Efficiency increase in carbon dioxide refrigeration technology with parallel compression

H. Fritschi[1], F. Tillenkamp[1*], R. Löhrer[2] and M. Brügger[3]

[1]*Institute of Energy Systems and Fluid-Engineering, Zurich University of Applied Sciences, Technikumstrasse 9, CH-8401 Winterthur, Switzerland*

[2]*Scheco AG, In der Euelwies 17, CH-8408 Winterthur, Switzerland;* [3]*Leplan AG, Technoparkstrasse 2, CH-8406 Winterthur, Switzerland*

Abstract

In this article, we will review the comparison between a CO_2 refrigeration system with a parallel compressor and a conventional carbon dioxide refrigeration machine. In order to carry out the comparison, a numerical model has been generated showing a good correlation to experimental data obtained with a fully instrumented test rig machine. We can determine under which conditions an increase in the energy efficiency ratio (EER) of 10% and more can be expected with a parallel compression system. The results will help users to decide upon economically viable refrigeration systems, depending on the operational conditions of the machine. In usual operating conditions, the achievable increase in the EER by the parallel compressor has been determined; furthermore, it showed that a low evaporation temperature and a high temperature at the outlet of the gas cooler have a positive effect on the parallel compression circuit. With respect to the intermediate pressure, an optimum can be specified at low evaporation temperatures.

Keywords: carbon dioxide; refrigeration; natural refrigerant; parallel compression; energy efficiency

*Corresponding author: frank.tillenkamp@zhaw.ch

Received 7 May 2015; revised 10 November 2015; accepted 13 January 2016

1 INTRODUCTION

A large number of the substances nowadays used as refrigerants are hydrofluorocarbons (HFC) with a high global warming potential (GWP). HFC's are the most common sort of fluorinated gases, the so-called F-gases. As a refrigerant, they find their way into the environment through leakages and improper disposal, thus contributing to global warming. Table 1 shows the amount of HFC, hydrochlorofluorocarbons (HCFC) and chlorofluorocarbons (CFC) used for refrigeration worldwide (not including air-conditioning units and heat pumps). The new development of both HCFC and CFC is worldwide restricted as a result of the Montreal protocol and its revisions.

The *European Union climate and energy package* [1] aims to reduce the EU's greenhouse gas emissions by 20% until 2020. Regulation (EU) No. 517/2014 [2]—also called F-gas Regulation—is a package of specific measures created to achieve this objective. It aims to gradually reduce the number of new appliances using F-gases down to 21% of current stocks by 2030. Given the conditions imposed by the legislator and, not least, our social obligation, it is crucial that alternatives to F-gases are being found. For obvious reasons, natural refrigerants have to be considered as substitutes. Table 2 shows four of the most common substances which can be used as natural refrigerants. For household appliances, such as fridges and freezers, natural refrigerants like R600a [3] or a mixture of R600a/R290 [4] can be used as replacements. Both are hydrocarbons (HC) with a very low GWP. However, for safety reasons, their application is limited to small appliances with a filling quantity of <150 g. Larger filling quantities require adherence as per the regulations of explosion prevention. In contrast, 90% of industrial applications in developed countries and 40% in developing countries are already today being operated with the natural refrigerant R717, which does not contribute to global warming [5, 6]. However, its use is limited to large systems which are able to conform to the strict safety regulations that are necessary due to the toxicity of R717.

The highest amount of environmentally hazardous refrigerants is being found in commercial refrigeration. Systems for this

International Journal of Low-Carbon Technologies 2017, 12, 171–180

doi:10.1093/ijlct/ctw002 Advance Access Publication 18 February 2016

Table 1. *Natural refrigerants and their properties.*

Refrigerant number	Chemical name	Safety group	Global warming potential
R744	Carbon dioxide	A1	1
R717	Ammonia	B2	0
R290	Propane	A3	3
R600a	Isobutane	A3	3

Table 2. *Global refrigerant bank of CFCs, HCFCs and HFCs by 2010 [5].*

Application	Quantity (t)
Domestic refrigeration	150 000
Commercial refrigeration	340 000
Industrial refrigeration	145 000
Transport refrigeration	51 110

purpose cover a wide performance range and are subject to a price-sensitive market. R290 can be used for ready-to-use freezer cabinets if safety requirements are taken into account [7]. However, while R290, R600a and R717 are not an option for most applications due to strict regulations, CO_2 lends itself as an environmentally friendly alternative classified in the lowest ASHRAE 34 safety group A1. Owing to a high critical pressure at a low critical temperature, the technical realization of refrigeration systems using CO_2 is subject to increased requirements. Special pressure-resistant components are needed, and manufacturers nowadays are able to provide a wide range of products. Moreover, the use of stainless steel piping requires complex welded joints. Solderable pipes and fittings made of high-strength copper alloy are available as an alternative; however, they are considerably more expensive. Despite these circumstances, it can be said that, on the whole, the biggest potential of CO_2 technology lies in commercial refrigeration.

The use of CO_2 as a refrigerant was rediscovered in the 1980s by Gustav Lorentzen. It was originally intended to be used in mobile air conditioning [8]. Kim *et al.* [9] and Cavallini and Zilio [10] give an overview of the CO_2 properties as a refrigerant and its uses in transcritical applications. The application of CO_2 refrigeration systems in commercial refrigeration [11], particularly for use in supermarkets [12–15], has been researched and has already been introduced in several European countries [6]. Normally, such facilities need to be heated as well as cooled. CO_2 applications are also highly suited to extracting useful heat, as described by Colombo *et al.* [16] and Sawalha [17]. In this case, systems using CO_2 compared with HFC are in addition to the considerably lower direct GWP, also resulting in higher total energy efficiency. Nevertheless, compared with conventional refrigerants in condensing mode [18], CO_2 appliances achieve a lower energy efficiency ratio (EER) for cooling in higher ambient temperatures (high ambient temperatures lead to transcritical operation). Ironically, under transcritical operation conditions in consequence of high ambient temperatures, there is usually also a reduced heat demand. This fact ultimately leads to increased demands on the system's cooling performance; thus, in view of the TEWI (total equivalent warming impact), the resulting negative impact in terms of the indirect greenhouse effect can be limited.

1.1 State of the art

Through theoretical studies, Bell [19] discovered that, under certain conditions, the efficiency of a refrigerating machine can be increased by more than 10% with a parallel compressor. It was observed that the EER and the refrigeration capacity depend on the temperature at the gas cooler outlet.

With the help of model calculations, Sarkar and Agrawal [20] discovered that a parallel compressor can improve the EER by up to 47.3% compared with the simple one-stage process. They noted that the parallel compressor is more profitable at low evaporation temperatures. They also reported that the optimal intermediate pressure varies only marginally.

Chesi *et al.* [21] studied the parallel compressor theoretically and practically by comparing it with the simple one-stage process, which was calculated from data of the parallel compressor cycle. Special attention was given to the efficiency of the flash tank, since the prerequisite for parallel compressors is the quality of how the mass flow is being separated. It was found that the efficiency of the separator decreases with increasing flow rates at its entrance.

1.2 Transcritical cycle with flash gas injection

Even in the temperate climate zone, the low critical temperature of CO_2 requires frequent supercritical operation of the refrigeration system. In this case, when re-cooling, the refrigerant is not condensed, but the gas is merely cooled. In the present study, we refer to the gas cooler instead of a condenser, even when, in some circumstances, no supercritical operation is involved. Only after a decrease in pressure, a phase change takes place in the refrigerant. Transcritical refrigeration systems with one-stage compression are often equipped with an intermediate pressure stage, i.e. with a two-stage expansion. This is common in systems with a single evaporation pressure [22] as well as in systems with two evaporation pressures, so-called booster machines [23].

In a separator with intermediate pressure, the gaseous refrigerant, which is present after the decrease in pressure, is being separated from the liquid and diverted. This refrigerant, also known as flash gas, constitutes the part of the refrigerant mass flow that cannot be used for refrigeration. Thus, the flash gas bypasses the evaporator, is superheated and injected directly into the suction pipe of the refrigeration compressor. The simple P&ID of the flash gas injection system is shown in Figure 1a.

A cycle which uses a flash gas bypass will hereinafter be referred to as flash gas injection. On the one hand, contrary to the one-stage process, the separation of the mass flow allows for smaller evaporators. On the other hand, the heat transfer in the evaporator is increased due to the lower steam content. The flash gas injection is state of the art and forms the basis for the present study.

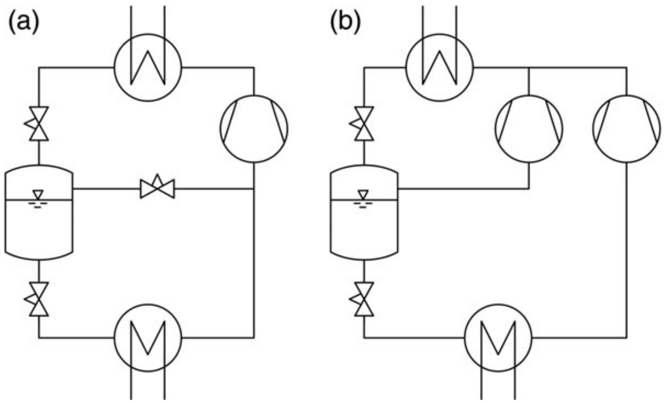

Figure 1. *Simple P&ID for flash gas injection (**a**) and parallel compression (**b**).*

1.3 Transcritical cycle with parallel compression

In order to compress the refrigerant gas in the separator to high pressure, the simplest technical amendment to the flash gas injection is to use an additional compressor rather than the flash gas bypass. Alternatively, the flash gas can also be compressed with the main compressor by using a certain number of its cylinders for parallel compression [19]. Another way of directly compressing the flash gas to gas cooler pressure is to lead the gas through a vent into the compression chamber of the main compressor [24]. In all three cases, the required compression power is reduced and furthermore, the compressor provides improved efficiency at smaller compression ratio operating conditions. This leads to the conclusion that refrigerating systems using parallel compression achieve at least the same level of efficiency, or are more efficient than refrigerating systems using flash gas injection. The simple P&ID of the parallel compression system is shown in Figure 1b.

The amendment of the cycle requires the inclusion of only one additional component. In practice, the expansion valve for the flash gas is usually also installed in systems with a parallel compressors, serving as the limiter for the maximum suction pressure of the parallel compressor. Depending on the system design, the parallel compressor might not be able to compress high quantities of flash gas, which results in an increase in the intermediate pressure. In this case, the expansion valve opens and hence prevents the suction pressure from exceeding the limit specified by the manufacturer of the compressor. With a parallel compressor, no additional adjustments are necessary, which is a major advantage of this refrigeration cycle. However, a parallel compressor, which includes a frequency converter, increases the price of a refrigeration system, and these additional investment costs have to be compensated by the operating cost savings. In order to provide clarity in this matter, the performances of both refrigeration cycles under the envisaged operating conditions need to be known.

1.4 Research objectives

This article explores efficiency increases regarding the EER in CO_2 refrigeration systems that can be achieved by using parallel compressors, which reduce the indirect contribution to the greenhouse effect. The impact of different operating conditions on the performance of the two presented transcritical cycles is investigated. Previous research shows that, using a parallel compressor results in EER improvements. The present study investigates the extent of these increases in efficiency under different conditions. In practice, there are mandatory minimum efficiency improvement levels for the use of parallel compressors. The aim of this research is to determine the ranges of the most important operating conditions that need to be investigated and to establish whether improvements in economic efficiency can be achieved with a parallel compressor. The performance of both cycles is determined on the basis of a fully instrumented transcritical CO_2 refrigerating machine. With the help of a calculation model, an extensive parametric study is conducted. The model is an exact representation of the CO_2 machine, which allows the accuracy of the calculations to be verified.

2 EXPERIMENTAL INVESTIGATION AND MODELLING

The object of the study is a conventional CO_2 refrigerating machine in the lower performance range, which is able to run at both subcritical and transcritical operating conditions. The cycle of the machine incorporates both flash gas injection and a parallel compressor. Since the operational behaviour of the machine is oscillating, it is difficult to obtain highly precise repetitions of measurements. This is relevant if both cycles are running under the same operating conditions. In order to obtain the best possible comparison of both cycles, a calculation model comprising the two configurations of the testing machine was developed.

2.1 Test facility

The testing infrastructure consists of two hydraulic cycles and the refrigerating machine. Two tanks, with a volume of 3 m^3 each, function as a heating and cooling source from which a water–glycol mixture is pumped to the refrigerating machine. This secondary fluid enters and exits the heat exchanger via a three-way valve. Thus, the flow temperature in the heat exchanger can be precisely adjusted.

The CO_2 refrigeration machine test rig is equipped with an internal heat exchanger in order to superheat the suction gas (*IHX*) and an internal heat exchanger for superheating the flash gas (*IHX FG*). An overall view of the test rig is given in Figure 2. The piping and instrumentation diagram is shown in Figure 3. In order to switch from operating with flash gas injection to operating with parallel compressor, the expansion valve (Pos. 12) simply needs to be closed and the parallel compressor activated. This shows that a refrigerating machine with parallel compressor is comparatively simple. The evaporator and gas cooler are connected to a hydraulic system; the inlet temperature and the volumetric flow rate can be set. This allows the testing of the machine with flash gas injection as well as with parallel compressor

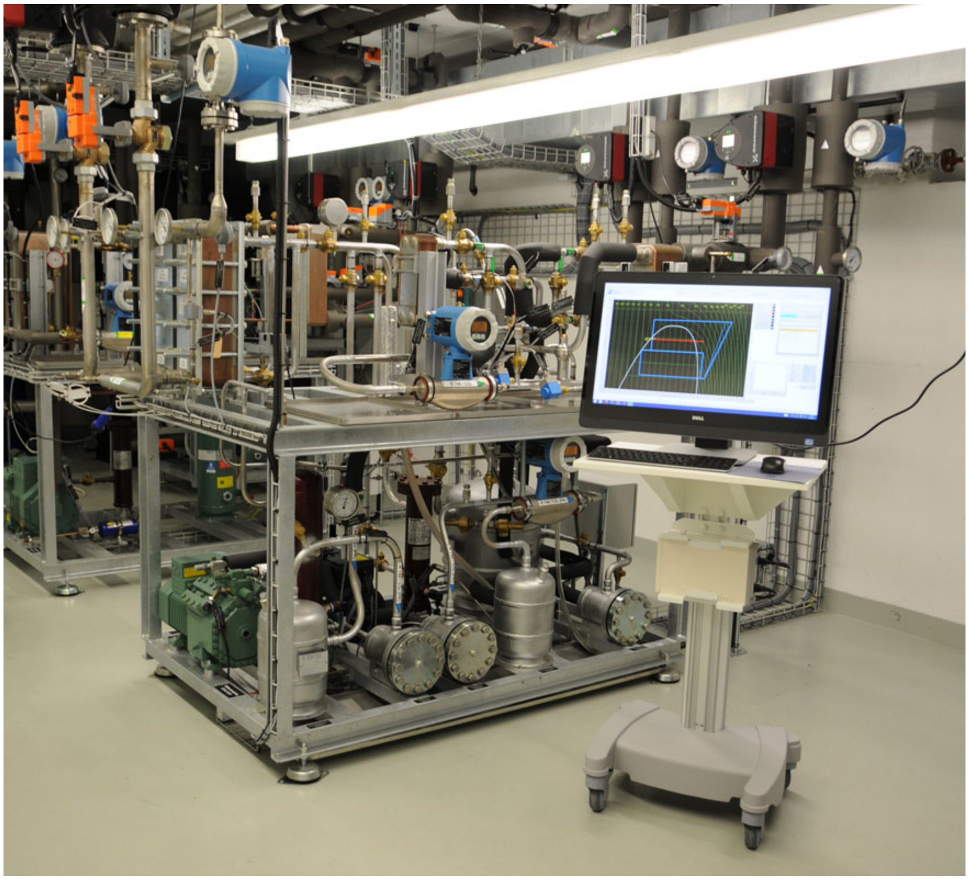

Figure 2. *Overall view CO_2 refrigeration machine test rig.*

under the same internal (piping and instrumentation) and external limiting conditions.

The refrigeration cycle is fully instrumented and equipped with 14 temperature measurement points (PT100) and four pressure taps. The temperature measurement points are located on the outer surface of the refrigerant pipeline. This results in relatively imprecise temperature measurements, as the thermal resistance of the pipeline is not taken into account. In addition, the heat transfer inside the pipeline is poor when measuring refrigerant gases. The mass flow rate is measured at two points with a coriolis flow meter: the mass flow rate at the evaporator and the mass flow rate of the flash gas. The performance of the refrigerating machine is ascertained via the hydraulic side, where temperatures are directly measured in the water–glycol mixture:

$$\dot{Q} = \dot{V} \times \rho \times c_{\mathrm{p}} \times (T_{\mathrm{inlet}} - T_{\mathrm{outlet}}) \quad (1)$$

To keep the measurement errors as small as possible, the machine is constantly operated with the largest possible ΔT. The experimental analysis of this study is usually conducted with $\Delta T = 10^{\circ}\mathrm{C}$. An electromagnetic flow meter measures the volume flow. Density and specific heat capacity are calculated based on the mean temperature specified by the glycol manufacturer. The electrical power of the compressor is measured as well as the frequency converter, in order to take the energy loss into account. All measuring instruments are equipped with analogue connections. The measurement data are digitized with a sampling rate of 1 S/s. Previous results from tests with the testing machine indicate that, in order to obtain a representative measurement, a stationary state has to be maintained for at least 10 min. According to the recorded data, the EER is as follows:

$$\mathrm{EER}_{\mathrm{PC,Measured}} = \frac{(1/n)\sum_{i=1}^{n} \dot{Q}_0(i)}{(1/n)\sum_{i=1}^{n} P_{\mathrm{el,A}}(i) + (1/n)\sum_{i=1}^{n} P_{\mathrm{el,B}}(i)} \quad (2)$$

$$\mathrm{EER}_{\mathrm{FG,Measured}} = \frac{(1/n)\sum_{i=1}^{n} \dot{Q}_0(i)}{(1/n)\sum_{i=1}^{n} P_{\mathrm{el}}(i)} \quad (3)$$

The relatively small size of the system results in additional restrictions on certain operating points. The operating limit for the low-pressure side of the testing machine is reached at an evaporation temperature of $-15^{\circ}\mathrm{C}$, while on the discharge side, maximum pressure is 90 bar. The operating range for intermediate pressure is between 35 and 45 bar. The above-mentioned conditions cover a wide range of applications of CO_2 refrigeration machines. Limitations are mainly caused by the size of the parallel compressor. When the amount of refrigerant gas is low after expanding at intermediate pressure, the compressor is too large, whereas it quickly becomes too small to exhaust all the flash gas when the steam quality increases. This results in an increase in the intermediate pressure. The capacities of the installed compressors and heat exchangers are shown in Table 3.

2.2 Model

A model was created in Excel VBA and the database REFLIB [25] is used to determine the refrigerant properties. The model represents the test rig machine with its state points as shown in Figure 3. The state points of the refrigeration cycle in the case of parallel compression in Figure 4 can also be seen in the log(p)−h diagram. Additionally, the inlet and outlet conditions at the heat exchangers can be seen in Table 4. Point 8 represents the state of pure saturated refrigerant gas in the evaporator, whereas the superheat control causes the evaporated refrigerant to exit the evaporator at point 9. Heat input leads to unusable superheating in the system which happens between the internal heat exchanger and the suction nozzle of the compressor. This is true for the suction side of the main compressor as well as for the suction side of the parallel compressor. This way, the occurrence of unwanted heat input is taken into account. This influence is relevant because superheating causes the temperature of the refrigerant at the outlet of the compressor to rise further. A limit of ∼125°C must be observed for this position. For reasons of simplification, superheating is assumed to be constant for the model calculations. The resulting decrease in effectiveness of the internal heat exchanger is justified through its marginal effect on system performance, which was proven by Agrawal *et al.* [26].

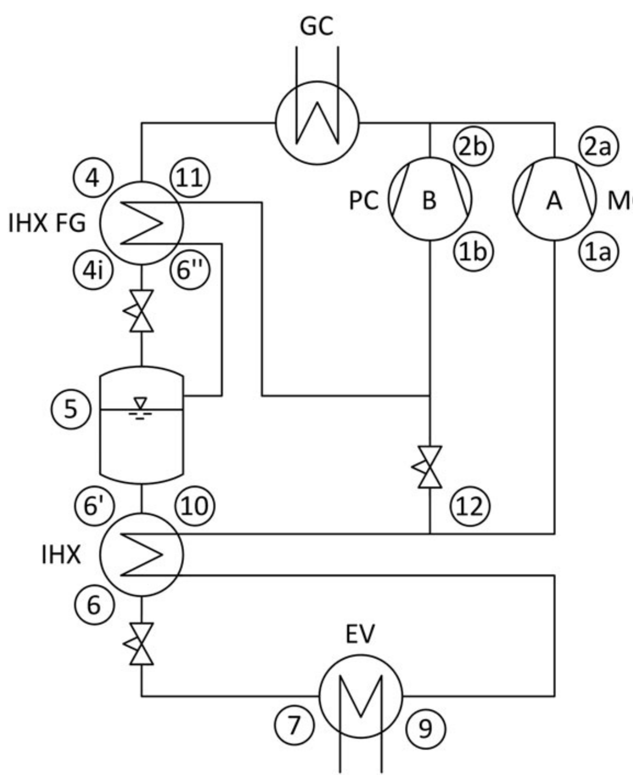

Figure 3. *P&ID of the test rig. EV, evaporator; MC (A), main compressor; PC (B), parallel compressor; GC, gas cooler; IHX FG, internal heat exchanger for flash gas; IHX, internal heat exchanger.*

The values for superheating in the heat exchangers and in the suction line can be seen in Table 5. The superheating in the internal heat exchanger (IHX) leads to a subcooling of the liquid between 6′ and 6, due to the energy balance. The gas cooling between 4 and 4i occurs by the energy balance in the internal heat exchanger for the flashgas (IHX FG), which is calculated by an iterative algorithm.

Iterative algorithm

(i) Set $k = 1$ and $h_{5i}(k-1) = h_4$
(ii) Calculate $x_5(k) = (h_{5i}(k-1) - h_{6'}/h_{6''} - h_{6'})$
(iii) Defining the enthalpy difference in the heat exchanger in reference to total mass flow $\dot{m}_5$ by $\Delta h_{\text{flashgas}} = (h_{1b} - h_{6''}) \cdot x_5(k)$
(iv) Defining the enthalpy at heat exchanger outlet $h_{5i}(k) = h_{5i}(k-1) - \Delta h_{\text{flashgas}}$
(v) Set $k = k + 1$ and go to (ii)

The iterative procedure terminates when $0.999 \leq (h_{5i}(k-1)/h_{5i}(k)) \leq 1.001$.

The EER for the parallel compressor (PC) or the flash gas injection (FG) cycle is calculated as:

$$\text{EER}_{\text{PC}} = \frac{\dot{Q}_0}{P_{\text{el,A}} + P_{\text{el,B}}} \tag{4}$$

$$\text{EER}_{\text{FG}} = \frac{\dot{Q}_0}{P_{\text{el}}} \tag{5}$$

Table 3. *Installed components in test facility.*

Component	Type	Model	Nominal capacity (kW)
Main compressor (MC)	Reciprocating compressor	Bitzer 4MTC-10K-40S	7.7
Parallel compressor (PC)	Reciprocating compressor	Dorin CD 180H Serie 200	1.4
Gas cooler (GC)	Plate heat exchanger	Alfa Laval AXP52-50H-F	40.85
Evaporator (EV)	Plate heat exchanger	Alfa Laval CBXP52-60MDS-F	17.3
Internal heat exchanger (IHX)	Plate heat exchanger	Alfa Laval CBX27-30H-F	1.7
Internal heat exchanger for flash gas (IHX FG)	Tube in tube heat exchanger	Klimal WGCO21WT	0.9

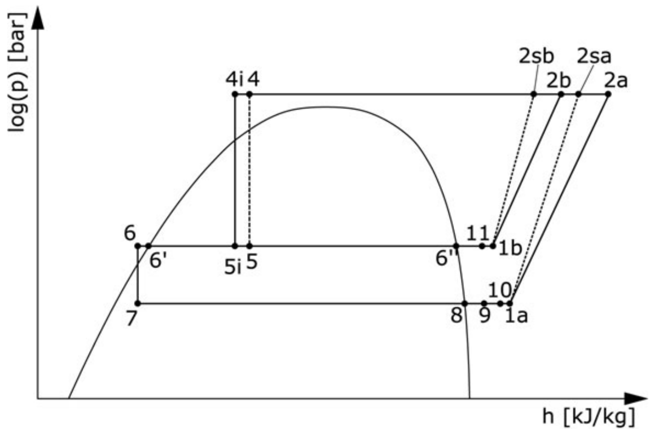

Figure 4. *Thermodynamic cycle of the refrigeration cycle with parallel compression.*

Table 4. *Inlet and outlet conditions of the heat exchangers.*

Heat exchanger	Inlet state	Outlet state
Evaporator	7	9
Gas cooler	2a	4
IHX, suction side	9	10
IHX, intermediate pressure	6′	6
IHX FG, intermediate pressure	6″	11
IHX FG, gas cooler pressure	4	4i
Not useable superheat suction side	10	1a
Not useable superheat intermediate pressure	11	1b

Table 5. *Values for constant superheat.*

Existing superheat	Superheat (°C)
Superheat in the evaporator (EV)	8
Superheat in the internal heat exchanger (IHX)	4
Not useable superheat in suction line	3
Superheat in the internal heat exchanger for flash gas (IHX FG)	12

Considering $\dot{m}_5 = \dot{m}_{6'} + \dot{m}_{6''}$, the following applies:

$$P_{\mathrm{el,A}} = \dot{m}_{6'} \frac{h_{2s,a} - h_{1,a}}{\eta_{\mathrm{A}}} \tag{6}$$

$$P_{\mathrm{el,B}} = \dot{m}_{6''} \frac{h_{2s,b} - h_{1,b}}{\eta_{\mathrm{B}}} \tag{7}$$

$$P_{\mathrm{el}} = \dot{m}_5 \frac{h_{2s,a} - h_{1,a}}{\eta_{\mathrm{A}}} \tag{8}$$

The energy conversion efficiency of the compressor refers to manufacturer's information regarding cooling capacity and power consumption under standard conditions. Therefore, this efficiency includes both isentropic and electric efficiency and a quantity of heat exchange with the ambient and allows conclusions to the effective electrical power required. A lookup search based on inlet and outlet pressure has been implemented to compute the compressor efficiency:

$$\eta_{\mathrm{A}} = f\left(p_1, p_2\right) \tag{9}$$

$$\eta_{\mathrm{B}} = f\left(p_5, p_2\right) \tag{10}$$

It must be noted that the DORIN CD180H, which is used as a parallel compressor, is one of the smallest available reciprocating compressors for transcritical applications, which therefore leads to lower compressor efficiency. The effect of this aspect on the validity of the present paper results in more conservative findings concerning the potential of the improvement by using a parallel compressor. It is to be assumed that bigger parallel compressors are used for installations in industry and therefore higher efficiency can be achieved by using parallel compression.

In transcritical CO_2 refrigerating machines, gas cooler pressure is regulated at the gas cooler outlet. For this purpose, the installation is equipped with a control system and a temperature sensor situated at the outlet of the gas cooler. The model uses the control curve of a commonly used gas pressure regulator [27]. The characteristics of this control curve were adequately determined through several series of measurements.

In summary, the model is based on the following assumptions:

- isobaric evaporation and gas cooling,
- isobaric and ideal separation of the mass flow in the separator,
- heat transfer with the environment is generally neglected except for the suction lines where an unusable superheat between state 10 and 1a for the main compressor and between 11 and 1b for the parallel compressor is considered,
- compression process is non-adiabatic and non-isentropic.

The inputs for the model are:

- cooling capacity,
- intermediate pressure,
- temperature at the gas cooler exit,
- evaporation temperature,
- quantities for superheat according to Table 5.

The outputs of the model are:

- state variables for every state point,
- electrical power demand,
- EER.

2.3 Validation

The validation of the model occurs in four series of measurements with a range of operating conditions which are as broad as possible. As the study does not focus on superheating, the variables that are of interest are the temperature at the outlet of the gas cooler T_{GCO}, the evaporation temperature T_0 and the intermediate pressure p_i. It became apparent that the dependency on T_{GCO} is of particular interest because the pressure in the

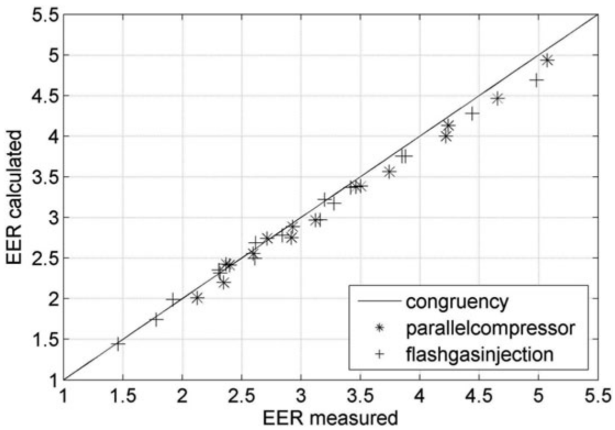

Figure 5. *Comparison of the computed EER with the experimental data obtained.*

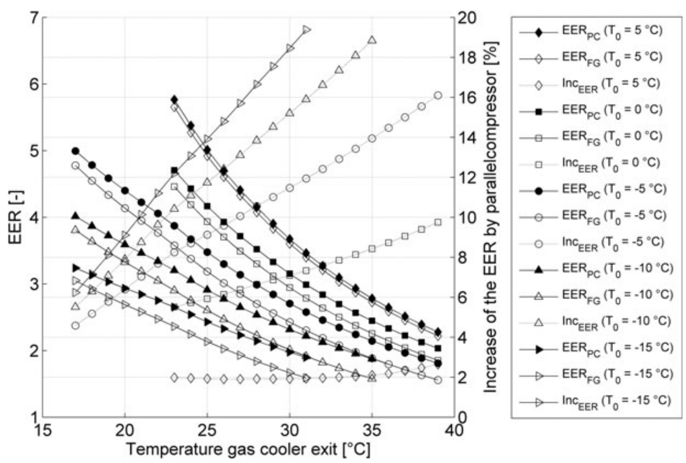

Figure 6. *Performance of the temperature at gas cooler exit at constant* $p_i = 45$ *bar.*

gas cooler is dependent on this temperature. Intermediate pressure values >45 bar are not included in the measurements, as the maximum intermediate pressure of the specific parallel compressor used is at 51 bar. Therefore, the operating point at 50 bar intermediate pressure, which is theoretically barely possible, is only calculated using the model. In order to validate the model calculations, the instances of measured superheating, the temperature at the outlet of the gas cooler, the evaporation temperature as well as the refrigerating capacity are given. Thus, the EER is then calculated based on these values. Figure 5 plots the calculated EER vs. the measured EER to recognize the deviations. The validation shows a deviation between the calculated and the measured values in the case of high values of EER. This can mainly be attributed to the fact that these operating points with a high EER reflect part-load conditions of the instalment and that therefore the efficiency of the reciprocating compressors is higher in such cases. The model does not consider part-load conditions. The overall finding is that the model generally calculates lower EER values and therefore, slightly more conservative predictions from model calculations are to be expected. The standard deviation between model calculations and measurements is 2.37%.

3 DISCUSSION

The system analysis was realized using the calculation model. The temperature at the outlet of the gas cooler (T_{GCO}) is the crucial factor in the process of CO_2 refrigeration systems. This temperature is determined by environmental conditions and, depending on the climatic conditions of the location, leads to a subcritical or transcritical operation of the system. The results show that, in the test machine, a transcritical operation starts at $T_{GCO} > 27.5^{\circ}C$.

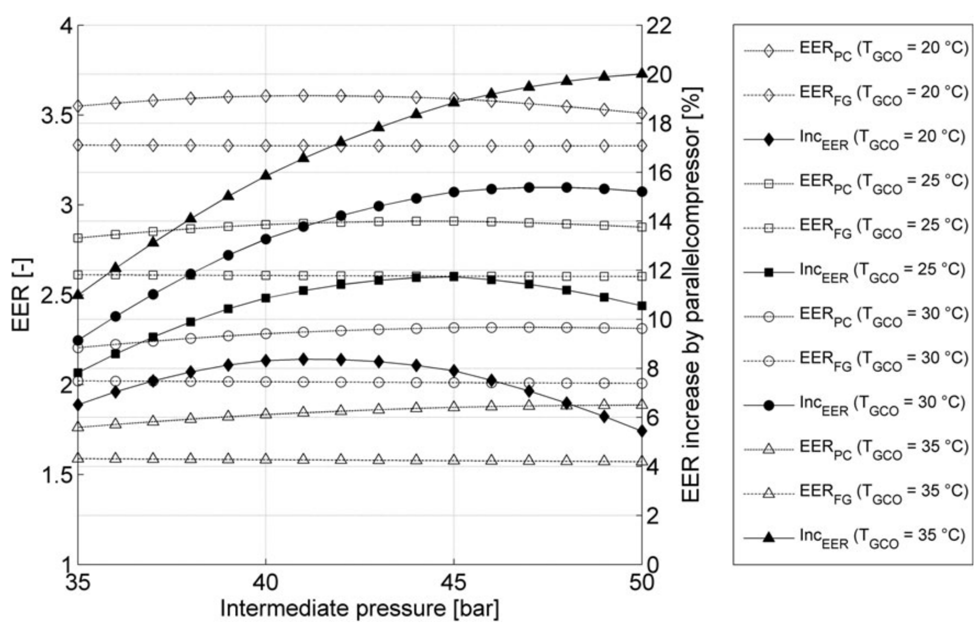

Figure 7. *Performance of the intermediate pressure at constant* $T_0 = -10^{\circ}C$

The range to be examined next concerns the minimum and maximum temperature at the gas cooler exit. In terms of the maximum, it can be said that a temperature at the gas cooler exit of 39°C leads to a maximum gas cooler pressure of 100 bar due to the gas cooler pressure control system [27]. If T_0 is low, the maximum temperature at the outlet of the gas cooler is additionally also restricted by the temperature at the outlet of the compressor. Given a constant evaporation temperature and constant values for the superheating and effectiveness of the compressor, the result is a maximum gas cooler pressure, at which a limiting value is met (state point 2a, Figures 2 and 3). In the present parameter study, a limit of 125°C was established for hot gas in the discharge line. Based on the gas cooler pressure, the temperature at the gas cooler exit can be deduced with the help of control characteristics. The lowest possible temperature at the gas cooler exit is restricted by the isenthalpic expansion from high pressure to intermediate pressure. The model relies on manufacturer specifications when calculating the efficiency of the compressors, which results in additional restrictions due to their limit of application and their known performance data.

Given a constant intermediate pressure and a constant evaporation temperature, there is a deterioration in the performance of the cycle with flash gas injection as well as of the cycle with parallel compressor when the temperature at the gas cooler exit increases. However, the performance of the parallel compressor cycle decreases less. This leads to an improvement compared with the flash gas injection cycle if the temperature at the gas cooler exit is rising. Along with increasing temperatures at the gas cooler exit, the steam quality at the intermediate pressure stage also increases. As the gaseous proportion of refrigerant

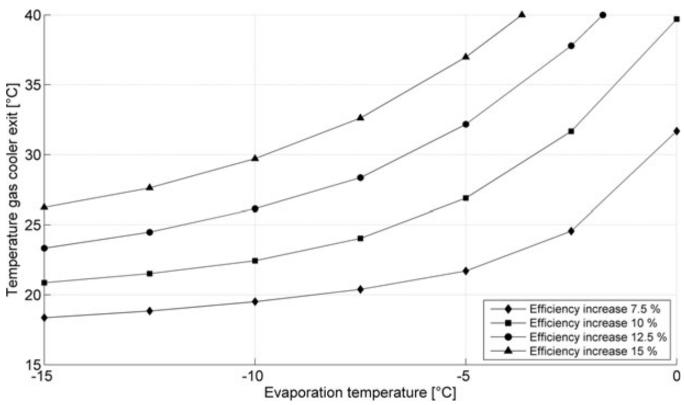

Figure 8. *Efficiency increase at constant* $p_i = 45$ *bar.*

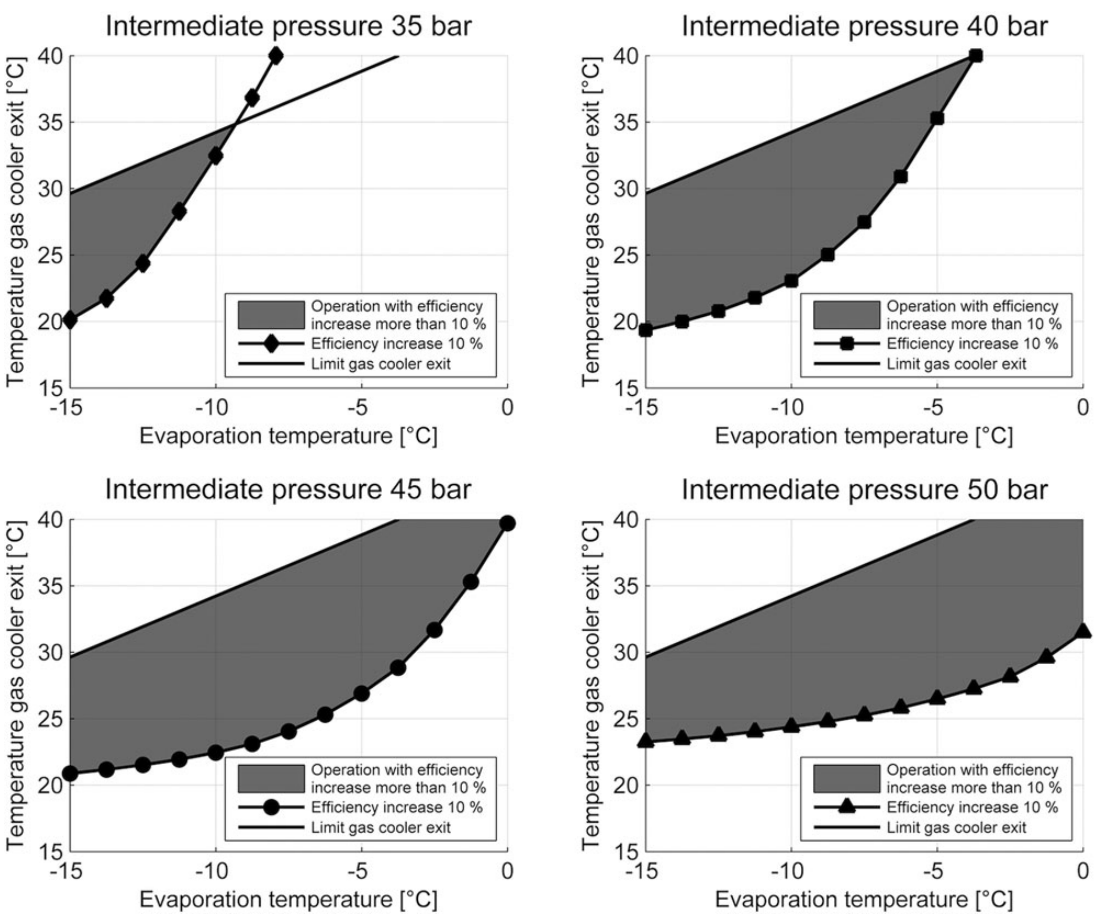

Figure 9. *Range for efficiency increase of more than 10%.*

cannot be used for evaporation, performance decreases. Using a parallel compressor cycle, this mass flow rate is directly compressed to high pressure, which results in a slower decrease in performance and therefore in an increase in efficiency compared with machines with flash gas injection. Figure 6 shows the performance and the percentage improvement with an intermediate pressure of 45 bar.

Examinations of intermediate pressure showed that, with respect to lower suction pressure there is an optimum intermediate pressure related to an efficiency increase gained through the parallel compressor. The optimum varies depending on gas cooler pressure and the temperature at the gas cooler exit. This is because with increasing intermediate pressure, the compression power of the parallel compressor is reduced, leading to a positive effect. On the other hand, increasing intermediate pressure leads to a decrease in evaporation enthalpy. The reasons are a narrowing region of wet steam at an increasing intermediate pressure, and the characteristics of the boiling point curve, where enthalpy increases with rising pressure. This results in a higher enthalpy at the evaporator inlet with higher intermediate pressure and therefore in a decrease in refrigerating capacity. The optimum intermediate pressure thus results from the maximum theoretical performance coefficient dependent on temperature and evaporation temperature at the gas cooler outlet. The influence of intermediate pressure on performance and efficiency increase is shown in Figure 7 with $T_0 = -10^{\circ}C$ at various temperatures at the gas cooler exit.

It is possible to show which increases in efficiency are feasible with a parallel compressor as a function of temperature at the gas cooler exit and evaporation temperature. Figure 8 shows the situation for an intermediate pressure of 45 bar. The curves show constant percentage improvements when using a parallel compressor. This figure shows that the biggest increases in efficiency are possible under circumstances that would, in any case, have a negative effect on the operation of any refrigeration system.

It can be assumed that a parallel compressor is required to increase efficiency by at least 10%. If this condition is met, it can be assumed that energy cost savings will offset the additional investment made. Depending on this 10% boundary condition, the range in which this increase in efficiency is achieved or surpassed can be determined. Figure 9 shows the different intermediate pressures and, in grey, the ranges in which an efficiency increase of at least 10% is achieved. It is clear that, if $T_0 > 0^{\circ}C$, efficiency increases can be achieved with potentially very high intermediate pressure. Assuming usual discharge temperatures that are $>25^{\circ}C$, the parallel compressor will generally be effective at low evaporation temperatures. These diagrams also show that an optimal intermediate pressure exists.

4 CONCLUSIONS

The study shows that, in specific cases, significant increases in efficiency (when compared with a machine with flash gas injection) are possible with the use of a parallel compressor. It was demonstrated that a high temperature at the gas cooler exit generally favours the use of a parallel compressor. If the intermediate pressure remains unchanged, the efficiency increase is raised as the evaporation temperature decreases. The maximum increases in efficiency are not the focus of this study, as these are attained under conditions that hardly ever occur in practice. Efficiency gains within the most frequent boundary conditions for refrigeration systems are more interesting.

With the increase in efficiency achieved by the parallel compressor, the use of CO_2 technology becomes more appealing. Of great interest in regard to practical applications are the boundary conditions under which the use of a parallel compressor becomes economically viable. Using the three most critical process parameters, ranges in which the use of a parallel compressor leads to an increase in efficiency of at least 10% have been defined. In summary, the conditions which must be met for the parallel compressor in order to be economical, based on the assumed 10%, are as follows:

- $T_{GCO} > 27.5^{\circ}C$ (each supercritical operating condition)
- $T_0 < -7^{\circ}C$
- Possibility of realizing intermediate pressures of up to 45 bar

Examination of intermediate pressure shows that there is an optimum which depends on operating conditions. This leads to the conclusion that a process improvement can be achieved by controlling intermediate pressure in systems which exhibit a fluctuating performance, e.g. changing gas cooler pressures and/or evaporating pressures. Regarding the intermediate pressure stage, systems in practice rarely realize pressures of over 40 bar. Operating points with $T_0 > -10^{\circ}C$ can be run efficiently if intermediate pressures of up to 45 or 50 bar are realized. By researching component manufacturers, it is clear that enough pressure-resistant components for the intermediate pressure stage are available on the market.

Refrigeration systems with parallel compressors are particularly well suited to systems with plus or minus cooling or booster systems. Application as a heating pump requires a case-by-case assessment and depends largely on the return flow temperature of the heat source. Characteristic seasonal fluctuations need to be considered when evaluating performance over a year, as they affect in particular the evaporating temperature conditions where efficient performance may only be possible with a high intermediate pressure. Based on the findings acquired here, the use of a parallel compressor in air-conditioning applications can be ruled out due to economic reasons.

ACKNOWLEDGEMENTS

The authors would like to acknowledge Scheco AG and Leplan AG for co-financing this study and for contributing practically relevant requirements.

REFERENCES

[1] Commission of the European Communities, 2020 by 2020—Europe's climate change opportunity, Brussels, 23 January 2008.

[2] The European parliament and the council of the European union, Regulation (EU) No 517/2014 of the European parliament and of the council on fluorinated greenhouse gases and repealing Regulation (EC) No 842/2006. Official J Eur Union, 16 April 2014.

[3] Joybari MM, Hatamipour MS, Rahimi A, *et al*. Exergy analysis and optimization of R600a as a replacement of R134a in a domestic refrigeration system. *Int J Refrig* 2013;**36**:1233–42.

[4] Mohanraj M, Jayaraj S, Muraleedharan C, *et al*. Experimental investigation of R290/R600a mixture as an alternative to R134a in a domestic refrigerator. *Int J Therm Sci* 2009;**48**:1036–42.

[5] United Nations Environment Programme (UNEP). 2010 Report of the Refrigeration, Air Conditioning and Heat Pumps Technical Options Committee. UNEP Ozone Secretariat, 2011.

[6] Chasserot M. Natural refrigerant market trends. In: *Bangkok Technology Conference*, 30 June 2013.

[7] Colbourne D, Espersen L. Quantitative risk assessment of R290 in ice cream cabinets. *Int J Refrig* 2013;**36**:1208–19.

[8] Lorentzen G. Trans-critical vapour compression cycle device. International Patent Publication Patent WO 90/07683, 1990.

[9] Kim MH, Pettersen J, Bullard CW. Fundamental process and system design issues in CO_2 vapor compression systems. *Prog Energ Combust* 2004;**30**:119–74.

[10] Cavallini A, Zilio C. Carbon dioxide as a natural refrigerant. *Int J Low Carbon Tech* 2007;**2**:225–49.

[11] Girotto S, Minetto S, Neksa P. Commercial refrigeration system using CO_2 as the refrigerant. *Int J Refrig* 2004;**27**:717–23.

[12] Campbell A, Maidment GG, Missenden JF. A refrigeration system for supermarkets using natural refrigerant CO_2. *Int J Low Carbon Tech* 2007;**2**:65–79.

[13] Sawalha S. Theoretical evaluation of trans-critical CO_2 systems in supermarket refrigeration. Part I: Modeling, simulation and optimization of two system solutions. *Int J Refrig* 2008;**31**:516–24.

[14] Sawalha S. Theoretical evaluation of trans-critical CO_2 systems in supermarket refrigeration. Part II: System modification and comparisons of different solutions. *Int J Refrig* 2008;**31**:525–34.

[15] Sharma V, Fricke B, Bansal P. Comparative analysis of various CO_2 configurations un supermarket refrigeration systems. *Int J Refrig* 2014;**46**:86–99.

[16] Colombo I, Maidment GG, Chaer I, *et al*. Carbon dioxide refrigeration with heat recovery for supermarkets. *Int J Low Carbon Tech* 2014;**9**:38–44.

[17] Sawalha S. Investigation of heat recovery in CO_2 trans-critical solution for supermarket refrigeration. *Int J Refrig* 2013;**36**:145–56.

[18] Aprea C, Greco A, Maiorino A. The substitution of R134a with R744: an exergetic analysis based on experimental data. *Int J Refrig* 2013;**36**: 2148–59.

[19] Bell I. Performance increase of carbon dioxide refrigeration cycle with the addition of parallel compression economization. In: *Proceedings of the 6h Gustav Lorentzen Conference on Natural Working Fluids*, Glasgow, UK, 2004.

[20] Sarkar J, Agrawal N. Performance optimization of transcritical CO_2 cycle with parallel compression economization. *Int J Therm Sci* 2010;**49**:838–43.

[21] Chesi A, Esposito F, Ferrara G, *et al*. Experimental analysis of R744 parallel compression cycle. *Appl Energy* 2014;**135**:274–85.

[22] Ge YT, Tassou SA, Suamir IN. Prediction and analysis of the seasonal performance of tri-generation and CO_2 refrigeration systems in supermarkets. *Appl Energy* 2013;**112**:898–906.

[23] Ge YT, Tassour SA. Thermodynamic analysis of transcritical CO_2 booster refrigeration systems in supermarket. *Energy Convers Manage* 2011;**52**:1868–75.

[24] Lambers KJ. Das Kältesystem mit Admission (Economizer) mit besonderer Betrachtung der Verdichtung nach Vorhees. PhD dissertation. Braunschweig University of Technology, 2008. Available from: Forschungsberichte des DKV (Deutscher Kälte- und Klimatechnischer Verein).

[25] ILK Dresden, REFLIBfxl Version 2.4, 11 March 2011.

[26] Agrawal N, Bhattacharyya S, Sarkar J. Optimization of two-stage transcritical carbon dioxide heat pump cycles. *Int J Therm Sci* 2006;**46**:180–7.

[27] Danfoss, Documentation capacity controller with heat recovery function AK-PC 781, 02-2014.