Water-lubricated journal bearings have been widely applied in the ships rotor-bearings system and often operates at low speed and heavy load condition. The traditional method to investigate the characteristics of water-lubricated journal bearing is solving Reynolds equation or N-S equation. However, as the poor operate conditions make the water film thickness is very thin under mesoscopic scale and the water-lubricated journal bearings work under boundary lubrication condition, the traditional method will be not suitable. In order to overcome the shortcoming of traditional method, a full scale method named lattice Boltzmann method is applied to study the characteristics of the water-lubricated journal bearing. In the present paper, basic principle of the lattice Boltzmann method, modeling process and how to deal with boundary conditions are presented in detail. The results of infinitely long water-lubricated journal bearing are obtained, including streamline, film pressure and stiffness coefficient. With the stiffness coefficient, the natural frequency of shaft system and the response of exciting force are simulated.

Keywords: Lattice Boltzmann method, Water-lubricated bearing, Rotor-bearing system
1. Introduction

As stern tube bearing on ship, water-lubricated journal bearing is the key support of stern shaft parts of the ship, and its dynamics performance has a significant influence on the dynamics response and the reliability of the whole system. Plenty of studies focus on the water-lubricated journal bearing. Many researchers modified the classical Reynolds equation, which is first derived by Osborne Reynolds [1-2]. To compute the Stribeck curves for water lubricated journal bearing, Kraker [3] used the finite element method to discretize the Reynolds-structure equations. Majumdar [4] utilized the finite difference method to solve the Reynolds equation numerically, the dynamics characteristics of three axial grooves were obtained and the dynamics parameters including stiffness and damping coefficients of water film were presented. Patir and Cheng [5-6] introduced an average Reynolds equation with flow factors, in which the surface roughness is considered. Kraker [7] proposed a novel texture averaged Reynolds equation. A multi-scale method is studied where the fluid flow in a single micro-scale texture unit cell is modelled using the Navier-Stokes equations, the results of which are used in the macro-scale texture averaged Reynolds equation he proposed. To assess the cavitation model of lubricated journal bearing, Ausas [8] found that the classical Reynolds model largely underestimates the cavitated area in the micro-textured situation, which leads to the inaccurate estimation of friction torque.

In addition, many scholars studied the water-lubricated bearing using the computational fluid dynamics (CFD) method. The CFD method was first introduced by Tucker [9], who applied a full three-dimensional thermohydrodynamics CFD approach to journal bearings. Hargreaves [10] studied fluid flow in journal bearing with three axial grooves using a computation fluid dynamics approach. Results of the circumferential and axial pressure distribution in the bearing clearance for different loads, speeds and supply pressures. Nassab [11] studied the influence of lubricant inertia on the thermohydrodynamics behavior of journal bearing. To eliminate many simplifying assumptions, he utilized CFD method to solve the exact governing equations and found that the fluid inertia term plays an important role in performance of journal bearing. Guo [12] applied CFD method to simulate various geometry fluid film bearings and study the static and dynamic characteristics of hydrodynamic, hydrostatic, and hybrid bearings. To determine the stiffness coefficients of hydrodynamic journal bearing, Zhang [13] used a three-dimensional CFD method to simulate the water-lubricated bearing. The relationships between the stiffness coefficients and the load for bearings with different relative clearances, different length-diameter ratios and different rotational speeds were presented. Gao [14] used three-dimensional CFD method to research the hydrodynamic load-carrying capacity of a water-lubricated journal bearing by a new bush structure. The result will give a design guideline for smart journal bearings.

This article presents a new computational fluid dynamics method named Lattice Boltzmann method (LBM), which is with roots in the kinetic theory of gases [15-19]. Unlike the traditional numerical schemes based on discretization of macroscopic continuum equations, the LBM is a mesoscopic method whose fundamental idea is to construct a simplified kinetic model on the mesoscopic system. The essential physics of the model obey the desired macroscopic equations. Kucinski [20] analyzed the thin film lubrication in infinitely wide wedge using LBM, considering inertia force which is neglect by Reynolds equations. With comparison the maximum pressure and load capacity obtained by LBM and the published papers, the LBM was verified. Solghar [21] studied the fluid flow within a single-groove journal bearing under laminar regime and steady-state condition using the lattice Boltzmann method. The relationship between pressure and eccentricity is presented.

In the present paper, the basic principles of the lattice Boltzmann method are presented in detail. The model of infinitely long journal bearing is built with LBM. The velocity, film pressure and stiffness coefficient of a small journal bearing are simulated. Based on the stiffness coefficient, the finite element model of shaft system is established. The natural frequency of shaft system and the response of exciting force are simulated.
2. Numerical method

2.1 The incompressible lattice Boltzmann model

In this section, the incompressible lattice BGK (Bhatnagar-Gross-Krook) model is introduced [22]. It is two-dimensional square lattice with nine velocity directions, as is shown in Fig. 1. Each particle moves between adjacent lattice nodes during one time step. The directions of the discrete velocity in the model are indicated by

\[
\begin{align*}
\mathbf{e}_i &= \begin{cases} (0,0), & i=0 \\ (\cos((i-1)\pi/2), \sin((i-1)\pi/2)), & i=1,2,3,4 \\ (\cos((i-5)\pi/2 + \pi/4), \sin((i-5)\pi/2 + \pi/4)), & i=5,6,7,8 \end{cases}
\end{align*}
\]

(1)

The evolution equation of the distribution function is

\[
f_i(x + ce_i\delta_j, t + \delta_j) - f_i(x,t) = -\tau^{-1}[f_i(x,t) - f_i^{eq}(x,t)].
\]

(2)

where \( c = \delta_x / \delta_t \) is the lattice speed, \( \delta_x \) is the lattice space and \( \delta_t \) is the time step. \( \tau \) is the dimensionless collision relaxation time, and \( f_i^{eq}(x,t) \) is the local equilibrium distribution function. In order to recover the Navier-Stocks equation correctly, the local equilibrium distribution function is given by

\[
f_i^{eq}(x,t) = \omega_i \left\{ \rho + \rho_0 \left[ \frac{\mathbf{e}_i \cdot \mathbf{u}}{c_s^2} + \frac{(\mathbf{e}_i \cdot \mathbf{u})^2}{2c_s^4} - \frac{\mathbf{u}^2}{2c_s^4} \right] \right\}.
\]

(3)

where \( \omega_i \) is weight coefficient, which depends on the lattice directions: \( \omega_0 = 4/9 \), \( \omega_i = 1/9 \) for \( i = 1 \sim 4 \), and \( \omega_i = 1/36 \) for \( i = 5 \sim 8 \). \( c_s^2 = c^2 / 3 \) is the sound speed of the model. \( \rho \) and \( \rho_0 \) are the flow density and mean density, respectively. \( \mathbf{u} \) is the flow velocity.

The flow velocity \( \mathbf{u} \) and pressure \( p \) are calculated from

\[
\rho_0 \mathbf{u} = \sum_i \mathbf{e}_i f_i.
\]

(4)

\[
p = c_s^2 \rho.
\]

(5)

with

\[
\rho = \sum_i f_i.
\]

(6)

For the LBM, the evolution equation Eq. (2) consists of two computational steps: streaming, in which each particle moves to the nearest node in the direction of its velocity, and collision, which occurs when two particle arrive at the same node simultaneously. Therefore, Eq. (2) can be divided into two steps: collision:

\[
f_i^{eq}(x,t) - f_i(x,t) = -\tau^{-1}[f_i(x,t) - f_i^{eq}(x,t)].
\]

(7)

streaming:

\[
f_i(x + ce_i\delta_j, t + \delta_j) = f_i^{eq}(x,t).
\]

(8)
2.2 Boundary conditions

2.2.1 The velocity and pressure boundary

As shown in Fig. 2, the line COA is the boundary, the node F, B and E are lying in the fluid region, and the node G, D and H are outside the fluid region. Generally, the macroscopic variables are provided as boundary conditions. Reasonable treatment of the distribution function at boundary is very significant. For the given macroscopic variable, a non-equilibrium extrapolation rule for velocity and pressure boundary conditions is presented in LBM [23]. The rule is based on the decomposition of the distribution function:

\[
 f_i(x_o, t) = f_i^{eq}(x_o, t) + \left[ f_i(x_B, t) - f_i^{eq}(x_B, t) \right].
\] (9)

2.2.2 The curved boundary

For the curved boundary, the locations of the physical boundaries and the grid nodes are not coincidence. Therefore, a special boundary treatment method must be used to deal with the curved boundaries. An effective method proposed by Guo [24] is used in this work, and this boundary treatment method is of second order accuracy in both time and space. As shown in Fig. 3, the node types are presented, including fluid node, solid node and boundary node. The link between the fluid node \(x_f\) and the solid node \(x_s\) intersects the physical boundary node \(x_b\). The fraction of the link in fluid region is:

\[
 0 \leq q = \frac{|x_f - x_b|}{|x_f - x_s|} \leq 1.
\] (10)

To specify the post-collision distribution function \(f_i^+(x_s, t)\), we decompose \(f_i(x_s, t)\) into two parts: the equilibrium part \(f_i^{eq}(x_s, t)\) and the non-equilibrium part \(f_i^{neq}(x_s, t)\), then the collision step can be obtained:

\[
 f_i^+(x_s, t) = f_i^{eq}(x_s, t) + (1 - \tau^{-1}) f_i^{neq}(x_s, t).
\] (11)

For the equilibrium distribution function \(f_i^{eq}(x_s, t)\), \(\rho_s\) and \(\mathbf{u}_s\) are given as:
For the non-equilibrium part, \( f_i^{\text{neq}}(x_i, t) \) is defined as:

\[
f_i^{\text{neq}}(x_i, t) = \begin{cases} 
    f_i(x_i, t) - f_i^{\text{eq}}(x_i, t), & q \geq 0.75 \\
    q \left[ f_i(x_i, t) - f_i^{\text{eq}}(x_i, t) \right] + (1 - q) \left[ f_i(x_{\text{eq}}, t) - f_i^{\text{eq}}(x_{\text{eq}}, t) \right], & q < 0.75
\end{cases}
\]  

(13)

3. Result and discussion

3.1 The performance of infinite long journal bearing

In order to simulate more convenient, the main parameters of journal bearing is employed in Table 1. The model is just used to verify the LBM in the application of hydrodynamics journal bearing. In the modelling, the grid is 80000x20, and the requirements is satisfied. As shown in Fig. 4, the influence of the eccentricity ratio on pressure distribution is described. In the picture, the varying parameter is the eccentricity ratio 0.4 and 0.5. It is obvious that pressure increases with increasing the eccentricity ratio. The stiffness of the bearing is \( 6.69 \times 10^8 \) N/m. The velocity contour over the journal bearing is shown in Fig. 5, under this condition, the fluid state is laminar flow.
Table 1: The infinitely long journal parameter value used for numerical analysis.

<table>
<thead>
<tr>
<th>Journal diameter/ m</th>
<th>Radial clearance/ m</th>
<th>Lubricant viscosity/Pa•s</th>
<th>Rotational speed /rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.02</td>
<td>1.0e-5</td>
<td>5.0e-2</td>
<td>3000</td>
</tr>
</tbody>
</table>

3.2 The result of shaft system

3.2.1 The nature frequency of shaft system

Based on the bearing stiffness calculated by LBM, the 2 meters long shaft with four journal bearings and one concentrated mass is built. With the finite element software Ansys17.2, the natural frequency of shaft system is simulated. The first three orders natural frequency is listed in Table 2. Due to the stiffness of the horizontal and vertical direction is consistent, the horizontal natural frequency is the same as the vertical natural frequency. As shown in Fig. 6, the vibration diagram of first three orders natural frequency is described. In the picture, the first order frequency appears in the concentrated point position.

Table 2: The first three orders natural frequency of shaft system.

<table>
<thead>
<tr>
<th>Directions</th>
<th>1st</th>
<th>2nd</th>
<th>3rd</th>
</tr>
</thead>
<tbody>
<tr>
<td>Horizontal</td>
<td>4.47</td>
<td>40.58</td>
<td>186.76</td>
</tr>
<tr>
<td>Vertical</td>
<td>4.47</td>
<td>40.58</td>
<td>186.76</td>
</tr>
</tbody>
</table>

3.2.2 The exciting force response of the shaft system

To study the exciting force response of the shaft system, 50N exciting force is loaded in the position of the concentrated mass point. As shown in Fig. 7, the response of the position of the concentrated mass point is presented. The x axis represents time, and the y axis represents vibration displacement. When the simulation is stable, the maximum amplitude of the concentrated mass point is 200µm.
Fig 7: The response of the concentrated mass point

4. Conclusion

In this paper, a new computational fluid dynamics method named lattice Boltzamnn method is introduced in the performance simulation of water-lubricated journal bearing. The streamline, film pressure and stiffness coefficient of infinitely long journal bearing is studied. Using the stiffness coefficient, a tested shaft system is built, and the natural frequency and response of exciting force are researched.

REFERENCES

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