# Design and performance evaluation of a novel $CO_2$ rooftop prototype

Francesco De Zan[a*], Mihir M. Hazarika[a], Simone Piovesan[a], Armin Hafner[b], Krzysztof Banasiak[c], Sara Bordignon[a]

a Clivet Spa, Via Camp Lonc, Feltre (Belluno) Italy
b NTNU Trondheim Norway
c Sintef Energy Research Trondheim Norway

**Abstract**

Air-to-air systems like rooftop units, combining mechanical ventilation and heating and cooling purposes, play a crucial role in the HVAC sector, mainly used in commercial buildings. In view of evolving regulatory frameworks on refrigerants, converting rooftop units from using synthetic refrigerants to natural refrigerants, becomes of great interest. This paper aims at presenting a study on a $CO_2$ rooftop unit, from the thermodynamic evaluations of the refrigerant cycle performance to the design of the components for a 45kW prototype. Different configurations and control parameters have been evaluated thanks to a simulation model developed in Modelica: different strategies for refrigerant mass flow regulation, fan speed and different expansion devices have been taken into account. The selected compressor is regulated both through frequency control and piston shut-off. The refrigerant expansion process is done either through the use of an ejector, or through an electronic expansion valve used as high pressure valve to evaluate the best configuration according to the boundary conditions (air temperature and thermal load). The highest seasonal energy efficiency ratio was estimated to be 4.55 when ejector is used. In the paper it is also presented the layout of a prototype to investigate the real operation of the unit and validate the simulation model. The prototype can operate in heating and cooling modes and is able to test different configurations and control parameters.


Selection and/or peer-review under the responsibility of the organizers of the 15th IEA Heat Pump Conference 2026.

*Keywords: Rooftop unit; $CO_2$; HVAC; Natural refrigerants; Commercial applications*

## 1. Introduction

In the HVAC sector, particularly for commercial buildings, air-to-air systems like rooftop units perform the important function of integrating mechanical ventilation, heating, and cooling purposes. According to the state-of-art, conventional HVAC units are responsible of almost half of the total of energy consumption in buildings [1]. In 2020, in EU27, 8 million continuously operating ventilation units (29% of the total) were for non-residential use, and the number is expected to grow [2]. On the other hand, 6.5 million air conditioners (ACs) were installed across the EU27 in 2020, of which, a significant majority (72%) were reversible units. These can both cool during the summer and provide heating in the winter [3]. The three main AC types are split/multi-split, rooftop, and variable refrigerant flow (VRF). Rooftop units are typically placed outdoors, often on the building's roof or side. In these systems, air is the heat carrier fluid, cooled by the AC unit and then moved throughout the building via ducts, which may also incorporate ventilation features. Current rooftop units in the market use hydrofluorocarbons (HFCs) as the refrigerant. However, due to F-gas regulation [4], by 2027 chillers and heat pump units with capacity lower than 50 kW must be equipped with refrigerants with GWP<150, and by 2030 all units must be equipped with refrigerants with GWP<150.

* Corresponding author. *E-mail address:* f.dezan @clivet.it

In view of evolving regulatory frameworks on refrigerants, converting rooftop units from using synthetic refrigerants to natural refrigerants, becomes of great interest. In this context, R744 ($CO_2$) is the most promising choice for air-to-air units due to safety regulations, which restrict the use of other natural refrigerants like hydrocarbons and ammonia, which are flammable and toxic.

The main challenges to face when using R744 as the refrigerant, are related to its high critical pressure, requiring high-pressure equipment. The higher suction vapor density and lower pressure ratio, compared to conventional refrigerants for HVAC applications, also necessitate different compressor displacements for the same cooling effect. The compressor usually needs thicker walls to contain the high operating pressure, but since the fluid has a large volumetric capacity, the compressor itself will actually be smaller than those used with conventional refrigerants for the same cooling capacity [5]. Moreover, due to its low critical temperature, air conditioning applications require a trans-critical cycle and a high temperature lift at the gas cooler side, to allow high efficiencies. Despites these technical issues, R744 is state of the art in commercial refrigeration [6], [7], and in the last decade it has been applied in residential heat pump units coupled to high temperature terminal units to replace gas boilers or for domestic hot water production [8], [9]. On the contrary, its application in the ventilation and air conditioning sector is still very limited, mainly due to relatively high cost of the components, compared to conventional HFCs (R32, R410a, R454B) and just a few studies have performed some performance analysis on R744 rooftops, which, so far, have shown performance near similar HFC systems and wider operating ranges [10].
This paper presents the thermodynamic study and component design of a 45 kW R744 rooftop unit prototype providing both heating and cooling. A key focus is the expansion process, which is evaluated using either an ejector or an high pressure valve (HPV) to determine the optimal configuration based on ambient conditions and thermal load. The study presents strategies for improving system efficiency, provides a description of the main components and of the experimental setup. The design focus on improving the efficiency in cooling operation; heating operation is usually less relevant in the design process given that rooftops are usually not intended to deliver the full heating load to the building but complement another heating source.

## 2. Method

Rooftop units are self-contained commercial air conditioning units using a refrigeration cycle; the rating of capacity and the efficiency is based on the standard EN14825. The efficiency is evaluated through the seasonal energy efficiency ratio (SEER) that is a weighted average of the EER in cooling in four operating conditions that are reported in Table 1. For each operating condition air temperatures and target capacities are defined.

Table 1: Boundary air conditions for cooling mode according to standard EN 14825

| Operative condition | Part load Temperature [°C] | Part load ratio [%] | Outdoor Heat Exchanger Outdoor Air Dry Bulb Temperature [°C] | Indoor Heat Exchanger Indoor Air Dry Bulb (Wet Bulb) Temperature [°C] |
|---|---|---|---|---|
| **A (Full load)** | 35 | 100 | 35 | 27 (19) |
| **B** | 30 | 74 | 30 | 27 (19) |
| **C** | 25 | 47 | 25 | 27 (19) |
| **D** | 20 | 21 | 20 | 27 (19) |

The objective of this study is to design the main components of the Rooftop unit prototype able to deliver a full load cooling capacity of 45 kW for a standard air flow of 9000 $m^3/h$ in Condition A and present a strategy to improve the SEER. In order to make more effective the prototype construction, the dimensional constrains of the metal frame of an existing rooftop unit is considered.

The work is organized into the following phases:
(a) Preliminary thermodynamic analysis referring to the full load operative conditions to define of the base thermodynamic parameters and cycle layout.
(b) Development of a simulation model in Modelica and sizing the major components.
(c) System architecture improvements to increase the SEER.
(d) Definition of the system layout of a rooftop prototype to be built with the aim to test the different solutions and validate the simulation model.

Experimental campaign and validation of the simulation model are still under execution and will be presented in future works.

## 3. Simulation models

In this section, the two simulation models are presented: first one is a simplified thermodynamic model and the second one is a detailed simulation model of the proposed unit developed in Modelica.

### *3.1. Simplified thermodynamic model*

In the first phase, a thermodynamic model is arranged to define thermodynamic properties and refrigerant mass flow rate in the design point corresponding to the full load cooling capacity (condition A). The major components considered in the thermodynamic cycle are: outdoor heat exchanger, indoor heat exchanger, compressor and expansion device. An internal heat exchanger (IHX) is included in this configuration to investigate its impact on the thermodynamic performance of the proposed R744 unit. The thermodynamic model clearly presents the benefits of using the IHX in the proposed unit as it allows to operate the evaporator in flooded mode and ensure decent superheat at the compressor suction. The boundary conditions for the thermodynamic model are presented in Table 2.

Table 2: Boundary conditions and assumptions for the thermodynamic model

| **Parameter** | **Value** |
|---|---|
| DBT of Outdoor air, $T_{outdoor}$ [°C] | 35 |
| DBT of return room air, $T_{return}$ [°C] | 27 |
| DBT of supply room air, $T_{supply}$ [°C] | 14 |
| Gas cooler approach temperature [K] | 2 |
| Evaporation temperature [°C] | $T_{supply} - 2$ |
| Vapor fraction at evaporator exit [-] | 0.85 |
| Cooling capacity [kW] | 45 |
| IHX effectiveness [-] | 0.6 |
| Compressor isentropic efficiency [-] | 0.7 |
| Compressor volumetric efficiency [-] | 0.8 |

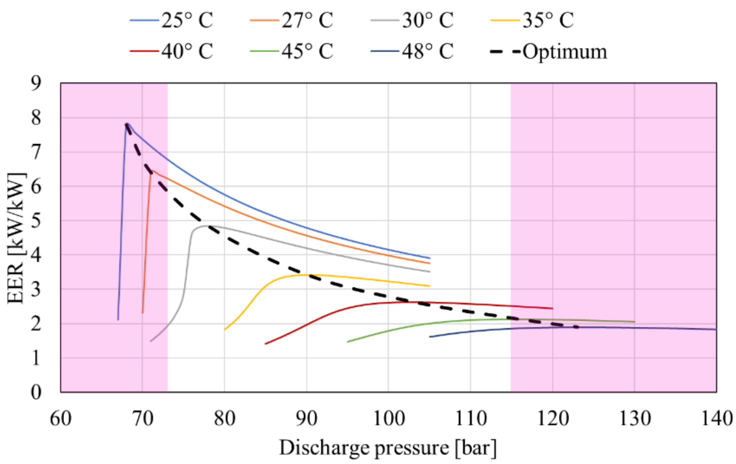


Figure 1: Optimum discharge pressures for different outdoor temperatures

Considering that R744 has low critical temperature, the cycle needs to be operated in transcritical mode at high outdoor temperatures. As a consequence, the compressor discharge pressure needs to be optimized to maximize the energy efficiency. The optimum discharge pressures for different outdoor temperatures (from 25° C to 48° C) are estimated using the present thermodynamic model and are presented in Figure 1. Figure 1 reports the optimum pressure curve (dotted line), which interpolates the optimum discharge pressure value for the different outdoor temperature conditions. The unit operates in subcritical mode below the critical point of 73 bar, while the maximum operating pressure limit is 115 bar which is selected considering the design pressure of the components (shaded areas). For condition A (Full load), the optimum discharge pressure is 90 bar and the estimated compressor swept volume is 10.8 $m^3$/hr.

*3.2. Detailed model using Modelica*

Once the suitable compressor is selected, the three heat exchangers (outdoor heat exchanger, indoor heat exchanger, and internal heat exchanger) can be designed. For this purpose, the simulation model is developed using object-oriented programming language Modelica in Dymola environment [11]. The commercial library package TIL is used to develop the simulation model [12]. A layout of the simulation model is presented in Figure 2 (a), while Figure 2 (b) shows the thermodynamic process state-points in a p-h plot.

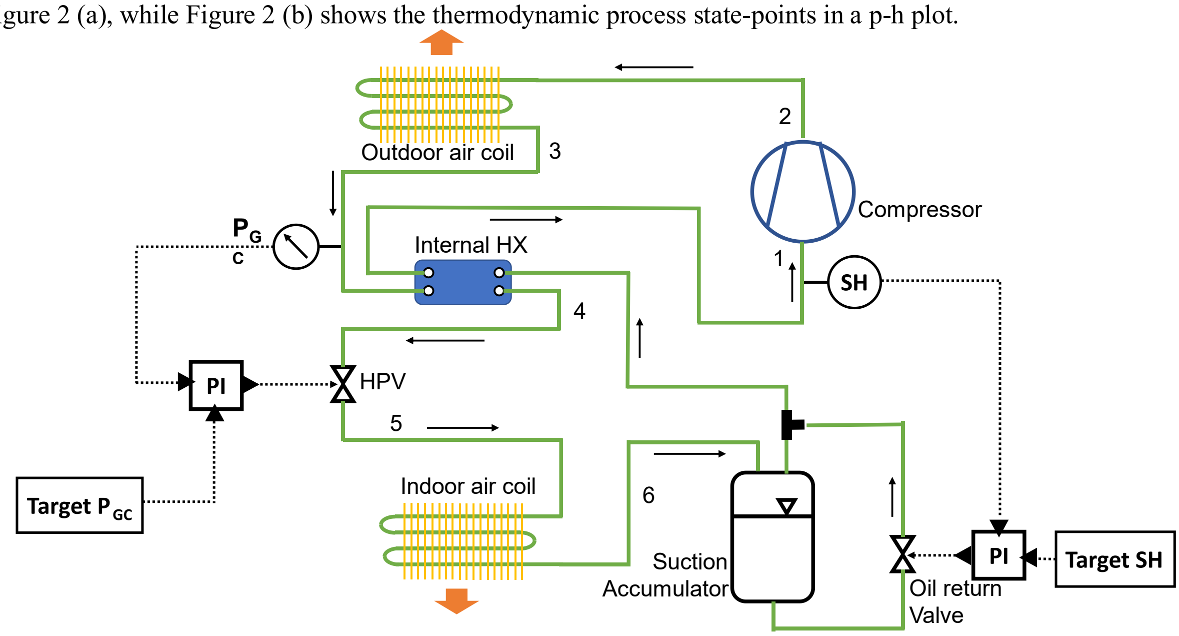


Figure 2: (a) Layout of the simulation model developed in Modelica

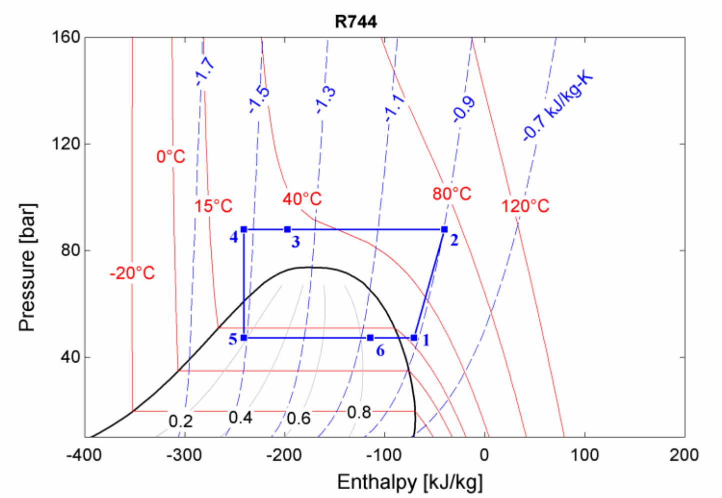


Figure 2: (b) p-h plot for the layout with HPV

Figure 2 (a) refers to the model for cooling operation, the green lines represent the refrigerant circuit, while yellow lines the air flows. The layout includes the heat exchangers, the electronic expansion valve, the suction accumulator, the liquid return valve and the compressor. In cooling operation, the outdoor heat exchanger works as a gas cooler, and the indoor heat exchanger as an evaporator. The IHX ensures subcooling of the refrigerant at the gas cooler exit and superheating of the refrigerant at the suction of compressor. The high-pressure level in the gas cooler is maintained using the electronic expansion valve that is regulated using a PI controller. In addition to high-side pressure control, this configuration has the possibility to regulate the liquid refrigerant returned from the bottom of the suction accumulator to the compressor and control the compressor suction superheating. This is achieved by using an oil return valve: in the real system it is used to regulate the oil return along with liquid refrigerant from the suction accumulator to the compressor and it is controlled using a PI controller to maintain the desired superheat.

The gas cooler and the evaporator used in this model are fin-and-tube heat exchangers. The models for fin-and-tube heat exchanger available in TIL library are nodal and allow to provide the geometrical dimensions as input parameters: fin thickness, fin pitch, finned tube length, tube diameter, tube pitch, row pitch, number of tubes in each row, number of tube rows, and number of tube circuits. To model the refrigerant-side and air-side heat transfer coefficients, suitable correlations are selected in the model. The refrigerant-side pressure drop during evaporation at subcritical pressure is estimated using available correlation, while the refrigerant-side pressure drop during gas cooling process at supercritical pressure is neglected.

The internal heat exchanger is a brazed plate heat exchanger. The geometrical dimensions of a plate heat exchanger are: total number of plates, plate length, plate width, wall thickness, pattern angle, pattern amplitude, and pattern wavelength. To estimate the heat transfer coefficients, suitable correlations are selected for subcooling of high-pressure refrigerant at supercritical pressure and superheating of low-pressure refrigerant at subcritical pressure.

The compressor is modelled using the polynomials from the compressor's manufacturer, estimating the compressor power consumption and refrigerant flowrate.

*3.2.1. Ejector as expansion device*

The use of an ejector as the expansion device to replace the HPV is also investigated. A second model is therefore developed, including the ejector, which nozzle is modeled using Brennen correlation, and a fixed ejector efficiency of 20% [12]. In this model, the high-side pressure is regulated by using a PI controller that is changing the nozzle cross-section to maintain the target high-side pressure under different conditions. The flowrate of liquid refrigerant from the suction accumulator to the evaporator is regulated by changing the cross-section of liquid feeding valve (LFV) to maintain the desired vapor fraction at the exit of evaporator.

In order to validate the ejector operation, a commercial model has been selected to be equipped in the prototype: commercial ejectors are usually designed for refrigeration conditions which pressure differences are larger and evaporating temperatures are lower so the one selected will mostly operate outside the declared pressure envelope and the actual efficiency will be measured during the test campaign.

## 4. Components' selection

4.1. *Heat exchangers*

Once the boundary conditions are finalized, the simulation model is used to estimate the dimensions for outdoor heat exchanger, indoor heat exchanger, and internal heat exchanger.

The factors that are most important for sizing the gas cooler are: the target approach temperature and refrigerant velocity at the outlet. In addition to that the overall dimension of the prototype unit impose limits on the gas cooler's height, width, and depth. These external limits directly constrain the internal geometry: width defines the maximum length of finned tube, height defines the maximum number of tubes in each row, and depth defines the maximum number of tube rows. Considering these factors, the only design parameters that can be varied during the sizing process for the outdoor heat exchanger (gas cooler) are: the number of tubes in each row, the number of tube rows, and the number of parallel flows. Simulations are carried out by varying these three parameters to obtain the target approach temperature and refrigerant velocity at the exit; given that the prototype is expected to operate and will be tested also in heating, also the reverse operation of the heat exchangers are checked for the selection of the number of circuits.

Table 3: Parametric study for definition of number of circuits of the outdoor heat exchanger

| *OUTDOOR COIL* | | | | | | | |
|---|---|---|---|---|---|---|---|
| **Number of circuits** | | **10** | | **15** | | **20** | |
| *COOLING (Gas cooler - Countercurrent)* | | | | | | | |
| **Air Outdoor DB / Indoor DB (WB)** | | **35/27(19)** | **25/27(19)** | **35/27(19)** | **25/27(19)** | **35/27(19)** | **25/27(19)** |
| Total capacity | kW | 66.0 | 26.6 | 66.5 | 26.6 | 66.4 | 26.6 |
| *HEATING (Flooded Evaporator - Cocurrent)* | | | | | | | |
| **Air Outdoor DB (WB) / Indoor DB** | | **-7°C(-8°C)/ 20°C** | **7°C(6°C)/ 20°C** | **-7°C(-8°C)/ 20°C** | **7°C(6°C)/ 20°C** | **-7°C(-8°C)/ 20°C** | **7°C(6°C)/ 20°C** |
| Vapor velocity | m/s | 5.6 | 1.8 | 4.8 | 1.3 | 3.8 | 0.9 |
| Refrigerant P Drop | kPa | 237.0 | 50.1 | 120 | 17.3 | 61.3 | 7.7 |
| Total capacity | kW | 22.7 | 12.6 | 29.1 | 12.9 | 31.1 | 12.8 |

The result of the parametric study is reported in Table 3, suggesting that the best choice of number of circuits is 15 where the capacity is the maximum and the gas velocity at the outlet of the evaporator is higher than 1 m/s for all the conditions to ensure proper oil entrainment to the evaporator outlet.

In cooling mode, the indoor heat exchanger works as an evaporator. The factors that are most important for sizing this indoor heat exchanger are: to maintain flooded operation during evaporation and achieve target refrigerant velocity at the exit. Like for the outdoor heat exchanger, the indoor heat exchanger is designed by varying the three design parameters: the number of tubes in each row, the number of tube rows, and the number of parallel flows.

The final dimensions estimated for the outdoor and indoor heat exchangers are presented in Table 4.

Table 4: Dimensions of Outdoor and Indoor heat exchanger

| | **Outdoor heat exchanger** | **Indoor heat exchanger** |
|---|---|---|
| Length of finned tube (mm) | 1750 | 840 |
| Number of tubes in each row | 60 | 52 |
| Tube pitch (mm) | 25 | 25 |
| Number of tube rows | 5 | 5 |
| Row pitch (mm) | 21.65 | 21.65 |
| Fin thickness (mm) | 0.1 | 0.1 |
| Fin pitch (mm) | 1.8 | 2.1 |
| Tube diameter (mm) | 7.2 | 7.2 |
| Tube wall thickness (mm) | 0.5 | 0.5 |
| Number of parallel flows (circuits) | 15 | 13 |

The internal heat exchanger is used to ensure subcooling of refrigerant at the gas cooler exit and superheating of refrigerant at the compressor suction. The dimensions of brazed plate heat exchanger are estimated to achieve 10 K degree of superheat at the compressor suction, and are reported in Table 5.

Table 5: Dimensions of Internal heat exchanger

| **Type** | **Brazed plate heat exchanger** |
|---|---|
| Number of plates | 18 |
| Plate width (mm) | 300 |
| Plate height (mm) | 100 |
| Wall thickness (mm) | 0.75 |

4.2. *Compressor*

R744 compressor landscape is dominated by semi hermetic compressors designed for commercial and industrial refrigeration. Compressor selection is critical and the detailed Modelica model allows the definition of the correct size. Beside the swept volume to satisfy the design cooling capacity, other requirements have to be considered: from one side the need to limit the system complexity to contain the unit's costs, on the other hand, the necessity to control the swept volume to low values to improve the efficiency at the part loads. To reduce complexity in system architecture and simplify the control strategy, compared to similar size units, the number of compressors has been changed from two to one. With variable frequency driver, the speed of a reciprocating compressor can be changed and typical range is from 70 Hz down to 30 Hz. However, this is not enough to meet the load variation requested by the proposed R744 unit. The simulation results from Modelica showed that the compressor speed should be changed from 70 Hz to 15 Hz to fulfill the requested load variations. A rigorous market survey of reciprocating compressors led to the selection of a 4-cylinder inverter compressor equipped with a 2-cylinder shut-off control mechanism. This additional control feature allows to reduce the effective compressor swept volume to a value which is equal to what could be achieved with 15Hz operation.

After selecting the commercial compressor model that provides the required control with suitable evaporating pressure and temperature, this component is simulated in Modelica using the manufacturer's performance polynomials [13].

4.3. *Fan power estimation*

Rooftop units are typically equipped with two sets of fans: outdoor fans circulate air across the outdoor coil, while indoor fans circulate air through the air treatment section. The indoor fans also generate the residual static pressure needed to distribute the conditioned air throughout the building. Fan power consumption is included in the overall power consumption of the unit and contributes to the EER value calculation. The air pressure drop through each coil was estimated through the detailed model, which also includes the fan performance curves to calculate the related power consumption.

Since rooftop units deliver a constant volumetric flow rate (equal to the design value) across all rating conditions, the supply fan power can be estimated, with good approximation, as constant across all working conditions. In the present study, the fan power for the selected fan, delivering 9000 $m^3/h$ over the indoor coil, is equal to 1.45 kW.

On the other hand, the air volumetric flow rate across the outdoor coil can be modulated according to the required heat load. Its value can be optimized in part loads, as its influence on the EER is relevant. In this study, the air flow through the outdoor coil was provided from an inverter controlled fan, which can modulate from 1000rpm down to 300 rpm. The detailed model allows optimizing the outdoor fan power consumption, while maintaining the desired heat rejection. The values of airflow, fan rotational speed and fan power consumption are reported in Table 6.

Table 6: Performance parameters of the fans in the different cooling operating conditions

| Operating conditions | **A** | **B** | **C** | **D** |
|---|---|---|---|---|
| Outdoor fan airflow [m3/h] | 23600 | 23600 | 9700 | 9700 |
| Outdoor fan speed [rpm] | 750 | 750 | 300 | 300 |
| Outdoor fan power [kW] | 1.45 | 1.45 | 0.10 | 0.10 |

## 5. Results and discussion

After sizing the main components, it was possible to select unit working parameters to optimize the EER in each of the cooling operating conditions, which can be used to calculate the SEER. Focusing on the system level, different configurations and control strategies can be identified:

- *HPV: t*he expansion device is the electronic expansion valve, used as a back pressure valve to control the system high pressure to the optimal value, and the swept volume is limited by the minimum frequency of the inverter compressor alone.
- *HPV (2 Pistons shut off)*: the swept volume can be further lowered in conditions C and D thanks to shut-off of 2 compressor pistons out of 4. This allows reducing the power consumption and increasing the energy efficiency at part loads.

- *HPV (2 Pistons shut off + Dsub)*: at low outdoor air temperatures, the HPV controls the high pressure to subcritical values in condition D, further decreasing the power consumption in this specific operating condition.
- *Ejector (2 Pistons shut off):* the use of the ejector as the high pressure control device, operating in transcritical mode in all outdoor air conditions, allows achieving good efficiency improvements in condition A-B and C compared to the system with HPV. No marked improvements can be achieved in condition D, when in transcritical operation.
- *Ejector (2 Pistons shut off + Dsub)*: in conditions D, the ejector is controlled as a simple expansion device not recovering any expansion work, and the system is run in subcritical mode. This configuration leads the system to the highest efficiency in all the conditions.

Figure 3 shows the comparison in terms of energy efficiency between the four proposed configurations.

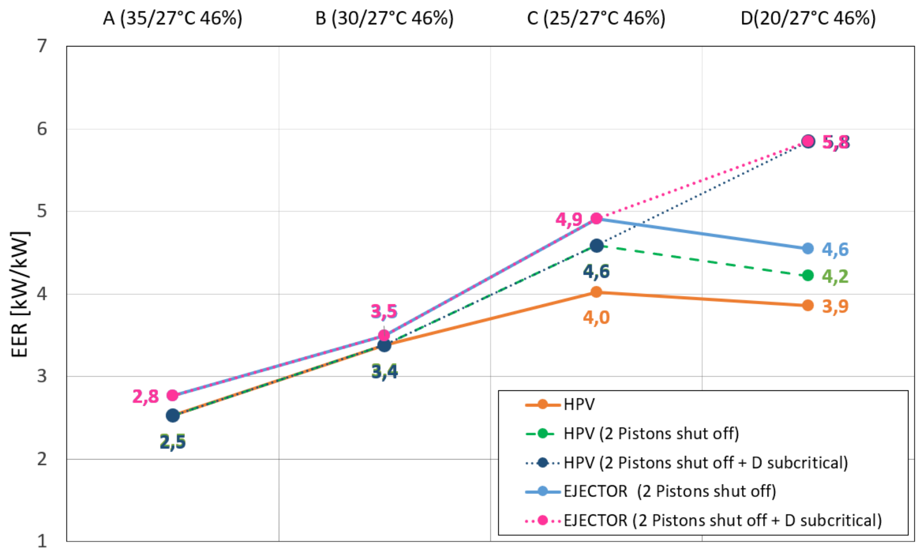


Figure 3. EER on all the cooling operating condition for each system configuration considered.

Table 7 reports the simulation results for the R744 rooftop both for *HPV (2 Pistons shutoff + Dsub)* and for *Ejector (2 Pistons shutoff + Dsub)* cases, which are those delivering the highest efficiencies.

Table 7: Overall simulation results for *HPV (2 Pistons shutoff + Dsub)* and *Ejector (2 Pistons shutoff + Dsub)*

| Air conditions | A | | B | | C | | D |
|---|---|---|---|---|---|---|---|
| Cycle Configuration | **HPV** | **Ejector** | **HPV** | **Ejector** | **HPV** | **Ejector** | **HPV / Ejector Sub critical** |
| Compressor speed (Hz) | 70 | 64 | 42 | 42 | 20 | 20 | 20 |
| Gas cooler pressure (bar) | 93 | 94 | 81 | 81 | 75 | 75 | 70 |
| Gas cooler approach (K) | 2.67 | 2.34 | 1.95 | 2.3 | 1.65 | 1.98 | 1.08 |
| Evaporator pressure (bar) | 45.6 | 45.9 | 48.9 | 48.48 | 53.9 | 53.93 | 52.59 |
| Suction superheat (K) | 10 | 10.8 | 10 | 9.3 | 5.2 | 4.8 | 2.4 |
| Cooling capacity (kW) | 48.53 | 48.95 | 36.36 | 36.97 | 22.47 | 22.62 | 25.06 |
| Compressor power (kW) | 15.02 | 14.30 | 7.45 | 7.30 | 3.05 | 2.77 | 2.49 |
| EER | 2.53 | 2.77 | 4.88 | 5.07 | 4.59 | 4.91 | 5.85 |

From the EERs reported in Table 7, it is possible to calculate $SEER_{HPV\ (2\ Pistons\ shutoff + Dsub)}$ and $SEER_{EJECTOR\ (2\ Pistons\ shutoff + Dsub)}$, equal to 4.1 and 4.55, respectively.

## 6. R744 Rooftop prototype

The previously designed R744 rooftop unit prototype will be experimentally tested in a second phase. The testing campaign aims to validate the proposed model and confirm the system's overall performance. The flexible setup will also allow for the testing of different configurations and the verification of oil return effectiveness, which is critical for ensuring long-term reliability. The P&ID of the rooftop R744 prototype is shown in Figure 4.

This prototype's design allows to operate the unit in both cooling and heating modes. As previously mentioned, during cooling mode, the outdoor coil works as a gas cooler/condenser and the indoor coil works as an evaporator to provide the required cooling capacity. During heating mode, the cycle is reversed – by means of the two 4-way valves – so that the outdoor air coil operates as an evaporator and the indoor air coil operates as a gas cooler to deliver heating capacity to the building. All the prototype components are selected and procured according to the sizes estimated using the described detailed simulation model. The components include:

- *Semi-hermetic reciprocating compressors* with permanent magnet motor. The compressor is equipped with switch-off valve that allows to switch off 2 pistons over 4. Maximum evaporating pressure up to 55 bars.
- *Oil separator with oil regulation system*. The set-up is also provided with a bypass circuit to allow excluding the oil separator.
- *Outdoor air heat exchanger*. The coil is meant to work as gas cooler during cooling operation and as flooded evaporator during heating operation; the circuiting is designed to ensure counterflow flow with the air while working as the gas cooler to achieve the target approach temperature. When working as the evaporator, the refrigerant will be co-current with the air.
- *Plate internal heat exchanger* circuit which includes a 3-way mixing valve to partially bypass the heat exchanger.
- *Indoor air heat exchanger* with coil circuiting design following the same principle of the outdoor air heat exchanger.
- *Suction accumulator* installed downstream of the evaporator to work as a liquid receiver under different operating conditions.
- *Oil return line* from the suction accumulator. It is used to regulate the compressor suction superheat and ensure oil return back to the compressor.

This prototype also enables the investigation of two different expansion devices: the HPV and ejector system. When the HPV is used as an expansion device for high-pressure regulation, the motive nozzle of the ejector is bypassed. Alternatively, when the ejector is used, the HPV is bypassed and the refrigerant flow is regulated through an additional valve (liquid feeding valve- LFV).

The set-up is equipped with pressure transducers and thermocouples, as shown in the P&ID, to measure the process parameters along the refrigerant circuit. In addition, a Coriolis mass flow meter is installed to measure the refrigerant flowrate and a liquid level meter is integrated with the suction accumulator to track the level of liquid inside. The airflow measurements are also performed, as the system is equipped with air temperature sensors, relative humidity sensors and air flow meters.

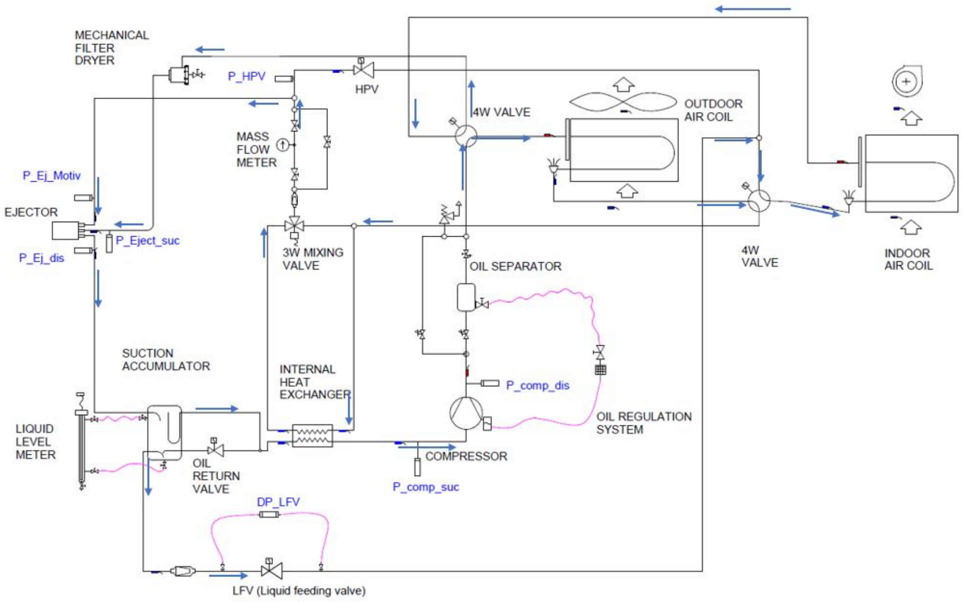


Figure 4. P&ID of the R744 rooftop prototype: Blue arrows are showing the refrigerant flow direction through the piping during cooling operation with the Ejector.

## 7. Conclusions

This study presents the methodology adopted to develop a rooftop air-to-air unit with R744. The major components are sized based on the results obtained from numerical simulations and the prototype is built. This structured procedure and the lessons learned can be summarized as follows:

(a) To reduce system complexity and use relatively simple control strategy it has been decided to use only one compressor instead of two. The selection of 4 cylinders inverter compressor, with the additional control feature of shut off control of 2 cylinders, allows to reduce the effective compressor swept volume to improve energy efficiency at part loads.

(b) The use of internal heat exchanger and the suction accumulator in the proposed R744 unit ensures flooded operation of the evaporator and maintains desired degree of suction of superheat under different operating conditions.

(c) The outdoor heat exchanger (fin-and-tube) is sized to achieve the target approach temperature and refrigerant velocity at the outlet, while the indoor heat exchanger (fin-and-tube) is sized to ensure flooded evaporation and target refrigerant velocity at the exit.

(d) The use of ejector as an expansion device improves the seasonal efficiency as compared to that with HPV. For condition A (Full load), EER of the R744 unit with ejector improves by 9.5% as compared to that with HPV.

(e) SEER for the R744 unit with HPV is estimated to be 4.10, while the R744 unit with ejector has a SEER of 4.55.

The next phase will focus on collecting the experimental data from the prototype's operation. This will lead to a comparison against the model's results and an analysis of possible areas for improvement.

*References*

[1] L. Belussi, B. Barozzi, A. Bellazzi, L. Danza, A. Devitofrancesco, C. Fanciulli, M. Ghellere, G. Guazzi, I. Meroni, F. Salamone, F. Scamoni and C. Scrosati, "A review of performance of zero energy buildings and energy efficiency solutions," *Journal of Building Engineering,* vol. Volume 25, no. ISSN 2352-7102, 2019.

[2] European Commission, «Ventilation Units,» 2024. [Online]. Available: https://energy-efficient-products.ec.europa.eu/product-list/ventilation-units_en#consumers-1. [Consultato il giorno 13 November 2025].

[3] L. Wierda e A. Zanuttini, «Ecodesign impact accounting, Overview report 2024,» European Union, Delft, 2024.

[4] European Parliament and Council of European Union, *Regulation (EU) 2024/573 of the European Parliament and of the Council of 7 February 2024 on fluorinated greenhouse gases,* 2024.

[5] M.-H. Kim, J. Pettersen e C. W. Bullard, «Fundamental process and system design issues in CO2 vapor compression systems,» *Progress in Energy and Combustion Science,* vol. 30, p. 119–174, 2004.

[6] A. Hafner, S. Försterling e K. Banasiak, «Multi-ejector concept for R-744 supermarket refrigeration,» *International Journal of Refrigeration,* vol. 43, pp. 1-13, 2014.

[7] S. Thanasoulas, «Investigation of heat recovery in CO2 refrigeration cycles for supermarkets: A mechanism for determining optimal discharge pressure,» *Applied Thermal Engineering,* Vol. %1 di %2Volume 274, Part B, 2025.

[8] E. Zanetti, S. Bordignon, R. Conte, A. Bisi, M. Azzolin e A. Zarrella, «Experimental and numerical analysis of a CO2 dual-source heat pump with PVT evaporators for residential heating applications,» *Applied Thermal Engineering,* vol. 233, 2023.

[9] Y. S. K. Reddy, A. Guruchethan, S. Singh, S. Kumar K, M. Maiya e A. Hafner, «CO2 heat pump integrated thermal storage for domestic hot water in hotels,» *Journal of Building Engineering,* vol. 89, 2024.

[10] C. Nicolandrea, M. Rita, W. M. Alfonso, R. Paola e T. Marcello, «Performance analysis of a rooftop, air-to-air heat pump working with CO2,» *Applied Thermal Engineering,* vol. 75, pp. 1046-1054, 2015.

[11] Dassault Systèmes, *Dymola,* Lund, Sweden, 2025.

[12] TLK-Thermo GmbH, *TIL (Thermal Insulation Library),* Braunschweig, Germany, 2024.

[13] «Bitzer Software,» [Online]. Available: https://www.bitzer.de/websoftware/calculate/HHK/?tab=results.