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IMPROVEMENT OF THE RELIABILITY OF A SINGLE SCREW COMPRESSOR

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ABSTRACT

The correct design of the engagement pair (mainrotor and gaterotor) profile is the foundation to improve the operating reliability of a single screw compressor. An introduction to take a straight line enveloping pair as the engagement pair of the main machine is presented after analysing the present engagement pair. A complete analysis is also made of the hydrodynamic lubrication property and its sealing. In order to further improve the engagement pair property, the necessary modification for the profile and its method are stated. All these was proved by test or practical operation.

INTRODUCTION

The single screw compressor is widely used in such fields as power and chemical industries and refrigeration owing to its unique advantages.

There are several kinds of the single screw compressor, the one concerned with is the cylindrical/planar type single screw compressor as shown in Fig.1. It consists of one cylindrical mainrotor 1 and two symmetrically disposed planar gaterotors 2. The engagement pair of its screw and pinions has very large relative sliding velocity in its operation. If we don’t find out the proper profile of the engagement pair, the working surface of the gaterotor will wear so soon that the inter-teeth sealing will lose its efficacy, and the compressor capacity will decrease, thus being unable to ensure the operating reliability of the machine.

We suggest taking a straight line enveloping pair as the engagement pair of the compressor. This enveloping pair has a reliable inter-teeth sealing and a stable dynamic oil film. The property of hydrodynamic lubrication is further improved after the modification for the screw profile at the beginning of engagement, thus making the operating process steadier and improving effectively the machine reliability.

THE PRESENT ENGAGEMENT PAIR

The sealing and the hydrodynamic lubrication are two main factors which affect the operating reliability of a single screw compressor. If we select the profile of engagement pair properly, a steady dynamic oil film will be produced after a certain amount of oil is injected into the compression chamber, so that the mainrotor and gaterotor will be separated from direct contact, and the wear between the working surfaces of gaterotor will decrease effectively. The simplified model of the engagement pair is shown in Fig.2[1]. The lubrication theory shows that the larger the
values of parameters $\gamma$, $R$, $V$ and $L$ are, the better conditions of hydrodynamic lubrication are created, especially the great effect of angle $\gamma$ on the lubrication. In order to make angle $\gamma$ approach a right angle, the contact lines should be located as far as possible at the radial position of the engagement pair when we select the profile of the engagement pair.

The inter-teeth sealing of the engagement pair depends on the contact lines and oil film, however, under the conditions of reasonable constructional parameters and better lubrication, it relies mainly on the position of contact lines. As shown in Fig. 3, owing to the existence of blowholes (see section A-A) and the clearance between the top of screw and the root of gaterotor (see section B-B), the main leakage paths are formed, resulting in the leakage of the compressed gas from the high-pressure chamber to the lower one. Therefore the section area of blowholes must be reduced as small as possible when we select the profile.

There are two kinds of profiles for the engagement pair of a single screw compressor, i.e. the type of straight line and the type of cylindrical enveloping. The screw helicoid of the former is a tracing surface formed by a certain meshing motion of straight generating line $\overline{MM}$, as shown in Fig. 4. The straight generating line $\overline{MM}$ goes through (or a little below) the screw axis, and is tangent to the basic circle with a radius $r_0$. Therefore the gaterotor engaged with this screw is really a straight line gear using line $\overline{MM}$ as its profile, and the thickness of this gaterotor is infinitely thin. Evidently the contact line of this engagement pair is the straight generating line $\overline{MM}$, and is fixed at the gaterotor flank. This kind of contact line is located on (or a little below) the gaterotor-teeth proof, and the sealing is better, but its contact line is fixed and the condition of hydrodynamic lubrication is poor, so that the gaterotor will wear easily.

The engagement pair of the cylindrical enveloping type takes a cylindrical surface as its generating surface, which envelops the screw to form the helicoid according to a certain meshing motion, then takes this screw to engage with the gaterotor whose flank is a part of the cylindrical surface.

If we have the system of coordinates shown in Fig. 5, the equation of gaterotor-working flank is

\[
\begin{align*}
x_1 &= -u \\
y_1 &= r_0 \cos \phi \\
z_1 &= r_0 \sin \phi
\end{align*}
\]

(1)

and the helicoid of the screw is

\[
\begin{align*}
x &= -ucos\phi_x \cos \phi_z - r_0 \cos \phi_z \cos \phi_x \\
y &= ucossin\phi_x + r_0 \cos \phi_z \cos \phi_x \\
z &= -usint\phi_z + r_0 \cos \phi_z \cos \phi_x
\end{align*}
\]

(2)

Fig. 6 shows the instantaneous contact lines of this engagement pair. It is clear that the contact lines are nearly radial, some oil wedge is formed, the conditions of hydrodynamic lubrication are better, and the contact line is moved in a limited range so as to disperse the wearing positions.

In the case of the main rotor has six threads, the gaterotor has eleven teeth, the main rotor and gaterotor each have a diameter $D$, the distance between the rotation axes is equal to $0.8D$, the contact point M on the gaterotor has a radius $xD$, we can get the thread helix angle

\[
\tan \omega = x/1.833(0.8-x \cos \phi_z)
\]

(3)

when $x = 0.5, 0.4, 0.3$, the relative curves between $\omega$ and $\phi_z$ are shown in Fig. 7. Therefore the working range of the gaterotor-
teeth flank is between $18^\circ$ and $42^\circ$, and the proof of gaterotor will be so far apart from the contact lines that larger blowholes are formed (see Fig.8a).

In order to decrease the section area of the blowholes, the axis of gaterotor is inclined with respect to the rotation axis of screw so as to make angle $B$ (see Fig.8b), or the axes of the cylindrical surfaces at both flanks do not coincide with each other, or the cylindrical surface is replaced by the conical surface. Both milling and grinding can be used to machine these engagement pairs, but the construction of the compressor or the machining process is complicated.

THE STRAIGHT LINE ENVELOPING PAIR

In order to develop the advantages of inter-teeth sealing of the lineartype engagement pair and to overcome its disadvantages of the fixed contact line and poor hydrodynamic lubrication, we have used the screw machined by the straight generating line $\overline{MM}$ of a primitive gaterotor to cut the gaterotor with a certain thickness (i.e. The enveloping gaterotor), and then this new gaterotor was made to engage with the screw (see Fig.9). We call this new engagement pair the straight line enveloping pair. With the coordinate system as shown in Fig.4, the following equations of the straight generating line $\overline{MM}$ are obtained

\[
\begin{align*}
  x_1 &= u \\
  y_1 &= r_0 \\
  z_1 &= 0
\end{align*}
\]

The equations of the screw helicoid cut by the primitive gaterotor are

\[
\begin{align*}
  x_1 &= -T\cos \phi, \\
  y_1 &= T\sin \phi, \\
  z_1 &= -s
\end{align*}
\]

where $T = u\cos \phi + r_0 \sin \phi - A$, $s = u \sin \phi - r_0 \cos \phi$.

The equations of gaterotor-teeth flank enveloped by the screw are

\[
\begin{align*}
  x_1 &= -T\cos \phi \cos (\phi - \theta) - S \sin \theta - A \cos \theta, \\
  y_1 &= T\sin \phi \cos (\phi - \theta) - S \cos \theta + A \sin \theta, \\
  z_1 &= T \sin (\phi - \theta).
\end{align*}
\]

\[
\begin{align*}
  u &= \frac{1}{T} \cos \phi \left[ \cos (\phi - \theta) - 1 \right] \\
  \tan \theta &= \frac{u}{T \left( u - A \cos \phi \right)}
\end{align*}
\]

The engagement condition formula (7) has two solutions

\[
\begin{align*}
  \phi &= \theta, \\
  \phi &= \theta - 2 \tan^{-1} \left( \frac{u}{T \left( u - A \cos \phi \right)} \right)
\end{align*}
\]

The contact line solved by equations (6) and (8) is the same as equation (4). Clearly it is the straight generating line $\overline{MM}$, a fixed contact line, which is shown by $A'$, $B'$, $C'$, $D'$ in Fig. 10.

The contact line solved by equations (6) and (9) is moved continuously as $\theta$ is changed, it approaches the fixed contact line gradually during engagement until they coincide with each other. This is shown by $A$, $B$, $C$, $D$ in Fig.10.

As aforesaid, the performance of the hydrodynamic lubrication created by the engagement is judged by angle $\gamma$. It is more reasonable that the calculated average values of $\gamma$ range from $75^\circ$ to $89^\circ$ at every contact point 1-5 in contact lines $C$ and $D$.

We have made use of Matin formula to calculate dynamic oil film thickness $h_0$ at contact line $D$, and its result is shown in Fig.11. If the load $p$ on unit length of the contact line is

\[10 \text{ kgf/cm}, \ h_0 = 15.54 \mu \text{m}.\]

In order to prove the existence of dynamic oil film, the dynamic oil film was tested repeatedly by the resistance method[2]. It indicates that the dynamic oil film is existent steadily in the normal operating condition, and only when the rotation speeds of screw are lower, such as starting or stopping, the direct contact of two frictional surfaces can take place (Fig.12)[3].
MODIFICATION FOR THE PROFILE

The theory of hydrodynamic lubrication shows that the key to produce the oil film pressure depends on the oil wedge with a certain ratio of clearance between two frictional surfaces. Clearance ratio $a$ is the ratio of inlet clearance $h_1$ to outlet clearance $h_2$ of the oil wedge. It is reasonable that $a$ is usually equal to 2.2 [4].

In the single screw compressor, if engagement clearance (one side) $h_2 = 0.05-0.10 \text{mm}$, then inlet clearance $h_1 = 0.11-0.22 \text{mm}$. In engaging process of straight line enveloping pair at the moment when gaterotor teeth successively engage with the screw, the first generating line on the helicoid contacts with the gaterotor flank, at this time $h_f = 0$. It is unfavourable to build dynamic oil film.

Modification for the screw-helicoid profile is to cut a slope at the inlet of the screw. Accordingly, a good hydrodynamic lubrication condition is founded at the beginning of engagement, and the engaging stability caused by the production and assembly errors of engagement pair is increased. In fact, after running-in the wearing trace can be clearly observed at the inlet of the original screw. Hence it is necessary to cut a slope at the screw inlet.

The easiest method for modifying the profile is to use the method of axial and radial displacement. If necessary, it can be filed out by a fitter.

As shown in Fig.13, if rotation center $O_2$ of tool (i.e. straight generating line $MM$) moves upward and the center distance decreases by $\Delta A$, then the tool edge cuts the thread groove. The depth of cut increases gradually from the thread of screw to the beginning of engagement. If the depth of cut at the beginning of slope is $A'$ and the finishing end is $A''$, the absolute value of slope $A = A' - A''$. According to the geometrical relationship, we obtain

$$A' = \Delta A \sin \theta'$$
$$A'' = \Delta A \sin \theta''$$

Where $\theta'$ and $\theta''$ depend on the beginning-finishing position of the slope. To ensure the inter-teeth sealing of the engagement pair, the beginning of slope must not go beyond the position where the screw groove begins to close. Thus

$$\Delta A = \frac{A}{\sin \theta'' - \sin (\theta' + \theta')}$$

(10)

In order to make the beginning position of slope $A' = 0$ and to ensure the absolute value $A$, the screw must be displaced axially in direction $I$

$$\Delta B = \Delta A \tan \theta'$$

(11)

As $A=0.11-0.22 \text{ mm}$, $\Delta A=0.39-0.78 \text{ mm}$, $\Delta B=0.19-0.38 \text{ mm}$. After the screw profile is modified, the operating state is well and the sealing property is further improved. The volumetric efficiency tested in practical operation is 0.9.

CONCLUSION

The results of theoretical analyses, calculations and tests show that the use of a straight line enveloping pair as the engagement pair of a single screw compressor will obtain good inter-teeth sealing and hydrodynamic lubrication and the operating reliability will be further improved.

It is necessary to modify the profile at beginning of screw engagement. When modifying the profile, the axial and radial displacement of tool and screw must be strictly controlled to ensure the position at which the thread groove starts to close.

In batch production condition, it is better to finish machining the gaterotor flank
with a special hob to raise the accuracy and finish of the flank.

NOTATION

A  Center distance of engagement pair
A  Absolute value of slope
ΔA  Displacement of center distance
ΔB  Axial displacement
D  Diameter of engagement pair
i  Drive ratio
L  Total length of contact line
R  Relative radius of curvature
r  Radius of basic circle
t  Angle parameter of gaterotor flank
u  Position parameter of gaterotor flank
w  Helix angle
α'  Engagement angle of inlet
α"  Closed angle of thread groove
γ  Angle between the relative velocity and the tangent of contact line
ϕ  Rotation angle of engagement pair
θ  Rotation angle when the screw envelops the gaterotor
θ  Angle (see Fig.13)

Subscripts
1  Mainrotor (screw)
2  Gaterotor

REFERENCES


[2] Masao Ozu, Tsugio Itami, Some Electrical Observations of Metallic Contact Between Lubricated Surfaces Under Dynamic Conditions of Rotary Compres-


Fig. 1 Sketch of single screw compressor

Fig. 2 Simplified model of engagement pair

Fig. 3 Leakage paths

Fig. 4 Straight line engagement pair

Fig. 5 Cylindrical generating surface gaterotor
Fig. 6 Contact line of cylindrical enveloping pair

Fig. 7 Relation of \( w = f(\phi_x) \)

Fig. 8 Cylindrical enveloping pair

Fig. 9 Straight line enveloping pair

Fig. 10 Contact line of straight line enveloping pair
Fig. 11 Relation of $h_o = f(p)$

Fig. 12 Text oil film

Fig. 13 Cut slope