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A STUDY OF THE SINGLE SCREW COMPRESSOR PROFILE

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ABSTRACT

Stable hydrodynamic lubrication film is needed between the tooth profiles for smooth running and long life of a single-screw compressor. This lubrication film is easily established with "straight line enveloping screw pair" rotors, but this kind of engagement pair is very difficult to machine. In this paper it is suggested that " a conical surface two-envelope engagement pair " be used as the single-screw compressor profile. The mathematical model, the working-surface characters, the structural characteristics and the machining method of this kind of engagement pair are elaborated. It has beem proven by practice and experiment that this kind of engagement pair can be machined conveniently either in small or large quantities.

SYMBOLS

u axial position variable of conical surface, cm

θ angle variable of conical surface, rad

$a, d_2, d_1 a', E, B, H$ const, cm

α, β const, rad

$\varphi_1, \varphi_2, \theta_1, \theta_2$ rotation angles, rad

i_{12} transmission ratio

θ_v slide angle, rad

σ^I, σ^{II} relatively-sliding factor

h_o Minimum dynamic oil film thickness, μm

p load on unit length, N/cm

INTRODUCTION

The engagement pair of a main rotor and a gate rotor is the key part of a single-screw compressor. The profile study of the engagement pair is a subject which is always paid close attention to. If we use a correct profile, stable hydrodynamic lubrication between the frictional surfaces of an engagement pair can be obtained while the compressor operates at a normal rotation speed, thus further improving the life and reliability of the compressor.

At present there are "straight line (the tooth profile of gate rotor) -- straight line enveloping surface (the tooth profile of main rotor) engagement pair "and" conical surface (the tooth profile of gate rotor) - conical enveloping surface (the tooth profile of main rotor) engagement pair " which are used as the teeth profiles of the single-screw compressor. In the paper[1], "straight line enveloping screw pair" is also used as the profiles. The stable hydrodynamic lubrication film is easily established with "straight line enveloping screw pair" rotors, but this kind of engagement pair is very difficult to machine. In this paper, "conical surface twice-envelop engagement pair" is used as the profiles. Its mathematical model and machining method are presented, and its tooth-surface characters and the minimum dynamic oil film thickness between two frictional surfaces are calculated. At last the oil film test results are indicated so as to prove that the theoretical analyses are correct indeed.

MATHEMATICAL MODEL

In Fig. 1 the system of coordinates $S_1(O_1-x_1y_1z_1)$ is fixed with a main rotor, and the system $S_2(O_2-x_2y_2z_2)$ is fixed with a gate rotor. S(O-xyz) and $S_p(O_p-x_py_pz_p)$ are two fixed systems of coordinates. Axis z coincides with the axis of the gate rotor, and axis zp, being at right angles to axis z , coincides with that of the main rotor.

Fig. 2 shows a conical surface in the system S_2 $(O_2-x_2y_2z_2$). The equation of main rotor tooth surface which is formed by the envelop of the conical surface with certain engaging movement can be shown as follows:

$$\left.\begin{aligned}
r_1 &= \left\{ [\, \vec{r}_2(\vec{k}, \varphi_2)\,]\ (\vec{i}_p, +\frac{\pi}{2}) + \overrightarrow{O_p\ O} \right\}\ (\vec{k}_1, -\varphi_1) \\
r_2 &= \Big[-u\cos\beta - \Big(\frac{d_2}{2} - u\,\mathrm{tg}\,\alpha\Big)\cos\theta\sin\beta - \\
&\qquad E\cos\beta \Big]\,\vec{i}_2 + \Big[-u\sin\beta + \Big(\frac{d_2}{2} - u\,\mathrm{tg}\,\alpha\Big) \\
&\qquad \cos\theta\cos\beta - E\sin\beta + B\Big]\,\vec{j}_2 + \Big[\Big(\frac{d_2}{2} - \\
&\qquad u\,\mathrm{tg}\,\alpha\Big)\sin\theta + a'\Big]\,\vec{k}_2 \qquad (0 \leq u \leq H, \\
&\qquad 0 \leq \theta < 2\pi) \\
\theta &= \arccos\left(\frac{A_3(\varphi_2)}{A_1^2(u,\varphi_2) + A_2^2(u,\varphi_2)}\right) - \mathrm{arc\,tg} \\
&\qquad \left(\frac{A_2(u,\varphi_2)}{A_1(u,\varphi_2)}\right) \\
A_1(u,\varphi_2) &= -C + B\sin\beta + a'\ i_{12}\sin(\varphi_2 + \beta) \\
A_2(u,\varphi_2) &= i_{12}\,C\cos(\varphi_2 + \beta) - i_{12}\,a + B\,i_{12}\sin\varphi_2 \\
A_3(\varphi_2) &= -i_{12}\,a'\,\mathrm{tg}\,\alpha\cos(\varphi_2 + \beta) - B\,\mathrm{tg}\,\alpha\cos\beta \\
C &= u + E - \Big(\frac{d_2}{2} - u\,\mathrm{tg}\,\alpha\Big)\mathrm{tg}\,\alpha
\end{aligned}\right\} \quad (1)$$

The equation of a gate rotor tooth surface which is formed by the envelop of the above-mentioned main rotor tooth surface, that is, by twice envelops of a conical surface, can be divided into two parts :

$$\left.\begin{aligned}
r_2^* &= \left\{ [\, \vec{r}_1(\vec{k}_p, \varphi_1)\,]\ (\vec{i}, -\frac{\pi}{2}) + \overrightarrow{OO_p} \right\}\ (\vec{k}_2, -\varphi_2) \\
\varphi_1 &= \theta_1 - 2\,\mathrm{arc\,tg}\left(\frac{C_2(u,\theta,\varphi_2)}{C_1(u,\theta,\varphi_2)}\right)
\end{aligned}\right.$$

$$
\left.\begin{aligned}
&C_1(u,\theta,\varphi_2)=-C\cos\theta+aC'+B\cos\theta\sin\beta+B\,tg\alpha\cos\beta\\
&C_2(u,\theta,\varphi_2)=C\sin\theta\sin(\varphi_2+\beta)+a'C'-B\sin\theta\cos\varphi_2\\
&C=u+E-\left(\frac{d_2}{2}-u\,tg\alpha\right)tg\alpha\\
&C'=\cos\theta\cos(\varphi_2+\beta)-tg\alpha\sin(\varphi_2+\beta)
\end{aligned}\right\}\quad(2)
$$

and

$$
\left.\begin{aligned}
r_2=&\left[-u\cos\beta-\left(\frac{d_2}{2}-u\,tg\alpha\right)\cos\theta\sin\beta-E\cos\beta\right]\vec{i}_2\\
&+\left[-u\sin\beta+\left(\frac{d_2}{2}-u\,tg\alpha\right)\cos\theta\cos\beta-E\sin\beta\right.\\
&\left.+B\right]\vec{j}_2+\left[\left(\frac{d_2}{2}-u\,tg\alpha\right)\sin\theta+a'\right]\vec{k}_2\\
\theta=&\arccos\left(\frac{A_3(\varphi_2)}{\sqrt{A_1^2(u,\varphi_2)+A_2^2(u,\varphi_2)}}\right)-\mathrm{arc\,tg}\left(\frac{A_2(u,\varphi_2)}{A_1(u,\varphi_2)}\right)
\end{aligned}\right\}\quad(3)
$$

Equations (2) and (3) correspond to two groups of instantaneous contact lines between two engagement teeth surfaces respectively. Their demarcating line is the engagement limit line.

In equations (1), (2) and (3), φ_1 and φ_2 are the rotation angles of the main rotor and the gate rotor respectively for the first envelop, and θ_1 and θ_2 for the second. $\varphi_1 / \varphi_2 = \theta_1 / \theta_2 = i_{12} =$ const. If $\alpha = 0$ in the above - mentioned equations, we can get the mathematical model of "cylindrical surface twice-envelop engagement pair " which is a special case of " conical surface twice-envelop engagement pair".

ENGAGEMENT - TOOTH - SURFACE CHARACTERS

1 The Instantaneous Contact Lines

As mentioned above there are two groups of instantaneous contact lines between two engagement teeth surfaces of "conical surface twice-envelop engagement pair". Fig. 3 shows these two groups of instantaneous contact lines on a tooth surface of the gate rotor. k-k' is the engagement limit line which divides the tooth surface into two parts, that is, Σ_2^{II} and Σ_1^{II} . The instantaneous lines L_1^{II} corresponds to equations (1) and (3), and L_2^{II}

corresponds to equations (1) and (2). In the engagement process of the gate rotor and main rotor, L_1^{II} always exists and seals a compressed gas, but L_2^{II} only exists within the certain limits of the rotation angle of the gate rotor. These two groups of instantaneous contact lines can result in the pressure effect for the lubrication oil film, as shown in Fig. 4, thus strengthening the bearing capacity and stability of the oil film.

2 Angle θ_v

Angle θ_v at a contact point is the directional angle from the tangent direction of the contact line to the direction of a relative velocity, It is better to let angle θ_v equal $\pi/_2$ (1.5708 rad) or approach $\pi/_2$ (1.5708 rad) in order to be advantageous to forming the thick dynamic oil film between two frictional surfaces of the engagement pair and improving the lubrication conditions.

3 Relatively - Sliding Factor σ^I and σ^{II}

The relatively - sliding factor between two engagement teeth surfaces is the limit value of the ratio of relatively-sliding arc length to the whole arc length sliding on one of the two engagement teeth surfaces. σ^{II} is the relatively - sliding factor of the gate rotor tooth surface at a contact point, and σ^I is that of the main rotor at the corresponding contact point.

The wearability of the engagement pair made up of certain material can be measured by the absolute value of the relatively-sliding factor. The larger the absolute value of the relatively-sliding factor, the poorer the wearability of the teeth surface, that is, the weaker the wearability of the engagement pair. The calculating results of "conical surface twice-envelop engagement pair" are shown as follows:

$1 < |\sigma^{II}| < 2.6$; $0.5 < |\sigma^I| < 1$ (Corresponding to equations (1) and (2))

MACHINING METHOD

According to the principle of the formation of "conical surface twice-envelop engagement pair" we have designed and produced a machine tool for its finish machining. The tool can grind the teeth surfaces of the main rotor and the gate rotor cutting hobbing which has been heat-treated, and can hob the teeth surfaces of the gate rotor also. The value of the engagement clearance between two engagement teeth surfaces can be controlled exactly.

OIL FILM CALCULATION AND TEST RESULTS

The stable hydrodynamic lubrication between two frictional surfaces of the engagement pair obtained under the normal operation of the compressor is the guarantee of long life of the compressor. The rotation of the gate rotor needs only to overcome the friction resistance of axle bearings, therefore the engagement process of the gate rotor driven by the main rotor is at high speed and light load, and the minimum dynamic oil film thickness h_o between two engagement teeth surfaces can be calculated by using Matin formula. The calculating results have shown when angle φ_2 is 0.025 rad, the oil film thickness is minimum, h_o equals 30.245 μm [2] . It is obvious that the dynamic oil film thickness is perfect.

The resistance method[3] has been used to prove the existence and stability of the dynamic oil film.The repeatedly-testing results have shown that the dynamic oil film exists steadily at the normal operating speed of 3000 r.p.m, and only when the rotation speeds of the main rotor are lower during starting or stopping the compressor, the direct contact of two engagement teeth surfaces can take place.

CONCLUSIONS

1) As shown in the calculating results, the engagement teeth surfaces of "conical surface twice-envelop engagement pair" have good lubrication conditions and wearability. The minimum dynamic oil film thickness calculated according to Matin formula is thick.

2) The oil film test has shown that the theoretical analyses and calculation are correct, and the stable hydrodynamic lubrication between two friction surfaces of the engagement pair can be obtained when the compressor operates at the normal rotation speed.

3) The engagement pair can be machined conveniently and mass-produced. Its engagement precision is high and its surface finish is smooth.

4) The engagement pair has successfully used in a refrigeration compressor which capacity is

30,000 kcal/hour.

5) "Conical surface twice-envelop engagement pair" can be used as the engagement pair of a single-screw machine for expansion of a gas, and the above-mentioned conclusions are still correct.

REFERENCES

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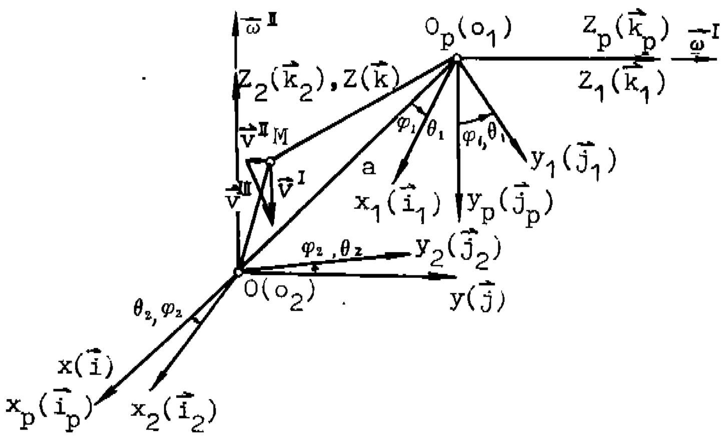

Fig.1 The systems of coordinates

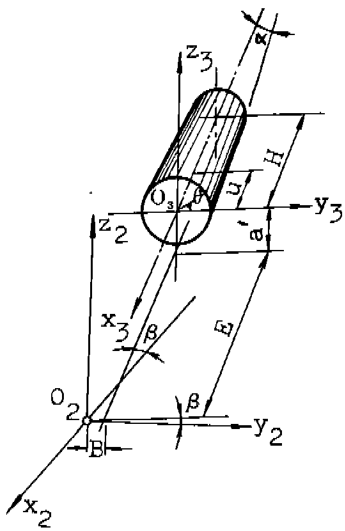

Fig.2 A conical surface in the system $S_2(O_2-x_2y_2z_2)$

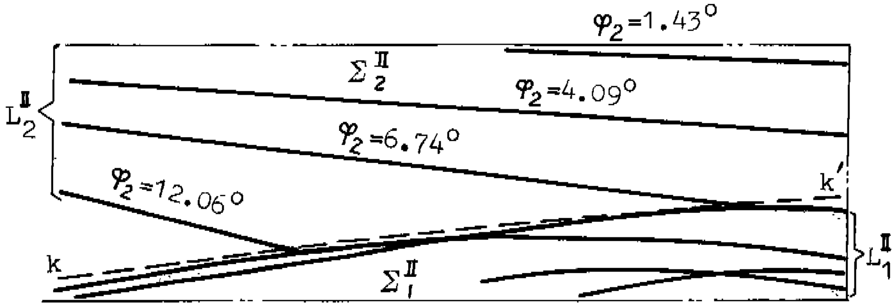

Fig.3 Instantaneous contact lines on a tooth surface of the gate rotor(the thickness of gate rotor is 0.6 cm)

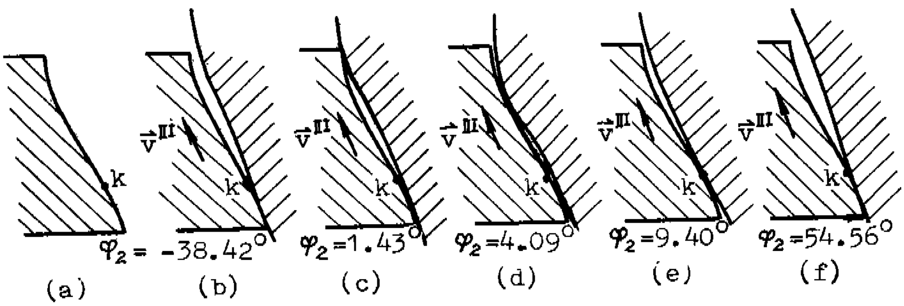

Fig.4 The pressure effect for the lubrication oil film(x_2=-4.25cm)

USE THE METHOD OF MAGNETIC SCORE TO MEASURE THE DYNAMIC PARAMETERS OF THE SCREW COMPRESSOR

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ABSTRACT

The dynamic measurement in connection with the working process of the screw compressor and the inspection of the whole rotor profile and engaged clearance have a great effect on improving the volumetric efficiency and saving energy, however this dynamic measurement was not a measurement under the assembling condition. The method of magnetic score can measure the dynamic parameters of the screw compressor in the assembling condition. The results measured by the method of magnetic score can reflect the performance of the whole set truly. The paper presents the principle of the method of magnatic score and the measuring process. The compensation for measuring error and the selection of parameter are discussed.

SYMBOLS

λ sine magnetic wave length

Z tooth number

N railing stripe number

$\bar{\omega}$ rotational speed

ω angular speed

V linear speed

f frequence

R rotational radium

l railing stripe distance