EVALUATION OF SHEAR MOUNTED ELASTOMERIC DAMPER

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SUMMARY

The safe and reliable operation of high speed rotating machinery often requires the use of devices that dissipate undesirable rotor vibrations. As an alternative to the more conventional squeeze film bearing damper designs, a Viton-70 elastomeric shear mounted damper was built and tested on a T-55 power turbine spool in the rotor's high speed balancing rig. This application of a shear mounted elastomeric damper demonstrated for the first time, the feasibility of using elastomers as the primary rotor damping source in production turbine engine hardware. The shear damper design was selected because it was compatible with actual gas turbine engine radial space constraints, could accommodate both the radial and axial thrust loads present in gas turbine engines, and was capable of controlled axial preload. The shear damper was interchangeable with the production T-55 power turbine roller bearing support so that a direct comparison between the shear damper and the production support structure could be made. Test results show that the Viton-70 elastomer damper operated successfully and provided excellent control of both synchronous and non-synchronous vibrations through all phases of testing up to the maximum rotor speed of 16,000 rpm. Excellent correlation between the predicted and experienced critical speeds, mode shapes and log decrements for the power turbine rotor and elastomer damper assembly was also achieved.

INTRODUCTION

With the advent of higher power density turbomachinery and the use of long unsupported shaft segments, rotor systems are operating near or above one or more critical speeds and are therefore susceptible to any number of destabilizing mechanisms. To insure the safe and reliable operation of this machinery, considerable effort is often expended in an attempt to control both synchronous and non-synchronous vibrations. Since balancing technology has matured to the point that the control of synchronous vibrations is not a limiting factor, emphasis has been directed at methods of dissipating undesirable vibrations and stabilizing rotor bearing systems. Squeeze film dampers provide one method of control but, close tolerance machining, oil supply requirements, and the associated hydraulic plumbing make these systems expensive and vulnerable. As an alternative to the squeeze film bearing damper a convenient, compact, self sufficient and inexpensive dry damper with a wide range of stiffness and damping characteristics that will provide comparable vibration control is desired. Elastomer dampers have many of the desirable features but, their use has been limited due to the lack of adequate design information and demonstrations of capability. With the research reported on in References 1 thru 5, considerable progress has been made in establishing the design guidelines for elastomer dampers. As a demonstration of elastomer damper capability in the control of rotor vibration an experimental investigation was performed at MTI. In this test program a Viton-70 shear mounted elastomeric damper was designed specifically to control the vibrations of a T-55 power turbine rotor mounted in a high-speed balancing rig. The high-speed balancing rig used as the test facility in this program was constructed by MTI under a NASA contract (NAS 3-20609) managed by Robert Cunningham for the U.S. Army's Corpus Christi Army Depot (CCAD) and was used with the Army's permission. The CCAD high speed balancing facility incorporates production engine bearings, support structure and is capable of achieving operational speeds. It therefore provided an excellent opportunity to test a production gas turbine engine hardware compatible, elastomer damper design for the first time.

Since many gas turbines are multi-shafted and radial design flexibility is inhibited, the damper was designed to be active in shear. This design configuration (fig. 1) minimizes the required radial envelope and accommodates axial thrust loading which may be required in typical gas turbine applications. Once an analytical model of the production CCAD balancing rig was established and calibrated against typical response characteristics for the rig, damper optimization studies were performed. The results of the optimization studies provided the required ranges of damper stiffness and loss coefficients for this application.

As documented, the shear elastomer damper operated successfully to the maximum operating speed of 16,000 RPM. Orbit control was reliable with no severe or large nonsynchronous components of vibration observed. Thermocouple monitoring showed that the temperature of the elastomer never exceeded 65°C during testing. Further, there was excellent agreement between the analytical predictions and measured data for critical speeds, mode shapes and logarithmic decrement which clearly demonstrated the ability of the T-55 power turbine elastomer damper to predictably control rotor vibration for safe and reliable operation.

ROTOR OPTIMIZATION STUDIES

A typical response profile of the CCAD rig with production hardware and a T-55 power turbine installed is shown in figure 2. This figure is a trace of the synchronous response of a slow acceleration pattern for a 90° pair of probes (vertical and horizontal) at approximately midshaft location. The two dominant features are the peaks at approximately 4,000 and 6,000 RPM. This highly elliptical horizontal orbit occurring at 4,000 RPM, rapidly changing to a dominant vertical ellipse at 6,000 RPM, is typical of characteristics of a retrograde and forward precession encouraged by asymmetric support characteristics. The response of the turbine end at these two speeds is dominant whereas the roller bearing end (cold end) showed little activity, indicating a strong precession of the turbine such as produced by a rigid body conical mode shape. A subsequent set of peaks occurred in the 8,000 to 9,000 RPM speed range, with all probe sets along the shaft indicating motion. The activity at this speed was not predicted analytically, but was generally subordinate to the two main peaks at 4,000 and 6,000 RPM and was most likely due to the structure.

To further assess possible rotor modes and determine the acceptability of the proposed rotordynamic model, a small exciter was attached to the CCAD rig to excite the nonrotating T-55 power turbine. Swept sine-wave excitation indicated three possible structural modes:

- 115 Hz bounce mode of turbine

- 210 Hz bending mode of shaft

- 600 Hz bending mode of shaft

Figure 3 indicates the location of these structural frequencies when compared to a tuned analytical model of the T-55 (whirl speed marked by "X" at zero spin speed axis). This analytic model also predicted a first retrograde (backward) precession turbine conical mode at 4,500 RPM, a first forward mode at 6,000 RPM and a first bending (2nd critical) at 21,000 RPM. The production engine support rigidity needed to calibrate this model was reasonable, with a roller bearing stiffness of 8.75×10^7 N/m (500,000 lb/in.) and a ball bearing (turbine) pedestal stiffness of 1.40×10^7 N/m (80,000 lb/in.).

Further evidence of the models validity was obtained by observing the super-synchronous excitation of the small 2/rev component of motion throughout a normal acceleration pattern to maximum speed. The 2/rev peaked while operating at 7,100 RPM indicating the presence of a mode as shown in figure 3 at the intersection of the 2/rev excitation line and the second critical speed line.

Therefore, from observation of rotor synchronous response, static shaker excitation data and observation of the 2/rev vibration, the rotor-dynamics model was considered acceptable for analytic examination of possible elastomeric-damper designs.

Previous elastomer material testing of shear specimens (ref. 1) was reviewed for candidate elastomer material selection. Viton-70 at 32°C showed the largest measured value of loss coefficient for the entire range of frequencies expected. As the loss coefficient is a measurement of the ratio of material damping to stiffness, a higher value of loss coefficient reflects an increased capacity to dissipate energy. Temperature increases make the selection of material for damping somewhat arbitrary as the loss coefficient for all materials, Buna-N, EPDM, Neoprene and Viton ranges between 0.1 and 0.2 for the entire frequency range at a temperature of 88°C. Therefore the decision to use Viton as the damping material was based upon the following:

- Viton-70 shows superior damping properties compared to the other materials tested for 32°C or 66°C operation (figs. 4 and 5).

- Viton-70's dissipation characteristics are as good as any of the other materials tested for operation at 88°C.

- Viton-70 was previously used as a damper in successful testing of high-speed rotor dampers (refs. 1 and 5).

- Unused Viton material was available from MTI elastomer test activity under NASA Programs NAS 3-18546.

Based on data obtained from shear specimen shaker testing of Viton-70 (ref. 5) a range of loss coefficients from 0.15 to 0.75 could conceivably be expected for the 0-16,000 RPM operation of the CCAD balancing rig (fig. 6). Accordingly, the T-55 power-turbine rotor-dynamics model was modified to determine the optimal damper design (figs. 7 and 8). All system damping was assumed to be due to the elastomer damper, consequently no structural damping was modeled for the turbine or gearbox bearing. However, the two disk pack couplings were included as elastic elements which offered a minimum of bending rigidity but offered full shear restraint.

As a result of the rotor damper optimization studies (figs. 7 and 8) a number of facts surfaced regarding the design of the elastomeric damper:

- The first critical speed is a conical precession of the turbine with its nodal point extremely close to the elastomer damper test bearing location. Accordingly, the first critical speed offers little damping and is insensitive to the elastomer damper stiffness or damping characteristics.

- The second and third mode shapes offer significant activity of the elastomeric damper. Consequently, the critical speed location and log decrement of those critical speeds are sensitive to damper selection.

- A best, or optimal, tradeoff between the second and third mode is obtained by a Viton-70 elastomeric damper of between 5.25×10^6 - 7.0×10^6 N/m (30,000-40,000 lb/in.).

- It is expected with this value of stiffness, the T-55 rotor mounted on the elastomeric damper would traverse the second critical, which is not within the operational range of the roller-bearing mounted production T-55 configuration (fig. 3).

- The range of log decrement expected for the second critical speed is between 0.2 to 1.1 dependent upon the loss coefficient (temperature of operation) for the elastomer damper.

DAMPER DESIGN AND FABRICATION

Past elastomer designs at MTI consisted of button configurations because of the need to have either interchangeability for comparisons of different elastomers or for direct comparisons with squeeze film bearing dampers. T-55 damper design however, offered uniquely challenging problems since this configuration had to be compatible with actual gas turbine design technology and constraints. Most gas turbines do not have the large amounts of radial envelope required by button designs, particularly if they are multishaft turbofan or turboshaft engines such as the T-55. On the other hand, design maneuverability and flexibility are often generally less restricted in the axial direction. Additionally, the design must accommodate possible thrust loading (although the T-55 roller bearing mount does not take thrust) in combination with radial loads and be capable of controlled preload. These requirements were satisfied by the shear damper configuration.

Figure 1 illustrates the concept used for the elastomer damper mount on the T-55 roller-bearing support. Two Viton-70 shear rings mounted along the entire circumference satisfied all requirements. Axial preload was accommodated by controlled machining of the outer flanges which attached to the housing by twelve equally spaced socket head cap screws. An overload protector was installed by using an O-ring with a prescribed clearance of 0.254 mm (0.010 in.) to avoid interference with normal operation of the damper while still providing backup protection should the shear damper or bond fail.

To obtain the required stiffness for the Viton-70 damper previous shear specimen data was used (ref. 5). The resulting design, 6.98 cm (2.75 in.) I.D. and 9.52 cm (3.75 in.) O.D. for a 3.175 mm (0.125 in.) thick Viton-70 specimen, produced a stiffness range in shear of approximately 6.12×10^6 - 7.0×10^6 N/m (35,000 to 40,000 lb/in.) depending upon strain amplitude experienced. These analytically predicted values of stiffness were validated prior to test through static load testing. The static values determined were between 6.12×10^6 and 6.47×10^6 N/m (35,000 to 37,000 lb/in.). Axial compression stiffness was also determined and was found to be approximately 1.26×10^7 N/m (72,000 lb/in.). In all cases the static loads were applied and held constant for up to 5 minutes to determine if any load relaxation would occur. No measurable load relaxation was observed.

TEST RESULTS

The baseline run of a balanced T-55 power turbine supported on the production roller bearing support housing showed the characteristics previously noted in fig. 2 and reconfirmed the existence of the turbine horizontal and vertical rigid body precessional modes at 4,000 and 6,000 RPM respectively. Additionally, the intermediate peaks in the 8,000-9,000 RPM range were again observed though not confirmed analytically (although suspected to be induced by the rigs' structure). The data collected during the initial elastomer damper tests showed that a substantial reduction in synchronous response occurred for both the 6,000 RPM and 8,000 RPM peaks but that the response near 14,000 RPM increased as can be seen in fig. 9. Through an unbalance sensitivity study this new peak was determined to be the second critical speed as was predicted analytically.

A comparison of the analysis performed for the elastomer-damper supported system with the measured data provides good correlation for mode shapes (fig. 10). Table I provides a comparison of critical speeds and logarithmic decrements for the two criticals observed.

There is excellent correlation in whirl speed, but the first critical's log decrement is significantly different in value between test and analysis. This is due to the fact that analysis has indicated a node at the damper for the first critical. Slight changes in rigidity of the shaft or location of roller-bearing support would alter the analytically predicted value of log decrement for the first critical appreciably. Further, consideration of structural damping which occurs at the turbine end bearing (which is heavily participating in the first critical, fig. 10) was not included in the analysis. Therefore, the analysis does not reflect this form of structural damping which can greatly effect the measured log decrement of the first critical speed.

Typical measured response indicated dominant 1/rev excitation of the rotor through the testing with the elastomer damper active. Figure 11 presents a typical frequency spectrum at 13,000 RPM. As shown, relatively minor subsynchronous and supersynchronous excitations were experienced and the operation of the T-55 power turbine was as predicted throughout the entire test phase.

CONCLUSION

This successful application of elastomer damper technology presents an opportunity to reflect upon the many uses for elastomers as alternatives to the conventional squeeze film dampers. While this work does establish an initial benchmark for elastomers as viable alternatives for damping of high speed rotating machinery, further efforts are required to firmly establish this technology. In particular, these efforts need to address the testing of elastomers in the rather hostile but realistic gas turbine environments. The authors are cognizant of the temperature limitations of elastomers however, newer and greater varieties are being introduced each year that do offer a wide variety of application opportunities particularly in the colder (compressor) section of gas turbines. Therefore, this contribution is offered in the hope that further exploration of "dry damper" technology be encouraged.

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TABLE I.

	Analysis		Measured	
	Speed (RPM)	Log Decrement	Speed (RPM)	Log Decrement
First Critical	6,022	0.001	6,007	0.090*
Second Critical	13,535	0.201	13,666	0.206*

* Average value for all data obtained.

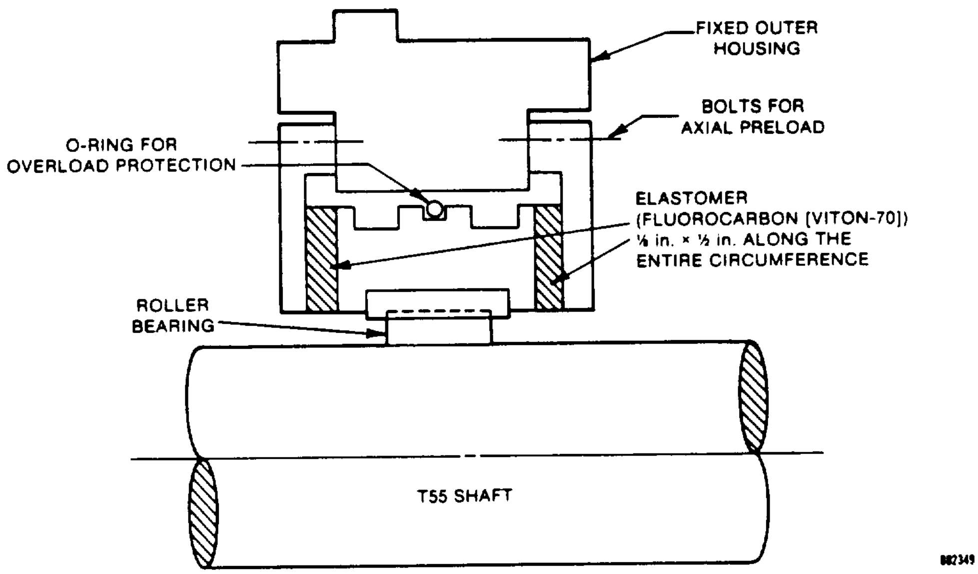

Figure 1. - T-55 Elastomer supported roller bearing cartridge.

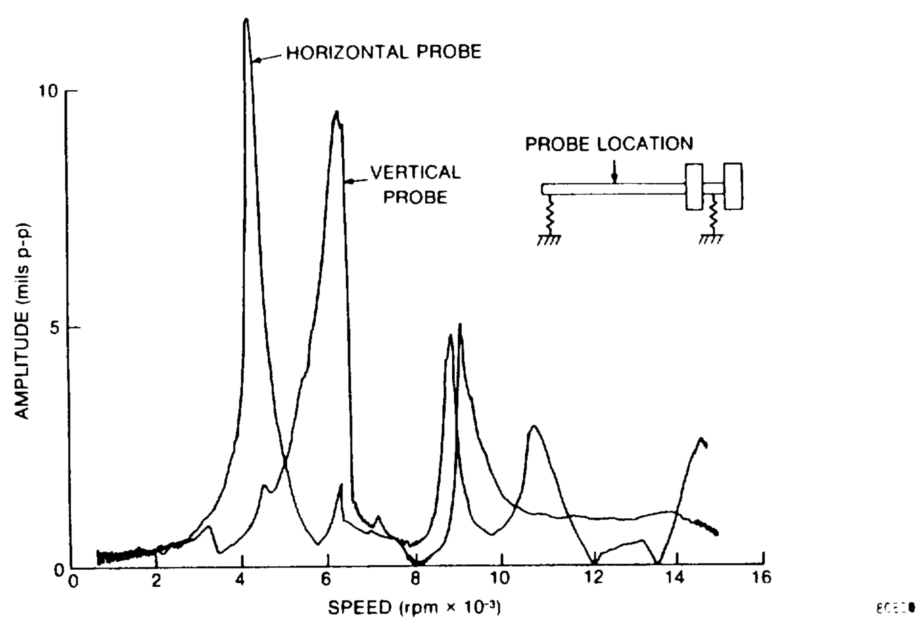

Figure 2. - Synchronous response of T-55 on CCAD rig.

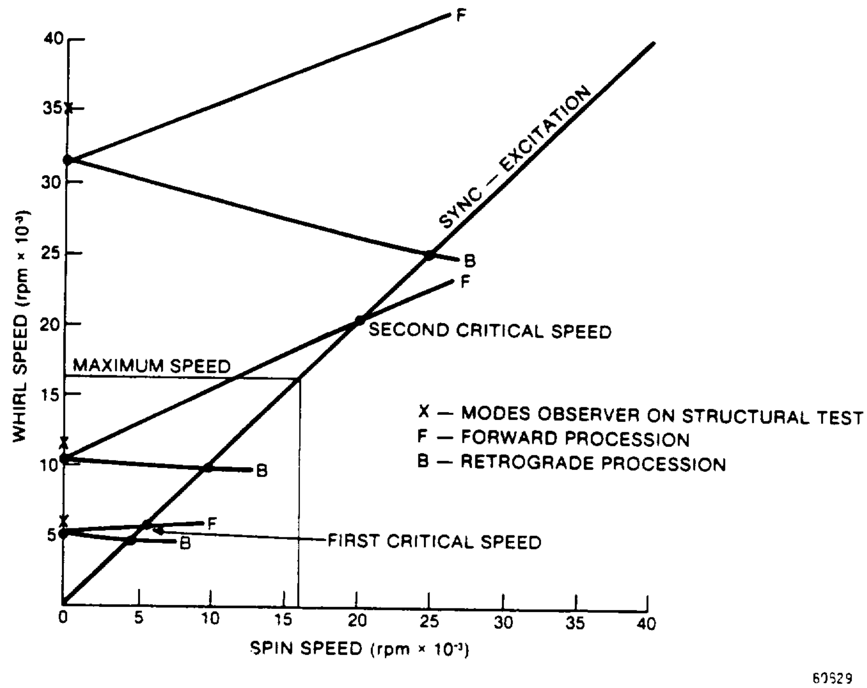

Figure 3. - Whirl speed versus spin speed.

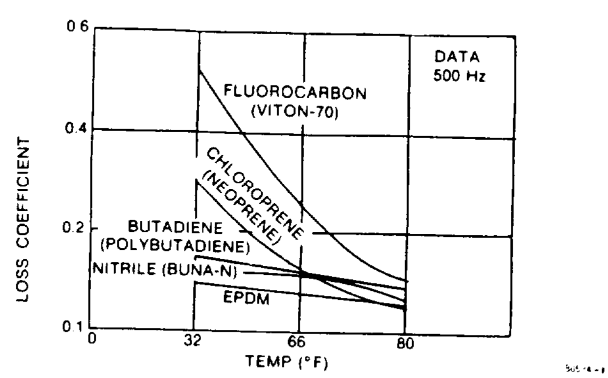

Figure 4. - Temperature variation of elastomer loss coefficient (at 500 Hz) for shear tests.

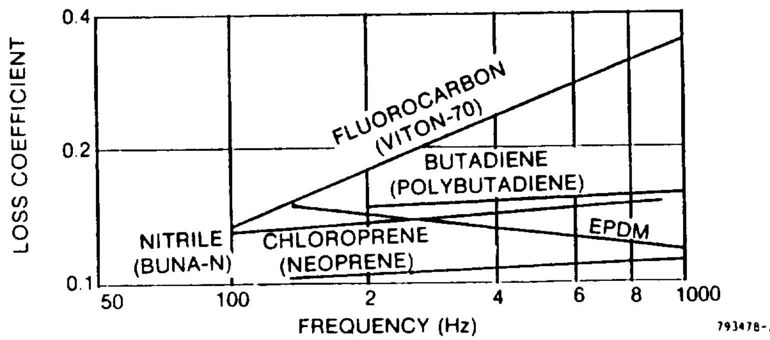

Figure 5. - Shear specimen data at 66°C.

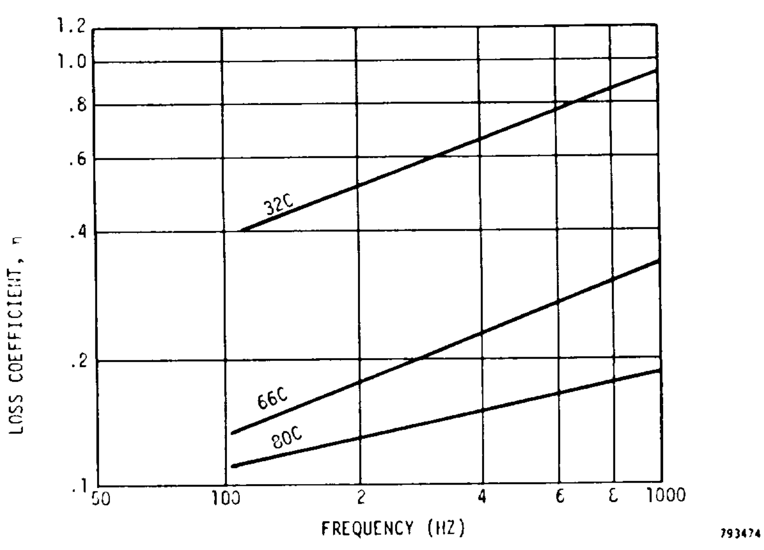

Figure 6. - Fluorocarbon (Viton-70) shear specimen data.

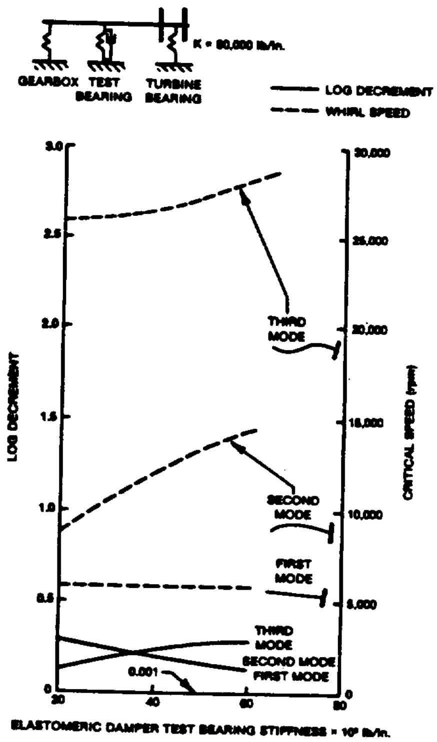

Figure 7. - T-55 power turbine - critical speeds and log decrement vs. bearing stiffness for Viton-70 elastomeric damper (η = 0.15).

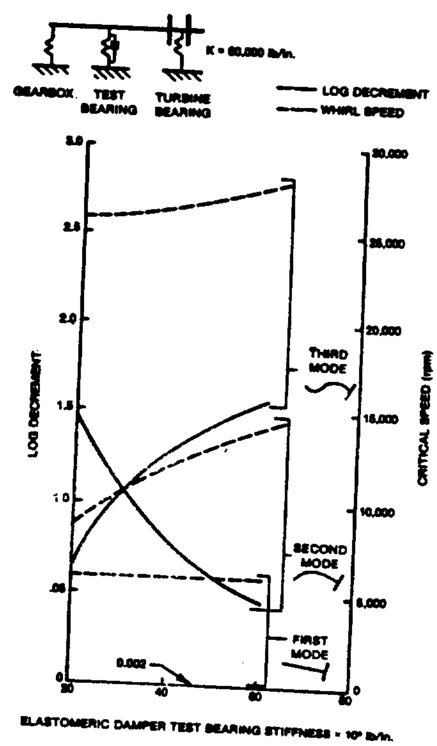

Figure 8. - T-55 power turbine - critical speeds and log decrement vs. bearing stiffness for Viton-70 elastomeric damper (η = 0.75).

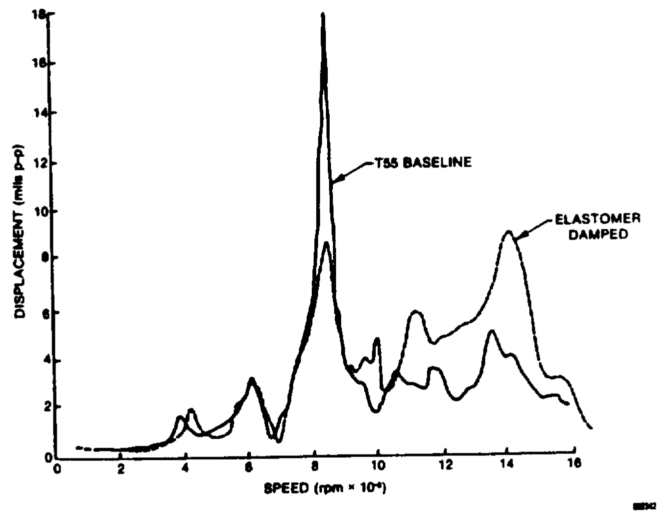

Figure 9. - Vibration plot comparing baseline and elastomer mounted rotor response.

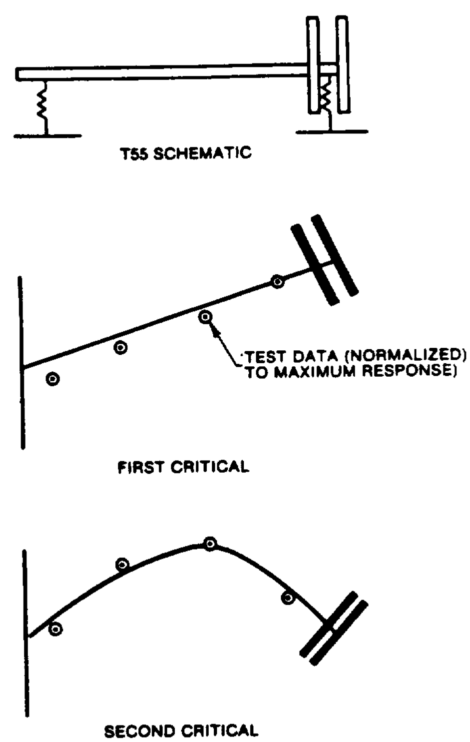

Figure 10. - Comparison of test data with analytic predicted modes.

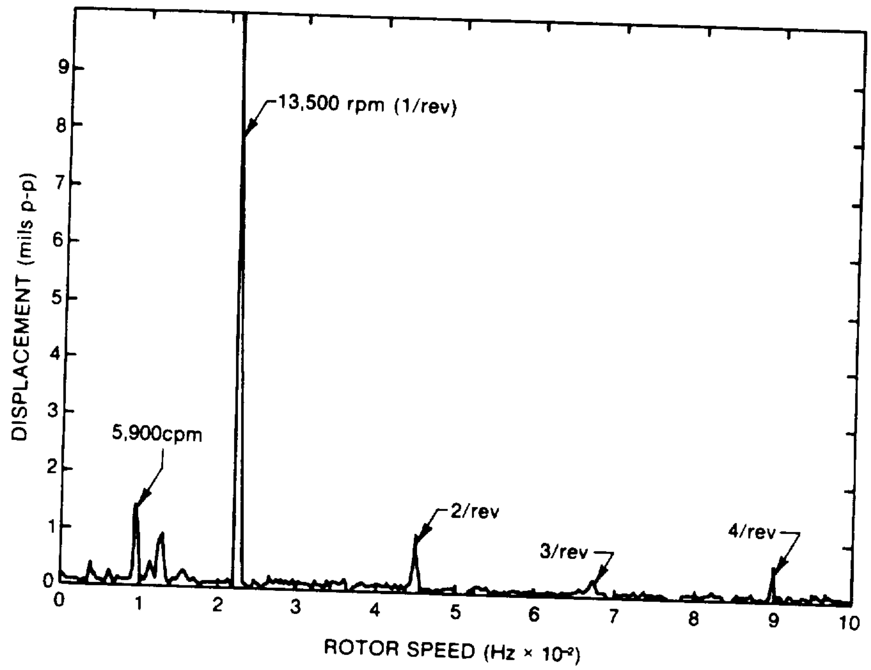

Figure 11. - Spectrum plot on elastomer probe #5; 3 gm @ 300^{o} plane 1.