A novel R744 multi-temperature cycle for refrigerated transport applications with low-temperature ejector: Experimental ejector characterization and thermodynamic cycle assessment

Nouveau cycle multi-température du R-744 à des fins de transport frigorifique intégrant un éjecteur à basse température : caractérisation expérimentale de l'éjecteur et évaluation du cycle thermodynamique

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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 providing cooling effect at two different temperature levels with only one stage of compression. The ejector was experimentally tested and its ability to effectively entrain mass flow rate from low pressure suction conditions (corresponding to a LT evaporation temperature of $-25\ °C$) was assessed. Lower external ambient temperatures and consequent lower expansion energy available at the ejector motive nozzle leads to a reduction of the maximum achievable pressure lift. Moreover, a significant degradation of the ejector performance towards the highest pressure lifts is experienced. Based on the ejector experimental data, a numerical evaluation of the proposed cooling unit performance has been performed, highlighting that in design conditions (LT evaporation at $-25\ °C$) the cooling unit provides a LT freezing power ranging between 1.1 kW and 2.3 kW and a corresponding minimum MT cooling power ranging between 5.1 kW and 3.8 kW, depending on the chosen ejector lift. The MT cooling power can be further increased by increasing the compressor mass flow rate. The system COP is maximized at the maximum available lift provided by the 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) and that weekly home delivery orders have registered a 38% increase following the COVID-19 outbreak, compared to the pre-pandemic period, leading to an expected additional 41% growth of the home delivery sector in the next five years (Yang et al., 2021).

Traditionally, temperature-controlled logistics was organized 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 (Muyldermans and Pang, 2010; Frank et al., 2021). The multi-compartment approach can lead to significant beneficial effects in

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Nomenclature

COP	coefficient of performance [-]
h	specific enthalpy [kJ kg^{-1}]
$\dot{m}$	mass flow rate [kg s^{-1}]
p	pressure [kPa]
Δp_{lift}	pressure lift [kPa]
P	power [kW]
Q	cooling effect [kW]
s	specific entropy [kJ kg^{-1} K^{-1}]
T	temperature [K]
SH	superheat [K]

Acronyms

FGV	flash gas valve
HPV	high-pressure valve
LT	low temperature
MT	medium temperature

Greek letters

η	efficiency [-]
ϕ	entrainment ratio [-]

Subscripts

comp	compressor
discharge	ejector discharge port
evap	evaporator
gc	gas cooler
max	maximum
min	minimum
motive	ejector motive nozzle
out	outlet
sat	saturation
suction	ejector suction nozzle

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logistics organization. The number of stops at customer locations or collection points, the number of necessary vehicles, and the total duration and mileage of all delivery drives can be significantly reduced (Ostermeier et al., 2021), resulting in significant economic advantages (Heßler, 2021). Besides the economic advantages, a multi-compartment delivery approach is also beneficial from an environmental point of view, as the increase of the fuel consumption and pollutant emissions of refrigerated vehicles compared to non-refrigerated ones has been proven to be significant (Fabris et al., 2022), and multi-temperature transport allows for the increase of flexibility in the logistics, the reduction of the number of vehicles on the road and the consequent improvement of the environmental sustainability of the sector, especially for the last-mile delivery of goods (Taniguchi and Thompson, 2018; Yao et al., 2019; Eshtehadi et al., 2020). In conclusion, the ability to satisfy customers' needs guaranteeing short delivery times, low delivery costs and low emissions of greenhouse gasses and air pollutants has become a very important success factor in the road temperature-controlled delivery sector (Savelsbergh and Van Woensel, 2016).

Multi-temperature transport refrigeration units currently available in the market employ HFC or HFO refrigerants, such as R452A and R404A (Thermo King, 2022; Daikin, 2022; Carrier Transicold, 2022) and are designed to provide a simultaneous cooling effect at a Medium-Temperature (MT) level (e.g. 0 °C, for fresh products) and at a Low-Temperature (LT) level (e.g. −20 °C, for frozen products). However, the approval of the EU F-Gas Regulation 517/2014 (European Commission, 2014) and the consequent progressive ban of commonly used synthetic refrigerants exponentially increased the interest in employing natural refrigerants (in particular carbon dioxide, R744, and hydrocarbons, HCs) in newly developed transport refrigeration units. A complete review on the employment of natural working fluids in refrigerated transport applications can be found in Minetto et al. (2023).

Multi-temperature units employing R744 as the refrigerant are developed and available in the market mostly for commercial stationary applications, as documented by several reviews and research papers available in literature (Gullo et al., 2018; Karampour and Sawalha, 2018; Azzolin et al., 2021; Tsimpoukis et al., 2021). The commonly implemented cycle for multi-temperature R744 stationary units is given by a booster cycle, conceived with double stage compression for LT appliances. The baseline booster cycle can be further modified with the implementation of parallel compression and the use of ejectors providing the lift from the MT evaporation pressure to the liquid receiver intermediate pressure (Gullo et al., 2019). In such a system, a subcritical compressor is needed to increase the refrigerant pressure from the LT evaporation pressure to the suction pressure of the transcritical MT compressors.

Artuso et al. (2020) evaluated the use of two-phase ejector in a mono temperature transport refrigeration unit, highlighting the benefits at high ambient temperature and introducing a dedicated configuration to extend the operations of the ejector. The dynamic behavior of such a cooling unit when employed in a daily long-distance delivery mission in hot European climatic conditions has then been numerically evaluated in Fabris et al. (2021a).

Two-temperature, ejector supported R744 systems are mostly employed for large stationary commercial refrigeration applications. Barta et al. (2021) developed a R744 multi-temperature refrigeration system test stand to compare the effect of different methods of expansion (electronic expansion valve or ejector) and economization at one or both the MT and LT evaporators pressure levels and performed an experimental validation of the test stand operation. Regarding transport refrigeration applications, Lawrence et al. (2020) proposed the design of a multi-temperature refrigerated container employing R744 as the refrigerant and integrating an ejector for expansion energy recovery, demonstrating experimentally the good performance of the system even under extremely harsh environmental conditions (up to 50 °C). On the other hand, Fabris et al. (2021b) numerically evaluated the performance of a R744 ejector supported multi-temperature system for refrigerated vehicles. The Authors employed a Pareto optimality criterion to determine the optimal operational parameters combination to maximize the system Coefficient of Performance (COP) and the MT cooling effect. However, all of the above-mentioned systems still implemented two stages of compression and the ejector was employed in the MT side of the unit to reduce the pressure ratio of the transcritical compressor.

Some experimental studies on the employment of ejectors in low-temperature operating conditions in R290 cooling systems are available in literature (Lin et al., 2019; Lu et al., 2020; Udroiu et al., 2023), demonstrating the improvement of the system efficiency given by the pressure lift provided by the ejector in units employing natural refrigerants. However, very few studies on the employment of R744 ejectors in low-temperature operating conditions can be found in the open literature.

Some studies focused on cascade refrigeration cycles, in which the R744 cycle is used for the LT evaporation stage, are available in literature (Megdouli et al., 2017; Rostamzadeh et al., 2021; Chi et al., 2023). Each of these studies highlight that the implementation of an ejector in the LT cycle can significantly improve both the energy and exergy efficiency of the whole cascade refrigeration system.

Yang et al. (2022a) proposed a novel transcritical R744 two-stage compression/ejector refrigeration system for low-temperature cold storage applications, focusing on the exergy destruction characteristics of the system. Numerical results highlighted that exergy destruction of

the system is primarily contributed to the irreversibility of components themselves and that the system exergetic performance is significantly affected by the efficiency of the ejector and compressors. However, in this study only LT load was considered, and two stages of compression were still required in the refrigeration cycle.

The same authors (Yang et al., 2022b) performed an experimental evaluation of the performance of a R744 two-stage compression cooling system for low-temperature applications (marine transport refrigeration) with and without an ejector, highlighting that the insertion of the ejector can lead to substantial system COP increases compared to a traditional two-stage compression system without the ejector; however, in this case the ejector was placed after the low-pressure compression stage, and therefore operating at an intermediate pressure level and, still, two stages of compression were necessary.

A R744 ejector-expansion refrigeration cycle design was developed with a numerical approach by Peris Pérez et al. (2021), for single low-temperature evaporation applications (from −10 °C to −38.1 °C). The Authors highlighted that the proposed cycle allows 5.5% improvement of the annual average COP compared to a traditional reference cycle with no ejector. However, also in this case, a double stage of compression with intercooler and LT-only load was considered.

To allow the complete removal of the LT compressor from the cooling unit cycle, Wang and Yu (2016) proposed a novel cooling unit schematic in which a small ejector (characterized by a motive nozzle throat diameter variable between 0.6 mm and 1.2 mm) placed after the MT evaporator was used to entrain mass flow rate from the LT evaporator outlet. The ejector was experimentally tested to highlight the influence of the geometrical parameters on the ejector performance. However, the refrigeration system considered in this study employed R600a as the refrigerant.

Bai et al. (2017) considered the single ejector schematic proposed by Wang and Yu (2016), changing the refrigerant from R600a to R744, and proposed an additional novel schematic in which two cascade ejectors are employed to enhance the performance of a dual-temperature refrigeration system, eliminating the LT compressor entirely. Numerical evaluations highlighted that, compared to the single ejector refrigeration cycle, the double ejector system improved the COP by 5.3–25.5%, considering typical evaporation temperatures for supermarket applications and ratio between MT and LT cooling load equal to 1.

A numerical evaluation of two R744 refrigeration unit concepts for supermarket applications, employing ejectors to provide pressure lift after the LT evaporation, thus replacing the whole LT compressor section, has been carried out by Banasiak et al. (2019). Numerical results highlighted that the introduction of the LT ejectors to replace the LT compressors involve a deterioration of the system energy performance due to the pressure lift to be overcome and the use of hot gas as motive flow for the ejector. The Authors claimed that proper design of a dedicated LT ejector geometry is crucial to enable efficient energy performance.

A R744 multi-temperature refrigeration cycle with ejector, subcooler and vapor-injection was presented by Zeng et al. (2022), comparing the performance with a subcooler vapor-injection cycle and an ejector-expansion refrigeration cycle. Evaporating temperatures down to −30 °C were numerically simulated. For gas cooler outlet temperature equal to 35 °C and evaporation temperature of −30 °C, the COP of the proposed cycle in the study temperature range is 26.7% higher than that of subcooler vapor-injection cycle, and 6% higher than that of ejector-expansion cycle, respectively.

Liu et al. (2021) proposed a modified dual-ejector and dual-evaporator transcritical R744 refrigeration cycle for supermarket application. Considering operation with gas cooler outlet temperature equal to 32 °C, MT evaporation temperature of −10 °C and LT evaporation temperature of −30 °C, numerical simulations highlighted that the proposed cycle can reduce the compressor pressure ratio by 19.1% and increase the system COP by 19.3% compared to a traditional

multi-temperature cycle with no ejectors.

The above-mentioned literature studies did not present any experimental study on R744 ejector operating in LT suction conditions. In addition, multi-temperature transport refrigeration applications present a significantly lower cooling demand compared to supermarket applications, resulting in small dimensions requirement for the ejector to be experimentally tested. The present study aims at filling this gap in the open literature, providing an experimental evaluation of a commercially available ejector, with suitable size for the cooling needs of a multi-compartment vehicle, when employed in LT suction conditions.

In this study, a novel R744 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). The ejector experimental results were then used to carry out a numerical evaluation of the performance of the proposed multi-temperature cooling unit in design operating conditions.

2. Refrigeration unit concept

The R744 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 at 0 °C air temperature (MT) and 1–2 kW at −20 °C air temperature (LT) 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. Differently from commonly developed multi-temperature R744 units, in which two stages of compression are needed, 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. The simplified schematic of the refrigeration unit concept is presented in Fig. 1a, while the pressure-specific enthalpy (p-h) diagram of the refrigerant during steady-state operation in conditions representative of the system purpose ($T_{out,gc} = 35$ °C, $T_{evap,LT}$ $T_{evap,LT} = -25$ °C and $\Delta p_{lift} = 9$ bar) is reported in Fig. 1b.

After heat rejection, the refrigerant flows through the ejector motive nozzle to provide the energy required to entrain mass flow rate from the suction nozzle. At this stage of the activity, a fixed geometry ejector was selected, due to its simplicity and robustness, as well as its availability in a relatively small size. Since a fixed-geometry ejector is considered, a high-pressure valve (HPV) is implemented in parallel to adjust the high-pressure according to environmental conditions and cooling load. The two-phase CO_2 stream at the outlet of the HPV-ejector stage is sent to the MT evaporator and then to a liquid separator. The liquid separator acts as a suction accumulator before the MT compressor, while the liquid phase is expanded to the LT evaporator inlet and then entrained by the ejector. Notably, the schematics presented in Fig. 1 also allows 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.

The presented evaluations are referred to the usual needs of a medium-size refrigerated van employed for short-range delivery, 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 experimental campaign
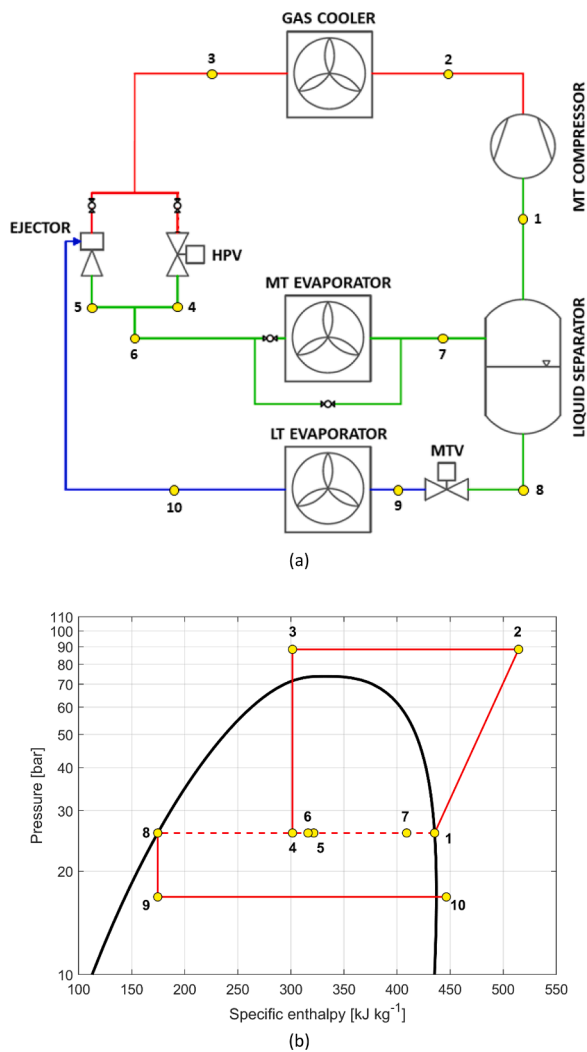
Fig. 1. Refrigeration unit concept: (a) operational schematic; (b) p-h diagram of the system operating points with $T_{out,gc} = 35\ ^{\circ}C$, $T_{evap,LT} = -25\ ^{\circ}C$ and $\Delta p_{lift} = 9$ bar.

and 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

An experimental campaign was carried out to verify whether the ejector is able to provide the required pressure lift from the LT load evaporation pressure (corresponding to a saturation temperature of down to $-25\ ^{\circ}C$) to the MT load evaporation pressure (to be evaluated depending on the available lift provided by the ejector).

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 et al., 2018). A simplified schematic of the unit is provided in Fig. 2, where the dashed lines and components were not

used during this experimental campaign.

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 vapor 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 $1 \bullet 10^{-3}$ m. Further information about the Multi Ejector can be found in Kalinski (2019), while the main geometric parameters of the VEJ1 ejector are reported in Banasiak et al. (2015).

With reference to Fig. 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 requested temperature at the outlet of the gas cooler section. The exceeding mass flow rate, not expanding through the ejector, is expanded in the high-pressure valve (HPV), granting at the same time the high-pressure control. The two-phase CO_2 stream at the outlet of the HPV and ejector discharge port converges into 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 ejector suction 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 evaporators and liquid receiver pressure levels represents the pressure lift provided by the ejector.

During the experimental tests which will be described in Section 4.1, the superheat at the compressors suction has no influence on the operation of the ejector. Referring to the schematic in Fig. 2, the ejector suction corresponds to the outlet of the liquid separator ("LIQ. SEP.") placed after the MT evaporators, whose pressure and superheat are controlled as described above. The part of the mass flow rate which can be entrained by the ejector flows through the ejector suction nozzle, while the remaining mass flow rate is superheated in an internal heat exchanger ("IHX 2″") and then sent to the MT compressors suction. Therefore, the compressors operating conditions did not affect any of the ejector operating parameters during experiments.

The data acquisition for this experimental campaign was based on the high-quality data acquired by the LabVIEW data acquisition system (sampling rate 1 s). The whole refrigerating system is equipped with sensors of various nature, but for this experimental campaign, 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, as reported in Table 1.

3.2. Test conditions

The specific objective of the experimental campaign was to verify ejector operations at relatively low suction nozzle pressures and the consequent constraints and limiting parameters. The selected test conditions are listed in Table 2.
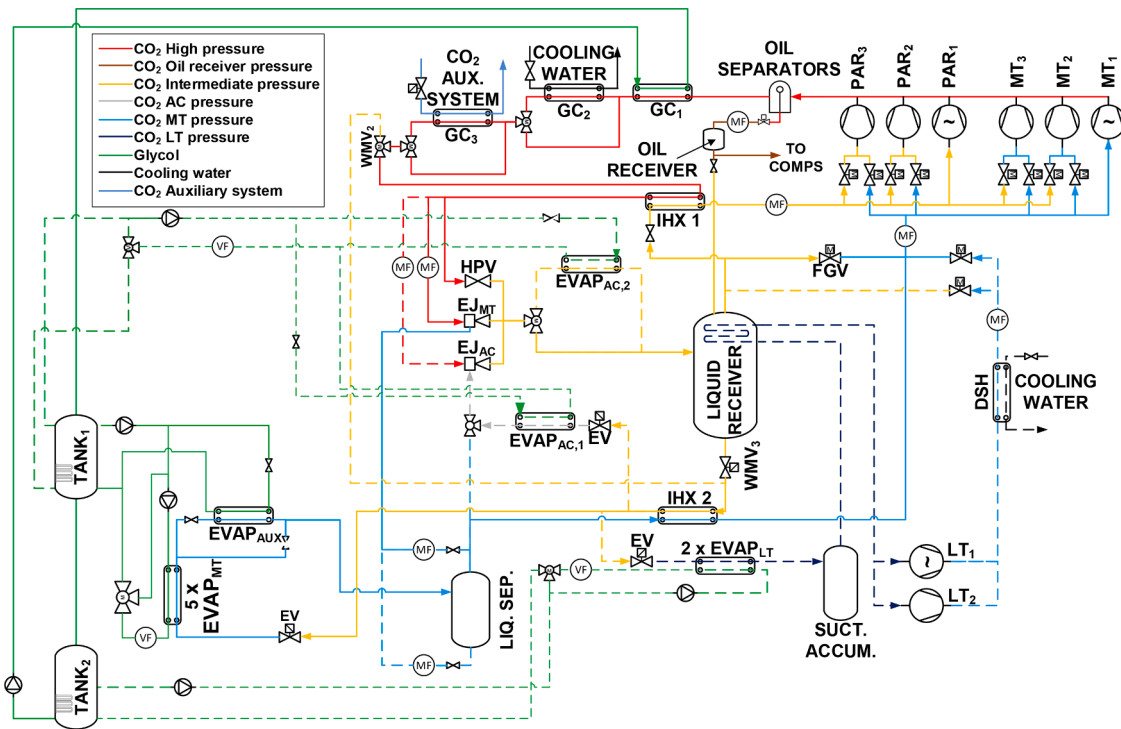
Fig. 2. Simplified schematic of the experimental setup used for the ejector performance tests. Dashed lines and components were not in use during the experiments.

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

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

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

Motive nozzle conditions	Suction nozzle conditions
T_{motive} = 35 °C, p_{motive} = 90 bar	$p_{suction}$ = 16.8 bar (T_{sat} = −25 °C)
T_{motive} = 25 °C, p_{motive} = 66 bar	$p_{suction}$ = 19.7 bar (T_{sat} = −20 °C)
T_{motive} = 15 °C, p_{motive} = 54 bar	$p_{suction}$ = 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 < 10 K

The ejector motive nozzle conditions (pressure and temperature) were selected through the control of the temperature at the outlet of the gas cooler. Once the temperatures at the outlet of the gas cooler were chosen (15 °C, 25 °C and 35 °C, to simulate operation of the ejector under different possible ambient temperature conditions in average European climatic conditions), the optimal gas cooler outlet pressure was set, allowing a slight subcooling (up to 5 K) in subcritical operation or optimizing the experimental system COP for transcritical operation. This approach is in accordance with previous experimental work related to ejector characterization (Banasiak et al., 2015).

Five different saturation pressures at the ejector suction nozzle (corresponding to the saturation temperatures of −25 °C, −20 °C, −15 °C, −10 °C and −5 °C) have been tested in separate moments, but for each single experimental point which will be presented in Section 4 the saturation pressure was fixed and not varying. The lowest suction nozzle pressure tested corresponds to a saturation temperature of −25 °C, corresponding to internal air temperature of −20 °C in the truck compartment. Other suction conditions were tested to evaluate the performance also in conditions typical of pulldown.

For each test condition, different pressure lifts were investigated, increasing the lift with a step equal to $\Delta p_{lift, \, step}$ = 1 bar from a minimum value of approximately 2 bar until the ejector was not able to entrain mass flow rate from the suction nozzle, thus allowing a complete evaluation of the ejector performance and the assessment of its optimal operating points.

3.3. Uncertainty analysis

The mean values of the measurement uncertainties registered during

Table 3
Mean values of the measurement uncertainties registered during the experimental campaign.

Measurement	Uncertainty
Motive pressure, p_{motive}	±0.62 bar
Motive temperature, T_{motive}	±0.34 °C
Motive mass flow rate, $\dot{m}_{motive}$	±0.094 kg min^{-1}
Suction pressure, $p_{suction}$	±0.24 bar
Suction temperature, $T_{suction}$	±0.86 °C
Suction mass flow rate, $\dot{m}_{suction}$	±0.036 kg min^{-1}
Discharge pressure, $p_{discharge}$	±0.16 bar
Pressure lift, Δp_{lift}	±0.21 bar

the experimental campaign, considering both the sensor accuracies and the time-averaged deviations from steady state, are reported in Table 3. Specific enthalpy and specific entropy, necessary for the calculation of the ejector efficiency, were evaluated from the measured values of pressure and temperature through the REFPROP 9.1 database (Lemmon et al., 2013), and their uncertainty was determined according to the procedure described in Aprea et al. (1997).

4. Results

4.1. Experimental evaluation of the ejector performance

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 ($\dot{m}_{suction}$), both measured by a dedicated mass flow meter, or, alternatively, by the ejector mass entrainment ratio, defined as:

$$\phi_{ejector} = \frac{\dot{m}_{suction}}{\dot{m}_{motive}} \tag{1}$$

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):

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

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 Fig. 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 nozzle conditions and mass flow rates are reported in Table 4.

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).

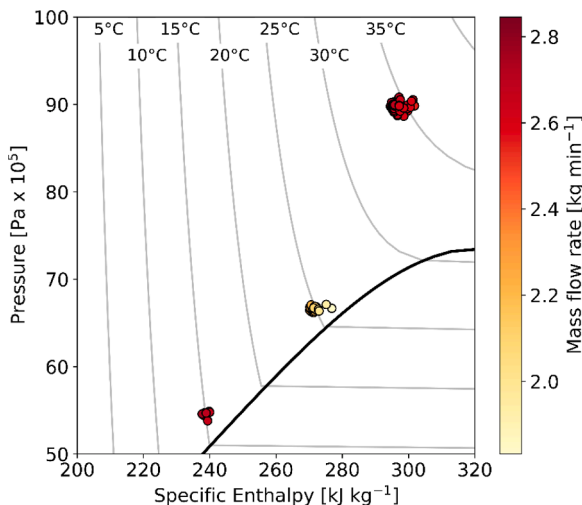As described in Section 3.1, for every experimental point, defined

Fig. 3. Motive nozzle inlet conditions and mass flow rate.

Table 4

– Motive nozzle average experimental mass flow rate.

Desired conditions	Average experimental motive conditions	Average experimental motive mass flow rate
$T_{motive}=35°C$ $p_{motive}=90$ bar	$\bar{T}_{motive}=34.53°C$ $\bar{p}_{motive}=89.70$ bar	$\dot{\bar{m}}_{motive}=2.688$ kg min^{-1}
$T_{motive}=25°C$ $p_{motive}=66$ bar	$\bar{T}_{motive}=24.92°C$ $\bar{p}_{motive}=66.66$ bar	$\dot{\bar{m}}_{motive}=2.173$ kg min^{-1}
$T_{motive}=15°C$ $p_{motive}=54$ bar	$\bar{T}_{motive}=15.05°C$ $\bar{p}_{motive}=54.52$ bar	$\dot{\bar{m}}_{motive}=2.736$ kg min^{-1}

fixing the evaporation saturation pressure and a specific pressure lift requirement, the condition of superheat at ejector suction $SH_{suction} < 10$ K was verified. Flooded operation of the evaporator was never experienced ($SH_{suction} > 0$ 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 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 Figs. 4 – 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. The average measurement uncertainties are equal to ±0.017 for the ejector mass entrainment ratio and ±0.074 for the ejector efficiency.

As it can be observed from Fig. 4, for $T_{motive} = 35 °C$ and $p_{motive} = 90$ bar 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 Fig. 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 monotonically as the pressure lift increases, with the exception of the data series at $p_{suction} = 16.8$ bar and 19.7 bar, which present a maximum entrainment ratio for intermediate pressure lifts. On the contrary, the ejector efficiency shows a regular parabolic trend for all the suction pressure conditions.

Fig. 5 presents the entrainment ratio and the ejector efficiency for $T_{motive} = 25 °C$ and $p_{motive} = 66$ bar. The reduced expansion energy available at the ejector motive nozzle reduces the achievable pressure lifts with respect to the previous data set. Conversely from the data set at $T_{motive} = 35 °C$, all the data series are monotonically decreasing with increasing pressure lift.

Fig. 6 reports the ejector performance for $T_{motive} = 15 °C$ and $p_{motive} = 54$ bar. In such conditions, the available expansion energy at the motive nozzle is so low that the ejector is not able to entrain mass flow rate from the lowest suction pressure ($p_{suction} = 16.8$ bar). Positive entrainment ratios are achievable for $p_{suction} = 19.7$ bar, but with negligible efficiencies. 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.

In case of low ambient temperature conditions, therefore, the desired temperature of the air inside the LT truck compartment (−20 °C) cannot be achieved employing this ejector cartridge in a standard ejector cycle.

The experimental results confirm that the maximum entrainment ratio and efficiency are obtained for higher suction pressure conditions. However, it can be observed consistently between each of the three motive conditions that this is not true in terms of the maximum achievable lift. Experimental data suggest, in fact, that the higher suction pressure data series are subject to a more intense degradation of the ejector performance once the maximum efficiency point is surpassed.
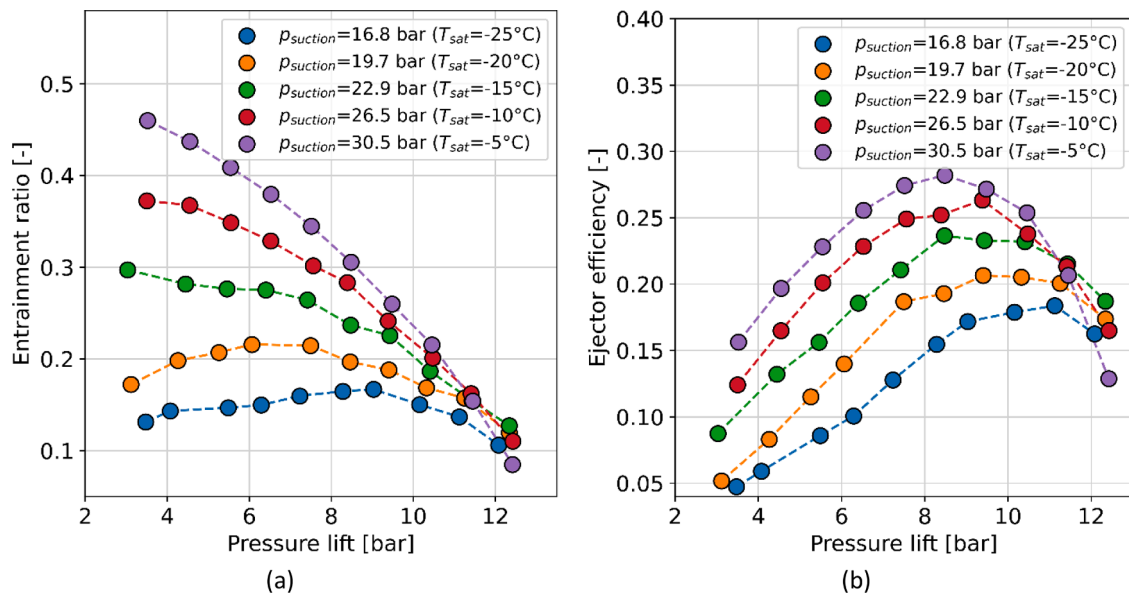
Fig. 4. Performance of the ejector with motive conditions equal to $T_{\mathrm{motive}} = 35\ ^\circ\mathrm{C}$, $p_{\mathrm{motive}} = 90$ bar: (a) Ejector entrainment ratio; (b) Ejector efficiency.
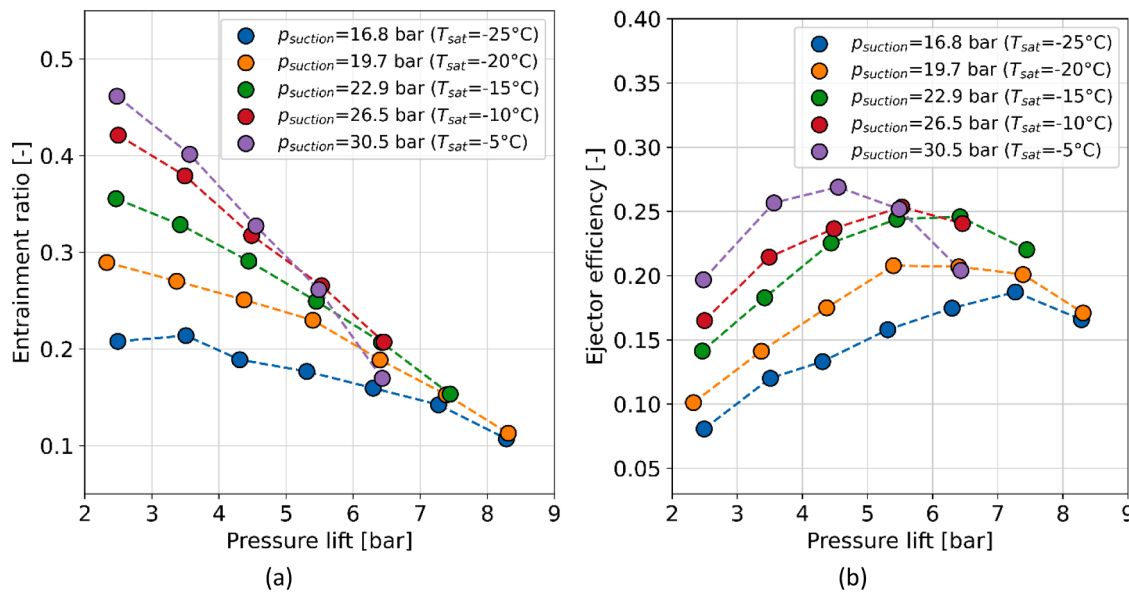
Fig. 5. Performance of the ejector with motive conditions equal to $T_{\mathrm{motive}} = 25\ ^\circ\mathrm{C}$, $p_{\mathrm{motive}} = 66$ bar: (a) Ejector entrainment ratio; (b) Ejector efficiency.

Despite all the data series are interrupted approximately at $\phi_{ejector} \sim$ 0.10–0.15, both the entrainment ratio-pressure lift curves and the efficiency-pressure lift curves suggest that the high suction pressure curves lead to lower maximum pressure lifts for $\phi_{ejector} = 0$.

It must be pointed out that the specific ejector considered in this experimental campaign was originally designed and optimized as a high pressure lift MT vapor ejector. Therefore, this experimental campaign, conducted at LT suction temperatures, is useful to set the minimum expectable performances in these conditions, while better performances could be certainly obtained with an ejector geometry designed specifically for the proposed application.

The complete ejector experimental data used in this section are available in Fabris et al. (2023).

4.2. Numerical evaluation of the refrigeration unit performance

After the experimental characterization of the ejector in LT suction

conditions, a thermodynamic evaluation of the performance of the system presented in Fig. 1 has been conducted, to numerically assess which would be the performance of the proposed cooling unit if the tested ejector was employed. In this way, a preliminary assessment of such a unit operation can be achieved, to determine whether the MT and LT cooling effects production could match the design needs for the specific application.

The ejector experimental data presented in Section 4.1 were used to model the ejector performance. In particular, the thermodynamic cycle evaluation is carried out according to the following procedure:

• The experimental motive nozzle conditions were used to set the ejector motive mass flow rate and the pressure and temperature at the outlet of the gas cooler. As for $T_{\mathrm{motive}} = 15\ ^\circ\mathrm{C}$ the ejector was not able to entrain mass flow rate from $p_{\mathrm{suction}} = 16.8$ bar (corresponding to the design LT evaporation temperature of $-25\ ^\circ\mathrm{C}$), only data sets for $T_{\mathrm{motive}} = 25\ ^\circ\mathrm{C}$ and $T_{\mathrm{motive}} = 35\ ^\circ\mathrm{C}$ were considered.

Fig. 6. Performance of the ejector with motive conditions equal to $T_{\text{motive}} = 15\ °\text{C}$, $p_{\text{motive}} = 54$ bar: (a) Ejector entrainment ratio; (b) Ejector efficiency.
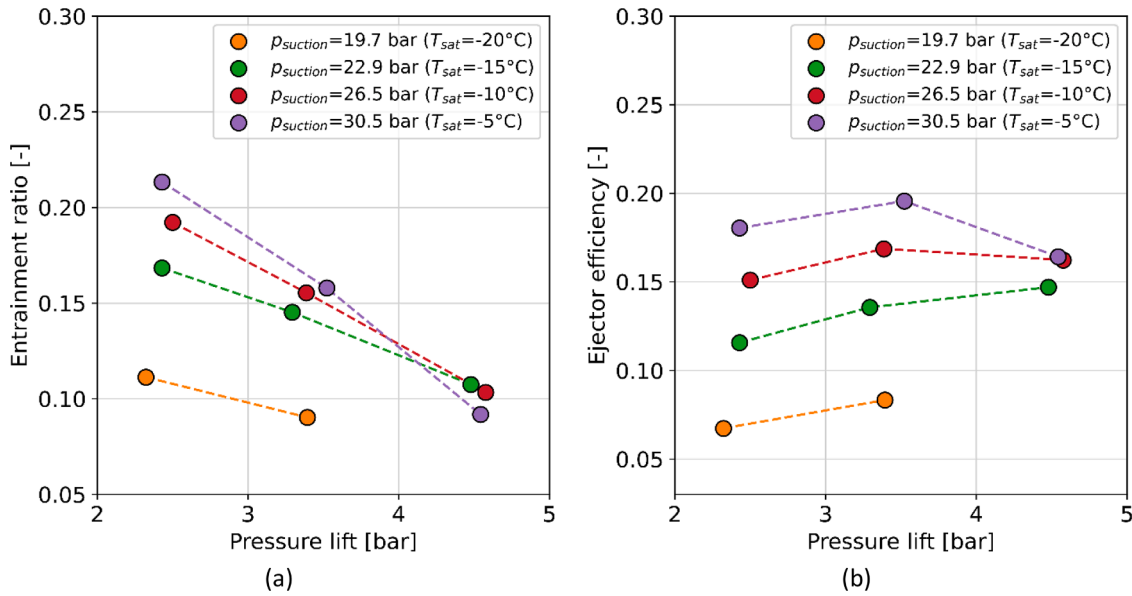
- The corresponding experimental ejector mass entrainment ratio registered the suction pressure of $p_{\text{suction}} = 16.8$ bar was used to determine the refrigerant mass flow rate in the LT evaporator. The superheat at the ejector suction was set according to the experimental one.

- The intermediate pressure in the MT evaporator and in the liquid separator was set equal to the experimental ejector discharge pressure. The mass flow rate in the MT evaporator was calculated as the sum of the mass flow rates at the ejector motive nozzle, ejector suction nozzle and, when used, at the high-pressure valve (HPV).

- The liquid separator was considered to have ideal performances (saturated liquid to the expansion valve before the LT evaporator and saturated vapor to the compressor suction). While the absence of superheating at the compressor suction can pose a problem in the actual system prototyping and will certainly be addressed in further work, for the numerical evaluation carried out in this study no internal heat exchanger was included in the unit schematic, in order to present the cycle layout and performances as clear as possible.

- The compressor isentropic efficiency was evaluated as a function of the compressor pressure ratio from the database of a compressor suitable for this kind of applications (Dorin, 2022), supplied by the manufacturer. The system operation directly resulting from the ejector experimental test points correspond to the condition in which the compressor elaborates only the mass flow rate experimentally registered at the ejector motive nozzle points. In this condition the ejector bypass HPV is closed ($\dot{m}_{HPV} = 0$). Increasing the mass flow rate elaborated by the compressor ($\dot{m}_{HPV} = 0.1\ \dot{m}_{motive}$, $\dot{m}_{HPV} = 0.2\ \dot{m}_{motive}$, …) results in expanding the exceeding mass flow rate in the HPV (as the ejector cannot elaborate more mass flow rate at the motive nozzle), leading to a corresponding increase of the MT cooling effect up to $Q_{\text{MT}} = 6$ kW without affecting the ejector operation and, consequently, the LT cooling effect.

- The cooling effect in the heat exchangers and the compressor power draw were then determined given the mass flow rates and cycle thermodynamic points calculated as described above.

Therefore, in the following numerical analysis no numerical prediction of the performance of the ejector is performed, as pressures and mass flow rates are fixed to the values obtained during the experimental campaign conducted on the ejector. The numerical evaluation of the refrigeration cycle performance only involves the calculation of the compressor power draw according to efficiency data and operating conditions and the assumptions of ideal operation of the liquid separator and isenthalpic expansion in the MTV.

The MT and LT cooling power achievable combinations are reported in Fig. 7. Colorbar is used to map the corresponding system COP, defined as:

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

In order to define these maps, when the same operating point (Q_{LT}, Q_{MT}) was realizable with more than one pressure lift, only the one leading to the higher cycle COP was considered. As a result, only the monotonic decreasing part of the $\phi_{\text{ejector}} - \Delta p_{\text{lift}}$ resulted in the best performing points, since higher pressure lifts lead to reduced compressor power draws.

Based on the ejector experimental data, the system can provide a LT cooling effect ranging between 1.1 kW and 2.3 kW, corresponding to a minimum MT cooling effect ranging between 5.1 kW and 3.8 kW respectively, depending on the chosen ejector lift. The increase of ejector lift (and the corresponding reduction of the entrainment ratio) reduces the system LT cooling power and increase the minimum MT cooling power. MT cooling power can be freely controlled above the minimum value, by increasing the compressor mass flow rate. The flow rate through the HPV valve is represented in Fig. 7 by means of isolines, normalized as a fraction of the ejector motive mass flow rate.

The system COP do not directly depend on $Q_{\text{MT}}/Q_{\text{LT}}$ ratio, but only on the ejector lifts (marked using dashed lines). This is clear observing that the overall cooling power $Q_{\text{MT}} + Q_{\text{LT}}$ can be assessed applying the first law of thermodynamics between the compressor suction and the gas cooler outlet:

$$Q_{\text{MT}} + Q_{\text{LT}} = \dot{m}_1(h_1 - h_3) \tag{4}$$

On the other side, the ejector pressure lift (at fixed LT evaporation pressure) has a direct impact on the system COP as it increases the intermediate pressure, thus reducing then the compressor work.

5. Conclusions

This paper presents the design of a novel R744 refrigeration unit conceived to serve the needs of a typical medium-size refrigerated truck for multi-temperature (MT and LT) goods delivery. An ejector is employed as the only component dedicated to the increase of the
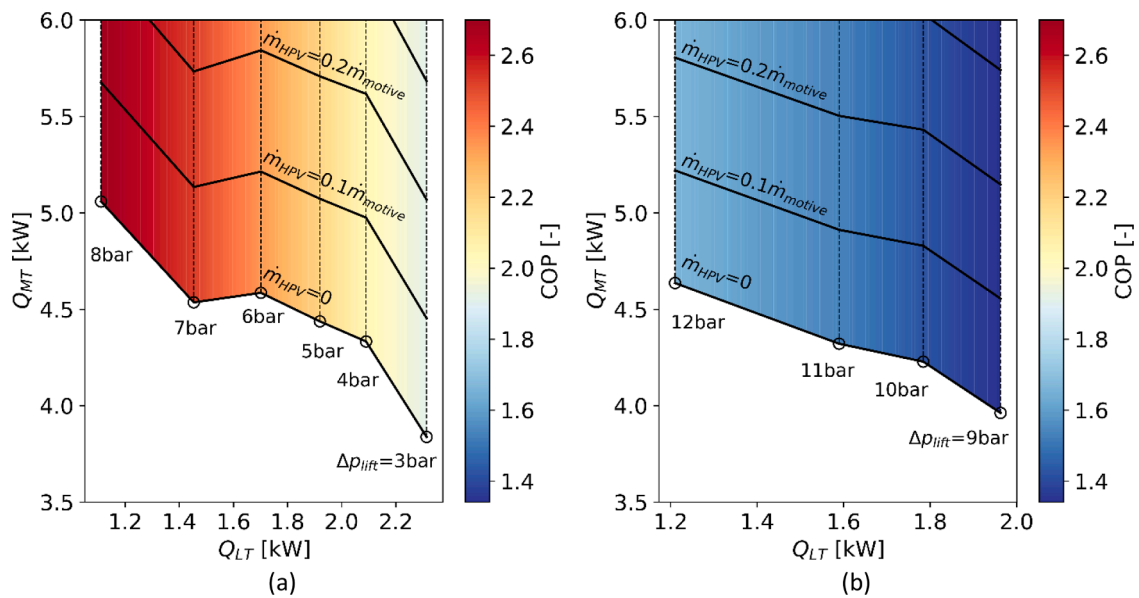
Fig. 7. Performance of the refrigeration unit concept presented in Fig. 1 for $p_{LTevap} = 16.8$ bar and different gas cooler outlet conditions: a)$T_{motive} = 25\,°C$, $p_{motive} = 66$ bar; b) $T_{motive} = 35\,°C$, $p_{motive} = 90$ bar.

refrigerant pressure from the LT to the MT evaporating pressure, thus allowing the complete removal of the LT subcritical compressor and enabling multi-temperature operation with the use of only one compressor.

An experimental campaign has been carried out to evaluate the feasibility of such design, testing an existing high pressure lift MT vapor ejector at LT suction conditions under three different refrigerant motive nozzle conditions ($T_{gc,\ out} = 35°C$, $T_{gc,\ out} = 25°C$ and $T_{gc,\ out} = 15°C$).

The experimental results demonstrated the capability of the tested ejector to provide acceptable efficiencies at the lowest suction pressure, down to 16.8 bar ($T_{sat} = -25\,°C$), for $T_{gc,\ out} = 35°C$ and $T_{gc,\ out} = 25°C$. These experimental data were then used to compute the multi-temperature system thermodynamic performance.

At the design LT evaporation pressure of 16.8 bar (corresponding to a saturation temperature of $-25\,°C$) the proposed cooling unit is able to provide a LT cooling effect ranging between 1.1 kW and 2.3 kW, depending on the chosen ejector lift. The overall system COP is maximized (reaching a value of 1.92 for $T_{gc,\ out} = 35°C$ and 2.70 for $T_{gc,\ out} = 25°C$) at the maximum available lift provided by the ejector for each gas cooler outlet condition, due to the reduced compressor pressure ratio.

To guarantee the system equilibrium a minimum MT cooling load is required when the LT evaporator is working, ranging from 3.8 kW to 5.1 kW. On the other side, the MT cooling power is not inherently limited by the ejector characteristics and can be increased by increasing the compressor mass flow rate.

In conclusion, while the ejector considered in the experimental campaign was not originally designed for LT operating conditions, the present study allowed to demonstrate the feasibility of the presented cooling unit. Furthermore, it highlighted the impact of the ejector characteristic on the system performance, allowing to focus on the development of ejector geometries designed to match the needs of the proposed application.

Declaration of Competing Interest

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests:

Silvia Minetto reports financial support was provided by European Union.

Supplementary materials

Supplementary material associated with this article can be found, in the online version, at doi:10.1016/j.ijrefrig.2023.05.003.

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