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Jason James Hugenroth
Inventherm, Baton Rouge, Louisiana, USA, jason.hugenroth@inventherm.com

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Oil-less Swing Compressor Development

Jason Hugenroth
Inventherm,
Baton Rouge, Louisiana, USA
(225) 247-3555, jason.hugenroth@inventherm.com

ABSTRACT
An overview of the development of an oil-less rotary swing compressor is presented. The compressor was developed for use in portable oxygen concentrators (POCs), which are used by patients that require supplemental oxygen. An extensive analytical and experimental program was conducted to develop the technology. Implementation of the oil-less swing compressor resulted in a POC system weight reduction of 24%, with a 24% reduction in power. In addition, the volume of the system was reduced by approximately 37% with a 2 dBA reduction in the noise level.

1. INTRODUCTION
This paper provides an overview of the development of an oil-less rotary swing compressor. The swing compressor is mechanically similar to the rolling piston compressor with the exception that the vane forms an integral part of the piston. This requires that the vane be rotatably and slidingly supported in the compressor cylinder. The earliest identified reference for a rotary machine of the type described herein was in US Patent 418,665, which was issued to T.R. Almond in 1890. The first known commercialization of this type of machine occurred in the early 1900’s by Kinney® (Tring, 2012), where it was developed as an oil-sealed vacuum pump and was referred to as a rotary piston pump. Vacuum pumps based on this technology are still manufactured today. In the mid 1990s Daikin began reporting on the development of swing compressor technology for use in refrigeration systems (Masuda et al., 1996). This work was motivated by the need to overcome reliability issues associated with excessive wear of the vane tip in rolling piston compressors, which became a concern with the introduction of chlorine-free refrigerants. The elimination of vane tip wear and other features of the swing compressor make it an attractive candidate for oil-less operation.

The oil-less swing compressor presented here was developed for use in portable oxygen concentrators (POCs). POCs separate oxygen from ambient air. Typically, this is accomplished via pressure swing adsorption (PSA) or vacuum pressure swing adsorption (VPSA). POCs are most commonly used by ambulatory (i.e. mobile) patients with chronic obstructive pulmonary disease (COPD). They provide the same function as oxygen bottles or liquid oxygen dewars, however, they have the advantage of providing an inexhaustible supply of oxygen as long as electrical power is available to operate the unit. POCs also eliminate safety risks associated with pressurized or liquefied oxygen reservoirs. Because of this they are permitted in the cabin of commercial airlines and are used by medical personnel on military land and air vehicles.

POCs operate by feeding a pressurized stream of ambient air to one or more zeolite beds that preferentially adsorb nitrogen and water vapor while allowing oxygen to pass through. The beds will eventually become saturated with nitrogen and water vapor. Lowering the pressure on a bed causes the gases to desorb. In the case of VPSA the pressure is reduced below ambient to provide more complete desorption. The system comprising the zeolite beds, and the valves and tubing used to control the flow over the beds is referred to as the fractionator. For a VPSA system a compressor and a vacuum pump are required to affect the operation as described. It is necessary for the compressor to be oil free to prevent fouling of the zeolite and the patient’s respiratory system.

It is the goal of POC manufacturers to minimize the weight and size of the system for a given oxygen flow rate. In addition, low vibration, low noise, high reliability and high efficiency are very important characteristics for POCs. The latter is especially important for maximizing operation on battery power, which has a significant impact on the patient’s quality of life. The compressor plays a dominant role in achieving these objectives. It has traditionally been a major stumbling block for improving system efficiency and providing further system miniaturization.
2. SWING COMPRESSOR MODELING AND ANALYSIS

Figure 1 shows a section view of a swing compressor. Rotation of the shaft causes the piston to move eccentrically inside the cylinder bore. The eccentric piston and vane divide the space between the piston and cylinder into a suction chamber and compression chamber. A discharge port and associated discharge valve (not shown) are located in the compression space to discharge the compressed gas. The vane is supported by bushings, which allows the required linear and rotational motion of the vane.

Figure 1: Swing compressor front section view.

To aid in the development process an Engineering Equation Solver (EES) (Klein, 2013) lumped parameter model of the swing compressor was developed. The compressor simulation includes a thermodynamic model, a dynamics model and a regression based leakage model. The leakage model can also be turned off and a specified volumetric efficiency can be provided as an input. Frictional losses were not directly included in the compressor simulation, although a separate model was developed to investigate friction related issues associated with the vane and bushing.

2.1 Thermodynamic model

The simulation calculates the volume in the compression space as a function of crank angle. The compression chamber area, excluding the space occupied by the vane, is identical to that of a rolling piston compressor. This area is (Pandeya and Soedel, 1978)

\[
A_{cl} = \frac{r_c^2}{2} \left[ (1 - a^2)\beta - \frac{(1 - a^2)}{2} \sin 2\beta - a^2 \arcsin \left( \frac{1}{a} - 1 \right) \sin \beta \right]
- \left( a(1 - a) \sin \beta \sqrt{1 - \left( \frac{1}{a} - 1 \right)^2 \sin^2 \beta} \right)
\]

(1)

where \( A_{cl} \) is the compressor chamber area excluding the vane, \( r_c \) is the radius of the cylinder, \( a \) is the ratio of the radius of the piston to the radius of the cylinder bore, \( \beta \) is the angle from top dead center to the contact point between the piston and cylinder. The area the vane occupies or isolates from the compression chamber is

\[
A_v = \frac{1}{2} w_v \left[ -a r_c + (l_{cb} + r_c) \frac{f}{g} + g \right] - \varepsilon (l_{cb} - r_c^2) \frac{f \sin \beta + \varepsilon (-a r_c + g)^2 \sin \beta}{f}
\]

(2)

where, \( f = l_{cb} - \varepsilon \cos \beta \), \( g = \sqrt{\varepsilon^2 + l_{cb}^2 - 2 \varepsilon l_{cb} \cos \beta} \)

and where \( w_v \) is the vane width, \( l_{cb} \) is the distance from the cylinder center to bushings center, and \( \varepsilon \) is the piston eccentricity. This area differs from that of a rolling piston compressor since the vane pivots in addition to sliding in the bushings. Figure 2 shows the areas \( A_{cl} \) and \( A_v \).
The final compression chamber volume is

\[ A_{cv} = (A_{ci} - A_v)h \]  

(3)

where \( h \) is the piston height. Polytropic relationships are used to determine the pressure and temperature in the compression chamber as the volume changes at successive crank angle intervals. This aspect of compressor modeling has been detailed in several other publications (Soedel, 1972, Chu et al., 1978, Pandeya and Soedel, 1978) and, therefore, will not be repeated here. Ideal gas properties of air were used, which was appropriate for the pressure and temperature range of interest. Specifically, pressures ranged from about 46 kPa to 165 kPa with inlet an inlet gas temperature of approximately 21 °C (i.e. room temperature).

2.2 Vane and bushings

The vane and bushing (VB) components – which are in dry sliding contact – are the dominant source of friction and wear in the compressor. These components have a significant impact on the efficiency and reliability of the compressor. In spite of the mechanical simplicity of the VB components, optimizing the design in this area is complex due to the number of variables involved.

With reference to Figure 3, forces acting on the VB components include gas forces from the compression chamber, suction chamber and vane chamber and friction forces between the moving components. Piston dynamics result in normal loads between the vane and bushing which affect the magnitude of the frictional forces. All of these forces vary with crank angle and compressor operating condition.

Figure 3(b) shows the static force balance on one bushing when gas forces are neglected. Using the Coulomb friction model \((Fr = \mu N)\) it can be shown that

\[ \mu_1 \sin \theta - \mu_1 \mu_2 \sin \theta - \mu_2 \cos \theta - \sin \theta = 0 \]  

(4)

where \( \mu_1 \) is the coefficient of friction between the vane and bushing, \( \mu_2 \) is the coefficient of friction between the bushing and bushing bearing and \( \theta \) is the wedge angle, which is defined as the angle between the bushing flat and the contact area between the bushing and bushing bearings (Figure 3(c)). Wedge angles are fixed by the bushing diameter, vane thickness, piston eccentricity and the distance between the cylinder centerline and the bushing bearing center line. The instantaneous wedge angle varies with crank angle (i.e. vane tilt). A plot of Equation (4) is shown in Figure 4 for \( \theta = 10 \). The line in the figure represents impending motion for the vane. For conditions above the line the vane will bind, while below the line the vane will slide.
Increasing the wedge angle decreases the mechanical advantage of the bushing, which shifts the line upward making vane binding less likely. If the net gas force on the bushing is in the same direction as the force on the vane (Upward in Figure 3 (a)) the line of impending motion will shift downward making vane binding more likely. It is advantageous to maximize the wedge angle in order to minimize friction forces on the vane.

Figure 5 shows an experimental torque trace from an early prototype compressor when vane binding was occurring. The vane angle, which is the vane’s angle of tilt from the vertical, is overlaid in the figure. It can be seen that the peak torque due to vane binding occurs near the point of maximum vane tilt. This is because the wedge angle for one of the bushings is at a minimum when the magnitude of vane tilt is at its maximum. In order to eliminate the potential for vane binding the vane width and piston eccentricity were reduced. Figure 6 shows the torque curve from a later prototype with vane binding eliminated.
2.3 Leakage
Lubricating oil in the compression space of compressors has the obvious benefit of reducing friction and wear. It is also well known that oil is very effective at inhibiting leaking through internal clearances. Therefore, operating without oil exacerbates friction and wear issues while tending to reduce volumetric and adiabatic efficiency due to increased leakage.

The net effect of leakage increases as the displacement volume of the compressor decreases. This occurs because the compressor’s displacement volume varies with the cube of the characteristic length, while leakage flow area tends to vary linearly. Thus, it is difficult to control leakage in small oil-less compressors. Contacting seals can be used to minimize leakage. However, the friction associated with the seals scales in a manner similar to leakage flow area. The leakage paths for the swing compressor are shown in Figure 7. Leakage in the compressor was controlled by minimizing clearance gaps. This approach is limited by practical machining and assembly tolerances.

CFD analysis was performed to quantify leak rates from the areas identified in Figure 7 with the exception of vane chamber leakage, which was neglected. The analysis was performed with dry air. Compression chamber pressure varied from 108.2 kPa to 163 kPa for pressure mode operation and from 39.3 kPa to 94.5 kPa in the suction chamber for vacuum mode operation. For both cases the pressure in the piston hub and the corresponding compressor chamber was 101.3 kPa. The clearance gap ranged from approximately 13 μm to 50 μm. Additional parameters related to the piston and cylinder geometry were also considered. The analysis was stationary. Ninety CFD simulation runs were performed. The data were used to generate regression curves which were then incorporated into the EES swing compressor simulation. Figure 8 shows pressure and velocity contour plots of leakage from areas 1-3. These plots correspond to a 13 μm clearance between mating parts.
Figure 8: Contour plots of internal compressor leakage between piston face and endplate.

Figure 9 shows leakage mass flow rates as a function of clearance gap for a suction pressure of 101.3 kPa and a discharge pressure of 163.4 kPa. The effect leakage has on adiabatic efficiency is also shown. The clearance gap is the same for all leakage paths. Leakage in the radial direction was found to be the dominant loss. This is contrary to what is typically reported for rolling piston compressors for refrigeration applications (Chu et al., 1978, Pandeya and Soedel, 1978). For refrigeration compressors the clearance gap is typically flooded with oil from the journal bearing, which will tend to minimize leakage in this area. This may explain the discrepancy between our results and other reported findings.

Figure 9: Flow rate of leaked air and the effect on adiabatic efficiency at a discharge pressure of 163 kPa.

2.4 Dynamics
Figure 10 shows a schematic of the dynamics model for the swing compressor simulation. The inertial loads predicted by the dynamics model are added to the gas loads predicted by the thermodynamics model. This makes it possible to predict the shaft torque, bearing loads and bushing loads. The predicted torque curves shown in Figures 5 and 6 are the net result of gas and inertial loads.
3. TESTING FACILITIES

An instrumented compressor load stand was built for swing compressor testing. The photographs in Figure 11 show an early prototype compressor on the test stand. A torque sensor was located between the motor and compressor to measure shaft torque and speed. In addition, the test stand included non-contact proximity probes, flow meters, pressure controllers, pressure transducers and thermocouples. Data were collected using a data acquisition system and associated software connected to a PC.

![Diagram of swing compressor dynamics model](image1.png)

**Figure 10:** Schematic representation of swing compressor dynamics model.

A principal development challenge was identifying materials that would allow the compressor to operate maintenance free in the POC for several years. Over thirty distinct materials and numerous material mating pairs were tested. To expedite the testing process several wear test stands were constructed. One of these is shown in Figure 12. The wear test stands had a vaned piston, bushings and bushing bearings in a configuration that replicates their arrangement in the compressor. To accelerate the testing a normal force was applied to the upper movable bushing bearing cradle using dead weight. Using this approach it was possible to increase the wear rate by a factor of up to about 40, depending on the load applied. Figure 13 shows a representative wear test sample and profilometer traces of the wear scar. A bearing test stand, shown in Figure 14, was also constructed to evaluate bearing performance. Dead weight was used to load the bearing. A torque sensor was used to measure the frictional torque from the bearings.
Figure 12: VB components wear test stand.

Figure 13: Profilometer traces of wear scar on a test vane.

Figure 14: Bearing test stand.
4. DUPLEX COMPRESSOR DESIGN

For operation of the POC system a compressor is needed to pressurize the ambient air that is fed to the fractionator where oxygen is concentrated. Another compressor operates as a vacuum pump with an inlet pressure that is below ambient. This is used to regenerate the zeolite in the fractionator. The design of the fractionator is such that the compressor and vacuum pump are run simultaneously to maximize oxygen separation rates in a compact device. Since most of the oxygen is separated from the incoming air, the mass flow rate of gas through the vacuum unit is lower than that of the pressure unit. However, the displacement rate of the vacuum unit is approximately twice that of the pressure unit, due to the high specific volume of the low pressure gas entering the vacuum unit.

The POC that this compressor was designed for produces continuous oxygen flow rates from 0.5 to 3.0 standard liters per minute (SLPM). The system can also operate in pulse mode where a bolus of oxygen is delivered upon sensing the patient’s breath. These operating modes require the compressor to operate at a fairly wide range of speeds. To maximize system efficiency it would be ideal to control the displacement rates of the vacuum and pressure units independently. This is because the optimal sorption and desorption processes in the fractionator vary nonlinearly with the rate of oxygen delivery. However, the weight and cost penalty of this approach could not be justified and a single motor was used to drive the vacuum and pressure units. To further reduce overall size, weight and part count, the pressure and vacuum units were arranged on one side of the motor. This is analogous to the arrangement of twin cylinder rolling piston and swing refrigeration compressors.

Figure 15 shows a photograph and cross section of the duplex compressor (i.e. combined vacuum and pressure unit). To minimize peak motor torque the pistons are 180° out of phase. This also minimizes the mass of the counterweights needed for balancing. The suction ports are located in the cylinders. The discharge valves are located in the endplates, which are covered by mufflers. A common divider plate isolates the pressure and vacuum units. Warm air that leaks from the pressure unit into the hub area of the divider plate will tend to be drawn into the vacuum unit which will decrease the efficiency of the vacuum unit. To minimize this effect the divider plate is vented to ambient pressure.

The overall length of the compressor is 125 mm with a maximum diameter of 88 mm. The compressor weighs approximately 1.1 kg. It is constructed primarily from aluminum with suitable materials used in areas of the compressor where wear and friction are a concern.

The required displacement rate of the pressure unit is approximately half that of the vacuum unit. Since both units operate on a common shaft, the displacement of the pressure unit must be reduced to achieve the proper flow rate. This was accomplished by providing a relief in the pressure unit cylinder which reduces the volume entrapped in the compression chamber. This approach minimizes the number of unique parts required for the duplex compressor, but it results in the pressure unit being larger and heavier than is strictly necessary.

Figure 15: Oil-less duplex swing compressor images.
5. CONCLUSIONS

Commercially available oil-less compressor technology was identified as the most significant obstacle preventing a substantial reduction in the size and weight of existing POCs. The swing compressor was identified as having several attributes that made it potentially viable for oil-less operation. To achieve the program goals it was necessary for the swing compressor to be smaller and lighter with better vibration and noise characteristics than competing technologies, while operating at a higher efficiency. It was also necessary for the compressor to be cost effective to manufacture while meeting or exceeding the service life of legacy POC compressors.

Implementation of the oil-less swing compressor resulted in a POC system weight reduction of 24%, with a 24% reduction in power. In addition, the volume of the system was reduced by approximately 37% with a 2 dBA reduction in the noise level. The performance improvements are in comparison to the manufacturer’s prior generation POC system, which produces 3 SLPM continuous oxygen flow using the same fractionator as the system utilizing the swing compressor. The prior generation system uses a wobble piston compressor.

While the oil-less swing compressor technology was developed for POCs, the technology can be used in any application where oil free operation is required. Examples include fuel cells and ventilators. In addition, the compressor can be adapted for use as a sump-less low lubrication compressor. This would extend its operating envelope and allow it to be used, for example, in orientation independent refrigeration systems (e.g. mobile or zero-g applications).

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>$a$</td>
<td>Ratio of piston radius to the cylinder bore radius</td>
</tr>
<tr>
<td>$A_{ci}$</td>
<td>Ideal compression chamber area</td>
</tr>
<tr>
<td>$A_{cv}$</td>
<td>Vane chamber area isolated by the vane</td>
</tr>
<tr>
<td>$A_c$</td>
<td>Actual vane chamber area</td>
</tr>
<tr>
<td>$F_r$</td>
<td>Friction force</td>
</tr>
<tr>
<td>$l_{cb}$</td>
<td>Distance between cylinder and bushing center lines</td>
</tr>
<tr>
<td>$h$</td>
<td>Piston height</td>
</tr>
<tr>
<td>$N$</td>
<td>Normal force</td>
</tr>
<tr>
<td>$P_{cc}$</td>
<td>Pressure in compression chamber</td>
</tr>
<tr>
<td>$P_{sc}$</td>
<td>Pressure in suction chamber</td>
</tr>
<tr>
<td>$P_{vc}$</td>
<td>Pressure in vane chamber</td>
</tr>
<tr>
<td>$r_c$</td>
<td>Cylinder radius</td>
</tr>
<tr>
<td>$w_v$</td>
<td>Vane width</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Angle from vertical to piston contact point</td>
</tr>
<tr>
<td>$\epsilon$</td>
<td>Piston eccentricity</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Wedge angle</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Friction coefficient</td>
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</table>

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