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ABSTRACT

This paper has simulated with computer to the lubrication state at piston rings, and presented the principle of lubrication oil selection based on the construction and performance parameters of reciprocating compressor. Also, an optimization programme has been presented. The lubricant selected by this programme can reduce the frictional loss to a minimum on the basis of an allowable minimum value of the oil-film thickness. The authors have made experimental measurements, and the results show that the selected oil meets the lubrication requirements with minimum work consumption and higher operating reliability of the compressor. The experimental results are in agreement with the theoretical studies.

INTRODUCTION

To bring the performance of compressors into full play depends largely on the suitable condition of lubrication. Especially with the development of high-pressure and high-rotation rate compressors, the requirement of better lubrication is quite necessary.

In the past, lubricating oil for compressors was selected mainly by experience, thus resulting in two defects: the increase of frictional loss between the piston rings and the cylinder wall caused by high viscosity of oil, or the increase of wear between frictional surfaces because the lower-viscosity oil can not maitain normal oil-film thickness.

After many years' research on oil selection, the authors have presented a programme for optimal oil selection based on the construction and performance parameters of reciprocating compressors. The lubricant selected by this programme can reduce the frictional loss to a minimum on the basis of an allowable minimum value of the oil-film thickness. The authors have made experimental measurements for above 2000 hours, and the results show that the selected oil meets the lubrication requirements with minimum work consumption and higher operating reliability of the compressor. The experimental results are in agreement with the theoretical studies.

MATHEMATICAL MODELS

1. The Viscosity Of Oil Reduces Sharply With The Increase Of The Oil Temperature

The temperature of oil film depends mainly on the temperature of the cylinder wall. The average temperature of the cylinder wall of a two-stage, water-cooling and double-acting compressor is defined by equation (1).

$$\bar{T} = 211 + 0.3\, T_{bc} + 0.017\, n + 8\,(\varepsilon - 1) - 5\,\frac{s(\varphi)}{S} \qquad (1)$$

where: T_{bc} is the temperature of intaking gas, K; n is the speed of revolution of crank, rpm; ε is the ratio of final pressure and initial pressure; S is the piston stroke, m; $s(\varphi)$ is the piston displacement, m/s

During computing the temperature of the cylinder wall of compressor of experamental facility, the constant and every coefficient in equation (1) are corrected according to the experamental result. When the pressure is less than 1.5 MP_a, the viscosity influenced by the pressure is negligible. Therefore, the viscosity of lubricating-oil is only determined by the temperature, i.e., the relation between viscosity and temperature:

$$l_g l_g(\nu + 0.8) = a\ l_g(T) + b \tag{2}$$

where: ν is kinematic viscosity, sentistoke; T is absolute temperature, K; a and b are constants.

If the viscosity of oil under two different temperatures are determined, the constants in equation (2) can be defined. The viscosity of oil in the mathematical model is defined by equation (2). The relation between viscosity and temperature of compressor lubricating oils with different viscosities is shown in Fig. 1.

2. Calculation Of Oil-film Thickness

The hydrodynamic equation of piston rings are derived from the general Reynold equation as follows:

$$\frac{\partial h}{\partial t}=\frac{\partial}{\partial X}\left(\frac{\partial P}{\partial X}\frac{h^3}{12\mu}\right)-\frac{\partial}{\partial X}\left(u\frac{h}{2}\right) \tag{3}$$

Equation (3) expresses the relation of pressure p, oil-film thickness h, piston speed u and dynamic viscosity of oil μ at time t and at the axial position x where piston rings and cylinder wall are in contact. The relation between dynamic viscosity and kinematic viscosity is defined by equation (4):

$$\nu=\frac{\mu}{\rho} \tag{4}$$

The boundary conditions of equation (3) are shown in the following equations:

$$\mu = f(T)$$

$$x = 0; \quad p = p_1$$

$$x = x_1; \quad p = dp/dx = p_2$$

where: ρ is the density of lubricating oil.

The cross section of piston rings in the experimental facility is shown in Fig. 2. The convex of the piston ring is determined by an inductance discrimination instrument, h' = 2-3 m.

3. Computation Of Friction

The velocity distribution of oil film at the position x is shown as follows:

$$U = \frac{1}{2\mu}\frac{\partial p}{\partial x}(Y^2 - hY) + u\left(1 - \frac{Y}{h}\right) \tag{5}$$

Equation (5) is differentiated to radial axis Y and shearing stress formula is substituted

$$\tau = -\mu\frac{\partial u}{\partial y} \tag{6}$$

Then solve it with equation (4) simultaneously. So the friction formula is obtained:

$$R_f=4\mu u\alpha(H) + 6\mu\frac{\partial h}{\partial t}\beta(H) + 6\mu C_o\psi(H) \tag{7}$$

where: $\alpha(H)$, $\beta(H)$ and $\psi(H)$ are the functions of oil-film form; C_o is the constant of integration.

The ratio of friction and force acting on the cylinder wall by piston rings is friction coefficient.

OPTIMAL SELECTION OF LUBRICATING OIL

By means of the above methematical models and computer calculation , we can get the variational curve of the oil-film thickness and friction coefficients of lubricating oils with various viscosities in relation to the crank angle within a working cycle. The compressor in the experimental facility is under normal pressure.

From Fig. 3 and Fig. 4, the larger the viscosity is, the thicker the oil film is. However, the friction coefficient is increased at the same time. The energy loss is related to the piston speed and friction. In turn the viscosity of oil sharply declines due to the larger amount of frictional heat, thus resulting in the thinning of oil film and the increase of friction because of the probable direct contact of piston rings and the cylinder wall. When the viscosity of oil is small, the oil-film thickness for fluid dynamical lubrication cannot be formed. Hence, the optimization programme is made on the basis of the above considerations. In this programme, the minimal consumption of frictional work is the objective function, and the friction coefficient, oil-film thickness and leakage rate, etc. are taken as constraints. In the meantime, when one thirds of the point-number of oil-film thickness is less than 1.6 μm in a working cycle, this condition is considered to be mixed-lubrication. After optimization, the relation curve of viscosity and frictional work is made and shown in Fig. 5. The curve shows that the consumption of frictional work is minimum when #32 oil is applied to the cylinder, and the consumption of frictional work increases when the oil applied to the cylinder is lower than #32 and higher than #150 (the number expresses the viscosity of oil at 40°C, sentistoke). This deduction is in accord with the experimental results which are shown in Table 1.

Table 1. The change of power per capacity with oil viscosity for the tested compressor

Oil Viscosity, centistoke (40°C)	32	68	100	150
Power Per Capacity, kw/(m3/min.)	5.0	5.11	5.14	5.34

Similarly, the optimal oil viscosity can be selected for other types of reciprocating compressors in accordance with this construction and performance parameters by means of this optimization programme.

THE INFLUENCE OF OIL VISCOSITY ON THE LEAKAGE OF WORKING MEDIUM AND THE FLASH POINT OF OIL

Generally, when the viscosity of oil is lower, the leakage will increase and the flash point will reduce. The authors have carried out practical measurements with the experimental facility of the sealed piston rings. When #150 oil is applied to the cylinder under the pressure difference of 0.5 MP_a and the speed of 600 rpm, the leakage of gas is 1130 ml/min. Whereas the application of #68 oil under same conditions makes the leakage increase to 1150 ml/min. Therefore there is almost no influence of oil viscosity on working medium leakage. Besides, the flash point of oil is increased with the rise of pressure. The intensified experiment also shows that the compressor can be put into safe operation when #32 oil is applied at the delivery temperature of 250°C and the pressure of 1.3 MP_a. Hence these two factors are negligible, and the calculating period of the programme will be greatly shortened.

CONCLUSIONS

1. The selection of lubricating oil for reciprocating compressors should be made according to the construction and performance para-

meters of the compressors.

2. When the working medium pressure is less than 1.0 MP_a, the optimization programme only takes the viscosity of lubricating oil into accoumt, and the working medium leakage and the flash point of lubricating oil can be neglected for simplicity.

3. Lower-viscosity oils are to be selected for normal-pressure power compressors.

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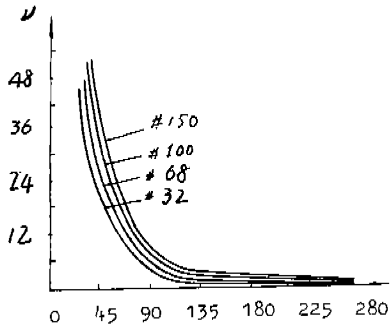

FIG. 1
The relation between viscosity and temperature of lubricating oils

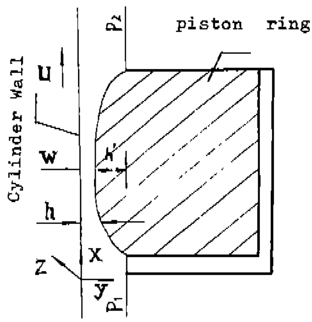

FIG. 2
The cross section of piston ring

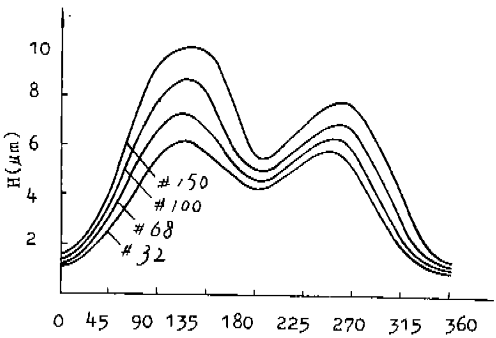

FIG.3 The relation between oil-film thickness and various viscosities with the crank angle

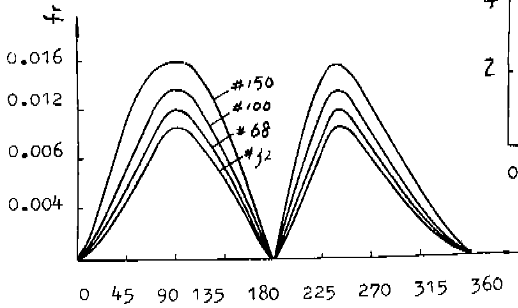

FIG. 4 The relaton between friction coefficients and various viscosities with the crank angle

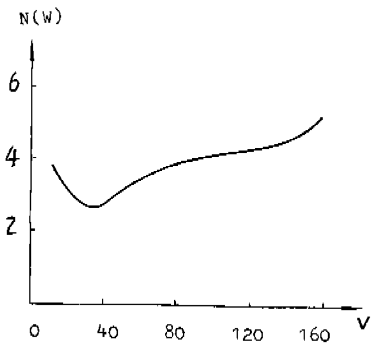

FIG.5
The relation of viscosities and frictionail work