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Multi Ejector and pivoting-supported R744 application with AC for supermarkets

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ABSTRACT

CO_2 refrigeration units are gaining market shares thanks to the ability to provide an energy efficient performance for industrial and commercial refrigeration applications in any climate. As with all the applications, the investment and operation costs should be kept as low as possible, to reduce the payback time and promote the introduction of innovative system solutions.

In this work the flexibility achieved by implementing the pivoting technology in a supermarket refrigeration application is investigated both at design- and partial load conditions. The air conditioning (AC) load is also considered within a wide range of ambient temperatures. The Multi Ejector block utilized will be analyzed in terms of performance and its effect on the compressor combinations at different operating conditions. The objective is to increase the flexibility of the centralized rack through a proper design and sizing of the compressor pack equipped with the pivoting technology, while maintaining the efficiency and reducing the investment costs. This work shows that a Multi Ejector pivoting-supported system will be beneficial from the flexibility and capital costs point of view, and the benefit will be more consistent if the AC load is part of the integrated system architecture. Furthermore, a thorough investigation has been conducted whenever the ejector capacity is too high compared to the load, proposing two alternative solutions.

Keywords: Trans-critical CO_2, Ejector, Pivoting compressor arrangement.

1. INTRODUCTION

The implementation of the EU F-Gas Regulation 517/2014[1] on fluorinated greenhouse gases has led to substitute these working fluids with less environmental-damaging refrigerants, such as CO_2. Carbon dioxide has proven its reliability in northern climates and even in warmer climates where the last developments allow to outperform HFC-based units [2]. These technological developments comprise overfed evaporators, mechanical subcooling and ejectors among others However the investment costs and less compactness of the new generation of CO_2 integrated refrigeration unit hold back their large-scale spread.

On the one hand, the vapour ejectors can transfer the load from the medium-temperature (MT) compressors to the parallel compressors enabling a power consumption reduction. On the other hand, they do not only increase the initial costs, but the parallel (IT) compressor swept volume considerably grows in summertime due to many reasons: flash vapor generated during the expansion process, the vapor entrained and pre-compressed by the ejector, the AC load which becomes more impactful at high temperatures. The considerable capacity needed both in the MT and IT side, can be reduced further with the adoption of the pivoting technology, as already discussed in [3, 4]. The pivoting compressors, i.e. technique that enables the MT and IT compressors to be widely interchangeable in order to reduce the total compressor capacity installed, allows a gap-free control of the refrigeration load. This has been investigated in [5] both numerically and experimentally, considering only MT- and LT (low-temperature) loads. The present article analyses numerically the pivoting compressors in a large supermarket ideally located in Southern Europe. The AC integration has been considered in wide range of operating conditions, as well as its effect on the ejector performance. The compressors combination chosen is described and verified for all the operating conditions, both at design and part-load conditions. Furthermore, the constraint of having AC load is discussed in the case where the ejector entrains too much refrigerant from the medium-temperature side allowing one or two solutions, depending on the ejector capacity utilized. The results are discussed focusing on the compressor-capacity installed, impact of ejector performance on the load distribution, two control strategies for the ejector capacity and a simplified cost analysis.

2. CO_2 SUPERMARKET REFRIGERATION UNIT WITH PIVOTING COMPRESSORS IN WARM CLIMATE

Figure 1 illustrates a CO_2 compressor rack for supermarket refrigeration including MT, LT and-AC load. The ejector supported system is equipped with an Internal Heat Exchanger (IHX) located downstream of the gas cooler, which subcools the high-pressure stream while superheating the vapour sucked by the parallel compressors in a controlled manner with a bypass valve.

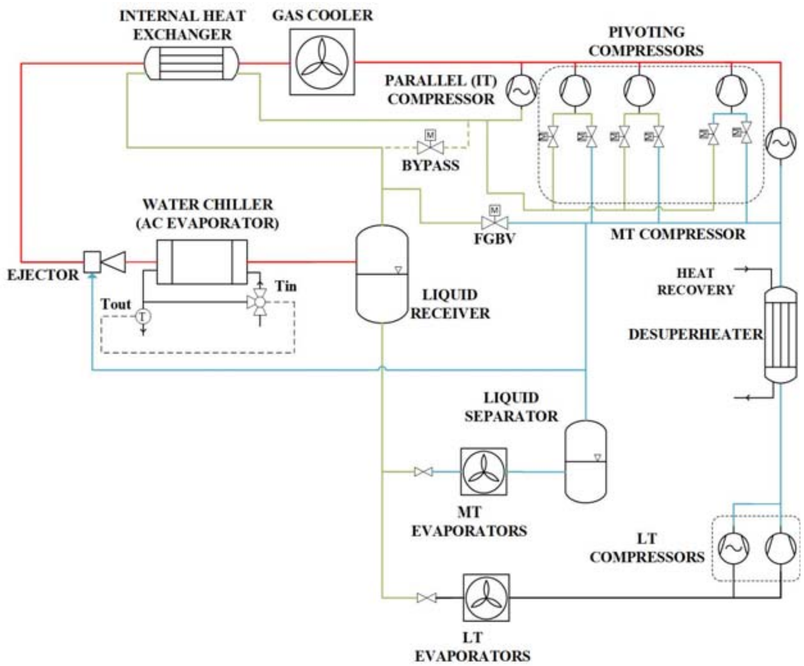

Figure 1: Ejector-supported CO_2 compressor rack with "pivoting" compressors.

The air conditioning demand is supplied through the water chiller (evaporator) which operates at the liquid receiver pressure and cools water in accordance to the AC demand. A three-way valve and a temperature sensor on the supply outlet temperature are present, and the bypass on the water circuit acts according to the requirements. The adjustment can be done both by adjusting the water mass flow rate or the water inlet temperature. The liquid receiver located downstream of the AC evaporator supplies the liquid to the different refrigeration cabinets. The two evaporating levels supply vapor to the two compression stages (LT-MT), where the LT compressors necessarily require cooling through a desuperheater because of the high discharge temperature of those compressors. The vapor at the outlet of the MT evaporators is stored in the liquid separator which supplies the refrigerant both to the MT compressors and the ejector. If the ejector capacity is such as to require more refrigerant flow, part or even all the flow coming from the desuperheater can be sucked by the ejector, after a mixing of the two flows. In the configuration illustrated, two valves upstream the compressor suctions ports are applied enable the so-called "pivoting" arrangement. Only one compressor is permanently dedicated to the MT side, as well as one to be a permanent parallel compressor while the other three compressors are "pivoting" compressors. This arrangement should provide an efficient capacity utilization of the compressors during the year. Regarding the low-temperature (LT) pressure level, two compressors work permanently with a given pressure ratio.

The shift regarding compressor capacities between the MT and AC temperature level becomes very important in an ejector-supported unit. The air conditioning is a considerable load in an integrated refrigeration system and demands large parallel compressor displacements. On the other hand, parallel compressors would stay idle during the cold months and therefore most of the installed capacity ends up unused. Therefore, a "pivoting" solution for compressors is a required function in the next generation of integrated R744 commercial refrigeration units.

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3. METHODOLOGIES

3.1 Operating conditions investigated and assumptions

A rack with defined parameters and capacities has been considered for all the simulations, which aimed at the typical refrigeration loads for a large-sized supermarket in a Southern European State. The CO_2 compressor rack is sized at the design condition, considering the maximum loads and the highest ambient temperature. In these conditions the number of compressors is selected and thereby the installed capacity can cover the cooling load, possibly without any disturbance in the system due to the variation of the pressures. The following data have been considered:

- Gas cooler outlet temperature varies in the range 10-40 °C, with the optimal high-pressure considered in each case (for transcritical conditions, https://www.ipu.dk/products/simple-one-stage-co2/).
- Receiver pressure 40 bar, i.e. AC evaporation at 5 °C.
- Multi Ejector unit (Multi Ejector HP 1875 LE 400 CTM 6 from Danfoss)[1] with the technical limits considered, i.e. maximum ejector suction superheat (15 K). Whenever the ejector capacity is such to require an amount of refrigerant which cannot be fully satisfied by the refrigerant flow at the outlet of the MT evaporators, the Multi Ejector unit can suck part or even the total flow coming from the outlet of the desuperheater. This mixing affects the thermodynamic properties of the flow rate at the suction port of the ejector.
- The IHX downstream the gas cooler unit allows to control the superheating of the parallel compressors by using a bypass valve, which is set to 10 K for each operating condition.
- MT Load 80 kW, at evaporation temperature -2 °C (approximate pressure of 33 bar, considering saturated vapour at the outlet of the evaporator.
- LT Load 20 kW, at evaporation temperature -26 °C (approximate pressure 16 bar, no superheat). Both LT compressors are in operation at each operating condition and they will not be considered in terms of capacity in the following sections. As discussed later, they will influence the ejector operation at part-load.
- AC Load is considered as a linear function of the gas cooler exit temperature (Ambient temperature) Figure 2:

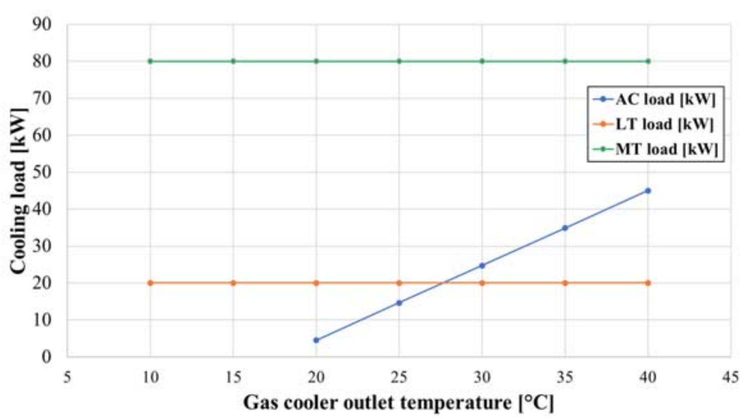

Figure 2: Cooling load profiles over the gas cooler outlet temperature range investigated.

The following assumptions are considered:

- A linear relationship that describes the temperature difference occurring between the CO_2 and the ambient air in the gas cooler/condenser, reaching 3 K and 0.5 K at 40 and 10 °C, respectively.
- Steady-state conditions are considered.
- The vapor stream and liquid stream exiting the liquid receiver are assumed to be saturated.
- Pressure drops and heat losses in pipeline and heat exchangers are neglected.

[1] (https://assets.danfoss.com/documents/DOC300732394440/DOC300732394440.pdf)

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- Superheat at the inlet of the LT compressor equal to 26 K, therefore with an inlet temperature of the LT compressors equal to 0 °C. The superheat is provided by a heat exchanger located inside the liquid receiver.
- Desuperheater located downstream the LT compressors is exchanging heat with the ambient air, considering a temperature approach of 1 K.
- 4 K of subcooling are considered in the condenser section at subcritical conditions.
- For the lowest investigated ambient temperature, the high-side pressure is kept at 55 bar to maintain a correct operation of the MT compressors, according to the operating envelope.
- Ejector efficiency has been calculated considering the CoolSelector software[2], for the Multi Ejector aforementioned. As it will discussed later, a trade-off between the IHX and the ejector exists [6, 7] and therefore an interpolation between the ejector efficiency and the subcooling degree is done.
- The Multi Ejector entrains part of the refrigerant from the MT suction at a gas cooler outlet temperature of 20 °C and above, while for lower gas cooler exit temperatures the ejector is acting as HPV.

3.2 Simulation model

This work explores the implementation of the "pivoting" compressor arrangement in a large-sized supermarket with a numerical analysis. The steady-state numerical model was programmed in EES (Engineering Equation Solver http://www.fchartsoftware.com/ees/). Concerning the Multi Ejector unit, the ejector efficiency was defined as in [8] with values that have been compared with CoolSelector. Realistic ejector efficiency values are essential to ensure a reliable simulation of the refrigeration unit, being the ejector the main component that affects the distribution of loads between the MT and IT (parallel) compressors. Compressors were modelled by applying the polynomial equations available in the software of the manufacturer (https://www.bitzer.de/websoftware/) as illustrated in Table 1. For compressors with Variable Speed Drive (VSD), the frequency range is also shown. The impact of the density in the mass flow rate displaced by a compressor is considered in the model, since the polynomials are evaluated for a reference superheating value equal to 10 K.

Table 1. Features of the compressors selected in the test-rig simulated and system where they are employed (P=pivoting-supported, NP=non pivoting-supported, VSD = Variable Speed Drive).

Compressor model	Reference Number	Operating mode (P)	Operating mode (NP)	Displacement [m^3/h] (at 50 Hz)	VSD	System
2FME-4K-40S	1	LT	LT	6.4	Yes (30-70 Hz)	P & NP
2GME-4K-40S	2	LT	LT	5	No	P & NP
4MTE-10K-40S	3	MT	MT	6.5	Yes (30-80 Hz)	P & NP
4MTE-10K-40S	4	Pivoting	IT	6.5	No	P & NP
4JTC-15K-40P	5	IT	IT	9.2	Yes (30-80 Hz)	P & NP
4HTE-20K-40P	6	Pivoting	IT	12	No	P &NP
4GTE-30K-40P	7	Pivoting	IT	15	No	P & NP
4MTE-10K-40S	8	/	MT	6.5	No	NP
4JTC-15K-40P	9	/	MT	9.2	No	NP

[2] (https://www.danfoss.com/en/service-and-support/downloads/dcs/coolselector-2/)

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4. RESULTS

4.1 Ejector performance

The performance of an ejector depends on its geometrical design and operating conditions, and it is very much influenced by the IHX located upstream of the motive port and the subcooling degree achieved in this component. In this study, the ejector efficiency and entrainment ratio were first evaluated in CoolSelector software, keeping fixed all the parameters except the subcooling degree in the IHX, which acts as a variable. Varying the high-side pressure and the relative gas cooler outlet temperature, a map for the ejector efficiency has been built for each operating condition as a function of the subcooling degree. This interpolation has been integrated later in the numerical model (Figure 3).

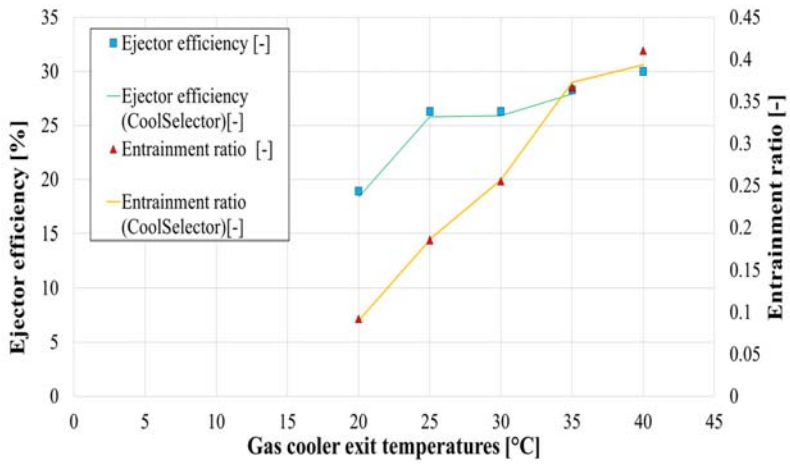

Figure 3:Ejector efficiency and entrainment ratio at the design load conditions.

A good agreement can be noticed between the simulation results and CoolSelector which has been used as a reference for the accuracy of the model, being based on an experimental campaign performed by SINTEF. The experimental campaign consists in 724 test points, divided as shown in Table 2. As it will be illustrated in the next section, lowers ambient temperatures lead towards a scenario where the ejector has a reduced performance and provides less unloading of the MT compressor load, i.e. the MT compressor gradually require more capacity.

Table 2: Number of points tested for the Multi Ejector HP 1875 LE 400 CTM 6 from Danfoss, by SINTEF.

Vapor cartridge	Cartridge 1	Cartridge 2	Cartridge 3	Cartridge 4
Number of points	463	126	58	77

4.2 Pivoting-supported system with Multi Ejector at the design conditions

Figure 4 shows how compressors must be distributed in the two different suction groups, MT and IT, as a function of the gas cooler outlet temperature and high pressure, with the aim to fully supply the cooling load and AC demand. Moreover, the “unused capacity” is illustrated at the top of each column.

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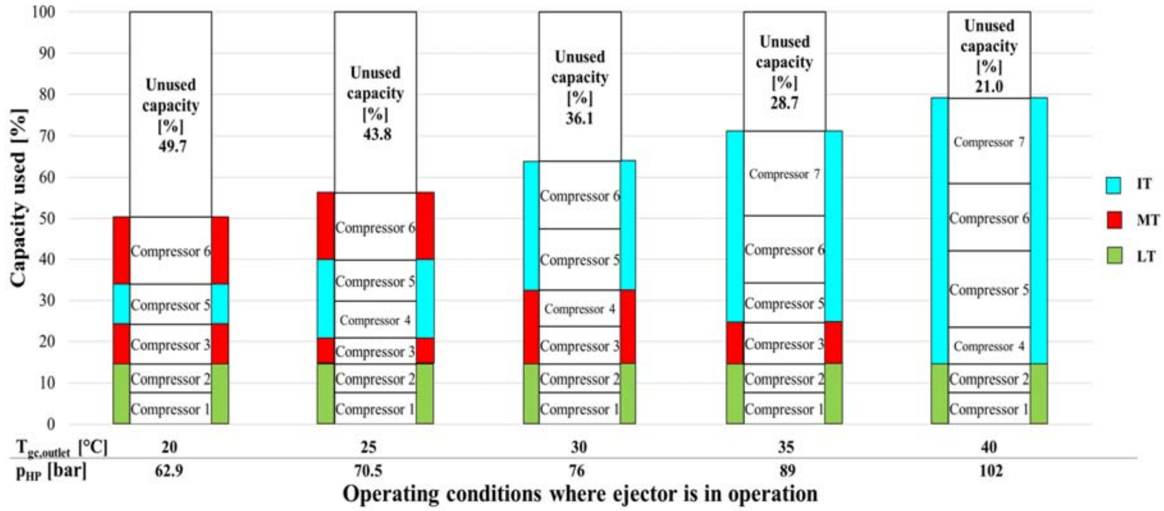

Figure 4: Effect of implementing "pivoting" compressors on the compressor-capacity used in a large-sized supermarket for such operating conditions where the Multi Ejector block is regulating the high-pressure.

As can be seen, the lowest "unused capacity" value appears at the highest temperature where the highest percentage of the compressor capacity installed is used. Four compressors (IT) and two compressors (LT) are needed in this case. It is interesting to notice that in this scenario the VSD compressor employed in the MT section is turned off, being the ejector able to deliver all the refrigerant from the MT pressure level to the receiver pressure level. Whenever the MT compressor is off, the MT evaporation temperature cannot be actively controlled, and it is a consequence of the ejector performance at the given conditions, as shown Figure 5.

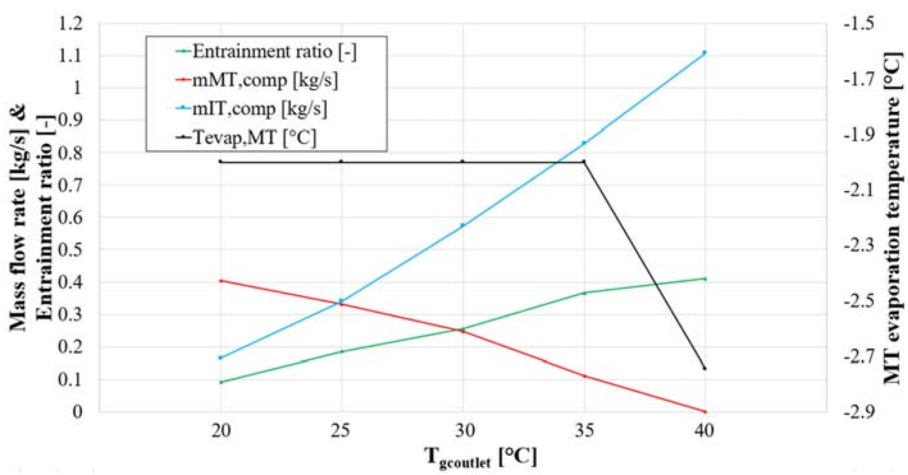

Figure 5: Mass flows, MT evaporation temperature and entrainment ratio as a function of the gas cooler outlet temperature.

Differently from a first impact, the relative high value of "unused capacity" at 40 °C is a result of two aspects: firstly the LT compressors must be sized in order to continuously supply a certain share of the cooling load in the event of a breakage of one of the them, secondly the VSD compressor employed in the MT section is off. This fact results in a great energy saving potential since all the flow is compressed with a lower pressure ratio, i.e. with the parallel compressors. The MT evaporation temperature slightly below the setpoint at 40 °C, which does not represent an issue happens because the ejector is able to remove more vapor towards the receiver

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pressure, if this pressure is kept fixed. An alternative approach would be to increase the pressure setpoint of the parallel compressors to reach higher pressure lift and lower entrainment ratio. It appears as an advantageous solution because of the lower pressure ratio at which the parallel compressors would work, but the constraint of air conditioning production can limit this alternative, depending on the design of the evaporator, which is strongly related to the method by which the heat is exchanged, i.e. direct or indirect cooling. Another interesting approach to increase the capacity in the MT section and enable the MT VSD compressor (3) would be CTES (Cold Thermal Energy Storage). The additional cooling load allows to avoid start/stop of the VSD compressors which is reflected in an oscillation of the evaporating temperature and consequently leading to a fluctuating air temperature inside the display cabinet.

At lower heat rejection temperature less refrigerant is pre-compressed by the ejector boosting the capacity required in the MT section. This trend becomes more marked proceeding from the right to the left (Figure 4). At 30 °C the MT compressors are not unloaded as much as at 40 and 35 °C, meaning that compressor 4 is employed in the MT section while the VSD (compressor 3) is adjusting its frequency. At 25 and 20 °C compressor 6 would be moved to supply the major capacity required in the MT compression stage, while the opposite happens to the IT compressors. Moving towards lower gas cooler outlet temperatures increases the "unused capacity" value. Below 20 °C the parallel compressors are not in operation, the flash gas is throttled by the flash gas bypass valve (FGBV) from the liquid receiver pressure level to the evaporator pressure level, and all the mass flow in the system must be compressed by the MT compressors (Figure 6).

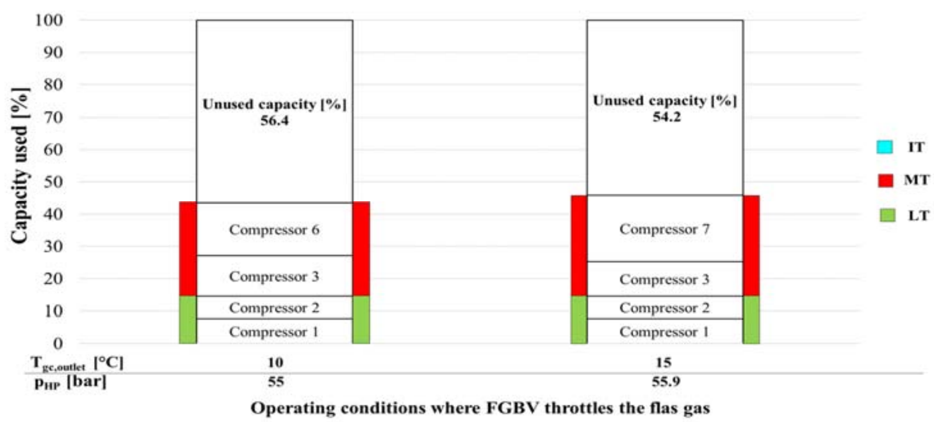

Figure 6:Effect of implementing "pivoting" compressors on the compressor-capacity used in a large-sized supermarket for such operating conditions when the ejector performs as a high-pressure control valve.

Figure 7 indicates that the MT and parallel compressors are efficiently used over a wide range of operating conditions. In this way, the cost of the rack can be reduced because of the lower number of compressors installed. Furthermore, since the definition of pivoting compressor implies a flexible utilization of the compressor pack, it can be seen how the compressor capacity utilization is relatively high both during summer and winter. By adopting the pivoting

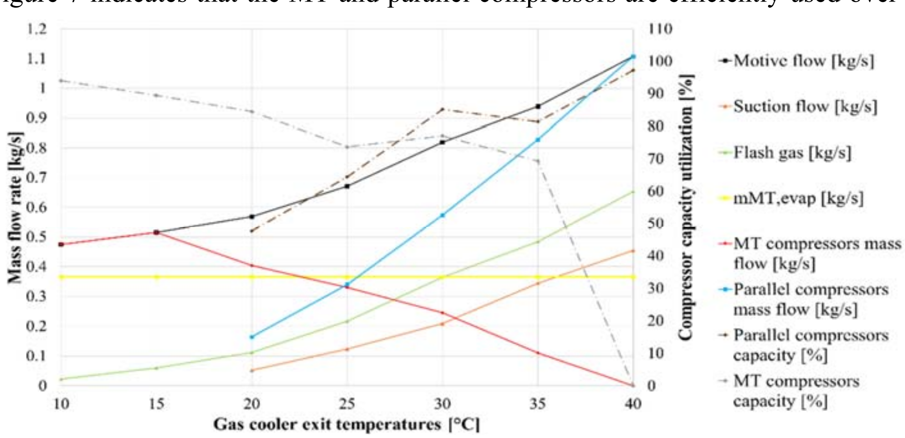

Figure 7:Mass flows and compressor utilization in a large-sized supermarket both during winter and summer.

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technology, the capacity of each section is never fixed, and it can vary according to the load and ambient conditions adjusting the compressor combinations based on what they really need. This allows to increase the hours of utilization of each compressor, overcoming a problem that is normally encountered in a large-sized supermarket. In fact, as already mentioned above, even with the pivoting technology if a large difference in terms of load is considered (i.e. AC during summer) some capacity ends to be unused but with a lower impact compared a Multi-Ejector unit without pivoting.

4.3 Pivoting-supported system with Multi Ejector at part-load conditions

The compressor pack chosen in this paper meets the refrigerating loads at MT part-load conditions from 100 to 20% of the design load. All these results are not shown here due to space constraints. The case at 35 °C was chosen as discussion scenario because it includes different dynamics that occur in the system. Figure 8 illustrates the actual capacity required in each scenario. Being the part-load operation occurring very often during the day, the performances of the compressor packs are very important to reach a good energy saving. Starting from the left to the right in the first depict, the ejector entrains too much refrigerant and therefore the evaporating temperature decreases to reach a new equilibrium where all the flow coming from the evaporators is entrained by the ejector. The reason why there are two cases in the second depict of Figure 8 at 60 and 70% is that in both cases the remaining amount of refrigerant not entrained by the ejector is insufficient to keep the MT VSD compressor in operation. However, a change of the evaporating temperature can handle the situation: in one scenario the MT evaporating temperature level increases slightly allowing the ejector to suck everything, keeping only the parallel compressors in operation. This scenario might be problematic since the evaporating temperature raises considerably (at 70%, it is equal to -0.9 °C), representing an issue for the food conservation.

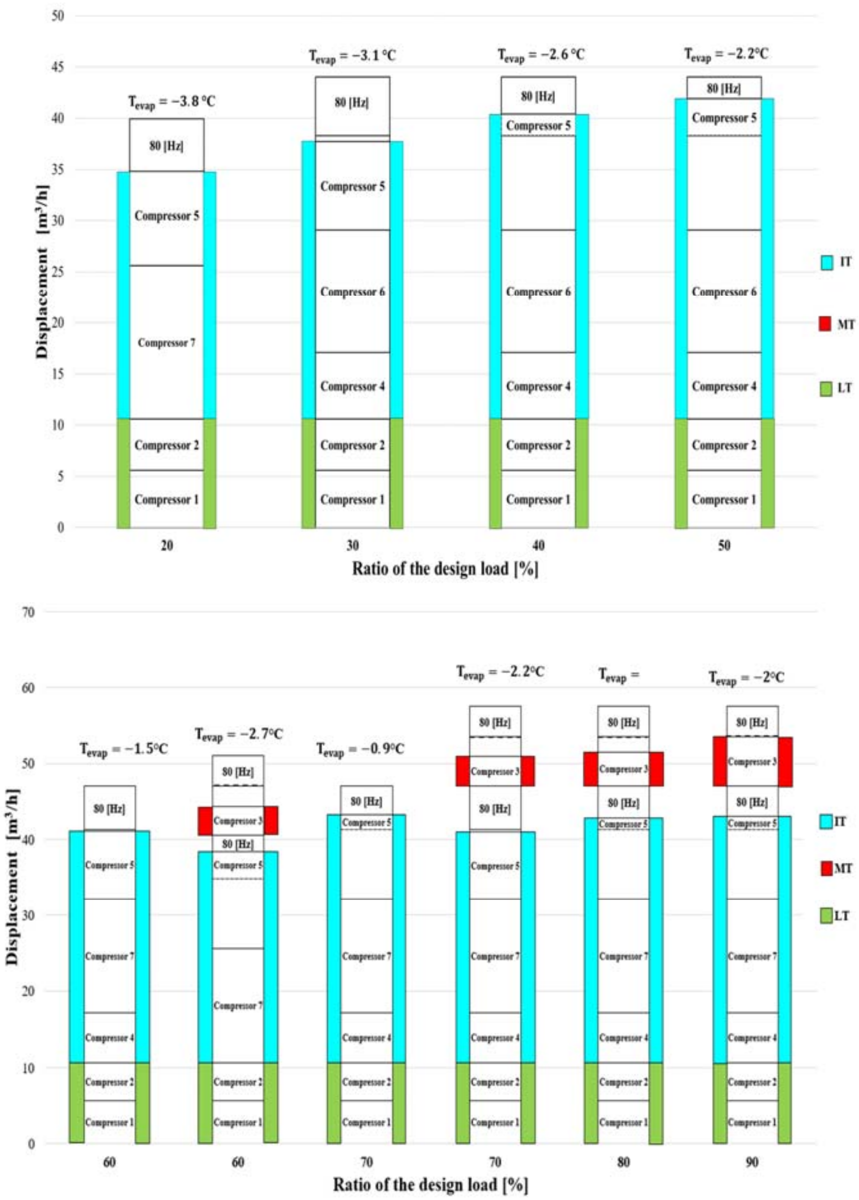

Figure 8:Compressor combinations adopted at part-load ($T_{gc,outlet}$ = 35 °C). At the top of each column the evaporation temperature.

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The other option is to further reduce the evaporating temperature (lower entrainment ratio) until the mass flow is such to maintain the VSD compressor at its minimum rotational speed (30 Hz). As it happened under the design conditions, at 80% only the VSD compressor is compressing the flow from the MT pressure level to the high-pressure level. It is interesting to notice the displacement in use in the parallel section at 80% of the design load is almost the same than what is required with 90%, due to the effect of the subcooling degree on the ejector. Firstly because of the lower motive flow the pressure losses inside the mixing chamber diminishes enhancing its performance. Secondly, although the suction temperature of the parallel compressors is fixed because of the bypass valve control, less refrigerant flows through the IHX at 80% of design load with a slight increase of the subcooling degree. Therefore, on one hand with a reduction of the MT load less flow is available in the MT pressure level but the trade-off between ejector-IHX still involves a significant amount of pre-compressed refrigerant.

Figure 9 illustrates how the MT evaporating temperature changes as a function of the percentage of the design load, with a look to the superheat of the ejector suction flow which can be as maximum the temperature reached when the two total mass flows (MT-LT) are mixed. For those conditions in which the ejector is entraining all the refrigerant from the MT-LT pressure level (from 20% to 70%), the entrainment ratio increases with the percentage of the design load because of the lower pressure lift given by higher values of the evaporating temperature. The lowest value of entrainment ratio can be recorded at 20%, with a drastic increment of the superheating of the suction flow. For a given percentage of the cooling load, a lower evaporating temperature would affect the superheat of the suction flow which consequently has an impact on the entrainment ratio. Therefore, if a fix evaporating temperature is considered, moving towards higher superheating degree would lead to reduced entrainment ratios. In the range 60%-90%, the system can work with the VSD compressor employed in the MT compression stage. At 80%, the evaporating temperature is still fixed to the setpoint, but the ejector capacity is such as to suck part of refrigerant flow coming from the desuperheater outlet, increasing the superheat of the suction flow. At 90%, the suction temperature is equal to the temperature of the saturated vapor exiting the MT evaporators and no flow needs to be taken from the outlet of the desuperheater. For part-load in the range 60-90%, the superheat at the MT compressors suction results very high since the LT mass flow is relatively high in comparison to residual MT evaporator's flow. This huge superheat will clearly affect the compressor performance due to the superheating losses at the suction line, resulting in high discharge temperatures at extreme ambient conditions and reducing the ability to compress a certain amount of refrigerant.

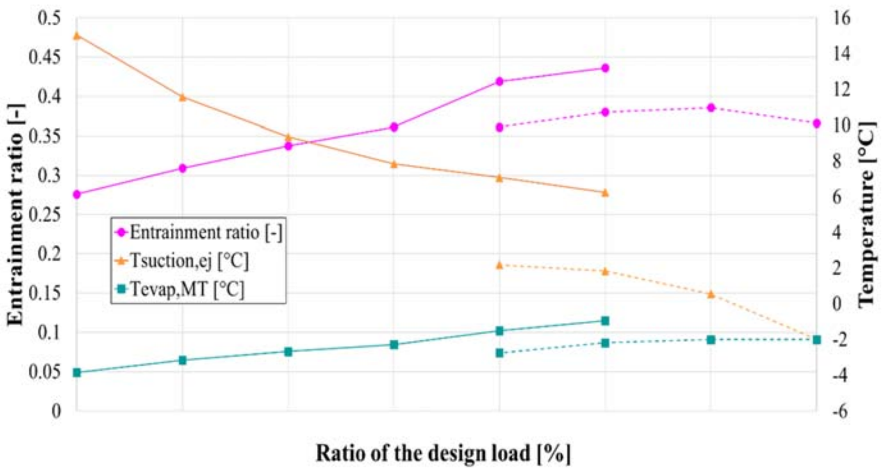

Figure 9: Evaporating temperature profile at part-load conditions with $T_{gc,outlet}$ = 35 [°C], p_{HP} = 89[bar] (straight line = only ejector sucking; dotted line = MT compressor in operation).

4.4 Floating control of the receiver pressure level

The evaporating temperature has a significant effect on the number of defrost cycles required and correspondingly on the shelf life of the products stored in the chilled food cabinets. When the evaporation temperature moves away from the setpoint towards lower values, more defrost cycles are required and this is particularly true at the highest temperature investigated when the ejector capacity is too high. An option to outflank this issue is a floating control of the receiver pressure level. It allows to remove completely the MT

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compressors and to keep in a way the ejector performance because of the tradeoff existing between entrainment ratio–pressure lift. Choosing this option, the control of the AC load is not prioritized and the outlet water temperature in the chilled evaporator increases, jeopardizing the thermal comfort inside the supermarket. However, two alternative solutions can extend the applicability of such strategy. Firstly, if the heat is exchanged with the internal environmental through a secondary medium (water), the AC evaporator must be sized according to the highest evaporating temperature. A bypass enables the adjustment of the capacity depending on the need, but the disadvantage of having several heat exchangers with their relative losses reduce the total efficiency of the heating and cooling system. Secondly, R744 as refrigerant permits to apply direct cooling, with a reduction of the total costs but with a special attention to the installation of the pipes due to high operating pressures.

Considering part-load conditions at 40 °C as gas cooler outlet temperature, Figure 10 illustrates how the receiver pressure behaves to reach a new equilibrium point where all the refrigerant flow is compressed by the IT compressors. A higher lift in the receiver pressure occurs when the ejector capacity becomes very high, therefore at very low part-load. A small effect has been noticed on the flow through the MT evaporators because of the higher vapor quality value at the evaporator's inlet, when the receiver pressure lifts to higher values. All in all, the total power consumption decreases due to the lower pressure lift in the parallel compressors, compensating for the increase of the total flow rate (Figure 11). The energy reduction becomes smaller when moving towards the design conditions. The different strategy impacts the compressors combination varying the speed of the VSD compressor.

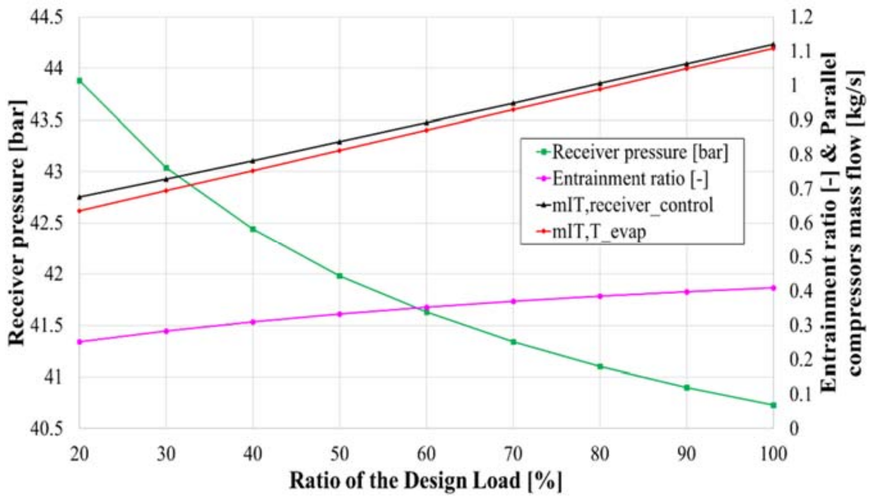

Figure 10: Fluctuation of the receiver pressure and difference in terms of refrigerant flow between the two strategies, for part-load at $T_{gc,outlet} = 40$ [°C].

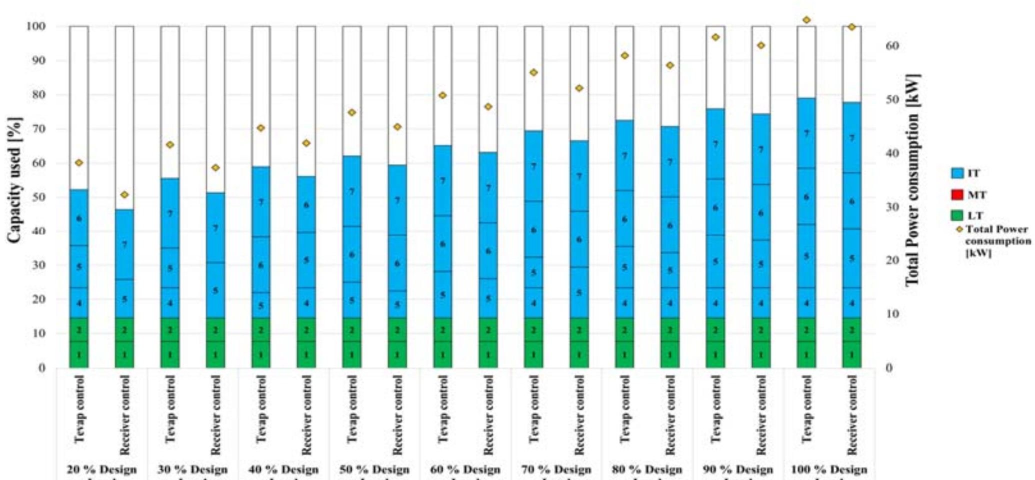

Figure 11: Comparison compressor combinations and power consumptions between the two strategies at part-loads ($T_{gc,outlet} = 40$ °C).

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4.5 Cost analysis

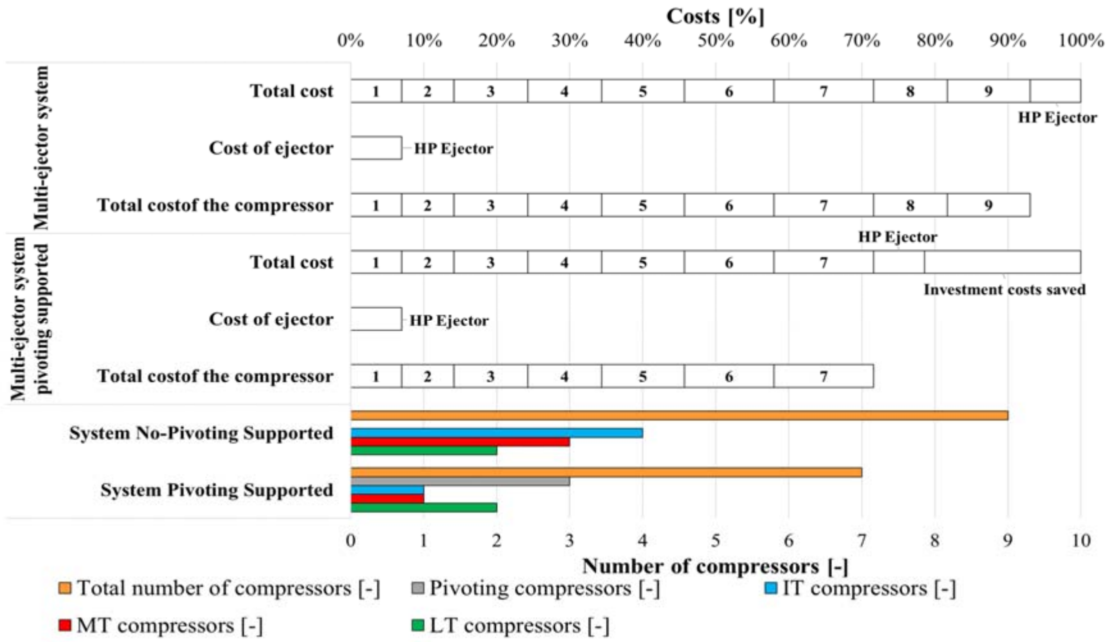

Figure 12: Simplified cost analysis.

A simplified cost analysis was done to point out the benefit of having pivoting compressors. Only the cost of compressors and Multi Ejector were considered, assuming that the cost of the Multi Ejector is in the same order of magnitude as a compressor and neglecting the costs of the other equipment (included the valves that turn a compressor into “pivoting”).

Figure 12 shows how the use of the pivoting arrangement allows to reduce the number of compressors and compensating the cost of the Multi ejector. In a non pivoting-supported system two additional compressors are required and their constraints of working permanently in a section reduces drastically the flexibility and the operating hours of each compressor. Figure 13 illustrates the consequent advantage of the pivoting technology: the average annual operation time of compressors is broadened, which should have a positive effect on the maintenance costs. Due to the large difference in terms of loads, ambient conditions and the use of ejector, the number of unused compressors in the non pivoting-supported system is larger over all the year. During summertime, two or three MT compressors would typically be off while during wintertime all the IT compressors are idle. The pivoting concept is perceived as the key to promote the compactness of the systems, their economic impact in a medium and large-size supermarket, the operational time of the compressor pack, while supporting the spread of the ejectors and the efficiency improvement linked to ejectors. Furthermore, the COP is not degraded by the pivoting system, but in some scenarios, it is even higher depending substantially on the efficiency of the compressor packs which is related to the rotational speed of the VSD compressors.

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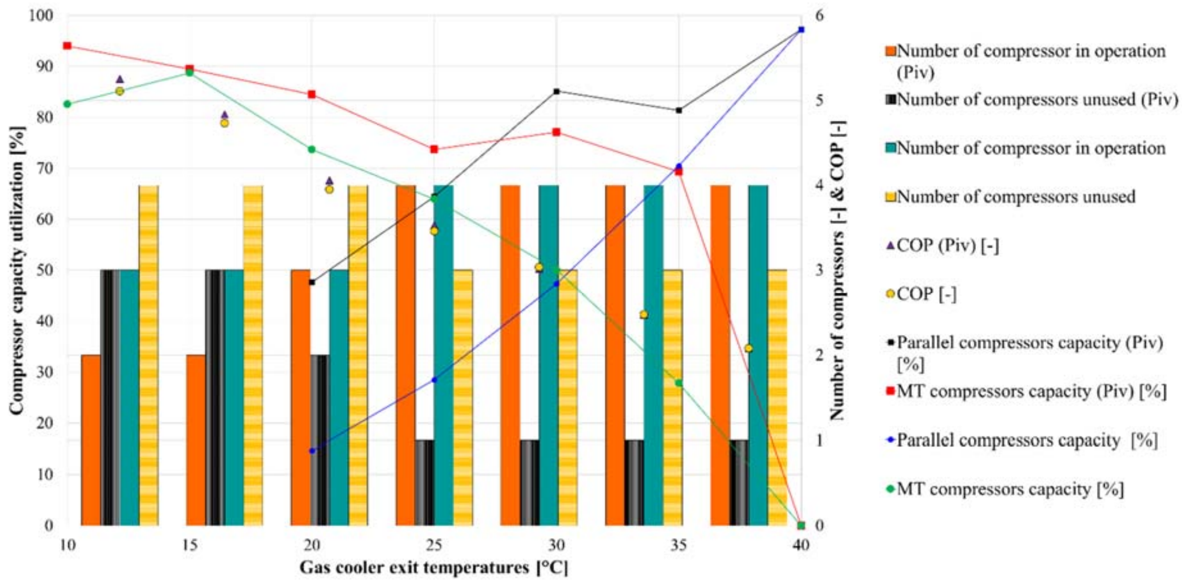

Figure 13: Compressors capacity utilization, number of compressors in operation and COP comparison between system pivoting and no pivoting supported.

An optional way to further reduce the number of compressors in the test-rig is to turn into pivoting even the VSD compressors (Figure 14). It allows to follow the refrigerating loads variations with a greater flexibility, as indicated by the unused capacity values. A deep investigation on the compressor stages efficiency should be performed in order to find the most suitable combinations.

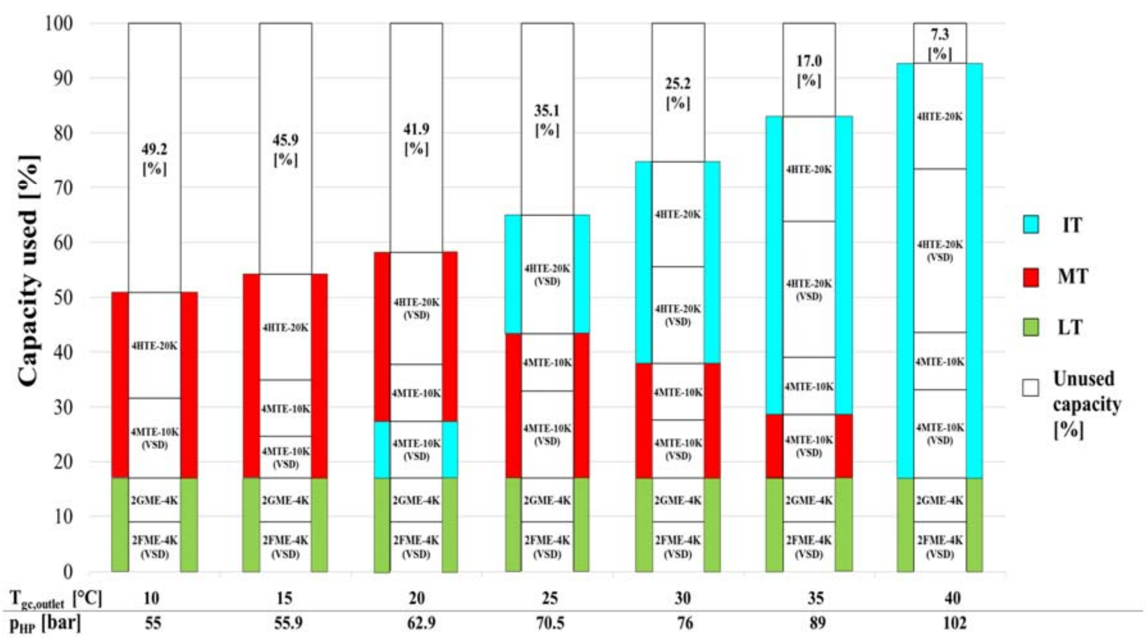

Figure 14: Effect of implementing "pivoting" in all the compressors (MT-IT) installed in the facility.

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5. CONCLUSIONS AND FUTURE WORK

This paper investigates the implementation of the pivoting concept in a large-size supermarket located in a South European region. It confirms the positive effect of applying pivoting in terms of installed capacity and thus of investment-maintenance costs. In the investigated scenarios, two compressors could be removed by implementing the pivoting concept. The main conclusion from this study is that when ejector technology is introduced, it's also advisable to implement a pivoting arrangement to reduce the total number of installed compressors. This is beneficial to the next generation of R744 refrigerating systems in warm climates. The ejector has a strong and positive impact on the system performance and efficiency, and it supports the vapour/load distribution between the two compressor stages, i.e. the active use of the pivoting principle becomes crucial to utilize the installed capacities. When AC demand is present, the capacity ratio between the parallel/AC and MT suction group is extended, due to the pivoting arrangement, compressor capacity is moved towards the elevated pressure suction group, which reduced the number of total compressors, as the ejectors are unloading the medium temperature suction group. Two strategic controls have been compared, pointing out that a floating control of the receiver pressure is usually not a viable solution. The next steps will be the analysis of the chance of having both the VSD compressors "pivoting", increasing further the flexibility of the rack and thus further reducing the number of installed compressors. Moreover, a suitable control system should be developed and verified in a demo plant.

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