R744 REFRIGERATING SYSTEMS
MULTI EJECTORS AND LIQUID OVERFEED

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EXPERIENCE WITH R744 REFRIGERATING SYSTEMS AND IMPLEMENTED MULTI EJECTORS AND LIQUID OVERFEED

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Abstract

Two CO_2 refrigerating systems with parallel compression and ejectors have been designed, installed and commissioned in 2014 in a Swiss supermarket. Additional components and measuring instruments are installed, so that different operation modes and control strategies can be investigated. Recorded data from the first year of operation are analyzed.

Since the end of 2014, the systems are running in the ejector operation mode. The ejectors recover expansion work, which normally is lost. The recovered work is applied to return liquid and vapour from the MT suction accumulator downstream of the MT evaporators back to the medium pressure receiver. Due to the liquid ejectors and the MT suction accumulators, the MT suction pressures of systems A and B were raised. Further, the systems are equipped with an LT suction accumulator and two internal heat exchangers allowing to increase the LT suction pressure of system A and B. Switching from booster operation mode to ejector operation mode, the energy consumption of the systems has been reduced by 15 to 20%. Depending on the application, climatic region and the demand of the heat recovery, the annual energy efficiency improvement of ejector supported CO_2 refrigerating systems compared to ordinary CO_2 systems with parallel compression is between 15 to 25%.

Keywords: Transcritical CO_2 booster, parallel compression, ejector, field test, heat recovery

1. Introduction

Transcritical CO_2-Booster systems are becoming the preferred refrigerating systems for more and more European supermarket chains. Whether the systems are considered for efficient heat recovery at low ambient temperatures or for operation at high ambient temperatures, the systems are characterized by high throttling losses in the high-pressure valve due to high system pressures. In the past years, several different system modifications have been tested to reduce these throttling losses in CO_2 refrigerating systems. Since the first systems with parallel compression (Girotto, Minetto 2008) have been built in 2009, the parallel compression has become well known and certain supermarket chains have defined it as a standard. Also other system modifications such as mechanical subcooling or expansion-compression-units have been implemented and tested. Another possibility to reduce the throttling losses is the implementation of ejectors. In 2013, a transcritical CO_2 refrigerating system with three ejectors has been commissioned (Schoenenberger, 2013).

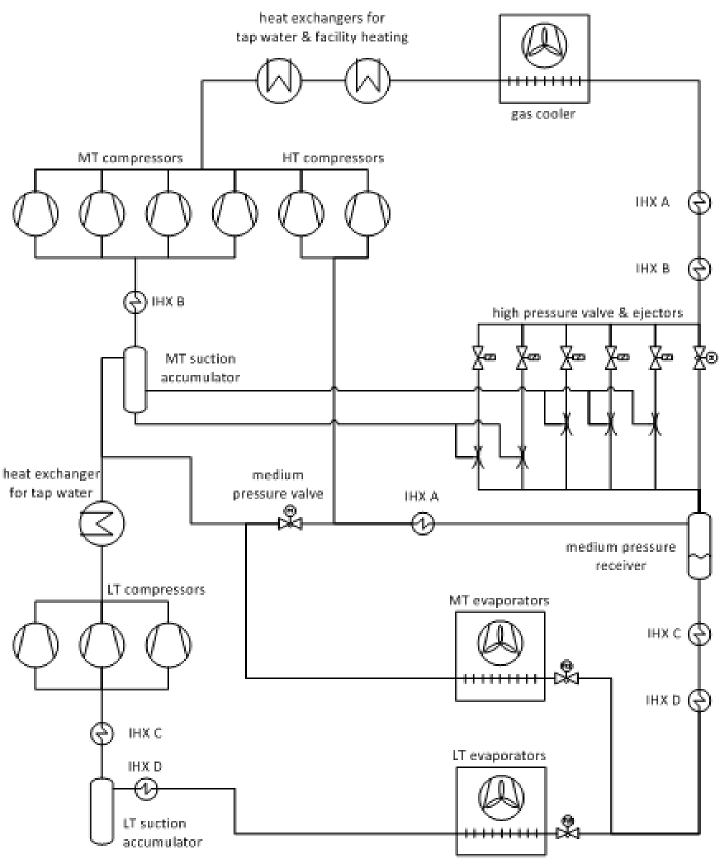
This system shows a simple and safe method to drastically reduce the energy consumption of a transcritical CO_2 refrigerating system. After one year of experience, two even more enhanced systems with each having five ejectors and flooded evaporators on the MT and LT side have been built. These systems are completely identical and are operating at the same job site. This allows a direct comparison of different operation modes, their behaviour and related energy consumption.

2. Job Site

The job site is located in central Switzerland, 450 meter above sea level, with an annual average temperature of +9.6°C. The supermarket has a total sales area of approximately 5250 square meters with a total length of 130 meter MT and 38-meter LT cabinets. The MT cabinets do not have glass doors. Further, there are five walk-in coolers with a total volume of 493 cubic meters and four walk-in freezers with a total volume of 128 cubic meters. The resulting cooling demand is split up in two identical systems. In addition, the systems are equipped with heat recovery heat exchangers for tap water and facility heating. Primarily, the heat is used for the supermarket; excess heat is passed on to other parties within the shopping mall.

3. Refrigerating Systems

The two enhanced refrigerating systems of the mentioned job site each consists of four MT compressors and two HT compressors. The discharged CO_2 of the MT compressors and HT compressors passes the heat exchanger for tap water and facility heat recovery before the rest of the heat is rejected to the ambient by the gas cooler. Downstream of the gas cooler, the CO_2 is expanded in the ejector or high-pressure valve from high pressure to medium pressure. The vapor phase of the medium pressure receiver is superheated by an IHX before it is compressed primarily by the HT compressors. Should the swept volume of the HT compressor not be sufficient, the rest of the vapor is expanded to the suction line of the MT compressors. The liquid phase of the medium pressure receiver is subcooled by two IHX in the suction line of the LT compressors before it is fed to the MT and LT evaporators. The CO_2 from the LT evaporators is superheated in the IHX before it is compressed by three LT compressors.



The suction line of two liquid ejectors is connected to the liquid phase of the MT suction accumulator. This allows to shift liquid from the MT suction accumulator to the receiver, driven by expansion work from high pressure to medium pressure. The suction line of the three vapor ejectors is connected to the vapor phase of the MT suction accumulator. The vapor ejectors are precompressing vapor from the suction pressure of the MT compressors up to the suction pressure of the HT compressors, also driven by expansion work from high pressure to medium pressure. Thereby, vapor mass flow is shifted from the MT compressors to the HT compressors. The HT compressors are running in a more efficient operation point than the MT compressors and need less energy to compress the same amount of CO_2.

Figure 1: P-I-Diagram of a transcritical CO_2 refrigerating system with parallel compression and multi ejectors

Liquid in the LT suction line is evaporated by the IHX D by subcooling the liquid line on medium pressure. Should the amount of liquid, returning from the LT evaporators, be more than the IHX D allows to evaporate, the liquid is trapped in the LT suction accumulator. The IHX C, in any case, superheats the CO_2 before it reaches the LT compressors. The possibility to remove liquid in the suction line of MT and LT, before it reaches the MT and LT compressors, allows reducing the superheat set point at the MT and LT evaporators or even run the systems with an overfeed of CO_2 in the evaporators. This leads to a higher heat transfer rate in the evaporators and finally the evaporation temperature may be raised.

3.1 System Design

The refrigerating systems are designed so that they can operate in conventional booster operation mode with standard superheat or in ejector operation mode with little or no superheat. In case of unexpected problems, they can be switched to the one or the other operation mode. Further, this ideally allows comparing two different operation modes or different operation set points. Therefore, additional components and measuring instruments are installed, so that different operation modes can be investigated. This equals an additional challenge designing the system, further it leads to additional costs. Due to the experiance gained, future systems can be simplified. Each of the refrigerating systems is designed to provide 95 kW MT cooling capacity and 29 kW LT cooling capacity. The following design points are considered:

	Booster Operation Mode	Ejector Operation Mode
HT compressors	±0°C / 35 bar_a	+3°C / 38 bar_a
MT compressors	-8°C / 28 bar_a	-2°C / 33 bar_a
LT compressors	-33°C / 13 bar_a	-26°C / 16 bar_a
Min. / max. gas cooler outlet temperature	+5°C / +36°C	+8°C / +36°C
Min. / max high pressure	45 bar_a / 92 bar_a	50 bar_a / 92 bar_a
Min. superheat set point at evaporators	8 K	0 K – 4 K

The evaporators of the cabinets, served counters, walk-in coolers and –freezers as well as ice machines are calculated and designed for conventional superheat operation mode with a standard superheat of 8 K. Due to that, the evaporators are the same size as in conventional systems. Further, the cabinets do not have glass doors; although glass doors would additional reduce energy consumption. In a standard booster system with parallel compression, the high pressure control is realized by the high pressure control valve. In the ejector operation mode, five ejectors are operated in parallel to the high-pressure control valve. Thereby it is possible to test different high-pressure control strategies. One strategy is to control the high pressure by the high-pressure control valve combined with the ejectors. This strategy has the advantage that the set-off between the high-pressure set point and real high pressure is in the same order of magnitude as in conventional operation modes. Another strategy is to control the high pressure completely by the ejectors. This strategy has the advantage that the whole mass flow is expanded by ejectors, which leads to a maximum benefit of the expansion energy, expecting that the set-off between the high pressure set point and real high pressure is higher than in conventional operation mode.
There are many influencing factors such as motive inlet conditions, suction inlet conditions as well as covering the total capacity of the system, to design ejectors and how to match them up to a multi ejector system (Banasiak et al. 2011, Hafner et al. 2012). It is obvious that there is not *the one* geometry for the whole application range of a system. Different

approaches can be considered to select and control the ejectors. Ejectors can be switched on and off and be prioritized for example according the liquid level in the suction accumulator, according the high-pressure control or the ejector design pressures. The ejectors of the mentioned systems have different nozzle diameters and are designed for different high pressures as well as to suck liquid or precompress vapour. Depending on the high-pressure set point, the real set-off, the high pressure design point of each ejector as well as the liquid level in the suction accumulator, the best suitable ejectors are activated within the system.

3.2 System commissioning

The systems were first commissioned and adjusted to the conventional booster operation mode with parallel compression and standard 8 K superheat in August 2014. In November 2014, system A was switched to the ejector operation mode with reduced superheat at the evaporators. After one month of experience with the ejector operation mode of system A, in December 2014 system B was switched to ejector operation mode as well. Since then, both systems are running continuously in ejector operation mode for more than one year.

4. Operating Experience

4.1 Energy Consumption during Commissioning

The diagram 1 shows the daily energy consumption, converted to annual energy consumption per meter length of cabinet during the period of switching to ejector operation mode. The shown energy consumption represents the total energy consumption of each system. The time axis is divided into three major periods: during the first period, both systems A and B are running in the conventional booster operation mode. In the middle period, system B is still running in the booster operation mode whereas system A is switched to ejector operation mode. During the last period, both systems are running in ejector operation mode. During the first period, before switching system A to ejector operation mode, the energy consumption of the systems differ by 5%. This is probably due to a slightly higher cooling demand of system B. Other major influences, such as ambient temperature, indoor temperature or humidity, demand on heat recovery, market opening hours, staff or consumer behaviour, are disregarded due to the fact that the systems are operated in the same supermarket e.g. sales area.
In the middle period, operating system A in ejector operation mode, the average energy consumption of system A drops from 2160 kWh/m-a to 1612 kWh/m-a. This equals a reduction of 25% energy consumption during ejector operation mode compared to conventional booster operation mode with parallel compression. During the same period, the average energy consumption of system B has dropped from 2284 kWh/m-a to 2170 kWh/m-a which equals a drop of 5%. Accordingly, the reduction of energy consumption during the period under review can be assumed to be 20%. In the third period, beginning from the date of switching system B to ejector operation mode, its energy consumption has dropped by 24% from 2170 kWh/m-a to 1657kWh/m-a. While the energy consumption of system A, during the same phase has dropped by 9% from 1612 kWh/m-a to 1465 kWh/m-a. Switching system B from conventional booster operation mode with parallel compression to ejector operation mode saves roughly another 15%. The difference of 5%, between switching system A and B to ejector operation model is due to different evaporation set points.
System A is running with a set point of -2°C MT evaporation temperature, whereas system B was running with a lower set point of -4°C, because the served counter with meat had difficulties to maintain the temperature range of -1°C to +1°C.
In the meantime, the evaporation set point of system B has been raised to -3°C. This was possible by changing the nozzle of the expansion valve of the served counter.

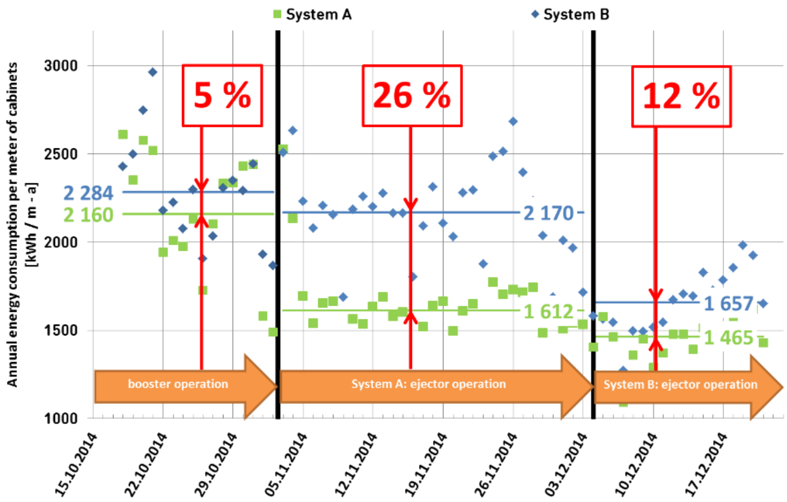


Diagram 1: Energy consumption of system A and B extrapolated and shown as annual energy consumption per meter of cabinets

4.2 Operation during Summer

Diagram 2 shows different operation parameters of system B during the store opening hours from 8 a.m. in the morning to 8 p.m. in the evening of the 2nd July 2015. Temperature parameters refer to the left y-axis and pressure parameters as well as the liquid level refer to the right y-axis. During this period, the outside temperature was between +23.2°C in the morning and +32.8°C in the afternoon.

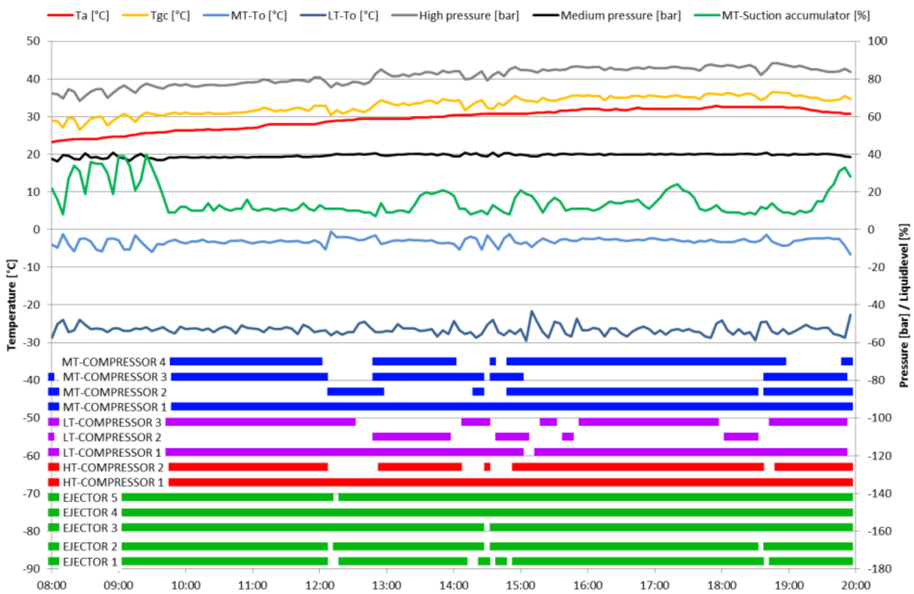


Diagram 2: Ejector operation parameters of system B of the 2nd July 2015

The gas cooler outlet temperature shifted parallel to the ambient temperature between +26.4°C and +36.5°C. The high pressure during this period was, as a function of the gas cooler outlet temperature, between 68.4 bar_a and 88.3 bar_a. The medium pressure was between 36.3 bar_a and 40.7 bar_a at an average pressure of 39.4 bar_a. The light blue line shows the MT evaporation temperature which was on average at -3.1°C. The LT evaporation temperature during this period was on average at -26.5°C, shown with the dark blue line. The green line represents the liquid level in the MT suction accumulator; the level varies between 7 and 40%. At the lower end of the diagram, the operation state of different compressors and ejectors are shown. It can be seen, that at least one ejector is always in operation. They are prioritized depending on the high pressure and the liquid level in the MT suction accumulator so that the high pressure can be maintained. The control quality of the high pressure is comparable with a conventional system. The on-off cycles of the compressors are low compared to conventional systems. Once in a while, the liquid level in the MT suction accumulator increases. It is assumed, that this relates to high cooling demand. The evaporation temperature set point of the MT compressors as well as the set point of the medium pressure are shifted according to the liquid level in the MT suction accumulator. It can be seen, that when the liquid level increases, the evaporation temperature of the MT compressors as well as the middle pressure drops. An alternative control strategy is to shift the superheat instead of the evaporation temperature, which leads to a further stabilization of the operation.

5. Conclusions

Two CO_2 ejector systems with parallel compression and each having 5 ejectors have been designed, installed and commissioned in 2014 in a Swiss supermarket. Each refrigerating system is designed to provide 95 kW MT cooling capacity and 29 kW LT cooling capacity. The installed refrigerating systems are designed so that they can operate as conventional booster systems with standard superheat or as ejector systems with little or no superheat. Additional components and measuring instruments are installed, so that different operation modes and controlling strategies can be investigated. The ejectors recover partly the expansion work, which is normally lost in the expansion from high to medium pressure. The recovered work is applied to return liquid and vapour from the MT suction accumulator downstream of the MT evaporators back to the medium pressure receiver which is located upstream of all evaporators. Further, the systems are equipped with an LT suction accumulator and two IHX. This allows to increase the LT evaporation temperature. The systems are operated all year round in the ejector operation mode. This leads to high efficiency at high ambient temperatures in summer and also during heat recovery operation mode during winter.

Due to the liquid ejector and an MT suction accumulator, the MT suction pressure of system A was raised from -8°C up to -2°C and of system B from -8°C up to -3°C. Due to the LT suction accumulator and the two heat exchangers for liquid subcooling, the LT suction pressure of system A and B was raised from -33°C up to -26°C. The superheat set point of the evaporators varies between from 0 K to 4 K, pending on the evaporator size. With these set points, all the cabinets and walk-in coolers and -freezers are able to maintain the required temperature range despite the increased evaporation temperature. It is assumed, that the increased evaporation temperature leads to less defrosting cycles of the MT evaporators.

It is shown, that the energy consumption of system A and B during commissioning was reduced by 15 to 20% by switching to ejector operation mode. Further it is shown, that the systems have a smooth and steady behaviour during ejector operation mode. The on-off cycles of the compressors are low compared to conventional systems. The evaporation temperature of the MT and LT level are showing a smooth behaviour. The evaporation set point of the MT and LT compressors is shifted by the liquid level in the MT and LT suction accumulator. This function can be replaced in future systems by shifting the superheat of the evaporators. Further, potential to optimize future systems by designing them slightly different and simplifying the system has been recognised. Calculating the system with proven evaporation temperatures and ambient temperatures, it allows forecasting an increased rack efficiency of 20%. Depending on the application, climatic region and the demand of the heat recovery, the annual energy efficiency improvement potential of ejector supported refrigeration systems compared to ordinary R744 systems with parallel compression is between 15 to 25%.

Nomenclature

HT	high temperature (parallel compressor)	LT	low temperature
MT	medium temperature	IHX	internal heat exchanger

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