Efficiency of the Indicated Process of CO_2 – Compressors[1]

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Abstract

The compressor of a refrigerant compression process is the component with the major influence on the efficiency and reliability of the entire system. Due to the fluid properties of carbon dioxide (CO_2), the pressure ratio of the refrigeration process with CO_2 as the working fluic0is in relation to common refrigeration processes rather low while the pressure difference is extremely high. From experimental and theoretical considerations it gets obvious that at these conditions a high volumetric and energetic efficiency of the compressor may be achieved if its design is appropriate. In the paper the effects on the efficiency of the indicated compression process of a CO_2-compressor are discussed and evaluated and a promising design concept for an efficient CO_2-compressor was derived.

Keywords: CO_2; compressor, compression process

CO_2 as a refrigerant

International regulations prescribe that CFCs should no longer be used as refrigerants in industrialised countries and also HCFCs seem to be an interim solution until the year 2020 world-wide while some national regulations prescribe even earlier phase out dates[1,2]. Looking for a final solution and taking furthermore regulations for greenhouse gases into account Gustav Lorentzen favoured substances with a negligible global warming potential (GWP) namely natural fluids as a promising alternative for the refrigeration technology[3]. Some of these refrigerants like the hydrocarbons and ammonia show an unfavourable safety behaviour. If intoxicity and non-flammability of the refrigerant are required Lorentzen focused on CO_2, which is close to being the ideal working medium, provided that a process to give competitive energy performance can be designed.

[1] *Delicated to the memory of Gustav Lorentzen*

Applications of CO_2 as a refrigerant in the field of one stage refrigeration technology can mainly be seen in automotive air conditioning and heat pump applications[4]. Expecting an installation rate of automotive air conditioning systems of about 50% for the coming years in Europe four to five million units are to be produced. CO_2 heat pumps may substitute water heaters and old central heating units for domestic heating systems with radiators with high temperatures in the hydronic system[5,6]. High temperatures can be achieved by a CO_2 heat pumping system as its working process is transcritical which means that the heat input is at a sub critical pressure in the two phase region while the heat output of the process proceeds at a supercritical pressure. The supercritical heat rejection results in a large temperature glide with energetic advantages for this application. The theoretical CO_2 process in a *T,s*-diagram with the isentropic compression (1$\longrightarrow$ 2), the isobaric heat output (2$\longrightarrow$ 3), the isenthalpic throttling (3$\longrightarrow$ 4) and the isobaric heat input (4$\longrightarrow$ 1) is shown in *Figure* 1. The

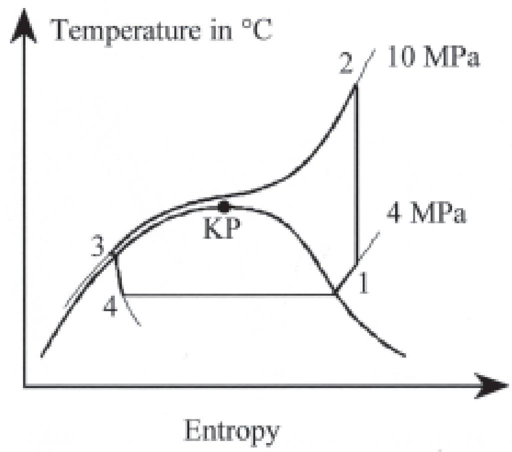

Figure 1 Theoretical CO_2 - process in a T,s-diagram

high working pressures of the process make exceptional demands on the lay-out of the compressor, while the design of the other plant components is less critical.

Indicated compression process

A compressor is designed to pump gas with the pressure p_{in} and the temperature T_{in} to the discharge pressure pout at the temperature T_{out}. For this process the indicated compression power P_i is required by the compressor. Due to throttling losses of the valves and the gas-chambers, leakage and heat transfer effects the power P_i exceeds the isentropic compression power P_t. The indicated compression process as well as the theoretical compression process are shown in *Figure* 2.

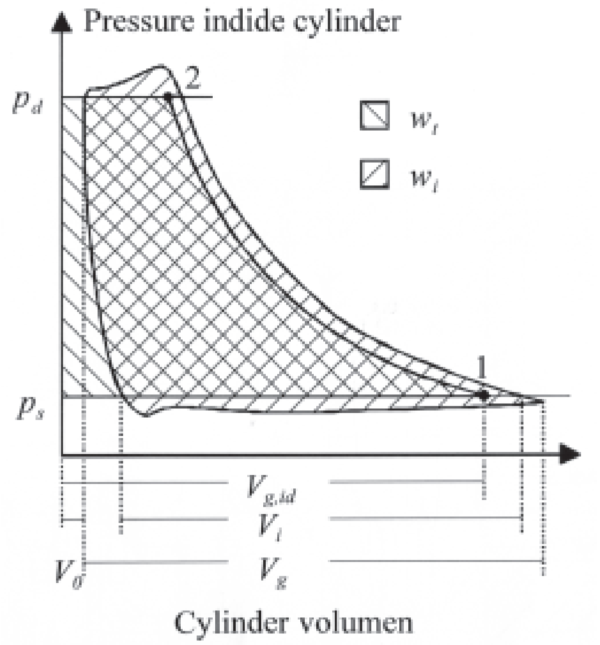

Figure 2 Theoretical and real indicated compression process

The indicated efficiency from the diagram $\eta_i = w_t / w_i$ is calculated with the theoretical work w_t that is required for the isentropic compression related to the indicated work w_i that is needed by the real compressor. The indicated efficiency is strongly influenced by valve losses. As the difference between the suction and the delivery pressure is extremely high when using CO_2, the pressure differences that are necessary to overcome the flow resistance inside the compressor may be negligible at an appropriate design. Therefore, relatively small pressure losses and consequently a high indicated efficiency must be achievable.

The volumetric efficiency $\lambda = V_i / V_g . \eta_{th}$ is calculated with the cylinder volume V_i that is actually filled with the suction gas at the end of the sucking, the geometric swept volume V_g and the thermometric efficiency h_{th}. The volumetric efficiency of the CO_2 compressor gains mainly from the small pressure ratio as it brings about a short re-expansion of the gas from the clearance volume of the cylinder and therefore an early opening of the suction valve. Blow-by and leakage of the valves as well as heat transfer phenomena between the gas and the cylinder are represented by η_{th}. Due to the thermodynamic properties of CO_2 a reconsideration of the influences of leakage of heat transfer phenomena on the process is advisable.

Characteristics of the Investigated Compressor Prototype

Schematic drawings of two investigated CO_2 compressors are shown in *Figure* 3 and *Figure* 4.

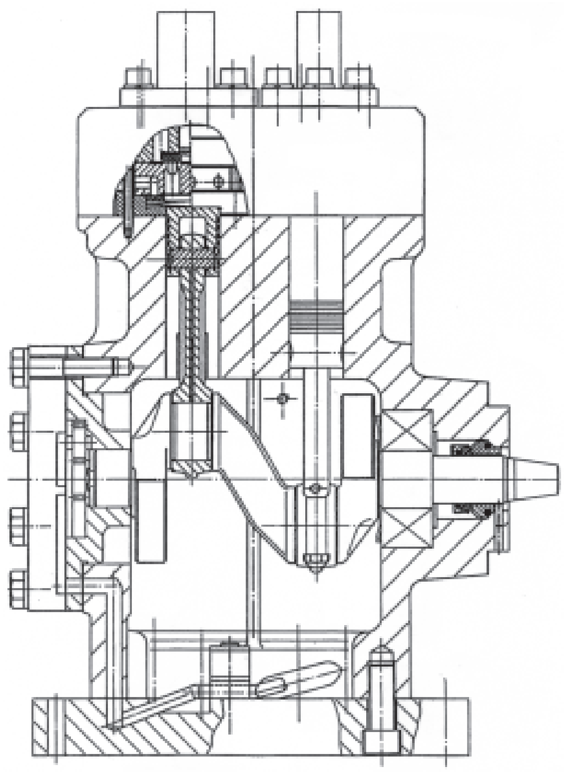

Figure 3 Prototype of a open type reciprocation compressor produced by Bock Kältlemaschinenfabrik GmbH, Germany

Figure 3 shows the prototype of an open type reciprocation compressor designed for bus air conditioning produced by Bock Kältemaschinenfabrik GmbH, Germany. The design of the oil lubricated machine was derived from a produced series of refrigeration compressors by reducing the piston diameter so far that the load on the compressor's crankshaft was kept constant while the cylinder pressures were higher when using CO_2. This method was selected to limit the designing and production efforts of this prototype. As the crankshaft itself was not modified, the stroke of the compressor was constant which resulted in a rather large stroke to bore ratio of about 1.7. To seal the cylinder from the crankcase four piston rings per piston are supplied[7].

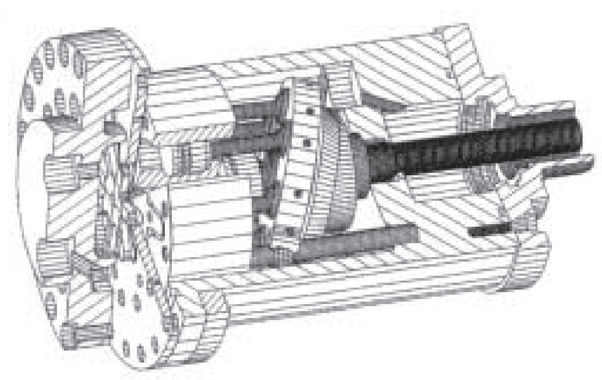

Figure 4 Wobble plate compressor test model no. 1 produced by Danfoss A/S, DK

In *Figure 4* the schematic drawing of test model no. 1 produced by Danfoss Nordborg A/S, Denmark is shown. The compressor of the wobble plate type was designed for car air conditioning. Capacity control by adjusting the wobble plate angle is possible[8].

Theoretical investigations of the influences on the compression process

To investigate the influences on the efficiency of the indicated compression process, a simulation program of the indicated compression process for CO_2 compressors was established. The basic equations and the program structure are explained in the following.

From the energy balance at the cylinder of a positive displacement compressor Röttger derived an equation for the temperature gradient of the gas inside the cylinder over the crank angle[9].

$$\frac{dT}{d\varphi}=\frac{\frac{dQ}{d\varphi}-\left(\left(\frac{\partial u}{\partial v}\right)_T+p\right)\cdot\frac{dV_{cyl}}{d\varphi}+\left(v\cdot\left(\left(\frac{\partial u}{\partial v}\right)_T+p\right)+h_V-h\right)\cdot\left(\frac{dm}{d\varphi}\right)_V}{m\cdot\left(\frac{\partial u}{\partial T}\right)_v} \qquad (1)$$

This equation pays respect to heat transfer phenomena between the gas and the cylinder walls, the compression and re-expansion of the gas as a function of the cylinder volume and the mass flow through the valves of the cylinder. The mass flow through the valves is a function of the actual valve position and geometry as well as the gas velocity depending on the fluid properties and the pressure difference.

Besides the mentioned factors, leakage through the gaps of the cylinder has an influence on the temperature gradient, which has to be added to equation (1) and is calculated as

$$\left(\frac{dT}{d\varphi}\right)_L=\frac{\left(v\cdot\left(\left(\frac{\partial u}{\partial v}\right)_T+p\right)+h_L-h\right)\cdot\left(\frac{dm}{d\varphi}\right)_L}{m\cdot\left(\frac{\partial u}{\partial T}\right)_v}. \qquad (2)$$

The leakage mass flow through the gaps of the cylinder is calculated with the following equation[10].

$$\left(\frac{dm}{dt}\right)_L=\left(\frac{2^{1+m_0}\cdot U^{2\text{-}m_0}\cdot h^3\left(\Delta p\right)\cdot\left(p_1^2-p_2^2\right)}{W\cdot\eta^{m_0}\cdot p_1\cdot v_1\cdot l}\right)^{\frac{1}{2\text{-}m_0}} \qquad (3)$$

With these equations the temperature course of the gas during the compression process of any compressor of the positive displacement type with a given geometry may be calculated in a numerical way.

To define the condition of the gas inside the cylinder, a second property for the one-state fluid is necessary. From the geometric cylinder volume and the mass inside the cylinder that can be calculated from the mass flow through the valves and the leakage mass flow, the specific volume of the gas inside the cylinder is known throughout the process. With the temperature and the specific volume an equation of state is employed to calculate all relevant properties.

By a variation of the simulation parameters it is possible to investigate the influences on the efficiency of the indicated compression process in a detailed manner. To obtain reliable results an adaptation of the simulation model to experimental data is required.

Experimental investigations of the influences on the compression process

As stated before, the efficiency of the indicated compression process is influenced by pressure losses of the valves and the gas chambers, leakage of the cylinder, and heat transfer phenomena. To individually assess the influence of these factors on the efficiency of the indicated process, a number of experimental investigations with CO_2-compressors at various running conditions were performed.

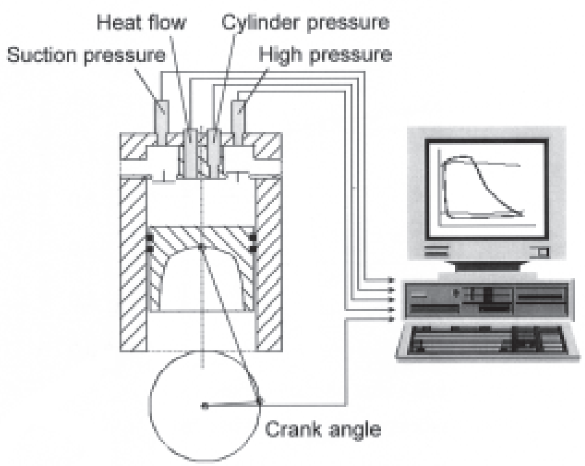

Figure 5 Installation of the pressure and heat flow sensors to investigate the indicated compression process

The most relevant way to estimate the efficiency of the compression process is to measure the dynamic pressure inside the cylinder as shown in *Figure 5*. From the indicated diagrams the pressure losses inside the compressor can be evaluated in detailed manner. To differentiate between the pressure losses at the valves and inside the gas chambers further pressure transducers were installed in the suction and the discharge chamber in the vicinity of the relevant valve. The valve losses were calculated from the pressure difference which was measured by the pressure transducer inside this cylinder and the corresponding pressure transducers in the gas chambers. The pressure losses inside the gas chambers were evaluated from the pressure difference between the indicated pressure inside the chamber and the nominal pressure of the test rig during the investigation.

To estimate the volumetric efficiency according to *Figure 5* additional experiments were performed to directly measure the heat transfer phenomena with a special sensor[11,12]. Furthermore, the leakagemass flow through the gap between piston and cylinder and at the cylinder valves was measured[13]. With the results of these investigations is possible to quote the thermometric efficiency and by knowing its value to determine the volumetric and indicated efficiency of the compression process.

Influences on the performance of the compression process

Pressure losses inside the compressor

Pressure losses inside the compressor influence the efficiency of the indicated compression process. Therefore, the influence of the pressure losses occurring at the valves and inside the gas chambers were evaluated. In *Figure 6* the suction valve losses in percent of the indicated compression work are exemplary shown for the reciprocating compressor as a function of the revolutions per minute and the pressure ratio. In addition to the measured values, the simulated suction valve losses are also plotted. Corresponding data are available for the discharge valve of the compressor.

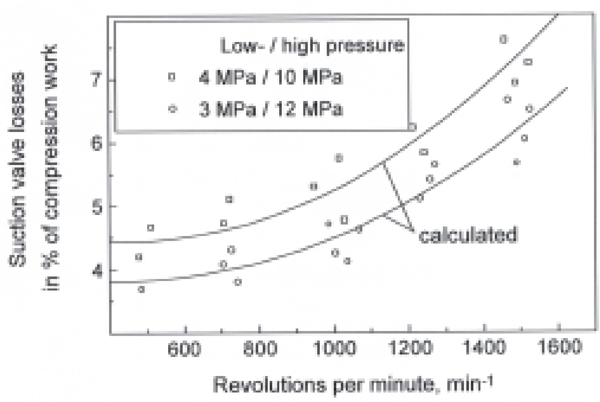

Figure 6 Suction valve losses for the reciprocating compressor

After having adopted the simulation model to the valve design of the reciprocating compressor, the programme was applied to theoretically investigate the influence of the flow area of the valves and the corresponding pressure losses on the efficiency of the indicated compression process. As the parameter of the simulation the coefficient of the flow area of the suction valve related to the geometric cylinder volume was defined and related to a value covering the range of common compressor valve designs. Leakage of the cylinder or heat transfer effects were not taken in consideration for this simulation. The basic simulation parameters were 1,000 rpm of the compressor, a suction pressure of 4 MPa, which corresponds to an evaporation temperature of approximately 5^{O}C and a supercritical high pressure of 12 MPa.

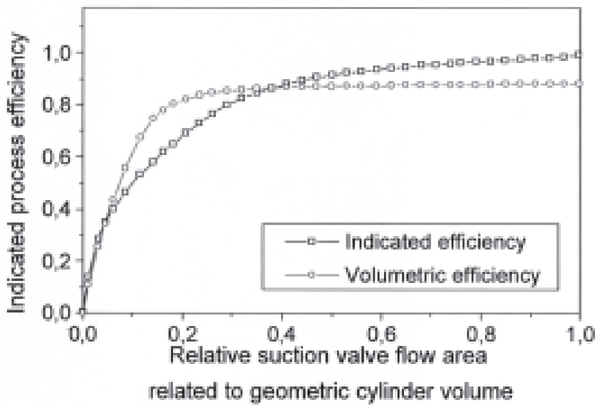

Figure 7 Simulation of the indicated and volumetric efficiency of the process as a function of the valve flow areas

The results of the simulation of the indicated and volumetric efficiency of the process are shown in *Figure 7*. Due to the definition the investigated reciprocating shown in Figure 3 compressor had a ratio of 0.3 which is due to the large stroke-to-bore ratio of the machine rather small. The investigated wobble plate compressor shown in Figure 4 and other common designs of piston compressors have due to their stroke to bore ratio a value of the defined ratio of about 0.6. From *Figure 7* it gets obvious that the indicated and volumetric efficiency of the compression process is not very sensible towards pressure losses of the valves, as even at rather small parameter values a good process performance is achievable.

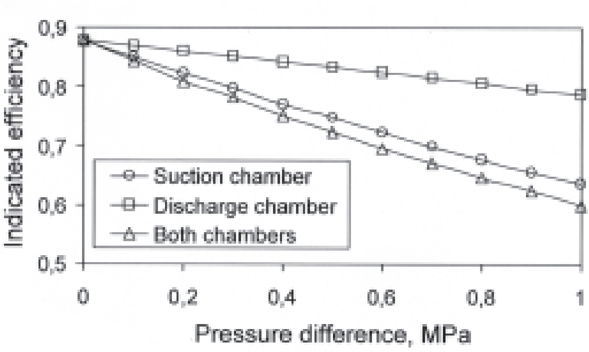

Figure 8 Simulation of the indicated efficiency of the process as a function of the gas-chamber pressure losses

A corresponding result is obtained from the theoretical investigation of the influence of pressure losses inside the gas chambers on the efficiency of the indicated process. In *Figure 8*, the indicated efficiency of the process is shown when pressure losses inside the gas chambers up to 1 MPa are assumed. From *Figure 8* it gets obvious that pressure losses on the suction side have a larger influence on the process performance than the losses on the discharge side have. This is explained by two phenomena. Firstly, the increase of the indicated compression work by a reduction of the suction pressure is larger than that resulting from a comparable increase of the high pressure on the discharge side. Secondly, a reduction of the suction pressure leads to an increase of the specific volume of the suction gas, reducing the volumetric efficiency and consequently the indicated efficiency of the compression process.

Nevertheless, the investigations have shown that pressure losses inside CO_2-compressors have by far a lower influence on the efficiency of the indicated compression process than is the case when applying other refrigerants leading to processes at lower pressures. Therefore, the influence of pressure losses on the efficiency of the CO_2 compression process is even without special precautions not substantial.

<u>*Leakage of the Cylinder*</u>

As a function of the pressure inside the cylinder and the suction or the discharge pressure, the gas leaks in either direction through the gaps at the cylinder valves and the gap between the piston and the cylinder. Results of former investigations of compressors and combustion engines have shown that leakage of the cylinder has only a negligible influence on the pressure and temperature at the end of the compression, but for small compressors working at high process pressures and large pressure differences a reconsideration of this statement is recommended.

The leakage mass flow through a gap is calculated with equation (3). Looking at this equations it gets obviously that the influence of leakage on the indicated compression process efficiency must not be neglected in general, as the leakage mass flow is a function of the difference between the squares of the pressures at each side of the gap. This difference gets high when applying CO_2 as the refrigerant. From detailed investigations of the influence of leakage on the efficiency of the indicated process of CO_2 compressors, it became obvious that it is possible to reduce its harmful influence on the efficiency with an appropriate design of the machine to a negligible value[13]. Due to the dimensions of the gap between the piston and the cylinder its negative influence on the performance of the compression process exceeds the effect of the gap at the cylinder valves.

To extend the considerations to a more general level, the leakage factor G_{VL} corresponding to the equation (3) was defined to describe the geometry of the gap between the piston and the cylinder in relation to the cylinder volume allowing a comparison of the suitability of various compressor types from an application in the CO_2-process.

$$G_{VL} = \frac{U \cdot h^3}{V \cdot l} \qquad (4)$$

Simulating the indicated process starting from a suction pressure of 4 MPa and a suction gas temperature of 25°C as a function of the defined coefficient, the high pressure that was achieved with the process at 1,000 rpm was recorded. Due to the application range of the CO_2 process, the maximum pressure was limited to 15 MPa. The simulated maximum pressure as a function of the leakage factor is shown in *Figure 9*.

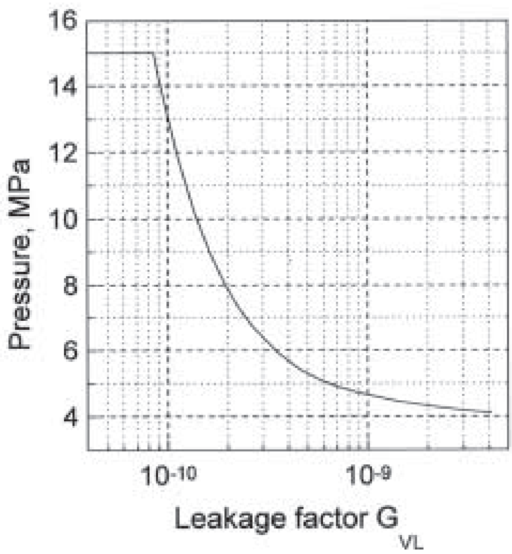

Figure 9 Simulated high pressure as a function of the defined leakage coefficient; suction gas condition: 4 MPa and 25°C, compressor 1,000 rpm

In *Figure 10* the leakage factors for various compressor types are given. The leakage factor for each type has been calculated under the assumption that a certain volume flow is required. According to the volume flow, the swept volume of the compressor was fixed by transferring geometric parameters of available refrigeration compressors. Regarding the leakage factors of rotating piston compressors it gets obvious, that these types of machines due to leakage are not suitable for the considered application as the process performance does not match the demands. Experimental investigations with a rotary vane compressor - originally designed for hydraulic equipment - have supported this statement rather impressively.

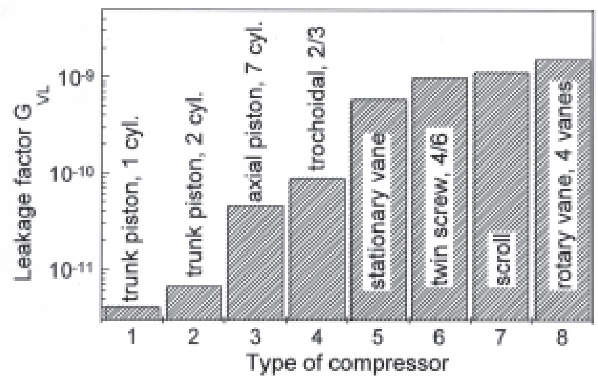

Figure 10 Leakage factors G_{VL} for the gap between the piston and the cylinder for different types of compressors

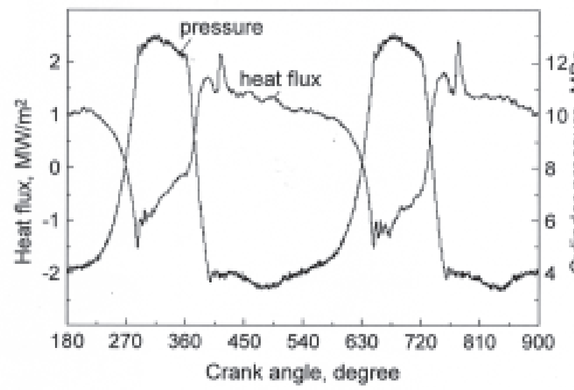

Figure 11 Measured heat flux and cylinder pressure over crank angle

Consequently, leakage of the cylinder of CO_2-compressors may have a rather strong influence on the performance of the indicated process, but with an appropriate design of the machine it is possible to reduce the influence of leakage on the performance to a reasonable value.

Heat Transfer between the Gas and the Cylinder

Apart from pressure losses and leakage, heat transfer phenomena between the gas and the cylinder walls have an influence on the performance of the compression process. To investigate the influence of heat transfer phenomena inside the cylinder on the efficiency of the indicated process the heat being exchanged between the gas and the cylinder wall in the vicinity of the sensor was measured. The results of this measurement are shown in *Figure 11*. Besides the heat flux, the pressure inside the cylinder was recorded.

The influence of heat transfer on the indicated efficiency of the compression process was theoretically investigated by simulating the process with and without the consideration of the measured heat flux.

In *Figure 12*, the indicated and volumetric efficiencies of the process are shown, when the heat flux is multiplied by a constant factor up to a value of 50. From the simulation it is evident, that heat transfer effects inside the cylinder have only a negligible influence on the performance of the process although the local heat transfer coefficient reaches a maximum

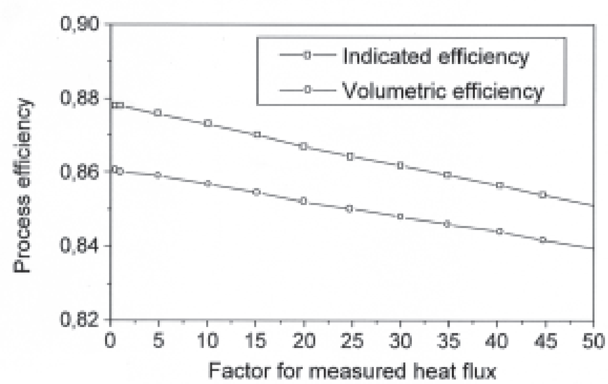

Figure 12 Simulated process efficiency at different factors for the measured heat flux

value of more than 35 kW/m^{2}K during the process. The average value of the heat transfer coefficient during the compression process is about 14 kW/m^{2}K.

Consequently, the geometry of the cylinder, namely the ratio between the cylinder surface and the cylinder volume has no major influence on the effect of heat transfer phenomena inside the cylinder and with that of the performance of the process. Therefore, no limits regarding the design of the working chamber of a CO_2-compressor must be respected due to heat transfer phenomena inside the cylinder /13/.

Design of a efficient CO_2-compressor

From the individual investigations of the parameters influencing the performance of the indicated process of CO_2-compressors, leakage of the cylinder has been identified to have the major effect. Therefore, it is essential to minimise the length of the leakage gaps and to apply an efficient sealing concept. It is noted that the lowest leakage rates are achieved by applying oil lubricated machines with seals sliding along the cylinder wall, e.g. piston rings. Complying this fact, rotating piston compressors are not a promising option for the application in CO_2 vapour compression processes as oil is used to seal the working chamber. An exception to this statement are trochoidal type compressors which have an efficient but rather long seal of the working chamber. With that the focus is on reciprocating piston type compressors, mainly trunk-and axial piston machines. To minimise the leakage mass flow the length of the seal must be kept short, pushing the concept towards a machine with a rather long stroke-to-bore ratio. The disadvantage of this concept is the little space left to apply valves with a sufficient flow area. On the other hand it had been shown, that the pressure losses inside a CO_2-compressor have a rather small influence on the energetic and volumetric performance of the compression process.

When simulating different CO_2 compression processes at 1,000 rpm and, a suction pressure of 4 MPa and a high pressure of 12 MPa for different stroke-to-bore ratios of the cylinder, the energetic and volumetric efficiencies of the indicated process shown in *Figure 13* are calculated. The geometric parameters of the compressor required for the simulation are corresponding to the data of the investigated reciprocating compressor.

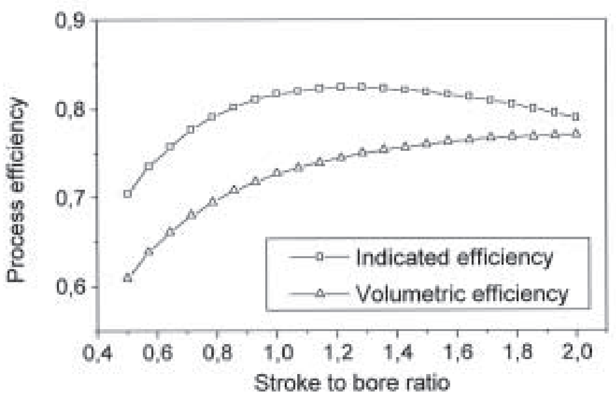

Figure 13 Simulated process efficiency as a function of the stroke to bore ratio

While the volumetric efficiency is due to the reduction of leakage at the piston gap increasing with the stroke-to-bore ratio, the efficiency of the process has a maximum for a stroke-to-bore ratio of about 1.3. This is explainable by the eclipsing of the valve losses and leakage. While the energetic losses at the valve are increasing with the stroke-to-bore ratio the influence of leakage decreases.

Conclusions

Due to the thermophysical properties of CO_2 the working pressures are rather high and require a special design of the system's components. On the other hand the application of CO_2 as the refrigerant offers various advantages for the performance of the refrigeration process. Especially for the compressor, the fluid properties for CO_2 are favourable for its energetic and volumetric efficiency as long as the influence of leakage can be reduced to a reasonable amount. Paying respect by an appropriate design of the machine, a high efficiency of the indicated compression process is achievable. Due to the results of the investigations there are from a technical point of view no serious limits for the CO_2 refrigeration technology to become a favourable alternative for certain future applications of refrigeration systems.

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Nomenclature

h	Enthalpy
h	Height of gap
l	Length of gap
m	Mass
m_0	Exponent of Reynolds equation
p	Pressure
Q	Heat flow
T	Kelvin temperature of high pressure fluid
u	Inner energy of fluid
U	Circumference of gap
V	Volume
v	Specific volume of fluid
W	Drag number
η	Dynamic viscosity of fluid
φ	Crank angle