Т. 10. №7. 2024
https://doi.org/10.33619/2414-2948/104

UDC 62-752.2 https://doi.org/10.33619/2414-2948/104/36

SIMULATION ANALYSIS OF NON-CONCENTRIC SQUEEZE FILM DAMPER

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МОДЕЛИРОВАНИЕ ДЕМПФЕРА ИЗ НЕКОНЦЕНТРИЧЕСКОЙ СЖИМАЮЩЕЙ ПЛЕНКИ

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Abstract. Squeeze film dampers are components that provide additional damping for rotating motors. In general, in order to achieve the best results of the squeeze film damper, it is necessary to appropriately increase the damping value, reduce the stiffness and set a reasonable eccentricity. In the process of designing the squeeze film damper, we can apply the rotor dynamics module of COMSOL software to model and simulate it, so as to achieve better simulation results and lay a foundation for the subsequent development and design of the product. In COMSOL software, in order to simplify the modeling of rotor components, we usually model squeeze film dampers based on the damping coefficient. The model calculates the damping coefficient for a short squeeze film damper and compares the results with the analytically calculated values. In addition, the damping coefficient depends on the position of the journal in the damper, i.e., the magnitude of the eccentricity. In this paper, by analyzing the dimensionless damping coefficient, bearing stiffness, journal displacement, and fluid load at different eccentricity, and plotting the fluid pressure distribution and the average oil film flow rate of the squeeze film damper, we can find the most suitable range of eccentricity and the damage-prone part of the damper, which can provide an important reference for the actual design of the squeeze film damper.

Аннотация. Пленочные демпферы представляют собой компоненты, обеспечивающие дополнительное демпфирование вращающихся двигателей. В общем, чтобы добиться наилучших результатов при использовании демпфера из сжимающей пленки, необходимо соответствующим образом увеличить значение демпфирования, уменьшить жесткость и установить разумный эксцентриситет. В процессе проектирования пленочного демпфера мы можем применить модуль динамики ротора программного обеспечения COMSOL для его моделирования и моделирования, чтобы добиться лучших результатов моделирования и заложить основу для последующей разработки и проектирования продукта. В программном обеспечении COMSOL, чтобы упростить моделирование компонентов ротора, мы обычно моделируем демпферы из сжимающей пленки на основе коэффициента демпфирования. Модель рассчитывает коэффициент демпфирования для демпфера с короткой пленкой сжатия и сравнивает результаты с аналитически рассчитанными значениями. Кроме того, коэффициент демпфирования зависит от положения шейки в демпфере, т. е. от величины эксцентриситета. В этой статье, анализируя безразмерный коэффициент демпфирования, жесткость подшипника, смещение шейки и нагрузку жидкости при различном эксцентриситете, а также строя график распределения давления жидкости и средней скорости потока масляной пленки демпфера с пленкой сжатия, мы можем найти наиболее подходящий диапазон эксцентриситета и подверженной повреждениям части демпфера, что может служить важным ориентиром для фактической конструкции демпфера с выжимной пленкой.

Keywords: squeeze film damper, simulation, damping factor, eccentricity.

Ключевые слова: демпфер из сжимающей пленки, моделирование, коэффициент демпфирования, эксцентриситет.

Squeeze Film Damper (SFD) is a device used to dampen and stabilize rotating machinery, commonly found in high-speed rotating equipment such as turbines, compressors and generators. Its main principle of operation is the formation of an oil film between two oppositely moving surfaces by means of a fluid, usually a lubricant, which provides a damping force that reduces vibration and stabilizes the system. It plays a key role in rotating machinery such as generators by providing an effective damping force, which reduces vibration, enhances stability, extends the life of the equipment and improves operational efficiency, which is important for ensuring the reliability and stability of the generator. In 1960s, the concept of squeeze film dampers was first introduced. Squeeze film damper structure is relatively simple, it consists of a movable inner ring, usually for the bearing in the rotor system; a fixed outer ring, usually for the bearing housing; and the oil film between them, as shown in Figure 1. Its structure is a rolling bearing and plain bearing composite system, but compared with plain bearings, squeeze film damper rolling bearing outer ring and the outer ring (bearing housing) is restricted to rotate, but allows the inner ring (bearing outer ring) in the oil film cavity of any vortex.

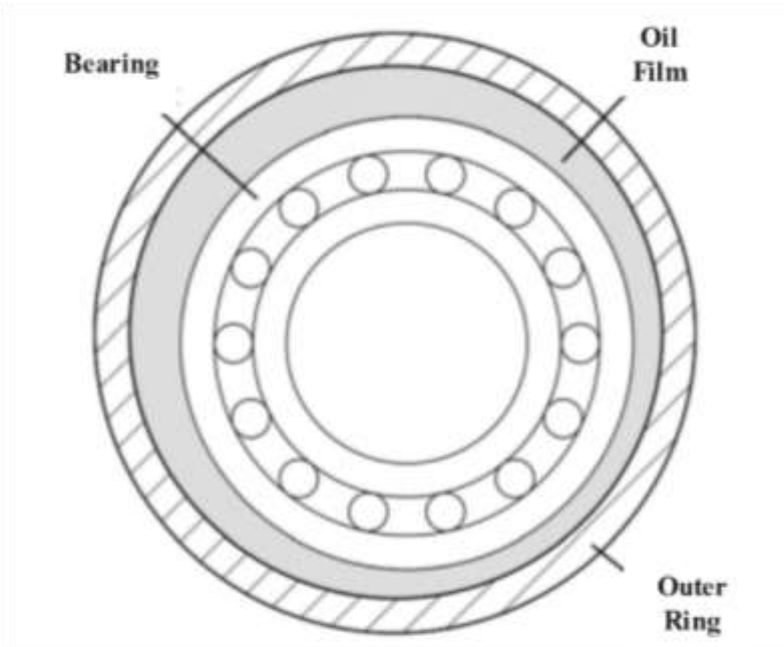


Figure 1. Squeeze film damper

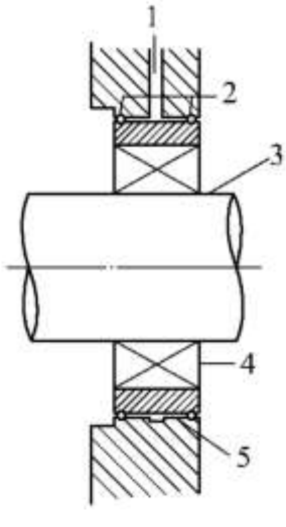
In 1963, engineer Cooper from Rolls Royce [1] published a technical paper detailing experimental investigations into utilizing squeeze film dampers (SFDs) for vibration control. Subsequently, Gunter, Barrett, and Allaire [2] conducted research on rotor vibration responses using the Reynolds equation under the assumption of short bearing, revealing the presence of multiple

Т. 10. №7. 2024
https://doi.org/10.33619/2414-2948/104

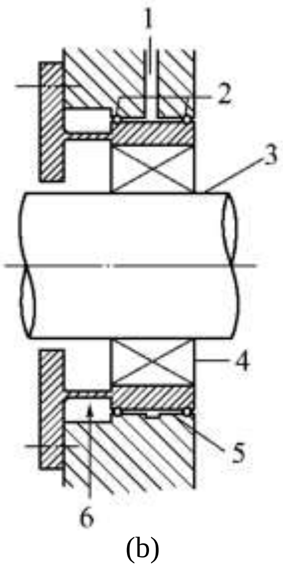
stable trajectories for the rotor under certain loads, with phenomena of transition from one trajectory to another. This jumping phenomenon, induced by the nonlinear characteristics of SFDs, remains a focal point of research. The concept of SFDs had been applied to steam engines as early as the Parsons [3] era, and later successfully implemented by Cooper on Conway engines, yielding significant effects. In the 1960s, the United States began adopting SFDs on J-69 engines. Following the 1970s, SFDs found extensive application in turbojet and turboshaft engines, leading to a proliferation of corresponding research.

Beck and Strodtman [4] utilized variational principles and numerical methods to study the motion equations of SFDs under the assumption of long bearings, while considering the compressibility of fluids. Their research indicated that the stability range of SFDs is influenced by various parameters. With the advancement of technology, an increasing number of engineering structures have started utilizing SFDs, leading to a surge in related research. As a commonly used vibration attenuation device in aircraft engines, the vibration attenuation capability of SFDs receives significant attention from research and design personnel. In practical engineering applications, the performance of SFDs is often evaluated through the dynamic characteristic coefficients of the oil film. Hence, accurate calculation and identification of these coefficients are critical research issues [5].

For the traditional squeeze film damper, it has two forms: concentric type and non-concentric type [6], as shown in Figure 2. The concentric type of squeeze film damper is equipped with centering spring, mainly squirrel cage type elastic support. Through the adjustment of the elastic bearing, when the bearing is in the static state, it can make the bearing have good centering function, and at the same time, the elastic bearing can also adjust the critical speed of the rotor system. The non-concentric squeeze film damper has no centering spring, so when the rotor system is in static state, there is no static bearing capacity, and the bearing is stranded in the bearing chamber, and when the bearing movement generates the oil film force, the rotor is lifted up, and the advantage of this non-concentric extruded oil film damper is that it occupies little space and is easy to install, so compared with concentric damper, the application of the non-concentric type is more widely used, so this paper selected the non-concentric oil film damper, which is not concentric. Therefore, this paper selects the non-concentric extruded oil film damper as the research object.



(a)



(b)

Figure 2. Two forms of squeeze film damper: (a) not concentric; (b) the concentric: 1-Lubricate 2-Seal 3-Shaft 4-Rolling bearing 5-Squeeze film 6-Centering spring

Figure 3 shows the Fixed and rotating coordinates systems for rotor and journal [7]. From Fig. 3, it can be seen that through the orthogonal decomposition, the damper will be subject to the force from the radial positive and negative directions and the tangential positive and negative directions, respectively, so it is necessary that the damping will be generated in these four directions, and we define the damping coefficients of these four directions as: C_{rr}, C_{rt}, C_{tr} and C_{tt}, and analyze the range of the above four damping coefficients under different eccentricity by COMSOL software to obtain a more reasonable range of eccentricity.

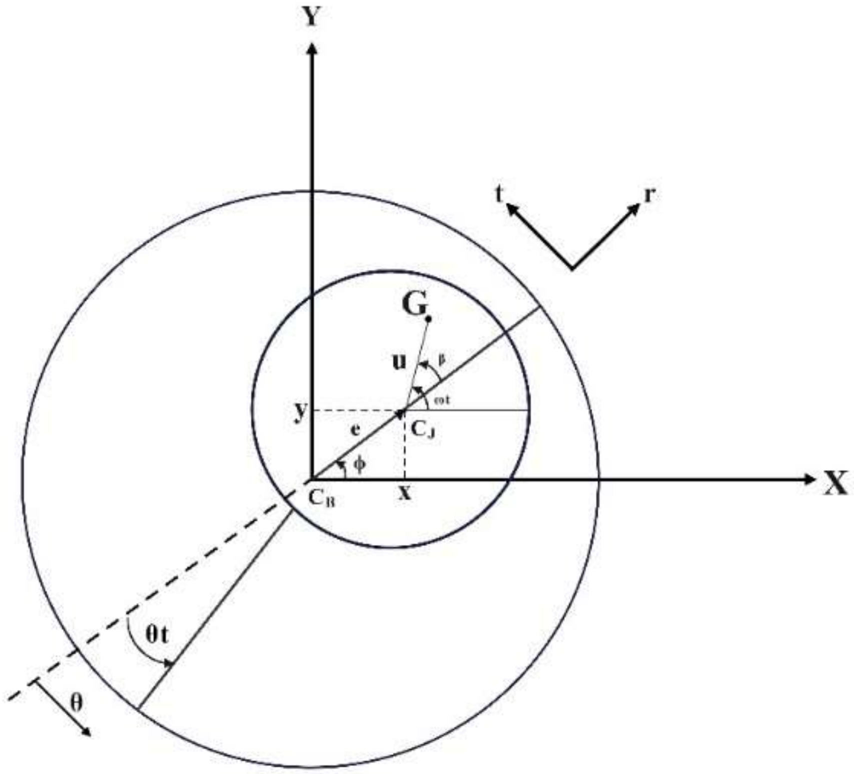


Figure 3. Fixed and rotating coordinates systems for rotor and journal

Boundary Condition Setting

COMSOL Multiphysics is a powerful multi-physics field simulation software widely used in engineering and science. It is capable of handling coupled simulation of multiple physical fields such as fluid, structure, heat, electromagnetism, etc. It supports a variety of specialized modules (e.g., structural mechanics, heat transfer, fluid dynamics, etc.), and provides flexible modeling tools and powerful numerical solvers for high-quality meshing and optimization of geometries ranging from simple to complex geometries. Squeeze film dampers can be simulated with COMSOL because it involves the coupled phenomenon of fluid dynamics and structural mechanics. COMSOL is able to accurately simulate the hydrodynamic effect of the oil film and its damping of structural vibration, providing detailed simulation results.

In the software, we set the boundary conditions as shown in Table and perform the modeling, and the picture after the modeling is completed is shown in Figure 4.

After modeling, the model is set to "structural mechanics-rotor dynamics-squeeze damper" type, mesh the model and start the computational simulation. Before starting the calculation, the initial eccentricity is set to 0, and the step size is 0.05, and the calculation ends at 0.95, in order to study the results of the variation of each parameter with the eccentricity in the range of 0 to 0.95.

Table

PARAMETERS OF SQUEEZE FILM DAMPER

Name	*Value*
Swirl speed	1.6667 1/s
Journal radius	0.2 m
Journal length	0.03 m
Clearance	0.001m
Attitude angle	0.17453 rad
Oil viscosity	0.02 Pa·s
Oil density	864 kg/m^3

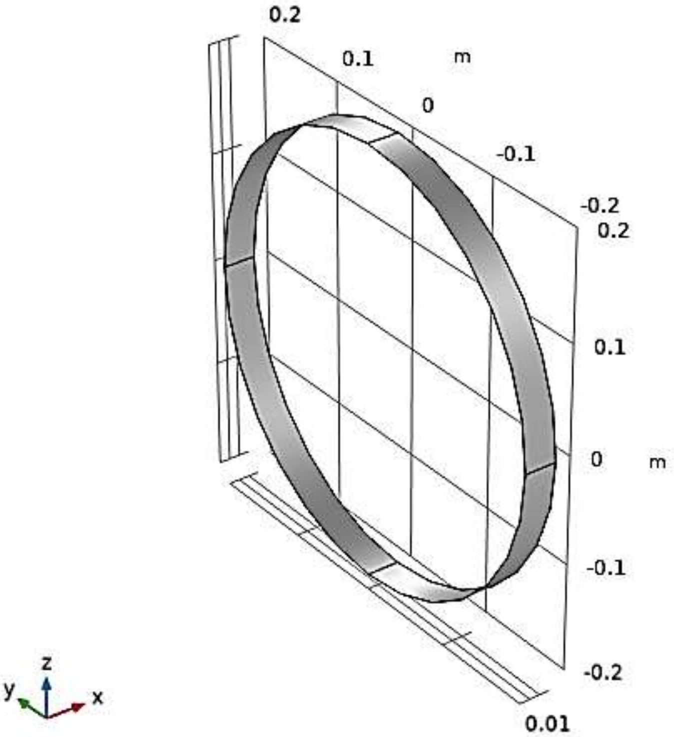


Figure 4. Modeling of squeeze film damper in COMSOL

Analysis of Results

In the study of squeeze film dampers, it is crucial to understand the relationship between the damping coefficient and the relative eccentricity because it directly affects the vibration control capability of the damper and the stability of the system. The relative eccentricity affects the oil film thickness and pressure distribution, and thus the damping coefficient, which is related to the optimization of the damper performance under different operating conditions. Studying this relationship not only optimizes the design and improves the operational reliability and efficiency of rotating machinery such as generators, but also aids in troubleshooting and maintenance by monitoring changes in the damping coefficient, detecting problems caused by changes in eccentricity at an early stage, and preventing more serious failures from occurring. The coordinates of the journal position within the damper in the localized direction are given by equation (1).

$$u_J = \begin{Bmatrix} 0 \\ C\varepsilon\cos(\theta) \\ C\varepsilon\sin(\theta) \end{Bmatrix} \tag{1}$$

where C is the clearance and ε is the relative eccentricity of the journal. The analytical values of the radial and tangential dimensionless damping coefficients are shown in equations (2)-(5) [8]:

$$C_{rr} = \frac{\pi(1+2\varepsilon^2)}{2(1-\varepsilon^2)^{5/2}} \tag{2}$$

$$C_{rt} = \frac{2\varepsilon}{(1-\varepsilon^2)^2} \tag{3}$$

$$C_{tr} = C_{rt} \tag{4}$$

$$C_{tt} = \frac{\pi}{2(1-\varepsilon^2)^{3/2}} \tag{5}$$

The damping coefficients are converted from global to local coordinates as shown in equation (6):

$$\begin{bmatrix} c_{22} & c_{23} \\ c_{32} & c_{33} \end{bmatrix} = \begin{bmatrix} \cos\theta & \sin\theta \\ -\sin\theta & \cos\theta \end{bmatrix} \begin{bmatrix} C_{rr} & C_{rt} \\ C_{tr} & C_{tt} \end{bmatrix} \begin{bmatrix} \cos\theta & \sin\theta \\ -\sin\theta & \cos\theta \end{bmatrix}^T \tag{6}$$

The variation of dimensionless damping number in four directions for different eccentricity is shown in Figure 5.

It can be seen from Figure 5, C_{22} and C_{33} increase with increasing eccentricity, C_{23} and C_{32} decrease with increasing eccentricity. In the range of eccentricity 0-0.4, the rate of change of the damping values in all four directions is very small; however, above 0.4, the damping values in all four directions show a large change, C_{22} changes the most，next is C_{33}，C_{23} and C_{32} curves coincide，It shows that the rate of change is almost the same from the beginning to the end.

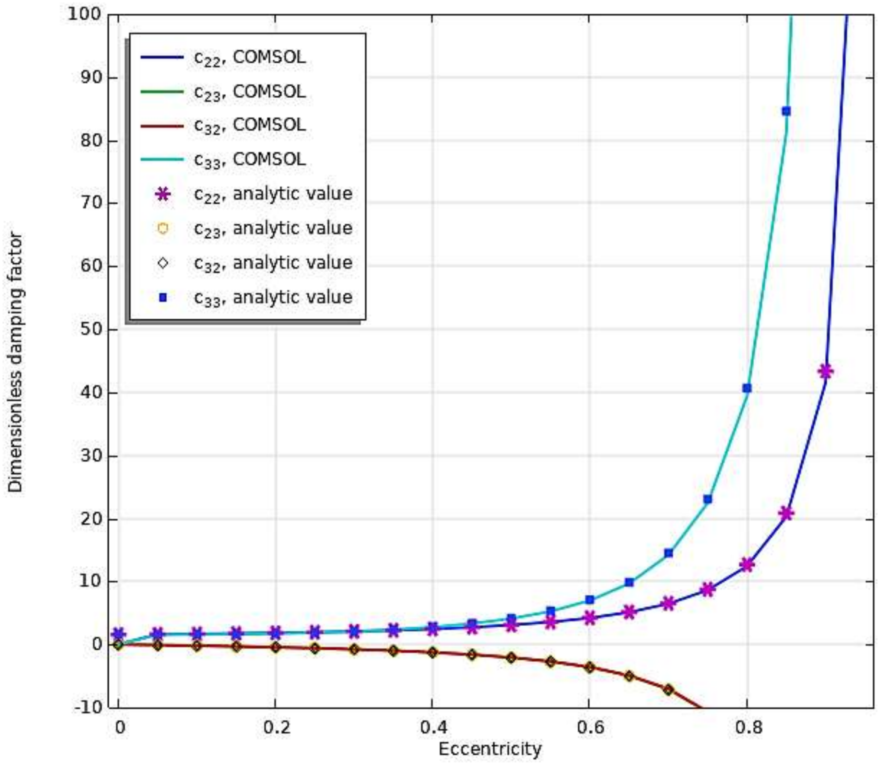


Figure 5. Plot of dimensionless damping coefficient versus eccentricity

In addition, the simulated value results are compared with the numerical analysis results, and it can be seen from the figure that there is almost no deviation of the simulated value results compared with the numerical analysis results, which indicates that the simulation has achieved good results.

The relationship between the stiffness coefficient and the eccentricity of the squeeze film damper (SFD) is typically determined through numerical simulation or experimental research. A common method involves using Rayleigh-Ritz [9] or finite element methods [10] to numerically simulate and derive the relationship between the stiffness coefficient and eccentricity of the SFD. In such cases, the stiffness coefficient is often a nonlinear function of the eccentricity, depending on factors such as the geometric shape of the SFD, fluid properties, and operating conditions. Understanding this relationship is crucial for comprehending the working principles of SFDs. In vibration control and mechanical system design, knowledge of how the SFD stiffness coefficient varies with eccentricity helps optimize designs and enhance system performance. Such research not only reveals the dynamic characteristics and fluid dynamics behavior of SFDs but also provides essential insights for practical engineering applications, ensuring system stability, reliability, and efficiency.

Figure 6 shows the variation of bearing stiffness with eccentricity. From Fig. 6, it can be seen that the stiffness values of the four directions have the value of 0 in the range of eccentricity 0-0.7, and after the eccentricity reaches 0.7, the stiffness values of zz and yy directions begin to increase, the stiffness values of zy and yz directions begin to decrease, and the rate of change of zz direction is larger than that of yy direction, and the curves of zy and yz directions overlap, which indicates that the rate of change is the same.

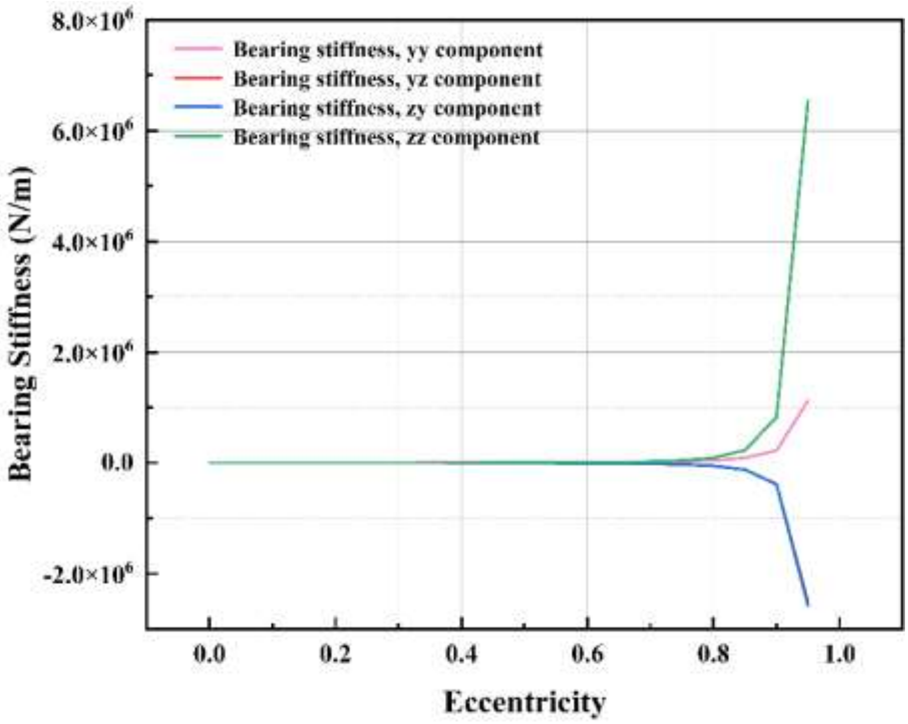


Figure 6. Plot of bearing stiffness versus eccentricity

Figure 7 shows the graph of displacement of journal relative to bearing with eccentricity. It can be seen from the figure that the displacement is linear in the x, y and z directions, and the displacement in the y direction gradually increases, the displacement in the x direction is zero, and the displacement in the z direction gradually decreases.

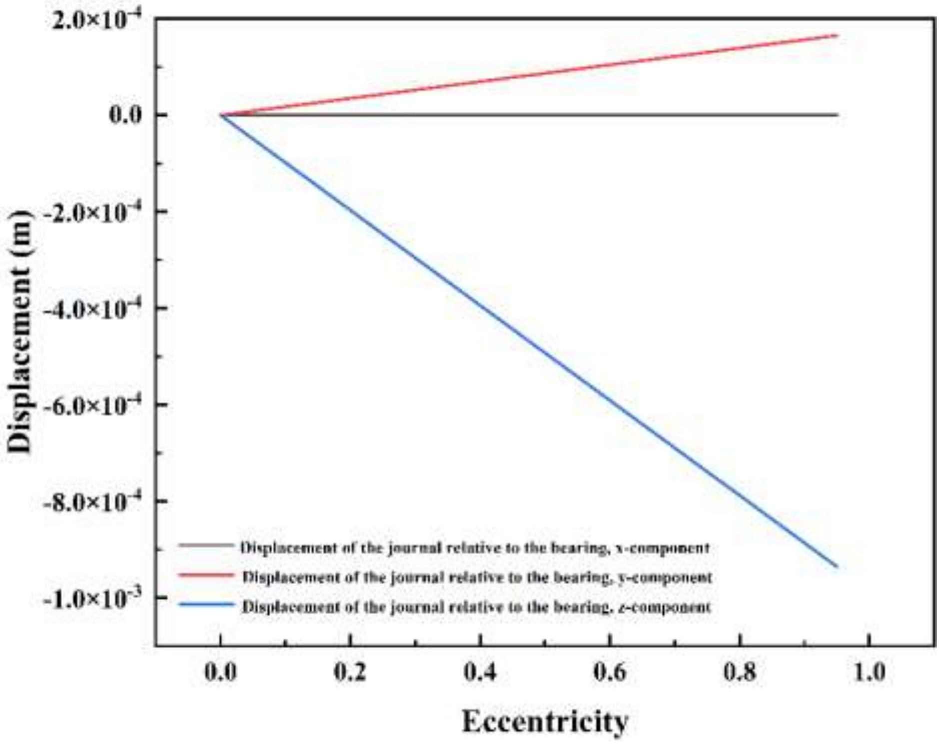


Figure 7. Plot of journal displacement relative to bearing versus eccentricity

Figure 8 shows the variation of fluid load on the journal. As can be seen from the figure, the fluid load in the x-direction is constant at 0. In the range of eccentricity 0-0.5, the fluid load in the y- and z-directions is also almost 0. However, after the eccentricity 0.5 a drastic change occurs: the fluid load in the z-direction starts to increase in the positive direction, and in the y-direction the fluid load starts to increase in the negative direction.

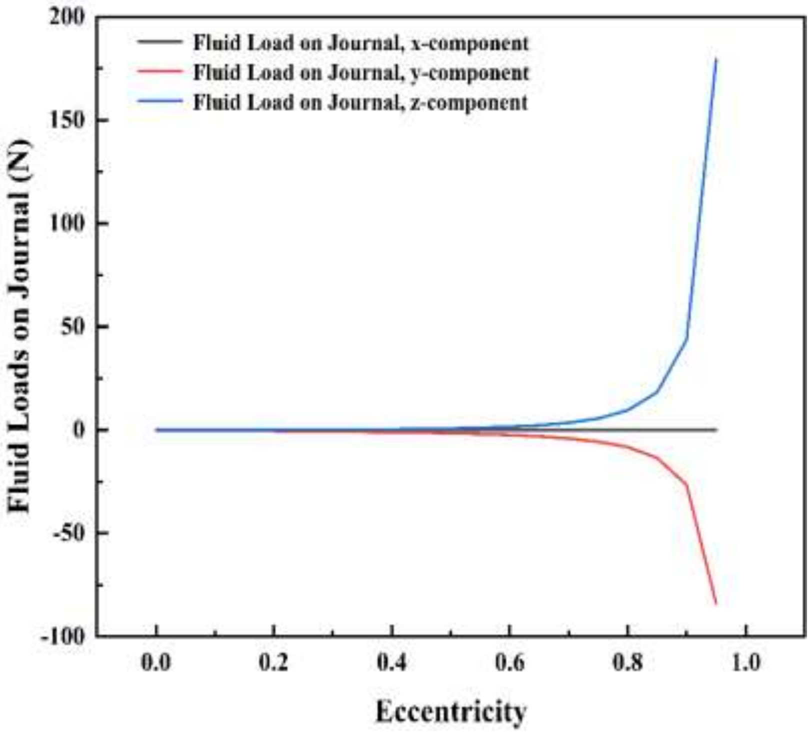


Figure 8. Variation of fluid loads on journal

Figure 9 shows the variation of fluid load on the bearing. As can be seen from the figure, the fluid load in the x-direction is constant at 0. In the range of eccentricity 0-0.5, the fluid load in the y-direction and z-direction is also almost 0. However, after the eccentricity 0.5 a drastic change occurs: the fluid load in the z-direction starts to increase in the negative direction, and in the y-direction the fluid load starts to increase in the positive direction.

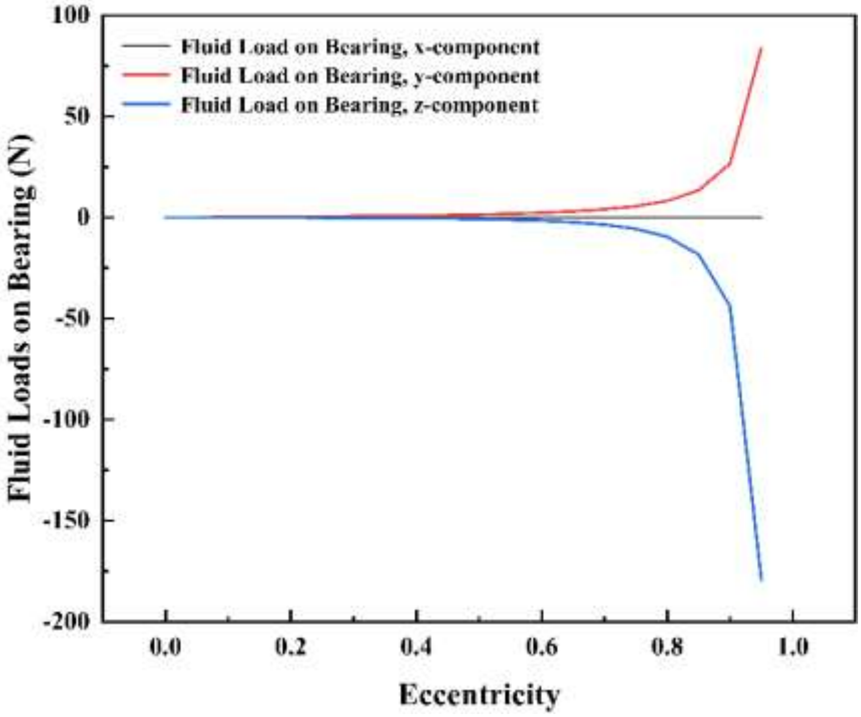


Figure 9. Variation of fluid loads on bearing

Figure 10 shows the fluid pressure distribution. From the figure, it can be seen that the presence of a large fluid pressure in one part of the damper causes a stress concentration.

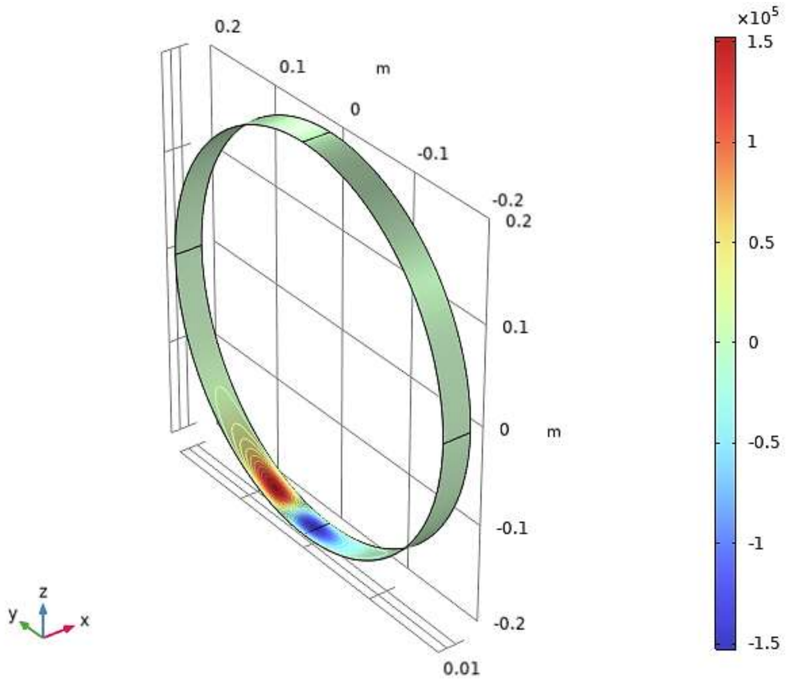


Figure 10. Fluid pressure

Figure 11 shows a plot of the average fluid rate. As can be seen from the figure, the places where there is a large change in velocity coincide with the places where the pressure is concentrated in Figure 10, indicating that the fluid mean rate has a large direct effect on the fluid pressure distribution.

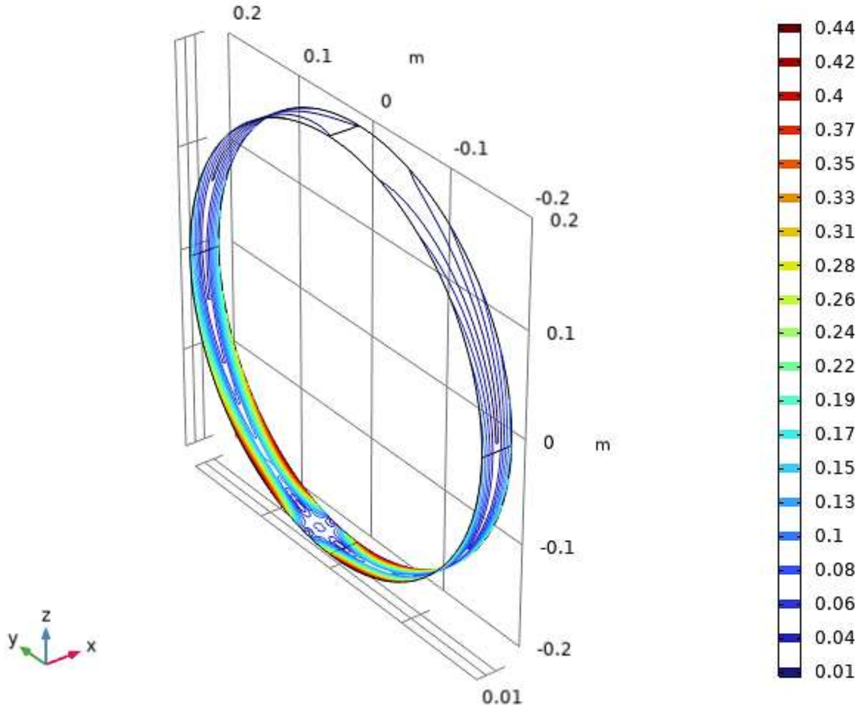


Figure 11. Mean flow rate

Conclusion

Based on the above simulation results, we can draw the following conclusions:

(a). Although the damping coefficients of C_{22} and C_{33} increase with the increase of eccentricity when the eccentricity is greater than 0.6, C_{32} and C_{23} will decrease with the increase of eccentricity, and the rate of change of each damping value is large, so if the eccentricity is chosen to be greater than 0.6, the damper is very likely to produce the consequences of uneven force and poor vibration damping, and it will not be able to satisfy the working conditions;

(b). When the eccentricity is greater than 0.7, the stiffness of the bearing in all directions will change dramatically, and the damper will undergo localized fracture, thus affecting the life of the damper;

(c). With eccentricity greater than 0.5, fluid loads can have a large effect on journals and bearings, which can lead to stress concentrations and potentially cause the damper to be scrapped;

(d). The flow rate of the oil inside the damper is inextricably linked to the pressure exerted by the fluid on the damper. To improve the damper's operating conditions and extend its service life, it is necessary to select a reasonable oil viscosity and density, improve the damper's material, and improve the damper's design structure.

In summary, although the design of the damper needs to uphold the principle of "increasing damping and reducing stiffness", but through the above analysis, with the change of eccentricity, the change of the parameters does not fully meet the design principles, so in the actual design process should be the first to pursue the balance, and on the basis of the balance of the improvement.

From the above analysis, it can be concluded that the eccentricity in the range of 0-0.5 is more appropriate, but taking into account in the 0.4-0.5 within certain parameters will occur some minor changes, so the eccentricity is set to less than 0.4 is the most appropriate.

References:

1. Cooper, S. (1963). Preliminary investigation of oil film for the control of vibration. *Institution of Mechanical Engineers*, (2), 305-315.

2. Gunter, E. J., Barrett, L. E., & Allaire, P. E. (1977). Design of nonlinear squeeze-film dampers for aircraft engines. https://doi.org/10.1115/1.3452990

3. Parsons, R. H. (1936). The development of the parsons steam turbine. *Nature*, 138, 567–568.

4. Beck, J. V., & Strodtman, C. L. (1967). Stability of a squeeze-film journal bearing. *Journal of Lubrication Technology, 89*(3), 369-373. https://doi.org/10.1115/1.3616994

5. Arauz, G. L., & Andres, L. S. (1996). Experimental study on the effect of a circumferential feeding groove on the dynamic force response of a sealed squeeze film damper. *Journal of Tribology*, *118*(4), 900-905. https://doi.org/10.1115/1.2831626

6. Xia, Nan, & Meng, Guang. (2002). A review and outlook on the dynamic characteristics of squeezed oil film damper bearings and rotating machinery rotor-squeezed oil film damper bearing systems. *Mechanical Strength, 24*(2), 216-224.

7. Heidari H. R., & Safarpourb P. (2016). Design and modeling of a novel active squeeze film damper. *Mechanism and Machine Theory*, (105), 235–243.

8. Chen, W. J. & Gunter, E. J. (2007). Introduction to the Dynamics of Rotor-Bearing Systems, sections 6.10 and 6.11. *Trafford Publishing*, 263–271.

9. Temple, G. & Bickley, W. G. (1933). Rayleigh 's principle andits applications to engineering, London.

10. Brenner, S. C., & Carstensen, C. (2004). Finite element methods. *Encyclopedia of computational mechanics, 1,* 73-114. https://doi.org/10.1002/9781119176817.ecm2003

Список литературы:

1. Cooper S. Preliminary investigation of oil films for the control of vibration. Institution of Mechanical Engineers, 1963.

2. Gunter E. J., Barrett L. E., Allaire P. E. Design of nonlinear squeeze-film dampers for aircraft engines. 1977. https://doi.org/10.1115/1.3452990

3. Parsons R. H. The development of the parsons steam turbine // Nature. 1936. V. 138. P. 567–568.

4. Beck J. V., Strodtman C. L. Stability of a squeeze-film journal bearing // Journal of Lubrication Technology. 1967. V. 89. №3. P. 369-373. https://doi.org/10.1115/1.3616994

5. Arauz G. L., San A. L. Experimental study on the effect of a circumferential feeding groove on the dynamic force response of a sealed squeeze film damper. 1996. https://doi.org/10.1115/1.2831626

6. Xia Nan, Meng Guang. A review and outlook on the dynamic characteristics of squeezed oil film damper bearings and rotating machinery rotor-squeezed oil film damper bearing systems // Mechanical Strength. 2002. V. 24. №2. P. 216-224.

7. Heidari H. R., Safarpourb P. Design and modeling of a novel active squeeze film damper // Mechanism and Machine Theory. 2016. №105. P. 235–243.

8. W.J. Chen and E.J. Gunter, Introduction to the Dynamics of Rotor-Bearing Systems, sections 6.10 and 6.11 // Trafford Publishing. 2007. P. 263–271,

9. Temple G., Bickley W. G. Rayleigh 's principle andits applications to engineering, London 1933.

10. Brenner S. C., Carstensen C. Finite element methods // Encyclopedia of computational mechanics. 2004. V. 1. P. 73-114. https://doi.org/10.1002/9781119176817.ecm2003

Работа поступила в редакцию 30.05.2024 г.

Принята к публикации 10.06.2024 г.

Ссылка для цитирования:

Dong Jiaqi, Bazhanov A. Simulation Analysis of Non-concentric Squeeze Film Damper // Бюллетень науки и практики. 2024. Т. 10. №7. С. 343-354. https://doi.org/10.33619/2414-2948/104/36

Cite as (APA):

Dong, Jiaqi, & Bazhanov, A. (2024). Simulation Analysis of Non-concentric Squeeze Film Damper. *Bulletin of Science and Practice, 10*(7), 343-354. https://doi.org/10.33619/2414-2948/104/36