Novel integrated CO_2 vapour compression racks for supermarkets. Thermodynamic analysis of possible system configurations and influence of operational conditions

Ángel Á. Pardiñas[a,*], Armin Hafner[a], Krzysztof Banasiak[b]

[a] Department of Energy and Process Engineering, NTNU, Trondheim, Norway

[b] SINTEF Energy Research, Trondheim, Norway

*Corresponding author. Department of Energy and Process Engineering, NTNU, Kolbjørn Hejes vei 1A, Trondheim, 7491, Norway. E-mail address: angel.a.pardinas@ntnu.no

ABSTRACT

An increasing number of supermarket chains are opting for the utilisation of centralised carbon dioxide refrigeration systems. However, the need to improve their efficiency in order to make them economically and environmentally sustainable worldwide becomes ever more important. One of the most common approaches is to integrate all the energy demands (cooling, air conditioning, heating, domestic hot water) into the same unit. In addition, operation at high ambient temperatures can be made more energy-efficient with the use of parallel compression and ejectors for expansion work recovery.

This paper describes the new proposals for the system architecture of integrated transcritical CO_2 refrigeration installations. A sample system with parallel compression with multiple ejectors has been modelled numerically. The results prove the decrease in the overall power consumption of the system when using parallel compression compared to a standard booster system (reduction of 19% at 30 ºC). The use of one group or two groups of ejectors allow the power consumption to be further diminished (around 5% and 8%, respectively). With the appropriate operating conditions (e.g. discharge pressure, liquid receiver pressure) and an optimized design of the air conditioning and cooling systems (which allows the evaporation pressure to be increased) an important reduction in power consumption can be achieved (for example greater than 7% when the evaporation pressure of the medium-temperature cabinets increases from 28 bar to 32 bar). In addition, the simulations allowed the comparison of three separate modes proposed for low-ambient-temperature conditions (temperatures close to or lower than 0 °C), which are challenging for current carbon dioxide refrigeration systems.

Keywords: Transcritical CO_2 refrigeration system; supermarket; ejector.

AC – air conditioning. CB – conventional booster. EJ_{AC} – AC multiejector. EJ_{MT} – MT multiejector. EVAP – evaporator. FGV – flash gas bypass valve. HPV – high pressure valve. HVAC – heating, ventilation and air conditioning. IESPC – integrated ejector supported parallel compression. IHX – internal heat exchanger. LT – low temperature. MOPD – maximum operation pressure difference. MT – medium temperature. PC – parallel compression. S – system. VSD – variable-speed drive. WM – winter mode. WMV – winter mode valve.

1. Introduction

The reintroduction of CO_2 (R744) in supermarket refrigeration is a fact today and will have a major impact on the environmental footprint of this sector. The works of Lorentzen [1] triggered, particularly in Northern Europe, the use of this refrigerant by food retailers [2]. Nevertheless, the development of cost- and energy-efficient units able to operate in all regions and for all climates constitutes a remarkable challenge.

The concept of integrated R744 vapour compression units for supermarkets is relatively new (2010 onwards). The principal idea is to meet with a single unit the needs of refrigeration at the temperature levels specific for supermarkets, as well as the heating, ventilation and air conditioning (HVAC) and hot water demands. Integrated units are normally tailor-made installations, designed specifically according to the individual requirements for a given store. There are a few examples of the successful application of such systems. The first installation was developed in the EnOB project, involving Aldi Süd in Germany in cooperation with the Fraunhofer Institute for Solar Energy Systems, which introduced a monovalent and geothermally supported R744 compressor rack [3]. The system provided commercial cooling for the chiller cabinets and freezers in the sales area and for the cold stores in the storage area, as well as space heating or cooling and pre-heating and pre-cooling for the ventilation system. Thus, neither separate heat generators nor an air conditioning (AC) system were required. The target of the EnOB project was to reduce the primary energy consumption by up to 29%, although the actual reduction reached 20% shortly after its implementation. Another example is the unit that resulted from the CREATIV project, developed by SINTEF, the Research Council of Norway and REMA 1000, and described in Hafner et al. [4]. The refrigeration system was utilised as a heat pump in the winter season and provided cooling for the air handling unit in the summer season. The surplus heat from the refrigeration system was applied for heating different areas of the supermarket or stored both in water tanks and in the ground (energy wells). The active integration of the energy wells allowed free cooling and the meeting of different heating and cooling demands with low energy consumption. This concept of the integrated unit led to annual energy saving of 30% compared to similar supermarkets in the area.

Neither of the previously mentioned examples comprised the use of ejectors for expansion work recovery. There are several works in the literature that showed that these devices improve the performance of transcritical CO_2 refrigeration systems [5-9]. The concept of ejectors in CO_2 vapour compression systems for supermarkets, particularly for hot and warm climates, was proposed by Hafner et al. [10]. They replaced the ordinary expansion valves and enabled the recovery of up to 30%–35% of the total expansion work [11]. With respect to the use of fixed geometry ejectors in such systems, the main challenge, which is to achieve a smooth control to adapt to the existing load and ambient conditions, was handled by the use of multiejector blocks, i.e. bunches of separate ejector geometries working in parallel. Multiejector blocks are step-wise controlled multiple ejectors with different ejector geometries (different cross sections in the motive nozzles) that can be enabled and combined as a function of the existing requirements [10].

There are a number of recent examples of ejector-supported parallel compression units for supermarkets. The first commercial (though of prototype-design) application of a prototype of a multiple-ejector system was carried out in the MIGROS Supermarket in Bulle in Switzerland in 2013 for a conventional, non-integrated CO_2 compressor rack [12]. On-site test campaigns proved that it is possible to raise the medium-temperature (MT) evaporation conditions from -8 °C to -2 °C with flooded evaporator operation and liquid ejectors. Liquid ejectors pump the liquid that accumulates at a liquid separator downstream of the evaporators, reducing the risk of liquid at the suction of the compressors. The increase of the evaporation temperature allows the power consumption of the compressors to be reduced. Both laboratory and field tests revealed that liquid ejectors at subcritical conditions are able to return the liquid refrigerant from the liquid separator back to the liquid receiver. Fredslund et al. [13] analysed experimentally the energy consumption of different installations (with and without AC load), with and without multiejectors and with flooded or direct expansion evaporators. The authors observed that energy savings of 4% (at 27 °C ambient temperature) were achieved in real supermarkets without AC load due to the use of the multiejectors. These values increased up to 15% (at 30 °C ambient temperature) in supermarkets with AC load. The authors stated that it is crucial to design compressor packs that match the load, as well as to reduce and adapt the required pressure lift of ejectors (difference between the pressure at the liquid receiver and at the MT evaporators) in order to maintain smooth operation throughout the year.

In order to design, manufacture and safely operate an integrated supermarket installation equipped with multiple ejectors, a number of technical challenges must be tackled. All the installations commissioned so far were dimensioned for given, individual load conditions and geographical location of the shops. Thus, no standardisation (and consecutive reduction of capital costs) has been reached to date. Therefore, new knowledge that could invoke the technological development and lower investment costs of integrated CO_2 vapour compression units for supermarkets is required. SuperSmart-Rack project (Research Council of Norway No. 244009/E20) aims to contribute in this line of research both from numerical and experimental points of view, focusing on the environmental conditions existing in Norway.

This paper introduces several innovative features to be applied in CO_2 transcritical commercial refrigeration units, such as:

- Integration of an ejector supported air conditioning suction group.
- Control strategies for the control of the pressure level at the gas coolers with ejector blocks working in parallel but with different functions (suction from the MT pressure level and air conditioning from an elevated pressure level).
- Pivoting the suction port of compressors from the base group to the parallel group and vice-versa.
- Increase of the evaporation temperature in refrigeration cabinets due to the operation under flooded conditions.
- Special configuration for efficient operation with low-temperature environmental conditions, or winter mode.

The energy performance of the novel solutions, solutions that are just emerging on the market as well as the solutions that have been well established in commercial refrigeration were modelled numerically in the Dymola-Modelica environment. The layout of the developed numerical model replicates the facility newly installed at the SINTEF/NTNU laboratory that will serve to experimentally verify the numerical findings in the nearest future. Multiple steady-state simulations were performed for different possible system configurations and in changeable operating condition considering maximum loads of a typical medium-sized supermarket in Norway.

2. General system architecture

2.1. Functional requirements

The concept of an R744 integrated ejector supported parallel compression (IESPC) refrigeration unit is an alternative solution for the less environmentally friendly installations that use individual systems to meet the different loads of supermarkets, normally operating with HFCs or using boilers. The approach is to employ a single unit to meet the demands of the shop, including cooling at medium temperature (MT) and low temperature (LT) for the different cabinets, as well as providing heating or air conditioning, if requested.

The simplified structure of an IESPC R744 refrigeration unit is shown in Figure 1. It consists mainly of three packs of compressors (base compressors, parallel compressors and LT compressors), a gas cooler section, a high pressure control section, AC evaporators, MT evaporators and cabinets, LT evaporators and cabinets and three accumulators (liquid receiver, liquid separator and suction accumulator).

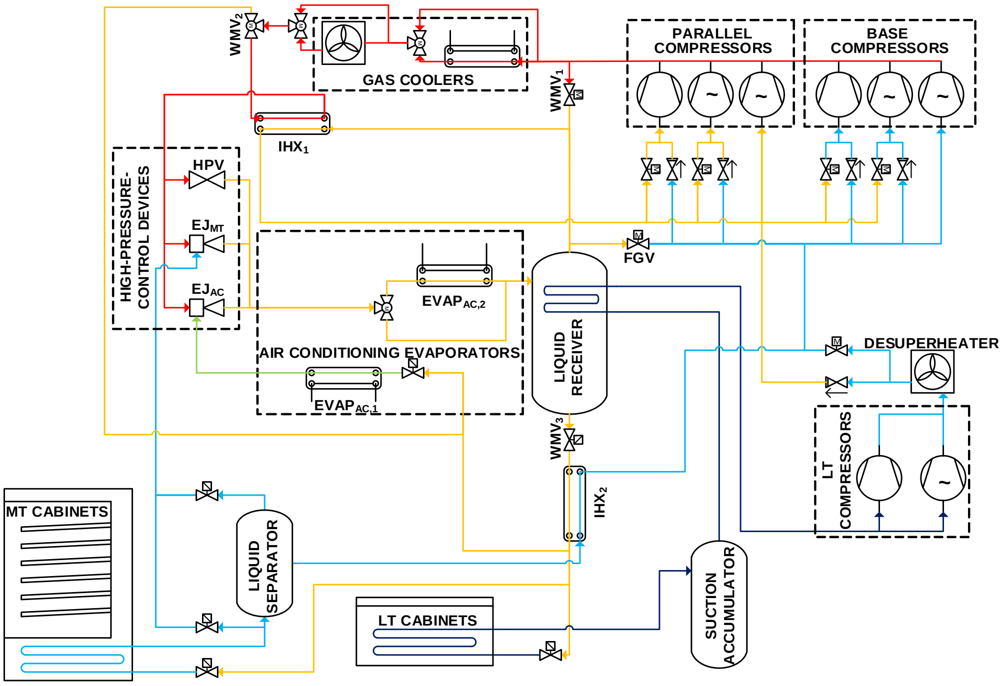


Figure 1. Simplified representation of the concept of R744 integrated ejector supported parallel compression (IESPC) refrigeration system (high-pressure lines indicated in red, intermediate-pressure lines in orange, medium-pressure lines in light blue, low-pressure lines in dark blue, AC lines in green). The symbol "~" in some of the compressors represents the frequency-controlled drive.

Evaporators

The basic functionality of an IESPC system is to meet the cooling loads at different temperature levels. 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.

Compressor packs

Base compressors suck vapour refrigerant from the liquid separator at the pressure of the MT cabinets and discharge it at the pressure of the gas coolers. Parallel compressors share the discharge manifold with base compressors, but the vapour is sucked from the receiver, at slightly elevated pressure level. LT compressors suck refrigerant from the suction accumulator at the pressure of the LT evaporators and can discharge either to the suction manifolds of base or parallel compressors.

The three packs of compressors are responsible for maintaining the requested pressure levels in the particular suction groups of the system. A good practice in integrated supermarket racks could be having at least one compressor per section for which the capacity could be modulated by integrating a variable speed drive, VSD, to adjust smoothly the compressor section capacity to the existing demand over a broad range. If the demand associated with any of these compressor sections rises, the corresponding pressure rises as well, and the controller increases the capacity of the suction group by speeding up the active VSD-compressor/s or by turning on additional compressors of the pack. The opposite occurs when the demand decreases. Base compressors aim to control the pressure level at the MT evaporators and liquid separator. LT compressors are responsible for the pressure at LT evaporators. Finally, parallel compressors control the pressure level at the liquid receiver, which depends mainly on the AC demand and temperature at the outlet of the gas coolers. If the amount of available vapour is too low for the minimum capacity of the parallel compressor section, these compressors are turned-off and the flash-gas bypass valve (FGV) regulates the pressure at the liquid receiver, by expanding vapour to the suction of the base compressors.

Gas coolers

Typically the heat of an R744 vapour compression system is rejected to the ambient air in the gas coolers, using a fin-and-tube heat exchanger. Heat recovery could also be applied with the use of heat exchangers in series with the fin-and-tube heat exchanger, as explained in Section 2.2. Downstream of the gas coolers the refrigerant at high pressure level and low temperatures reaches the high pressure control devices via an internal heat exchanger (optional).

High pressure control devices

The high pressure control section proposed in an IESPC system, being the first innovation proposed, consists of a high pressure valve (HPV in Figure 1) and two multiejector blocks.

Either the HPV or one of the multiejector blocks is in charge of the high pressure control. If the HPV is used, its opening degree is regulated to adjust the pressure at the gas coolers; it will be opened by the controller if the pressure is higher than the setpoint and vice versa. However, the tendency is that the high pressure valve should be a safety device only and could be also removed. If the multiejector block regulates the high pressure, the controller will combine conveniently the opening of the four or six fixed geometry ejectors (cartridges) per block; meanwhile, expansion work is being recovered. Each fixed geometry ejector is independently opened by means of its respective solenoid coil. If the pressure level at the gas coolers is higher than the setpoint, the controller increases the motive nozzle cross section area by selecting an appropriate combination of active ejector cartridges. The opposite occurs if the pressure is lower than the setpoint. From our understanding, the multiejector block intended to regulate the high pressure, the MT multiejector block (EJ_{MT} in Figure 1), should be dimensioned for low-entrainment-ratio and high pressure lift to suck vapour from the liquid separator downstream of the MT cabinets. It may also pump liquid from this same separator, using the dedicated liquid ejectors, if the liquid level rises due to flooded operation of the MT evaporators (in order to increase the evaporation temperature and reduce the formation of frost).

The second ejector block, the AC multiejector block (EJ_{AC} in Figure 1), is dimensioned for high-entrainment-ratio and low pressure lift and delivers the refrigerant from the AC evaporator ($EVAP_{AC,1}$ in Figure 1) to the higher pressure level of the liquid receiver. It controls the pressure lift between the evaporator and the liquid receiver in an analogous way as the MT multiejector block. If there is an increase of the AC load, the controller changes the combination of ejector cartridges in order to increase the motive nozzle cross section area and the ejector is able to suck all the refrigerant coming from the AC evaporator keeping the pressure levels at the setpoint (and therefore the pressure lift). In case there is a reduction of the AC load, the motive nozzle area is diminished and the effect is the opposite.

Since both multiejector blocks have their motive nozzles connected to the same high side pressure level, it is not technically true that only the MT multiejector block is regulating this high pressure level. There are two possible approaches to control the discharge pressure with both multiejector blocks: an independent strategy and a synchronised strategy. The independent strategy involves that the MT multiejector block controls the discharge pressure directly, while the AC multiejector block tries to adjust the pressure difference between the AC evaporators and the receiver in accordance to AC cooling demands. The disadvantage of this solution is that the control of the discharge pressure could be compromised at conditions of relatively high share of the AC load to the total load. In contrast, the synchronised solution prioritises the control of the high pressure by adjusting the overall opening degree of both multiejector blocks. The controller distributes the total opening between both blocks in order to adjust the pressure lift at the AC multiejector block. The higher the AC load, the more refrigerant flows through the motive nozzle of the AC multiejector block to keep the pressure lift. The counterpart of this strategy is that the AC load might reach a cut-off boundary that guarantees the high pressure control, but this is unlikely if the system is properly sized.

2.2. Optional features

AC production

AC production can be a major part of the total load of a supermarket, particularly in warm climates, and, if needed, it should be integrated into the refrigeration system. The proposed solution includes two evaporators located in two positions of the system, each at a different pressure level (see Figure 1). The first option is with the evaporator positioned downstream of the high pressure control devices, at the pressure of the liquid receiver. The second, innovative, option is with the evaporator positioned at the suction port of the AC multiejector block, described in the previous section, at a lower pressure level than the receiver. The entire R744 mass flow rate through this second evaporator has to be entrained by the AC multiejector block. The pressure level in the second evaporator allows to deliver AC cooling capacity at a lower evaporation temperature than with the first solution, if the liquid receiver pressure is fixed, or preferably to elevate the liquid receiver pressure for the same AC evaporation pressure.

Hot water production

The production of hot water for heating, domestic hot water or other purposes, by using heat rejected from the refrigeration system, is another optional functionality of integrated units that results in an enhancement of the overall energy efficiency of supermarkets. In order to achieve this, at least an additional heat exchanger, operating as a heat reclaim gas cooler/condenser, should be placed in the high pressure line, upstream of the conventional gas cooler (Figure 1). It must be taken into account when designing the heat reclaim system that the lower the return temperature of the water, the more efficient the CO_2 rig will be. This justifies the use of radiant floor emission systems, which operate with lower temperatures than fan-coils, for example, due to their high heat transfer area.

Pivoting suction port of base/parallel compressors

The pivoting suction port of base/parallel compressors refers to the possibility of choosing the suction manifold connection for several compressors and integrating them in either the base or parallel section. The main utility of this novel feature, not available in any commercial compressor packs on the market yet, lies in the possibility to swiftly adapt the number of compressors assigned to a particular suction group. Such a feature enables greater pack flexibility in terms of ability to follow the actual load profile, dependent on the time of day and the ambient conditions, without the need of additional compressors or even with the possibility of reducing the total number of compressors. At high temperature conditions and with low activity in the supermarket, the AC load may exceed by far the needs at MT, and the number of compressors dedicated to the parallel section should increase. With fixed configuration in each section, it might happen that the number of parallel compressors is insufficient; meanwhile, the base compressors are not in operation. On colder days with no AC load, parallel compressors could be switched off completely, while at the same time the actual capacity of the base compressors would be too low to meet the existing MT load. Therefore, the possibility of pivoting the suction ports of compressors (at least some in the pack) gives new degrees of freedom and increases the flexibility of the refrigeration system.

One possible technical solution for pivoting compressors in an IESPC refrigeration unit is represented in Figure 1, where only one compressor per section is connected in a fast way to the corresponding suction manifold. The remaining compressors are connected to the liquid receiver pressure level through a solenoid valve and to the liquid separator pressure level through a check valve. Therefore, any 'pivoting' compressor will suck from the liquid receiver when the solenoid valve is open and from the liquid separator when it is closed (the check valve is activated in this case). This solution is inexpensive and simplifies the switching process. In addition, it should not affect the oil return to the compressors, since it is controlled by an independent system that connects the carter of each compressor and the oil receiver.

Turning a compressor into a base or parallel compressor section is not a trivial issue to trigger and control. For given distribution of MT and AC loads, it could occur that one of the setpoints of pressure, for either the base or the parallel compressors, cannot be achieved with the default number of compressors in the section. In this case, if there is any compressor available (in standby and with the set of valves for pivoting) in a section, this compressor should be incorporated into the neighbouring section, requesting more capacity. The corresponding controller should adopt the new machine and regulate it using the corresponding settings and measurements. Another situation to be tackled by the controller might occur when one of the packs is not able to meet the demand, i.e. to reach the pressure setpoint, but there are no compressors available in the other set. In this case, the strategy is to prioritise demands, focusing on the MT load first in order to guarantee the proper conservation of the products.

In practice, situations involving a lack of capacity would be rather infrequent, given that the compressor pack was dimensioned in the right way, according to the specified load profile. In turn, the option of pivoting compressors could potentially decrease the overall number of compressors installed in the entire pack (summarised over LT, base and parallel section) in cases where the designer could expect operational challenges in maintaining the assumed division of the load between the base and parallel sections. In addition, the average annual operation time of compressors should be extended. Consequently, the overall ownership costs should decrease.

Pivoting discharge of LT compressors

Another innovative feature of IESPC refrigeration systems is the pivoting discharge of LT compressors, i.e. connecting them to the suction line of either the base or parallel compressors. This utility can be particularly useful if one of the sections is not in operation under certain conditions. Practical implementation of this type of pivoting might also be carried out with a solenoid valve and check valve. The solenoid valve is connected to the suction line of the base compressors, and the check valve to the parallel compressors. The refrigerant from LT compressors is directed to the base compressors if the solenoid valve is open, and to the parallel compressors if this valve is closed (the check valve is activated in this case). A simple control approach of this feature is to connect the LT discharge to the active compressor section, if only one is in operation. If both base and parallel compressors are working, the decision should be taken, focusing on efficiency and requested capacity. The effect on efficiency has been studied numerically and the results are reported later in the paper.

Winter mode

Another new configuration introduced to R744 IESPC refrigeration systems is one of the so-called winter modes (WM), specific for locations that operate at low ambient temperatures (lower than 0 °C) in longer periods. Low ambient temperature entails low temperature at the outlet of the gas cooler, which consequently invokes a low setpoint for discharge pressure in the main pack controller. In such conditions, if the high pressure is controlled with the multiejector blocks (with the MT multiejector block) or with the HPV, the pressure inside the liquid receiver sinks. Therefore, without any extraordinary system that would maintain either the elevated gas cooler outlet temperature (e.g. by regulating the amount of cooling air flowing through the gas cooler) or the elevated receiver pressure (e.g. by bypassing a fraction the high-pressure and high-temperature refrigerant from the compressors directly to the receiver), the MT cabinets could malfunction due to the too small pressure difference over the expansion valve (greater than 4 bar in any case to prevent it [14]).

Three possible solutions for winter mode are proposed, each of them entailing specific control issues:

- WM_I is based on the liquid receiver operated as high pressure separator, with the full opening of all the expansion devices assembled between the gas cooler and receiver, i.e. HPV and/or multiejector blocks (Figure 2, left). The individual expansion devices of the MT evaporators are supposed to control the mass flow in the system, while the discharge pressure would be dictated by the conditions of heat rejection in the heat sink (i.e. condenser in subcritical mode). At very low ambient temperatures, the temperature at the outlet of the condenser might be so low that the corresponding discharge pressure would be too low for compressors available on the market today. Therefore, the controller should maintain a certain minimum temperature at the outlet of the condenser, e.g. by adjusting the secondary fluid mass flow rate or bypassing selected sections of the condenser refrigerant channels.
- WM_{II} solution operates almost as a standard booster circuit, with the HPV or multiejectors regulating the pressure at the gas coolers, but with an additional valve (WMV_1), which allows the pressure of the liquid receiver to be increased by expanding a part of the vapour discharged by the base compressors directly to the receiver (Figure 2, middle). The controller associated with this valve maintains the minimum pressure drop required in the cabinets' expansion devices. Thus the expansion devices of MT and LT evaporators work without any disturbance. This winter mode requires no additional control of the condenser capacity.
- WM_{III} approach (Figure 2, right) is based on a revival of the idea of multiple high-pressure expansion devices [15]. For the integrated system considered this configuration would entail bypassing the liquid receiver, together with all the high-pressure expansion devices, by the use of an additional three-way valve installed downstream of the condenser (WMV_2), together with a shut-off valve installed downstream of the receiver (WMV_3). As indicated above, this mode uses all the individual expansion devices of the MT evaporators in parallel to adjust the discharge pressure level to the requested value. To ensure saturated conditions at the MT evaporators' outlet, as well as to compensate for charge migration in floating operating conditions, the liquid separator acts as an MT-pressure accumulator, in which the liquid level could be adjusted. The liquid level in the liquid separator could be reduced by transferring high-pressure refrigerant to the liquid receiver through the WMV_2 (in the position that connects the gas coolers and the HPV), opening the HPV and keeping closed the WMV_3. To increase the liquid level in the separator with refrigerant from the liquid receiver, the WMV_2 should connect the gas coolers with the HPV, the HPV should be slightly open to prevent a dangerous increase of the high-pressure level, and the WMV_3 should be totally open. In the case of unbalanced operation on the cabinets' side (some cabinets operated at substantial superheating, some cabinets operated in flooded mode), a special algorithm to effectively equalise the evaporator outlet conditions would have to be developed, e.g. by overriding the requested opening degree of expansion valves in all the cabinets where substantial superheating was detected. No control of the condenser capacity is required.

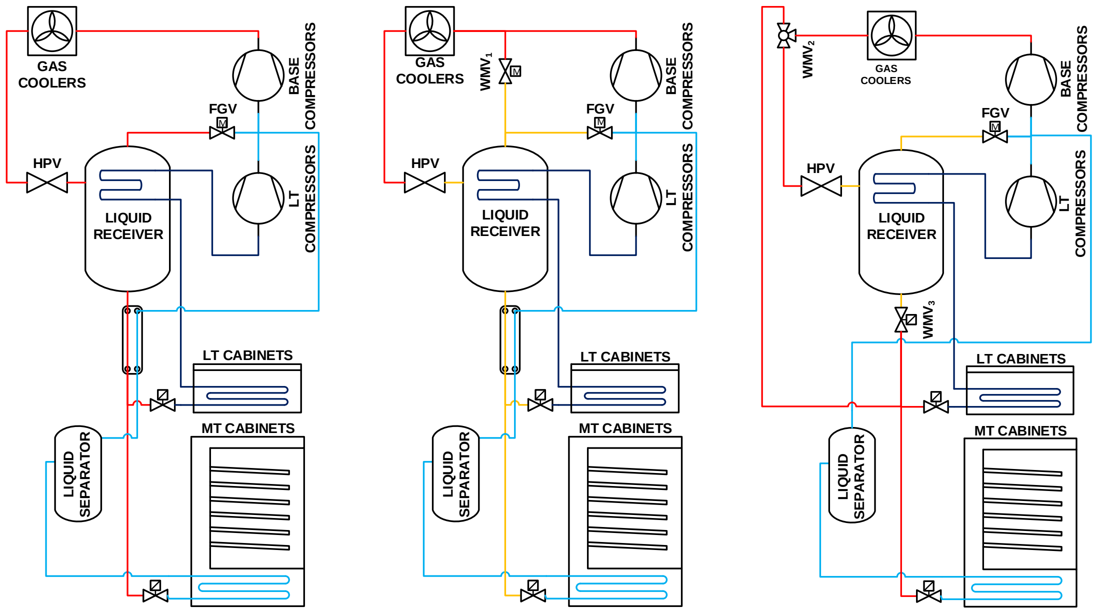


Figure 2. Winter mode concepts proposed. Left scheme represents WM_{I}, with bypass of the high pressure control devices (FGV in the figure). Middle diagram shows WM_{II}, which distributes part of the hot refrigerant from the discharge of base compressors to the liquid receiver to maintain the pressure level through valve WMV_1. Right scheme depicts WM_{III}, where the liquid receiver is “removed” from the loop under low temperature conditions with the positions of WMV_2 and WMV_3.

2.3. Additional control aspects

Discharge pressure setpoint

The discharge pressure of base and parallel compressors is a crucial parameter for the efficiency of CO_2 vapour compression systems, particularly in transcritical operation [16]. The optimisation of the discharge pressure has been the subject of many studies that can be found in the specialised literature. Yang et al. [17] reviewed the results of previous works found in the literature of the optimum discharge pressure for a transcritical simple compression system. Each of these works proposes a different correlation, usually a function of the temperature at the outlet of the gas cooler and, sometimes, of the evaporation temperature. In addition, Yang and co-workers observed that these correlations may lead to important deteriorations in the overall efficiency, depending on the conditions. Another approach is presented in Gullo et al. [18] for commercial refrigeration systems. The authors defined up to four zones for the calculation of the optimum discharge pressure, not only for transcritical but also for subcritical operation. In addition, the authors established different limits for different regions as a function of the configuration (conventional booster, parallel compression, parallel compression with mechanical subcooling, etc.).

The use of multiejector blocks may also affect the optimum pressure curve, especially since the performance of ejectors will be influenced strongly by the conditions of the motive flow (pressure and temperature), pressure lift, etc. Therefore, for complex systems with a large number of parameters involved, the most accurate and convenient solution is the use of an online control strategy [19]. Such an approach was proposed in several works for simple transcritical refrigeration systems [20-23]. To the best of our knowledge, real-time optimisation should be further studied in order to apply it in IESPC refrigeration systems, seeking a minimisation of the electric consumption of the refrigeration system, which can be measured online. The optimisation algorithm should also be constrained by the requested loads to be met at each time step.

Another issue concerning the control of the discharge pressure is associated with the operational envelope of compressors. Compressor manufacturers recommend a minimum/maximum discharge pressure, which is a function of the suction conditions. At low ambient temperatures, where subcritical operation occurs, the requested condensation pressure (considering a certain degree of subcooling) might be lower than the minimum operational discharge pressure allowed. At high ambient temperatures, the opposite situation might take place. Therefore, any setpoint value for the gas cooler pressure enforced should not fall outside the operational envelope.

Receiver pressure and pressure lift

Concerning the receiver pressure, a typical operational practice is to maintain it at a constant value (around 38 bar), mainly for two reasons: (i) to simplify the control, and (ii) to guarantee that the pressure of the refrigerant leaving the pack and directed to the shop's cabinets is never greater than the nominal pressure for the part of the installation located inside the supermarket's envelope. However, the enforced value of the receiver pressure affects the actual performance of this kind of refrigeration system [24]. Generally, the greater the temperature of the ambient and of the refrigerant at the outlet of the gas cooler, the higher the receiver pressure should be, always minding the production of AC, which is linked with the receiver pressure in integrated systems. New refrigeration systems with R744 should be built to allow pressure at the receiver of at least 60 bar, close to the operation limits for the suction pressure in the transcritical compressors available today [25]. However, from our experience and the observation from industrial partners and system manufacturers, with certain types of polyalkylene glycol (PAG) oils and oil management systems, if the liquid receiver pressure exceeds 46 bar, the amount of liquid CO_2 dissolved in lubricant increases, and when the oil-refrigerant mixture returns to the crankcase of base or LT compressors, it begins to foam.

The proposed innovative controlling strategy should comprise adjustment of the setpoint for the receiver pressure in order to optimise the energy performance of the refrigeration system. A simplified approach could be based on a correlation dependent on the temperature at the outlet of the gas cooler. In practice this correlation would be influenced by the system itself. Thus, the most convenient regulation solution is online control, in a similar way as described above for the discharge pressure (minimisation of the power consumption for the existing loads).

The setpoint of the pressure lift at the AC multiejector block, used for the control of the multiejector block, in new controlling approaches should also depend on ambient conditions (temperature at the outlet of the gas cooler). The reason for this is that the performance of the multiejector block is strongly influenced by the conditions of the refrigerant at the motive flow. Therefore, the setpoint for the pressure lift could be elevated as the temperature of the refrigerant at the outlet of the gas cooler rises.

Flooded evaporators and liquid level at the separator

A standard control approach for expansion devices in cabinet evaporators is to control outlet superheating, typically maintained at 5 or 8 K. This guarantees safe operation of the base compressors (no liquid at the suction). However, an important fraction of the total heat transfer surface of each evaporator, normally between 10 to 30%, is charged with superheated refrigerant. The major consequence of this is that the evaporation temperature needs to be reduced in order to maintain refrigeration capacity and to keep the temperature of the product inside the cabinets within a correct range. Therefore, the efficiency of the refrigeration system diminishes and the formation of frost increases, leading to more frequent defrosting periods.

There are two ways to tackle the aforementioned problems. The first option is to increase the size of the evaporators, but this raises the capital costs and consumes space available in the cabinet for foodstuffs. The second solution is to use the available surface more efficiently, by replacing the superheat-mode heat transfer with the use of flooded evaporators. This option allows the evaporation temperature to be increased by at least 4 K [13]. In Hafner et al. [12], the evaporation temperature of the MT cabinets was successfully raised 6 K (from -8 to -2 °C). The main disadvantage of this approach is the need for a separator downstream of the evaporators to collect the resting liquid, since it is not possible to meter the refrigerant to cabinets in such a way that the evaporator outlet conditions match precisely saturated vapour (quality equal to 1, no superheating). In addition, as the liquid level inside the liquid separator might increase with time, the operational safety of the base compressors might become compromised, which brings a new challenge to the control of the units. In an IESPC refrigeration system such as the one described, liquid ejector cartridges are available in the MT multiejector block to control the liquid level at the separator, pumping liquid from the separator to the receiver, if needed [26].

3. Methods

3.1. Numerical model

The numerical model, represented in a simplified manner in Figure 1, describes the experimental facility newly installed at the SINTEF/NTNU laboratory, which consists of an IESPC refrigeration system and several auxiliary circuits. The IESPC system is a scale 1:1 system that could be found in a medium-sized supermarket. It is versatile and allows experiments to be performed with different system configurations (conventional R744 booster, R744 parallel compression, ejector supported parallel compression, etc.) and for a wide range of operating conditions.

The system was modelled in detail using Modelica object-oriented programming language in the Dymola 2016 environment (Dassault Systems, Vélizy-Villacoublay, France [27]). The models developed are based on the commercially available library TIL 3.4 from TLK-Thermo GmbH (Braunschweig, Germany [28]). The thermodynamic and transport properties of the refrigerant (R744) are provided by the TILMedia 3.4 library, also by TLK-Thermo. The effects of oil dissolved in the refrigerant and operation of the oil management system were neglected.

The system consists of eight compressors: two compressors in the LT section, one directly connected to the MT evaporators' pressure level (base compressor), one directly connected to the liquid receiver (parallel compressor), and four machines that can work either as parallel or base compressors, thanks to the pivoting suction lines. They were modelled by their swept volumes (Table 1) and by the use of correlations for the mass flow rate circulated and electric power consumption. These correlations, expressed as functions of the suction and discharge conditions, as well as of the rotational speed of the frequency converter (for VSD compressors), were individually formulated for each compressor model, based on the data published by the manufacturer [29].

Table 1. Characteristics of the compressors utilised in the supermarket mock-up facility [25].

Compressor pack	Model	Swept volume [$m^3 \cdot h^{-1}$] at 30 Hz	Swept volume [$m^3 \cdot h^{-1}$] at 50 Hz	Swept volume [$m^3 \cdot h^{-1}$] at 87 Hz
LT	2JME-3K + VSD	2.1	3.5	6.1
	2GME-4K		5.0	
Base	4MTC-10K-40S + VSD	3.9	6.5	11.3
Parallel	2KTE-7K-40S + VSD	2.9	4.8	8.4
Pivoting compressors	4MTC-10K-40S + VSD	3.9	6.5	11.3
	2KTE-7K-40S + VSD	2.9	4.8	8.4
	4JTC-15K-40P		9.2	
	4JTC-15K-40P		9.2	

The gas cooler section comprises three plate heat exchangers models and a constant temperature approach (expressed by the difference between the ambient and refrigerant outlet temperature) of 3 K at the gas cooler outlet was assumed. Downstream of the gas cooler section there is an internal heat exchanger, modelled with a plate heat exchanger, with the task of superheating the suction flow to the parallel compressors.

The high pressure control devices were modelled with an orifice-valve model for HPV, and ejector models for the MT multiejector and AC multiejector block. For the sake of simplicity of the steady-state 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. Such an approach is certainly appropriate for the high-resolution step-motor-controlled HPV but should be upgraded in future works for the limited-resolution (16 possible values of the overall opening degree) solenoid-valve-controlled multiejector blocks. The vapour ejectors of the MT multiejector block (EJ_{MT} in Figure 1) were modelled by correlations provided by Banasiak et al. [11], who reported an experimental study on a multiejector block designed for low entrainment ratio and high pressure lift. Since the AC multiejector block (EJ_{AC} in Figure 1), designed for higher entrainment ratio and lower pressure lift, consisted of the same motive nozzles as in the MT block, it was possible to calculate the motive nozzle mass flow rate by using the same equation as for the MT multiejector block (the motive nozzle mass flow rate is dependent purely on the motive nozzle geometry and motive nozzle inlet conditions). As for the modelling of the secondary flow, the actual performance of an ejector is related to the geometrical design of mixer and diffuser, and these sections were designed in a different way for the AC multiejector block. Therefore, taking into account the results from Kriezi et al. [26] for high entrainment ratio ejectors, the efficiency of the AC ejector was set to a constant value of 30%. Modelling of liquid ejectors was avoided in this work, since the paper covers results generated only for steady-state operation, while liquid ejectors should in practice operate in a transient pattern, i.e. removing the excessive liquid from the liquid separator, on demand.

The liquid receiver positioned downstream of the high pressure control devices (and optionally of $EVAP_{AC,2}$) was modelled as a single ideal separator of 260 L. The helical coil positioned inside the liquid receiver, allowing the vapour sucked by the LT compressor pack to be superheated, was approximated by a heat port at the tank connected to a tube. The parallel compressor section sucks vapour from the liquid receiver, with a final superheating provided by the internal heat exchanger IHX_1. If the minimum capacity of the parallel compressors is excessive to maintain the pressure at the liquid receiver or with conventional booster configuration (with no parallel compressors), the flash gas bypass valve is in charge of adjusting this pressure by expanding vapour to the suction of the base compressors. The liquid port of the receiver is connected to the expansion devices of different evaporators and cabinets, through another internal heat exchanger, IHX_2, which subcools the liquid from the receiver and superheats the vapour sucked by the base compressor(s).

The MT and LT evaporators are constituted of five and two helical coaxial tube-in-tube heat exchangers assembled in parallel, respectively. Each evaporator has its individual expansion device. The refrigerant flowing out from MT evaporators and evaporator $EVAP_{AC,1}$ reaches the liquid separator (ideal separator model, 60 L capacity). The base compressor(s) suck vapour refrigerant from the top part of this tank, after passing through the internal heat exchanger IHX_2 and being superheated. An auxiliary suction accumulator was applied downstream of the LT evaporators (ideal separator model, 25 L capacity). The vapour from the outlet of this accumulator is superheated with a heat exchanger placed in the liquid receiver before being sucked by the LT compressor section.

The refrigerant discharged by the LT compressors is cooled down in a plate heat exchanger, denoted as DESUPERHEATER in Figure 1. This loop simulates the heat exchanger that is usually found in refrigeration systems and that releases heat to the air at the machine room. The model for this heat exchanger was fin-and-tube, with air at the temperature at the machine room, which was considered to be identical to the ambient air, but always higher than 20 ºC. The air flow rate was adjusted so that the temperature difference between the refrigerant at the outlet of the desuperheater and the air from the machine room was 3 K. Downstream of the desuperheater, the refrigerant may be delivered to either base or parallel compressors (pivoting discharge of LT compressors).

Two evaporators were modelled (both as plate heat exchangers) and compared for the production of AC. The first AC evaporator ($EVAP_{AC,1}$) uses an expansion valve to control the conditions of the refrigerant at the outlet of the evaporator. As aforementioned, if the AC evaporator is operated as ejector-supported, vapour is sucked by the AC multiejector block. However, it is also possible to connect the same evaporator to the liquid separator at MT pressure level (Figure 3). The second solution consists of an evaporator ($EVAP_{AC,2}$) connected upstream of the liquid receiver This evaporator allows the implementation of AC in a conventional booster configuration.

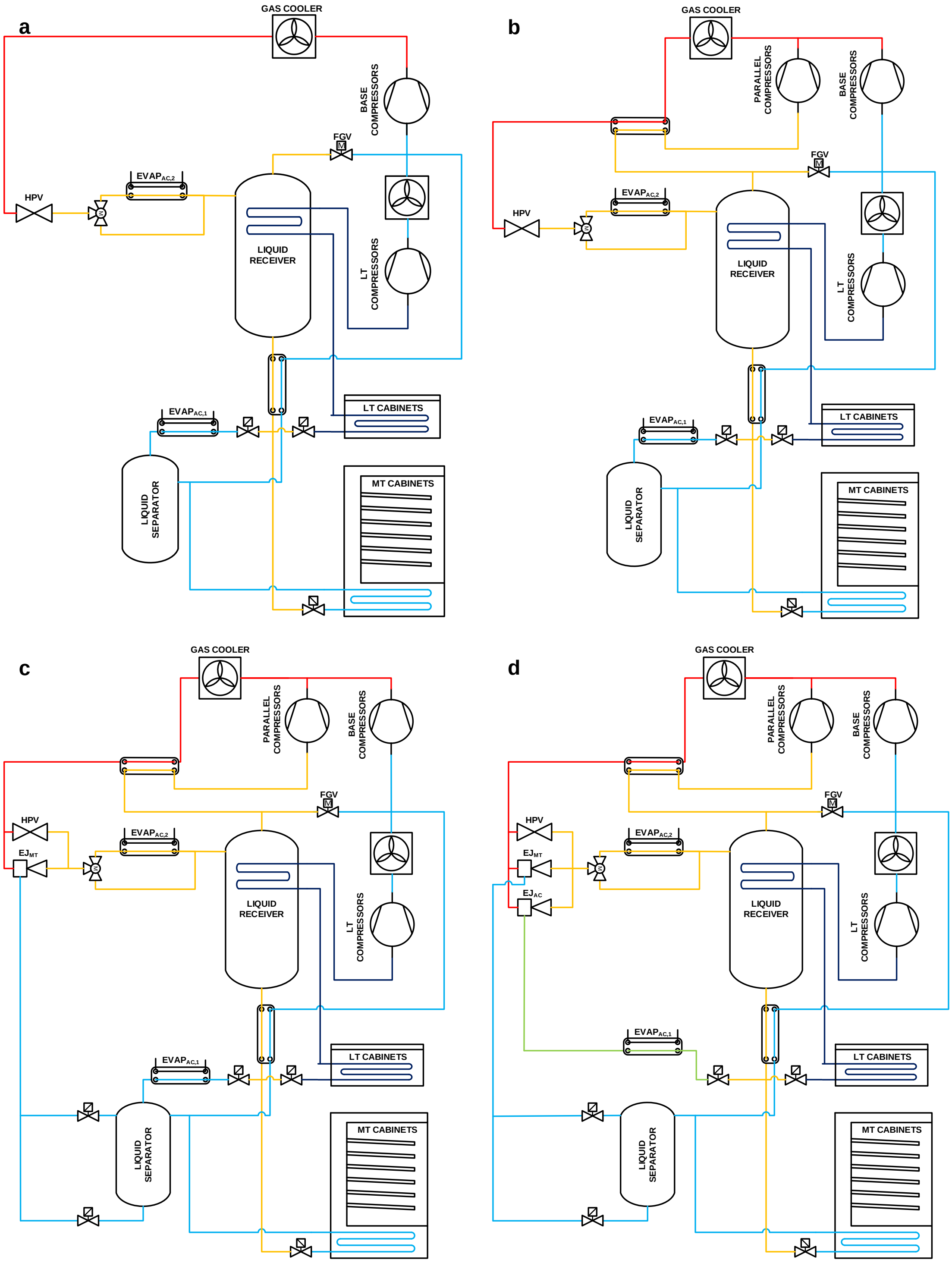


Figure 3. Configurations of transcritical CO_2 refrigeration systems considered in this study. a) Conventional Booster (CB). b) Parallel Compression (PC). c) Parallel Compression with MT multiejector block (PC+EJ_{MT}). d) Parallel Compression with both multiejector blocks (PC+EJ_{MT}+EJ_{AC}).

Further details:

- All plate heat exchangers were taken from the TIL 3.4 library. The pressure drop in refrigerant was calculated with the correlation available in the VDI Heat Atlas [30] for heat exchangers with Chevron plates.
- IHX_1 and IHX_2 were sized considering the actual dimensions of the plate heat exchanger used in systems with the capacity of this one, resulting in heat transfer areas of 1.82 m^2 and 1.53 m^2,

respectively. The heat transfer coefficients for the refrigerant at both sides of each exchanger were estimated to be 2500 W·m^{-2}·K^{-1}.

- The helical coaxial tube-in-tube evaporators were modelled with a tube-in-tube parallel-flow heat exchanger model. The pressure drop was estimated according to the correlation by Konakov [31].
- Valves (flash gas bypass valve, expansion valves, etc.) were modelled by using orifice-valve models available in the aforementioned library.

3.2. Operating ranges

Two simulation stages were performed during this work: Stage 1 comprised system analysis of various configurations of transcritical CO_2 compressor packs working at nominal load at various ambient temperatures, while Stage 2 encompassed study of the effect of floating operating conditions on the power consumption of the two most efficient configurations.

In Stage 1, the configurations considered were Conventional Booster (CB, Figure 3a), Parallel Compression (PC, Figure 3b), Parallel Compression with MT multiejector block (PC+EJ$_{MT}$, Figure 3c), and Parallel Compression with both multiejector blocks (PC+EJ$_{MT}$+EJ$_{AC}$, Figure 3d). In addition, the winter modes described in Section 2.2 (Figure 2) were also studied. Simulations were performed for ambient temperature ranging between 30 °C and -10 °C (with steps of 5 °C), which captures the vast majority of conditions in inhabited areas of Norway throughout the year. Table 2 presents the assumed loads as a function of the ambient conditions, reflecting the maximum loads for a typical Norwegian supermarket. As stated in Walker and Baxter [32], it is justified that MT and LT loads are greater at high ambient temperature conditions. However, due to the increasing tendency to use air-tight LT counters, the maximum LT load was considered as a constant. In addition, AC load depends on the ambient temperature and exists only when this is greater than 20 °C.

Table 2. The assumed MT, LT and AC load profiles, and main conditions used for Stage 1 simulations as a function of the ambient temperature.

T_{AMB} [°C]	30	25	20	15	10	5	0	-5	-10
MT load [kW]	60	55	50						
LT load [kW]	10								
AC load [kW]	45	25	0						
Discharge pressure [bar]	88	74	68.9	61.4	54.6	48.4	42.8	42.8	42.8
Liquid receiver pressure [bar]	40						-		
AC multiejector pressure lift [bar]	5		-						
Pressure MT evaporators [bar]	28								
Pressure LT evaporators [bar]	15								
Superheating setpoint for MT, LT and AC1 evaporators [K]	8								

No optimisation of the operational degrees of freedom was performed in Stage 1. Therefore, the setpoint values of the main pressure levels in the refrigerant cycle were set to values typically used in conventional refrigeration systems with R744 as refrigerant [33], as stated in Table 2. Concerning the discharge pressure, in subcritical conditions a constant subcooling degree (set at 5 K) was maintained. The lower limit of the condensation pressure setpoint was 42.8 bar, due to the minimum discharge pressure allowed by the compressor's manufacturer for suction pressure equal to 28 bar [25]. At transcritical operation, the selection of the discharge pressure setpoint was not explicitly optimised for each configuration in every operating condition but followed a straight line in a temperature-pressure diagram given by two points: 25 °C/74 bar and 30 °C/88 bar.

Due to the pivoting suction ports of base and parallel compressors, it was possible to meet the loads with different combinations of compressors in each section. For the sake of comparison, the combination that led to the lowest power consumption was chosen. In addition, the pivoting discharge of the LT compressors allowed the effect of discharging to the base or parallel compressors on the overall power consumption to be simulated. Another possibility to consider was air conditioning load met either with the evaporator placed downstream of the liquid receiver ($EVAP_{AC,1}$) or with the evaporator located between the high pressure control devices and the liquid receiver ($EVAP_{AC,2}$).

In Stage 2, the two configurations that led to lowest power consumptions in Stage 1, i.e. the system with parallel compression and both multiejector blocks (S_1) and another with parallel compression, MT multiejector block and AC production upstream of the liquid receiver (S_2) were chosen and the effect of floating operating conditions on the power consumption were analysed. They can both be considered one system when there are no AC loads, i.e. when the ambient temperature is lower than or equal to 20 °C. Simulations were performed considering the same ambient temperatures as in the previous case (from 30 °C to -10 °C, steps of 5 °C). The ranges of the operating conditions considered for the simulations are also included in Table 3.

Table 3. The assumed MT, LT and AC load profiles, and changeability range used for Stage 2 simulations as a function of the ambient temperature.

T_{AMB} [°C]	30	25	20	15	10	5	0	-5	-10
MT load [kW]	60	55	50						
LT load [kW]	10								
AC load [kW]	45	25	0						
Discharge pressure operating range [bar]	78 - 83	69 - 75	68.9	61.4	54.6	48.4	42.8; 45.6; 48		
Liquid receiver pressure operating range [bar]	35 - 50								
AC multiejector pressure lift [bar]	5		-						
MT evaporator conditions: pressure [bar]/superheating [K]/quality	28/8/-; 30/0/0.95; 32/0/0.95								
LT evaporator conditions: pressure [bar] (superheating [K])	15 (8)								

The discharge pressure in transcritical conditions was optimised for each set of conditions simulated, and is a function of the different pressure levels, the configuration and the ambient temperature. In subcritical conditions, it was calculated with a constant subcooling degree (set at 5 K). The operational envelope of compressors [25] sets the lowest discharge pressure of the compressors used as a function of the conditions at the suction. In the case of the calculated pressure being less than the technical limit, extra simulations were performed, taking into account this constraint.

The lower limit for the receiver pressure in this study is a result of the MT pressure level and of maintaining a minimum pressure drop at the expansion devices of the MT cabinets that guarantees their proper operation [14]. Two issues were considered for the definition of the higher limit. The first concerns the maximum operational pressure difference (MOPD) of expansion valves, particularly in LT cabinets. Taking as an example a MOPD of 35 bar [34] and LT evaporation pressure of 15 bar, the receiver pressure should not exceed 50 bar. The second issue is associated with the production of AC, since the temperature of the secondary fluid at the outlet of the AC evaporator should match the requirements of the AC system. With chilled-water systems, it should range from 4 to 13 °C, usually being 7 °C [35-37]. Therefore, the AC evaporation pressure should not exceed 45 bar (10 °C evaporation temperature, approximately). As a consequence, 45 bar is the maximum receiver pressure allowed if the AC evaporator is upstream of the liquid receiver, S_2 configuration. It can be lifted to 50 bar with the other AC production system, S_1, due to the constant pressure lift (pressure difference between the AC evaporation pressure and the receiver pressure) of 5 bar considered for the AC multiejector block.

As base value for the analysis, 28 bar (-8 °C evaporation temperature) was considered as the MT evaporation pressure, with 8 K superheating at the outlet of the evaporator. The positive impact of overfeeding the evaporators on the power consumption of the refrigeration system was evaluated with pressures of 30 and 32 bar (-5.6 and -3.2 °C evaporation temperature, respectively). This evaluation was restricted to simulations with the discharge pressure at the optimal value and different receiver pressures and ambient temperatures. With flooded conditions, it was assumed that the expansion valves are able to maintain the requested values of the quality at the outlet of the evaporators (0.95).

Concerning the LT evaporation pressure, 15 bar (-28.6 °C evaporation temperature) was considered for the analysis, with 8 K superheating at the outlet of the evaporator. No other values of pressure were taken into account, since the effect on the overall power consumption is expected to be low due to the low share of the LT load in the total.

4. Results

4.1. Configurations

Figure 4 presents the overall compressor power consumption for the CB-mode simulation, labelled with the ambient temperature and the AC evaporator and AC load (if any). The power consumption increases as the ambient temperature conditions increase. As depicted in the graph, at 30 °C, the maximum AC load that could be met with the CB configuration was 35 kW (lower than the original requirement of AC load included in Table 2), even with the maximum base compressor capacity (one compressor directly connected and the four compressors with pivoting suction).

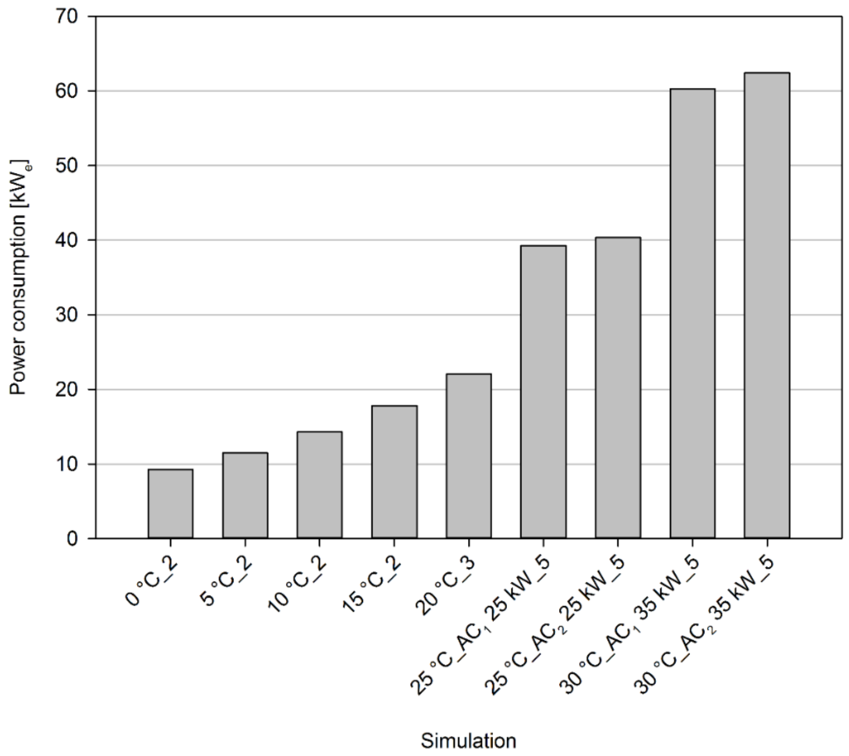


Figure 4. Electric power consumption of the compressors for conventional booster system. The labels on abscissae include the ambient temperature, the AC evaporator used and the AC load (if any), and the number of compressors in operation.

It can be observed that the power consumption is slightly higher with the evaporator placed downstream of the high pressure control devices (AC_2) than with the evaporator that operates in this case at the same pressure level as the MT evaporators (AC_1), at the same ambient temperature. This behaviour could be seen as contradictory, since the AC load is produced at the liquid receiver pressure. However, the mass flow rates of refrigerant circulated by the base compressors and the mass flow rate bypassed by the flash-gas bypass valves are greater with the AC_2 configuration than with the AC_1 configuration.

Figure 5 compares the overall electric power consumption of the compressors of a PC refrigeration system at 30 °C of ambient temperature, as a function of: (i) the combination of compressors in the base and parallel compressor sections, (ii) the AC evaporator used and AC load, and (iii) the compressor section that is connected to the discharge of the LT compressors.

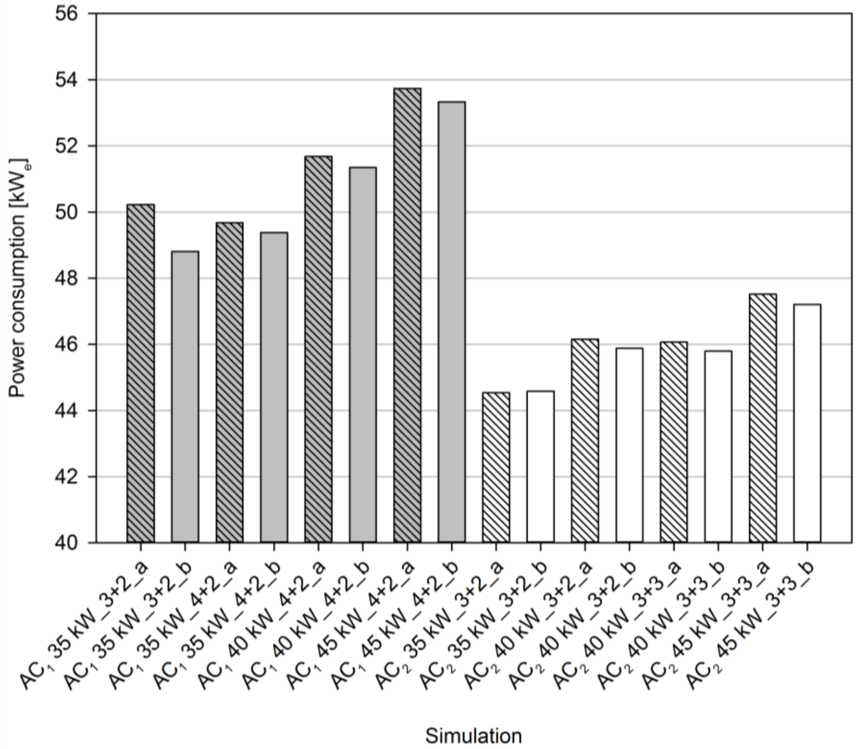


Figure 5. Electric power consumption of the compressors with parallel compression configuration, at 30 °C of ambient temperature, and with different combinations of compressors in the sections and pivoting discharge of the LT compressors. The labels on abscissae include the AC evaporator used (AC_1, downstream of the liquid receiver, in grey bars, or AC_2, upstream of the liquid receiver, in white bars); the AC load (35 kW, 40 kW, 45 kW); the number of compressors per pack (“number of base compressors” + “number of parallel compressors”); and where the discharge of the LT compressors is delivered (“a” to the base compressors, indicated with diagonally-patterned lines, and “b” to the parallel compressors, clean bars).

The effect of the compressor configuration in both sections on the electric power consumption between AC_1 35 kW_3+2_a (simulation with 35 kW of AC load at the AC_1 evaporator, three compressors in the base pack and two in the parallel pack, LT compressors discharging to the suction of the base pack) and AC_1 35 kW_4+2_a (as the previous one but with four compressors in the base pack) is lower than 550 W. The difference does not reach 90 W between AC_2 40 kW_3+2_a (simulation with 40 kW of AC load at the AC_2 evaporator, three compressors in the base pack and two in the parallel pack, LT compressors discharging to the suction of the base pack) and AC_2 40 kW_3+3_a (with three compressors in the parallel pack instead). These variations are mainly related to the fact that the rotational speed at which the VSD compressors of each section operate depends on the number of active compressors in the section, and the overall compressor efficiency is highly dependent on the rotational speed.

The overall compressor power consumption is lower when the discharge of the LT compressors is connected to the suction port of parallel compressors. With the rest of the parameters kept constant, the difference was approximately 400 W, if comparing AC_1 45 kW_4+2_a (simulation with 45 kW of AC load at the AC_1 evaporator, four compressors in the base pack and two in the parallel pack, LT compressors discharging to the suction of the base pack) with AC_1 45 kW_4+2_b (with the only difference being that the LT compressors discharge to the suction of the parallel pack). The difference rose to 1400 W, when the comparison was between AC_1 35 kW_3+2_a (simulation with 35 kW of AC load at the AC_1 evaporator, three compressors in the base pack and two in the parallel pack, LT compressors discharging to the suction of the base pack) and AC_1 35 kW_3+2_b (as the previous one but with the LT compressors discharging to the suction of the parallel pack). This general reduction is associated with the fact that a greater share of the total mass flow rate is compressed by the parallel compressors, which operate with a lower pressure ratio and higher efficiencies, and compensate for the increase in the consumption of the LT compressor pack due to greater pressure ratio. In addition, such a configuration allows the parallel compressors to work at lower ambient temperatures, when not so much vapour needs to be drawn from the liquid receiver.

The implementation of pivoting suction of base/parallel compressors might not be decisive, from the point of view of power consumption optimisation, but it does improve the flexibility of the unit. A system with pivoting suction at several base/parallel compressors allows the operational range of the refrigeration system to be enlarged, keeping the capacity and number of compressors in the test rig constant. This solution is particularly interesting for the configuration with parallel compression and both multiejector blocks ($PC+EJ_{MT}+EJ_{AC}$), since MT and AC evaporators operate at different pressure levels in this case, each of them controlled by a different section of compressors (base and parallel, respectively). Therefore, changing the share of the capacities of base and parallel compressors has an important effect on the maximum AC and MT loads that can be met by the system. Figure 6 illustrates these maximum loads for a system with $PC+EJ_{MT}+EJ_{AC}$ configuration and the different combinations of compressors allowed by the pivoting suction system. With the combination of two base compressors and three parallel compressors (and LT discharge to the parallel compressors), it is possible to meet loads lower than or equal to those limited by the solid black line, including the assumed specifications of 60 kW MT load and 45 kW AC load (shown with an X in Figure 6). If the parallel compressor section capacity increases due to the pivoting suction system (one base compressor and four parallel compressors), the area of MT/AC loads is that delimited by the dashed black line. In this case, the pressure lift of the AC multiejector block is limited to 3 bar (lower than the 5 bar considered for the simulations), due to the coordinated control strategy of the MT and AC multiejectors, i.e. the AC multiejector is requesting a greater motive nozzle opening than that which can be provided, in order to control the high pressure level. Similarly, if the capacity of base compressors increases, reaching three base compressors and two parallel compressors, the new region of MT/AC loads is that limited by the dashed-double dotted black line.

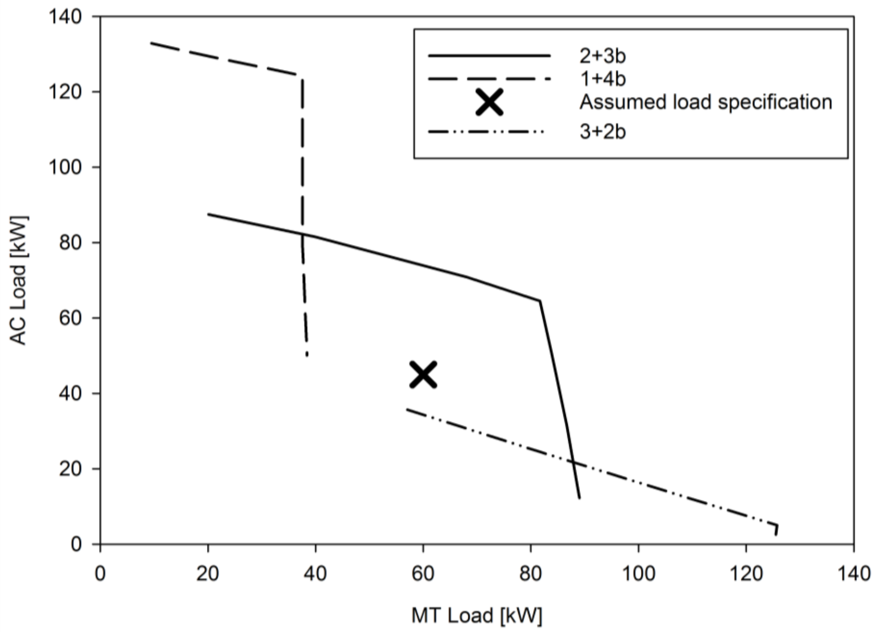


Figure 6. Maximum operating range (AC load vs. MT load) for systems with parallel compression and MT and AC multiejector blocks (PC_EJ$_{MT}$+EJ$_{AC}$), at 30 °C of ambient temperature; with different combinations of compressors in the base and parallel sections, allowed by the pivoting suction solution.

Figure 7 shows the overall compressor power consumption as a function of the ambient temperature, of the different configurations considered in this study and with AC loads of 35 kW (at 30 °C ambient temperature) and 25 kW (at 25 °C) at the evaporator placed downstream of the liquid receiver (AC$_1$). For each condition and configuration, the bar illustrated in the graph corresponds to the combination of compressors in the base and parallel compressors in the packs and selection of the position of the pivoting discharge of LT compressors that leads to the lowest power consumption.

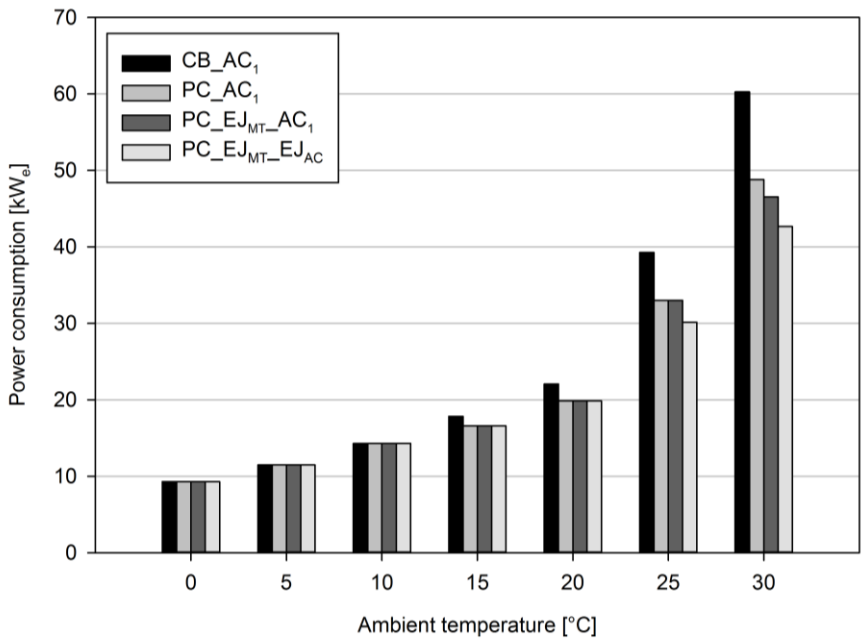


Figure 7. Electric power consumption of the compressors against the ambient temperature for: conventional booster (CB), parallel compression (PC), parallel compression with MT multiejector block (PC_EJ$_{MT}$) and parallel compression with MT and AC multiejector blocks (PC_EJ$_{MT}$+EJ$_{AC}$); AC production at the evaporator downstream of the liquid receiver (AC$_1$), and with AC loads of 35 kW and 25 kW at 30 °C and 25 °C, respectively.

The PC mode improves the performance and reduces the electric consumption of the refrigeration unit, if compared with a CB, for ambient temperatures equal to or higher than 15 °C. At 15 °C, this reduction is 1260 W (7% approximately) and reaches 11460 W (around 19%) at 30 °C of ambient temperature. The minimum capacity of the parallel compressor pack (with only one compressor) is too large at ambient temperatures lower than or equal to 10 °C and, therefore, the system should operate as a conventional booster, with the pressure at the liquid receiver controlled by the flash-gas bypass valve. The implementation of the MT multiejector block to the PC system has a beneficial effect only at an ambient temperature of 30 °C, with the conditions considered during the simulations, diminishing the electric power consumption from 48810 W with the PC system to 46550 W with the PC+EJ_{MT}. With lower ambient temperatures, the conditions at the motive nozzle of the MT multiejector block are not sufficient to suck any vapour from the liquid separator and overcome the pressure lift of 12 bar (from 28 bar of the liquid separator to 40 bar of the liquid receiver). Finally, the use of both multiejector blocks for MT and AC allows the power consumption to be reduced, not only at 30 °C but also at 25 °C, if compared with the system using only the MT multiejector block. The reason for this is that the pressure lift for the AC Multiejector block is only 5 bar (the AC evaporator operates in this case at 35 bar). The difference between them was almost 3870 W (around 8.3%) at 30 °C and 2850 W (around 8.6%) at 25 °C.

Figure 8 focuses on the production of AC and on how the different configurations and the use of one or the other evaporator affects the electric power consumption of the refrigeration system. Three different AC loads were considered for the simulations with ambient temperature of 30 °C, 35 kW, 40 kW and 45 kW, since it was not possible to meet the initial requirement of 45 kW with CB configuration, independently of the evaporator used or the number of compressors.

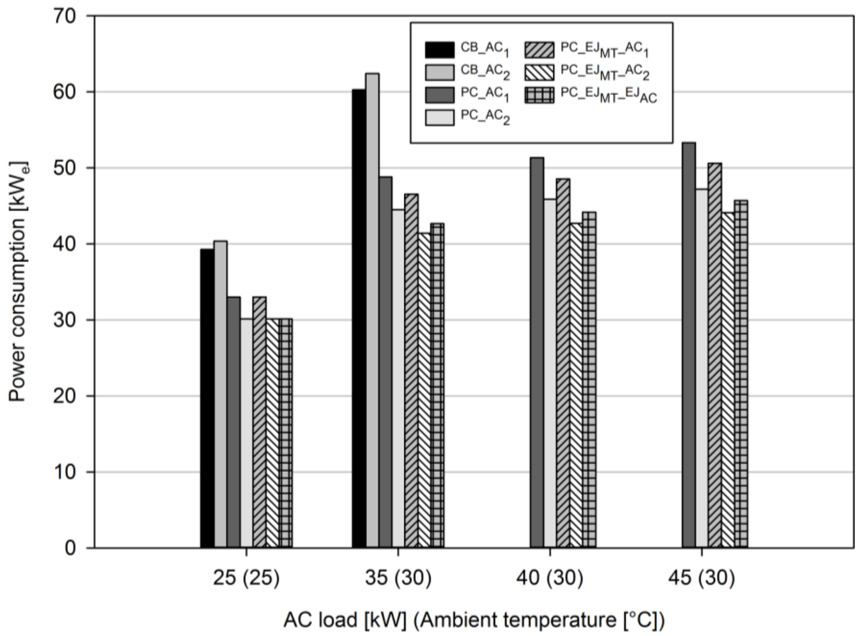


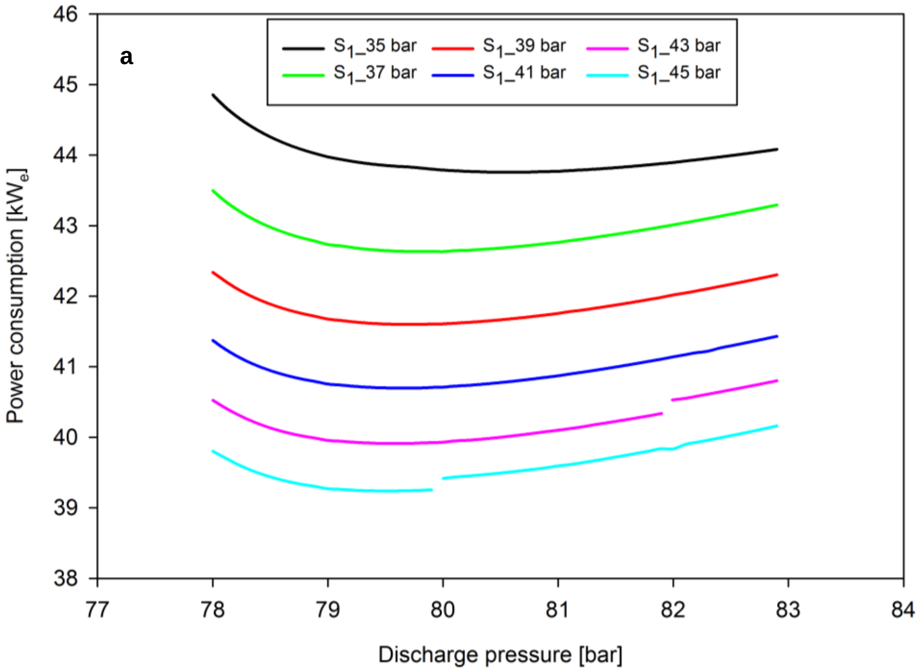
Figure 8. Electric power consumption of the compressors against the AC load at ambient temperatures of 25 °C and 30 °C for: conventional booster (CB), parallel compression (PC), parallel compression with MT multiejector block (PC_EJ_{MT}) and parallel compression with MT and AC multiejector blocks (PC_EJ_{MT}+EJ_{AC}). The two AC production solutions were considered, either with the evaporator downstream of the liquid receiver (AC_1) or with the evaporator upstream of the liquid receiver (AC_2).

In contrast to the trend presented in Figure 4 for CB systems, the power consumption of the PC refrigeration system is lower with the AC evaporator at the pressure level of the liquid receiver (AC_2) than with the evaporator downstream of the liquid receiver (AC_1). The explanation is that the vapour produced at the evaporator is sucked by the parallel compressors, which operate at a lower pressure ratio and higher efficiencies than the base compressors, and not expanded to the suction of the base compressors and recompressed. However, it is necessary to verify whether the conditions in the evaporator AC_2 can ensure the required temperature of the secondary fluid at the outlet (depending on the characteristics of the AC system). At 30 °C of ambient temperature, the system with the lowest power consumption, independently of the AC load, is the $PC+EJ_{MT}$ using the evaporator AC_2, closely followed by the $PC+EJ_{MT}+EJ_{AC}$ (it only works with AC_1 evaporator) and the PC using the evaporator AC_2. At 25 °C, the performance of those three systems is very similar, since the MT multiejector block is not able to overcome the pressure lift of 12 bar, as aforementioned, with those conditions at the motive nozzle. Therefore, the use of the AC multiejector block compensates for the "penalisation" of using the evaporator AC_1, which works at a lower pressure level than AC_2.

4.2. Operating conditions

Transcritical conditions

Figure 9 presents the results of the simulations for the optimisation of the discharge pressure at 30 °C ambient temperature of both systems studied, S_1 (Figure 9a) and S_2 (Figure 9b), with MT evaporation pressure of 28 bar. This optimisation was performed by confronting overall compressor power consumption and discharge pressure, as a function of the pressure at the AC evaporator.



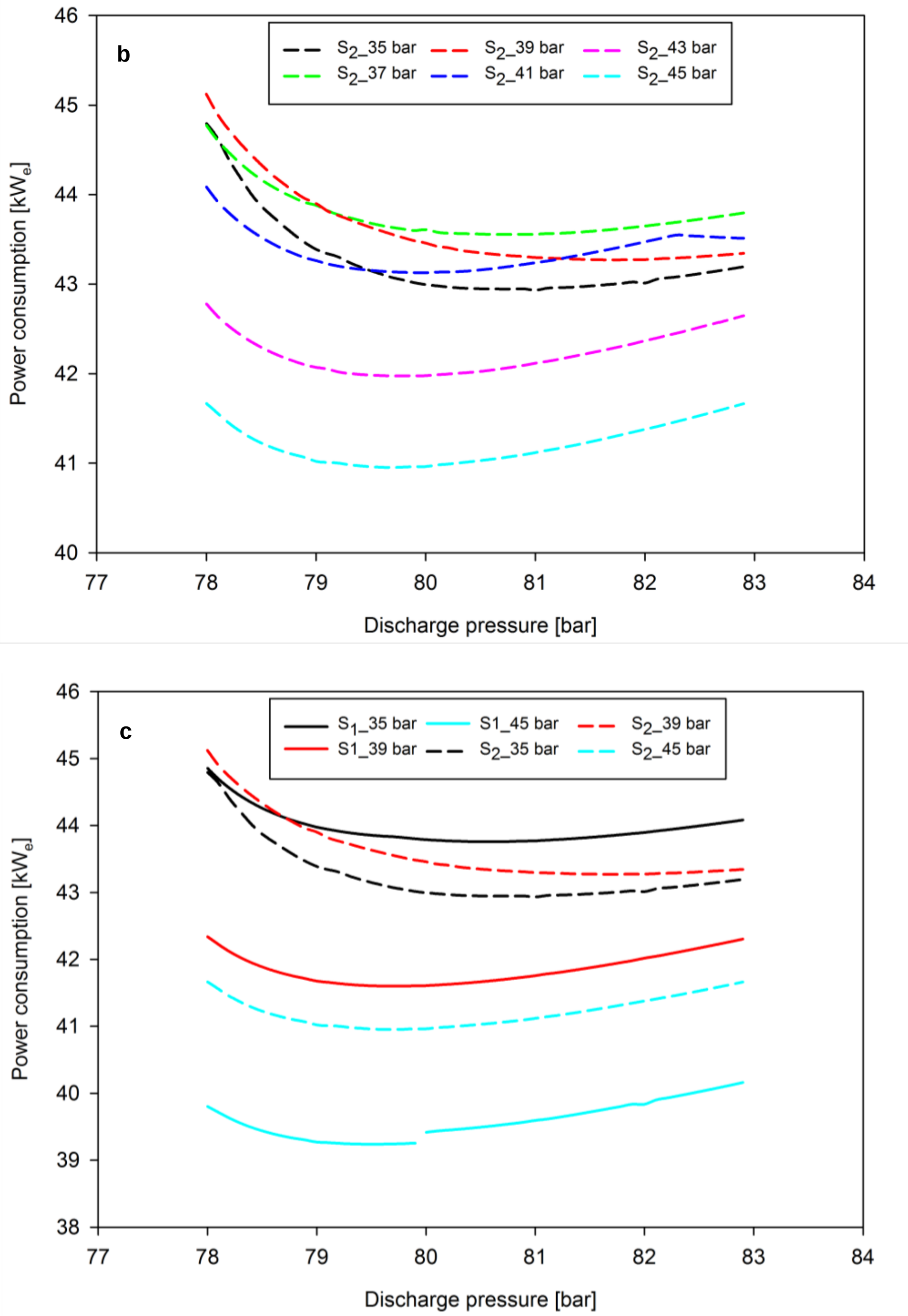


Figure 9. Overall compressor power consumption against discharge pressure of the base and parallel compressors, with the system using both multiejector blocks S1 (a), with the system with one multiejector block S2 (b), and a comparison of both (c), as a function of the AC evaporation pressure (in the legend); with constant values of MT evaporation pressure equal to 28 bar and ambient temperature equal to 30 °C. The discontinuities observed in three of the curves (figures a and c) are caused by the need of changing the number of active compressors throughout the range represented in order to meet the existing demands at those conditions.

In the range studied and with the S_1 refrigeration system, the power consumption of the refrigeration system decreases as the pressure at the AC evaporator increases, independently of the discharge pressure (Figure 9a). This occurs since the parallel compressors operate with a lower pressure ratio as the AC evaporator pressure increases, which has a positive effect on their overall performance. The trend is generally the same with the second system S_2 (Figure 9b) and the upper range of AC evaporator pressure (from 41 to 45 bar). However, it is not so straightforward when it is less than 41 bar, since there is a positive effect of the MT multiejector block that compensates for the increased pressure ratio of parallel compressors. As the AC evaporator (and liquid receiver) pressure decreases, the pressure lift at the MT multiejector (from the MT pressure level to the liquid receiver pressure level) diminishes, and the mass flow rate sucked by the block increases, shifting part of the load from the base compressors to the parallel compressors. As a consequence, the overall compressor power consumption decreases.

The power consumption decreases first, reaches a minimum and then increases as the discharge pressure rises, independently of the system (S_1 or S_2). This minimum (optimum) occurs at slightly greater discharge pressure with S_2 and in the lower range of AC evaporator (liquid receiver) pressures (from 35 to 39 bar). The reason for this is that the MT multiejector block increases its efficiency and entrainment ratio with higher discharge pressure and compensates partially for the increase of the pressure ratio of operation of the base and parallel compressor sections. The optimum discharge pressures, as a function of the AC evaporator pressure and system, at 30 °C ambient temperature, are included in Table 4. The table incorporates these values not only for an MT evaporation pressure of 28 bar but also for 30 bar and 32 bar (with flooded conditions of the evaporators), since it was observed that this pressure also affects the optimal discharge pressure.

Table 4. Optimum discharge pressures and overall compressor power consumption for the systems using both multiejector blocks S_1 and one multiejector block S_2, as a function of the pressure at the AC evaporator used by each system, of the MT evaporation pressure, and with ambient temperature of 30 °C.

System	p_{MT} [bar]	p_{AC} [bar]	$p_{dis,opt}$ [bar]	W_e [W]	System	p_{MT} [bar]	p_{AC} [bar]	$p_{dis,opt}$ [bar]	W_e [W]
S_1	28	35	80.6	43759	S_2	28	35	81	42935
		37	80	42633			37	80.8	43557
		39	79.7	41601			39	81.7	43272
		41	79.6	40697			41	80	43128
		43	79.6	39912			43	79.8	41974
		45	79.5	39237			45	79.7	40953
	30	35	80.3	42385		30	35	80.6	41937
		37	79.9	41681			37	80.5	40762
		39	79.8	40610			39	82.3	40783
		41	79.7	39672			41	82.1	41541
		43	79.9	39074			43	79.8	40873
		45	79.7	38312			45	79.7	39783
	32	35	80.1	40829		32	35	79.9*	42027
		37	80.3	40184			37	80.5	39445
		39	79.8	39332			39	80.5	38589
		41	79.7	38333			41	81	39439
		43	80	37683			43	79.9	39760
		45	79.7	36849			45	79.8	38615
*p_{MT} lower than the setpoint, even with minimum base compressor capacity.									

When comparing both systems, S_1 and S_2, it is important that the pressure of the refrigerant at the AC evaporator (not the liquid receiver pressure) is identical. In this way and considered identical evaporators, both systems should meet the same AC cooling loads at the same secondary fluid temperature levels. Figure 9c depicts this comparison in terms of overall compressor power consumption, at an ambient temperature of 30 °C and MT and LT evaporation pressures of 28 bar and 15 bar, respectively. In the range considered for the simulations, the lowest values were achieved with S_1 and AC evaporation pressure of 45 bar, followed by S_2 at the same pressure level. The difference in power consumption between these two situations was around 1720 W, if the optimal discharge pressure is considered for each of them (79.5 bar and 79.7 bar, respectively). As the AC evaporation pressure decreases, the trend changes, and system S_2 even outperforms S_1, reaching approximately 820 W power consumption difference at 35 bar and optimal discharge pressure (81 bar and 80.6 bar, respectively).

From the power consumption point of view, food retailers should be encouraged to use air conditioning systems designed for the highest temperature possible, allowing the pressure at the AC evaporator to increase as much as possible. This could be done, for example, by implementing the AC evaporators directly into the air handling units or in the shop, eliminating the use of an intermediate fluid [38]. In addition, it is indispensable to evaluate whether the performance enhancement of the refrigeration system, achieved due to the use of the AC multiejector block, compensates for the cost of this component and the more complex control of the discharge pressure due to the coordinated action of both multiejector blocks. If the air conditioning system is designed in a way that requires low secondary fluid temperature, leading to AC evaporation pressures close to 35 bar, the system with only MT multiejector block seems more convenient and simpler. However, the parallel compressor section in this case needs considerable capacity (up to four compressors). Therefore, it is an advantage to have the capability of moving compressors between the base and parallel compressor sections with the pivoting suction solution, in order to adapt the capacity to the conditions.

The minimisation of the power consumption as a function of the discharge pressure, with different AC evaporation pressures, was also studied at 25 °C ambient temperature. According to the simulations, reducing the discharge pressure (and reaching the subcritical region) leads to lower power consumption. Taking into account that a temperature approach of 3 K was considered at the outlet of the gas coolers, at these ambient conditions the temperature at the outlet of the gas cooler is 28 °C, which corresponds to a condensation pressure of approximately 68.9 bar, and it should be possible to have a discharge pressure as low as this value. Nevertheless, a typical requirement for the control of such systems is to keep a subcooling around 5 K, and this would mean that the condensation pressure should correspond to a temperature of 33 °C, which is impossible since these are already transcritical conditions with CO_2. Therefore, 74 bar was high enough to be clearly in the transcritical region but low enough in order to guarantee an acceptable power consumption (the power consumption keeps increasing as the discharge pressure rises).

Figure 10 illustrates the effect, of increasing the pressure at the MT evaporators, on the overall compressor power consumption of the refrigeration unit, at ambient temperatures of 30 °C (Figure 10a) and 25 °C (Figure 10b) and as a function of the AC evaporator pressure. Simulations were conducted, considering the discharge pressure that leads to minimum power consumption, detailed in Table 4.

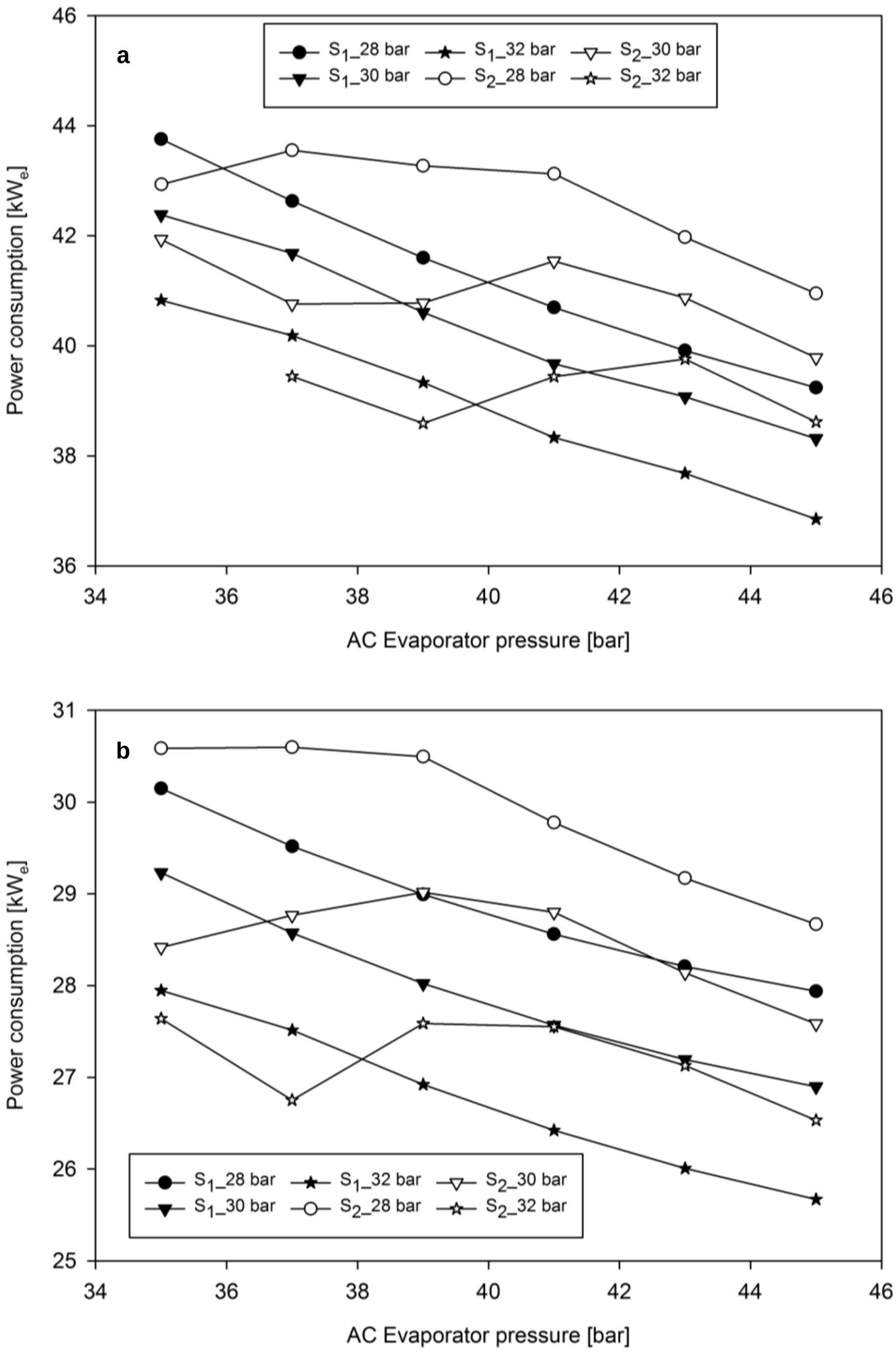
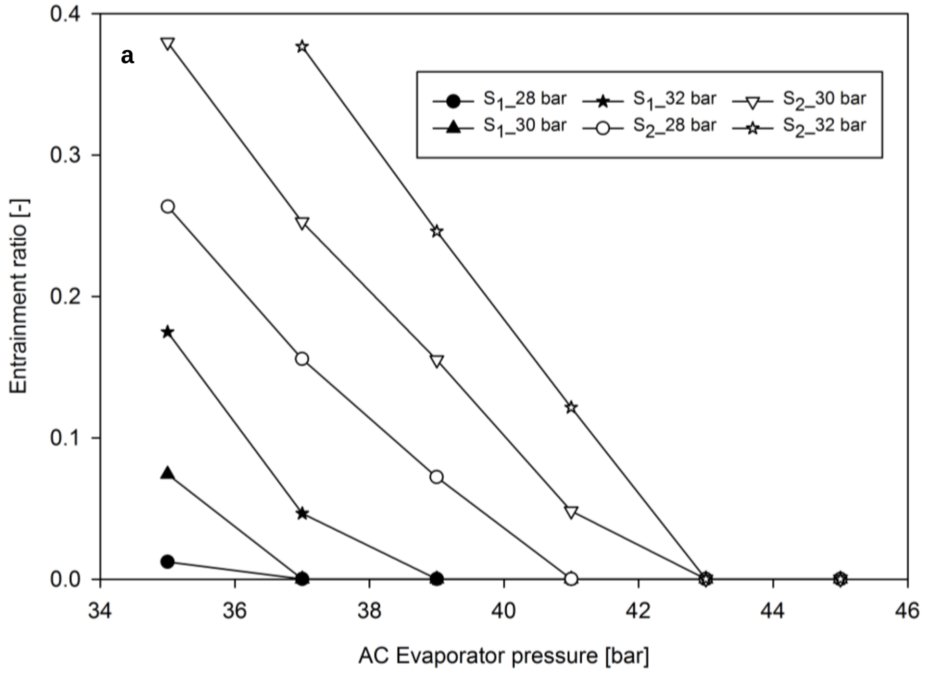


Figure 10. Overall compressor power consumption against pressure at the AC evaporator, with the systems using both multiejector blocks, S_1, and one multiejector block, S_2, as a function of the ambient temperature: 30 °C (a) and 25 °C (b). Pressure values in the legend refer to the MT evaporation pressure.

Independently of the system or temperature, the power consumption decreases if the MT evaporation pressure rises. The reason for this is twofold: on the one hand, by increasing this parameter, the pressure ratio of the base compressors decreases and their power consumption diminishes; on the other hand, the pressure lift at which the MT multiejector block operates decreases as well, if the liquid receiver pressure and AC evaporation pressure is kept, increasing the mass flow rate of refrigerant that can be transferred from the liquid separator (at MT evaporation pressure) to the liquid receiver, as depicted in Figure 11. With S_1 and ambient temperature of 30 °C, the difference can be as large as 1370 W, when it increases from 28 to 30 bar, and 2930 W from 28 to 32 bar (both at 35 bar AC evaporation pressure). If the ambient temperature is 25 °C, the difference is approximately 1040 W, when it goes from 28 to 30 bar, and 2270 W from 28 to 32 bar (both at 45 bar AC evaporation pressure). Similarly, with S_2 and ambient temperature of 30 °C, the reduction of the power consumption reaches 2800 W and 4110 W when the MT evaporation increases from 28 to 30 bar and from 28 to 32 bar, respectively (both at 37 bar AC evaporation pressure). If the ambient temperature is 25 °C, the difference is approximately 2170 W, when it goes from 28 to 30 bar, and 3850 W from 28 to 32 bar (at 35 and 37 bar AC evaporation pressure, respectively). It should be pointed out that, with S_2, 35 bar of AC evaporation pressure and 30 °C of ambient temperature, the minimum capacity of the base compressors was too great to maintain 32 bar at the MT evaporator, and therefore this point is not represented.



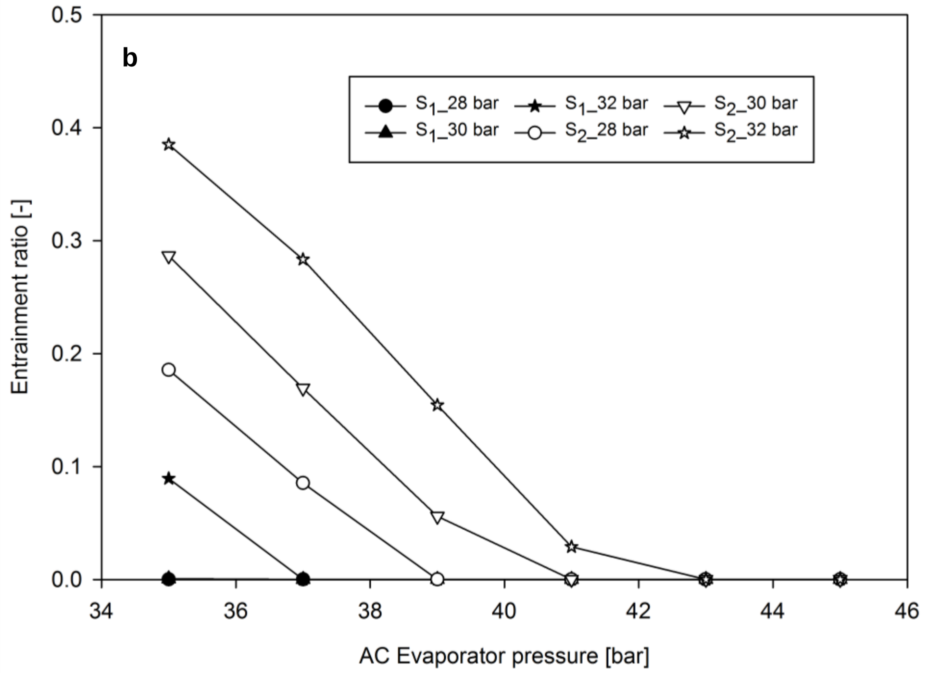


Figure 11. Entrainment ratio of the MT multiejector block vs. AC evaporation pressure, with the system using both multiejector blocks, S1, and that using one multiejector block, S2, as a function of the ambient temperatures: 30 °C (a) and 25 °C (b). Pressure values in the legend refer to the MT evaporation pressure.

The comparison between systems generally favours S_1, since it can have a greater liquid receiver pressure than S_2 with an identical AC evaporator pressure. However, for the same conditions of MT evaporator pressure and ambient temperature, the lower the AC evaporator pressure, the better S_2 performs, compared to S_1. This behaviour is explained by the enhancement of the entrainment ratio of the MT multiejector block, as shown in Figure 11. For instance, with S_2, ambient temperature of 30 °C, MT evaporation pressure of 28 bar and 39 bar AC evaporation pressure, the entrainment ratio is 0.072 and reaches 0.156 and 0.264 if AC evaporation pressure is reduced to 37 and 35 bar AC, respectively. At 30 °C ambient temperature (Figure 11a), the entrainment ratio is in general higher than at 25 °C (Figure 11b), since the multiejector block is performing better. As an example, it falls from 0.380 to 0.287 at 35 bar and 30 bar of AC and MT evaporation pressures, respectively. This justifies the fact that more simulations with S_2 had lower power consumption than those with S_1 at 30 °C and at 25 °C. In addition, by increasing the MT evaporation pressure, it is possible to delay the boundary of AC evaporation pressure between the region at which it is more convenient to use S_2 and that at which S_1 is better. Again, this is associated with the reduction in the pressure lift, at which the MT multiejector block operates, due to the rise of the MT evaporation pressure and the increase in its entrainment ratio. As seen in Figure 10a, at 30 °C and 28 bar MT pressure, S_2 seems convenient from the power consumption point of view, only when the AC evaporation pressure is around 35 bar, but if the MT pressure rises to 32 bar, it becomes the preferred option at 37 or 39 bar as well. When the ambient temperature is 25 °C the behaviour is similar, with the only difference that S_1 requires always less power consumption than S_2 if the MT evaporation pressure is 28 bar (Figure 10b).

Subcritical conditions

With subcritical conditions (ambient temperatures from 5 to 20 °C), there is no AC cooling load and, thus, it is unnecessary to differentiate between system S_1 and S_2 (the AC multiejector block will never be in operation).

Figure 12 illustrates the power consumption of the refrigeration system under subcritical conditions, as a function of the liquid receiver pressure and MT evaporation pressure. A first conclusion drawn from the figure is that the power consumption decreases as the ambient temperature drops, independently of the other parameters. The reason is that the condensation pressure that corresponds to each ambient temperature (and gas cooler outlet temperature) is also reduced, and the pressure ratio of both base and parallel compressors falls too.

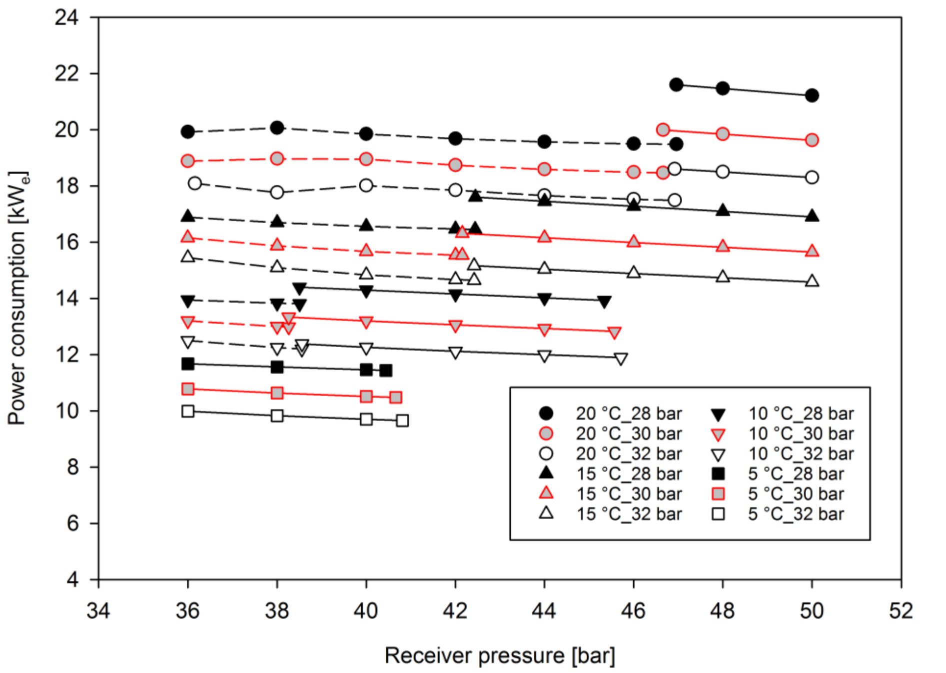


Figure 12. Overall compressor power consumption vs. pressure at the liquid receiver, as a function of the MT evaporation pressure and ambient temperatures from 5 °C to 20 °C. There is a discontinuity in the data of each series caused by the change from conventional booster (solid line) to parallel compressor (dashed line) configuration.

Figure 12 also shows that, if we choose a group of data at the same ambient temperature and MT evaporation pressure, the power consumption decreases as the receiver pressure rises. Nevertheless, each group of data is broken at a certain value of liquid receiver pressure, and there is a gap caused by the change from parallel compression operation (dashed line) to conventional booster (solid line). When the liquid receiver pressure is in the low range, the parallel compression section can be kept active, but when it is the higher range, the smallest capacity of the parallel compressor section is too high in order to keep the receiver pressure at the setpoint value. The limit of liquid receiver pressure between the parallel compression and conventional booster operation decreases as the ambient temperature diminishes. There is an exception at 5 °C ambient temperature, at which it is only possible to operate in conventional booster configuration, independently of the liquid receiver or MT evaporation pressures. The limit of pressure at 20 °C ambient temperature and 28 bar MT evaporation pressure takes place at approximately 47 bar, and at that point the power consumption with the parallel compressor configuration is 9.8% lower than conventional booster configuration. Similarly, at 10 °C ambient temperature, 30 bar MT pressure and 38 bar liquid receiver, the parallel compression section can be enabled and the power consumption is 13010 W, but it is necessary to reach a liquid receiver pressure of 44 bar to have a similar power consumption operating as conventional booster. However, the liquid receiver pressure for parallel compression at 10 °C ambient temperature is technically limited at a value slightly greater than 37 bar due to the operational envelope of compressors [25] and given the discharge pressure of 54.6 bar.

The effect of increasing the MT evaporation pressure on the power consumption with subcritical conditions, illustrated in Figure 8a, is analogous to that observed with transcritical conditions. When this pressure increases from 28 to 30 bar and from 28 to 32 bar, at a liquid receiver pressure of 42 bar and

15 °C ambient temperature, the beneficial effect on the power consumption can be as high as 5.6% and 10.9%, respectively. If the temperature is 10 °C and with the same liquid receiver pressure, this reduction of the power consumption is 7.7% and 14.4%, respectively.

4.3. Winter modes

The results of the simulations performed with the different winter modes (configurations) proposed in Section 2.2 are shown in Figure 13a. The graph illustrates the electric power consumption of compressors against the ambient temperature, comparing the CB configuration with configurations WM_{I}, WM_{II} and WM_{III} (Figure 2). WM_{I} is only possible at a temperature of 5 °C. At this ambient temperature, the discharge pressure, which is indirectly controlled by means of the expansion devices of the MT evaporators, is already at the minimum discharge pressure allowed by the compressors (43 bar, approximately, with a suction pressure of 28 bar). The power consumption of compressors with WM_{I} at 5 °C is 16.3% lower than with the CB system, but due to this indirect control of the discharge pressure, the conditions at the outlet of the gas coolers are too close to saturated liquid and this can be a challenge if there is some vapour reaching the liquid receiver. The second winter mode proposed, WM_{II}, can be useful with ambient temperatures lower than or equal to 0 °C. At 0 °C, both the CB and WM_{II} perform exactly in the same way, since the CB maintains the pressure at the receiver at a value that is lower than the setpoint (40 bar in this case) but greater than the minimum pressure required to guarantee the operation of the expansion devices at the MT evaporators. WM_{III} is convenient at ambient temperatures lower than 0 °C. The electric power consumption of the compressors at -5 °C is almost identical between WM_{II} and WM_{III}, but WM_{III} outperforms WM_{II} at -10 °C, with approximately 400 W lower power consumption (≈ 4.4% difference). Therefore, this last approach appears to be the most suitable solution for the operation under winter conditions (temperatures lower than 0 °C) in terms of both power consumption and the lack of need to limit the heat exchange at the gas coolers in order to increase the temperature approach at their outlet. In addition, the refrigerant at the outlet of the MT evaporators with WM_{III} was very close to saturated vapour for the range of conditions simulated and steady state.

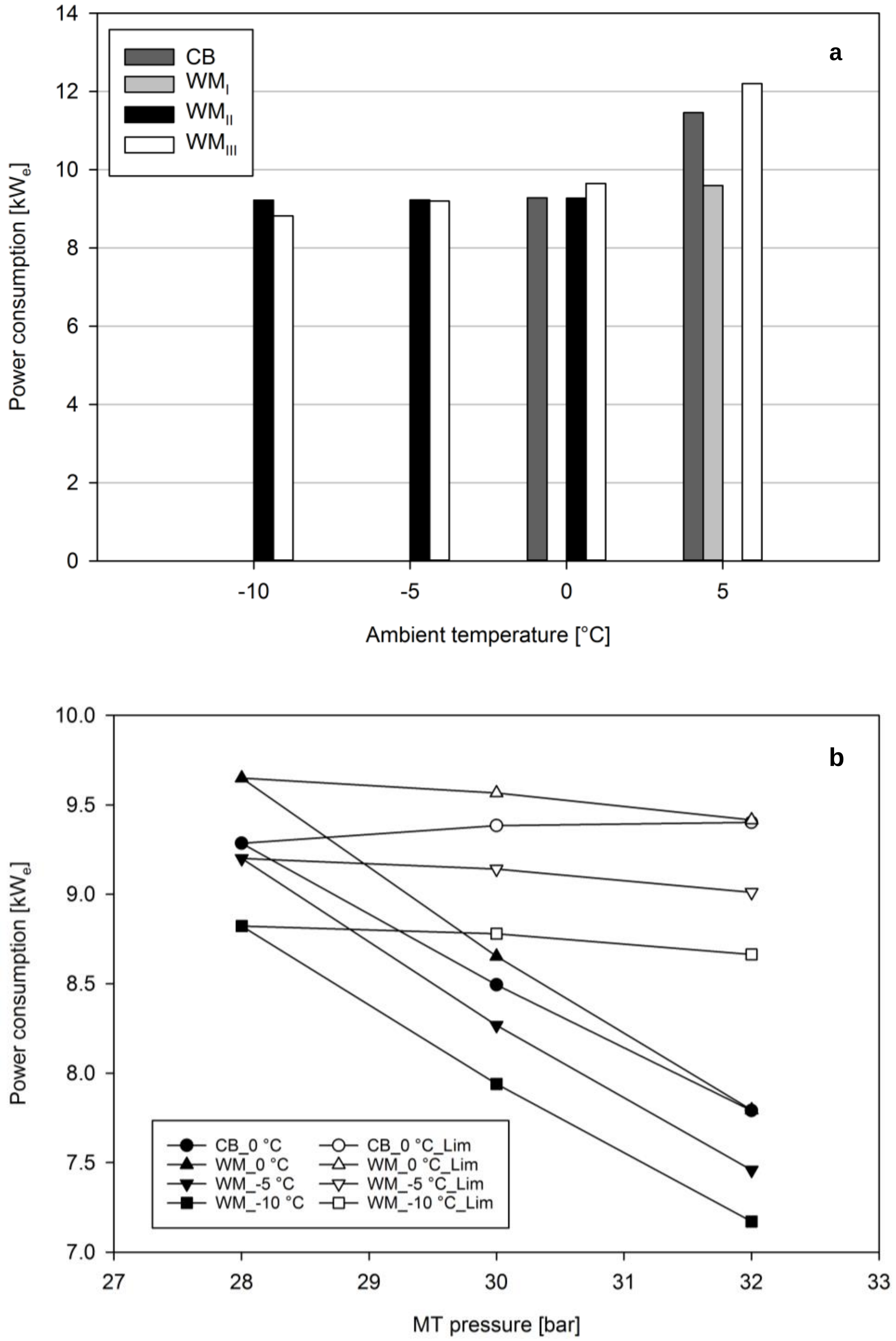


Figure 13. Winter mode figures. Electric power consumption of the compressors against the ambient temperature for conventional booster configuration (CB) and other winter modes (WM_I, WM_{II} and WM_{III}) (a), at an MT evaporation pressure of 28 bar. WM_{III} was seen to be a convenient solution for low temperatures without regulation of the gas cooler capacity and it was compared with conventional booster, in terms of power consumption, at ambient temperatures of 0 °C, -5 °C and -10 °C and with MT evaporation pressures of 28, 30 and 32 bar (b). Results were obtained both considering the limitation of discharge pressure due to the operational envelop of compressors (Lim in the legend) and not considering it.

Figure 13b illustrates the effect of increasing the pressure of MT evaporators at low ambient temperatures with WM_{III} (0 °C, -5 °C and -10 °C) and compares it at 0 °C with a conventional booster (CB). Independently of this pressure, the power consumption with conventional booster at 0 °C is slightly less than with the winter mode developed. At 28 bar (42.8 bar discharge pressure), they use 9250 W and 9650 W, respectively, and at 32 bar (42.8 bar discharge pressure) these values are approximately

7790 W in both cases. The conventional booster configuration is not technically possible at lower temperatures, since the pressure difference between the liquid receiver and the MT evaporators is too low to operate the expansion valves that control them.

With the winter mode configuration, the power consumption decreases as the ambient temperature falls, even when the discharge pressure is not adjusted to the level that should correspond to the temperature at the outlet of the gas cooler, i.e. calculated taking into account the consideration of 5 K subcooling degree and 3 K temperature approach at the outlet of the gas cooler. The reason for this is that the manufacturer of the compressors recommends minimum values for the discharge pressure which are a function of the suction conditions [25]. The lowest suction pressure in this case is 28 bar, which corresponds to the minimum MT evaporation pressure, has an associated value of discharge pressure of approximately 43 bar. At this level of MT evaporation pressure, the power consumption goes from 9650 W to 9200 W, when the ambient temperature drops to -5 °C, and reaches 8820 W at -10 °C.

The effect of increasing the MT evaporation pressure on the power consumption is also beneficial in case the required increase in the discharge pressure due to the change in suction pressure is not taken into account. With conventional booster configuration and ambient temperature of 0 °C, the power consumption is 9280 W, 8490 W and 7790 W with 28 bar, 30 bar and 32 bar, respectively. However, it might not be possible to increase the MT evaporator pressure, since the pressure difference at which the MT expansion valve would operate would be too low (at 0 °C ambient temperature, the liquid receiver pressure is 36 bar). With winter mode, the increase in MT evaporation pressure at 0 °C ambient temperature has an effect on power consumption of 1000 W, when passing from 28 bar to 30 bar, and of an additional 860 W when going from 30 bar to 32 bar. These differences are similar at -5 °C and -10 °C ambient temperatures.

If the discharge pressure of the compressors is adjusted to the value that corresponds to each suction pressure due to the recommendation of the manufacturer, namely to 45.6 and 48 bar when the MT evaporation pressure is 30 bar and 32 bar, respectively, the effect on power consumption differs. There are opposed consequences between the benefit obtained from increasing the pressure and the increase in power consumption due to a greater discharge pressure. On the one hand, with the winter mode configuration, the power consumption remains almost constant as the MT pressure increases, independently of the ambient temperature. On the other hand, with the conventional booster configuration at 0 °C, the power consumption increases. Therefore, such a solution is not convenient at such low temperatures, even with winter mode, unless the increase in MT evaporation pressure and temperature notably reduces the formation of frost on the evaporators and the number of defrost cycles.

5. Conclusions

The results of the simulations conducted allow the definition of how standardised refrigeration racks should be designed, taking into account the maximum loads for Norwegian supermarkets and as a function of the existing system configuration.

CB configuration does not fulfil the maximum loads stated with the compressors considered for the simulation, but the implementation of parallel compressors (PC configuration) allows these demands to be met and the power consumption to be reduced, up to 19%. The implementation of MT and AC multiejector blocks involves that the power consumption is further reduced with respect to the solution without ejectors, particularly at high ambient temperature conditions.

Simulations have demonstrated that implementing the novel concept of pivoting suction appears as an attractive solution in order to have flexible standardised units that can cover a large range of MT and AC loads, even though the effect on reducing the power consumption of the system is not so important. On the other hand, the use of the proposed pivoting system at the discharge of LT compressors in standardised units allows the power consumption to be reduced when parallel compressors are on. This also increases the number of hours the parallel compressors are in operation.

The study of operating conditions (Stage 2) was performed only in those configurations with the best performance, based on the results obtained from the study of configurations (Stage 1), i.e. S_1, with two multiejector blocks (one for MT pressure and one for AC pressure) and AC production downstream of the liquid receiver, and S_2, just with the MT multiejector block and AC production upstream of the liquid receiver. Both systems are identical when there is no AC cooling load.

At 30 °C ambient temperature, the system operates in transcritical conditions, and it is essential to determine the discharge pressure that allows operation with the lowest power consumption possible. The optimal discharge pressure is not only a function of the ambient temperature but also of the MT evaporation pressure and AC production system (AC evaporation pressure and position of the evaporator in the system). The optimal discharge was as low as 79.5 bar and as high as 82.3 bar, depending on the parameters used.

Another conclusion from these simulations is that food retailers should be encouraged to use integrated air conditioning systems designed for the highest temperature possible, in order to increase the pressure at the AC evaporator as much as possible and reduce the overall power consumption. The idea of setting floating liquid receiver pressure for the controllers, and therefore floating AC evaporation pressure, should be considered in the future in order to adapt its value to the AC load and reduce as far as possible power consumption. The two systems for the production of AC, S_1 and S_2, were compared. It was observed that the AC multiejector block (S_1) reduces the overall power consumption if the AC system was designed in a way that allows a secondary fluid with a relatively high temperature and, consequently, high AC evaporation pressure. When a relatively low AC evaporation pressure is required, S_2 appears to be the most convenient solution, caused by the improved performance of the MT multiejector block under these conditions.

Under transcritical and subcritical conditions, the power consumption of IESPC refrigeration systems can be further reduced if the cabinets are designed to operate at flooded conditions, with increased evaporation pressure and temperature, guaranteeing the safety of the goods stored. In addition, this might beneficial from the point of view of reducing the formation of frost on the evaporators and the number of defrost cycles. However, this involves the implementation of a liquid separator after the evaporators and a system to control the liquid level of this tank, for example liquid ejectors. Generally, the power consumption also decreases as the liquid receiver pressure rises, but it might be interesting to limit this increase in the liquid receiver pressure, in order to maximise the time the system operates with parallel compressors.

A final feature to be included in standard units for cold-climate regions is a winter mode to avoid the malfunction of R744 CB systems at low ambient temperatures. The most attractive solution, from a point of view of safe operation and power consumption, is WM_{III} (Figure 2 right), which bypasses and isolates the liquid receiver and connects the outlet of the gas coolers to the inlet of the expansion devices of the different evaporators. Such a system can operate with no need to limit the heat exchange at the gas coolers to increase the temperature approach at their outlet at ambient temperatures at least as low as -10 °C. Further improvements could be achieved with winter mode, if compressor manufacturers could enlarge the operational envelope of their machines, which would allow the setting of the discharge pressure that corresponds to each ambient temperature and the increase of the MT evaporation temperature.

Future works on this topic will be focused on two main lines. A first line is the conduction of experimental campaigns in order to validate the results obtained from the simulations. The second concerns the development of the model by including performance maps that characterise in a better way the AC (high entrainment ratio) multiejector block. In addition, transient simulations will be performed with the model and, for that, the implementation and correct definition of liquid ejectors will be crucial.

6. Acknowledgements

The authors gratefully acknowledge the financial support of the Research Council of Norway through project No. 244009/E20.

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