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# INTEGRATED R744 EJECTOR SUPPORTED PARALLEL COMPRESSION RACKS FOR SUPERMARKETS. EXPERIMENTAL RESULTS

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## ABSTRACT

R744 is currently the preferred solution in the commercial refrigeration sector for most regions, but the limited efficiency of such systems under high temperatures constrains its expansion. Technologies such as parallel compression and ejectors allow the $CO_2$ horizon to be pushed closer to the equator, making $CO_2$ systems an interesting solution even under these environmental conditions.

An experimental facility that replicates the refrigeration system for a medium-size supermarket has been built and instrumented. The purpose of this setup is to study which of the potential $CO_2$ vapour compression configurations (booster, parallel compression, ejectors) leads to the lowest power consumption as a function of the ambient temperature, with given refrigeration (and AC) loads. The paper also focuses on the benefits that are associated with increasing the medium-temperature evaporation level, possible due to the combined use of flooded evaporators and liquid ejectors, and with modifying the intermediate pressure (downstream of the ejectors and high-pressure valve).

**Keywords:** Commercial refrigeration, R744, Ejector, Parallel compression

## 1. INTRODUCTION

$CO_2$ transcritical refrigeration systems have proved their reliability and good performance in Northern Europe, using mainly booster systems and heat recovery to meet the heating demands of the supermarket (Karampour and Sawalha, 2017; Sawalha, 2013). According to Karampour and Sawalha (2018), there are two main lines to spread these systems to warmer locations: integration of other demands, such as air conditioning, in a compact unit, and modification of the booster system to adapt it to the warmer temperatures. In this second line is included for example parallel compression, flooded operation of evaporators or mechanical subcooling.

Some simulation studies can be found where different configurations of $CO_2$ transcritical refrigeration systems are compared at different operating conditions, such as Karampour and Sawalha (2018) or (Pardiñas et al. 2018). The former quantify the reduction of annual energy use with parallel compression, compared to standard booster system, in a range between 3% and 7%. The actual value within this range depends on the location of the unit. The latter determined that the power consumption reduction due to the introduction of parallel compressors in the booster system could account for 17% at 30 °C ambient temperature. A further 6% reduction was calculated with the implementation of high pressure lift vapor ejector.

Despite it is possible to find some experimental studies that compare the different configurations of $CO_2$ refrigeration systems for commercial applications, it is useful to enlarge the existing database with reliable results of actual supermarket refrigeration units under controlled conditions. In NTNU/SINTEF laboratory in Trondheim (Norway) there is such a compressor pack available which was sized for maximum loads at medium temperature (MT), low temperature (LT) and AC of 60 kW, 15 kW and 60 kW, respectively. This paper describes first the refrigeration system installed. It continues with the experimental setup used for the adjustment and measurement of the loads of the system. Later on, the procedure to determine the key performance factors is explained. Finally, the first group of results comparing the different configurations (booster, booster + parallel compression, booster + parallel compression + vapor ejectors) are presented and discussed.

## 2. EXPERIMENTAL SETUP

Figure 1 shows a simplified diagram of the experimental facility newly installed at the SINTEF/NTNU laboratory, which consists of the $CO_2$ refrigeration system and several auxiliary circuits. It is versatile and allows experiments to be performed with different system configurations (booster, parallel compression, ejector supported parallel compression, etc.) and for a wide range of operating conditions.

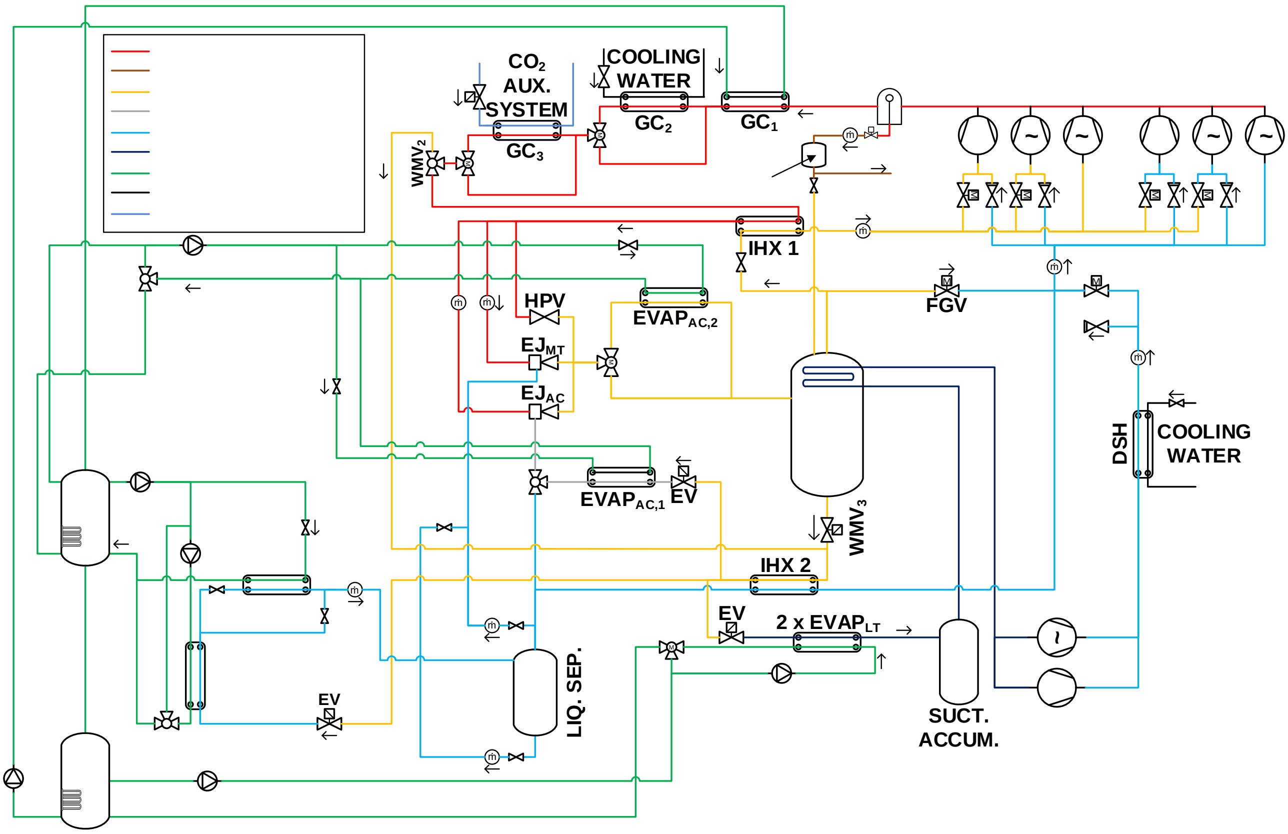


*Figure 1. Simplified diagram of the refrigeration system and secondary loops in NTNU/SINTEF laboratory (Trondheim, Norway).*

Eight semi-hermetic piston compressors were installed in the unit (Table 1), all of them manufactured by Bitzer (Bitzer Kühlmaschinenbau GmbH, Sindelfingen, Germany). There are two compressors in the low temperature (LT) section ($LT_1$ and $LT_2$), one directly connected to the medium temperature (MT) evaporators' pressure level ($MT_1$), one directly connected to the liquid receiver ($PAR_1$), and four machines that can work either as parallel or base compressors ($MT_2$, $MT_3$, $PAR_2$ and $PAR_3$). These "pivoting" compressors have a system with two valves, a check valve and a solenoid valve, to select the suction level for each of these compressors. In this case, $MT_2$ and $MT_3$ are always set as MT compressors; $PAR_2$ and $PAR_3$ as parallel compressors.

*Table 1. Characteristics of the compressors utilised in the supermarket mock-up facility*

| Model | Name | Swept volume [$m^3 \cdot h^{-1}$] at 50 Hz | Inverter driven? |
|---|---|---|---|
| 2JME-3K | $LT_1$ | 3.5 | Yes |
| 2GME-4K | $LT_2$ | 5.0 | No |
| 4MTC-10K-40S | $MT_1$ / $MT_2$ | 6.5 | Yes |
| 2KTE-7K-40S | $PAR_1$ / $PAR_2$ | 4.8 | Yes |
| 4JTC-15K-40P | $MT_3$ / $PAR_3$ | 9.2 | No |

The high pressure part of the test facility allows safe operation up to 130 bar. It consists of three gas coolers (plate heat exchangers) implemented downstream of the compressors to reject heat at different temperature levels. In the first heat exchanger, $GC_1$, R744 transfers heat to the glycol circuit (that is also used to supply load to the different evaporators). The second heat exchanger, $GC_2$, allows the heat rejection from the refrigerant to a cooling water loop. The third gas cooler, $GC_3$, uses an auxiliary R744 refrigeration system as heat sink, which makes it possible to reach relatively low refrigerant temperatures at the outlet of the gas coolers. The temperature at the outlet of the gas coolers can be adjusted in two ways. The first is based on bypassing partially or totally $GC_2$ and $GC_3$ by means of the motorized three-way valves positioned downstream of each heat exchanger. The second option is to regulate the mass flow rate and inlet temperature of the secondary fluids. An internal heat exchanger, $IHX_1$, is located downstream of $GC_3$ and upstream of the high pressure control devices, to superheat the suction flow to the parallel compressors at elevated $CO_2$ gas cooler outlet temperatures.

Concerning the high pressure control devices, there are several alternatives. A normal (high pressure valve) HPV enables operation in booster configuration, coordinated with the flash-gas bypass section, FGV. Two multiejector blocks are also installed. The MT multiejector block, $EJ_{MT}$, is a low entrainment ratio and high pressure lift block, with six fixed-geometry ejector cartridges that can suck either vapor or liquid from the separator at the MT pressure level. Two cartridges are specifically designed for liquid pumping (liquid ejectors). Concerning the AC multiejector block, $EJ_{AC}$, it has six fixed-geometry ejector cartridges and was conceived as a high entrainment ratio block, able to suck all the vapor produced at the AC evaporator positioned at its suction port, $EVAP_{AC,1}$. Further information about the ejector blocks can be found in Kriezi et al. (2016).

The liquid receiver positioned downstream of the high pressure control devices (and optionally of AC evaporator $EVAP_{AC,2}$) consists of two reservoirs of 130 L each. In booster operation, the pressure level at the receiver is controlled by means of the FGV. In parallel configuration, it is adjusted with the parallel compressors if the amount of vapor is enough compared to their minimum capacity. The parallel compressor section sucks vapor from the liquid receiver, with a final superheating provided by $IHX_1$. The liquid port of the receiver is connected to the expansion devices of the different evaporators and cabinets, through another internal heat exchanger, $IHX_2$, which subcools the liquid from the receiver and superheats the vapor sucked by the MT compressor(s).

The MT evaporators are constituted of five helical coaxial tube-in-tube heat exchangers assembled in parallel. Each evaporator has its individual expansion device. The refrigerant flowing out from MT evaporators reaches the liquid separator (60 L). The MT compressor(s) suck vapor refrigerant from the top part of this tank. In addition, the top and bottom ports of the liquid separator are connected to the suction of the MT multiejector block, enabling entrainment of vapor, liquid or both fractions simultaneously, depending on the test. The tank is equipped with a liquid level indicator that controls the operation of the liquid ejectors and prevents liquid from reaching the MT compressors.

The LT evaporators are two helical coaxial tube-in-tube heat exchangers of the same characteristics as the MT evaporators, also with individual expansion devices. An auxiliary suction accumulator (25 L) was applied downstream of the LT evaporators. The main goal of this tank is to guarantee safe and undisturbed operation of the LT compressors. As an extra safety measure, the vapor from the outlet of the accumulator is superheated with a heat exchanger placed in the liquid receiver.

The refrigerant discharged by the LT compressors is cooled down in a plate heat exchanger, denoted as DSH, by a cooling water loop. This loop simulates the desuperheater that is usually found in refrigeration systems to reduce the temperature of the refrigerant at this point. Downstream of the desuperheater, the refrigerant may be delivered to either MT or parallel compressors with a system of solenoid valve/check valve. In this study it was always discharged to the suction of the MT compressors.

In addition, there are two AC evaporators in the unit, which were not used in the tests performed for this paper. The first AC evaporator ($EVAP_{AC,1}$) is downstream of the liquid receiver and uses an electronic expansion valve as a metering device to control the conditions of the refrigerant at the outlet of the evaporator. This AC evaporator can be operated as ejector supported, but it is also possible to connect it to the liquid separator through a manually operated three-way valve. The second AC evaporator ($EVAP_{AC,2}$) allows AC production to be integrated into any standard booster system and is placed at the liquid receiver pressure level, downstream of the high pressure control devices. In this unit it is possible to bypass it with a 3-way valve.

The oil management system consists mainly of two oil separators, an oil reservoir, solenoid valves that connect each oil separator with the oil reservoir, and oil metering devices that feed the returning oil to particular compressors. In addition there is a pressure-differential driven valve between the oil receiver and the liquid receiver that keeps the pressure of the former at a value slightly higher than the liquid receiver pressure (3 bar approximately). This guarantees a good distribution of oil to the parallel compressors.

The glycol loop (propylene glycol-water solution 30 %vol.), represented in green in Figure 1, is used to both absorb heat from and reject heat to the main refrigerant loop. It consists of two buffer tanks, and four individual sub-circuits. The capacity of each tank is 800 L, and four electric heaters per tank, with a total capacity of 24 kW are installed and help to regulate the temperature of the glycol. The first glycol sub-circuit absorbs heat from the refrigerant in $GC_1$. The second sub-circuit provides load to both AC evaporators. The third sub-circuit releases heat to the MT evaporators. The fourth and last sub-circuit is identical to the previous one and provides heat to the LT evaporators. Each individual evaporator has a system with inverter driven pump and motorized valve to adjust the conditions of the glycol and regulate the load.

The refrigerant and secondary fluids' loops were instrumented with different kinds of sensors. The mass flow of refrigerant in different locations was measured with Coriolis effect mass flow meters (Rheonik RHM of different sizes). The locations are indicated in Figure 1 with $\dot{m}$ inside a circle. Pressure transducers (Danfoss MBS8250) measure the pressure at different points of the refrigerant loop, and in combination with the different temperature sensors (Pt 100 Class B DIN 1/3 on the tube) allow the cycle to be characterized. Differential pressure sensors (Endress+Hauser Deltabar PMD75) are used to measure the pressure lift of each ejector block. The power consumption of each compressor is measured individually with an active power meter (Schneider Electric A9MEM3150). Pt100 temperature sensors are also placed on different locations of the secondary loops to measure the conditions of these fluids. Finally, the volumetric flow of glycol reaching each evaporator could be measured by means of ultrasonic energy meters (Honeywell EW773).

Two data acquisition systems were synchronized to register the data from the sensors. The first provided by Danfoss, responsible as well of the control of the refrigeration unit, had a sampling rate of 5 s. To this acquisition system were connected the pressure transducers, the differential pressure sensors, as well as Pt1000 temperature sensors used internally by the controllers. The second was a LabVIEW acquisition system, with a sampling rate of 1 s, in charge of measuring data from the Pt100 temperature sensors, active power meters, refrigerant mass flow meters and energy meters. The LabVIEW program was also used to control the different components of the secondary loops (pumps, valves, electric heaters) and adjust the loads and operation conditions of the refrigeration system.

## 3. EXPERIMENTAL CONDITIONS

The aim of the study presented in this paper was to compare the power consumption of the compressors of three configurations of $CO_2$ transcritical refrigeration systems (booster, booster + parallel compression, booster + parallel compression + MT vapor ejectors) at different conditions and with set loads at MT and LT. The conditions chosen for this study were:

- MT load: 30 kW approximately. This value corresponds to the maximum MT load that could be met with the three MT compressors ($MT_1$, $MT_2$ and $MT_3$) running at full capacity, with the temperature at the outlet of the gas cooler equal to 45 ºC, receiver pressure 39 bar and discharge pressure calculated by the controller (curve 40 °C to 100 bar gage).
- LT load: 15 kW approximately. This value corresponds to a cooling load close to the maximum that could be reached with the two LT compressors ($LT_1$ and $LT_2$).
- MT evaporation temperature: -8 °C. Superheat control of the expansion devices, setpoint of 10 K.
- LT evaporation temperature: -30 °C. Superheat control of the expansion devices, setpoint of 10 K.
- Temperature at the outlet of the gas cooler: 45 °C, 40 °C, 35 °C and 30 °C.
- Liquid receiver pressure: 39 bar.
- Discharge pressure setpoint: Calculated with optimization curves as a function of the temperature at the outlet of the gas cooler. These curves are set with the temperature at the outlet of the gas cooler

that corresponds to 100 bar gage. The setpoints for the temperature considered were 40 °C, 39 °C, 38 °C, 37 °C and 36 °C.

- Temperature outlet of the desuperheater: 25 ºC.

Under these conditions, two of the five MT evaporators and one of the two LT evaporators were activated.

## 4. DATA ANALYSIS

The loads at the different evaporators were calculated both in the $CO_2$ side, following Eq. (1) and in the glycol side, with Eq. (2). The enthalpies for the refrigerant were determined using REFPROP 9.1 database (Lemmon et al., 2013). The properties of the glycol solution came from the ASHRAE Handbook Fundamentals (2009).

$$\dot{Q}_{ref,evap} = \dot{m}_{ref,evap}\,(h_{ref,out,evap} - h_{ref,in,evap}) \quad (1)$$

$$\dot{Q}_{glycol,evap} = \dot{V}_{glycol,evap}\,\rho_{glycol}\,cp_{glycol}\,(T_{glycol,in,evap} - T_{glycol,out,evap}) \quad (2)$$

Figure 2 shows the comparison of the loads calculated with these equations for each test performed. It can be seen that in the case of LT loads (orange) there is a very good agreement between the results obtained with both equations. In the case of the MT evaporators it is still acceptable, with deviations that range mostly between -5% and 15%. Taking into account the uncertainty of the different sensors, it was assumed as the right value of load that obtained in the $CO_2$ side.

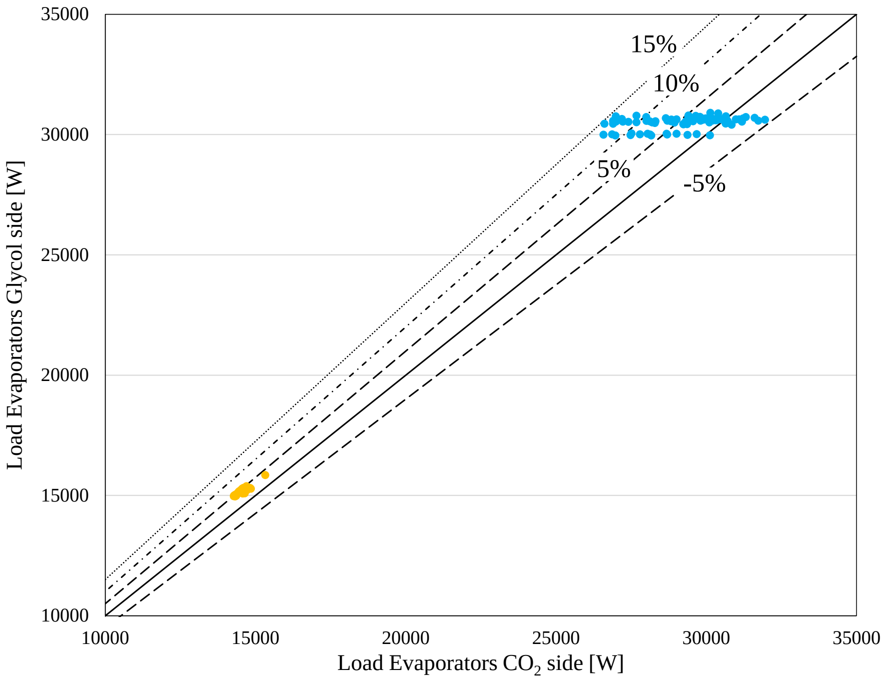


*Figure 2. Load calculated in the glycol side vs. load calculated in the $CO_2$ side in the MT (blue) and LT (orange) evaporators.*

Figure 2 represents as well that the loads, particularly the MT load, differed for the different tests, and therefore it would not be fair to compare the different configurations with the power consumption of the compressors. This explains the need of defining a performance factor, *COP*, calculated as in Eq. (3).

$$COP = (\dot{Q}_{ref,MT} + \dot{Q}_{ref,LT}) / (\dot{E}_{comp,MT} + \dot{E}_{comp,PAR} + \dot{E}_{comp,LT}) \quad (3)$$

The performance of the vapor ejectors was also calculated through the ejector efficiency, defined through Eq. (4) as in Elbel and Hrnjak (2008). $\Phi_m$ stands for the mass entrainment ratio, which is the ratio of the mass flow rate sucked by the ejector (secondary flow) to the mass flow rate at the motive nozzle (primary flow).

$$\eta_{ej} = \Phi_m\,(h_{s=suct,p=disc} - h_{suct}) / (h_{mot} - h_{s=mot,\,p=disc}) \quad (4)$$

Finally, the overall efficiency of each group of compressors ($\eta_{comp}$), i.e. MT compressors, parallel compressors and LT compressors, was determined through Eq. (5).

$$\eta_{comp} = \dot{m}_{comp}\, \Delta h_{is,comp} / \dot{E}_{comp} \qquad (5)$$

## 5. RESULTS

Figure 3 shows the COP of the refrigeration system vs. the high pressure, as a function of the configuration tested (booster, booster with parallel compression or ejector supported booster with parallel compression) and of the temperature at the outlet of the gas cooler. The implementation of parallel compression is clearly beneficial in terms of COP improvement, if kept constant the rest of the conditions. The positive effect decreases as the temperature at the outlet of the gas cooler is reduced. Meanwhile it could account for more than 20% at 45 °C, the improvement is reduced to 11% at 30 °C.

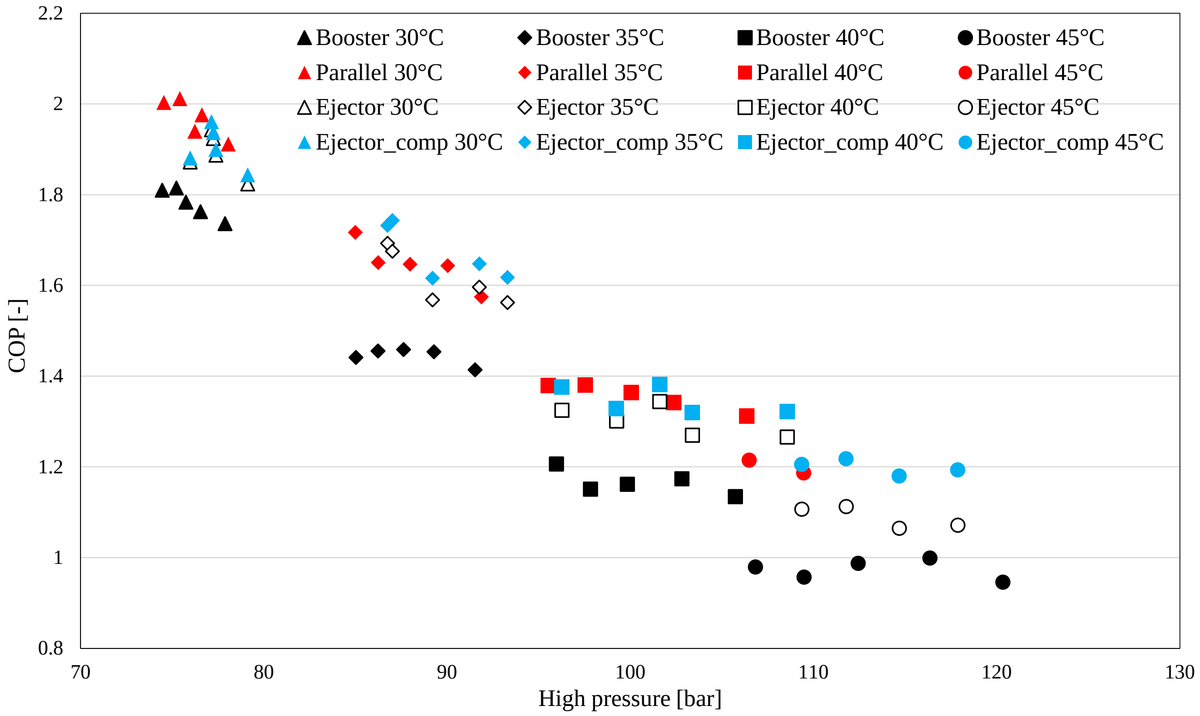


*Figure 3. COP of the refrigeration system vs. high pressure, as a function of the configuration tested (booster, booster with parallel compression, ejector supported booster with parallel compression) and of the temperature at the outlet of the gas cooler. In blue it is also shown the COP that would result if the deterioration of the overall efficiency of the MT compressors could be prevented.*

Concerning the MT ejector configuration, an enhancement of the performance would also be expected, or at least no deterioration compared to the booster system with parallel compression. However, the experimental results indicate that the performance of the ejector supported system was in many ocassions lower. This means that, even though part of the refrigerant from the liquid separator, at MT pressure, is compressed for free up to the pressure of the receiver, the overall power consumption is not reduced. Two potential causes could explain this phenomenon: low ejector efficiency or worse performance of the compressors when the ejector is in operation.

The ejector efficiency was calculated as indicated in the previous section and the results are included in Figure 4 left, as a function of the temperature at the outlet of the gas cooler. It can be seen that the efficiency is quite high with temperatures of 45 °C and 40 °C (greater than 25%), but it is actually in this cases when the ejector supported system is deteriorating more the COP. At 30 °C or 35 °C the ejector is not able to entrain much refrigerant due to the relatively high pressure lift of 11 bar (from 28 bar at the liquid separator to 39 bar at the liquid receiver) and the performance with and without ejector is very similar. Thus, we could conclude that the explanation is not coming from the ejector itself.

Figure 4 right compares the overall efficiency of the MT and Parallel compressors working in parallel compression configuration with the same overall efficiencies in equivalent tests (same temperature at the outlet of the gas cooler and discharge pressure) but in ejector supported configuration. The solid black line represents the y = x equation. Most of the points in the graph are to the left hand side of this line, which means that the overall efficiency of the compressors is in most cases better without the ejectors than with the ejectors. Thus, it seems clear that the deterioration of COP in the ejector supported configuration is caused by the worse performance of the compressors, particularly of the MT compressors. The reason that explains this worsening is that the MT compressors, once unloaded due to the ejector, operate at significantly lower frequencies.

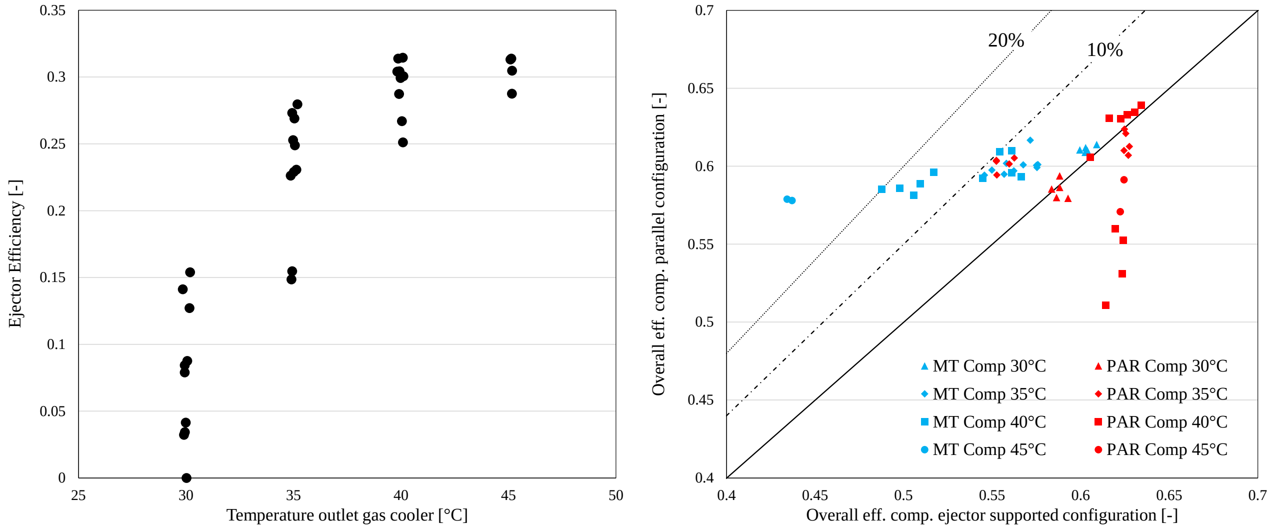


*Figure 4. Left, ejector efficiency as a function of the temperature at the outlet of the gas cooler. Right, overall efficiency of the MT and parallel compressors in the tests with parallel compression configuration, vs. overall efficiency of the MT and parallel compressors with ejector supported configuration under equivalent conditions (temperature at the outlet of the gas cooler and discharge pressure).*

The conclusion that arises at this point is that an ejector supported system, even with efficient vapor ejectors, is not reducing the energy consumption as expected if the compressors are not efficient in their range of operation when ejectors boost parallel compressors and unload MT compressors. The blue points in Figure 3 show what could be done if the MT compressors were able to operate, when unloaded by the ejector, at least with the efficiency they had in the equivalent parallel compression test. A slight improvement of the COP is observed in some of the tests with temperature at the outlet of the gas cooler from 35 °C to 45 °C. In the worst case scenario, the COP of the ejector supported system is almost the same as with parallel compression only.

## 6. CONCLUSIONS

This paper concerns the low capacity $CO_2$ transcritical refrigeration system installed and instrumented in the NTNU/SINTEF laboratory in Trondheim (Norway). The aim of this experimental setup is to measure how the refrigeration systems perform under different configurations (booster, parallel compression, ejector, etc.) and it is possible to simulate the conditions of operation (loads, temperature at the ambient of the gas cooler, etc.) for any global location, however with a limited refrigeration capacity, i.e. small semi hermetic compressors.

The results of the tests included in this study show that the implementation of parallel compression is very beneficial in terms of power consumption reduction in the range of gas cooler ambient conditions considered (from 30 °C to 45 °C). The expected enhancement of the COP due to the use of the MT ejector block was not observed in the test facility, due to the compressor configuration. It could be determined that the deterioration does not come from the ejector itself, but from the worse overall efficiency of the MT compressors when the ejector entrains part of the refrigerant from the liquid separator to the liquid receiver. In conclusion, manufacturers of compressors and inverters should focus on an efficiency improvement of their products when applied in these kind of applications, even for smaller capacity ranges as the case for the laboratory unit.

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## NOMENCLATURE

| | | | | | |
|---|---|---|---|---|---|
| comp | compressor | is | *isentropic* | ref | medium temperature |
| COP | *performance factor (-)* | *LT* | *low temperature* | s | *entropy* (J kg$^{-1}$ K$^{-1}$) |
| cp | spec. heat cap. (J kg$^{-1}$ K$^{-1}$) | $\dot{m}$ | *mass flow rate (kg s$^{-1}$)* | *suct* | suction ejector |
| disc | discharge ejector | *mot* | *motive* | *T* | temperature (K, °C) |
| $\dot{E}$ | power consumption (W) | MT | *medium temperature* | $\dot{V}$ | volumetric flow rate ($m^3$ $s^{-1}$) |
| ej | ejector | out | *outlet* | η | efficiency (-) |
| evap | evaporator | p | *pressure (Pa)* | $\Phi_m$ | mass entrainment ratio (-) |
| h | specific enthalpy (J kg$^{-1}$) | PAR | *parallel* | ρ | *density (kg m$^{-3}$)* |
| in | inlet | $\dot{Q}$ | *load* | | |

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