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THEORETICAL AND EXPERIMENTAL INVESTIGATIONS ON CARBON DIOXIDE COMPRESSORS FOR MOBILE AIR CONDITIONING SYSTEMS AND TRANSPORT REFRIGERATION

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1. ABSTRACT

In this paper different types of CO_2 compressors for mobile air conditioning systems, transport refrigeration and heat pump applications are presented and analyzed: a pressure controlled reciprocating compressor, a pressure controlled swash-plate compressor and a path controlled swash-plate compressor. The investigations are based on both experimental data of the compressor prototypes and computational simulation.

2. NOMENCLATURE

f	compressor speed	[min-1]
h	specific enthalpy	[kJ/kg]
$\vec{M}$	torque	[N m]
$\dot{m}$	mass flow rate	[kg/s]
P	power	[W]
p	pressure	[bar]
T	temperature	K
$\dot{V}$	volumetric flow rate	[m³/s]
η	efficiency	[-]
λ	volumetric efficiency	[-]
ρ	density	[-]
π	pressure ratio	[-]

3. INTRODUCTION

In the last few years a renaissance of supercritical vapor compression cycles (so called transcritical cycles) using carbon dioxide as a refrigerant took place. One important part of current investigations is the development of new compressors for carbon dioxide applications. A variety of compressors types have been developed and tested like reciprocating compressors (Kaiser 1999, Fagerli 1997, Süß 1998, Nekså 2000), rolling piston compressors (Tadano 2000), scroll compressors (Hasegawa 2000), wobble-plate and swash-plate compressors (Parsch 2002). In order to achieve high compressor efficiencies measurements at test rigs and comparative studies are of great importance to optimize the compressor efficiencies.

In this connection this paper presents experimental data and theoretical results of different compressor types developed for mobile applications: two types of swash-plate compressors with a pressure controlled (type A) and a path controlled (type B) valve system and a reciprocating compressor (type C) with a pressure controlled valve system. The prototype compressors were tested on a test rig under conditions expected in air-conditioning, refrigeration, and heat pump applications. The test rig consists of a gas cycle with hot gas heat exchange and compressor drive and is equipped with pressure transducers in the cylinder, the discharge, and the suction chamber measuring transient quantities. To generate indicator diagrams the rotation angle is measured at the crankshaft.

For the numerical simulation an object-oriented simulation platform was used to solve the basic compressor and valve equations to calculate indicator diagrams and theoretical compressor efficiencies.

4. COMPRESSOR PROTOTYPE DESIGN

Compressor type A

This prototype compressor was constructed by Obrist Engineering GmbH in Austria and is designed for CO_2 air conditioning systems in small buses and transport refrigeration in small trucks and vans. It is a swash-plate compressor with a pressure controlled valve system. The axial piston unit consists of seven cylinders alternatively with fixed or variable displacement. The maximum displacement is 45cc. The displacement of the tested compressor prototype was fixed to a reduced value of 33.5cc. One of the seven cylinders as well as the suction and discharge chamber are equipped with a pressure transducer to generate indicator diagrams.

Compressor type B

This compressor is the first generation of a path controlled axial swash-plate compressor, derived from a serial produced hydraulic pump for mobile applications. In contrast to common swash-plate compressors the nine cylinders rotate with the pistons (Figure 1). The swash-plate is fixed with respect to the rotation of the shaft and the cylinders. Its possible to vary the displacement by adjusting the swash-plate angle. The displacement of the test compressor was fixed at a value of 105cc, the maximum stroke. As the cylinders rotate the cylinders are pressed by the exerted gas forces against the fixed port plate. A movement from one dead center to the next constitutes one complete stroke during which a volume of gas, corresponding to the piston area and the stroke, is either sucked in or discharged via the suction or discharge area in the port plate. To achieve high efficiencies its important to design a port plate according the specific application. The design point of the port plate areas depends on the expected pressure ratio. The rotating cylinder chamber made it impossible to mount pressure transducers in the cylinder chamber at least this project. Therefore only theoretical indicator diagrams are presented.

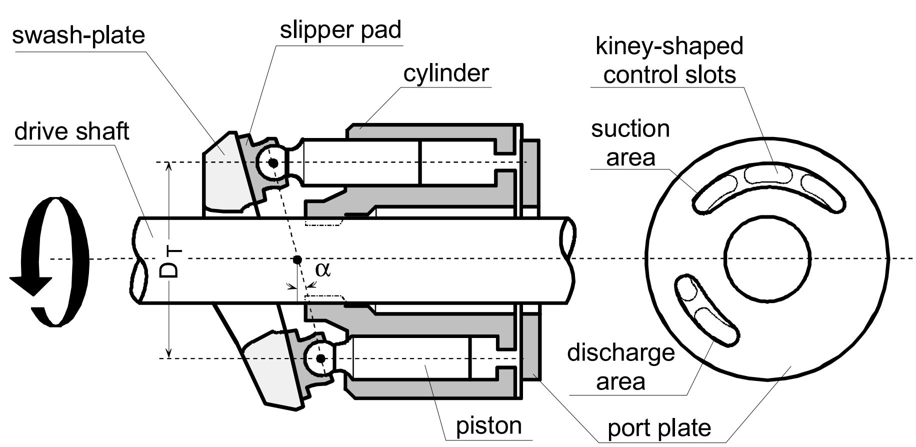


Figure 1: Schematic drawing of the path controlled swash-plate compressor.

Compressor type C

This compressor prototype is the second generation of an open two cylinder oil lubricated reciprocating compressor with a pressure controlled valve system, derived from a serial produced compressor for mobile air conditioning (Kaiser 1999, Försterling 1999). The originally series compressor was designed for R134a as refrigerant. Due to the high volumetric refrigeration capacity of carbon dioxide the displacement was reduced from originally about 116cc to 30cc per cylinder by reducing the piston diameter from 55mm to 28mm. The crankshaft itself and the stroke of the compressor were not altered. Therefore, the stroke-to-bore ratio of 1.71 is rather high leaving only little space to install cylinder valves.

5. CARBON DIOXIDE COMPRESSOR TEST RIG

The prototype compressors type A and type C where investigated in a test rig consisting of a gas cycle and driving machine. To achieve constant suction and discharge conditions at the compressor inlet and outlet the operating conditions of the gas cycle are controlled as shown in Figure 2. The cycle contains a water cooled gas cooler, a manually operated expansion valve and an electrically heated heat exchanger. Most important purpose of this heat exchanger is the control of the superheat at the suction inlet. The oil separator with oil return cycle was used only for the swash-plate compressor type A. The path controlled swash-plate compressor type B was tested with a similar test rig in the test lab of the compressor manufacturer.

The test rig is equipped with sensors for temperature, pressure, mass flow, shaft torque and compressor speed, which are applied to measure the mean quantities. These sensors measure the mean gas inlet and the gas outlet conditions at the compressor and also the refrigerant mass flow rate. The compressor prototype itself is equipped with pressure transducers in the cylinder, the discharge chamber, and the suction pipe measuring transient quantities. To generate the indicator diagram the rotation angle is measured at the crankshaft.

The top dead center is determined with a dial gauge under static conditions. The rotational speed of the driving machine can be adjusted by a frequency converter.

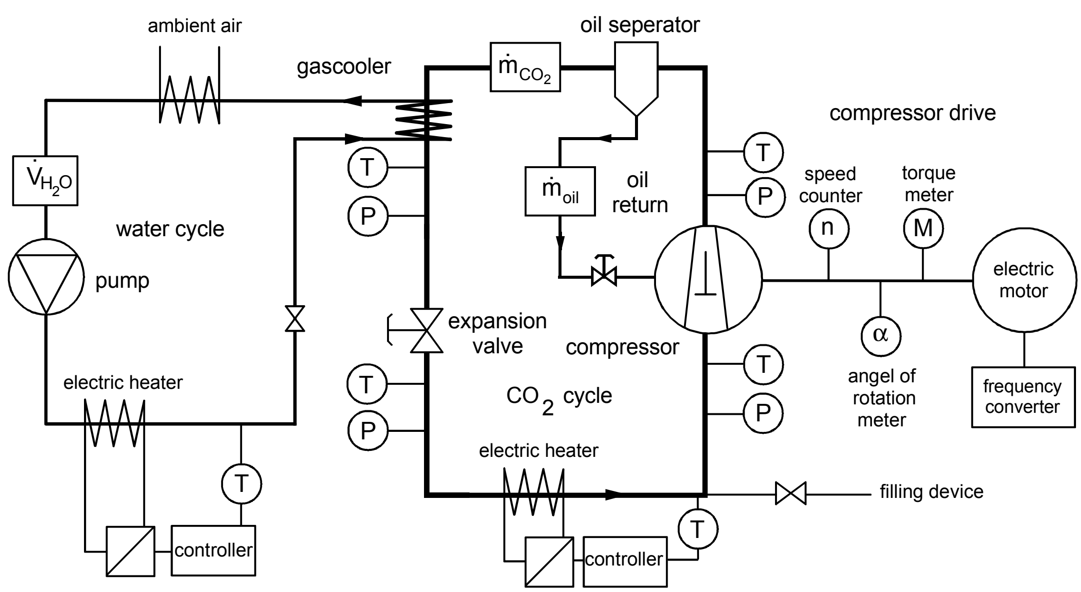


Figure 2: Test rig cycle with points of measurement.

6. MATHEMATICAL MODEL

For the numerical simulation an object-oriented simulation platform (Tegethoff 1999) is used to solve the basic compressor and valve equations to calculate indicator diagrams and theoretical compressor efficiencies. In the cylinder chamber vapor conditions are numerically analyzed based on balances of energy and mass, piston stroke and flow equations taking into account real gas properties. The model accounts the suction and discharge valve leakage and heat transfer between cylinder gas and cylinder wall. The flow through the valves is treated incompressible. The motion of the valves is approximated as one dimensional mass-spring system including the influence of adhesive forces.

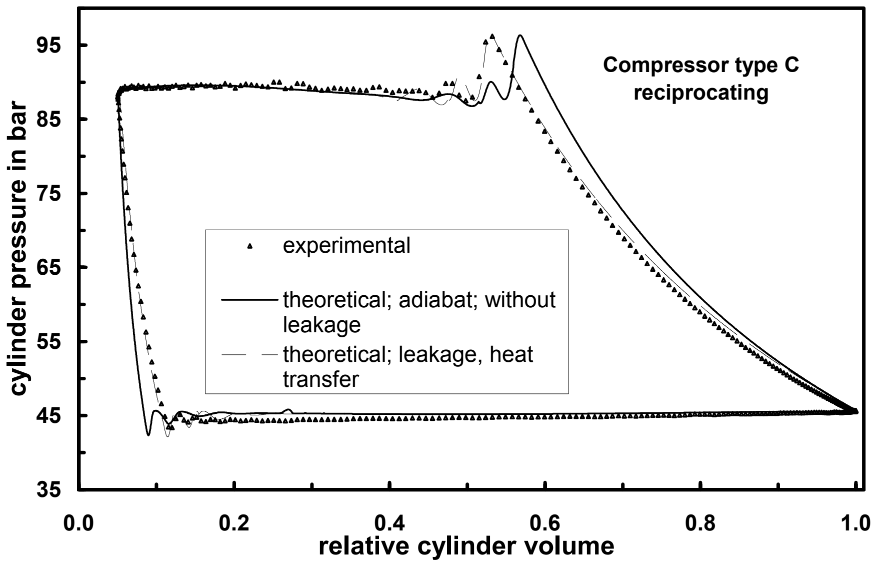


Figure 3: Comparison of experimental and theoretical indicator diagrams.

Figure 3 shows a good agreement between the measured and simulated indicator diagram for the reciprocating compressor type C.

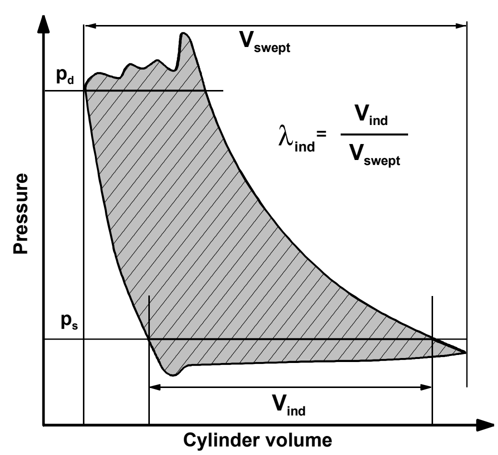
Compressor efficiencies

The indicated volumetric efficiency is defined in Figure 4 as

$$\lambda_{ind} = \frac{V_{ind}}{V_{swept}}$$

and the indicated isentropic efficiency is the ratio the theoretical isentropic work based on the indicated volumetric efficiency $P_{ind-isen}$ and the indicated work P_{ind} (area of the indicator diagram):

$$\eta_{ind-isen} = \frac{P_{ind-isen}}{P_{ind}} \ .$$



The effective isentropic efficiency is defined as the ratio of the theoretical isentropic compression power to the effective power supplied to the crankshaft as

Figure 4: Definition of the indicated volumetric efficiency.

$$\eta_{eff-isen} = \frac{P_{isen}}{P_{eff}} = \frac{\left((h_d)_{s=const} - h_s\right) \cdot \dot{m}_{CO2}}{2\pi \cdot \left|\vec{M}\right| \cdot f} .$$

The isentropic compression work P_{isen} was calculated based on the pressure and the temperature measured in the suction pipe.

The calculation of the effective volumetric efficiency was carried out based on the following equation

$$\lambda_{eff} = \frac{\dot{m}_{CO2}}{\dot{V} \cdot \rho(p,T)} .$$

Comparison of path and pressure controlled compressors

One important objective of this paper is to compare theoretically the two different principles of path and pressure controlled compressor design. Therefore a simple model (taking no leakage and heat transfer into account) was developed for compressor type C (pressure controlled) based on experimental data and compressor type B (path controlled) based on geometrical shape of the port plate areas. Results of this study are presented in Figure 5-10.

The advantage of the path controlled principle are the relative low losses in the suction and discharge channels compared to the losses of a pressure controlled valve system especially at high compressor speeds. The expected indicated isentropic and indicated volumetric efficiencies (Figure 5 and Figure 6) show nearly constant values over the whole speed range while the efficiencies for the compressor with pressure controlled valve system are decreasing due to the higher valve losses.

The disadvantage of the path controlled principle is the fixed opening and closing time of the discharge and suction areas that is optimized for only one pressure ratio π_0. In this operating point very similar indicator diagrams and efficiencies are expected compared to the pressure controlled valve system. For a lower pressure ratio a supercharging before the discharge stroke and an under-expansion before the suction stroke takes place (Figure 7). For a higher pressure ratio the valve areas are opening before the discharge pressure or respectively the suction pressure is reached and a backflow over the discharge and suction port takes place (Figure 8).

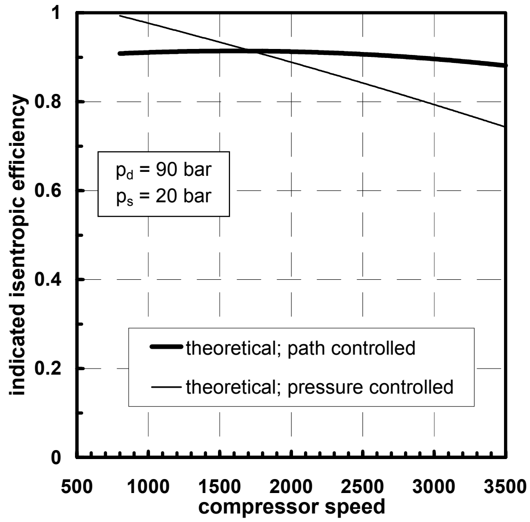


Figure 5: Calculated indicated isentropic efficiencies versus compressor speed.

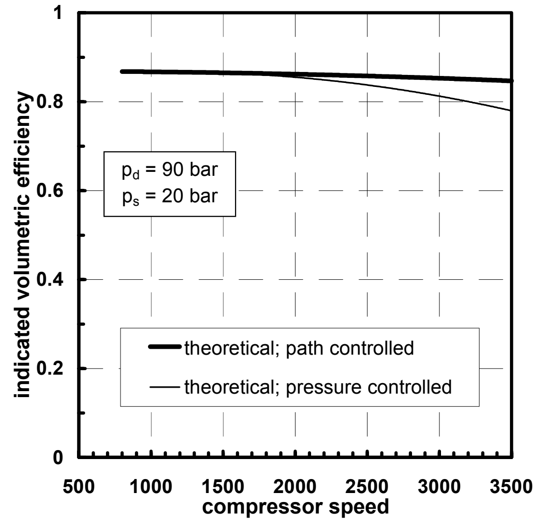


Figure 6: Calculated indicated volumetric efficiencies versus compressor speed.

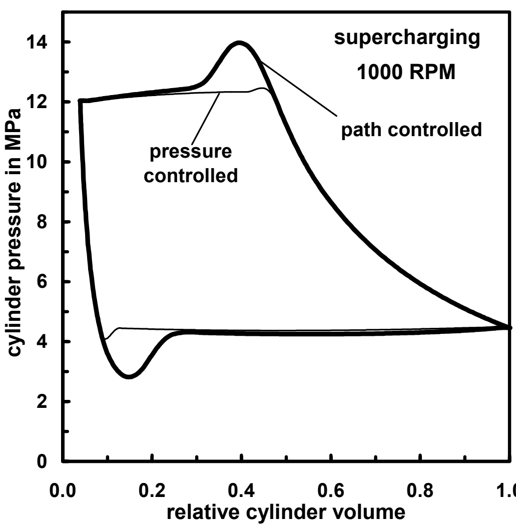


Figure 7: Calculated indicator diagrams for the case of supercharging $\pi = 2.5$.

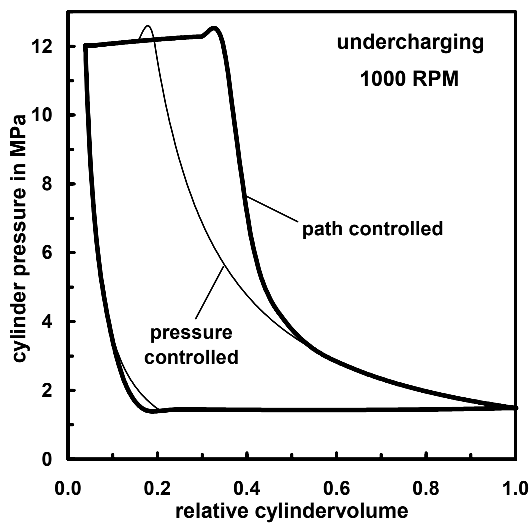


Figure 8: Calculated indicator diagrams for the case of undercharging at $\pi = 8$.

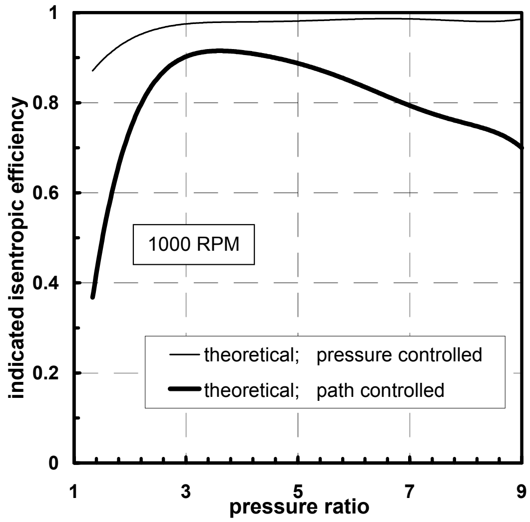


Figure 9: Calculated indicated isentropic efficiencies versus pressure ratio.

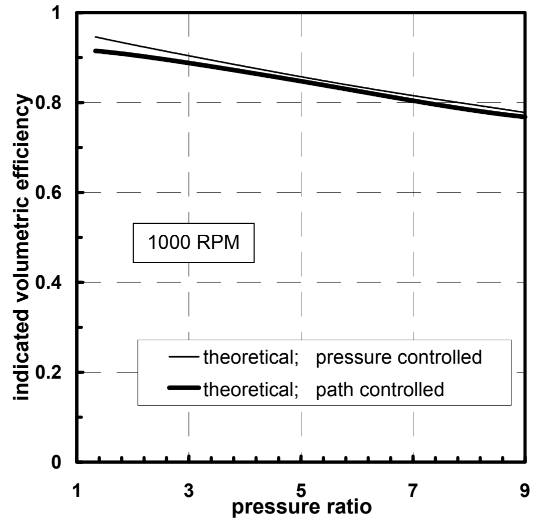


Figure 10: Calculated indicated volumetric efficiencies versus pressure ratio.

The main consequence of both, supercharging and backflow is an additional need of compression work for pressure ratios lower or higher π_0. Therefore the indicated isentropic efficiency decreases with a pressure ratio lower or higher π_0 as it is shown in Figure 9. The expected volumetric efficiency (Figure 10) of the path controlled compressor is nearly as high as the efficiency of the pressure controlled prototype.

7. EXPERIMENTAL RESULTS

In the first set of measurements the compressors type A, type B and type C were tested under different suction and discharge pressure conditions. The saturation temperature (pressure) was varied between 10°C / 45 bar (air conditioning) and -30°C / 14.5 bar (refrigeration). The discharge pressure was varied in a range between 60 bar and 130 bar. The compressor speed was set to 1000 RPM and superheat adjusted to 10 K.

Figure 11 and Figure 12 show indicator diagrams and the corresponding dynamic pressures in the discharge and suction chambers for the compressors type A and type C. The steady discharge and suction pressure in Figure 11 reflects the smooth running 7 cylinder swash-plate compressor. The 2 cylinder reciprocating compressor shows high pulsations especially in the discharge chamber (Figure 12). The cylinder pressure shows a pulsation when the discharge valve opens but there are no large losses in the valve system at the speed of 1000 RPM. The pulsations of the cylinder pressure of compressor type A are probably augmented by the capillary tube between the pressure transducer and the cylinder chamber.

The plot in Figure 13 shows the effective isentropic efficiency for the three tested compressor types at a relatively low compressor speed of 1000 RPM. Both pressure controlled compressor types achieve a 10% to 20% higher isentropic efficiency than the path controlled type and the efficiency of the reciprocating compressor with two cylinders is about 5% higher than the swash-plate compressor with 7 cylinders.

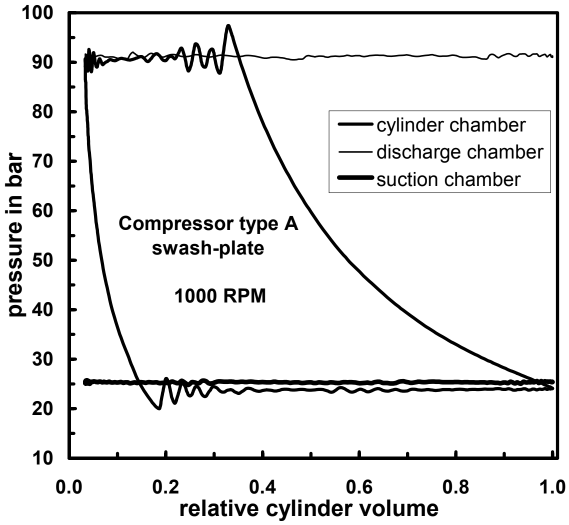


Figure 11: Measured indicator diagram.

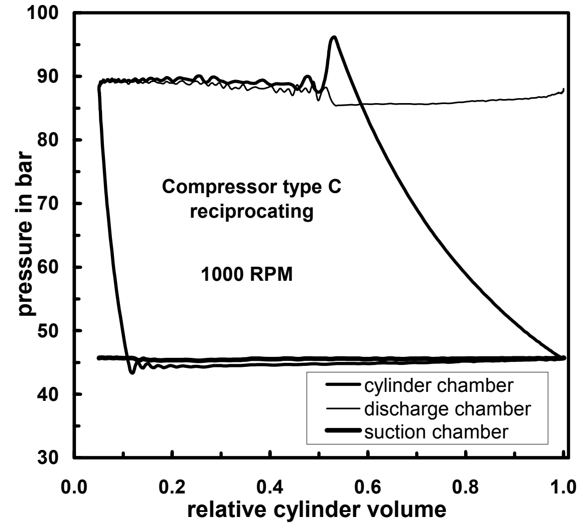


Figure 12: Measured indicator diagram.

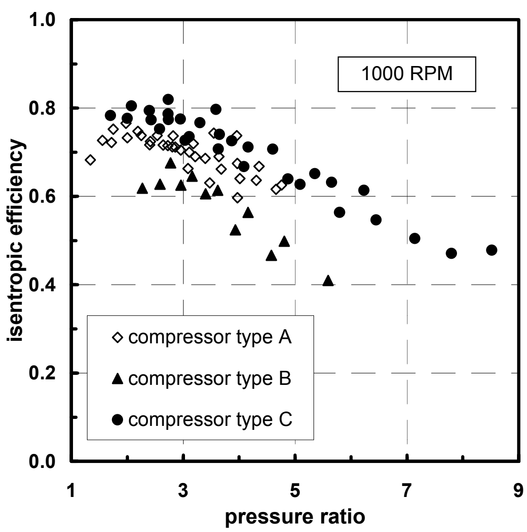


Figure 13: Measured isentropic efficiencies.

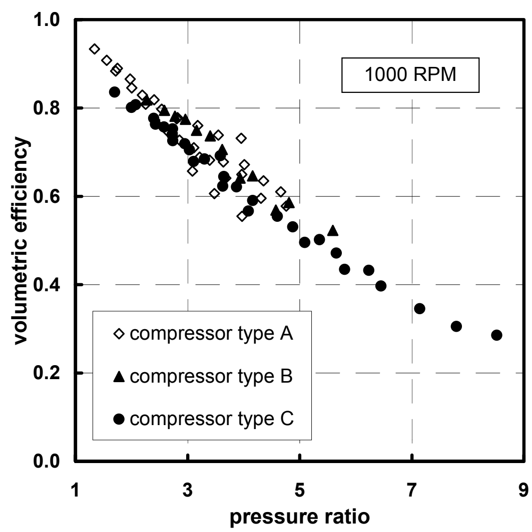


Figure 14: Measured volumetric efficiencies.

First experiments at higher compressor speed show that the isentropic efficiency of the path controlled compressor does not decrease with increasing compressor speed. At a compressor speed of about 2500 RPM both compressor types have similar isentropic efficiencies. The gas leakage between the cylinder and port plate is not negligible and is probably one reason for the low isentropic efficiency at low compressor speeds.

Concerning the volumetric efficiency (Figure 14) all three compressors show very similar results. Due to the higher relative clearance volume of compressor type C the efficiency is about 5% lower compared to type A and type B. Compared with the theoretical results of the indicated efficiencies presented in Figure 9 and Figure 10 a high potential of improvement concerning the valve system or the port plate is revealed.

8. CONCLUSION

Three different types of carbon dioxide compressors (pressure and path controlled) that have been tested with a test rig show fairly good efficiencies. Concerning the volumetric efficiency both compressor principles pressure and path controlled achieve similar results, the isentropic efficiency of the first generation of a path controlled axial compressor is about 10%-20% lower at 1000 RPM than both compressors with self acting valve system. At higher speeds similar efficiencies are observed. Theoretical investigations reveal a potential of improvement for both principles. Especially the path controlled principle is an interesting option to achieve good efficiencies over a wide range of compressor speed as needed for mobile applications.

9. ACKNOWLEDGEMENTS

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