Two-Stage Rolling Piston Carbon Dioxide Compressor

Nelik Dreiman
Tecumseh Products Company

Rick Bunch
Tecumseh Products Company

Yun Horn Hwang
University of Maryland

Reinhard Radermacher
University of Maryland

Follow this and additional works at: http://docs.lib.purdue.edu/icec

Dreiman, Nelik; Bunch, Rick; Hwang, Yun Horn; and Radermacher, Reinhard, “Two-Stage Rolling Piston Carbon Dioxide Compressor” (2004). International Compressor Engineering Conference. Paper 1630. http://docs.lib.purdue.edu/icec/1630

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information. Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
TWO-STAGE ROLLING PISTON CARBON DIOXIDE COMPRESSOR.

Nelik Dreiman\textsuperscript{1}, Rick Bunch\textsuperscript{1}, Yunho Hwang\textsuperscript{2}, and Reinhard Radermacher\textsuperscript{2}

\textsuperscript{1} Tecumseh Products Company, 100E. Patterson St., Tecumseh, MI 49286 USA
Tel.: 517/423 – 8582; FAX: 517/423 – 8426; E-mail: ndreiman@tecumseh.com
\textsuperscript{2} CEEE, Department of Mechanical Engineering, University of Maryland, College Park, MD 20742
Tel.: 301/405 – 5247; FAX: 301/405 – 2025; E-mail: yhwang@eng.umd.edu

ABSTRACT

A prototype hermetic rotary compressor has been developed for use with carbon dioxide as natural working fluid. Two-stage compression cycle has been employed to reduce effect of the high pressure usual for the compression cycle of carbon dioxide compressors. The paper presents detailed description of the compressor lubrication system, suction and discharge gas flow pattern, discharge valve design, results of the analytical study of the journal and thrust bearings, and some experimental test data. The modular design of the compressor elements helps to simplify assembly, reduce production cost, and improve performance of the compressor.

1. INTRODUCTION

In the last few years’ carbon dioxide received increasing attention as possible replacement of fluorocarbon-based refrigerants used at present for vapor compression cycle technology. One concern with R–744 refrigerant is the effect of high operating pressure. Large difference between compressor discharge and suction pressures can trigger higher leak losses, increased load on the bearings, etc.. Fatigue problems due to higher impact velocities (lift and closure of the valve, for example) when compressing such a relatively dense gas as CO\textsubscript{2} (density is \approx 5 times higher than that of R-22) must be also considered. The large pressure difference also required a larger torque of the compressor motor. The fittings, piping, bolts, mufflers, and housing of the compressor has to be stronger to meet the high working pressure of R–744 and satisfy ASTM Standard Specifications for Construction of Pressure Vessels and UL Requirements for a compressor enclosure strength. Developed concept of the compressor aims at eliminating the drawbacks specified above by use of two-stage compression process that eases the compressor workload [Dreiman and Bunch, 2003].

2. DETAILED DESCRIPTION OF THE COMPRESSOR DESIGN.

The modular design compressor shown in Fig.1 provides a positive displacement two cylinder, two-stage rotary hermetic compressor comprising the lower end compression module and the upper end compression module which are coaxially coupled to the opposite axial ends of the electric motor module to form vertically upstanding compressor. The two-stage compression cycle operates with three levels of pressure, which will be referred to as low, intermediate and high. The low-pressure suction gas is supplied directly to the lower end compression module 2 (first stage). As the refrigerant gas is compressed in the first stage it is discharged directly through the discharge port (not shown) and discharge pipe 100 to the unit cooler (not shown) located outside of the compressor casing. Subsequently, the cooled refrigerant gas is introduced into the intermediate pressure cavity 102 through the fitting 115 of the electric motor module 4 which is in fluid communication with the cavity 103 of the lower end compression module 2 through the oil passages 101 which allow lubricant to be reclaimed by the oil sump 18. Introduction of the cooled refrigerant gas into the electric motor cavity 102 helps to cool the motor and the lubricant, and reduce the heat transfer to the suction gas of the first stage. The refrigerant gas introduced to the upper end compression module 3 through the suction port 104 made in the wall 105, discharged in cavity of the module 3, and distributed to a unit through the pipe 120. In the exemplary embodiment of the compressor, the lower end module are similar in design to the upper end module and both modules comprise integral parts combining main bearing 1, housing segment 5, optional axial or radial suction and discharge ports, compressor external mounting 6 (lower module), and rotary pumps 7.
2.1 Lubrication system
The positive displacement oil pump 8 is operable associated with the end of crankshaft 17, all of which are submerged in the oil sump 18. The crankshafts of the compressor rotary pumps are couplet so that the oil delivery channels 30 of both crankshafts along the axis of rotation are in fluid communication and connected also to the recess 31 machined at the top end of the upper crankshaft [Dreiman and Bunch, 2003].

![Diagram of lubrication system]

Fig. 1: Two-stage carbon dioxide hermetic compressor

Oil delivered to the crankshafts lubrication channel 30 is able under centrifugal forces to flow into a series of radially extending passages of the crankshafts to the journal bearings surfaces. The rest of the lubricant supplied to the cavity 31 (see Fig. 2a), is directed by a downwardly inclined channel 29 machined in the upper outboard bearing.

![Diagram of lubrication system]

Fig. 2: Lubrication system of the compressor

Oil delivered to the crankshafts lubrication channel 30 is able under centrifugal forces to flow into a series of radially extending passages of the crankshafts to the journal bearings surfaces. The rest of the lubricant supplied to the cavity 31 (see Fig. 2a), is directed by a downwardly inclined channel 29 machined in the upper outboard bearing.
axial channel 32 machined in the upper cylinder, downwardly inclined channel 33 machined in upper pump main bearing, to the annular cavity 34 defined by recessed area in upper crankshaft 35 and upper journal 36. The system of such interconnected channels are used for delivery of the oil to the bearings and final return of the oil to the sump 18. The oil sump of the upper end compression module 3 separated by partition 105 from the first stage oil sump 18 and used to provide lubrication of the vane. To improve lubrication at start, blind holes 40 have been machined at the outlet part of radially extended lubrication channels of the crankshaft and in the upper planar surfaces of the lower and upper kit shaft eccentrics 37 (see Fig. 2b, 2c) in the way that remaining oil fill dead-end holes cavity at the compressor stops, and supplied oil immediately to the inner surface of the bearings surfaces due to centrifugal force at the compressor start.

2.2 Journal Bearings
The main-, eccentric-, and outboard bearings load of lower and upper pump have been analyzed by application of ROTBALCL and HYDAUX software programs (developed by Tecumseh Products Company) assuming ARI operating conditions of the compressor, with oil viscosity of 3 centipoises and refrigerants R – 744 and R-410A. The minimum film thickness, maximum and average loads magnitude are summarized for R-744 refrigerant in Table 1.
Relatively high load magnitudes and comparable thinner oil film have been predicted for the outboard bearing of the lower pump, eccentric and outboard bearing of the upper pump. Fig. 3 shows maximum load magnitudes for modified main-, eccentric-, outboard bearing of the lower and upper pumps. Increase of the bearings diameter slightly reduces max. load and increases oil film thickness, but value of the specified above parameters still higher for the compressor with R-744.

<table>
<thead>
<tr>
<th>Bearing* Position</th>
<th>Load Magnitude, kgm</th>
<th>Minimum Film Thickness, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Maximum</td>
<td>Average</td>
</tr>
<tr>
<td>MBL</td>
<td>25.8551</td>
<td>11.6882</td>
</tr>
<tr>
<td>EL</td>
<td>74.9550</td>
<td>33.3372</td>
</tr>
<tr>
<td>OBL</td>
<td>49.4441</td>
<td>22.1001</td>
</tr>
<tr>
<td>MBU</td>
<td>14.2992</td>
<td>6.3574</td>
</tr>
<tr>
<td>EU</td>
<td>45.2896</td>
<td>20.2157</td>
</tr>
<tr>
<td>OBU</td>
<td>31.5346</td>
<td>14.8638</td>
</tr>
</tbody>
</table>

*MBL-main bearing; EL-eccentric bearing; OBL- outboard bearing (lower pump)
MBU-main bearing; EU-eccentric bearing; OBU- outboard bearing (upper pump)

2.3 Thrust bearing
A single cylinder rotary compressor crankshaft thrust surface has (as usual) a half-moon shape and located on the side of the eccentric inside of the compression chamber. Due to the limited space the thrust area is relatively small. It creates conditions for partial or total overloading of the bearing. The total axial force F applied to the thrust surface is

\[ F = F_M + F_R + F_C, \]  

Fig. 3: Modified bearings max.load

International Compressor Engineering Conference at Purdue, July 12 – 15, 2004
where $F_M$ is the motor axial (solenoid) force, $F_R$ and $F_C$ are gravitational forces of the rotor and two crankshafts, respectively. The motor axial (solenoid) force can be computed from the equation below:

$$F_M = 0.0117P \ (60f) \ \left( \frac{I_{M0} E_0}{L_0} \right) \ \left( \frac{L_0}{L} \right)^2 \ [1 - 2p^{-1} (h/g)]$$  \hspace{1cm} (2)

where $P$ - phase number (for single phase = 2), $f$ - line frequency, $I_{M0}$ - magnetizing current in amperes, $E_0$ - line voltage, $L_0$ - stator core stock height, $L$ - effective core height, $h$ - misalignment, and $g$ - rotor-stator air gap. Others factors which significantly affect performance, radiated sound, and reliability are metal to metal contact due to poor oil film generation caused by saturation of the refrigerant in the oil (holes in the oil film). The dynamics of the thrust bearing during start and operation of the compressor is governed by the torques exerted on it. Since the configuration of this thrust bearing is a parallel face, a geometric converging wedge for fluid friction is not shaped. The boundary fricction loss $F_L$ is

$$F_L = 2\mu W_S \left( R_{S2}^3 - R_{S1}^3 \right) / 3 \left( R_{S2}^2 - R_{S1}^2 \right)$$  \hspace{1cm} (3)

where $\mu$ - coefficient of friction, $R_{S1}$ and $R_{S2}$ - inside and outside radius of the thrust surface, $W_S$ - weight of rotor and crankshafts. With the addition of the axial solenoid downward force the loss factor will be significantly higher. The following parameters have been used in calculations: $P = 4W/p \ (D^2 - d^2)$ is the static loading per unit area, psi (kg/cm$^2$); $W$ - static load, lb (kg); $A$ - area of the thrust surface, in$^2$ (cm$^2$); $D_m$ - medium dia. in (cm); $N$ - speed of rotation, rpm (cycles/min); $V = p \ (D \ N)$ is surface velocity, in/min (cm/min); $PV$ - psi-ft/in$^2$ min (kg-m/cm$^2$ sec). New thrust bearing has up to 3 times larger thrust area than the single cylinder compressor. Low friction coefficient of SP-21 material has been beneficial at start and operation of the compressor.

2.4 Discharge valve

Developed check-type discharge valve provides an integral one piece valve-spring-retainer assembly that can withstand higher pressure difference across the valve, has little re-expansion loss, permits an enlargement of the discharge port to reduce the over-compression and designed as integral multifunctional part to reduce cost and simplify assembly [Dreiman and Bunch, 2004]. Referring now to Fig.5, the valve member 1 includes integral support, spring and a semi-spherical end portion 2 facing an exposed semi-spherically shaped seating surface 3 machined in the cylinder block 4. A valve seat 3 is formed about annular port 5. The spherical portion of the discharge valve member 2 substantially fills at closing annular port 5, reducing the gas re-expansion volume of the discharge port. This also maximized exposure of spherical valve surface 2 to discharge refrigerant flow, accelerates the discharge valve opening thereby increasing compressor efficiency. The valve member reciprocates between a
first closed position engaging the semi-spherical seat 3 and a second open position spaced longitudinally from the valve seat. The member 6 is provided as a guide and retainer. The arrangement of holes 7 permits easy assembly since alignments of valve member 1 and guide member 6 are located upon pin 8.

3. PERFORMANCE MEASUREMENTS

3.1 Test Setup
The stand developed by CEEE (University of Maryland) has been used for the performance tests of the two-stage compressor. The stand system shown in Fig. 6 has been equipped with regular four-cycle components and consists of evaporator 10, gas cooler 6, compressor 1, expansion valve 9, and such additional components as intercooler 2 and oil separator 5. Connected to the second stage discharge of the compressor was the oil separator 5, where the oil recirculated after separation through the retention valve 12 and the oil filter 13 to the suction line of the compressor second stage. The mass-flow meter 8 has been installed at the gas cooler outlet where the refrigerant density was the highest, to achieve stable mass flow rate readings. The electric heater provided heat to the evaporator to ensure that the suction gas was superheated gas. The prototype CO₂ compressor was placed in the environmental chamber 4, which can provide a stable temperature condition up to 60°C. Since partially miscible oil, PAG, to CO₂ was used in this study, the oil separator was installed after the 2nd stage discharge of the compressor.

![Fig. 5: Discharge valve mounted in the cylinder block.](image)

![Fig. 6: Schematic of the Test Setup.](image)
3.2 Test Conditions
The performance of the compressor was measured under the conditions shown in Table 2. For each test condition data sets were collected at three different discharge pressures around the optimum pressure. In this table, \( \Delta T \) Intercooler means the temperature decrease across the intercooler, which was cooled down by the dedicated fan. As the saturated suction temperature decreases, \( \Delta T \) Intercooler was increased since the temperature difference between the 1st stage discharge gas and the ambient became higher. However, the approach temperature, which is the temperature difference between the gas cooler outlet and ambient temperature, was maintained at 5 K by assuming the same gas cooler performance.

3.3 Performance Evaluation
The performance of the compressor was evaluated in terms of its capacity, coefficient of performance (COP), and volumetric efficiencies. Here, the compressor efficiency is defined as the ratio of the isentropic compressor work to the actual compressor work based on the definition from the ARI standard 540 (1999).

3.4 Test Results
Test conditions specified in Table 2 can be classified in two groups. The test conditions 1 though 4 have the same ambient temperature and test conditions 4 and 5 have the same saturated suction temperature. Therefore, test results were compared in two groups as described next.

<table>
<thead>
<tr>
<th>Test Condition</th>
<th>Saturated Suction Temperature</th>
<th>Suction Superheat</th>
<th>Chamber Temperature</th>
<th>( \Delta T ) Intercooler</th>
<th>Gas Cooler Outlet Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>-17.8 °C</td>
<td>11°C</td>
<td>35.0°C</td>
<td>25…36 °C</td>
<td>30.0°C</td>
</tr>
<tr>
<td>2</td>
<td>-6.7 °C</td>
<td>11°C</td>
<td>35.0°C</td>
<td>15…25 °C</td>
<td>30.0°C</td>
</tr>
<tr>
<td>3</td>
<td>4.4 °C</td>
<td>11°C</td>
<td>35.0°C</td>
<td>10…22 °C</td>
<td>30.0°C</td>
</tr>
<tr>
<td>4</td>
<td>15.5 °C</td>
<td>11°C</td>
<td>35.0°C</td>
<td>10…15 °C</td>
<td>30.0°C</td>
</tr>
<tr>
<td>5</td>
<td>15.5 °C</td>
<td>11°C</td>
<td>43.3°C</td>
<td>10…15 °C</td>
<td>38.3°C</td>
</tr>
</tbody>
</table>

**Test Results at Test Conditions 1 Through 4.** The comparison of test conditions 1 through 4 shows how the performance of the prototype compressor changed by increasing the saturated suction temperature. Fig.7 shows the trend of the measured performance under the test conditions 1 though 4. It shows that the capacity and COP of the compressor increased by raising the saturated suction temperature. Each test condition shows a similar trend, which began with a relatively low value, increased to its maximum, and slightly decreased again as the discharge pressure increased. The highest measured COP was 2.3 at a saturated suction temperature of 15.5°C. The compressor efficiency was in the range of 0.4 to 0.6. The compressor efficiency decreased as the pressure ratio increased at the same saturated suction temperature. The volumetric efficiency of the 1st stage was 0.8 in maximum during any test, and decreased as the discharge pressure increased. The volumetric efficiency of the 2nd stage was lower than that for the 1st stage and decreased as the saturated suction temperature decreased. This behavior can be understood from the effect of the overall pressure ratio on the volumetric efficiency. As the saturated suction temperature decreased, the pressure ratio increased, which resulted in the reduced volumetric efficiency. The same results were obtained by increasing the discharge pressure at the given saturated suction temperature.

**Comparison of Test Results at Test Conditions 4 and 5.** The difference between test conditions 4 and 5 was that the compressor surrounding temperature was 35°C and 43°C for test conditions 4 and 5, respectively. Therefore, the comparison between tests results for these two conditions demonstrated how much the performance is affected by the ambient temperature. As illustrated in Fig. 8, the capacity is much lower at the increased ambient temperature. This change can be attributed to the reduction of the heat transfer through the shell of the compressor. The COP is associated with the capacity and therefore it follows the similar trend. Hence, the highest COP within the tests is 2.3 for an ambient temperature of 35°C. The compressor efficiency was not affected appreciably by the change of the ambient temperature. As ambient temperature increased, the 1st stage volumetric efficiency was increased, but that of the 2nd stage was decreased.
**Fig. 7: Comparison of Test Results under Test Conditions 1-4**

**Fig. 8: Comparison of Test Results under Test Conditions 4-5**
5. CONCLUSIONS

Developed two-stage two cylinder rotary hermetic compressors with carbon dioxide as working fluid has the following advantages:

- Use of the same tooling as for production of single cylinder rotary compressors due to the modular design.
- Extended variation of the capacity range by combining single cylinder production rotary kits with different value of capacity.
- Location of the lower end compression module and the upper end compression module on the opposite axial ends of the electric motor module drastically reduce the degree of the first stage intake gas superheat.
- Use of the compressor motor is more efficient due to continuous loading by two out-of-phase compression strokes and cooling of the motor by supplied intermediate pressure gas.
- The volumetric and mechanical efficiency of the two-stage multi-unit compressor will be improved due to minimized leakage losses and because of reduced distortion strain associated with the smaller pressure difference acting on vane, crankshaft, and discharge valve. Greater compressor capacity is achieved without sizable increase of the bearings.
- Smaller size multi-unit compressor is achieved by the provision of a single drive motor coupled to two pumps and single accumulator.
- Quieter operation with less vibration is achieved due to the fact that the pumps eccentric portions counterbalance one another.
- Better-balanced system due to motor-rotor support from both ends. In a single or multistage compressors with kits located on one side of the motor, rotor exhibit radial variations (whip). Such phenomena increase bearing wear, reduce performance, and increase noise and vibrations.
- The Test Stand developed by CEEE (University of Maryland) has been used for the performance measurements of the two-stage compressor operating under various saturated suction temperatures.
- The tests have shown that the compressor is able to accomplish all desired test conditions. The volumetric efficiency ranged between 0.4 and 0.8 and the compressor efficiency was as large as 0.6. Furthermore, the discharge temperature could be maintained within safe design range by using an intercooler.

REFERENCES