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ACTIVELY CONTROLLED BEARING DAMPERS FOR AIRCRAFT ENGINE APPLICATIONS

John M. Vance
Mechanical Engineering Department
Texas A & M University

Daniel Ying
Turbine Manufacturing Division
ABB Power Generation, Inc.

Jorgen L. Nikolajsen
Staffordshire University

ABSTRACT

This paper describes some of the requirements for bearing dampers to be used in an aircraft engine and briefly discusses the pros and cons of various types of dampers that were considered as candidates for active control in aircraft engines. A disk type of electrorheological (ER) damper was chosen for further study and testing. The paper explains how and why the choice was made. For evaluating potential applications to aircraft engines, an experimental development engine (XTE-45) was used as an example for this study.

Like most real aircraft engines, the XTE-45 ran through more than one critical speed in its operating speed range. There are some speeds where damping is desirable and other speeds where it is not. Thus, the concept of a damper with controllable forces appears attractive. The desired equivalent viscous damping at the critical speeds along with the available size envelope were two of the major criteria used for comparing the dampers.

Most previous investigators have considered the ER damper to produce a purely Coulomb type of damping force and this was the assumption used by the present authors in this study. It is shown in a companion paper (Vance and San Andres, 1999), however, that a purely Coulomb type of friction cannot restrain the peak vibration amplitudes at rotordynamic critical speeds and that the equivalent viscous damping for rotordynamics is different from the value derived by previous investigators for planar vibration. Control laws for Coulomb damping are derived in Vance and San Andres, (1999) to achieve minimum rotor vibration amplitudes in a test rig while avoiding large bearing forces over a speed range that includes a critical speed. The type of control scheme required and its effectiveness was another criterion used for comparing the dampers in this paper.

INTRODUCTION

A disk type of electroviscous damper, utilizing fluid with electrorheological (ER) properties, was invented in 1988 by J. Nikolajsen (see Nikolajsen and Hoque, 1988). Figure 1 shows a cross section of the damper, with six thin nonrotating disks moving with the outer race of a ball bearing and with five nonrotating disks attached to the housing and sandwiched in between. Since the ball bearing is on a squirrel cage (flexible) support, transverse rotor motion at the bearing shears the fluid between the disks. Nikolajsen and Hoque (1990) constructed a rotordynamic test rig and demonstrated experimentally that the damper could substantially reduce the peak rotor vibration amplitude at a low critical speed (1400 rpm). They also demonstrated that varying the voltage applied across the disks could control the effective damping. They hypothesized that active control could eliminate the critical speed response by changing the effective system stiffness (not the damping) at appropriate speeds during runup of the rotor. With regard to the damping, they state that "the electroviscous fluid provides Coulomb-type friction damping at a variable rate controlled by a DC voltage applied to the fluid".

It has been known for about a century (Duff, 1896) that electrorheological (ER) fluids can be

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concocted with shear properties that are controllable, by using an imposed electric voltage. Nikolajsen and Hoque (1990) give some of the history of their development and give a useful description of one class of these ER fluids as shown in Figure 2. In such fluids, the applied voltage converts them from a Newtonian fluid to a Bingham plastic, where the viscosity is not changed but a positive shear stress must be applied before the fluid breaks loose and begins to shear. A high enough voltage makes the fluid appear to be solid.

Shortly after Nikolajsen's invention, a study was carried out to determine the feasibility of developing actively controlled bearing dampers for aircraft engines. The ER damper was one of several candidate devices to be considered. This paper describes some of the requirements for bearing dampers to be used in an aircraft engine and briefly describes the pros and cons of the various dampers that were considered for active control. Vance and San Andres (1999) explain how active control could be implemented with the ER damper if the Coulomb model is a correct representation.

Vance and Ying (1999), describe experiments with the actively controlled ER damper, carried out after this study was completed, that explored some of the practical rotordynamic considerations by conducting higher speed tests and implementing two different control schemes. Their experiments revealed the complex nature of the damping forces obtained from an ER bearing damper. The results of the experiments were not particularly encouraging in a practical sense, neither for the ER damper nor for the idea of actively controlled dampers. Therefore there was no real incentive for publication at the time. However, the first author continues to receive requests for information about this research from people who have heard of it. The contemporary enthusiasm for active control of almost every thing is not lost on bearing researchers, especially in the academic community. It begs a description of the challenges faced, the lessons learned, and the disappointments encountered in the study to be reported here, and in the paper describing the experiments (Vance and Ying, 1999).

Most previous investigators considered the ER damper to produce a purely Coulomb type of damping force and this was the assumption initially used by the present authors in this study, before the experiments were performed.

BEARING DAMPER REQUIREMENTS FOR AIRCRAFT ENGINES

Squeeze-film dampers (SFD) have been used extensively in almost all aircraft turbines designed since 1970 to damp imbalance response, and are probably a major contributor to the rarity of rotordynamic instability encountered in these engines. However, the SFD has some shortcomings that are difficult to overcome. For example, recent testing of an aircraft engine rotor-bearing system by the first author showed a large increase in rotor response when the oil temperature was raised from 80°F to 247°F. According to well established SFD theory, the damper actually increases the response whenever the local rotor imbalance exceeds 2.3 times the damper clearance. Also, nonlinear phenomena (bistable jump up) has been reported and extensively investigated (Vance 1988). Twenty-five years of applied SFD research has failed to produce reliable design tools that can accurately predict the performance of any except the simplest geometry under laboratory conditions. Realistic conditions include high speeds and low supply pressures relative to the peak pressure in the oil film. Cavitation and air entrainment under these conditions produce a two phase air-oil mixture that greatly reduces the direct damping coefficient (Zeidan and Vance, 1989, 1990) and that is not highly repeatable even under apparently identical operating conditions.

Rotor damping at the bearings is most needed near critical speeds and is relatively ineffective at other speeds as shown by the classic family of response curves in Figure 3. The three curves are for three different values of damping. Rotor damping increases the dynamic bearing loads at supercritical speeds as shown in Figure 4, so an ideal damper should have its damping disappear at speeds where the viscous forces are deleterious. The SFD actually has this characteristic due to the air entrainment, a feature that has not been widely appreciated. To the contrary, several researchers including the first author have suggested in the past that raising the supply pressure could make the SFD more effective and more predictable by eliminating cavitation. (Zeidan and Vance, 1989, 1990). But this would lose the desirable characteristic of low bearing forces at high speeds.

ACTIVE DAMPERS FOR AIRCRAFT ENGINES

The first experiments with the ER damper by Nikolajsen and Hoque (1990) demonstrated that the ER damper has a high damping capability and offers controllability of the rotor response at the first critical speed (which was a rigid body pitching mode in their experiments). One year later a funded research project titled "Development of Active (Variable Rate) Rotor System Dampers" was carried out to investigate the feasibility of active controlling bearing damping in aircraft engines. The ER damper concept with the control system was eventually selected as the prototype active damper from seven active damper concepts (SFD, Hydrostatic SFD, ER damper, Piezoelectric Pusher Damper, DC Electromagnetic Damper, AC Electromagnetic Damper and Gas Damper).

The No.1 and No.3 bearings of General Electric's XTE45-1 engine were chosen for evaluating the seven candidate devices including the ER damper. The XTE-45 was an experimental aircraft gas turbine engine with dual rotors, high pressure (HP) and low pressure (LP). It was an early test bed for the "Integrated High Performance Turbine Engine Technology" (IHPTET) program, a long-range development project jointly funded by government and industry (Valenti, 1995). Bearings 1 and 3 are on the LP rotor and HP rotor respectively.

One of the important criteria in selecting a damper was sizing, to fit within an available envelope and still be able to produce the desired bearing forces. The engine used for the feasibility study had approximate design parameters and requirements for bearing dampers as shown in Table I. Like most real aircraft engines, the XTE-45 ran through more than one critical speed in its operating speed range. There are some speeds where damping is desirable and other speeds where it is not. The desired damping in Table I refers to the value of equivalent viscous damping at the critical speed. Bearing damping (and radial stiffness) in any aircraft engine also becomes more desirable under maneuver and shock loads (e.g. landings). The available restraint force in aircraft maneuvers is also important (see Ying, 1993). Thus, the concept of a damper with controllable forces appears attractive.

CONTROL SCHEMES FOR BEARING DAMPERS

Controllable bearing dampers can be conveniently categorized into two classes: Class I has indirect control in the sense that a design parameter (such as the viscosity in a passive viscous damper) is varied based on intermittent measurements of rotor vibration (or speed if the response curve is known *a priori*). For example, the rotor orbit amplitude might be measured once per rotor revolution and the measurement used to control a linear damping coefficient. A more appropriate name for the class I type might be parametric control. Class II has instantaneous forces on the bearing directly and continuously controlled by feedback signals from the rotor motion with very rapid response to the signals. Both classes were initially included in the study reported here, but the class II candidates were dropped early on because of: 1) challenging requirements for rapid data acquisition and rapid actuator response, 2) stability issues, and 3) the expected higher cost.

The squeeze film damper was a Class I candidate because it had been found in previous research that the direct damping coefficient is controllable with the oil supply pressure to the damper (Zeidan and Vance, 1989, 1990). Increasing the supply pressure reduces the air entrainment and cavitation, thus increasing the damping. However, the precision with which this control could be achieved with variable speed was (and still is) questionable and this was considered to be a major disadvantage of the SFD as a controllable device. The hydrostatic SFD would require a high pressure oil pump and had never been tested in the laboratory. It was extensively analyzed (San Andres, 1992) but was eliminated from consideration mainly because of a lack of any experimental knowledge about its performance.

The disk type ER damper was an obvious Class I candidate. At the time of the comparison study it was generally believed that it produced Coulomb damping, the strength of which could be easily controlled by varying the voltage across the disks.

If one believes that the ER damper produces Coulomb damping then it is logical to question whether it would be more cost effective to use a damper with dry friction elements, rather than dealing with high electrical voltage and exotic fluid concoctions. Therefore, a dry friction damper with controllable friction force was also considered and a laboratory test rig was built by San Andres (see Figure 5 and see Vance and San Andres, 1999, for a theoretical analysis). The experiments yielded some good damping results but some of the demonstrated practical disadvantages are 1) wear of the friction surfaces and 2) changes of the friction coefficient with temperature and surface contaminants. Both of these disadvantages would make a reliable control scheme difficult to implement.

As described in the next section, the eddy current damper would be best suited to Class I control. The AC and the DC electromagnetic dampers were Class II candidates since they are essentially magnetic bearings with continuous velocity feedback. The piezoelectric pusher damper was also a Class II candidate and had some stability problems at the time of this study. It was the subject of fairly intensive research by Palazzolo (1989). The gas damper was a Class I candidate. It was controllable by bleedoff air from the compressor and the engine performance penalty from this was unattractive. A prototype was built and demonstrated considerable damping capability in the laboratory (Sundararajan and Vance, 1995), but the hardware was more than twice as heavy as a traditional SFD for the same size application.

DESIGN CALCULATIONS AND TEST RESULTS FOR SOME OF THE DAMPERS

The calculation objective for each of the candidate bearing dampers was to find the lightest and smallest configuration that could meet the conditions of Table I. Realistic design calculations for DC-, AC-, and EC-dampers require sophisticated electromagnetic finite element analyses for which the facilities and expertise were not immediately available. Therefore, the field was narrowed through a preliminary evaluation, including a literature survey of the operating principles.

DC-Damper: The actuator of a simple DC electromagnetic damper is shown in Figure 6. The shaft is supported by a rolling element bearing whose outer race is secured in a ferromagnetic sleeve that is flexibly supported by a squirrel cage and surrounded by electromagnets. Both the sleeve and the electromagnet cores are made of sufficiently thin radial laminations to ensure a flat frequency response up to the maximum rotor speed. The electromagnets are the actuators in a feedback control system (not shown) which senses the shaft vibrations and produces radial forces on the shaft proportional to the velocity of the shaft vibrations, i.e., viscous damping forces. A considerable body of literature exists concerning the design of DC-dampers. Nikolajsen et al. (1979) performed one of the first such studies which confirmed very good damping performance, both analytically and experimentally.

The size of DC-dampers, required for the XTE-45, was approximated based on sales literature on DC electromagnetic *bearings* (Magnetic Bearings Inc. 1986). The resulting damper sizes exceeded the available envelope. The required actuator masses were 16.6 kg and 32.7 kg for the #1 and #3 bearings respectively resulting in specific damping capacities of 4.2 kNs/m/kg and 2.1 kNs/m/kg. The permissible temperature range for the actuator is -250°C to 450°C according to the sales literature. The reliability limited by the control electronics and the power consumption is relatively high due to power electronics and actuator losses. This is a Class II damper.

AC-Damper: This damper looks similar to the DC-damper in Figure 6 but the ferromagnetic sleeve is replaced with a conducting sleeve (e.g., Aluminum) and the electromagnets are fed with AC power instead of DC power. This results in repulsion between the electromagnets and the sleeve instead of attraction. The advantage is inherent stability, thus, simpler and more reliable electronics and the possibility of Class I operation. The AC-damper is inherently stable because the repulsive force diminishes as it pushes the sleeve away. The DC-damper must be stabilized electronically because the attractive force increases as it pulls the sleeve nearer. Class II operation, however, would be required to ensure adequate damping since the inherent damping capacity of the AC-damper is extremely low. Nikolajsen (1989) demonstrated this experimentally for an AC-bearing. Also, the repulsive AC force is only approximately half the size of the attractive DC force. This is due to (1) substantial flux leakage and high reluctance of the magnet-core/sleeve magnetic circuit, (2) the alternating nature of the flux density created by the AC-current, and (3) the resistivity of the conducting sleeve, leading to high eddy-current losses. The required weight and actuator envelope around the XTE-45 bearings were both estimated to be more than twice the DC-damper requirements and the power consumption was estimated to be an order of magnitude higher than for the DC-damper. The AC-damper was rejected on this basis.

EC-Damper: The actuator of a simple EC (eddy current) damper is shown in Figure 7. The shaft is supported by a rolling element bearing whose outer race carries a conducting disk (e.g., aluminum) which is flexibly supported by a squirrel cage. A ring of DC electromagnets face both sides of the disk. The electromagnets are the actuators in a Class I feedback loop (not shown) which senses the shaft vibrations intermittently and produces a proportional DC-current in the coils which creates a magnetic flux path through the magnet cores and the disk. When the disk vibrates, the magnetic flux lines penetrating it gives rise to eddy-currents in the disk which are directed such that the resulting electromagnetic forces try to prevent the vibrations. The result is radial restoring forces on the disk which are proportional to the velocity of the vibration, i.e., viscous damping forces. A considerable body of literature exists on the design of EC-dampers, see for example Cherry (1960). One obvious advantage of both the EC-damper and the ER-damper over the DC-, and AC-dampers and SFD for aircraft applications is that the damping capacity does not deteriorate at large radial displacements. This is useful for maintaining the damping capacity during shock loading (e.g., hard landing) and after a blade loss when high and potentially destructive vibration amplitudes need to be attenuated.

The size of the EC-dampers needed for the XTE-45 was approximated based on experimental work with a simple EC-damper rig, see Figure 8. The cantilevered Aluminum beam has a perpendicular aluminum plate attached at the free end. The plate moves freely in the air gap between the permanent magnets, cutting the magnetic flux lines when the beam vibrates in its first bending mode. Thus, eddy-currents are generated in the plate resulting in viscous drag on the plate. The natural frequency (f_n) in Hz and the logarithmic decrement of the first bending mode are measured by impact testing. The beam is regarded as a 1 D.O.F. system such that the damping coefficient (c) can be written as

$$c = \zeta k/(\pi f_n) \qquad (1)$$

where ζ is the critical damping ratio, and

$$k = 3EI/l^3 \qquad (2)$$

is the static beam stiffness. The eddy-current damping is determined by subtracting the results from testing with and without the magnets.

Beams of different lengths were used to produce natural frequencies f_n of 50 Hz, 192 Hz and 225 Hz corresponding to 3,000 rpm, 11,500 rpm, and 13,500 rpm. Two plate thicknesses of 2.54 mm and 5.08 mm were used. The vibration amplitude was kept of the order of 0.25 mm to simulate gas turbine engine conditions. The results are shown in Figure 9. Note the following typical characteristics of EC-damping:

- The damping coefficient increases with vibration amplitude (except at 50 Hz where the accelerometer sensitivity is believed to have deteriorated to the extent where the trend is considered unreliable).

- The damping coefficient increases with plate thickness. The depth of penetration of eddy-currents into the plate at 250 Hz was estimated to be about 6 mm. The plate therefore needs to be at least 6 mm thick to allow maximum eddy-current generation, thus, maximum damping.
- The damping coefficient decreases with increasing vibration frequency. The damping reduction is about 15% from 50 Hz to 192 Hz.

The damping coefficient was also estimated numerically as a function of flux density for the experimental damper design with the 5.08 mm plate thickness, Choi and Nikolajsen (1989). This required a number of simplifying assumptions, including no reduction in damping with increasing frequency. The results are shown in Figure 10. The flux density in the airgap of the experimental damper was measured to be approximately 0.2 Tesla, thus, according to Fig. 10, the damping coefficient should be 3.7 Ns/m, which is surprisingly close to the measured values.

These results were used to estimate the size of an EC-damper capable of producing a damping coefficient of 70 kNs/m at 15,000 rpm. The best estimate gave an actuator mass of 907 kg, thus a specific damping coefficient of 77 Ns/m/kg, which is unacceptably low, and the damper had to be rejected on this basis alone.

Further background on the electrically related damper principles discussed in this paper can be found in Nikolajsen and Palazzolo (1992).

THE ER DAMPER - THE FINAL CHOICE

Detailed design calculations for the ER damper are lengthy and can be found in Ying (1993). Assuming aluminum construction and an ER fluid shear strength of 8274 Pa (1.2 psi), the specific damping is calculated to be 48.9 kN-s/m per kg (127 lb-s/in per pound) and 26.6 kN-s/m per kg (69 lb-s/in per pound) for the number 1 and 3 bearings respectively. The total damper weights calculated for each bearing are 1.45 kg (3.2 lb) and 2.63 kg (5.8 lb) respectively. Figures 11 and 12 show the damper weight and the specific damping versus ER fluid shear strength calculated with the assumption of pure Coulomb damping represented by equivalent viscous damping. They lead one to conclude that shear strength is the most important characteristic of an ER fluid for this application.

These results indicated that the ER damper is far superior to any of the Class I candidates except the traditional squeeze film damper with variable supply pressure, and the latter has questionable control capability. The main concerns about the ER damper were of a type that could only be answered by testing, such as high voltage arcing, the true nature of the damping model, and ER fluid robustness. Hence the ER damper was chosen to explore further with laboratory experiments in a rotordynamic test rig (see Vance and Ying, 1999).

CONCLUSIONS OF THIS STUDY

1. Controllable bearing dampers for aircraft engines appear to offer some advantages, although the traditional squeeze film damper without control has an intrinsic desirable characteristic that automatically reduces bearing forces at high speeds. The weight, size, and cost penalties for all of the damper design candidates considered are probably unacceptable when compared to the passive squeeze film damper as commonly found today in most aircraft engines.
2. The ER damper invented by Nikolajsen was found to meet the technical requirements of the designated aircraft engine and had already been shown to be controllable by electric voltage. The main concerns about it were high voltage arcing and ER fluid robustness. The ER damper was chosen for further study and experimental evaluation with active control.
3. Assuming the Coulomb model is correct, the most important parameter of an ER fluid for producing required levels of damping is the yield shear stress with voltage applied. But the experiments performed after this study (Vance and Ying, 1999) show that the voltage also changes the viscosity of the fluid.

ACKNOWLEDGEMENT

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Table 1: Parameters of the XTE-45 engine

Bearing Box Parameter	#1 bearing	#3 bearing
Length	12.7 cm (5")	15.24 cm (6")
Max. Diameter	33.02 cm (13")	38.1 cm (15")
Critical and max. speeds	4700 rpm, 11,500 rpm	11,000 rpm, 13,600 rpm
Operating temperature	121° C (250° F)	177° C (350° F)
Radial displacement	.0356 mm (.0014")	.0305 mm (.0012")
Desired damping	70 kN-s/m (400 lb-s/in)	70 kN-s/m (400 lb-s/in)

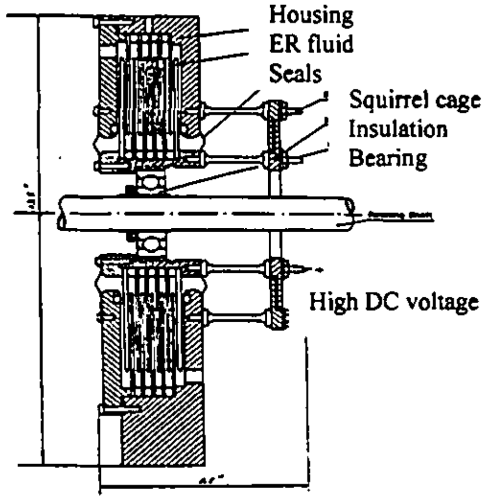

Figure 1: Cross section of the ER damper

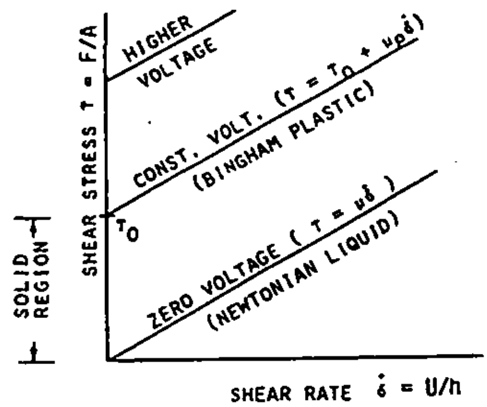

Figure 2: Electroviscous fluid behavior (from Nikolajsen and Hoque, 1990)

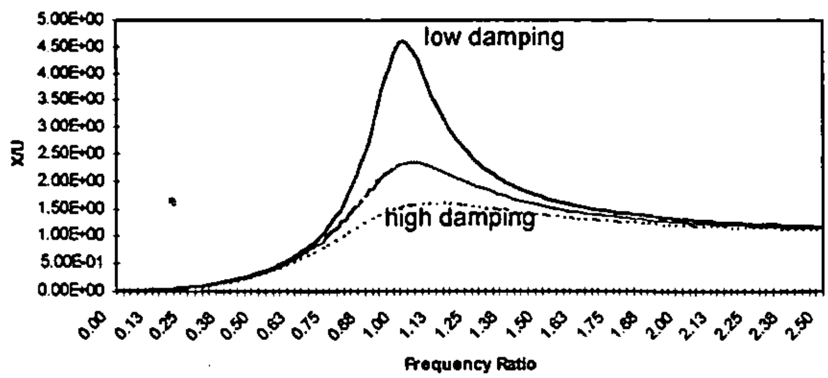

Figure 3: Effect of viscous damping on vibration amplitude

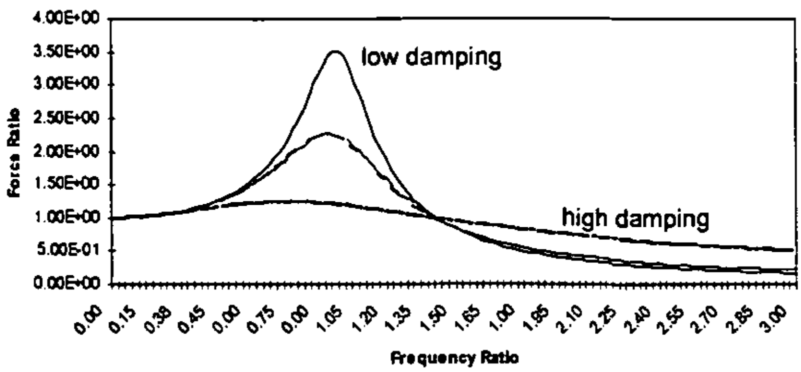

Figure 4: Dimensionless bearing force vs. frequency ratio

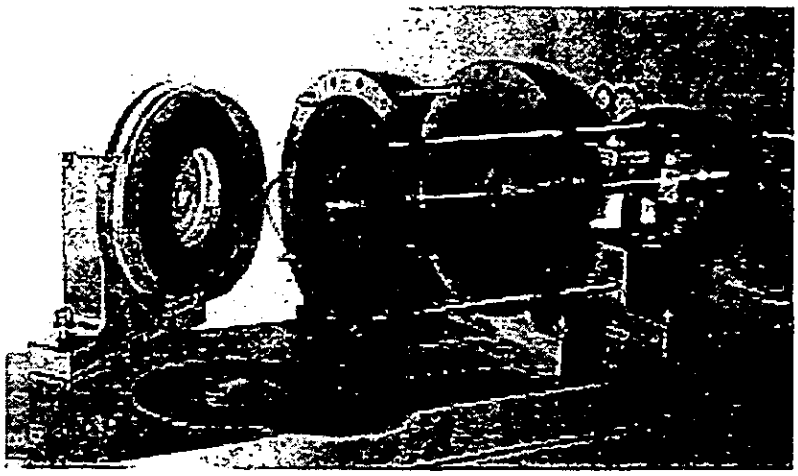

Figure 5: Coulomb friction test rig with dry friction disks

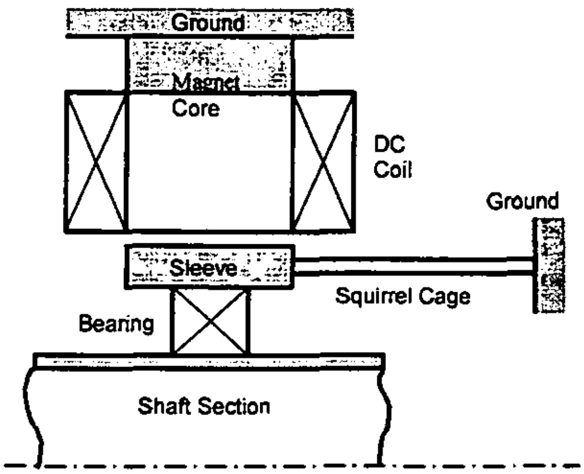

Figure 6: DC electromagnetic damper operating principle

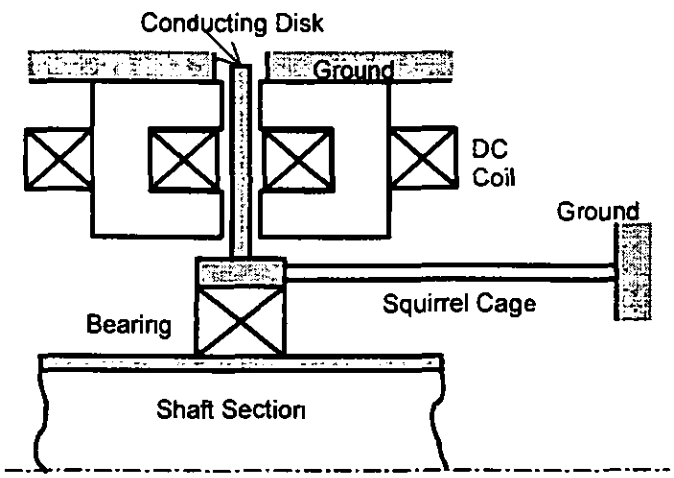

Figure 7: Eddy-current damper operating principle

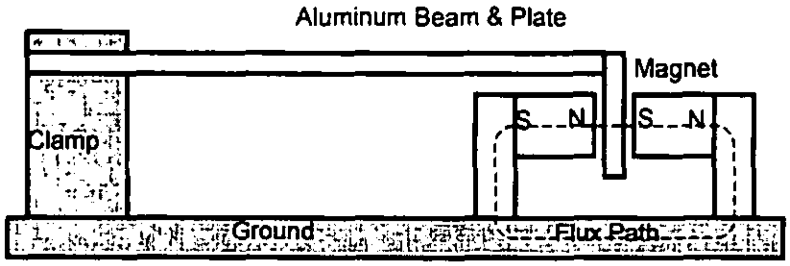

Figure 8: Eddy-current damper rig

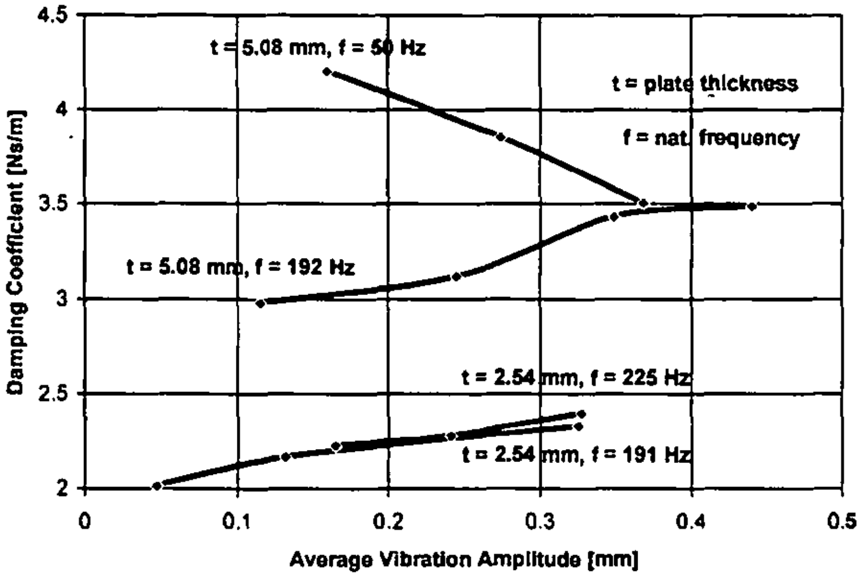

Figure 9: Measured eddy current damping

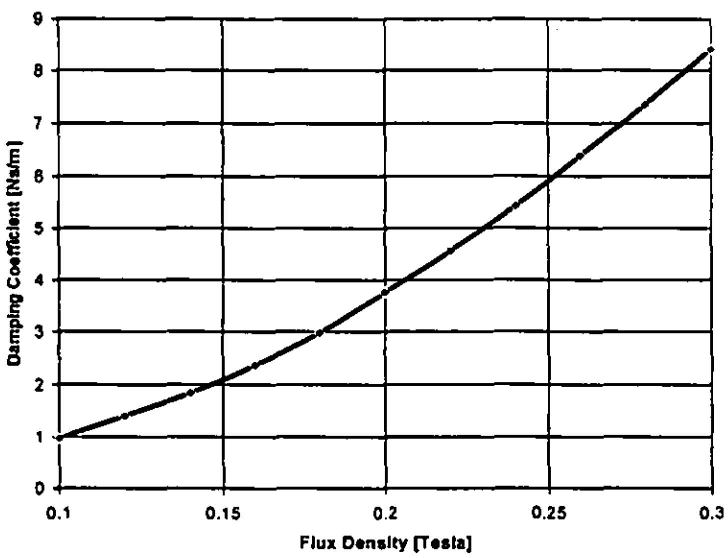

Figure 10: Predicted eddy current damping

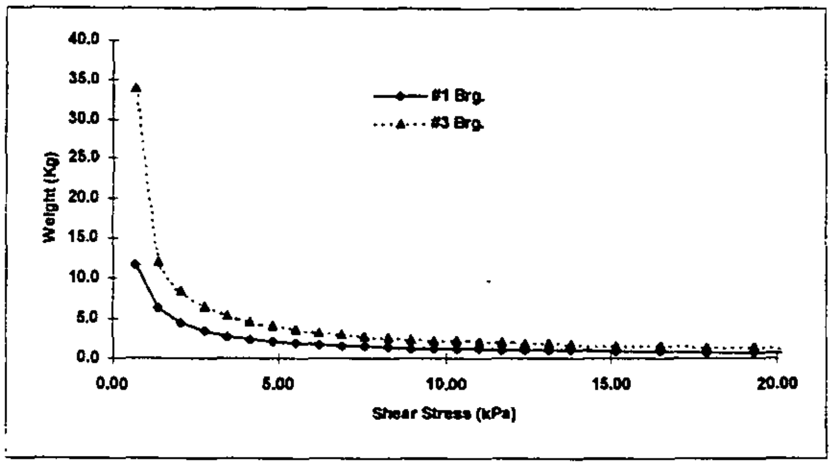

Figure 11: ER damper weight vs. yield stress of the ER fluid

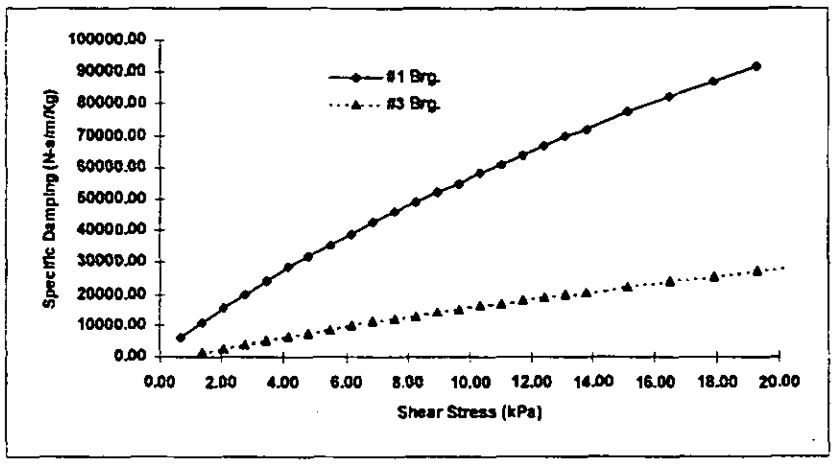

Figure 12: Specific damping of the ER damper vs. yield shear stress of the ER fluid