Research program
Ambient- and waste heat;
Combined heat and power

Small oilfree CO_2-compressor

prepared by
Heinz Baumann
Baumann Engineering
Bürglistrasse 49
8400 Winterthur
hpbaumann@bluewin.ch

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Abstract

The objective of the project is to prove the feasibility of a small oilfree semi hermetic piston type CO_2 compressor for supercritical heatpump applications with large temperature spans. These processes involve high pressures like 35 bar suction pressure and 80 to 150 bar delivery pressure. The compressor design is based on two key elements which are :

- clearance seal piston/cylinder combination and
- PEEK-plate valves with flat valve springs

The feasibility study covers :

- design and manufacturing of a functional model
- performance tests over the full range of speed and pressure
- manufacturing cost estimate for a modified serial model design.

The compressor was designed for domestic water heating applications and has therefore no external cooling. The cooling of the motor is effected by the suction CO_2-gas flow through the motor, crankcase and cylinder head to the suction valves.

The tests confirmed the feasibility of the technology for the use in small oilfree compressors. The compressor which has a scotch yoke drive operates very quietly and without vibrations. It is able to handle outlet pressures up to 150 bar and outlet temperatures up to 200 °C .

Two kinds of cylinder-heads from stainless steel and from plastic were tested in order to find out the influence of heat exchange and heat conduction. The stainless steel cylinder-heads conduct a lot of heat to the compressor housing which results in a considerable preheating of the suction gas (see Diagr.4). The process of preheating and compression follows different courses for the two cylinder head materials (see Diagr.5). However, the characteristic values of compression are practically the same for both (see Diagr.8-12).
The Isentropic Efficiency (see Diagr.10) is comparable to the efficiency of oil-lubricated compressors on the market, despite the very small cylinder volume of 1.25 cm^3. The Volumetric Efficiency (see Diagr.12) is rather low due to the large design-related dead volume of 18%.

This CO_2 compressor technology is a possible alternative to oil lubricated compression systems in

- automotive air conditioning (heating and cooling)
- domestic water heating
- applications in the food industry where oilfree compression is a must.

Precondition for most of the applications however are competitive costs in comparison to oil-lubricated systems. In order to reach this target further development steps will be needed.

This work was done on behalf of the Swiss Federal Office of Energy. The author alone is responsible for the content and the conclusion of this report.

Content page

1 Introduction

The synthetic refrigerants HCFC will be phased out in the near future due to their ozone depletion effect. The substitutes HFC are not less problematic due to their Global Warming Potential.

The real substitutes for these synthetic refrigerants are Natural Working Fluids like :

Ammonia
Air
Water
Hydrocarbons (for example Isobutane, Propane, Butane etc.)
Carbon dioxide

The Carbon dioxide is interesting for applications like

Automotive air conditioning (heating and cooling)
Domestic water heating and
Drink refrigeration

because it is neutral to the environment, has no smell, is non-toxic, non-flammable and is not limited in nature, and furthermore has interesting thermodynamic properties.
These applications require supercritical processes with high pressures.
The problem is the handling of the high pressures with oil-lubricated compressors and the interaction between the oil and the CO_2-gas.
The use of oilfree compressors is proposed in order to overcome these problems.

2 Objectives of the project

The objective of this project is to proof the feasibility for a small oilfree semi hermetic piston type CO_2 compressor for supercritical heat pump applications with large temperature spans.

3 Procedure

A Functional Model was designed and manufactured, with the use of two serial proven key elements which are

- high pressure clearance seal piston/cylinder combination and
- PEEK-plate valves with flat valve springs

The Functional Model was tested and its performance characteristics evaluated over a wide range of speed and pressure.
The project concludes with a Manufacturing Cost Estimate for a modified Serial Model.

4 Design

The compressor was designed as a heat pump for domestic water heating applications. In order to use all the available heat for the heat pump process the CO_2 gas was used to cool the motor. The CO_2 gas enters at the bottom end of the motor and flows through the motor and crankcase into the cylinder heads. A high efficiency motor should guarantee a high overall efficiency of the compressor and keep the preheating of the suction gas as low as possible.

4.1 Data of the compressor :

• Number of stages	1
• Number of cylinders	4
• Cylinder diameter	10 mm
• Stroke	16 mm
• Suction pressure	35 bar
• Delivery pressure	80 – 150 bar
• Speed variabel	500 – 3000 U/min
• Power consumption at 1500 RPM	ca. 500 W
• Dead volume	18 %

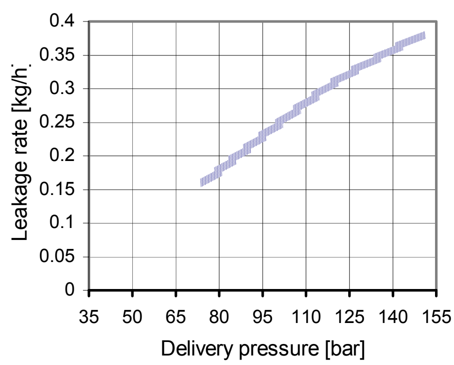
4.2 Design features of the Functional Model and the Serial-Model :

- Semihermetic compressor with integrated motor
- 4 cylinders in cross arrangement
- Scotch yoke drive with complete mass balance, no vibrations
- Simple shaft with one crank pin; arranged on two sealed ball bearings
- Piston/cylinder: Clearance seal
- Valves: Plastic-plate valves with flat spring
- Driven by Permanent Magnet Synchron Motor
- Cooling effected by working media CO_2

The cross section of the Functional Model is shown in Fig. 1.

The following design features deserve particular attention:

➢ ***High pressure clearance seal***
The clearance seal seals with a minimal gap between piston and cylinder of 4÷6 µm in diameter. The leakage flow through the gap is laminar and according to pressure a few percent of the flowrate (see Diagr. 1). The piston moves practically frictionless back and forth and does therefore not wear. (In similar applications with other gases the service period of piston/cylinder used to reach 10`000 hours)



Diagr. 1: Estimated clearance seal losses

- ***Valves***
  The compressor valves are flat sealing plates from PEEK, pushed against the metallic seat by a flat spring. The sealing plates have good sealing qualities, are light and operate quietly without any wear. The service period of the valves exceeds the one of the clearance seal.
- ***Scotch yoke drive***
  This well known drive configuration produces sinusoidal movements. Advantageous is a combination with 4 cylinders in cross arrangement with the two opposed pistons coupled together by a yoke. The addition of the two 90° phase shifted mass forces of the first order results in a rotating force which can be completely balanced by two rotating counterweights on the shaft. Machines with this drive configuration operate very quietly and without vibrations.

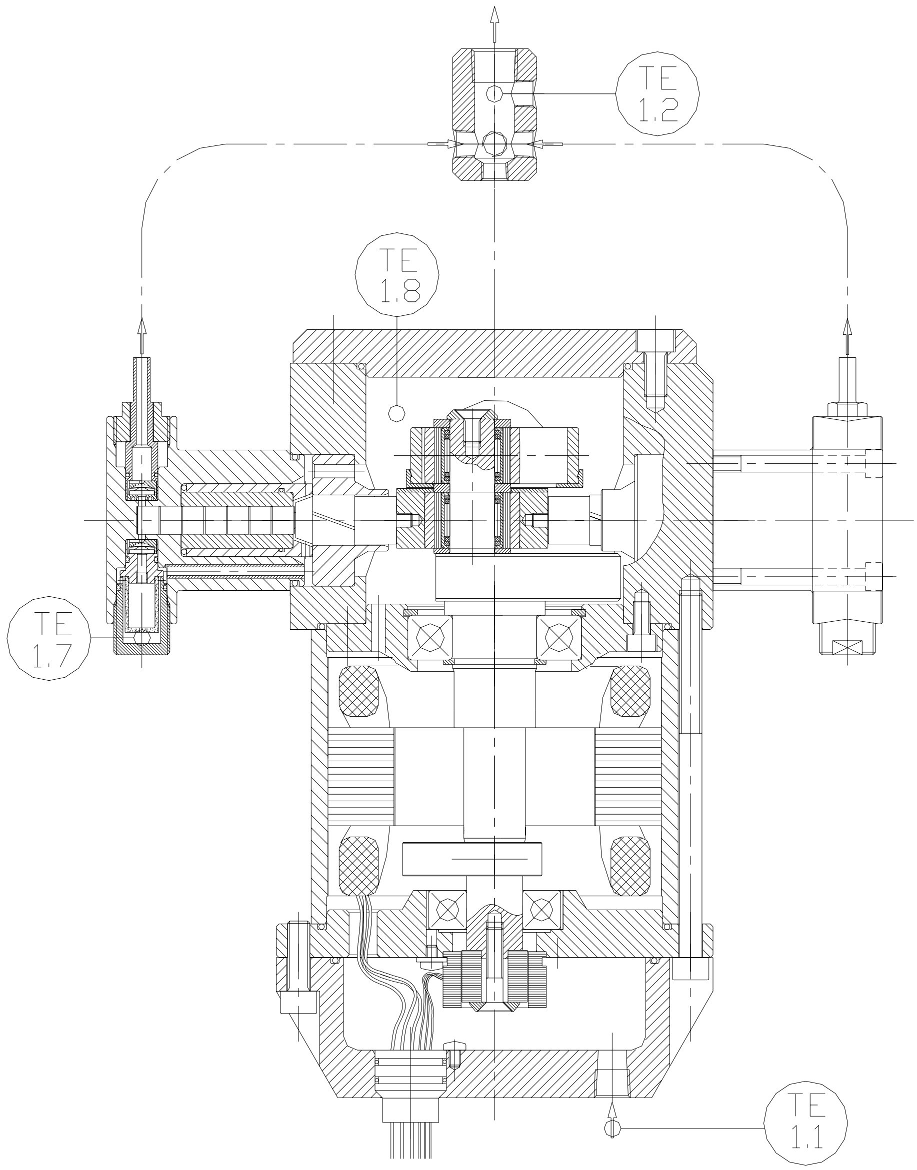


Fig. 1 : CO_2-Compressor, Functional Model
Temperature measuring points on the compressor
(Measuring points list see Tab.1)

4.3 **Functional Model** (see Fig.1)

The compressor housing parts are all manufactured in aluminium. The cylinder-heads are from stainless steel in order to keep the heat conduction losses low. Cylinder-heads from a temperature resistant plastic were manufactured too in order to investigate the influence of heat conduction.
The resolver on the motor side shaft end serves the control of speed.

4.4 Serial Model

The Serial Model is different from the Functional Model in shape and manufacturing method of the housing parts which are aluminium castings.

4.5 Motor

A Permanent Magnet Synchron Motor was chosen for highest efficiency over a wide range of speed and torque. The speed can be varied by a frequency converter between 500 and 3000 RPM.

5 Motor measurements

For a detailed analysis of the compressor efficiency it was necessary to know the efficiency of the drive train <frequency converter – motor – shaft> at the respective operating conditions. The performance of the motor together with the frequency converter was measured on a motor teststand (see fig.2 below and pict.1 in annex). For this the cylinder heads and the scotch yokes were removed and a shaft extension piece was fixed onto the crank pin of the shaft. An elastic coupling connected it to the torque transmitter and the motor brake.

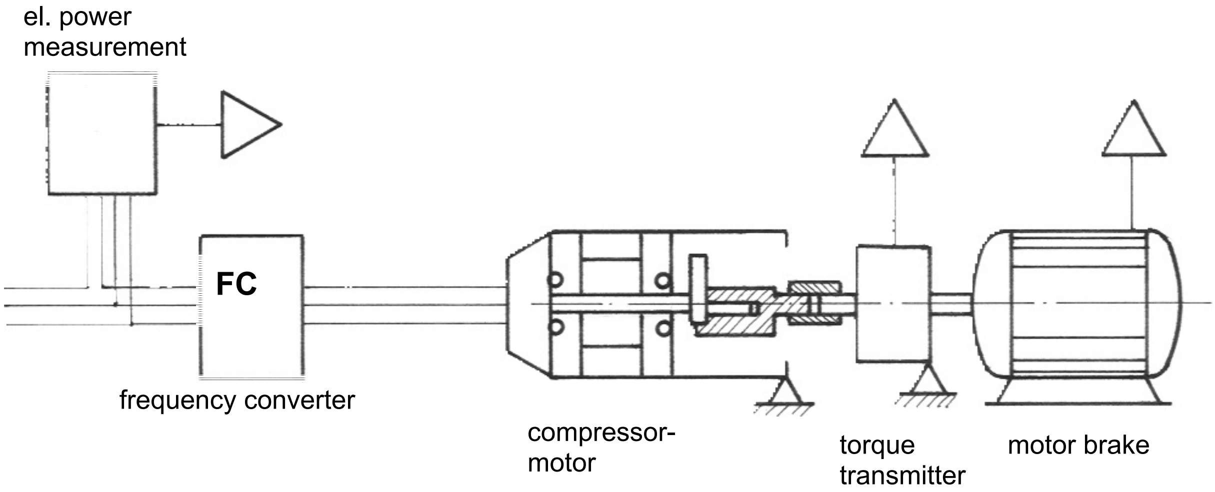
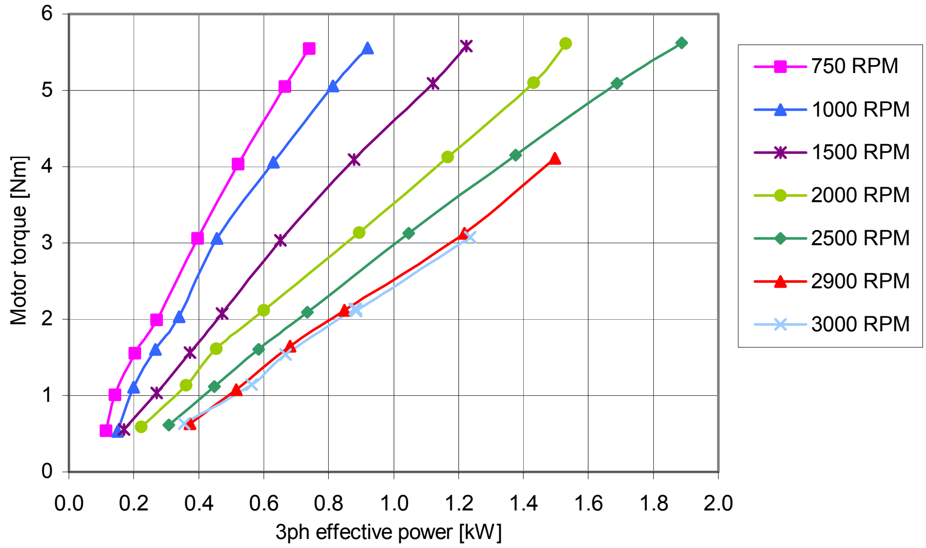
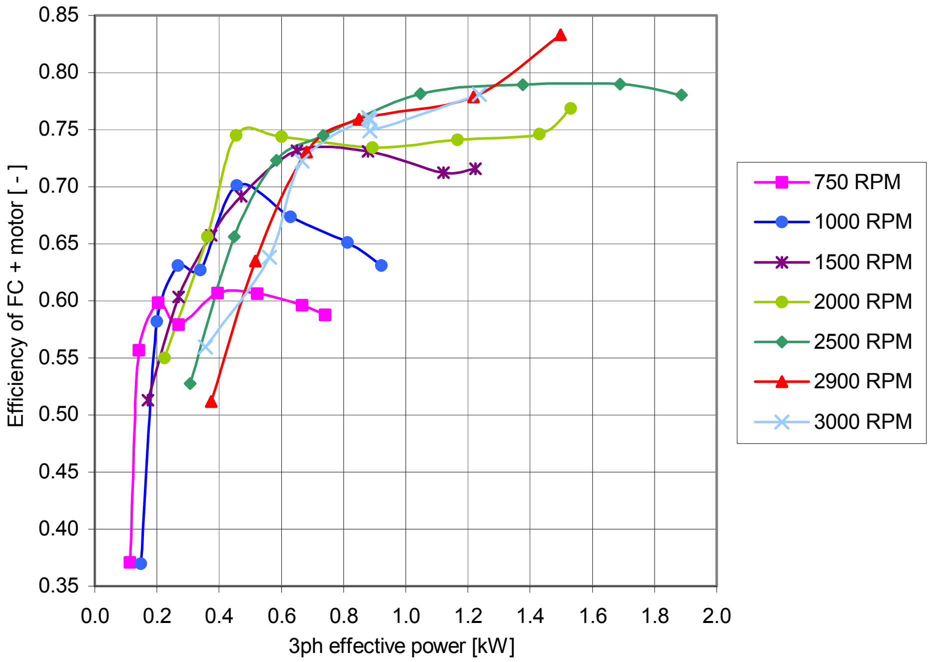


Fig.2 : Motor measuring arrangement

The data were stored on a PC and afterwards converted into the torque and efficiency curves in the following diagrams 2 und 3:



Diagr.2 : Motor measurements; Torque on compressor shaft



Diagr. 3 : Motor measurements; Efficiency of Frequency Converter + Motor

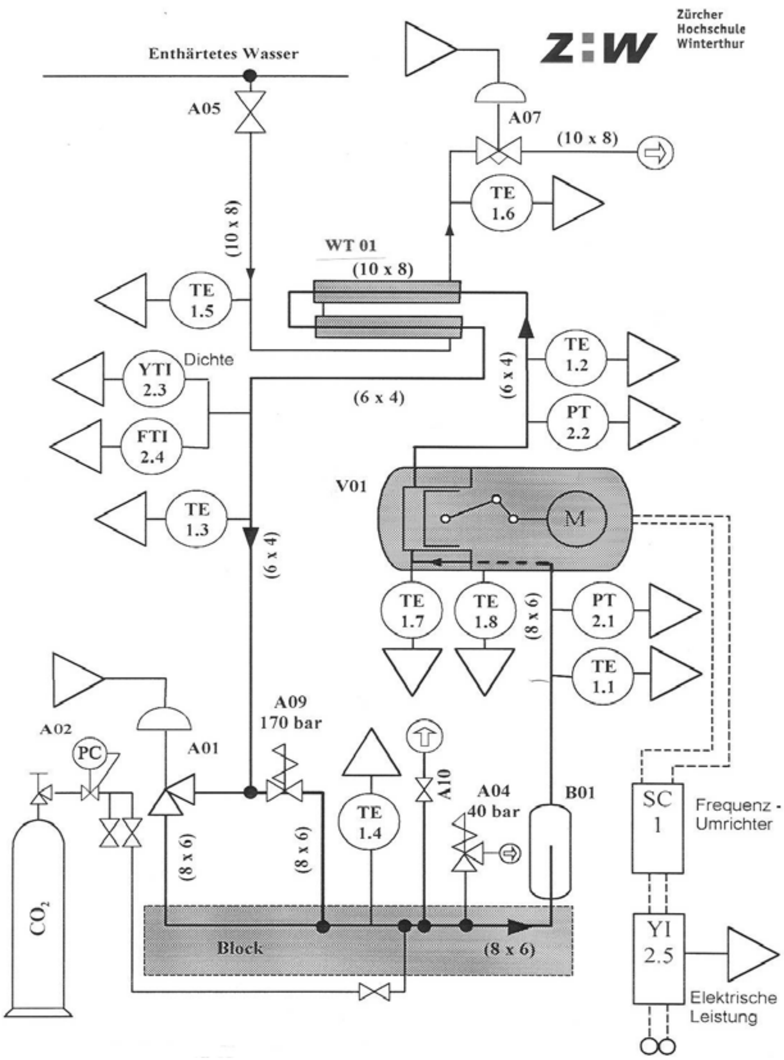
The motor efficiency was not as good as expected; especially at part load and at low speeds the efficiency was rather low.
(The compressor testruns covered the power range between 160 and 950 Watt)

6 Compressor testloop

The testloop (see pict. 2 in annex) was planned as a so called Hot Gas Loop, which was more practical for performance measurements on the compressor than a complete heatpump process with condensation and evaporation.
Figure 3 shows the (P+I) - diagram of the testloop.
The measuring data were all conducted to the PC. The monitoring and the storage of the measuring data and the control of the process was done with the software program DASYLab32. The following variables were adjusted or controlled:

- The suction pressure was set to 35 bar for all the test runs. (Reason: 35 bar is the vapour pressure of CO_2 at 0°C which is the evaporation temperature of the assumed domestic water heat pump process)
- The motor speed and the position of the expansion valve A01 were adjusted to a preset value for each measurement.
- The process was controlled only by the cooling water meetering valve A07; the cooling water flowrate was controlled in this way, that the gas temperature reached +5°C after the expansion valve, which corresponds to a superheating of 5°C. (i.e. the controlled variable was the gas temperature TE1.4 after the expansion valve).

Fig.3 : (P+I) – diagram of the testloop



Tab.1 Measuring points list

Measuring point	Measured variable				Principle of Measurement	Channel assignement	
	Symbol	Unit	Description	Signal			
TE 1.1	T	°C	Gas-temperature compressor inlet		Thermocouple	1	Analog inputs Board DS-16-8-TC-AO
TE 1.2	T	°C	Gas-temperature compressor outlet		Thermocouple	2	
TE1.3	T	°C	Gas-temperature expansion valve in		Thermocouple	3	
TE1.4	T	°C	Gas-temperature expansion valve out		Thermocouple	4	
TE1.5	T	°C	Cooling water heat exchanger in		Thermocouple	5	
TE 1.6	T	°C	Cooling water heat exchanger out		Thermocouple	6	
TE 1.7	T	°C	Gas-temperature at suction valve		Thermocouple	7	
TE 1.8	T	°C	Gas-temperature in crankcase		Thermocouple	8	
PT 2.1	p	bar g	Suction pressure	mA	Pressure transmitter	1	Analog inputs Board DS-16-8-GP-AO
PT 2.2	p	bar g	Delivery pressure	mA	Pressure transmitter	2	
YTI 2.3	ρ	kg/m^3	Density of gas	mA	Coriolis-flowmeter	3	
FTI 2.4	m	kg/h	Massflow of gas	mA		4	
YI 2.5	P_{el}	W	Electric power consumption	V	el. power measurement	5	

Tab.2 Equipment list

Designation	Description	Channel assignement		
A01	Expansion valve	V	2	Analog outputs Board DS-16-8-GP-AO
A07	Cooling water meetering valve	mA	1	
A02	Gas cylinder pressure control valve			
A04	Low-pressure relief valve 40 bar			
A05	Cooling water shut-off valve			
A09	High-pressure relief valve 170 bar			
B01	Vessel 300 cm^3			
WT01	Double-pipe heat exchanger			
V01	Compressor			

7 Compressor test results

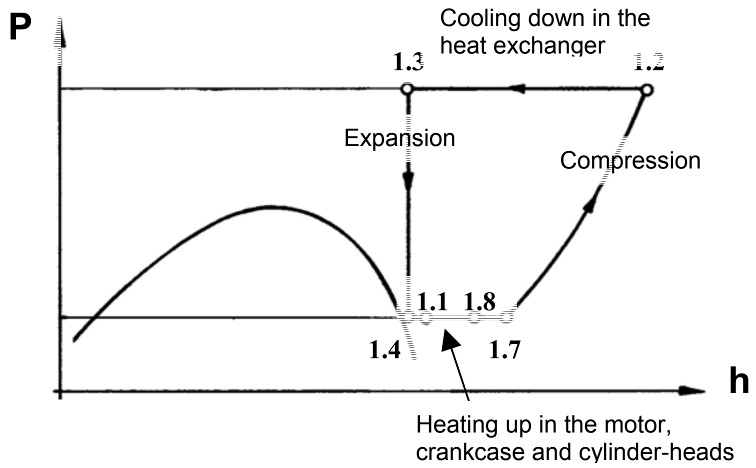
7.1 Functionality

The key components piston/cylinder and the valves show no sign of wear or fatigue after approx. 300 hours of operation. These components are excellent for the use in CO_2 compressors at high pressures.
We did not run any endurance tests so far, but from experience with other applications and gases, we expect these components to last several thousand hours of operation with CO_2.

7.2 Test results

The compressor performance test results are presented in diverse diagrams. The compressor measurements were carried out both with cylinder-heads from stainless steel and with cylinder-heads from plastic.

The basic course of a testloop process with its process points is shown in Fig.4 .

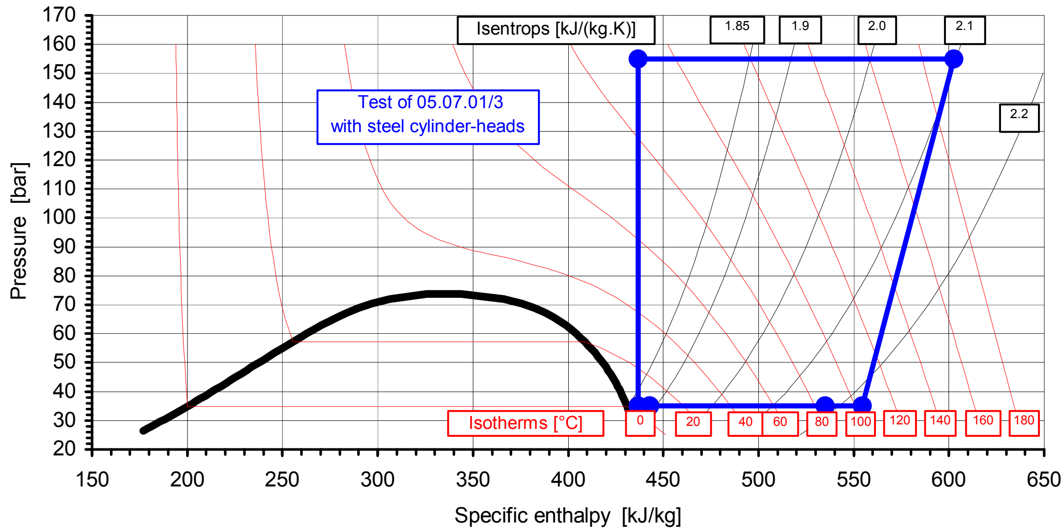


Process points:

1.1	Compressor inlet
1.8	Crankcase
1.7	Compr. suction valve
1.2	Compressor outlet
1.3	Expansion valve in
1.4	Expansion valve out (=aimed fix condition 35 bar / 5°C)

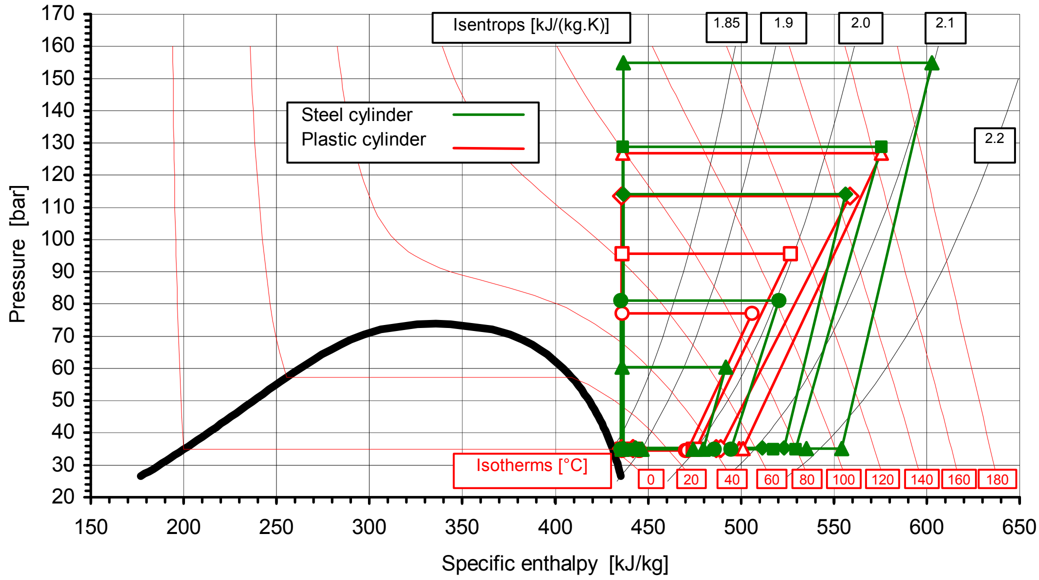
Fig.4 : Basic course of a testloop-process in the (p-h)-diagram

A typical course of a process is shown in diagram 4. It represents a measurement with steel cylinder-heads. The suction gas is heated up in the motor and in the crankcase quite strongly by about 70°C and in the cylinder-head again almost 20°C ! The reason for this is the high crankcase temperature of about 85°C which is caused by the heat conduction from the steel cylinder-heads to the compressor housing. (The compressor is not cooled externally!).



Diagr.4 : Typical course of a process in the (p-h)-diagram (with steel cylinder-heads)

Diagram 5 shows a comparison of a few process-courses with steel cylinder-heads against courses with plastic cylinder-heads.



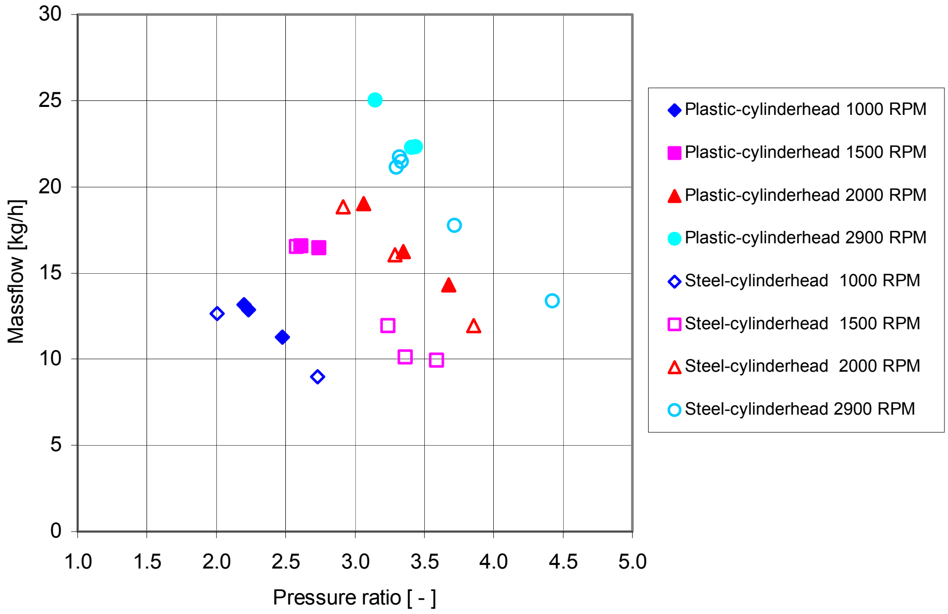
Diagr.5 : Comparison of process-courses with steel cylinder-heads and with plastic cylinder-heads in the (p-h)-diagram

The remarkable difference between the two cylinder-material executions can be recognized in the process-courses:

- The compressor with steel-cylinderheads shows a considerable heating up of the gas before it enters the cylinder, but compresses the gas with heat rejection.
- The plastic-cylinders prevent the heat exchange and the heat conduction to the compressor housing, which shows in lower housing temperatures and less heating up of the gas before the compression. The compression itself is almost isentropic which means there is less heat exchange with the cylinders.
- **The result at the end stays the same!** The compressor outlet temperatures are approximately the same, independent of cylinder-head material. All the characteristic values of compression, presented in the diagrams 8 – 12 show only very small differences between the two executions of cylinder-head material. It seems that above effects are compensating each other.

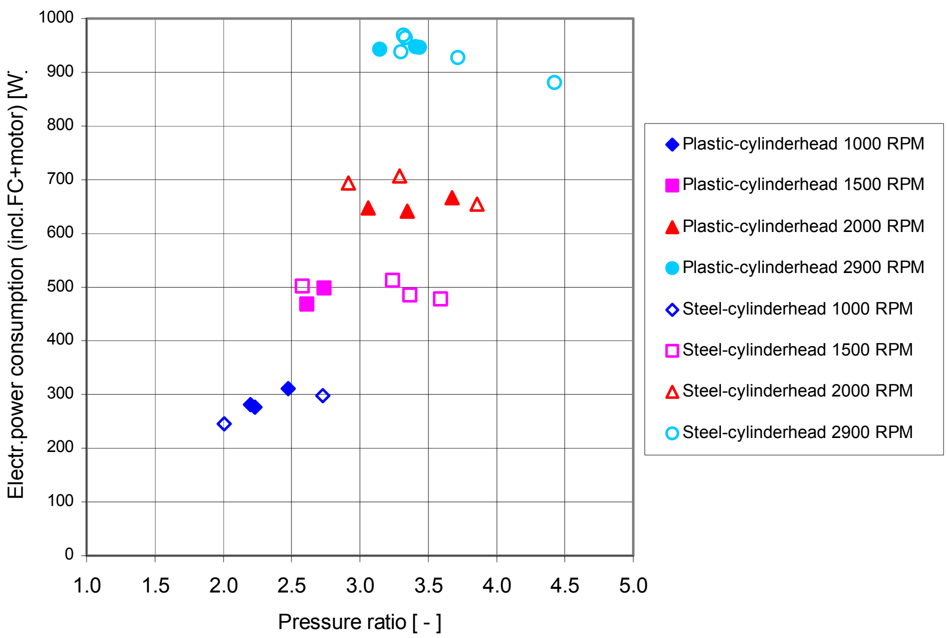
The evaluation of the numerous measurements are shown in the following diagrams 6 – 12, diagrams 6 and 7 with the speed as parameter.
The diagrams 8 –12 apply for the whole range of speed from 750 to 2900 RPM. The speed has practically no influence on the compression specific values, i.e. the valve losses do not weight within our range of operation.

Diagram 6 shows the massflow of CO_2 in function of the pressure ratio, with the speed as parameter.



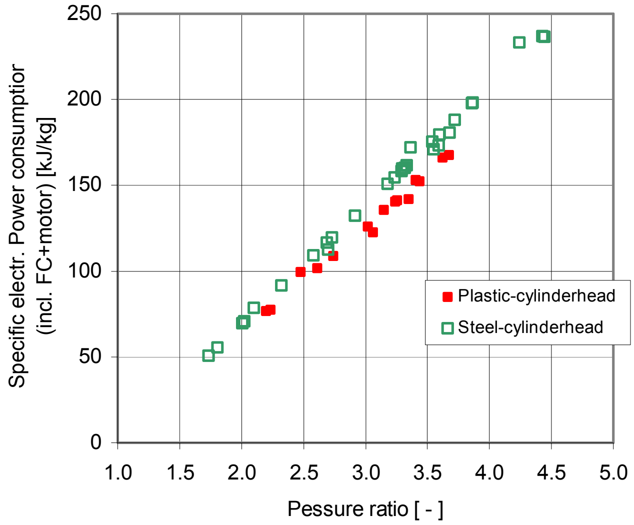
Diagr. 6 : Massflow of CO_2

Diagram 7 shows the electrical power consumption measured at the input of the frequency converter, in function of the pressure ratio, with the speed as parameter.



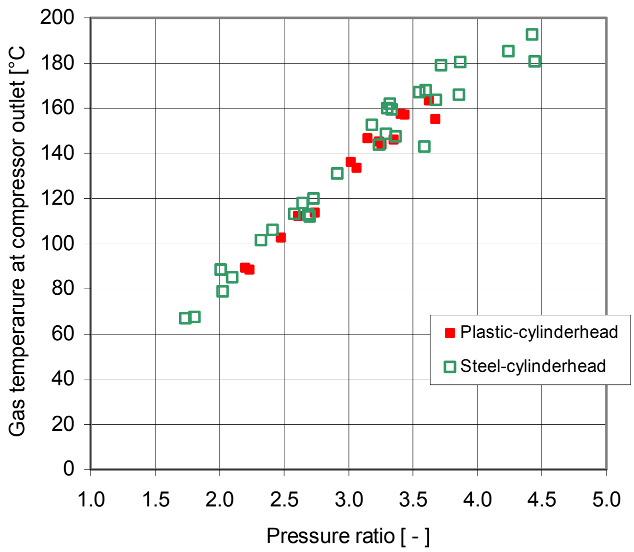
Diagr. 7 : Electrical power consumption (inclusive frequency converter + motor)

Diagram 8 shows the specific electrical power consumption measured at the input of the frequency converter, in function of the pressure ratio.



Diagr. 8 : Specific electr. power consumption inclusive frequency converter + motor

Diagram 9 shows the gas temperatures on the compressor outlet in function of the pressure ratio.



Diagr. 9 : Gas temperature at compressor outlet (TE1.2)

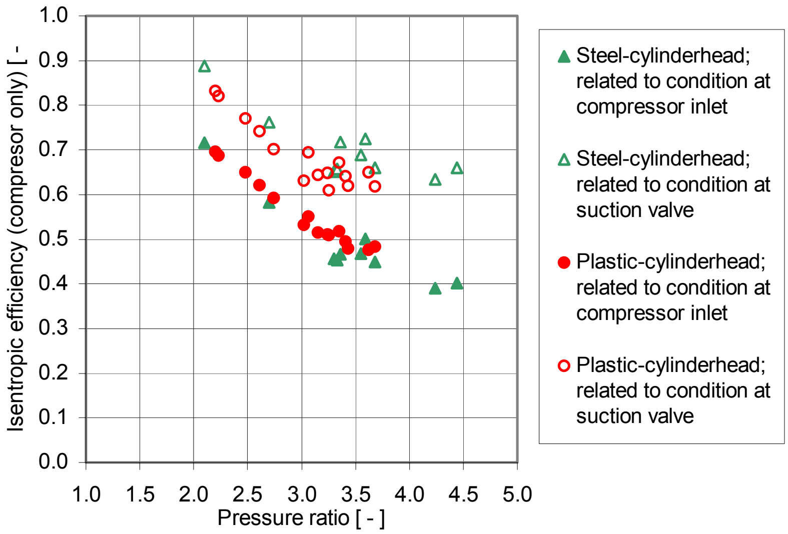
7.3 Efficiencies

The various efficiencies of the compression are defined in chapter 10, page 16.

In order to be able to compare the "quality" of the compression with values of compressors on the market, the Volumetric Efficiency as well as the Isentropic Efficiency were each related to the condition at ***compressor inlet*** and to the condition at the ***suction valve***. With the latter the negative effect of the heating up of the suction gas in the motor and crankcase is excluded.

Diagram 10 shows the Isentropic Efficiency of the compressor without frequency converter and motor, in function of the pressure ratio.

(The Isentropic Efficiency compares the theoretical isentropic compression power with the shaft power).

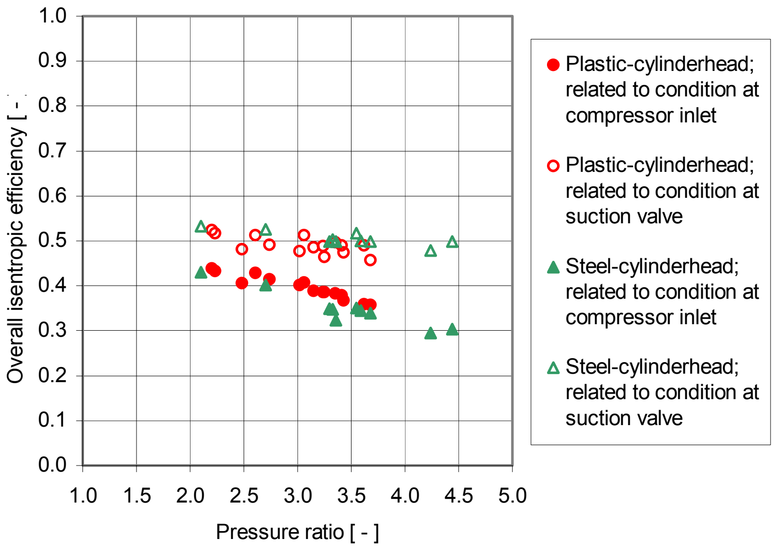


Diagr. 10 : Isentropic efficiency (compressor only)

Diagram 11 shows the Overall Isentropic Efficiency (incl. motor + frequency converter), in function of the pressure ratio.

(The Overall Isentropic Efficiency compares the theoretical isentropic compression power with the electric power input).

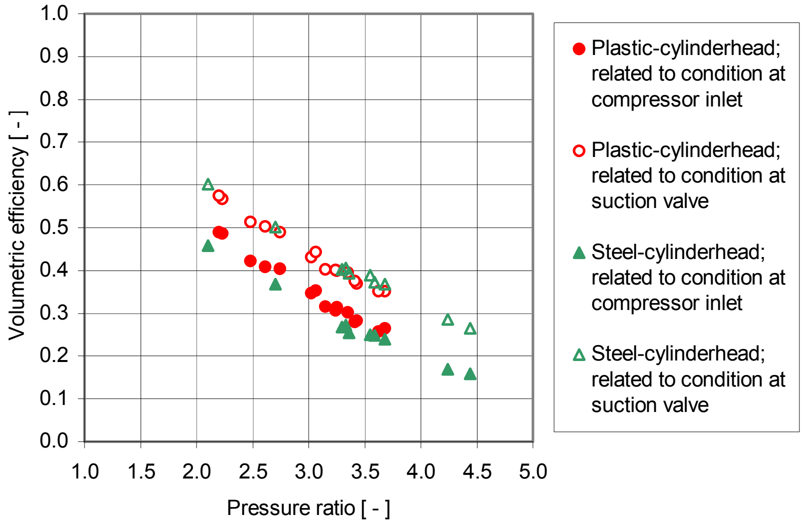
The Overall Isentropic Efficiency is low due to the low motor efficiency (see diagr.3, page 8), especially at low pressure ratios.



Diagr. 11 : Overall isentropic efficiency (inclusive motor+FC)

Comparison with compressors on the market:
The few CO_2-compressor-reports found in Literature [1;2] give an Isentropic Efficiency of about 0.7 (for lubricated compressors) in the pressure-ratio range of 3÷4.
As in Diagr.10 the Isentropic Efficiencies of our oilfree CO_2 compressor in the same pressure-ratio range varies between 0.6 and 0.72 , related to the condition at the suction valves, which compares to compressors with the suction line directly into the cylinder. This is quite a remarkable result for a small compressor with 1.25 cm^3 cylinder volume.

Diagram 12 shows the Volumetric Efficiency in function of the pressure ratio.



Diagr. 12 : Volumetric efficiency

The Volumetric Efficiency is rather low due to the large dead volume of 18%. The dead volume could not be reduced any further for design reasons. It is large in relation to the very small cylinder volume of only 1.25 cm3.

Comparison with compressors on the market:
In Literature [3] a Volumetric Efficiency of 0.6 is given for a lubricated CO_2 compressor with 10% dead volume and a pressure ratio of 3. This compares to a Volumetric Efficiency of 0.45 for our compressor. The influence of the larger dead volume makes at least 0.1 from the total difference of 0.15 .

8 Manufacturing costs for a small series of the Serial Model

The manufacturing costs for a small series of 5000 pieces of the Serial Model was evaluated. (The tooling costs for the castings and the Injection moulds are not included.)

Low Cost execution:

Compressor with Asynchronmotor, motor efficiency 78%, fix speed; assembled and tested — CHF 1320.-

High Tech execution:

Compressor with Permanentmagnet-Synchronmotor of highest efficiency of 88%, variable speed; with Frequency Converter and Netfilter; assembled and tested — CHF 1860.-

(The Frequency Converter without housing, only as board, integrated in an external control box would reduce the price by approx. CHF 150.-)

The reasons for the relatively high manufacturing costs are:

- The size of the series of 5000 is relatively small.

- Although it is called the Serial Model it is still more or less based on conventional mechanical engineering technics. For large series a redesign would be needed focused on cheap manufacturing methods.

- The manufacturing costs of the 4 Piston/Cylinder combinations alone makes almost 40% of the total manufacturing costs. A cheaper material combination suitable for dryrun should be evaluated. The reduction of the amount of cylinders would have a positive effect on the costs but would cause disadvantages like stronger vibrations, higher pressure pulsations and higher bearing loads.

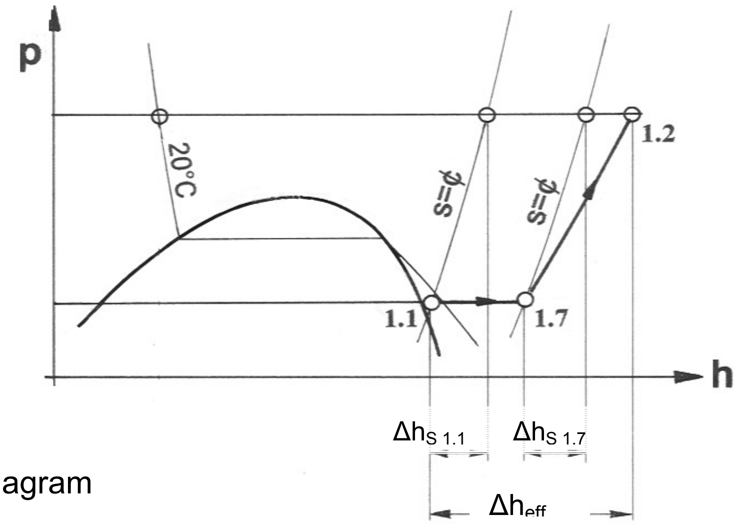
9 Conclusion and outlook

The test series proved the feasibility of the used technology for small oilfree high-pressure CO_2-compressors. The Efficiencies and the Functionality show that there is a lot of potential in this technology.
A redesign should concentrate on a more compact design and lower production costs.
Where the main focus of future developments will be is dependent on the kind of application.

Future development areas could be:

- Development of cheaper material combinations for the dryrunning Piston/Cylinders.
- Detailed analysis of the thermodynamic processes and energy flows in the cylinders. The concept with the cooling of the unit by the gasflow alone is to be questioned and according to the application possibly another gasflow direction to choose.
- Design study and development of a dryrunning multi-axialpiston-compressor for compact arrangements.
- Redesign focused on cheap mass manufacturing methods.

10 Definition of the compressor efficiencies



1.1 Condition at compr. inlet
1.7 Condition at suction valve
1.2 Condition at compr. outlet

Fig.6 : p-h-Diagram

Isentropic Efficiency (without motor) (Diagram 10)

related to condition at compressor inlet $$\eta_{is\,1.1\,Compr} = \frac{\Delta h_{S\,1.1}}{w_{el} \cdot \eta_{Motor}}$$ (Eq.1)

related to condition at suction valve $$\eta_{is\,1.7\,Compr} = \frac{\Delta h_{S\,1.7}}{w_{el} \cdot \eta_{Motor}}$$ (Eq.2)

Overall Isentropic Efficiency (incl. FC+motor) (Diagram 11)

related to condition at compressor inlet $$\eta_{is\,1.1\,overall} = \frac{\Delta h_{S\,1.1}}{w_{el}}$$ (Eq.3)

related to condition at suction valve $$\eta_{is\,1.7\,overall} = \frac{\Delta h_{S\,1.7}}{w_{el}}$$ (Eq.4)

Volumetric Efficiency (Diagram 12)

related to condition at compressor inlet $$\eta_{Vol\,1.1} = \frac{\dot{V}_{eff\,1.1}}{\dot{V}_H} = \frac{\dot{m}}{\rho_{1.1} \cdot \dot{V}_H}$$ (Eq.5)

related to condition at suction valve $$\eta_{Vol\,1.7} = \frac{\dot{V}_{eff\,1.7}}{\dot{V}_H} = \frac{\dot{m}}{\rho_{1.7} \cdot \dot{V}_H}$$ (Eq.6)

$\dot{V}_H$ = theoretically displaced volume-flow

$$= \frac{D^2 \cdot \pi \cdot H \cdot n \cdot 4}{4 \cdot 60}$$

11 Symbols

p	pressure
T	absolut temperature
ρ	density
h	specific enthalpy
h_S	specific enthalpy difference at isentropic compression
h_{eff}	real specific enthalpy difference in the compressor
s	specific entropy
w_{el}	specific electrical energy consumption measured at the input of the FC
η_{Motor}	efficiency of motor+FC
η_{is}	isentropic efficiency
η_{Vol}	volumetric efficiency
$\dot{V}_H$	theoretically displaced volume-flow
$\dot{V}_{eff}$	effective flowrate
z_S	dead volume (= 0.18)
$\dot{m}$	massflow
D	cylinder diameter
H	stroke
n	speed

12 References

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Annex

Content

Sample of a process calculation

The sample refers to the measurement of 5.7.01/03 with steel-cylinderheads. The course of the process is shown in the (p-h)-diagram 4, page 11.
The measured data of the process are listed in the following table "Process data". The table "Evaluation" below represents the calculated values which were used for the diagrams in the report.
The tables show also process data which have not been used in this report but could be interesting for other considerations.

Process data

Date	Electr.	Massflow	Cooling water		Ambient	Density bef	Valve position		Speed		Motor
05.07.01/3	Power		in	out	pressure	exp.valve	expansion	cooling w.			efficiency
Bla / Czm	[W]	[kg/h]	[°C]	[°C]	[mbarg]	[kg/m^3]	[V]	[mA]	[V]	RPM	[--]
Mean value	881	13.390	18.83	73.24	961.9	373.6	1	7.75	7.9	2900	0.75
Standard dev.	10	1.378	0.34	3.08	---	6.67	---	0.25	---	---	---

Process points

Meas. point	Temperature mean value	Temperature stand.deviat.	Pressure (gauge) mean value	Pressure (gauge) stand.deviat.	Pressure (absolute)	Density	Spec. Entropy	Spec. Enthalpy
	[°C]	[°C]	[bar]	[bar]	[bar]	[kg/m^3]	[kJ/(kg.K)]	[kJ/kg]
Compressor in	7.49	0.23	34.06	0.25	35.02	90.28	1.8880	442.89
Crankcase	85.14	2.01	34.06	0.25	35.02	57.35	2.1800	535.10
Sution valve	103.42	1.91	34.06	0.25	35.02	53.56	2.2325	554.38
Compressor out	192.60	1.64	154.01	0.88	154.97	201.52	2.0977	602.71
before Exp. valve	91.54	0.42	154.01	0.88	154.97	383.94	1.6909	436.74
after Exp.valve	4.96	0.27	34.06	0.25	35.02	92.69	1.8739	438.94
do. korrigiert	---	---	34.06	---	35.02	---	1.8659	436.74
Dummy	---	---	---	---	---	---	---	---
Isentropic compression from compressor inlet					154.97	---	1.8880	512.05
Isobare cooling down to 20 °C					154.97	---	1.0814	236.12

Compressor		
Cylinders	4	[---]
Diameter	10	[mm]
Stroe	16	[mm]
Displ.vol.	5.027	[cm^3]
Dead vol.	0.18	[---]
Heat exchanger		
d_outside	6	[mm]
d_inside	4	[mm]
Length	0.88	[m]
Area	0.0166	[m^2]
eta FC [--]	Cooling water massflow	
0.95	[kg/h]	6.6

Specific energy - consumptions		
Electrical (FC + motor)	[kJ/kg]	236.9
Specific enthalpy - differences		
From compressor inlet to outlet	[kJ/kg]	159.8
From compressor inlet to crankcase	[kJ/kg]	92.2
From crankcase to suction valve	[kJ/kg]	19.3
From suction valve to compr. outlet	[kJ/kg]	48.3

Evaluation		Date/No.	05.07.01/3
Energy balances			
Energy consumption of FC and motor		[W]	881
Specific enthalpy-increase of CO2 in the compressor		[kJ/kg]	237
(1)	Energy consumption motor	[W]	837
(2)	Enhalpy difference cooling water	[W]	419
(3)	(1) - (2) Heat convection from testloop o ambient	[W]	418
(4)	Enthalpy difference of CO2 from compressor inlet to outlet	[W]	594
(5)	(1) - (4) Heat convection from compressor to ambient	[W]	243
(6)	Enthalpy difference of CO2 on high pressure side	[W]	617
(7)	(6) - (2) Heat convection from high pressure side to ambient	[W]	199
(8)	Heat absorption of expansion valve from ambient	[W]	8
(9)	Heat absorption of suction line from ambient	[W]	15
Heat exchanger			
(10)	Power of the heat exchanger	[W]	518
(11)	Mean logarithmic temperature difference	[grd]	94
(12)	Heat transfer coefficient of the heat exchanger	[W/(m^2.grd)]	332
Energy balance of otor and compressor			
(13)	Losses of motor to CO2	[W]	176
(14)	Heat absorption of CO2 from compr. inlet to to crankcase	[W]	343
(15)	(14) - (13) Heat absorption of CO2 in the crankcase	[W]	167

Heat pump process with cooling down of CO2 to 20°C		
Isentropic efficiency from inle to outlet of compressor	[---]	0.43
Theor. COP with isentropic compression after the evaporator	[---]	3.99
Real COP according to diagram	[---]	2.29
Overall COP including FC and motor	[---]	1.55
Comparison overall COP with isentr. compr. COP	[---]	0.39
Compression		
Pressure ratio	[---]	4.42
Polytropic exponent (estimated)	[---]	1.17
Effective flowrate at suction valve	[l/h]	250
Theor. displaced volume-flow	[l/h]	875
Effective volumetric efficiency	[---]	0.29
Theor. Volumetric efficiency	[---]	0.54

Comparison diagrams for the Isentropic Efficiency

Diagram 1:

Oillubricated reciprocating compressor,
open design,
compressor-type unknown,
displaced volume 20.7cm^3
number of cylinders unknown

Compressor efficiency means:
Isentropic compression power
compared with shaft power.

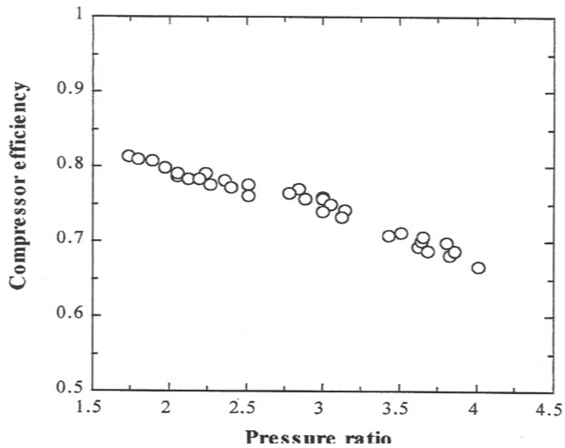

Diagram 2:

Oillubricated axialpiston compressor,
open design,
displaced volume and number of
cylinders unknown,
speed 950 and 1800 RPM

Isentropic efficiency:
Same definition as above.

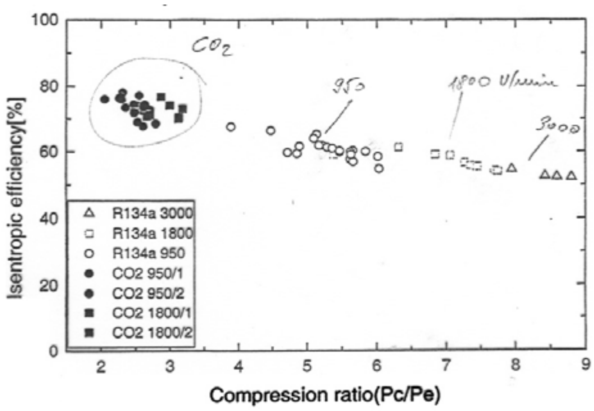

Diagram 3:

Oillubricated reciprocating compressor,
open design,
compressor-type unknown,
displaced volume unknown,
speed 1600-1700 RPM

Isentroper Wirkungsgrad:
Same definition as above

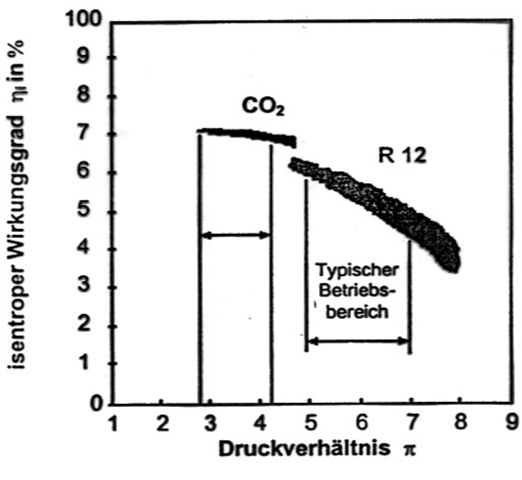

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Diagram 1 P.S.Hrnjak et al., University of Illinois, USA
Experimental Investigation of an Automotive Heat Pump prototype for Military, SUV and compact cars.
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Diagram 2 P.Hrnjak, University of Illinois, USA
Automotive and residential air conditioners and heat pumps
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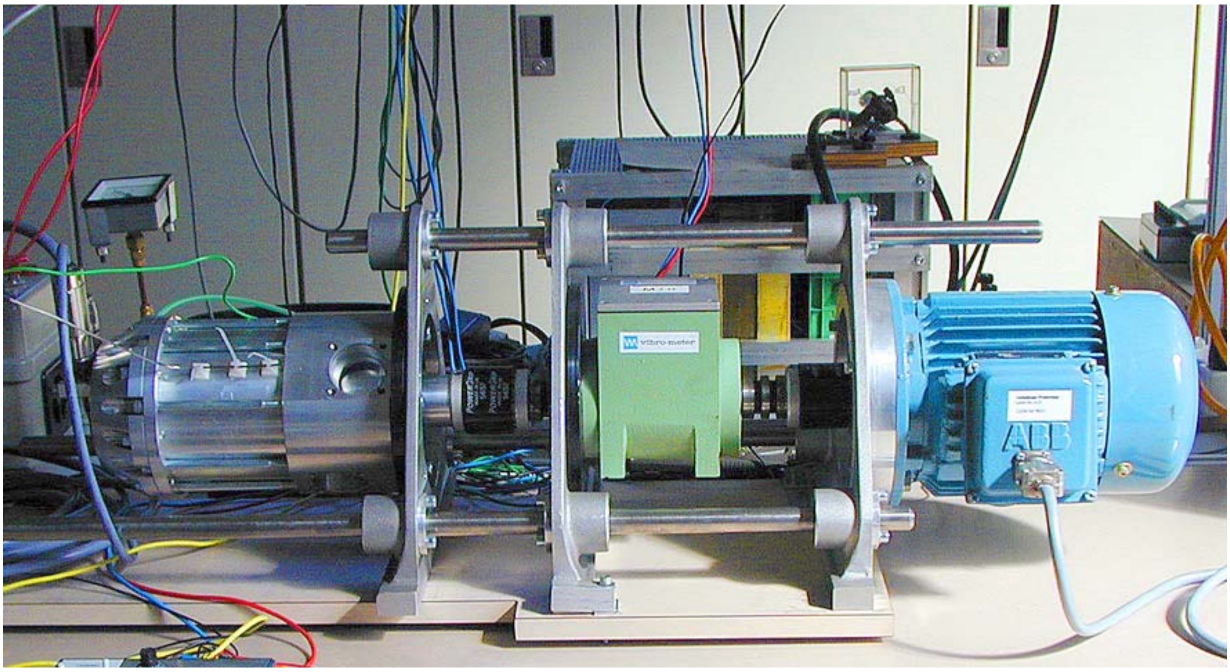
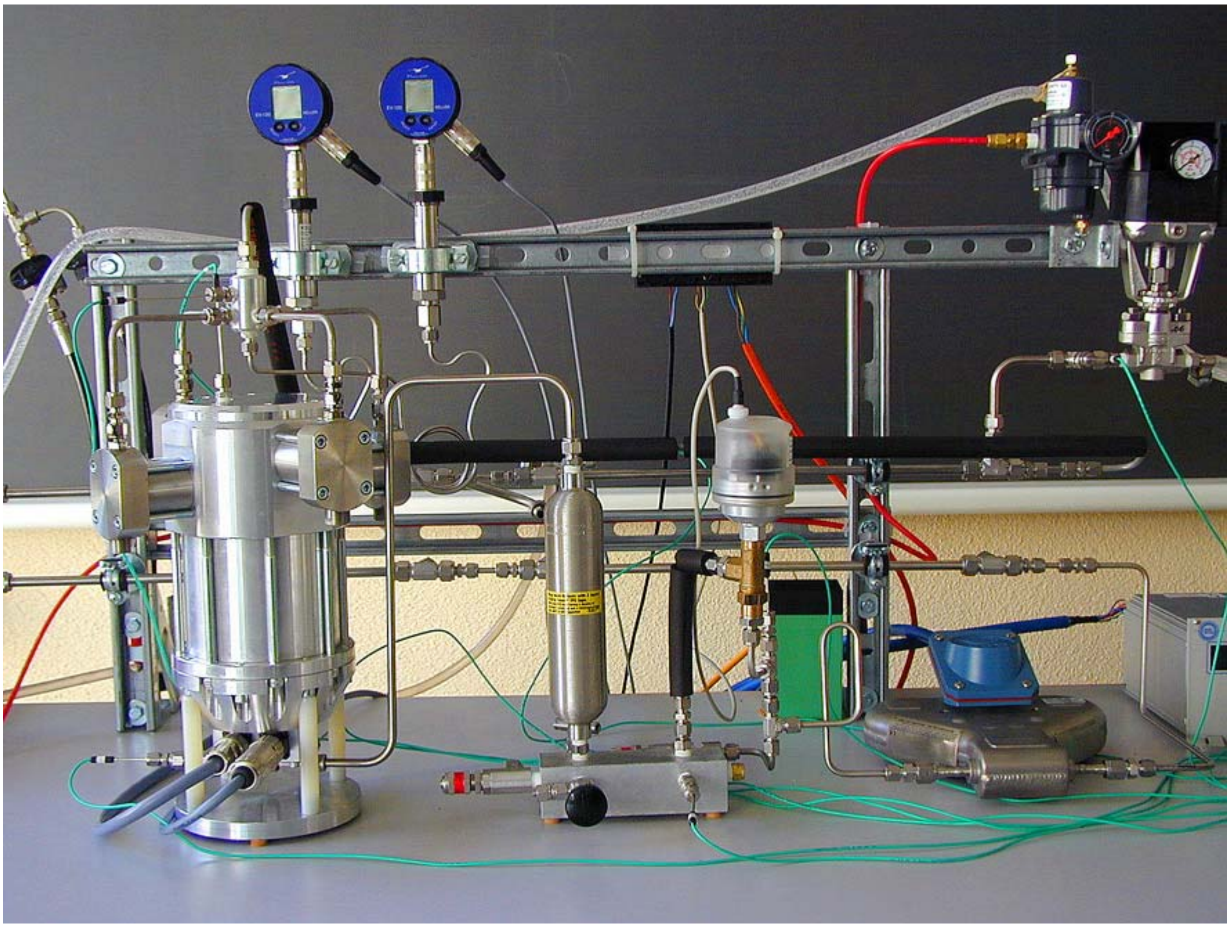
Purdue University, USA, July 2000

Diagram 3 Pettersen, J., G. Lorentzen: Eine neue, effiziente und umweltfreundliche Pkw-Klima-Anlage mit CO_2 als Kältemittel. Luft- und Kältetechnik, 29 (1993), H.3,S.105-111



Pict. 1 Motor measuring set up with torque transmitter and motor brake



Pict. 2 Compressor testloop