ISSN:2790-1688 DOI: 10.56028/aetr.3.1.775

Research on the influence of lubrication performance of hydrodynamic journal bearings based on orthogonal test method

Xinliang Hu[1,a],Jun Wang[2,b],Kaiping Yu[1],Quande Chi[1]

[1]Department of Mechanical EngineeringPLA Army Academy of Artillery and Air Defense

Hefei Anhui 230000,China.

[2]Department of Arms EngineeringPLA Army Academy of Artillery and Air Defense

Hefei Anhui 230000,China.

[a]huxinliang99@yeah.net, [b]386166085@qq.com

Abstract. The orthogonal experiment design method is used based on analysis of lubrication of hydrodynamic journal bearing. The effect of structural parameters (bearing diameter, wide diameter ratio and clearance ratio) and operating parameters (linear velocity and specific pressure) on lubrication performance of hydrodynamic journal bearings is studied. The results show that the linear velocity and specific pressure have significant influence on maximum oil film pressure of the bearings. The influence of specific pressure and bearing diameter on load capacity of the bearings is significant. The bearing diameter and linear velocity have significant influence on discharge flow of the bearings. The linear velocity has the most significant influence on friction power loss of the bearings when the low viscosity lubricant is used, and the clearance ratio, specific pressure and bearing diameter have a great influence on friction power loss of the bearings when the high viscosity lubricant is used.

Keywords: hydrodynamic journal bearing; structural parameter; operating parameter; lubrication performance; orthogonal test

1. Introduction

Hydrodynamic journal bearing have high load-bearing capacity, high vibration resistance, stable working process, long service life, etc. They are widely used in the transmission system of mechanical equipment and devices, and their working conditions directly affect the working performance and reliability of mechanical equipment. Lubrication can reduce the friction and wear on the working surface of the journal bearing, improve the efficiency and service life of the bearing, and also play a role in cooling, vibration absorption and rust prevention. Journal bearing can work normally and its lubrication condition is closely related. With the continuous improvement of lubrication theory and calculation technology, the factors considered in the study of hydrodynamic journal bearing lubrication gradually increased, and the prediction of bearing lubrication performance has become more accurate. Reference [1] studied bearing lubrication under severe conditions; reference [2] explored the treatment of bearing mixed lubrication; reference [3] studied the influence of surface weaving and thermal effects on bearing lubrication; reference [4] explored the numerical solution of bearing elastofluid lubrication; reference [5] analyzed the influence of oil supply conditions on bearing lubrication; reference [6] studied the influence of thermal boundary conditions on bearing thermal fluid lubrication analysis; Reference [7] conducted the analysis of elastofluid lubrication of heavy-duty bearings; reference [8] studied the effect of operating conditions on the lubrication of crankshaft bearings in internal combustion engines; reference [9] analyzed the effect of lubricant viscosity on bearing lubrication; reference [10] studied the orthogonal test method for the performance of bearings in internal combustion engines. Although a lot of research work has been carried out on the lubrication of dynamic-pressure plain bearings, there is a lack of comprehensive and systematic research on the influence of structure and operating parameters on the lubrication performance of bearings. To this end, the hydrodynamic journal

ISSN:2790-1688 DOI: 10.56028/aetr.3.1.775
bearing as the research object, in the analysis of bearing lubrication based on the application of the orthogonal test design method, to study the structure and operating parameters on the hydrodynamic journal bearing lubrication performance.

2. Hydrodynamic Journal Bearing Lubrication Analysis

2.1 Reynolds Equation

$$\frac{\partial}{\partial\theta}(h^3\frac{\partial p}{\partial\theta})+R_b^2\frac{\partial}{\partial y}(h^3\frac{\partial p}{\partial y})=6\eta R_b(u\frac{\partial h}{\partial\theta}+2R_b\frac{\partial h}{\partial t}) \tag{1}$$

where p is oil film pressure, h is oil film thickness, η is lubricant viscosity, $u=u_j+u_b$, $uj=R_j\omega_j$ is speed of journal surface, R_j is radius of journal, ω_j is journal rotational angular velocity , $u_b=R_b\omega_b$ is velocity of the bearing surface , R_b is bearing radius, ω_b is bearing rotational angular velocity.The *Reynolds* boundary conditions are applied. the *Reynolds* equation is solved by the finite difference method. The nodes inside the solution domain are in central difference format, and the nodes at the boundary of the solution domain (located at the front and rear end faces of the bearing) are in front or back difference format along the axis of the bearing.

2.2 Oil Film Thickness Equation

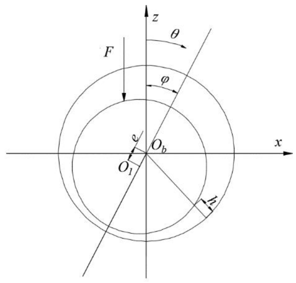

Fig.1 Schematic diagram of journal bearing in the Cartesian coordinate system

In Fig.1, φ is eccentricity angle and e is eccentricity distance. θ is the circumferential coordinate of bearing expansion, and θ=0° is the vertical direction. Since the bearing clearance value c is much smaller than the bearing diameter, after a series of simplifications, the expression of oil film thickness for the general case is:

$$h \approx c + e\cos(\theta-\varphi) \tag{2}$$

2.3 Bearing Oil Film Counterforce

For a finite width bearing, let the oil film reaction force in the horizontal direction be F_x and the oil film reaction force in the vertical direction be F_z. The component of the oil film reaction force can be obtained by integrating the oil film pressure.

$$F_x = -\int_0^L \int_{\theta_1}^{\theta_2} pR\sin\theta d\theta dy \tag{3}$$

$$F_z = -\int_0^L \int_{\theta_1}^{\theta_2} pR\cos\theta d\theta dy \tag{4}$$

$$F = \sqrt{F_x^2 + F_z^2} \tag{5}$$

2.4 Discharge Flow

The flow rate of lubricant from the front face and rear face of the bearing are:

$$Q_1 = -\int_0^{2\pi} \frac{h^3}{12\eta}\frac{\partial p}{\partial y}\bigg|_{y=0} \mathrm{d}\theta \tag{6}$$

ISSN:2790-1688 DOI: 10.56028/aetr.3.1.775

$$Q_2 = -\int_0^{2\pi} \frac{h^3}{12\eta} \frac{\partial p}{\partial y}\bigg|_{y=L} \mathrm{d}\theta \tag{7}$$

$$Q = Q_1 + Q_2 \tag{8}$$

2.5 Friction Power Loss

$$F_{\mathrm{j}} = \int_0^L \int_0^{2\pi} \left(\frac{h}{2} \frac{\partial p}{R\partial\theta} + \frac{u\eta}{h} \right) R\mathrm{d}\theta\mathrm{d}y \tag{9}$$

$$N_{\mathrm{F}} = F_{\mathrm{j}} \cdot u \tag{10}$$

2.6 Solution Method

The *Reynolds* equation is solved by the finite difference method [11]. The nodes within the solution domain are in the central difference format, and the nodes at the boundary of the solution domain (located at the front and rear end faces of the bearing) are in the forward or backward difference format along the bearing axis. The solution domain is taken as the full length of the bearing in the axial direction and 360° in the circumferential direction, and is divided into equally spaced grids along the circumferential and axial directions. The grid division of the bearing surface along the plane is shown in Fig. 2. The circumferential direction θ = 0° to 360° is divided into m nodes, i.e., i = 1 to m, with i = 1 corresponding to θ = 0° and θ = 360°; the axial direction y = 1 to B is divided into n nodes, i.e., j = 1 to n.

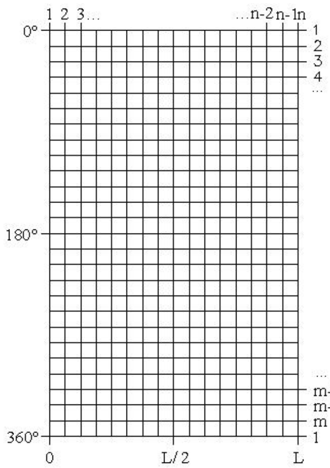

Fig.2 Meshing

Calculation of bearing load capacity, end leakage flow and friction power consumption in the formula for the integration of the application of Simpson's formula for numerical integration, the partial derivative of the application of the four-point difference formula.

3. Orthogonal Tests

Based on the lubrication analysis of hydrodynamic journal bearing, the influence of structural parameters (bearing diameter, width-to-diameter ratio and clearance ratio) and operating parameters (journal surface linear velocity and specific pressure) on the lubrication performance of hydrodynamic journal bearing is analysed by means of an orthogonal test design. Two lubricants with two viscosity values were selected, corresponding to a high viscosity lubricant (lubricating oil) and a low viscosity lubricant (fuel oil).

ISSN:2790-1688 DOI: 10.56028/aetr.3.1.775

Table1. Bearing structure and working condition parameter and its value range

Parameters	Range
Width to diameter ratio L/D	0.5 ~ 1
Clearance ratio ε	0.001 ~ 0.003
surface linear velocity V /m/s	1 ~ 60
Specific pressure P /MPa	0.1 ~ 4
Bearing diameter D /mm	20 ~ 120
Lubricant viscosityη/Pa·s	0.43032×10-3 (Low viscosity) 9.0546×10-3 (High viscosity)

3.1 Orthogonal Test Design

In practical problems, consider one or two influencing factors (structure and working conditions parameters) on the test results (sliding bearing lubrication performance parameters) significance analysis, you can choose a one-dimensional or binary ANOVA, but the bearing lubrication calculation process to consider more than two influencing factors, it is necessary to consider multiple influencing factors (bearing diameter, width to diameter ratio, clearance ratio, journal surface linear velocity and specific pressure) on the results at the same time Whether there is a significant role, will use the orthogonal test method to analyze the calculation results. The basic principle of the orthogonal test method is to use statistical principles and the law of orthogonality to extract some of the test points that meet the principles of orthogonality from a very large number of test points, and use the "orthogonal table" to arrange the test points in accordance with the principles of the scientific test method. The construction criteria of the orthogonal table are "balanced dispersion method" and "neat comparability", which has the advantage of greatly reducing the number of tests, and can summarize the law of test results more accurately.

For the dynamic plain bearing structure and working parameters analyzed in this paper, five factors of the orthogonal test were determined. In order to analyze as accurately as possible, the number of levels selected for each factor should be appropriate more. Due to the large span of each parameter range of the bearing, each parameter range is divided equally and five levels are taken for each factor. Table II shows the selected L25(5^6) standard orthogonal table.

Table 2. Factor level setting table

Number of levels	L/D	ε	V/(m/s)	P/MPa	D/mm
1	0.5	0.001	1	0.1	20
2	0.625	0.0015	15	1	45
3	0.75	0.002	30	2	70
4	0.875	0.0025	45	3	95
5	1	0.003	60	4	120

3.2 Orthogonal Test Results

According to the test points determined by the orthogonal table, the bearing lubrication performance parameters under each test point (bearing structure and working condition parameters) are calculated by bearing lubrication analysis, and the results are shown in Table III Where, 'low' represents the low viscosity lubricant case, and 'high' represents the high viscosity lubricant case.Results and analysis

Since the number of levels selected for each factor in the analysis was 5, the results of the orthogonal tests were analyzed in this paper using ANOVA [12] in order to accurately estimate the importance of the effects of the test results for each factor.

ISSN:2790-1688 DOI: 10.56028/aetr.3.1.775

Table 3. Calculation results table

Serial No.	*L/D*	*ε*	*V(m/s)*	*P (MPa)*	*D(mm)*	*Pmax (L) (MPa)*	*F (L) (N)*	*Q (L) ($m^3/s\times10^7$)*	*N (L)(kW)*	*Pmax (H) (MPa)*	*F (H) (N)*	*Q (H)($m^3/s\times10^7$)*	*N (H)(kW)*
1	0.5	0.001	1	0.1	20	0.3964	19.98	0.73464	0.00005	0.2067	19.99	0.24776	0.00059
2	0.5	0.002	15	1	45	6.3472	1011.79	114.8	0.02652	2.9403	1012.01	90.39	0.19842
3	0.5	0.003	30	2	70	18.52	4902.61	739.27	0.19243	7.941	4904.06	809.92	1.17642
4	0.5	0.004	45	3	95	37.0153	13524.45	2374.1	0.66937	14.9013	13547.99	3125.1	3.67735
5	0.5	0.005	60	4	120	63.5388	28795.06	5574.2	1.67661	23.8101	28779.91	8244.7	8.64118
6	0.625	0.001	15	2	95	11.7189	11271.61	298.32	0.13884	5.3099	11290.02	222.74	0.99205
7	0.625	0.002	30	3	120	26.8356	27010.33	1630.6	0.65019	10.9946	27010.92	1868.3	3.71597
8	0.625	0.003	45	4	20	25.1272	999.07	116.95	0.09733	11.1901	999.6	95.392	0.66352
9	0.625	0.004	60	0.1	45	0.3122	126.61	956.18	0.10748	0.1909	126.69	165.79	1.45298
10	0.625	0.005	1	1	70	42.3124	3064.84	21.826	0.00098	10.8553	3062.16	42.126	0.00462
11	0.75	0.001	30	4	45	17.4392	6071.63	155.33	0.22234	8.4195	6074.41	69.715	1.88378
12	0.75	0.002	45	0.1	70	0.244	367.38	764.21	0.17968	0.1867	367.28	67.718	3.02842
13	0.75	0.003	60	1	95	5.6451	6773.2	3990.3	0.91927	2.4899	6774.34	2912.2	6.20413
14	0.75	0.004	1	2	120	169.7625	21609.98	80.347	0.00391	28.3335	21603.42	92.37	0.01746
15	0.75	0.005	15	3	20	33.2501	900.13	54.539	0.0185	12.2578	900.13	76.449	0.09433
16	0.875	0.001	45	1	120	3.2443	12607.29	1724.1	1.08584	1.8872	12607.23	354.46	12.2837
17	0.875	0.002	60	2	20	6.4788	699.6	127.71	0.16079	3.7724	700.03	26.243	1.81979
18	0.875	0.003	1	3	45	101.9201	5312.23	6.7006	0.00123	25.975	5315.08	13.222	0.00576
19	0.875	0.004	15	4	70	68.5854	17136.88	449.72	0.16201	21.2542	17161.99	790.26	0.76772
20	0.875	0.005	30	0.1	95	0.4428	789.6	3776.6	0.12101	0.2078	789.42	1775.7	0.93684
21	1	0.001	60	3	70	9.8749	14707.38	857.63	1.39785	5.6246	14709.71	186.39	14.6008
22	1	0.002	1	4	95	152.2841	36098.11	19.52	0.00496	36.6316	36117.26	39.818	0.02324
23	1	0.003	15	0.1	120	0.4068	1438.78	1945	0.0686	0.1971	1440.39	753.14	0.54613
24	1	0.004	30	1	20	4.6053	400.04	141.65	0.0412	2.0795	400.34	73.329	0.28779
25	1	0.005	45	2	45	15.8542	4049.37	1082.1	0.35293	5.8962	4046	1260.4	1.81699

3.3 Maximum Oil Film Pressure of the Bearing

The variance results of the maximum oil film pressure of the bearing at low viscosity and high viscosity lubricants, respectively. As seen from the F values in the table, the effects of linear velocity and specific pressure on the maximum oil film pressure of the bearing are highly significant. In the case of low viscosity lubricant, the degree of influence of specific pressure on maximum oil film pressure is lower than that of linear velocity; in the case of high viscosity lubricant, the degree of influence of specific pressure is significantly higher and exceeds that of linear velocity. The clearance ratio has a significant effect on the maximum oil film pressure only in the case of high-viscosity lubricant, and the effect of width-to-diameter ratio and bearing diameter on the maximum oil film pressure is not significant.

3.4 Bearing Load Capacity

The bearing load capacity under low viscosity and high viscosity lubricant. From the table F value, it can be seen that the degree of influence of specific pressure and bearing diameter on bearing load capacity is highly significant, and the degree of influence of wide diameter ratio, clearance ratio and linear speed is not significant. In the case of low viscosity and high viscosity lubricant all the structure and working parameters on the bearing load capacity almost did not change, it can be seen that the change of lubricant viscosity almost will not change the structure and working parameters on the bearing load capacity degree of significance.

3.5 Figures and Tables

The bearing frictional power consumption under low viscosity and high viscosity lubricants. From the F value in the table, it can be seen that in the case of low viscosity lubricant, the influence

ISSN:2790-1688 DOI: 10.56028/aetr.3.1.775

of linear velocity on the frictional power consumption of the bearing is highly significant, the influence of bearing diameter on the frictional power consumption of the bearing is significant, and the influence of width-to-diameter ratio, clearance ratio and specific pressure on the frictional power consumption of the bearing are not significant. In the case of high viscosity lubricant, all the structure and working condition parameters have a certain degree of influence on the frictional power consumption of the bearing, the influence of linear speed is still highly significant, the influence of clearance ratio and specific pressure is significantly increased, the influence of bearing diameter, the degree of influence are significant. It can be seen that the increase of lubricant viscosity obviously affects the influence of structure and working condition parameters on the frictional power consumption of the bearing, with the increase of lubricant viscosity, the original influence degree is not significant structure and working condition parameters on the frictional power consumption of the bearing end leakage flow has a significant effect.

4. Conclusion

In this paper, based on the lubrication analysis of dynamic pressure plain bearings with different structure and working parameters, the influence of structure and working parameters on the lubrication performance of dynamic pressure plain bearings is studied by applying the orthogonal test design analysis method.

(1)The influence of bearing structure parameters (bearing diameter, width to diameter ratio and clearance ratio) on the lubrication status of dynamic-pressure plain bearings is analyzed. Bearing diameter has a more significant effect on the bearing friction power consumption, end drainage flow and load capacity, is the bearing structure parameters on the bearing lubrication performance of the greater influence of factors; wide diameter ratio on all bearing lubrication state have no significant effect; clearance ratio only in the case of high viscosity lubricant on the bearing friction power consumption has a significant effect, most cases on the bearing lubrication performance also has no significant effect.

(2)On the bearing operating parameters (linear velocity and specific pressure) on the sliding bearing lubrication state influence. Linear velocity and specific pressure on the overall degree of influence on the bearing lubrication state are very significant. The influence degree of linear speed and specific pressure is much lower than the influence degree of lubricant flow and friction power consumption, and the influence degree of specific pressure on bearing load is higher than the influence of linear speed. When the lubricant viscosity is low, the influence degree of specific pressure is less than the influence degree of linear speed, while the opposite result when the lubricant viscosity is high.

(3)About the influence of different lubricant viscosity. With the increase of lubricant viscosity, the influence degree of working condition parameter on the maximum oil film pressure increases, and the influence degree of structure parameter on the maximum oil film pressure does not change much; the structure parameter and working condition parameter have no influence on the bearing load capacity basically; the influence degree of structure and working condition parameter on the bearing end drainage flow will decrease significantly; the influence degree of structure and working condition parameter on the bearing friction power consumption increases to some extent.

References

[1] CHATTERTON S,DANG P V，PENNACCHI P．Experimental evidence of a two － axial groove hydrodynamic journal bearing under severe operation conditions［J］．Tribology International，2017，109: 416 － 427.

[2] HAN Y，XIONG S，WANG J, A new singularity treatment approach for journal － bearing mixed lubrication modeled by the finite difference method with a herringbone mesh［J］.ASME Journal of Tribology,2016,138(1) : 1-10.

ISSN:2790-1688 DOI: 10.56028/aetr.3.1.775
[3] TALAIN,FILLON M. A numerical investigation of both thermal and texturing surface effects on the journal bearings static characteristics ［J］.Tribology International,2015,90: 228-239.

[4] MENG F,CHEN Y. Analysis of elasto － hydrodynamic lubrication of journal bearing based on different numerical methods[J]. Industrial Lubrication and Tribology,2015,67(5): 486-497.

[5] BRITOFP,MIRANDAA S,CLAROJC P.The role of lubricant feeding conditions on the performance improvement and friction reduction of journal bearings[J].Tribology International,2014,72: 65-82.

[6] ZHANG Z S,YANG Y M,DAI X D. Effects of thermal boundary conditions on journal bearing thermos hydrodynamic lubrication[J].Tribology Transactions,2013,56(5) : 759-770.

[7] BENDAOUD N,MEHALA K,YOUCEFI A,An experimental and numerical investigation in elastohydro dynamic behaviour of a plain cylindrical journal bearing heavily loaded［J］. Proceedings of the Institution of Mechanical Engineers Part J:Journal of Engineering Tribology,2012,226(10) : 809-818.

[8] Suryawanshi Ganesh L., Patil Sachin K., Desavale Ramchandra G. Dynamic model to predict vibration characteristics of rolling element bearings with inclined surface fault[J] Measurement, 2021, 184.

[9] Zou Yisheng, Li Zhixuan, Liu Yongzhi et al. A method for predicting the remaining useful life of rolling bearings under different working conditions based on multi-domain adversarial networks[J] Measurement, 2022, 188

[10] Chen Peng, Ding Jianguo, Zhuang Hui et al. Influence of manufacturing errors and misalignment on the performances of air journal bearings considering inertia effects based on SUPG finite element method[J] Measurement, 2022, 189 57 － 60，63.

[11] Suryawanshi Ganesh L., Patil Sachin K., Desavale Ramchandra G. Dynamic model to predict vibration characteristics of rolling element bearings with inclined surface fault[J] Measurement, 2021, 184.

[12] Yukiyoshi Iwata, Tomoyoshi Hirota, Takeshi Suzuki et al. Comparison of soil frost and thaw depths measured using frost tubes and other methods[J] Cold Regions Science and Technology, 2011, 71