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Structural Dynamic Measurement Practices for Turbomachinery at the NASA Lewis Research Center

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SUMMARY

Methods developed for measuring blade and rotor-shaft system response include optical systems, transient instruments, and special digital data processing equipment. Optical methods offer some distinct benefits for blade vibration measurement. Transient and steady state measurements of the response of rotor-shaft systems strongly affect development of design analysis methods. Digital computing systems allow processing of large volumes of high speed data from rotating blade sets. Also, digital systems develop useful vibration response signatures from randomly excited systems. Research facilities include the spin rig facility and the transient rotor response laboratory.

BACKGROUND

Structural Dynamics at Lewis

The development of future propulsion systems requires improved dynamic analysis of high speed and flexible rotating systems. Propulsion system efficiency gains come from higher temperature operation, higher rotating speeds and decreased system weight. Each of these efficiency gains increases propulsion system flexibility and therefore, increases the requirement for better design methods. Dynamic coupling between components also increases and becomes a greater concern. Improved design analysis methods require more effective use of costly experimental data.

The structural dynamics research program at Lewis emphasizes development of design analysis methods. The focus is on problems associated with high speed, flexible turbomachinery systems. The research has four major thrusts: bladed systems, vibration control of blades and rotor-shaft systems, system dynamics, and computer methods.

In the bladed system dynamics area, we develop improved methods for analyzing the flutter and forced vibration response of bladed-disk assemblies. In the vibration control area, we develop new methods for controlling system vibration levels. Systems dynamics modeling studies develop understanding of the dynamics of ensembles of components. We are developing better computer algorithms and math models to describe dynamic systems. Finally, we exploit modern computer science practice to find better ways to solve structures problems.

Longer term research focuses on supporting future NASA missions and the widest range of end users. We give high priority to developing methods that will support the future needs of the United States.

We want to replace costly experimental engine development procedures with numerical methods. We need to assure end users that our methods are correct and useful. The development of confirmed design analysis methods supports our role of providing technology for United States interests. Also, better understanding of fundamental physical processes requires development of new measurement technology.

Paper submissions and symposia distribute our methods to the university community and to industry. Industry further confirms the usefulness of our methods by applying them to specific engine system problems.

Experimental measurements provide the basis for predictions of dynamic phenomena within engines. These measures are difficult to obtain from full scale engine systems. Therefore, we conduct most of our experimental work on small test rigs that simulate specific phenomena. Even with these experimental rigs, it is difficult to measure enough points to define the phenomena fully. We have begun using computer predictions along with measured data to compensate for sparse data.

Experimental Needs

Complex vibrations occur within bladed-disk assemblies. One cannot predict these patterns from the vibration characteristics of single blades alone. Bladed-disk assemblies vibrate as coupled structural systems. The white nodal lines in the hologram of figure 1 show a typically complex modal vibration pattern. Some of the blades are bending while others are twisting. This behavior is typical of bladed-disk vibration modes. These modal vibration patterns can rotate about the bladed-disk assembly. When the assembly is rotating, pairs of similar patterns can rotate in opposite directions. Blade flutter occurs when these vibration patterns interact with and extract energy from the gases flowing through the bladed-disk. Fatigue damage due to excessive forced vibration is also possible.

It is difficult to measure the response of a complete set of blades within an engine or even within a test rig. This is because of restricted physical access and high rotation speeds. The high speeds limit conventional strain gauge placement and the possible number of data channels in slip ring systems.

Low level damping within blade joints and small differences between blades can greatly affect vibration response levels. These otherwise small effects act to interfere with the energy exchange between blade vibrations and the gases flowing through the stage.

First, we needed to measure the vibration response of a bladed-disk assembly. Next, we needed to make precise measurements of single blade properties. Low level blade damping and small differences between blades significantly affect the overall response of bladed-disk assemblies.

Higher speed, flexible rotor-shaft systems also require better methods for predicting and controlling rotor response. For example, super critical shaft systems perform better and are light in weight. Their application requires precise knowledge of possible instability modes. Many factors affect rotor-shaft system dynamic instability modes and most are difficult to measure or predict. Elastic properties of rotor-shaft systems vary with rotor speed.

Complicating factors include large Coriolis effects, gyroscopic accelerations, and variable damping within the rotor-shaft assembly. Also, speed varying properties of bearing assemblies and the supporting structure influence the dynamics.

Many kinds of rotor forcing result in dynamic instability. These included blade rub induced rotor whirl, Alfred turbine forces, or hydro-pneumatic forces within the seals. Most vibration control strategies require predictions of the rotor dynamic response. Mathematical models of the physical processes that can affect rotor dynamic response require careful measurement to assure their effectiveness.

Flexibility increases dynamic coupling between the bladed stages of a rotor-shaft system. This has also proven difficult to measure. Rotor whirl modes can couple with blade vibrations on overhung fans. Dynamic coupling between the lower weight and highly integrated engine case structures and the engine core is significant. It is difficult to measure enough points to define the overall system response. Measurements coupled with digital signal processing and computer predictions are necessary.

Research Facilities

The spin rig facility (fig. 2) allows study of the dynamics of rotating blade sets in a vacuum. We use this facility (ref. 1) to study the effects of rotation on bladed stages in the absence of aerodynamic forces. The facility consists of a large evacuated tank. The lid of the tank supports the test chamber. All services to the test chamber pass through the lid. An air turbine spins 20 in. diameter bladed-disk assemblies at speeds of up to 18 000 rpm. The bladed-disk test section bearings mount against vertically compliant pads.

Shakers or air jets excite blade vibrations. The shakers work by pushing against a thrust yoke to load the top set of bearings against the compliant pads. Strain gauges measure blade vibrations at as many as 50 locations using a 100 channel freon cooled slip ring assembly. Also, noncontact photo-optical probes measure the vibration response of all the blades at once.

We evaluate blade damping mechanisms in our dynamics laboratory complex or as part of sponsored research (ref. 2) within similar industrial laboratories. Figure 3 diagrams a typical platform damper test. Figure 4 is a photograph of a platform damper test in our dynamics laboratory. Many blade damping tests use a massive support table and noncontact exciters. Otherwise, whole test fixtures mount to massive shaker systems and use noncontact sensors.

The transient rotor response laboratory (ref. 3) allows study of transient excitation and rotor instability limits. The test rotor-shaft system is a lumped mass equivalent of a typical small engine (figs. 5 and 6). An air turbine drives the rotor to 16 000 rpm. Independent squeeze film dampers act at the two bearing housings. Two squirrel cage centering springs control the effective support stiffness. Segmented seal rub strips placed around the disks simulate engine blade tip seals. The dynamic rubbing forces can cause rotor instability.

The rotating systems dynamics rig (ref. 4) allows study of coupled vibration between blades, shafts, and case support structures (figs. 7 and 8).

Shaft whirl mechanics and gyroscopic force studies contribute to better design methods for overhung fans. Dynamic bearing flexibility measurements and bearing alignment studies enable better rotor-shaft system analyses. Distributed rotor flexibility and damping measurements allow better system response predictions.

The rotating systems dynamics rig is an elastic shaft lumped mass equivalent of a typical engine mounted within a flexible engine case. An aluminum shaft with two large disks and an overhung slotted disk rotate as fast as 6000 rpm. Motions of the slotted disk couple with the shaft motions under proper conditions. Air jets can excite slotted disk vibration. Shakers at the bearing housings can simulate various engine operating conditions. With some changes, this rig also allows active vibration control studies. In figure 25 an actively controlled unbalanced steel shaft rotates as fast as 10 000 rpm.

Similar rigs measure other dynamic phenomena. Several rigs determine basic blade damping mechanisms. These include the platform damper studies above. Also, shroud slip, root friction, blade material, and centrifugal stiffening studies contribute to understanding blade response.

Shaft bushing seal and high speed rotating shaft seal rigs measure the forces between seal and shaft surfaces. Transient seal rub force rigs measure short duration transient rub forces between blades and tip seals. Shaft damper rigs evaluate several damper configurations. These include squeeze film, stacked squeeze film, and hydrostatic damper configurations. Also included are elastomeric, high load, thrust bearing, and magnetic dampers.

EXAMPLE MEASUREMENT METHODS

Blade Dynamics

Damping. - The small size and complex shape of jet engine blades complicate the measurement of blade damping mechanisms. Low damping levels and the presence of other damping sources add to the difficulty. Modal damping factor measures are also difficult. These difficulties determine the fixture design, excitation used, and the measurement method chosen.

Decay measurements, frequency response analysis or direct methods measure damping levels. Vibration decay response signatures result from suddenly stopping an initial excitation. Good modal damping estimates require initial sinusoidal excitation at a single resonant frequency. The envelope of the resulting decaying response predicts the logarithmic decrement loss factor. This measures net damping levels. A variation involves averaging the vibration response of a randomly excited blade over many cycles.

Decay response methods determine damping levels from the initial slope of the response envelope. It is often difficult to measure this slope. The damping may depend on other factors. Figure 9 illustrates these problems for shroud dampers. Here, the damping is a function of the initial stress level. Other problems include the difficulty of generating a transient response that contains a single modal frequency. Specially developed frequency domain techniques for transient data aid with these problems.

Frequency response test methods apply sinusoidal sweep or dwell at or near resonances. They measure damping at the half power point or as the inverse of the transmissibility at the resonance. These methods have not been as successful as the decay response or the direct measurement methods.

The direct method measures the input power required by an exciter to maintain a constant amplitude steady state vibration at resonance. The energy dissipated per cycle is equal to the average energy input per cycle. The ratio of this value with a computed maximum kinetic energy within a cycle determines the loss factor. Figure 10 diagrams typical instrumentation for a shaker excited test.

Blade platform friction dampers control the vibration stress in many blade designs. The device (ref. 5) in figure 11 assesses the importance of damper loading forces. This device rotates as fast as 12 000 rpm in the spin rig facility. Electromagnets vary the damper loading on each blade during a test. Shakers and air jets excite blade vibrations. Strain gauge data predicts blade stress levels. Figure 12 reveals optimal normal platform loadings for a given operating condition. These results confirm design methods for platform dampers.

<u>Bladed-disk assemblies</u>. - A digital system using photo-optical sensors measures bladed-disk vibration (refs. 6 and 7) in the spin rig. A set of 16 optical sensor assemblies mount to the spin rig test chamber (fig. 13). The probe assembly consists of three optical sensors arranged to sense three positions on the passing blade tips (fig. 14). Each sensor assembly (fig. 15) contains an optical sensor and a self contained light source (fig. 16). Each blade tip on a bladed-disk (fig. 17) reflects light when it rotates past the sensor.

Records of the time when reflected light pulses trigger each probe assembly provide the measurement. The system computes the time when each blade should pass each sensor if it were not vibrating. The system also measures the time that each blade actually does pass each sensor. The difference in these times is a measure of blade deflection at the time of the measurement. The data recorded as a blade passes each sensor assembly is a time history of the blade vibration. Simultaneous recordings of all blade deflections fully define the bladed-disk vibration response. A probe assembly or sample port consists of three optical sensors. Figure 18 diagrams the measurement principle. The relative difference between the blade passage at each of the three locations allows discrimination between bending and twisting blade vibrations.

Since each probe assembly can sense every blade passage, this system is equivalent to having three displacement gauges mounted on every blade. The direct formation of blade data in digital format allows immediate sorting, storing, and further processing.

We wanted to be able to sense as many as 64 blades spinning as fast as 18 000 rpm. This allows an average of 52 μs between each blade passage (table I). The reflected light pulse from each blade passage latches the current value of a high speed digital counter. This angle clock counter always counts to the same high value during each revolution of the test article. The angle clock principle varies the clock rate to match the rotor speed.

The angle counter operates at rates of up to 24 MHz with emitter coupled logic to meet our resolution requirements. Each sensor latches the counter to a separate Z-80 microcomputer data acquisition module (DAM) board. The system shown in figure 13 has 16 probe assemblies, 48 sensor assemblies, and has 48 independent DAMs. Each DAM shares a common angle clock system and common control minicomputer. Each microprocessor stores blade vibration data in its 4096 words of memory. This memory fills as each blade passes in sequence. The control computer sorts the data from each DAM to assemble the vibration histories of the blades.

The DAM associated with each probe links to adjacent DAMs in daisy chain fashion (fig. 19). This allows many alternative modes of operation (table I). For example, the disabling of selected sensors and sharing of the associated DAM memories allows extended sample length records. Each sample port can ignore the passage of selected blade sets. Many variations exist by mixing these two modes. For example, two probes measuring the response of a single blade shares the blade sample memory of all DAMs. Frequency resolution greatly increases at the expense of possibly aliased and less than fully defined data.

The least significant eight bits from the angle clock counter measure each blade passage. A blade detect signal latches the count from a back plane bus to the associated DAM. Subtraction of a normalizing constant provides the value stored in the DAM memory. This value represents the absolute deflection of the blade when it passed the sensor. The normalizing constant is unique for each blade and sensor combination. The normalizing constant corrects for nominal misalignments of the probes and the blades. Initial spin testing with minimal vibration generates the set of normalizing constants.

The control computer assembles vibration histories by sorting the data from independent DAM boards according to the sampling method chosen. Successful sorting requires that all data be in proper sequence in the individual DAM memories. Any data lost during start up or operation prevents this sorting. Complex arming and window logic maintains data integrity.

During system start up, the first DAM looks for blade number one after a specified arming delay. A one per revolution timing signal synchronizes the initial arming delay. The first DAM board signals the second DAM board to begin an arming delay count when a blade detection occurs. The second DAM remains disabled until the arming delay completes. Then, passage of blade number one signals the third DAM to begin its counter. This process repeats until each DAM senses blade number one for the first time. Once armed, each DAM senses every blade that passes. DAMs continue taking data for a prescribed set of blade passages. Some extended record length operations disable the storage of blade passage data within a DAM. The slaved memory of the disabled DAM fills with data taken from the previous DAM in the daisy chain.

Sample window hardware logic on each DAM board only allows blade detection during specified window periods. The logic generates an artificial blade detect signal if a blade detection does not occur during the window period. Also, the window logic does not allow more than one blade detect during a window period. Each blade detection synchronizes the data window. The window logic insures that the proper number of data samples exist in the proper order in the DAM memories.

The system has a resolution of eight ten thousandths of an inch (0.0008) with an 18 000 rpm 20 in. diameter rotor. With 32 sample ports, 393 216 data samples occur in 70 msec. Digital sample point records store the simultaneous vibration response of up to 64 blades. Records of at least 2048 points define the vibrations of each blade tip at each of three separate locations. This would otherwise require 192 strain gauges.

Figure 20 shows an expected modal response of the test rotor shown in figure 17. The results agree with strain gauge data. Figure 21 reveals a new technique (ref. 8) for examining the modal response. Each blade has a different vibration amplitude. This results in a complex modal pattern. Small differences between blades can greatly affect system response. Complex aerodynamic modes may couple with these structural modes and cause severe vibrations.

Rotor-Shaft Systems

Vibration control. - Shaft damper research focuses on controlling large unbalance due to severe service conditions. Lumped mass elastic shaft systems operate at critical speeds with prescribed unbalance. The measurements are simple and direct. Two load cells, installed at right angles, measure forces at each damper housing. Likewise, two eddy current proximity probes sense the displacement of each damper housing. The phase between the radial and tangential forces (with respect to radial motions) determines the stiffness and damping characteristics. The precession rate of elliptical orbits determines cross coupled stiffness and damping terms as well. The four resulting factors define the performance characteristics of particular damper designs.

Rotor balancing in real time increases the performance of rotor-shaft systems in an economic way. Shaft motion measurements permit balancing by removal of rotor material during rotation. Balancing occurs at a rotation speed less than that of unbalance testing. A high power laser with a moving lens system removes material from a selected surface of the rotor (fig. 22). Balancing operations do not break the vacuum in the test chamber nor do they stop the rotation. Electronic servo systems track the unbalance, move the lens systems and fire the laser. Conventional eddy current probes measure the shaft motions before and after the operation.

Transient rotor rub dynamics. - Blade tip seal rubs can cause shaft instability. The transient rotor response rig (figs. 5 and 6) confirms predictions of instability modes and limits. The rig (ref. 9) creates a short duration transient rotor instability. Ten eddy current proximity probes measure shaft motions. Four eddy current proximity probes also measure damper housing motions. Sets of two probes measure horizontal and vertical motions in a plane. The 14 probes take data at each of seven planes along the shaft. Two load cells measure the horizontal and vertical forces at each of the dampers as well.

A transient unbalance excites shaft vibrations. Blade tip seal rubs cause the instability. The initial unbalance starts with a solenoid impacting and removing a balance weight. The balance weight easily breaks off from a specially machined screw. The rig is re-balanced during testing the same way.

The test starts with the suddenly unbalanced shaft. Shaft vibration grows until the disks begin to rub the outer seal segments. With the right conditions, an unstable backward whirl occurs after a few shaft rotations. The control computer senses this by analyzing many channels of transient data. Within a revolution, solenoids fire to re-balance the shaft. Simultaneously, the seal segments move away from the disks.

A digital recorder saves 14 channels of shaft motion and four channels of force measurement taken during test. Digital sample rates near 2000 Hz form 4096 fourteen bit sample point records. A minicomputer uses some of this data to control the test. Shaft run out data from a low speed test corrects the high speed test data. Plots of the corrected data in figure 24 show the shaft motions for eight revolutions during a seal rub event. Comparisons between these motions and those predicted allow corrections of the analysis. Differences are often due to inexact predictions of the transient rub forces.

The modeling of transient rub forces is difficult. Many factors affect these forces. Rub force mechanisms vary with particular seal materials, operating temperatures, blade properties, and rub speeds. Figure 23 shows a turbine blade tip seal rub force experiment. In this experiment, a rigid carriage supports a sample turbine blade tip seal segment. The seal segment slowly moves into a simulated rotating blade tip.

Transient force and acceleration reaction measurements allow computation of the transient rub forces. The measured mass of the carriage, the acceleration, and the transient force permits this computations. Reliable predictions of the carriage dynamics allow detailed rub force model development. Bi-axial reaction force and acceleration measurements determine radial and tangential rub forces. These rub models update the shaft dynamic instability predictions.

Coupled Systems

Rotating systems dynamics. - The rotating systems dynamics rig confirms coupled system response predictions. The rig permits study of coupled bladed-disk, shaft, and flexible support vibration (figs. 7 and 8). Proximity probes measure shaft motions. Accelerometers measure support structure vibrations. The measurements are simple and direct.

Two eddy current proximity probes measure shaft motions within a plane. Four probes measure motions in two planes. Four additional eddy current probes measure planer bearing motions. Four accelerometers measure bearing housing response. An additional 18 accelerometers measure the response of the support structure. The four shakers apply planer forcing at both bearings. Load cells measure shaker forces. Patch panels distribute measured data. A 14 channel tape recorder, an array of eight four channel oscilloscopes and control computers record results. Digital transient recorders capture transient phenomena.

The shakers can simulate various engine operating conditions. Also, they can control vibration. The four shakers in figure 25 actively cancel the vibration (ref. 10) of an unbalanced shaft. The shakers act to cancel unbalance forces and velocities at the bearing housings. In theory, the motionless housings cannot dissipate energy. Knowledge of the shaft vibration modes allow velocity feedback control with sparse measurements. The known modal shaft response predicts unmeasured variables from sparse data. This allows optimal active feedback control (fig. 26). A large vibration reduction at a critical speed (fig. 27) uses only velocity feedback control.

Random decrement signatures. - The random decrement vibration signature (ref. 11) reveals net damping levels in vibrating systems. It represents the equivalent linear system response of otherwise nonlinear systems. It has application to system identification methods and to real time failure detection. A high speed digital implementation of the random decrement algorithm (ref. 12) allows real time signature development.

Randomly excited systems with ergodic response can produce random decrement signatures. The signature sums each portion of signal that follows a specified threshold level. In figure 28, record 1 crosses the threshold and starts at time t1. Record 2 starts at time t2. These records sum together and form the initial estimate of the signature. The origin of each record shifts to zero for summing. The third record starts at time t3, when the signal crosses the chosen threshold again. The random decrement signature results form the sum of many such records.

The signature looks like a damped system response to a step input. Figure 29 shows why this occurs. Each summed record is a superposition of linear system responses. These include step responses, impulse responses, random responses, and random noise. All responses but the step response sum to zero because there is a threshold level. If a threshold velocity were chosen instead of a threshold level, the impulse response would remain. The decay of the equivalent step response provides the damping measurement.

A fixed number of sample points define the digital random decrement signature. Each sample can belong to many different records. Each samples contribution to the resulting signature sums to several positions at once. In figure 30, sample 14 is sample 13 of record 1. It is also sample 7 of record 2, and so on. Therefore, this sample sums to position 13, 7, 3, and 1 of the signature. The process discards the sample after it contributes to forming the signature.

Figure 31 shows an algorithm for doing this in real time. A computer built from random logic performs this algorithm. It develops 1024 point random decrement signatures in real time. A threshold crossing rate of 5000 Hz is possible. The algorithm uses wrap around counters to eliminate time spent on record length comparisons. The computer also uses a stable averaging procedure not shown. Stable averaging allows monitoring of the signature development in real time.

SUMMARY AND CONCLUSIONS

State of the Art

The increased flexibility of propulsion systems requires better analysis and experimental methods. For example, the analysis of short duration seal rubs, rotor whirl, and coupled system dynamic response needs development. Small perturbations in system properties can greatly affect energy transport during an instability. Precise measures of these perturbations is difficult. Restricted physical access results in the sparse data problem. Vibration control strategies require better mathematical models of the controlled system.

Digital Systems

Digital signal processing enhances the quality of many forms of measured data. With knowledge of the process being measured, digital processing often extracts more information than otherwise possible. This is important for complex systems where various restrictions limit the possible measurement points.

Some complex system analyses require experimentally based computer models. Methods for broader and more timely use of costly experimental data need development. Digital systems allow coupling of computer predictions with measured results in real time for controlling experiments. Digital signal processing and predictive digital filtering methods reduce the impact of sparse data. Digital techniques such as comb filtering can enhance either synchronous or nonsynchronous vibration data. Adaptive active vibration control of coupled dynamic systems needs development.

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TABLE I. - PHOTO OPTICAL SYSTEM CHARACTERISTICS FOR DIFFERENT EXPERIMENT SETUPS

Experiment[a] setup	32 Ports 64 blades 1885 rad/sec	16 Ports 64 blades 1885 rad/sec	32 Ports 32 blades 1885 rad/sec	32 Ports 64 blades 942.5 rad/sec	16 Ports 32 blades 1885 rad/sec
Port sample time, μsec	52	52	104	104	104
Port sample frequency, Hz	19 200	19 200	9600	9600	9600
Blade sample frequency, Hz	9600	4800	9600	4800	4800
Memory/blade	2048	2048	4096	2048	4096
Time to fill memory, sec	0.213	0.426	0.426	0.426	0.852
Frequency resolution, Hz	4.69	2.35	2.35	2.35	1.17
Maximum frequency, Hz	4800	2400	4800	2400	2400

[a]All conditions are taken for a system using 32 ports and the specification rotor. Lesser ports imply skipping ports while lesser blades imply skipping blades. Smaller number of ports mean additional memory available per port.

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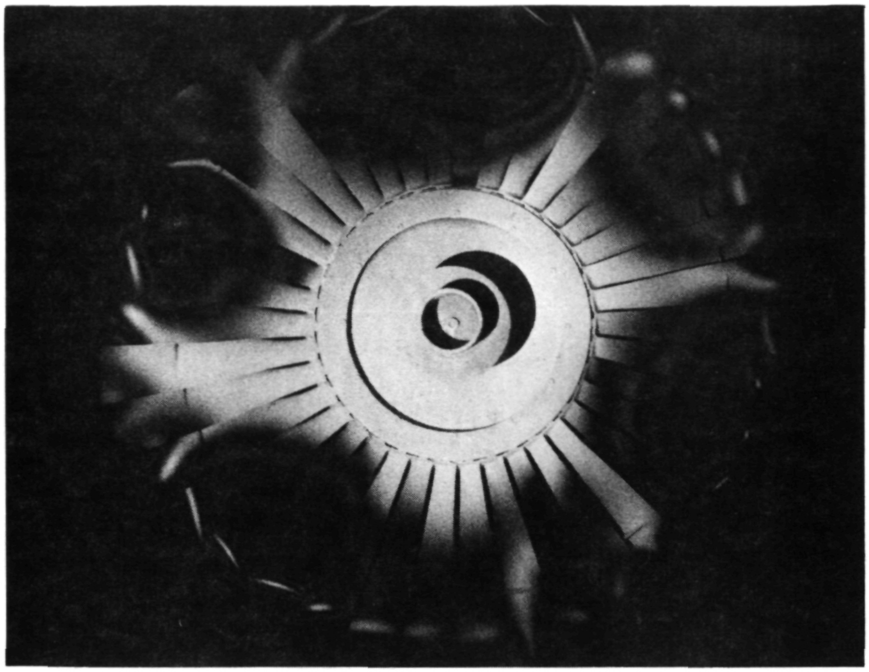

FIGURE 1. - TYPICAL BLADED-DISK VIBRATION MODE.

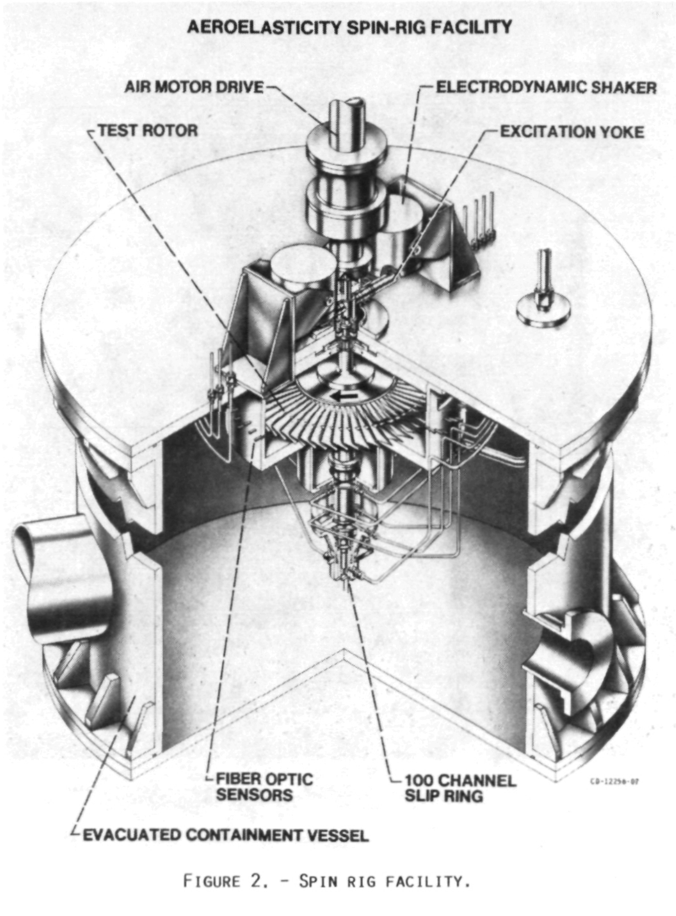

FIGURE 2. - SPIN RIG FACILITY.

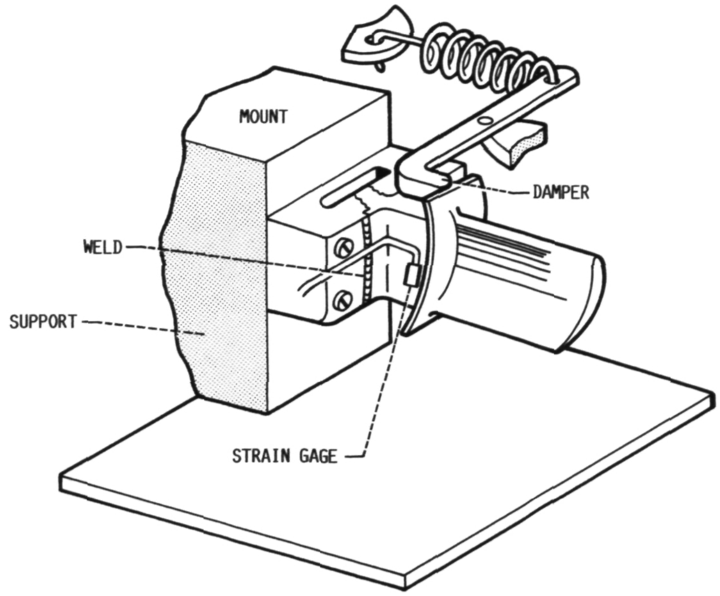

FIGURE 3. - BLADE PLATFORM DAMPER.

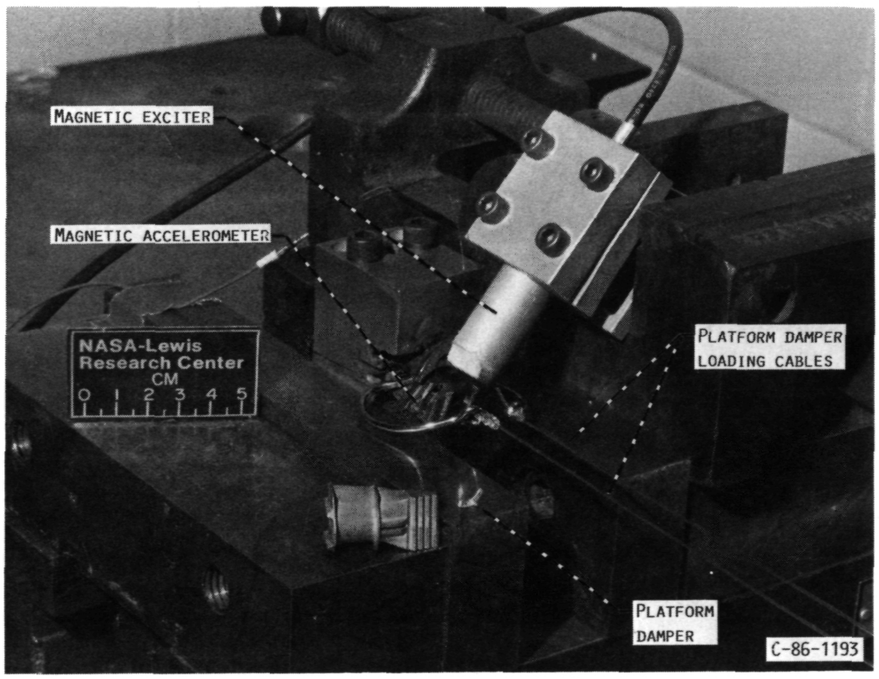

FIGURE 4. - PLATFORM DAMPER TEST.

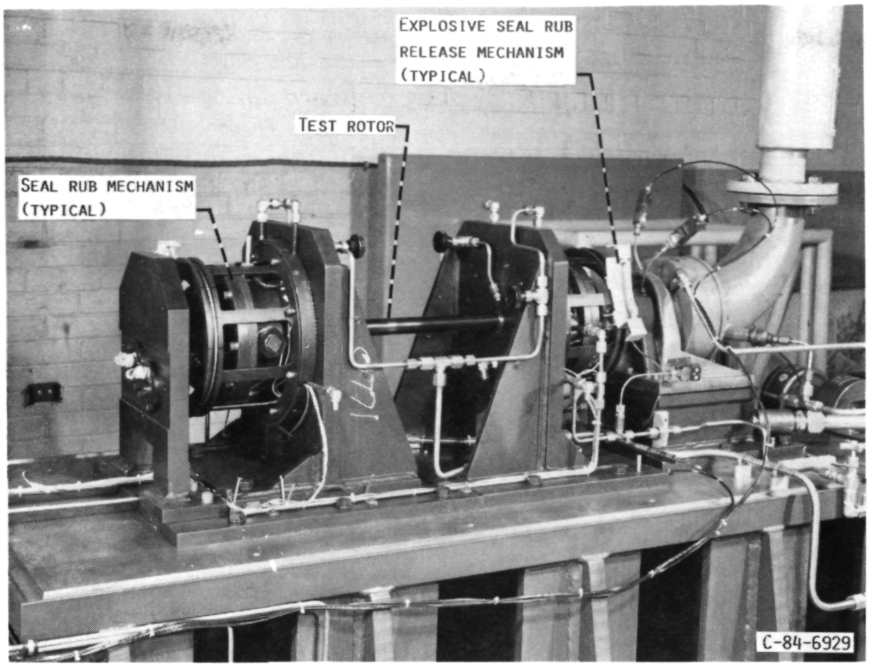

FIGURE 5. - TRANSIENT ROTOR RESPONSE LAB.

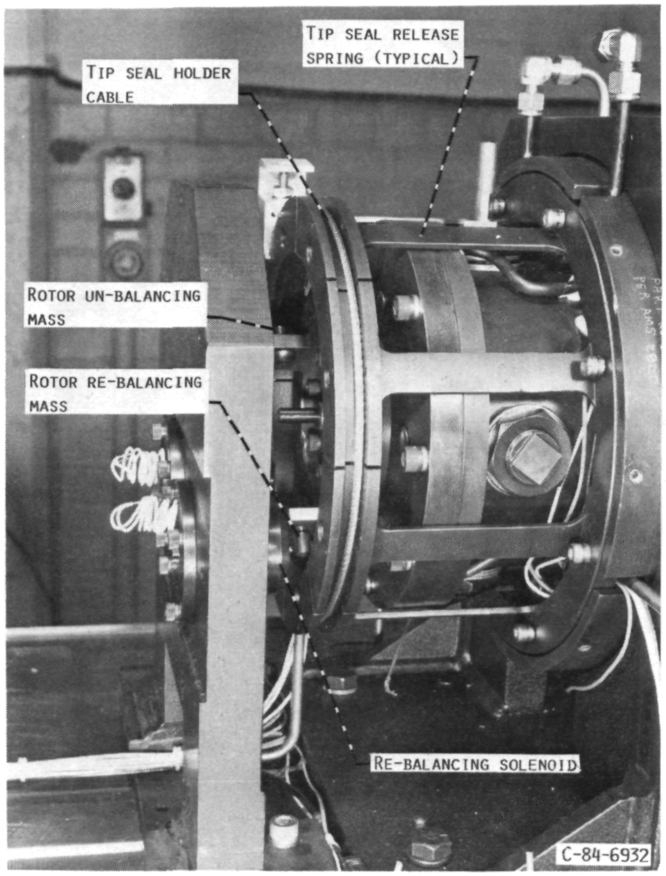

FIGURE 6. - TRANSIENT ROTOR SEAL RUB MECHANISM.

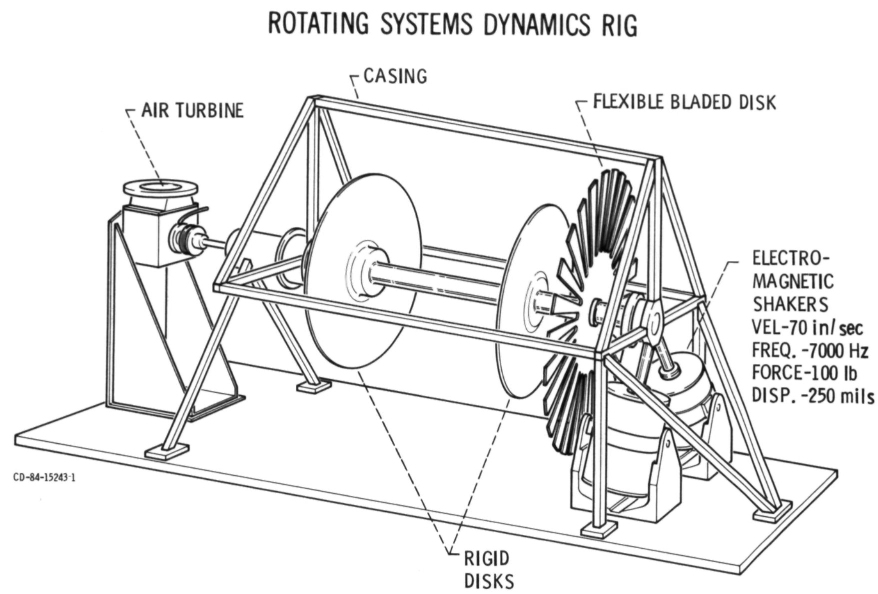

FIGURE 7.- ROTATING SYSTEMS DYNAMICS RIG.

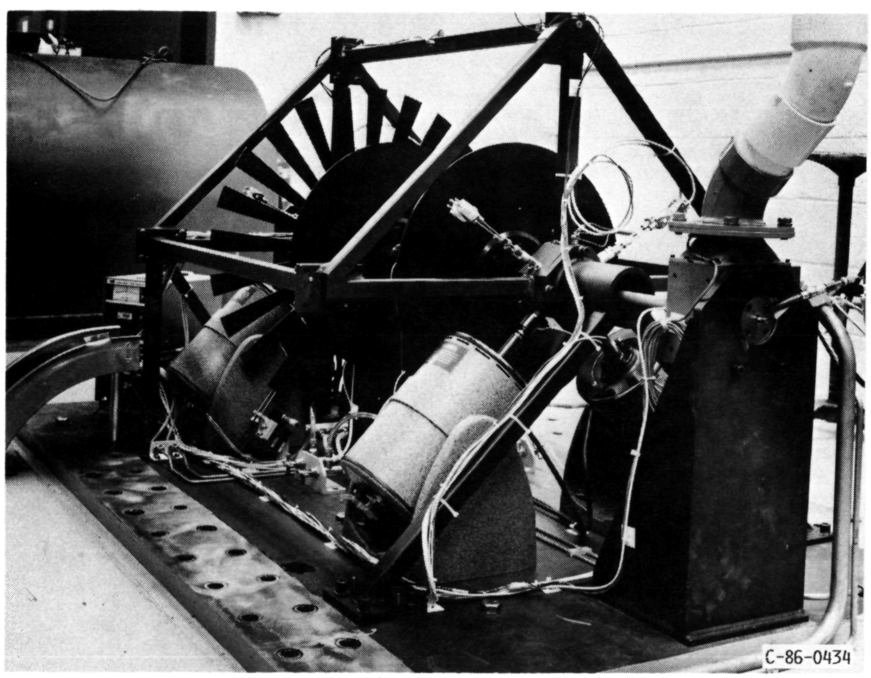

FIGURE 8. - ROTATING SYSTEMS DYNAMICS RIG.

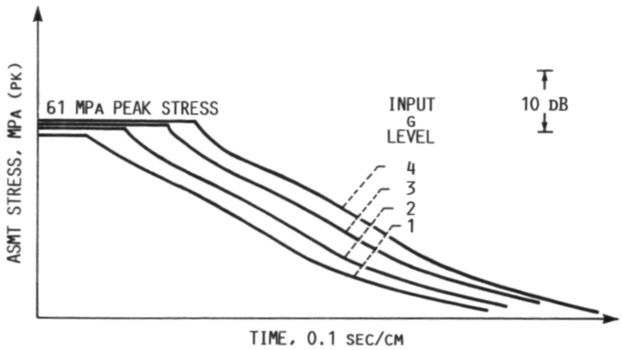

FIGURE 9. - DAMPING DUE TO RUBBING AT SHROUD FOR A TYPICAL FAN BLADE - VARIATION OF RESPONSE DECAY WITH INITIAL STRESS. NORMAL SHROUD LOAD: 150 N. MODE: FIRST FLAP. FREQUENCY: 279 Hz.

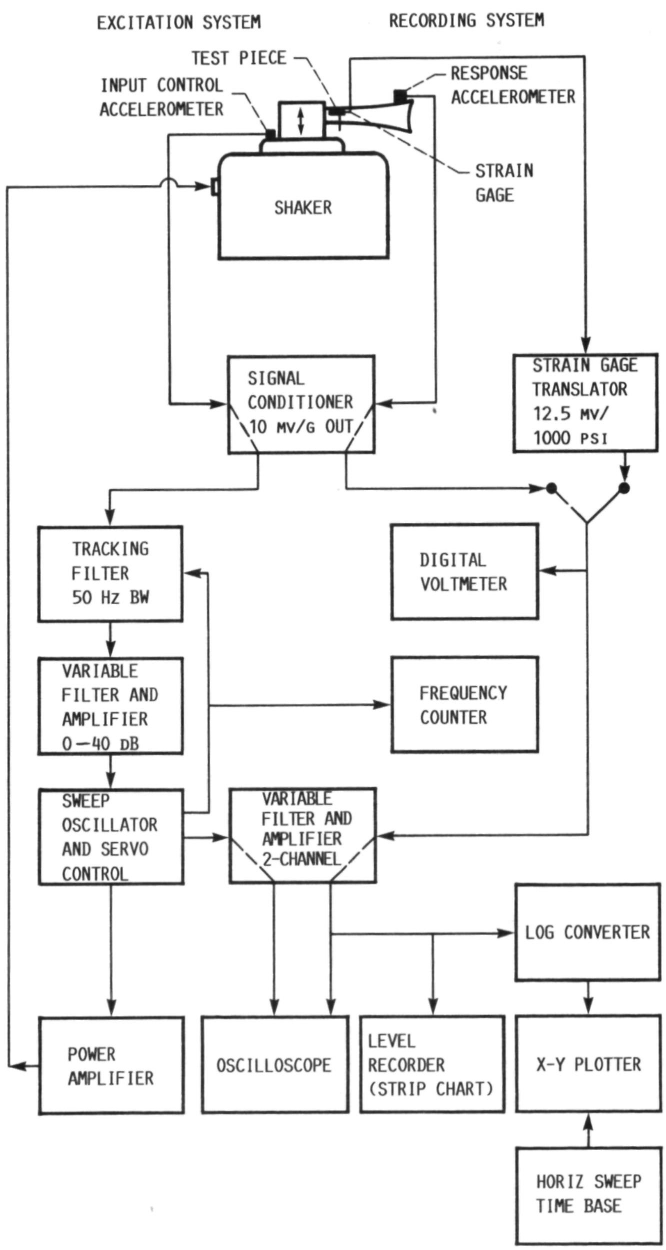

FIGURE 10. - BLOCK DIAGRAM OF BASIC EXCITATION AND RECORDING SYSTEMS USED TO OBTAIN STRAIN AND ACCELERATION RESPONSE DATA FROM TEST ITEMS EXCITED BY THE ELECTRO-DYNAMIC SHAKER.

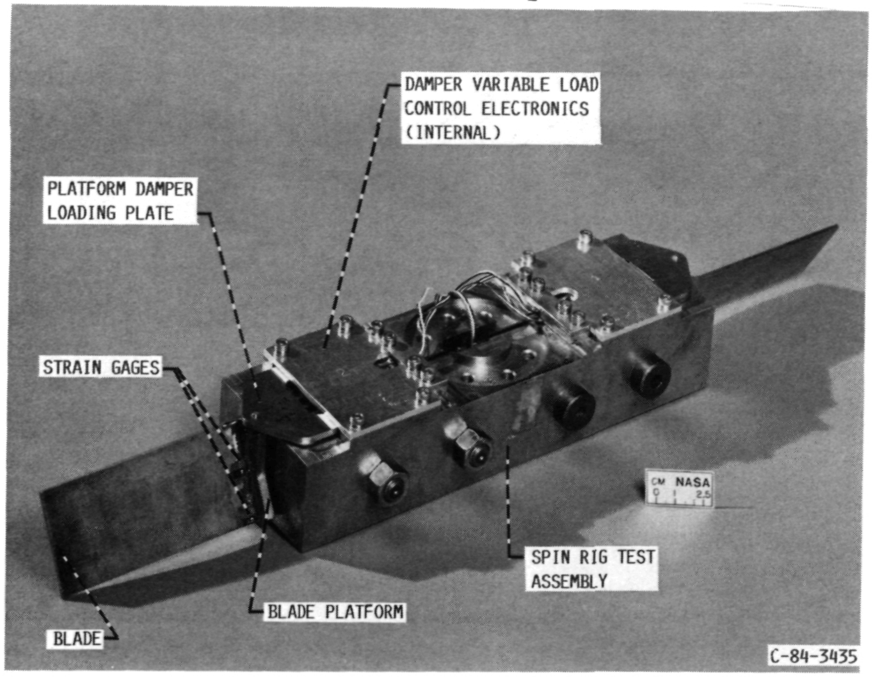

FIGURE 11. - ROTATING VARIABLE PLATFORM DAMPER LOAD EXPERIMENT.

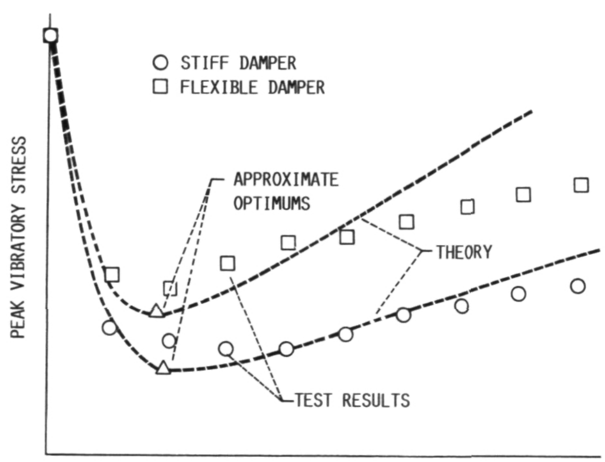

FIGURE 12. - STRESS REDUCTION VERSUS DAMPER LOAD.

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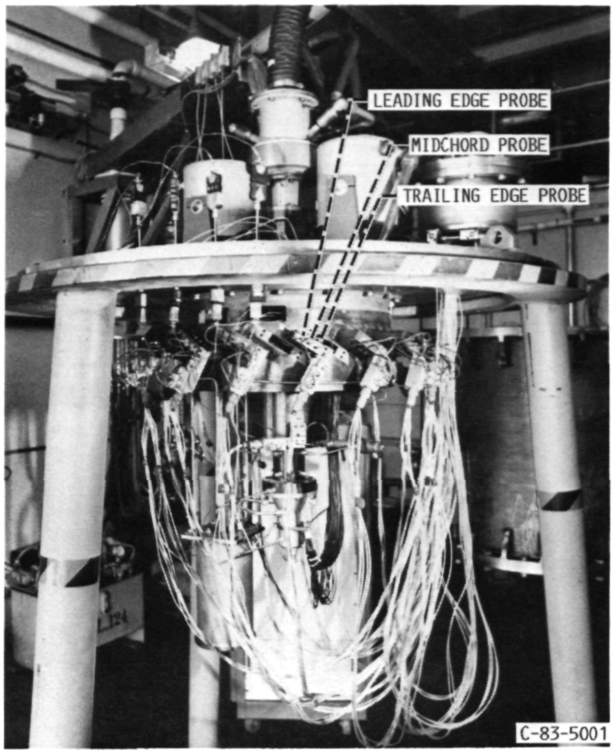


FIGURE 13. - OPTICAL DATA AQUISTION SYSTEM.

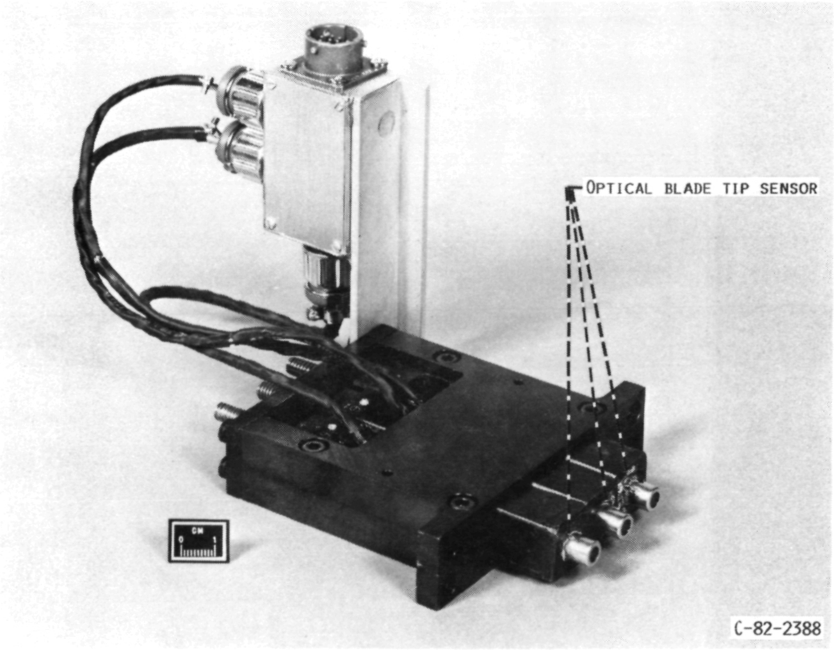


FIGURE 14. -OPTICAL PROBE ASSEMBLY.

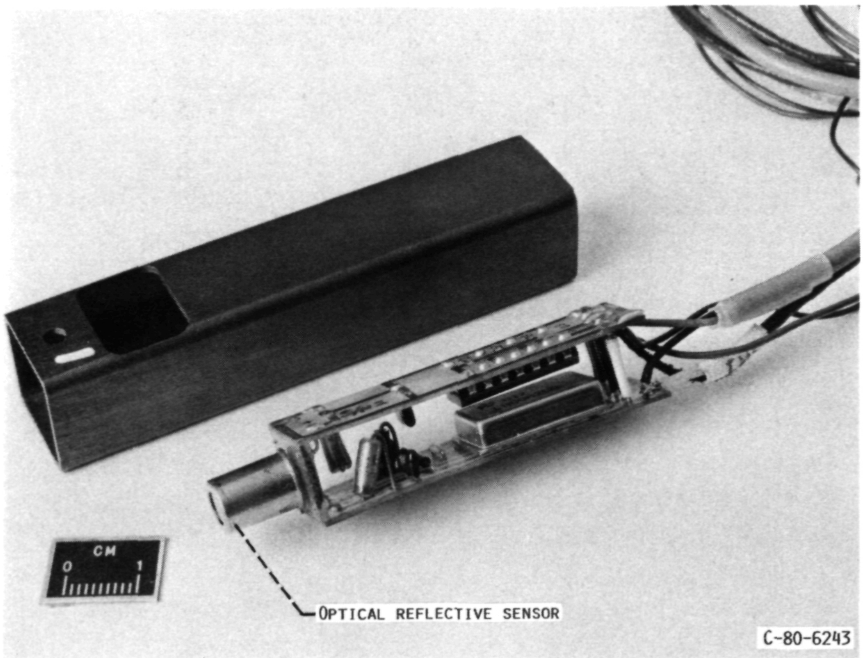


FIGURE 15. - OPTICAL BLADE TIP SENSOR.

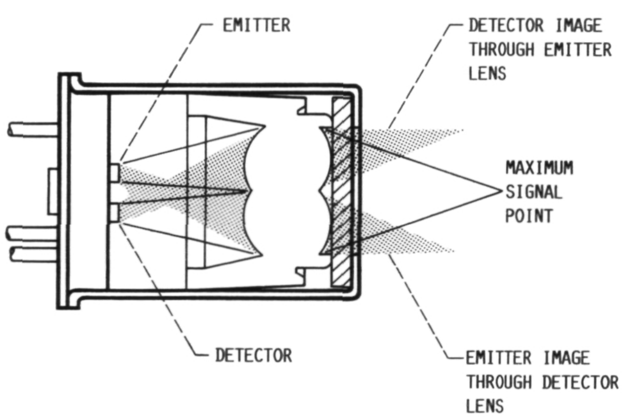


FIGURE 16. - OPTICAL REFLECTIVE SENSOR.

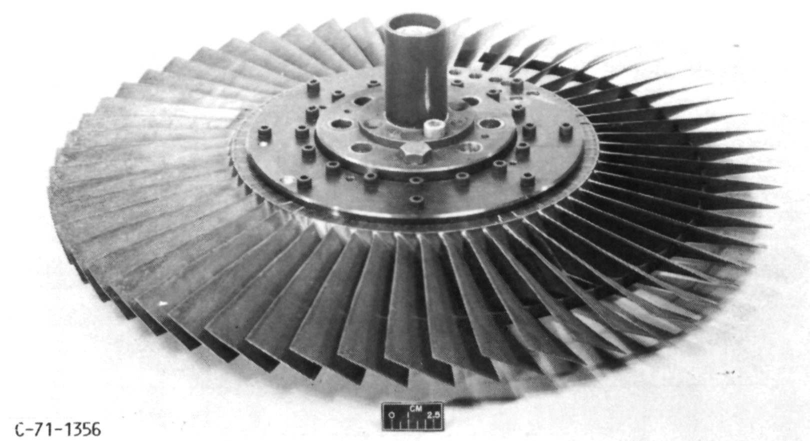


FIGURE 17. - 56 BLADE ROTOR.

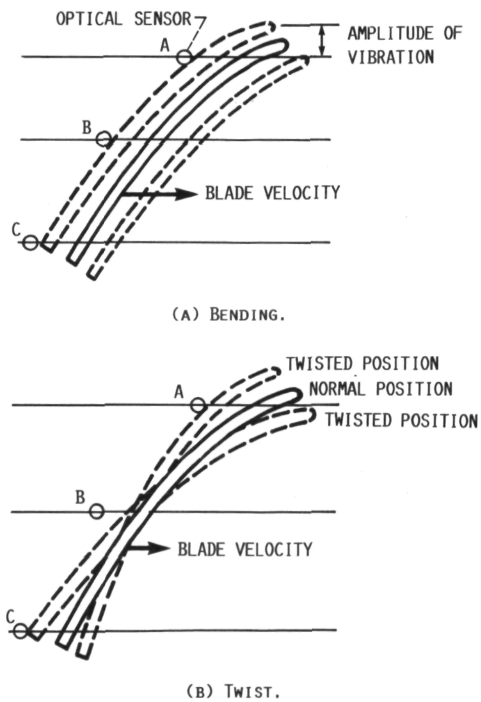


FIGURE 18.- BLADE MOTIONS OBSERVED WITH 3 PROBES LOOKING AT THE BLADE TIP (RADIALLY INWARD).

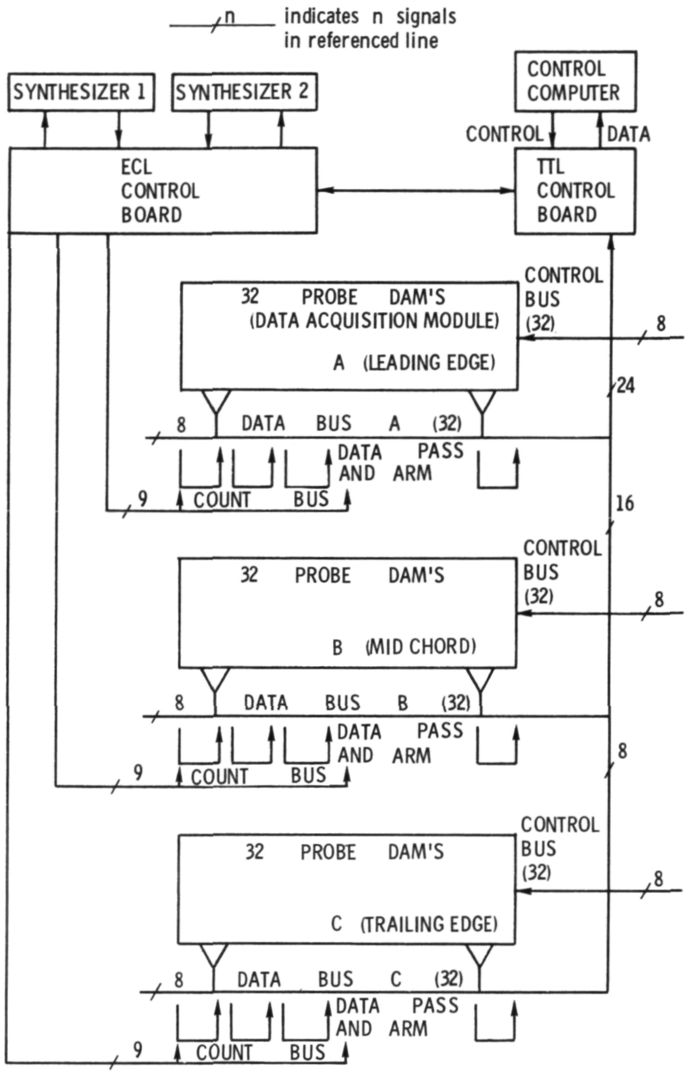


FIGURE 19.- PHOTO-OPTICAL SYSTEM BLOCK DIAGRAM.

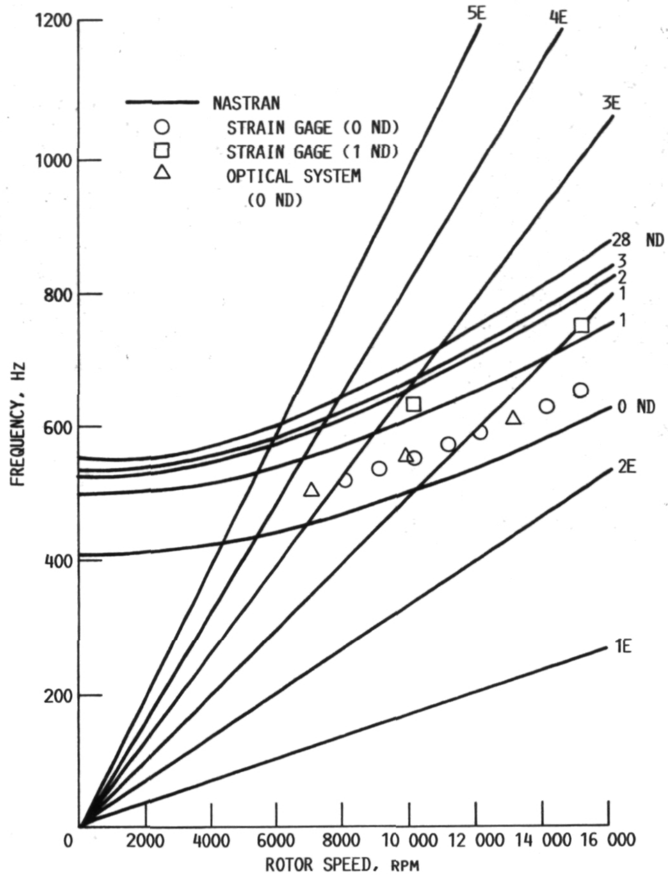


FIGURE 20. - ROTOR CAMPBELL DIAGRAM FOR FIRST BENDING FAMILY OF MODES.

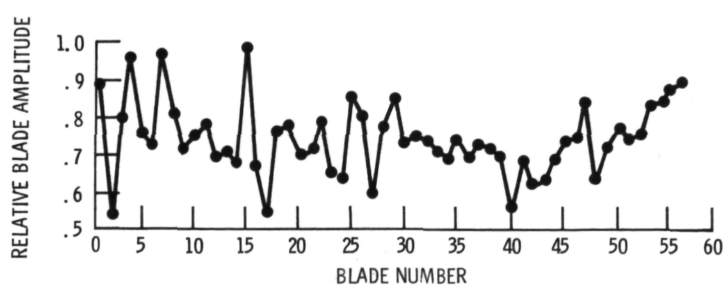


FIGURE 21.- 0 NODAL DIAMETER MODE SHAPE AT 551 Hz (10 000 RPM).

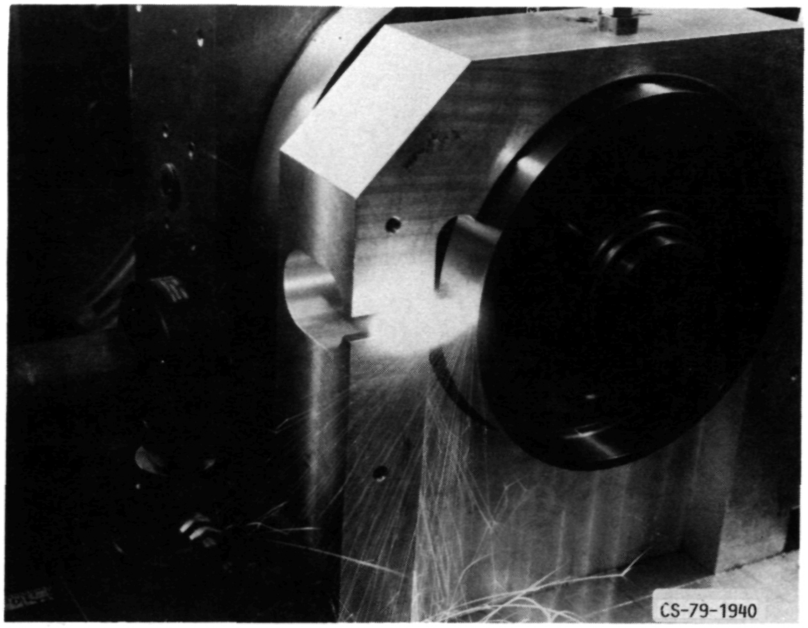


FIGURE 22. - LASER REMOVING METAL FROM TEST ROTOR.

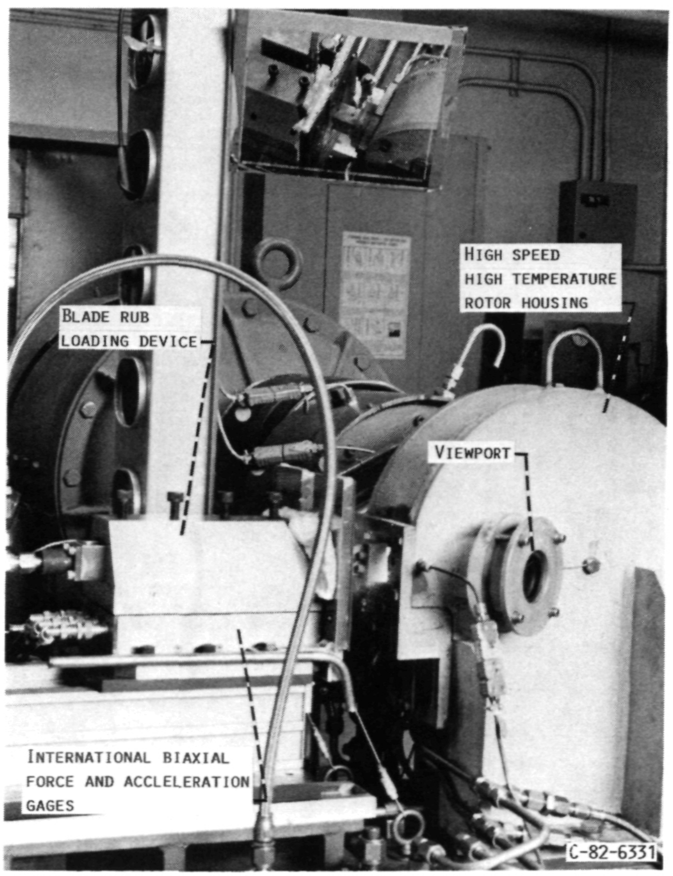
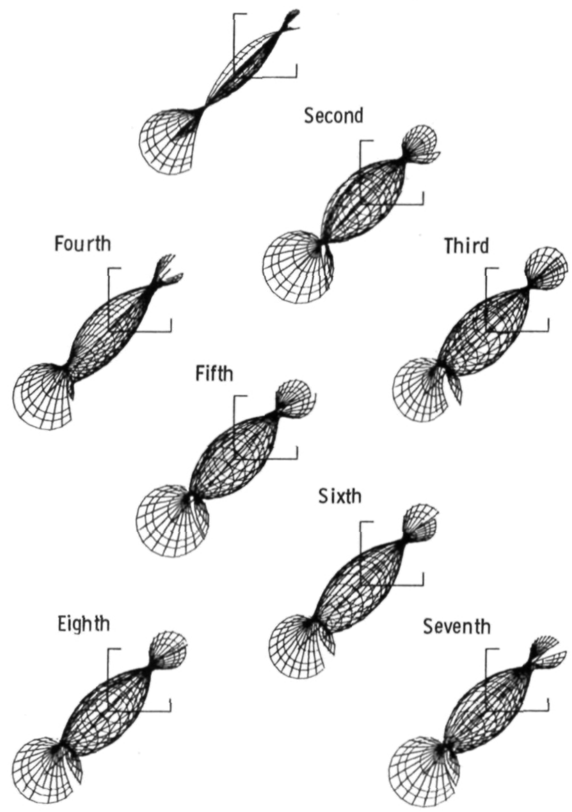


FIGURE 23. - HIGH TEMPERATURE TRANSIENT RUB FORCE MEASUREMENT.



$\nu = 7.5\times10^{-6}$ LB SEC/IN.^{2}

FIGURE 24. - ROTOR RUB RESPONSE - ENVELOPE OF CENTER LINE MOTION.

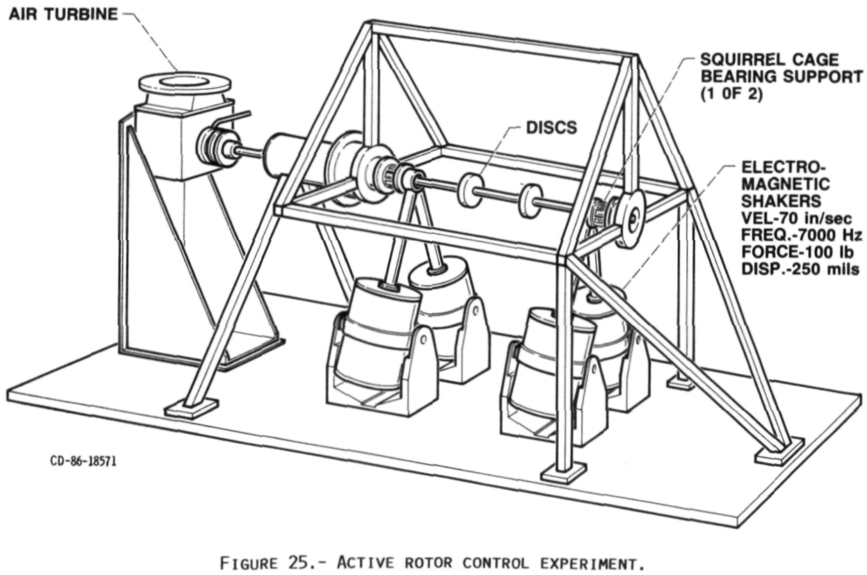


FIGURE 25.- ACTIVE ROTOR CONTROL EXPERIMENT.

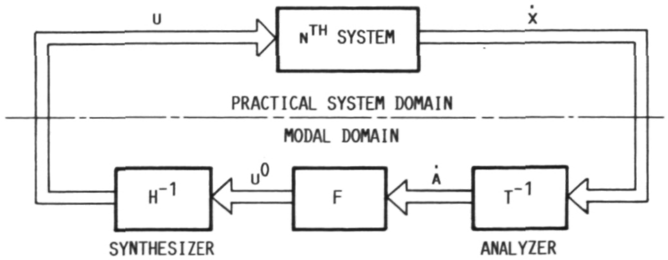


Figure 26. - Modal control system with velocity feedback.

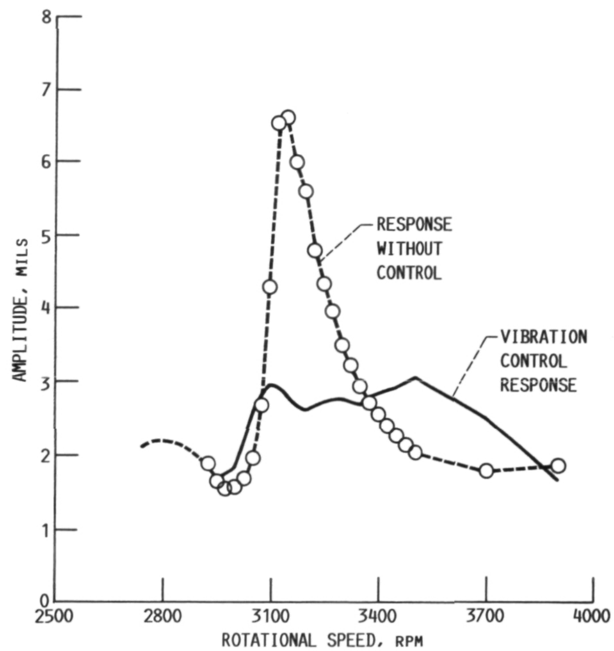


Figure 27. - Steady state unbalance response using optimal regulator active bearing control.

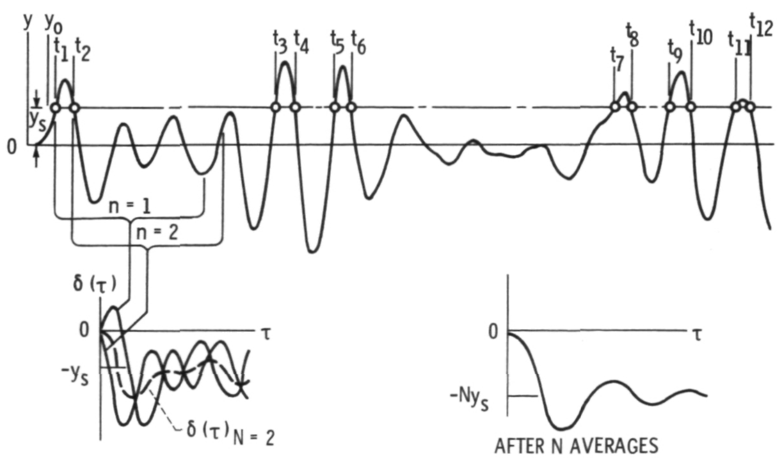


Figure 28.- Random decrement process.

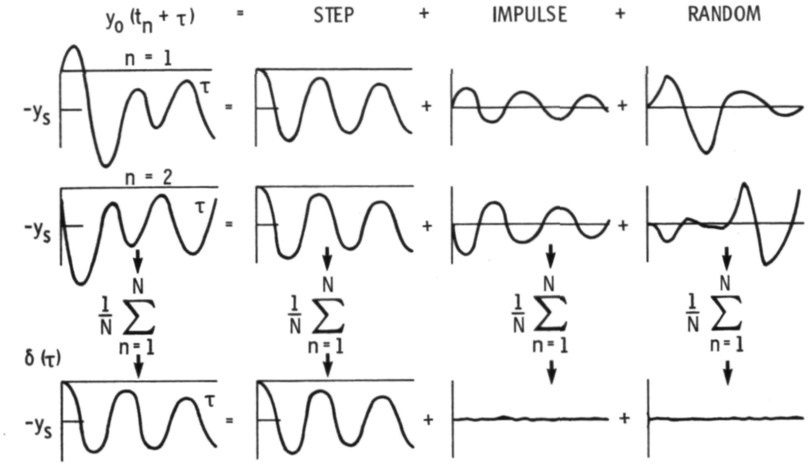


FIGURE 29.- REMOVAL OF IMPULSE AND RANDOM COMPONENTS WITH AVERAGING.

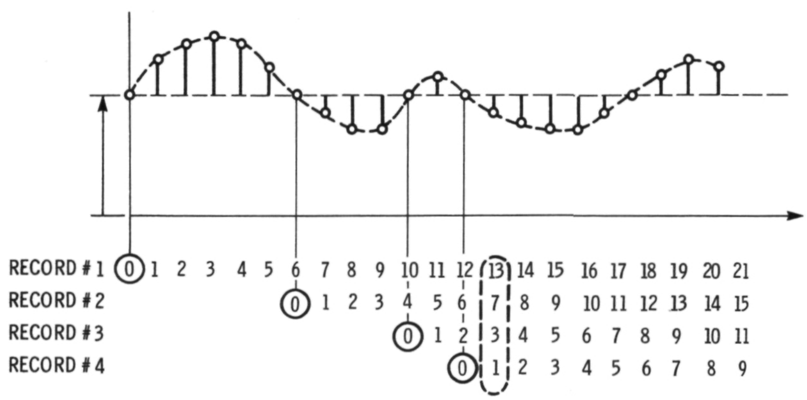


FIGURE 30.- RANDOM DECREMENT WITH DIGITIZED DATA.

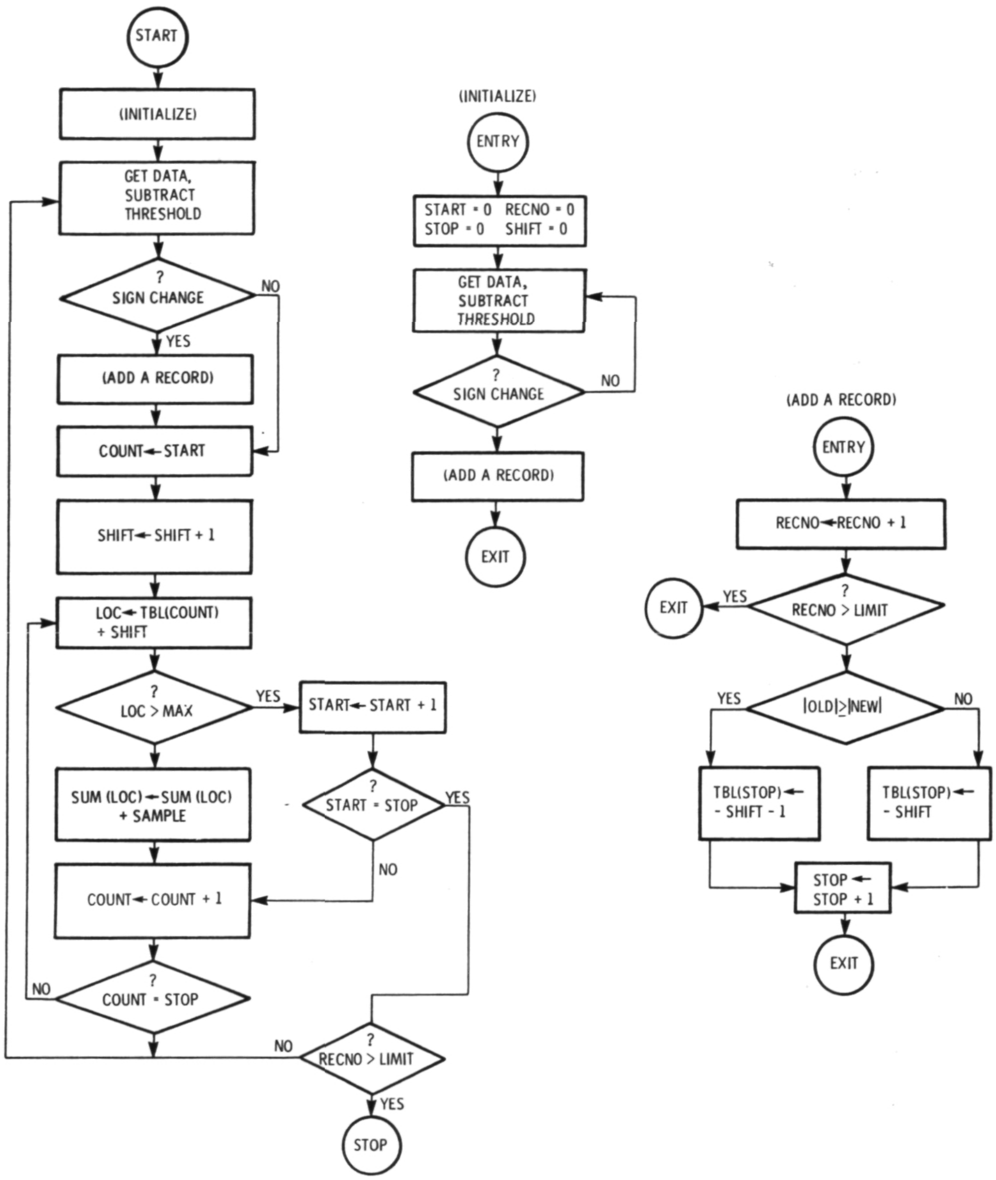


FIGURE 31.- DIGITAL RANDOM DECREMENT ALGORITHM.

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16. Abstract

Methods developed for measuring blade and rotor-shaft system response include optical systems, transient instruments, and special digital data processing equipment. Optical methods offer some distinct benefits for blade vibration measurement. Transient and steady state measurements of the response of rotor-shaft systems strongly effect our analytical methods development. Digital computing systems allow processing of large volumes of high speed data from rotating blade sets. Also, digital systems develop useful vibration response signatures from randomly excited systems. Research facilities include the spin rig facility and the transient rotor response lab.

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