



# Experimental Methodology for Determining Turbomachinery Blade Damping Using Magnetic Bearing Excitation and Non-Contacting Optical Measurements

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## Abstract

Experiments to determine the effects of turbomachinery fan blade damping concepts such as passively shunted piezoelectric materials on blade response are ongoing at the NASA Glenn Research Center. A vertical rotor is suspended and excited with active magnetic bearings (AMBs) usually in a vacuum chamber to eliminate aerodynamic forces. Electromagnetic rotor excitation is superimposed onto rotor PD-controlled support and can be fixed to either a stationary or rotating frame of reference. The rotor speed is controlled with an air turbine system. Blade vibrations are measured using optical probes as part of a Non-Contacting Stress Measurement System (NSMS). Damping is calculated from these measurements. It can be difficult to get accurate damping measurements using this experimental setup and some of the details of how to obtain quality results are seemingly nontrivial. The intent of this paper is to present those details.

## 1.0 Introduction

NASA's Fundamental Aeronautics Program is interested in pursuing research in technology areas which facilitate the development of lighter, quieter and more efficient fans for propulsion applications. One such application is commercial fixed wing, powered flight. High performance fan blades designed to achieve such goals will be subjected to higher levels of aerodynamic excitations which will undoubtedly lead to more serious and complex vibration problems (Ref. 1). There is concern over the levels of vibratory stresses, the reduction of high cycle fatigue (HCF), and aeroelastic instabilities. Furthermore, blade material and mechanical damping become issues of concern when material is removed from the blade design and when advanced stage design increases the number of vibration modes within the engine operating speed range (Ref. 1).

Historically, NASA Glenn Research Center has used the Dynamic Spin Rig (DSR) to characterize the vibrations of fan blades and bladed structures and to study damping. In the past several years, this facility has been used to assess the effects

of advanced blade damping concepts on flat plates and subscale fan blades. Damping concepts, such as viscoelastic material patches (Ref. 2), impact dampers (Ref. 3), plasma-sprayed damping coatings (Ref. 4), and high-damping high-temperature shape memory alloy materials (Ref. 5) are typically investigated in this facility. Recent experiments explore the possibility of using piezoelectric materials in the form of patches to dampen problematic blade vibration modes (Refs. 6 and 7).

To facilitate the effective testing of damping concepts, an active magnetic bearing (AMB) support system was designed for the DSR with the intent of using it to excite blade vibrations (Refs. 8 to 10). Modifications to the DSR bearing electronics and controls since last publication, including some used by Jansen, et al. (Ref. 11) and Dever, et al. (Ref. 12) will be discussed. Maturation of AMB techniques to excite the blades along with those historically used to obtain damping measurements are also presented.

## 2.0 Dynamic Spin Rig Facility

The DSR can be used for a variety of turbomachinery studies which include:

- Determining the effects of centrifugal loading on blade structural properties and behavior.
- Determining the feasibility of new damping concepts.
- Determining the effects of rotation on candidate damping system components utilized in the rotating frame.
- Determining structural damping coefficients versus rotational speed, which can be used in aeroelastic finite element analysis codes. (Structural damping data determined experimentally can be used analytically to determine blade and vane response to unsteady loading).

## 2.1 Test Facility

Test articles, up to 80 cm diameter, usually consisting of a rigid hub and two blades, are mounted on a vertical rotor simply connected to an air turbine with a pin in slot. The rotor is supported on three AMBs (Fig. 1). An axial magnetic

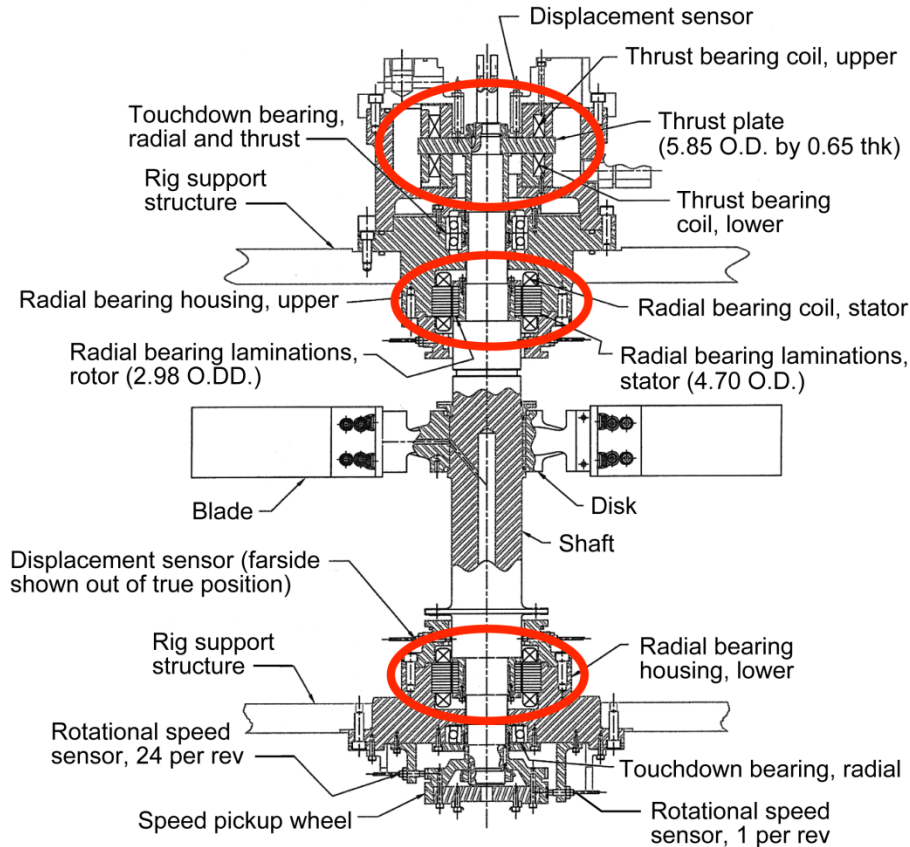


Figure 1.— Spin rig assembly schematic showing AMB locations in red.

bearing carries the weight of the rotating system. The two other magnetic bearings provide radial support. Grease-packed ball bearings provide back-up rotor support in the event of an AMB failure or instability which results in rotor excursions beyond a 0.25 mm radius. All rotating and support hardware are attached to the underside of the vacuum tank lid. Together, the lid and test hardware are lowered into a 1.22 m diameter, 2.54 cm thick steel vacuum chamber for testing (Fig. 2). The tank has a 7.62 cm thick safety ring at the top to contain hardware in the event of a failure while spinning. A vacuum level of 70 Pa is typical.

Blade excitation is achieved using the same radial AMBs which maintain rotor support and control. Details of this excitation system will be discussed in Section 5.

## 2.2 Other Excitation Methods

Several excitation methods have been used in the past, and are still available for use. These include a permanent magnet eddy current excitation system, air jet excitation, and electro-mechanical shakers. The intent of this work is to use AMBs to replace these other methods.

Permanent magnets have been used to excite titanium blades at fixed engine orders. The excitation frequency is fixed to the engine order defined by the number of magnet



Figure 2.—Test article lowered into dynamic spin rig.

circuits. To investigate other engine orders requires a mechanical reconfiguration of the magnets. A ring, holding these magnet circuits arranged uniformly around the circumference, is lifted using precision linear actuators towards the blades to excite them (Fig. 3). Eddy currents are generated within the blades to combat the change in magnetic flux encountered as they pass. This produces a force on the blades and on the rigidly supported magnets. The excitation force magnitude is a function of the conductivity of the blade material, the magnitude and direction of the magnetic field, and the speed of the blade relative to the stationary magnet circuit. Excitation force increases as the magnets are lifted closer to the blades because the reduction in air gap between the blade and circuit increases the density of the magnetic field. Often, to achieve blade resonant responses large enough to be effectively measured, highly electrically conductive coatings such as silver are added to the excitation regions of the blades. These coatings however, can alter blade damping. The main drawback of this method is blade heating. In an evacuated tank, there is negligible convective and radiant cooling. The blades rely on conduction to dissipate the localized heating. It is possible to melt the tips of blades using this method (Ref. 13).

Air jets placed in the vacuum chamber have been used. In previous tests, two jets 180° apart were used to excite blade vibrations. Disadvantages of this system include the reestablishment of aerodynamic loads and rig vacuum pump inability to maintain a vacuum pressure lower than 300 Pa.

Electromechanical shakers have also been used to excite blade vibration in a stationary frame. Axial oscillatory forces or moments are applied to the shaft through soft axial mechanical bearing supports (Ref. 14). Blade motion is induced by shaft motion and blade vibration amplitude varies with blade circumferential position. As with magnetic bearing excitation, blade natural frequencies can be excited independent of rotor speed. The soft mounting of the rotor supports, which facilitates shaft motion, introduces high levels of damping which contributes in some unknown amount to damping measurements of the test blade article response (Ref. 14).

The magnitudes of blade response to this form of excitation were often too low making it difficult to extract relevant information from strain gage data (Ref. 15). Problems like these led to the design of the current AMB excitation system.

Oil-jet spray excitation is a method not used at GRC, but has been used successfully by others. The U.S. Navy rotor spin facility uses multiple oil-jets with externally-controllable valves to excite various Campbell Diagram crossings up to 64E without stopping to reconfigure (Ref. 13). When using oil-jets, NSMS measurements are made with eddy current probes as opposed to light probes which are affected by the oil mist in the vacuum chamber.

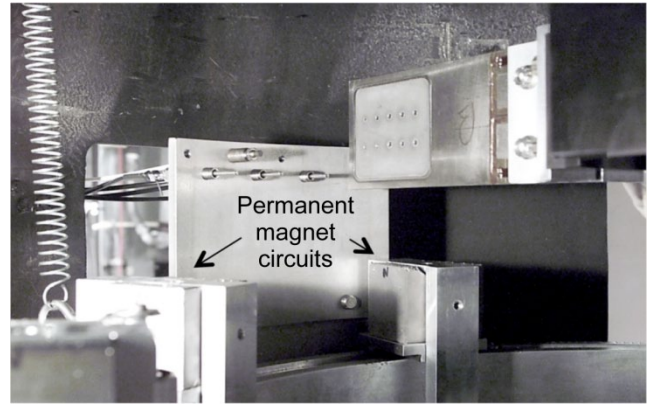


Figure 3.—Permanent magnet ring used to excite blades.

### 3.0 Magnetic Bearings and Control

Figure 1 shows the location of the magnetic bearings, eddy current displacement sensors, speed pickups, and auxiliary bearings. The thrust bearing, located at the top of the rotor system, is constructed of non-laminated components made from a soft magnetic alloy called Hyperco. Since the flux carrying components are not laminated, this thrust bearing has a low bandwidth, on the order of 50 Hz. This bearing was not intended to provide excitation to the rotor, but to simply support the rotor to eliminate possible frictional heating due to physical contact and to eliminate the potential for axial support effects on blade vibration damping. Axial bearing control derivative, which provides AMB control damping, is minimized to reduce its potential effects on same. The radial bearings are 4 pole with laminated cores and a force output capacity of approximately 1 kN each. AMB cores, like transformers, are laminated to increase bandwidth by hindering the development of large time constant eddy currents. This has the side benefit of reducing power loss and thus heat generation as well.

Prior magnetic bearing control algorithms were coded in Turbo Pascal or C and shared resources on the resident computer CPU. Control loop times, which were much faster than DSP-based systems, were unfortunately not constant. This made prescribing rpm tracking excitation from within the controller difficult. As a result, an extra input channel was needed to facilitate use of an external function generator. A dSPACE DS1005 system is currently used to establish control and facilitate robust excitation of the rotor. Compiled controllers run on a dedicated PowerPC chip. The AMB controller is coded in Simulink and also C inside of S-Functions. Fixed loop times on the order of 10 μsec are typical using this approach.

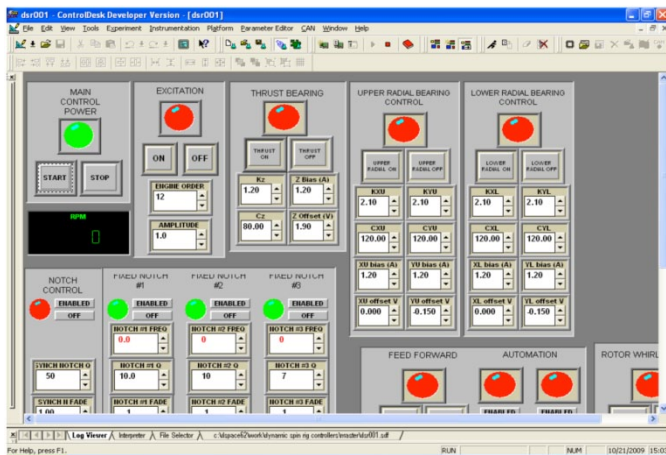


Figure 4.—dSPACE ControlDesk AMB user interface.

### 3.1 Controller Tuning

A conventional linearized independent PD control scheme is implemented. Figure 4 shows the dSPACE ControlDesk interface for manipulating the control parameters. For each AMB, the bias current, rotor position, stiffness (P), and damping (D) can be set here for each axis of operation. For example, each radial bearing has two axes and the PD values are independently set. Typically, the bias is set higher than the excitation amplitude to maximize excitation level and to avoid excitation harmonics. Stiffness and damping values are set to provide a stable rotor support and to minimize rotor excursions during excitation. Damping values are also minimized to reduce the possible effects of the controller in damping blade vibrations. All control parameters are kept consistent to avoid possible effects of AMB control on damping measurements.

### 3.2 Notch Filtering

Notch filtering of feedback control signals has proven to be an effective way to prevent the excitation of fundamental rotor system modes by sensor and other control system noise sources (Ref. 12). In this application, the shaft which floats in its support, is often lightly damped. Noise in the feedback control system tends to excite the rotor's first and/or second bending modes. Shaft ringing depends heavily on the construct of the rotor system. In a passive shunt experiment where several inductor coils were stacked with rubber disk separators and housed at the bottom of the rotor, no notch filters were required. This heavy mass at the bottom of the rotor shifted these frequencies lower and provided sufficient internal damping. To circumvent problems of this nature, fixed infinite notches centered about these frequencies are used to remove signal content at resonance frequencies from control feedback sensor signals. Figure 4 shows the user controls for these

notches. Although provisions for a moving notch are coded, only fixed notches are currently used. The location of the notch is set by inputting the center frequency. The width of the notch, can also be adjusted.

In simple PD control, the derivative of rotor position is used to generate damping. This derivative serves to amplify noise present in the eddy current position sensors. This may be the culprit in bending excitation. Alternative control approaches, such as that described in (Ref. 16), could improve AMB control and simplify it by eliminating the need for fixed notches located at rotor resonances.

### 3.3 PWM Switching Amplifier Filtering

The eddy current displacement sensors measure the rotor shaft excursions near each radial AMB in x and y directions. Eddy current sensor noise is influenced by high frequency flux generation around the bearing's electromagnetic coils. Pulse Width Modulation Amplifier (PWM) switching effects on eddy current sensors are discussed in detail in (Ref. 11). PWM output edge filters were installed in the DSR to reduce high-frequency magnetic field content broadcast by the AMBs and picked up by the position sensors.

### 3.4 Imbalance Compensation

The control software includes a manual imbalance compensator for both radial bearings. A thorough description of feed forward imbalance compensation can be found in (Ref. 12). A force vector synchronous with the rotor speed is generated by first prescribing its magnitude and then searching for the proper phase angle (referenced to a 1/rev) that minimizes the radial imbalance magnitude, at the radial bearing proximity probe locations, calculated using a digital vector filter. Force vector magnitude is then optimized to reduce the orbit further. Rotor orbits at the sensor locations are typically kept under 5  $\mu\text{m}$  during operation. Although the shaft has been spin balanced outside of the DSR, test articles and damping hardware of varying mass and uniformity are interchanged frequently during testing. The rotor system, as a result, is often out of balance and either needs to be removed completely from the rig and rebalanced, balanced in-situ using trial and correction weights, or balanced directly during testing using AMB feed forward compensation. The first two options can be time consuming and extend test program duration. The third saves time and has proven to be successful. Recent tests have had a rotational speed limit of 4,000 rpm, and compensation for any imbalance has been successful.

Minimizing rotor orbits during rotation and excitation should improve NSMS blade displacement measurements, facilitate a greater transmission of excitation force to the blade hub, and reduce the contribution of other damping mechanisms from the blade vibration measurements.



### 3.5 Excitation Limitations

The spin rig radial magnetic bearings were salvaged from a homopolar bearing used in prior research. These stators each have a force production capability of about 1 kN. Because the radial bearings operate in a vacuum, stator coil temperatures must be continuously monitored to assure that the wire insulation does not melt, producing shorts which could cause instability while spinning and possible permanent bearing and test rig damage. The radial bearing coils do heat up during testing. The higher the excitation amplitude, the faster they heat up. AMB coil temperatures have posed a problem in spin rig testing. When coil temperatures reach 80 °C, excitation is turned off and the bearings are allowed to cool. In addition, the AMBs have a frequency bandwidth of approximately 1 kHz, limiting the blade modes that can be excited.

### 4.0 NSMS Measurement System

The current spin rig setup, which utilizes full magnetic suspension, does not have a slip ring or telemetry system installed. No capability to monitor blade responses using strain gages exists. In this configuration, blade responses are determined solely using a non-contacting technique using lasers. IFOSYS optical laser spot probes are used to provide blade tip time of arrival information relative to an optical 1/rev located at the rotor mid-span close to the test article axial plane of rotation. Multiple probes are aligned radially with the center of rotation and normally equally spaced around a portion of the circumference of the stage. As the blades rotate by, tip timing data is sent to a non-contacting stress measurement system where the data is converted into displacements. From the blade displacements, blade response and damping can be determined. Hood Technologies NSMS Analyze software can be used to calculate a damping quality factor (Q). Time history data of blade ringdown after the excitation is removed can be exported and damping calculated from the vibration decay.

The title non-contacting stress measurement system is really a misnomer—at least in the way it is used in this application. Herein, response at the blade tips is what is required. In other applications, tip response is correlated with blade strain acquired simultaneously using strain gages attached to the blade. Predicted blade stresses associated with an experimental blade strain field are then coupled to tip responses which allows for the sole use of a NSMS system and elimination of the need for strain gages over the entire duration of a test.

### 5.0 Blade Base Excitation Techniques

Blades are excited by direct excitation of the hub. The radial magnetic bearings serve a dual purpose in both maintaining rotor position and control and also providing

blade excitation. Single frequency sinusoidal excitation is most commonly used, however, random or white noise is available and has been used to excite many modes at once. Since two bearings are used to generate excitation, various combinations of both forces and moments can be generated. Operating in a cylindrical or bounce mode, both bearings produce excitation force in phase. In a conical or tilt mode, the forces are out of phase and produce a resulting moment. Prior experience indicates that bounce excitation is better at exciting blade bending modes and tilt is better for torsional modes. However, this conclusion is for vertical flat plate testing and may not apply to twisted or pitched blades.

Hub-based excitation using radial AMB's as described herein is used for rotor configurations which include only two blade specimens (Fig. 1). The current focus is on individual blade damping. All fundamental modes for this configuration are observable and controllable at the hub (Ref. 17). If they are controllable, then they are excitable. However, certain blade modes in rotor configurations with more than three blades are not excitable using this method. Any circumferential modal pattern which produces no net inertial force or moment on the hub will not be excitable. Axial excitation using AMBs could be added to excite modes where a net axial force is required (Ref. 17).

#### 5.1 Fixed Excitation

Excitation can be fixed to a stationary or rotating frame of reference. In the fixed frame, the excitation that the blades experience, is related to the excitation frequency  $f_c$  and the shaft rotation speed  $f_r$  in the following way:  $f_b = f_c \pm f_r$ , resulting in two distinct excitation frequencies on the blades. This reduces the force level that the blades experience at the frequency desired. For this reason, the rotating excitation method was developed.

In fixed excitation (see Fig. 5),  $F_{excite}$  is stationary and,

$$F_{excite} = A \sin(2\pi f_c t) \quad (1)$$

where A represents excitation force amplitude.  $F_{normal}$ , which is the component of fixed excitation (shown perpendicular to the blade row) that excites blade vibration, can be expressed as,

$$F_{normal} = F_{excite} \sin(\Omega t) \quad (2)$$

Substituting (1) into (2) with  $\Omega=2\pi f_r$ ,

$$F_{normal} = A \sin(2\pi f_c t) \sin(2\pi f_r t) \quad (3)$$

Using a trigonometric identity,

$$F_{normal} = A/2[\sin(2\pi f_c t - 2\pi f_r t) + \sin(2\pi f_c t + 2\pi f_r t)] \quad (4)$$

Since  $f_c = N*f_r$  where N is an integer representing the desired engine order (EO) excitation,

$$F_{\text{normal}} = A/2[\sin[2\pi(N-1)f_r t] + \sin[2\pi(N+1)f_r t]] \quad (5)$$

Note that in order to achieve the proper excitation frequency in this mode of operation,  $f_c$  must be offset.  $N$  must be prescribed one EO above or below that desired. The blades will still be excited at two frequencies. Also note that the initial amplitude,  $A$ , is shared by the two sine functions.

## 5.2 Rotating Excitation

A unique feature of the AMB system is the ability to rotate the excitation with the blades. In this case,  $F_{\text{normal}}$  is computed directly, thus the excitation frequency  $f_c = f_b$ . There is no compensation for rotor speed. In the rotating frame, the blades will experience a single excitation frequency. A radial or transverse excitation force which follows the rotation of the rotor system is generated in the stationary frame of reference. This force vector is locked synchronously to the rotational speed and is phase controllable. An optimum excitation phase angle that produces the highest blade vibration amplitude for a given rotational speed and resonant frequency is currently determined manually.

Typically, a small to moderate excitation amplitude is prescribed while the rotor is spinning in the rpm range of interest for the proposed experiment at a speed where an engine order line crosses a fundamental resonance of the test article (Fig. 9). First bending is typically used because it is the easiest to excite and produces a relatively large tip displacement. The excitation vector phase angle is altered until the maximum blade response is achieved. The manual procedure described was implemented for bounce and tilt excitation. This method could easily be automated if strain gage or NSMS computation feedback signals were available to the controller.

In fixed excitation, an excitation force generated at a specific frequency in the stationary frame produces an excitation force with two frequencies symmetric about  $Nf_c$  offset by  $f_r$ . The inverse is also true and will be shown below: An excitation force with a single frequency generated in the rotating frame is produced by a force with two distinct frequencies symmetric about  $Nf_c$  and offset by  $f_r$ .

In Figures 6 and 7, orthogonal excitation current command components produced by the magnetic bearing controller are plotted for representative even and odd engine orders, respectively. These patterns are generated to produce a zero median sinusoidal excitation which is synchronized with rotor speed. In the 6-times-the-rotational-speed (6EO) example (Fig. 6), 12 lobes are visible. This is expected as for every sine wave there is a plus and a minus lobe. However, in the 5 EO example (Fig. 7), only 5 lobes are visible. In this case, there are actually 10 lobes—the pattern for the second half of a revolution has simply been repeated and sits directly on top of the first.

Figure 8 gives reassurance that the proper excitation is being produced in the rotating frame. This Fast Fourier Transform (FFT) magnitude and envelope plot provides the

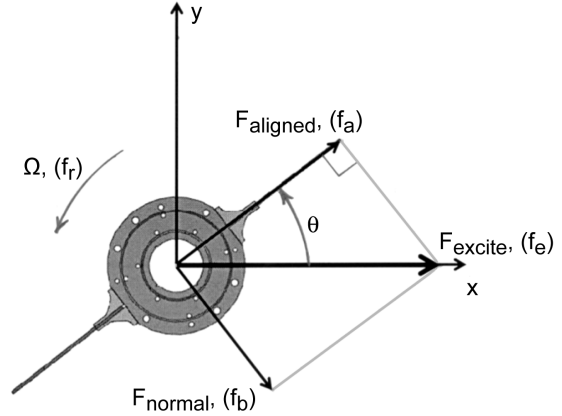


Figure 5.—Fixed excitation force and components.

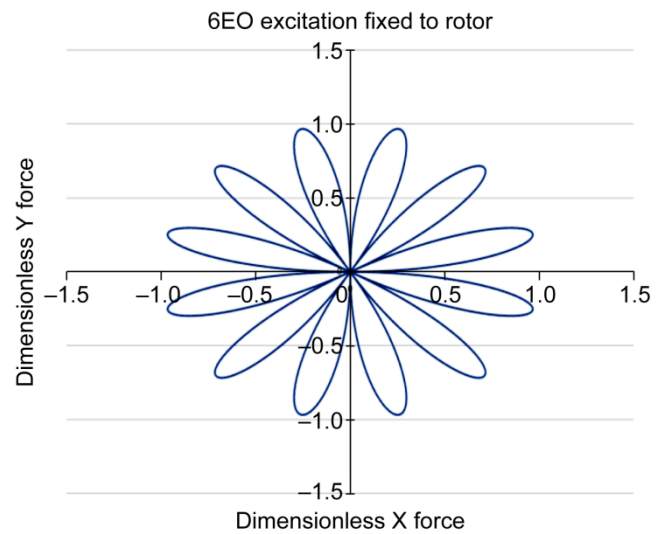


Figure 6.—Even EO AMB rotating force pattern.

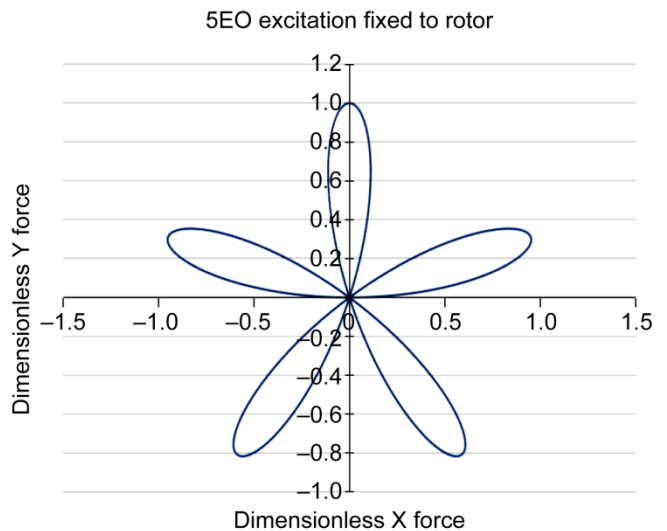


Figure 7.—Odd EO AMB rotating force pattern.

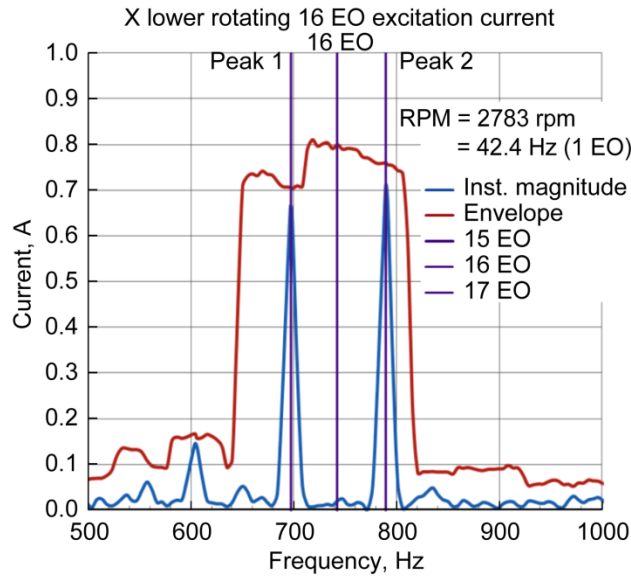


Figure 8.—FFT of commanded radial AMB current to produce a rotating 16 EO excitation at 2783 rpm.

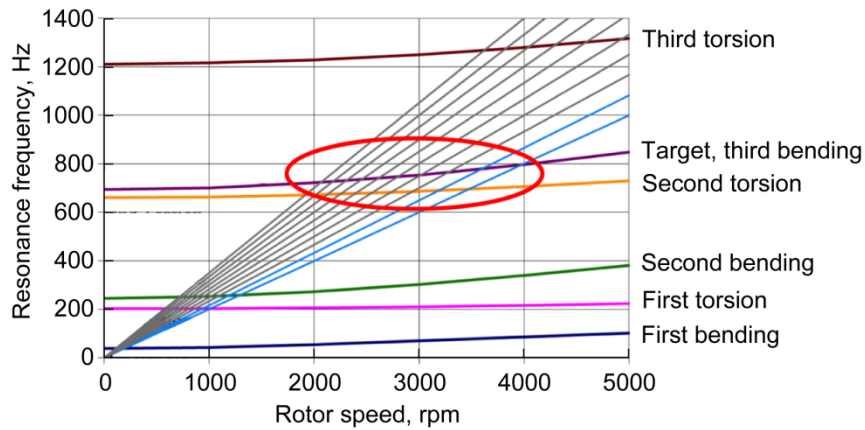


Figure 9.—Campbell diagram for flat Ti plates showing ten EO lines and third bending target intersections.

frequency content of the magnetic bearing control current (generated in the stationary frame) measured at the output of the PWM amplifier associated with a lower bearing control axis. Two side band peaks are shown straddling the rotating 16 EO excitation prescribed. In this example, the rotor speed ( $f_r$ ) was 2783 rpm (46.4 Hz). Excitation frequency,  $f_b$ , was 16 times that or 742.5 Hz. Peak 1 is 696 Hz and Peak 2 is 789 Hz, which are exactly  $16 \text{ EO} \pm f_r$ .

The red curve in Figure 8 shows FFT peak history information, over a specified rotor speed range, extracted from the entire data file. It has been included to qualitatively show the effects of magnetic bearing control and excitation system bandwidth on rotating excitation system amplitude. The negative sloping lines above the two excitation frequency peaks

show that for a fixed, prescribed excitation amplitude, actual applied current amplitude will decrease with rotational speed. Over a set test rpm range, this effect can be accommodated to preserve a fixed excitation either manually or in software. This feature has not yet been adopted in the DSR.

The phase angle between rotor position and synchronous excitation also changes with rpm. As explained, excitation force angle with respect to the 1/rev signal is set to maximize response. The optimum is determined manually at a single rotor speed and excitation frequency. As the excitation frequency changes, so too does this angle. As the angle changes the force vector effectiveness in producing a response will change. This effect also could be compensated for manually or by using an automated scheme.

### 5.3 Engine Order Jumps

A technique being called engine order jumps was developed to improve the acquisition of blade response data in a timely manner. Magnetic bearing excitation makes this rotating force technique possible. Engine order jumps, while using rotating forced excitation, cannot be easily implemented with other previously mentioned techniques (magnetic ring, air jets). It should be possible to use piezoelectrics to achieve similar capability, and ways of making this possible in a rotating frame are being explored.

In this technique, turbine acceleration and deceleration are set as low as practically possible which is approximately  $\pm 3$  rpm/sec in this facility. At speeds midway between locations where an EO line will intersect the target blade resonant frequency line (Fig. 10), the integer excitation EO multiplier value is incremented or decremented opposite the direction of rotation speed change.

The excitation tracks the EO line as rpm changes causing the blade to vibrate at the resonance of interest, then continues on until commanded to jump to the next line. Figure 11 shows an example of blade response versus time generated using this technique during a slow acceleration from 2000 to 3800 rpm.

### 5.4 RPM Hovering

Another technique developed to get useable blade response data is called RPM Hovering. When blade modes are lightly damped, 3 rpm/sec can be too fast for the Engine Order Jump Technique to produce blade responses which represent a full steady state response (Ref. 18). In RPM Hovering, rpm dwells with very small rpm excursions up and down are used to trigger a response at a slower ramp rate. RPM ramp rates lower than the 3 rpm/sec are achievable when the difference between the actual rotor speed and commanded rotor speed are on the order of 10 rpm. When the amplitude of a synchronous blade response jumps on the Hood Technology Acquire Software Display (Fig. 12), the AMB excitation is immediately turned off. The ringdown of the blade is captured and used to provide an alternative measure of damping. Figure 13 shows an example of multiple ring downs recorded during an RPM Hovering test. In this example rpm hovered around 1975 rpm.

### 6.0 Damping Calculations

Response peaks from rpm sweeps during engine order jumps, such as shown in Figure 11, are used to determine the level of damping. The Analyze software directly determines the quality factor Q and resonance frequency from the engine order response data. Q levels from damped and undamped blades are compared to determine the effectiveness of a particular damping technique. It is necessary to assume a single-degree-of-freedom system for this calculation, so the modes under investigation should be well-spaced. For

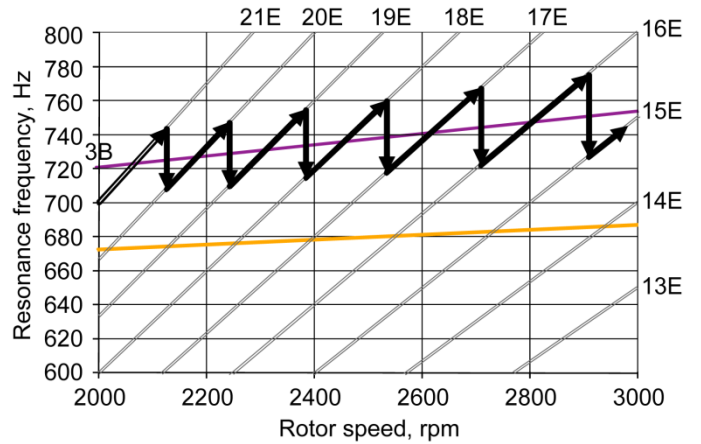


Figure 10.—Engine order jumping implementation example for a slow acceleration from 2000 to 3000 rpm.

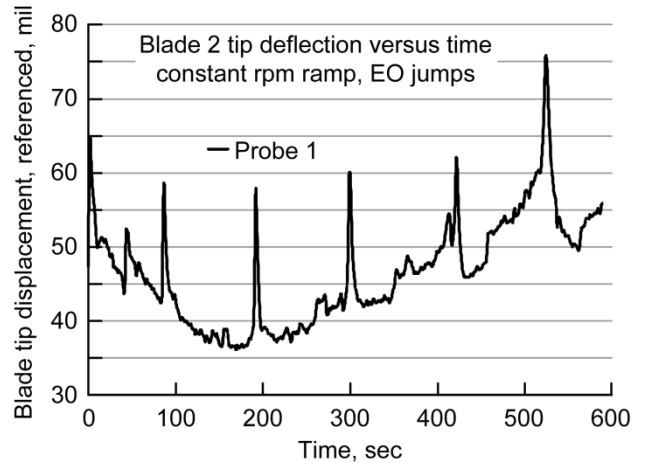


Figure 11.—Blade response versus time for experiments utilizing EO jumps.

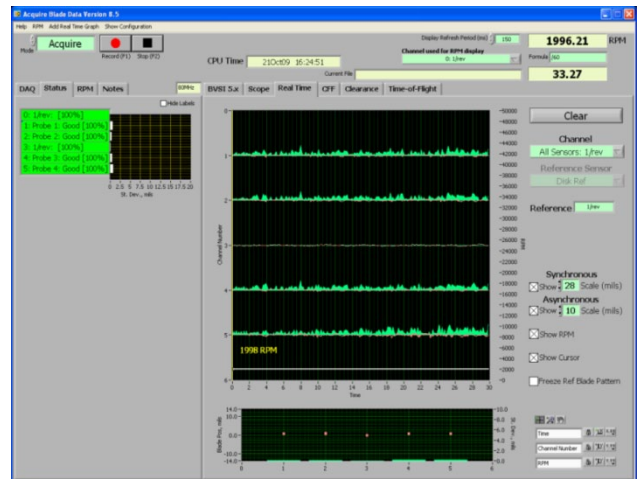


Figure 12.—Acquire interface showing in the middle synchronous blade responses in green.

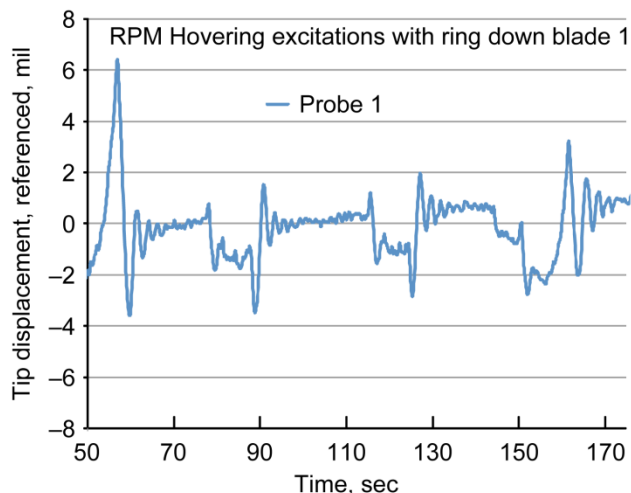


Figure 13.—Example of RPM Hovering excitations with ring down.

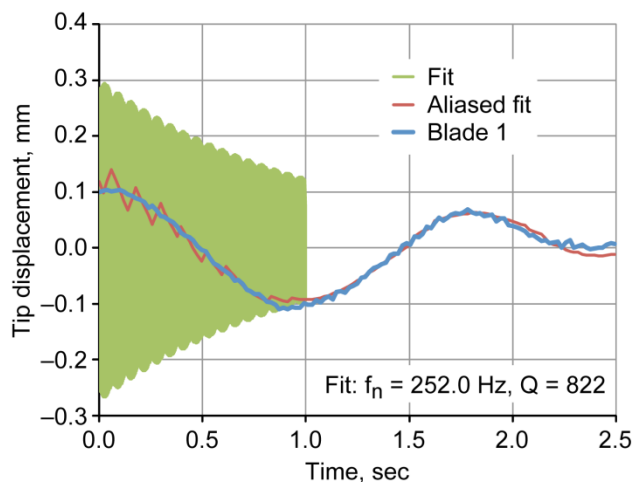


Figure 14.—Fit to hovering technique ringdown, blade 1 (aliased NSMS ringdown), fit (full ringdown fit), and aliased fit (aliased ringdown fit).

example, the plates tested in this study were designed to have no frequency veerings within the speed range of this test (e.g., third bending and second torsion modes in Fig. 9).

If the blade damping is very low, the rpm ramp rate may be too high to capture a good response peak. In this case, ringing will occur in the response. Therefore, the rpm hovering technique was used. The ringdowns shown in Figure 13 can be used to determine resonance frequency and damping. This is not a typical ringdown, because the NSMS measurement technique yields highly aliased data. There may be several blade oscillations between data points. However, since the excitation frequency is known at each data point, a fit can still be made to this ringdown, as shown in Figure 14. The “blade 1” data is the NSMS measurement with a time between data points of 0.0198 s. The “fit” data is the least-squares single-

degree-of-freedom fit with time between data points of 0.0005 s. The “aliased fit” is the same least-squares single-degree-of-freedom fit with data points only at the NSMS  $\Delta t$  of 0.0198 s. The Q of 822 calculated is very low, and would have required an rpm sweep rate of less than 1 rpm/sec. Comparisons between ringdown and frequency response data have been made, with fairly good correlation.

This method has been applied to linear damping treatments only. Extending it to a nonlinear damping system such as impact damping (Ref. 3) may be problematic. Impact damping is a nonlinear function of displacement amplitude rather than a constant value. The aliased fit calculates displacement amplitude versus time, and a single damping value for the specified time period. It would be difficult to accurately calculate both amplitude versus time and damping versus time, although the technique could yield qualitative information (e.g., vibration envelope shape).

The next set of tests planned for the DSR will include a slip ring, allowing strain gage and accelerometer measurements to be made. It is expected that more accurate damping calculations can be made with this instrumentation; however, the tests described in this paper do show that measuring damping with NSMS alone can be accomplished.

## 7.0 Concluding Remarks

The DSR, which uses a five-axis AMB system to support and excite rotating test articles to 80 cm in diameter and 20,000 rpm in a vacuum, is being used for structural turbomachinery blade vibration damping experiments at GRC. Details of excitation and speed control techniques developed to obtain quality damping measurement were discussed. Two such techniques, Engine Order Jumps and RPM Hovering, were used in recent damping tests of passively-shunted piezoelectric patches glued to rotating flat plate specimens. Blade tip timing data is obtained using NSMS probes and converted to displacement and damping quality factor Q using commercially available software or exported and post-processed by other means. Comparisons between ringdown and frequency response data have been made, with fairly good correlation. Possible excitation control improvements to preserve amplitude and phase during changes in rotational speed have been briefly discussed and yet to be implemented in the DSR.

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<b>14. ABSTRACT</b> Experiments to determine the effects of turbomachinery fan blade damping concepts such as passively shunted piezoelectric materials on blade response are ongoing at the NASA Glenn Research Center. A vertical rotor is suspended and excited with active magnetic bearings (AMBs) usually in a vacuum chamber to eliminate aerodynamic forces. Electromagnetic rotor excitation is superimposed onto rotor PD-controlled support and can be fixed to either a stationary or rotating frame of reference. The rotor speed is controlled with an air turbine system. Blade vibrations are measured using optical probes as part of a Non-Contacting Stress Measurement System (NSMS). Damping is calculated from these measurements. It can be difficult to get accurate damping measurements using this experimental setup and some of the details of how to obtain quality results are seemingly nontrivial. The intent of this paper is to present those details.					
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