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Development of Experimental Test Rig for Internally Geared Screw Compressors

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ABSTRACT

Gerotor pumps operate with two rotors, each rotating in the same direction about parallel but offset axes. Commercial applications for these machines include oil pumps, fuel pumps and hydraulic power transmission. It is also possible to use the gerotor configuration to achieve internal compression by specifying appropriate discharge port geometry, and the addition of helical twist to the rotors has been shown to achieve further benefits of reducing porting losses and power transfer between rotors. These internally/geared screw compressors have a number of potential advantages over conventional twin-screw configurations, including reduced leakage areas, co-directional thermal expansion, reduced rotor deflection, reduced viscous losses, and higher swept volume for a given machine envelope. Many of the loss mechanisms in these machines are similar in nature to conventional twin-screw machines but must be characterised due to the unique aspects of sealing line geometry and co-rotation (leading to much lower sliding velocities at contact points). Experimental testing is an essential step for model validation and appraisal of the potential benefits of this technology. This paper describes the test rig that is currently being developed to investigate the operation and performance of internally/geared machines, and how the how the geometry of a prototype machine has been specified.

1. INTRODUCTION

Internally geared positive displacement machines are widely used in pumping and hydraulic power transmission applications. The concept dates back to an original patent by Moineau (1934), and a large amount of research has been conducted over the years focussed on optimising the design and operation of these machines, key aspects of which are discussed by Rundo (2017). The idea of using such machines with helical rotors was also considered by Moineau, who described rotors with both constant or variable pitch, as well variable profile rotors with intersecting axes. The constant pitch version is essentially the configuration used in progressing cavity pumps, but with the outer ‘rotor’ fixed and an orbiting inner rotor. There has been recent interest in the idea of using the variable profile rotors with intersecting axes for compressor applications resulting in development of prototype machines, although the manufacturing challenges for producing such rotors with tight geometrical tolerances are significant. The concept proposed by Read et al. (2020) maintains the relative simplicity of constant rotor pitch, but introduces stationary end plates with porting designed to control the periods when fluid is able to directly enter and leave the working chambers created between the co-rotational inner and outer rotors. This is the subject of a current research project aimed at developing numerical tools to design and optimise these machines, along a programme of experimental testing to investigate the performance, characterise the loss mechanisms, and provide data for validation.

Crucial prerequisites include complete test rig design tuned for internally/geared machines, along with the development of the design protocol for prototype oil-injected screw compressor including consideration of machine configuration, rotor geometry and component selection. Challenges in this area include an appropriate selection of material and manufacturing methods as outer rotor cannot be made using common techniques known from the twin-
screw machine manufacturing process. Another challenge to overcome is methods of oil injection as well as bearings and sealing arrangements.

2. SUMMARY OF BASIC DESIGN PROCEDURE

To investigate the performance of internally-gearred screw machines, we need to characterise the relationship between geometric parameters and the machine operation.

The key geometrical parameters that define the machine can be reduced to:
   i. the rotor profile (including the number of lobes on each rotor),
   ii. the maximum diameter of the outer rotor profile, \( D \),
   iii. the length of both rotors, \( L \),
   iv. the wrap angle of each rotor, \( \Phi \), (note that these are related via the gear ratio between the rotors).

The key parameters that characterise the operation of the machine include:
   i. the volumetric flow rate of the working fluid,
   ii. the built-in volume ratio of the machine, \( e_v \),
   iii. the fluid filling velocity, \( u \),
   iv. the rotational speed, \( \omega \),
   v. the suction and discharge pressures (and hence the bearing loads).

Previous work by the authors (Read, 2021) has considered how the relationship between these parameters can be characterised in a simple way purely via consideration of the machine geometry. By neglecting the leakage flows, port losses and heat transfer effects, a simple analysis can be performed to calculate non-dimensional groups that are a function of the rotor geometry:

\[
\Pi_1 = \frac{Q_{id}}{D^2 u^*} = f(\lambda, \sigma, N, \Phi)
\]

\[
\Pi_2 = \frac{u^*}{\omega L} = f(\lambda, \sigma, N, \Phi)
\]

As both the flow area and rate of change or working chamber volume vary continuously during rotation, \( u^* \) is defined as the minimum velocity that occurs during filling of the working chamber, and can be considered representative of the flow for a particular rotor geometry. In equations 1 and 2 the parameter \( u^* \) is therefore used to provide a representative value of the fluid velocity during filling of the working chamber.

The values of \( \Pi_1 \) and \( \Pi_2 \) only depend on the rotor geometry. In this study pin generated profiles have been used as described by Litvin and Feng (1996) and Vecchiato et al. (2001). The basic geometry needed to define such profiles is the size and position of the circle that defines part of the outer rotor profile; these can be defined in non-dimensional form as a normalized circle radius, \( \sigma \), with its center at a normalized radius, \( \lambda \), from the center of the outer rotor. The actual axis spacing, circle radius, \( \rho \), and center distance, \( a \), can then be defined relative to the maximum outer rotor profile diameter, \( D \), as follows

\[
\frac{E}{D} = \frac{1}{2N_1(\lambda - \sigma) + 4}, \quad \frac{\rho}{D} = \sigma N_1 \left( \frac{E}{D} \right), \quad \frac{a}{D} = \lambda N_1 \left( \frac{E}{D} \right)
\]

For a given application the value of the required volumetric flow rate will be known. If the volumetric efficiency is initially assumed to equal 100\%, the ideal volumetric flow rate for the machine, \( \dot{Q}_{id} \), is then known. A sensible value of \( u^* \) can also be specified based on consideration of allowable filling losses. Using equation 1 the required value of \( D \) can then be found as a function of the rotor profile and wrap angle.

Valid rotor profiles are only produced using the pin generation method if the value of \( \sigma \) is less than a maximum allowable value \( \sigma_{lim} \) which is itself a function of \( \lambda \) and \( N \). The value of \( \sigma_{lim} \) is found to be zero when \( \lambda = 1 \), and no
valid profiles can be created for $\lambda < 1$. To simplify the presentation of results, a normalized parameter $\bar{\sigma} = \sigma / \sigma_{lim}$ is used; i.e. when $\bar{\sigma} = 1$ the generated profile uses that maximum possible pin radius for the corresponding values of $\lambda$ and $N$.

The profile also influences how much rotation occurs between a working chamber first appearing and it being closed off from the inlet end face. The wrap angle of the rotor is therefore normalized by this total rotation angle as follows;

$$\Phi = \Phi / (\varphi_{\text{start}} - \varphi_{\text{end}}) = f(\lambda, \bar{\sigma}, N, \Phi)$$

This is useful as a value of $\Phi = 1$ indicates that the working chamber reaches maximum volume at the point where it becomes closed from the inlet end face and is about to be exposed to the discharge end face. In this case no porting is required at the inlet end, but the discharge end face must be partly closed to achieve compression as the working chamber volume decreases. If $\Phi < 1$ the maximum volume occurs when the working chamber is still exposed to both the inlet and discharge end faces; porting is then required to block both ends to prevent back flow and allow compression as volume decreases. The shape of the inlet port is therefore defined by the rotor profile $(\lambda, \bar{\sigma}, N)$ and the rotor wrap angle $(\Phi)$, while the shape of the discharge port will additionally depend on the volume index for the machine $(\varepsilon_v)$.

2.1 Results of basic design procedure

While using a lower values of $N$ is known to increase the swept volume, it leads to larger pressure difference across the leakage paths. A value of $N_1 = 6$ was selected as a reasonable compromise for the initial investigation of machine performance.

The maximum swept volume of machine is achieved when $\bar{\sigma} = 1$, but this results in areas of high curvature of the rotor profiles. A value of $\bar{\sigma} = 0.8$ has therefore been considered as a suitable compromise between maximizing the volume created per revolution for a given volume enclosing the rotor profile (i.e. $\pi LD^2 / 4$), and limiting the contact stresses that will occur. The other main consideration for the initial design is to minimize the power transferred between the rotors, which will again limit contact stresses and frictions losses. In general, the power transfer is seen to decrease as the rotor wrap angle increases. Increasing the wrap angle does however reduce the swept volume for a given machine volume. Hence an appropriate compromise needs to be identified.

Power is transferred between the rotors during operation due to the net torque exerted on each rotor by the working fluid. In order to calculate this, the pressure variation during operation must be known. For the basic design procedure we can neglect losses due to port flows and leakages, and consider just a simple polytropic process for compression of the gas. This process can be defined purely in terms of a volume ratio $(\varepsilon_v)$, the inlet temperature $(T_{\text{suc}})$, and the allowable discharge temperature $(T_{\text{dis}})$. The polytropic exponent can then be found:

$$n_{\text{poly}} = 1 + \frac{\ln(T_{\text{dis}}/T_{\text{suc}})}{\ln(\varepsilon_v)} \quad (4)$$

The working chamber pressure can then be found as a function of the volume, and the net torque acting on the rotors can be calculated. The maximum proportion of input power being transferred between the rotors, $\Pi_3$, is defined as follows, when the driven rotor is chosen such that $\max(|P_{\text{dis}}/P_{\text{in}}|)$ is minimised.

$$\Pi_3 = \min \left( \max \left( \frac{P_1}{P_{\text{in}}} \right), \max \left( \frac{P_2}{P_{\text{in}}} \right) \right) = f(\lambda, \bar{\sigma}, N, \Phi) \quad (5)$$

It is also possible to relate the machine rotation speed and diameter, as this will be constrained by the bearing used for the outer rotor. The limiting speed in the bearings, $v_{\text{lim}} = \omega D / 2$, can be substituted into equations 1 and 2 to define a non-dimensional machine volume:

$$LD^2 v_{\text{lim}} \sqrt{\frac{\rho}{\gamma}} = \frac{1}{2\Pi_2 \sqrt{\Pi_1^3}} = f(\lambda, \bar{\sigma}, N, \Phi) \quad (5)$$
The power transfer ratio is shown in figure 1 for the case when $N_1 = 6$ and $\bar{\sigma} = 0.8$. The value is seen to reach a minimum when $\lambda = 1.4$ and $\bar{\Phi} \geq 0.5$. Higher values of $\bar{\Phi}$ do however lead to a larger machine volume for fixed values of $v_{lim}$, $u^*$, and $Q_{id}$. A value of $\bar{\Phi} = 0.5$ will therefore be used in initial prototype design.

In order to size the machine for a particular application, the values of $Q$, $u^*$ and $v_{lim}$ must be specified. Using data for a conventional twin screw compressor, the values shown in Table 1 have been used, resulting in the contour plots of required rotor $D$, $L/D$ and $\omega$ values for the machine.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value (m$^3$/s)</th>
<th>Rationale</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Q$</td>
<td>0.034</td>
<td>Conventional screw compressor</td>
</tr>
<tr>
<td>$u^*$</td>
<td>23.1</td>
<td>Conventional screw compressor</td>
</tr>
<tr>
<td>$v_{lim}$</td>
<td>30</td>
<td>Typical rolling element bearing limit</td>
</tr>
</tbody>
</table>

**Table 1:** Parameters defining compressor application

2.1 Selection of geometry for experimental testing

Based on the previous analysis, a machine has been specified for initial testing of the internally-geared compressor concept. The geometry and volume ratio for the prototype are described in Table 2. The corresponding rotor profile are illustrated in figure 3, and the predicted bearing forces are shown in figure 4.
### Table 2: Parameters defining selected internally-geared compressor geometry

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$N_1$ (-)</td>
<td>6</td>
<td>Outer rotor lobe number</td>
</tr>
<tr>
<td>$N_2$ (-)</td>
<td>5</td>
<td>Inner rotor lobe number</td>
</tr>
<tr>
<td>$D$ (mm)</td>
<td>100</td>
<td>Max diameter of outer rotor profile</td>
</tr>
<tr>
<td>$L$ (mm)</td>
<td>130</td>
<td>Length of both rotors</td>
</tr>
<tr>
<td>$E$ (mm)</td>
<td>6.712</td>
<td>Axis spacing distance</td>
</tr>
<tr>
<td>$\lambda$ (-)</td>
<td>1.4</td>
<td>Normalised profile shape parameters</td>
</tr>
<tr>
<td>$\sigma$ (-)</td>
<td>0.8</td>
<td></td>
</tr>
<tr>
<td>$\Phi$ (-)</td>
<td>0.5</td>
<td>Normalised wrap angle</td>
</tr>
<tr>
<td>$\epsilon_v$ (-)</td>
<td>2</td>
<td>Built-in volume ratio</td>
</tr>
</tbody>
</table>

**Figure 3:** Rotor profiles generated using values in Table 2

**Figure 4:** Bearing forces for machine defined using values in Table 2, shown for the inner and outer rotors at the high and low pressure ends of the machine

### 3. INITIAL THERMODYNAMIC MODELLING

#### 2.1 Generation of input data

Using the machine geometry defined in Table 2 it is possible to define the shape of the porting that will allow:

- maximum flow area for filling the working chambers while $dV_{wc}/dt > 0$
- no direct flow to suction or discharge while $dV_{wc}/dt < 0$ and $V_{max} \leq V_{wc} \leq V_{max}/\epsilon_v$
- maximum flow area for discharge of the working chamber while $dV_{wc}/dt < 0$ and $V_{wc} < V_{max}/\epsilon_v$

This porting is defined by the rotor geometry and by the loci of contact points between the two rotors, and is illustrate in Figure 5 along with the resulting flow area as a function of rotor position.
The machine geometry can also be used to calculate the leakage line lengths for a single working chamber, which will influence the leakage flow into different working chambers within the machine. As the rotors both have helices in the same direction, there is no blow hole area, and the leakage area is mostly due to the rotor-to-rotor contact points that connect to adjacent working chambers; this is analogous to the tip leakage areas in a conventional machine (except that here there are only a single leading edge and trailing edge, rather than leading and trailing edges for each rotor). There are however contact points between the rotors in the region of the pitch point which create leakage paths similar to the inter-lobe leakage in a conventional machine, where the pressure difference can be high as the tip of a low pressure chamber has a region linking to the tail of a high pressure chamber. This flow path does however converge and diverge gradually compared to the other ‘tip’ leakage paths; an important question for the internally geared machine is how this geometry and the rotor-to-rotor clearance gaps influence the machine performance.

2.2 Results of chamber modelling
The geometry data discussed in the previous sections can be used to provide the necessary input data for chamber modelling of the compression process using the SCORG software (Rane et al., 2019). This has been developed to apply the necessary analysis to the conservation of internal energy through the compression of air with the use of oil injection. This analysis is still in the early stages, as the nature of the leakage flows and mechanical losses for the internally-geared configuration are not well understood. The influence of rotor-to-rotor clearance gaps as a function of operating speed have however been investigated using the prototype geometry (see table 2), and are presented in figure 7.
A test rig is currently being developed to investigate this prototype machine, and is discussed in the next section. While the bearing speed limit for the prototype is around 6,000rpm, this initial testing will be performed at lower speeds using a direct drive motor. The performance of the machine at an inner rotor tip speed of 15 m/s (corresponding to 3,300rpm) has therefore been calculated as shown in figure 8.

Further optimisation of the machine geometry will be conducted using the chamber modelling approach. Development of the necessary leakage and loss models will be guided by both experimental studies and computational fluid dynamics analysis of the machine performance across a range of operating conditions. The description of the test rig currently being developed for the investigation of internally geared screw compressors is provided in the following section.

**4. DEVELOPMENT OF EXPERIMENTAL TEST RIG**

**4.1 Elements of the test rig**

The test rig under development consists of several separate subcomponents. The design includes a frame made of 100x100 mm and 100x50 mm steel sections. Once the frame is welded, precisely machined plates will be bolted on top. This is done to avoid warping and deflection of the components’ mounting surface during the welding process. The machined plates will feature a grid of M12 mounting threads and dowel holes for precise alignment of the machine.

**Figure 7:** Predicted indicated power as function of mass flow rate and rotor-to-rotor clearance gaps for selected internally-geared compressor with inner rotor tip speeds of 10-50 m/s

**Figure 8:** Predicted efficiency as function of discharge pressure for selected internally-geared compressor with inner rotor tip speed = 15 m/s and clearance gaps = 50μm
with the motor axis. The separate mounting plate has been designed for the motor to locate it precisely with respect to the dowels grid.

The drive part of the test rig includes an induction motor coupled with a 3-phase inverter allowing for variable speed control up to 3500 RPM and 11kW of motor power. This is sufficient for prototype testing with a relatively low-pressure ratio. With the further development of optimised prototypes, the drive chain can be upgraded to a variable high-speed motor. Between two couplings, a torque meter has been placed for measurement of speed, power and torque delivered to the compressor. This is rated at 10,000 RPM and 50Nm. Under the testbed, oil management components have been located. These will include, an oil separator paired with an oil filter as well as the heat exchanger to cool oil before injecting it again into the compressor.

![Figure 9: Illustration of test rig under construction for investigation of internally geared screw compressor](image)

The aim of this initial testing is to understand the practical challenges in this compressor configuration, to investigate the losses due to leakage, bearings and seals, and to provide performance data to validate the numerical modelling.

An important aspect of the research will be to develop understanding of the leakage that occurs between conjugate rotor profiles. This type of leakage path is similar to the interlobe clearance of conventional screw compressors (which is important due to the high pressure difference present) but is very difficult to study experimentally due to its position within a conventional machine.

### 4.2 Bearing Arrangement

While the inner rotor bearing arrangement is common for the conventional twin-screw machine, the outer rotor imposes challenges. There is no shaft as the helical shape is made on the inside of the rotor body. This means it has to be suspended in the housing relying on the outer surface of the rotor. Bearing selection will depend on the outer diameter of the rotor. This creates a challenge as with a bigger bearing diameter minimum load required for correct operation is increasing. Where there are very light loads, failure mechanisms other than fatigue, such as skidding and smearing of raceways or cage damage, often prevail. One way of dealing with that is the use of low-friction bearings, where rolling elements and/or races are coated with wear-resistant carbon. This coating reduces friction and improves resistance against wear and smearing, even in bearings where only the rolling elements are coated.
4.3 Future work

The next steps will be to manufacture and assemble the test frame with auxiliary systems. The prototype machine design has been finalised and possible manufacturing methods are investigated for the outer rotor. One of the focuses is on the optical methods for flow investigation inside the compressor. This requires clear optical access to the working chamber, thus meaning most likely a transparent outer rotor and/or end plates. Future work also includes the development of design tools as well as the investigation of possible mass production manufacturing methods.

6. CONCLUSIONS

The paper describes the project made in the ‘Internally-Geared Screw Compressor’ research project. The key conclusions from the current study are as follows.

- Initial sizing and geometry selection of an internally geared machine can be performed on the basis of required volumetric flow, and limiting speed for the fluid during filling and limiting bearing speeds.
- Once geometry of the rotor and porting is defined, thermodynamic analysis of the compression process can be performed using a chamber modelling approach.
- A small scale prototype of the internally-geared compressor has been designed along with a test rig to investigate the operation of performance of an oil-injected air compression.
- Experimental data from this test rig is intended to support development of loss models for internally-geared machines, and provide data for validation of 1D and 3D numerical analysis.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tbody>
<tr>
<td>$L$</td>
<td>Rotor length</td>
<td>m</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume</td>
<td>$m^3$</td>
</tr>
<tr>
<td>$A$</td>
<td>Area</td>
<td>$m^2$</td>
</tr>
<tr>
<td>$D$</td>
<td>Max rotor profile diameter</td>
<td>m</td>
</tr>
<tr>
<td>$a$</td>
<td>Radial distance to pin centre</td>
<td>m</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Radius of pin</td>
<td>m</td>
</tr>
<tr>
<td>$E$</td>
<td>Distance between rotor axes</td>
<td>m</td>
</tr>
<tr>
<td>$N$</td>
<td>Number of lobes on rotor</td>
<td>-</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>Normalised radial distance to pin centre</td>
<td>-</td>
</tr>
</tbody>
</table>
σ  Normalised radius of pin  (-)

Subscript
1, 2  Outer or inner rotor
wc  Working chamber
sw  Swept area or volume

REFERENCES


ACKNOWLEDGEMENT

The ‘Internally-Geared Screw Compressor’ research project is supported by Carrier Corporation and PDM Analysis.