

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

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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. In design conditions (LT evaporation at -25 °C) the proposed 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.

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) 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 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, allowing 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 gases 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.

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. While two-temperature, ejector supported R744 systems are employed for large stationary commercial refrigeration applications, Fabris et al. (2021) numerically evaluated the performance of such a system for refrigerated transport applications. 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, the analysed unit still implemented two stages of compression.

Very few studies on the employment of R744 ejectors in low-temperature operating conditions can be found in the open literature.

Yang et al. (2022) 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.

A R744 ejector-expansion refrigeration cycle design was developed with a numerical approach also 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 $-25\text{ }^{\circ}\text{C}$ LT saturation temperature).

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 $0\text{ }^{\circ}\text{C}$ air temperature (MT) and 1-2 kW at $-20\text{ }^{\circ}\text{C}$ 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 Figure 1a, while the pressure-specific enthalpy (p-h) diagram of the refrigerant during steady-state operation in conditions representative of the system purpose ($T_{\text{out,gc}} = 35\text{ }^{\circ}\text{C}$, $T_{\text{evap,LT}} = -25\text{ }^{\circ}\text{C}$ and $\Delta p_{\text{lift}} = 9\text{ bar}$) is reported in Figure 1b.

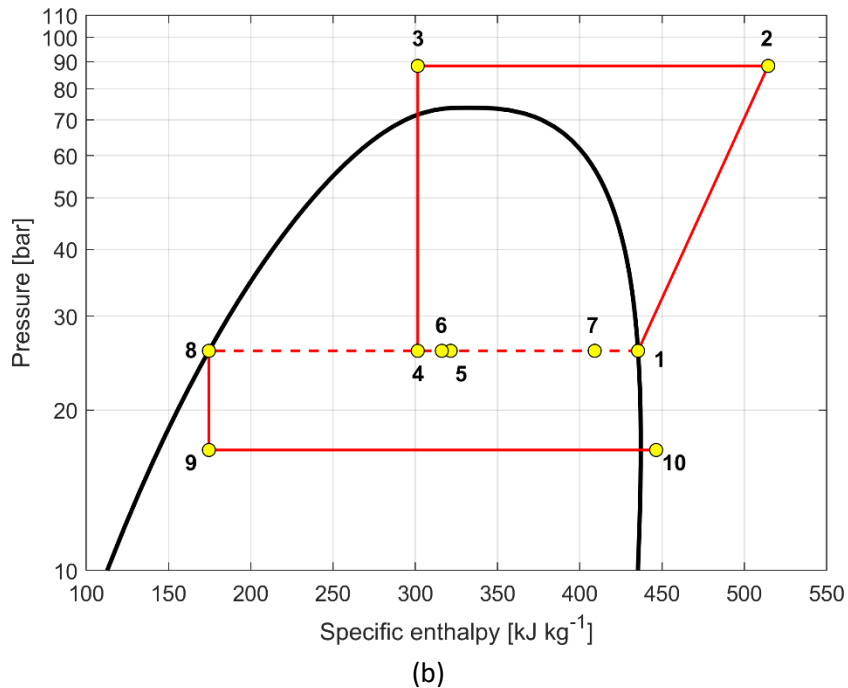
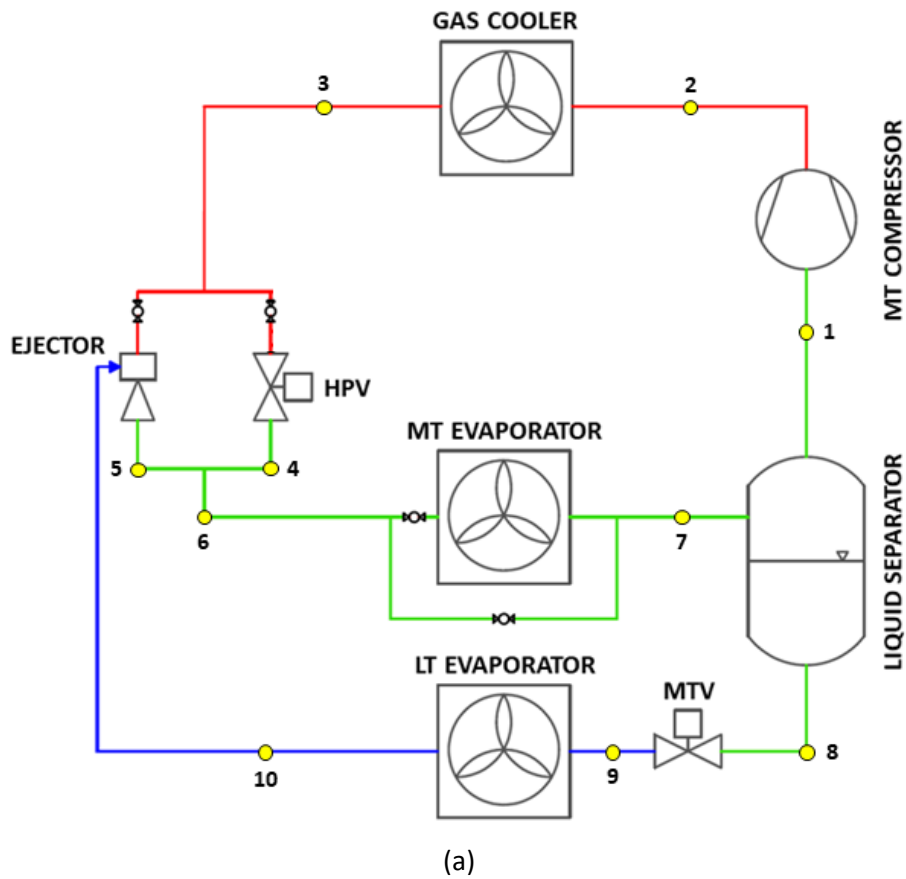


Figure 1 – Refrigeration unit concept: (a) operational schematic; (b) p-h diagram of the system operating points with $T_{out,gc} = 35^{\circ}\text{C}$, $T_{evap,LT} = -25^{\circ}\text{C}$ and $\Delta p_{lift} = 9 \text{ bar}$.

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. Since a fixed-geometry ejector will be considered for this application due to the limited cooling load compared to commercial refrigeration applications, a high-

pressure valve (HPV) is implemented in parallel to adjust the high-pressure according to environmental conditions and cooling load. The two-phase CO₂ 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 Figure 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 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 °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 Figure 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 $1 \cdot 10^{-3}$ 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).

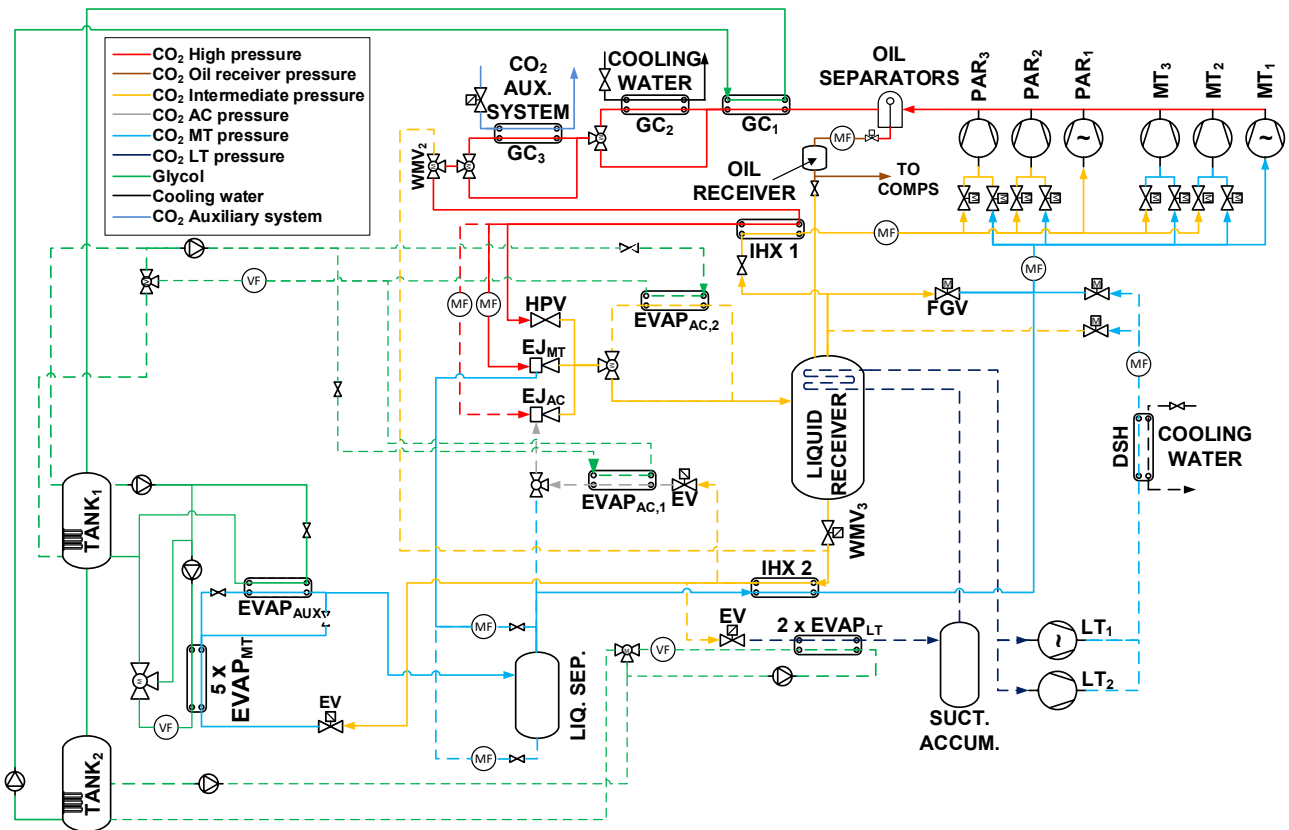


Figure 2 – Simplified schematic of the experimental setup used for the ejector performance tests. Dashed lines and components were not in use during the experiments.

With reference to 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 CO₂ 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 CO₂ 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.

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.

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))

Uncertainties on the measured variables and the error propagation on the derived quantities will be presented in Section 4.1.1 after the discussion of the experimental results.

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.

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

Motive nozzle conditions	Suction nozzle conditions
<ul style="list-style-type: none"> • $T_{\text{motive}} = 35 \text{ }^{\circ}\text{C}$, $p_{\text{motive}} = 90 \text{ bar}$ • $T_{\text{motive}} = 25 \text{ }^{\circ}\text{C}$, $p_{\text{motive}} = 66 \text{ bar}$ • $T_{\text{motive}} = 15 \text{ }^{\circ}\text{C}$, $p_{\text{motive}} = 54 \text{ bar}$ 	<ul style="list-style-type: none"> • $p_{\text{suction}} = 16.8 \text{ bar}$ ($T_{\text{sat}} = -25 \text{ }^{\circ}\text{C}$) • $p_{\text{suction}} = 19.7 \text{ bar}$ ($T_{\text{sat}} = -20 \text{ }^{\circ}\text{C}$) • $p_{\text{suction}} = 22.9 \text{ bar}$ ($T_{\text{sat}} = -15 \text{ }^{\circ}\text{C}$) • $p_{\text{suction}} = 26.5 \text{ bar}$ ($T_{\text{sat}} = -10 \text{ }^{\circ}\text{C}$) • $p_{\text{suction}} = 30.5 \text{ bar}$ ($T_{\text{sat}} = -5 \text{ }^{\circ}\text{C}$) • $SH < 10 \text{ K}$

The three motive nozzle inlet conditions were selected as representative of the average European climatic conditions. The lowest suction nozzle pressure tested corresponds to a saturation temperature of $-25 \text{ }^{\circ}\text{C}$, corresponding to internal air temperature of $-20 \text{ }^{\circ}\text{C}$ 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 $\Delta p_{\text{lift,step}} = 1 \text{ bar}$ 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.

4. RESULTS

4.1 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:

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

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):

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

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 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 nozzle conditions and mass flow rates are reported in Table 3.

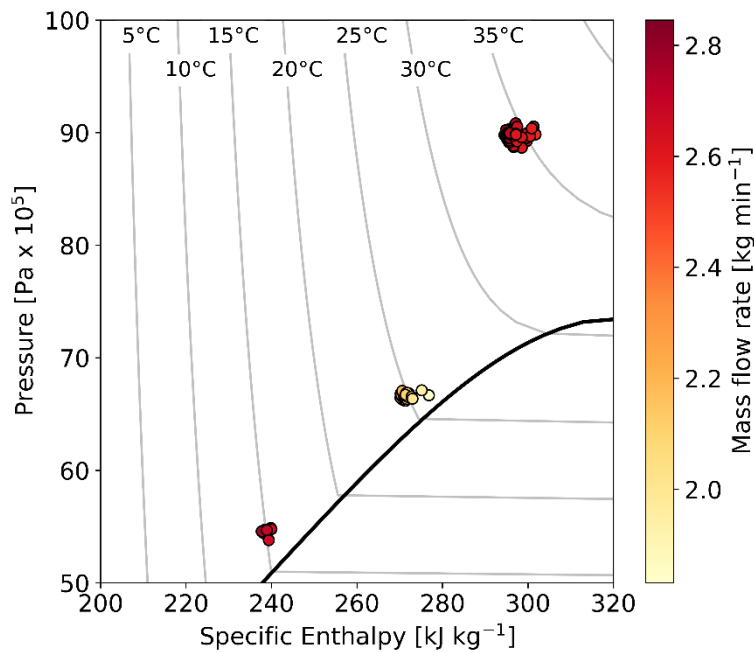


Figure 3 – Motive nozzle inlet conditions and mass flow rate.

Table 3 – Motive nozzle average experimental mass flow rate.

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

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 described in Section 3.1, the superheat at the ejector suction nozzle was maintained under 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 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 $T_{\text{motive}} = 35 \text{ }^\circ\text{C}$ and $p_{\text{motive}} = 90 \text{ bar}$ 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 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 monotonically as the pressure lift increases, with the exception of the data series at $p_{\text{suction}} = 16.8 \text{ bar}$ 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.

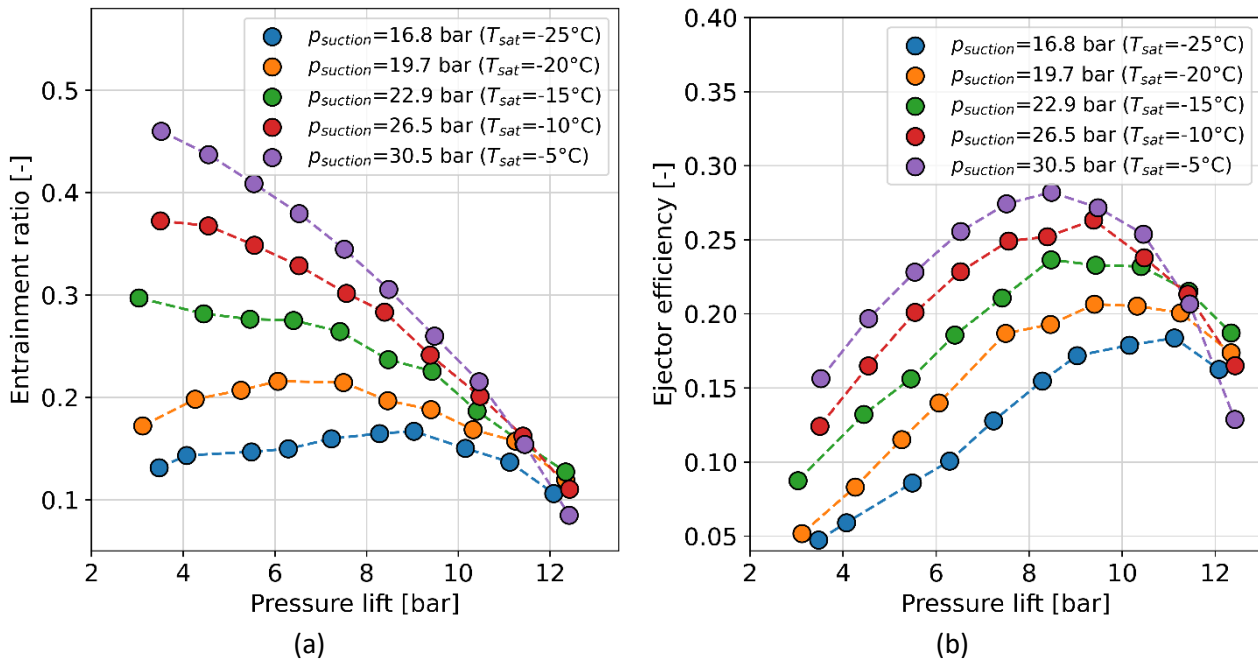


Figure 4: Performance of the ejector with motive conditions equal to $T_{\text{motive}} = 35 \text{ }^\circ\text{C}$, $p_{\text{motive}} = 90 \text{ bar}$: (a) Ejector entrainment ratio; (b) Ejector efficiency.

Figure 5 presents the entrainment ratio and the ejector efficiency for $T_{\text{motive}} = 25 \text{ }^\circ\text{C}$ and $p_{\text{motive}} = 66 \text{ bar}$. 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 $T_{motive} = 35\text{ °C}$, all the data series are monotonically decreasing with increasing pressure lift.

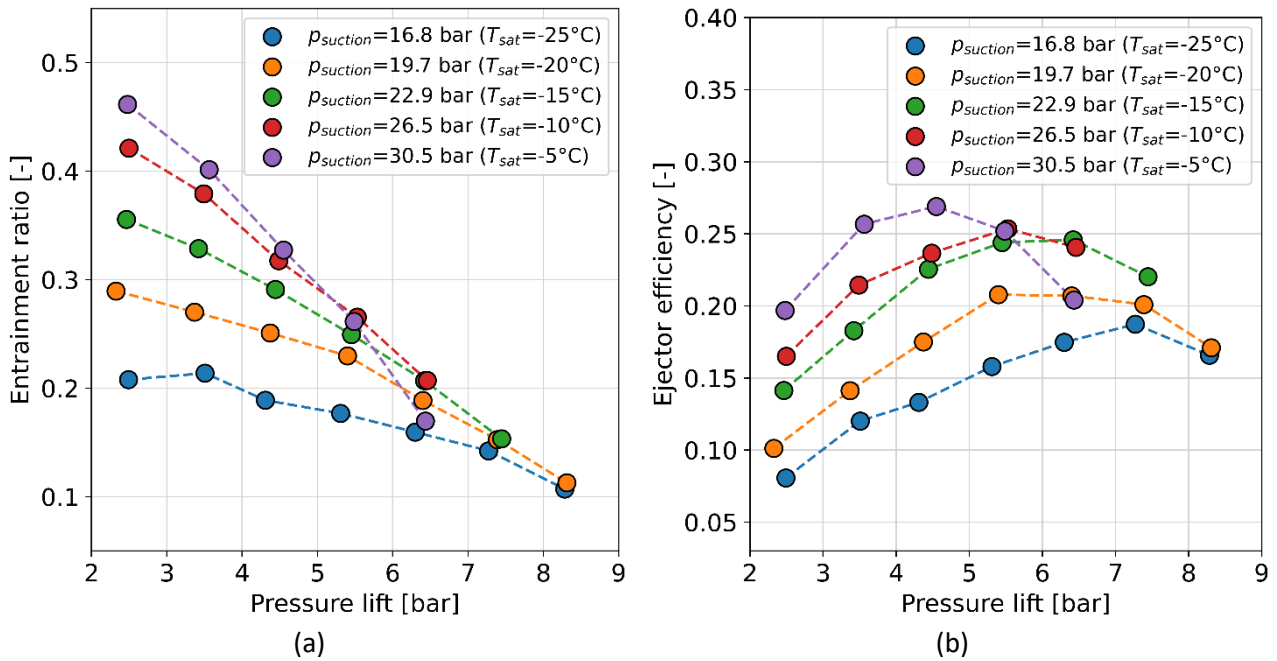


Figure 5: Performance of the ejector with motive conditions equal to $T_{motive} = 25\text{ °C}$, $p_{motive} = 66\text{ bar}$: (a) Ejector entrainment ratio; (b) Ejector efficiency.

Figure 6 reports the ejector performance for $T_{motive} = 15\text{ °C}$ and $p_{motive} = 54\text{ bar}$. 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 ($p_{suction} = 16.8\text{ bar}$). Positive entrainment ratios are achievable for $p_{suction} = 19.7\text{ bar}$, 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.

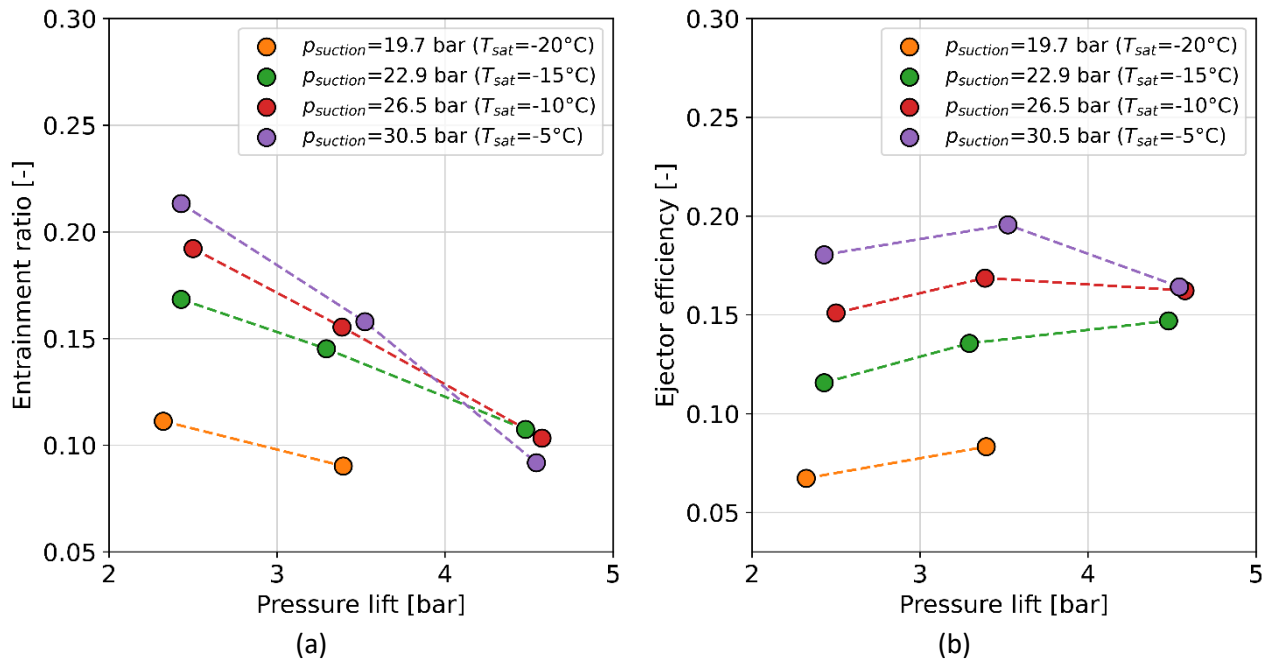


Figure 6: Performance of the ejector with motive conditions equal to $T_{motive} = 15^{\circ}\text{C}$, $p_{motive} = 54$ bar: (a) Ejector entrainment ratio; (b) Ejector efficiency.

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. 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.

4.1.1 Uncertainty analysis

The mean values of the measurement uncertainties registered during the experimental campaign, considering both the sensor accuracies and the time-averaged deviations from steady state, are reported in Table 4. 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).

Table 4 – 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, $p_{\text{discharge}}$	± 0.16 bar
Pressure lift, Δp_{lift}	± 0.21 bar
Ejector mass entrainment ratio, ϕ_{ejector}	± 0.017
Ejector efficiency, η_{ejector}	± 0.074

4.2 Refrigeration unit performance evaluation

After the experimental characterization of the ejector in LT suction conditions, a thermodynamic evaluation of the performance of the system presented in Figure 1 has been conducted.

The ejector experimental data were used to model the ejector performance. In particular, the design LT evaporation temperature of -25°C was chosen, corresponding to an ejector suction pressure of $p_{\text{suction}} = 16.8$ bar. Superheat at the ejector suction was set according to the experimental one. As for $T_{\text{motive}} = 15^\circ\text{C}$ the ejector was not able to entrain mass flow rate from $p_{\text{suction}} = 16.8$ bar, only data sets for $T_{\text{motive}} = 25^\circ\text{C}$ and $T_{\text{motive}} = 35^\circ\text{C}$ were considered.

The intermediate pressure was obtained from the experimental LT pressure and ejector lift. 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).

The compressor isentropic efficiency has been 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. For each point the minimum flow rate considered was the one required to choke the motive nozzle, as measured during the experimental campaign. In this condition the ejector bypass HPV is closed. Compressor flow rate was then increased ($\dot{m}_{\text{HPV}} > 0$) to map the system performance up to $Q_{\text{MT}} = 6$ kW.

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

$$COP = \frac{Q_{\text{MT}} + Q_{\text{LT}}}{P_{\text{comp}}} \quad (3)$$

In order to define these maps, when the same operating point ($Q_{\text{LT}}, Q_{\text{MT}}$) 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.

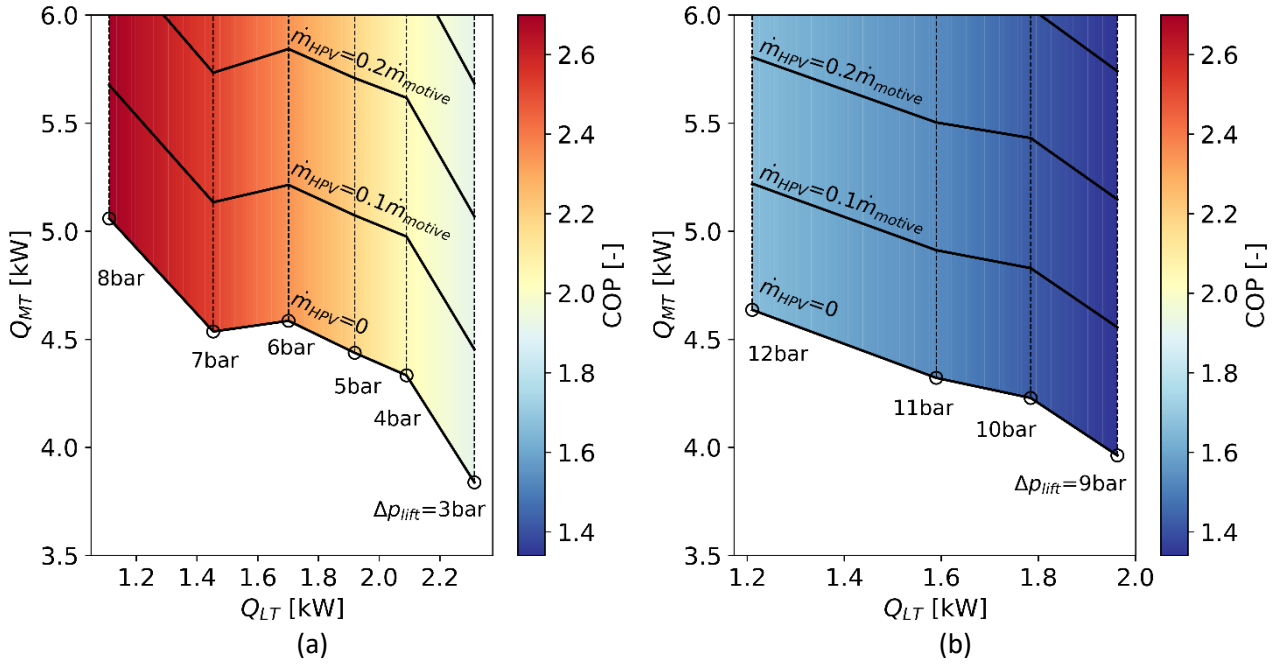


Figure 7: Performance of the refrigeration unit concept presented in Figure 1 for $p_{LT\ evap} = 16.8$ bar and different gas cooler outlet conditions: a) $T_{motive}=25$ °C, $p_{motive}=66$ bar; b) $T_{motive}=35$ °C, $p_{motive}=90$ bar.

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 Figure 7 by means of iso-lines, normalized as a fraction of the ejector motive mass flow rate.

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

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

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 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 ($T_{gc,out} = 35\text{ °C}$, $T_{gc,out} = 25\text{ °C}$ and $T_{gc,out} = 15\text{ °C}$).

The experimental results demonstrated the capability of the tested ejector to provide acceptable efficiencies at the lowest suction pressure, down to 16.8 bar ($T_{sat} = -25\text{ °C}$), for $T_{gc,out} = 35\text{ °C}$ and $T_{gc,out} = 25\text{ °C}$. 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 $T_{gc,out} = 35\text{ °C}$ and 2.70 for $T_{gc,out} = 25\text{ °C}$) 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.

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NOMENCLATURE

COP coefficient of performance [-]

h specific enthalpy [kJ kg⁻¹]

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

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

LT low temperature suction ejector suction nozzle
MT medium temperature

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