FREE PISTON EXPANDER-COMPRESSOR FOR CO_2 – DESIGN, APPLICATIONS AND RESULTS

P. Heyl, H. Quack,
Technische Universität Dresden,
Lehrstuhl für Kälte- und Kryotechnik,
Dresden 01062,
Germany

ABSTRACT

The range of application of CO_2 as refrigerant is presently limited by energetic disadvantages. The thermodynamic efficiency can be improved by the use of an expander-compressor-unit. The paper contains process calculations, the design and first results of this new machine.

INTRODUCTION

The application of the natural refrigerant carbon dioxide is ecologically desirable, safety-wise harmless and represents a long-term refrigerant solution. The range of applications is limited however by energetic disadvantages, essentially by outlet temperatures of the high pressure heat exchanger greater than 25°C. The COP of CO_2 – cycles is quite low, compared with classical compression cycles (Perkins-Cycle) using R134a or comparable refrigerants. This is in contrast to the situation with lower outlet temperatures and applications for water heating with a temperature glide of 50 to 70 K, where carbon dioxide is energetically equal or better than R134a and other refrigerants /1/ and /2/.

The aim of the following investigation is to reduce the energetic disadvantages with suitable and economical justifiable measures in order to enable a broad application of carbon dioxide. Thermodynamic calculations had indicated that with of a two-stage process and an integration of a combined expansion-compression-machine the efficiency of competing systems could be reached.

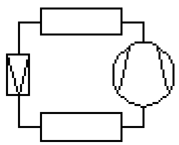
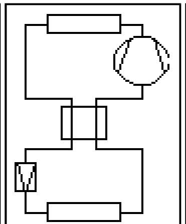
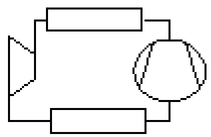
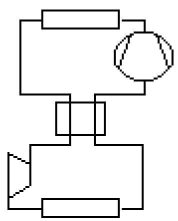
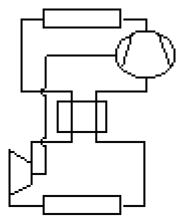
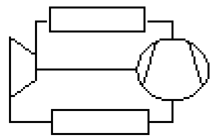
1 IMPROVEMENTS OF CO_2 CYCLES

We investigated several modifications of the simple transcritical process to find out whether a significant improvement of the efficiency would be possible. The COP for different circuits was calculated using the following parameters (see flow diagrams in Table 1 and Table 2):

- evaporation temperature: 0°C and superheating prior to compressor inlet: 10 K
- CO_2 outlet temperature of the medium (p_{MP}) and high (p_{HP}) pressure heat exchanger: 40°C
- temperature difference at the warm end of the internal heat exchanger: 5 K
- isentropic efficiency of the compressor with external actuation: 0.7, also for process (5) and (6)
- isentropic efficiency expander-compressor-unit (internal actuation): compressor- and expansion stage respectively 0.85
- cycle (3) and (4) without the use of the expansion work
- cycle (10), (11) and (12): technical expansion work equal to the technical compression work.

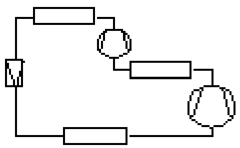
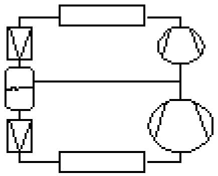
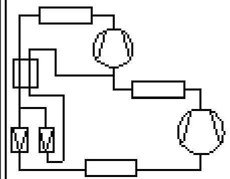
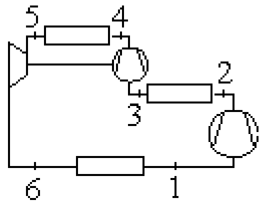
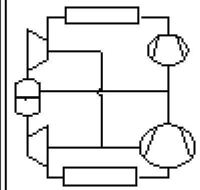
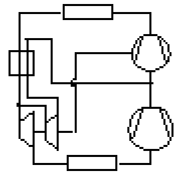
The use of an internal heat exchanger increases the COP negligibly. The use of the work extracting expansion without recovery of the work (Table 1, cycle 3 and 4) instead of the isenthalpic throttling allows an increase of the COP of 10%. The next qualitative steps are the use of the expansion work in these single stage cycles, see Table 1 cycle (5) and (6). The improvement of COP in cycle (6) is already 38,7% compared with cycle (1). With the two stage compression it is possible to reduce the specific compression work. The COP of cycles (7), (8) and (9) in Table 2 exceeds the COP of the single stage cycles given in Table 1, except cycle (6). Furthermore the use of work extracting expansion in the two stage compression cycle, Table 2 cycle (10), (11) and (12), increases the COP up to 3. These COP´s are comparable with conventional systems using R134a, R290 or similar.

Table 1: Single stage CO_2 - cycles

cycle	**(1)**	**(2)**	**(3)**	**(4)**	**(5)**	**(6), see /3/**
flow sheet						
$p_{HP, Optimum}$ [MPa]	10.05	9.81	9.90	9.71	9.45	9.49
COP_{max}	1.99	2.06	2.19	2.19	2.53	2.76

The idea of the new cycle is to realise compression and expansion in one machine. The benefits are small transfer losses between expansion and compression. For a first development of this expansion and compression machine we favour cycle (10) in Table 2. This cycle is easier to realise then cycle (11) or (12). The COP of cycle (10) is improved on 3.28, i.e. 64.8% better than the COP of cycle (1). The rate of the specific refrigeration capacity is only 4.3%.

Table 2: Two stage CO_2 - cycles

cycle	(7), see /4/	(8)	(9), see /5/	(10)	(11)	(12)
flow sheet						
$p_{HP, Optimum}$ [MPa]	11.35	9.50	9.39	9.60	9.11	9.34
$p_{MP, Optimum}$ [MPa]	8.79	5.95	7.68	7.00	5.18	7.88
COP_{max}	2.22	2.30	2.64	3.28	3.18	3.50

2 EXPANDER-COMPRESSOR-UNIT

The application of expander-compressor-units is well known in cryogenic systems like air separation plants. Additionally, its use has also been proposed for normal refrigeration systems with the refrigerant R134a /7/.

2.1 REQUEST AND SELECTION OF SUITABLE DESIGN

Prerequisites for the economic implementation of an expander-compressor-unit are that it is variable over a broad range of parameters (evaporation temperature, outlet temperature of the high and medium pressure heat exchanger), that it has a simple design and is easy to operate.

The following conditions are characteristic for the application in CO_2 cycles:

- high volumetric cooling capacity; this results in relatively small volumetric flow rates
- high pressure differences (e.g. 70 bar for the expansion, 25 bar for the compression).

These conditions recommend the use of a piston machine; even though other authors have discussed the use of a turbomachine, /3/. We propose the basic design shown in Figure 1. The expander-compressor-unit is constructed as a free piston machine. It consists of two double-acting pistons, which are connected by a piston rod. The two cylinders define four working volumes, which are divided in two compression (2 and 3) and two expansion volumes (1 and 4). Each piston divides a cylinder into an expansion and a compression volume.

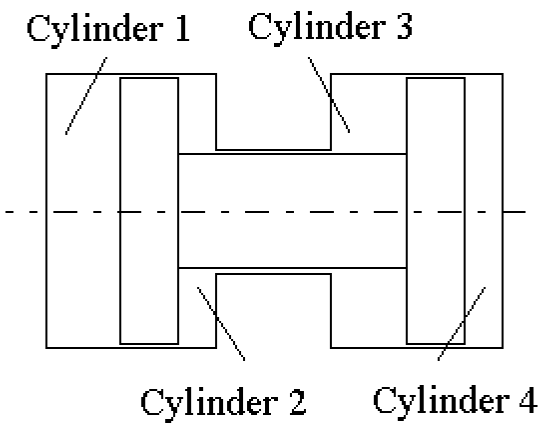


Figure 1: Free piston expander-compressor-unit

2.2 EXPANSION - COMPRESSION - BALANCE OF FORCES

The prerequisite for the functioning of the expander-compressor-unit is

a. that the overall thermodynamic condition $w_{t5,6} \geq w_{t3,4}$ is fulfilled, Table 2 cycle (10) and
b. that the balance of forces of the four pistons is positive over the entire piston stroke in the direction of motion, eq. (1).

$$F_g = \sum_{i=4}^{n} p_{n,Piston} * A_{n,\,Piston} \qquad (1)$$

Whereas condition a) is simple to realise, condition b) is quite tricky. For the complete expansion of the refrigerant the balance of forces is represented in Fig. 2.

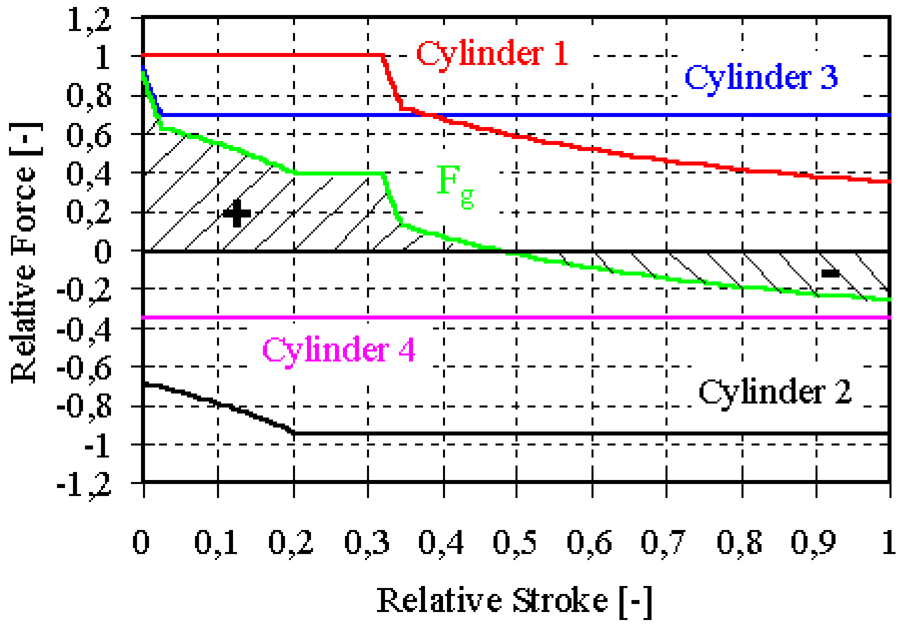


Figure 2: Force-stroke - diagram with the complete expansion of the CO_2

During a piston movement from the left to the right, in cylinder 1 the refrigerant is expanded, in cylinder 2 it is compressed and discharged, in 3 one finds a back expansion of the remaining refrigerant in the cylinder and a sucking in for the next compression process, and in 4 the expanded refrigerant is discharged. From the balance of the four forces (F_g in Fig. 2) it follows that over the length of the piston stroke the resulting force F_g changes its sign. A continuous operation therefore requires an energy-storage system in the first part of the piston stroke, which releases the stored energy in the second part of the stroke. The energy storage could take place for example by means of an inertia of moving mass, by a mechanical spring or by a gas spring. So far we were not able to find a simple straight forward solution. One solution is to terminate the expansion in that position of the stroke where the resulting balance of forces stops the movement of the piston. In this case the expansion remains incomplete. Our present solution is, that after the filling of the expansion volume the working process is concluded and the exhaust valve is opened. The expander operates according to the "full pressure principle" (Fig. 3; "Volldruckmaschine").

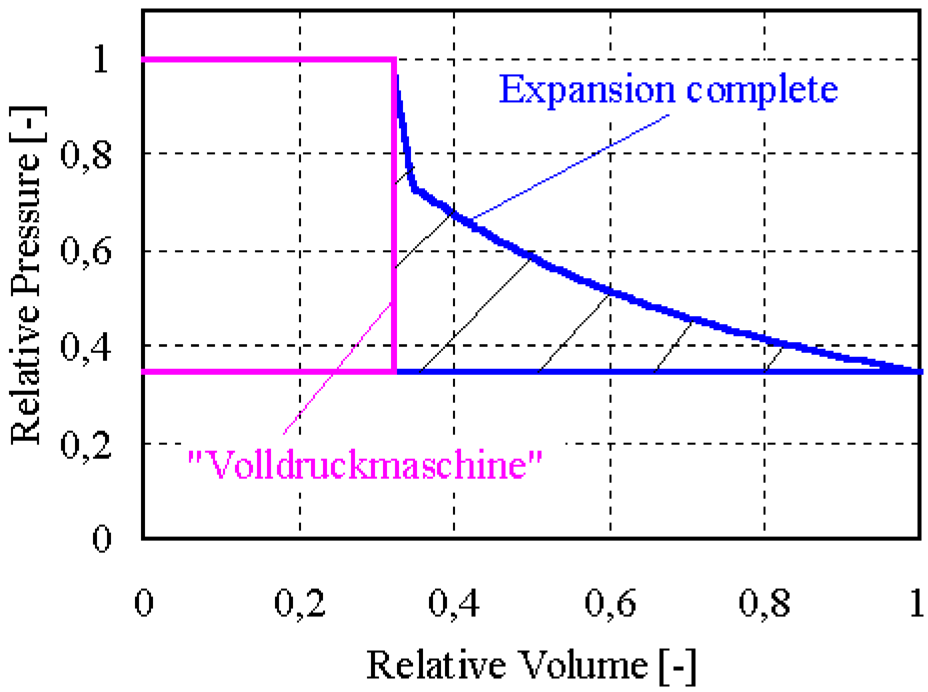


Figure 3: Pressure-volume - diagram

The shaded area represents the unavoidable loss in case of an expansion which is executed in the "Volldruckmaschine" instead of a complete expansion.

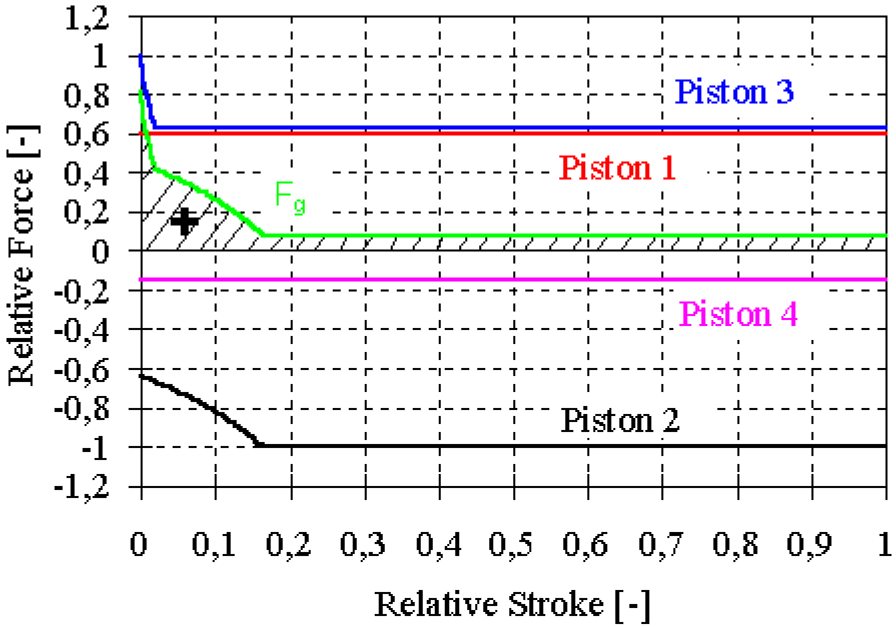


Figure 4: Force-stroke - diagram of the "Volldruckmaschine"

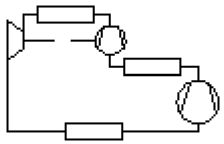
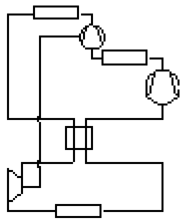
The resulting balance of forces of the expander-compressor-unit operating with the "full pressure principle" is positive over the entire stroke (Fig. 4). The functioning of the machine is ensured.

2.3 EFFECTIVENESS OF THE "VOLLDRUCKMASCHINE"

Concerning COP, the "full pressure principle" ("Volldruckmaschine") cannot achieved the COP of a complete expansion, see Table 3.

Table 3: COP with "Volldruck" - Principle

cycle	**(13)**	**(14)**
flow sheet		

		
p_{HP} [MPa]	14..31	12.52
p_{MP} [MPa]	8.97	8.81
COP_{max}	2.88	2.78

The "Volldruck"-principle uses only about 78% of the available power of the expansion cycle (13). Multiplicated with the efficiency of the expansion side of the expansion-compression-machine (85%) a total efficiency of 0.78*0.85 = 0.66 results compared to the isentropic expansion. The COP is approximately 12% smaller then in cycle (10) and the pressure at medium and high pressure level is much higher. It is important to note, that the COP is nearly independent of the high pressure level.

3 "VOLLDRUCKMASCHINE" - Prototype

3.1 OPERATIONAL BEHAVIOR

The functioning of the "Volldruckmaschine" has to be possible over a broad parameter range, i.e. for different evaporation temperatures and different outlet temperature of the medium and high pressure heat exchanger. Once a machine has been built, it has a fixed volume ratio between the expansion side and the compression side. Thus this machine demands a fixed ratio between the density at the second compression stage suction side and the density at the end of the expansion, eq.(2). With the "Volldruckmaschine" this is equal to the density on the inlet of the expansion stage, Table 2 cycle (10).

$$Z = \frac{\rho_3}{\rho_5} \quad \textit{fixed density ratio} \qquad (2)$$

This relation determines the operating points for parameters deviating from the design point. A second variable is the frequency of the machine. This can be used for the adjustment of the mass flow. The design point of the machine is selected in such a way that a broad parameter range is possible with optimum COP and a moderate high pressure level. Actually, the deviations from the maximum COP at outlet temperatures from 35°C to 45°C and a evaporation temperatures from -10°C to +10°C are less than 1.7%.

3.2 Construction

In cooperation with Maximator, a medium sized company for hydraulic and pneumatic components, a expansion-compression-machine for carbon dioxide has been developed. Some details are shown in Figure 5. There is a symmetry plane in A-A (two more cylinders, not shown here). In cylinder I the carbon dioxide compressed and in cylinder II expanded. The expansion valves are opened and closed according to the piston position monitored via B.

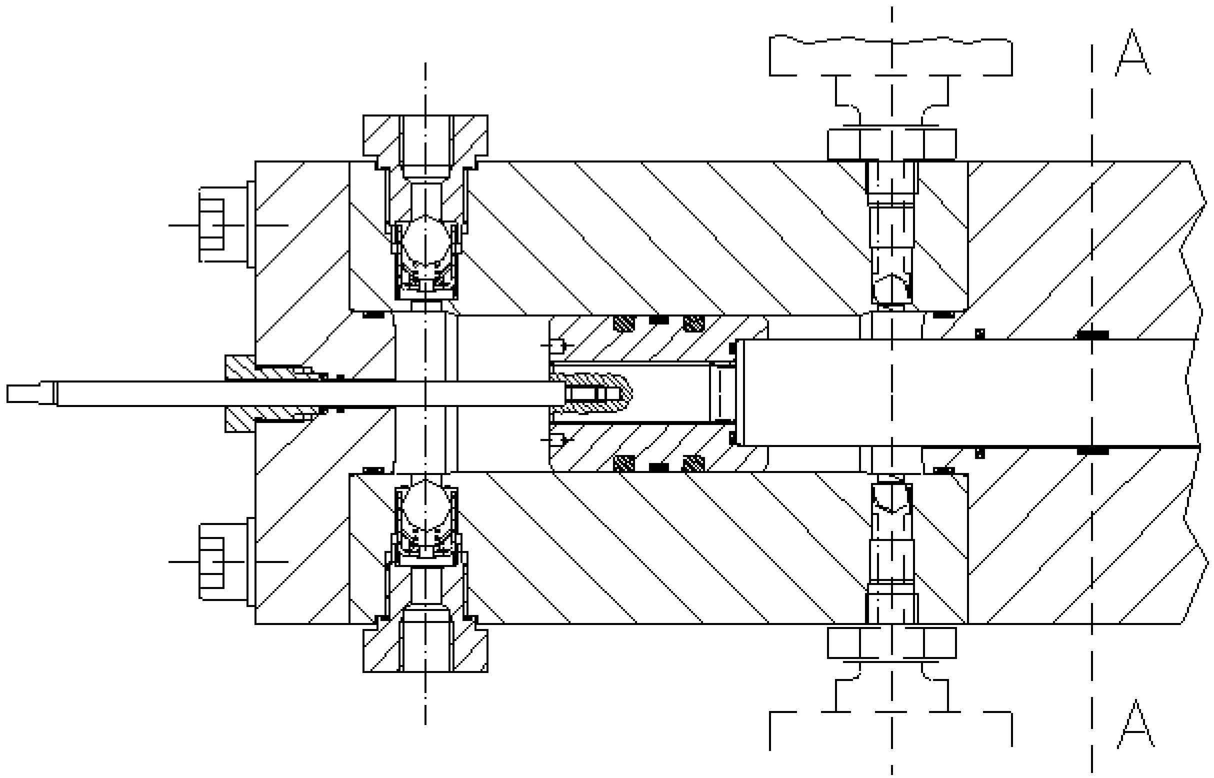


Figure 5: Detail of the expansion-compression-machine

The important parameters of the expansion-compression-machine are:

- piston stroke 100 mm
- piston and piston rod diameter 50 and 34 mm respectively

- frequency at nominal condition 0.7 Hz
- fixed density ratio 0.55
- the oil free operation (without lubricant) is possible.

3.3 First Results

In Figure 1 the pressure-volume - diagram for the compression of carbon dioxide from 8 MPa to approximately 11 MPa and the expansion from 10 MPa to approximately 3 MPa is shown. The throttle losses of the different inlet and outlet valves are negligible. The reason for the pressure difference between the high pressure level in the expansion and the compression cylinder is a filter in front of the inlet valves of the expansion side. The small sized tubes diameter leading into the expansion-compression-machine are responsible for the pressure decrease from 8 MPa to 7.5 MPa for the suction of the carbon dioxide in to the compression cylinder and also in to the expansion cylinder.

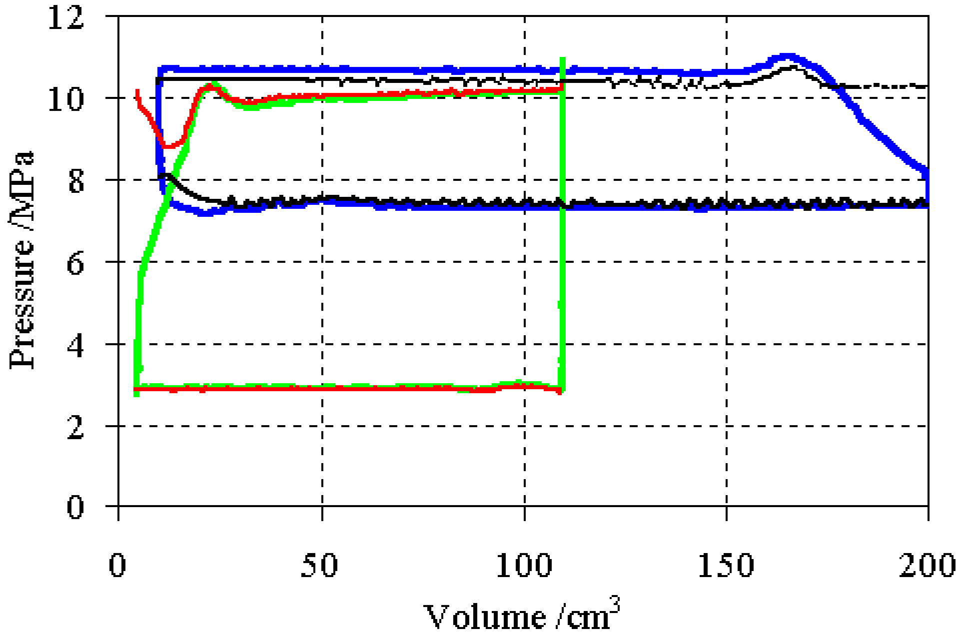


Figure 6: p, V - Diagram of the expansion and compression

87% of the energetically available expansion work is used for the compression.

4 CONCLUSION

The thermodynamic efficiency of the transcritical process with carbon dioxide can be improved by the application of an expander-compressor-unit. The test of this machine will be continued for different applications in refrigeration and heat pump cycles.

ACKNOWLEDGEMENTS

The authors would like to thank the German Ministry of Education and Research (BMWi, project number: 0327221) and the companies Bock-Kältemaschinen, Behr Industrietechnik Mylau, Egelhof, Fuchs/DEA, Konvekta, Leser, Maximator and Wilo for supporting this project.

NOMENCLATURE

Nomenclature:			**Subscripts:**	
ρ	density	(kg/m^3)	0	evaporation
h	stroke	(mm)	g	total
A	surface	(mm^2)	s	isentropic
COP	coefficient of performance	(-)	HP	high pressure
F	force	(kN)	LP	low pressure
p	pressure	(MPa)	MP	medium pressure
Δt	temperature difference	(K)		
w_t	technical work	(kJ/kg)		

REFERENCES

/1/ Neksa, P.; Rekstad, H.; Zakeri G. H., 1997, CO_2 Prototype Hot Water Heat Pump characteristics, system design and experimental results, Proceedings: Heat Pump Systems, Energy Efficiency, and Global Warming, p. 165-172, Linz, Austria

/2/ Heyl, P.; Kraus, W.E.; Quack, H., 1997, CO_2-Wärmepumpe zur Warmwasserbereitung - Ergebnisse und Erfahrungen, DKV-Tagungsbericht 24 Bd. II/2, p. 87-100, Hamburg, Germany

/3/ Robinson, D.M.; Groll, E.A., 1997, Efficiencies of Transcritical CO_2-Cycles with and without an Expansion Turbine, International Journal of Refrigeration, 1998, Volume 21, Number 7

/4/ Horst, L., 1911, Verbesserte CO_2 - Expansions - Kältemaschine, Zeitschrift für Sauerstoff- und Stickstoffindustrie 12, p. 241 - 246

/5/ Inagaki, M., Sasaya, H., Ozakli, Y.: Pointing to the Future: Two-Stage CO_2 Compression. Proceedings Heat Transfer Issues in "Natural Refrigerants", University of Maryland, USA, Part I, p. 58 - 74

/6/ Quack, H.; Kraus, W.E., 1994, Carbon Dioxide as a Refrigerant for Railway Refrigeration and Air Conditioning, Proceedings: New Application of Natural Working Fluids in Refrigeration and Air Conditioning, p. 489 - 494, Hannover, Germany

/7/ Hartmann, K., 1994, Kreisverbesserung durch Zwei-Phasen-Entspannungsturbine, DKV-Tagungsbericht 21, vol. II/1 p. 241 - 252, Bonn, Germany

---

LIBRE EXPANSION-COMPRESSEUR POUR CO_2-CONSTRUCTION, UTILISATION ET RESULTATS

RÈSUMÈ: L'étendue des applications du dioxyde de carbone comme réfrigérant est actuellement limitée par les inconvénients énergiques. L'efficacité thermodynamique peut-être améliorée par l'utilisation d'une machine à extension - compression. Le papier contient de différentes possibilités de calculs de processus, la conception et les premiers résultats d'essai de cette nouvelle machine.