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The Use of an Optical Data Acquisition System for Bladed Disk Vibration Analysis

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THE USE OF AN OPTICAL DATA ACQUISITION SYSTEM FOR BLADED DISK VIBRATION ANALYSIS

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SUMMARY

A 56 bladed rotor was studied using the newly developed and installed optical data acquisition system. The optical system was built so that experimental vibration data from multi-bladed rotors could be obtained. The optical system was shown to provide unique experimental capabilities unattainable with conventional strain gauge systems. In this study resonant frequencies, mode shapes, and vibration amplitudes are determined by the optical system.

By using sinusoidal excitation it was possible to investigate mistuned mode shapes. These arise in the rotor because of both frequency and damping mistuning. The mistuned mode shape for the zero nodal diameter, first bending mode was determined.

An air jet excitation which produces engine order excitation impulses by striking each blade as it passes in front of the air jet was found to be useful for exciting higher nodal diameter modes. The fourth and fifth nodal diameter modes were successfully detected by the optical system.

Random noise was used for locating the rotor resonant frequencies for the zero nodal diameter modes. The first bending resonant frequencies were identified by the optical system. A comparison among the resonant frequencies determined by the optical system, strain gauge measurements, and those computed by NASTRAN was made and found to be in fairly good agreement.

INTRODUCTION

An important consideration in the design of modern turbomachinery is the dynamic response of the multi-bladed disks such as compressor stages. The dynamic response of the compressor stage is important because it is subjected continuously to dynamic forces. Requirements for increased efficiency has led to lighter, more flexible compressor stages operating under more severe conditions. This has resulted in a growing need to reliably predict dynamic events such as flutter and forced response. Before these events can be predicted an accurate characterization of the dynamic properties of the bladed-disk is required. Unfortunately, currently available computer codes and analytical techniques are incapable of predicting dynamic properties of modern compressor stages with sufficient precision. Although considerable research has been conducted in an attempt to discover the deficiencies in existing analytical techniques, additional work is required.

The Lewis Research Center Spin Rig (ref. 1) was constructed so that experimental data could be generated to help understand the dynamic characteristics of rotating turbomachinery components. The spin rig was designed so that vibration data could be obtained on a variety of bladed-disk assemblies. Thus far a simplified two bladed rotor with variable blade setting angle and sweep adjustment, and a 56-bladed compressor stage have been studied. Results from these investigations are in reference 1.

A cutaway view of the Lewis spin rig along with a multi-bladed rotor is shown in figure 1. The test rotor is spun in an evacuated chamber using the air drive motor shown on top of the rig. The rig can achieve speeds up to 16 000 rpm. Dynamic excitation is provided by using either a pair of electromagnetic shakers or an air jet which produces a once-per-rev impulse force near the rotor blade tip. The slight imperfection of the rotor-shaft also generates a natural whirling excitation. A more detailed description of the physical characteristics of the rig and its capabilities is provided in reference 1.

Strain gauges are used as the primary instrumentation for monitoring the vibration response of the test rotor blades. Output from the individual gauges can be viewed on oscilloscopes or fed through a spectrum analyzer so that the frequency content of the signal can be measured. The output can also be stored on magnetic tape for processing at a later time.

There are several limitations of using a strain gauge measurement system in the spin rig. When a multi-bladed rotor is studied in the rig it is impractical to adequately instrument every blade. This is because the strain gauge signal must be transmitted through a slip ring assembly that mates the rotor to the stationary rig. The slip ring currently installed in the rig can only accommodate 50 strain gauges. Since it is usually desirable to have at least three gauges on any given blade, only a fraction of the rotor blades can be fully instrumented. The large number of blades left uninstrumented leaves the response of most of the rotor unknown. Another limitation of strain gauges is that it is difficult to relate strain measurements to blade deflections. Consequently, amplitudes of vibration, and steady state deflections from centrifugal forces, are difficult to obtain. In addition to the above limitations, the weight of the strain gauge and wire on the blade will alter the blades mass distribution which can slightly affect its dynamic properties. Also, damage to the strain gauge and loss of strain gauge signals is very common in a rotating environment.

A new concept in instrumentation was developed by engineers at NASA Lewis to overcome the problems associated with a strain gauge measurement system (refs. 2 to 4). This new concept uses optical transducers located around the perimeter of the spin rig case to measure blade tip deflections. Each optical transducer (or "probe") is individually controlled by a microprocessor which digitally records blade deflection data. This new concept allows the tip vibration of every blade on the rotor to be monitored. After a data collection run the vibration data can be easily transformed into a frequency domain representation where the vibrating frequencies and mode shapes of the blade and rotor can be determined.

The purpose of the present study is to evaluate the performance of the newly installed optical data acquisition system and to investigate the dynamic characteristics of a 56-bladed rotor. Several vibration properties were studied. The optical system was first used to identify resonant frequencies and blade mode shapes at various operating conditions and using different excitation schemes. Nodal diameters were also identified from the dynamically excited rotor. The resulting frequencies and mode shapes were then compared to results obtained from strain gauge measurements and finite element analysis. Amplitudes of vibration were also examined. Finally, frequency mistuning between blades was investigated. Because it is impossible to manufacture identical blades there will always be structural differences among blades on the rotor. The optical system was used to measure the extent of the effect these manufacturing differences have on the rotors response to dynamic excitation.

OPTICAL DATA AQUISITION SYSTEM

The optical data acquisition system is made up of three interconnected systems. These systems are the set of optical probes which sense blade passage, the microcomputer array which receives data from the probes, and an executive control computer. The optical probes monitor the passage of the vibrating rotor blades by reflecting light beams off the tips of the blades as they pass in front of the probe. Each probe contains an optical transmitter and receiver. The transmitter emits a light beam radially inward towards the center of the rotor. When a blade tip passes in front of the light beam the beam is reflected back to the probe where the receiver detects a reflected signal (fig. 2).

The microcomputer array is used to record the "time" that the reflected signal is detected by the probe. A dedicated microcomputer is required for every probe to keep up with the rapid occurrence of the reflected signals. Each microcomputer has a 4K memory space for storing reflected signals. The microcomputer array also contains a high speed clock which provides a constant number of clock pulses in each revolution of the rotor. A counter is reset to zero at the start of a new revolution. By using the number of counts per revolution and the circumferential distance of the rotor a conversion factor can be computed that will convert the blade passage time to blade deflection.

The data stored in the microcomputer rack is transmitted to a Hewlett Packard HP 1000 computer at the conclusion of a data collection run. At this point the data is sorted and used to construct a complete vibration record for each blade on the rotor. The HP 1000 is also used for transforming the data into the frequency domain (i.e., Fourier transforms and power spectrums). It also can display interblade phase diagrams and any of the spectral analyses on a graphics terminal.

The optical system uses an array of 16 optical ports equally spaced around the perimeter of the spin rig case (fig. 3). Each of the 16 ports contain three probes allowing for a total of 48 probes in the system. The top probe in the port is used to monitor the position of the leading edge of the tip, the middle probe the tip midchord, and the bottom probe the blade tip trailing edge. This arrangement can distinguish bending and torsional vibration modes. An example of the modes that the probes can distinguish are shown in figure 4. The 16 groups of probes provide enough resolution to analyze most of the modes that are of practical interest.

The performance of the optical system is dependent on the number of blades on the rotor, the rotor speed, and the number of data points that the optical system can store. Since there are 48 microcomputers and each microcomputer has 4K of memory a total of 196 608 data points can be stored for a single data collection run. Since there are 56 blades on the test rotor, a maximum of 3510 data points can be stored for each blade or 1170 data points for each of the leading edge, midchord, and trailing edge probe locations. More data points per blade can be stored if there are fewer blades on the rotor.

The rotor rotational speed determines the rate at which data points are taken. The faster the rotor is spinning the quicker each blade passes from port to port where data points are taken. For example, when the rotor is spinning at 15 000 rpm it will take 0.25 msec for a blade to pass from one port to the next. This results in a sampling rate of 4000 data points/blade/sec. At this rate the optical system can detect unaliased frequencies up to 2000 Hz (ref. 5). At slower rotor speeds the maximum detectable frequency will decrease. At 15 000 rpm it would take about 1 sec to fill all the available memory and complete the data collection run.

The deflection resolution obtainable by the optical system is determined by the number of pulses/revolution generated by the system clock and the circumference of the test rotor. For a rotor with a 62.83 in circumference (test rotor) and a maximum clock rate of 72 192 counts/revolution the smallest discernable blade tip deflection is one count or 0.00087 in. This is the highest deflection resolution obtainable by the optical system in its current configuration.

TEST ROTOR DESCRIPTION

A photograph of the test rotor (a compressor stage) used for this study is shown in figure 5. This rotor was originally designed for an aerodynamic performance study (ref. 6) and was later used for an evaluation of the Lewis Research Center Spin Rig (ref. 1). The rotor is typical of an axial-flow transonic compressor stage except for the disk which is considerably thicker than normal. The test rotor is 20 in in diameter and has 56 blades. The blades are structurally coupled by the hub and by a shroud located at approximately blade midspan. At low rotational speeds, adjacent shrouds are free to slide against one another. This tends to create a considerable amount of friction damping. The damping is eliminated at higher rotor speeds (above 8000 rpm) where the shrouds lock up.

SINUSOIDAL SHAKER EXCITATION

Initial tests using the optical system were performed using the electromagnetic shakers and sinusoidal motion. The shakers excite the rotor by enforcing an axial motion in the rotor shaft. A wave generator was used to sinusoidally drive the shakers at 525 Hz while the test rotor was spinning at 8700 rpm. The sampling parameters used were 1024 data points/blade and a maximum detectable unaliased frequency of 1160 Hz. The resulting vibration signal obtained by the optical system for the leading, midchord and trailing edge of blade 1 on the rotor is shown in figure 6. Although 1024 data points were collected from the vibration signal only the first 128 are shown in the figure. For this rotational speed 128 data points corresponds to 1 msec. Sixteen data points corresponds to one revolution since there are 16 ports around the rotor perimeter. The figure indicates that the blade response is approximately sinusoidal with an amplitude of vibration of approximately 3 mils. It is evident that the vibration signals at the three probe positions are in phase indicating a blade bending mode.

The auto spectrum of the vibration signals are shown in figure 7. As expected the signal has a strong frequency content corresponding to the excitation frequency of 525 Hz. The vibration signals for the rest of the blades are similar to blade 1. This exercise verified that the optical system and its post processing software was operating properly.

Driving the shakers with sinusoidal input is not a convenient approach for locating the rotor resonant frequencies with the optical system. The only way this type of excitation can be used to identify resonances is to excite the rotor at many different frequencies, collect data at each individual frequency with the optical system, and then identify the resonances by picking out the frequency with the maximum response. The data from each individual run provides the magnitude of the response at the excitation frequency but does not provide enough information to determine whether the excitation frequency is a resonant frequency. This approach for locating resonances is both inefficient and time consuming. A more efficient approach using random excitation to identify resonances is discussed later.

AIR JET EXCITATION

The air jet induces dynamic excitation by striking the face of each blade near the tip as it passes past a small tube inside the rig that emits a steady flow of air. The magnitude of the impulse on the blade can be regulated by varying the air pressure in the tube. The rate at which each blade receives impulses is related to the rotor speed. The rotor speed frequency and its integer multiples (engine orders), appear as dominant excitation frequencies.

An example of an air jet induced vibration signal detected by the optical system and its corresponding auto spectrum are shown in figure 8. In reality the blade deflection data oscillates about zero, but it appears slight fluctuations in rotor speed cause the optical system to shift from the zero reference. This explains why the deflections computed by the optical system drift away from the correct abscissa. This effect does not appear to hinder interpretation of the results.

As expected, the dominant frequency components of the vibration response signal shown in the auto spectrum plot appear at the engine order frequencies. This is expected since the frequency of the air impulse excitation is also at the engine order frequencies. The power for the fourth engine order (733 Hz) exceeds the power for the rest of the engine order responses. It was thought that this was due to the fact that the fourth engine order was close to a rotor resonant frequency. An effort to pin point the exact crossing of the fourth engine order with rotor resonant frequency was unsuccessful.

The air jet excitation provides a convenient means for exciting rotor modes other than the zero nodal diameter (umbrella) mode. The umbrella mode which is characterized by all blades vibrating in phase can not be excited by the air jet because the air jet strikes each blade at a different time. This produces nodes which are fixed in space but which can be viewed as backward traveling waves in the bladed disk. Some of these nodal diameter backward traveling waves, which are characterized by an interblade phase angle of (360 by N)/56 blades (N is the number of nodal diameters), were excited by the air jet.

Nodal diameter modes are usually grouped into families (e.g., first bending, second torsion). The commonality between modes in the family is that the individual blades vibrate with much the same mode shape. The difference between nodal diameter modes in the family is that they have a different number of nodal diameters. The frequencies of the modes in the family will be slightly different but are usually packed into a relatively small frequency band. The calculated frequency distribution for the first bending family of modes for the test rotor is shown in figure 11.

Each nodal diameter mode can only be excited by the frequency component in the air jet impulse that corresponds to that nodal diameter number. For example, the fourth nodal diameter mode can be excited by the fourth engine order frequency. Engine orders other than the fourth are not capable of exciting this mode. This is because the generalized force for any nodal diameter computed at any engine order is zero unless the mode number is equal to the engine order number.

The interblade phase angle between each blade for the fourth and fifth nodal diameter modes are shown in figure 9. Both the theoretical and experimentally obtained interblade phase angles are shown in the figure. The experimental phase angles were determined by computing the cross spectrum between blade 1 and each of the other blade vibration signals and then plotting the phase angles at the desired frequencies. For this figure the fourth and fifth engine order at 600 and 750 Hz were chosen.

RANDOM EXCITATION

In addition to driving the electromagnetic shakers with sinusoidal input the shakers were driven with random noise. The random noise generator was used in conjunction with a bandpass filter for eliminating excitation frequencies outside the range of interest. It was advantageous to limit the random excitation to a band of 2000 Hz or less so that the shakers would have adequate energy in the desired frequency range. When a wide band excitation was attempted the energy level was too small to sufficiently excite the rotor.

The random noise excitation was very useful for locating the rotor resonant frequencies. Since the random noise input signal contains an evenly distributed energy level throughout the frequency band, any rotor resonant frequency lying in the band will be excited. The dynamic characteristics of the rotor will filter out off resonant responses so that the resonant responses will dominate the blade vibration signal. The auto spectrum of the vibration signal will have peaks at the resonant frequencies. In practice, it was necessary to make only one or two data collection runs to locate the resonant frequencies at a single rotor speed. For the initial run a relatively wide band random signal is used to determine the general frequency range where the resonances are located. The frequency band is then narrowed and a subsequent run is made so that the resonances can be computed more accurately. This procedure can be repeated at several operating speeds.

Figure 10 shows the results from a data collection run where narrow band random noise was used to excite the rotor. For this run the energy in the excitation was evenly distributed in the frequency band of 500 to 600 Hz. A rotor resonant frequency was found at 550 Hz by computing the auto-power spectrum for the vibration signal. The resulting power spectrum is shown in the figure. The resonant frequency at 550 Hz along with two more computed at 7000 and 13 000 rpm are plotted in figure 11. Figure 11 also shows the location of the resonant frequencies detected by strain gauge measurements and those computed by the finite element computer program NASTRAN. The NASTRAN analysis is discussed in reference 1. The finite element model used for the analysis is shown in figure 12.

All of the nodal diameter modes computed by NASTRAN for the first bending family are shown in the figure. There are 28 nodal diameters possible since there are 56 blades on the rotor. Higher frequency modes other than the first bending family were computed by NASTRAN but are not shown in the figure. The other modes are at higher frequencies.

The strain gauge data and the optical system data correlate very well for the first bending, zero nodal diameter mode. Both sets of experimental values are higher than the frequencies predicted by the finite element analysis. This is somewhat surprising because it was thought that the finite element model was on the stiff side which would result in higher not lower frequencies. The model was thought to be overly stiff because the blade base was fully constrained and the shrouds were assumed to be rigidly connected when they were actually free to slide against each other.

An attempt to excite and locate the one nodal diameter mode with random noise by driving the shakers 180° out of phase was unsuccessful. Previous attempts using sinusoidal excitation have also been less than satisfying. It is impossible to excite modes higher than the first nodal diameter mode with the shakers because the excitation is induced by the shaft motion and the shaft can not excite the blades with the necessary phase differences. The one nodal diameter mode can be excited by driving the shakers out of phase which exerts a moment on the shaft. The moment will excite the one nodal diameter mode while the axial shaft motion excites the zero nodal diameter mode. Neither of the shaft motions can excite the higher nodal diameter modes.

The second bending family of modes which NASTRAN predicted to lie between 800 and 2800 Hz could not be detected by either the strain gauges or the optical system. The reason the strain gauges could not pick up the second mode frequencies is probably because the gauges are located near a point of low strain for the mode shape. The reason the optical system did not detect the second family of modes is probably because the deflections for this mode are too small. The rest of the family of modes could not be discovered by the optical system because their frequencies are higher than the systems maximum detectable frequency capability. The blade tip deflections for these higher modes would also be to small for the system to detect. For future studies it would be desirable to use a more flexible rotor which would have lower resonant frequencies and larger displacements.

ROTOR MODE SHAPES

The mode shapes for the test rotor can be found by determining all the blade deflections and interblade phase angles at a resonant condition. The deflection of each blade is dependent on the damping in the blade and the ratio between the blade alone resonant frequency and the frequency of the entire bladed assembly (ref. 7). Because manufacturing procedures produce unidentical blades and because the shroud contact and root connection vary among blades, each blade will have a different resonant frequency and level of damping. A rotor such as the test rotor, composed of blades with variable structural properties and resonant frequencies is known as a "mistuned" rotor. Blades that have blade alone resonant frequencies similar to the resonant frequency of the assembled bladed disk will exhibit large forced responses when the excitation frequency is at the resonant frequency, while blades with blade alone frequencies far from the rotor resonances will have weak responses. A rotor such as rotor-12 composed of blades with variable structural properties and resonant frequencies is known as a "mistuned" rotor.

The zero nodal diameter mode in the first bending family of modes is shown in figure 13. This mode shape was produced by exciting the rotor at 551 Hz. There were no phase differences between any of the blade vibration signals which verified that the rotor was vibrating in a zero nodal diameter mode. For an ideal rotor the amplitudes would all be equal. As expected the blade amplitudes are not uniform, but are irregular. The irregularity is attributed, at least partially, to frequency mistuning. Besides the frequency mistuning effects, the large variation in blade amplitudes could be caused by a difference in the amount of damping in each blade. The variation in damping can have as much or more effect on the blades response than frequency mistuning.

AMPLITUDES OF VIBRATION

A comparison between measured blade tip vibration amplitudes for the third and eleventh blade are shown in table I. This comparison was made to evaluate the accuracy of the displacements computed by the optical system. In this table optical system displacements are compared to displacements derived from strain gauge measurements. The optical system displacements and strain gauge measurements were recorded while the blade was vibrating in its first bending mode at 530 Hz and 9100 rpm.

Stain gauge data was converted to tip displacement using two approaches. In the first approach, a static, nonrotating, experimental calibration was made that determined the conversion between the strain in the blade midspan gauge and the blade tip displacement. The conversion factor was found by approximating the first bending mode shape by manually deflecting the blade tip, measuring the tip deflection with a micrometer, and recording the strain in the midspan gauge. The resulting relationship between strain and tip deflection was then used to convert the midspan strain gauge reading for the vibrating, rotating blade to the tip deflections shown in the table.

In the second approach a finite element analysis was used to find the relationship between the blade strain and the tip deflection. The correlation was found by first performing a NASTRAN modal analysis on the rotating blade model. The resulting first bending mode shape was then applied to a static blade model as an enforced displacement and a subsequent NASTRAN stress analysis was performed. The relationship between strain and tip displacement was found from this analysis by correlating the resulting stain with the magnitude of the mode shape at the blade tip. The relationship was then used to convert the strain data for the rotating, vibrating blade into tip deflections. Considering the inaccuracies in deriving the calibration factors and the small vibration amplitudes being examined, the match up between the optical system deflections and the strain gauge measurements are fairly good. The correlation between the optical system data and the experimentally derived deflection is especially good. The main reason for the difference between the optical system deflections and the NASTRAN computed deflections is the mesh coarseness. Although this mesh is adequate for frequency analysis it is inadequate for determining stresses and strains.

OPTICAL SYSTEM EVALUATION

There are several advantages that the optical system has over a strain gauge measurement system. For one, the optical system can monitor the response of three positions of every blade on a multi-bladed rotor while the strain gauge system is limited by the slip ring capacity to a total of 50 gauges. Another advantage is that the optical system measures vibration amplitudes directly unlike strain gauges, which measure strain that can not be related to amplitude without some type of calibration and knowledge of the mode shape. The optical system is also advantageous because data is taken in digital form which facilitates analyzing the vibration signal on a digital computer with readily available signal processing software.

There are also disadvantages associated with the optical system. First, the system is relatively complex making it comparatively unreliable and difficult to repair and use. Second, the maximum unaliased vibration frequency that the system can detect is limited by the rotor speed. At 15 000 rpm the maximum detectable unaliased frequency is only 2000 Hz. Frequencies approaching 10 kHz have been detected with the spin rig strain gauge system. Finally, the optical system cannot be easily adapted to rotors with geometries dissimilar to the test rotor. A new probe mounting case and some modifications to the optical system software would be required for studying other test rotors.

CONCLUDING REMARKS

A complete study was conducted using the optical data acquisition system to determine the vibration properties of a test rotor. Resonant frequencies, mode shapes, and vibration amplitudes were determined for this rotor. From this study it was determined that the optical system is a useful and accurate measurement system for studying multi-bladed rotor vibrations. It was also found that even though the optical system is a valuable vibration measurement system, it does have limitations. Because of these limitations it is desirable to continue to use strain gauges as a supplemental data acquisition system. The use of both systems jointly will provide a unique experimental capability for rotor vibration analysis.

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TABLE I. - COMPARISON OF BLADE TIP VIBRATION AMPLITUDES

[530 Hz, 9100 rpm.]

Blade	Optical system	Experimental[a]	Experimental[b]
3	2.5 mils	5 mils	2.8 mils
11	2.2	4.5	1.7
15	2.5	6	2.7

[a]F.E./gauge calibration.
[b]Defl/gauge calibration.

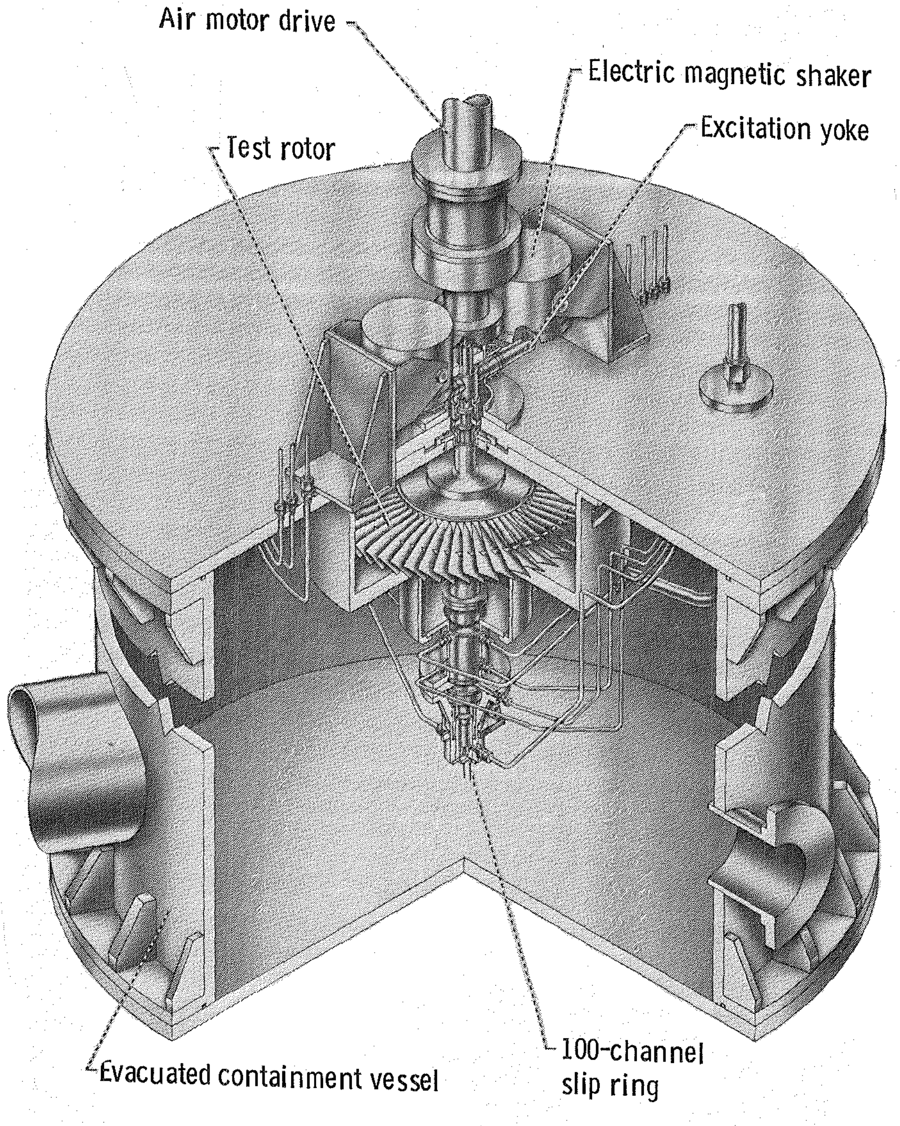


Figure 1. - Cutaway drawing of the spin rig.

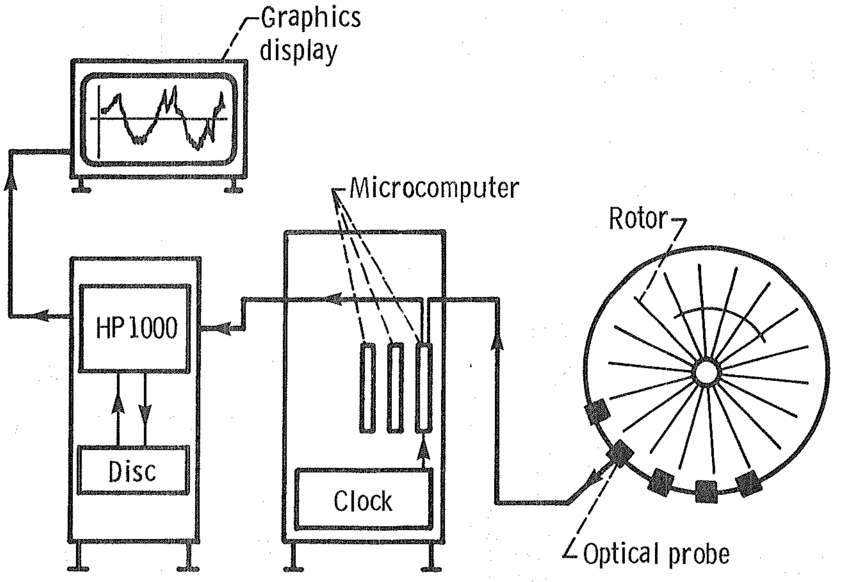


Figure 2. - Data aquisition system.

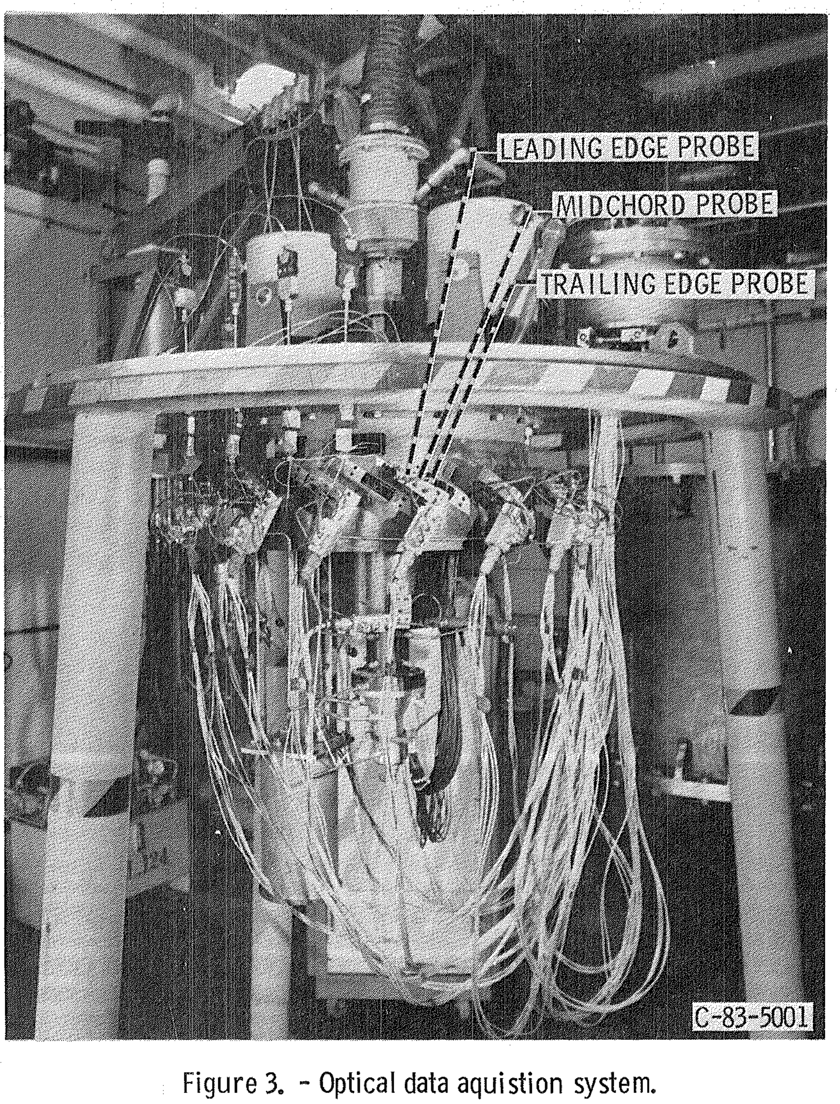
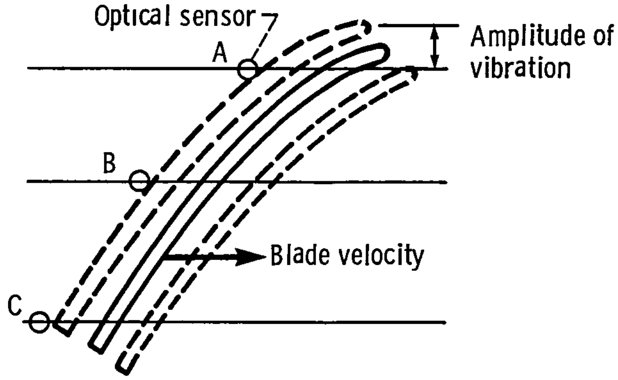
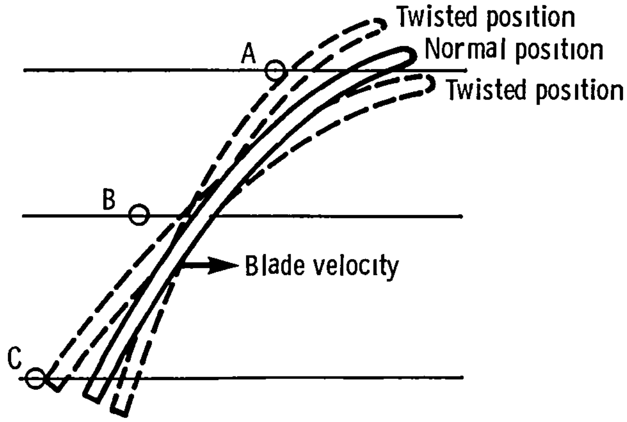


Figure 3. - Optical data aquistion system.



(a) Bending.



(b) Twist.

Figure 4. - Blade motions observed with 3 probes looking at the blade tip (radially inward).

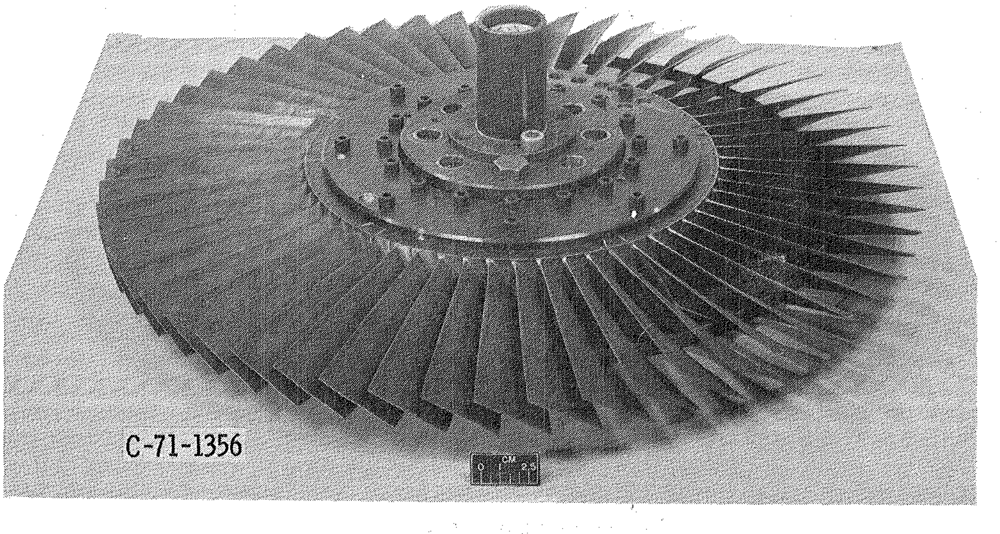
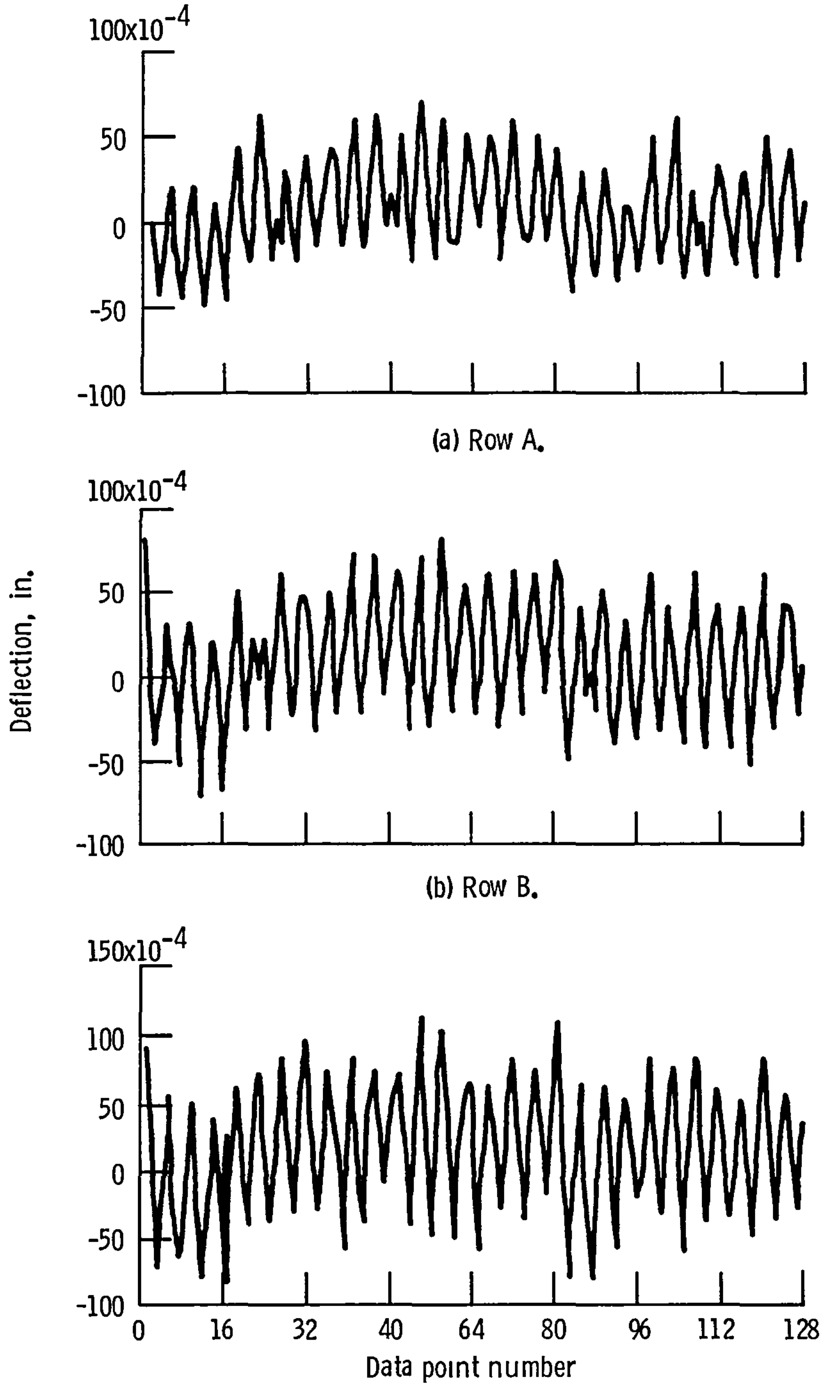
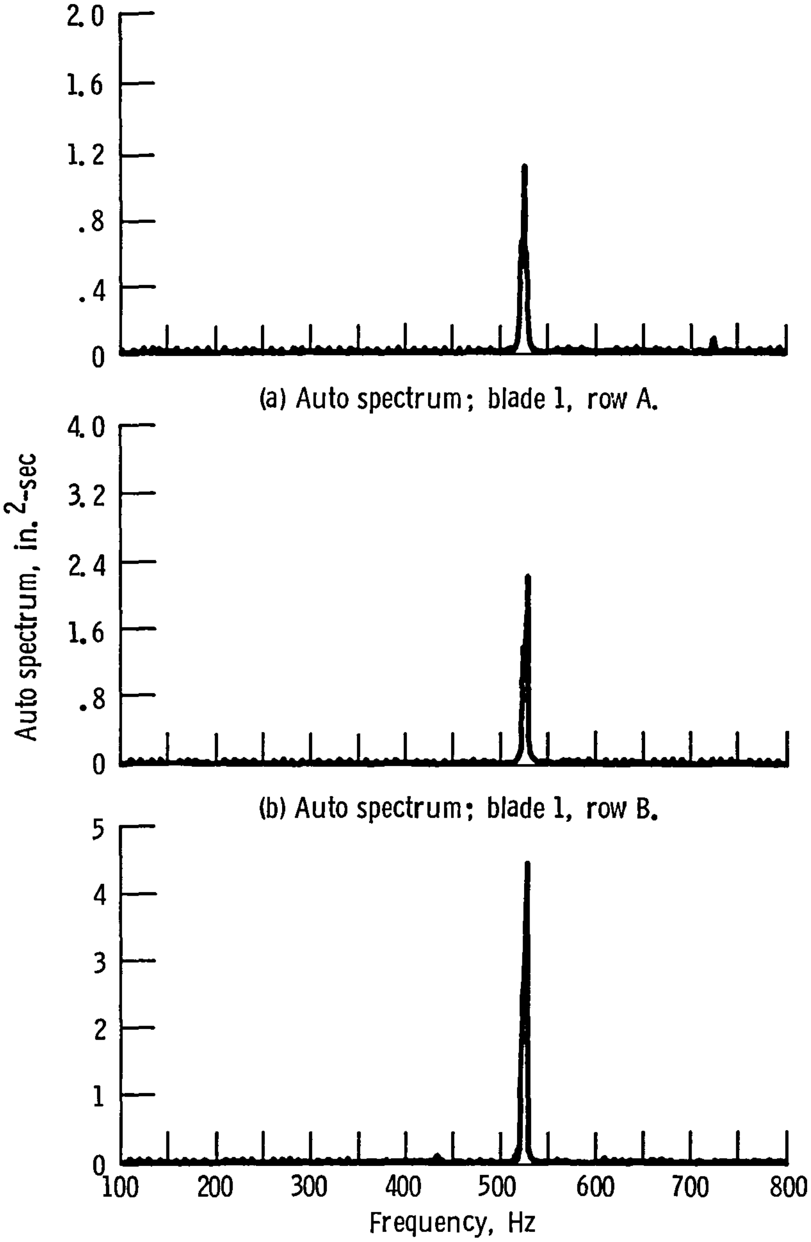


Figure 5. - 56 blade rotor.



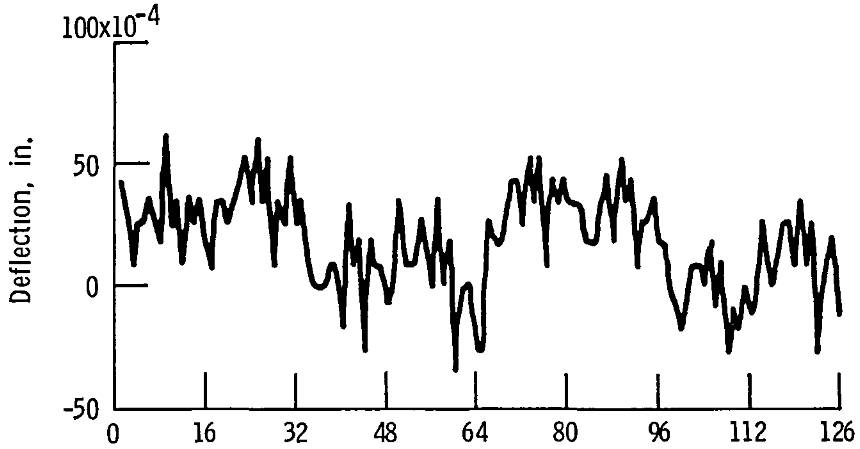
(c) Row C.

Figure 6. - Blade 1 leading, midchord and trailing edge vibration signals at 8700 rpm (sinusoidal excitation).

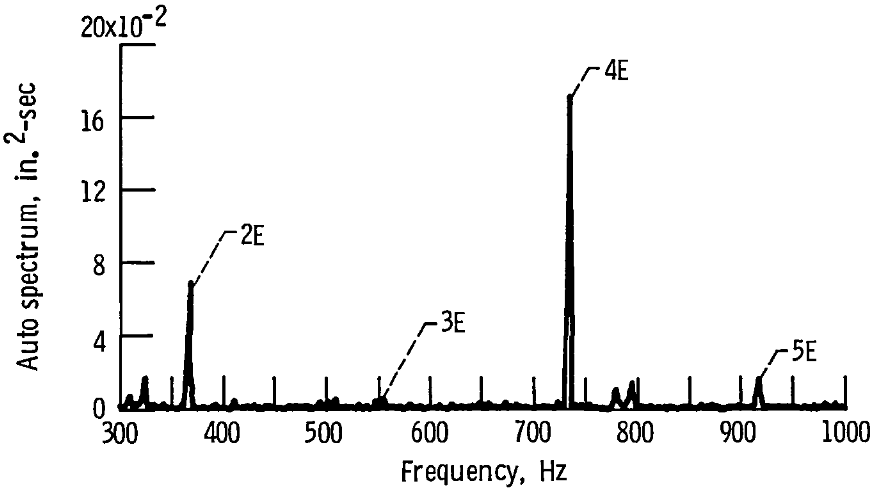


(c) Auto spectrum; blade 1, row C.

Figure 7. - Blade 1 auto spectrum (sinusoidal excitation).

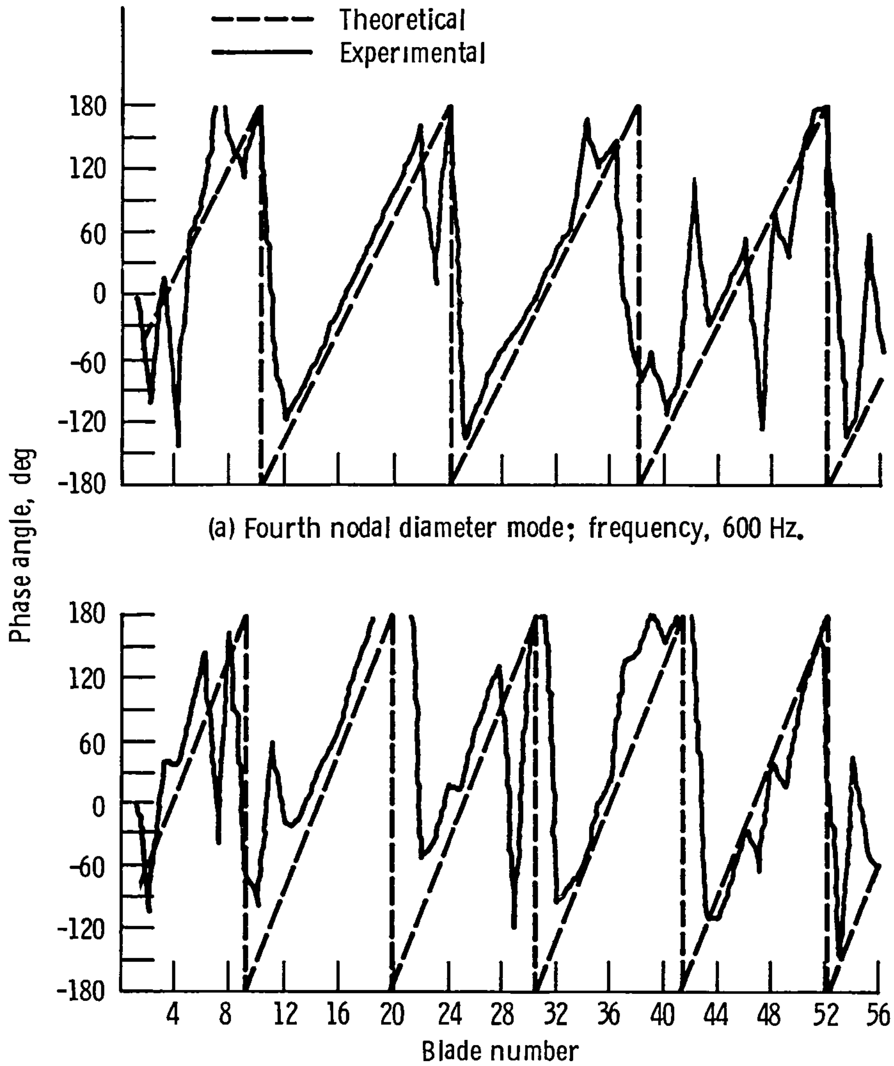


(a) Deflection data; blade 1, row C.



(b) Auto spectrum, blade 1, row C.

Figure 8. Blade response from air jet excitation at 11 000 rpm.



(b) Fifth nodal diameter mode; frequency, 750 Hz.

Figure 9. - Rotor interblade phase diagram for fourth and fifth nodal diameter mode at 9000 rpm (150 Hz).

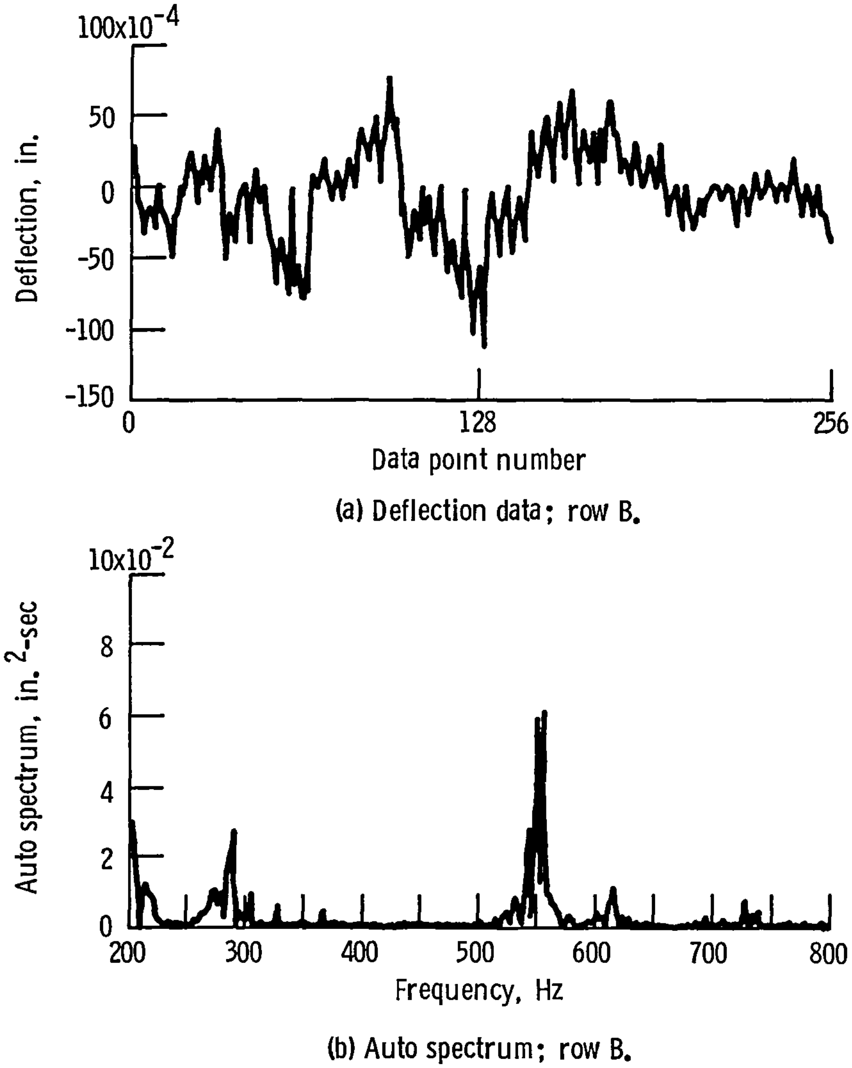


Figure 10. - Narrow band random excitation (500-600 Hz) at 9800 rpm, blade 11.

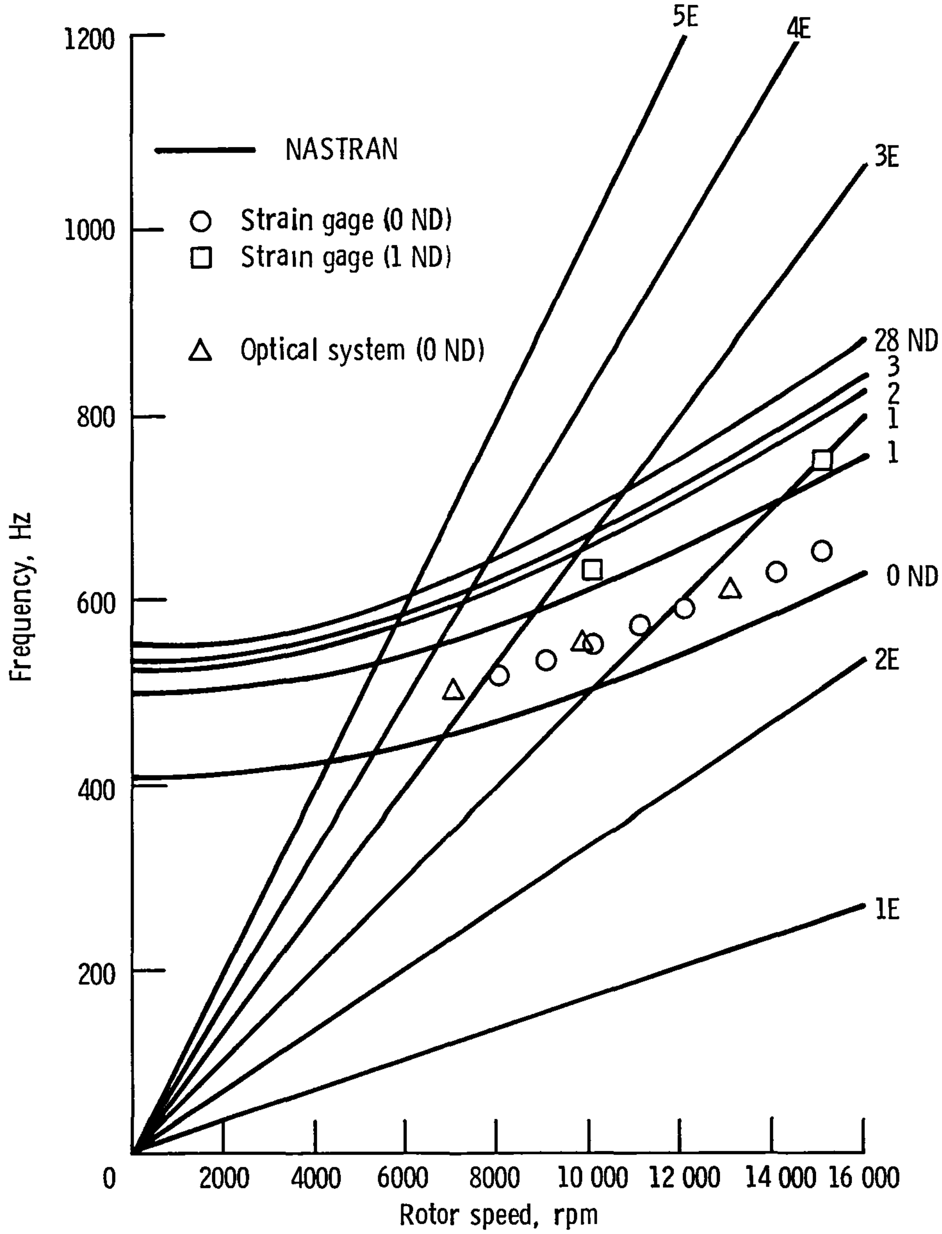


Figure 11. - Rotor Campbell diagram for first bending family of modes.

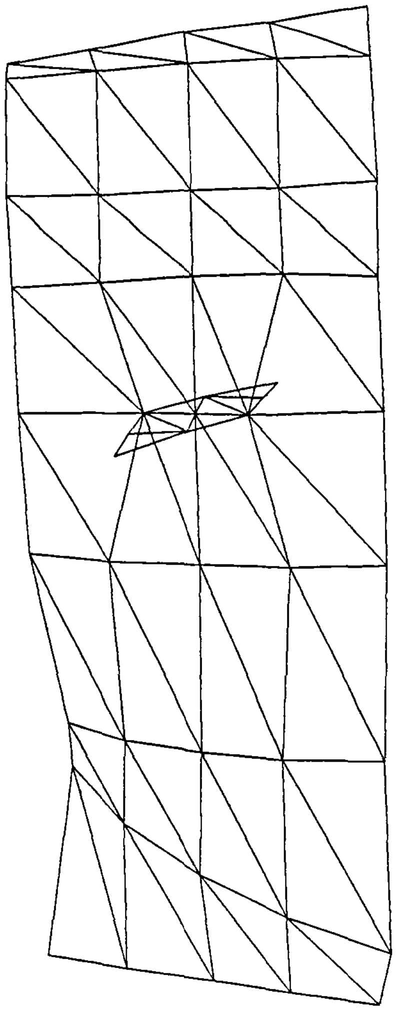


Figure 12. - Compressor blade finite element model.

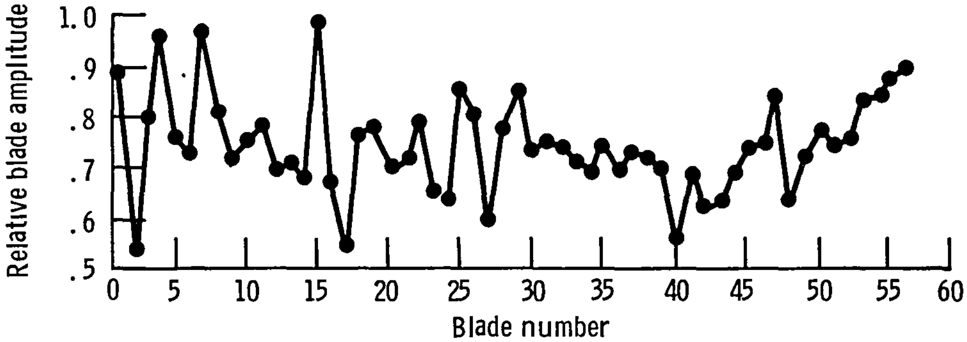


Figure 13. - 0 nodal diameter mode shape at 551 Hz (10 000 rpm).

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15 Supplementary Notes

16 Abstract

A new concept in instrumentation was developed by engineers at NASA Lewis Research Center to collect vibration data from multi-bladed rotors. This new concept, known as the optical data acquisition system, uses optical transducers to measure bladed tip delections by reflecting light beams off the tips of the blades as they pass in front of the optical transducer. By using an array of transducers around the perimeter of the rotor detailed vibration signals can be obtained. In this study, resonant frequencies and mode shapes were determined for a 56 bladed rotor using the optical system. Frequency data from the optical system was also compared to data obtained from strain gauge measurements and finite element analysis and was found to be in good agreement.

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