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RESEARCH ON THE LUBRICATION CONDITIONS OF THE PACKING/ROD SYSTEM IN COMPRESSORS

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A simulating test for packing-rod systems in reciprocating compressors is conducted to measure the pressure distribution in the packing, the frictional resistance, the temperature on the sliding surface of sealing elements and the gas leakage through the packing. Mathematical simulations for this system is also achieved. The results show that in order to maintain an excellent lubricating condition the sealing pressure drop in one set of sealing elements should not exceed a certain magnitude, which can be determined according to the rod speed, the temperature of oil film and the surface profile of the sealing element, etc.

INTRODUCTION

In reciprocating compressors the cylinder packing is liable to lose their sealing function because of rapid wear. For a compressor lubricated with oil this means that the oil film between the piston rod and sealing elements often splits.

In comparison with piston rings the sealing elements in a packing often bear much larger sealing pressure drop and work with poorer cooling. On the other hand, the gas leakage through the packing elements will cause a loss of a valuable gas and environmental pollutions. Therefore to achieve a better comprehension for the sealing mechanism and lubricating conditions of packing/rod systems is very important.

There are two aspects in the research of the packing/rod systems. One is the investigation of the gas leakage and influences of related factors. The other is the analysis of the lubricating conditions in the packings. In recent years few papers concerning the sealing and lubrication of compressor packings have been found. However, the published papers dealing with that of piston rings is quite common. In principle, the function of sealing elements in packing is similar to that of piston rings, though the rings slide against the liner surface with their outer surface, and the sealing elements slide with their inner surface against the piston rod. Under certain assumptions the effective gas pressure governing the loading of the rings can be calculated according to the sealing condition of the rings. For a packing the gap clearances of tangential sealing rings are concealed by a three-piece front ring(Fig.1). In this way, not a fixed passage cross-section is left free. Moreover, a greater pressure drop across a certain set of sealing elements tends to minimize the leakage clearance more effectively, which would further increase the pressure drop across this set of sealing elements. Hence this gives rise to another sealing mechanism, and the pressure drop can no longer be calculated because the geometry of the leakage cross-section can not be determined now. It may happen, for example, that a certain set of sealing elements provides the entire sealing/1/. Consequently, only measurements can reveal the actual pressure distribution in the packing. If the pressure distribution in the packing is obtained, the loading on a sealing ring can be determined, and then the lubricating condition can be simulated with mathematical equations.

EXPERIMENTAL TECHNIQUES

In order to measure the main parameters which characterize the sealing and lubricating condition in a packing, a rig is constructed by the authors. The rig consists of a high-pressure compressor, a crank-link system, a packing/rod system and a measuring system. It is used to simulate the working process of a packing/

rod system in compressors and to test the frictional resistances, the sealing pressure distribution, the surface temperature and the leakage in this system. The compressed air from the compressor enters the packing from the middle box. Then it leaks along the piston-rod towards the top and bottom of the packing in other words, the sealing system consists of two packings—upper and lower one. The piston-rod is connected to the crank-link system, which is driven by a motor whose rotational speed is adjustable. In order to test the frictional resistance the packings connect only to a wheel, on whose spoke several strain sensors are mounted. The outer edge of the wheel is fixed to the frame of the rig (Fig.2). Each sealing chamber is provided with a pressure sensor to measure the pressure distribution in the packing. Several thermocouples are mounted on the sliding surface of the sealing elements to measure the temperature there. The air leaked out from the top of the packing box enters a gas collector, and then is measured with a float-type flow meter.

At first, two sets of sealing elements type T made of cast-iron are tested on the rig. One set is in the upper packing, and the other in the lower one. Under the condition of a sufficient oil supply the measured frictional resistance between the rod and sealing elements is shown in Fig.3. In order to compare the characteristics of the resistance, the same experiment as mentioned above is carried out without the oil supply. The measured resistance is also given in Fig.3. In all the above experiments the total pressure drop across the packing is 6 MPa. The diameter of the piston rod is 45 mm, and the rod stroke is 120 mm.

Because the loading acted on the above-tested sealing elements is constant in a working cycle, the factors affecting the frictional resistance are expected to be the speed of the rod, the lubricating condition in the packing and the temperature of the sliding surfaces. The frictional resistance in Fig.3a varies with the rod speed. Resistance in Fig.3b is not closely related to the rod speed. In the case of dry friction the resistance does not vary with the rod speed.

On the principle of tribology/2/ a frictional state is divided into three types as shown in Fig.4: 1. hydrodynamic lubrication: in this case two sliding surfaces are separated by a hydrodynamic oil film, and the resistance increases directly with the sliding speed; 2. mixed lubrication, in which the loading is supported by both the hydrodynamic film and microprotuberances on the sliding surfaces, and the resistance partly relates to the sliding speed; 3. boundary lubrication, in which the microprotuberances on the sliding surface apparently contact each other, and the resistance is nearly independent of the sliding speed.

Obviously, the resistance curve shown in Fig.3a characterizes a condition of hydrodynamic lubrication, and the curve shown in Fig.3b indicates a mixed lubricating condition. In order to achieve a hydrodynamic lubricating condition for packing/rod systems, mathematical simulations for this lubricated system are necessary.

THEORETICAL ANALYSIS

Many researchers have carried out mathematical simulations of the lubricating conditions in compressor cylinders. Considering the similarity of the lubricating mechanism of packing elements to that of piston rings, Reynolds equation is expected to be available to the packing/rod system, though these are some differences between these two systems, some of which will be discussed later.

For a segment of a three-piece sealing ring the forces acting on it in radial direction are gas pressure, the strain force of the spring, the supporting force of the oil film and the frictional resistance on the flank of the sealing ring (Fig.5). According to the equilibrium condition of these forces, the force acting on the oil film can be determined.

Under the condition of hydrodynamic lubrication, the simulating equation for a sealing ring is described as follows:

$$\frac{\partial}{\partial x} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial y} \right) = 6U \frac{\partial h}{\partial x} + 12 \frac{\partial h}{\partial t} \quad (1)$$

where p is pressure in the film, h is film thickness, μ is oil viscosity, x and z are axial and circumferential coordinates on the sliding surface. For analysis of the pressure distribution of the oil film, the neglect of the "squeeze" term in equation(1) will not cause a remarkable error 1/3/. Consequently, equation(1) becomes

$$\frac{\partial}{\partial x} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial y} \right) = 6U \frac{\partial h}{\partial x} \quad (2)$$

The results calculated from this equation show that the regions at the ends of the segment in which the pressure drop of the oil film occurs are very small(Fig.6).

On the other hand, the circumferential length of the segment is, in general, over 4 times larger than its axial height. Obviously, on the assumption of a whole ring, the simulating calculation will be still valid. That is to say, the second term at the left side of equation(1) can be neglected. The equation applied to simulating the lubricating condition of packing elements is

$$\frac{\partial}{\partial x} \left(h^3 \frac{\partial p}{\partial x} \right) = 6\mu U \frac{\partial h}{\partial x} + 12\mu \frac{\partial h}{\partial t} \quad (3)$$

The frictional resistance between the packing and rod is calculated from equation(3) and the following formula

$$\tau = -\mu \frac{du}{dy} \quad (4)$$

where τ is shearing stress, u is velocity of oil film, and y is radial coordinate on the sliding surface. The calculated resistance is shown in Fig.3. In comparison with the measured results the difference between them is small. Therefore the validity of the simulation can be confirmed.

ANALYSIS OF RESULTS

Again experiments are carried out with a constant rotating speed. For different sealing pressures a series of curves of the frictional resistance VS. the crank angle are obtained. The results show that with the increase of the sealing pressure the resistance curve changes from a sine wave to a rectangular one. Therefore a critical sealing pressure under which the shape of the resistance curve begins to vary can be determined by the experiments. A group of the critical sealing pressure drops can be measured for different rod speeds (as shown in Fig.7).

According to reference[4], the contact of the microprotuberances on sliding surfaces occurs when the oil film between them is less than 1.6 μm . At the same time the resistance increases sharply. Therefore a critical size can be taken as 1.6 μm . The film thickness between the rod and sealing elements is calculated for different sealing pressure drops in a given rotating speed. Of course, the larger the sealing pressure drop, the thinner the film. If the sliding surface of the rod on which the calculated film thickness is less than 1.6 μm exceeds one third of the whole sliding surface, its lubricating condition can't be accepted. Thus a group of critical sealing pressure drop can be calculated for different rotating speeds (Fig.7). In the calculation all the geometric parameters of the surface profiles of the sealing ring come from the measurement of the used sealing elements. Though there are some differences between these two results, which probably come from some assumptions, the results can give designers important information in the design of compressor packings, for example, under the condition of an average rod speed $c_m = 3 \text{ m/s}$ the sealing pressure drop across a set of sealing elements should not exceed about 7 MPa to maintain a better lubricating condition. Of course, there are several sets of sealing elements in a packing. In the light of the results measured on the rig the pressure distribution in the packing, which consists of five sets of sealing elements, is quite uneven (Talbe 1). This will cause sets of the sealing elements fail one after another.

It should be pointed out that in the above calculations the temperatures of the oil film come from the measurements on the rig. The flow of lower-temperature gas and good heat conduction in the packing of the rig allow the sliding surfaces to maintain a lower temperature. The effect of the frictional heat on the lubricating condition is not remarkable. In practical compressors the temperature of the sliding elements in the packing is much higher than that in the rig. The oil film will thin out, or the loading-capacity of the film will go down with the increase of speed within the range of high speeds.

CONCLUSIONS

On the basis of the measured and calculated results the following conclusions can be drawn:

1. Many factors affecting the sealing and lubrication of the packing/rod system can be conveniently clarified by simulating tests. The frictional process in this system satisfies hydrodynamic lubricating conditions. Mathematical simulation of the lubricating condition for this system can be conducted with Reynolds equation.

2. The sealing pressure distribution in packing is quite uneven, which will cause sets of the sealing elements to fail one after another. Therefore the effective method to solve this problem is to achieve a packing whose pressure distribution is even.

3. A critical sealing pressure drop under which the lubricating condition rapidly gets poor can be determined by experiments or mathematical simulation. This pressure drop varies with the rod speed, the temperature in the packing and the surface profile of the sealing elements, etc.

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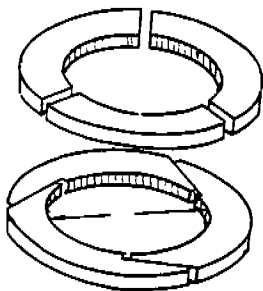


Fig.1 Sealing elements type T

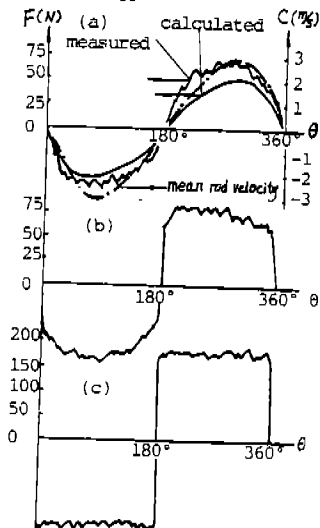


Fig.3 Frictional resistances between the packing and rod

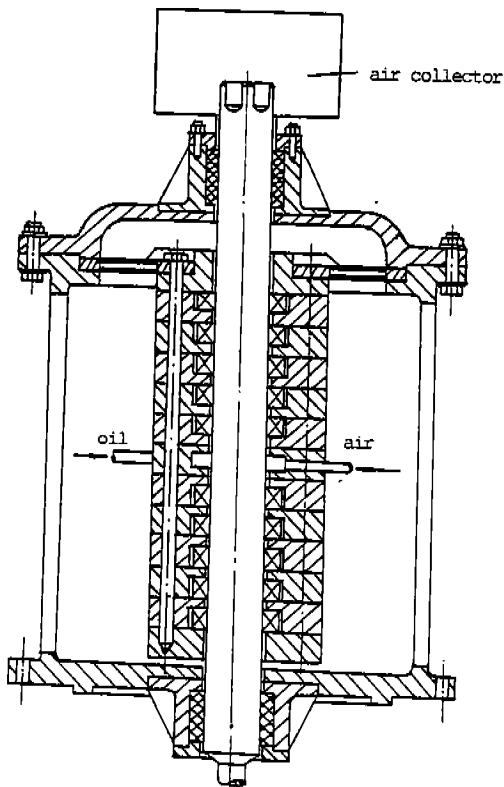


Fig.2 Installation of the tested packing

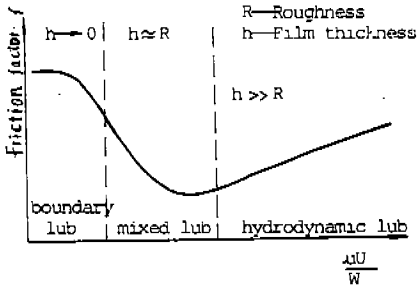


Fig.4 Stribeck curve and lubricating state

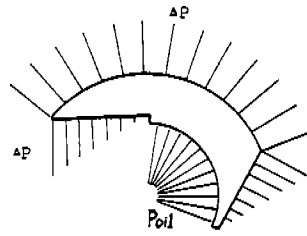


Fig.5 Forces acted on a segment of the sealing ring

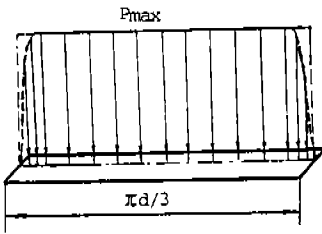


Fig.6 Circumferential distribution of the oil film pressure of ring segment

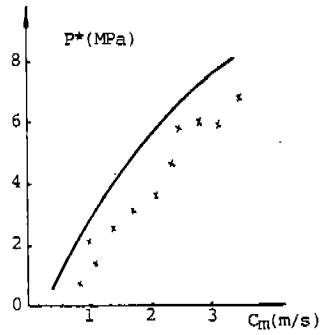


Fig.7 Critical pressure drops at different rod speeds

Table.1 Pressure drop across each set of the sealing elements

set number	sealing pressure drop (MPa)	Temp of sliding surface °C
1	0.15	75
2	0.23	73
3	4.64	110
4	4.7	105
5	0.28	65

($P=10$ MPa)