THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
Three Park Avenue, New York, N.Y. 10016-5990

99-GT-15

EXPERIMENTAL EVALUATION OF A METAL MESH BEARING DAMPER

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ABSTRACT

Metal mesh is a commercially available material used in many applications including seals, heat shields, filters, gaskets, aircraft engine mounts, and vibration absorbers. This material has been tested by the authors as a bearing damper in a rotordynamic test rig. The test facility was originally used to support the design of a turboprop engine, developing squirrel cages and squeeze film dampers for both the gas generator and power turbine rotors. To design the metal mesh damper, static stiffness and dynamic rap test measurements were first made on metal mesh samples in a specially designed nonrotating test fixture. These property tests were performed on samples of various densities and press fits. One sample was also tested in an Instron machine as an ancillary and redundant way to determine the stiffness. Using the stiffness test results and equations derived by a previous investigator, a spreadsheet program was written and used to size metal mesh donuts that have the radial stiffness value required to replace the squirrel cage in the power turbine. The squirrel cage and squeeze film bearing damper developed for the power turbine rotor was then replaced by a metal mesh donut sized by the computer code.

Coast down tests were conducted through the first critical speed of the power turbine. The results of the metal mesh tests are compared with those obtained from previous testing with the squeeze film damper and show that the metal mesh damper has the same damping as the squeeze film at room temperature but does not lose its damping at elevated temperatures up to 103 °C. Experiments were run under several different conditions, including balanced rotor, unbalanced rotor, heated metal mesh, and wet (with oil) metal mesh. The creep, or sag, of the metal mesh supporting the rotor weight was also measured over a period of several weeks and found to be very small. Based on these tests, metal mesh dampers appear to be a viable and attractive substitute for squeeze film dampers in gas turbine engines. The advantages shown by these tests include less variation of damping with temperature, ability to handle large rotor unbalance, and the ability (if required) to operate effectively in an oil free environment. Additional testing is required to determine the endurance properties, the effect of high impact or maneuver loads, and the ability to sustain blade loss loads (which squeeze films cannot handle).

INTRODUCTION

Squeeze-film dampers (SFD) are used extensively in almost all aircraft turbine engines designed since 1970, and have been installed in more than 300 multistage industrial compressors to raise stability thresholds. However, the SFD has some shortcomings that are difficult to overcome. If the local rotor imbalance exceeds 2.3 times the damper clearance, the SFD actually increases the response. Also, nonlinear phenomena (bistable jump up) may occur. Twenty years of applied research on SFD has failed to produce analysis tools that can accurately predict the performance of any except the simplest geometry under laboratory conditions. (Zeidan, et. al., 1996). Testing of a full-scale prototype aircraft engine rotor with squeeze film dampers in the authors' laboratory showed a large increase in rotor response when the oil temperature was raised from 27°C to 119°C.

Two researchers in China, Wang and Zhu (Wang, 1996) publicized a replacement for the SFD that did not require fluid. They described this damper as a short hollow cylinder made of woven metal material. Xin and Zi-Gen called the woven metal "metal rubber", and reported that it had good internal damping properties. They tested a metal mesh damper in a rotordynamic rig and a SFD in the same rig for comparison. The results are remarkable; they show the metal mesh damper controlling almost three times more unbalance than the SFD.

Joe Tecza (Tecza, 1997) also reports tests of a metal mesh damper in a rotordynamic rig in the fall of 1991 The project was internally supported by Tecza's company in an effort to obtain an Air Force contract. The results were encouraging but the external funds were not obtained to continue the project so it was dropped. Although previously performed research has shown great promise for the metal mesh bearing dampers, test results have not been published to show the effect of temperature or the effect of an oil environment. Prediction tools for sizing a metal mesh damper are yet to be developed and made available to machine designers.

TEST APPARATUS

A nonrotating test apparatus was used to determine the static stiffness and damping of two densities of metal mesh. The test apparatus consisted of two concentric cylinders that were used to constrain the inner and outer boundary of the metal mesh. A steel ring was constructed to fix the outer dimension of the metal mesh contained between it and the inner cylinder. The diameters of these inner cylinders varied in size to put different press fits on the metal mesh. The inner cylinder was fixed to the top of a rigid concrete and metal slab. To fix the inner cylinder, an aluminum cap was placed on top of the inner cylinder and a bolt was placed through the center of the top into the table. The outer steel ring of

Presented at the International Gas Turbine & Aeroengine Congress & Exhibition
Indianapolis, Indiana — June 7–June 10, 1999
This paper has been accepted for publication in the Transactions of the ASME
Discussion of it will be accepted at ASME Headquarters until September 30, 1999

the assembled rig did not rest on the table, therefore; it was free to move in the horizontal plane, parallel to the tabletop. This test setup is shown in Figure 1.

A power turbine test rig was used in the rotordynamic testing of the metal-mesh damper. The power turbine rotor assembly is made up of two turbine wheels, an Inconel shaft, and a steel spline. The assembly is press fitted together and fastened with two tie-bolts. A cross section of the assembled power turbine rotor is shown in Figure 2.

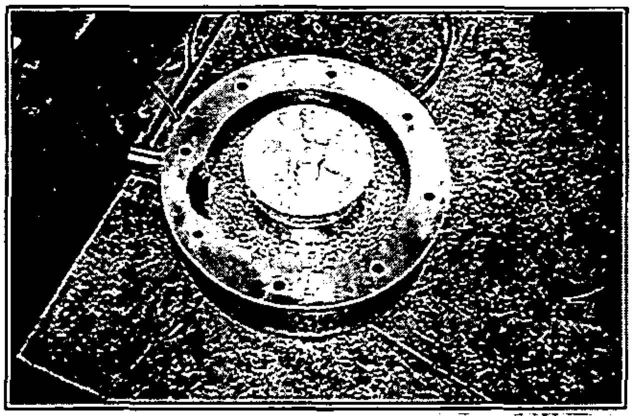

Figure 1, nonrotating test apparatus

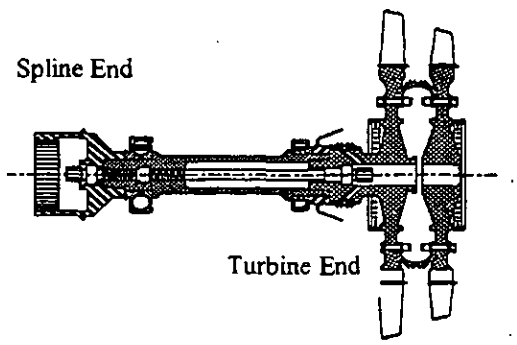

Figure 2, Power Turbine Rotor

The weight of the fully assembled rotor is 147.6 N. It is driven by an air turbine in the test cell. The power turbine test rotor has a ball bearing at the spline end and a roller bearing at the turbine end. The original test rotor used two squeeze film dampers to provide damping. A squirrel cage, which provided a specific stiffness, accompanied each squeeze film damper. The squirrel cage located at the spline end had a stiffness of approximately 5,253 kN/m. The squirrel cage at the turbine end had a stiffness of approximately 4,990 kN/m. For the metal-mesh testing, the squirrel cage at the spline end of the roller-bearing system was retained but the land of the squeeze film damper was removed to insure that this bearing had no damping. Thus, there was no damping in the system except that provided by the metal-mesh donut at the turbine end. The assembled power turbine test rotor can be seen in figure 3.

PRELIMINARY ANALYSIS AND RESULTS

To try and match the stiffness and damping of the squeeze-film dampers previously installed, stiffness testing, damping tests, and computer analysis were performed on two densities of metal mesh samples. Samples of 29% density were tested for stiffness using the nonrotating test apparatus in Figure 1, with a hand held force gauge, and a dial indicator. The 29% density sample had dimensions 57.15 mm ID, 99.57 mm OD, and 12.7 mm thickness. The test was performed with press fits ranging from (1.27 mm) to (3.048 mm). These tests indicate that the stiffness of this material increases as the press fit is increased. The stiffness values for the two extremes of fit were not close to our target stiffness of 4,990 kN/m, (for rotordynamic testing to be done later).

To obtain a stiffness value closer to the target value, a sample of 57% density was tested. The 57% density sample had dimensions 58.42 mm ID, 95.76 mm OD, and 11.176 mm thickness. This sample was tested in an Instron machine where both loading and unloading could be graphed. To perform this test, the inner and outer surface of the mesh was fixed by cylinders. A solid rod was placed through the center of the inner cylinder (see Figure 4). This test yielded a stiffness value close to 3,502 kN/m, which was input into a spreadsheet program to obtain an equivalent modulus of elasticity. The metal-mesh spreadsheet program had been previously developed with support from a research consortium (see the Acknowledgements). It takes the dimensions and equivalent modulus of elasticity of the metal-mesh as input, and outputs the stiffness by curve-fitting a number of points. The equations used in these computer programs may be the subject of a future paper after refinement by further testing.

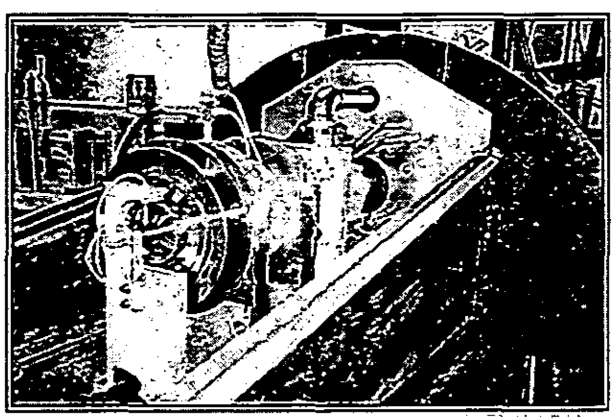

Figure 3, Assembled power turbine test rig

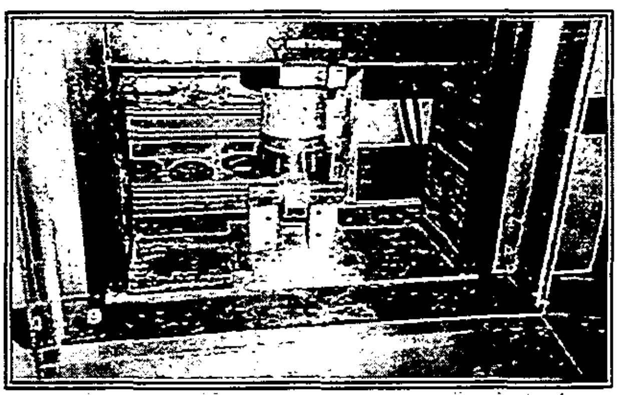

Figure 4, Instron machine and test setup for stiffness testing

Since the stiffness was measured, the modulus of elasticity for the 57% density sample could be backed out of the program as approximately 1,628 kN/m. This value along with the desired inner and outer diameter were input back into the equations leaving the stiffness as a function of width only. The width was then altered to obtain an acceptable value for stiffness. Using the dimensions for inner and outer diameter that are required for the power turbine hardware, a stiffness of 5,734 kN/m was obtained from the metal mesh program. The dimensions for the final metal-mesh damper were then 76.2mm ID, 108.6mm OD, and 12.7mm thickness.

In order to measure the damping of the metal mesh, the outer steel ring of the nonrotating test rig was rapped with a plastic hammer to obtain a time response and frequency spectrum. The time response was used to obtain the log decrement and the frequency spectrum identified the natural frequency of the system. The log decrement and damping ratio were obtained for several different press fits. The results show that the damping decreases with increasing press fit. Table 1 shows the results of several different tests with different press fits. Experiments are currently underway to determine how the damping varies with frequency and to determine what mechanism is responsible for the damping. It is not currently know whether or not the damping and stiffness parameters would by effected by a static eccentricity due to an offset load.

Table 1, Effects of press fit on damping

Configuration	Press fit	Log Dec	Damping Ratio
Press fit #1	1.27 mm	0.988	0.155
Press fit #2	1.905 mm	0.815	0.129
Press fit #3	2.54 mm	0.534	0.085
Press fit #4	3.048 mm	0.451	0.0702

The free vibration data in Table I was collected using an accelerometer attached to the outer ring. An average of ten raps were taken using a Hewlett Packard FFT analyzer.

Computer analysis of the power turbine mode shapes was performed to determine if the metal mesh would be needed at both bearings. The first critical speed of the rotor bearing system with the original squirrel cage configuration was at 3200 RPM. The mode shape associated with it shows large amplitude of vibration at the turbine end. It became apparent that metal-mesh would only need to be installed at the turbine end for a complete rotordynamic evaluation at the first critical speed. With high levels of vibration at the turbine end and no other form of external damping in the system, the metal-mesh donut is responsible for providing the damping necessary to traverse the first critical speed.

ROTORDYNAMIC TESTING AND RESULTS

The power turbine test rig was run up to a speed of 7000 RPM, since it was only necessary to coast down through the first critical speed. The second mode shape has the largest amplitude at the spline end where there is no damping so the second critical speed was to be avoided. All data presented is from the horizontal and vertical proximity probes at the turbine end of the rotor (X, Y).

The metal-mesh was tested under several conditions. including hot, cold, balanced rotor, and unbalanced rotor. For tests requiring high temperatures the metal-mesh was heated with oil that flowed over it. The mesh was first tested dry and then soaked with room temperature oil to determine whether oil would affect the damping. It did not. A heating strip was then used to heat the oil that flowed from the pump to the test rig and through the metal mesh. A valve could be closed for tests not requiring high temperature oil. To monitor the temperature of the oil entering, a thermocouple was placed at the inlet of the bearing pedestal containing the metal-mesh donut.

A set of tests was performed with the rotor balanced. The rotor was balanced in one plane to achieve an acceptable level of vibration. The amplitude of vibration was reduced to 30% of its original unbalanced state. The test rig was run with the metal-mesh dry and with oil at 54°C, 71°C, and 82°C. The results of the balanced tests are shown in Figures 5 and 6. They show the response at the turbine end where the highest vibration amplitudes exist. The critical speed peaks in Figures 5 are slightly shifted to the left as the temperature increases. The RY amplitude increased approximately 20% at temperatures of 71°C and 82°C. Both Figures 5 and 6 (RX, RY) show split critical speeds. This may be due to stiffness asymmetry in the metal-mesh.

The second and most important test condition was the unbalanced condition. This condition is important since the squeeze film damper had much larger levels of vibration with an unbalanced rotor at high temperatures. These tests were run with oil at 54°C, 71°C, 82°C, and 99°C. Results of these tests can be compared with those obtained with the squeeze film damper previously installed. Figures 7 and 8 show the measured response with the squeeze film dampers at an oil supply pressure of 1.034 bar. They show the response at the X and Y probes (turbine end). The amplitude of vibration with the squeeze film dampers increases dramatically at elevated temperatures. For comparison, tests were run at similar temperatures with the metal-mesh installed. Figures 9 and 10 show the results. Although the rotor unbalance is probably not exactly the same as it was with the squeeze-film dampers installed, the results can be compared as a percent increase in the level of vibration with increase in temperature. Observation of the response bandwidths suggests that the metal-mesh damping at all temperatures is about the same as the squeeze-film damper at room temperature. The results with metal-mesh show that there is no correlation between temperature and amplitude of vibration.

CONCLUSION

The following conclusions apply only to the parameters tested. However, it should be remarked that they are full scale parameters of a power turbine test rig used to develop a turboprop aircraft engine.

a.) Metal-mesh has useful damping properties that can be used to reduce rotordynamic amplitudes of vibration.
b.) The damping provided by the metal-mesh is not significantly temperature dependent over the range of 54-99°C.
c.) The damping provided by the metal-mesh is not affected by the presence of turbine oil.
d.) Higher temperatures appear to have a de-stiffening effect on the metal-mesh bearing damper under balanced conditions
e.) Increasing the radial press-fit interference of the metal-mesh damper ring has a stiffening effect.
f.) Increasing the radial press-fit interference of the metal-mesh damper ring decreases the free vibration damping coefficient.
g.) A spread sheet program is being developed that shows promise as a design tool for metal-mesh bearing dampers.

ACKNOWLEDGEMENTS

The help of Metex Corporation in providing damper rings of several different densities is gratefully acknowledged. The help of Air Tractor Inc. in providing the test rotor and associated hardware is gratefully appreciated. This research was supported by

the Turbomachinery Research Consortium at Texas A&M University.

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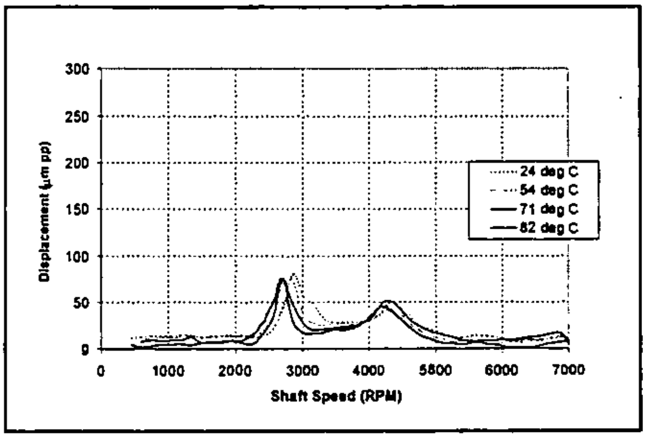

Figure 5. Balanced Condition @ Rx-probe With Metal-Mesh Installed, Multiple Temperatures

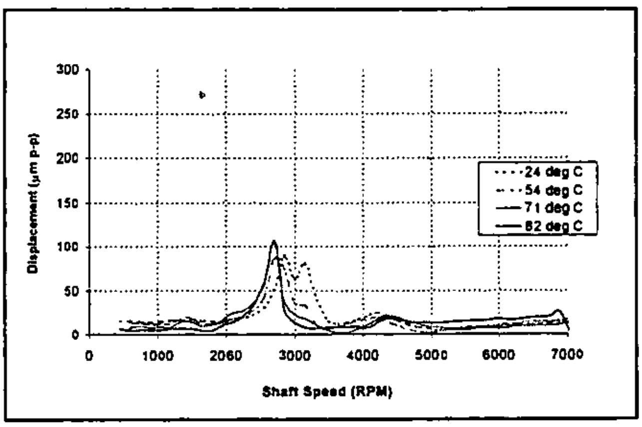

Figure 6. Balanced Condition @ Ry-probe With Metal-Mesh Installed, Multiple Temperatures

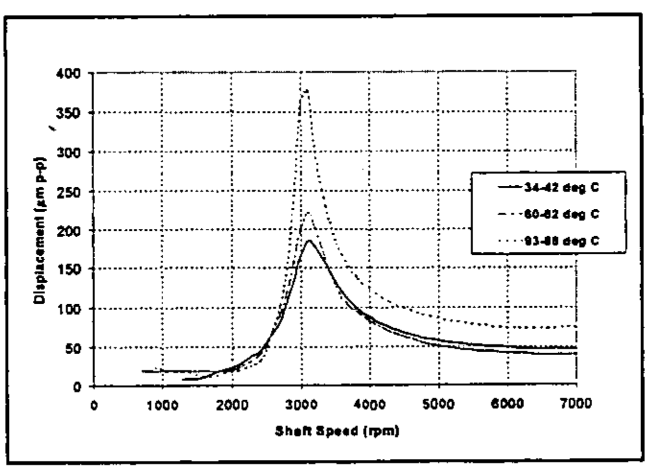

Figure 7. Unbalanced Condition @ Rx-probe With Squeeze-Film Damper Installed, Multiple Temperatures

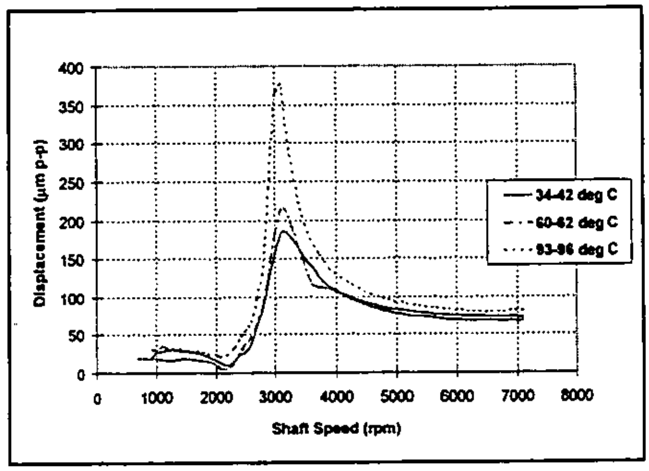

Figure 8. Unbalanced Condition @ Ry-probe With Squeeze-Film Damper Installed, Multiple Temperatures

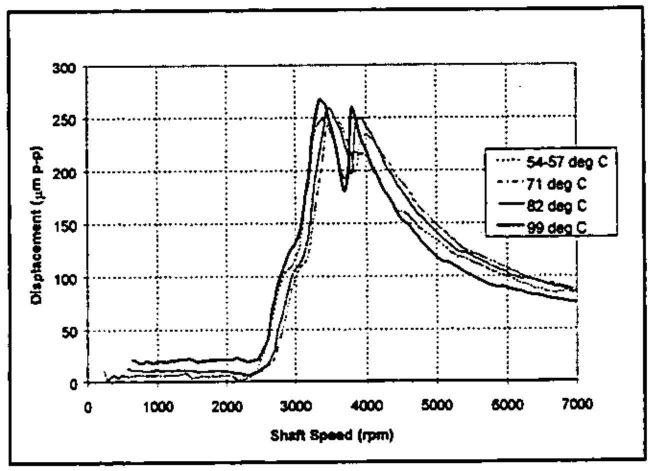

Figure 9. Unbalanced Condition @ Rx-probe With Metal-Mesh Installed, Multiple Temperatures

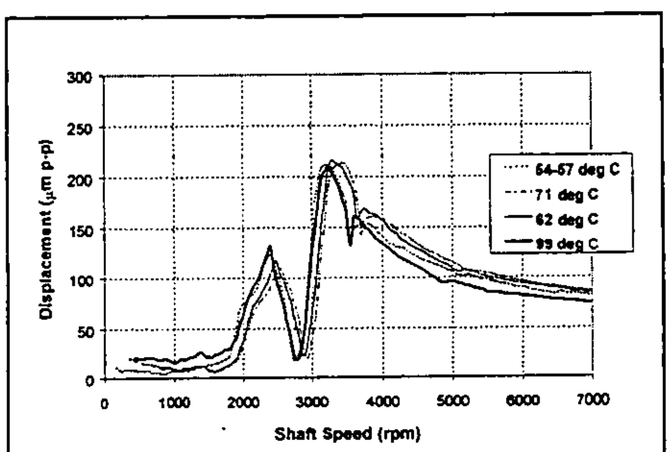

Figure 10. Unbalanced Condition @ Ry-probe With Metal-Mesh Installed, Multiple Temperatures