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Analysis of Factors Affecting Lubrication Performance of Main Bearings of Service Diesel Engine Based on Orthogonal Design Method¹

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Abstract. Based on the established numerical lubrication simulation model of a main bearing of a certain type of diesel engine, combined with the orthogonal design method, the effects of engine speed, cylinder pressure, oil supply pressure and clearance on bearing lubrication characteristics are studied. The results of lubrication characters show that cylinder pressure and clearance are first and secondary factor of influence degree, oil supply pressure has the least impact, and the most effective way to improve bearing lubrication is to reduce the explosion pressure appropriately.

1. Introduction

Diesel engines are widely used in transportation, engineering machinery, and national defense. They are highly efficient and are moving toward high power density and reliability. The main journal of crankshaft and main bearing is one of the most important friction pairs in diesel engines, affecting the efficient transmission and output of power, plays a key role in the vibration, noise and reliability of the whole machine.

The sliding bearing in diesel engine works in a very harsh environment, under periodic impact load caused by cylinder combustion pressure. With its power density increasing, its lubricating performance faces greater challenges. Research shows that the existence of the clearance has a great influence on the contact mechanism between the sliding friction pairs, and the influence becomes more obvious with the increase of the clearance[1]. During the service of the internal combustion engine, the working conditions are variable, and the bearing clearance increases due to wear of the bearings under dynamic load, changes the dynamic load and lubrication characteristics, which in turn aggravates the wear and forms a vicious circle.

In this paper, the numerical simulation of the main bearing lubrication of a diesel engine is carried out. Based on this, the orthogonal test design method is used to study the influencing factors and influence laws of the main bearing lubrication characteristics in the service process, and then explore the method of extending the remaining life of the equipment.



2. Numerical simulation of the main bearing lubrication

2.1 Hypothesis

The structure and load of the internal combustion engine are complex, and the Numerical simulation model is appropriately assumed to ensure the calculation,

- The journal-bearing clearance is regular,
- Assumes that wear of the clearance pair along the circumferential is equivalent at a certain wear clearance,
- The dynamic process of wear is not considered when calculating the dynamic response because the amount of wear is a slower variable parameter relative to the kinetic process,
- The bearing shell adopts the same property as a whole,
- Assumes a rough peak Gaussian distribution, and the stress does not increase at a single rough peak when it reaches the yield limit in bearing lubrication analysis,
- Assumes that the supply pressure is constant.

2.2 governing equations of numerical simulation

The simulation mainly combines the multi-body dynamics equation and the extended Reynolds equation of lubrication. According to the Lagrangian dynamic equation, the governing equations of the multi-flexible dynamic of the elastic axis are as below

$$M\ddot{q} + \dot{M}\dot{q} - \frac{1}{2} \left[\frac{\partial M}{\partial q} \dot{q} \right]^T \dot{q} + Kq + f_g + D\dot{q} + \left[\frac{\partial \psi}{\partial q} \right]^T \lambda = F$$

where q is generalized coordinates, M , K , D are generalized mass, stiffness and damping matrix, f_g is generalized force matrix, F is generalized force matrix, λ is lagrangian operator, and ψ are system constraint equations.

As the oil film thickness direction is much smaller than the other two directions, the sliding bearing is simplified into two directions for the Navier-Stokes equation. The elastic fluid lubrication model is established by the extended Reynolds equation [2] as below

$$\frac{\partial}{\partial x} \left(\theta \frac{h^3}{12\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left(\theta \frac{h^3}{12\mu} \frac{\partial p}{\partial z} \right) = \frac{\partial(\theta h)}{\partial x} \frac{U_1 - U_2}{2} + \frac{\partial(\theta h)}{\partial t}$$

Where p , h are the oil film pressure and thickness, θ and μ are the oil filling rate and viscosity respectively, U_1 and U_2 the speed of the journal and the bearing in the circumferential direction, z and x are the axial and circumferential coordinates, respectively, and t represents time.

Due to the rough surface of the journal and bearing, the oil film thickness is small sometimes, and even reaches the roughness level (several microns), the rough contact equation uses the Greenwood/Tripp model[3] as follows

$$P_a = \frac{16\sqrt{2}\pi}{15} (\eta_s \bar{\beta} \sigma_s)^2 \sqrt{\frac{\sigma_s}{\beta}} E^* F_{5/2}(H_s)$$

Where

$$\text{Deformation function } F_{5/2} = \begin{cases} 4.40861^{-5} (4 - \frac{h}{\sigma})^{6.804}, & \frac{h}{\sigma} < 4 \\ 0, & \frac{h}{\sigma} \geq 4 \end{cases}$$

$$\text{Comprehensive elastic modulus } E^* = \frac{1}{\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right)}$$

E_k ($k=1,2$), ν_k ($k=1,2$) and η_s are the elastic modulus of the two materials, the Poisson's ratio and the number of rough peaks in the nominal area.

2.3 Physical models and boundary conditions

The engine characteristic parameters and bearing material properties are shown in table 1 and table 2, and the mesh of oil film is shown in figure 1.

Table 1. Engine Characteristic Parameters

Name	Unit	Value
Bore diameter	mm	116
stroke	mm	139
Cylinder distance	mm	144
Design pressure	MPa	20
Rated speed	r/min	1900
rated power	kW	247
Maximum torque	Nm	1600
High torque speed	r/min	1000-1400
Fire order	—	1-5-3-6-2-4

The numerical simulation model uses the Runge-Kutta method and the finite difference method to solve the multi-body dynamic equations and the Reynolds equation.

Table 2. Bearing Material Properties

Name	Crankshaft	Bearing
Elastic Modulus (MPa)	210000	212000
Poisson's ratio	0.3	0.3
Roughness	0.1	1
Bearing clearance	0.051	

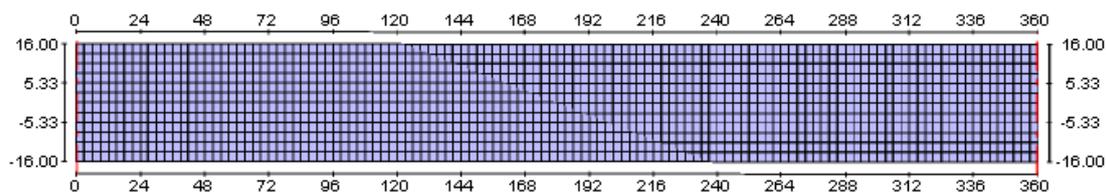


Figure 1. Mesh of the bearing film

2.4 Grid independence

In the numerical simulation of bearing lubrication, the number of grids plays an important role in the simulation results, thus the grid-independent verification is performed. The selected grids are 12×120 , 18×120 , 24×120 , 24×240 , 48×240 and 24×480 , and then the maximum oil film pressure and minimum oil film thickness are calculated in different grid numbers. Results are shown in figure 2.

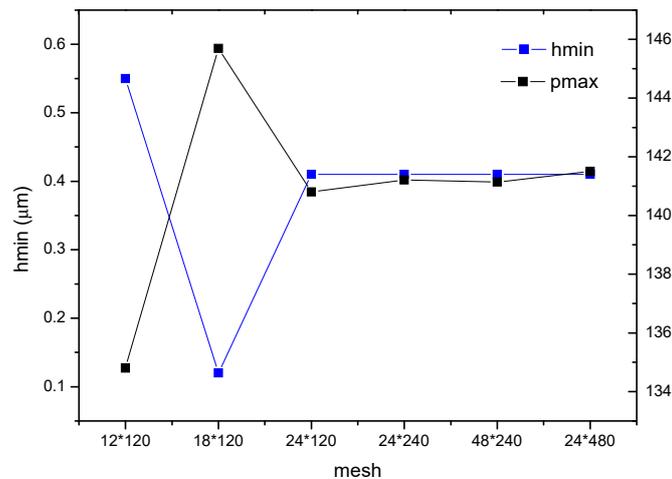


Figure 2. Grid independence

It can be seen that as the meshing density increases, the calculation result tends to be stable, the calculated error after 24×240 is less than 1%. Considering the calculation accuracy and efficiency, the fourth group is adopted.

3. Analysis of Factors Affecting Lubrication Characteristics of Bearings and Orthogonal Test Design

3.1 Analysis of Factors Affecting Lubrication of Main Bearings in Service Diesel Engines

There are many factors affecting the lubrication of the main bearings of diesel engine in service, which are mainly divided into the following categories [4-8],

- Bearing load dynamic parameters (speed, ignition sequence, balance weight arrangement and size, V-angle in V-type engine, cylinder pressure, etc.),
- Bearing running characteristics factors (bearing width, bearing diameter, bearing clearance, the position angle of oil holes, etc.),
- Lubricating factors (lubricating oil viscosity, lubricating oil temperature, lubricating oil viscosity, oil supply pressure, etc.),
- Surface characters (surface texturing, wear profile, surface roughness).

There are too many factors affecting the lubrication characteristics of the main bearing, and the influence mechanism of these factors on lubrication is complicated. In this work, for a certain existing model in the service process, the main bearing lubrication condition is mainly affected by the working condition (engine speed and cylinder pressure), lubrication conditions (oil supply pressure) and the effects of wear clearance during service.

3.2 Orthogonal array

The main purpose of the paper is to study the effects of the engine speed, cylinder pressure, oil supply pressure and radial clearance on the bearing lubrication characteristics. The five levels of these four factors (regardless of their interaction) of the orthogonal tests are shown in table 3. The parameters cannot be too large or too small. As for the target engine, the selection of the rotational speed is based on the five typical engine speeds. The upper and lower limits of the cylinder pressure and the oil supply pressure are measured during test experiments, the get the uniform values within the range the five levels, the clearance is based on the design clearance of $50 \mu\text{m}$ (statistics average value) and take one smaller value and three greater values. The Orthogonal Array is shown in table 4.

Table 3. Orthogonal Factor and Level Table

Level	Factors			
	engine speed/rpm	cylinder pressure/Mpa	oil supply pressure/bar	clearance / μm
1				
2				
3				
4				
5				

1	600	15.00	200	45
2	1000	17.25	250	50
3	1400	20.00	300	55
4	1900	22.25	350	60
5	2150	25.00	400	65

4. Results and discussion

Using the numerical simulation model established in section 2, and the simulation conditions are corresponding to the design conditions in table 5 of section 3.2, the influence degree and influence law of each factor on the lubrication characteristics of the main bearing of the service diesel engine are analyzed.

Table 4. $L_{25}(5^5)$ Orthogonal Array

Level	Factors					Evaluation index
	A	B	C	D	E	
1	1	1	1	1	1	
2	1	2	2	2	2	
3	1	3	3	3	3	
4	1	4	4	4	4	
5	1	5	5	5	5	
6	2	1	2	3	4	
7	2	2	3	4	5	
8	2	3	4	5	1	
9	2	4	5	1	2	
10	2	5	1	2	3	
11	3	1	3	5	2	
12	3	2	4	1	3	
13	3	3	5	2	4	
14	3	4	1	3	5	
15	3	5	2	4	1	
16	4	1	4	2	5	
17	4	2	5	3	1	
18	4	3	1	4	2	
19	4	4	2	5	3	
20	4	5	3	1	4	
21	5	1	5	4	3	
22	5	2	1	5	4	
23	5	3	2	1	5	
24	5	4	3	2	1	
25	5	5	4	3	2	

4.1 Orthogonal experimental scheme and analysis

The orthogonal test conditions and results are shown in table 5. A total of 25 sets of tests are designed, with the maximum total pressure and the minimum oil film thickness being the evaluation indexes of bearing lubrication characteristics.

4.2 Effect of factors in different levels on the maximum total pressure

The maximum total pressure of the bearing is an important index to evaluate the bearing lubricating character and load capacity, and the maximum total pressure value at different levels of each influencing factor is extracted for analysis, as shown in the figure 3.

Figure 3 shows the effect of engine speed on the maximum total pressure. As the engine speed increases, the fluctuation of the maximum oil film pressure becomes obvious, showing a trend of

decreasing firstly, then increasing and later decreasing again. According to the dynamic lubrication theory, when the rotating speed is low, it is not good for the formation of oil film, causing poor load capacity. With the rotational speed gradually increases, the dynamic lubrication

Table 5. Simulation Condition and Results

Level	Factors				Evaluation index	
	engine speed/rpm	cylinder pressure/Mpa	oil supply pressure/bar	clearance / μm	Pmax Mpa	Hmin μm
1	600	15.0	200	45	85.013	0.831
2	600	17.5	250	50	102.455	0.642
3	600	20.0	300	55	122.272	0.505
4	600	22.5	350	60	146.739	0.398
5	600	25.0	400	65	264.774	0.321
6	1000	15.0	250	55	86.865	0.973
7	1000	17.5	300	60	101.500	0.809
8	1000	20.0	350	65	117.510	0.604
9	1000	22.5	400	45	130.723	0.513
10	1000	25.0	200	50	149.219	0.422
11	1400	15.0	300	65	87.111	1.027
12	1400	17.5	350	45	88.040	0.921
13	1400	20.0	400	50	108.870	0.720
14	1400	22.5	200	55	130.391	0.450
15	1400	25.0	250	60	159.560	0.228
16	1900	15.0	350	50	79.285	0.878
17	1900	17.5	400	55	94.442	0.631
18	1900	20.0	200	60	135.214	0.441
19	1900	22.5	250	65	172.325	0.575
20	1900	25.0	300	45	206.249	0.332
21	2150	15.0	400	60	82.998	0.838
22	2150	17.5	200	65	102.141	0.671
23	2150	20.0	250	45	101.848	0.678
24	2150	22.5	300	50	127.687	0.471
25	2150	25.0	350	55	136.806	0.512
Pmax1	144.251	84.254	120.396	122.4		
Pmax2	117.163	97.716	124.660	113.5		
Pmax3	114.794	117.143	124.707	114.2		
Pmax4	137.503	141.573	113.971	125.2		
Pmax5	110.296	183.322	126.504	148.8		
PmaxR	33.9546	99.067	12.533	35.27		
Hmin1	0.540	0.720	0.56305	0.655		
Hmin2	0.568	0.735	0.619102	0.627		
Hmin3	0.601	0.590	0.629	0.614		
Hmin4	0.621	0.481	0.663	0.543		
Hmin5	0.644	0.363	0.604	0.450		
HminR	0.104	0.372	0.100	0.205		

condition becomes better, and the local maximum total pressure is gradually reduced. After that, as the rotational speed increases, the heat generated by the high-speed rotation gradually increases, which

decreases the viscosity of the oil, and the load capacity decreases, causing p_{\max} increasing. As the engine speed further increases, the flow rate of the lubricating oil becomes larger, and the heat generated by the rotation is taken away by the lubricating oil, then the viscosity of the oil is increased as the temperature is lowered, so the carrying capacity is enhanced, and the maximum total pressure is gradually reduced.

Figure 4 shows that the influence degree of the cylinder pressure on the maximum total pressure of the bearing is obvious, as the cylinder explosion pressure increases, the maximum total partial pressure of the bearing also increases, which conforms to the force principle of the engine, because the load on the main bearing is mainly derived from the cylinder pressure, passing through the piston, connecting rod and crankshaft.

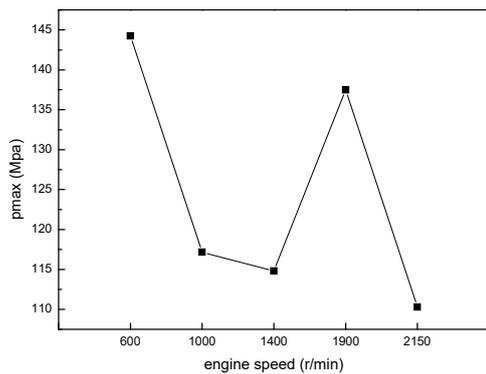


Figure 3. Effect of engine speed on the maximum total pressure

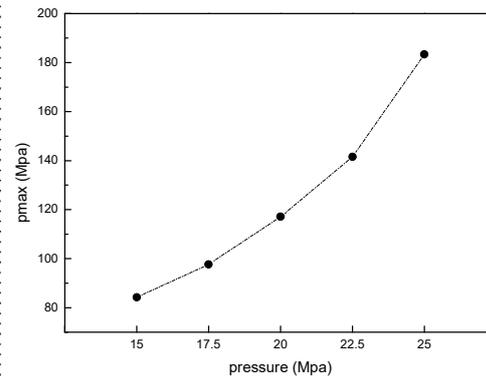


Figure 4. Effect of cylinder pressure on the maximum total pressure

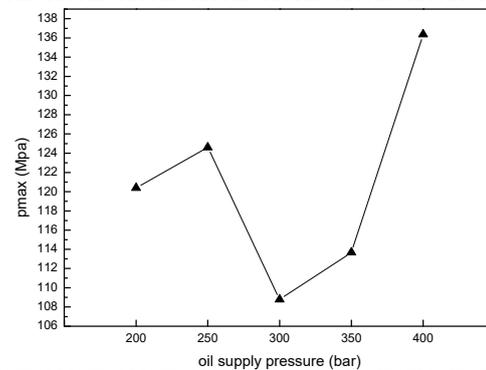


Figure 5. Effect of oil supply pressure on the maximum total pressure

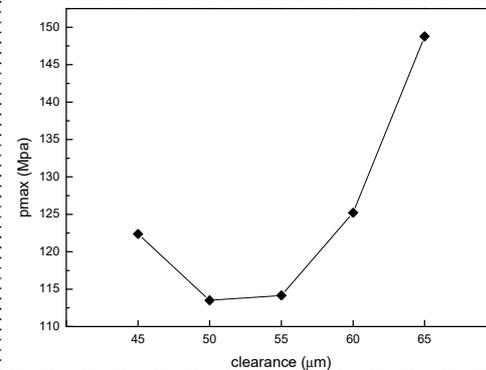


Figure 6. Effect of clearance on the maximum total pressure

The oil supply pressure is mainly controlled by the pressure relief valve of the lubricating oil pipe line. From the degree of influence on the maximum total pressure (Figure 5), the influence of different oil supply pressures below 350 bar on the maximum total pressure of the main bearing has a small fluctuation, which becomes larger at 350-400 bar. The maximum total pressure has a more obvious variety, which is the difference between dynamic lubrication and static lubrication.

The racial clearance is the second factor affecting the maximum total pressure of the bearing. It can be seen from figure 6 that as the clearance increases, the maximum total pressure decreases first and increases later, because a small gap is not conducive to heat dissipation due to the poor flow of lubricating oil. The increase of the viscosity of the lubricating oil increases the bearing capacity, and the local maximum total pressure decreases, as the clearance increases, the movement range of the journal increases, and the contact collision behavior increases, resulting in an increase of the local maximum total pressure, that is, during the service progress, the local maximum total pressure of the bearing is getting larger and larger, which may lead to a decrease in the remaining useful life of the bearing and even the whole engine.

4.3 Effect of factors in different levels on the minimum oil film thickness

The minimum oil film thickness is the general index for evaluating the quality of the bearing oil film. The minimum oil film thickness at different levels of each influencing factor is extracted for analysis. Generally speaking, the maximum oil film pressure and the minimum oil film thickness of the main bearing are not in the same position. As shown in the figure 4, the order of influence degree of these factors on the minimum oil film thickness are cylinder pressure, clearance, engine speed and oil supply pressure.

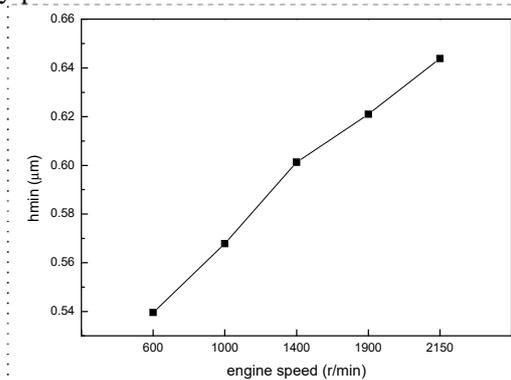


Figure 7. Effect of engine speed on the minimum oil film thickness.

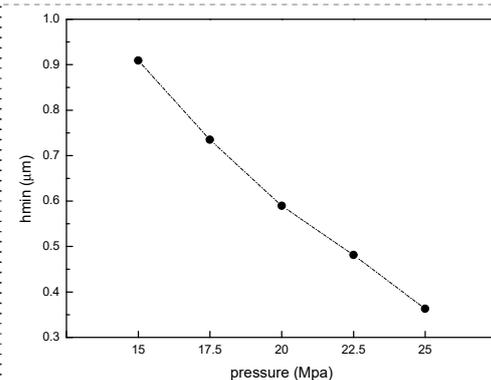


Figure 8. Effect of cylinder pressure on the minimum oil film thickness

As the relative rotation speed of the journal-bearing increases (figure 7), which is more conducive to the formation of the wedge-shaped oil film, the minimum oil film thickness increases with the increase of the rotational speed. As the cylinder pressure increases, the pressure of the oil film needs to be carried increases, and the minimum oil film thickness is correspondingly reduced.

The effect of cylinder pressure on the minimum oil film thickness is shown in figure 8. The increase of oil supply pressure is more conducive to the formation of oil film, so the minimum oil film thickness increases, but the excessive oil supply pressure will also cause the increase of hydraulic friction loss, which may cause changes in oil film flow and distribution, resulting in the reduction of the local oil film thickness.

From the perspective of the thickness of the oil film thickness (figure 9), as the clearance increases, the range of the shaft diameter of the bearing increases, and the minimum thickness of the oil film decreases gradually under the action of the explosion pressure, that is, with the service progresses undergoing, the bearing clearance increases due to the wear which will result in a decrease in oil film thickness, and the degree of reduction becomes more and more obvious, this law is consistent with the clearance influence on the maximum total pressure in section 4.2.

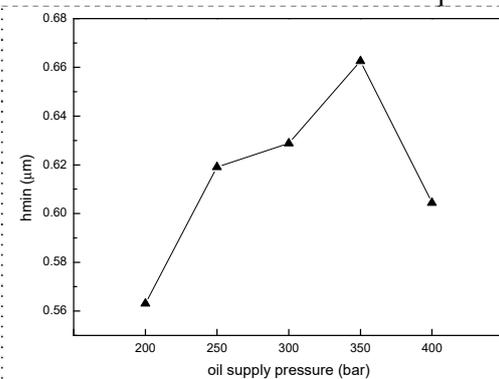


Figure 9. Effect of oil supply pressure on the minimum oil film thickness

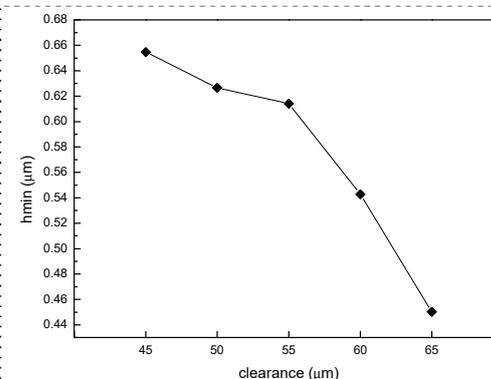


Figure 10. Effect of clearance on the minimum oil film thickness

5. Conclusion

Cylinder pressure and clearance are first and secondary factor of influence degree for bearing lubrication, oil supply pressure has the least impact.

The most effective way to improve bearing lubrication is to reduce the explosion pressure appropriately.

Acknowledgment

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