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Dynamic numerical modelling of a single stage dual temperature R744 ejector-supported refrigeration unit for last mile delivery

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ABSTRACT

A novel design of a R744 vapor-compression refrigeration unit is proposed and developed to provide cooling capacity at two different temperature levels, in the range of 4-5 kW of MT refrigeration at 0°C and 1-2 kW of LT refrigeration at -20°C, inside the insulated box mounted on a typical medium-size refrigerated truck. Such a system can be employed for multi-temperature (chilled and frozen) goods transport to enhance the range of ambient conditions served with highly efficient operation during last-mile deliveries in urban environment.

The main innovation of the proposed cycle is related to the presence of a single compression stage with two different evaporation levels. Firstly, the cooling system design is based on the implementation of a MT ejector, with the purpose of energy efficiency improvement of the system due to the consequent reduction of the compressor pressure ratio. Secondly, an implemented LT ejector allows for the production of LT cooling effect with a single stage compressor. The system can switch to a back-pressure cycle once the ambient temperature is not sufficient to sustain the ejector cycle. On the other hand, the use of an ejector for the LT cycle allows to extend the application field of the single stage compressor to high ambient temperatures.

A dynamic numerical model of the refrigerating system has been developed to assess the system performance in ejector and back pressure mode. The numerical model was also used to characterize the system pull-down from ambient temperature.

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

1. INTRODUCTION

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 (e.g. fresh products at 0 °C and frozen products at -20 °C) in separate chambers of the same truck, especially for last mile delivery in urban environment (Frank et al., 2021).

Multi-temperature transport refrigeration units currently available in the market employ HFC or HFO refrigerants, such as R452A and R404A. However, sustainability challenges, supported by regulations are exponentially increasing the interest in natural refrigerants (in particular carbon dioxide, R744, and hydrocarbons, HCs) in newly developed transport refrigeration units.

Being food systems responsible for 20 to almost 40 % of total greenhouse gas emissions, the ENOUGH project (https://enough-emissions.eu/) supports the EU farm to fork sustainable strategy by providing technical, financial, and political tools and solutions to reduce GHG emissions (by 2030) and achieve carbon neutrality (by 2050) in the food industry. In this context, ENOUGH also addresses transport refrigeration, by proposing solutions which use natural refrigerants (CO_2) and are suitable for integration with renewables (PV), for use of Thermal Energy Storage (TES) and ready for electrification.

Multi-temperature units employing R744 as the refrigerant are developed and available in the market mostly for commercial stationary applications (Gullo et al., 2018; Karampour and Sawalha, 2018) and the employment of ejectors for the partial recovery of expansion work, to provide a pressure lift to the refrigerant mass flow rate at the evaporators outlet, has been proven as a consolidated solution to enhance the R744 transcritical cycle efficiency (Gullo et al., 2019). However, in such multi-temperature stationary systems, a subcritical compression stage is always included to increase the refrigerant pressure from the LT evaporation pressure to the suction pressure of the transcritical MT compressors, thus implementing a two-stage compression cycle for the LT operations.

Being compactness and weight reduction a crucial factor on the overall carbon footprint of a road transport refrigeration unit (Fabris et al., 2023a), the removal of the subcritical compressor from a multi-temperature system schematic would allow a significant reduction in weight, as well as in size and cost. To this extent, Fabris et al. (2023b) presented a first simplified R744 multi-temperature cooling unit schematic which employed an ejector to replace the subcritical compression stage and provide the necessary pressure lift from the LT evaporation pressure to the MT evaporation pressure and performed an experimental assessment of a commercially available ejector when employed in such operating conditions.

In this study, to avoid operational issues and to comply with the test procedures defined in Annex 1 of the ATP agreement (United Nations, 2020), which require the possibility of operating MT and LT evaporators independently, an improved version of the preliminary R744 multi-temperature schematic presented in Fabris et al. (2023b) is proposed, in which the employment of two separate ejectors (an MT ejector and a LT ejector) allows the necessary operational flexibility. A dynamic numerical model of the cooling unit has been developed. Simulation results help providing an assessment of the proposed cooling unit performance when operating under different conditions (with or without ejectors, with MT or LT cooling effect production) and understanding the system dynamic behaviour during configuration switches.

2. THE REFRIGERATION SYSTEM

The R744 refrigeration unit proposed in this study is designed to fulfil the refrigerating needs of a multi-temperature refrigerated vehicle, employed for last mile delivery of chilled and frozen goods in urban environment. The design cooling power is 4-5 kW of Medium-Temperature (MT) refrigeration at 0 °C air temperature and 1-2 kW of Low-Temperature (LT) refrigeration at -20° C air temperature.

The schematic of the cooling unit is presented in Figure 1. The proposed design allows operations by a simple back-pressure cycle or through the employment of an MT ejector and of an LT ejector for MT or LT operation, respectively; in both MT and LT operations, the same single-stage compressor is used. The schematic in each of these configurations is highlighted in Figure 2, in which the red colour is used for the high-pressure level in the gas cooler, the green colour for medium-pressure level, for MT refrigeration, and blue refers to low-pressure level, for LT refrigeration. In the ejector cycle schematics, the light green and light blue colour are used to highlight the pressure lift provided by the dedicated ejector between evaporation and compressor suction.

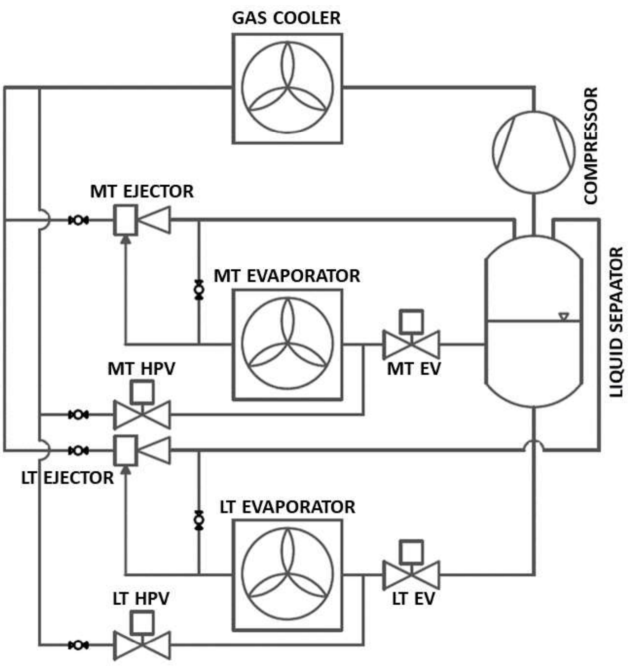


Figure 1 – Simplified schematic of the refrigeration system.

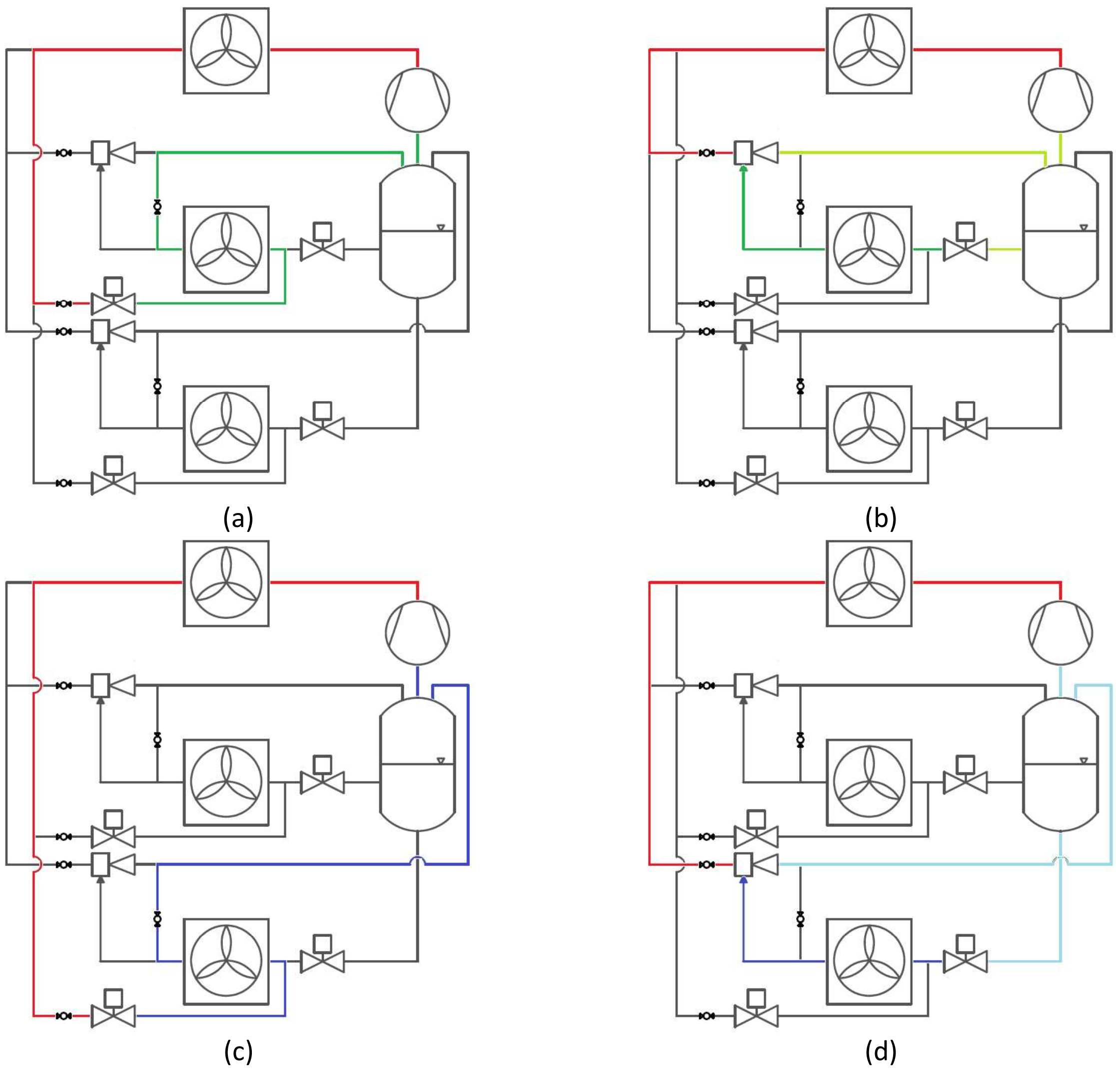


Figure 2 – Operational schematic of the refrigeration system: (a) MT back-pressure cycle; (b) MT ejector cycle; (c) LT back-pressure cycle; (d) LT ejector cycle.

In the simple back-pressure cycles (Figure 2a and 2c), a high-pressure valve (HPV) is used for the expansion of the refrigerant mass flow rate to the evaporation pressure level, while controlling the correct pressure in the gas cooler. The refrigerant evaporates than reaches the liquid separator as the ejector ports and the EV valves are closed. In this configuration, the compressor speed can be varied through an inverter with frequency between 30 Hz and 70 Hz with effects on the cooling effect production and cycle efficiency, as it will be further discussed in Section 4.1. On the other hand, in the ejector cycles (Figure 2b and 2d), the compressor speed is adjusted to control the high-pressure value while the HPVs are fully closed. The opening of the expansion valve (EV) is adjusted to modify the pressure lift provided by the ejector.

The ejectors reduce the compressor pressure ratio, providing a positive effect on the system Coefficient of Performance (COP), as it will be further discussed in Section 4.2. Despite the use of a single-stage compressor, the pressure lift provided by the ejectors avoids excessive compressor pressure ratio, which could exceed the compressor application limits, under harsh environmental temperature conditions, as it will be highlighted in Section 4.1

It is worth mentioning that, in case of simultaneous requirement of MT and LT cooling effect, the MT side can be engaged together with the LT side of the system, operating with the separator at a low-pressure level until the LT cooling effect requirement ceases, and the liquid separator pressure can be increased to medium-pressure level again.

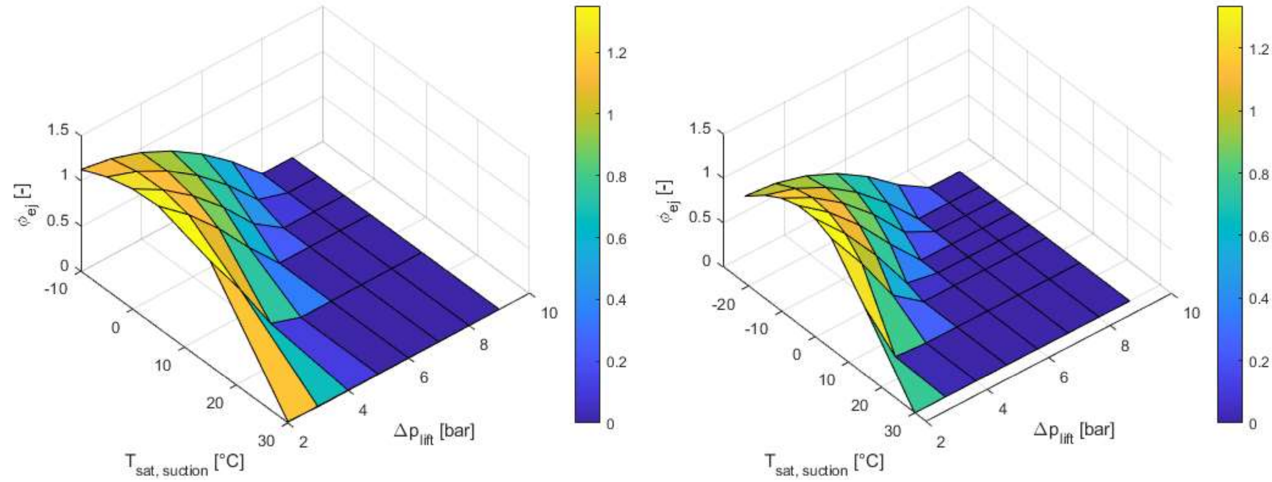
2.1. Components dimensioning

Compressor and heat exchangers are commercially available components. In particular, a R744 semi-hermetic compressor with a displacement volume equal to 16.8 cm^3 is considered. The internal and external convective surfaces of the heat exchangers are equal to 2.1 m^2 and 16.9 m^2 for the gas cooler, 3.8 m^2 and 39.4 m^2 for the MT evaporator and 1.3 m^2 and 8.7 m^2 for the LT evaporator, respectively.

Conversely, the MT and LT ejectors have been specifically designed for this application, as further described in the next section. The ejectors are characterized by a fixed geometry, with a throat diameter at the motive nozzle equal to 0.95 mm for the MT ejector and 0.46 mm for the LT ejector.

2.2. Ejectors design and performance

Following the experimental results obtained with a commercially available non-optimized ejector (Fabris et al., 2023b), a specific design process of the MT ejector and the LT ejector to be used in this refrigeration system was performed and their characteristic dimensions were optimized in order to provide the design cooling power at rated operating conditions (ambient temperature of 30 °C). 3-D CFD-based numerical simulations were carried out to numerically assess the performance of the ejectors (entrainment ratio, ejector efficiency) under various operating conditions (motive nozzle temperature and pressure, suction nozzle temperature and pressure, pressure lift) and the performance maps of the ejectors were then obtained through interpolation of the numerical results. As an example, the interpolated performance maps of the MT and LT ejectors for motive nozzle conditions equal to T_{motive} = 35 °C and p_{motive} = 86 bar are reported in Figure 3. The design and optimization approach adopted is in accordance with the one described in previous paper published by the same authors: Smolka et al. (2013), Palacz et al. (2015), Bodys et al. (2017) and Haida et al. (2018).



(a) (b)

Figure 3 – Interpolated performance maps of the ejectors for T_{motive} = 35 °C and p_{motive} = 86 bar: (a) MT ejector; (b) LT ejector.

3. NUMERICAL MODEL

A dynamic numerical model of the refrigeration system is developed using the commercial multi-physics software Simcenter Amesim v.17. The numerical approach of the software is based on the discretization of real components in lumped parameters elements, connected to describe the entire system. Each element is described by nonlinear time-dependent differential equations involving the state variables. Equations are then assembled in a system of differential equations, according to the elements connection. The model dynamic is then solved by integrating the system of differential equations over time. This approach allows to solve fast dynamics without the limiting assumptions of the quasi-stationary formulation.

The compressor is modelled using a fixed displacement compressor model. From the compressors operating data available in the commercial data sheets, the different volumetric efficiency and the overall compression efficiency are interpolated as functions of the pressure ratio and the rotational speed. The compressor is assumed to allow operation at variable speed, with frequency between 30 Hz and 70 Hz.

The gas cooler is discretized into N = 18 lumped volumes, to better describe the sharp properties changes of the CO_2 refrigerant in supercritical conditions inside the heat exchanger, while the MT evaporator and the LT evaporator are discretized into N = 6 and N = 4 lumped volumes, respectively. For all the heat exchangers, each discretized volume is then sub-divided in 3 nodes, one referring to the refrigerant flow, one to the state of tube wall and fins and one referring to the state of the air. The geometric characteristics of the heat exchanger, such as exchange areas and mass, are equally distributed in each lumped element. Internal convection between the refrigerant and the internal wall, conduction through wall and fins and external convection between the fins and the outside air are considered for each of the lumped volumes in which the heat exchangers are discretized. The heat transfer coefficients are evaluated through empirical correlations, available in the literature. For the refrigerant, mass and energy balances are evaluated in each discretized element.

The liquid separator is modelled as a cylindrical-shape tank with constant cross-sectional area, homogeneous pressure in the entire volume and homogeneous densities for the liquid phase and the vapor phase, in their respective volumes.

A detailed description of the dynamic numerical modelling approach, including the formulation of the differential equations used to describe the dynamic behaviour of the refrigeration system and the empirical correlations used to determine the heat transfer coefficients, can be found in Artuso et al. (2020) and Fabris et al. (2021).

4. NUMERICAL RESULTS

The cooling unit is firstly characterized in steady-state conditions, to critically discuss the dimensioning and the performance of the system. The results are assessed through evaluation of the steady-state response of the system to different set-points (compressor speed in case of back-pressure cycle operation and ejector pressure lift in case of ejector cycle operation).

After that, the pulldown of the system from environmental temperature conditions towards the set-point internal temperatures is discussed as an example of the dynamic response of the system.

4.1. Back-pressure cycle operation

The steady-state performance of the system is firstly evaluated in its simplest configuration, i.e. back-pressure cycle operation.

In MT operation, the cooling unit provides the necessary cooling effect to achieve the temperature set-point of the internal air (T_i = 0 °C). The steady-state MT cooling power (Q_{MT}), the compressor power draw (P_{comp}), the system COP (defined as COP = Q_{MT}/P_{comp}) and the evaporation temperature (T_{ev}) of the back-pressure cycle are reported in Figure 4 as functions of the compressor speed, for ambient temperature equal to T_{amb} = 15 °C (Figure 4a) and T_{amb} = 30 °C (Figure 4b).

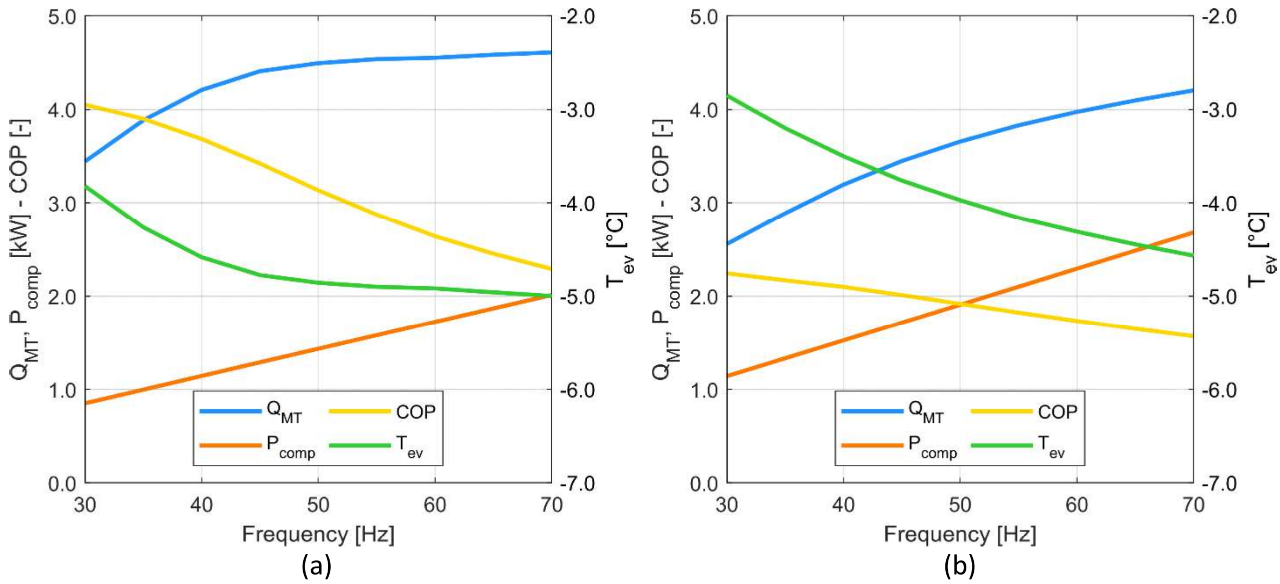


Figure 4 – Performance of the cooling unit in MT operation with back-pressure cycle: (a) with ambient temperature equal to T_{amb} = 15 °C; (b) with ambient temperature equal to T_{amb} = 30 °C.

For both ambient conditions, the cooling power and the compressor power draw obviously increase with increasing compressor speed. An MT cooling effect ranging from 3.4 kW to 4.6 kW can be provided for T_{amb} = 15 °C, while ranging from 2.6 kW to 4.2 kW for T_{amb} = 30 °C. The cooling unit COP decreases with increasing compressor speed, ranging from 2.3 to 4.1 for T_{amb} = 15 °C, and from 1.6 to 2.2 for T_{amb} = 30 °C, as the evaporation temperature decreases for increasing compressor speed, with a value between -3.8 °C and -5.0 °C for T_{amb} = 15 °C and between -2.9 °C and -4.6 °C for T_{amb} = 30 °C.

In LT operation, the temperature set-point of the internal air is equal to T_i = -20 °C. The steady-state performance of the back-pressure cycle in LT operation is reported in Figure 5 as a function of the compressor speed, for ambient temperature equal to T_{amb} = 15 °C (Figure 5a) and T_{amb} = 30 °C (Figure 5b).

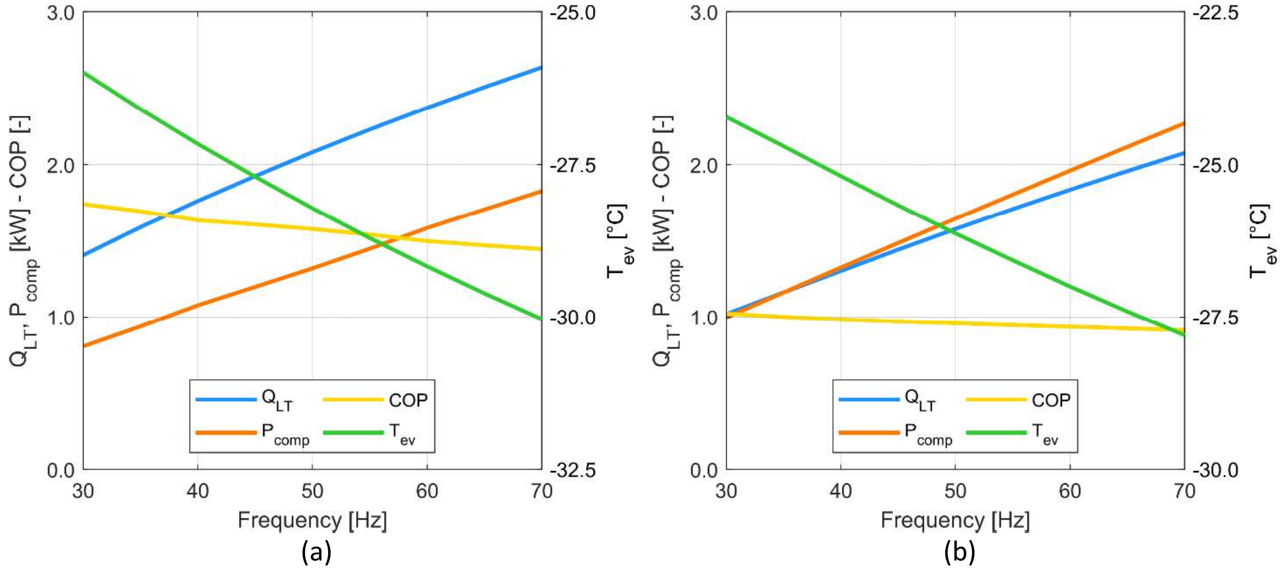


Figure 5 – Performance of the cooling unit in LT operation with back-pressure cycle: (a) with ambient temperature equal to T_{amb} = 15 °C; (b) with ambient temperature equal to T_{amb} = 30 °C.

The system performance as a function of the compressor speed follows the same trend discussed in Figure 4, with an increase of the compressor power draw and LT cooling effect (1.4 kW – 2.6 kW for T_{amb} = 15 °C and 1.0 kW – 2.1 kW for T_{amb} = 30 °C) with increasing compressor speed, while the cooling unit COP (1.4 – 1.7 for T_{amb} = 15 °C and 0.9 – 1.0 for T_{amb} = 30 °C) and the evaporation temperature (-26.0 °C – -30.0 °C for T_{amb} = 15 °C and -24.2 °C – -27.8 °C for T_{amb} = 30 °C) decrease with increasing compressor speed.

It is important to point out that, in LT operations at high ambient temperature (T_{amb} > 30 °C) the compressor pressure ratio overcomes the maximum allowed value, thus posing a technological limit to the implementation of the back pressure cycle. The introduction of the LT ejector, providing a pressure lift between evaporation pressure and compressor suction pressure, can help overcoming this issue, as it will be described in next section.

4.2. Ejector cycle operation

The use of an ejector in the cooling unit schematic has the main objective of recovering part of the expansion work of the refrigerant mass flow rate after heat rejection to the environment to provide a pressure lift of the mass flow rate at the outlet of the evaporator up to the liquid separator pressure level, thus reducing the compression power requirement. Moreover, in LT operations, the ejector helps in decreasing the compression ratio at high ambient temperature, thus allowing the use of a single stage compressor, as it will be detailed in the following discussion.

In the ejector cycle the compressor speed is controlled to provide to the ejector the exact mass flow rate which can be elaborated by its motive nozzle. On the contrary, the set-point parameter is the pressure lift, which is adjusted through the opening of the evaporator expansion valve.

The performance of the cooling unit operating in ejector configuration for ambient temperature equal to T_{amb} = 30 °C is presented in Figure 6. Motive energy at T_{amb} = 15 °C is insufficient to allow the ejector operation. Figure 6a refers to the performance in MT cooling effect production, while Figure 6b refers to LT cooling effect production.

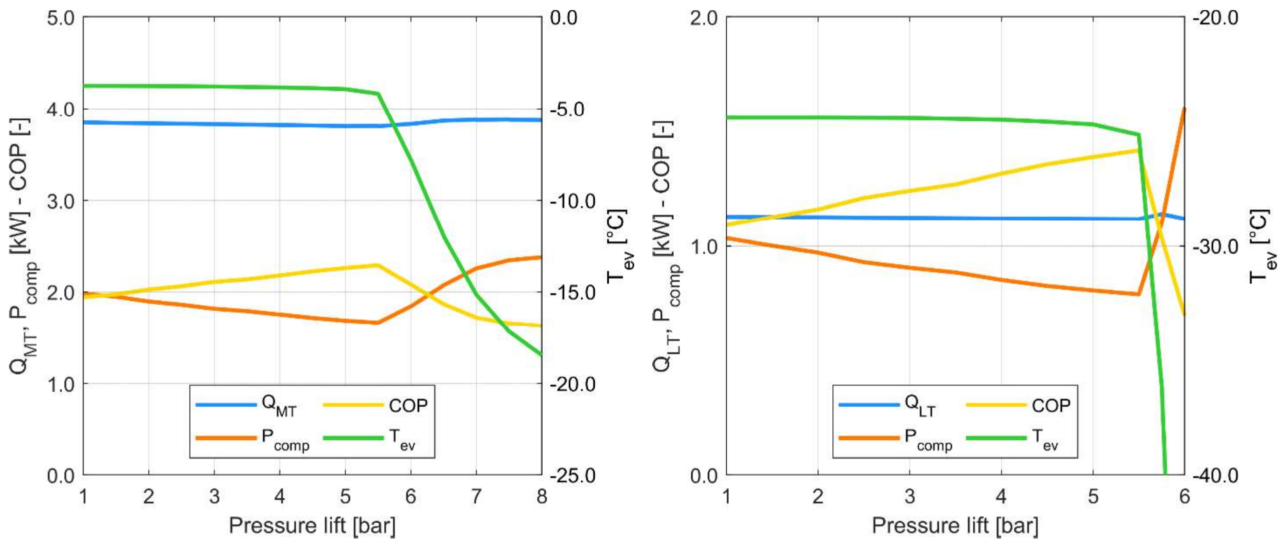


Figure 6 – Performance of the cooling unit with ejector cycle, with ambient temperature equal to T_{amb} = 30 °C: (a) MT operation; (b) LT operation.

For MT operation (Figure 6a), the cooling power is almost unchanged and equal to approximately 3.8 kW regardless of the ejector pressure lift, while the compressor power draw decreases for increasing lift, therefore increasing the maximum system COP, which reaches its maximum value (equal to 2.3) for a pressure lift of 5.5 bar. However, for pressure lifts higher than 5.5 bar, the decrease of the refrigerant mass flow rate inside the evaporator, due to the reduction of the ejector entrainment ratio with increasing lift, leads to a reduction of the heat transfer coefficient and increase of superheating and to a consequent sudden reduction of the evaporation temperature. Therefore, the separator pressure, corresponding to the compressor suction pressure, starts decreasing, progressively reducing the system COP. The same behaviour can be observed in LT operation (Figure 6b), with an even more significant drop of the system performance for high lifts. In LT operation, a cooling effect production of approximately 1.1 kW and a maximum unit COP of 1.4, corresponding to a pressure lift equal to 5.5 bar, are obtained.

To determine whether the ejector cycle can help improving the performance of the system in MT operation compared to the simple back-pressure cycle for high environmental temperatures, the cooling power, compressor power draw, unit COP and evaporation temperature are compared between the two configurations in Table 1. To ensure a fair comparison, operating conditions characterized by the same MT cooling power are compared. The ejector cycle presents a COP increase equal to 25.8% compared to the back-pressure cycle; nevertheless, it must be pointed out that the back-pressure cycle allows a significantly higher flexibility in the range of MT cooling power compared to the ejector cycle, which can be useful during pulldowns or part-load operation.

Table 1 - Performance comparison between back-pressure cycle and ejector cycle in MT operation, with ambient temperature equal to T_{amb} = 30 °C and equal MT cooling effect production.

	Q_{MT} [kW]	P_{comp} [kW]	COP [-]	T_{ev} [°C]	Operating conditions
Back-pressure cycle	3.81	2.09	1.82	-4.18	Frequency = 55 Hz
Ejector cycle	3.81	1.66	2.29	-4.02	Pressure lift = 5.5 bar

The same performance comparison between back-pressure cycle and ejector cycle is reported for LT operation in Table 2. In this case, the ejector cycle presents an even higher COP increase compared to the back-pressure cycle, equal to 42.0%.

Moreover, as it was described in Section 4.1, the pressure lift provided by the ejector can help extending the range of environmental conditions in which a single stage compressor can be applied. The evaporation pressures, as well as the compressor suction, discharge and maximum discharge pressure for the operating condition of table 2 are reported in Table 3.

Table 2 - Performance comparison between back-pressure cycle and ejector cycle in LT operation, with ambient temperature equal to T_{amb} = 30 °C and equal LT cooling effect production.

	Q_{LT} [kW]	P_{comp} [kW]	COP [-]	T_{ev} [°C]	Operating conditions
Back-pressure cycle	1.16	1.16	1.00	-24.70	Frequency = 35 Hz
Ejector cycle	1.12	0.79	1.42	-25.15	Pressure lift = 5.5 bar

Table 3 – Pressure and pressure ratio comparison between back-pressure cycle and ejector cycle in LT operation.

	p_{ev} [bar]	p_{suc} [bar]	p_{Hp} [bar]	r_p [-]	$p_{HP\ max}$ [bar]
Back-pressure cycle	17.0	17.0	83.0	4.9	95
Ejector cycle	16.7	22.2	83.0	3.74	115

The increase of the suction pressure given by the ejector allows to extend the operability of the compressor toward the high pressure up to 115. This is crucial to guarantee the refrigeration system functionality at design high pressure at ambient pressure up to 40°C, while the back-pressure cycle can operate with this evaporation pressure at maximum ambient temperature of 34°C. While for stationary application in temperate or continental climate areas such high temperatures can be considered extreme cases but for the given application it is critical to consider such conditions.

4.3. Dynamic example: system pull-down

After the assessment of the steady-state performance of the cooling unit, the dynamic behavior of the system is evaluated through the simulation of a system startup and consequent pull-down from thermal equilibrium with the external environment (T_{amb} = 30 °C) to the MT and LT temperature set-points. Since the data required for the dynamic characterization of a specific insulated body are not available at this point, the only thermal capacities considered for the following assessment of the system dynamic response are the air volumes inside the MT and LT compartments (approximately 24 m^3 and 1 m^3, respectively) and the mass of the evaporators (including fans and structural supports). The thermal capacities initialization is done assuming thermal equilibrium with the environment.

The pull-down from thermal equilibrium with the environment at T_{amb} = 30 °C, performed in ejector configuration, is highlighted in Figure 7. At the cooling unit startup, the system operates in MT cooling effect production, to reach the set-point temperature $T_{i,MT}$ = 0 °C inside the MT compartment. After reaching the MT set-point, the cooling unit is turned off and the system switches to LT cooling effect production, to reach the set-point temperature $T_{i,LT}$ = -20 °C inside the LT compartment.

Figure 7a reports the saturation temperature in the liquid separator, in the MT evaporator and in the LT evaporator. Dashed lines are used to highlight the portion of the system which is not operating in each of the sections of the pull-down. The MT pulldown (air inside the MT compartment from 30 °C to 0 °C) takes around 23 minutes, while the LT pulldown after the configuration switch (air inside the LT compartment from 30 °C to -20 °C) takes around 40 minutes. Excluding the initial minutes after the system is switched on, it can be observed that both the MT ejector and the LT ejector are able to maintain the optimal pressure lift during dynamic operation.

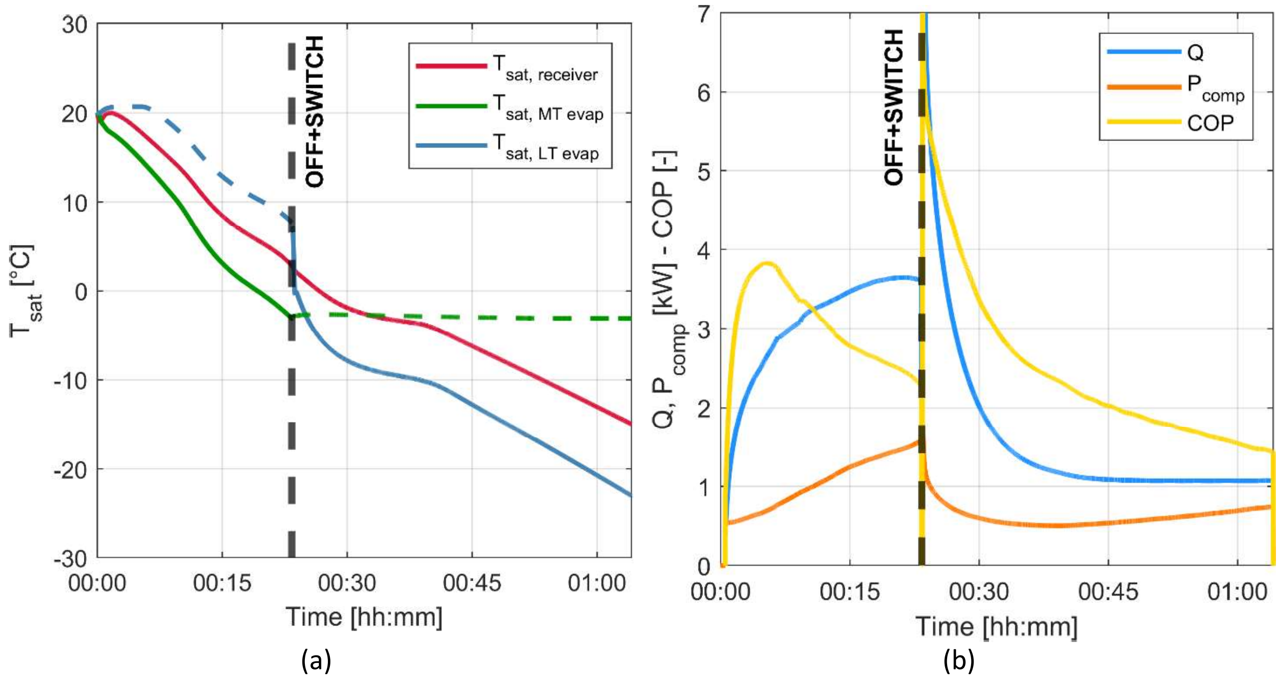


Figure 7 – System pull-down from thermal equilibrium with environment at T_{amb} = 30 °C: (a) saturation temperatures; (b) cooling unit performance.

Figure 7b reports the performance parameters of the system during the pull-down. In the first minutes after the first system switch on in MT operation, the system COP is high because the compression ratio is still limited; on the other hand, after the switch to LT operation, the system COP presents high values because of the high heat transfer in the LT evaporator in the first minutes of operation, due to the significant temperature difference between the air inside the LT compartment (still at 30 °C at the beginning of the LT operation) and the saturation temperature of the refrigerant in the evaporator coils (which is already at low values, thanks to the MT pull-down previously performed). However, in both MT and LT operation, as the temperature set-points are approached, the cooling unit performance reaches values which are in agreement with the steady-state values evaluated in Section 4.2.

5. CONCLUSIONS

In this study, a novel R744 cooling unit schematic for multi-temperature road transport applications is proposed. The cooling unit is designed to provide MT cooling effect at a temperature equal to 0 °C for chilled products and LT cooling effect at a temperature equal to -20 °C for frozen products. The system can operate following a simple back-pressure cycle or an ejector cycle, with dedicated MT and LT ejectors. The use of an ejector for LT operations is intended for both performance improvement and for allowing the use of a single stage compressor in LT operations at high ambient temperature. A dynamic numerical model of the cooling unit has been developed to assess at first the steady-state performance of the system under different environmental temperature conditions and with both the possible operating configurations, and then to evaluate the dynamic response of the system during a pull-down from equilibrium with the external environment to the MT and LT temperature set-points.

For ambient temperature equal to 30°C, the cooling unit in back-pressure configuration and for MT operation can provide an MT cooling effect ranging between 2.6 kW to 4.2 kW, with a COP ranging from 1.6 to 2.2, while for LT operation it can provide a LT cooling effect ranging between 1.0 kW to 2.1 kW, with a COP ranging from 0.9 to 1.0. On the other hand, the cooling unit in ejector configuration can provide for MT operation an MT cooling effect of approximately 3.8 kW, with a maximum COP equal to 2.3, while for LT operation it can provide a LT cooling effect of approximately 1.1 kW, with a maximum COP equal to 1.4. Comparing the back-pressure and ejector configuration performance for the same cooling effect production, the use of the ejectors can lead to a COP increase of 25.8% in MT operation and 42.0% in LT operation. After a switch on from thermal equilibrium with the environment at 30 °C, the system takes around 23 minutes to reach the MT set-point temperature (0 °C) inside the MT compartment and around 40 minutes to reach the LT set-point temperature (-20 °C) inside the LT compartment of the refrigerated vehicle.

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