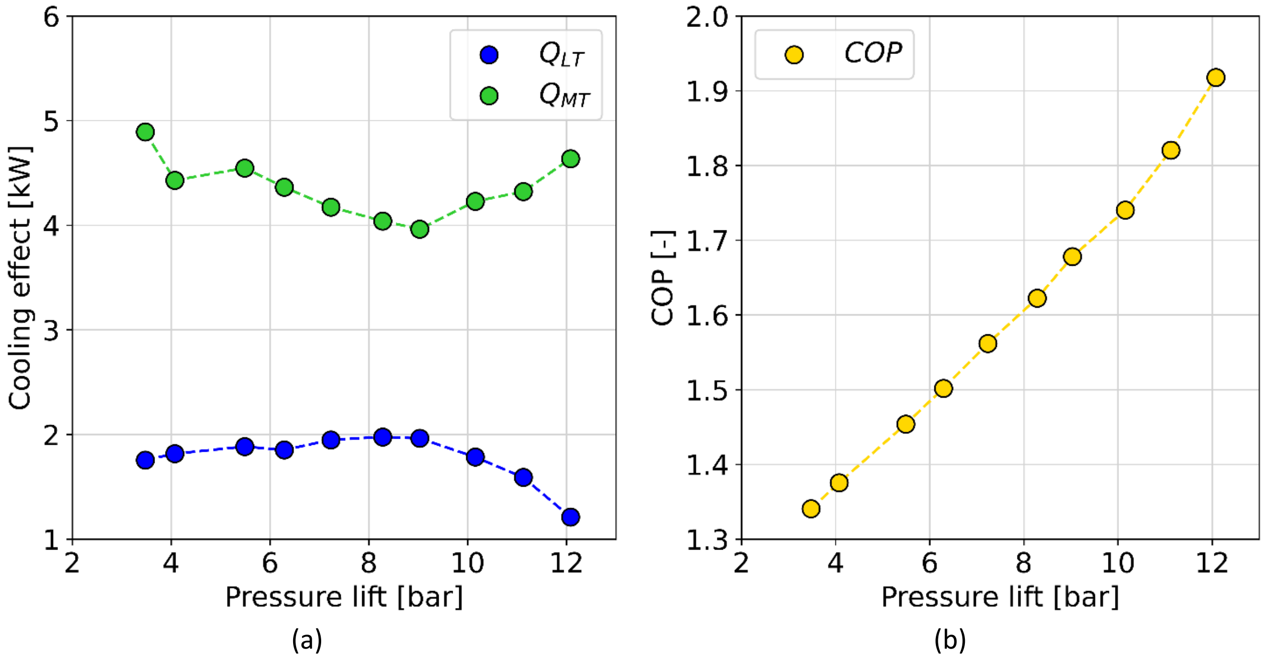

Figure 7: Performance of the refrigeration unit concept presented in Figure 1 at 16.8 bar LT evaporation pressure and 35°C gas cooler outlet: (a) MT and LT cooling effect; (b) COP.

5. CONCLUSIONS

This paper presents the design of a novel CO_2 refrigeration unit conceived to serve the needs of a typical medium-size refrigerated truck used for multi-temperature (MT and LT) goods delivery. In this cooling unit

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concept, an ejector is employed as the only component dedicated to the increase of the refrigerant pressure from the LT to the MT evaporating pressure, thus allowing the removal of the LT subcritical compressor, usually necessary in traditional multi-temperature unit configurations, and enabling operation with the use of only one compressor. An experimental campaign has been carried out on the ejector to evaluate its ability to effectively entrain mass flow rate from the LT evaporating pressure conditions at the suction nozzle (down to 16.8 bar) under three different refrigerant motive nozzle conditions (35°C, 25°C and 15°C). The experimental results demonstrate the ability of the ejector to entrain mass flow rate from very low suction conditions but, at the same time, the crucial influence of the pressure lift requirement on the ejector entrainment ratio and efficiency is highlighted. At an LT evaporation pressure of 16.8 bar (corresponding to a saturation temperature of -25°C) and at 35°C gas cooler outlet, the proposed system is able to provide up to 4.9 kW of MT cooling and up to 2.0 kW of LT cooling, depending on the desired ejector lift. At the maximum available lift, the system COP is equal to 1.92.

NOMENCLATURE

h	specific enthalpy (kJ kg^{-1})	$\dot{m}$	mass flow rate (kg s^{-1})	p	pressure (kPa)
P	power (kW)	Q	cooling effect (kW)	s	specific entropy (kJ kg^{-1} K^{-1})
T	temperature (K)	η	efficiency (-)	ϕ	entrainment ratio (-)

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