# 고속 안정 운전을 위한 스퀴즈 필름 댐퍼

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# Squeeze Film Damper for High Speed Stable Operation

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**Abstract** : Four new types of squeeze film dampers have been experimentally investigated to develop a practical and effective damper for flexible rotors. The cylindrical bush damper was selected as the best one among them and applied in a test rotor supported on ball bearings. A stable running of the rotor exceeding the critical speed was observed.

**초 록** : 탄성회전축을 위한 실제적이고 효과적인 댐퍼를 개발하기 위하여 4개의 새로운 스퀴즈 필름 댐퍼들이 실험적으로 연구되었다. 이들 중에서 원통형 부쉬 댐퍼가 가장 효과적인 것으로 선택되었고 볼베어링으로 지지된 실험 회전축에 적용되었다. 이러한 회전축은 위험속도를 지나서 안정된 운전이 관찰되었다.

**Key Words** : cylindrical bush, squeeze film damper, rotor bearing system, critical speed

## 1. Introduction

Increasing the rotational speed of machine tools and turbo machinery for high power to weight ratio has always been a problem because of the critical speed or unstable speed range of the rotor-bearing system. A rotor supported on rolling bearings shows extremely large amplitude near the critical speed due to poor damping of the bearings, so that the operating speed is limited by this critical speed. On the other hand, the amplitude of the rotor supported on journal bearings is relatively small at the critical speed because the damping ratio of the bearings is large. There exists, however, an unstable speed range above the critical speed.

A drastic increase of the critical speed or the unstable speed limit by changing design parameters related to the shape of rotor and bearings is difficult to achieve. Among several approaches, nevertheless, the application of a squeeze film damping device for the reduction of the magnification of amplitude due to critical speed or unstable vibration is known to be the simplest and most acceptable solution.

Since Coopers[1] demonstrated the squeeze film in practice, many researchers have investigated the squeeze film damping of a rotor dynamic system[2-10]. They have developed the mathematical models of the rotor-bearing system with the squeeze film damper and analyzed the damping ratio of the damper using the hydrodynamic lubrication theory for an infinitely short or long journal bearing.

The results have shown that the damper has excellent damping characteristics when the system is in the range of the critical speed. Some problems, however, still remain in the design of the optimal squeeze film damping device. For example, the flexible support element shown in Fig. 1 must be stiff enough to maintain the oil film gap between the support element and the outer surface of the shaft-supporting bearing. The rotor amplitude depends on the smallest gap height of the squeeze film damper and the smaller the gap height, the

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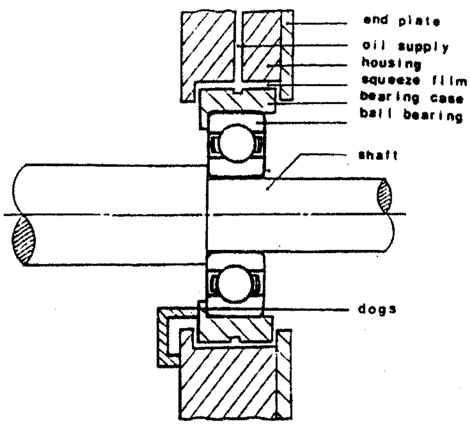


Fig. 1. Rotor with a squeeze film damping element

larger is its damping ratio. At the same time, the support element should have enough flexibility because it also has a positive effect on the rotor amplitude. Therefore, it is not easy to find the optimal shape and dimensions of the flexible support element and squeeze film damping device.

In this paper four damping elements are applied in a rotating rotor-bearing system and their damping characteristics are developed into a compact damping element for high damping effects with the proper modeling to calculate the damper characteristics. Such an element should easily be applicable to many rotating machine without much revision to the original design.

## 2. Comparsion of the Damping Characteristics

For a compact design of a damping element between the rolling element bearing and its support housing, the bending bar of the flexible support element in Fig. 1 can be replaced by some slender pins around the bearing[2]. The bearing can easily be located at the center by these pins. However, the flexibility of the support element is limited by the minimum diameter of the pin which has to be determined by considering the fatigue limit of the material. The mounting of the pins can also be a problem.

Squeeze film dampers without axial alignment devices can also operate in the rotating rotor-bearing system. In this case the vibration amplitude of the bearing will be limited by the squeeze film gap.

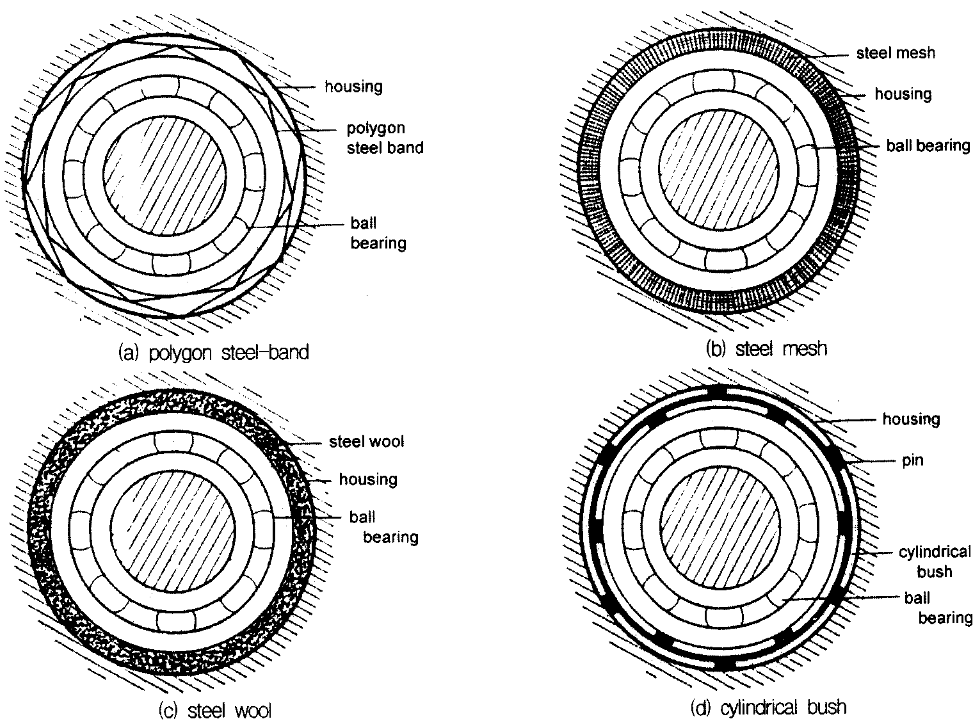


Fig. 2. Squeeze film damping elements

A schematic of the squeeze film damper without axial alignment devices is shown in Fig. 2. The polygon steel band damper is made by wrapping the edged steel band on the bearing surface; the edge of the band in the first layer lies on the straight line of the band in the second layer. The oil in the triangular space between the band layers operates as a squeeze damper.

The steel mesh and the steel wool dampers shown in Fig. 2 operate as effective dampers while the oil exits through the many narrow passages of the steel mesh or steel wool by compressing the mesh layer. The fourth one is made by inserting the cylindrical steel band and a few blocks between the bearing band and the housing. These blocks make a double layer of thin squeeze film between the bearing and the housing. This type of damper was developed by Glienickel[1] for a self-acting air bearing. Although the stiffness of these four types of squeeze dampers is negligibly small, the vibration amplitude depends on the gap height, and therefore, can be controlled as a function of the gap height.

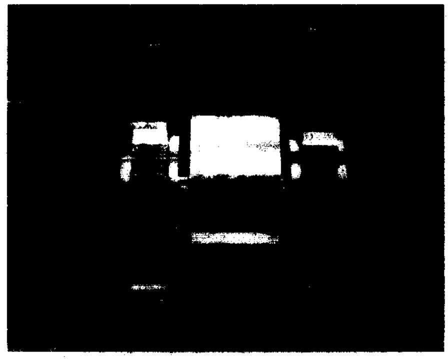

Fig. 3. Test rig for measuring the damping characteristics

For measuring the damping characteristics of the dampers a test rig of a mass-spring-damper system was set up as shown in Fig. 3. A 24.3kg short rotor fitted on the ball bearings with 45mm inner diameter(KBC6209) is supported on the squeeze film dampers and installed in the housings. A gap measuring pickup was mounted on one end of the rotor and the vibrating motion of the rotor due to an impulse at the middle of the rotor was measured. The signal from the pickup was moni-

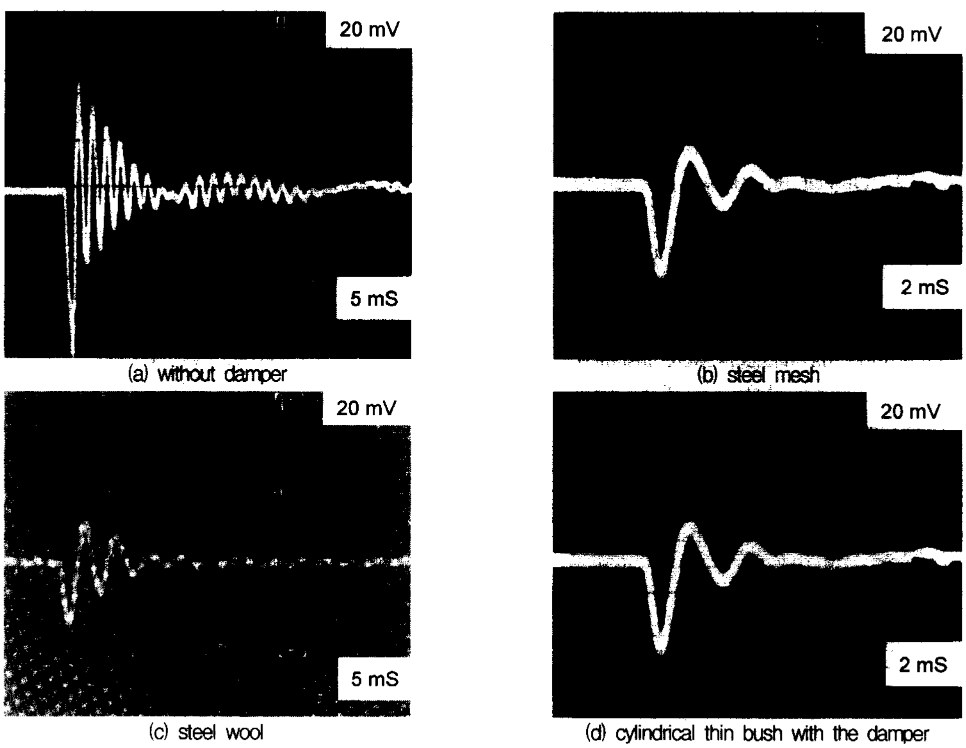


Fig. 4. Vibratory radial motion of the rotor supported on the ball bearing KBC 6209 (a) without damper and on (b) steel mesh, (c) steel wool and (d) cylindrical thin bush with the damper

Table 1. Measured damping ratio of different dampers installed on ball bearing 6209(d = 45mm)

| Dampers | Radial Gap [mm] | Damping Ratio $\zeta$ | Oil |
|---|---|---|---|
| Ball bearing without damper | 0 | 0.003 | SAE 30 |
| Polygon steel band | 0.4 | 0.009 | " |
| Steel mesh | 0.5 | 0.05 | " |
| Steel wool | 0.8 | 0.11 | " |
| Cylindrical bush | 0.4 | 0.15 | " |

tored on the oscilloscope.

Fig. 4 shows the vibrating radial motion of the rotor supported on a different damper for the impulse test. It can be seen that the polygon steel band damper has poor damping characteristics, while the others have the desired damping ratio. Using the test data and the response characteristics, damping ratios for different dampers were estimated Table 1 shows the radial gap and the measured damping ratio $\zeta = \frac{d}{2cm}$ for various types of dampers.

Since the damper with a cylindrical thin bush of these tests is shown to have the best damping effect and the damping characteristics of the squeeze film can be calculated explicitly, this type of damper was selected as a practical damper model for further investigation. Installation of the damper is very easy. One inserts the thin bush first and then the blocks. Depending on the thickness of the blocks, the shape of the bush can be circular or wavy.

## 3. Application in a Rotating Shaft

To confirm the quality of the bush damper in a practical application, the dampers were installed in the bearings of a rotor and the behavior of the rotor vibration was measured.

The dimensions of the rotor are shown in Fig. 5. The calculated stiffness of the rotor is 7924 N/mm and the natural frequency is $543/2\pi$ cps.

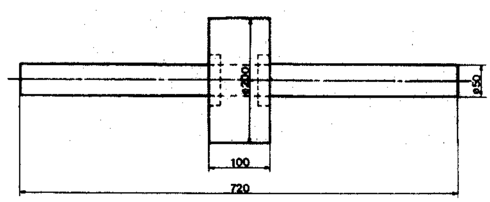


Fig. 5. Dimensions of the test rotor

The outer diameter of the ball bearings is 85mm and the width is 19mm. The stiffness of the bearings is about 68,000N/mm. Fig. 6 shows the installation of the bush damper between the bearing and the housing. The inner diameter of the bush is 0.2mm. The height of the blocks is 0.125mm. These blocks are made of plastic tape and inserted into the gap between the bearing, the band and the housing.

The instrumentation of the rotor test rig is shown in Fig. 7. The vibrating motion of the rotor is measured by the gap measuring pickups in the horizontal and vertical direction. The reaction force of the bearing is measured by a dynamic load cell.

The rotating speed of the test rotor varied from 0 to 10,000rpm as shown in Fig. 8. The measured amplitude curve of the rotor for this speed range is plotted in Fig. 9. The critical speed was 4,800rpm and the amplitude peak of the journal measured about 100 $\mu$m, which is half the total radial clearance of the squeeze film damper. The amplitude of the rotor over the speed range of 7,000rpm was about 7 $\mu$m, which is believed to be the unbalanced radius of the rotor. Using these test data, the estimated damping constant of the damper is $d_a$ = 350Ns/mm. This is larger than the estimated damping constant of $d_a$ = 290Ns/mm for the earlier test shown in Fig. 3.

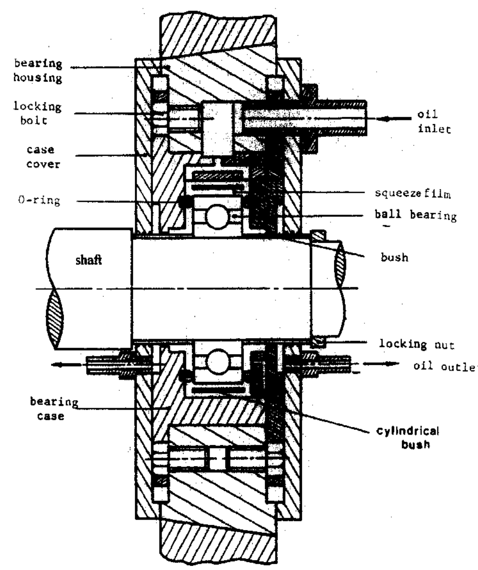


Fig. 6. Cross section of the bush damper in a bearing

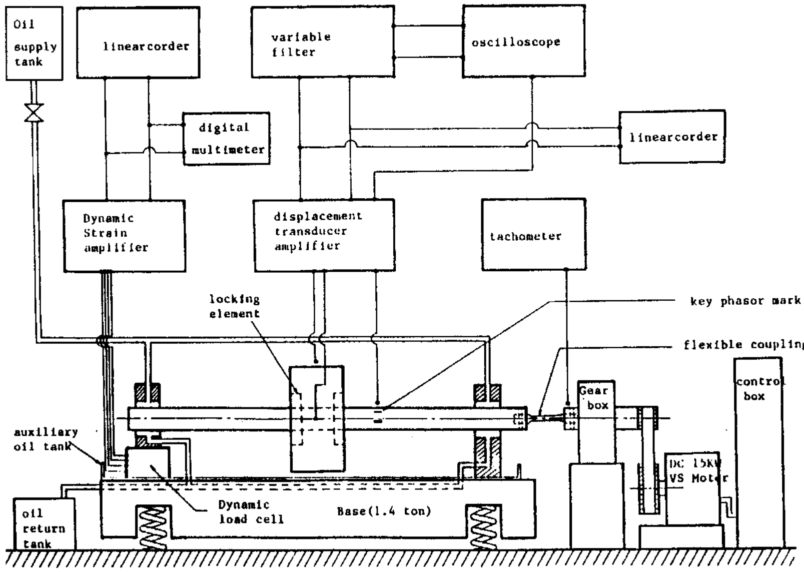

Fig. 7. Instrumentation of the rotor test rig.

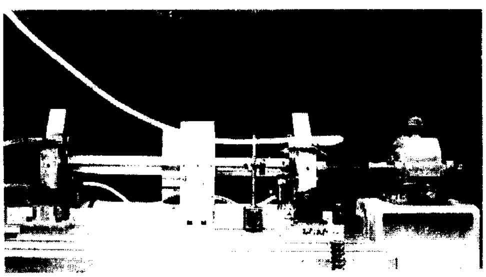
Fig. 8. Test rig of rotor bearing system

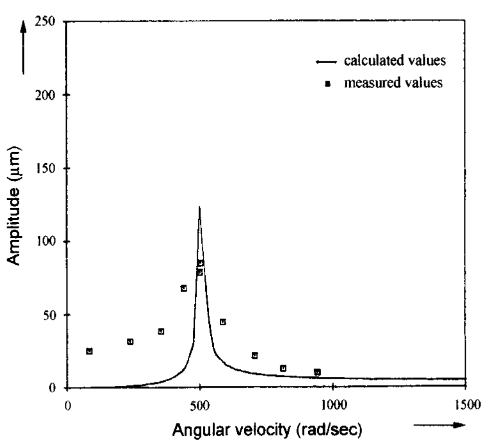

Fig. 9. Measured and calculated amplitude curve of the test rotor

The calculated amplitude curve of the test rotor is compared with the measured curve in Fig. 9. It can be seen that the calculated amplitude curve is in good agreement with that measured at the critical speed. The analysis of the forced vibration was done by the transfer matrix method[12].

## 4. Conclusion

Among the various squeeze film dampers the thin shell bush damper has shown the largest damping effects and therefore, chosen as the most practical one for design and installation on rolling element bearings.

The amplitude peak of journal in the damper at the critical speed of a flexible rotor supported on these dampers was limited by the gap height of the squeeze film thickness. A quiet running of the rotor up to 10,000rpm was observed. Optimal design of squeeze film dampers for a rotor-bearing system can be achieved by changing the gap height, the damper width and the oil viscosity.

## Reference

1) Cooper, S., "Preliminary Investigation of Oil films for the Control of Vibrations", IMechE, Lubrication and Wear Convention, Paper 28, pp. 305~315, 1963.

2) R.E. Cunningham, D.P. Fleming, E.J. Gunter, "Design of a Squeeze-Film Damper for a Multi Mass Flexible Rotor", ASME, J. of Engineering for Industry, 1975.
3) A.Z. Szeri, A.A. Raimondi, A. Giron Duarte, "Linear Force Coefficients for Squeeze film Dampers", ASME, J. of Lubrication Technology, 1983.
4) D.P. Fleming, "Dual Clearance Squeeze Film Damper for High Load Conditions", ASME, J. Tribology, 1985.
5) 윤석철, "고속 회전축 베어링계의 스퀴즈 필름 댐퍼에 관한 연구", 서울대학교, 1989.
6) 윤석철, "절단 원추형 전기점성 SFD 베어링연구", 산업안전학회지, Vol. 18, No. 3. pp. 200, 2003.
7) R.A. Cookson, S.S. Kossa, "The Vibration Isolating Properties of Uncentralized Squeeze film Damper Bearing Supporting a Flexible Rotor", ASME, J. of Engineering for Power, 1981.
8) J.W. Lund, "Review of the concept of Dynamic Coefficients for Fluid Film Journal Bearings", ASME, J. of Tribology, 1987.
9) R.D. Ramli, J.B. Rogerts, J. Ellis, "Determination of Squeeze film Dynamic Coefficients from Experimental Transient Data", ASME, J. of Tribology, 1987.
10) D.W. Parkins W.T. Stanley, "Characteristics of an Oil Squeeze Film", ASME, J. of Lubrication Technology, 1982.
11) Glienicke, J., "Aerodynamisches federndes Mehrgleit flachenlager", Deutsches Patent DE 2909973 C2 C21. 12 1982.
12) D.C. Han und A. Meyer, "Querschwingungen", Forschungsheft Forschungs -vereinigung Antriebs technik Heftl, 1983, Frankfurt.
13) G.L. Arauz and L.A. San Andres, "Experimental Pressures and Film Forces in Squeeze Film Damper", ASME, J. of Tribology, Vol. 115, 1993.