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Basic design procedure for an internally geared screw compressor

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Abstract. The internally geared gerotor type of positive displacement machine is widely used for pumping applications, but can also be used to achieve internal compression or expansion of a working fluid. A further development of this configuration is the application of helical twist to the inner and outer rotors. Previous studies have shown that this can decrease the power transferred between rotors, and increase the relative maximum flow areas at the inlet and discharge ports. There are possible benefits to this configuration when compared to conventional twin-screw machines, including reduced leakage area, stiffer rotors, and co-directional thermal expansion of rotors. However, no experimental data is currently available for this configuration, and optimising the geometry of the gerotor screw machine requires a detailed understanding of the porting, leakage flows and mechanical losses. An experimental programme is being conducted to investigate the influence of these factors on performance. This paper discusses preliminary results from geometrical analysis leading to a simple design procedure. The basic design of an internally geared air compressor is demonstrated, based on the operation characteristics of a conventional twin screw machine.

Keywords: screw, internally geared, gerotor, compressor

1. Introduction

Internally geared helical twin screw machines have been proposed for application as positive displacement compressors and expanders. The use of stationary end plates with carefully shaped holes acting as inlet and discharge ports can allow internal compression or expansion to occur using fixed profile and helical pitch for the rotors [1–4]. These machines are then analogous to conventional twin screw compressors, but have potential advantages in terms of stiffer rotors, co-directional thermal expansion of rotors, and reduced sealing line length for a given working chamber displacement. Comparing these two configurations requires a fundamental understanding of the relationship between geometric parameters including the rotor profile, maximum diameter, D, length, L, and wrap angle, Φ, and the operation of the machine characterised by the volumetric flow rate, $\dot{Q}$, built-in volume ratio, ϵ_v, fluid filling velocity, u, rotational speed, ω, and bearing loads. This paper focusses on characterising these parameters for the internally-geared machine, in order to develop a design procedure for a prototype machine.

2. Relationships between geometric and operational parameters

The current study considers only simple composite epi-hypo cycloid rotors, where the rotor profile is defined by a shape parameter, ν, and the number of lobes, N. These profiles are fully described in ref. [3]. This rotor profile can then be scaled by the maximum profile diameter, D. The swept volume of a machine is an important parameter which depends on the geometry of the rotors; once normalised by the volume of a cylinder enclosing the rotors as shown in Equation 1, it is a function of the rotor profile and the wrap angle of the helical rotor [4]. The swept volume can then be related to the ideal volumetric flow rate of the machine, $\dot{Q}_{id}$ (Equation 2). Assuming the fluid is incompressible during the filling process, the rate of change of volume divided by the port area provides an indication of the fluid inlet velocity. This value tends to infinity as the port area tends to zero, but quickly drops to an approximately constant value for most of the filling process. A further performance parameter, u^*, is therefore defined here as the minimum ideal velocity of the fluid at the inlet port during filling (Equation 3). It is possible to define a non-dimensional group, λ_1, relating rotor diameter, volumetric flow rate, and u^* as shown in Equation 4. A second non-dimensional group, λ_2, relating rotational speed, rotor length and u^* can also be defined (Equation 5). As with the normalised swept volume, $\bar{V}_{sw}$, these non-dimensional groups are only functions of the rotor profile shape (defined by ν and N, but not depending on the scaling factor D) and the helical wrap angle.

$$\bar{V}_{sw} = \frac{V_{sw}}{\frac{\pi}{4}LD^2} = f(\nu, N, \Phi) \tag{1}$$

$$\dot{Q}_{id} = V_{sw}\left(\frac{\omega}{2\pi}\right) \tag{2}$$

$$u^* = \omega \times \min\left(\frac{1}{A_{lp}}\frac{dV_{wc}}{d\phi}\right) \tag{3}$$

Non-dimensional groups:

$$\lambda_1 = \frac{\dot{Q}_{id}}{D^2 u^*} = f(\nu, N, \Phi) \tag{4}$$

$$\lambda_2 = \frac{u^*}{\omega L} = \frac{\bar{V}_{sw}}{8\lambda_1} = f(\nu, N, \Phi) \tag{5}$$

The following relationships therefore apply:

- λ_1 relates D to the ratio of volumetric flow rate to nominal filling velocity. For a given value of u^*, the volumetric flow rate is proportional to D^2.
- λ_2 relates L to the ratio of nominal filling velocity to rotational speed. For a given value of u^*, the rotational speed is inversely proportional to the length.

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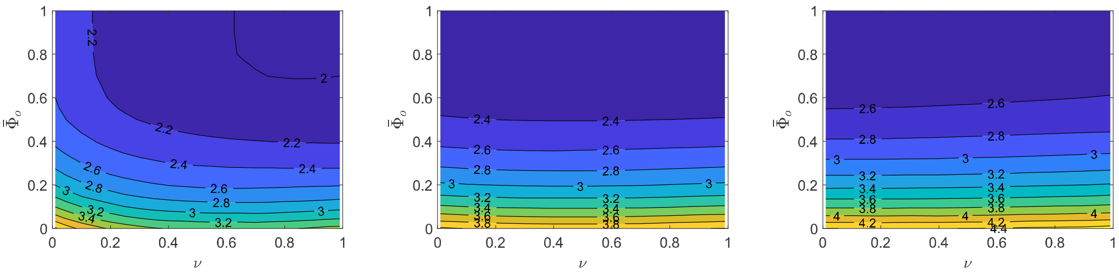
2.1. Selection of compressor operating speed

The rotational speed of the outer rotor is related to the diameter, due to the operating limits of the bearings. The speed limit, $v_{lim} = \omega D/2$, depends on the type of bearings used; based on reference speed data for single row cylindrical deep groove and roller bearings, an appropriate rotational speed limitation is $v_{lim} = 30m/s$, while for hydrodynamic bearings much higher surface speeds of 150-250 m/s are reported.

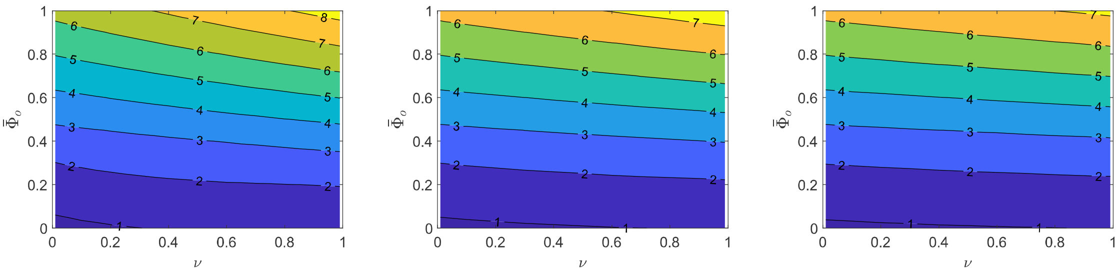
2.2. Non-dimensional parameters as functions of rotor geometry

The non-dimensionalised values of rotor diameter and L/D ratio are shown below as function of the profile shape parameter, ν, and the normalised wrap angle, $\bar{\Phi}_o$ (defined as the actual wrap angle divided by the angle of rotation required for a working chamber volume to go from zero to maximum and back to zero), for cases when $N_o = 3, 5$ and 7.



(a) (b) (c)

Figure 1: Contour maps showing $D\sqrt{\frac{u^*}{\dot{Q}_{id}}} = \sqrt{\frac{1}{\lambda_1}}$ as function of ν and $\bar{\Phi}_o$, for cases when $N_o = (a)3, (b)5$ and $(c)7$



(a) (b) (c)

Figure 2: Contour maps showing $\frac{L}{D}(\frac{v_{lim}}{u^*}) = \frac{1}{2\lambda_2}$ as function of ν and $\bar{\Phi}_o$, for cases when $N_o = (a)3, (b)5$ and $(c)7$

The results shown in Figures 1 and 2 can be used to produce a further plot of the non-dimensionalised machine volume ($\propto LD^2$), which is shown below for the same cases.

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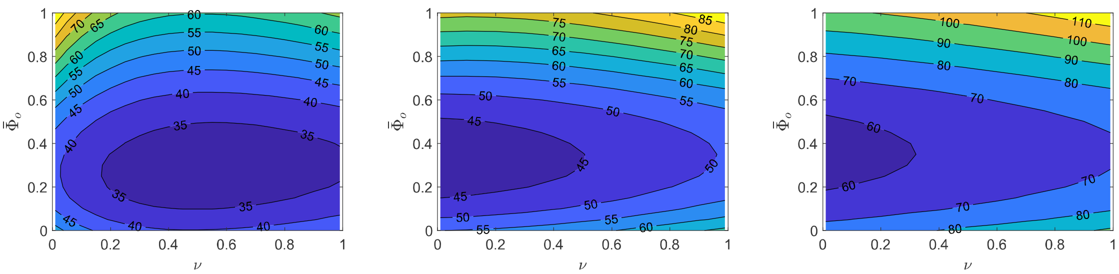

(a) (b) (c)

Figure 3: Contour maps showing $LD^2\sqrt{\frac{v_{lim}^2 u^*}{\dot{Q}_{id}^3}}$ as function of ν and $\bar{\Phi}_o$, for cases when $N_o = (a)3, (b)5$ and $(c)7$

The results in Figure 3 show that the machine volume can be minimised by using lower values of N_o, and that for the values considered ($N_o = 3, 5$ and 7) the volume is minimised with a value of $\bar{\Phi}_o = 0.3 - 0.4$. This may be an important requirement for particular applications, but there are other factors which should also be considered when selecting an appropriate geometry and wrap angle.

2.3. Power transfer between rotors

The pressure of the working fluid acting on the exposed faces of the rotors results in torque applied about the axes of rotation. As these axes are offset, the torque is different during rotation. If the variation of pressure in the working chamber is known, the rotor torques due to the fluid in a single working chamber can be calculated. The effect of all simultaneous working chambers can then be found, and by considering the sign and magnitude of these net rotor torques and the gearing ratio between them, the maximum proportion of input power being transferred between the rotors, $\Pi_{\max} = \min(\max(|P_o/P_{in}|), \max(|P_i/P_{in}|)$, can be found. Note that the choice of driven rotor should be made to ensure that $\max(|P_{idle}/P_{in}|)$ is minimised in order to reduce the power transferred via rotor-to-rotor contact. Once an application is characterised by specifying the pressure ratio, this can be used to estimate a required value of ϵ_v. In this paper a simple polytropic compression cycle has been assumed with no pressure losses during filling and emptying, and discharge pressure equal to maximum chamber pressure (i.e no over- or under-compression). The inlet conditions have been taken as 1.01 bar and 288K. Using Equation 6, the polytropic exponent has been chosen to allow a temperature rise of 100K for the compression process.

$$n_{poly} = 1 + \frac{\ln(T_{dis}/T_{suc})}{\ln(\epsilon_v)} \tag{6}$$

The examples shown in Figure 4 illustrate that once the working chamber pressure is defined, this is all the information required to make an initial assessment of suitable values of ν and $\bar{\Phi}_o$ that will limit the rotor-to-rotor power transfer for particular values of N_o. In all cases, the local minima shown at lower values of ν correspond to a driven

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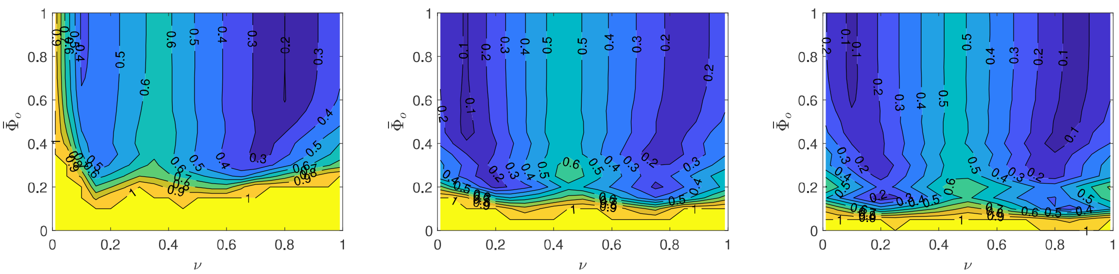

(a) (b) (c)

Figure 4: $\Pi_{\max}$ as function of ν and $\bar{\Phi}_o$, for cases with $\epsilon_v = 5$ when $N_o = (a)3, (b)5$ and $(c)7$ (note that contours are capped at a maximum value of 1)

inner rotor, while the local minima at higher values of ν correspond to a driven outer rotor.

2.4. Rotor-to-rotor leakage line lengths

The leakage paths that form between the rotors are an important geometrical characteristic of twin screw machines, as they will have a strong influence on the volumetric and isentropic efficiency of the machine. This can be characterised by considering the maximum total combined length of these rotor-to-rotor contact lines for single working chamber, $\max(\ell_{tot})$. This is only an approximate guide to leakage effects, as it does not consider other paths (such as end-face leakage), and does not consider the net effect of increasing and decreasing leakage areas between multiple working chambers during the compression process. It does however provide a useful indication of how this varies with different geometrical and operational parameters. A non-dimensional group, λ_3, can again be defined by dividing $\max(\ell_{tot})$ by the length of the helical path following the pitch circle radius of the outer rotor along its length. The value of λ_3 can be shown to always equal 2 for the epi-hypo composite rotor profiles considered in this paper; this can most clearly be seen in the case of straight rotors, where $\Phi_o = 0$, and the total length of the rotor-to-rotor contact paths will have a constant value of $2L$ throughout the compression cycle.

$$\lambda_3 = \frac{\max(\ell_{tot})}{\sqrt{\{L^2 + (\Phi_o \rho_{p,o})^2\}}} = \frac{\max(\ell_{tot})}{D\sqrt{\{(\frac{L}{D})^2 + (\frac{N_o \Phi_o}{2N_o + 4\nu})^2\}}} \tag{7}$$

$$\Phi_o = \bar{\Phi}_o \frac{2\pi}{N_o}(N_o + \nu) \tag{8}$$

$$\lambda_3 = \frac{\max(\ell_{tot})}{D\sqrt{\{(\frac{L}{D})^2 + (\frac{(N_o+\nu)\pi\bar{\Phi}_o}{N_o + 2\nu})^2\}}} = 2 \tag{9}$$

It is clear from Equation 9 that the values of D and L must both be known before the value of $\max(\ell_{tot})$ can be found for specific values of N_o, ν and $\bar{\Phi}_o$. The procedure for calculating these values and assessing the suitability of a design is discussed in Section 3.

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2.5. Bearing forces

The net forces exerted on the rotors can be considered in terms of the net torques exerted about a rotor's global x, y and z axes. In each case the torque about one axis is due to components of the net force aligned with the other two axes. In order to assess the bearing forces acting at each end of a rotor the torques about the x and y axes must be known. If we take the example of the torque about the x axis, it will consist of a torque due to a force in the y direction ($\propto LD^2$), plus a torque due to a force in the z direction ($\propto D^3$). It is therefore only possible to evaluate these bearing loads once values for L and D have been specified.

The maximum diameter of the outer rotor profile, D, can be used as the minimum allowable bore diameter for the outer rotor bearings. The minimum diameter of the inner rotor profile, $d = D(1 - \frac{3}{N_o+2\nu})$, can be used as the maximum allowable outer diameter for the inner rotor bearings.

The axial and radial forces exerted on the inner and outer rotor are of similar magnitude, and both fluctuate with period $2\pi/(\omega_o N_o)$ during machine operation. The significantly larger size of the outer rotor bearings imposes a speed limitation on the machine, and generally leads to these being lightly loaded. Ensuring that these bearings achieve the required minimum load is therefore a key design requirement. Although the inner rotor rotates faster than the outer due to the gearing ratio, the smaller bearing diameter means that these operate well below their speed limit. In both cases the operating life of the bearing under the load conditions is a key consideration. The results for bearing loads presented for the design example in the following section therefore consider both the minimum and maximum axial force and radial forces (at the high and low pressure ends of the machine) for the rotors. The results also assume that the bearing centres are located at the corresponding end face (hp or lp) for both the inner and outer rotor. Bearing offsets can easily be applied to this calculation for more accurate calculation of bearing loads in a later stage of the design process once suitable components have been identified.

3. Design procedure

Specifying the required operational parameters $\dot{Q}_{id}$, u^*, and υ_{lim} allows the required dimensions of the machine to be identified for particular values of ν and $\bar{\Phi}_o$. The initial design procedure can therefore be applied as follows:

$$D = \sqrt{\frac{\dot{Q}_{id}}{\lambda_1 u^*}} \tag{10}$$

$$\frac{L}{D} = \frac{u^*}{2\upsilon_{lim}\lambda_2} \tag{11}$$

$$\max(\ell_{tot}) = 2D\sqrt{\left(\frac{L}{D}\right)^2 + \left(\frac{(N_o+\nu)\pi\bar{\Phi}_o}{N_o+2\nu}\right)^2} \tag{12}$$

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(i) Assume initial values for profile shape (N, ν) and wrap angle $(\bar{\Phi})$

(ii) Calculate or lookup non-dimensional parameters λ_1, λ_2 and $\Pi_{\max}$ for specific profile shape and wrap angle

(iii) Specify required $\dot{Q}_{id}$ and allowable u^*

(iv) Use Equation 9 to find D

(v) Specify v_{lim} and calculate corresponding ω_o

(vi) Use Equation 10 to find L

(vii) Use Equation 11 to find $\max(\ell_{tot})$

(viii) Define fitness function based on relative importance of machine volume, Π_{max}, rotational speed, $\max(\ell_{tot})$, and bearing loads

(ix) Iterate profile shape and wrap angle to optimise fitness function

3.1. Example of geometry selection

In order to investigate the initial design of an internally geared screw compressor, a conventional twin screw machine has been used to specify the basic operating parameters, and as a basis for comparison. This machine is a typical design for a small oil-injected air compressor, and the key geometrical parameters are as follows:

Table 1: Example of conventional twin screw compressor geometry

Main/gate rotor lobe no.	4/5
Rotor centre distance	71.0 mm
Rotor length	158.0 mm
Main rotor outer diameter	101.9 mm
Gate rotor outer diameter	80.2 mm
Main rotor wrap angle	306°
Swept volume	0.720 litres/rev of the main rotor

Based on this screw compressor geometry, the software SCORG [5] has been used to assess the volume and port areas as functions of rotor angular position. For a main rotor tip speed of 40m/s (corresponding to $\omega = 785$rad/s) this results in values of $\dot{Q}_{id} = 0.0895\text{m}^3/\text{s}$ and $u^* = 23.1$m/s. The value of the bearing limiting speed, v_{lim}, assumed for rolling element bearings is 30m/s. These input values have been applied to a internally geared geometry with $N_o = 5$, resulting in the following dimensional contour plots shown in Figures 5-7. These figures show the relevant geometric and operational parameters as functions of ν and $\bar{\Phi}_o$ for this specific application.

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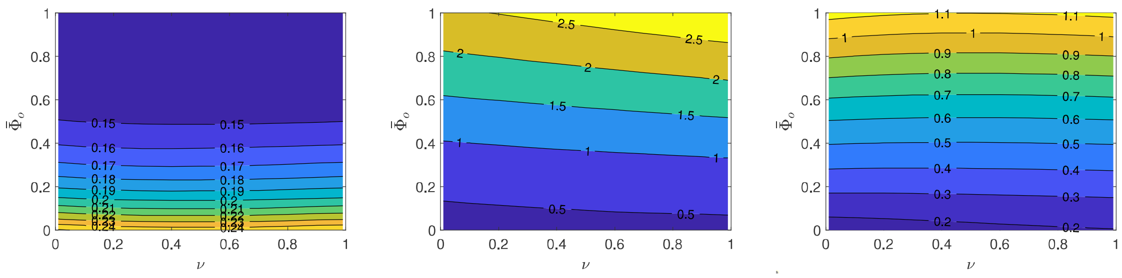
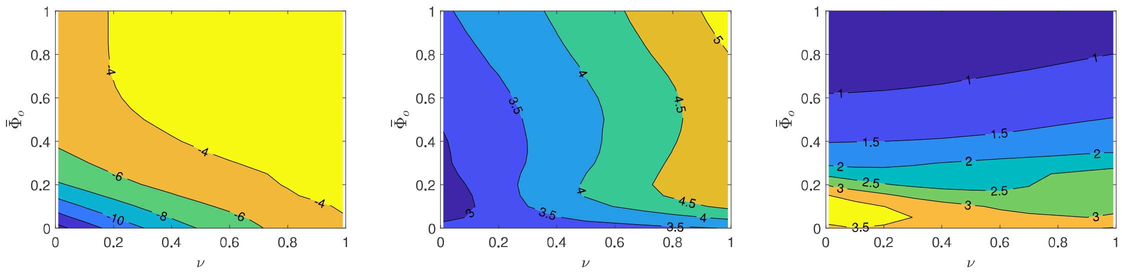


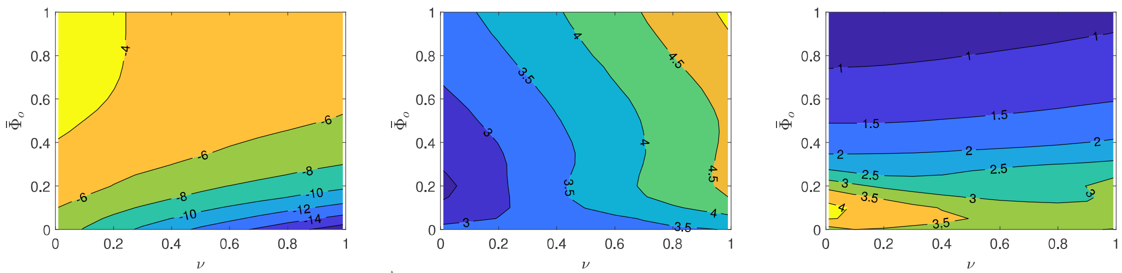
(a) (b) (c)

Figure 5: $(a) D$, $(b) L/D$, and $(c) \max(\ell_{tot})$ as functions of ν and $\bar{\Phi}_o$ for $N_o = 5$, $\dot{Q}_{id} = 0.0895\text{m}^3/\text{s}$ and $u^* = 23.1\text{m/s}$



(a) (b) (c)

Figure 6: Outer rotor radial and axial minimum bearing forces: $(a) \max(|B_{z,o}|)$, $(b) \max(B_{rad,o})_{hp}$, $(c) \max(B_{rad,o})_{lp}$ for $N_o = 5$, $\dot{Q}_{id} = 0.0895\text{m}^3/\text{s}$, $u^* = 23.1\text{m/s}$ and $\epsilon_v = 5$



(a) (b) (c)

Figure 7: Inner rotor radial and axial maximum bearing forces: $(a) \max(|B_{z,i}|)$, $(b) \max(B_{rad,i})_{hp}$, $(c) \max(B_{rad,i})_{lp}$ for $N_o = 5$, $\dot{Q}_{id} = 0.0895\text{m}^3/\text{s}$, $u^* = 23.1\text{m/s}$ and $\epsilon_v = 5$

If the aim of the design exercise is chosen to be minimising the value of $\Pi_{\max}$ for an outer-driven configuration, the results in Figure 4 for the case when $N_o = 5$ show that this occurs when $\nu = 0.8$ and $\bar{\Phi}_o = 0.5$. This results in the geometric design parameters shown in Table 2. In this case the results suggest that, compared to the

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Table 2: Example of initial specification of internally geared compressor geometry

Input parameters	$\dot{Q}_{id}$	0.0895 m^3/s
	u^*	23.1 m/s
	N_o	5
	ϵ_v	5
Conditions for min power transfer	ν	0.8
	$\bar{\Phi}_o$	0.5
	$\Pi_{\max}$	0.0982
	Driven rotor	Outer
Geometrical parameters	Φ_o	209°
	D	149.4 mm
	d	81.5 mm
	L	209.6 mm
	$\max(\ell_{tot})$	588 mm
Operational parameters	ω_o	402 rad/s
	ω_i	502 rad/s
	n_{poly}	1.185
	p_{suc}	1.01 bar
	p_{dis}	6.74 bar
Bearing loads	$\min / \max(B_{z,o})$	-2.85 / -2.70 kN
	$\min / \max(B_{rad,o})_{hp}$	3.62 / 4.44 kN
	$\min / \max(B_{rad,o})_{lp}$	1.18 / 1.44 kN
	$\min / \max(B_{z,i})$	-5.74 / -5.53 kN
	$\min / \max(B_{rad,i})_{hp}$	3.45 / 4.21 kN
	$\min / \max(B_{rad,i})_{lp}$	1.33 / 1.69 kN

conventional machine, the internally geared configuration has a larger maximum rotor diameter, lower rotational speed, and longer rotors. The wrap angle of the rotors is significantly lower for the internally geared configuration, leading to significantly lower maximum total radial leakage line length.

The bearing loads and rotational speeds defined in Table 2 have been used to select appropriate bearings using the SKF 'Bearing Select' tool [6], as shown in Table 3. For both rotors, the axial loads are assumed to be supported by bearings at the high pressure (hp) end of rotors, in combination with the radial load. This is to allow the end face clearance gap to be accurately set during assembly of the machine, and in order to accommodate possible thermal expansion of the rotors in operation, the design must allow non-locating displacement on the seat for the deep groove bearing at the low pressure (lp) end of the outer rotor. For the inner rotor, non-locating displacement occurs within the cylindrical roller bearing selected. The basic rating life of the bearings (a factor only of the load and speed) has been assessed, along with the SKF rating life which is also dependant on lubrication conditions and the bearing fatigue load limit. The lubricant is assumed to be ISO 46 oil. The rated life of the hp bearings is significantly lower due to the higher combined radial and axial loads, although this could be increased if necessary through the use of multiple bearings to distribute these loads.

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Table 3: Rolling element bearings selected for loads identified in Table 2; bearing types are deep groove (DG), angular contact (AC) and cylindrical roller (CR).

Bearing location	-	-	o, lp	o, hp	i, lp	i, hp
Designation	-	-	61832		N207ECP	QJ306MA
Type	-	-	DG		CR	AC
Bearing bore	d_b	mm	160		35	30
Bearing outer diameter	D_b	mm	200		72	72
Bearing width	B_b	mm	20		17	19
Basic dynamic load rating	C	kN	49.4		56.0	53.0
Basic static load rating	C_0	kN	64.0		48.0	41.5
Fatigue load limit	P_u	kN	2.0		6.1	1.8
Minimum load	F_m	kN	0.731		0.271	0.117
Reference speed	n_{ref}	r/min	6300		11000	-
Limiting speed	n_{lim}	r/min	4000		12000	17000
Basic rating life	L_{10h}	h	175000	1360	$> 2e^5$	793
SKF rating life (ISO 46)	L_{10mh}	h	$> 2e^5$	63400	$> 2e^5$	12800

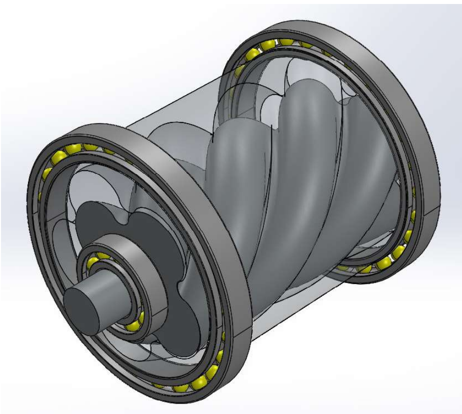

Figure 8: Model illustrating basic geometry of rotors and bearings as specified in Tables 2 and 3 (note that outer rotor is shown as transparent for clarity)

4. Conclusions

Basic design of a internally geared gerotor type screw compressor has been performed. Key operational parameters considered are ideal volumetric flow rate, nominal port flow velocity, and bearing velocity limit. Through the definition of non-dimensional shape parameters, relationships showing rotor length and diameter required as a function of profile shape and helical wrap angle are found. Once the rotor geometry is defined, the leakage line length and the bearing forces can be calculated. A design procedure is described in which the optimum geometry can be chosen based on a fitness function involving power transfer ratio, machine volume, leakage line length, operating speeds and bearing loads. A conventional twin-screw compressor is used to illustrate this design process for an equivalent gerotor-type machine. Rotor geometry and bearing selection are characterised. While this does not allow direct comparison of the performance of the conventional and internally-geared, detailed thermodynamic chamber modelling and experimental testing are planned to investigate the operation of these machines. Future work will focus on applying this method to design an experimental machine to investigate the performance of this novel compressor configuration.

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