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Development of a Hermetic Rotary Carbon Dioxide Compressor

Zer Kai Yap
Tecumseh Products Company

David K. Haller
Tecumseh Products Company

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Zer Kai Yap, Project Eng.

David K. Haller, P.E., Mgr. Adv. Eng.

Tecumseh Compressor Company
100 E. Patterson St. Tecumseh, MI 49286, USA

ABSTRACT

Cooling systems using natural refrigerants are receiving more and more attention worldwide. Carbon dioxide (R 744) has reappeared, as it is nontoxic, nonflammable, and widely available as a byproduct of industrial processes. However, carbon dioxide (CO_2) systems operate at much higher pressures than systems using HCFC, HFC, or HFC-blend refrigerants. Therefore, there is a need to develop compressors that operate efficiently and reliably at these higher pressures. A two-stage hermetic rolling piston compressor was developed for these applications. The compressor has dimensions such as 207.8 mm outer diameter, 445.8 mm tall, and displacement of 21.8 cc. In the first stage, the suction gas is compressed to a pressure between 7 – 8 MPa, and in the second stage, the gas is compressed to a pressure between 11-12 MPa. The housing, bearings, and crankshaft designs were some of the critical items in developing the compressor.

1. INTRODUCTION

For conventional commercial or residential heat pump applications, a hermetic motor driven compressor has been traditionally been used. Therefore, a two stage hermetic motor driven rotary compressor utilizing rolling piston technology was developed for application with CO_2. By utilizing the two-cylinder rotary mechanism and optimizing mechanical parts dimensions, the compressor can run under high working pressure and large pressure differences reliably. The refrigerant cycle using CO_2 is transcritical. Therefore, the design of the compressor has to account for the transcritical phenomena of CO_2. Compressor housing, bearings, and crankshaft designs are studied. It is further verified by running the compressor in a hot water heat pump for performance evaluation.

2. BASIC CONSTRUCTION OF THE COMPRESSOR

The compressor comprise of an overall height of 445.8 mm. The outer diameter is 207.8 mm. It has a displacement of 21.8 cc. It operates on 230 V, three phase, 60 Hz power. The nominal output for this compressor is 12 kW. The layout of the compressor mechanism is shown in Figure 1. As shown in Figure 1, the compressor developed was a two-stage compressor with the intermediate pressure being the internal housing pressure. The two stages of compression decrease the work required compared to a single stage mechanism. This decreases the electrical input required. This compressor has direct suction into its first stage. The second stage suction is the internal pressure in the housing. Both discharge pressures are direct discharge with a built in muffler located on their outboard bearing.

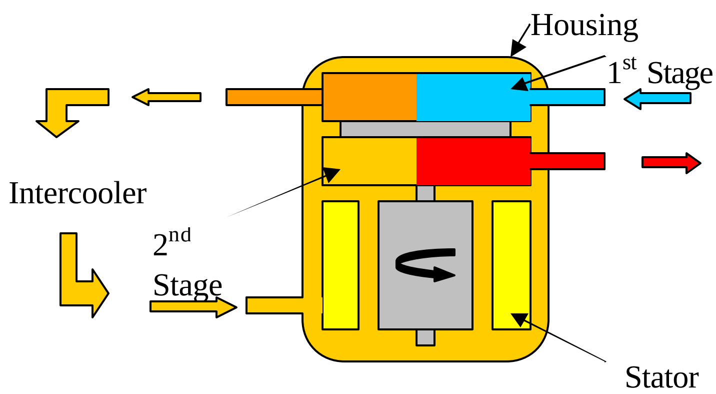

Figure 1. Compressor Mechanism Layout

3. HOUSING DESIGN

As the compressor housing will be seeing intermediate internal pressure, conventional material for the housing cannot be utilized. The housing of the compressor was designed in accordance with UL fatigue test requirements. The housing material selected was high strength low alloy carbon steel, which allows the usage of housing thickness comparable to production R410A housings.

Figure 2 shows the stress analysis conducted on the housing design. Stress analysis allows us to optimize parameters such as wall thickness, the shape of transition into piping, as well as reducing excess material and areas of high stresses. Figure 3 shows the safety factors on the housing design, which are at least 2.5 based on the maximum operating conditions. The results were then verified by undergoing a burst test.

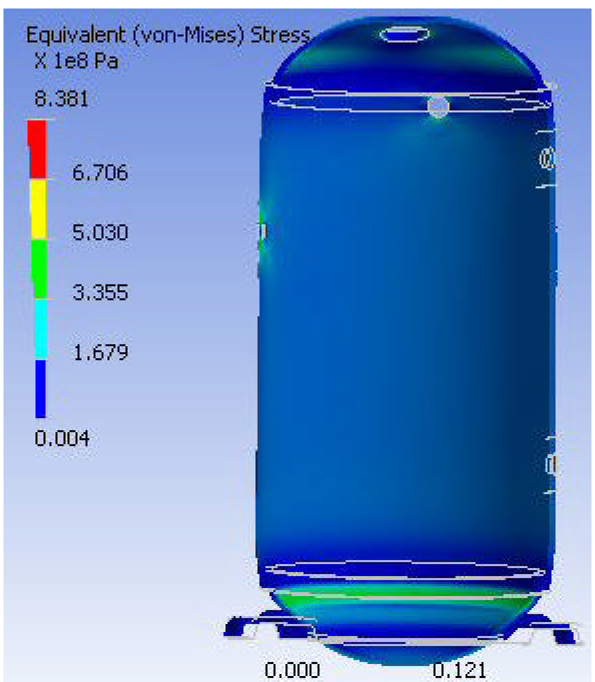

Figure 2. Housing Stress Analysis

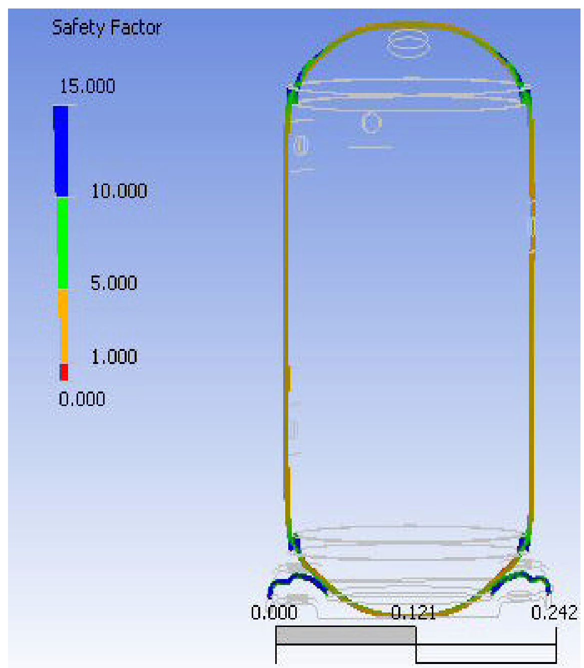

Figure 3. Housing Safety Factor

4. BEARINGS

Bearing lifespan during the course of application is most critical as it can shut down the compressor if it fails. Therefore the design of the bearings has to exc eed the estimated lifetime of the compressor.

The throw bearings used in the compressor development were needle bearings. This is due to the increase of bearing load over the conventional numbers we had seen in simulation as well as bearing wear in tests. Figure 4 shows the total bearing load distribution across a complete cycle at a typical operating condition. Figure 5 shows the radial load distribution on the bearing. The oil supply hole was located at the unloading cycle of the radial bearing force.

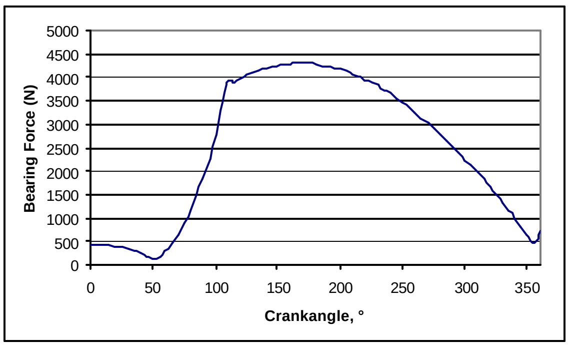

Figure 4. Total Bearing Loads

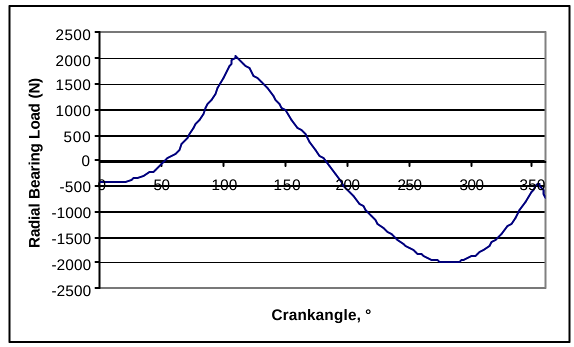

Figure 5. Radial Bearing Loads

Given that the bearing loads were known, we can determine the equivalent bearing load under variable load. Then, the equivalent bearing load is used to obtain the basic rating life utilizing the basic dynamic load rating from bearing catalogs [9]. The bearing operational L_{10} life was found to be 26,500 hours.

5. CRANKSHAFT

Crankshaft is the most critical design items in compressor design. Therefore, a successful crankshaft design will ensure the durability, performance, and balance the compressor. The compressor utilized a two rotary compression kit with a 180° phase difference by a single crankshaft that is driven by a motor mounted in the bottom part of the shaft. As the load for both the eccentric bearings acting on the crankshaft are in an offset of 180° degrees of each other, the geometry between the two eccentrics needs to be designed to withstand the high pressure loading deformation.

Figure 6 presents the shaft analysis results after undergoing several design iterations to reduce areas of high stresses and deformation. The deformation of the geometry was reduced to 1.42μm. This is a comparable value with current existing production crankshafts. On the basis of our rotary compressor experience, we are confident of the reliability of the stress analysis result. This enables us to produce a prototype crankshaft for testing.

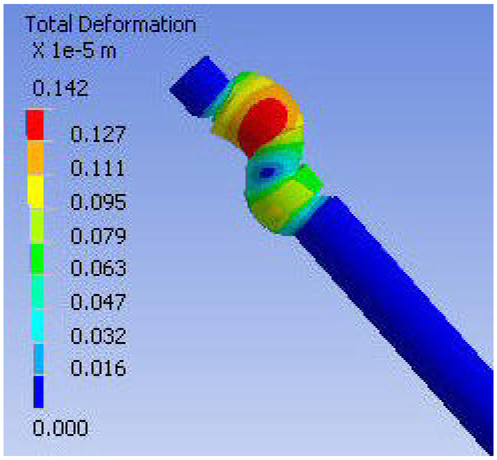

Figure 6. Crankshaft Analysis Results

6. PERFORMANCE EVALUATION

The prototype compressor was tested in a load stand to verify the design. It was then installed into an application for further testing. The performance of the CO_2 compressor was measured in an application in terms of its capacity and COP. The application selected for testing was a hot water heater. This seems to be a promising application to pursue in North America. The hot water heater utilizes a micro channel evaporator and tube-to-tube heat exchanger for a gas cooler. The hot water heater was instrumented and an automated data acquisition system was used in collecting data. Testing was conducted in a room ambient of 24 °C. The suction pressure tested ranges from 3 MPa to 4.5 MPa. The discharge pressure tested ranges from 7.5MPa to 13 MPa.

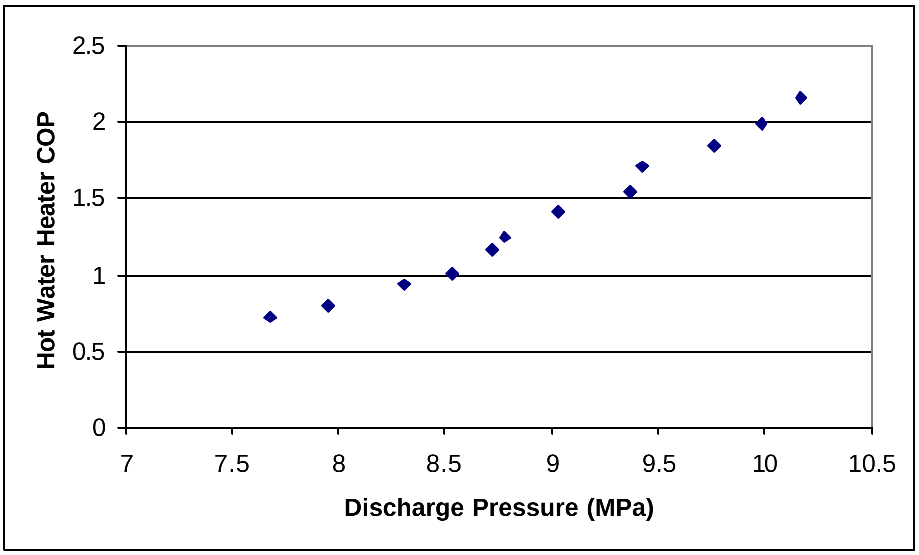

Figure 7. Hot Water Heater COP Curve With Various Discharge Pressures

Figure 7 shows the relations between the discharge pressure and system COP. The COP becomes higher as the discharge pressure increases. A COP of 2.25 was achieved under the conditions of 4.48MPa suction pressure and 10.17 MPa discharge pressure. System power consumption was used for COP calculations.

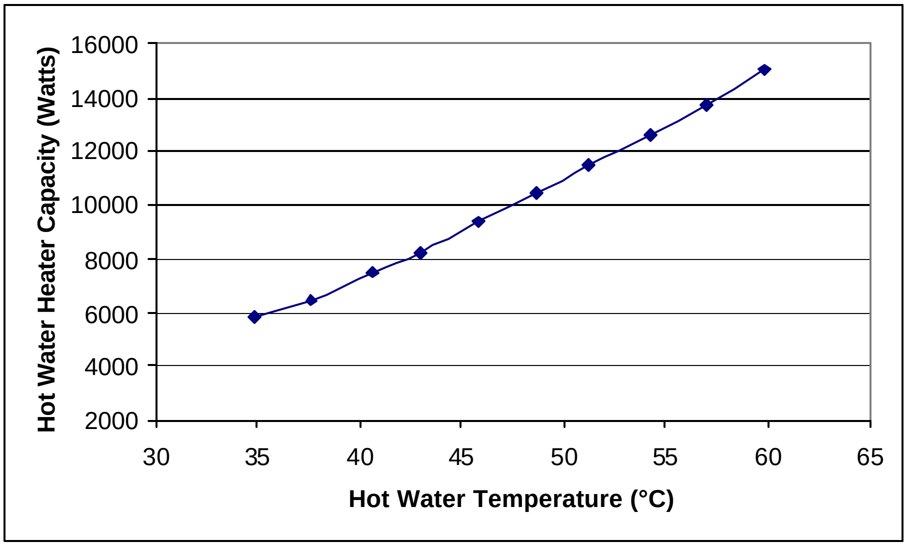

Figure 8. Relation Between System Capacities and Hot Water Temperatures

Figure 8 provides an insight to the relation between system capacities and the hot water temperature. From both graphs, we noticed that the higher the water temperature, the better the performance than the lower water temperature. However, due to the application constraints we were not able to continue testing to obtain the maximum efficiency operating point. Besides, the evaporator used needs to be optimized to provide more heat exchange capability. Therefore, the results shown are just an indication that the compressor works in an actual application rather than test data from a loadstand. Based on the above assumption, the nominal capacity of the hot water heater was found to be 12kW at suction pressure of 4.3 MPa and a discharge pressure of 10 MPa.

Both the performance of the compressor and the heat pump hot water system can be further improved. Compressor efficiency can be increased by obtaining the maximum motor efficiency thru matching of the motor torque to maximum load. An optimum gas cooling pressure needs to be obtained as any deviations from this pressure will reduce the efficiency of the system.

7. CONCLUSION

A working compressor prototype was built based on the above design and was tested in an application. The results indicate that the prototype compressor has promise to be developed into a commercial heat pump hot water heater. We believe this compressor can be further developed, such as increasing its efficiency. As part of the hot water heater system improvement efforts, a variable pump will be used to vary the water flow; the heat exchanger will be optimized as well as the refrigerant charge for the system.

SPECIAL THANKS

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