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Formation and Control of a Piston Ring Profile During Operation

Q. Feng
Xi'an Jiaotong University

S. Yang
Xi'an Jiaotong University

FORMATION AND CONTROL OF A PISTON RING PROFILE
DURING OPERATION

Quanke Feng Shaokan Yang

Department of Power Machinery Engineering
Xi'an Jiaotong University, China

ABSTRACT

In the paper an excellent surface profile of a piston ring is selected by means of the simulating calculation of the film thickness and the oil consumption between the rings and liners. On the basis of dynamics and tribology of rings stated by the authors, a practical measure is put forward in which the local surface of a common ring is hardened by laser. Installed in a Z-0.2/10 air compressor, the har- dened rings were tested for 3000 hours. The measured results show that the hardened rings do form and keep a desirable surface profile, which is provided with excellent lubricating condition and less oil consumption.

INTRODUCTION

In a single-acting reciprocating compressor the lubrication of the cylinder is usually carried out by oil splash. The oil consumption through the clearance between rings and liners is not easily controlled. As a result, a lot of oil is carried into compressed air. This not only gives rise to an excessive waste of lu- bricating oil but also causes carbon deposits in the discharge system of the com- pressor. As is well known, the carbon deposits may result in many disastrous troubles in compressors. Therefore, manufacturers and users of compressors pay close atten- tion to oil consumption. Every effort has been made to reduce the oil consumption in compressors. The widespread application of tapered and twisted rings (Fig.1), which are effective on reducing oil consumption, is one of the achievements.

In practice the tapered rings and the twisted rings have the excellent function of controlling oil consumption only in several hundred hours of their intial opera- tion. As time goes on, the original tepered profile of the rings turns into a barrel one (Fig.1), thus loosing the function of scraping oil from the cylinder wall.

Besides the clearance between rings and liners, of course, there are two other channels for oil to escape, namely, the ring/groove flank clearance and the clear- ance at the ends of the ring. However, these two channels can be reduced to small enough sizes by reasonable design and precise machining. Thus the oil consumption passing through these two channels can be cut down as less as possible. In com- parison with the oil consumption from the ring/liner clearance, the amount of oil passing through these two channels is quite small under normal conditions. Therefore studying the ring's lubricating conditions and the oil consumption through the ring/ liner clearance is far more important.

In order to understand the mechanism of lubrication in the tribological system "piston/ring/liner" and achieve the optimization of this tribological system, many researchers have carried out a lot of theoretical and experimental investigations. as a continuation of works of Kruse/1-2/, F. Wrede/3/compared the mathematical and experimental simulations of lubricating conditions in the system of piston, piston ring and liner. U. Todsen /4/ put forward an idea of hydrodynamic optimization of the tribological system of piston ring and liner, and compared the lubricating characteristics of a barrel shaped ring with that of a wedge shaped ring at dif- ferent sliding speeds. A. Baker /5/ analyzed the piston ring dynamics in internal combustion engines and proposed a feasible method of theoretical determination of lubricant consumption. However, a comprehensive study concerning ring's lubrication, ring's wear feature and its dynamics should be carried out to achieve a practical optimization of ring/liner system.

MATHEMATIC SIMULATION

In recent years there have been significant advances in the thoeretical analysis of piston ring performance. By considering ring as a dynamically-loaded bearing, and solving the simplified Reynolds' equation applicable to the oil film seperating the ring and liner, much has been learnt about the principal features of piston ring lubrication. According to the well known assumption for incompressible Newtonian fluids the mathematical model can be described by the Reynolds equation as follows:

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

where p is pressure in oil film, h is oil film thickness, μ is dynamic viscosity of lubricating oil, U is sliding speed, t is time, x,y and z represent axial, radial and circumferential coordinates respectively. This equation cannot be solved within an acceptable amount of computational time for a ring/liner system. Since the circumferential length of a ring is much longer than its axial height, the second term at the left of Eq.(1) can be neglected. Then the equation can be smplified as follows:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right)=6\mu U\frac{\partial h}{\partial x}+12\mu\frac{\partial h}{\partial t} \tag{2}$$

This two-dimensional model describes nonsteady conditions including variable piston velocities and squeeze effects within the fluid.

In order to calculate the oil consumption through the clearance of the ring/liner the rings are reparded as motionless. Thus the liner can be considered as reciprocation with velocity U in relation to the rings. The distributing velocity of the lubricating oil on the ring profile can be given as follows:

$$u(x\cdot y)=\frac{1}{2\mu}\frac{\partial p}{\partial x}(y-h)y-\frac{h-y}{h}U \tag{3}$$

because $\frac{\partial p}{\partial x}=0$ at $x = x_0$, the distribution of velocity u is linear along the direction of coordinate y. Therefore u can be represented as:

$$u(x_o,y)=\frac{h-y}{h} \tag{4}$$

The oil displacement in a unit circumferential length of the ring profile is discribed as follows:

$$g=\int_0^{h_{x_o}}\gamma\, u(x_o,y)\,dy \tag{5}$$

where γ is the specifc gravity of oil. The oil consumption in a working circle will be

$$G=\pi D\left(\int_0^{\pi}g\,d\alpha-\int_{\pi}^{2\pi}g\,d\alpha\right) \tag{6}$$

where α is the instantaneous crank angle and D is the diameter of the cylinder. In calculating the film thickness and oil consumption of the ring/liner system, the instantaneous velocity of the ring, the pressure in the cylinder, the instantaneous viscosity of the oil and the parameters of the ring profile are necessary, all of which are determined in terms of the operating conditions and structural parameters of the ring/liner system.

INFLUENCES OF RING PROFILE ON LUBRICATION AND OIL CONSUMPTION

In order to clarify the influences of a ring profile on lubricating conditions and oil consumption, five different models of ring profiles (Fing.2) are taken into account. Generally, most of the ring profiles in compressors can be roughly simplified into one of these models. Models A,B and C represent the ring profiles formed after normal running-in. Models D and E represent the surface shapes of newly machined rings. For every piston ring its profile can be simulated as Fig.3, in which section L_1 and L_3 are parabolic curves and section L_2 is a straight line.

If the magnitudes of L_1, L_2, L_3 and the curvature of the parabolic curves are changed, different kinds of ring profiles can be obtained. On the basis of the operating conditions of a Z-0.2/10 air compressor and its ring profiles shown in Fig.2, the oil film thickness and oil consumption of the ring/liner system are calculated in terms of the equation given above. Their results are shown in Fig.4 and Table 1. Because many researchers /3-5/ have proved the validity of the above-mentioned mathematical simulations, it is no longer necessary to test the calculated oil film thickness by experiments.

From Fig.4 and Table 1 the following conclusions can be drawn. Model E is provided with an excellent function of scraping oil from the liner surface and a poor lubricating condition in the downstroke. Model A benefits the formation of thicker oil film, but its oil consumption is not satisfactory. By comparison with Model C, Model B can be provided with less oil consumption and thicker film. Therefore it is desirable for a piston ring to be turned into this model during its running-in, and to keep this model unchanged over a long period of operation. However the profiles of the piston rings manufactured on the basis of the existing technological standards of twisted or tapered rings are similar to Model E. After running-in they approach Model B. Over a long period of operation they will be turned into the similartype of Model A.

It should be noted that the negative value of oil consumption by calculation does not mean that the lubricating oil flows continuously from the compression chamber to the crankcase through the clearance between the rings and the liner. In the simulation of oil consumption, it is assumed that there is an adequate oil supply either in the upstroke or in the downstroke. Actually, the oil resource for most of small compressors is in the crankcase. The oil supply for lubricating the rings is sufficient in the downstroke, and insufficient in the upstroke. Therefore the negative amount of oil consumption does not exist. Though the calculated results of oil consumption are not real, they can indicate the extent to which different rings are able to control the oil consumption.

MECHANISM OF THE FORMATION OF A RING PROFILE

When a compressor is put into operation, the wear of its rings and liners is unavoidable. This fact shows that the split of hydrodynamic oil film in the ring/liner system often takes place, exspecially near to TDC and BDC , this breakdown is more frequent. It is the emergence of such a lubricating condition that makes possible the formation of various ring profiles. On the other hand, the fact that the low opecific wearability of many piston rings which have been used for 8000 hours in dicates that the lubricating condition in the ring/liner system approaches perfect hydrodynamic lubrication. A breakdown of oil film occurs only at a few points on sliding surfaces.

Several years ago, the profiles of many piston rings used in compressors were measured by the authors. The results showed that most of the ring profiles approach barrel shape. To form proper profile, in general, a piston ring has to tilt against the cylinder surface, and thus wearing off its upper and lower edges. The tilting angles for an aptimum profile are about 10'-20'. Many kinds of tilting action of the piston ring are to be expected as follows:

(a) The ring is twisted by the gas pressure on the piston ring surface.
(b) Twisting by inertia force of the ring.
(c) Twisting by frictional force of the ring against the cylinder wall.
(d) Thermal stress in the ring, the piston and the cylinder.
(e) Geometrical condition of the upper and lower surface of the ring groove.
(f) Tilting of the piston against the cylinder wall, of course, a piston ring would be tilted against the cylinder wall as a result of combined action of these factors.
(g) Rotation of the piston ring in the cylinder, which will make more complex the the influence of tilting on the formation of the ring profile.

It is obvious that most of the above-mentioned factors are random. Their total effect——tilting of the piston ring against the cylinder wall, of course, it still more random. According to the ivestigations of Furuhama /6/, the probable occurrence of larger tilting angles is less frequent, wheseas that of smaller tilting angles is more frequent. Therefore a normal distribution can be used to describe the regular pattern of the variations of the tilting angles. If tilting angle θ is taken as an independent variable, the probability distribution of the tilting angles can be described as follow:

$$f(\theta) = \frac{1}{\sqrt{2\pi}\,\sigma} e^{\frac{-(\theta-\mu)^2}{2\sigma^2}} \quad (7)$$

where μ is the mathematic expectation, σ is mean square deviation. Assuming that the probabir occurrence of positive tilting angles is equal to that of negative tilting angles, the probability distribution of tilting angles can be shown as Fig.5. The tilting angles, in general, are very small. Hence an excellent lubricating condition is expected. When the larger tilting angles occur, the lubricating condition is poor. This makes the upper and lower deges of the ring wear quickly. It is the high specific wearability of ring edges that leads to the formation of a desirable ring profile.

CONTROL OF A RING PROFILE

In consideration of the dynamical factors and wear feature of the rings, a practical measure has been taken to control the ring profiles. i.e. a narror band of the outer surface of a rectangular or tapered ring is hardened with laser to get different hardness. Then the ring is assembled on the piston with the hardened band facing the crankcase. Different hardness on the same surface can cause different wear. Therefore the formation of profile Model B in Fig.2 can be expected in the compressor operation. In this way, 2 rectangular rings are hardened and mounted in Z-0.2/10 air compressor. After running in the compressor for 1000 and 3000 hours, the hardered rings are carefully measured with a tricoordinate measuring machine (Fig.6). In order to reduce measuring errors, the two ends of a ring are butted together with a cuff. The ring is fixed on a gauge block. The flank surface of the ring is laid so horizontally that differences of the heights measured at any position on it are less than 5 um. Then a profile curve along the axis of the ring is measured out by raising and descending the gauge pin. Fig.7 shows the position of the measuring points and the profiles measured at 6 points on the ring, which has been tested for 3000 hours. The measured profiles of another ring are similar to this one. The profiles of these two rings after running for 1000 hours are also similar to the results shown in Fig.7. Thus it can be seen, that these special profiles formed automatically in the ring's operation are very stable. Further results have been obtained from the authors' recent investigations. A desirable profile for different type of rings can be achieved by choosing appropriate location, proper hardness, moderate width and depth of the hardened band on the ring surface.

CONCLUSION

1. Optimization of a ring profile can be achieved by simulation of the lubricating condition and oil consumption of the ring/liner system.

2. A ring profile has important influences on the lubricating condition and oil consumption of the ring/liner system. After running in a compressor for several hundred hours, the surface profile of a common rectangular ring, twisted ring and tapered ring turns completely into barrel shape, which provides a better lubricating condition but a larger oil consumption.

3. When a ring tilts against the liner within a range of minor anlges, its lubricating condition is perferct. Of course, the ring works mainly in this condition. When a ring tilts within a range of larger angles, its lubricating condition is poor, and its edges wear rapidly. Consequently, a ring profile benefiting lubricating condition is formed.

4. For a ring hardened in a given surface, its upper edge is easier to wear its lower edge because of their different hardness. As a result, an asymmetrical barrel profile can be obtained, which benefits both the excellent lubricating condition and the remarkable reduction in oil consumption.

5. An asymmetrical barrel profile can be controlled by adjusting the position, width and hardness of hardered hand on the ring. The profile obtained in this way will remain unchanged until all the hardened band wears off.

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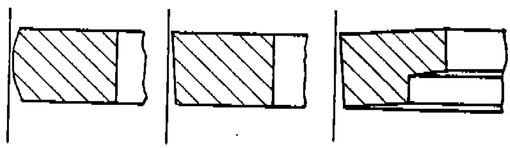

Fig.1 Three types of common rings

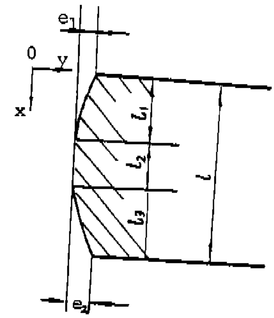

Fig.3 Geometric parameters of a ring profile

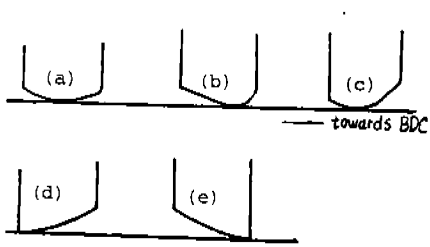

Fig.2 Five models of ring profiles

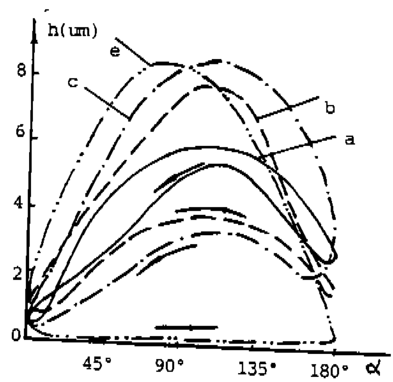

Fig.4 Calculated oil film thicknesses for different profiles

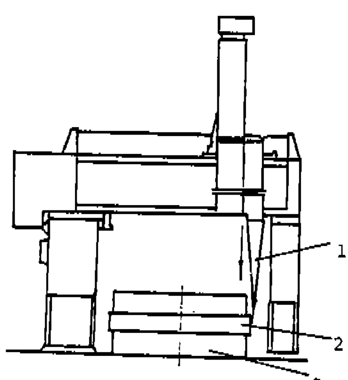

Fig.6 Tricoordinate measuring machine

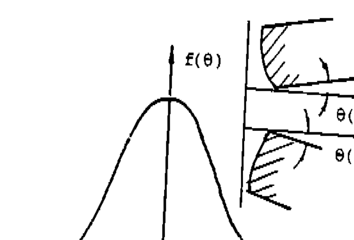

Fig.5 Probability distribution of tilting angles

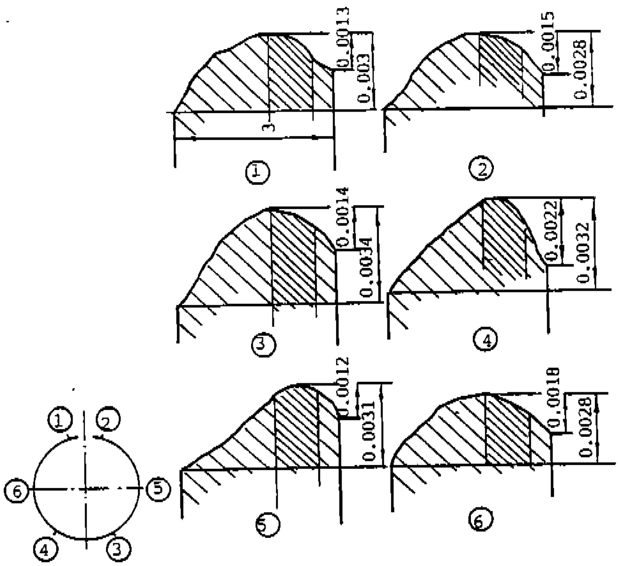

Fig.7 Measured profiles of a ring

Table 1 oil consumption (O.C)

Type of profile	a	b	c	d	e
O.C cal-culated	101	-908	1205	1825	-2040
O.C. mea-sured	70	9	520	730	3