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# STRATEGY FOR THE CONTROL OF TWO GROUPS OF EJECTORS OPERATING IN PARALLEL IN INTEGRATED R744 REFRIGERATION SYSTEMS

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## ABSTRACT

Commercial refrigeration systems using $CO_2$ as refrigerant are becoming common in the South of Europe, due to technological solutions such as the use of ejectors or parallel compression that allow their efficiency to be enhanced even at high-temperature conditions. Under such conditions, there is also an AC demand and an interesting opportunity is to integrate it within the same refrigeration unit. A system with two groups of ejectors is proposed in order to achieve a successful integration of the loads in the same system: one dedicated to driving refrigerant from the medium-temperature to the intermediate pressure level and another to circulating the refrigerant through the AC evaporators. However, there is a challenge concerning the control of the high pressure of the cycle when there is a high ratio of AC load to refrigeration load or the AC ejectors perform inefficiently. Several solutions have been analyzed in this work.

**Keywords:** Commercial refrigeration, R744, Ejector, Control

## 1. INTRODUCTION

Transcritical $CO_2$ refrigeration systems are the preferred refrigerant for the commercial refrigeration sector in Northern Europe and this technology is spreading throughout Europe and in other locations in the world. According to Shecco's database, there are more than 9000 of such systems in Europe, more than 2700 in Japan, and the numbers are rising in other locations such North America, South America, South Africa or even Australia. It has been possible to reach these new regions with higher ambient temperatures due to the advances that have been introduced into the standard booster system, such as parallel compression, ejector technology and increase of the evaporation temperature due to flooded operation of the heat exchangers.

A further potential of $CO_2$ transcritical systems is the integration in the same unit of different demands, in addition to the needs of refrigeration at the temperature levels specific for supermarkets. Among these and depending on the location we can have heating, ventilation and air conditioning (HVAC), hot water demands or even snow melting. Tailor-made installations with integration exist and several successful examples can be found in Réhault and Kalz (2012) or in Hafner et al. (2014).

The most common trend concerning the production of AC with the $CO_2$ transcritical system is to produce chilled water and distribute it to the ventilation system or to the different heat exchangers (fan coil units) in the supermarket. A new tendency is to utilize directly $CO_2$, in a similar way as it is done with the cabinets and remove the intermediate fluid (chilled water), as mentioned in Girotto (2016). This allows the evaporation temperature at the AC evaporator to be increased, which should have a very positive effect on the power consumption of the unit (Pardiñas et al. 2018). In addition, this study also proved that it could be justified to have dedicated AC ejectors (low pressure lift and high entrainment ratio). In this way, liquid refrigerant from the receiver would be expanded to the AC evaporators and returned to the liquid receiver with the ejector.

The fact of having a dedicated ejector for AC production, in charge of adjusting the flow of refrigerant as a function of the load, implies that a different component needs to be in charge of the control of the high pressure, either a high-pressure valve or another ejector. Independently of the component chosen, a challenge is expected concerning the right control of this high pressure when there is a relatively high ratio of AC load to the total refrigeration load, or if the AC ejectors perform inefficiently. In this paper and by means of numerical simulations we analyse these situations and we propose and evaluate different alternatives to solve this challenge.

## 2. SYSTEM PROPOSED AND CONTROL CHALLENGE

$CO_2$ transcritical systems for refrigeration can be represented as in Figure 1 left. The simplest system, called booster system, consists of two groups of compressors, a gas cooler, evaporators at different temperature levels, a high pressure valve (HPV), a liquid receiver and a flash gas bypass valve (FGV). The main objective of such system is to meet the refrigeration demands at two temperature levels: medium temperature (MT) and low temperature (LT). Typically, the evaporation temperature in MT cabinets/evaporators ranges between -10 °C and -2 °C, while in LT cabinets it is between -40 °C and -24 °C. Two groups of compressors regulate these evaporation temperatures: MT compressors and LT compressor, respectively. A common and good practice to adjust smoothly the capacity of the compressor group to the demand and have a good regulation of the evaporation temperature is to implement at least one compressor per section with a variable speed drive, VSD.

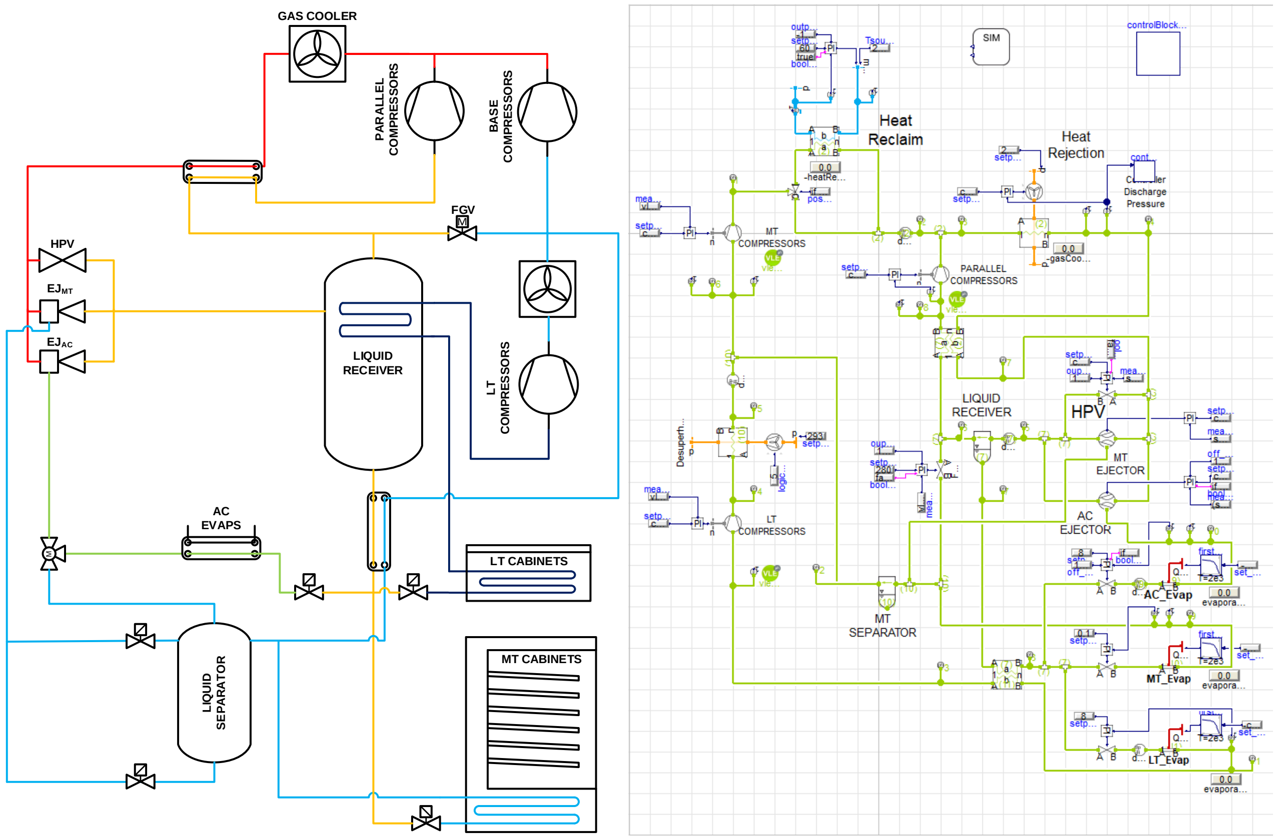


*Figure 1. Left, sketch of a transcritical $CO_2$ refrigeration system with integrated AC load and two ejector blocks in parallel. Right, caption of the model developed in Dymola/Modelica.*

The control of the intermediate pressure, or pressure at the liquid receiver, depends on the location where the unit is. In regions with low or mild temperatures throughout the year, a flash gas bypass valve (FGV) is used. It expands as much vapour from the liquid receiver to the suction of the MT compressors as is needed in order to keep the pressure at the receiver. The amount of vapour to be removed from the liquid receiver increases with the ambient temperature, which is also related to the temperature at the outlet of the gas coolers. Therefore, in locations with higher ambient temperatures, the solution of FGV is not efficient and parallel compression (PC) appears as the best possibility to remove the vapour from the receiver and keep the pressure.

The high pressure level has a very important effect on the performance of transcritical $CO_2$ refrigeration systems (Yang et al., 2015). For each system and conditions there should be a certain relation between high pressure and temperature at the outlet of the gas cooler that leads to the best performance. The simplest option to control the high pressure is the HPV, but the tendency nowadays is to implement ejectors to do the task. These devices not only adjust the high pressure, but also can use the expansion work available to suck refrigerant from a lower pressure level and lift it to an intermediate pressure. The case shown in Figure 1 left is what we would call a high pressure lift and low entrainment ratio ejector ($EJ_{MT}$), bringing refrigerant from the pressure level of the MT evaporators to the liquid receiver.

When a $CO_2$ transcritical refrigeration system has an integrated air conditioning (AC) load, it could make sense to install another multiejector block in parallel to the $EJ_{MT}$. This second one, designed for high entrainment ratio and low pressure lift, circulates all the refrigerant from the receiver through the AC evaporators and back to the receiver ($EJ_{AC}$) and maintains the conditions of the refrigerant at the AC evaporator in order to guarantee that the AC load is met.

There could be a challenge to maintain the discharge pressure when both ejectors are in operation, particularly when there is a high ratio AC/MT loads, i.e. when the AC load is relatively high compared to the MT load. In order to meet the AC load, the AC ejector could request an opening that would prevent the discharge pressure to be maintained even with the MT ejector fully closed. In addition, the decrease of the performance of the AC ejector as the temperature at the outlet of the gas cooler ($T_{GC,out}$) diminishes could further complicate this situation.

Several potential control strategies have been proposed and in order to restore the control of the discharge pressure by the MT ejector:

- force a higher setpoint of the high pressure
- decrease the setpoint of the receiver pressure
- reduce the setpoint of the pressure lift of the ejector
- force a reduction of the AC load, prioritizing the MT load and the discharge pressure control.

Some of these solutions have been simulated with a model explained below.

## 3. SIMULATION MODEL AND CONDITIONS

The challenge studied in this paper arose from a future commercial refrigeration system with integrated AC load that will be installed soon in CMD Marechal Gomes da Costa, in Portugal, and which will be part of the European Project MultiPACK (7231237). The system is basically that described in Figure 1, with parallel compressors and two ejector blocks in parallel. It was modelled in detail using Modelica object-oriented programming language in Dymola 2016 environment (Dassault Systems, Vélizy-Villacoublay, France). The models developed are based on TIL 3.4 library, and R744 properties are provided by the TILMedia 3.4 library, both from TLK-Thermo GmbH (Braunschweig, Germany). The effects of oil and of the oil management system were neglected.

The main aim of the model was to study the behavior of the ejectors. For the sake of simplicity of the simulations, continuous modulation of the opening degree (cross-section area of the flow-restricting channel, i.e., throat of the motive nozzle) was assumed for all the devices. The vapour MT multiejector block was modelled by correlations provided by Banasiak et al. (2015), both for the motive nozzle mass flow rate mass flow rate and for the compression performance of the ejector. Since the AC multiejector block was manufactured with the same motive nozzles as the MT block, the same correlation was used to calculate the motive nozzle mass flow rate. For the modelling of the secondary flow, different constant values for the ejector efficiency were considered (from 20% and 30% mainly). The ejector efficiency is defined as in Elbel and Hrnjak (2008). Modelling of liquid ejectors was avoided in this work, since we assumed that the expansion devices of the MT evaporators were able to adjust a very low superheat (very close to 0 K).

The rest of the components, such as compressors, heat exchangers, expansion valves, were modelled in a very simple way to guarantee the refrigerant flows and loads, but no special attention was drawn on the determination of the energy consumption of the system.

The design ambient temperature of the system that we are replicating with the model was 38 ºC was, with a temperature approach at the outlet of the gas cooler of 2 K ($T_{GC,out}$ = 40 ºC). The loads estimated during the design phase were:

- MT load. At the design temperature, the nominal load was 112.7 kW. The evaporation temperature in this case should be -2 ºC (almost 0 K superheat in the evaporators). A linear decay of the load as a function of the ambient temperature was assumed, with a minimum of the 60% of the nominal at 20 ºC ambient temperature.
- AC load. At the design temperature, the nominal load was 282.4 kW. The evaporation temperature depends on the simulation as explained later (control of the expansion device with superheating of 8 K). We assumed a linear decay of the load as a function of the ambient temperature, being 0 kW at 15 ºC ambient temperature.
- LT load. It was assumed as constant and equal to the nominal, i.e. 28.6 kW. The evaporation temperature in this case should be -25 ºC (control of the expansion device with superheating of 8 K).

The relation between the high pressure level and the temperature at the outlet of the gas cooler was obtained differently for the subcritical region than for the transcritical region, and considering a transition in between both. A subcooling degree of 5 K was considered at subcritical conditions, with maximum condensation temperature of 30 ºC. Transcritical conditions occurred whenever the temperature at the outlet of the gas cooler was greater than or equal to 30 ºC, and a linear correlation was established between the high pressure and this temperature defined by the points [30 ºC; 76 bar] and [42 ºC; 107 bar]. In this way, the temperature at the design point (ambient temperature 38 ºC and $T_{GC,out}$ = 40 ºC) would be lower than the maximum pressure of the system we are replicating (110 bar).

The first group of simulations performed had the aim of determining if the challenge described of losing control of the high pressure due to the operation in parallel of both blocks would ever occur. The parameters taken into consideration in this preliminary analysis were the ambient temperature (30 ºC and 20 ºC), AC ejector efficiency (20% and 30%) and of the receiver liquid receiver pressure (from 44 bar to 38 bar).

The second group of simulations was a more detailed parametrical study. The ambient temperature was changed from 20 ºC to 40 ºC in 5 ºC intervals. The aim of the AC ejector was to maintain the AC evaporation temperature at 5 ºC (39.7 bar), a value which is quite low taking into account direct AC production with $CO_2$, but still realistic and not unusual. The AC ejector was considered to be operating with pressure lifts from 2 bar to 5 bar, at 1 bar intervals. Therefore, the setpoint for the parallel compressor control, which is the liquid receiver pressure, depended on the test and was calculated as the sum of the AC evaporation pressure and the pressure lift at the ejector.

## 4. RESULTS

Figure 2 and Figure 3 show the values at which the high pressure stabilized for different simulations at ambient temperatures of 30 ºC and 20 ºC, respectively. In other words, they describe if the high pressure could be controlled by the MT ejector at the setpoint (function of the ambient temperature), even with the effects resulting from the AC ejector working in parallel. Figure 2 shows that, at 30 ºC and with the same ejector efficiency (20%), this challenging situation was achieved with receiver pressure of 44 bar, but not with 40 bar or 38 bar. This means that at 44 bar receiver pressure the amount of motive flow requested by the AC ejector to keep the sucked flow from the AC evaporator was almost the total coming from the gas coolers, leaving nothing for the MT ejector and the control of the high pressure. In contrast, the high pressure could be back into control at 44 bar if the ejector efficiency were 30%, since this improved performance would increase the entrainment ratio and reduce the need of motive flow.

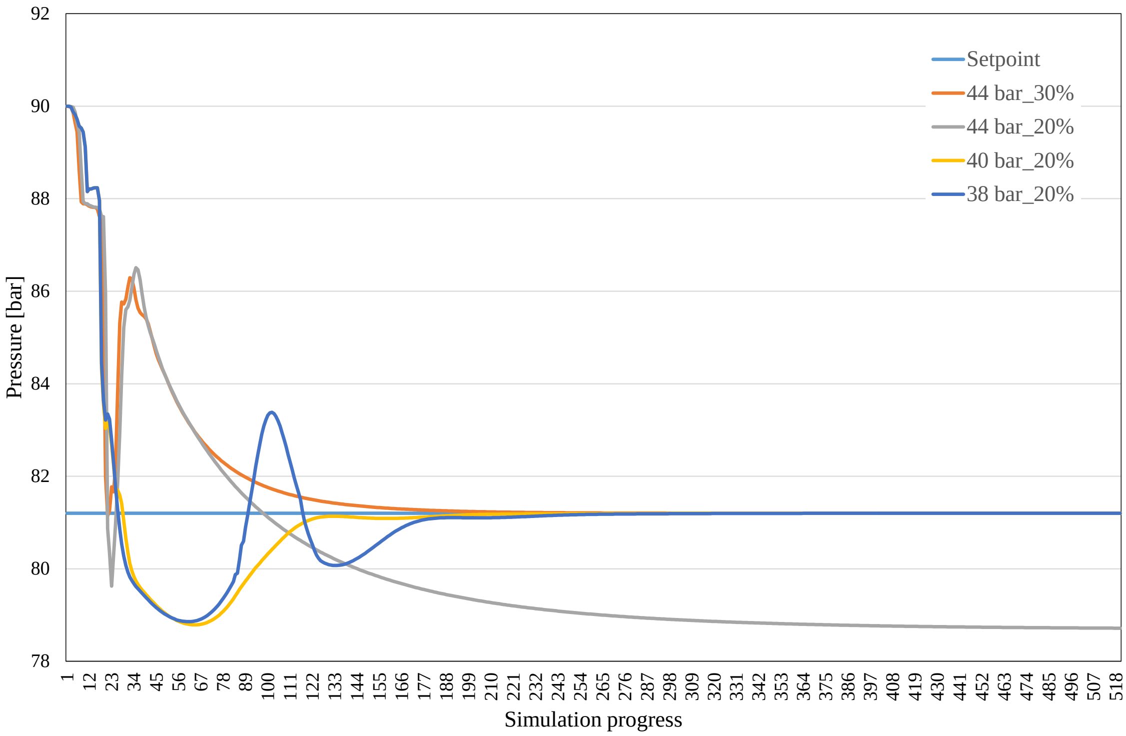


*Figure 2. Discharge pressure control in time, as a function of the receiver pressure and AC ejector efficiency and with ambient temperature of 30 ºC.*

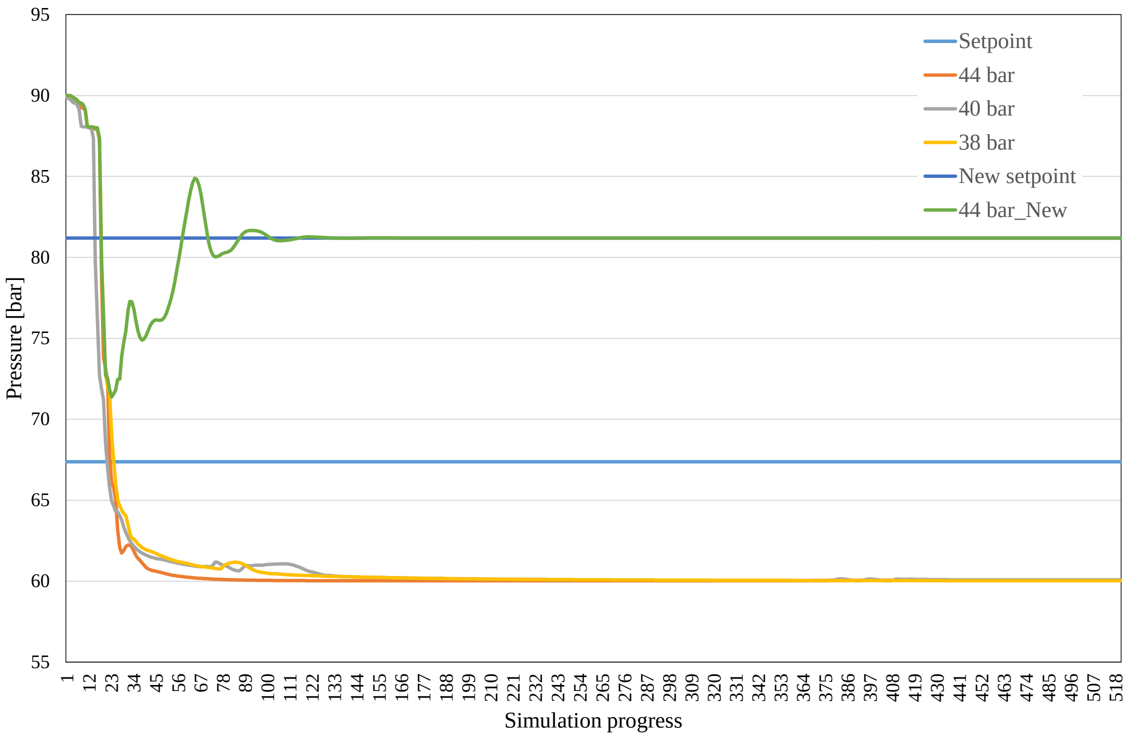


*Figure 3. Discharge pressure control in time, as a function of the receiver pressure and with reduced heat rejection in the air-cooled gas coolers (New setpoint and 44 bar_New in the legend), being the ambient temperature 20 ºC and the AC ejector efficiency 20%.*

Figure 3 represents similar results, but with ambient temperature of 20 ºC and AC ejector efficiency of 20%. In this case, the simple fact of reducing the receiver pressure is not enough in order to allow the regulation of the high pressure with the MT ejector. For such conditions, our simulations have shown that a potential solution is to force an increase of the high pressure setpoint ("New setpoint" and "44 bar_New" lines in Figure 3), for example by reducing the heat rejection at the gas coolers. If the heat rejection is decreased, the temperature approach at the outlet of the gas coolers could be increased and $T_{GC,out}$ lifted, with the consequent rise of the high pressure. This increases the expansion work available at the ejector motive nozzle and solves the challenge of the high pressure control. However this solution should be considered as a backup, since it leads to a significant increase of the power consumption of the system.

Concerning the parametrical study, some of the results on the capability of controlling the high pressure, as a function of the ambient temperature, and pressure lift and efficiency of the AC ejector, are shown in Figure 4. At ambient temperatures of 40 ºC or 35 ºC no challenge was observed, independently of the rest of the parameters, since the expansion work recoverable was always enough to guarantee the right operation of both ejectors. Thus and for the sake of simplicity, these results have not been included in Figure 4.

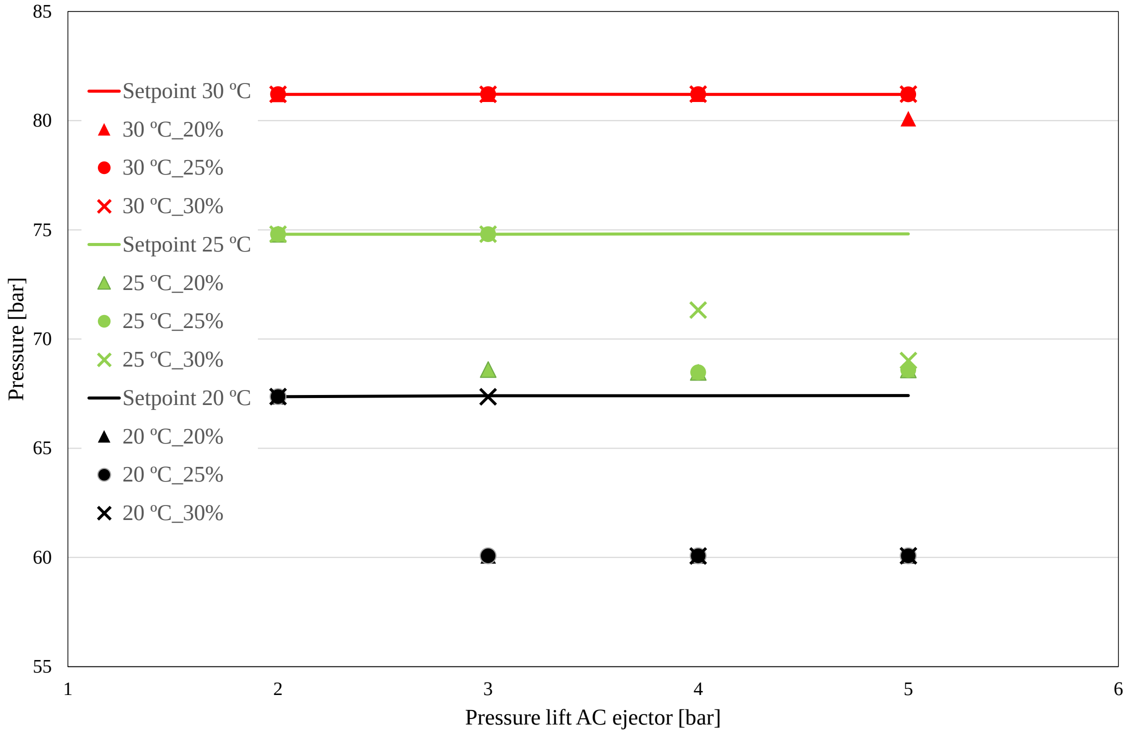


*Figure 4. Capability of discharge pressure control as a function of the ambient temperature and of the AC ejector pressure lift and efficiency.*

As the ambient temperature decreases, it becomes more common that the high pressure setpoints (represented with lines in Figure 4) cannot be reached with this parallel operation of the MT and AC ejector blocks. These results are in line with those from the first group of simulations, but now the AC evaporation pressure was set (39.7 bar). From the figure it could be drawn that, if fixed the ejector efficiency and ambient temperature, forcing a reduction of the pressure lift at the AC ejector allows the high pressure to be regulated. For instance at 25 ºC and AC ejector efficiency 30%, the system is not properly controlled with 5 bar or 4 bar pressure lift, but it is with 3 bar.

The way to cut down the pressure lift with the AC evaporation pressure set by the AC ejector, is to reduce as much as needed the setpoint of the receiver pressure. This increases the capacity at which the parallel compressors need to operate. This solution is not perfect since it is convenient to keep the receiver pressure high to reduce the power consumption of the system (Pardiñas et al. 2018). It would be better in terms of performance to guarantee that the AC ejectors are able to operate with relatively high efficiency in all the range. In this line and as shown in Kvalsvik et al. (2017), the low pressure lift ejector (AC ejector) efficiency can be maximized with a correlation between the pressure at the motive nozzle ($p_{mot,AC}$ and in bar) and its pressure lift ($p_{lift,AC}$ and in bar), Eq. (1).

$$p_{lift,AC} = 0.075\ p_{mot} - 2.1 \quad (1)$$

As a result of all the simulations performed, the control strategy for an integrated $CO_2$ refrigeration system with 2 ejector blocks in parallel would be the following. The AC ejector block is controlling the AC evaporation temperature. The optimal pressure lift of the AC ejector is calculated as a function of the motive nozzle pressure, which depends also on the conditions at the outlet of the gas cooler and will be given by the controller. This optimal pressure lift is added to the AC evaporation pressure to obtain the liquid receiver pressure setpoint, used to control the capacity of the parallel compression. In the case the conditions lead to a situation where it is not possible to regulate the high pressure, meaning that the MT ejector is fully closed and the setpoint is not achieved, the controller will start to cut down the pressure lift (by reducing the setpoint of the receiver pressure) until the situation is solved. As a result, the system operates with a liquid receiver pressure setpoint that floats with time. Just as a backup solution, when the pressure lift cannot be reduced any more, the high pressure setpoint could be lifted as indicated above.

## 5. CONCLUSIONS

This paper concerns the study of $CO_2$ refrigeration systems for commercial applications with integrated AC production. In particular, it has been proposed to use in such units two different ejector blocks operating in parallel: one with the aim of controlling the high pressure level, the task conventionally assigned to the high pressure valve, and another to control the conditions at the AC evaporator. However, a challenge was expected and confirmed related to the control of the high pressure of the cycle when there is a high ratio of AC load compared to the MT load and AC ejectors perform inefficiently. Another parameter that contributes to this situation according to the simulations was the temperature at the outlet of the gas cooler, since the expansion work available at the ejector diminishes when this temperature decreases.

The results of a more detailed parametric analysis led to a proposal of control strategy for such systems with two ejector blocks, based on setting the AC evaporation pressure, calculating the ideal pressure lift for the AC ejector at those ambient conditions and using it to determine the setpoint for the control of parallel compressors. A reduction of this pressure lift should be applied if the high pressure control became impossible. This strategy will be simulated in the near future using models that replicate better the actual behavior of the different ejector blocks, as well as with an experimental facility currently available in the NTNU/SINTEF laboratory in Trondheim (Norway).

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## NOMENCLATURE

| | | | |
|---|---|---|---|
| AC | air conditioning | MT | medium temperature |
| EJ | ejector | $p_{lift,AC}$ | pressure lift AC ejector (bar) |
| FGV | flash gas bypass valve | $P_{mot,AC}$ | motive nozzle press. AC ejector (bar) |
| HVAC | heating ventilation and AC | $T_{GC,out}$ | temperature outlet gas cooler (ºC) |
| HPV | high pressure valve | VSD | variable speed drive |
| LT | low temperature | | |

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