The design of eccentricity ratio, relative clearance and length-to-diameter ratio of water-lubricated journal bearings under different working conditions

Ying Liu, Gengyuan Gao*, Zhongwei Yin, Dan Jiang, Ning Ding
School of Mechanical Engineering, Shanghai Jiao Tong University, Shanghai, China
Corresponding author's mailbox: gaoqi_1118@163.com

Abstract: This paper investigated the eccentricity ratio, relative clearance, and length to diameter ratio under different load carrying capacities and journal rotation speeds, aiming to provide design reference of eccentricity ratio, relative clearance, and length to diameter ratio in water-lubricated bearing design process. The computer fluid dynamics was adopted to analyze eccentricity ratio, relative clearance, and length to diameter ratio of water-lubricated bearing. The cavitation effect of water was considered in the analysis. The simulation results of eccentricity ratio were verified by experiments. In different journal rotation speeds, the relative clearance had little influence on the maximum water film pressure and load carrying capacity. The numerical number of maximum water film pressure and load carrying capacity was in a small range. Designing chart of eccentricity ratio, relative clearance, and length to diameter ratio was provided.

1. Introduction
The water-lubricated bearing is widely used for improving the disguise of naval ships as the oil leakage can be avoid. As the viscosity of water is smaller than that of oil, the load carrying capacity of water lubrication is smaller. Therefore, improving load carrying capacity and hydrodynamic lubricating property was the research direction of water-lubricated bearing (Zhang X et al., 2015; Lv F et al., 2017a; Wang Y et al., 2016; Gao G et al., 2015).

Zhang X et al. (2016) investigated the load carrying capacity of misaligned water-lubricated plain journal bearings. Providing the methods for selecting the appropriate radial clearance and length for misaligned water-lubricated plain journal bearings. Wang Y et al. (2017) investigated the lubrication performance of water-lubricated journal bearings with CFD and FSI method. The reference whether considered elastic deformation under different working conditions was given. Gao G et al. (2016) investigated the design parameter of water-lubricated bearing. The design chart of the bearing diameter was given. The experiment was carried out to find the bearing operating well when the eccentricity ratio (ε) was about 0.6. Lv F et al (2019a, 2019b) investigated bearing structure with axial grooves and enlarged axial end to improve the hydrodynamic lubrication. Lv F et al. (2018, 2017b, 2019c) investigated the mixed-lubrication analysis of misaligned bearing considering turbulence and wall slip. The result showed turbulence and wall slip decrease the hydrodynamic lubrication of water-lubricated bearing. Wang Y et al. (2014) investigated the hydrodynamic lubrication of bearing bush structure with two cavities, the result showed two cavities improved the hydrodynamic lubrication for constructing continuous water film. Xie Z et al. (2021) studied the performances and regimes of lubrication of water lubricated bearing. The results guided further study of the effect of surface topography on bearing design.
For bearing parameter design, the select of eccentricity ratio (ε), relative clearance (ψ), and length to diameter ratio (L/D) was critical for the bearing performance (Zhang Z, 1986). The research about the influence mechanism of eccentricity ratio (ε), relative clearance (ψ), and length to diameter ratio (L/D) on the load carrying capacity was less. In this article, the mainly work was used the computational fluid dynamics (CFD) programs to analyze the bearing performance. The influence of ε ψ L/D on the bearing load carrying capacity and rotation speed was studied. The reference of ε ψ L/D design chart was given.

2. Simulation

2.1. Governing equations

The water fluid is considered isothermal, incompressible Newtonian fluid. The viscosity model is definite to laminar model. The governing equations are mass conservation equation and momentum conservation equation, as follows:

\[ \nabla \cdot \vec{u} = 0 \]  
\[ \frac{\partial \vec{u}}{\partial t} + (\vec{u} \cdot \nabla) \vec{u} = -\frac{1}{\rho} \nabla p + \rho \nabla^2 \vec{u} + f \]  

When the water film pressure runs under the cavitation pressure of water, the liquid water turns into water vapor. The liquid-to-vapor transport equation is:

\[ \frac{\partial}{\partial t} (\alpha_v \rho_v) + \nabla \cdot (\alpha_v \rho_v \vec{v}_v) = R_g - R_c \]  

Where \( \alpha_v \) is vapor volume fraction, \( \rho_v \) is the vapor density, \( R_g \) and \( R_c \) represent mass transfer between liquid and vapor phases in cavitation. The above model is based on Rayleigh-Plesset equation which describes the growth of single bubble in liquid showing in the following equation:

\[ R_b \frac{d^2 R_b}{dt^2} + \frac{3}{2} \left( \frac{dR_b}{dt} \right)^2 = \frac{\rho_v - \rho}{\rho_l} - \frac{2 \sigma}{\rho_l R_b} - \frac{4 \mu_l}{\rho_l R_b} \frac{dR_b}{dt} \]  

For water lubricated bearings, the Rayleigh-Plesset equation can be simplified as:

\[ \left( \frac{dR_b}{dt} \right)^2 = \frac{2 p_v - p}{3 \rho_l} \]  

The load carrying capacity (F) can be calculate by integrating the pressure over the journal surface as follows:

\[ F_x = \int_{-L/2}^{L/2} r^2 \int_0^{2\pi} p \cos \theta Rd\theta dz \]  
\[ F_y = \int_{-L/2}^{L/2} r^2 \int_0^{2\pi} p \sin \theta Rd\theta dz \]  
\[ F = \sqrt{F_x^2 + F_y^2} \]  

2.2. Geometric model

![Schematic and coordinate system of plain journal bearing](image)
Fig 1 showed the schematic and coordinate system of plain journal bearing. Eccentricity (e) was the distance between bearing center ($o_0$) and journal center ($o_1$). The bearing clearance (c) was bearing radius ($R_b$) minus journal radius ($R_j$). The ratio of e to c was eccentricity ratio ($\varepsilon$). The ratio of bearing clearance (c) to the nominal diameter of journal (R) was the relative clearance ($\psi$).

For a water-lubricated bearing, eccentricity (e), bearing clearance (c), bearing length (L), and diameter (D) were the main design parameters. In the bearing design, the mainly work was to select $\varepsilon$, $\psi$, and L/D according to the bearing load (F), journal rotation speed (N), and journal diameter (D). At the present work, the used water-lubricated bearing parameter and the water properties at 20℃ is shown in Table I:

| Parameters of the bearing flow model          | Value       |
|----------------------------------------------|-------------|
| Bearing diameter (D)                         | 80mm        |
| Eccentricity ratio ($\varepsilon$)           | 0.4 - 0.9   |
| Relative clearance ($\psi$)                  | 0.06% - 0.3%|
| Length to diameter ratio (L/D)               | 0.5 - 2     |
| Journal rotation speed (N)                   | 500 - 3000 r/min |
| Saturation water vapor pressure              | 2340 Pa     |
| Saturation density of water                  | 998.2 kg/m³ |
| Saturation density of water vapor            | 0.5542 kg/m³|
| Dynamic viscosity of water                   | 10 - 3 Pas  |
| Dynamic viscosity of water vapor             | 1.34 × 10⁻⁵ Pas |

2.3 Solution method

The clearance between journal and bearing was the water flow. The load carrying capacity was offered by the water flow. One side of the flow was considered as inlet and the other as outlet. The pressure of inlet and outlet used the atmospheric pressure. The outer surface was station wall and the inner surface was moving wall. The rotation speed of moving wall equaled to that of the journal. The flow model was established and meshed in Gambit2.3 software. As the water film thickness was three orders of magnitude smaller than the diameter and length of the bearing. The flow model was separated into several parts before finite element division. The line grid of each direction was defined, then the volume grid was generated by the way of map. The water film was divided into five layers across the film thickness, and the unit size in the circumferential and axis direction was set to 1 mm.

The meshed flow model was calculated in FLUENT 19.0. The numerical analysis used pressure-based solver. The pressure-velocity coupling was select the SIMPLEC algorithm due to fast convergence speed. The momentum equation was solved by the second order upwind discretization scheme. The volume fraction equation was solved by QUICK discretization scheme. After many tries, the relaxation factors of pressure, momentum and gasification mass were 0.1 and 0.01. For greater accuracy, the residual error was set to $10^{-6}$, and the number of calculation steps was set to 1000.
3. Results and discussion

Fig 2 the pressure distribution in the circumferential and axial direction under different ε, ψ, and L/D.

Fig 2(a) showed the pressure distribution for different eccentricity ratio (ε) in the circumferential direction of middle section. ε was a working condition parameter, Fig 2(a) showed ε effect bearing property by increasing the pressure value. Fig 2(a) explained that the stress concentration occurred as the ε above 0.9, the load carrying capacity was too small as ε under 0.4. In order to obtain good operating conditions, ε should be in a range of 0.4 - 0.9.

Fig 2(b) showed the pressure distribution for relative clearance (ψ) in the circumferential direction of middle section. ψ influent the bearing property mainly by improving the bearing pressure. Fig 2(b) showed when ψ less than 0.06%, the pressure was high and the bearing load carrying capacity was large. However, small ψ lead to worse lubrication performance and poor heat dissipation. When the ψ was large than 0.3%, the pressure falls off and the bearing load carrying capacity became worse. However, large bearing clearance could store more lubricating water. The stored water can improve the bearing lubrication performance. The heat dissipation becomes better. Therefore, ψ should be in a range of 0.06% - 0.3%.

Fig 2(c) showed the pressure distribution for different length to diameter ratio (L/D) at axial direction. Fig 2(c) showed L/D influent the load carrying capacity (F) mainly by raising the pressure area. For example, when L/D was 2, the stress area was 4 times of that when L/D was 0.5. However, the bearing with large L/D occupied big space. A large L/D bearing became more sensitive to the misaligned errors. Therefore, the L/D should be selected in a range of 0.5 - 2.
Fig 3 (a), (b), and (c) are the $\varepsilon$, $\psi$, and L/D and $P_{\text{max}}$ versus N for $F=1000N$.

The left y-axis curve of Fig 3 (a), (b), and (c) showed the change of $\varepsilon$, $\psi$, and L/D under different rotation speeds for $F=1000N$. The right y-axis showed the change of $P_{\text{max}}$ for $F=1000N$. Fig 3 (a) showed $\varepsilon$ decrease from 0.92 to 0.46. The $P_{\text{max}}$ decreased from 0.49MPa to 0.1Mpa. The trends of $\varepsilon$ and $P_{\text{max}}$ were consistent. Fig 3 (b) showed $\psi$ increase from 0.1 to 0.25. $P_{\text{max}}$ was in a small range from 0.14MPa to 0.15MPa. $\psi$ was increase and the $P_{\text{max}}$ had little change. Fig 3 (c) demonstrated L/D decreased from 2 to 0.85, $P_{\text{max}}$ increased from 0.06MPa to 0.19Mpa. The trends of L/D and $P_{\text{max}}$ was opposite. The right y-axis curve of Fig 3 had the same coordinate. $\varepsilon$ had the largest effect on the $P_{\text{max}}$.

Fig 4 (a), (b), and (c) are $\varepsilon$, $\psi$, and L/D, F versus N, for $P_{\text{max}}=0.1\text{Mpa}$
Fig 4(a), (b), and (c) showed the change of $\varepsilon$, $\psi$, and $L/D$ under different rotation speeds ($N$) for $P_{\text{max}}=0.1\,\text{MPa}$. Fig 4(a) showed $\varepsilon$ reduced from 0.79 to 0.42, $F$ moved up from 513N to 912N. The trend of $\varepsilon$ and $P_{\text{max}}$ was opposite. Fig 4(b) showed $\psi$ raised from 0.12% to 0.3%, $F$ in a small ranged from 768 to 714N. $\psi$ increased and the $F$ had little change. Fig 4(c) showed the $L/D$ reduced from 1.6 to 0.5 and $F$ decrease from 1290 to 308N. The trends of $L/D$ and $F$ were consistent. The right y-axis curve of Fig 4 had the same coordinate. $L/D$ had the largest influence on $F$.

In Fig 3(b), $\psi$ had the smallest effect on the $P_{\text{max}}$ for $F=1000\,\text{N}$. In Fig 4(b), $\psi$ had the smallest effect on $F$ for $P_{\text{max}}=0.1\,\text{MPa}$. A conclusion obtained that the designer could find different data pair of $\psi$ and Rotation speed($N$) made $F$ and $P_{\text{max}}$ change in a small range. For example, when the $\psi$ was 0.1%, 0.2%, 0.25% and $N$ was 500(r/min), 2000(r/min), 3000(r/min). The $P_{\text{max}}$ was 0.146(MPa), 0.148(MPa), 0.145(MPa) and the $F$ was 1007(N), 1002(N), 1000(N).

Fig 5 the design chart of $\varepsilon$, $\psi$, and $L/D$
Fig 5 (a), (b), and (c) showed the design chart of ε, ψ, and L/D. With the bearing diameter D=80mm, the load carrying capacity (F) and rotation speed (N) were given. The value of ε, ψ, and L/D can be selected in Fig 5. For example, when the needed rotation speed was 2000rpm, the load carrying capacity was 1160N. As shown in the points marked in Fig 5, three groups of design data were obtained. Group 1, L/D=1, ε=0.65, ψ=0.2% and P_max =0.193MPa; group 2, L/D=1, ε=0.6, ψ=0.18% and P_max =0.179MPa; group 3, L/D=1.1, ε=0.6, ψ=0.2% and P_max =0.162MPa. When the operating condition need low P_max, large L/D could be selected, such as group 3. If the working condition had excellent heat dissipation conditions, small ψ could be selected, such as group 2. If the bearing support a higher maximum pressure value, big ε could be selected, such as group 1.

4. Experimental verification of eccentricity ratio
The experiments aimed to verify the simulation analysis results of the left y-axis curve in Fig 3 (a): for F=1000N, the rotation speeds (N) were set 1000(r/min), 1500(r/min), 2000(r/min), 2500(r/min), 3000(r/min) correspondingly. ε was calculated by measuring the displacement sensor 1# and 2#. The eccentricity ratio (ε) calculated method was same in Gao et al. (2014).

Fig 6 (a), (b) showed the physical drawings of the tested bearing and displacement sensor. The bearing bush material was white Thordon material. The journal diameter was 80.002mm and the diameter of bearing inner was 80.082mm. So ψ was 0.2%. Experiments result of the two sensors were listed in table II and table III. Fig 7 showed the results comparison between experiment and simulation. The experiment results were the ε of sensor 1# and sensor 2# in Table II and Table III. The experimental result was fit to the simulation result. The error maybe caused by the runout of the journal in the process of rotation.

![Displacement sensor and test bearing](image)

**Fig 6 (a) displacement sensor, (b) test bearing**

| Table II Experiment results of sensor 1# |
|-------------------------|---------|---------|---------|---------|---------|
| N(r/min)                | 1000    | 1500    | 2000    | 2500    | 3000    |
| d of 1#                 | 0.408   | 0.413   | 0.420   | 0.428   | 0.437   |
| h of 1#                 | 0.0071  | 0.0096  | 0.0131  | 0.0171  | 0.0216  |
| ε of 1#                 | 0.83    | 0.76    | 0.67    | 0.57    | 0.46    |
| ε of simulation         | 0.81    | 0.71    | 0.6     | 0.52    | 0.46    |
| Error of 1#             | 2.4%    | 6.6%    | 10.4%   | 8.8%    | 0%      |

| Table III Experiment results of sensor 2# |
|-------------------------|---------|---------|---------|---------|---------|
| N(r/min)                | 1000    | 1500    | 2000    | 2500    | 3000    |
| d of 2#                 | 0.535   | 0.526   | 0.518   | 0.505   | 0.502   |
| h of 2#                 | 0.0051  | 0.0096  | 0.0136  | 0.0201  | 0.0216  |
| ε of 2#                 | 0.87    | 0.76    | 0.66    | 0.50    | 0.46    |
| ε of simulation         | 0.81    | 0.71    | 0.6     | 0.52    | 0.46    |
| Error of 2#             | 6.9%    | 6.6%    | 9.1%    | 4%      | 0%      |
5. Conclusion
This study investigated the influence mechanism of $\varepsilon$, $\psi$, and L/D on $F$, $N$, and $P_{\text{max}}$. $\varepsilon$ and $\psi$ effect bearing capacity by raising the pressure value mainly, the length to diameter ratio (L/D) by the stress area mainly. When $F$ is constant, $\varepsilon$ has the greatest impact on $P_{\text{max}}$. The $\psi$ influent $P_{\text{max}}$ smallest and the numerical number range small. For a certain $P_{\text{max}}$, the L/D influent $F$ most. $\Psi$ has the smallest effect on $F$, and the change in value is also small. Simulation results were summarized and the $\varepsilon$, $\psi$, and L/D design parameter chart was given to guide the bearing design.

| Notation          | Definition                                      |
|-------------------|-------------------------------------------------|
| $\varepsilon$     | eccentricity=$o_b-o_j$(mm)                     |
| $c$               | bearing clearance=$R_b-R_j$                    |
| $D$               | bearing diameter(mm)                           |
| $N$               | Journal rotation speed (r/min)                  |
| $\psi$            | relative clearance=$c/R$                       |
| $\varepsilon$     | eccentricity ratio=$\varepsilon/c$             |
| $P_{\text{max}}$  | maximum water film pressure (Pa)               |
| $o_b$             | bearing center                                 |
| $o_j$             | journal center                                 |
| $R_b$             | bearing radius(mm)                             |
| $R_j$             | journal radius(mm)                             |
| $L$               | bearing length(mm)                             |
| $d$               | displacement of sensor(mm)                     |
| $h$               | thickness of water film(mm)                    |

Acknowledgements
The work was financially supported by the National Natural Science Foundation of China (Grant no. 51705310) and Marine Low Speed Engine Project-Phase I (Grant No. CDGC01-KT11).

References:
[1] Zhang, X., Yin, Z.(2016), “Load carrying capacity of misaligned hydrodynamic water-lubricated plain journal bearings with rigid bush materials”, *Tribology International*, Vol. 99, pp. 1-13.
[2] Zhang, X., Yin, Z.(2015), “Determination of stiffness coefficients of hydrodynamic water-lubricated plain journal bearings”, *Tribology International*, Vol. 85, pp. 37-47.
[3] Wang, Y., Yin, Z., Jiang, D., Gao, G., Zhang, X. (2016) “Study of the lubrication performance of water-lubricated journal bearings with CFD and FSI method”, *Industrial Lubrication and Tribology*, Vol. 68 No. 3, pp. 341-348.
[4] Wang, Y., Yin, Z., Gao, G., Zhang, X.(2017), “Analysis of the performance of worn hydrodynamic water-lubricated plain journal bearings considering cavitation and elastic deformation”, *Mechanics & Industry*, Vol. 18 No. 5, pp. 508.
[5] Gao, G., Yin, Z., Jiang, D., Zhang, X., Wang, Y.(2016), “Analysis on design parameters of water-lubricated journal bearings under hydrodynamic lubrication”, *Proceedings of the Institution of*
[6] Gao, G., Yin, Z., Jiang, D. (2015), “CFD analysis of load-carrying capacity of hydrodynamic lubrication on a water-lubricated journal bearing”, *Industrial Lubrication and Tribology*, Vol. 67 No. 1, pp. 30-37.

[7] Gao, G., Yin, Z., Jiang, D. (2014), “Numerical analysis of plain journal bearing under hydrodynamic lubrication by water”, *Tribology International*, Vol. 75, pp. 31-38.

[8] Lv, F., Ta, N., Rao, Z. (2017a), "Analysis of equivalent supporting point location and carrying capacity of misaligned journal bearing", *Tribology International*, Vol. 11, pp. 626-38.

[9] Lv, F., Jiao, C., Zou, D., Ta, N., Rao, Z. (2019a), "Analysis of misaligned water-lubricated polymer bearing with axial grooves", *Industrial Lubrication and Tribology*, Vol. 71 No. 3, pp. 411-419.

[10] Lv, F., Zou, D., Ta, N., Rao, Z. (2019b), "Improvement of lubrication performance of water lubricated polymer bearing via enlarged axial end bearing diameter", *Industrial Lubrication and Tribology*, Vol. 71 No. 4, pp. 564-572.

[11] Lv, F., Zou, D., Ta, N., Rao, Z. (2019c), "Influence of local turbulent flow on the performance of a mixed-lubrication bearing", *Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology*, Vol. 233 No. 7, pp. 1029-1035.

[12] Lv, F., Jiao, C., Ta, N., & Rao, Z. (2018), "Mixed-lubrication analysis of misaligned bearing considering turbulence", *Tribology International*, Vol. 119, pp. 19-26.

[13] Lv, F., Rao, Z., Ta, N., & Jiao, C. (2017b), "Mixed-lubrication analysis of thin polymer film overplayed metallic marine stern bearing considering wall slip and journal misalignment", *Tribology International*, Vol. 109, pp. 390-397.

[14] Wang, Y., Shi, X., Zhang, L. (2014), “Experimental and numerical study on water-lubricated rubber bearings”, *Industrial Lubrication and Tribology*, Vol. 66 No. 2, pp. 282-288.

[15] Xie, Z., N. Shen, et al. (2021), "Theoretical and experimental investigation on the influences of misalignment on the lubrication performances and lubrication regimes transition of water lubricated bearing", *Mechanical Systems and Signal Processing*, Vol. 149 No. 107211, pp. 107211.

[16] Zhang, ZM. (1986), “Theory of hydrodynamic lubrication of sliding bearings”, Beijing.