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Numerical Analysis and Study of Crankshaft System Hydrodynamic Lubrication and Dynamic Coupling System

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Abstract: Using finite element method and multi-body dynamics method, this study established an engine digitalized virtual prototype embraced tribological characteristics in virtual prototype and MATLAB platform, and introduced Elastohydrodynamics theory into the system, wrote program with difference equation method, and solved the Reynolds Equation through shafting-bearing self-feedback control system, obtained the crankshaft main bearing reaction force which considered the system's oil film dynamic lubricating friction behaviour, and also obtained data of main bearing's load capacity, path of axle center and the minimum thickness of oil film. Though analysis, it is found that engine crankshaft system's oil film dynamic lubricating friction has certain impact on system's dynamic characteristics, where using hydrodynamic lubrication coupling multi-body dynamic system for simulation could better reflect the engine's actual working conditions, therefore the oil film dynamic lubricating friction behaviour of the system should be considered in dynamic simulation calculations.

Keywords: hydrodynamic lubrication, coupling, dynamics, numerical simulation.

1 Introduction

Spurred by the more and more intense market competition, the major car companies feel more pressure on the engine products' development cycle, upgrading and updating. Therefore more and more attention is paid on engine system's numerical simulating technology, expecting to complete the designing of products, virtual testing, analysis and manufacture, as well as program improvement under digitalized means-based conditions [1,3]. With the analysis and research on modern engine digitalized simulation getting more in-depth, the coupling problem of tribological and dynamic characteristics of the engine system needs to be resolved urgently; and the dynamic and kinematic transitive relation and boundary conditions among components are important data affecting the accuracy of simulation analyze results. During the digitalized simulation analysis, the dynamic and kinematic transitive relation and boundary conditions among components are especially important data affecting the correctness of simulation analyze results. And as an important dynamic behavior and boundary condition, tribological system behavior commonly exists in major systems of engines; its features will undoubtedly affect the accuracy of the engine system's digitalized simulation result [4,5]. However, in the engine digitalized simulation analysis process, the friction behavior is often simulated with simple mathematical models and defined with empirical values, which will lead to error between simulation analyze result and the actual result. Therefore it is necessary to carry out systematic study on engine dynamic system's tribological behavior in order to better guide the engine digitalized design work. As the pivot of engine, crankshaft system's components and body's dynamic characteristics are not only related to the system's dynamic characteristics, but also related to the tribological characteristics among them. The main bearing oil film plays a role in supporting crankshaft and carrying the pressure load of the cylinder, which is a very important part of engine crankshaft system's dynamic analysis; its dynamic lubrication characteristics directly affect the dynamic coupling relation between crankshaft

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system and the frame [6,7]. However, traditional analysis usually ignores the influence of hydrodynamic lubrication, and simplifies the main bearings to constraint bearings or linear bearings to analyze, thereby isolate the tribological problems and system dynamic problems for study. Because of the complexity of the load changes of actual shafting, the coupling effect between the shaft and bearing should not be ignored, thus this kind of isolated simplified analysis has considerable error. This paper takes crankshaft system as study object, and engine flexible virtual prototype model and MATLAB mathematic calculation software as simulation calculation platform, carries out numerical calculation analysis on crankshaft shafting and sliding bearing's hydrodynamic lubrication and dynamic coupling behaviors; and predicts the engine crankshaft system's dynamic and kinematic characteristics fairly accurately, allowing a more accurate evaluation of engine dynamic characteristics and vibration levels.

2 Crankshaft shaft - bearing coupling dynamic model

The first thing of studying the crankshaft and the body's hydrodynamic lubrication effect using numerical analysis and virtual prototype is to establish the crankshaft system's shaft-bearing flexible virtual prototype model. In crankshaft systems, crankshaft is undoubtedly the pivot of the engine. When the engine is working, the elastic vibration deformation of crankshaft directly affects the filmatic bearing's loading and movement. Meanwhile, the main bearing's bending moment caused by crankshaft's elastic deformation would affect the body and the cylinder's loading. Therefore, this study uses flexible crankshaft while constructing the crankshaft system virtual prototype. Figure 2.1 presents the crankshaft's finite element model.

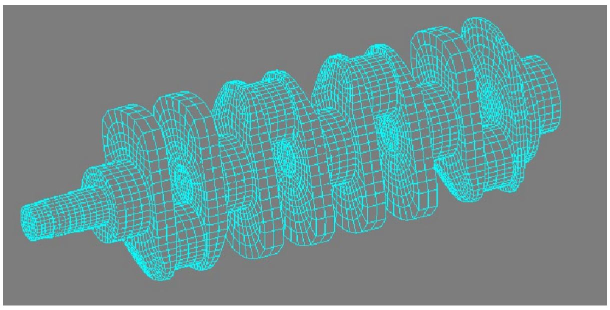

Fig. 2.1 Crankshaft finite element model

Considering that the flywheel in the system is also low stiffness component with lower connatural frequency, it should not be treated as rigid body simply when analyzing crankshaft system's vibrating characteristics, therefore infinite element model for flywheel is also constructed. In past multi-body dynamic models, connecting rod is usually simplified as rigid component with big and small head qualities. However due to the uneven quality distribution of connecting rod, such simplification would definitely affects the correctness of the model. Therefore, same with crankshaft, connecting rod is defined as flexible component. Other components are all defined as rigid centralized quality. This study uses MSC.PATRAN to divide each component to finite element grid in hexahedral units, and uses MSC.Nastran to solve the free orthotropic modality.

When introducing finite element model into multi-body dynamic software for virtual prototype simulation, mode synthesis method is usually used. For components with complex structure, to reflect their dynamic characteristics accurately, finite element model with large amount of degrees of freedom needs to be constructed; hence the solving is very large-scale. The advantage of mode synthesis method is using a small number of degrees of mode freedom to describe components' macroscopic deformation, and encapsulate components' connatural features retained by modal truncation in the calculation result of modal analysis. The finite element model of the flexible crankshaft, the flywheel and the connecting rod mentioned in this chapter are all introduced into the virtual prototype model using mode synthesis technology. Figure 2.2 and 2.3 show the finite element models of the flywheel and the connecting rod respectively.

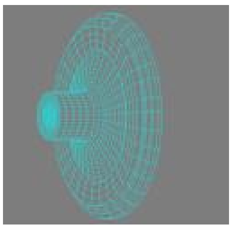

Fig. 2.2 Flywheel finite element model

Fig. 2.3 Connecting rod finite element model

By determining kinematic pairs according to actual kinematic relation, and simplifying each kinematic pair to ideal constraints in multi-body dynamic software MSC/ADAMS, a complete kinematic structure could be constructed. The main bearing constraints in the structure are replaced by oil film force, which is obtained by solving the Reynolds Equation. Figure 2.4 represents the virtual prototype model of this crankshaft system.

In this virtual prototype model, MSC/ADAMS Solver is used for solving and analysis. The solver constructs

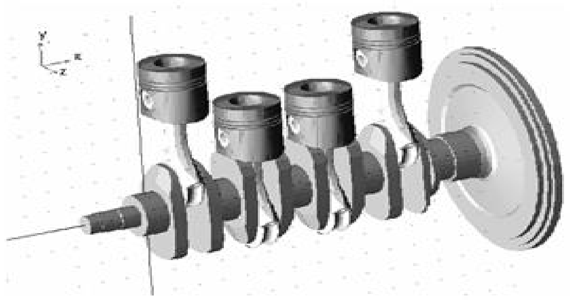

Fig. 2.4 Crankshaft three-dimensional model

kinematic constraint equations for multi-body system in the form of a set of non-linear equations:

$$C(q,t) = 0 \tag{2.1}$$

In which, $q = \left[q^{1T} q^{2T} \ldots q^{n_b T}\right]^T$ are the system's generalized coordinates, n_b is the number of the generalized coordinates; and $C = [C_1 C_2 \ldots C_{n_c}]^T$ is the number of constraint equations. For the i -th flexible body or rigid body, list Lagrange Equation in following form:

$$\frac{d}{dt}\left(\frac{\partial K}{\partial \dot{q}^i}\right)^T - \left(\frac{\partial K}{\partial q^i}\right)^T + C_q^{iT} \lambda = Q^i \tag{2.2}$$

Where, K is the system's kinetic energy, q^i is the generalized coordinate describing the system, Q^i is the generalized force including the generalized force caused by unit flexibility deformation and the generalized force caused by additional loads (including loads among units), λ_i is Lagrange multiplier array, and C_q^{iT} is kinematic constraint. For the series of non-linear algebraic equations above, ADAMS solves them using amended Newton-Raphson iterative algorithm.

3 Establishment of main bearing oil film mathematic model

The crankshaft main bearings are radial cylinder filmatic bearings using dynamic lubrication; and the main bearing oil film perform supporting function for the crankshaft. Using cylindrical filmatic bearings for simulating the crankshaft system's main bearing constraints and hydrodynamic model for calculation could greatly improve the model's accuracy.

3.1 Operating principle of hydrodynamic lubricating friction

In order to calculate the load capacity of the lubricating film, its pressure distribution needs to be calculated. The basic content of all kinds of hydrodynamic lubrication calculation is the applying and solving of a special form of Navier-Stokes Equation – Reynolds Equation. The general form of Reynolds Equation is:

$$\begin{aligned}\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \\ \frac{\partial}{\partial x}\left(\frac{\rho U h}{2}\right) + \frac{\partial}{\partial z}\left(\frac{\rho W h}{2}\right) - \rho V + h\dot{\rho}\end{aligned} \tag{3.1}$$

Where, U,V,W are the relative units of the velocities on x,y,z directions respectively.

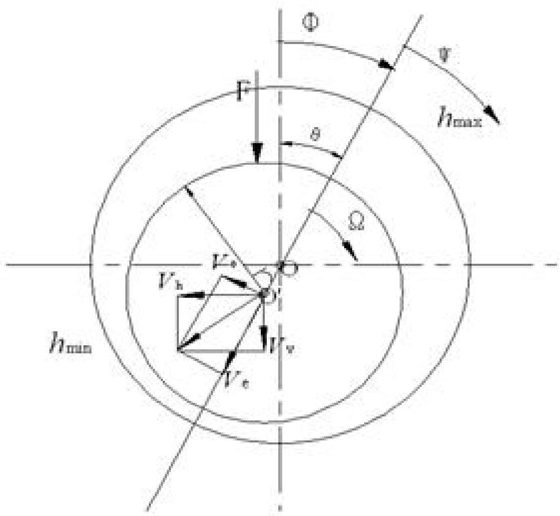

Fig. 3.1 Radical bearing diagram

For the radical bearing presented in Figure 3.1, $x = r\Phi$ indicates the circumferential direction coordinate, y is the radical direction coordinate, and z is the axial coordinate. Assume the bearing journal moves around its center with angular velocity of Ω , horizontal shift velocity of V_h and vertical shift velocity of V_v . Thereby, U,V,W in equation (3.1) could be expressed as:

$$U = r\Omega + V_v \sin\Phi - V_h \cos\Phi$$
$$V = -V_v \cos\Phi - V_h \sin\Phi$$
$$W = 0$$

Substitute them into Equation (3.1) and considering $\frac{V_v}{r}, \frac{V_h}{r} << \Omega$, following equation is obtained:

$$\begin{aligned}\frac{1}{r^2}\frac{\partial}{\partial \Phi}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial \Phi}\right) + \frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) \\ = \frac{\Omega}{2}\frac{\partial(\rho h)}{\partial \Phi} + \rho\left(V_v \cos\Phi + V_h \sin\Phi\right) + h\dot{\rho}\end{aligned} \tag{3.2}$$

When using the velocity on the eccentric direction V_e and the velocity of the axle center moves around the bearing's center V_θ to indicate the axle center velocity, the above equation could be transformed to:

$$\frac{1}{r^2}\frac{\partial}{\partial \Phi}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial \Phi}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial z}\right) = \frac{\Omega}{2}\frac{\partial(\rho h)}{\partial \Phi}+\rho\left(V_e\cos\phi+V_\theta\sin\phi\right)+h\dot{\rho} \tag{3.3}$$

3.2 Dimensionless form of the Reynolds Equation

Assuming the operating temperature is 85 ^oC , and the viscosity coefficient is a constant. Usually the analytical calculation of oil film dynamic pressure bearings is carried out in dimensionless form. Using dimensionless form for calculation could simplify the calculation equation, and prevent the value of the variables in the equation from oversize or too small as far as possible, so that error doesn't appear in the process of submitting to calculation. Reynolds Equation in the form of Equation (3.3) uses dimensionless form to represent the variables in the equation.

$$\lambda = \frac{z}{l/2}\ ;\quad H = h/c\ ;$$

$$h = c + e\cos\phi \ P = p/p_0 \ \ p_0 = 2\Omega\mu/\psi^2$$

$$\left.\begin{array}{l}\bar{V}_v = V_v/(c\Omega), \bar{V}_h = V_h/(c\Omega)\\ \bar{V}_e = V_e/(c\Omega), \bar{V}_\theta = V_\theta/(c\Omega)\end{array}\right\}$$

$$\frac{\partial}{\partial \phi}\left(H^3\frac{\partial P}{\partial \phi}\right)+\left(\frac{d}{l}\right)^2\frac{\partial}{\partial \lambda}\left(H^3\frac{\partial P}{\partial \lambda}\right) = 3\frac{\partial H}{\partial \phi}+6\left(\bar{V}_e\cos\phi+\bar{V}_\theta\sin\phi\right) \tag{3.4}$$

In the above equations: h is the thickness of the oil film, z is the bearing axial coordinate, p is the oil film pressure, r is the bearing radius, c is the bearing radius interval, and $\psi = c/r$; p_0 is the relative pressure, with the value of $2\Omega\mu/\psi^2$, and e is the eccentricity, θ is the angle of displacement.

The oil film pressure could be obtained by solving the above equation, and the oil film reaction force can be obtained through integration:

$$\left.\begin{array}{l}F_x = f_x(e,\theta,V_e,V_\theta) = \int_{\phi_1}^{\phi_2}\int_{-l/2}^{l/2} p(\phi,\lambda)dz\sin\phi d\phi\\ F_y = f_y(e,\theta,V_e,V_\theta) = \int_{\phi_1}^{\phi_2}\int_{-l/2}^{l/2} p(\phi,\lambda)dz\cos\phi d\phi\end{array}\right\} \tag{3.5}$$

Where, l is the bearing width. And perform following coordinates changes to Equation (3.5):

$$\left.\begin{array}{c}e = \sqrt{x^2+y^2}\\ \theta = \arctan(y/x)\\ V_e = \dot{x}\cos\theta+\dot{y}\sin\theta\\ V_\theta = (\dot{x}\sin\theta-\dot{y}\cos\theta)/e\end{array}\right\}$$

In which x,y and z are the crankshaft main bearing's horizontal, vertical and axial coordinates respectively. Then Equation (3.5) could be transformed to:

$$\left.\begin{array}{l}F_x = f_x(x,y,\dot{x},\dot{y}) = \int_{\phi_1}^{\phi_2}\int_{-l/2}^{l/2} p(\phi,\lambda)dz\sin\phi d\phi\\ F_y = f_y(x,y,\dot{x},\dot{y}) = \int_{\phi_1}^{\phi_2}\int_{-l/2}^{l/2} p(\phi,\lambda)dz\cos\phi d\phi\end{array}\right\} \tag{3.6}$$

In the above equation, $x, y, \dot{x}, \dot{y}$ are the axle center movement parameters; they can be given by the crankshaft system's virtual prototype simulation calculations. And then, through user-defined modules and ADAMS virtual prototype combined simulation in MATLAB, the dynamic lubrication main bearing's load capacity can be obtained.

3.3 Boundary conditions of oil film

When using MATLAB for integral calculation to solve the pressure distribution of Reynolds Equation's, pressure boundary conditions must be applied to determine the integration constants, where the selection of different boundary conditions would directly affect the accuracy of the solving result. For cylinder bearings of 360°, convergent and divergent oil wedges may exist at the same time, therefore there are several different assumptions about the pressure boundary conditions.

The first kind of boundary condition is the Sommerfeld boundary condition. This boundary condition considers that there is complete oil film in the whole oil film interval, with pressure distribution as shown in Figure 3.2. Positive pressure forms in the convergent area, while negative pressure forms in the divergent area, and the pressure distribution is anti-symmetric, i.e. pressure is 0 at the maximum interval $h_{\max}$ and minimum interval $h_{\min}$. Thus the pressure is periodic function at circumferential direction:

$$p(\Phi) = p(\Phi+2\pi) \tag{3.7}$$

Obviously, error of this assumption is large. The actual oil film could not bear continuing negative pressure, and lubricating oil film can only bear shock wave of higher load or very small continuing negative pressure. Therefore it is not possible for complete oil film to be existed in negative pressure area. However, since Sommerfeld boundary condition could solve pressure distribution conveniently, it can be used for the qualitative analysis of lubrication problems.

Another boundary condition is Reynolds boundary condition assumption which is more reasonable. It considers the ending edge of complete oil film is

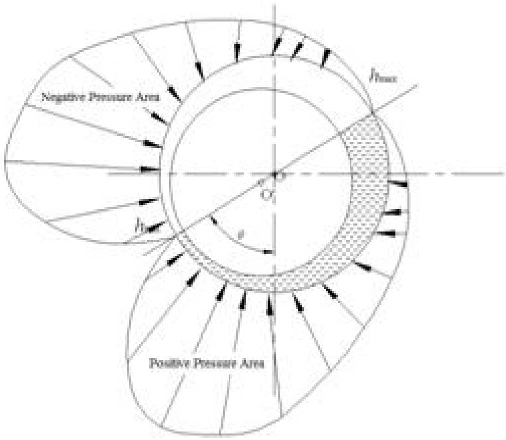

Fig. 3.2 Cylinder bearing pressure distribution

determined by the Reynolds boundary condition $p = p_a$, and

$$\frac{\partial p}{\partial \Phi} = 0 \tag{3.8}$$

The oil film starting point of this boundary condition is at the maximum interval $h_{\max}$, and $p = p_a$. The ending point of the oil film is determined by the natural break of the film, its location is at some point in the divergent area after the minimum interval, and the point meets the two conditions in equation (3.8) at the same time.

Reynolds boundary condition is the closest to actual condition, however its oil film ending point must be determined by calculation and the calculation precision is high, while the model is relatively complex, and each time step requires repeating iterations of large amount of calculation. And since the iteration process requires the time steps to be very small, the number of time steps in one work cycle of the engine would increase. Hence for this method, the biggest challenge for current software and hardware conditions is the calculation size. Meanwhile when using half-Sommerfeld boundary condition method to calculate, although the accuracy would be affected, this boundary condition is already quite close to actual condition and the calculation size is much smaller. Therefore using this method would save calculation time and improve the prediction efficiency. For the analysis in this study, the calculation of oil film pressure makes use of half-Sommerfeld boundary condition:

$$\left.\begin{matrix} when\Phi = \theta and \theta + \pi, & p = 0 \\ when\Phi = \theta \sim \theta + \pi, & p \geq 0 \end{matrix}\right\} \tag{3.9}$$

And when $\Phi = \theta + \pi \sim \theta + 2\pi$, $p = p_a$

4 Sub-system Models

The crankshaft virtual prototype model is constructed with MSC/ADAMS, the pressure distribution based on crankshaft system's virtual prototype model and the solving of Reynolds Equation for oil film dynamic lubricating bearings is a self-feedback control system. Meanwhile, the Reynolds Equation for dynamic lubrication bearings is an elliptic partial differential equation, which cannot be solved with analytical methods in ordinary conditions. Thus using efficient and precise mathematical calculation software for numerical solving becomes an effective way for solving this problem.

Figure 4.1 represents the model of the crankshaft system's control system. Under the cylinder firing pressure and the oil film supporting reaction force, the movement parameters of the main bearing's axle center will change accordingly. And the location change of the filmatic bearings' axle center would definitely cause change of the oil film pressure. Apparently, the changes of the axle center's location and the oil film reaction force will keep being fed back between the axis and the bearing, leading to further change of the entire system, and so forth. Hence, the system constructed by crankshaft system virtual prototype and main bearing is a typical self-feedback system. The control system of crankshaft system and main bearing is presented by Figure 4.1.

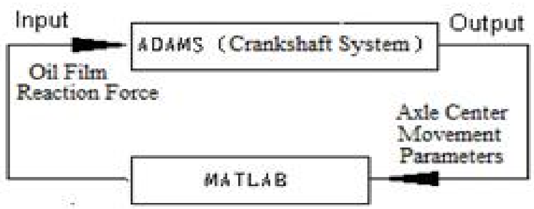

Fig. 4.1 Control system model

Define the crankshaft system's output variables as the bearing's axle movement parameters, and the input variable is the oil film supporting reaction force obtained by MATLAB calculation. Each change of the simulation step axle center's path will be calculated by ADAMS to get the axle center movement parameter at the moment, and input into MATLAB by ADAMS/CONTROL module. Build the main bearing sub-system model in MATLAB, and get the oil film pressure of the moment by calculating equation (3.6), and thereby obtain the changed oil film reaction force. Feedback this oil film reaction force to the crankshaft system through CONTROL module, and act it on the main bearing to get the new axle center path as the input of the next calculation. In this way, through the signal feedback between the crankshaft system virtual prototype and the main bearing controller, the main bearing's support for crankshaft can be realized, and the crankshaft system's dynamic and kinematic parameters within the simulation time could be obtained.

Set the Output as axle center movement parameters and Input as oil film reaction force in ADAMS/CONTROL module to generate the crankshaft system's sub-system module in MATLAB. Write the program to solve Reynolds Equation in MATLAB, and pack it into bearing sub-system module in SIMULINK. Use the packed crankshaft sub-system module and the bearing sub-system module to construct a complete crankshaft axle center- filmatic bearing self-feedback control system.

5 Numerical calculation method of dynamic lubrication bearings

Usually for the calculation of the crankshaft system's dynamic response, the dynamic oil film lubricating main bearings are simplified to linear spring - damping model or nonlinear - spring damping model. This is because the pressure distribution of the main bearings with hydrodynamic model could not be solved by analytic methods, and barely be solved by approximate analytical methods. Numerical methods to solve dynamic pressure bearings are mainly finite difference method and finite element method. Usually finite difference method could already allow obtaining accurate calculation results with little calculating time.

Using the finite difference method for solving is a transformation way to transform Reynolds partial differential equation to algebraic equation group. The solving principle is: divide the solution area into limited number of units as shown in Figure 5.1, and make each unit as small as possible so that the oil pressures in the units could be considered equal to each other or changes linearly without causing substantial error. Then transform the partial differential equation to be solved to a group of discrete linear algebraic equations. And finally, get the oil film pressure distribution by solving this group of algebraic equations.

First of all, divide the solution area into equidistant grids. As illustrated in Figure 8, column number along the bearing direction ϕ is indicated by i , and it is divided into m grids evenly on the direction; row number along the bearing direction λ is indicated by j , and it is divided into n grids evenly on the direction; there are $(m+1)\times(n+1)$ nodes in total. Usually in calculation $m = 12 \sim 25$, $n = 8 \sim 10$ would fulfill the precision requirement, while sometimes to improve the calculation precision, thin the grids in the solution area that the numbers of unknown amount are dramatic, which means to use uneven spacing grids or even proportion grids. In Figure 8, the location of each node is indicated by (i, j) , and the dimensionless pressure value P at node (i, j) is indicated by $P(i, j)$. If using the central difference

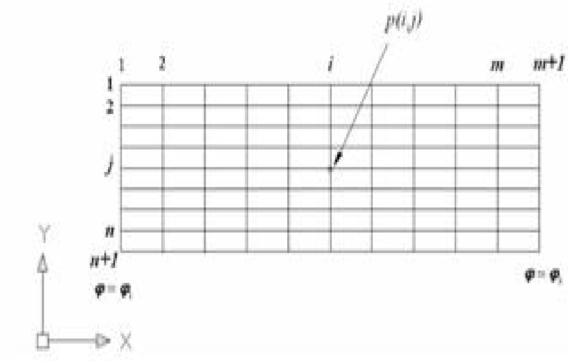

Fig. 5.1 Grid Division

formula, then the first derivatives $\frac{\partial P}{\partial \phi}$ and $\frac{\partial P}{\partial \lambda}$ at node (i, j) can be expressed in the difference quotients of the value its adjacent node's P :

$$\left(\frac{\partial P}{\partial \phi}\right)_{i,j} \approx \frac{P_{i+1,i} - P_{i-1,j}}{2\Delta\phi}; \quad \left(\frac{\partial P}{\partial \lambda}\right)_{i,j} \approx \frac{P_{i,j-1} - P_{i,j+1}}{2\Delta\lambda} \tag{5.1}$$

The second derivative on node (i, j) can be expressed in the central difference of the first derivatives of the insert point which is half step away:

$$\left[\frac{\partial}{\partial\phi}H^3\frac{\partial P}{\partial\phi}\right] \approx \frac{\left(H^3\frac{\partial P}{\partial\phi}\right)_{i+\frac{1}{2},j} - \left(H^3\frac{\partial P}{\partial\phi}\right)_{i-\frac{1}{2},j}}{\Delta\phi} \tag{5.2}$$

And the first derivatives in the equation can be expressed by central difference as:

$$\left(H^3\frac{\partial P}{\partial\phi}\right)_{i+\frac{1}{2},j} \approx H^3{}_{i+\frac{1}{2},j}\frac{P_{i+1,j}-P_{i,j}}{\Delta\phi} \quad \left(H^3\frac{\partial P}{\partial\phi}\right)_{i+\frac{1}{-2},j} \approx H^3{}_{i-\frac{1}{2},j}\frac{P_{i,j}-P_{i-1,j}}{\Delta\phi} \tag{5.3}$$

Likewise, the isothermal and uncompressible dimensionless Reynolds Equation could be transformed to:

$$A_{i,j}P_{i+1,j} + B_{i,j}P_{i-1,j} + C_{i,j}P_{i,j+1} + D_{i,j}P_{i,j-1} - E_{i,j}P_{i,j} = F_{i,j} \tag{5.4}$$

Where the coefficients are:

$$\left.\begin{aligned} A_{i,j} &= H^3_{i+\frac{1}{2}} \\ B_{i,j} &= H^3_{i-\frac{1}{2}} \\ C_{i,j} &= \left(\frac{d}{l}\cdot\frac{\Delta\phi}{\Delta\lambda}\right)^2 H^3_{i,j+\frac{1}{2}} \\ D_{i,j} &= \left(\frac{d}{l}\cdot\frac{\Delta\phi}{\Delta\lambda}\right)^2 H^3_{i,j-\frac{1}{2}} \\ E_{i,j} &= A_{i,j}+B_{i,j}+C_{i,j}+D_{i,j} \\ F_{i,j} &= 3\Delta\phi\left(H_{i+\frac{1}{2},j}-H_{i-\frac{1}{2},j}\right) \end{aligned}\right\}$$

Based on the above equation, a set of equations which apply to all internal nodes of $i = 2 \sim m, j = 2 \sim n$ can be obtained, thus constituted a group of non-homogeneous algebraic equations for $(m-1)(n-1)$ internal nodes' $P_{i,j}$ values, and solved the value of each interval nodes' $P_{i,j}$. Then, solve the equation group through iteration method. And to determine if the algebraic equation group's iteration result fulfill the precision requirement in order to decide the termination of iteration, following relative convergence principle is usually used:

$$\frac{\sum\limits_{j=2}^{m}\sum\limits_{i=2}^{n}\left|P^{(k)}_{i,j}-P^{(k-1)}_{i,j}\right|}{\sum\limits_{j=2}^{m}\sum\limits_{i=2}^{n}\left|P^{(k)}_{i,j}\right|} \le \delta \tag{5.5}$$

Relative error value δ usually takes 10^{-3} . The flow chart in Figure 5.2 shows the process of solving oil film supporting reaction force using finite difference method.

6 Multi-body dynamic simulation and data analysis

In the engine's operation process, the main bearing load caused by crankshaft force is the most important drive that the body bears. The driving force generated by the cylinder firing is passed to the body through piston - connecting rod - crankshaft, thus the main bearing's mathematical model has important effect on the simulation analysis of the body's vibration and the radiated noise. Through co-numerical calculation of ADAMS/CONTROL and MATLAB, the parameters of the crankshaft system using oil film bearing for simulation could be obtained.

During co-simulation, when solving the Reynolds equation, in order to prevent irregular matrix causing unable to calculate, it is very important to define a suitable and minimal solving step length for the crankshaft multi-body dynamic system. The fixed step length INTERGRATOR \GSTIFF&FORMULATION \I3 solver in MSC \ADAMS is used here for calculation. The engine virtual prototype model uses flexible crankshaft for dynamic simulation; using flexible crankshaft could better reflect the crankshaft's flexible deformation, and also obtain the crankshaft's stresses and strains information, as well as the maximum dynamic stresses and strains location and amplitude, thereby provide reference and determining criteria for the structure's improvement.

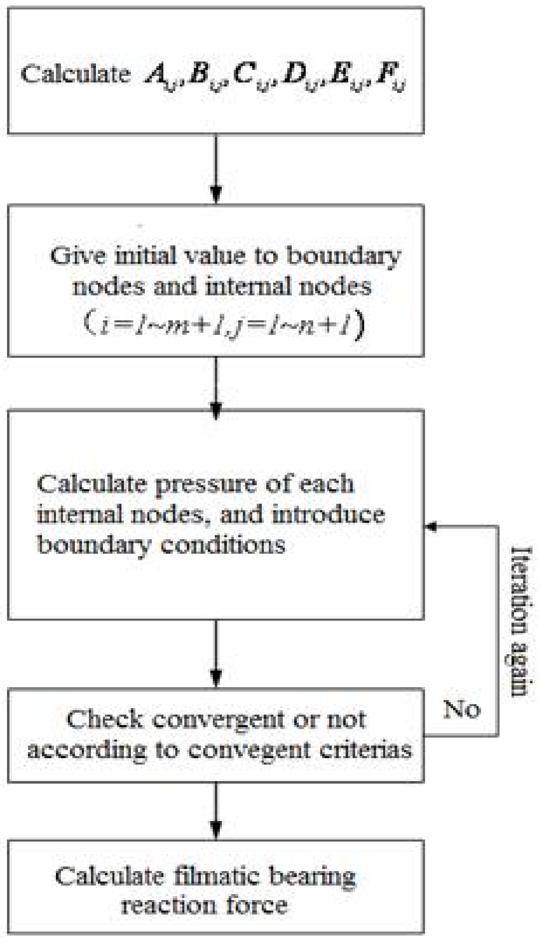

Fig. 5.2 Filmatic bearing pressure calculations

Figure 6.1 shows the main bearing loads of both the simulation using oil film lubrication bearing and the simulation turning the vice constraint bearing under the standard condition of 2200rpm rotation speed. Main bearing peak load value on y direction of the constraint bearing is 10.7% larger than the one of filmatic bearing, while the main bearing peak load value on x direction of the constraint bearing is 15.5% lower than the one of filmatic bearing. This shows that because of the filmatic bearing's dynamic lubrication effect, the difference of the main bearing loads decreases, and main bearing loads tend to be stress-even in the working cycle time period.

Figure 6.2 shows the main bearing axle center's paths of the linear bearing under the speeds of 1000rpm and 2200rpm, while Figure 6.3 shows the main bearing axle center's path of filmatic bearing under the speeds of 1000rpm and 2200rpm. As illustrated in the figures, there

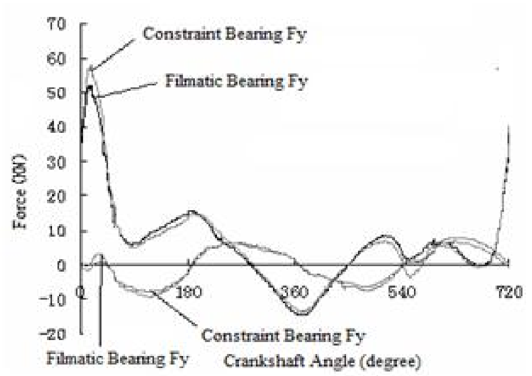

Fig. 6.1 Load comparison of main bearing

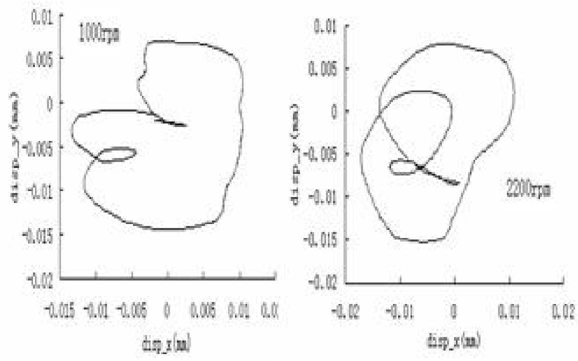

Fig. 6.3 Axle center paths of filmatic bearing

is obvious difference between the axle paths of linear bearing and filmatic bearing. Meanwhile under high and low speeds, linear bearing shows more different paths, while filmatic bearing's paths are more consistent, and the overall amplitudes under the two rotation speeds show small difference. This indicates that oil film bearing's mathematical model is more close to actual model than the model of linear bearing, therefore considering bearing's dynamic lubrication behavior in dynamic simulation would help to better simulate the engine crankshaft system under actual working conditions, and have significant meaning for guiding engine's dynamic design and structure design.

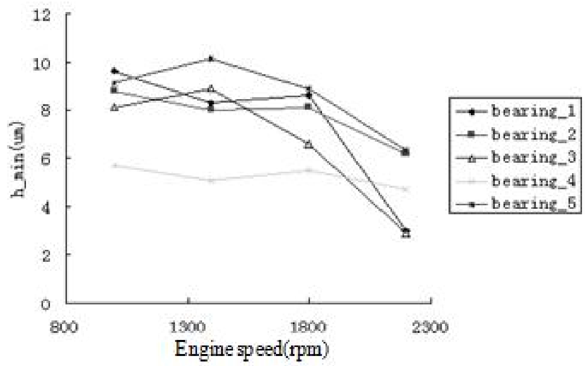

Fig. 6.4 Pattern of each main bearing's minimum oil film thickness' changes with the rotation speed change

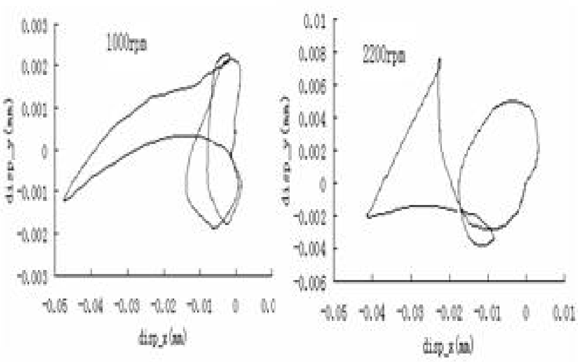

Fig. 6.2 Axle center paths of linear bearing

Figure 6.4 shows the pattern that engine crankshaft's main bearings' minimum oil film thickness changes along with the rotation speed. As shown in the picture, along with the speed's increase, each main bearing's minimum oil film thickness decreases. Take the main bearing 1 as example, when speed is 1000rpm, the minimum oil film thickness is 9.6μm, and when the speed increases to 2200rpm, the thickness decreased to 3μm.

Meanwhile, the load film thickness of the hydrodynamic lubrication bearing is not simply determined by the bearing load and the journal speed, it must be obtained by calculate each bearing's axle center path. Figure 6.5 is the comparison chart of each bearing's maximum load and minimum film thickness. Seen from the chart that the second main bearing's maximum load is 119KN while the first main bearing's maximum load is only 52KN, but the second main bearing's minimum film thickness is 6.2μm while the first main bearing's minimum film thickness is 3μm. Although the first main bearing bears smaller load, its minimum film thickness is 3.2μm thinner than the second main bearing's; hence it can be seen that the first main bearing is easier to be damaged.

Conclusion

With virtual prototype technology as simulation platform, this study made use of MSC/ADAMS and MATLAB and proposed a co-numerical calculation technology for the

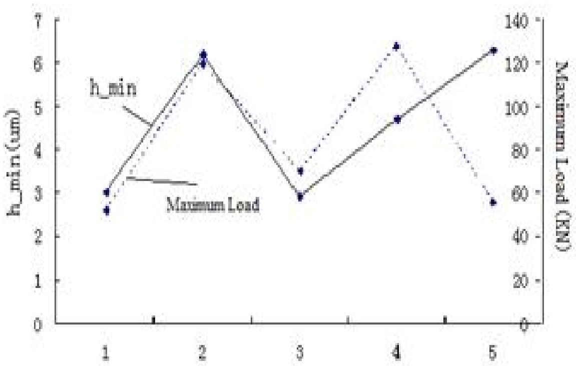

Fig. 6.5 Comparison of each main bearing's minimum oil film thickness and maximum load

coupling of crankshaft system and oil film dynamic lubricating friction. The study introduced EHD (Elastohydrodynamics) theory into multi-body dynamic simulation technology, using self-coded program, and considered the system's oil film dynamic lubricating friction behavior, solved the Reynolds Equation and obtained the crankshaft's main bearing reaction force. Through coupling calculation with the crankshaft system virtual prototype, this study also made relatively accurate prediction about the engine crankshaft system's dynamic and kinematic characteristics. Following conclusions were obtained:

First, the oil film dynamic lubricating friction behavior of the engine crankshaft system produces certain impact on the system's dynamic characteristics. It should be considered in dynamic simulation calculations.

Second, simulations with hydrodynamic lubrication coupling multi-body dynamic system could better reflect the actual working conditions of engines, and provide effective technology support for engine's design and development.

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