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ABSTRACT

The performance of scroll compressor depend a great extent on the conditions of wear and energy consumption of the plain bearing. In this paper, the frictional behaviour of the plain bearing was analyzed, the operating conditions of the plain bearing were discussed. In addition, author presented parameter selection and some material of the plain bearing.

INTRODUCTION

Plain bearings are often employed as the main bearing of scroll compressors and shaft bearing of the orbiting scroll. (Fig. 1) These bearings support or transmit loads, slide relative to the main axial or the crankpin, and operate over a long period of time under the conditions of great oil-film pressure, high oil-film temperature and corrosion of the organic acid formed by the high-temperature oxidation of the lubricating oil. The reliability and service life as well as some other important economic and technical indexes of scroll compressors all depend a great extent on the conditions of wear and energy consumption of the plain bearing. [1]

To reduce friction and wear, we have tested on some geometric and operating parameters. As the plain bearing of scroll compressors is a typical hydrodynamic lubrication system, its working condition is not only related to the properties of the materal, but also depends on the lubrication of the system. Furthermore, the structure, machining and assembly precision for bearing, and the quality of the lubricating oil are all closely connected with lubrication, and together they affects on the reliability and durability of the bearing system. Therefore, both lubrication and the material should be taken into consideration in order to guarantee the long-term reliable operation of the plain bearing of scroll compressors.

ANALYSIS OF THE FRICTIONAL BEHAVIOUR OF THE PLAIN BEARING

Most of the scroll compressors that are used today for air-conditioning are hermetic. The medium cooling capacity scroll compressor in our test has the pressure of the plain bearing of about 8 MPa, linear velocity of the journal of 5.4 M/S, surface temperature of the bearing of 110 ^{o}C. In such a demanding condition, the plain bearing may fail due to wear, fatigue, corrosion and fluid erosion if the operating parameters go beyond certain limits. The most important form of failure is wear, which is closely connected with other forms of failure, and with the performance and economy of the bearing as well.

As we know, wear is caused by friction, and the most effective way of re-

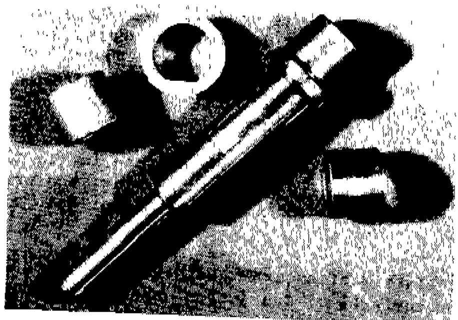

Fig. 1 Bearing and crank of scroll compressor

ducing friction wear is lubrication. According to the lubricant storage between the friction surfaces, friction of the plain bearing can be categorized into such four conditions as dry friction, boundary friction, mixed friction and fluid friction [2]. Fig. 2 shows the frictional behaviour curve drawn from the test results.

In Fig. 2, f is the friction coefficient, η is the dynamic viscosity of the lubricating oil, MPa.s; n is the rotating velocity of the journal, rad/s; p is the pressure of the bearing, N/mm^2, $\eta n/p$ is the characteristic number of the bearing, dimensionless parameter.

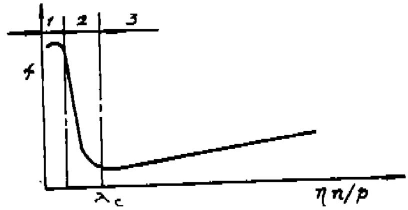

Fig. 2 Frictional behaviour curve of the plain bearing

1. Boundary friction area
2. mixed friction area
3. fluid friction area

It can be seen from Fig. 2 that when $\eta n/p$ is greater than λ_c, the oil film is thick enough to separate the two metal surfaces of the journal and the bearing completely, and friction occurs inside the fluid. Although the friction coefficient f increases with $\eta n/p$, the values of f are very small ($f < 0.005 \sim 0.01$) when $\eta n/p$ is smaller than λ_c, it enters the mixed friction area first, and the oil film gets thinner so as not to separate the two metal surfaces. Friction depends on both the property of the lubricating oil and the roughness peak of the contact surfaces. The friction coefficient increases apparently ($0.005 < f < 0.1$). When $\eta n/p$ continues to decrease, only a very thin oil film is absorbed on the metal surfaces, the contact ratio of the roughness peaks of the two surfaces increases rapidly and the friction coefficient increases greatly as well. ($0.1 < f < 0.3$). This time it enters the boundary friction area. If there is no lubricating oil between the two metal surfaces, it is in the dry friction area, and under the action of the load, pick-up will occur between the contact peaks of the surface, causing serious friction and wear.

To guarantee the long-term steady operation of the scroll compressor, the plain bearing must not work in the dry friction condition. It has been proved in practice that even slight carelessness with the friction condition may result in the pick-up and even "seizure" between the surfaces of the journal and the bearing, forcing the compressor to shut down.

Therefore, fluid lubrication or friction is the aim of the wear-free design of plain bearings. Efforts should be made to secure a fluid friction condition for the plain bearing, and thus steady operation state free from surface wear [2]

OPERATING CONDITION AND PARAMETER SELECTION OF THE PLAIN BEARING

The load of the plain bearing in the scroll compressor remains almost the same magnitude, and keeps its direction on the centre line between the journal and the bearing. Frozen machine oil is used as the lubricant; therefore, the bearing can be regarded as a hydrodynamic lubrication journal bearing carrying a static load. Acted on by a load, one side of the static journal will stick to the surface of the bearing. When the rotating velocity of the journal reaches a certain value, the dynamic pressure produced by the wedge effect can bring in enough lubricating oil to part the contact metal surfaces and make the pressure inside the oil layer carry the load on the journal. This shows that the oil film has alrady formed, and the bearing can operate in the liquid friction condition. When the rotating velocity keeps increasing, the pressure in the oil layer increases as well and the centre of the journal gets closer to the center of the bearing.

There must be a oil film of enough thickness between the two surfaces in order to separate the surfaces of the journal and the bearing. The thickness of the oil film is determined by the geometric parameters, load, viscosity of the lubricating oil and the oil feed; therefore, the oil film of enough thickness can only be built by comprehensive measures.

The operating condition of the hydrodynamic lubrication journal bearing is shown in Fig. 3.

In Fig. 3, R and r are the ratio of the bearing bore and the journal respectively, m; F is the load on the bearing, N; ω is the rotating velocity of the journal, rad/s; and p is the pressure field of the oil-film under the dynamic pressure.

When the resultant of forces of the pressure field is in equilibrium with the load, the centre of the journal is at the static equilibrium position O, a eccentricity of e from the centre of the bearing O_b. φ is the angle between the connecting line $O_b O$ and the acting line of the external load; h_{min} is the minimum oil-film thickness, m.

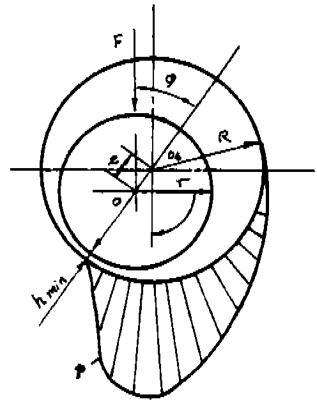

Fig. 3 The operating condition of the plain journal bearing

Let the bearing radial clearance $C = R-r$, m;

the bearing relative clearance $\psi = c/r$;

and the eccentricity ratio between the journal and the bearing bore $\varepsilon = e/c$,

then the minimum oil-film thickness

$$h_{min} = c - e = c(1-\varepsilon) = \psi r(1-\varepsilon) \quad (1)$$

From (1) we know that we can get h_{min} if we know .

In liquid friction theory, when the lubricating oil film forms continuously, we can get the expression of the load-carrying capacity of the plain bearing by solving the Reynolds equation of the limited length plain bearing,

$$F = \frac{\eta\omega}{\psi^2} BDS_0 \quad (2)$$

where B is the operating length of the bearing, m;

D is the nominal diameter of the bearing bore, m;

S_0 is Sommerfeld Number, a dimensionless operation characteristic number to express the load-carrying capacity, and the other symbols have the same meaning and dimension as before.

S_0 can be expressed as

$$S_0 = p\psi^2/\eta\omega \quad (3)$$

where p is the pressure of the bearing, MPa, and the other symbols have the same meaning and dimension as before.

S_0 is a function of the length-radius ratio B/D and the included angle β of the bearing , the latter of which is determined by the axial section of the bearing, and is usually 360° or 180°. The load-carrying oil film can only form in the scope of the included angle. It should be emphasized that once the hydrodynamic lubrication oil film is formed, the greater S_0, the greater the load-carrying capacity of the oil film. All the bearings with the same geometric parameters will operate at the same eccentricity if only the S_0 numbers are the same, regardless of the rotating velocity of their journals, load, clearance, and the viscosity of the lubricating oil. [3]

Calculation of S_0 is very complex. For given values of β and B/D, the calculation results of S_0 and can be made into a chart and table for application in design. Given the geometric parameters, and parameters of operating condition such as load and rotating velocity, we can get the eccentricity ratio of the bearing in steady operation state by these charts, and then calculate the minimum oil-film thickness by (1). For the plain bearing we have used in the compressor, $\beta = 360°$, $B/D = 0.7$, $F = 5000$ N, $n = 2850$ rpm, the environment temperature $t = 90$ °C, and it is lubricated by 25# frozen oil. Accordingly the minimum oil-film thickness $h_{min} = 2.1\ \mu m$.

It should be pointed out that the calculated h_{min} is the theoretical value under the ideal condition of smooth surface and parallel bus lines on the surfaces of the journal and the bearing. In practice, both surfaces have roughness and bad circularity. In addition, since assembly error or deformation may lead to the relative inclination of the journal and the bearing, the theoretical minimum oil-film thickness must be greater than or equal to the minimum allowable film thickness, i.e.,

$$h_{min} \geqslant [h_{min}]$$

$[h_{min}]$ can be estimated by the following equation,

$$[h_{min}] = (1.5 \sim 2)\ (RZ_1 + RZ_2) \tag{4}$$

where RZ_1 and RZ_2 are the average heights of the surface unevenness of the journal and the bearing respectively.

The load-carrying capacity of the hydrodynamic lubrication bearing is mainly confined by h_{min} and the service temperature of the lubricating oil. Given h_{min}, the load-carrying capacity of the bearing will increase with the linear velocity of the journal surface v. However, when the velocity is too high, the heat produced by friction will greatly reduce the viscosity of the lubricating oil, which will limit or reduce the load-carrying capacity. The maximum velocity limit is determined by the oil-film oscillation of the bearing.The operation limit curve of the plain bearing is shown in Fig. 4.

The performance of the hydrodynamic lubrication bearing is connected with such parameters as the bearing pressure, rotating velocity of the journal, viscosity of the lubricating oil and the bearing structure. The rotating velocity of the journal is determined by the general design of the machine and cannot be changed, but all the other parameters can be changed whenever necessary.

A greater bearing pressure p can reduce the dimensions of the bearing and help it operate in steady state. However if the pressure is too high, the oil film will get thinner and the performance of the lubricating oil and the machining, and assembly precision of the parts must be improved, it may even result in the failure to make liquid lubrication. When the pressure is too low, however, the dimensions of the bearing must be increased, and oil-film oscillation may occur due to the very small eccentricity ratio at great velocity, making the bearing lose steadiness.

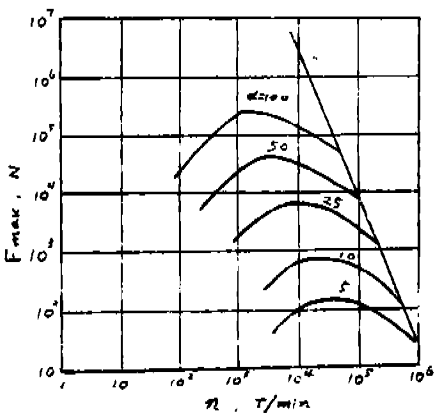

Fig. 4 Operation limit curve of the plain bearing

The smaller the value of B/D, the greater the end flux, the smaller the friction power loss and the bearing temperature rise, but the smaller the load-carrying capacity. If B/D increases, the load-carrying capacity can be improved, but the power loss and the temperature rise will also increase.

Furthermore, the inclination of the journal may result in the edge contact. Therefore, B/D of the plain bearing in scroll compressors generally ranges from 0.1 to 1.

The relative clearance ψ has a significant effect on the load-carrying capacity, rotating precision and the temperature rise. A smaller ψ may improve the load-carrying capacity, but the power loss and temperature rise will increase, and the lubricating oil feed will decrease. Too small a ψ value may even result in the failure to build the lubricating oil film. Generally, can be selected by the following empirical formula [4]

$$\psi = (0.6 \sim 1.0)\sqrt[4]{v} \times 10^{-3} \tag{5}$$

where v is the journal surface linear velocity, m/s.

SELECTION OF THE BEARING MATERIAL

As said above, liquid friction for the plain bearing is only an ideal condition. In practice, the minimum oil film thickness of the bearing is only 1 ~ 3 μm, and such a thin oil film cannot overcome the effects produced by shutdown, starting, slow-running, machining and assembly error of the related parts, and the roughness of the working surface. Local metal surface contact caused wear and scoring is thus inevitable. When the specific power of compressor is increased, the performance characteristic number S_0 of the plain bearing will decrease. According to our testing results, the plain bearing frequently operates in the mixed or boundary friction condition. To guarantee the lasting and reliable operation of the bearing under these two conditions, the bearing material must possess the following properties: [5]

1. Good mechanical properties at high temperature —— to be pressure - resistant and shock-resistant, and have enough fatigue strength.

2. Good surface properties —— small friction coefficient of the surface and not easy to wear.

3. Good compatibility with the journal material —— adhesive-wear-resistant and scuffing-resistant when the journal runs at a low velocity or the oil film is very thin, thus avoiding journal scoring or "seizure".

4. Good conformability —— able to adapt to the inclination of the bearing and other geometric errors.

5. Good embedability —— able to accept and hold in hard foreign grains and prevent scoring the journal.

6. Good lubrication ability —— great affinity to the lubricant and able to form uniformly-absorbed oil film on the surface of the material.

7. Good wear-in behaviour —— The surface roughness of the bearing material is easy to decrease in operation, which ensures the fit of the journal and bearing surfaces.

8. Anti-corrosion —— able to resist the corrosion of the organic acid produced by the aging and oxidation of the lubricating oil.

No existing bearing material can satisfy all the above property requirements, and selection must be made by taking into consideration the specific conditions. The multi-layer composite material, whose overall performance is more successful, represents one of the trends in the development of bearing material The Tinned bronze is an extensively applied bearing material, but we have found out that the bearing of this material is liable to adhesive wear and scoring due to the poor conformability and embedability of the material. When the leaded bronze is in the boundary friction condition, the lead in free state is easy to be coated on the bearing surface to form the anti-friction film. Besides, its shock-resistant and fatigue-resistant performance is superior to that of the bearing alloy. Therefore leaded bronze is suitable for the plain bearing in scroll compressors. In our test, the leaded bronze with the lead content of 25% was used, and after 2400 hours of operation, no adhesive wear occurred, and the wear of the frictional surfaces is 8 μm. To further improve the anti-friction and wear-resistant performance of the bearing, DU and DX bearings made of composite materials may be adopted.

CONCLUSION

1. Plain bearing is one of the key parts of the scroll compressor. The wear and energy consumption condition holds great influence on the reliability and service life of the compressor.

2. Wear is the main form of failure of the plain bearing and lubrication is the most effective way to reduce friction and prevent wear.

3. The lubricating oil film must be built between the bearing and the journal in order to guarantee the liquid friction condition for the bearing. The minimum oil-film thickness must be greater than or equal to the minimum allowable thickness. $h_{min} \geqslant [h_{min}]$

4. The minimum oil-film thickness is usually very small, and the bearing often operate in the mixed or boundary friction condition in practice. Therefore, the bearing material must possess good mechanical properties and anti-friction and wear-resistant properties.

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