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Evaluation of a Prototype Rotating Spool Compressor in Liquid Flooded Operation

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ABSTRACT

The design and operation of a novel rotating spool compressor are reviewed. A first generation rotating spool compressor prototype is constructed and tested utilizing refrigerant R134a with PAG 150 oil in liquid flooded operation. The rotating spool compressor is open drive constructed allowing direct instrumentation of actual shaft rpm and torque as well as in chamber dynamic pressure monitoring. Experimental performance testing is conducted utilizing a compressor load stand based on a hot-gas bypass design. The load stand includes oil separation and oil cooling circuits to facilitate liquid flooded compression. Testing is done at steady state conditions collecting performance data at various oil mass flow rates. The compressor is fitted with an in chamber dynamic pressure sensor to record compression pressure and rotor angular position during the compression process.

1. INTRODUCTION

Vapor compression refrigeration systems are a reliable technology, having been used for many years to meet the needs of residential, commercial, and industrial applications. The efficiency of refrigeration equipment has improved substantially throughout the years due to advances in component design, material sciences and manufacturing practices. However, the most efficient vapor refrigeration equipment available is approaching practical limits that cannot be surpassed without modifications to the base cycle and components.

Research by Bell et al. (2009) indicates that a liquid-flooded compression process can significantly reduce thermodynamic losses associated with desuperheating of the refrigerant in the condenser. A traditional approach for reducing these losses involves the use of two-stage compression with intercooling. However, because of the high cost of this approach, it has only been commercially successful in large-scale equipment (e.g., large chillers). This cycle requires the use of a compressor that is specifically designed to operate in the presence of large quantities of liquid during the compression process. Flooded compression operation represents a significant challenge and cost premium to most commercially available compressor technologies. However it is contemplated that flooded compression can be achieved with the novel rotating spool compressor (Kemp, 2008) at a minimal cost premium.
Liquid flooded compressors must be capable of withstanding substantial component loading in the event of an over-flooded hydraulic compression condition. The purpose of this project was to determine the baseline viability of the rotating spool compressor in liquid flooded operation. Thus, it was determined that the most economical first step was to modify and test an existing rotating spool compressor prototype for liquid flooded operation.

2. ROTATING SPOOL COMPRESSOR OVERVIEW

The rotating spool compressor has a fixed volume between the spool rotor and fixed stator housing as shown in Figure 1. The spool rotor rotates about a fixed axis which is parallel and eccentric to the stator housing bore. The “spool”, as shown in Figure 2 is formed by mounting opposing endplates to both sides of the spool rotor planar faces. Figure 2 illustrates a gate placed in a diametric bore which bisects the spool rotor. The gate translates radially through the bore as the spool rotor turns.

![Figure 1: Rotating spool compressor design](image1)
![Figure 2: Rotating spool](image2)

A stationary eccentric cam mechanism that is shown in Figure 1 controls the gate position to maintain a near constant distance between the gate tip and the stator bore. The gate distal end (or tip) and stator bore do not touch at any point during the rotation. As the spool rotor turns, the trapped volumes ahead of and behind the gate change as a function of the rotor angular position. The volume behind the gate increases creating a suction volume. The volume ahead of the gate decreases creating a compression volume.

The suction volume is fluidically connected to the fluid inlet through a suction port in the stator housing as shown in Figure 1. The compression volume is fluidically connected to the fluid discharge through discharge ports in the stator housing. A check valve allows fluid to be discharged at the appropriate point and prevents backflow.

3. PROTOTYPE ROTATING SPOOL COMPRESSOR

An existing prototype compressor shown in Figure 3 was utilized for testing in liquid flooded operation. It was determined that the main rotor bearings and eccentric cam should be modified to handle the additional loading of liquid flooding.

Analysis showed that the diameter of the main rotor bearings should be increased for liquid flooded operation. Physical constraints precluded the existing prototype main rotor bearings from being increased to the required diameter. It was decided that the main rotor bearing diameter would not be increased. The risk of over-loading the
bearings would be mitigated through careful oil temperature and flow control as well as machine speed control. Modifications were made to integrate an external oiling circuit to provide oil cooling and oil flow control.

![Figure 3: Prototype rotating spool compressor](image)

![Figure 4: Eccentric cam with needle bearing](image)

Experimental analysis showed that the sliding friction between the eccentric cam and gate was unacceptably high. The eccentric cam was modified, placing a needle roller bearing between the eccentric cam lobe and the gate assembly see figure 4. Prototype compressor specifications are detailed in Table 1.

| Table 1: specifications of prototype rotating spool compressor |
|---------------------------------|-----------------|
| Refrigerant                     | R134a           |
| Lubrication Oil                 | 150 PAG         |
| Compressor Displacement / Revolution | 64 cm³      |
| Speed Range                     | 1200 – 1800 rpm |
| Materials                       |                 |
| Stator                          | Aluminum & Ductile Iron |
| Rotor                           | A2 Steel & Ductile Iron |
| Gate                            | Aluminum & Ductile Iron |
| Seals                           | Ductile Iron & Elastomer |

Initial reliability testing of the rotating spool compressor prototype was conducted at 1200 to 1800 rpm and 10% mass flow of oil. Early in testing the compressor developed spool instability. The instability caused vibration, knocking, overheating and unacceptable performance. Upon disassembly and inspection it was determined that the rotor journals were not maintaining full film lubrication separation from the journal bearings. The bearings were honed and the journal polished to resume testing. The eccentric cam, equipped with a needle bearing showed only signs of polishing with no apparent wear. It was decided that testing would continue at low speed below the point at which spool instability became apparent.

4. EXPERIMENTAL TEST STAND

A hot gas bypass load stand with integrated oil separation and oil cooling was used to test the rotating spool compressor. This design is based on the work by Hubacher and Groll (2003). A schematic diagram of the hot gas bypass load stand is shown in Figure 5.

The hot gas concept relies on an intermediate system pressure as a stable pressure anchor for the test stand by condensing a fraction of the refrigerant flow. The stable anchoring pressure allows the suction and discharge pressures to be controlled using flow valves in the discharge line and bypass line. The superheat at the compressor inlet is controlled by metering a small amount of condensed liquid refrigerant flow into the bypass flow. Low-
pressure, slightly superheated refrigerant vapor enters the compressor (state point 1). The compressor discharges high-pressure, high-temperature refrigerant (state point 2), which is throttled to the intermediate pressure (state point 3). At this point, any oil in the discharge flow is separated from the refrigerant using an oil separator. This is done to reduce the concentration of oil in the refrigerant flow before it passes through the coriolis mass flow meter in the vapor line so that only the refrigerant mass flow rate is measured. The refrigerant flow is then split with most of the refrigerant flow entering the bypass loop where it is throttled to the suction pressure (state point 4), then throttled to the suction pressure and mixed with the bypass flow. State pressures and temperatures are monitored and recorded using pressure transducers (p.s & p.d) and thermocouples (t.s & t.d) at the suction state (state point 1) and discharge state (state point 2) as illustrated in figure 5.

Separated oil is cooled and returned to the compressor through a bearing lubrication circuit at the intermediate system pressure. Thus the compressor housing is at intermediate system pressure rather than the common high-side pressure of a hermetically sealed rotary compressor. Oil collected in the housing sump at intermediate pressure exits through an oil port at the base of the compressor housing. This oil is passed through an oil cooler prior to being metered for return to the suction line. The oil flow is metered by a precision throttle valve as seen in figure 5. Oil flow is measured after the oil flow passes through the oil cooler utilizing a rotameter.

The compressor is powered by an electric DC motor. Motor speed is controlled by DC motor controller. The DC motor is coupled to a non-contacting rotary torque sensor which is coupled directly to the compressor input shaft. The rotary torque sensor is equipped with an integral speed sensor.

5. EXPERIMENTAL PROCEDURE AND PERFORMANCE CALCULATIONS

5.1 Volumetric and Overall Isentropic Efficiencies
Prior to data collection the system is allowed to run until the oil in the separator reaches a steady state temperature. Temperature and pressure are measured at the coriolis mass flow sensor to ensure the refrigerant is in a superheated vapor state. The system is then adjusted until the compressor speed, suction pressure, suction temperature and oil flow rate are at steady state.

To the degree possible experimental tests were conducted at a constant suction state. Actual states are listed on the test matrix in Table 2. The rotational speed is set for the first test point then changes slightly based on load. The following parameters are directly measured and recorded using a computer data acquisition system and instrumentation as describe above: 1) Suction pressure and temperature, 2) discharge pressure and temperature, 3) refrigerant mass flow rate, 4) compressor input shaft rotational torque (by means of a rotating torque sensor), 5) compressor rotational speed (by means integrated in the rotating torque sensor). Oil flow is recorded manually. Data
is collected at one second sample intervals over a thirty second period. The data average and standard deviation are analyzed to assure the test was conducted at steady state conditions.

Table 2: Test series operating conditions

<table>
<thead>
<tr>
<th>Test Number</th>
<th>Oil Mass Flow (%)</th>
<th>Compressor Speed (rpm)</th>
<th>Suction Pressure (bar)</th>
<th>Suction Temperature (°C)</th>
<th>Pressure Ratio [-]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A[1]</td>
<td>3.4%</td>
<td>592</td>
<td>4.4</td>
<td>16.4</td>
<td>3.2</td>
</tr>
<tr>
<td>A[2]</td>
<td>8.3%</td>
<td>575</td>
<td>4.2</td>
<td>17.8</td>
<td>3.4</td>
</tr>
<tr>
<td>A[3]</td>
<td>14%</td>
<td>548</td>
<td>4.1</td>
<td>16.7</td>
<td>3.4</td>
</tr>
<tr>
<td>A[4]</td>
<td>18%</td>
<td>522</td>
<td>4.1</td>
<td>16.9</td>
<td>3.4</td>
</tr>
<tr>
<td>A[5]</td>
<td>23%</td>
<td>522</td>
<td>4.1</td>
<td>17.8</td>
<td>3.4</td>
</tr>
<tr>
<td>A[6]</td>
<td>26%</td>
<td>511</td>
<td>4.1</td>
<td>18.0</td>
<td>3.4</td>
</tr>
<tr>
<td>A[7]</td>
<td>30%</td>
<td>500</td>
<td>4.1</td>
<td>16.2</td>
<td>3.4</td>
</tr>
<tr>
<td>B[1]</td>
<td>~ 5%</td>
<td>1100</td>
<td>3.2</td>
<td>7.1</td>
<td>4.1</td>
</tr>
<tr>
<td>B[2]</td>
<td>~ 15%</td>
<td>1100</td>
<td>3.2</td>
<td>7.3</td>
<td>4.7</td>
</tr>
</tbody>
</table>

The following performance parameters are calculated:

- **Volumetric efficiency** - The volumetric efficiency was determined using Equation (1), where the theoretical volume flow was obtained based on speed measurements and the displacement volume:

  \[ \eta_{vol} = \frac{\dot{m}_{act} \cdot V_1}{V_{th}} \]  

  (1)

  The suction specific volume is calculated as a function of suction temperature and pressure. The compressor displacement volume is obtained from the compressor design.

- **Overall isentropic efficiency** - The overall isentropic efficiency is a frequently used measure for the first law efficiency of compressors by using an overall control volume, i.e., an evaluation by using the thermodynamic states at the compressor inlet and outlet. The overall isentropic efficiency is obtained based on Equation (2):

  \[ \eta_{is,o} = \frac{\dot{m}_{act} \cdot (h_2 - h_1)}{W_{comp}} \]  

  (2)

  The suction enthalpy is calculated as a function of the suction temperature and pressure. The isentropic discharge enthalpy is calculated by assuming constant entropy for the compression process.

**5.2 In Chamber Pressure Trace**

A test was also performed to determine the in-chamber pressure trace as a function of spool angular position. A high speed dynamic pressure sensor was positioned through the stator housing near the discharge ports. The dynamic pressure sensor was positioned such that its sensing face was approximately 0.5 mm from the stator housing bore. A rotary encoder was coupled between the motor and compressor, replacing the rotary torque sensor in figure 5. The rotary encoder was configured with the data acquisition system to trigger data collection at each degree of rotation (360 data points per spool revolution).

As described earlier the system was allowed to reach steady state prior to collecting pressure profile data. In chamber pressure trace data was collected at a low oil mass flow of approximately 5% and a higher mass flow of approximately 15%.
6. RESULTS

6.1 Volumetric and Overall Isentropic Efficiencies
The overall isentropic and volumetric efficiency for Test Series A are shown in Figure 6 as a function of the percentage of oil mass flow rate relative to the total flow rate.

![Figure 6: Prototype efficiencies as a function of oil mass flow at 550 RPM](image)

6.2 Compression Chamber Pressure Trace
The compression chamber pressure traces measured during Test Series B are shown in Figure 7 at low and high oil mass flow rates as a function of the rotating angle.

![Figure 7: Compression chamber pressure as a function of spool angle](image)
7. CONCLUSIONS

7.1 Performance
Overall isentropic efficiencies of 50% and volumetric efficiencies of 98% at a pressure ratio of 3.5 were obtained at 550 rpm. Considering the development stage and operating conditions of the prototype rotating spool compressor this performance is considered to be very good. The ability to attain a high pressure ratio at low speed indicates that the compression volume is well sealed.

It is noted that the loading on the eccentric cam and spool main bearings is substantial. It appears that providing a rolling element bearing to the eccentric cam lobe reduces friction between the eccentric cam and gate to an acceptable level. Detailed consideration of main bearing type and design will be required for liquid flooded operation at high spool speeds. It is contemplated that the stator seal pressure balance may be reduced without significant degradation in compression process sealing. Reduction in the stator seal pressure balance will significantly reduce frictional losses and irreversible losses due to refrigerant fluid heating. This should substantially improve the overall isentropic efficiency.

7.2 Follow-on Development
It is clear that the rotating spool compressor has potential for application in liquid flooded compression operation. The existing prototype main bearings will be reviewed for modification or replacement. If modification is not possible a new prototype will be designed and constructed.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( h_1 )</td>
<td>Enthalpy at state point 1 [kJ/kg]</td>
</tr>
<tr>
<td>( h_{2s} )</td>
<td>Enthalpy at state point 2 for an isentropic compression process [kJ/kg]</td>
</tr>
<tr>
<td>( \dot{m}_{act} )</td>
<td>Measured mass flow rate [kg/s]</td>
</tr>
<tr>
<td>( \dot{V}_{th} )</td>
<td>Theoretical volume flow rate [m³/s]</td>
</tr>
<tr>
<td>( \nu_1 )</td>
<td>Specific volume at state point 1 [m³/kg]</td>
</tr>
<tr>
<td>( \eta_{vol} )</td>
<td>Volumetric efficiency [-]</td>
</tr>
<tr>
<td>( \eta_{o,ist} )</td>
<td>Overall isentropic efficiency [-]</td>
</tr>
<tr>
<td>( W_{comp} )</td>
<td>Input power to the compressor [W]</td>
</tr>
</tbody>
</table>

REFERENCES


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