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RESEARCH ON THE BEHAVIOR
OF REFRIGERATION COMPRESSORS USING CO₂ AS THE REFRIGERANT

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0. ABSTRACT

Due to the thermophysical properties of CO₂, the pressures of the refrigeration process are significantly higher than for commonly used fluids. This brings about particular working conditions for the compressor from which it may benefit if the design is adapted to these conditions. The research activities, which are carried out as an ongoing PhD thesis and are generously supported by the Joule Research Program of the European Commission and international refrigeration compressor manufacturers, are a contribution to develop a highly efficient semi-hermetic type compressor for CO₂ in heat pump applications. To be able to design such a machine, detailed knowledge on the inner compression process and the mechanical efficiency of the compressor drive must be achieved. In this paper results from first experimental and theoretical investigations on an open type CO₂ compressor are discussed. In a further step the gained experience can be applied for a semi-hermetic or hermetic type compressor.

1. INTRODUCTION

The regulations of the Montreal Protocol prescribe that CFCs should no longer be used as refrigerants in industrialized countries and HCFCs are only an interim solution until the year 2040 worldwide [2]. In the European Union the year 2015 is proposed as a phase-out date and in some countries like Germany R22 must be abandoned before the 1.1.2000 [12]. Looking for a final solution and taking furthermore possible future regulations for greenhouse gases into account, substances with negligible global warming potential (GWP) like the natural fluids turn out to be a promising alternative for certain fields of refrigeration technology. Some of them like the hydrocarbons and ammonia show an unfavorable safety behavior. If intoxicity and non-flammability of the refrigerant are required a choice could be R744 (CO₂), that was once widely used in ship refrigeration before being substituted by R12 and later by R22 after the 1950s [6, 7].

Table 1 Selected characteristics and properties of various refrigerants

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>R12</th>
<th>R22</th>
<th>R134a</th>
<th>NH₃</th>
<th>CO₂</th>
</tr>
</thead>
<tbody>
<tr>
<td>Natural fluid</td>
<td>No</td>
<td>No</td>
<td>No</td>
<td>Yes</td>
<td>Yes</td>
</tr>
<tr>
<td>Ozone Depletion Potential</td>
<td>1.0</td>
<td>0.05</td>
<td>0.0</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>Global Warming Potential</td>
<td>7100</td>
<td>1500</td>
<td>1200</td>
<td>-</td>
<td>1 (0)</td>
</tr>
<tr>
<td>Critical temperature</td>
<td>112.0</td>
<td>96.2</td>
<td>101.2</td>
<td>132.4</td>
<td>31.1</td>
</tr>
<tr>
<td>Critical pressure</td>
<td>603.3</td>
<td>4.99</td>
<td>4.07</td>
<td>11.27</td>
<td>7.38</td>
</tr>
<tr>
<td>Flammable or explosive</td>
<td>No</td>
<td>No</td>
<td>No</td>
<td>Yes</td>
<td>No</td>
</tr>
<tr>
<td>Toxic/irritating decomposition products</td>
<td>No</td>
<td>No</td>
<td>No</td>
<td>Yes</td>
<td>No</td>
</tr>
<tr>
<td>Approximate relative price</td>
<td>1</td>
<td>1</td>
<td>0.2</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>Relative volumetric refrigeration capacity</td>
<td>1</td>
<td>1.6</td>
<td>1.6</td>
<td>8.4</td>
<td></td>
</tr>
</tbody>
</table>

In Table 1 selected characteristics and properties of various refrigerants are listed. It is obvious that the natural substances are preferable from an environmental point of view. The low price of CO₂ and its high volumetric refrigeration capacity are strong arguments for the application of this substance.
Possible applications of CO₂ as a refrigerant in the field of refrigeration technology can be seen in automotive air conditioning and heat pump applications [1, 4, 7]. Expecting an installation rate of automotive air conditioning systems of 50% for the coming years in Europe four to five million units are to be produced.

Heat pump systems working with CO₂ may substitute old central heating units for domestic heating systems with radiators with high temperatures in the hydronic system [1, 4]. High temperatures can be achieved by a CO₂ heat pumping system as its working process is transcritical. Transcritical means that the heat input is at a subcritical 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 and therefore in energetic advantages for the process in a heating task with a big temperature glide.

The theoretical transcritical process is shown in Figure 1. Looking at this process with its isentropic compression (1→2), its isobaric heat output (2→3), its isenthalpic throttling (3→4) and its isobaric heat input (4→1) the remarkably high pressures resulting from the CO₂ properties are obvious.

2. CO₂ TEST RIG

To examine especially the performance of a CO₂ compressor a test rig has been set up. With this test rig the working process shown in Figure 1 has been investigated. The energetic and volumetric compressor performance has been obtained from energetic balances at the heat exchangers and by directly measuring the inner compression process [11].

3. COMPRESSOR FOR CO₂

3.1 Designing Steps for a Hermetic Type Compressor

Compared to open-type compressors the semi-hermetic or hermetic type compressors have the advantage that a shaft seal is not required, as the driving motor is arranged together with the compressor unit in a completely sealed housing. Especially for high crank case pressures an effective shaft seal for open type compressors is costly [5]. Therefore, it is recommendable to design a semi-hermetic or hermetic type compressor when using CO₂ as the refrigerant. To be able to optimize the set of construction parameters for a CO₂ compressor the various power losses ΔP_v, ΔP_m and ΔP_a shown in Figure 2 have to be specified and assigned to their causes which are explained in the following.

A compressor is designed to compress the suction gas with the pressure p_in and the temperature T_in to the pressure p_out at the temperature T_out. For this process the inner or indicated compression power P_i is required by the compressor. Due to throttling losses in the valves, leakage of the gas in the cylinder and heat transfer effects the power P_i exceeds the theoretical compression power by the losses ΔP_v. Due to mechanical losses ΔP_m of the realistic compressor the shaft power P_s is larger than the inner power P_i. At a semi-hermetic or hermetic type compressor, the electrical power P_e is required. This electrical power exceeds the shaft power P_s since the motor of the compressor has the energetic losses ΔP_a. The total isentropic efficiency η_tot of a semi-hermetic or hermetic type compressor can be calculated as η_tot = η_i · η_m · η_e [5].
3.2 Ideal and Real Compression Process

From theoretical considerations the indicated isentropic efficiency $\eta$ and the volumetric efficiency $\eta_{vul}$ of the inner compression process are expected to be extraordinarily high when using CO$_2$ as the refrigerant. The inner compression process as well as the theoretical compression process are shown in Figure 3.

The isentropic efficiency $\eta_{i} = \frac{w_i}{w}$ is the coefficient of the theoretical work $w_i$ that is in the ideal case necessary to compress the gas and the indicated work $w_i$ that is actually needed by the real compressor. The isentropic 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 of the valves can be expected to be negligible. Therefore, the relatively small valve losses and consequently a high isentropic efficiency will be achieved [5, 14].

The volumetric efficiency $\eta_{vul} = \frac{V_i}{V_g}$ is the coefficient of the cylinder volume $V_i$ that is actually filled with the suction gas at the end of the sucking and the geometric stroke volume $V_g$. The volumetric efficiency of the CO$_2$ compressor gains mainly from the small pressure ratio, as given when using CO$_2$ as it brings about a short re-expansion of the gas from the clearance volume of the cylinder and therefore a early opening of the suction valve [14].

3.3 Characteristics of the Investigated Compressor Prototype

A schematic drawing and some construction data of the investigated CO$_2$ compressor are shown in Figure 4. The design of the oil lubricated CO$_2$ compressor 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$_2$. The former piston diameter of the original construction is shown in Figure 4 with a dotted line. This method was selected to limit the designing and production cost of the prototype CO$_2$ compressor. As the crankshaft itself was not modified, the stroke of the compressor was constant. Therefore, the stroke to bore ratio of 1.71 is rather huge leaving only little space to install the cylinder valves. To seal the cylinder from the crankcase four piston rings per piston are supplied. The maximum crankcase overpressure is limited to 13 MPa (1885.4 psi) controlled by two overpressure valves [3].

3.4 Inner Compression Process

To measure the indicated compression process, a pressure transducer was installed in one of the cylinders of the compressor [11]. The measured data went online together with a signal of the actual piston position into a personal computer where the isentropic and volumetric efficiency of the compression process are indicated. Figure 5 gives an example of a measured indicated diagram of the investigated CO$_2$ compressor at a number of revolutions of 840 min$^{-1}$, a suction pressure of 4 MPa (580 psi) and a delivery pressure of 10 MPa (1450 psi).
The isentropic efficiency of the process shown in Figure 5 is 0.83. Its volumetric efficiency is 0.89. Although the efficiency of the compression process is already comparatively high, it still has a potential for improvement [11]. From Figure 5 it gets obvious that the delivery valve losses are rather large. Furthermore, it can be derived from the measured compression line that leakage at the cylinder occurs. The simulation of the process in Figure 6 was calculated without the consideration of leakage and heat transfer effects. The isentropic efficiency of the simulated process is 0.82 and the volumetric efficiency is 0.89. When comparing the compression line of the measured process shown in Figure 5 with the simulated one shown in Figure 6, the effect of leakage gets obvious.

At the compression chamber leakage may occur at the suction valve, the delivery valve, and the piston rings. It is common sense that leakage affects the efficiency of the compression process [9]. Detailed experimental and theoretical investigations are under way to estimate the influence of leakage on the energetic improvement of the process. An attempt to reduce the losses of the delivery valve is to increase its free-space sectional area. Another option is to increase the maximum valve lift and the valve bore of a delivery reed valve. If the second of these methods is applied and the bore and maximum lift of the delivery valve are increased by 25% the process shown in Figure 7 is obtained from the simulation. With the modified data set the energetic losses of delivery valve were reduced by 70.5% which increased the isentropic efficiency of the process to 0.90. The volumetric efficiency of the process is not remarkably influenced.

![Figure 5 Measured indicated diagram](image)
![Figure 6 Simulated working process](image)
![Figure 7 Simulated working process with new data set for delivery valve](image)

Another factor that has to be investigated is the effect of heat transfer phenomena between the gas and the cylinder. The superheating of the gas in the suction chamber of the compressor needs to be regarded, too, especially when concentric valves are supplied. The heat transfer properties of CO₂ are by far higher than of commonly used refrigerants. Consequently, it is interesting to investigate the effect of heat transfer phenomena in a CO₂ compressor as they influence the design of the compressor, such as the geometry of the cylinders, the gas in- and outlet as well as the cooling of the cylinders and the driving motor [5].

3.5 Driving Mechanism of the Compressor

Due to the large difference between the suction and the delivery pressure of the CO₂ process, the load on the driving mechanism of a CO₂ compressor is rather high even at a small piston diameter. To predict the reliability of the journal bearings of the investigated compressor prototype and to obtain theoretical knowledge on their design the moving shaft centers of the journal bearings were calculated in a cyclic orbit [8] with a method showing accordance with measured steady-state eccentricities [13]. The Figures 8 to 13 show the results of these simulations of the moving shaft centers in a cyclic orbit. Major wear of the bearing has to be expected when a contact of the surface roughness occurs, normally at a
relative eccentricity of around 0.95 [13]. Figure 8 shows the calculated shaft center orbit of one of the crankshaft bearings and Figure 9 of the large connection rod bearing at the conditions of the measured compression process as shown in Figure 5. It has been expected due to the way the investigated compressor was designed as described in 3.3 by reducing the cylinder diameter of an already produced refrigeration compressor that the dimensions of these bearings match the load on the driving mechanism. No major wear of these bearings must be expected especially when the number of compressor revolutions is increased to above 1000 rpm.

Figure 8 Calculated shaft center orbit moving of crankshaft bearing
Figure 9 Calculated shaft center orbit of large connection rod bearing
Figure 10 Calculated shaft center orbit of piston pin bearing

More critical is the design of the piston pin bearing as the piston diameter with decreasing cylinder bore is rather small and does therefore not offer much space. Furthermore the relative movement of the piston pin in this bearing is quite poor which is unfavorable for hydrodynamic lubrication. As shown in Figure 10 the calculated moving of the piston pin center in a cyclic orbit exceeds the value of 0.95, meaning critical conditions. Experiences from experimental investigations have already confirmed these simulation results [3].

Figure 11 Calculated shaft center orbit of crankshaft bearing in a cyclic orbit for a 25 % increase of piston diameter
Figure 12 Calculated shaft center orbit of large connection rod bearing for a 25 % increase of piston diameter
Figure 13 Calculated shaft center orbit of piston pin bearing for 25 % increase of piston diameter allowing an increase of the bearing dimensions

Trying to reduce the critical conditions at the piston pin bearing a parameter variation was carried out. As having proposed earlier in this paper an increase of the piston diameter improves the compressor performance. Besides the reduction of energetic losses of the compression process by larger valves, the piston pin bearings can be redesigned. Looking at a corresponding 25 % increase of the piston diameter according to the assumption of the delivery valve enlargement the diameter of the piston pin may be increased from 10 mm (0.39 in) to 15 mm (0.59 in) and the length of the piston pin bearing can be increased from 17 mm (0.67 in) to 25 mm (0.98 in). With respect to the increase of the piston mass the simulated shaft center orbit of the concerned bearings are shown in Figures 11 to 13. To allow a direct comparison of the effects of these parameter variations to Figures 8 to 10 the remaining set of parameters for the bearings was kept constant. Figure 13 shows that an enlargement of the piston pin bearing is a promising attempt to reduce the critical conditions of this bearing. Due to the assumed increase of the piston diameter the load on the driving mechanism of the compressor rises.
as well. Figures 11 and 12 confirm that the dimensions of the unmodified crankshaft bearing and the large connection rod bearing still match the requirements concerning lifetime under these modified conditions. Parallel to these attempts to improve the bearing performance investigations of a suitable lubricant for the CO₂ compressor are underway.

CONCLUSION

CO₂ is a safe and environmentally friendly refrigerant. Especially for refrigeration applications with high leakage rates, like automotive air conditioning systems, the use of the environmentally benign fluid R744 (CO₂) as the refrigerants is a very promising option. Applying CO₂ in a heat pumping system its energetic efficiency could under certain conditions benefit from the temperature glide of the transcritical CO₂ process.

Due to the thermophysical properties of CO₂ the working pressures of a CO₂ cycle are rather high and require a special design of the systems components. Especially for the compressor the fluid properties of CO₂ are favorable for its energetic and volumetric efficiency. On the way to design a hermetic type compressor for CO₂ that gains from these advantages it is recommendable to start the investigations with an open type compressor to be able to analyse the occurring losses of this machine. Parallel to the experimental investigations, theoretical models of the CO₂ compressor for means of optimizations must be applied.

The current investigations are a contribution to designing a highly efficient compressor for CO₂ as the refrigerant and by doing so, to support the CO₂ refrigeration technology to become a sensible alternative in tomorrow’s world of refrigeration.

REFERENCES