COMMERCIAL REFRIGERATION SYSTEMS WITH CO_2 AS REFRIGERANT

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ABSTRACT

With the focus on supermarket applications, this paper provides an overview of various system solutions with CO_2 as refrigerant, such as cascade-, booster- and parallel compression systems, and discusses their advantages and disadvantages, along with a comparison of system COPs. The application of CO_2 requires appropriate measures with respect to the system design, -control, etc. to ensure reliable and efficient operation. These measures are discussed from the application engineering point of view, with the emphasis on the challenging issues regarding capacity regulation, operating temperatures and oil distribution of CO_2 systems.

1. INTRODUCTION

After the rediscovery of CO_2 as a refrigerant by Gustav Lorentzen in 1988, this traditional coolant has gained relevance in the world of refrigeration. A remarkable number of studies, researches and developments have been carried out since then. Together with newly developed components and system solutions, CO_2 has been again used for refrigeration and will be in the future. Relevant issues for the application of CO_2 are discussed with this paper.

2. SYSTEM SOLUTIONS WITH CO_2 AS REFRIGERANT

Apart from favorable hybrid systems, e.g. R134a or R290 in a chiller stage and combined with a cascade stage including pump circulated CO_2 for medium temperature (MT) and direct expansion for low temperature (LT), different system solutions with only CO_2 as refrigerant can be applied for commercial applications. Five CO_2 system possibilities are described in the following.

2.1 Flash gas bypass (FGB)

The FGB concept is a system with single stage compression and two stage expansion. Typically this type of system is applied with the MT stage of a cascade system. After the rejection of heat energy in the gas cooler, the refrigerant flows through the valve for high pressure control and expands into a receiver. Inside the vessel liquid CO_2 and flash gas, generated by the expansion process, become separated. Liquid flows on to the expansion devices of the evaporators and through the evaporators after the expansion to evaporating pressure. The amount of flash gas bypasses the evaporators; an additional expansion device downstream the receiver expands the flash gas to evaporating pressure. Upstream the compressors, both mass flows create the total mass flow of the system again, which is then taken in by the compressors; on basis of equal suction and discharge conditions for a single stage and an FGB system, the resulting total mass flows are equal as well. The FGB solution shows various advantages. The pressure between the valve for HP control and the expansion devices of the evaporators can be reduced to values that allow the application of 40 bar components, such as copper fittings, pipe work, valves, etc. Besides this the amount of flash gas inside the MT evaporators is reduced, which offers the potential for an increase of the heat transfer coefficient and a reduced pressure drop on the CO_2 side [1]. Ebel and Hrnjak showed that especially the decreased pressure drop of an FGB system has a high influence on the improvement of cooling capacities and COPs for microchannel evaporators. Larger cross sections and different distribution in evaporators for commercial applications versus investigated evaporator design might decrease the positive effect. Due to the separation of liquid and flash gas, the refrigerant mass flow through the evaporators is reduced in comparison to a single expansion system. With respect to the evaporator capacity this influence is compensated to a certain extent by the advantages of a higher available evaporating enthalpy and a more efficient heat transfer on the refrigerant side of the evaporators. The system shows a higher expenditure than a standard single expansion concept, due to the additional FGB valve and its control.

2.2 Parallel Compression

A system with parallel compression can be applied either with two individual compressors or a single compressor. This paper focuses on the solution with a single compressor, hereafter called ECO compressor, which is designed to compress two different refrigerant mass flows in parallel.

Between gas cooler, valve for HP control and receiver, the solution is equal to the system described before. Again two different mass flows are generated inside the receiver, but the amount of flash gas streams directly to the suction port of the ECO compressors and is processed by the ECO cylinders from medium to high pressure level. Liquid CO_2 from the receiver flows to the expansion devices and after expansion and evaporation, it enters the main suction port of the ECO compressors. After compression to the common high pressure level, flash gas and evaporator mass flows create the total mass flow inside the common discharge channel of the ECO compressors. Considering equal suction and discharge conditions in systems with FGB and parallel compression, the mass flow of the cylinders under evaporating pressure is reduced by the ratio between the displacements on evaporating and ECO pressures; the mass flow under evaporating pressure corresponds to the total mass flow of an FGB system multiplied by the ratio of displacements. Nevertheless, the resulting refrigerant mass flow through the evaporators is intensified in comparison to the FGB system. The ECO stage has higher suction densities and consequently higher mass flows for the ECO cylinders. Thus the total mass flow of the compressors is increased as well. This fact results in a higher absolute liquid fraction inside the receiver and in the end on the evaporator side of the system. FGB reduces the mass flow through the evaporators in comparison to a single stage system, whereas parallel compression increases the mass flow. In addition to this, the specific work for the compression process of flash gas is reduced by the lower pressure ratio. Due to the separation of liquid and flash gas inside a receiver, the system shows the benefits of higher available enthalpy differences and reduced amounts of flash gas on the evaporator side. This applies to an FGB system as well. The control and the design of the system are more demanding than it is for an FGB concept.

In dependence on the displacement ratio of the ECO compressor between evaporating- and ECO pressure levels, the pressure inside the receiver can be varied. A ratio of 3:1 is applied with the ECO compressors of Bitzer; three cylinders work on evaporating- and one cylinder operates on ECO pressure level. Measurements confirmed the first simulations. Considering the compressor type 4HTC-20K-ECO operating at -10 °C evaporating-, 35 °C gas cooler outlet temperature, 90 bar discharge pressure and 10 K suction gas superheat at the main and ECO suction ports, the ECO pressure corresponds to 40 bar. In comparison to an FGB system with 40 bar, the efficiency improvement corresponds to 10 per cent. The increase is lower than in theory; reduced mass flows for the motor cooling affect the motor temperatures and the volumetric efficiencies. Another influence shows the ECO valve plate arrangement, but the decreased mass flow is within the tolerances of the measurements. The potential for optimisations was not consulted yet.

2.3 Cascade system for MT and LT

Cascade systems are characterised by independent refrigerant and oil circuits and an exchange of heat energy due to the cascade heat exchanger(s). The discussed system contains single stage compression for the MT and LT stage. Typically the MT stage shows a two stage expansion process with receiver and FGB valve for pressure reduction, whereas the LT stage just contains an ordinary single stage expansion. Liquid CO_2 out of the receiver supplies the cascade heat exchanger(s) on the evaporator side. The system was introduced to commercial applications for the first time in November 2004 by Haaf et al. [2]. The system offers the benefits of an FGB system. Two independent refrigerant and consequently oil circuits show an advantage for the oil distribution of such a system, whereas the cascade heat exchanger(s) need to be applied in the way that material stress caused by high temperature gradients and pressure fluctuations is reduced to a minimum. This might include the application of a discharge gas de-superheater.

2.4 Booster system for MT and LT

Booster systems have a common refrigerant circuit and are characterised by direct exchange of refrigerant, oil and heat energy. The described option is an externally compounded two stage compression and expansion concept, with receiver and FGB; it is shown in Figure 1. The medium temperature stage operates similar to the FGB system described under 2.1, despite of the facts, that liquid CO_2 from the receiver supplies the LT evaporators and the LT compressors discharge directly into to the suction header of the MT stage. There, mass flow from the MT evaporators, flash gas and the LT stage mass flow generate the total mass flux of the system. The booster system does not contain a cascade heat exchanger which provides a reduced intermediate pressure for the LT compressors and therefore a lower specific work for the process of compression. Besides the reduced number of involved assemblies, these are the benefits in comparison to the cascade system. On the other hand the LT stage of the booster system has a lower available evaporator enthalpy caused by the fact that the CO_2 is not condensed on LT condensing pressure level before the expansion.

Very important for booster systems is the influence of different load conditions on the resulting operating conditions. During the design stage of such systems, the extreme load variations should be calculated and evaluated. The two worst case scenarios are: Low load for MT evaporators and high load for the LT evaporators and vice versa. The first scenario leads to high suction gas temperatures for the MT compressors with influence on the motor cooling and discharge temperatures whereas the second scenario results in low suction gas temperatures with low oil sump temperatures and possible wet operation due to the influence of the FGB. In dependence on the results and the feasibility of such extreme conditions, it must be decided whether additional measures like heat exchangers or hot gas bypass should be provided.

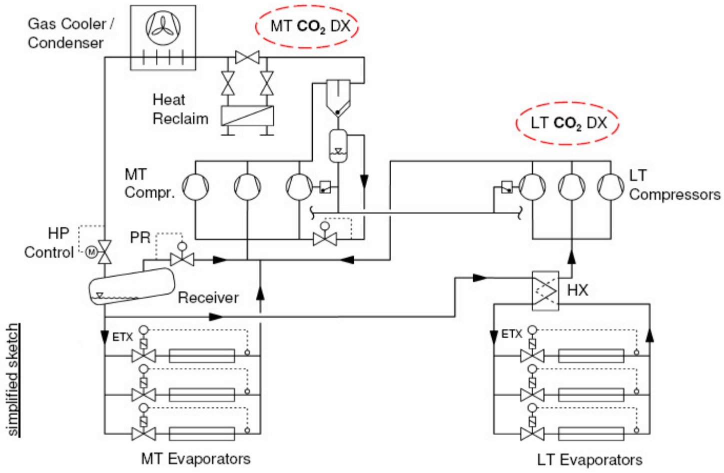

Figure 1. Simplified sketch of a commercial booster system for MT and LT load

2.5 Enhanced booster system for MT and LT

An enhanced option for a booster system is further described as a solution, which contains externally compounded two stage compression and three stage expansion, with two receivers and parallel compression. A simplified sketch is described in Figure 2. The difference to the concept described under 2.4 is the application of parallel compression and an additional receiver. Downstream the first vessel, hereafter called ECO receiver, the liquid is split into two different mass flows for the MT and LT stage. The medium temperature evaporators are supplied with liquid out of the ECO receiver. After expansion to MT pressure and evaporation, the MT mass flow streams back to the MT compressors. For the LT stage a three stage expansion is applied. After expansion into the ECO receiver and separation of liquid and flash gas, a second stage expansion takes place, where the low temperature mass flow expands from ECO pressure to MT pressure. Inside the additional receiver, called LT receiver, liquid and flash gas becomes separated again. In an ideal system the pressure inside the low LT vessel corresponds to the MT evaporating and LT discharge pressure. The flash gas is led back to the MT compressors, whereas the liquid flows further to the upstream LT evaporators. After the third stage of expansion, evaporation in the evaporators and compression in the LT compressors, the discharged LT mass flow streams back to the second compressor stage. Before the second stage of compression, the mass flows from the MT evaporators, the LT flash gas and LT evaporators merge again. After the compression from MT to HP level and in parallel from ECO pressure to HP level, the total mass flow of the system streams through the gas cooler.

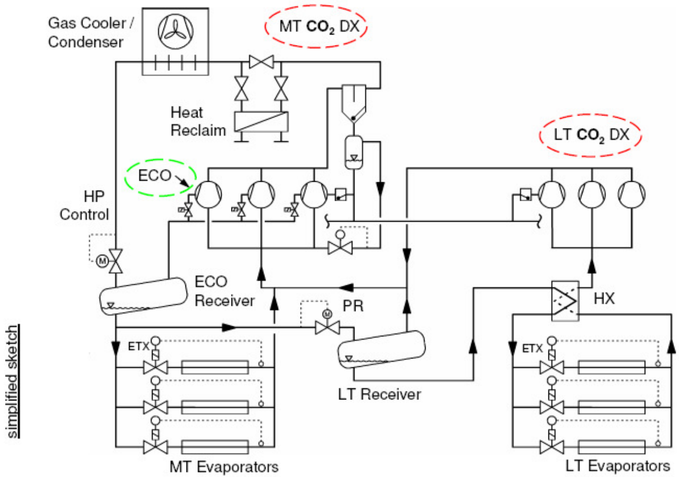

Figure 2. Simplified sketch of an enhanced commercial booster system for MT and LT load

In comparison to the system described under 2.4, the enhanced booster system shows the advantages of the parallel compression process as mentioned under 2.2. In addition to this, the three stage expansion for the LT evaporators plus the additional separation of liquid and flash gas provide the benefits of a reduced amount of flash gas for the LT evaporators plus an increased evaporating enthalpy. This leads to smaller displacement of the LT compression stage. Depending on the requested capacities, this offers smaller diameters for the pipe work and an electrical installation for lower maximum operating currents. In order to keep the pressure inside the receiver independent from the displacement ratio of the ECO compressors, the ECO operation can be combined with an FGB. This measure allows the adjustment of defined pressure levels.

2.6 Comparison of system COPs

Differences of the highlighted commercial refrigeration systems for MT and LT applications with emphasis to the theoretical system COPs are discussed in the following. Compressor efficiencies, the beneficial properties in heat transfer on the evaporator side as described by Pearson A. [3], heat rejection on the gas cooler side, system pressure drops, different load conditions, ambient temperature distribution and air humidity are excluded from this comparison but would need to be applied for a simulation of the seasonal efficiencies.

The systems 2.3 to 2.5 are options for commercial applications with MT and LT load. The basis for the comparison are: Evaporating temperatures of -10 °C and -35 °C for MT and LT, discharge pressure of 90 bar for the MT stage, 35 °C gas cooler outlet temperature, 10 K useful superheat for MT and LT and net evaporator capacities in a ratio of 4:1 between the MT and LT stage. An all CO_2 cascade system as described under 2.3 is considered as benchmark. Furthermore 35 bar flash gas pressure and 3 K temperature difference inside the cascade heat exchanger are assumed.

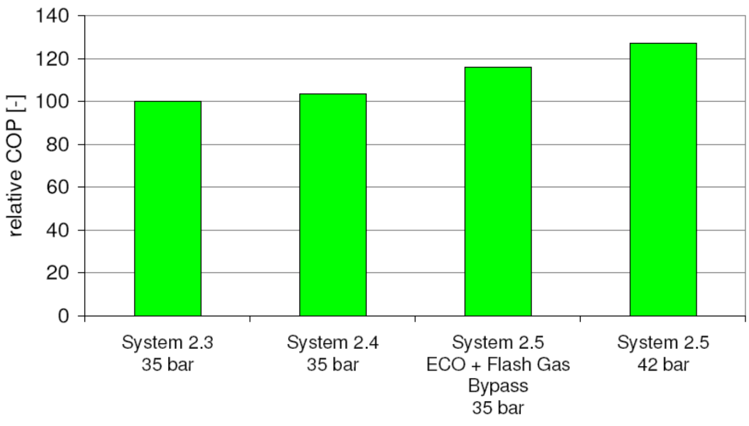

Figure 3. Comparison of system COPs

The comparison with a booster system as mentioned under 2.4, offers an improvement in COP of 4 per cent. A flash gas pressure of 35 bar is considered as before. The required displacement for the MT stage is 3 per cent lower whereas a displacement of 107 per cent is required for the LT stage. A significant improvement of COP can be achieved with an enhanced booster solution as mentioned under 2.5. For the theoretical comparison the system is considered with a combined ECO and FGB option to achieve a pressure inside the ECO receiver of 35 bar under the assumed gas cooler outlet conditions. A superheat of 10 K is applied for the ECO ports. This configuration offers an increase of 16 per cent for the COP. The effect on required displacements is positive as well; the reductions for the MT and LT stages are equal to 13 and 2 per cent. In case the system 2.5 is not applied with an FGB, the ECO pressure inside the receiver corresponds to 42 bar for the given gas cooler outlet conditions and assumptions as described in the beginning of this chapter. The higher ECO pressure provides an additional improvement with respect to the COP of 11 percent and a further reduction of the displacement of the LT stage of 2 per cent. The comparison of COPs is summarized in Figure 3.

3. THE APPLICATION ENGINEERING POINT OF VIEW

The application of CO_2 offers some specifics and pitfalls that require attention. Basic issues like cleanliness and dryness are the same as with standard HFCs but more important with CO_2. Safety issues with regard to maximum operating pressures, trapped liquid and leak detection are even more demanding. Within this paper additional issues are highlighted.

3.1 Capacity regulation

A major issue for the application of CO_2 is the necessity for capacity regulation; part load is the challenging part and not full load. Considering the thermodynamic properties of CO_2, it is obvious that the expansion into the two phase area from high discharge pressures and gas cooler outlet temperatures offers only small liquid fractions to the process. In the opposite of this, the increase of enthalpy difference with reduced discharge pressures and gas cooler outlet / condensing temperatures is significant. The following example highlights the differences between full load and part load operation. The refrigeration capacity of the compressor type 4HTC-20K corresponds to 23.2 kW when operating at -10 °C evaporating, 35 °C gas cooler outlet temperature, 90 bar discharge pressure and 10 K suction gas superheat. An operation at 20 °C condensing temperature with the same suction conditions shows an increase in refrigeration capacity of 50.2 per cent. This growth is shared between an increased enthalpy difference of 28.8 percent and an enhanced volumetric efficiency of 16.6 per cent, with simultaneous effect on the mass flow. At 15 °C condensing temperature the refrigeration capacity is even 68.1 per cent higher. Thus CO_2 systems demand a good adjustment of capacity steps in order to adapt the required capacity to the ambient temperature distribution over the year and the requested load conditions at the evaporators. A good capacity regulation minimizes the pressure fluctuations on the suction side and avoids wet operation of the compressors. Focusing on CO_2 systems for MT and LT applications (e.g. systems 2.3 – 2.5), capacity regulation is also mandatory to reduce the interactions between MT and LT stages. Suitable options to achieve a sufficient capacity regulation are the parallel compounding of compressors in each stage and the use of variable speed drive for one or several compressors. Cylinder unloading is more difficult to apply than with standard refrigerants, because of the unfavourable temperature conditions inside a blocked cylinder and high differential pressures for the components. Another aspect for the capacity regulation is the influence of the gas cooler outlet temperature (and condensing temperature respectively) on the operating characteristic of CO_2 systems. For the selection of a gas cooler it should be considered in general that the temperature differences for condensing conditions are in the range of approximately 10 K to ensure a stable condensing operation with ambient temperatures around 15 °C. This requires larger surfaces for the gas cooler; experience shows that this leads to temperature differences of approximately 2 K between ambient and gas cooler outlet temperatures during full load operation with high ambient temperatures.

3.2 Operating temperatures

Essential for a reliable operation of CO_2 compressors are the operating temperatures. Of special interest are the maximum and minimum discharge temperatures and the minimum oil sump temperature. These temperatures are related to the utilized lubricant. In case of Bitzer compressors for trans-critical applications a

POE with a basic viscosity of 85 cSt based on 40 °C is used. POEs are used for commercial installations because of a high miscibility, lower hygroscopicity than PAG and higher di-electric strength. Of course the application of POE shows the drawback of higher solubility of CO_2 in the lubricant. The maximum discharge temperature for standard applications corresponds to 140 °C measured on the discharge pipe close to the compressor. An ageing process of the pure lubricant and a cracking of the chemical structure starts with temperatures higher than 160 °C under 50 bar pressure and a water content < 100 ppm [4]. Considering that the water content of CO_2 systems could be in the range of 100 ppm and discharge pressures up to 100 bar, the limit of 140 °C should be observed. The minimum discharge temperature is of interest especially in connection with the crankcase pressure and oil sump temperature. Besides the mass flow, which has an influence on the internal cooling effect, the discharge temperature shows the strongest influence on the oil sump temperature. The resulting discharge temperature itself is dependent on electrical, mechanical, internal pressure losses plus heat radiation to the ambient. The Figure 4 shows the relationship between the discharge and oil sump temperatures over the pressure ratio. The measurements were performed for the compressor types 4FTC, 4HTC, 4JTC, 4KTC and 4MTC with a suction gas superheat of 10 K, discharge pressures of 40, 62, 84, 106 bar and evaporating temperatures of -5 to -20 °C in temperature steps of 5 K. The ambient temperature around the compressor corresponded to 32 °C. The results show that the oil sump temperatures are related to the discharge temperatures. Oil sump temperatures are different for each individual compressor type but show similar trends. The minimum discharge temperature for sub-critical MT operation has to be 40 K above the condensing temperature to avoid low oil sump temperatures. Considering crank case pressures between 20 and 30 bar, the amount of CO_2 solved in the lubricant with 20 °C temperature varies between 9 and 13 per cent. These values are not critical in connection with the resulting kinematic viscosities. However, the high amounts of solved refrigerant in the lubricant and pressure fluctuations result in strong degassing effects and oil foaming inside the crank case.

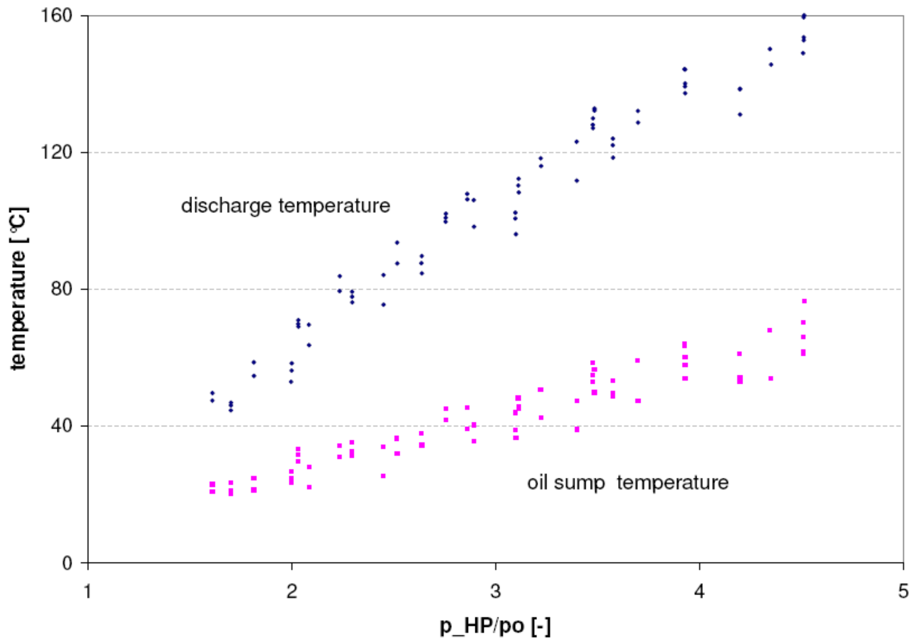

Figure 4. Discharge- and oil sump temperature in dependence on pressure ratio

3.3 Oil distribution

For a reliable oil distribution under all load conditions, the application of a suitable oil management system is essential. Based on the combination of CO_2 and POE, the return of lubricant out of the system is not challenging. High miscibility in liquid CO_2 and high vapor densities on the suction side of the systems lead to an easy return of the lubricant to the compressors, provided that similar velocities like with R404A are applied. Considering a commercial application, only a small number of compressors is in operation during part load conditions, e.g. during night time in winter. Consequently mass flows and velocities inside the evaporators and suction lines are reduced and the return of lubricant becomes more challenging. A sufficient oil supply to the compressors under various operating conditions can be ensured with oil management systems, which could include an oil separator with level sensor, a valve for pressure reduction from e.g. 100

to 40 bar, an oil reservoir with level sensor and adjustable differential pressure valve, oil strainer and oil level regulators mounted on the compressors. The valve for the pressure reduction is controlled by the level sensor inside the oil separator. An additional level sensor inside the low pressure oil reservoir ensures that the oil level regulators on the compressors only open if the oil level inside the reservoir is sufficiently high. This is to avoid a hot gas bypass. An adjustable differential pressure valve between oil reservoir and the common suction header ensures the degassing of CO_2. The differential pressure must be higher than with standard applications for an efficient oil supply to the compressors. For MT and LT booster systems, the oil level regulators are assembled on the compressors for both temperature levels and connected to the common low pressure oil reservoir. Important for booster systems is the fact that the MT compressors have a higher absolute oil carryover rate in comparison with the LT compressors. This is caused by the higher suction vapor densities and mostly higher cooling loads on MT level. This could lead to the situation that the oil level inside the crankcases of the LT compressors rises above the maximum. Thus the effectiveness of the oil separator must be very high and in the range of the oil carry over rate of the LT compressors to ensure a balanced condition. Specially designed LT booster compressors with a controlled oil carryover rate at higher oil levels are operating in field trials. Oil equalization between compressor crankcases as a simplified method for oil management is very demanding in terms of layout. This principle requires a certain level difference between equalizing port and equalizing line as well as special adaptation of cross sections. The design is dependent on the suction and discharge conditions, oil temperatures, amount of CO_2 solved in the lubricant, resulting densities of the oil / CO_2 mixture, internal pressure losses of the compressor and inside the suction header. Therefore, an application requires intensive testing, optimization and standardization.

3.4 Flash gas bypass

For the FGB process it has to be considered, that the expansion process of flash gas back to evaporating pressure ends inside the two-phase area of CO_2. The generated amount of liquid has to be evaporated, e.g. inside a heat exchanger, before it enters the suction ports of the compressors. This is to avoid wet operation. Considerably high vapour densities of CO_2 lead to a lower slip between the liquid and the vapour phase in comparison to HFC applications. Consequently the remaining time of liquid on the suction side of CO_2 systems is reduced, even when similar flow velocities are applied.

3.5 Parallel compression

Parallel compression offers the possibility to increase cooling capacities and efficiencies of MT systems or stages during peak load operation and high ambient temperatures. In connection with reduced gas cooler outlet temperatures and discharge pressures it must be considered that the amount of flash gas inside the receiver is reduced as well. This leads to a lower medium pressure, higher enthalpy differences on evaporator side and increased refrigeration capacities. In this case the growth of cooling capacity is not demanded and requires capacity regulation. Consequently, parallel compression is redundant with lower ambient conditions. In case that the parallel compression system operates in condensing mode and the valve for HP control is opened fully, a controlled changeover from parallel compression mode to standard operating mode is possible. A solenoid valve designated as “ECO on/off” in Figure 5 closes when the port is not charged by flash gas anymore. Consequently, the ECO pressure inside the ECO suction chamber shown in Figure 5 is reduced with every stroke of the ECO cylinder and finally to a value lower than the evaporating pressure. In this case an additional valve on the ECO valve plate opens and through a suction channel in the compressor housing, refrigerant on evaporating pressure streams into the ECO cylinder. Consequently all four cylinders operate between evaporating- and discharge pressure.

The medium pressure inside the receiver is a function of the gas cooler outlet conditions and the displacement of the ECO cylinders. Consequently the pressure is both system and compressor related. In case of ECO pressures higher than requested, parallel compression can be combined with an FGB to achieve a defined pressure level inside the receiver. The surplus of flash gas that cannot be processed from the ECO cylinders is then bypassed to the suction side again. The benefit of a constant ECO pressure on a certain level shows the disadvantage of lower system efficiencies due to reduced evaporator mass flows.

Another aspect of the application of parallel compression is the required superheat at the ECO port. Pressure fluctuations inside the receiver can lead to boiling effects and a carryover of liquid drops. Therefore a heat exchanger between ECO suction line and e.g. gas cooler outlet has to be provided to avoid wet operation of the ECO cylinders under all operating conditions.

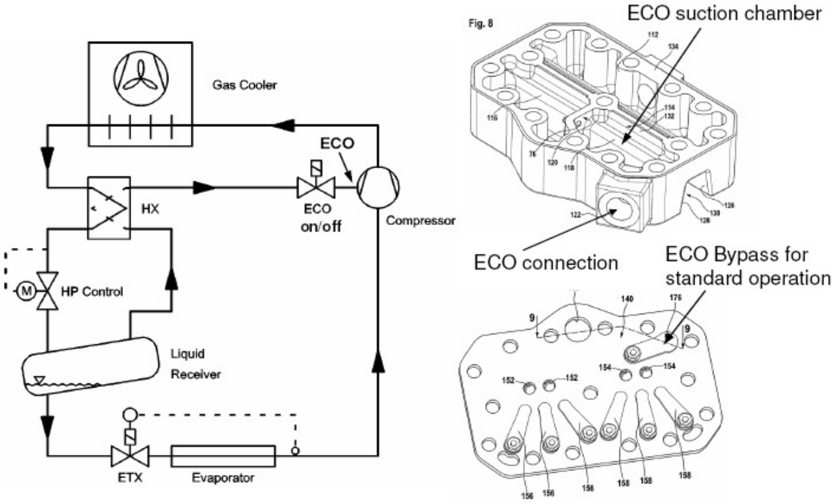

Figure 5. Simplified sketch of a parallel compression system
and details on cylinder head and valve plate design

4. CONCLUSION

The application of CO_2 introduced other system designs to the world of commercial refrigeration. Considering the described systems it is obvious that the complexity is significant. Further field tests have to prove that such complexities are justified under consideration of ecological and economical aspects. FGB and parallel compression add additional benefits to the application of CO_2. The use of parallel compression shows promising results; but field test experience is not sufficient. A correctly balanced capacity regulation, an observation of the operating temperatures and a sufficient oil distribution are the premise for a successful CO_2 installation besides safety, cleanliness and dryness of such systems.

NOMENCLATURE

COP	coefficient of performance
ECO	economizer
FGB	flash gas bypass
HFC	hydrofluorcarbons
HP	high pressure
LT	low temperarature
MT	medium temperature
PAG	polyalkylene glycol
POE	polyol ester

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