POLISH MARITIME RESEARCH Special Issue 2018 S2 (98) 2018 Vol. 25; pp. 30-34
10.2478/pomr-2018-0070

# MARINE FOUR-STROKE DIESEL ENGINE CRANKSHAFT MAIN BEARING OIL FILM LUBRICATION CHARACTERISTIC ANALYSIS

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**ABSTRACT**

*The Craig-Bampton modal synthesis method was used to establish the dynamic model of marine four-stroke diesel engine body and crankshaft. Based on the Greenwood / Tripp microlong contact theory considering the surface roughness and the generalized Reynolds equation considering the oil filling rate, the elastohydrodynamic lubrication model of the main bearing of the four - stroke diesel engine is found. At the rated speed, the lubrication performance of the main bearing is simulated and analyzed by the maximum dynamic pressure, the minimum oil film thickness and the friction power. The results show that the oil pressure of 4 # main bearing is the largest and the maximum oil film pressure is in the 4 # main bearing position. The friction load of 4 # main bearing is the largest. The average oil film thickness of 4 # main bearing is the smallest and the minimum oil film The thickness also occurred in the 4 # main bearing position; it can be seen 4 # bearing the most bad lubrication conditions.*

**Keywords:** Marine four – stroke diesel engine; Main bearing; Elastic hydrodynamic lubrication.

## INTRODUCTION

Marine four stroke diesel engine is usually used as the active power of the medium and small power ship or the generator set of the large ship. The ship in the ocean or inland coastal navigation process, usually in a trim and heel of the complex working conditions, so compared to land-based diesel engine, four stroke diesel engine crankshaft main bearing for ship lubrication reliability requirements more stringent. At present, the simplified method is usually used in the hydrodynamic lubrication of the main bearing, which does not take into account the deformation of the engine block and crankshaft, surface roughness and so on. With the progress of the complex work condition and the numerical simulation technology, the main influence factors of the main bearing of the modern marine diesel engine are taken into account, and the lubrication characteristics of the main bearing are judged more comprehensively.

Ebrat O [1] present a comprehensive formulation for the dynamics of a rotating flexible crankshaft coupled with the dynamics of an engine block through a finite difference elastohydrodynamic main bearing lubrication algorithm. Jun Sun [2] investigate the effect of deformation of the whole engine block on the hydrodynamic lubrication performance of main bearings. Cai, X. X [3] analysis the crankshaft bearing elastic fluid dynamic lubrication for a four-stroke four-cylinder internal combustion engine crankshaft main bearing, considering the influence of the deformation of the body. Lei Jilin [4] built a three-dimensional lubrication simulation model of the main bearing with AVL Excite Power Unit software, based on the elastic fluid dynamic lubrication theory. Wei, Lidui [5–6] put forward a two-stroke diesel engine main bearing thermo-elastic hydrodynamic TEHD mixed lubrication model, considering the deformation of flexible body. At present, there are few reports on the study of the mixed lubrication characteristics of marine four stroke diesel engine main bearings.

In this paper, based on the theory of elastohydrodynamic lubrication and the theory of structural dynamics, the lubrication simulation model of marine 6 cylinder diesel engine crankshaft bearing system is established. Under the rated speed, the lubrication performance of the main bearing is simulated and analyzed, which provides the basis for the optimal design of the bearing.

## MULTI BODY DYNAMICS MODEL OF CRANKSHAFT BODY

According to the Craig-Bampton modal synthesis method, the number of freedom degrees of flexible body and crankshaft structure is reduced. The relationship between the physical coordinates and modal coordinates is as follows:

$$x^b=\begin{pmatrix} x_i^b \\ x_r^b \end{pmatrix}=\begin{pmatrix} X^b & T^b \\ 0 & I \end{pmatrix}\cdot\begin{pmatrix} \alpha^b \\ x_r^b \end{pmatrix}=\phi\cdot\begin{pmatrix} \alpha^b \\ x_r^b \end{pmatrix} \tag{1}$$

The equation of motion of the body is,

$$M^b\cdot\ddot{x}^b+C^b\cdot\dot{x}^b+K^b\cdot x^b=I \tag{2}$$

Substituting equation(1) into equation (2), the body modal reduction motion equation is given as follows,

$$\overline{M}^b\cdot\begin{bmatrix} \ddot{\alpha}^b \\ \ddot{x}_r^b \end{bmatrix}+\overline{C}^b\cdot\begin{bmatrix} \dot{\alpha}^b \\ \dot{x}_r^b \end{bmatrix}+\overline{K}^b\cdot\begin{bmatrix} \alpha^b \\ x_r^b \end{bmatrix}=I \tag{3}$$

Considering the influence of the large range of rigid body motion rotation inertia crankshaft, crankshaft motion equation is reduced to,

$$\overline{M}^c\cdot q_a^c+\overline{C}^c\cdot q_a^c+\overline{K}^c\cdot q_a^c=\overline{Q}^c \tag{4}$$

Where, $x$ is the physical coordinates, subscript $r$, $i$ are reservations of degrees of freedom (including all load points and degrees of freedom) and internal degrees of freedom respectively, $\theta$ is the crankshaft rotation coordinates, $\alpha$ is the main modal coordinates (also known as modal participation factor, $q$ is a generalized displacement vector. Superscript $b$, $c$ represent the body and the crankshaft respectively. $M$, $C$, $K$ represent mass matrix, damping matrix and stiffness matrix respectively, and the upper line represents reduced physical quantities. $T$ is a static reduction matrix. $\varphi$, $\Phi$ are the coordinate transformation matrix of the body and the crankshaft. The force $F$ includes the external load vector, the oil film force and the asperity contact force.

## MAIN BEARING ELASTIC HYDRODYNAMIC IUBRICATION THEORY

Main bearing of crankshaft in the actual operation, can produce hydrodynamic pressure of oil film to support the crankshaft rotation. Local high pressure makes the oil film of lubricating oil viscosity increases sharply, as well as the engine block of significant local deformation. These effects will significantly change the geometry of the oil film area, which in turn affect the contact area pressure distribution at the same time. Therefore, in the elastic hydrodynamic lubrication, the sliding bearing oil film pressure distribution, distribution of oil film thickness and surface roughness, influence and elastic deformation distribution is a coupling problem.

### THE OIL FILM THICKNESS

In the main bearing lubrication characteristics analysis, the elastic deformation and surface roughness factors are considered, the main bearing oil film thickness is expressed as:

$$h(\theta)=h_{\min}(\theta)+\Delta h(\theta)+\delta h(\theta)+\sigma h(\theta) \tag{5}$$

Where, $h_{\min}(\theta)$ is the minimum oil film thickness under rigid bearing hypothesis. $\Delta h(\theta)$ is the difference between the oil film thickness and $h_{\min}(\theta)$, only with the geometry factors. $\delta h(\theta)$ is elastic deformation caused by the oil film. $\sigma h(\theta)$ is the lubrication oil film thickness produced by surface roughness. $\theta$ is for the oil filling rate.

### THE GENERALIZED REYNOLDS EQUATION

Main bearing lubricating oil film thickness in the process of work only a few microns, the influence of surface roughness on the lubrication performance need to be considered. So the elastic generalized Reynolds equation of hydrodynamic pressure lubrication are as follows [7]:

$$\frac{\partial}{\partial x}\left(\frac{1}{12\mu}h^3\theta\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{1}{12\mu}h^3\theta\frac{\partial p}{\partial z}\right)= \theta\frac{u_1+u_2}{2}\frac{\partial h}{\partial x}+h\frac{u_1+u_2}{2}\frac{\partial\theta}{\partial x}+\frac{\partial\ h\theta}{\partial t} \tag{6}$$

$p$ is the oil film pressure, $MPa$. $\mu$ is the oil dynamic viscosity, $Pa\cdot s$. $u_1$, $u_2$ are the journal, bearing the circumferential speed, $u_1=V$, $u_2=0$, m/s. Generalized Reynolds equation shows the relationship between the dynamic bearing force and the dynamic external load, which shows the relationship between the pad diameter, the pad width, the axial trajectory, the oil dynamic viscosity and the oil film pressure in time and space. Boundary conditions are extended boundary conditions.

### ASPERITY CONTACT MODEL

Greenwood / Tripp microlite contact theory is applied, bearing surface asperity contact pressure equation [8]:

$$P_b(h)=KE^*F(4-h/\sigma_s) \tag{7}$$

$P_b$ is the contact pressure; $K$ is the elastic contact factor; $\sigma_s$ is the integrated surface roughness; $F$ is the film thickness function; $E^*$ is the integrated elastic modulus.

When $(4-h/\sigma_s)\geq 0$, it is determined that the contact area is rough and in the mixed friction state. When $(4-h/\sigma_s)<0$, it is determined that the contact area is fully lubricated and in the fluid dynamic lubrication state.

## THE BODY – CRANKSHAFT – THE MAIN BEARING SIMULATION MODEL

Solidworks is used to build three-dimensional model of marine four-stroke diesel engine, including diesel engine body, crankshaft, main bearing, connecting rod and so on, as shown in Fig.1.

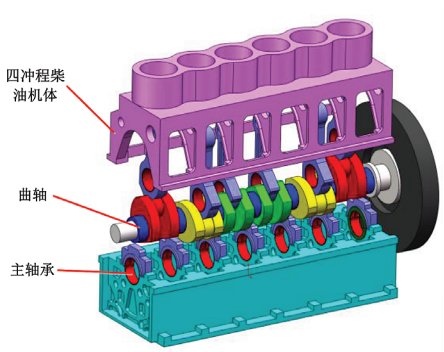


*Fig. 1. Three – dimensional model of marine four – stroke diesel engine*

The finite element model of the body-bearing is established by using the finite element software Abaqus, as shown in Fig. 2.

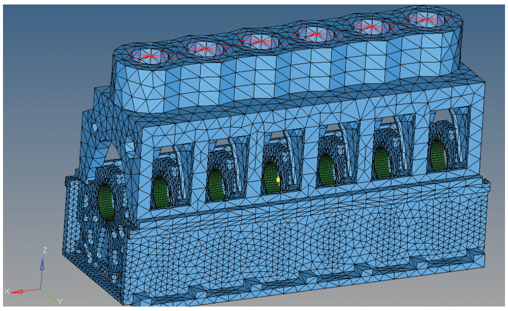

*Fig. 2. Finite element model of body – bearing*

The AVL-EXCITE SHAFTER MODELER module is used to build the crankshaft reduction dynamics model. The finite difference model of main bearing is established in ALV-EXCITE.The main bearing oil film surface mesh is divided into axial 17 × circumferential 97. The finite element model of diesel engine body is introduced into AVL-EXCITE by modal reduction. Crankshaft power reduction model and the main bearing finite difference model are combined, the marine four-stroke diesel engine body – crankshaft bearing oil film dynamic lubrication model was established using AVL-EXCITE module, as shown in Fig. 3.

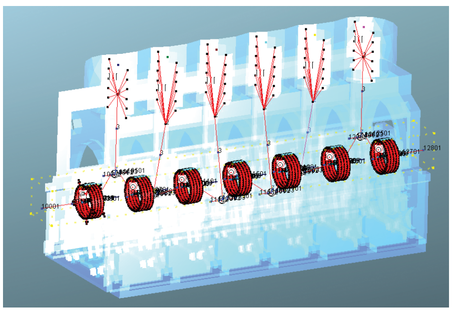

*Fig. 3. Marine four-stroke engine block-crankshaft-main bearing lubrication and multibody dynamics model*

## THE CRANKSHAFT MAIN BEARING MIXED LUBRICATION CHARACTERISTICS ANALYSIS

The simulation calculation, the explosion gas pressure piston, piston components, and the calculation in the inertial force of connecting rod load was applied on the crankshaft crank pin. The average reverse torque on the crank flywheel end are imposed. Crankshaft ignition order is 1-5-3-6-2-4. Each cylinder gas force curve as shown in figure 4.

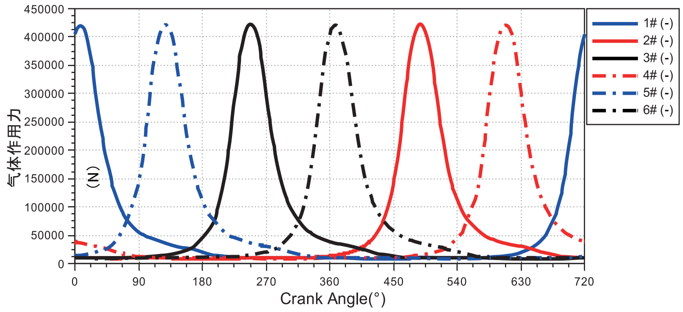


*Fig. 4. Each cylinder indicator diagram*

The specific structure parameters and working condition of the main bearing as shown in Tab. 1.

*Tab. 1. Main bearing operating parameter*

| Parameter | Value |
|---|---|
| Journal radius | 90 mm |
| Bearing radius clearance | 0.135 mm |
| Bearing width | 70 mm |
| Oil groove width | 8 mm |
| Oil groove angle | 220° |
| Bearing pad / Journal Surface roughness | 0.8/0.4 |
| Lubricating oil grades | SAE-15W-40 |
| Oil supply pressure | 0.45 |
| Oil supply temperature | 40° |
| Journal speed | 900 rpm |

The above parameters are input to the simulation model. Four stroke diesel engine main bearing oil film pressure, the minimum oil film thickness and friction power consumption data is obtained by AVL multi-body dynamics simulation iterative calculation.

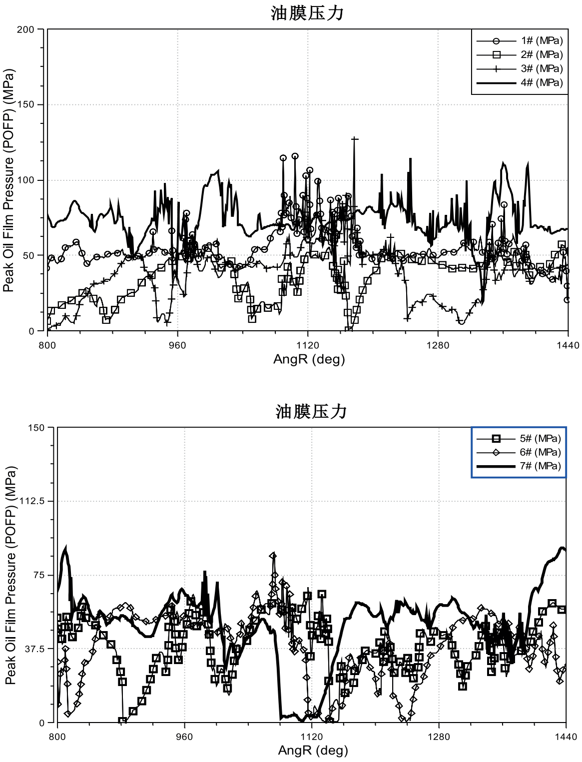


*Fig. 5. The peak bearing oil film pressure distribution*

As can be seen from Fig. 5., the average oil film pressure from large to small sort, 4#(72.9 MPa), 1#(50.8 MPa), 7#(50.3 MPa), 5#(39.7 MPa), 6#(39.6 MPa) > 3#(37 MPa) > 2#(34.6). It can be seen that the 4# main bearing average oil film pressure is the largest, reaching 72.9 MPa, and the maximum oil film pressure occurs in the 4# main bearing position, reaching 114.4 MPa. 1# and 7# main bearing average oil film pressure is close to the 4# main bearing.

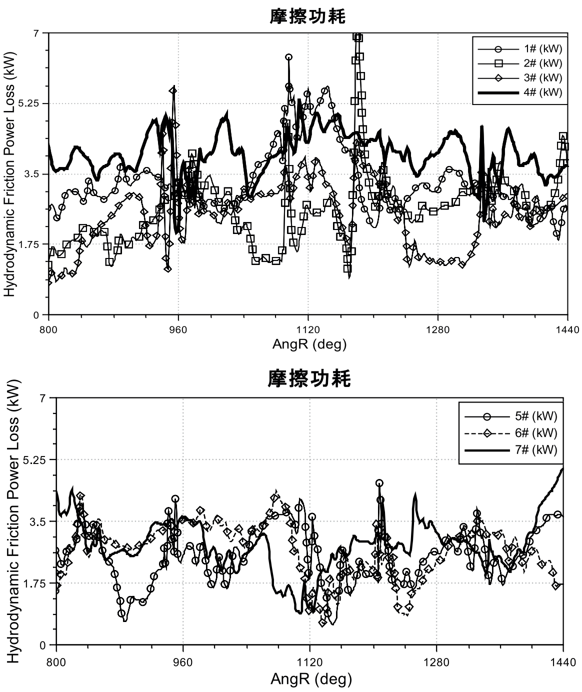


*Fig. 6. The bearing friction power*

As shown in Fig. 6., the order of the average frictional power consumption is 4#(4.27 KW)> 1#(3.2 KW)> 7#(3.0 KW)> 2#(2.78 KW)> 6#(2.7 KW) 5#(2.73 KW)> 3#(2.71 KW). It can be seen, 4 # main bearing oil friction power consumption of the largest, reaching 4.27 KW. 1 # and 7 # main bearing average oil film pressure is similar, only less than 4 # main bearing. The rest of the main bearing friction power consumption is not much difference.

As can be seen from Fig. 7., the average minimum film thickness from large to small sort, 4#(0.0049 mm) <1#(0.0073 mm) <5#(0.0085 mm) <3#(0.01 mm)、7#(0.01 mm) <6#(0.0114 mm) <2#(0.0127 mm). It can be seen that the 4 # main bearing average minimum film thickness is the smallest, reaching 0.0049 mm, and the minimum film thickness occurs in the 4 # main bearing position, reaching 0.0014 mm. 1 # and 5 # main bearing average minimum film thickness is similar, only more than 4 # main bearing.

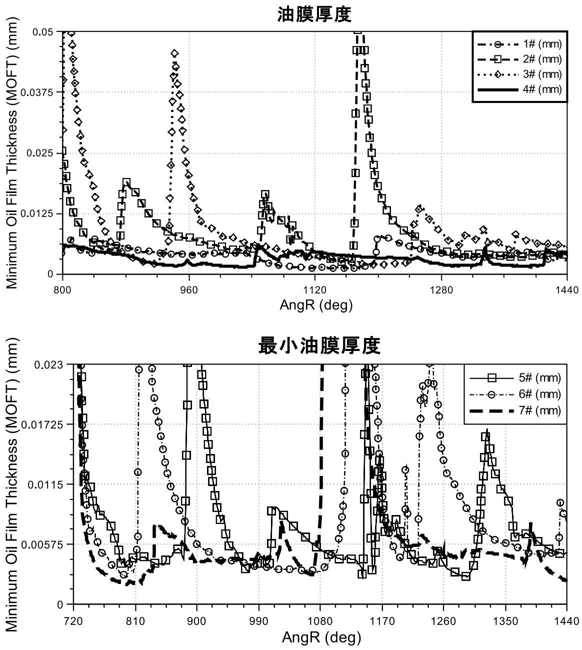


*Fig. 7. Each bearing minimal oil film thickness*

## CONCLUSIONS

Elastic hydrodynamic lubrication theory and structural modal reduction dynamics theory are applied. The roughness of the bearing body and crankshaft is considered. The lubrication simulation model of the 6-cylinder four-stroke diesel engine crankshaft-bearing was established considering the flexible deformation of the body and the crankshaft.

At the rated speed, the lubrication performance of the main bearing is simulated and analyzed by the maximum dynamic pressure, the minimum oil film thickness and the friction power. The results show that the oil pressure of 4 # main bearing is the largest and the maximum oil film pressure is in the 4 # main bearing position. The friction load of 4 # main bearing is the largest. The average oil film thickness of 4 # main bearing is the smallest and the minimum oil film The thickness also occurred in the 4 # main bearing position; it can be seen 4 # bearing the most bad lubrication conditions.

This paper provides a reference for the optimization design of the main bearing of 6-stroke diesel engine. According to the above results of the main bearing simulation, 4 # main bearing structural parameters, materials and lubrication characteristics should be focused on the optimization design.


**ACKNOWLEDGEMENTS**

This project is partially supported by the Fundamental Research Funds for the Central Universities (No. 3132015337).


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