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MATERIAL EFFECT ON THE PERFORMANCE OF SQUEEZE FILM DAMPERS

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Abstract

This work is devoted to the fabrication and investigation of the Squeeze Film Dampers (SFDs) which are widely used in many applications. This include the fabrication of a test rig and several dampers with different sizes and two different materials which composite and non-composite. Composite dampers (Glass/epoxy), each consists of 30 layers, were fabricated by hand lay-up method. Outer and inner diameters of all the fabricated dampers were maintained as 60 and 40 mm respectively. Non-composite dampers (Steel) were fabricated and tested using turning machine. Three damper lengths of 18 mm, 30 mm and 42 mm were examined for both materials. A rotor-bearing system for the analysis has been designed and fabricated. The test rig consists of mild steel shaft of 700 mm length and 12 mm diameter. Two supports, oil pressure system and two self-alignment ball bearings were fixed on each end support. Two squeeze film dampers were used for the two support ends. Vibration amplitude has been examined for all the fabricated dampers at different shaft rotational speeds. The first resonance speed was examined for all the dampers tested. Results show that the vibration amplitude of the steel damper with L/D ratio 0.7 is less 14% than Glass/epoxy dampers with the same L/D ratio. On the other hand, saving weight of 79.6% has been achieved by using Glass/epoxy composite dampers with L/D ratio 0.7. It has been found that the performance of squeeze film damper improved with increasing length /diameter ratio (L/D) within the range tested.

INTRODUCTION

(Cooper 1963) was published first experimental demonstration of the use of a squeeze film for controlling shaft vibrations in 1963. He showed that the problem of "oil whip" can be eliminated by constraining the oil-film journal to prevent rotation. Since Cooper's investigation, the adoption of the SFD for turbo-machinery has increased, extending its application from the original field of aircraft turbine engines to

industrial. A method of linearization was noted by (Hahn 1984), who used energy concepts to obtain equivalent stiffness and damping coefficients for a squeeze-film damper. His approach involved a numerical iteration procedure to evaluate the equivalent coefficients that is no closed-form expressions were derived for the oil-film coefficients. In most aero-engine applications, the purpose of squeeze-film damper is to introduce damping as a series element between the outer race of a rolling element bearing and its rather flexible housing, so that the rotor can safely negotiate any critical speeds and operate smoothly at higher speeds. (Holmes and Dogan 1985). The vast investigations performed on SFDs have shown that damper force response is greatly influenced by conditions such as: oil feed mechanisms, end seals to restrict axial leakage, levels of inlet (supply) pressure and cavitation pressure of the lubricant, coupling of the damping device to the rotor system, and, in some circumstances, fluid inertia effects. Some experimental investigations involving SFDs with a circumferential feeding-groove have reported significant levels of dynamic pressure at the groove as well as higher damping capacity than that predicted by a classical theory. (San Andres and Vance 1987), observed dynamic pressures at a circumferential groove of magnitudes about 1/3 of those measured at the thin film lands. The damper configuration tested has a c/R ratio of 0.025 and a groove to land clearance ratio (cg/c) equal to 5. (Zeidan and Vance 1990) also measured circumferential feeding groove dynamic pressures of magnitudes as large as 60 percent of those recorded at the film lands. For this investigation the (c/R) and (cg/c) ratios were 0.0072 and 7 respectively.
The main objectives of this study are to study the effect of using Glass/epoxy composite squeeze film dampers on vibration response of a rotor-bearing system. This would include the vibration amplitude, eccentricity ratio, and effect of length/diameter (L/D) ratio, also to compare the result with the mild steel squeeze film dampers. Resonance speed at 1st mode examined also for all the tested dampers.

EXPERIMENTAL SETUP

Test rig design and fabrication

AutoCAD Mechanical software has been used to design a model for the test rig as shown in Figure 1. A schematic diagram of the designed test rig is shown in Figure 2. It consists of **General base (1),** it is a plate steel and fixed to the ground and an absorber is placed between the plate and the flour. **Base support (2),** it is divided into two parts, the first part is fixed on the general support by two bolts and the second part is fixed on the two support shafts which are mounted between them. **Support Shafts (3),** they are two shafts of steel and the use of these support shafts are to hold the bearing support and to provide a variable distance between the two bearing system. **Motor support (4),** it is assembled with self-alignment ball bearing and fixed on the general support. **Electrical motor (5),** this motor is used to provide the required rotational speed to the test shaft. **Pulley system (6),** this system is used to increase the test shaft speed to 10.000 rpm. **Flexible coupling (7),** it is used to transmit the motion to the test rig shaft. **Bearing support (8),** two bearing supports are mounted on the support shaft in both sides. The use of these supports is to support the rotor bearing system to operate properly. **Tested shaft (9),** it is a steel shaft with 12mm diameter and 760mm length. It is supported by two fabricated squeeze film bearing on both ends. **Bearing case/hosing (10),** the bearing case is used for holding the squeeze film damper. There also a thin oil film layer between the bearing case and damper. **Bolts (11)** there are four bolts in each side (in each bearing support), the use of these bolts to align the shaft. **Squeeze film damper (12),** it is the fabricated dampers with self-alignment ball bearing inside each damper. Vibration response was taken by two transducers connected to the bearing case in X and Y directions. The response is sent to the data acquisition system and be analyzed by the DASY LAB software.

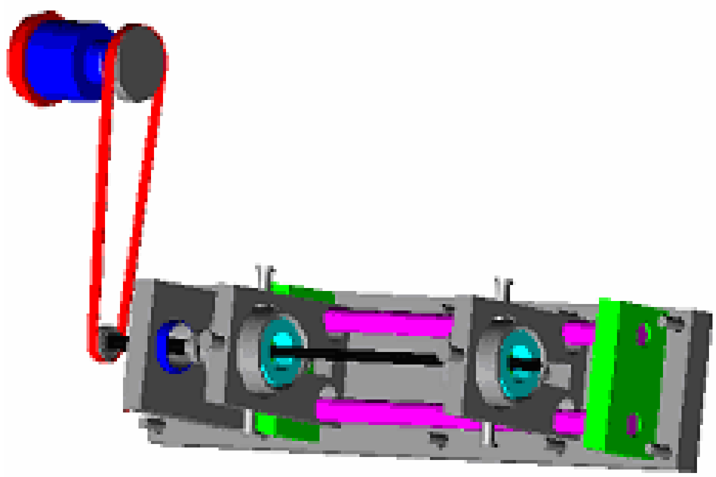

Figure 1 - The test rig

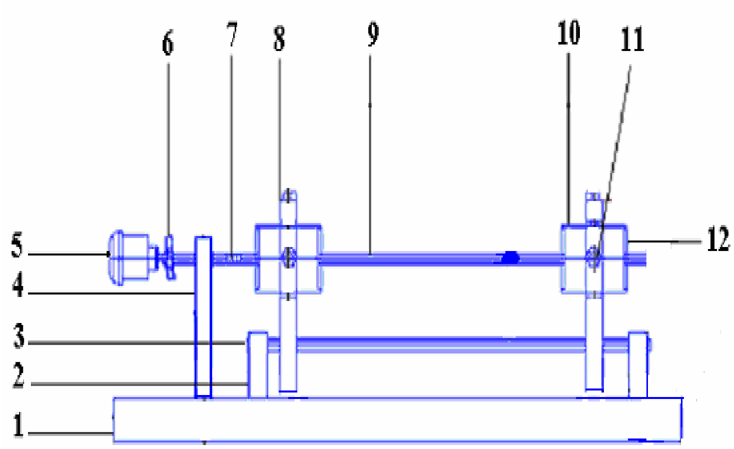

Figure 2 - Schematic diagram

Dampers Fabrication

Steel and Glass/epoxy dampers was fabricated using turning machine. Samples of the fabricated dampers are shown in Figure 4. Two "O" rings seals were used for each damper to provide the required damper length and to prevent the oil from coming out from the damper surface. The fabricated dampers specification is shown in table1.

(a) Steel dampers

(b) Glass/epoxy dampers

Figure 3 - Fabricated dampers

Table 1 fabricated dampers specification

No	Material	l/d ratio	Damper parameters		
			Length (mm)	Outer diameter (mm)	Dampers Weight (g)
1	Steel	0.3	18	60	350
		0.5	30	60	560
		0.7	42	60	760
2	Glass	0.3	18	60	67
		0.5	30	60	106
		0.7	42	60	155

EXPERIMENTAL RESULTS

Natural Frequency of the shaft

The natural frequency of the shaft has been determined for the simply supported shaft using the following Equation:

$$\omega_n = (n\pi)^2 \sqrt{\frac{EI}{\rho A_s L_s{}^4}} \qquad (1)$$

where,

$n = 1$ for the first mode

ω_n = natural frequency of the shaft (rad/s)

Then the natural frequency for the test shaft having the following specification:-

Ls = 0.700m, Ds = 0.012m, E = 207 x 10^9 N/m, ρ = 7840 kg/m^3

$A_s = \pi\,(0.012)^2/4 = 1.150\times10^{-4}\,m^2$

$I = \pi\,(0.012)^4/64 = 1.05\times10^{-9}\,m^2$

Is: -

$$\omega_n = (\pi)^2\sqrt{\frac{(207\times10^9)\times(1.05\times10^{-9})}{(7840)\times(1.15\times10^{-4})\times(0.7)^4}}$$

ω_n = 312.74 rad/sec

ω_n = 312.74 X 9.54 = 2983.5 rpm

Figure 4 shows the vibration amplitude of the fabricated simply supported shaft. The response was taken in x and y directions. As shown from this Figure, displacement in y direction is higher than that in the x direction at all the test shaft rotational speeds. Experimentally, the resonance frequency of the shaft was found to be 2800 rpm. The fabricated test rig percentage error was found to be 6.15 %.

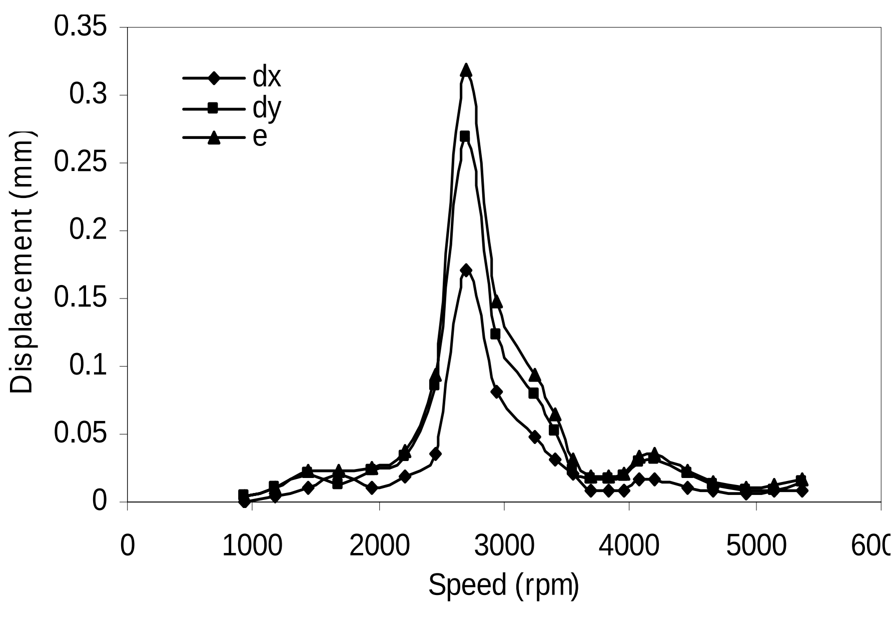

Figure 4 - Speed – displacement relation for simply supported shaft (Mild steel shaft of 0.7 m length)

As shown in Figure 5and 6, with increasing the L/D ratio from 0.3 to 0.7, the vibration amplitude decreased. The vibration amplitude at resonance decreased about 28.38% for steel damper and about 10.94% for Glass/epoxy damper, which means L/D ratio affected Steel damper than Glass/epoxy damper, on the other hand the L/d ratio introduced more weight on Steel damper than on Glass/epoxy damper. Figures 7, 8 and 9 show the effect of damper material with different (L/D) ratio on vibration amplitude. As shown in these figures the amplitude of vibration reduced by 37.73%, 33.41%, and 40.24% for Steel damper with L/D ratio 0.3, 0.5, and 0.7 respectively compared with Glass/epoxy damper with the same L/D ratio. On the other hand saving weight by 80.86%, 81.10%, and 79.61% achieved by using Glass/epoxy damper with L/D ratio 0.3, 0.5, and 0.7 respectively.

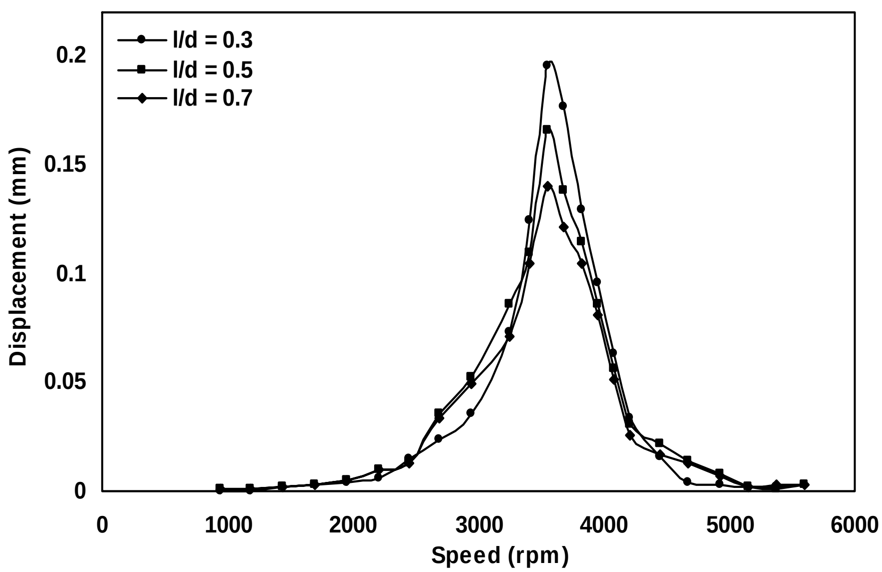

Figure 5 - L/D ratio effect on vibration amplitude (Glass/epoxy damper)

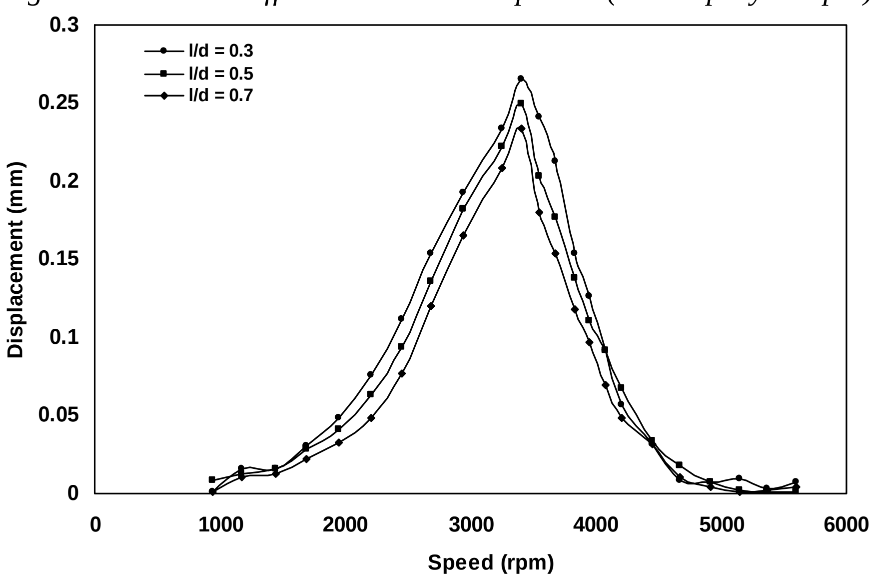

Figure 6 - L/D ratio effect on vibration amplitude (Steel damper)

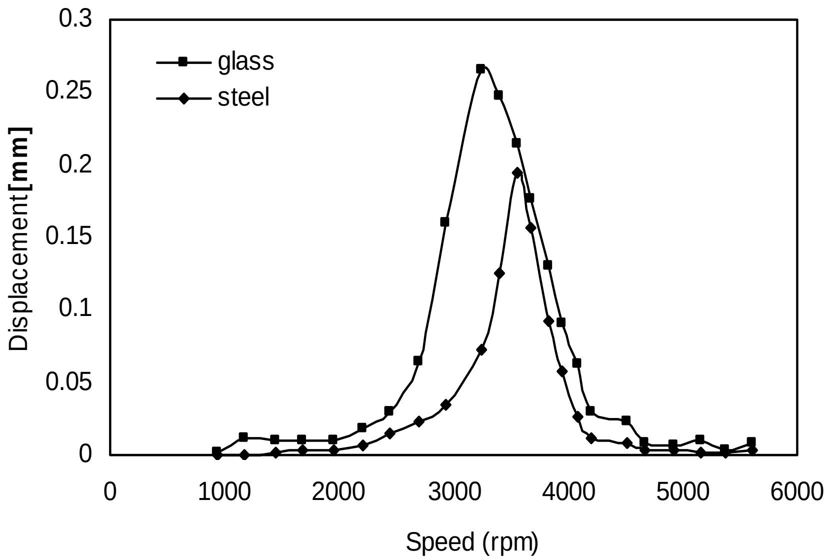

Figure 7 - Damper Material effect on vibration amplitude (l/d=0.3)

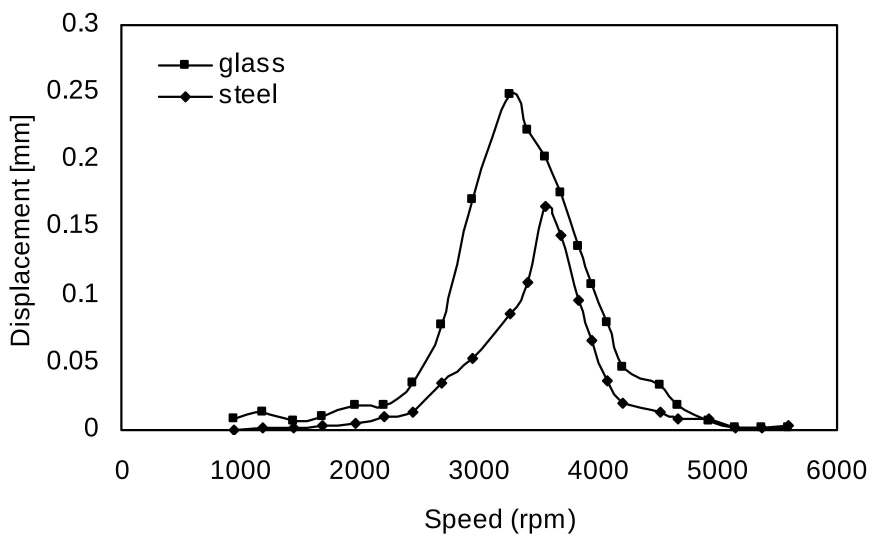

Figure 8 - Damper Material effect on vibration amplitude (l/d =0.5)

CONCLUSIONS

Dampers and The test rig have been designed and fabricated carefully. Three different length-diameter (L/D) ratio dampers for each material were tested to study the effect of (L/D) ratio on both materials. Rotational speed at resonance frequency was found to be 2703 with an acceptable error of 6.15%. Results show that the vibration response in y-direction was higher than in x direction for both steel and Glass/epoxy dampers and with increasing (L/D) ratio for the range tested, the vibration amplitude improved 28.38% for steel damper and 10.94% for Glass/epoxy damper. From results, it is clear that (L/D) ratio affected the steel dampers more than the Glass/epoxy dampers. The amplitude of vibration reduced by 37.73%, 33.41%, and 40.24% for Steel damper with L/D ratio 0.3, 0.5, and 0.7 respectively compared with Glass/epoxy damper with the same L/D ratio. On the other hand saving weight by 80.86%, 81.10%, and 79.61% achieved by using Glass/epoxy damper with L/D ratio 0.3, 0.5, and 0.7 respectively.

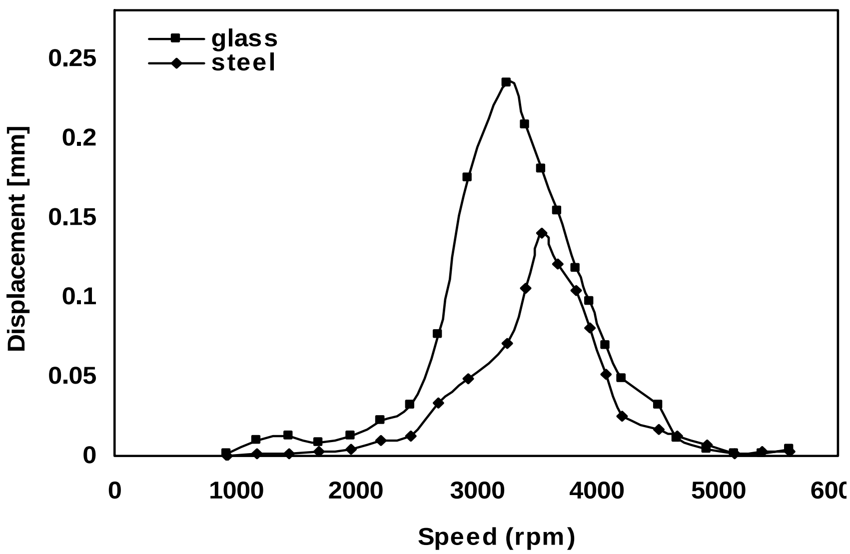

Figure 9 - Damper Material effect on vibration amplitude (l/d=0.7)

NOMENCLATURE

C = radial clearance (m)
D = bearing diameter (m)
h = film thickness (m)
P = pressure (N/m^2)
L = bearing length (m)
c/R = clearance to radius ratio
cg/c = groove to land clearance ratio
ω_n = natural frequency of the shaft (rad/s)
Ls = length of tested shaft (m)
Ds = tested shaft diameter (m)
E = (N/m)
ρ = (kg/m^3)

A_s = (m^2)
I = (m^2)

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