Calculation of Rotor Interference in Screw Compressors

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Abstract: Small rotor clearances are today a vital prerequisite for an efficient screw compressor. The clearances are responsible for compressor leakages, small clearance will minimise them and thereby increase both, the volumetric and adiabatic efficiency. A continuous improvement in the rotor manufacturing equipment has been achieved during the recent years and tight production tolerances may now be applied to compressor rotors. This imposes a new approach upon the design of compressor elements, such as the rotor and bearing housings and bearings to respond properly to this challenge.

A mathematical apparatus to quantify a change in rotor position in screw compressors due to the bearing clearance and the imperfections in compressor housing manufacturing is presented in this paper an applied for identification of their significance and influence upon the screw compressor behaviour. The analysis based on this approach has shown that the change in rotor position may have negative, either static or dynamic effects on the compressor efficiency, safety, endurance and reliability and enhancing compressor noise.

1. INTRODUCTION

Compressor efficiency depends on the rotor profile and clearances between the rotors and between the rotors and compressor housing. For such machines to perform effectively the rotors must meet meshing requirements and should maintain a seal along the entire band of rotor contact.

Rotors of screw compressors are usually manufactured on specialised machines by the use of formed milling or grinding tools. Machining accuracy achievable today is high and tolerances in rotor manufacture are of the order of 5 μm around the rotor lobes. *Holmes and Stephen, 1999* reported that even higher accuracy was achieved on the new Holroyd vitrifying thread-grinding machine, thus keeping the manufacturing tolerances within 3 μm even in large batch production. This means that, as far as rotor production alone is concerned, clearances between the rotors can be as small as 12 μm.

To obtain the full potential advantages derived from such accuracy in rotor manufacture, the design and manufacture of all other components must be reviewed. Thus rotor to housing clearances, especially at high pressure must be properly selected. This in turn requires either precise bearings with smaller clearances or looser bearings with their clearances reduced to an acceptable value by preloading.

Screw compressor rotors, especially in oil flooded machines which operate with high pressure differences, are heavily loaded by axial and radial forces. These are transferred to the housing by 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. 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.

A reliable procedure is required to analyse rotor movement within the range of the bearing clearances and manufacturing imperfections of the rotor and bearing housings. This will enable bearing clearance and housing manufacturing tolerance limits to be correctly specified in order to avoid rotor interference when inter lobe clearances are reduced to the levels now possible. The following is a simple and effective approach which does this.

2. IDENTIFICATION OF ROTOR MOVEMENT IN COMPRESSOR BEARINGS

The system of rotors in screw compressor bearings is presented in Fig 1. The rotor shafts are parallel and their positions are defined by axes Z_1 and Z_2.

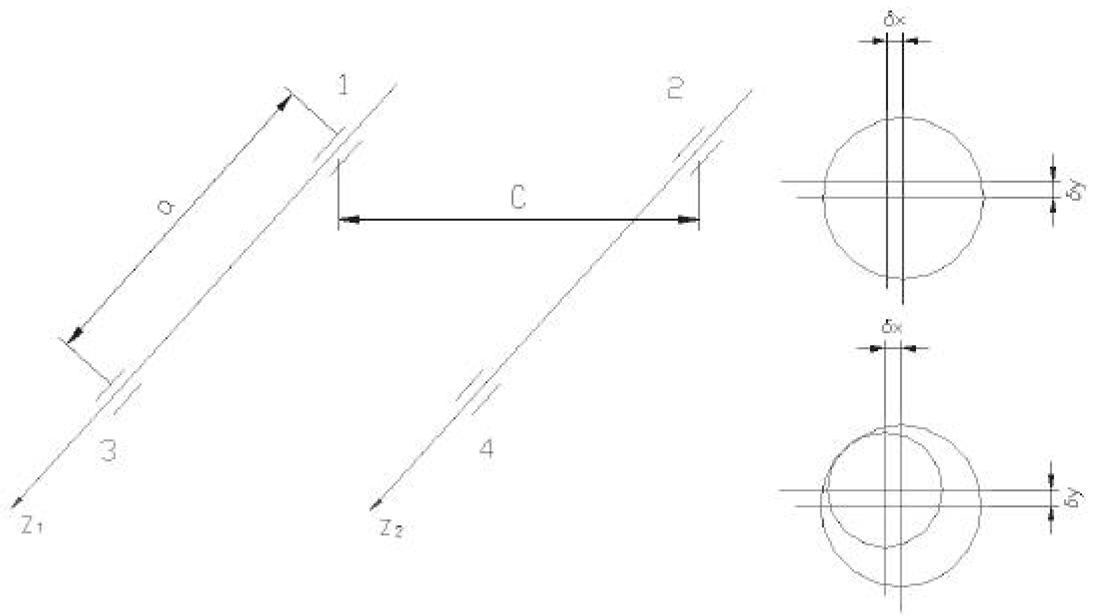

Fig 1. Rotor shafts in the compressor housing and displacement in bearings

The bearings are labelled 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 distance span between the bearings is a.

Vectors $\mathbf{r}_1$ and $\mathbf{r}_2$ represent the surfaces of rotor helicoids of the main and gate rotor respectively. x_{01}, y_{01} and x_{02}, y_{02} are the point coordinates at the end rotor section in the coordinate systems fixed to the main and gate rotors. The angle θ is obtained by the envelope method which is in detail described by *Stosic, 1998*.

$$\mathbf{r}_1 = \mathbf{r}_1(t,\boldsymbol{\theta}) = [x_1, y_1, z_1] = [x_{01}\cos\boldsymbol{\theta} - y_{01}\sin\boldsymbol{\theta}, x_{01}\sin\boldsymbol{\theta} + y_{01}\cos\boldsymbol{\theta}, p\boldsymbol{\theta}] \tag{1}$$

$$\mathbf{r}_2 = [x_2, y_2, z_2] = [x_1 - C, y_1, z_1] = [x_{02}\cos\boldsymbol{\tau} - y_{02}\sin\boldsymbol{\tau}, x_{02}\sin\boldsymbol{\tau} + y_{02}\cos\boldsymbol{\tau}, p\boldsymbol{\tau}] \tag{2}$$

where p is the rotor lead per unit angle of rotation.

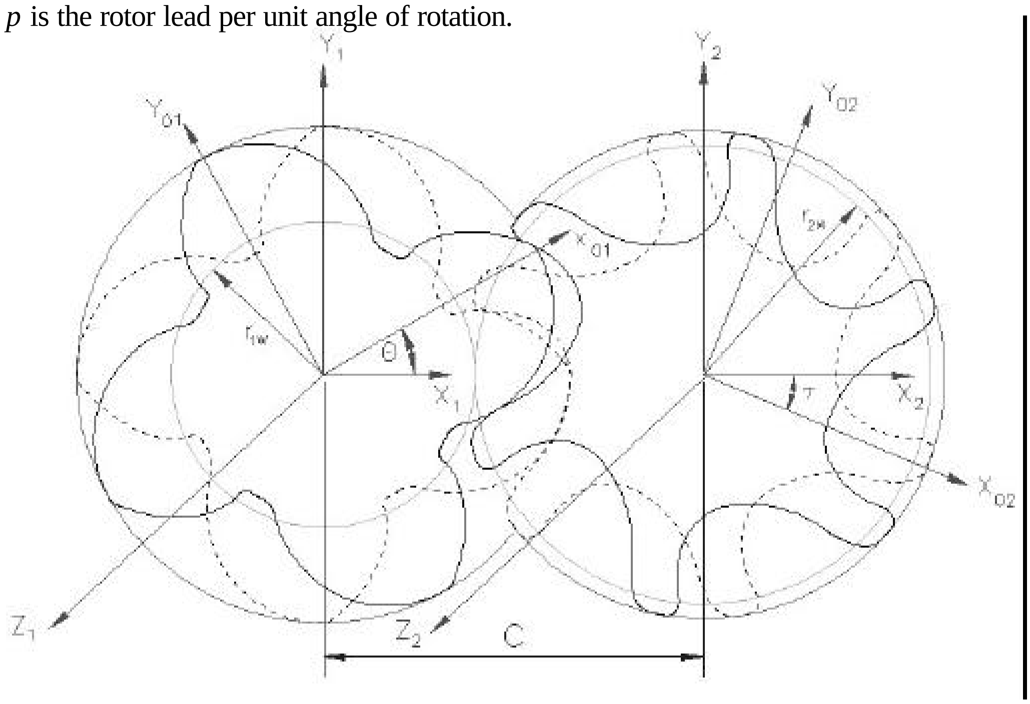

Fig 2. Rotors with parallel shafts and their coordinate systems

Equations (1) and (2) give the relation between the main and gate rotor point coordinates for a rotation angle θ. Rotation of the rotor shaft is the natural rotor movement in its bearings. While the main rotor rotates 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.

Andreev 1961 and, more recently, Xing 2000 also use the same method for screw compressor rotor generation in their books on screw compressors. A similar procedure has been described by Tang, 1995.

All imperfections in the manufacture of screw compressor rotors are expected to fall within and can be accounted for by production tolerances. These are the wrong position of the bearing bores, eccentricity of the rotor shafts, bearing clearances and imperfections and rotor misalignment and together, they account for the rotor shafts not being parallel. Let a rotor movement δ_y in the y direction contain all displacements, which are presented in Fig 1, and cause virtual rotation of the rotors around the X_1, and X_2 axes, as shown in Fig 3. Let a rotor movement δ_x in the x direction cause rotation around the Y_1, and Y_2 axes, as shown in Fig 4. 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 mathematical description describes the rotor movement and accounts for these changes.

As shown in Fig 3, vectors $\mathbf{r}_1=[x_1,y_1,z_1]$ and $\mathbf{r}_2$ given in equation (3) now represent the helicoid surfaces of the main and gate rotors on intersecting shafts. The shaft angle ζ is a rotation around Y.

$$\mathbf{r}_2 = [x_2, y_2, z_2] = [x_1 \cos V - z_1 \sin V - C, y_1, x_1 \sin V + z_1 \cos V] \qquad (3)$$

$$tgV = \frac{d_x}{a} \qquad (4)$$

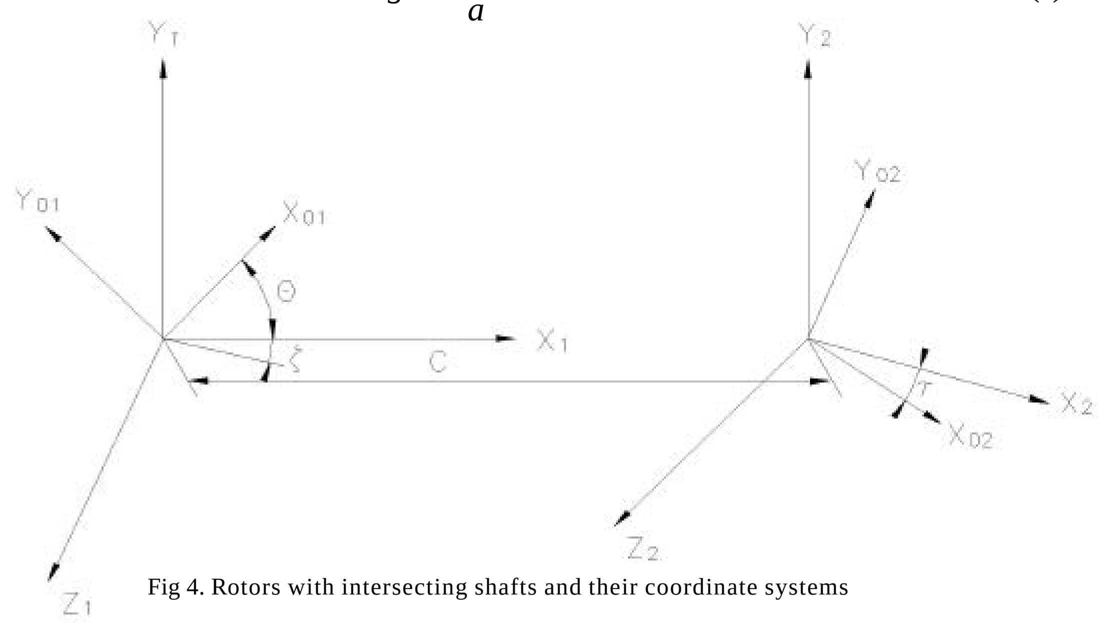

Fig 4. Rotors with intersecting shafts and their coordinate systems

Since this rotation angle is usually very small, equation (4) can be rewritten in a simplified form as equation (5), which can be used more easily for further analysis.

$$\mathbf{r}_2 = [x_2, y_2, z_2] = [x_1 - z_1V - C, y_1, x_1V + z_1] \qquad (5)$$

The rotation z will result in a $-z_1z$ displacement in the x direction and a displacement x_1z in the z direction, while there is no displacement in the y direction. The displacement vector becomes:

$$\Delta\mathbf{r}_2 = [-z_1V, 0, x_1V] \qquad (6)$$

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

Since the rotation angle z caused by displacement within the tolerance limits is very small, a two-dimensional presentation in the rotor end plane can be applied, as is done in the next section.

As shown in Fig 5, vectors $\mathbf{r}_1=[x_1,y_1,z_1]$ and $\mathbf{r}_2$, given by equation (7) 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 (8).

$$\mathbf{r}_2 = [x_2, y_2, z_2] = [x_1 - C, y_1 \cos\Sigma - z_1 \sin\Sigma, y_1 \sin\Sigma + z_1 \cos\Sigma] \tag{7}$$

$$tg\Sigma = \frac{d_y}{a} \tag{8}$$

Since angle Σ is very small, equation (7) can be rewritten in simplified form as equation (9), which is simplifies further analysis.

$$\mathbf{r}_2 = [x_2, y_2, z_2] = [x_1 - C, y_1 - z_1\Sigma, y_1\Sigma + z_1] \tag{9}$$

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:

$$\Delta\mathbf{r}_2 = [0, -z_1\Sigma, y_1\Sigma] \tag{10}$$

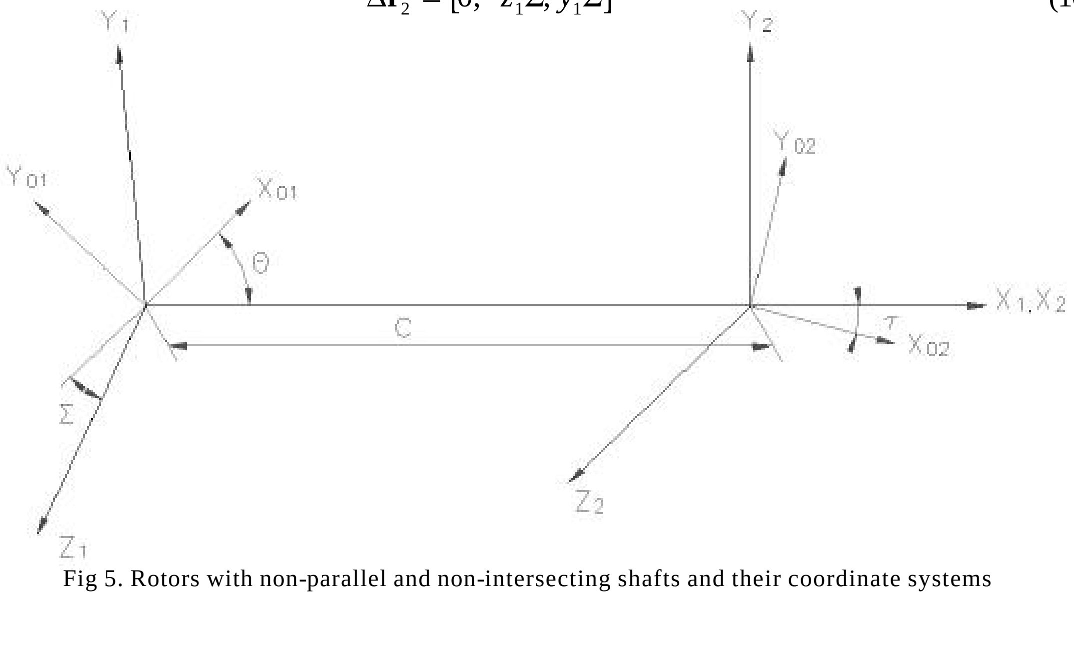

Fig 5. Rotors with non-parallel and non-intersecting shafts and their coordinate systems

Although, in the majority of practical cases, a displacement in the z direction is very small and therefore not so important 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 $z_1\Sigma$.

Such an approach substantially simplifies the analysis and allows the problem to be presented in two dimensions in the rotor end plane.

3. ROTOR POSITION IN THE ROTOR END PLANE

Although the displacements described in this paper are entirely three-dimensional, a two-dimensional presentation of them in the rotor end plane section can be used for analysis.

Equation (2) serves to calculate both the coordinates of the rotor meshing points x_2, y_2 on the rotor helicoids and x_{02}, y_{02} in the end plane from the given rotor coordinates points x_{01} and y_{01}. It may also be used to determine the contact line coordinates and paths of contact between the rotors. Since there exists a clearance gap between rotors, their contact line is a line consisting of points of the most proximate rotor position. A convenient practice to obtain a clearance gap between the rotors is to consider the gap as the shortest distance between the rotors in a section normal to the rotor helicoids.

A 4/6 rotor pair is chosen as an example to present the calculation of variation of rotor position caused by imperfections in the compressor manufacturing. The rotor centre distance is 75 mm and the outer rotor diameters are 99.00 and 93.36 mm for the main and gate rotors respectively. The rotor length is 153.5 mm and the unit lead is 25.24 mm/rad. The bearing span is 200 mm. The uniform clearance between the rotors is 100 mm.
For the purpose of this analysis, the rotors are fixed in their suction bearings and the whole bearing displacement is applied to one of the discharge bearings. The aggregate displacement of all the bearings is $\delta_x=\delta_y=100$ μm, which reduces the rotor centre distance by a maximum of 88 μm and imposes a rotation of 0.11^o on the gate rotor.

A clearance distribution between the rotors is scaled 50 times and both of the rotors with the corresponding clearance are presented in the same figure.

As can be seen in Fig. 6, contact is established on the lower part of the rotor lobe at the rotor end close to the suction bearings. The rotor contact belt is somewhat too long on the main rotor round side, indicating that rotor seizure might occur in this area. The clearance distribution close to the mid span of the rotors is presented in Fig 7. It can be seen that the clearance around the rotor lobe is almost uniform. In Fig 8, the clearance distribution is presented at the discharge end of the rotors. The reduced backlash is visible in this position.

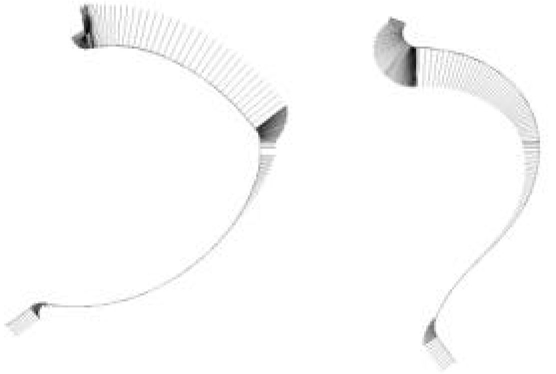

Fig 6. Clearance distribution between the rotors close to the rotor suction end

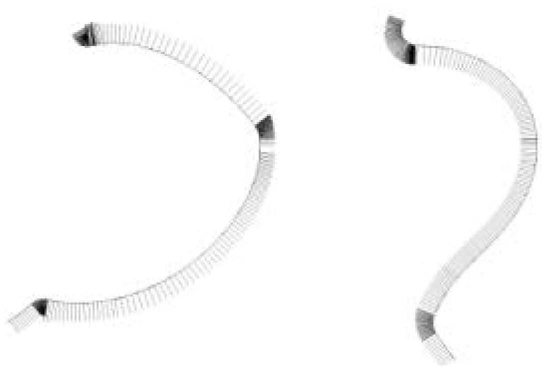

Fig 7. Clearance distribution near to mid of the rotors

If by any means, the rotors change their relative position, the clearance distribution at the discharge end of the rotors may be reduced to zero at the flat side of the rotor lobes, as presented in Fig 9. In such a case, rotor contact will be prohibitively long on the flat side of the profile indicating that rotor seizure will almost certainly occur in this area if the rotors come into contact with each other.

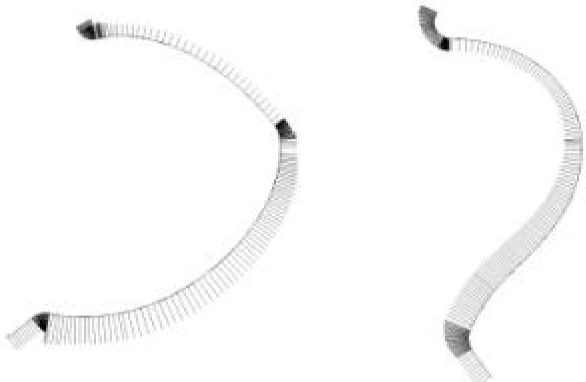

Fig 8. Clearance distribution between the rotors at the discharge rotor end

Fig 9. A possible clearance distribution at the discharge end of the rotor

This situation indicates that a non-uniform clearance distribution should be applied to avoid hard rotor contact at rotor areas where a sliding motion between the rotors is dominant.

4. DYNAMIC EFFECTS OF CHANGE OF ROTOR POSITION

Apart from the static effects of change of the rotor position, which are described in the previous section, some dynamic effects may be caused by the incorrect rotor position in its housing. Two examples are related to oil flooded compressors.

The first example, presented here, is the non-uniform movement of the gate rotor caused by the displacement of the rotor contact point in the z direction. Since such displacements x_1z and y_1S, presented by Eqs. (6) and (10), are variable, they cause variable relative rotation of the rotors, which will result in a non-uniform movement of the gate rotor regardless of whether or not the rotors are conjugate. This will inevitably cause oscillations of the gate rotor and eventual appearance of the rotor noise.

The second dynamic effect, which is mentioned here, is the intermittent contact of the main and gate rotors along the rotor length. It can be seen from Figs 6 and 11, that the necessary driving contact between the rotors is maintained only for a short length at the suction end of the rotors. By rotation of the rotors, this contact will soon terminate, leaving the gate rotor in a position to rotate freely until it again contacts the main rotor along the next rotor lobe. This will inevitably be connected with discontinuity of the gate rotor movement and its eventual impact with the main rotor, which may cause excessive noise and premature rotor wear.

5. CONCLUSIONS

Screw compressor rotors are manufactured today by formed grinding tool within very tight tolerances which permit small rotor clearances. These lead to more efficient screw compressors. However, small rotor clearances result in production imperfections in the screw compressor housings and bearings having a far greater significance.

A simple mathematical analysis, derived in the paper, was applied to quantify the static and dynamic effects of the change of rotor position in screw compressors caused by the manufacturing imperfections of the screw compressor housings and bearings which will certainly help compressor designers to choose appropriate tolerances for the compressor parts to meet new requirements introduced to the design by implementing the high precision compressor rotors.

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