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# Development and Performance Measurements of a Small Compressor for Transcritical CO2 Applications

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# DEVELOPMENT AND PERFORMANCE MEASUREMENTS OF A SMALL COMPRESSOR FOR TRANSCRITICAL $CO_2$ APPLICATIONS

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## ABSTRACT

The use of carbon dioxide ($CO_2$) as a refrigerant is considered for an increasing number of various applications including low capacity transcritical processes. Due to the fluid properties of $CO_2$, the pressure ratio of the refrigeration process is rather low compared to common refrigeration processes while the pressure difference is high. Furthermore, the volumetric capacity of $CO_2$ is higher than for traditional refrigerants. These facts result in special demands regarding the design of suitable components and especially compressors to be able to meet the demands regarding the overall systems performance. It is found that different strategies for suction and discharge processes can be applied for transcritical $CO_2$. This paper describes the development and resulting performance characteristics of such a small compressor using $CO_2$ as the refrigerant.

**Keywords:** $CO_2$; compressor, compressor design, compression process, efficiency

## 1. INTRODUCTION

The increasing interest in CO2 as refrigerant in light commercial applications has created a demand for small compressors. For CO2 cooling at ambient temperatures above 30 ºC, the thermodynamic properties of CO2 requires the transcritical cycle, which has gas cooling instead of condensation. This implies, not only a different control strategy compared to normal systems, but also a compressor capable of operating with high differential pressures. As found by Süss /1/ this implies that a favorable compressor choice is a reciprocating type with piston rings, as any leakage of the cylinder needs to be minimized to achieve a high efficiency /2/. The high volumetric efficiency of CO2 in combination with the need for good cylinder sealing practically limits the possibility to design the compressors for refrigeration capacities below 400 W at -10 ºC evaporation temperature.

As mentioned /3/, a development of a small $CO_2$ compressor for cooling capacities in the range of 400 W to 1,2 kW refrigeration capacity at -10 ºC was launched. The development has progressed in sequential steps since the first initiative in 2001. These steps and their individual areas of attention are listed below.

Step 1: Feasibility: The first prototypes was a compressor based on a conventional compressor platform as described in /3/.

Step 2: Simplification and durability: Based on the acquired results from step 1. A more simple drive mechanism was chosen and its durability confirmed in life time test.

Step 3: Performance: Based on the experiences made in steps 1 and 2, a new compressor platform (TN) was designed to obtain a good performance while keeping the same driving mechanism. Additional parameters for the development of this version were safety considerations demanding a minimum internal volume and compactness.

Step 4: Industrialization: In the current process the TN prototype is redesigned to help manufacturing. Pilot production lines are being implemented in parallel to this development process.

This paper focuses on the concepts underlying the development of the $CO_2$ compressor as implemented in step 3 above. It describes the solutions to some of the main design parameters influencing the efficiency of the compressor and it also outlines the efficiency improvements by these design modifications.

## 2. COMPRESSION PROCESS AND DESIGN REQUIREMENTS

Due to the thermodynamic properties of $CO_2$ the effects influencing the compressor efficiency is somewhat different from those found using conventional refrigerants. Static pressure losses and heat transfer losses in the cylinder are of minor importance for the efficiency /2/. However, pressure losses due to non-stationary flow such as pulsations in the suction plenum needed to be considered and resulted in relatively large suction and discharge chambers. Also the leakage is known to be a driving factor in the determination of a suitable type of compression mechanism. Due to the large difference between the suction and the discharge pressure, cylinder leakage is extremely critical for the compression process performance. This requirement led to the choice of a single piston compressor with piston rings. Another important factor was found to be the heat transfer outside the cylinder- especially the suction gas heating inside the suction plenum. The efficiency of the TN compressor was - compared to earlier versions - significantly increased by the reduction of heat transfer between the cold suction gas and the compressor.

## 3. FEASIBILITY

The feasibility test of a small compressor for transcritical $CO_2$ applications was based on an existing compressor platform with changes applied to appropriate functions. *Figure 1* shows a schematic section of the first prototype.

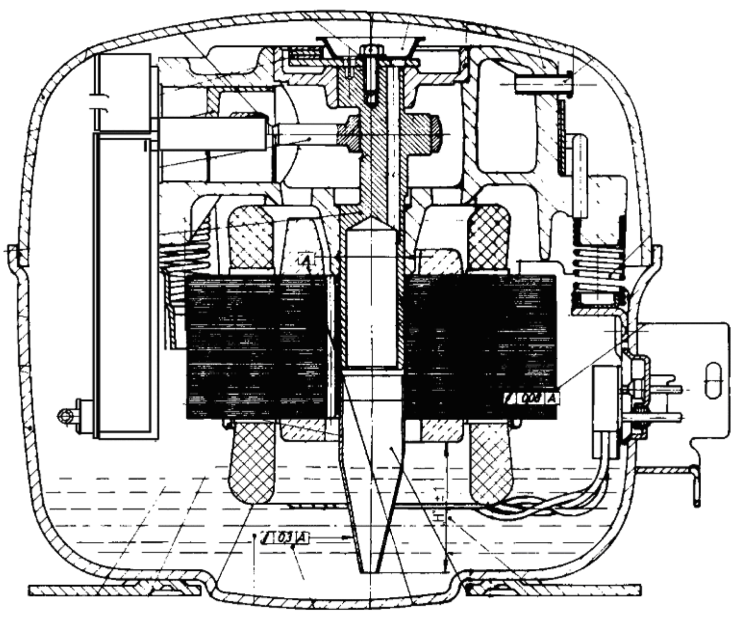

Figure 1: $CO_2$ compressor based on a state of the art hermetic type compressor.

The shell, which is under suction pressure, acts as the first suction chamber. Its original strength was increased to sustain the higher pressures. The piston/cylinder and valve system was modified and the suction and discharge chambers adapted to the requirements specified by the non-stationary flow analysis /3/.

The suction chamber was made of steel and had an approximate volume of 25 cc. *Figure 2* shows a sketch of the suction and discharge chamber arrangement. The suction gas enters the suction chamber directly from the suction line positioned in the upper left corner of the plenum. An oil separation feature is included in suction line telescope. An additional small hole in the bottom of the plenum allows caught oil to drain back to the oil sump. The suction chamber was designed with the focus on minimization of suction gas heating and at the same time of reducing non-stationary pressure losses due to gas accelerations. The main portion of the gas, which is inside the suction plenum chamber, remains there throughout the processes and just act as a gas suspension to reduce the suction pressure amplitude and the acceleration of the gas in the systems low-pressure side.

The suction plenum was mounted onto the valve plate, which is a part of the arrangement. The suction chamber was thermally separated from the discharge chamber as much as possible.

The discharge plenum was designed to resist the high inside pressure and temperature, which may exceed 140 bars and 150 °C respectively. To avoid leakage from the discharge to the suction plenum, the valve plate and the plenum chamber were made as one part. The discharge plenum chamber design is shown in *figure 2* on the right hand side of the muffler arrangement.

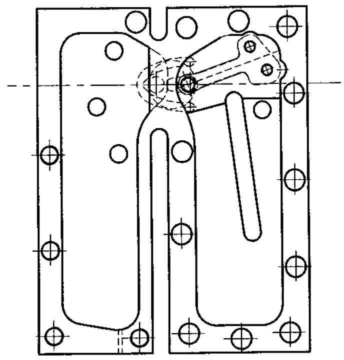

Figure 2: Suction and discharge chambers

Like the suction plenum, the discharge plenum war designed to a volume of approximately 25 cc. It consisted out a main part including the valve plate and a cap. This cap is mounted firmly and tight on the plenum avoiding leakage from the discharge to the suction side plenum. The discharge plenum has an inside structure guiding the discharge directly towards the discharge line. Additionally, this inside structure is used to increases the plenum's strength. The plenum is designed with the prior focus of minimizing heat transfer by taking certain disadvantages regarding noise and vibrations.

As described /1,2/, the leakage of the working chamber, and especially between the piston and cylinder is a major parameter, which determines the overall success of the compressor development. To further investigate this impact, 3 different types of pistons were produced having none, one and two piston rings. The piston/cylinder tolerance of the piston without a ring was machined to high accuracy, as being applied in today's HFC/HC piston ring free compressors with the same piston diameter. The gaps of the pistons with one or two rings were a magnitude larger. These three pistons were tested on the same compressor of the type as shown in *Figure 1* to investigate the impact of cylinder liner leakage on the compressor's performance. *Figure 3* shows the results of the measurements for the different piston arrangements as a function of the discharge pressure. Suction conditions were kept constant throughout all testing at 30 bars and 15 ºC.

Limiting leakage through the piston/cylinder gap by applying a minimum gap, results in the poorest isentropic and volumetric efficiency for the compressor. One piston ring offers already significant performance improvements while two rings could further reduce the harmful leakage, which results in a further performance increase. Higher numbers of piston rings don't offer adequate further performance improvements, as this increases friction and piston mass /1/.

The obvious efficiency reduction with increase of the discharge pressure results mainly from suction gas temperature increase, which goes up together with the pressure ratio of the compression process. The amount of suction temperature increase is depending on the compressor design, and was more significant for this compressor than for the later TN version.

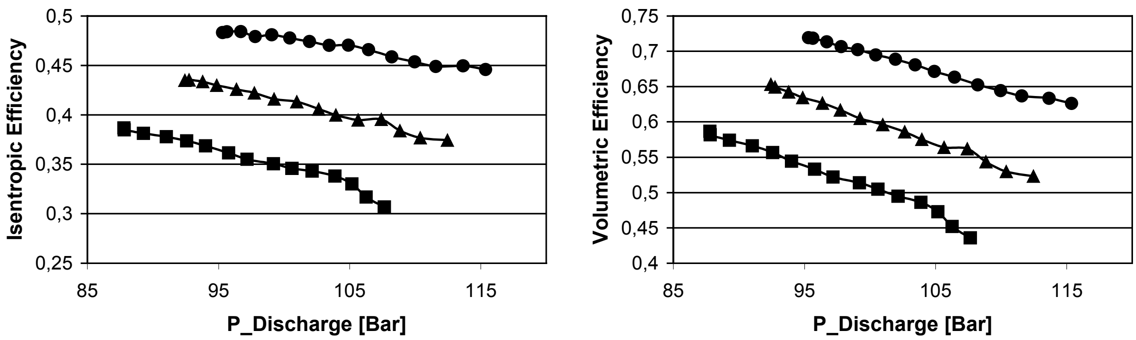


Figure 3: Impact of cylinder leakage on the isentropic and volumetric compressor efficiency
■ no piston ring and minimum gap; ▲ one piston ring; ● two piston rings

# 4. THE COMPRESSOR CONCEPT

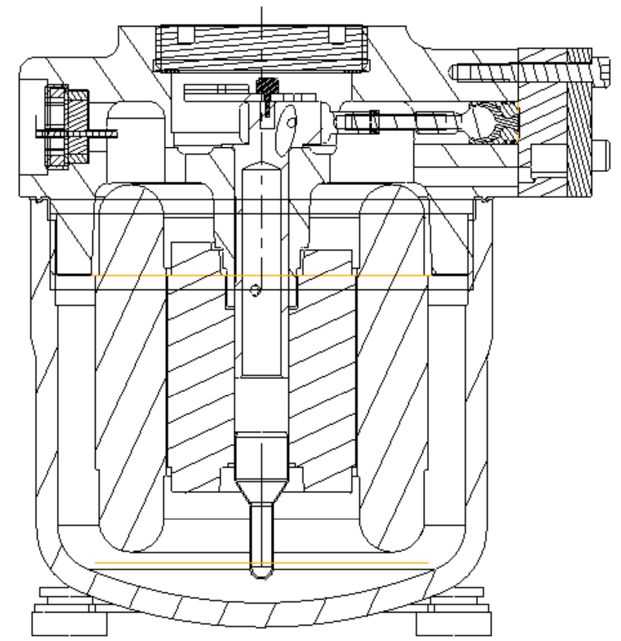

Figure 4: $CO_2$ compressor concept

The compressor, as being developed up to step 3 is shown in *Figure 4*. It is a more compact design compared to the first prototype as shown in *Figure 1*. The drive mechanism, the piston and cylinder arrangements as well as the valves are basically kept unchanged from the earlier compressor versions /3/ reusing the experiences regarding lifetime and reliability. Various strokes in 3 different models presently offer a capacity range from 400 W to 1200 W refrigeration at -10 ºC evaporation and 32 ºC compressor return and high side heat exchanger outlet temperatures.

As shown in *Figure 4* the present compressor is of a semi-hermetic type. A large thread connects the compressor block and the shell. The power feed through and the lid on top of the compressor are also hold to the block by applying a thread Furthermore, the valve plate and its cap are bolted to the block.

Apart from this, the overall compressor design and especially the drive and lubrication concept correspond to the design of traditional HFC/HC hermetic type compressors for light commercial application. A standard single-phase motor, as today used in HFC/HC hermetic type compressor, is applied. Motors for different voltage and frequency are available.

As mentioned before, suction gas heating is besides cylinder leakage a major source of energetic and volumetric losses of the CO2 compression process /2/. Not considering valve losses, the pressure drop in the suction chamber is negligible due to the high suction pressure and therefore hardly contributes to volumetric process losses. On the other hand, heat transfer resulting in suction gas heating, may reduce the energetic and volumetric efficiency by 10-20 %, especially when the temperature difference between the suction gas and the surrounding is large. Therefore, they have to be considered and heat transfer needs to be minimized.

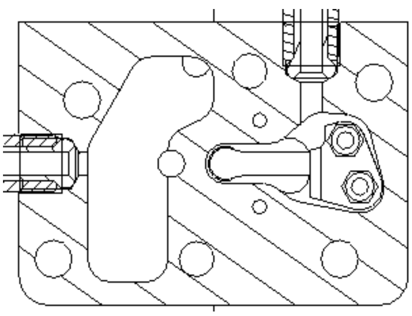

Figure 5: Suction and discharge plenum for the TN compressor.

In the applied valve plate design, the suction gas spends as little time as possible in the suction plenum before entering into the cylinder. This was implemented as shown in *Figure 5*. The suction gas enters directly a chamber with a volume of only a few times the cylinder volume. Oil returning from the process gets separated and drained to the compressor housing through a connection in the bottom of the suction plenum. The housing volume, which contains suction gas, acts as a buffer. When the suction valve opens, only the gas close to the suction hole gets sucked into the cylinder. During this process the pressure in the suction chamber decreases and flows to the chamber from the suction line and from the compressor shell through the connection line. During this a significant amount of gas is sucked directly through the cold suction line into the cylinder. By the end of the suction process, the refilling of the suction chamber is still in progress, but now with colder gas out off the suction line. Since the housing is a fixed volume with no other connections, a backflow occurs into the shell - draining separated oil form the suction chamber back to the oil sump.

The same concept is chosen for the discharge chamber as the gas gets directly guided towards the discharge line passing only a small volume. Depending on the system design, this may result in pressure pulsations, which will propagate into the discharge line and further on to the high-pressure side of the system. Any noise implication due to these pulsations is, however, quite small due to the robust nature of the compressor and the system tubing.

## 5. PERFORMANCE EXPERIENCES

So far, more than a hundred TN compressors were manufactured and various independent performance tests have been undertaken. Furthermore, the compressors were integrated in various applications mainly in beverage coolers, vending machines and heat pumps with satisfactory results. In addition, a calorimetric test rig was set up to investigate the performance of the compressor. Due to the high miscibility of the applied oil and CO2, the oil circulation rate (OCR) is hard to quantify. In order to reduce the impact of oil on the energy balance, measurements were conducted in this way that the outlet temperature of the evaporator was equal to the expansion valve inlet temperature. In this way oil has a very limited influence on the total energy of the evaporator.

The test rig, which was set up, is shown schematically in *Figure 6*. The gas cooler is of a counter flow water-cooled type. The expansion valve (J17) of the process automatically controls the discharge pressure to a given value, set by the data acquisition software. The evaporator is electrically heated and the heater is controlled to obtain a right superheat value. Additionally, a thermal bath after the oil separator arrangement sets the suction temperature. As indicated in *Figure 6*, various temperatures and pressures are recorded. The compressor power consumption and the electrical power input of the evaporator are also measured. Furthermore, on the suction line a set of flow meters are installed to indicate the refrigerant mass flow. Between the two flow meters an oil separator is installed to allow a flow measurement with and without oil. This allows quantifying the OCR.

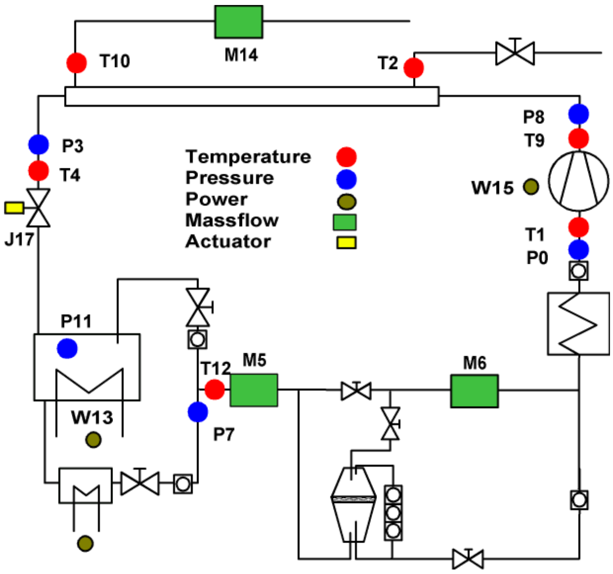


Figure 6: Calorimetric test rig

The isentropic and the volumetric efficiency of the present compressor design version as shown in *Figure 4* is given in *Figure 7*. The isentropic efficiency is defined as the ratio of an isentropic compression from certain suction conditions to a certain high pressure related to the measured electric power input to the compressor at the same working point. The volumetric efficiency indicates the ratio of the compressed mass related to the theoretical mass through the compressor, which is calculated from the compressor's stroke volume and the current frequency at the certain running condition. All data were recorded at 230V and a frequency of 50 Hz. The data refer to the biggest compressor presently available with 1200 W refrigeration at -10 ºC / 32 ºC / 32 ºC.

The plotted data range covers a suction pressure ranging from 20 to 50 bars and a discharge pressure ranging from 50 to 110 bars. All shown performance data is based on mass-flow measurements and confirmed by mass-flow calculations from energy balances of the high- and low side heat exchangers. The isentropic efficiency increases with the suction pressure and reaches maximum values of around 60 %. The volumetric efficiency reaches especially at small compression ratios also rather high values of up to 90 %. This high efficiency is possible, as suction gas heating inside the compressor is minimized. The drop of volumetric efficiency with increasing compression ratio is mainly due to re-expansion from the clearance volume of the cylinder. Leakage - which could also cause the drop in volumetric efficiency, is only affecting the compression process at a negligible degree, as the isentropic efficiency is hardly depending on the compression ratio. The minor efficiency change is rather due to valve losses. Furthermore, the compressor performance is not strongly depending on the suction gas temperature, but a slight efficiency increase with rising temperature was noticed. The presented data were taken at an OCR of about 5 mass %, which seems to be typical for the compressor applied in light commercial application without the use of an oil separator.

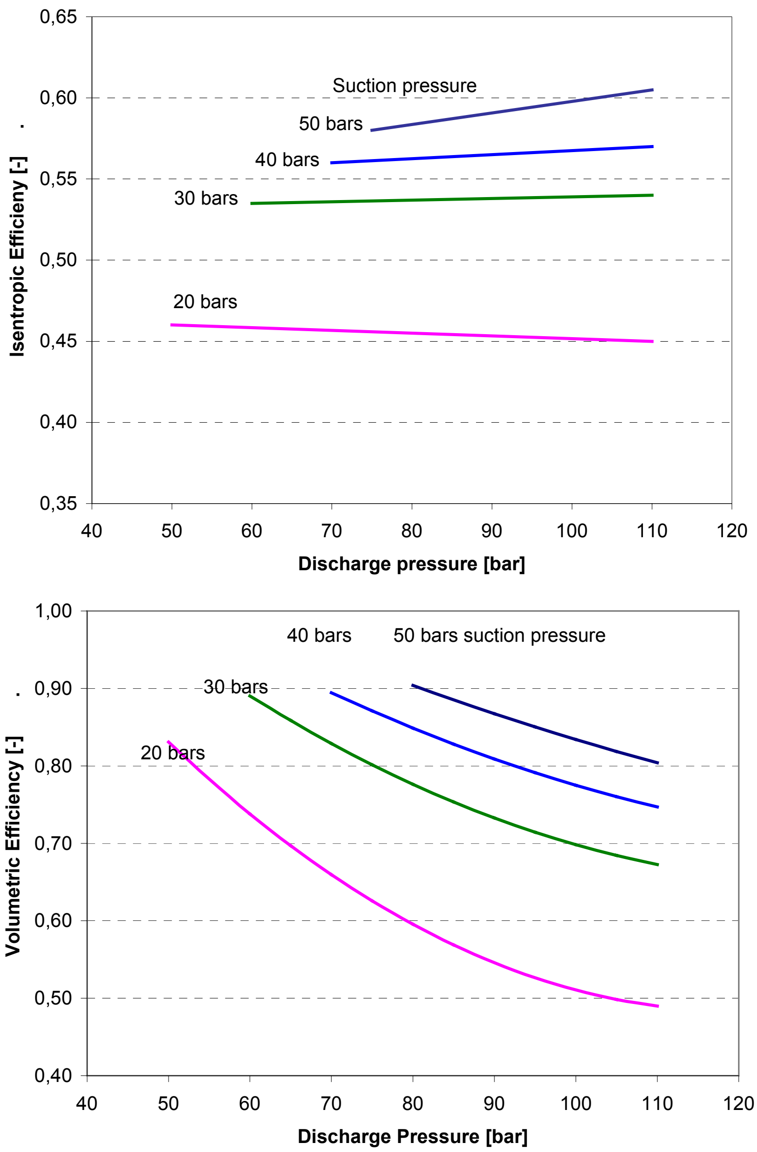


Figure 7: Isentropic and volumetric efficiency present compressor design version

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