2008

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Novel Rotary Spool Compressor Design and Preliminary Prototype Performance

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ABSTRACT

The “Rotary Spool Compressor”, a positive displacement rotary machine, is introduced. The compressor design fundamentals are described. A single stage prototype was designed and manufactured. A test stand was constructed to confirm the rotary spool compressor's mechanical operation and to determine if adequate sealing could be achieved. The prototype operated successfully achieving pressure ratios greater than 35:1 with air as the working fluid.

Preliminary test results are presented including: attainable dead-head pressure ratio, volumetric efficiency and isentropic efficiency. The experimental performance was found to compare favorably with current positive displacement compressors. Early evaluation suggests that the rotary spool compressor may offer performance improvements and reduced costs relative to current positive displacement compressor technologies.

1. INTRODUCTION

TORAD Engineering is currently developing a patent pending rotary spool machine invented by Kemp (2007, 2008). A two phased development approach was utilized. Phase 1 was validation of key novel components and validation of critical performance criteria prior to investment in Phase 2 engine development.

Specific Phase 1 Objectives
1. Validate that the novel design could be easily manufactured
2. Validate that the key elements of the novel design would function reliably
3. Achieve a pressure ratio necessary for Phase 2 Engine development
4. Achieve a volumetric efficiency necessary for Phase 2 Engine development

Phase 1 development commenced in June 2006 with the design, construction and testing of a prototype rotary spool compressor. The initial compressor, Compressor Prototype 1 (CP1), was competed in April 2007. Development and testing of CP1 is ongoing. Introduced in this paper is one embodiment of the rotary spool compressor development, hereafter referred to as the “spool compressor”.

2. SLIDING VANE COMPRESSORS

The spool compressor utilizes the same compression process as a conventional sliding vane compressor. However, the spool compressor’s novel design addresses several problems inherent to both the sliding vane and controlled (restrained) vane compressors. Following is a discussion of several issues the spool compressor overcomes.
Conventional sliding vane compressors achieve sealing and controlled vane position by forcing the vane tip to interfere with the housing bore. Lee and Oh (2003) reported that this results in significant friction and component wear, which are detrimental to machine efficiency and machine life. The vane tip friction adds heat to the surrounding working fluid, which further reduces the process efficiency.

Sealing of the vane-housing interface is achieved by a combination of either direct interference and/or hydrodynamic sealing. Centrifugal forces cause the vane to slide axially into interference with the housing sealing the working chambers. As the fluid pressure increases, an opposing or lifting force is exerted on the vane which causes it to retract and lose its sealing ability. Internal vane springs are often incorporated to augment centrifugal force and thus sealing. This exacerbates the problem of friction losses and component wear. Relying on hydrodynamic sealing permits increased leakage reducing process efficiency and attainable pressure ratios. Bloch (2006) and Brown (2005) reported that single stage sliding vane air compressors with atmospheric inlet pressure are generally limited to a 50 psig discharge pressure; a pressure ratio of approximately 3.5:1.

To achieve high pressure ratios the difference between the stator bore diameter ($D_{sb}$) and rotor diameter ($D_{rt}$) are increased. The result is an increase of the bottom dead center gap between the housing and rotor, requiring an increase in the maximum vane height $V_{ht}$ (ignoring TDC clearance $V_{ht} = D_{sb} - D_{rt}$) beyond the rotor as seen in Figure 1. The higher pressure and greater vane extension result in larger bending torques on the exposed vane ($V_{ht} \times V_{\text{width}} \times F_{\text{press}} \times V_{ht}/2$). This requires stiffer, typically thicker, vanes which increase the vane mass, centrifugal force and friction. Increasing the vane thickness also increases the vane axial profile ($V_{ht}/2$) and thus, the fluid pressure lifting forces ($V_{ht}/2 \times V_{\text{width}} \times F_{\text{press}}$). Depending on the geometry and mass, the increased lifting force may be greater than the counter-acting centrifugal force, which, in turn, requires an increase in spring force to maintain tip sealing. This of course translates into increased friction as previously discussed.

Controlled vane compressors such as the Groll Compressor (Smith et al. 1990) and Orbital Vane™ Compressor (Edwards 1994) offer solutions by controlling the vane position without vane/housing interference. These designs incorporate a combination of roller bearing cam followers, precision endplate slots and slider crank mechanisms. These designs have demonstrated that a vane compressor can achieve outstanding performance rivaling that of reciprocating compressors. However, these solutions substantially increase the machines’ complexity, precision manufacturing requirements, and ultimately the total cost of ownership.

Sliding vane compressors require a precision fit between several components to achieve adequate sealing. In particular, a precision fit between the vane/rotor slot and end plate/rotor/vane is required. These precision fits require extremely accurate machining and usually secondary grinding operations to assure flatness and finish. As such, the manufacturing costs are quite high.

The issues of vane tip friction, vane sealing, and manufacturing cost have limited the broad application of sliding vane compressors.

### 3. SPOOL COMPRESSOR

The spool compressor contains a spool assembly rotating about an eccentric axis fixed within a stator bore as shown in Figure 2. The void between the stator bore and the spool hub defines the working volume. The working volume is divided by a gate extending axially from the spool hub nearly intersecting the stator bore creating independent chambers on either side of the gate as shown in Figure 2.
Rotation of the spool increases the chamber volume behind the gate (suction chamber) while reducing the chamber volume ahead of the gate (compression chamber). The chambers are closed front and back by rotating endplates forming a spool as shown in Figure 3, hence the name “spool compressor”. Incorporating the rotating endplates improves sealing and reduces precision machining requirements.

The spool hub has a radial bore in which the gate is inserted, causing the gate to rotate with the hub. The gate and hub bore can be of any shape limited only by manufacturing constraints. A cylindrical gate and bore are easily manufactured and sealed with common piston rings. The gate has contoured control surfaces which interface with an eccentric cam attached on a fixed eccentric shaft about which the spool rotates as shown in Figure 4.
The contour of the eccentric cam and gate control faces determine the position of the gate throughout the spool’s rotation. Thus, the radial position of the gate tip, or apex, relative to the spool can be controlled throughout spool rotation. A complimentary stator bore is created such that the gap between the stator bore and gate apex is precisely controlled. Gate (vane) control without stator bore interference is achieved by the straightforward addition of the eccentric cam/gate mechanism.

Fluid flow through the machine is controlled by a combination of inlet and outlet ports and/or controlled valves positioned in the spool and/or stator. A detailed discussion of fluid control is beyond the scope of this paper.

The spool compressor primary sub-assemblies, spool, stator, eccentric and gate are illustrated in Figure 5.

4. FLUID SEALS

Fluid seals are used which minimize leakage of the working fluid from and between the suction and compression chambers. Fluid seals fill the clearance gaps between two or more machine components whose interfaces are dynamic. They are small elements inserted into edges, faces or perimeter of carrier components protruding, typically 0.5 mm or less, to interface with the mated component. It follows that fluid seals have small profiles exposed to fluid forces resulting in reduced shear forces and bending torques on the seals. This allows fluid seals to be much thinner than their carrier component. In some instances fluid seals may be made of low friction materials such as PTFE.
Because of their small size the fluid seals have low mass and require minimal, if any, spring force, which results in reduced friction. Selective inclusion of each or all fluid seals, seal material and seal design are application dependent.

Operating parameters such as the fluid lubricity, viscosity, pressure ratio, operating temperature, friction and tolerance dictate the inclusion and design of fluid seals. Example fluid seals and their respective components are designated in Table 1 and illustrated in Figure 6. Application specific seal design is ongoing. A full discussion of fluid seals is beyond the scope of this document.

![Sealing Elements](image)

Figure 6: Sealing Elements

Table 1: Fluid seal examples and their respective components

<table>
<thead>
<tr>
<th>Seal Designation</th>
<th>Seal Carrier</th>
<th>Intersecting Component</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gate Ring(s)</td>
<td>Gate</td>
<td>Spool Bore</td>
</tr>
<tr>
<td>Gate Side Seal(s)</td>
<td>Gate</td>
<td>Spool End-Plates</td>
</tr>
<tr>
<td>Gate Apex Seal</td>
<td>Gate</td>
<td>Stator Bore</td>
</tr>
<tr>
<td>TDC Seal(s)</td>
<td>Stator Housing</td>
<td>Spool Hub &amp; End-Plates</td>
</tr>
<tr>
<td>Stator Main Seal</td>
<td>Stator Housing</td>
<td>Spool End-Plates</td>
</tr>
</tbody>
</table>

5. COMPRESSOR PROTOTYPE 1 (CP1) – DESIGN & SPECIFICATIONS

A photograph of CP1 is shown in Figure 7. As mentioned above, the primary purpose of CP1 was the design validation and achievement of Phase 2 performance requirements. CP1 was not designed to be a commercial compressor prototype. As such, only minor consideration was given to the issues of part count, materials, frictional losses and efficiency.

CP1 utilizes all of the aforementioned seals including an actuated TDC seal between the stator housing and spool hub, the TDC clearance. A full description of the actuated TDC seal is beyond the scope of this paper.

CP1 utilizes a permanent uncontrolled suction intake port configured within the spool. The port is located behind the gate extending from the spool hub surface through the rear spool endplate which contains an intake cavity. The spool intake cavity is sealed from the stator assembly interior and open to the atmosphere through holes in the stator rear cover.

CP1 utilizes a permanent uncontrolled compression outlet port. The port is located before the TDC seal extending from the stator bore to a check valve cavity in the stator housing. A custom check valve in this cavity allows pressurized fluid flow while preventing backflow.
CP1 is coupled to an electric motor through a shaft integral to the front spool endplate which extends through and is supported by the front stator cover.

The specifications of CP1 are as follows:
- Approximate Dimensions: 235 mm x 190 mm x 108 mm
- Approximate Dry Mass: 8.5 kg
- TDC Clearance: 0.38 mm
- Maximum Possible Suction Volume: 98 cm³
- Minimum Possible Compressed Volume: 3.8 cm³
- Maximum Possible Volumetric Ratio: 26:1

The suction and compression volume calculations consider the locations of the intake and outlet ports as well as the timing of the articulated TDC seal. The suction and compression volumes are indicated in Figure 8.
The maximum possible intake volume represents the maximum suction chamber volume achieved while the intake port is the only open fluid path. The maximum possible compressed volume represents the minimum volume of the compression chamber prior to the gate apex passing the outlet port plus the outlet port volume and plus the compressor side check valve volume. CP1 was designed and simulated utilizing solid modeling software. The suction and compression volumes were created as complementary components in the machine assembly. The suction and compression volume measurements were derived through software analysis of these models.

The major components, seals and materials utilized in the construction of CP1 are listed in Table 2.

<table>
<thead>
<tr>
<th>Component</th>
<th>Material</th>
<th>Seal</th>
<th>Material</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stator Front Cover</td>
<td>Aluminum</td>
<td>Gate Side Seals</td>
<td>Ductile Iron</td>
</tr>
<tr>
<td>Stator Rear Cover</td>
<td>Aluminum</td>
<td>Gate Apex Seal</td>
<td>Ductile Iron</td>
</tr>
<tr>
<td>Stator Housing</td>
<td>Ductile Iron</td>
<td>TDC Seals</td>
<td>Ductile Iron</td>
</tr>
<tr>
<td>Spool Hub</td>
<td>Ductile Iron</td>
<td>Stator Main Seal</td>
<td>Bronze</td>
</tr>
<tr>
<td>Spool Endplates</td>
<td>Alloy Steel</td>
<td>Stator Oil Seal</td>
<td>Ductile Iron</td>
</tr>
</tbody>
</table>

* no surface or heat treatment utilized

### 6. COMPRESSOR PROTOTYPE 1 – PERFORMANCE TESTING

#### 6.1 Test Stand

A compressor test stand was constructed and instrumented to test the performance of CP1 at the specified Phase 1 requirements. A schematic of the test stand is included as Figure 9.

The test stand incorporates three systems: (1) compressor power and speed control, (2) fluid flow piping with oil separation and a capacitance tank, and (3) a monitoring and data collection system.

The power section utilized a 3.7 kW AC induction motor controlled by a variable frequency drive. The motor is directly attached to the test compressor (CP1) via a shaft coupling. The compressor rotary speed is monitored and recorded directly utilizing a photoelectric frequency sensor.

The fluid piping incorporates an oil separation, cooling and return circuit prior to emptying into a tank with a volume of 40 l. An additional liquid separator and filter is positioned after the tank to insure a clean fluid prior to the interaction with electronic transducers. Manual ball valves are used to control the flow for various tests. A needle valve is incorporated as the final element of the flow path to control the system pressure and flow.

Atmospheric pressure is recorded from a nearby environmental monitoring station. Redundant manual gauges and electronic sensors are used to monitor and record discharge and mass flow meter fluid pressures. The electronic
pressure sensors have a full range accuracy of ±0.25%. Internal compressor, suction, discharge and mass flow meter temperatures are monitored by means of J-type thermocouples. The mass flow is measured by an electronic thermal mass flow meter of the heated tube design. The mass flow meter provides a temperature compensated output of standard liters/min (SLPM, calibrated for Nitrogen at 21.1°C at 1atm) with a k factor correction for air of 1.0000. The mass flow meters range is 0-500 SLPM with a full range accuracy of ±1.5%. A thin film polymer capacitor type humidity sensor, accurate to ± 2.5%, is integrated to record relative humidity. An electronic AC current sensor is integrated prior to the variable frequency AC drive to monitor and record power consumption. Current readings are adjusted per the AC drive and AC motor manufacturer's power factors of 0.92 and 0.96, respectively. Electronic sensors are integrated with a programmable data acquisition and control system that is connected to a personal computer. The computer software allows process data to be continuously monitored and/or recorded.

6.2 Test Results
6.2.1 Dead Head Pressure Tests
Dead head pressure tests were performed to determine seal effectiveness and maximum achievable pressure ratio. The dead head pressure tests were performed as follows:

- The compressor was brought to test speed with the discharge circuit open.
- BV-1 was then closed, allowing the pressure to build between the compressor and BV-1. The pressure is monitored at PG-1, typically reaching a maximum value within a few seconds.
- Discharge pressure, compressor speed and current draw are recorded.

Figure 10 presents the dead head pressure on the left and the current draw on the right y-axes as a function of the time in seconds. It can be seen that 3800 kPa or a pressure ratio greater than 38:1 was reached. This surpasses the Phase 2 pressure ratio performance requirements. It is seen that the attainable pressure ratio exceeds the theoretical maximum volumetric ratio. This is due to the displaced volume of incompressible lubricating fluid.

6.2.2 Volumetric and Isentropic Efficiency Tests
The tests to obtain the volumetric and isentropic efficiencies were performed as follows:

- System was charged by an external source to assure there is no leakage then discharged.
- The compressor was brought to test speed with all ball valves open and the needle valve 2 (NV-2) closed.
- The system pressure was allowed to build until the desired pressure was reached.
- NV-2 was opened allowing flow out of the system.
- NV-2 was adjusted until the desired pressure was at steady state.
Relative humidity, suction and discharge pressures, suction and discharge temperature, mass flow rate, compressor speed and current draw are recorded.

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

$$\eta_{vol} = \frac{m_{act} \cdot V_1}{V_{th}}$$  \hspace{1cm} (1)

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):

$$\eta_{is, o} = \frac{m_{act} \cdot (h_{2s} - h_1)}{W_{comp}}$$  \hspace{1cm} (2)

Figure 11 presents the volumetric efficiency and the overall isentropic efficiency as a function of the pressure ratio for two cases: 1800 rpm with no suction valve (case 1) and 2000 rpm and a new suction valve (case 2). It can be seen from Figure 11 that the measured volumetric efficiency decreases relatively linearly in both cases. The volumetric efficiency decreases from its highest value of 0.95 at a pressure ratio of 1.5 (case 2) to a value of 0.35 at a pressure ratio of 8.4 (case 1). The measured overall isentropic efficiency reaches a maximum value in both cases and decreases towards lower and higher pressure ratios as typically seen for positive displacement compressors. For case 1, the maximum value is 0.35 at a pressure ratio of 5.5. For case 2, the maximum value is 0.65 at a pressure ratio of 4.

![Figure 11: Volumetric efficiency and overall isentropic efficiency versus pressure ratio](image)

### 6.3 Performance Factors

As noted previously, CP1 was designed and constructed to test the design’s mechanical viability and sealing ability. As such, several factors can be addressed to significantly improve the compressors volumetric and isentropic efficiencies.

#### 6.3.1 Flow Reversion

The compression volume (clearance volume) of CP1 is much larger than necessary. Minor changes in geometry and the incorporation of a reed valve in the inlet port will easily correct this problem. The suction port of CP1 is always
open between atmospheric pressure and the suction chamber. The suction port is also in direct fluid communication with the clearance volume at the immediate end of the compression cycle. Thus, high pressure fluid is able to flow back through the suction port. This causes significant reversion of the fluid flow detrimental to volumetric efficiency. This will be addressed in the next prototype machine.

6.3.2 Fluid Heat Rejection
CP1 compression components were constructed of iron and alloy steel both having poor thermal conductivity. The use of aluminum and production surface treatments will permit greater heat rejection and possibly increase isentropic efficiency.

6.3.3 Friction Losses
The sealing and spring elements of CP1 have yet to be optimized for performance. Friction losses can be significantly reduced through the use of alternate materials, material treatments, and spring rate optimization.

7. CONCLUSIONS
A novel “Rotary Spool Compressor” is introduced. A single-stage prototype was designed, manufactured, and tested with air as the working fluid. The compressor design and performance are presented. The rotary spool compressor provides significant advantages with respect to reducing the frictional losses experienced in a conventional sliding vane compressor. The test results indicate that the prototype operated successfully achieving pressure ratios greater than 38:1, volumetric efficiencies of up to 0.95, and isentropic efficiencies of up to 0.65. The experimental performance was found to compare favorably with other positive displacement compressors at this early design stage. Further design optimization suggests that the rotary spool compressor may offer performance improvements and reduced costs relative to current positive displacement compressor technologies.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
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<tbody>
<tr>
<td>$h_1$</td>
<td>Enthalpy at state point 1</td>
<td>[kJ/kg]</td>
</tr>
<tr>
<td>$h_{2s}$</td>
<td>Enthalpy at state point 2 for an isentropic compression process</td>
<td>[kJ/kg]</td>
</tr>
<tr>
<td>$m_{act}$</td>
<td>Measured mass flow rate</td>
<td>[kg/s]</td>
</tr>
<tr>
<td>$\dot{V}_{th}$</td>
<td>Theoretical volume flow rate</td>
<td>[m$^3$/s]</td>
</tr>
<tr>
<td>$\nu_1$</td>
<td>Specific volume at state point 1</td>
<td>[m$^3$/kg]</td>
</tr>
<tr>
<td>$\eta_{vol}$</td>
<td>Volumetric efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>$\eta_{p,ie}$</td>
<td>Overall isentropic efficiency</td>
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</tr>
<tr>
<td>$W_{comp}$</td>
<td>Input power to the compressor</td>
<td>[W]</td>
</tr>
</tbody>
</table>

REFERENCES


ACKNOWLEDGEMENT

TORAD Engineering would like to thank Charles Paparelli and Regi Campbell for their commitment and ongoing support of the Kemp Rotary Machine. We would also like to recognize Valley Tool of Water Valley Mississippi for their craftsmanship and valuable contributions to this project.