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## EXPERIENCES WITH APPLICATION OF A CO<sub>2</sub> RECIPROCATING PISTON COMPRESSOR FOR A HEAT PUMP WATER HEATER

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### ABSTRACT

This paper describes some of the challenges experienced in the application of a carbon dioxide reciprocating piston compressor to a water heating heat pump. Due to the characteristics of CO<sub>2</sub> in a trans-critical cycle, including high pressures, high discharge temperatures, and high solubility in oil, particular difficulties may be experienced such as reduced bearing reliability caused by bearing wear and reduced performance caused by valve and port size limitations. These challenges were overcome using analyses from compressor simulation and component models to diagnose the problems and correct them, and experimental testing to validate models, root causes, and design improvements.

### 1. INTRODUCTION

The use of carbon dioxide (CO<sub>2</sub>) as a refrigerant is becoming widely investigated among compressor manufacturers for various applications. One particularly suitable application for trans-critical CO<sub>2</sub> is in heat pumps, where the temperature glide of the refrigerant during the heat rejection process allows better matching of the refrigerant temperature with the secondary fluid compared to sub-critical cycles. The benefits of trans-critical heat pumps have been analyzed by numerous authors (Lorentzen, 1994; Neksa et al., 1998; Neksa, 2002; Richter et al., 2000; Saikawa et al., 2000; Bullard and Rajan, 2004).

Due to the characteristics of CO<sub>2</sub> as a working fluid, especially in trans-critical cycles, a number of difficulties may be encountered for compressors because of the high pressures, high discharge temperatures, and high solubility of CO<sub>2</sub> in oils. These topics have been discussed to various depths by some authors as well (Kruse, 1996; Neksa, 1999; Seeton, 2000; Neksa, 2000; Li, 2002; Hubacher, 2004). Experience has shown that bearing reliability can be a particular problem due to the high pressure loads and reduced oil viscosity from CO<sub>2</sub> solubility. In the area of performance, problems can arise from leakage, heat transfer, friction, and flow losses. Some of the main challenges experienced in the application of a CO<sub>2</sub> reciprocating piston compressor to a water heating heat pump were due to bearing wear and valve losses. Examples of these challenges will be described in what follows.

### 2. DESCRIPTION OF HEAT PUMP WATER HEATER SYSTEM

A sanitary heat pump water heater (HPWH) was developed based on a trans-critical CO<sub>2</sub> cycle to provide nominal 60kW capacity of hot water at 60°C to 80°C temperatures suitable for small to medium commercial businesses. Other operating characteristics are given in Table 1, and a schematic diagram of the heat pump system is shown in Fig. 1. The system uses a single stage, reciprocating piston compressor to provide hot, high pressure CO<sub>2</sub> to a gas cooler where the heat is transferred to sanitary water. The cooled CO<sub>2</sub> is then expanded back to low pressure through an electronically controlled expansion valve. The saturated, low pressure CO<sub>2</sub> is then heated through an evaporator before passing through an accumulator where liquid CO<sub>2</sub> can be held while the vapor passes on to the inlet of the compressor. The compressor is shown in Fig. 2, supplied by Dorin, and is further described below.

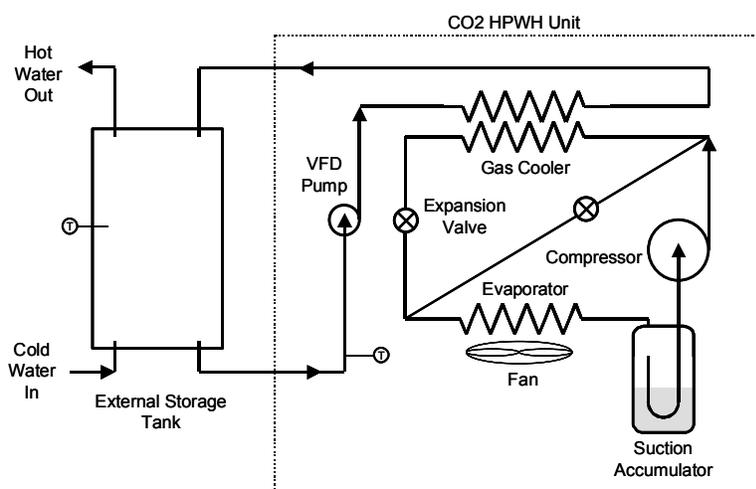


Figure 1: Schematic diagram of CO<sub>2</sub> HPWH system.

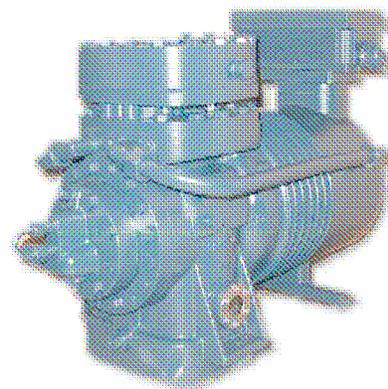


Figure 2: Dorin compressor applied to CO<sub>2</sub> HPWH.

Table 1: Operating characteristics of the sanitary hot water heat pump.

Nominal Heating Capacity: 60 kW, 7571L/day (2000 gal./day)
Ambient Air Temperature: -15 to 46 °C (5 to 115 °F)
Leaving Water Temperature: 60 to 80 °C (140 to 176 °F)
Entering Water Temperature: 1 to 40 °C (33 to 104 °F)
Optional Cooling Capacity: 3.5 to 17.5 tons

An example of a transcritical CO<sub>2</sub> cycle typical for this application is shown in Fig. 3 where it can be seen that the high side pressure affects the heating capacity of the heating cycle due to the shape of the isotherms in the super-critical region. During heat pump operation at the extreme cold ambient temperature, suction pressures can go as low as about 15 bar, with discharge pressures ranging from 80 to 110 bar. During operation at the extreme hot ambient temperature, suction pressures can go as high as approximately 55 bar, with discharge pressures ranging from 115 to 135 bar. Consequently, the compressor must operate over a very wide range of pressures and temperatures for this heat pump application.

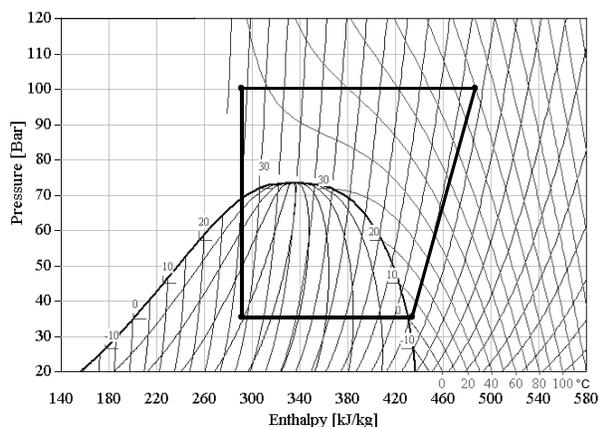


Figure 3: Example CO<sub>2</sub> transcritical cycle in HPWH.

### 3. EXPERIENCES WITH THE COMPRESSOR

#### 3.1 Description of Compressor

The compressor used in the heat pump is a semi-hermetic, two cylinder, single stage, reciprocating piston type compressor (henceforth referred to as a recip compressor), and is made by Officine Mario Dorin S.p.A – model TCS362. The compressor, shown above in Fig. 2, has the basic geometric and capacity parameters given in Table 2. This compressor has gone through development and improvements over the years since the early prototypes were first under evaluation (Neksa, 1999; Neksa, 2000).

Table 2: Basic geometric & capacity parameters of compressor.

Compressor: Dorin TCS362
Volume Flow Rate: 10.7 m <sup>3</sup> /hr
Bore: 34 mm
Stroke: 34mm
Cylinders: 2
Motor: 2-pole, 2900 rpm, 50 Hz, 18kW

#### 3.2 Computer Simulation Model

To assist with assessing and improving performance and reliability of recip compressors, a computer simulation has been developed (Nieter and Chen, 2002) that models the major physical processes occurring in a recip compressor. This simulation has been validated with detailed data measured in instrumented compressors for applications such as this. Examples of the good comparison between measured and predicted pressures possible with the simulation are shown by the p-V diagrams in Figs. 4 and 5. Performance parameters such as flow rate, power consumption, losses, and efficiencies are generally predicted within +/- 5% of the measured values. The simulation has proven to be a very useful tool in the design and development of various compressor component details.

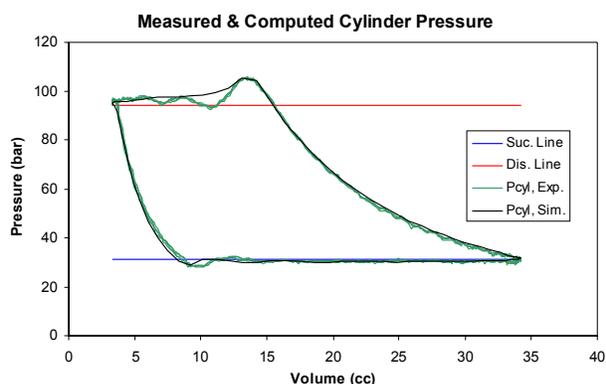


Figure 4: Comparison of measured and predicted p-V diagram for pressure ratio of 3.

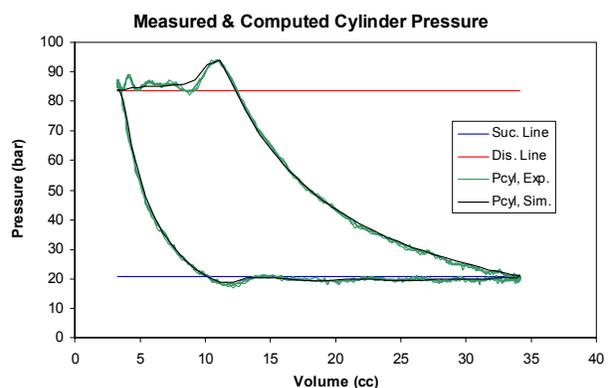


Figure 5: Comparison of measured and predicted p-V diagram for pressure ratio of 4.

#### 3.3 Discharge Valve and Port Design

The original version of the compressor tested in the prototype heat pump was a prototype design delivered in early 2002 with a valve plate having a ring-type suction valve that was smaller than the earlier prototype design, and allowed for less clearance volume. However, this smaller suction valve forced the change from two discharge ports to a single one. The initial result at that time was a valve plate design with less clearance volume, but greater over and under pressure losses.

Analysis of the original prototype discharge port and valve revealed excessive over-pressure loss. Consequently, a new discharge valve and port design was pursued that could provide low over-pressure loss and high reliability, while keeping clearance volume acceptable. A drawing of the redesigned discharge valve and port design is shown in Fig. 6. The new design is of the cantilevered reed type valve and provided greatly reduced over-pressure loss, as shown in Figs. 7 - 10, with nearly identical clearance volume. In Fig. 7 is shown the predicted compression work over a range of pressure ratios for the original and new valve plate designs. The higher compression work required by the original design is clearly shown, and is due in large part to the over-pressure loss, shown as work in Fig. 8.

The predicted P-V diagrams for two pressure ratios is shown in Figs. 9 and 10. From these diagrams, it can be seen that while the over-pressure loss is greatly reduced by the new design, it also introduces delayed valve closure that results in somewhat reduced volumetric efficiency.

Besides the performance considerations described above, the reliability of the discharge valve design must also be addressed. For this a combination of analyses using handbook calculations and FEM were used to arrive at the final redesigned discharge valve. Major reliability issues for the valves are due to peak stresses and fatigue failure when the valves are in the closed position and during opening. The peak stress when the valves are closed is due to the peak pressure difference across the valves. For a circular port and valve seat such as the above discharge valve, the valve reed undergoes an “oil can” type deflection with corresponding stresses. An example of this type of valve stress is shown in Fig. 11.

The peak stress when the valves are opening may occur at different valve positions depending upon the valve dynamic behavior and operating condition. If valve flutter or some type of valve oscillation occurs, the peak stress will likely be found at some point

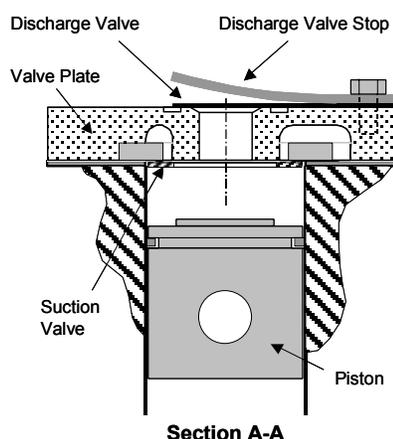


Figure 6: Redesigned discharge valve and port in CO<sub>2</sub> transcritical compressor for HPWH.

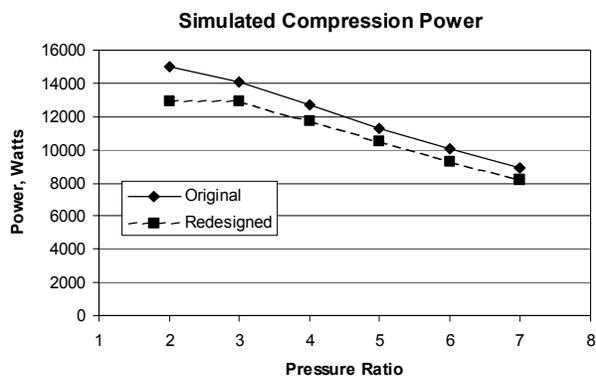


Figure 7: Comparison of compression work predicted for original and redesigned valve plate.

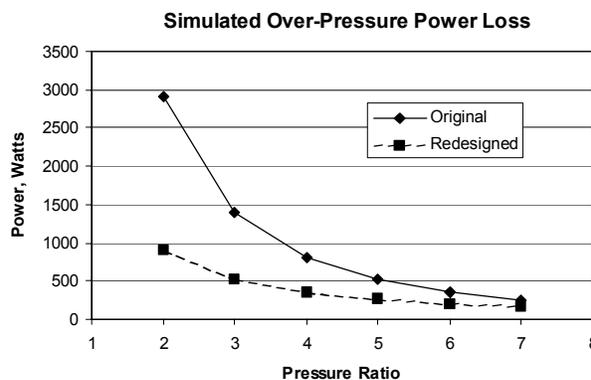


Figure 8: Comparison of over-pressure work predicted for original and redesigned valve plate.

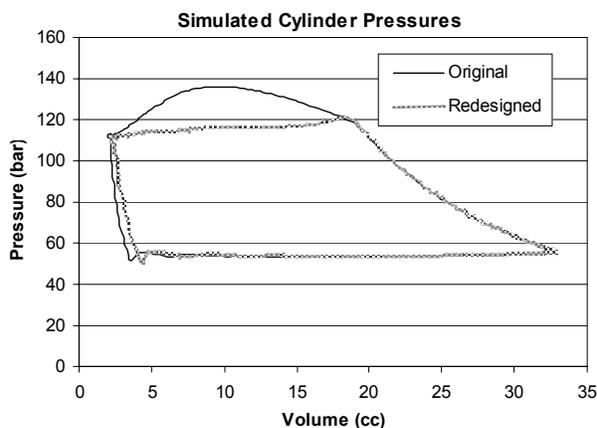


Figure 9: Comparison of cylinder pressures predicted for original and redesigned valve plate at 55/110 bar.

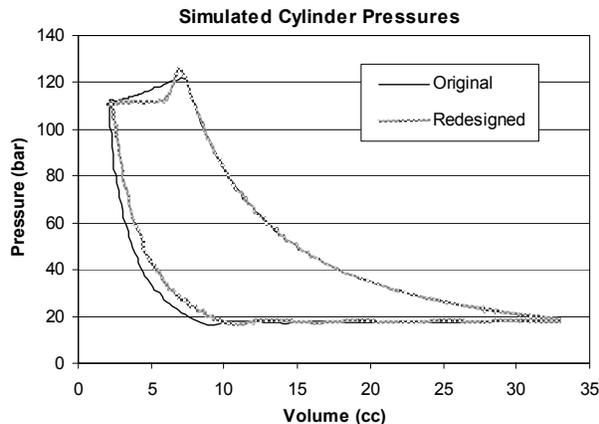


Figure 10: Comparison of cylinder pressures predicted for original and redesigned valve plate at 18/110 bar.

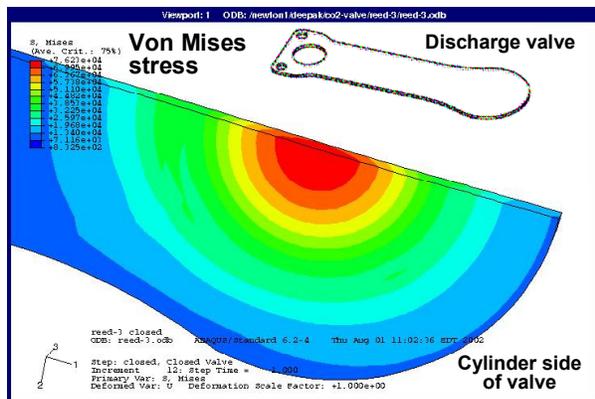


Figure 11: Example of computed Von Mises stress in discharge valve when in closed position.

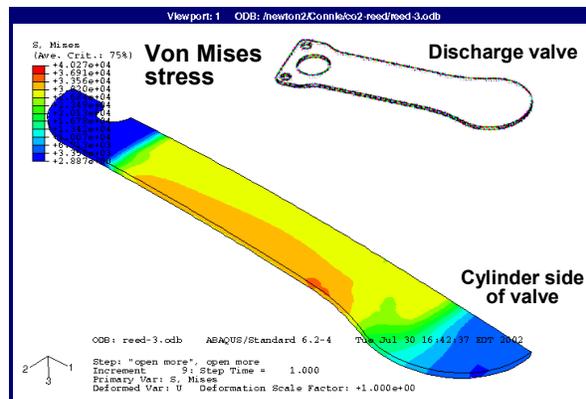


Figure 12: Example of computed Von Mises stress in discharge valve when in full open position.

during this behavior of higher order valve deflection. In a well designed valve system, the valve will not oscillate or flutter significantly, and the high stress levels typical of higher order valve deflections are avoided. In this case, the peak valve stress during opening will occur when the valve is fully deflected against a stop or backer. An example of this type of stress is shown in Fig. 12.

In this compressor for the CO<sub>2</sub> HPWH, the redesigned discharge valve and valve plate resulted in both improved performance and high reliability, as well as retaining the reduced clearance volume associated with the change to a smaller suction ring valve.

### 3.4 Crank Bearing Reliability

The original version of the prototype compressor tested in the prototype heat pump was found to experience some crank bearing wear and occasional failures due to several factors. An example of the kind of severe wear experienced is shown in Fig. 13 for the crank journals of the crankshaft, and in Fig. 14 for the shell bearings used in the big-end of the connecting rod. This kind of severe wear was found to occur after repeated starting and stopping of the compressor, and became especially significant during accelerated start/stop testing such as for qualification testing. After an extensive root cause investigation, it was determined that episodes of this excessive wear occurred due to two main factors: 1) insufficient surface finish of the crank journals, and 2) foreign particle contamination.



Figure 13: Example of severe crank journal wear on crankshaft of CO<sub>2</sub> HPWH compressor.



Figure 14: Example of severe crank bearing wear on shell bearings from connecting rod of CO<sub>2</sub> HPWH compressor.

Due to the high loading on the crankshaft with this CO<sub>2</sub> application, particular attention must be given to the journal surface finishing. The prototype compressor was assembled with a crankshaft that had insufficient surface finishing on the crank journals. In Fig. 15 is shown a magnified view of a poorly finished crank journal. After focusing much attention on this issue, an improved finishing process was put in place to give satisfactory surface finish on the crank journals, such as is shown in Fig. 16.

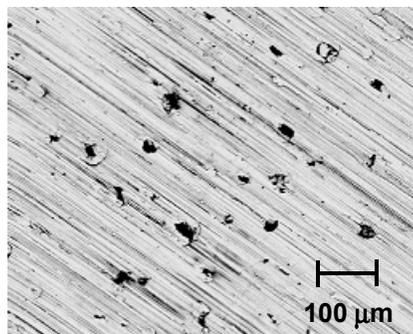


Figure 15: Example of a journal surface with poor surface finish.

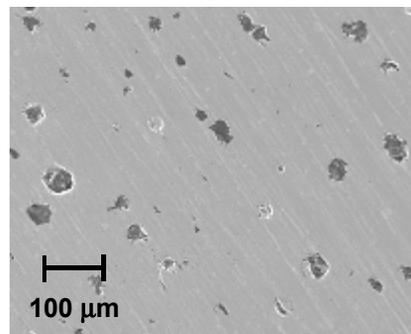


Figure 16: Example of a journal surface with good surface finish.

Another unusual problem occurred from the design and manufacturing of the prototype pistons. The pistons are made of aluminum that is oxide coated to provide a hard surface to resist wear. Unfortunately, in the prototype compressor the interior free surfaces of the piston that do not contact another part did not get proper attention to surface finish. This situation, coupled with the oxidation process used to produce a hard aluminum oxide surface layer, resulted in aluminum oxide scales or flakes on the interior surface of the piston that eventually broke free and became abrasive particles in the lubricating oil. In Fig. 17, the area on the interior surface of the piston is shown, along with a magnified picture of this surface showing the oxidized aluminum scales. These freed aluminum oxide particles, or alumina, naturally found their way into the crank bearings and acted as abrasive particles, contributing to excessive bearing wear. This problem was corrected in early 2003 by improving the surface finish on the free surfaces of the piston to eliminate the possibility of aluminum oxide scales or flakes. Also, improved cleaning of the pistons and the addition of an oil filter to the system eliminated this abrasive particle problem from causing any more bearing wear.

In addition to correcting the above problems, some effort was put into making the bearings more tolerant of particles and journal roughness by changing the bearing design in a way that would increase the minimum oil film thickness. This was accomplished by eliminating the oil groove from the upper half, or compressive load half, of the shell bearing. Since these bearings are heavily loaded

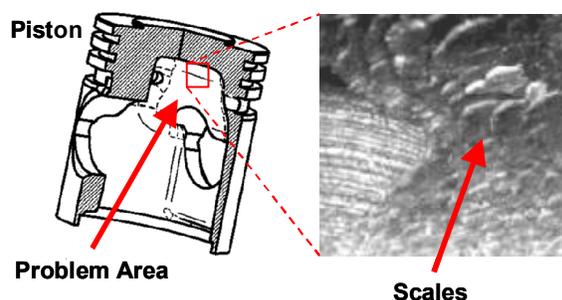


Figure 17: Picture of aluminum oxide scale on interior surface of piston.

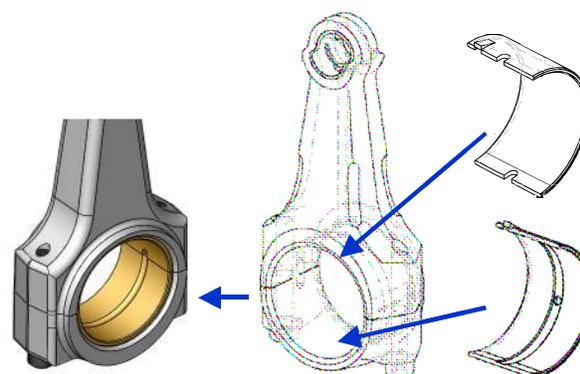


Figure 18: Depiction of change in connecting rod crank bearing design to one having no oil groove in the upper half bearing shell.

due to the high pressures associated with CO<sub>2</sub>, the increased bearing surface area that results from removing the oil groove from the upper half shell is beneficial to supporting a thicker oil film.

The force loading on this bearing as predicted by the simulation is shown in Fig. 19 where the forces acting in the local x-y coordinates of the bearing are represented. This force diagram is for the maximum pressure condition of the compressor. The predicted bearing eccentricity or orbit diagram at this condition is shown in Figs. 20 and 21, respectively, for the original full-grooved bearing design and the new design without an oil groove. These results were predicted by a standard hydrodynamic journal bearing analysis code.

From comparing these eccentricity diagrams, one can see the desired result of increasing the oil film thickness was achieved. This change in bearing design, along with the above improvements to avoid abrasive particles and rough journal surfaces, has enabled much improved reliability of the crank bearings in the connecting rods of this compressor for the CO<sub>2</sub> HPWH.

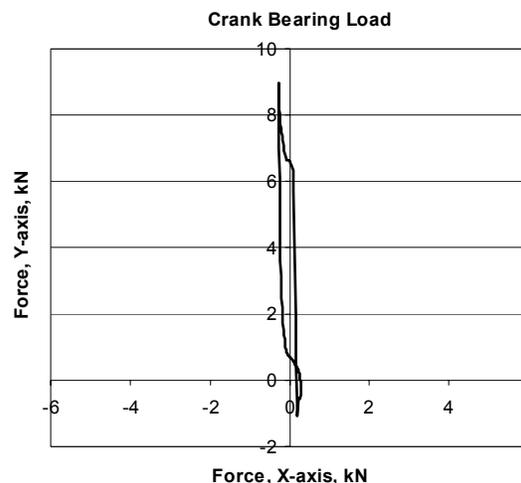


Figure 19: Predicted loading on crank bearing at maximum pressure condition.

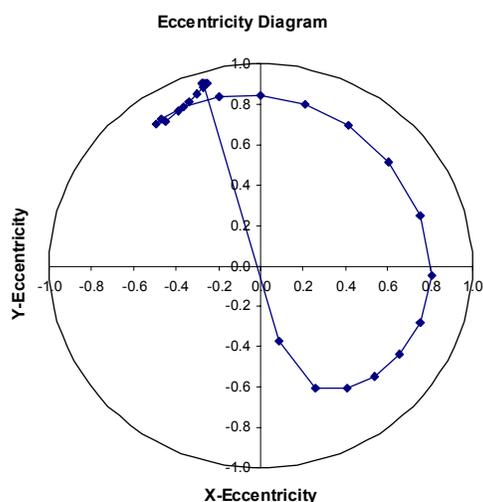


Figure 20: Predicted bearing eccentricity at maximum pressure condition for original full-groove bearing design.

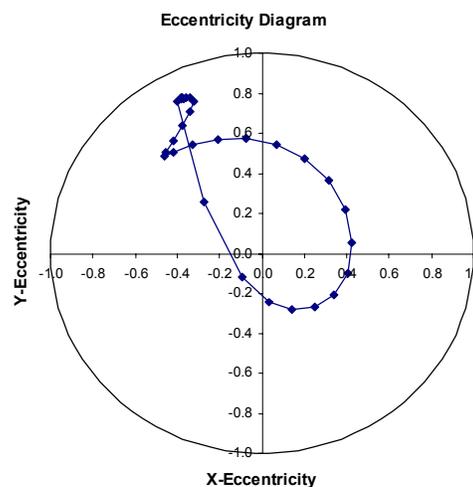


Figure 21: Predicted bearing eccentricity at maximum pressure condition for new bearing design with no oil groove.

#### 4. CONCLUSIONS

A recip compressor has been successfully implemented in a CO<sub>2</sub> heap pump for commercial sanitary hot water applications. Through the use of various modeling tools, such as a compressor simulation, FEA, and a hydrodynamic journal bearing program, the performance and reliability of this compressor has been improved. The discharge valve and port design was improved for better performance and reliability. The crank bearing design in the connecting rod was improved for increased minimum oil film thickness, and along with improved

crankshaft journal finish, and improved piston manufacturing and cleanliness, has resulted in a more reliable crank bearing.

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