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Introduction of an Ejector for Industrial Scale CO2 Systems

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ABSTRACT

The European F-gas Regulation EU 517/2014 demands the incremental phase out of the most common F-gases such as R134a, R410A, R404A, leading to the need of finding future proven refrigerants. Transcritical CO_2 systems are in the meantime state of the art in commercial refrigeration in Europe. The natural refrigerant with a global warming potential of one is now gaining attention for industrial scale applications as well. Facing strongly increasing energy costs and a growing environmental awareness, energy efficiency becomes one of the most important factors for successfully designing an HVAC&R system. One approach to fulfil both requirements is the usage of a CO_2 vapor compression cycle in combination with an ejector. The ejector utilizes the energy, usually lost during the throttling process of an expansion valve, to create a pressure hub, thus unloading the compressors and increasing the energy efficiency. The presented paper discusses the implementation of different ejectors into a CO_2 system calculation selection software. The nominal mass flow of said ejectors at gas cooler outlet conditions of 31°C/92 bara, varies between 800 kg/h and 9500 kg/h. In combination with newly developed CO_2 compressors with eight cylinder design, the software allows the calculation of industrial scale CO_2 booster systems with flash gas bypass (FGB), with parallel compression and high lift ejector. Furthermore, the software is then used to examine the impact of ejectors on industrial scale CO_2 systems by comparing different CO_2 cycle layouts and the respective coefficients of performances. Additionally, the impact on the seasonal energy efficiency ratio is investigated.

1. INTRODUCTION

It is confirmed in the daily project business that the global implementation of Carbon Dioxide (CO_2) refrigeration systems continues to grow. Considering discounters, super- and hypermarkets, cash and carry projects, CO_2 applications in the retail sector established themselves worldwide from 2004 to today. According to Shecco, the number of transcritical installations in the US is over 900 (Koegelenberg, 2022). In Europe, 40,000 systems are reported. In this context, the systems convince with a good seasonal energy efficiency and thus support the industry's efforts to reduce the CO_2 footprint. In addition to the efficiency and the reduction of the cooling capacity, the avoidance of direct F-gas emissions is another starting point to reduce the industries CO_2 footprint. Within Europe, the revised F-Gas regulation (European Union, 2014) pushes the application of low GWP refrigerants, especially CO_2. The so-called Phase Down, which is anchored in the regulation, is in practice a Phase Out of HFC refrigerants with a Global Warming Potential >150, such as R134a, R410A, R404A. In recent years, distribution centers, warehouses, food processing plants become the focus of operators, OEM's and the refrigeration industry. Applications with 2 to 5 MW cooling capacity are already numerous in operation. New developments in the area of the compressors applied and the use of ejectors represent additional possibilities to further improve the system efficiency and thus reduce the energy demand. The paper gives a brief overview over ejector technology and the used ejector model. It describes the implementation of the ejector model into a CO_2 selection software. An empirical study is conducted with a Beta version of the BITZER selection software (BITZER 2022), which addresses the positive impact of high pressure gas ejectors in an industrial system with a new generation of compressors.

All calculations are performed with preliminary data and are subject to change.

2. EJECTOR DESCRIPTION

The working principle of an ejector is based on a jet pump. It consists out of a motive nozzle, suction nozzle, mixing chamber and diffusor. The refrigerant of the high pressure side leaving the condenser / gas cooler outlet is accelerated through the motive nozzle, thus resulting in an increase in kinetic energy, while the according pressure of the fluid is decreasing. Depending on the operating conditions, the mass flow through the motive nozzle might be limited by choking. If a converging-diverging nozzle is used, the refrigerant can reach supersonic speed. A pressure at motive nozzle exit lower than the suction pressure leads to a secondary refrigerant mass flow, usually the refrigerant exiting the evaporator, being pulled in through the suction nozzle. The secondary mass flow might also be limited by choking. The primary and secondary mass flow then mix in the first part of the mixing chamber, which is often accompanied by shockwaves, resulting in an instant pressure increase. At the diffusor, the kinetic energy of the refrigerant mixture is then transformed into potential energy, thus increasing the pressure. The outlet pressure at the ejector lies between the pressure of the motive- and suction nozzle inlet. The two main characteristics of the ejector are pressure hub and entrainment ratio. The pressure hub is the pressure difference between the pressure at ejector outlet and the pressure of the secondary mass flow at suction nozzle inlet. The entrainment ratio describes the ratio of suction nozzle mass flow in relation to the motive nozzle mass flow. The ideal ejector has a high pressure hub while having a high entrainment ratio.

2.1. An ejector for industrial scale CO_2 system

As CO_2 is entering the market for industrial applications, the need for larger ejectors becomes substantial. Schönenberger *et al*. (2014) showed with their work that ejectors offer a great opportunity to increase overall system energy efficiency by decreasing the compressor power consumption. This work considers the software implementation of seven ejector geometries with a nominal mass flow between 800 kg/h and 9500 kg/h at inlet conditions of 31°C and 92 bara. The ejector with a nominal mass flow of 9500 kg/h is referred to as HDV-E95 and is used throughout the paper for further analysis. A needle valve manages the mass flow through the motive nozzle, thus controlling the transcritical pressure at the gas cooler outlet. No additional high pressure control valve is needed. By closing the needle valve, the motive nozzle throat area decreases, resulting in a decreased mass flow across the motive nozzle and therefore in a reduction of the suction mass flow. The position of the needle valve, which is controlled by a stepper motor, is referred to as utilization. A utilization of 100 % corresponds to a maximum open needle valve. A combination of different sized controllable ejectors allows COP improvements for not only full load, but also part load conditions.

2.2. Ejector model

The ejector model consists out of three submodels: geometric, motive nozzle and pressure hub model. It can be used for different ejector geometries. As this paper focuses on the software implementation and the system analysis, the model will be only briefly illustrated. All empirical factors are obtained from laboratory measurements and ejectors being installed in the field.

The motive nozzle model describes the mass flux *G* of an expansion device. Along a pressure-temperature correlation, the mass flux can be defined in dependency of the motive nozzle inlet temperature displayed by Equation 1, respectively Equation 2.

$$ln\frac{p^*}{p_{crit}} = -6.5908 * \frac{T_{crit}}{T_m^* + 4K} + 6.587 \tag{1}$$

$$G_m^* = \left[-706.73 + 482.6 * \frac{T_m^*}{100K} - 81.845 * \left(\frac{T_m^*}{100K}\right)^2\right] * 10^4 \tag{2}$$

Since not all inlet conditions follow the reference function described in Equation 1, any motive nozzle inlet condition can be calculated as a function of G_m^* and ΔG_{is} as described in Equation 3,

$$G_m^2(T_m, p_m) = G_m^{*2}(T) + \Delta G_{is}^2(p) \tag{3}$$

where ΔG_{is}^2 is the alternation in the squared mass flux along an isentropic phase change.

$$\Delta G_{is}^2(p) = 2\rho_m^2 \int_{p^*}^{p} v(p)dp \tag{4}$$

As shown in Equation 5, the specific volume along an isentropic phase change is a function of speed of sound and specific volume at motive nozzle inlet conditions.

$$v = v_m - \frac{v_m^2}{a_m^2}(p - p^*) \tag{5}$$

An iteration is used to determine for a given entropy s_m the according temperature T_m* along the pressure-temperature correlation, considering the corresponding state to be isentropic. The calculation of the motive nozzle mass flow follows Equation 6.

$$\dot{m}_m = A_m * G_m \tag{6}$$

The motive nozzle area A_m depends on the motive nozzle angle, its diameter and the stroke of the throttling element.

As shown in Equation 7, the pressure hub of the ejector is described as a function of an empirical function F_o, friction and inertia. Friction and inertia depend on ejector geometry, as well as fluid data, and are considered for motive nozzle, suction nozzle, mixing chamber and diffusor.

$$-\Delta p_H = -p_d + p_o = F_o(\rho_m; \rho_d; p_s) + \sum \xi_i R_i + k_T \sum T_i \tag{7}$$

F_o and the frictional coefficient ξ are obtained out of measurements. The diffusor outlet state, which is needed for Equation 7, can be determined using Equation 8 and Equation 9, the mass- and energy balance.

$$\dot{m}_d = \dot{m}_m + \dot{m}_s \tag{8}$$

$$h_d * \dot{m}_d = h_m * \dot{m}_m + h_s * \dot{m}_s \tag{9}$$

For a given diffusor outlet- and suction pressure, the pressure hub function is solved numerically by varying the entrainment ratio.

3. IMPLEMENTATION OF A MODEL FOR DIFFERENT SIZED EJECTORS INTO A CO_2 SELECTION SOFTWARE

Fourteen years ago, the so-called flash gas bypass (FGB) system became a standard solution for the application of CO_2 (Heinbokel 2009). It is quite simple in its design and is therefore considered the "baseline" type of system. Thousands of installed units in many parts of the world verify this. Presuming that the concept is applied to only one temperature level, the FGB concept is a system with single-stage compression, two-stage expansion, and flash gas bypass. The system design comprises a medium pressure receiver in which the liquid and gas phases are divided. The pressure within the medium pressure receiver is controlled and regulated by a flash gas bypass valve and is above the required evaporating pressure level. For two different temperature levels, such as medium temperature (MT) and low temperature (LT), the systems are typically used in a booster configuration, shown in Figure 1. The parallel compression concept reduces the pressure ratio required to return flash gas to a high pressure level. Therefore, flash gas is drawn in at a higher pressure level from a dedicated compressor or compressor stage that is directly connected to the intermediate pressure receiver. Transcritical CO_2 booster systems with parallel compression typically have four different pressure stages. In addition, the system includes a low temperature, medium temperature and parallel compression stage (IT). As pointed out in Figure 1, the high pressure (HP) gas ejector, or ejectors, are installed upstream of the intermediate pressure receiver at the outlet of the gas cooler and exploit the compressor mass flow with the largest possible pressure differential in the system. The liquid from the intermediate pressure receiver is fed as usual to the refrigeration and normal refrigeration evaporators. After expansion and heat absorption in the evaporators, the LT evaporator mass flow is taken in by the LT compressor stage and recompressed to suction pressure level of the MT stage. The circuit considered here can distinguish between two modes of operation. In operation with suction effect, the ejector(s) reduces the load on the MT compressor stage. Driven by the potential and kinetic energy of the motive mass flow, and thus without auxiliary electrical energy, a pressure lift from MT suction to intermediate pressure is generated for a suction mass flow taken from a receiver downstream the MT evaporator(s). The FGB valve, which is installed between the intermediate pressure receiver and the MT suction side, is closed during this operating condition. Consequently, the MT stage is unloaded. Due to the generated pressure lift of the ejector stage, the pressure ratio for the compression of the mass flow is reduced with a simultaneous increase in the suction gas density, whereby the energetic advantage of parallel compression takes effect. In this condition, the pressure in the intermediate pressure receiver is controlled by the parallel compression stage. In contrast, when the ejector is operated without suction, the control valve upstream of the ejector suction port is closed. The parallel compression stage is thus out of operation and the FGB valve regulates the pressure in the intermediate pressure receiver solely by expanding the flash gas to MT suction pressure. Consequently, the ejector(s) operate just as high-precision, high-pressure control valve(s).

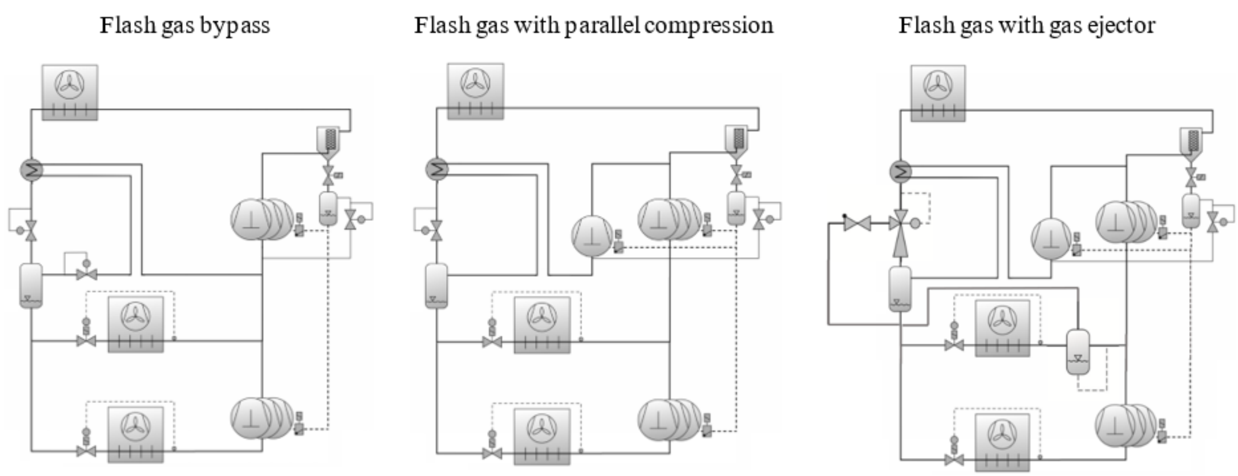

Figure 1: CO_2 flash gas booster systems

3.1. Integration of the ejector model into the system environment

This section describes the implementation of the presented ejector model into the previously discussed system environment in which the ejector reduces the load on the MT compressor stage by drawing in refrigerant after the according evaporator(s). The system calculation is performed in a dynamic-link library and programmed in Delphi. The graphical user interface of the selection software then calls the according calculation library.

It has to be distinguished between cooling demand calculation and type specification. For the cooling demand calculation the required cooling demand, the amount of compressors and ejectors, as well as the operating parameters are given. Operating parameters are $t_{o,MT}$, $t_{o,LT}$, $toh_{evap,MT}$, $toh_{evap,LT}$, $toh_{sl,MT}$, $toh_{sl,LT}$, p_d, p_c and t_c. The algorithm then picks the required compressor and ejector configuration. For the type specification, the ejector(s), the compressor(s) and the operating conditions are given. The algorithm adjusts the utilization of the chosen ejectors, so they can be operated with the selected compressor setup. If an operation is not possible, a warning is given. While the cooling demand calculation helps to design a system for full load conditions, the type specification allows to calculate the performance for part load conditions for a given ejector and compressor setup. To facilitate the description of the calculation process, a system without internal heat exchanger and heat recovery is described. Since the type specification is still under development, the cooling demand calculation process is described.

As the intermediate system pressure (diffusor outlet pressure) is given, the enthalpy at the evaporator inlet(s) can be calculated using Equation 10. Isenthalpic expansion is assumed.

$$h_{evap,in} = f(p_d, x = 0) \tag{10}$$

The evaporator outlet enthalpy is a function of the according pressure and temperature, thus the required mass flow through the evaporator(s) can be derived as described in Equation 11.

$$\dot{m}_{flash,l} = \frac{Qo_{MT}}{h_{evap,MT,out} - h_{evap,in}} + \frac{Qo_{LT}}{h_{evap,LT,out} - h_{evap,in}} \tag{11}$$

The presented ejector model returns the entrainment ratio and motive nozzle mass flow, as a function of the ejector model E_{Mod}, p_c, t_c, p_d, $t_{o,MT}$, $toh_{evap,MT}$, $toh_{suc,MT}$ and the utilization UT, stated in Equation 12 and Equation 13.

$$ER = f(E_{Mod}, p_c, t_c, p_d, t_{o,MT}, toh_{evap,MT}, toh_{sl,MT}, UT) \tag{12}$$

$$\dot{m}_M = f(E_{Mod}, p_c, t_c, UT) \tag{13}$$

The diffusor outlet mass flow is calculated by Equation 14 and is a function of motive nozzle mass flow and entrainment ratio.

$$\dot{m}_d = \dot{m}_M + \dot{m}_M * ER \tag{14}$$

$\dot{m}_{flash,l}$ is the fraction of the total ejector diffusor outlet mass flow(s), which is required to be liquid phase when entering the flash tank. The amount of liquid phase leaving the ejector is characterized by the according outlet quality and diffusor mass flow. An energy balance around the ejector, specified in Equation 15, leads to the according enthalpy,

$$h_d = \frac{\dot{m}_M * h_M + \dot{m}_S * h_S}{\dot{m}_d} \tag{15}$$

where the corresponding vapor quality is determined using the lever rule.

$$x = \frac{h_d - h'(p_d)}{h''(p_d) - h'(p_d)} \tag{16}$$

Knowing the mass flow at the diffusor outlet and the according vapor quality, the amount of liquid refrigerant entering the flash tank is computed using Equation 17.

$$\dot{m}_{d,l} = \dot{m}_d * (1 - x) \tag{17}$$

The vapor phase refrigerant out of the flash tank is taken in by the parallel compressor(s). Often, more than one ejector is needed to deliver the required liquid phase mass flow into the flash tank to feed the evaporator(s). An algorithm is used to obtain the amount of needed ejectors. In most cases, having the ejector(s) run with a fully open needle valve does not match the needed liquid phase mass flow, hence an adjustment of the needle position is required. Since closing the needle valve lowers ejector efficiency, the algorithm targets to control the utilization of as few ejectors as possible, while having the remaining ejectors running with full capacity. Equations 12 to 17 are performed for the according ejector model(s) to obtain the corresponding mass flow(s) for a given pressure hub. Once the ejector(s) suits the required cooling capacity(ies), the parallel compressor(s), MT compressor(s) and, if applicable, LT compressor(s) are selected to meet the demand of the required mass flows. For this purpose, the algorithm is applied that is already used for the calculation of flash gas systems with parallel compression. The according lead compressor for each stage is controlled by a frequency inverter.

4. INVESTIGATION OF THE IMPACT OF THE EJECTOR ON THE COEFFICIENT OF PERFORMANCE FOR DIFFERENT CO_2 CYCLE LAYOUTS

4.1. Introduction of an example for the case study

The example for the case study could be a state of the art distribution center in central Europe. The required cooling load at an evaporation temperature of -5°C is 3.6 MW. To heat up water from 30°C to 40°C, a heat recovery of 1.5 MW is needed. A flooded system with a central surge drum and a pump station will be installed to feed the evaporators.

The following operational assumptions are made. To ensure a heat recovery of 1.5 MW the system has to be operated transcritical all year. To assure system stability for transcritical operation, a minimum pressure of 80 bara, as well as a flash tank vapor quality of minimum 10%, are required. To avoid damage to the parallel compressors and to increase the mass flow through the ejectors, an internal flash gas heat exchanger with a superheat of 3K, is applied. The superheat at the evaporator outlet is assumed to be 1K. The superheat of the suction line is estimated with 2K. Pressure losses for piping and heat exchangers are neglected. As the power input for fans and pumping are considered equal for operation with or without an ejector, it is neglected for any kind of energy efficiency calculation. To lower the amount of applied compressors and in order to improve energy efficiency, newly developed components for industrial scaled applications with CO_2 are considered: A semi-hermetic piston compressor, designated 8CTE-140K, and an adjustable ejector, termed HDV-E95.

4.2. Description of the eight cylinder compressor

The compressors, considered for the prevailing work, were developed to meet the requirements of industrial scaled applications, which are high efficiency, capacity regulation and low oil carry-over rates. The prevailing work is based on a compressor with a displacement of 99.2 m³/h at 50 Hz or 110.6 m³/h at 60 Hz grid frequency. It can either be equipped with a further developed mechanical capacity regulation, which avoids the principle of blocked suction channels, or be operated by a frequency inverter between 30 and 60 Hz. Performance and efficiency are at a very high level as described in Javerschek and Mannewitz (2021).

4.3. Analysis and comparison of different CO_2 cycle layouts

The three systems which shall be examined are the FGB (flash gas bypass) system, the flash gas system with parallel compression and the ejector system with parallel compression, as shown in Figure 1. Throughout the rest of the paper, the system with flash gas and parallel compression will be referred to as FGP system, the system with ejector as FGE system, respectively. The different systems are compared in terms of COP for a set of four different operating conditions. While the evaporation temperature and superheats are kept constant, the cooling load as well as the ambient temperature are varied according to the SEPR calculation method following EN13215 (2020). The corresponding conditions are shown in Table 1. The following chapter *Calculation of the seasonal performance ratio* provides further information. For transcritical operation the temperature difference between gas cooler outlet and ambient temperature is assumed to be 2K. Since a minimum vapor quality of 10% is desired to guarantee system stability, the gas cooler outlet temperature for point D is raised to 14°C. The gas cooler pressure for point A is determined in accordance with Equation 1, respectively with the BITZER software (BITZER 2022) for the FGB and FGP system.

Table 1: Cooling load demands and ambient temperatures in accordance with EN13215 (2020) and the resulting operating parameters for the case study

Point	A	B	C	D
Qo in MW	3.60	3.23	2.69	2.16
t_{amb} in °C	32.0	25.0	15.0	5.0
t_c in °C	34.0	27.0	17.0	14.0
p_c in bar	85.2 // 84.0	80.0	80.0	80.0

The intermediated pressure of the systems is optimized for the different points of operations in terms of the highest possible coefficient of performance while using the fewest amount of compressors. The calculations are performed with a Beta version of the BITZER selection software (BITZER 2022).

As demonstrated in Table 2, Point A requires 14 eight cylinders compressors for the FGB system, while the flash gas system with parallel compression requires 12 compressors, eight MT compressors and four parallel compressors. The ejector system demands four MT compressors, seven parallel compressors as well as 16 HDV-E95 ejectors of which one is controlled. The lead compressor is operated with a frequency inverter to adjust to the required mass flow. The COPs and the respective improvements for different systems are shown in Figure 2. While the use of parallel compression leads to a COP improvement of 12.4% in comparison to the FGB system, the ejectors add an additional 7.4% improvement. Percentagewise, the ejector shifts the most refrigerant from the evaporator to the parallel compressor for Point A, thus the COP improvement for this point of operation is the highest.

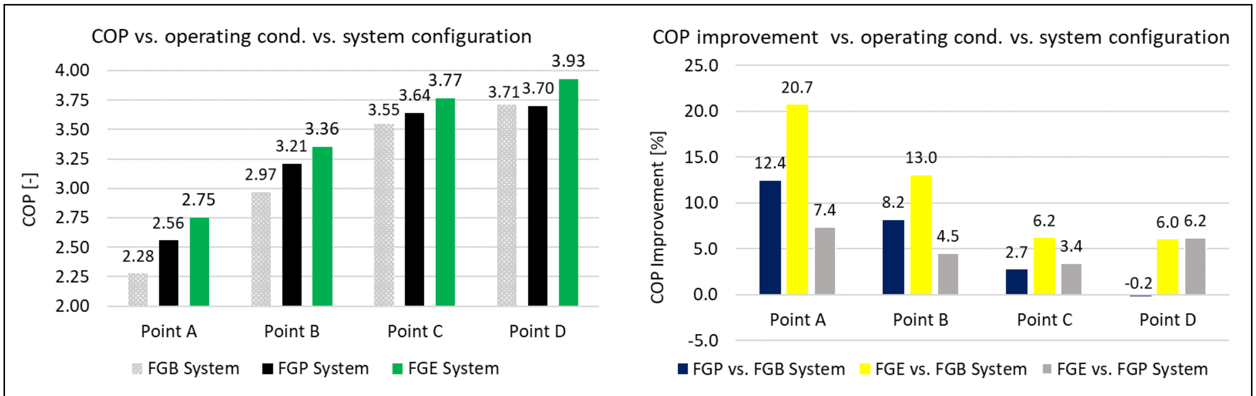

Figure 2: COP comparison for different investigated CO_2 Cycle Layouts

The COP improvement through the ejector for Point B is 4.5% in comparison to the operation with parallel compression. As already pointed out for Point A, the FGB system shows the smallest coefficient of performance. The ejector system has a 13.0% higher COP than the FGB system. As the cooling capacity decreases, the total amount of compressors needed decreases. One of the eight needed ejectors is controlled to adjust to the required mass flow. All ejectors are HDV-E95 ejectors. Using an ejector and parallel compression instead of parallel compression results in a COP improvement of 3.4% for Point C, which is the lowest observed for all investigated conditions. Operating the lead MT as well as the lead parallel compressor with a relatively low frequency to match the required cooling load, the compressor efficiency slightly decreases, thus the system COP decreases. One out of the four HDV-E95 ejectors is controlled to match the required mass flow. Point D shows a COP improvement of 6.2% for the comparison of the ejector system with the FGP system. If a parallel compressor stage is used without the ejector, the COP decreases by 0.2% in regards to the FGB system.

Table 2: System configuration, respective efficiencies and cooling loads for Point A, B, C and D

	p_d [bar]	$f_{MT,comp}$ [Hz]	$f_{par,comp}$ [Hz]	$no_{MT,comp}$ [-]	$no_{par,comp}$ [-]	UT_{ejec} [%]	no_{ejec} [-]	$Qo_{evap,MT}$ [kW]	COP [-]
Point A (FGB)	40.0	37	-	14	-	-	-	3603	2.28
Point B (FGB)	40.0	34	-	10	-	-	-	3173	2.97
Point C (FGB)	40.0	45	-	7	-	-	-	2702	3.55
Point D (FGB)	40.0	60	-	5	-	-	-	2133	3.71
Point A (FGP)	40.0	60	49	8	4	-	-	3598	2.56
Point B (FGP)	40.0	60	38	7	2	-	-	3189	3.21
Point C (FGP)	38.0	49	36	6	1	-	-	2698	3.64
Point D (FGP)	35.0	34	30	5	1	-	-	2168	3.70
Point A (FGE)	38.7	53	55	4	7	74.0	16	3600	2.75
Point B (FGE)	37.5	35	49	5	4	34.9	8	3214	3.36
Point C (FGE)	37.0	34	40	5	2	89.5	4	2698	3.77
Point D (FGE)	37.0	40	59	4	1	88.0	3	2170	3.93

The FGB system requires for full- and part load operation 14 MT compressors in total. Eight MT compressors and four parallel compressors are selected for the FGP system. The ejectors lead to a shift from some MT compressor mass flow to the parallel compressors, resulting in the need of five MT compressors and seven parallel compressors.

5. CALCULATION OF THE SEASONAL ENERGY PERFORMANCE RATIO

Over the years, more attention has been paid to the part-load coefficient of performance of refrigeration systems and condensing units. As part of a regulation mandated by the Ecodesign Directive, the EU hast established minimal requirements for the Seasonal Energy Performance Ratio (SEPR) for condensing units for commercial refrigeration. This allowed the EU to define thresholds for the energy efficiency of products sold to the EU market. Ecodesign Regulation (No) EU 2015/1095 on professional refrigeration (ENTR Lot 1) is at the moment under revision and defines these thresholds and the corresponding calculation procedures. This procedure focuses on the actual energy consumption of a refrigeration system, taking part load behavior of the system as well as seasonal temperature profiles into account. The calculation method was defined for condensing units but also delivers reasonable results for other refrigeration systems.

5.1. Calculation of the SEPR according to ENTR Lot 1

The calculation process of the SEPR is described in the latest revision of the EN13215 (2020). The calculation is based on the bin temperature profile of Strasbourg to reflect the seasonal behavior of a refrigeration system in central Europe. In addition, a load profile from 100-60% for medium temperature applications is mapped with the temperature profile reflecting the part load behavior of a refrigeration system. In the SEPR calculation, four rating points at 32°C/25°C/15°C/5°C ambient temperature have to be calculated at -10°C evaporating temperature for medium temperature applications. The four rating points are calculated at four different load conditions, 100%/90%/75%/60%. If a refrigeration system is not able to reach the required part load capacities, a degradation factor is applied to the COP. The degradation coefficient is assumed as 0.25. This mechanism takes different methods of capacity regulation into account and evaluates its efficiencies. COPs are interpolated for each 1K temperature bin between these 4 rating points. The SEPR is calculated by summing up cooling capacity and dividing it by the total power consumption. The calculations in this paper are based on the SEPR calculation while the evaporating conditions are customized to certain industrial applications.

5.2. SEPR comparison for different CO_2 cycle layouts

The SEPR calculation is performed for previously described system layouts. To adjust the calculation to the presented example, the evaporation temperature is altered to -5°C. The parameters and cooling loads presented in Table 1 are in accordance with SEPR calculation described in EN13215 (2020). The required cooling loads are matched using a frequency inverter. Figure 3 shows, that the SEPR for the FGE system is 3.73 and therefore the highest value for all compared system solutions. An improvement of 7.6% in comparison with the FGB system, respectively an improvement of 4.8% in comparison with the FGP system, are achieved. Even though the FGP system offers a great COP improvement for Point A and Point B, the SEPR improvement in regards to the FGB system is 2.7%. This is based on the relatively moderate improvement for the coefficient of performance for Point C and Point D. As the Strasbourg temperature profile shows a significant concentration in temperatures close to Point C (t_{amb} = 15°C) and Point D (t_{amb} = 5°C), the influence of those two conditions is accordingly high. While the FGB system needs a total of 14 compressors, the ejector as well as the parallel system needs 12 compressors.

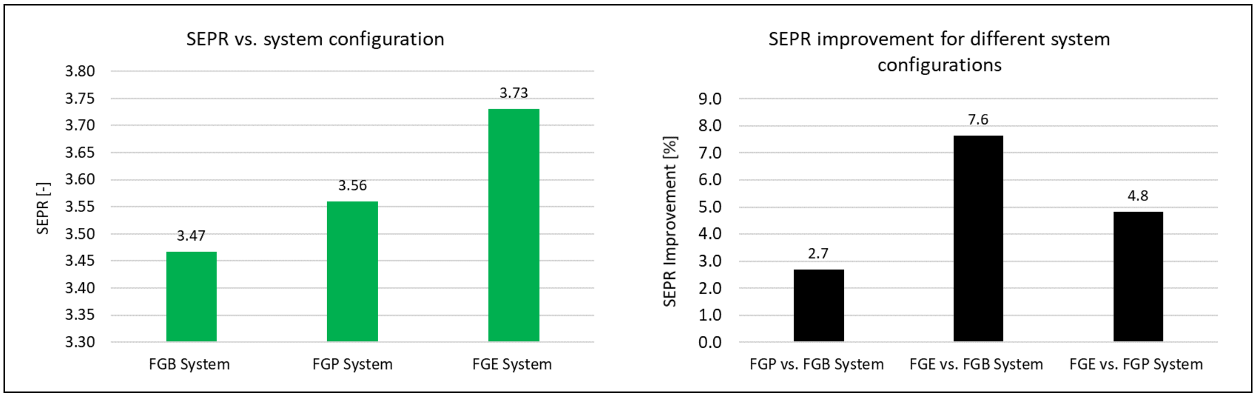

Figure 3: SEPR comparison for different investigated CO_2 Cycle Layouts

6. CONCLUSION

An ejector model which can be used for different ejector geometries has been presented. The implementation of the ejector model into a selection software enabled to conduct an empirical study to investigate the impact of ejectors on a CO_2 system in regards of coefficient of performance and seasonal energy performance ratio. The case for which the analysis has been performed is an example for an distribution center in central Europe requiring a cooling capacity of 3.6 MW for an evaporation temperature of -5°C as well as a heat recovery of 1.5 MW, resulting in a transcritical operation. To ensure high energy efficiency and to limit the number of compressors, newly developed eight cylinder compressors have been deployed. The COP comparison has been examined for the SEPR rating points as described in EN13125 (2020). The evaporation temperature for the according calculations has been altered to -5°C. Being able to control the needle position of the ejector as well as the lead compressors via frequency inverter, all required cooling capacities could be matched. The analyzed operating conditions showed the highest COP improvement for a gas cooler outlet temperature of 34°C and a gas cooler pressure of 85.2 bara. The improvement was 7.4% in regards to a flash gas system with parallel compression and 20.7% for a flash gas bypass system. In comparison to a flash gas bypass system and a system with parallel compression, the ejector led to a SEPR improvement of 7.6% and 4.8%, respectively. As shown in Table 2, the FGB system needed with a total amount of 14 compressors, two more than the FGP and FGE system. The total amount of compressors needed for ejector operation could be lowered to 11 by using one compressor as MT as well as a parallel compressor. Future work will focus on the model development and software integration for liquid ejectors. Additionally, a CO_2 system calculation will be developed for ejectors applied in low lift applications.

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NOMENCLATURE

a	Speed of sound	(m/s)
A	Area	(m²)
COP	Coefficient of performance	(-)
dp	Change in Pressure	(Pa)
E	Ejector	(-)
ER	Entrainment Ratio	(-)
f	Frequency	(Hz)
F_0	Empirical Function	(Pa)
G	Mass Flux	(kg/m²-s)
h	Specific Enthalpy	(kJ/kg)
k_T	Coefficient Inertia	(-)
$\dot{m}$	Mass Flow	(kg/s)
p	Pressure	(Pa)
no	Number	(-)
Qo	Cooling Capacity	(kW)
R_i	Friction Term	(Pa)
s	Specific Entropy	(kJ/kg-K)
SEPR	Seasonal Energy Performance Ratio	(-)
t	Temperature	(°C)
T	Temperature	(K)
T_i	Inertia Term	(Pa)
toh	Superheat	(K)
UT	Utilization	(-)
v	Specific Volume	(m³/kg)
x	Vapor Quality	(-)

Greek symbols

ρ	Density	(kg/m³)
Δ	Change	(-)
ξ	Frictional Coefficient	(-)

Abbreviations and Subscripts

amb	Ambient
c	Condenser / Gas Cooler Outlet
comp	Compressor
crit	Critical Point
d	Diffusor Outlet / Intermediate Pressure
ejec	Ejector
evap	Evaporator
FGB	Flash Gas Bypass
FGE	Flash Gas Ejector
FGP	Flash Gas Parallel
flash	Flash Tank
H	Lift
HP	High Pressure
in	Inlet
is	Isentropic
IT	Parallel Compression
l	Liquid Phase
LP	Low Pressure
LT	Low Temperature
m	Motive Nozzle Inlet
Mod	Model
MT	Medium Temperature
o	Evaporating
out	Outlet
par	Parallel
s	Suction Nozzle Inlet
sl	Suction Line

Superscripts

*	Reference Function