DEVELOPMENT OF A ROTOR PROFILE FOR SILENT SCREW COMPRESSOR OPERATION

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ABSTRACT

There is an increasing need for reduced noise levels in screw compressors. The theoretical principles used for generation of a rotor profile for silent screw compressor operation are presented paper. A pair of oil injected air compressor rotors, designed for efficient compression and currently in use are compared with another made to run more silently. Experimental results obtained from both sets of rotors showed that the new rotors reduced the compressor noise without any significant performance penalty.

1 INTRODUCTION

As environment protection legislation becomes stricter, the demand for reduced noise levels from all forms of machinery increases and hence the need for silent or low noise levels from compressors becomes more significant. Since rotor profiles for silent compressor operation are generally associated with a loss in efficiency, it is necessary to seek a means of generating them in a manner that minimises performance loss. An optimization procedure is therefore needed to balance these conflicting requirements. This must be based both on analytical techniques commonly used for screw profile development and experimental data to confirm the method.

A significant part of the noise generated in a screw compressor originates from contact of its moving parts. These are principally the rotors, the gears and the bearings. This mechanical noise is caused by contact between the rotors due to pressure and inertial torque, together with torque caused by oil drag forces, acting circumferentially upon the driven rotor. It is also due to contact between the rotor shafts and bearings due to the radial and axial pressure and inertial forces. These forces should be as constant as possible to minimise noise. Unfortunately, the radial and axial forces and rotor torque, which create the rotor contact forces, are not constant, due to the periodic character of the pressure loads. Imperfections in the rotor manufacture and compressor assembly also contribute to this. *Soedel, 2006*, has described these effects fully.

If the torque on the driven rotor changes its sign, contact between the rotors will be lost and impact will occur due to their repeated contact. This is known as rotor rattling and its effects on noise may be severe.

The method of analysis of pressure loads acting upon the screw compressor rotors, originally presented by *Stosic et al*, 2005 has been used to identify the effects which the rotor profile imposes upon the driven rotor torque. Also included, is the influence of machining imperfections, originally presented by *Stosic 2006*.

2 PRESSURE FORCES ACTING ON SCREW COMPRESSOR ROTORS

Screw compressor rotors are subjected to high pressure loads. For any instantaneous angle of rotation θ . The pressure $p(\theta)$ creates radial and torque forces at any cross section, as shown in Fig 1 The pressure p acts in the corresponding interlobes normal to line AB, where A and B are on the sealing line either between the rotors or on the rotor tips. Thus their position is fully defined by the rotor geometry.

At position 1, there is no contact between the rotors. Since A and B are on the circle, the overall forces F_1 and F_2 act towards the rotor axes and are purely radial. Thus there is no torque caused by pressure forces in this position. At position 2, there is only one contact point between the rotors at A. Forces F_1 and F_2 are eccentric and have both radial and circumferential components. The latter cause the torque. Due to the force position, the torque on the gate rotor is significantly smaller then that on the main rotor. At position 3, both contact points are on the rotors, with overall and radial forces equal for both rotors. These also cause torque, as at position 2.

The radial force components are:

$$R_{x} = -p \int_{A}^{B} dy = -p(y_{B} - y_{A}), \quad R_{y} = -p \int_{A}^{B} dx = -p(x_{B} - x_{A}), \quad R_{z} = -p \int_{A}^{B} dx = -p(x_{B} - x_{A}) \quad (1)$$

The torque can be expressed as:

$$T = p \int_{A}^{B} x dx + p \int_{A}^{B} y dy = 0.5p \left(x_{B}^{2} - x_{A}^{2} + y_{B}^{2} - y_{A}^{2} \right)$$
(2)

The axial forces are:

$$Z = p \int_{A}^{B} x dx + p \int_{A}^{B} y dy = 0.5 p \left(x_{B}^{2} - x_{A}^{2} + y_{B}^{2} - y_{A}^{2} \right)$$
(3)

The above equations are integrated along the profile for all profile points. Then they are integrated for all angle steps to complete one revolution, given the pressure history $p=p(\theta)$. Finally, the sum for all rotor interlobes is obtained after taking account of both the phase and axial shift between the interlobes.

The rotor axial and radial forces are transferred to the housing through the bearings. Rolling element bearings are normally chosen for small and medium screw compressors and these must be carefully selected to obtain a satisfactory design. Usually, two bearings are employed on the discharge end of each of the rotor shafts in order to absorb the radial and axial loads separately. Also, the distance between the rotor centre lines is in part determined by the bearing size and internal clearance. Any production imperfection in the bearing housing, like displacement or eccentricity, will change the rotor position and thereby influence the compressor behaviour.



Fig 1 Pressure forces acting on screw compressor rotors

3 ROTOR MOVEMENT IN THE COMPRESSOR BEARINGS

The natural movement of rotors in their bearings is the rotation of their shafts around their axes. While the main rotor rotates around the Z_1 axis through angle θ , the gate rotor rotates through angle $\tau = r_{1w}/r_{2w}\theta = z_2/z_1\theta$, where r_w and z are the pitch circle radii and number of rotor lobes respectively. However, there may also be rotation of the rotors about the axes perpendicular to their shafts due to the bearing clearances and manufacturing imperfections. This may cause additional contact noise.

A procedure is presented here, which calculates the rotor movement within the range of the bearing clearances and manufacturing imperfections of the rotor and bearing housings. This enables the bearing clearance and housing manufacturing tolerance limits to be correctly specified in order to prevent rotor interference when interlobe clearances are small and consequently to avoid rotor contact noise. More details can be found in *Stosic et al*, 2003.

The location of screw compressor rotors in their bearings is presented in Fig 2. The rotor shafts are supposed to be parallel and their positions are defined by axes Z_1 and Z_2 .



Fig 2 Rotor shafts in the compressor housing and displacement in bearings

The bearings are labeled 1 to 4, and their clearances, as well as the manufacturing tolerances of the bearing bores, δ_x and δ_y in the x and y directions respectively, are presented in the same figure. The rotor centre distance is C and the axial span between the bearings is a.

All imperfections in the manufacture of the rotors should be accounted for by production tolerances. These include wrong position of the bearing bores, eccentricity of the rotor shafts, bearing clearances and imperfections and rotor misalignment. Together, they account for the rotor shafts not being parallel.



Fig 3 Rotors with non parallel shafts and their coordinate systems, with intersecting axes above and non-intersecting axes below

Let the rotor movement δ_y in the y direction include all displacements. These are presented in Fig 2, and cause virtual rotation of the rotors around the X₁, and X₂ axes, as shown in Fig 4. Let rotor movement δ_x in the x direction cause rotation around the Y₁, and Y₂ axes, as shown in Fig 2. The movement δ_x can cause the rotor shafts to intersect. However, the movement δ_y causes the shafts to become non-parallel and non-intersecting. These both change the nature of the rotor position so that the shafts can no longer be regarded as parallel. The following analytical approach enables the rotor movement to be calculated and accounts for these changes. It was first presented by *Stosic et al*, 2001.

In the case of non parallel intersecting shafts, the angle between the axes, obtained by rotation about the Y₁ axis, is ζ . As shown in Fig 3, vectors $\mathbf{r}_1 = [x_1, y_1, z_1]$ and \mathbf{r}_2 , given in equation (4) now represent the helicoid surfaces of the main and gate rotors on intersecting shafts.

$$\mathbf{r}_{2} = [x_{2}, y_{2}, z_{2}] = [x_{1} \cos \varsigma - z_{1} \sin \varsigma - C, y_{1}, x_{1} \sin \varsigma + z_{1} \cos \varsigma]$$
(4)

$$tg\varsigma = \frac{\delta_x}{a} \tag{5}$$

Since this rotation angle is very small, equation (4) can be rewritten in a simplified form as equation (6), which can be used more easily for further analysis.

$$\mathbf{r}_{2} = [x_{2}, y_{2}, z_{2}] = [x_{1} - z_{1}\varsigma - C, y_{1}, x_{1}\varsigma + z_{1}]$$
(6)

The rotation ζ will result in a displacement $-z_1\zeta$ in the x direction and a displacement $x_1\zeta$ in the z direction, while there is no displacement in the y direction. The displacement vector becomes:

$$\Delta \mathbf{r}_2 = \begin{bmatrix} -z_1 \varsigma, 0, x_1 \varsigma \end{bmatrix}$$
(7)

In the majority of practical cases, x_1 is small compared with z_1 and only the displacement in the x direction need be considered. This means that rotation around the Y axis will practically change only the rotor centre distance. Displacement in the z direction may be significant for the dynamic behaviour of the rotors. Displacement in the z direction will be adjusted by the rotor relative rotation around the Z axis. This can be accompanied by significant angular acceleration and may cause the rotors to lose contact at certain stages of the compressor cycle, thus create rattling.

As shown in Fig 3 below, vectors $\mathbf{r}_1 = [x_1, y_1, z_1]$ and \mathbf{r}_2 , given by equation (8) now represent the helicoid surfaces of the main and gate rotors on the intersecting shafts. Σ is the rotation angle around the X axes given by (11).

$$\mathbf{r}_{2} = [x_{2}, y_{2}, z_{2}] = [x_{1} - C, y_{1} \cos \Sigma - z_{1} \sin \Sigma, y_{1} \sin \Sigma + z_{1} \cos \Sigma]$$
(8)
$$tg\Sigma = \frac{\delta_{y}}{a}$$
(9)

Since angle Σ is very small, equation (8) can be rewritten in simplified form as equation (10), which facilitates further analysis.

$$\mathbf{r}_{2} = [x_{2}, y_{2}, z_{2}] = [x_{1} - C, y_{1} - z_{1}\Sigma, y_{1}\Sigma + z_{1}]$$
(10)

The rotation Σ will result in displacement $-z_1\Sigma$ in the y direction and displacement $y_1\Sigma$ in the z direction, while there is no displacement in the x direction. The displacement vector can be written as:

$$\Delta \mathbf{r}_2 = \begin{bmatrix} 0, -z_1 \Sigma, y_1 \Sigma \end{bmatrix}$$
(11)

Although, in the majority of practical cases, displacement in the z direction is very small and therefore unimportant for consideration of rotor interference, it may play a role in the dynamic behaviour of the rotors. The displacement in the z direction will be fully compensated by a regular rotation of the rotors around the Z axis. However, the angular acceleration involved in this process may cause rotors to lose contact at some stages of the compressor cycle.

Rotation about the X axis is effectively the same as if the main or gate rotor rotated relatively through angles $\theta = -z_1 \Sigma / r_{1w}$ or $\tau = z_1 \Sigma / r_{2w}$ respectively and the rotor backlash will be reduced by $z_1 \Sigma$.

4 DESIGN IMPLICATIONS OF GATE ROTOR TORQUE AND MOVEMENT

Oil flooded compressors have direct contact between their rotors. In well designed rotors, the clearance distribution will be set so that this is first made along their, so called, contact bands, which are positioned close to the rotor pitch circles to reduce the danger of their seizing. As shown in Fig 5, the contact band may be either on the rotor flat flank, left, or on the rotor round flank, right.

The traditional approach is to maintain a high, so called, positive gate rotor torque, which ensures round flank contact. However, there are advantages to be gained from maintaining a negative gate rotor torque to ensure that contact, when it occurs, will be on the flat face. The reason for this can be understood by examination of the sealing line lengths, shown as item 5 in Fig 4. Here it can be seen clearly that the flat flank sealing line is much longer than that of the round flank. Thus, minimising the clearance on the flat flank will reduce the interlobe leakage more than minimising the round flank clearance. Also, negative gate torque is achieved by making the gate rotor lobes thicker and the main rotor lobes correspondingly thinner. The displacement is thereby increased. Thus both these effects lead to higher compressor flows and efficiencies.

However, it is essential to keep the torque direction constant to prevent any loss of rotor contact and avoid eventual chatter. Since the torque caused by oil drag is always positive, it may overwhelm the negative pressure torque. Therefore, a good advice is to keep the negative pressure torque always smaller in absolute value than the oil drag torque on the female rotor to avoid a change in the torque sign. Since it is difficult to predict the oil drag, a careful calculation must precede the rotor design and thorough experimentation has to follow to be sure that the design met the requirements. A safe solution is to start with a positive pressure torque which, supported by the positive drag torque will always cause rotor contact on its round lobe side

There must be some clearance in the shaft bearings, therefore the pressure loads will tend to push the rotors apart and displace their centres from their design position with respect to the compressor housing. Thus, if the bearing centres are set to be the same as those of the rotors, the rotors will be eccentric and as a result the clearance between the rotors and housing will be smaller at the low pressure side of the rotors and larger at the high pressure side. This will keep the rotors parallel. Moreover, since leakage is caused by the pressure difference, this displacement creates the most favourable rotor position for efficient compressor operation.



Fig 4 Screw compressor rotors, 1-main, 2-gate, 3-rotor external and 4-pitch circles, 5sealing line, 6-clearance distribution, 7- rotor displacement area

Therefore, precise rotor and housing manufacture is an essential prerequisite for silent operation of screw compressor rotors. The involute rotor profile in the contact area, as well as reduction of bearing centre distance in the discharge area may help to minimise the effects of the movement caused by the intersecting axes misalignment. These factors were taken fully into account and a family of rotors dedicated to quieter screw compressor operation has been introduced. One example is presented here.

5 ROTOR MODIFICATION FOR SILENT SCREW COMPRESSOR OPERATION

A standard set of compressor rotors 4/5-102 mm was modified to accommodate the principles for reducing compressor noise, presented in the previous sections. The pressure angle on the round side was increased for that purpose. The standard and modified rotors are presented on the left and right, respectively in Fig 5.

The calculated pressure torque is presented in Fig 6, left for the standard rotors and right for the modified ones. The standard rotors have a negative female torque, while the new ones have a positive female torque. This causes contact on the flat side for the standard rotors and on the round side for the new rotors, as presented in the left and right, respectively, of Fig 5.



Fig 5 Standard screw compressor rotors, left, Rotors modified for silent operation, right

6 NOISE AND PERFORMANCE MEASUREMENTS

A pair of rotors, designed to reduce noise level, were designed and manufactured. A standard compressor was used to accommodate the old and new rotors to measure and compare their performance and noise. The measurements were performed at City University compressor laboratories in the period January-March 2007 and the test installation is presented in Fig 7.

The noise and performance results obtained for standard and new rotors are presented in Tables 1 and 2 respectively, while a comparison between the standard and new rotor noise and performance characteristics is given in figures 8 and 9 respectively.



Fig 6 Torque, main rotor above, gate rotor below, standard rotors left, new rotors right



Fig 7 4/5-102 compressor in test bed

Table 1 Standard N rotors

Sound Pressure Level [dB]

					Flow	Power	Spec Power
Speed [rpm]	Left	Back	Right	Suct.	[m3/min]	[kW]	[kw/m3/min]
2000	87	90	88	107	1.361	8.50	6.24
3000	93	95	94	114	2.005	12.89	6.43
4000	95	95	96	116	2.703	17.63	6.53
5000	99	101	101	119	3.320	22.42	6.75
6000	103	104	104	123	3.969	28.08	7.08

 Table 2 NSilent rotors

Sound Pressure Level [dB]

					Flow	Power	Spec Power
Speed [rpm]	Left	Back	Right	Suct.	[m3/min]	[kW]	[kw/m3/min]
2000	86	87	86	105	1.295	8.51	6.57
3000	90	91	91	110	2.001	12.45	6.22
4000	93	92	93	114	2.698	17.29	6.41
5000	98	99	99	118	3.373	22.39	6.64
6000	103	104	104	120	4.018	27.39	6.82

Sound Pressure Level - Compresssor Speed



Fig. 8 Sound improvement N – NSilent Rotors at 8 bars

Specific Power - Compressor Speed



Fig. 9 Performance comparison N – NSilent Rotors at 8 bars

The sound pressure level was measured around the compressor at its left and right side and rear, as well as in its suction pipe. The noise reduction introduced by NSilent rotors compared with the standard rotors was up to 3 dB, while the average was 2.08 dB. The measures indicate that the difference in performance between the standard N and NSilent rotors appears to be insignificant.

7 CONCLUSION

The rotor contact component of screw compressor noise has been reduced by careful redesign of standard rotors to change the sign of the gate rotor torque. As a result, the contact force between the rotors became more uniform, which reduced chatter and prevented rattling. At the same time, the performance of the compressor with modified rotors was unchanged.

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