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Effect of roundness and coaxiality error on the lubrication performance of ICE main bearings

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Abstract. Nowadays, with higher power density trend and lightweight design requirements in internal combustion engine, more factors are considered in bearing lubrication analysis to improve the lubrication performance and to decrease the risk of failure. The effect of roundness and coaxiality on the lubrication performance of a V-type diesel engine main bearings were investigated systematically and comprehensively using numerical simulation method based on elasto-hydrodynamic lubrication theory. The average flow model, the asperity contact theory and the multi-body dynamics based on component mode synthesis were used to improve simulation accuracy and efficiency. The results showed that out-of-roundness has an apparent negative effect on the lubrication performance of main bearings and the out-of-roundness wave height and the wave number should not exceed 2μm and 10 respectively. In addition, the lubrication performance increased slightly when the bearing house coaxiality tilt angle is the same as the journal load bending direction. The study has some guiding significance for improving the lubrication performance and tolerance requirements design of main bearings.

1. Introduction

Plain bearings are machine elements with widespread application, which are used in internal combustion engine (ICE) due to their simple structure and high load carrying capacity. Engine main bearing is one of the main sources of power loss in ICE, and its lubrication performance has an important impact on engine's operation. Nowadays, with higher power density trend and lightweight design requirements in ICE, more factors are considered in the lubrication analysis to improve lubrication performance and to decrease the risk of failure. Due to deviation of machining and assembly, the wear and environmental changes during work, etc., the journal and house of main bearing has some shape and position tolerance which may have an impact on the lubrication performance. The shape and position tolerance of main bearings mainly include roundness and coaxiality error because both journal and shell are cylindrical rotary parts.

Some work were dedicated to the study of the influence of journal out-of-roundness and misalignment on the bearing lubrication performance. D Vijayaraghavan [1] analysed the effect of out-of-roundness on the dynamic performance of a diesel engine connecting-rod bearing. The journal profiles of circular, elliptical, semi-elliptical and three lobe epicycloid were analysed and compared. D S Mehenny [2] theoretically analysed the influence of circumferential waviness of a rigid journal on the Lubrication of Dynamically Loaded Journal Bearings. The influence of lobe number and size on operating parameters such as maximum lubricant pressure and minimum film thickness is

examined. Xiang [3] analysed the effect of oval and tooth circularity errors of the journal on the lubrication performance of sliding bearings, which show that the tooth errors would bring a periodic fluctuation of lubricant film at the load bearing region and the profile of film pressures has a multi-peak distribution. Based on assumption of smooth surface and rigid bearing, P Maspeyrot [4-5] analysed the influence of journal misalignment on a big-end bearing lubrication performance including minimum film thickness, the friction torque and the axial flow. A comparison of the dynamic performance characteristics of two misaligned crankshaft bearings belonging to a four-stroke petrol engine was carried out by M Lahmar [6] which showed that the presence of even a small skew of the main journal axis can cause asperity contact at one of the bearing edges.

However, few studies have been done on the effect of roundness and coaxiality error of bearing house on lubrication performance of ICE main bearings. In addition, early lubrication calculation accuracy is not high because roughness and flow factor were not considered. In this paper, the effect of roundness and coaxiality error on the lubrication performance of a V-type diesel engine main bearings were investigated systematically and comprehensively using numerical simulation method based on elasto-hydrodynamic lubrication theory. The average flow model, the asperity contact theory and the multi-body dynamics based on component mode synthesis were used to improve simulation accuracy and efficiency. The study has some guiding significance for improving the lubrication performance and tolerance requirements design of main bearings.

2. Problem formulation and theory

The two-dimensional modified Reynolds equation is used as lubrication governing equation. Neglecting the centrifugal effects for an incompressible Newtonian lubricant the Reynolds equation can be expressed as in equation (1):

$$\frac{\partial}{\partial \theta}\left(\frac{1}{12\eta}\gamma\phi_\theta h^3 \frac{\partial p}{\partial \theta}\right) + \frac{\partial}{\partial x}\left(\frac{1}{12\eta}\gamma\phi_x h^3 \frac{\partial p}{\partial x}\right) = \frac{V_j - V_s}{2}\frac{\partial}{\partial \theta}\left(\gamma h + \gamma\sigma_s\phi_s\right) + \frac{\partial(\gamma h)}{\partial t} \qquad (1)$$

Where θ and x are circumferential and axial coordinates, respectively. η is the oil film viscosity, and γ is the fill ratio which describes the oil percentage in the gap between shell and journal. p and h are hydrodynamic pressure and oil film thickness, respectively. ϕ_θ and ϕ_x are circumferential and axial pressure flow factor, respectively. ϕ_s is the shear flow factor and σ_s is integrated surface roughness. V_j and V_s are circumferential velocity components of journal and shell.

According to Patir/Chang average flow model [7], the pressure flow factor and shear flow factor are determined when the surface direction parameter is 1. The total friction of bearing is composed of hydraulic friction force and summit friction force. The isotropic surface summit asperity contact pressure is determined using Greenwood/Tripp contact model [8]. Finite difference method and JFO mass conservation boundary are used to solved equation (1). The oil film thickness h is affected by elastic deformation and surface roughness of journal and shell which can expressed as in equation (2) :

$$h = h_m + \Delta h + h_\delta + h_\sigma \qquad (2)$$

Where h_m is the minimum oil film thickness under the assumption of rigidity, $h_m + \Delta h$ is the oil film thickness under the assumption of rigidity, h_δ and h_σ are changes in oil film thickness caused by elastic deformation and surface roughness, respectively. The condensation of degree of freedom method in the dynamic substructure reduction technique is used to improve the multibody dynamic calculation efficiency and the finite element model of crankshaft and bearing house are condensed using Guyan method.

3. Simulation model

A numerical simulation model of main bearing of a V-type diesel engine was established. Figure 1 shows the finite element model of bearing house and crankshaft. The condensation of degrees of freedom of bearing house and crankshaft retain the first 30 and the first 60 modes respectively. The

global parameters, nonlinear connection joint parameters and main bearing parameters were defined firstly. The combustion gas pressure as a function of crank angle was predicted using a cycle simulation code which the peak gas pressure is 22MPA. Inertia load and output torque were calculated by mass information, geometry information and combustion gas pressure. Using Gear prediction emendation multi-step algorithm and Newton-Raphson to solve flexible multi-body dynamic equations. Elastic deformation equations and modified Reynold equation were solved by iterative calculation using the finite element method and the finite difference method respectively. The node pressure is the boundary of the equilibrium equation, and the node displacement is the boundary of the Reynold equation. The global and bearing parameters are shown in table 1 and numerical methodology is shown in figure 2 which was based on a program named EXCITE, developed by AVL.
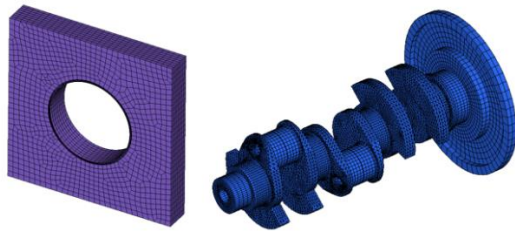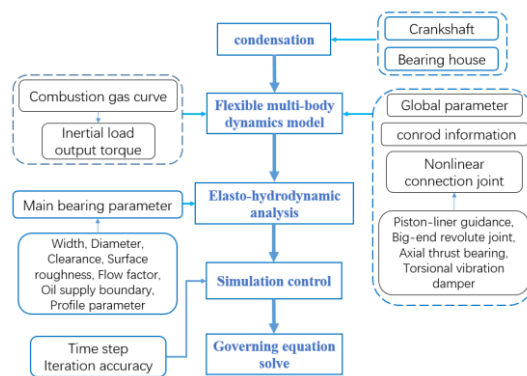

Figure 1. Finite element model.   **Figure 2.** Numerical methodology.

Table 1. Global and bearing parameter

Parameter name	Valve	Parameter name	Valve
Engine speed/(r/min)	4200	Ambient pressure/(MPA)	0.1
Bore/(mm) & Stroke/(mm)	110&110	Cavitation pressure/(MPA)	0.098
Bearing width/(mm) & diameter/(mm)	30&98	Supply pressure/(MPA)	0.5
Radial clearance/(μm)	35	Oil type	SAE-5W30
Journal/shell Young's modules/(MPA)	210000/206000	Groove position/(deg) & width/(mm)	330~30& 7
Journal/shell Poisson's ration	0.3/0.33	Bore position/(deg) & diameter/(mm)	270, 90&8
Journal/shell surface roughness/μm	0.1/0.3		

4. Results and discussion

4.1. Effect of roundness error

Due to the machine shake during machining process and the deformation of the workpiece, the journal and shell of the main bearing may form roundness error of tooth profile. Measurement data shows that some engine crankpins are machined with up to 21 circumferential lobes and some lobes have amplitudes even exceeding 5μm [2]. Bearing are subjected to cyclic circumferential load, so tooth roundness error could decrease lubrication performance which may lead to premature damage. The radial height change of the bearing surface caused by the roundness error can be expressed by the Fourier formula, as shown in equation (3):

$$\delta(\theta) = \frac{a_0}{2} + \sum_{n=1}^{n=k}(a_n \cos\theta + b_n \sin\theta) \tag{3}$$

$\delta(\theta)$ is the radial height change of different circumferential angle θ, a_n and b_n are the Fourier series. The frequency and amplitude of machine shake during machining process usually close to constant, so

the tooth profile is usually close to the regular wave shape shown in figure 3, which can be represented by few Fourier series. The height between wave peak and trough and wave number are two parameters of the waveform roundness error. Generally, the wave height is about 3μm and the wave number is between 5 and 50. Two sets of numerical simulations were carried out to investigated the influence of these two parameters separately, and each set includes five cases. In the first set, the wave number (n) is 20, and the wave height (h) varies from 0μm to 5μm. In the second set, the wave height is 3μm, and the wave number varies from 0 to 40. Details of the two sets are shown in table 2.

Table 2. The details of set 1 and set 2.

	Set 1, n=20						Set 2, h=3μm				
	Case1	Case2	Case3	Case4	Case5		Case1	Case2	Case3	Case4	Case5
$h/$μm	0	2	3	4	5	n	0	10	20	30	40

The peak total pressure (PTP), the minimum oil film thickness (MOFT) and the load uniformity were selected as evaluation index of lubrication performance in this paper. The most prominent types of wear in diesel engine bearings are abrasive wear, adhesive wear and surface fatigue wear [9]. The abrasive wear is mainly caused by oil contamination with foreign particles and wear products which not been considered here. The adhesive wear is caused by oil film breakdown which results in poor lubrication conditions like boundary lubrication even dry friction which can be reflected by the minimum oil film thickness. The surface fatigue wear is mainly influenced by the total pressure, and too large total pressure will accelerate surface fatigue wear. The load uniformity includes circumferential and axial uniformity, which is important for bearing reliability.
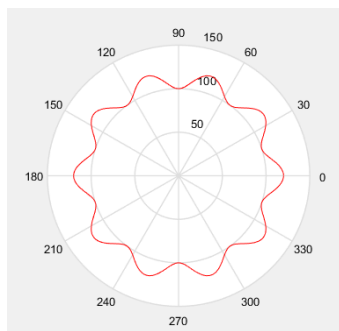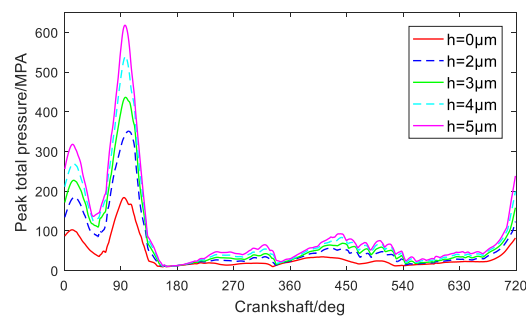

Figure 3. Regular wave shape.

Figure 4. Variation in the PTP with different height.

4.1.1. Effect of the weight height. The first main bearing has the worst lubrication performance, so the performance of the first main bearing was analyzed here. The PTP of the bearings with roundness error is much higher than cylindrical bearing at the ignition moment of first two cylinders (about 10 degrees and 100 degrees in cycle) shown in figure 4. The PTP increased significantly with increasing wave height and it is even higher than 300MPA when the height is greater than 3μm.
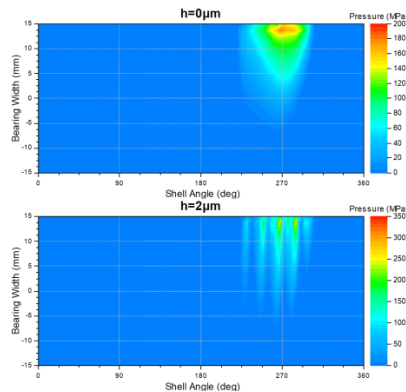
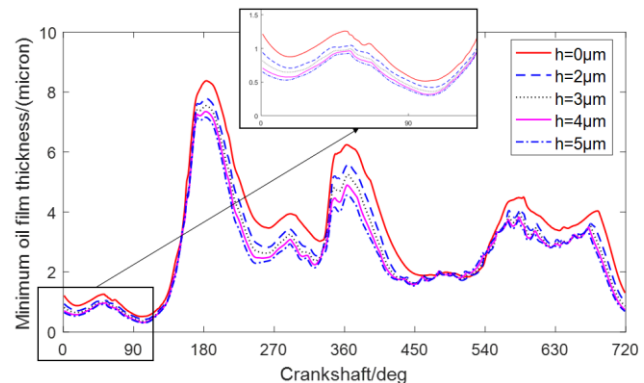**Figure 5.** The total pressure distribution with *h*=0μm and *h*=2μm.

Figure 6. Variation in the MOFT with different height

The figure 5 shows the total pressure distribution of the cylindrical bearing and the bearing with roundness error (*h*=2μm) at 100° in cycle. It indicates that the wave trough area was not loaded, and the wave peaks area was subjected to most of the load, so the loaded area of bearing with roundness error significantly reduced and the total pressure distribution is multi-peak which approximate strip. The total pressure in wave peak area is very high, and it even exceeded the limits in some regions. As the wave height increase, the MOFT decreased by different extent within the cycle shown in figure 6. The MOFT is less than 1μm within 100 degrees of cycle, so the film thickness ratio is less than 1 and the main bearing is in the boundary lubrication condition which will reduce the bearing efficiency and reliability. In addition, the load density in wave peak area is extremely high and asperity contact occurs in peak area which decreased the lubrication performance. The total pressure distribution of different roundness error bearings at 100 degree in cycle are shown in figure 7. As the wave height increase, the total pressure at the wave peak increased significantly and the area subjected to load reduced, so the circumferential load uniformity decreased obviously with the wave height.
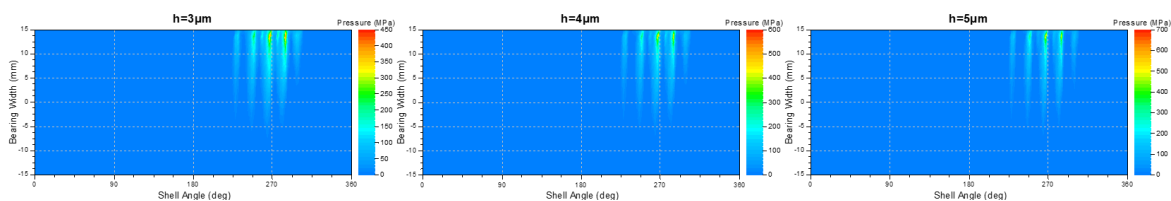

Figure 7. The total pressure distribution with weight equals to 3, 4 and 5μm.

In conclusion, the roundness error of bearing has a significant negative impact on lubrication performance of main bearing. As the wave height increase, the peak total pressure increased sharply and the minimum oil film thickness decreased, so the risk of adhesive wear and surface fatigue wear increased with wave height. In addition, the circumferential load uniformity decreased obviously with the wave height which reduce the bearing reliability.

4.1.2. Effect of the wave number. The PTP of bearings with roundness error is much higher than that of cylindrical bearing within 150 degree in cycle and it increased significantly with the wave number shown in figure 8. The reason for the increase of the PTP is that the metal-to-metal contact occurs in bearing peak region when the wave number is greater than 10, so the peak asperity contact pressure (PACP) increased sharply with the number shown in figure 9. Moreover, the PACP even greater than the limit value of 200MPA when the number is greater than 20.
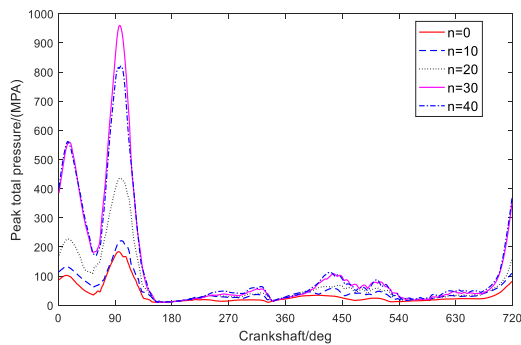
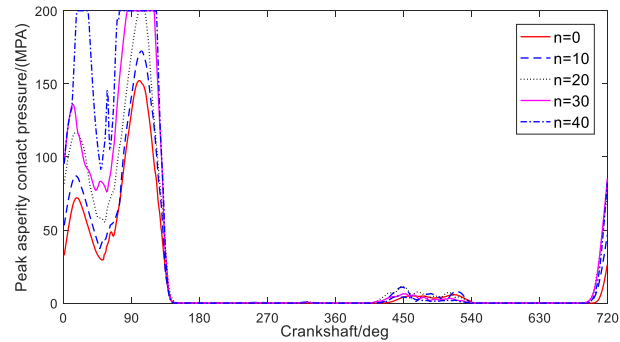**Figure 8.** Variation in the PTP with different number.

Figure 9. Variation in the PACP with different number.

The MOFT decreased with the wave number within 150 degree shown in figure 10 which is corresponding to the PACP. When the wave number is greater than 40, the MOFT is 0 in some moment which mean the lubrication condition is dry friction and the bearing reliability will greatly reduce. The enlargement of the total pressure distribution of bearings with different wave number roundness error at 100 degree in cycle are shown in figure 11. It shows that the area subjected to load decreased sharply with the wave number and the peak total pressure increased significantly. When the wave number is greater than 10, the circumferential load is in the shape of multi-peak and the circumferential load uniformity decreased obviously with the wave number.

In conclusion, the roundness error wave number has a significant negative effect on main bearing lubrication performance in terms of the peak total pressure, the minimum oil film thickness and the load uniformity. When the wave number is greater than 10, the bearing reliability and performance decrease significantly so the wave number should not exceed 10.
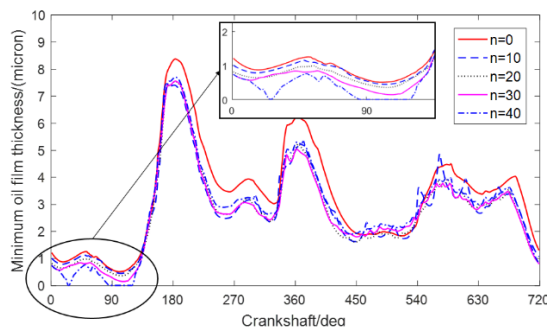

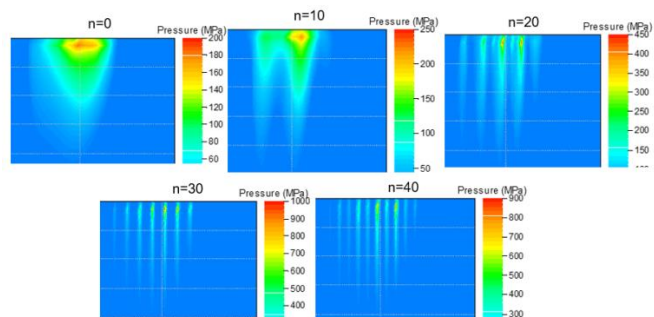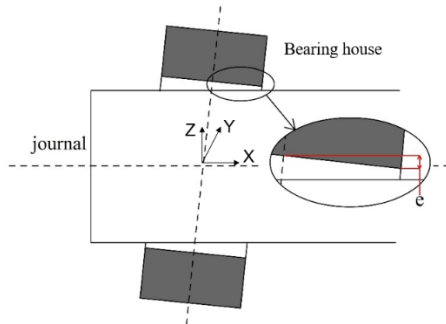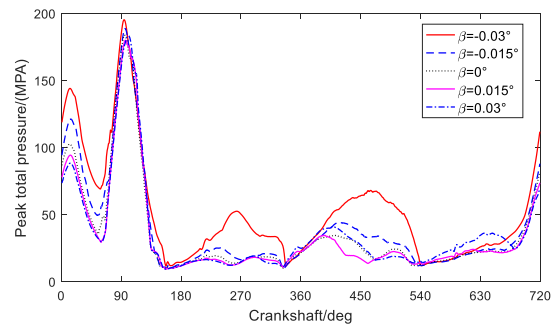**Figure 10.** Variation in the MOFT with different number.

Figure 11. The enlargement of the total pressure distribution of bearings with different number.

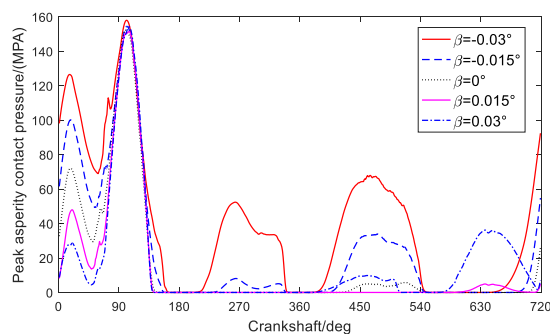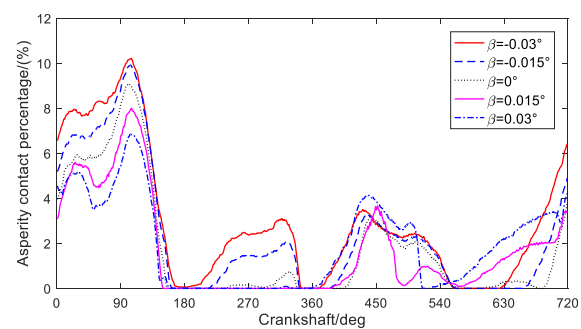*4.2. Effect of coaxiality error*

The main journal is bent due to load during operation, resulting a higher pressure at the edge of the bearing. If the main bearing house has a certain coaxiality error due to machining assembly error, the coaxiality error causes the bearing clearance at different axial position to be different, which may increase the asperity contact and total pressure in some bearing areas, resulting in severe friction and wear of bearing. Therefore, the influence of coaxiality error on main bearing lubrication performance was analysed in this part. Five cases of numerical simulations with different bearing house tilt angle were performed. The tilt angle (β) varies from -0.03° to 0.03° which is positive in the clockwise direction around the Y-axis, and e is the maximum bearing clearance change of each case shown in figure 12. The details of each case are shown in table 3.

Table 3. The details of each case.

	Case1	Case2	Case3	Case4	Case5
$\beta/°$	-0.03	-0.015	0	0.015	0.03
$e/\mu m$	7.85	3.93	0	3.93	7.85

Figure 12. Bearing house tilt coaxiality error.

Figure 13. Variation in the PTP with different tilt angle.

The PTP of each bearing with different tilt angle are shown in figure 13. When the tilt angle is negative, the PTP increased significantly in the ignition moment of the left side cylinder like 10 degree, 250 degree and 480 degree. The reason of the phenomenon above is that the left cylinder piston moves in Z direction, so the asperity contact occurs in the bearing edge area in ignition moment. By contrast, when the tilt angle is positive, the tilt direction is consistent with the crankshaft journal bending direction, so the area subjected to load increased and the PTP decreased at 10 degree moment in cycle. Therefore, as the tilt angle increase from negative to positive, the peak asperity contact pressure (PACP) and the asperity contact percentage (ACP) decreased steadily within 100 degree in cycle shown in figure 14 and figure 15. In addition, both the PACP and the ACP increased slightly at ignition moment of the left second and third cylinder when the tilt angle is 0.03°.

Figure 14. Variation in the PACP for various tilt angle.

Figure 15. Variation in the ACP for various tilt angle.

Affected by the above factors, the average peak total pressure (APTP) decreased sharply with tilt angle firstly, reaching its lowest value when tilt angle is 0.015° (drop by 4.5% compared to no tilt bearing), then increased slightly with tilt angle, which is approximately a concave quadratic curve shown in figure 16. As the tilt angle increase from negative to positive, the MOFT increased within 90 degree in cycle from about 0.8μm to 1.6μm shown in figure 17, so the lubrication condition change from boundary lubrication condition to mixed lubrication condition which will raise the lubrication efficiency.
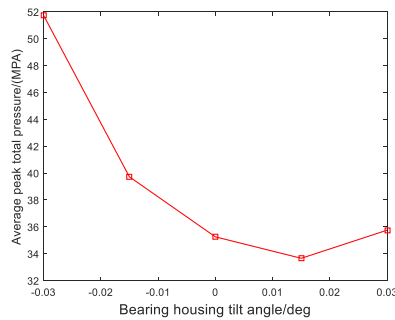
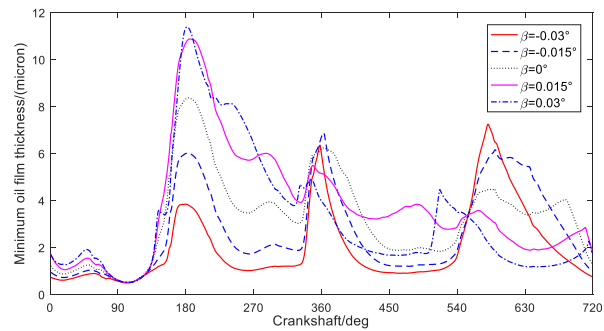**Figure 16.** Variation of the APTP for various tile angel.

Figure 17. Variation of the MOFT for various tile angel.

The figure 18 shows the total pressure distribution of main bearings with different tilt angle at ignition moment of left first cylinder (about 10 degree in cycle). It indicates that the area subjected to load increased with the tilt angle and the peak total pressure decreased with the tilt angle. In addition, the central of load translates from edge region to central with the increase of tilt angle, so the edge asperity contact pressure dropped with the tilt angle. The average total pressure (ATP) of each axial section are shown in figure 19 which indicates that the ATP increased with tilt angel at left side of bearing and it decreased with tilt angle at right side, so the load percentage in central area increased and the axial load uniformity raised with the tilt angle.

In conclusion, when bearing house tilt angle is opposite to the journal bent direction, the lubrication performance decreased significantly, however, the performance increased slightly when the tilt angle is the same as the bend direction. Therefore, for V type 6 cylinder diesel engine, the first main bearing house and the fourth main bearing house could be tilted by a small angle clockwise and counterclockwise, respectively. In addition, the tilt angle should not be too large which is recommended to be within 0.02°.
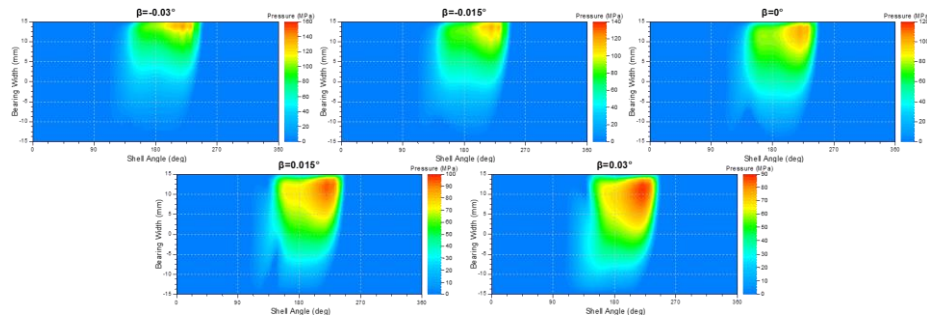

Figure 18. The total pressure distribution of bearings with different tilt angle.
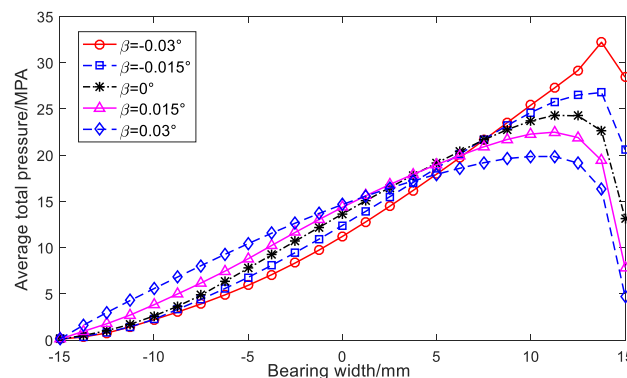

Figure 19. The ATP of each axial section of bearings with different tilt angle.

5. Conclusion

Out-of-roundness error has an apparent negative effect on the lubrication performance of the main bearings. Lubrication performance such as the peak total pressure, the minimum oil film thickness and the circumferential load uniformity decreased significantly with the wave height and the wave number. The reliability of main bearings reduced significantly when the roundness error wave number is greater than 10 and the wave height is greater than 2μm, which can be used as a basic machining tolerance limit requirement. When bearing house coaxiality error tilt angle is opposite to the journal load bending direction, the main bearing lubrication performance decreased significantly, however, the performance increased slightly when the tilt angle is the same as the bending direction. Therefore, for a V type six-cylinder diesel engine, the first and the fourth main bearing houses can be tilted a small angle clockwise and counterclockwise to improve lubrication performance, respectively.

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