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Development of a Noninterference Technique for Measurement of Turbine Engine Compressor Blade Stress

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ARO, Inc.

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ARNOLD ENGINEERING DEVELOPMENT CENTER
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APPROVAL STATEMENT

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A noninterference technique for measuring stress in compressor blades of turbine engines is being developed to alleviate disadvantages associated with conventional strain-gage measurement systems. This technique utilizes blade tip deflection measurements and special data processing algorithms to infer local blade stress. A prototype noninterference processing system for inferring blade stress from a single compressor stage, from blade vibrations

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20. ABSTRACT (Continued)

nonintegral to engine speed has been developed. Blade stress amplitude and spectral information are displayed on conventional strain-gage-type displays with which the blade stress analyst is intimately familiar. The prototype system has been checked by comparison with existing recorded strain-gage test data sampled to simulate blade tip deflection measurements.

PREFACE

The work reported herein was conducted by the Arnold Engineering Development Center (AEDC), Air Force Systems Command (AFSC), and E. L. Hively was the Air Force project manager. The results were obtained by ARO, Inc., AEDC Division (a Sverdrup Corporation Company), operating contractor for the AEDC, AFSC, Arnold Air Force Station, Tennessee, under ARO Project Nos. R35P-B3A, R35I-04A, R32I-00A, E32I-P2A, and E32L-02A. The manuscript was submitted for publication on September 25, 1979.

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1.0 INTRODUCTION

1.1 BACKGROUND

Compressor rotor blades in turbojet engines are subjected to static and dynamic stresses caused by centrifugal and aerodynamic forces, fixed-vane wake excitations, aeroelastic instabilities, airflow disturbances, and engine vibrations. Resulting stresses imposed on the blades adversely affect blade life. Stress caused by static components can be analyzed mathematically, whereas the dynamic forces acting on the blades are less predictable. Therefore, experimental determination of the dynamic vibratory stress component is required.

Dynamic stresses on compressor blades during engine operation are measured exclusively with strain-gage systems. The strain-gage system is the only field proved system for satisfying the performance requirements for these measurements. However, there are significant problems associated with the installation, operating life expectancy, and cost of strain-gage systems.

Installing strain gages and connecting leads requires disassembly of the engine compressor stages for access to the blades. Placement of the gages and connecting leads is a tedious task, and total system installation time for a few blades per stage may take up to several months. Slipring or telemetry access to strain gages on the rotor blades of inboard compressors of multicompressor engines is also troublesome and expensive.

The operating life expectancy of the strain-gage system is limited, with failures of the connecting leads at the blade root/disc interface and at the strain gages being the major failure modes. Also, the higher temperatures in present compressors require the use of welded strain gages on many stages, and the weld sometimes modifies the physical properties of the blade sufficiently that crack initiation and airfoil failure result.

The cost problem associated with strain-gage systems relates to the manpower/material resources and also the impact of the time delay between field definition of an engine compressor blade stress problem and having an instrumented engine ready for testing to resolve the problem. This delay time is in the order of months, whereas, in the interim, aircraft using a problem engine are either grounded, fly under restricted operations or, if used in a normal mode, may be vulnerable to failure.

The noninterference blade stress measurement system is based on a technique which requires the measurement of blade tip deflections and special data processing algorithms to infer local blade stresses. The blade tip deflection sensors are mounted in the engine case on the periphery of the compressor stage. Thus, the problems associated with strain-gage system installation and short operating life expectancy are alleviated. Also, the long delay time between problem definition and engine "ready for test" is significantly reduced.

This report presents a brief review of conventional strain-gage measurement systems and describes the noninterference blade stress measurement technique in perspective with the widely known strain-gage system. Also, a single-stage noninterference prototype data processing system is described which has been developed for blade vibration stress measurements nonintegral to engine speed. Recent online evaluation tests of the prototype system were successful, and the results will be published in a separate report.

1.2 OBJECTIVE

The ultimate objective is to develop a viable alternate to the strain-gage system for measuring engine compressor blade dynamic stress. The objective to date was to develop an algorithm for processing data from the noninterference compressor blade stress measurement system for nonintegral blade vibrations and assemble a single-stage prototype processing system for field evaluation of the noninterference technique. Further, the display of blade stress information for the noninterference technique was to be identical in data format and content as for the conventional strain-gage system with which the blade stress analyst is intimately familiar.

1.3 APPROACH

The approach to developing the noninterference compressor blade stress measurement technique is as follows:

1. Develop the data processing algorithms required to infer local blade stress for nonintegral vibrations from the measurements of blade tip deflections.

2. Verify the noninterference technique data processing algorithms using existing analog-recorded strain-gage stress data from a previous Arnold Engineering Development Center (AEDC) test program.

3. Assemble a single-stage prototype system for processing noninterference blade tip deflection measurements and displaying blade stress information in a format identical to the conventional strain-gage system stress data display.

4. Validate the noninterference stress measurement system online using the prototype noninterference system and an engine instrumented with both conventional strain-gage systems and compressor blade tip deflection measurement sensors.

5. Develop the data processing algorithms required to infer local blade stress for integral vibrations from the measurements of blade tip deflections.

6. Prepare specifications for a multistage noninterference stress measurement system for both nonintegral and integral-type blade vibrations based on experience with the prototype system and evaluation of the prototype system online test results.

7. Procure the multistage noninterference stress measurement system for routine use in engine testing to identify compressor blade-stress-related problems.

Currently, the program has progressed through Step 4 above. Recent online evaluation tests using an F101 engine tested at the AEDC were successful and the results will be reported separately.

2.0 STRAIN-GAGE STRESS MEASUREMENT SYSTEM

Engine stress tests are performed to assess performance or problems related to dynamic stresses on the blades of one or more compressor stages. This is accomplished using the strain-gage measurement technique by instrumenting three to six blades per compressor stage with one or two strain gages each to obtain a representative sample of blade stress behavior per stage (Ref. 1). Blade stress levels are monitored during online operation to protect the compressor from excessive stress levels.

A simplified strain-gage stress measurement system is shown in Fig. 1. Strain gages are installed directly on the blades and resulting signals are extracted using sliprings or telemetry techniques. Signal conditioning equipment, located external to the compressor, provides interfacing between the strain-gage signals and stress analysis, display, and recording equipment.

First, a brief theory of the strain-gage system will be given followed by a description of each major functional component.

2.1 THEORY

Compressor blades, when subjected to dynamic forces, vibrate in specific vibrational patterns and modes typical of those shown in Fig. 2. The strain-gage measurement technique is based on the ability to infer resulting maximum blade stress from measurements made with strain gages at one or two predetermined locations on the blade.

Bench tests of heavily instrumented blades (Fig. 3) provide the comprehensive vibrational mode and amplitude characteristics required to correlate measured stress at the selected strain-gage location to maximum blade stress which usually occurs at some other point on the blade. A typical set of rotor blade bench test results is shown in Table 1. Selection of the strain-gage location on the blade (Fig. 4) takes into consideration the maximum blade stress versus mode, strain-gage survivability, and potential of the gage installation altering blade characteristics.

2.2 HARDWARE

Figure 1 depicts the major components of the strain-gage measurement system. A brief description of each component will be given along with significant disadvantages associated with the system installation, operating life expectancy, and cost.

2.2.1 Sensors and Lead Wires

Installation of strain gages on compressor blades requires disassembly of the engine. The blades are then individually prepared for the strain gages and lead wires which are attached by a welding or adhesive bonding process. The installed strain gages and lead wires are then coated with an epoxy for environmental protection. The bonding process and protective coatings can modify the blade physical properties and affect aerodynamic performance of the blade. Figure 5 shows an assembled instrumented rotor with lead wires from the gages routed into the hub.

Strain gages and lead wires are exposed to hostile environments of extremely high temperatures, centrifugal forces, and vibrations which result in excessive failure rates. Most lead wire failures occur where the wires traverse high-stress points or transition between movable parts internal to the engine (Ref. 2).

2.2.2 Signal Transmission Devices

Sliprings or telemetry techniques are required for transferring the strain-gage signals from rotating to stationary components of turbine engines.

Sliprings are typically 3 to 10 in. in diameter and provide 10 to 50 data channels. Electrical noise generated by the slipring brushes and/or contamination of the rings can degrade data quality thereby limiting operating time without periodic extensive slipring cleaning operations.

Telemetry techniques require installation of miniature transmitters within the engine rotor. The transmitter outputs typically operate near 100 MHz, are frequency modulated by the strain-gage signals, and transmit to a stationary receiving antenna. The strain-gage analog signals are reconstructed by receivers tuned to specific transmitter frequencies. The receiver output analog signals are then processed in the same manner as slipring output signals. Typically, fewer than 20 data channels per compressor can be acquired using telemetry techniques. Time multiplexing schemes, sometimes used to increase the number of data channels, can decrease system reliability but do not increase the number of channels which can be viewed simultaneously.

2.2.3 Signal Analysis and Display Equipment

Typically, stress testing subjects the engine to operations near critical blade stress limit conditions. Therefore, stress levels are monitored online to protect the compressor. Conventional data display formats (Fig. 6) have been developed from direct operating experience acquired during engine aeromechanical test programs, and stress analysts have developed recognition patterns which relate to the stress conditions and the compressor design changes necessary to correct stress problems.

During stress testing, the data analyst monitors overall signal amplitudes from selected strain gages on either the bargraph or oscilloscope displays. When the signals become active, he selects the most active gage and determines the blade vibration frequency directly from the spectral analysis display. He then identifies the vibratory mode and applies stress limits based on bench tests of representative blades (Table 1).

Viable alternatives to the strain-gage system should alleviate existing problems associated with sensing and signal transmission devices internal to the engine while retaining conventional stress analysis and display features of the strain-gage system external to the engine. The noninterference stress measurement system described next meets these criteria.

3.0 NONINTERFERENCE STRESS MEASUREMENT SYSTEM

A simplified noninterference stress measurement system is shown in Fig. 7 and is functionally comparable to the strain-gage system (Fig. 1). Blade tip sensors are located on the periphery of the compressor case in line with the rotor stage of interest. Signal conditioning equipment provides interfacing required between the blade deflection sensors and special processing algorithms which provide output data compatible with conventional strain-gage-type analysis and display equipment.

The noninterference compressor blade stress measurement technique is an attractive alternative to the strain-gage system in that it has the potential to alleviate weaknesses of the strain-gage system and yet retain stress data analysis and display features of the strain-gage system with which the blade stress analyst is intimately familiar.

First, a brief theory of the noninterference stress measurement technique will be given followed by a description of each major functional component.

3.1 THEORY

A simplified schematic showing the conceptual configuration of the noninterference compressor blade stress measurement system is shown in Fig. 8. Stationary noninterference sensors located on the periphery of the compressor case sense the deflection of compressor blade tips with respect to the blade roots. A reference for measurement system synchronization and rotor stage blade identification information is provided by a one/rev sensor. The processor contains the special alogrithms for extracting blade vibration amplitude and frequency information from the blade tip deflection measurements. Maximum blade stress is inferred from this information in the same manner as for the strain-gage measurement system.

As is the case for the strain-gage system, prior knowledge of blade characteristics is required to correlate blade tip deflections and maximum blade stress. Blade tip deflection versus stress must be obtained from bench tests of representative blades in much the same manner as currently done for strain-gage measurements. In fact, the procedure would be the same except for the added measurements of blade tip deflections.

Noninterference blade tip deflection measurements are comparable to strain-gage measurements in terms of information content if certain sampling theory criteria are satisfied. If a vibrating blade is sampled at a sufficiently high rate, every detail (amplitude and frequency content) of the vibration can be reconstructed exactly. If the vibration has periodic or random content, sampling theory states that a minimum of two samples per wavelength of the highest frequency is necessary to completely identify the vibration amplitude and frequency content.

To meet the above sampling criteria, the minimum number of equally spaced measurement stations required around the periphery of a compressor stage would be equal to the highest blade vibration frequency of interest divided by the compressor revolutions per second times two sampling stations per vibration cycle. The number of measuring stations for the rotor stage of a typical compressor is calculated below.

Example: Highest blade vibration frequency of interest = 3,500 Hz

Compressor Speed = 16,600 rev/min ≈ 277 rev/sec

Number of Measuring Stations:

$$\frac{3,500}{277} \times 2 \approx 25$$

Such a scheme would fully satisfy classic sampling theorem requirements and thus provide data compatible with accepted data analysis techniques. Because of the large number of sampling stations required, this scheme is not mechanically acceptable for a standard developmental engine test, but may be acceptable in a research effort.

Using a more practical number of peripheral measurement stations results in tip deflections being sampled at a sampling rate considerably less than the blade vibration frequency. This condition of the sampling frequency being less than the blade vibration frequency results in a sampled data frequency that is alias to the blade vibration frequency (Fig. 9). This phenomenon is well described in the literature (Refs. 3 and 4). Measurement of blade tip deflection to infer blade maximum stress having an assumed vibration mode has been reported in the literature (Refs. 5, 6, 7). In fact, only with unique processing techniques can the blade vibration frequency be extracted from the data sampled less than a two sample-per-vibration cycle rate.

The two categories of blade vibration to be measured are (1) nonintegral and (2) integral. The nonintegral vibrations category is where the blade vibration frequencies are not an integer multiple of engine rotor speed. The integral case is where the blade vibrations are an integer multiple of engine rotor speed. The number and location of noninterference sensors may be dependent on whether the vibrations are nonintegral or integral with respect to engine rotor speed.

When nonintegral vibrations are present, the blade tip deflection with respect to the blade root is different each time the blade passes the stationary sensors; thus, blade tip deflection amplitude and frequency information can be extracted using a single-measurement station per rotor stage.

When integral vibrations are present, the blade tip deflection with respect to the blade root is the same each time the blade passes the stationary sensors; thus, extraction of blade vibration information may require multiple measurement stations per rotor stage.

A noninterference blade stress measurement prototype system for a single compressor stage has been developed to extract nonintegral blade vibration amplitude and frequency from blade tip deflections obtained from a single compressor case periphery measurement station. The prototype system has been verified in the laboratory by electronically simulating blade tip deflection measurements.

The next phase of this effort will develop the processing algorithm requirements to extract integral blade vibration information from blade tip deflection measurements.

3.1.1 Tip Deflection Amplitude Determination

For a typical engine compressor, the time a given rotor blade can be viewed from a stationary point on the compressor case is small compared to the period of a blade's primary vibration cycle. The blade viewing time for a given rotor stage is inversely proportional to the compressor speed and the number of blades. The blade viewing time and blade resonant vibration characteristics are depicted in Fig. 10 for two typical rotor stages with the lowest and highest primary vibration frequencies shown for each stage.

For example, for a rotor stage with 25 blades at 100-percent rated compressor speed, the blade viewing time per revolution is approximately 150 μsec.

$$\frac{1}{16{,}600\ \frac{\text{rev}}{\text{min}} \times \frac{1\ \text{min}}{60\ \text{sec}} \times 25\ \frac{\text{blades}}{\text{rev}}} \approx 150\ \mu\text{sec}$$

The period of this rotor stage first flex vibration frequency is approximately 1,000 μsec; thus, several revolutions of the compressor are required to obtain blade deflection amplitude data from a single-measurement station.

Sensors located on the periphery of the compressor case provide measurements of instantaneous blade tip deflections each revolution of the engine rotor. Overall blade vibration amplitude is extracted by observing instantaneous deflections over several engine revolutions as the blade passes the sensors in different positions within its vibration cycle.

3.1.2 Tip Deflection Frequency Determination

Sampling theory (Ref. 3) states that for a sampling frequency (f_s) data frequencies up to a cutoff frequency (f_{co}) equal to $f_s/2$ can be sampled and the spectral information identified using conventional techniques. Data frequencies above f_{co} will "fold-over" into the 0 to f_{co} frequency band, and the result aliased. "Fold-over" occurs at multiples of f_s and is illustrated in Fig. 11. For example, the input data frequency of 3.75 kHz results in an apparent output indication of 1.25 kHz. Of course, in the general case, any number of input frequencies could exist which would produce the same apparent output frequency. In the case of a turbine engine rotor blade, the primary vibration frequencies are known from laboratory tests and are the predominant component of the blade's dynamic deflection at a stress boundary condition experienced during testing as typified in Fig. 12. Utilizing this predominant vibration frequency characteristics along with sampling rate information, the apparent output or alias frequency can be determined as shown in Fig. 13. Figure 13a, for example, is based on 100-percent rated compressor speed which yields a sampling rate of 277 samples/sec (revolutions/sec). If the primary vibration modes are entered on this chart (shown in rectangles) along the input frequency axis, the resulting apparent output frequency (shown in circles) for each vibration mode is found by projecting vertically to the curve, then projecting horizontally to the output frequency axis. The apparent output frequency is 124 Hz for the 984-Hz first flex vibration mode. In this case, each vibration mode has a unique apparent output frequency. Thus, when the baseline output data increases as a result of a stress boundary condition, the apparent output frequency obtained indicates the vibration mode. For the case where vibration modes yield the same or nearly same apparent output frequency, as a result of rotor blade or vibration mode frequency tolerance and/or compressor speed stability tolerance, then the sampling rate (compressor speed in revolutions/sec) would be changed and the apparent output frequency determined as shown in Fig. 13b. In this case, the magnitude of the apparent output frequency change between the two sampling rates indicates the vibration mode. Fig. 13c shows the total change in apparent output frequency for each vibration mode resulting from a 1-percent sampling rate change from Fig. 13a to Fig. 13b. For example, the third flex vibration mode resulted in apparent output frequencies of 58 Hz for N_{100} (Fig. 13a) and 87 Hz for N_{101} (Fig 13b) for a difference of 29 Hz which is approximately 1 percent of the vibration mode frequency. Note that the blade vibration mode resonant frequencies increased at the higher compressor speed as a result of the stiffening effect of centrifugal blade forces.

Intentional changes in rotor speed to determine the blade vibration frequency may not be required. Most engine rotors exhibit some minor variations in speed of a few rpm even during steady-state operations. These minor speed changes (sampling rate variations) cause corresponding changes in the aliased frequency depending on the blade vibration frequency and the speed variation as illustrated in Fig. 14. In this example, a blade vibrating at 2,630 Hz will vary from 27 to 33 Hz when sampled from a single-measurement station with a rotor speed of 12,000 rpm with variations of ±15 rpm.

The blade predominant vibration frequency(s) can be uniquely identified by the above method. A signal of known frequency sampled synchronously with the blade tip deflections will be affected by the sampling rate in the same way as the blade vibration frequency. Therefore, one can identify the blade vibration frequency by observing the spectral content of both sampled signals while adjusting the frequency of the known frequency signal until the two resulting frequency spectra are similar. The output spectra will be similar only when the known frequency signal is at the same frequency as the predominant vibration frequency. Additional information concerning the sampling process is included in Appendix A.

3.1.3 Validation of Noninterference Technique

The noninterference blade stress processing algorithm has been demonstrated in the laboratory using existing analog-recorded strain-gage stress data from a previous AEDC F-100 engine aeromechanical test program and an instrumentation system as shown in Fig. 15. The strain-gage data were sampled using a sample/hold amplifier with the sampling frequency controlled by engine rotor speed to simulate blade tip deflection measurements from a single measurement station.

A known frequency signal was also sampled synchronously with engine rotor speed. Both sampled signals were simultaneously observed on a dual channel spectrum analyzer operated in a peak hold mode in the frequency range from zero to one-half the engine rotation frequency. The known frequency signal was set to predicted modal frequencies for 12,000 rpm. A match in spectral outputs was obtained when the known frequency signal was adjusted to near one of the predicted modal frequencies. The apparent frequency of the strain-gage signal was 90 Hz (Fig. 16) with a 6-Hz deviation (bandwidth) and the known frequency signal (710 Hz) produced the same apparent frequency of 90 Hz and bandwidth of 6 Hz indicating that the blade vibration frequency was also 710 Hz. The above results were checked by processing the strain-gage signal using conventional spectral analysis equipment which verified that the strain-gage signal was in fact at 710 Hz.

3.2 PROTOTYPE NONINTERFERENCE SYSTEM

A prototype system including the processing algorithm has been developed to extract blade nonintegral vibration amplitude, frequency, and waveform from blade tip deflections using a single compressor case periphery measurement station.

3.2.1 Functional Description

A functional block diagram of the noninterference stress measurement system is shown in Fig. 17. Stress data are displayed online for monitoring of compressor health and recorded on tape for later detailed analysis. Stress data analysis and display features of the strain-gage system have been retained to minimize problems for the data analyst in transitioning between strain-gage and noninterference stress measurement schemes. Major components of the system include the signal conditioning. system controller, data processor, and data displays.

The signal conditioning equipment was designed based on earlier work reported by NASA-Lewis (Ref. 6) and shown functionally in Fig. 18 for flexural-type deflections. Signals from the sensors which represent time of passage of the blade roots and midtips are conditioned to provide measurements of instantaneous blade deflections each revolution of the rotor. System synchronization is provided by a one/rev signal which is also used to identify individual blades.

When a blade root passes the root sensor, the resulting output event triggers the start circuit of a digital counter. The counter is stopped when the tip of the same blade later passes the midtip sensor. During the time interval between root and midtip events, the counter totalizes pulses from a clock signal whose frequency is a known multiple of the rotor rotational frequency. Each clock pulse therefore represents a finite circumferential distance around the blade tip periphery which scales the output of the counter proportional to instantaneous blade tip deflection independent of rotor speed (Ref. 6). Torsional vibrations are extracted similarly by observing time interval differences between midtip and edge-tip events.

The functions performed by the system controller are illustrated in Fig. 19. The controller uses software logic to interpret and execute operator commands, route blade deflections to the data processor, and maintain synchronization of all system components in the different operating modes. Functional software, illustrating major sequencing operations including sampling of the known frequency signal during both data acquisition and data playback, is shown in Fig. 20. These software algorithms execute the noninterference technique theory described in Section 3.1 of this report.

The data processor includes an H/P 2100 series minicomputer with 32k of 16 bit word memory and operating software stored on floppy disks. The primary function of the data processor is to process instantaneous blade deflections compatible with the stress analysis and display equipment. Spectral analysis of the blade deflections data and the known frequency signal are performed using a Nicolet Model 660 dual channel spectral analyzer.

Development of suitable tip deflection sensors was not included in this project; however, a promising type appears to be those using optics (Fig. 21). Both NASA Lewis Research Center. Cleveland and General Electric of Evendale and possibly others have had success with these type sensors.

The blade root signal typically can be electronically generated using the one/rev synchronization signal (Ref. 6) thereby avoiding the potential mechanical problems associated with a blade root sensor.

Since all blades on each instrumented stage are visible to the tip deflection sensors, individual blades of interest need not be predetermined as is the case when using strain-gage measurement systems. The number of blades viewed simultaneously depends only on the capacity of the measurement system external to the engine.

3.2.2 Operational Description

Features of the prototype noninterference stress system include provisions for online monitoring, data recording, and data playback. During online monitoring operations, instantaneous deflections from five blades selected by the operator are acquired each revolution of the rotor. Overall deflection amplitudes are derived from multiple data samples of each blade and displayed on a CRT in bargraph format. Waveform and spectral information from one selected blade are displayed on a storage oscilloscope and spectral analyzer, respectively. The known frequency signal is sampled once per rotor revolution, and the resulting spectra are displayed simultaneously with the spectra of the blade deflection data. The data recording mode is identical to the monitoring mode except blade deflections and synchronization information are stored on digital tape. During data playback operations, data are retrieved from tape and displayed similarly as during online monitoring operations.

Blade selector switches permit the operator to manually scan all blades (five simultaneously) and select particular blades for monitoring of overall amplitude. When the signals become active, he can select one blade for identification of vibratory mode by comparing spectra of the blade deflections and the known frequency signal. The storage oscilloscope retains amplitude/time history information from one blade and provides an effective means of classifying engine instability and stall events by permitting observation of the waveform of the blade deflections as an instability or stall condition is approached.

4.0 SUMMARY

A noninterference blade stress measurement system is being developed which appears to be a viable alternative to the conventional strain-gage system. It is based on a technique which measures blade tip deflections and uses special data processing algorithms to infer local blade stresses. The blade tip deflection sensors are mounted in the engine case on the periphery of the compressor case. Display of blade stress information is identical in data format and content as for the conventional strain-gage system.

To date, the development has (1) provided algorithms for extracting blade stresses of nonintegral-type vibrations from measurements of blade tip deflections, (2) checked the algorithms using existing strain-gage data recorded during previous AEDC stress test programs, and (3) assembled a prototype single compressor stage measurement system.

Online evaluation testing of the prototype system was completed recently using an engine instrumented with both strain-gage and blade tip deflection sensors. Results of the online evaluation tests with the prototype system will be included in a separate report.

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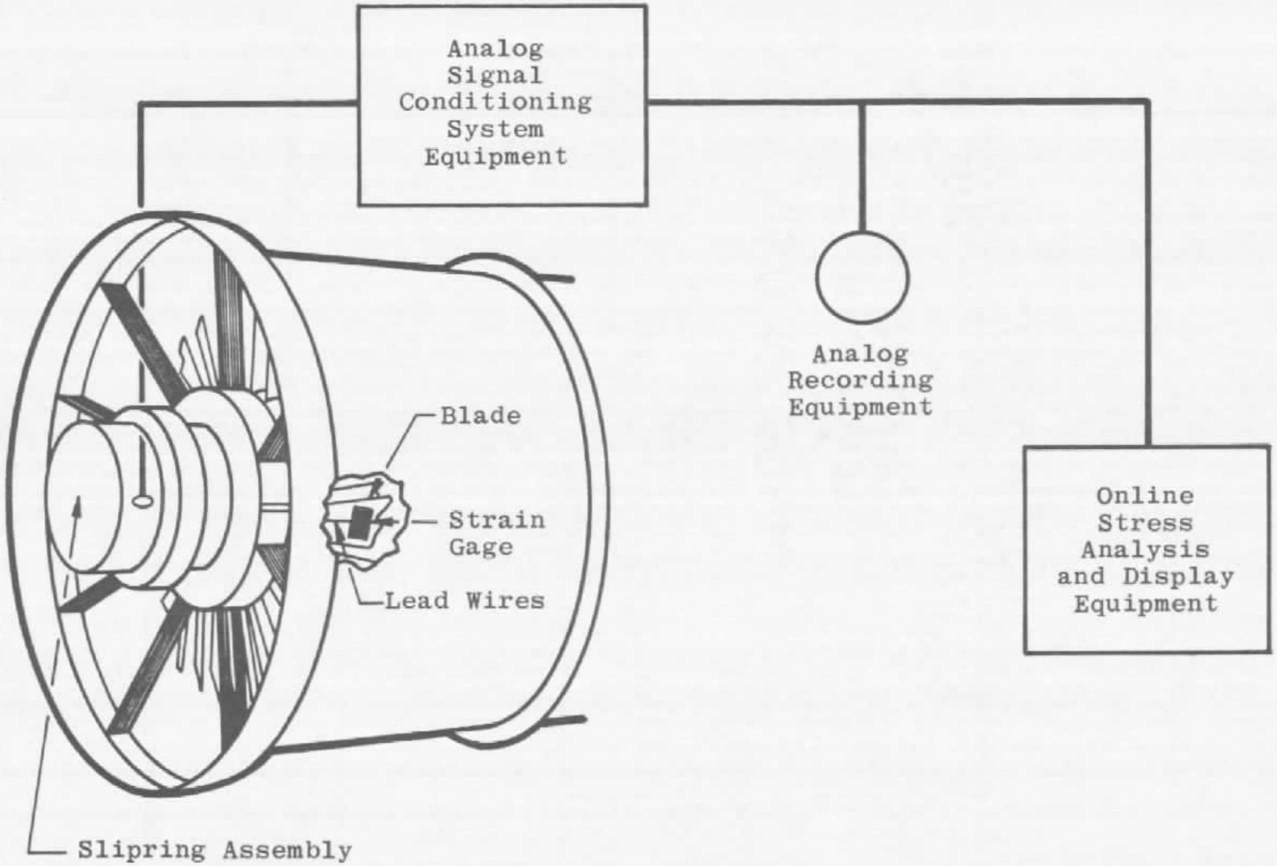


Figure 1. Simplified strain-gage stress measurement system.

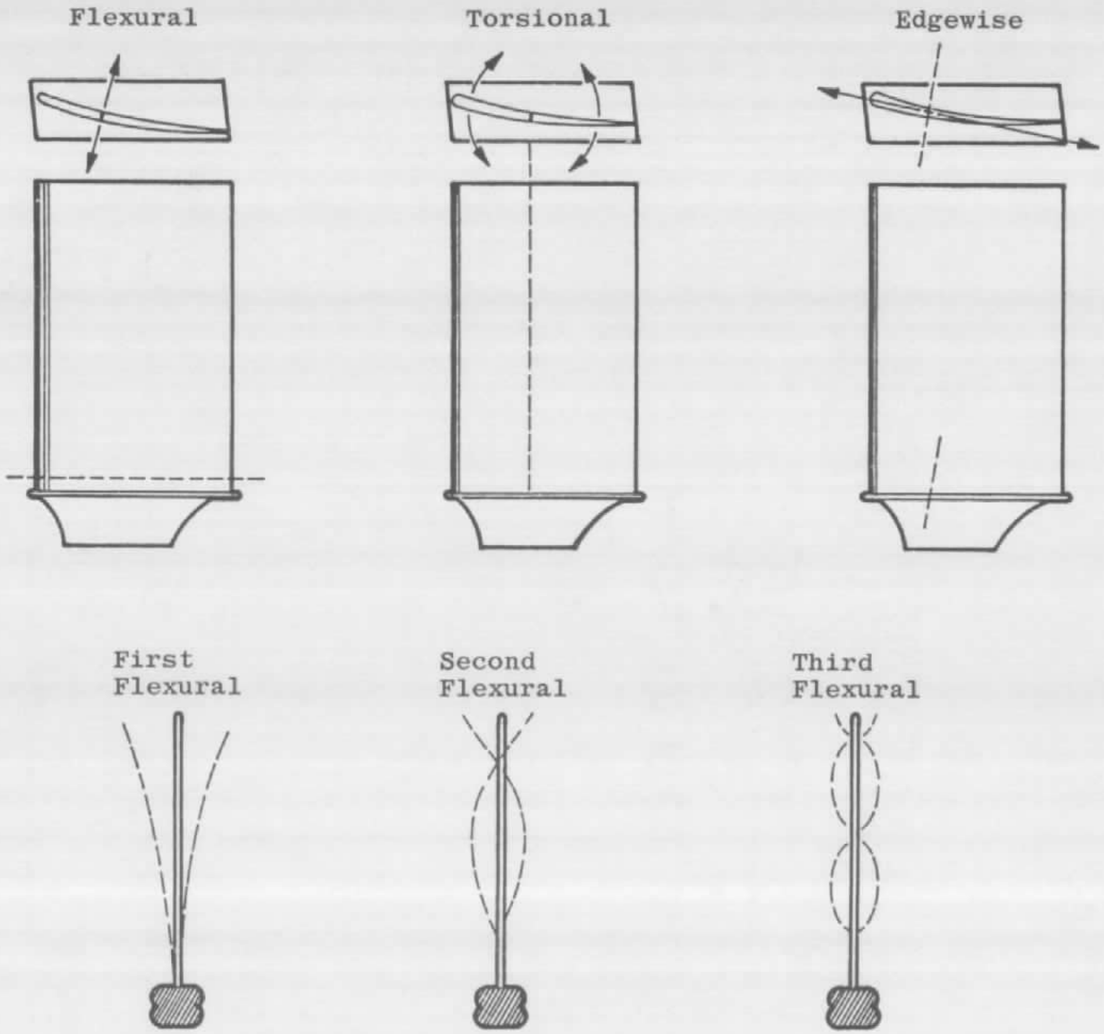
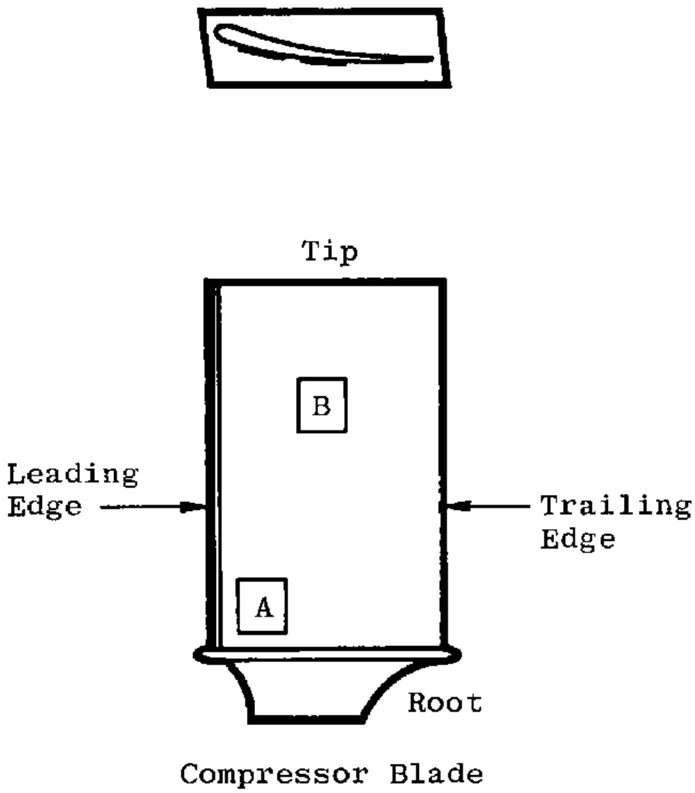


Figure 2. Typical compressor blade vibration responses.

Figure 3. Rotor blade strain-gaged for laboratory stress tests.



[A] Gage for First Flex

[B] Gage for First Torsion, Second Flex, First Edgewise

Figure 4. Typical strain-gage locations on rotor blade.

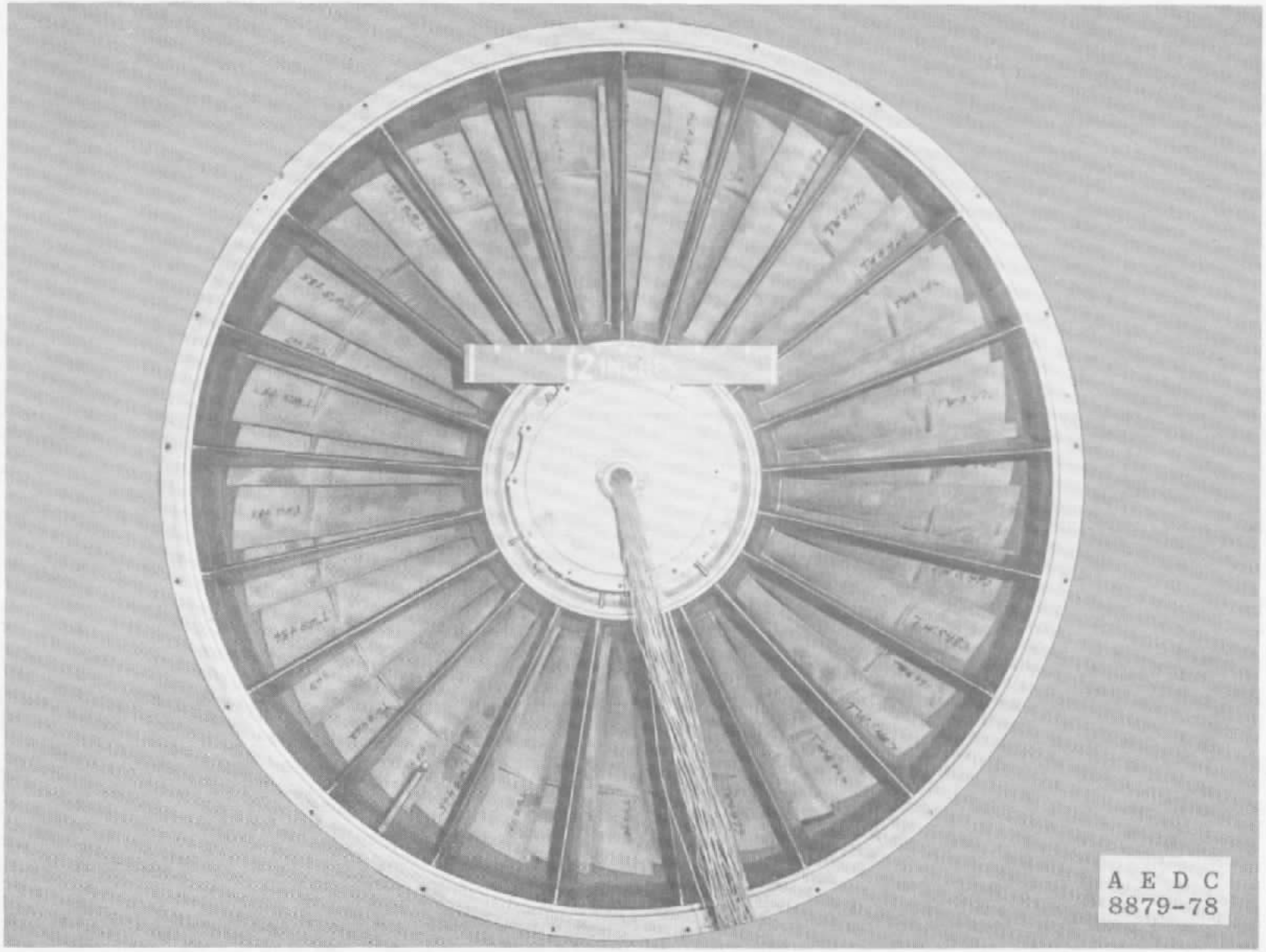


Figure 5. Rotor stage strain-gaged for engine stress tests.

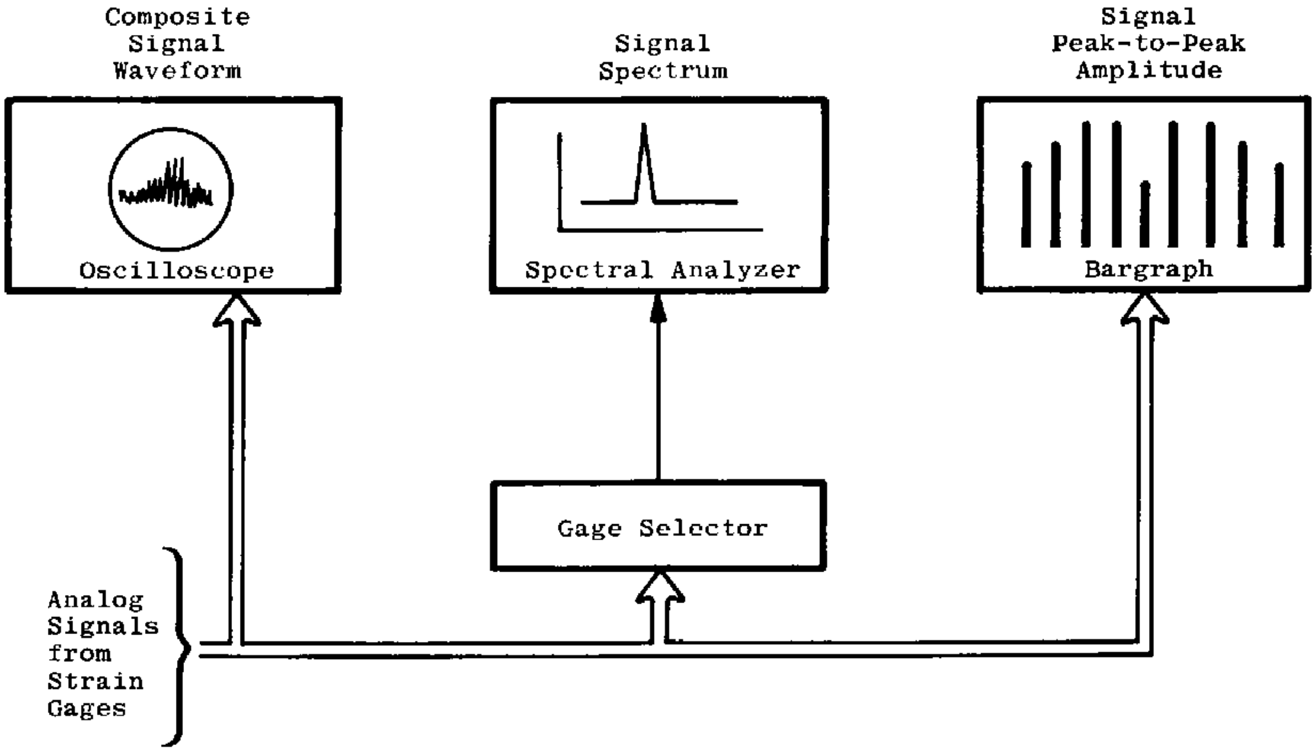


Figure 6. Typical strain-gage online stress analysis and display equipment.

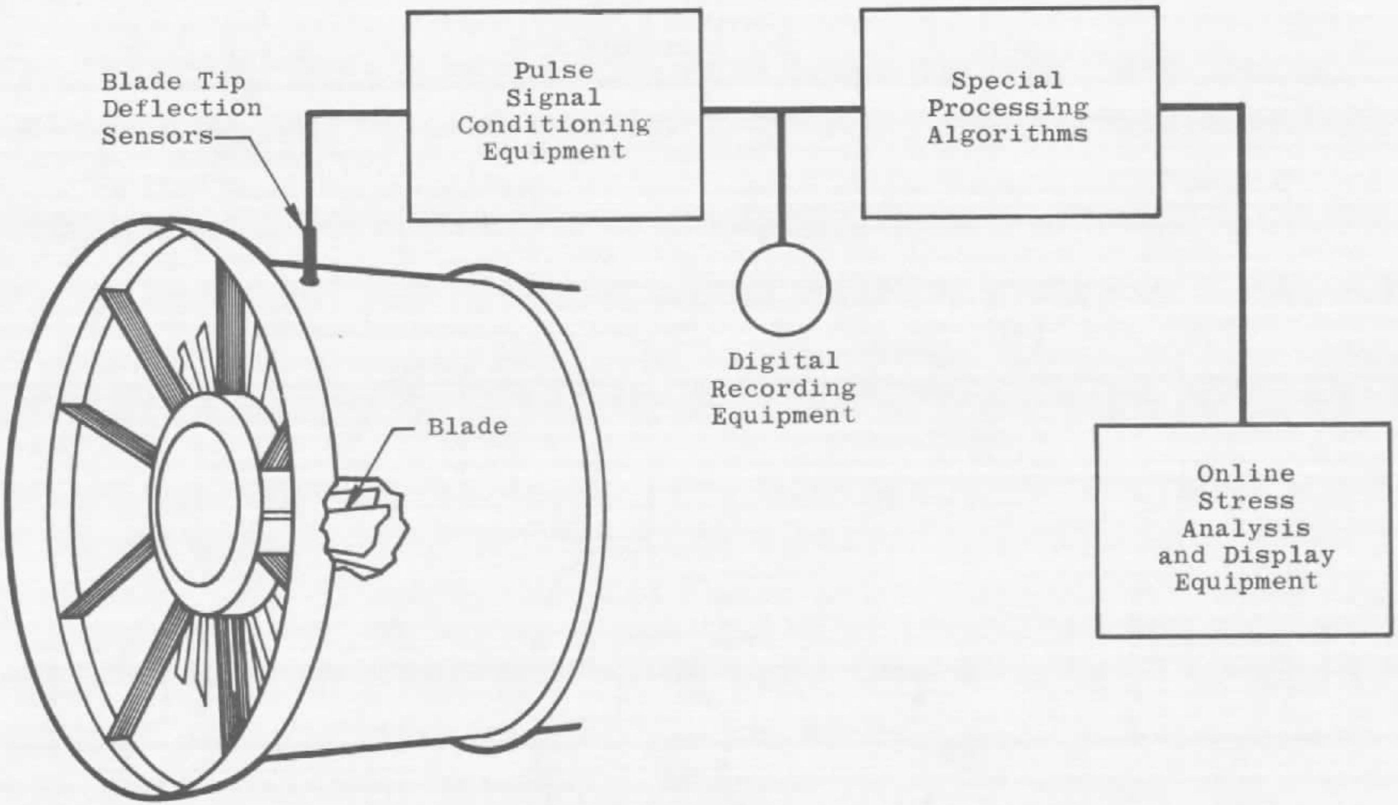


Figure 7. Simplified noninterference stress measurement system.

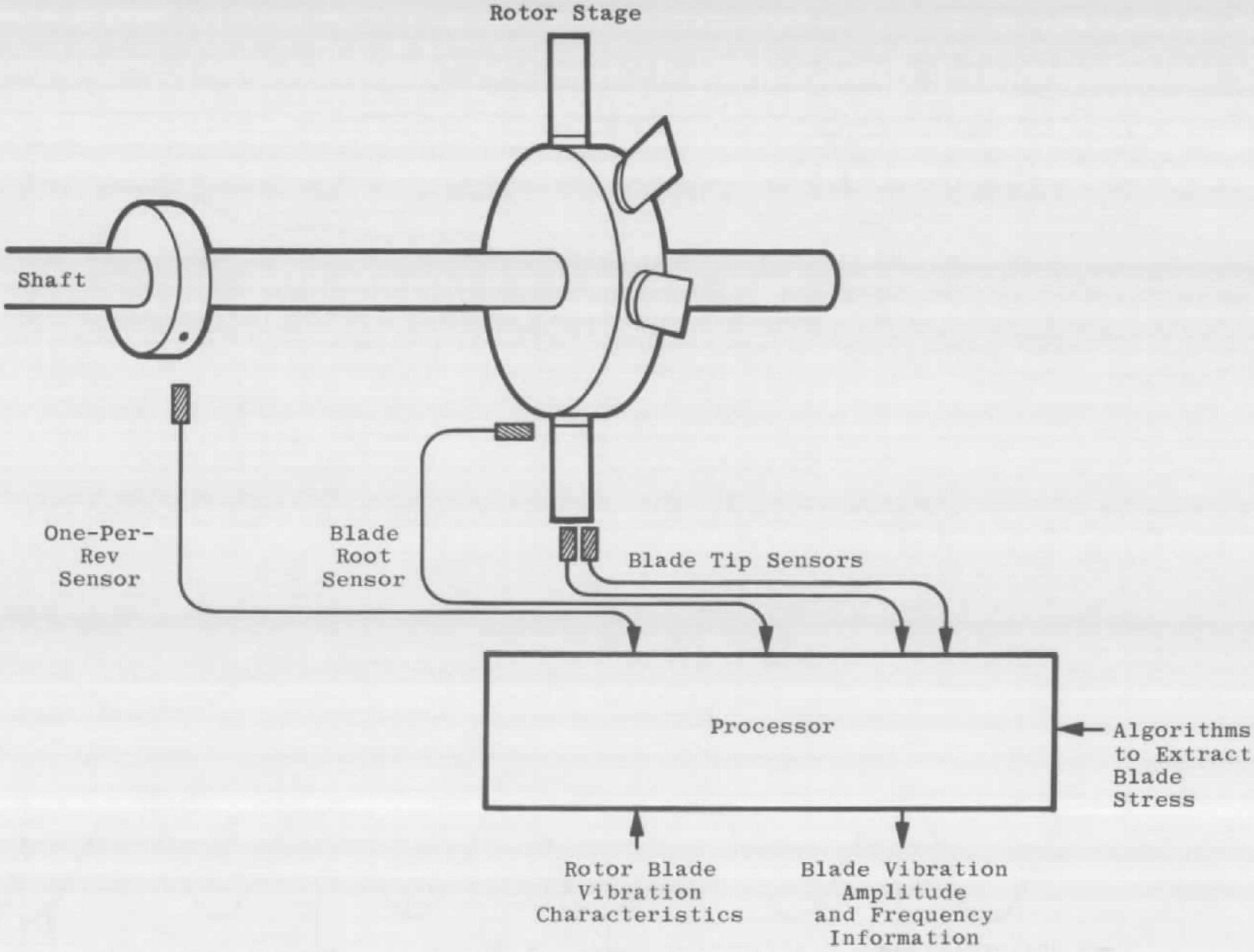


Figure 8. Conceptual configuration of noninterference stress measurement system.

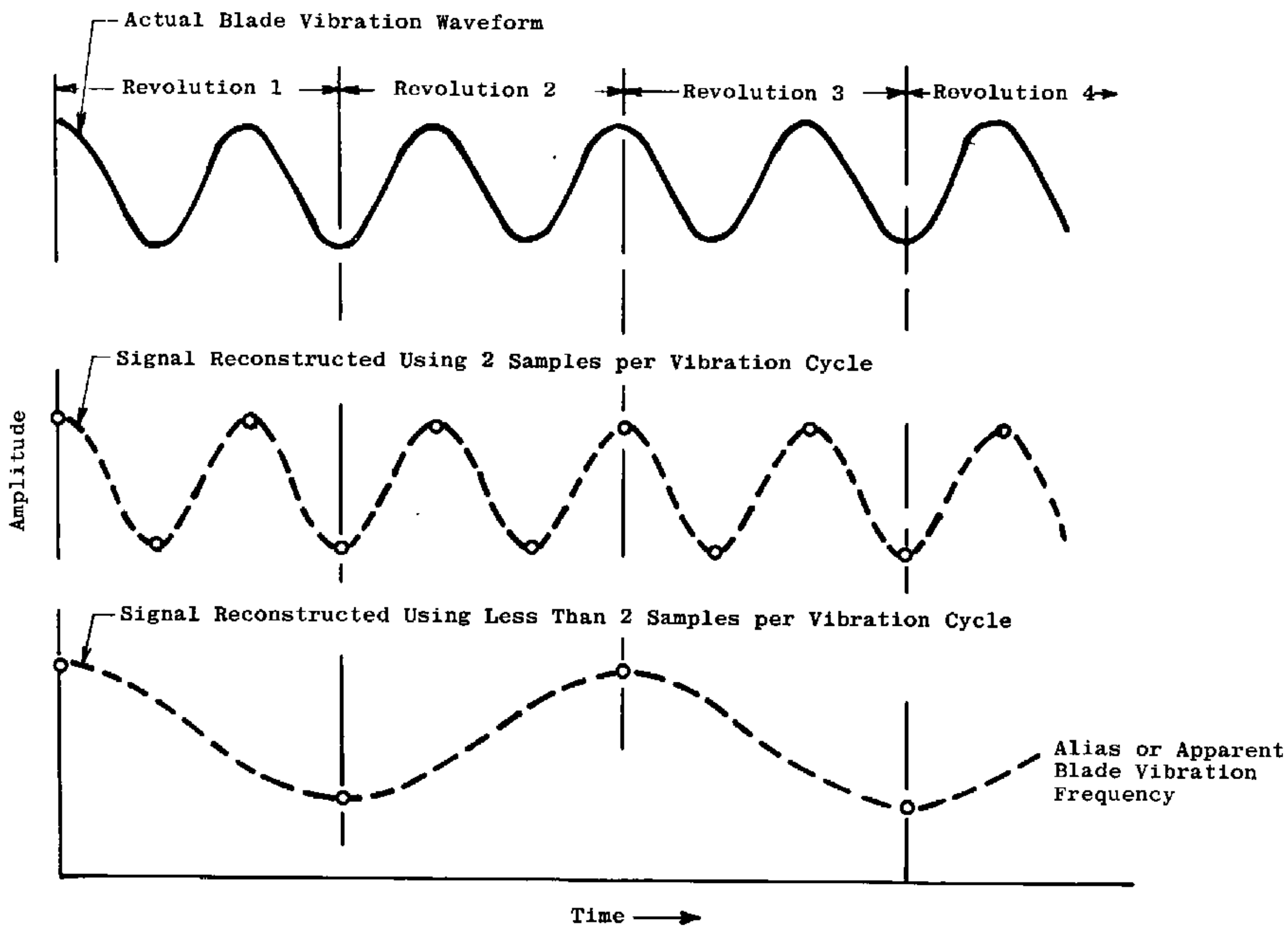


Figure 9. Example of apparent blade tip deflection frequency as influenced by the sampling rate.

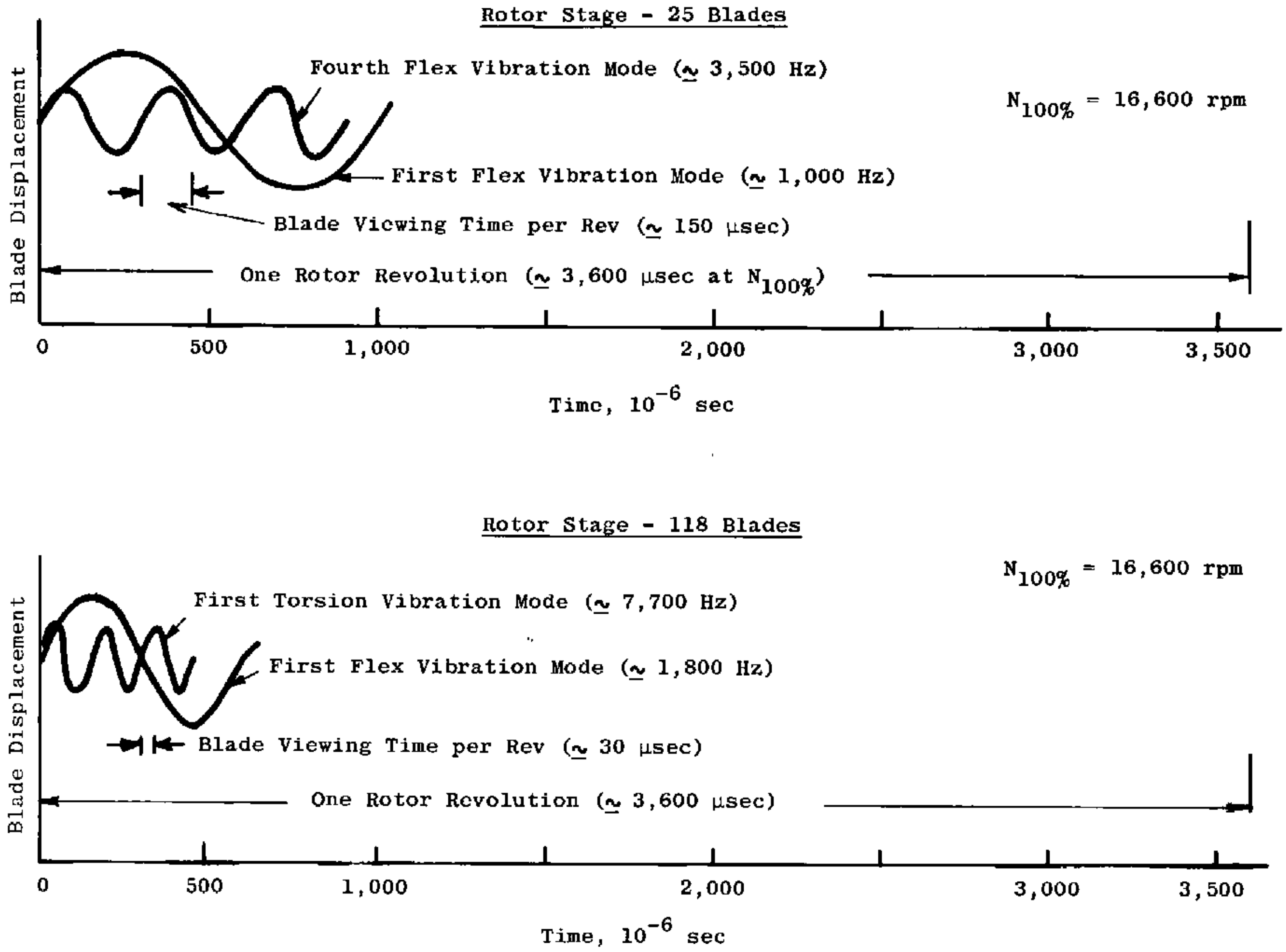


Figure 10. Typical compressor blade vibration characteristics.

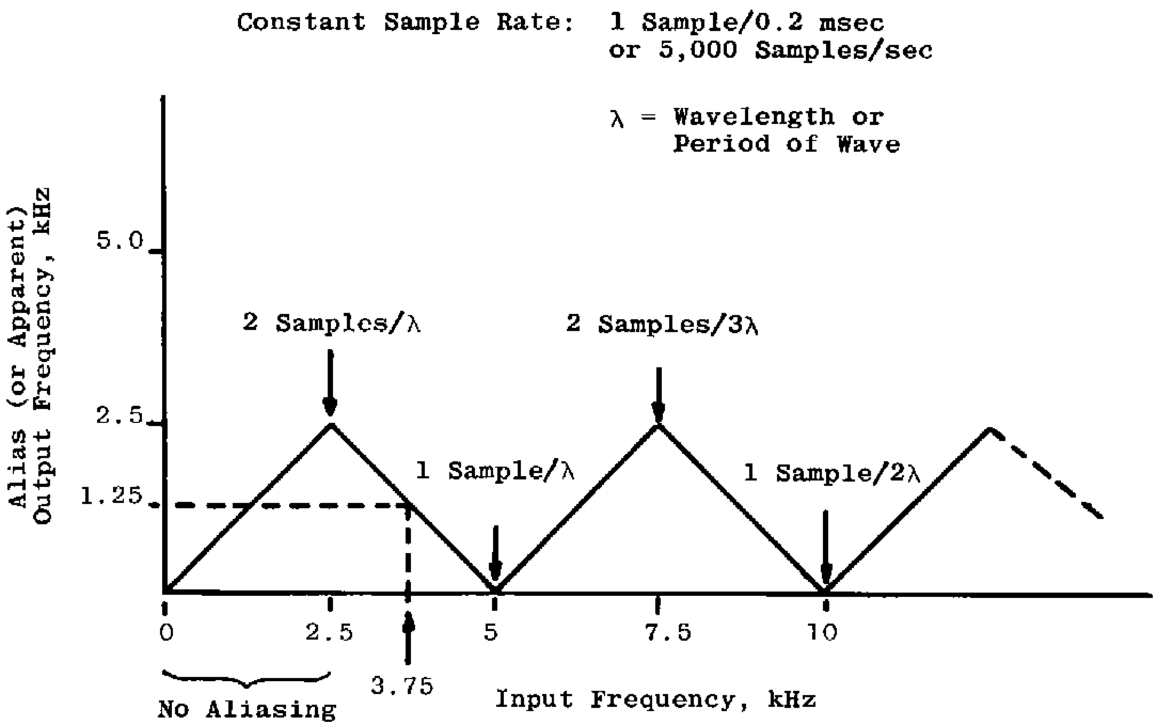


Figure 11. Example of frequency aliasing.

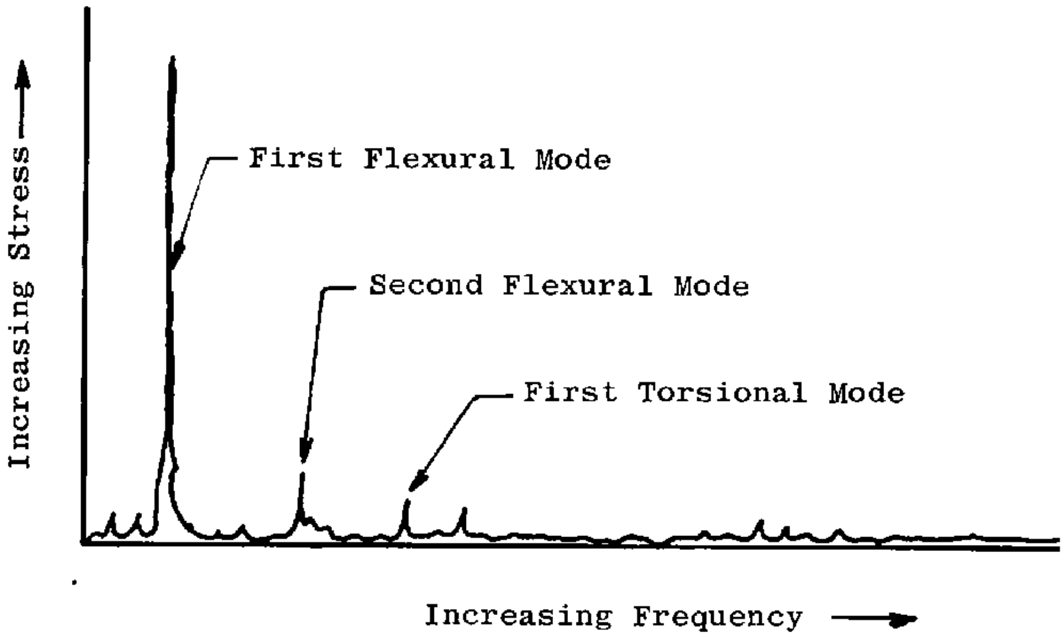
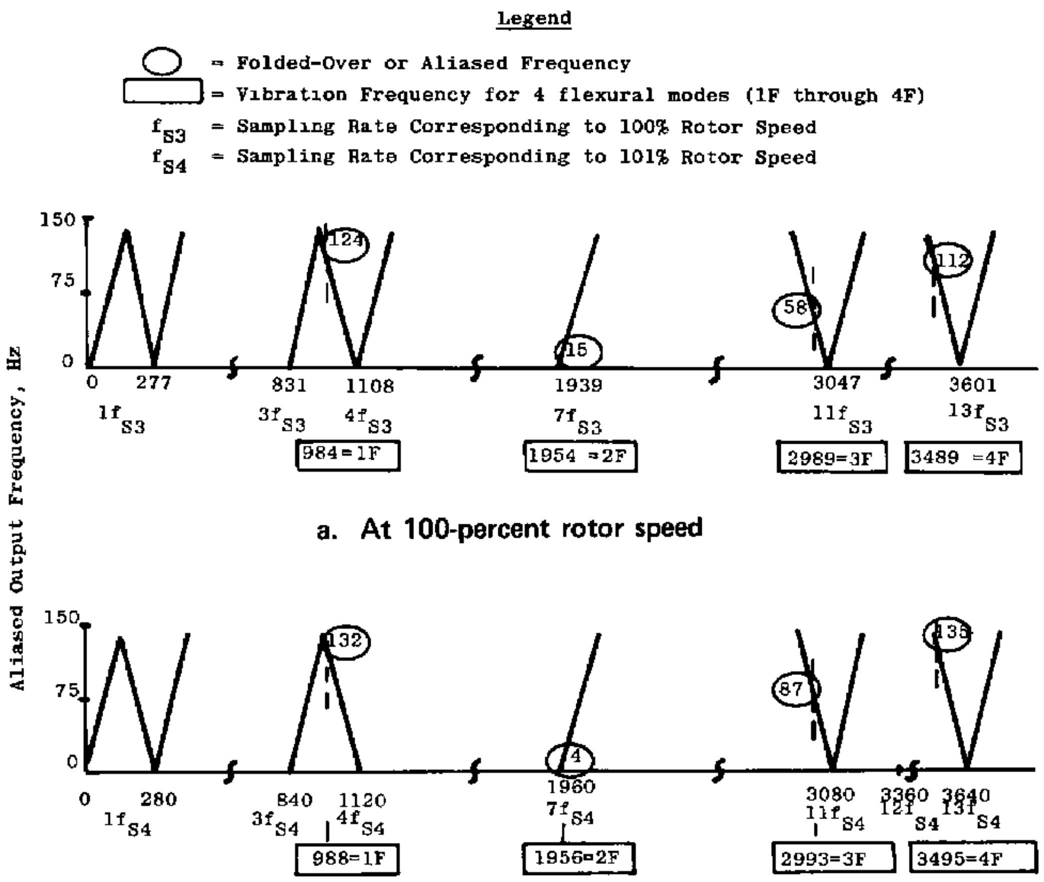


Figure 12. Typical amplitude spectrum of strain-gage signal at a stress boundary condition.



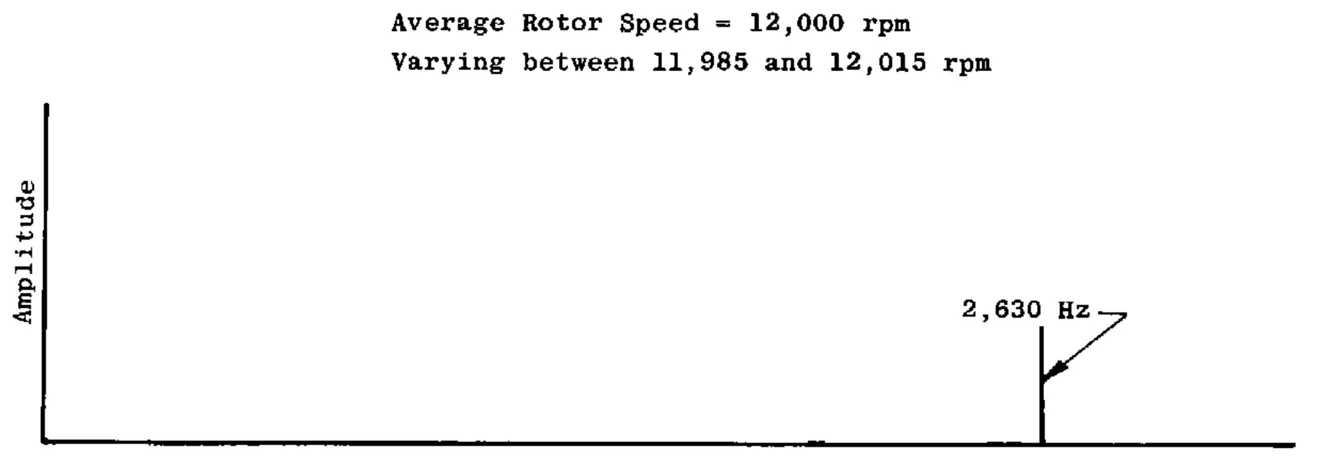
a. At 100-percent rotor speed

b. At 101-percent rotor speed

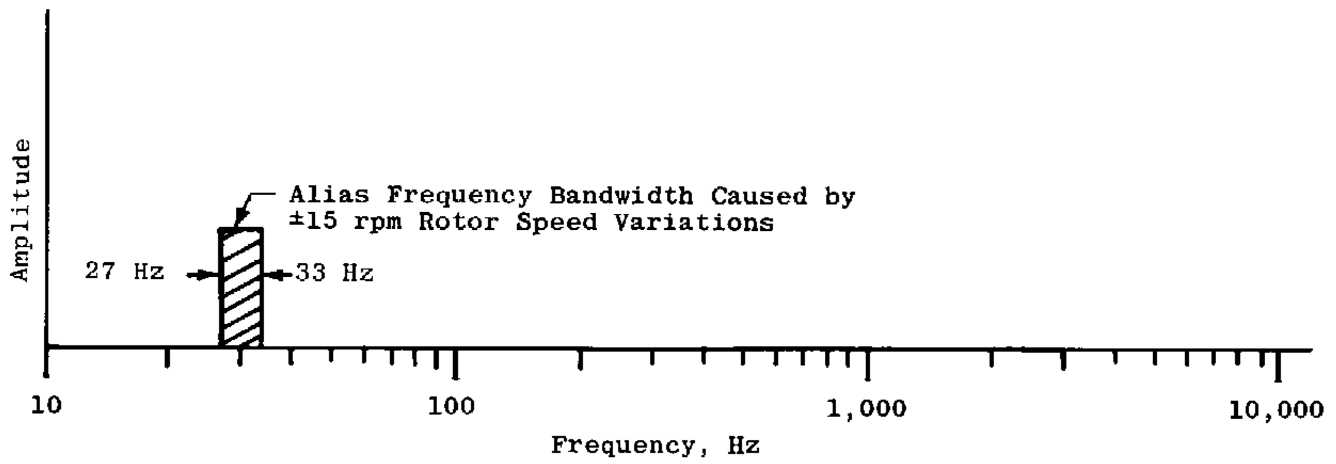
$\lvert f_{S4} - f_{S3} \rvert$	Δ 8 @ 1F	Δ 19 @ 2F	29 @ 3F	33 @ 4F

c. Change in apparent frequency between 100- and 101-percent rotor speeds

Figure 13. Example of determining apparent blade vibration frequencies for four typical vibration modes.



a. Actual blade vibration frequency



b. Apparent frequency

Figure 14. Effect of variations in rotor speed on apparent blade vibration frequency.

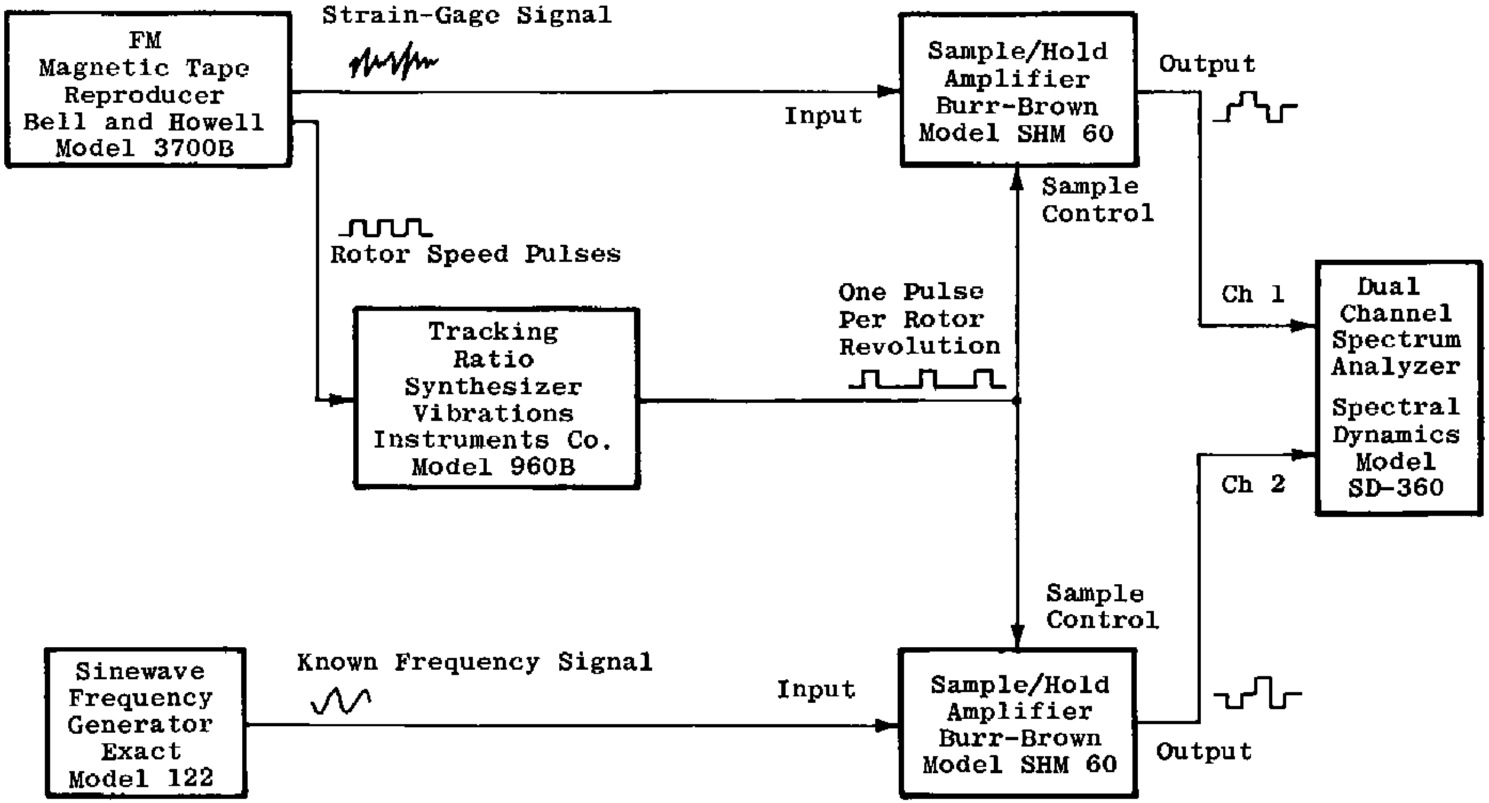


Figure 15. Schematic of instrumentation used to validate the vibratory mode extraction technique.

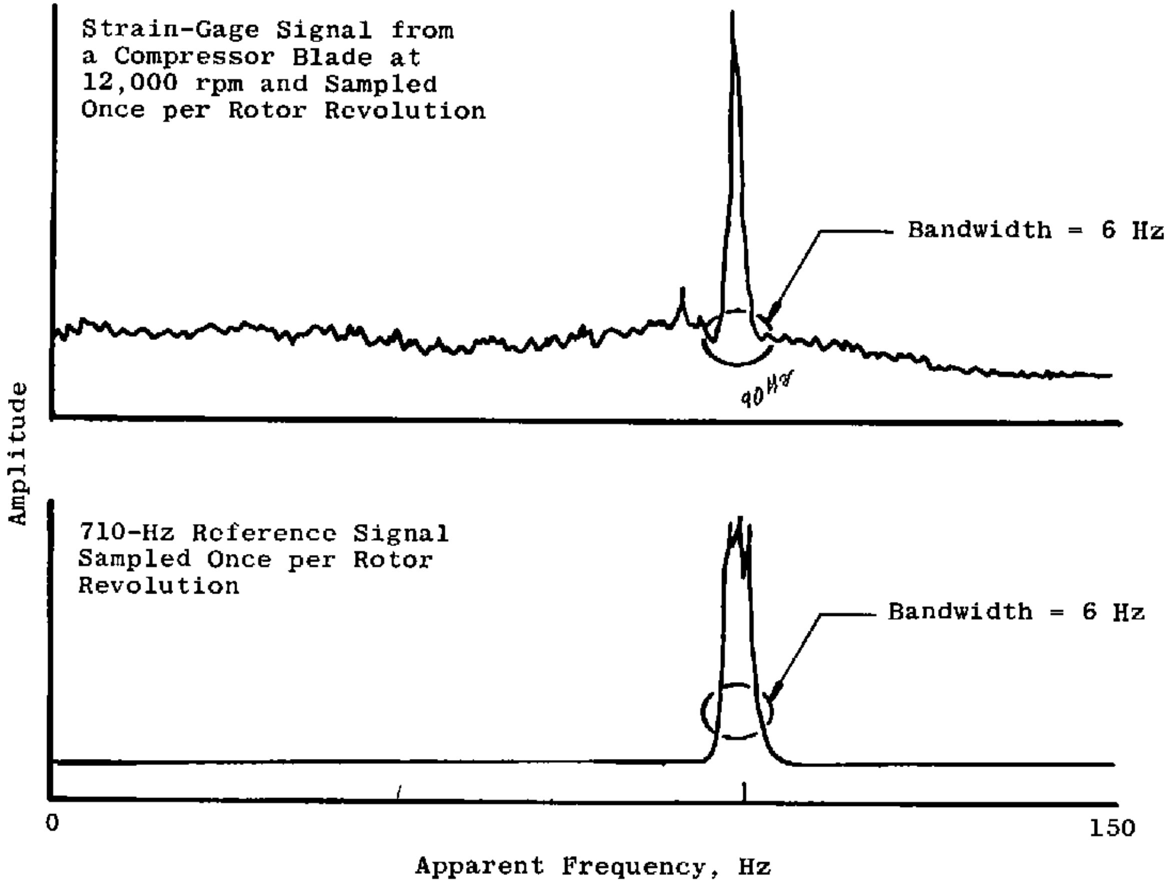


Figure 16. Determination of blade vibration frequency using the noninterference technique.

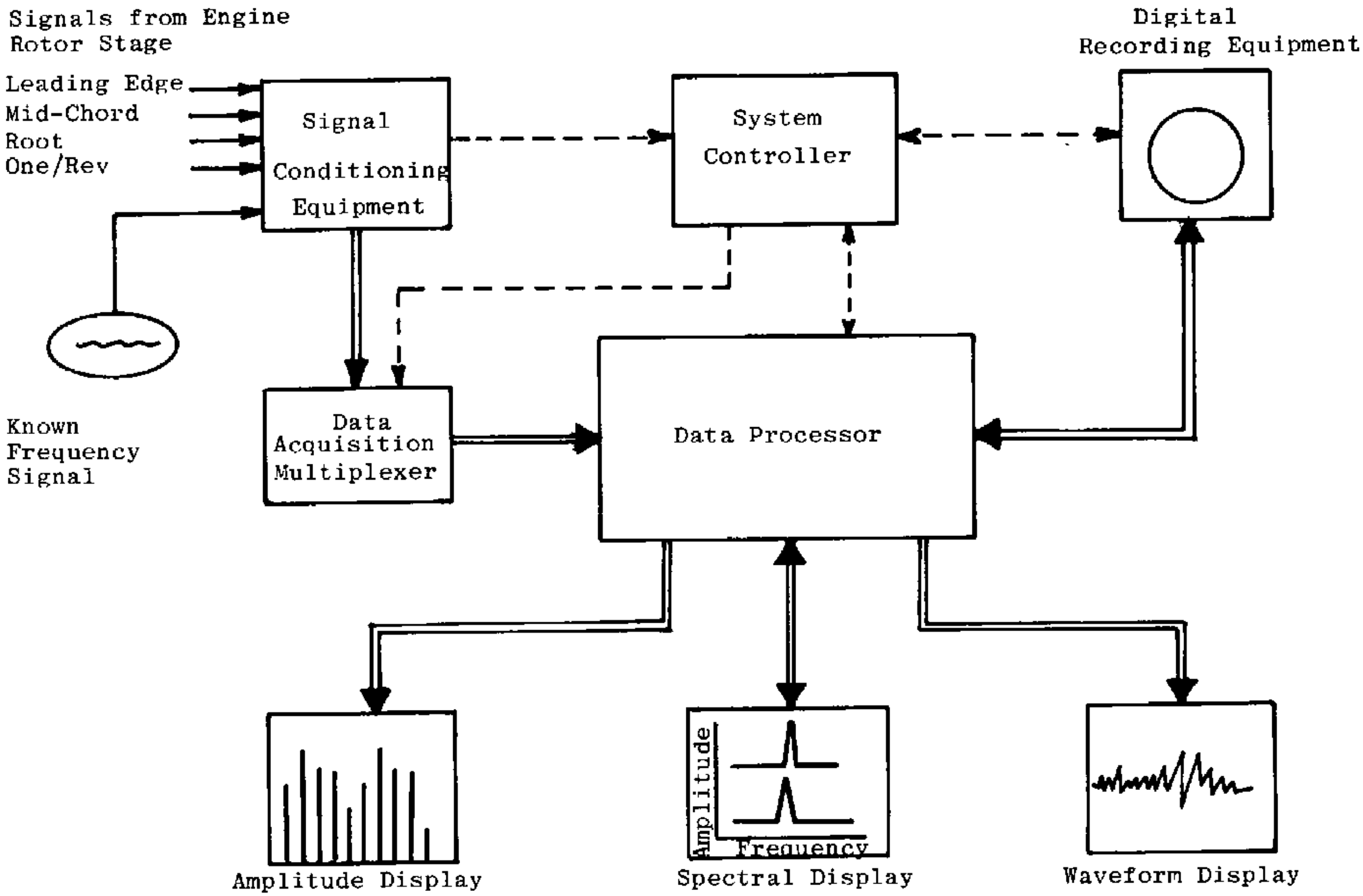


Figure 17. Functional block diagram of noninterference system.

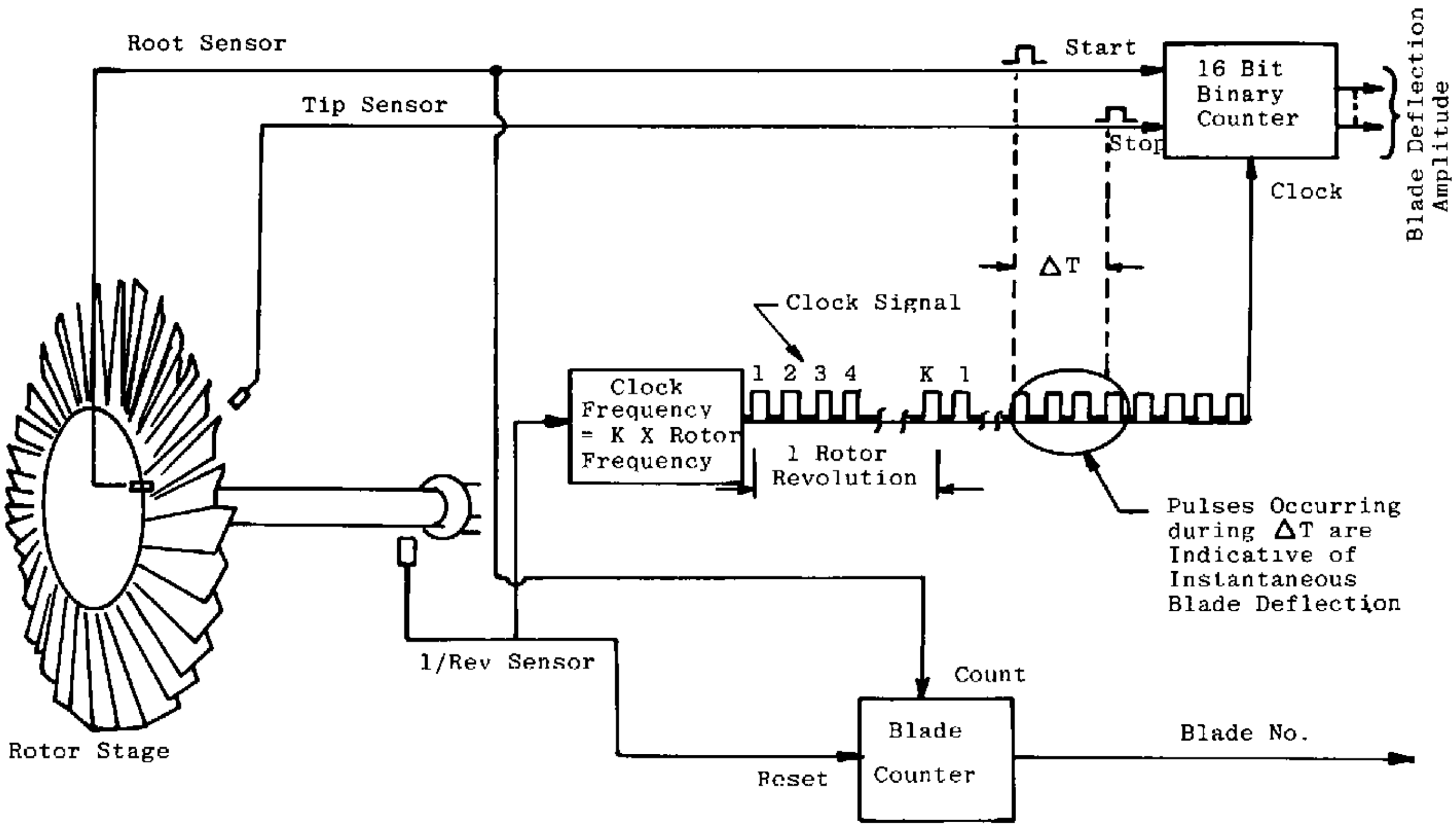


Figure 18. Functional block diagram of signal conditioning for blade deflection signals.

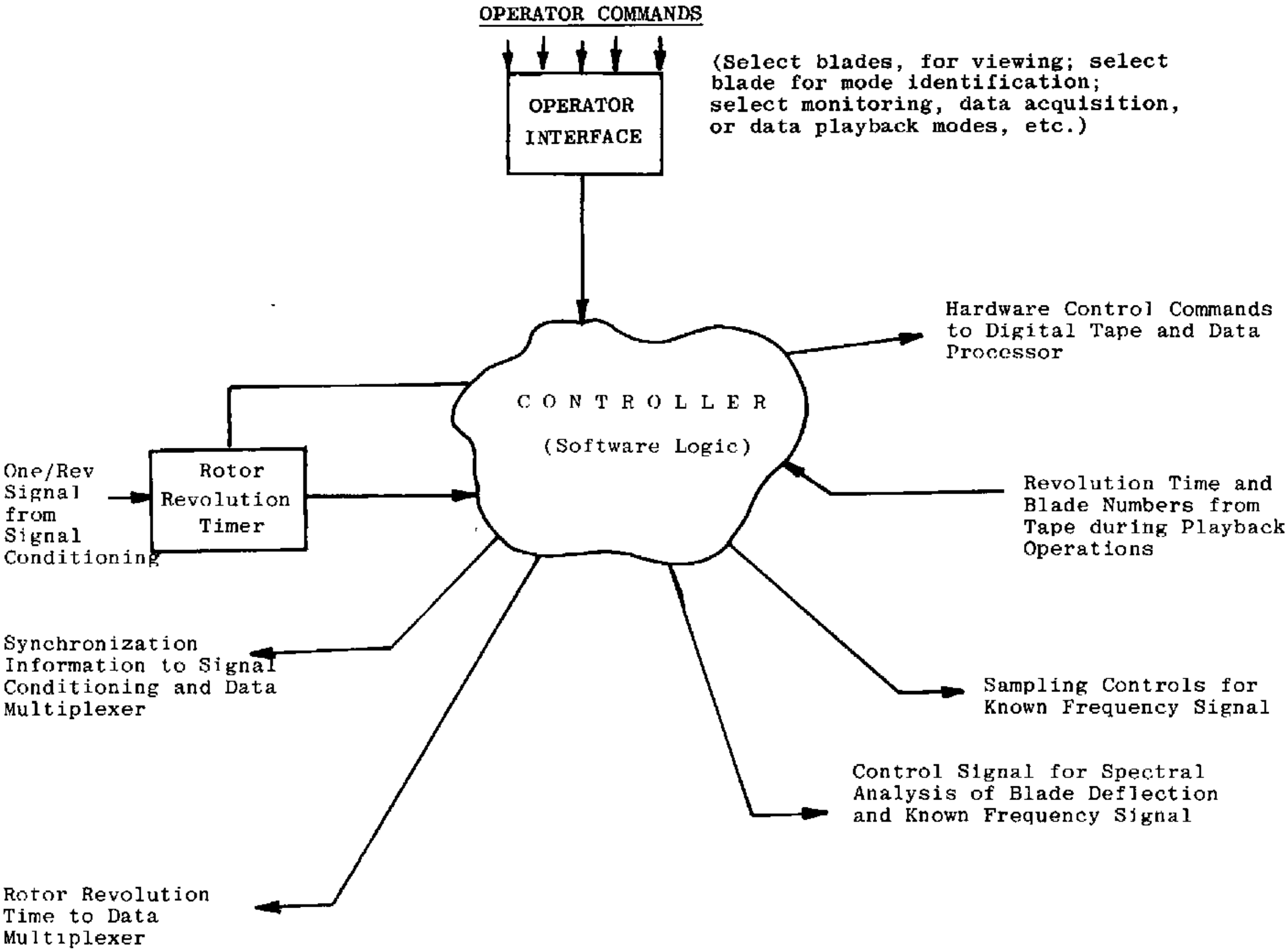
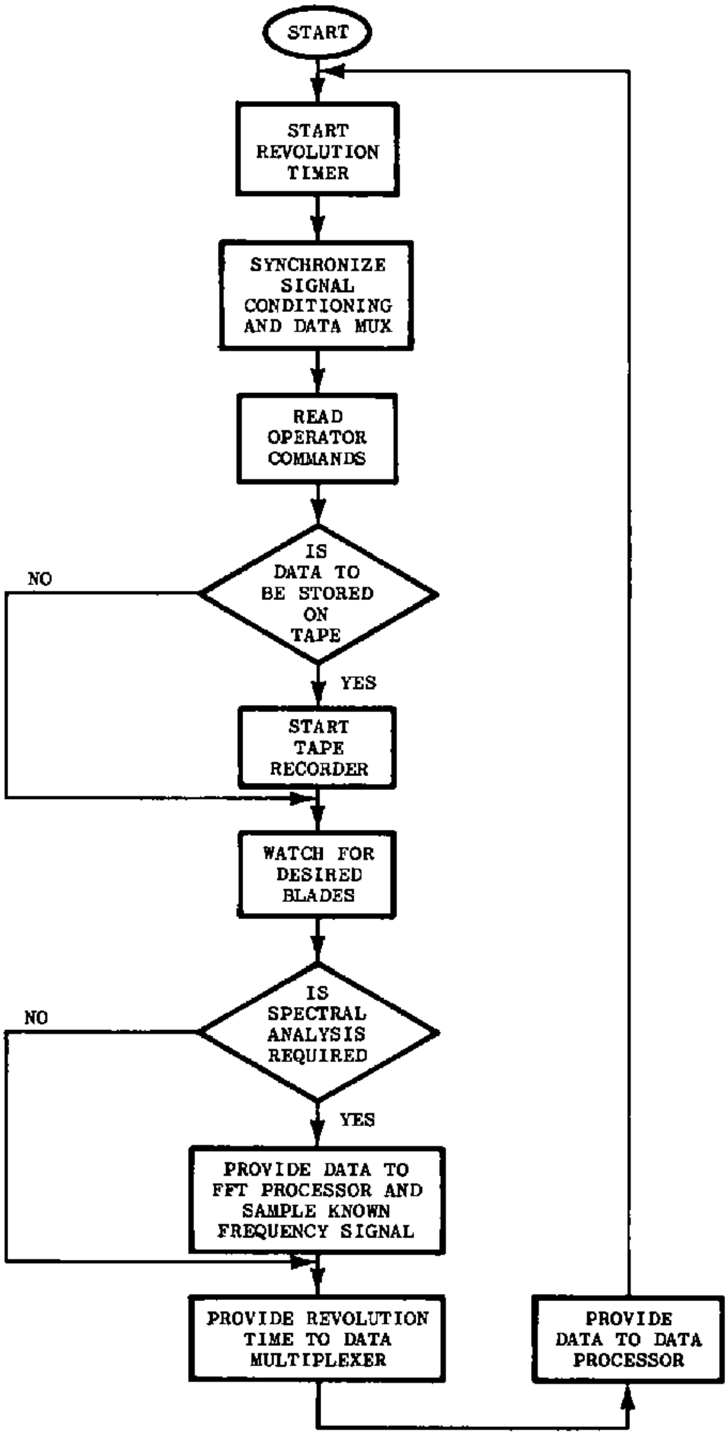
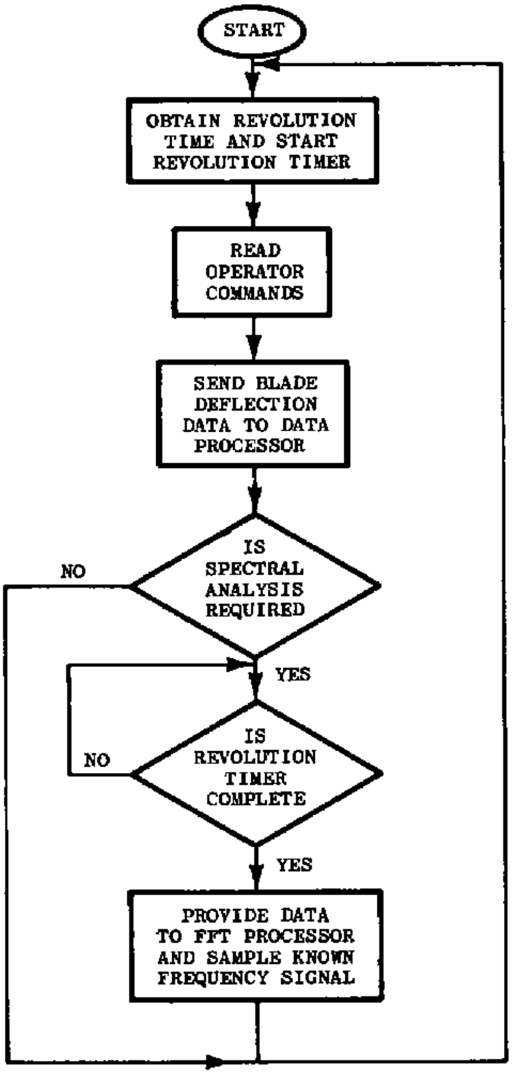


Figure 19. Functions of the system controller.



a. Data acquisition
Figure 20. System controller simplified software flow chart.



b. Data playback
Figure 20. Concluded.

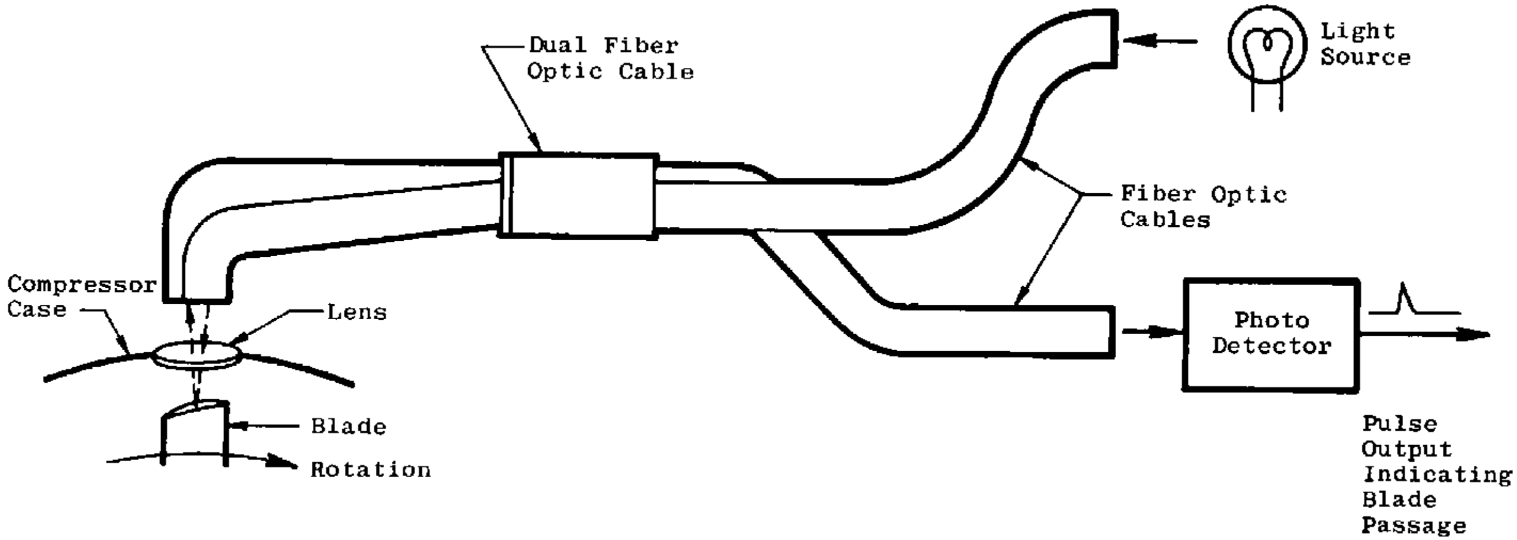


Figure 21. Schematic of typical blade tip deflection sensor using fiber optics.

Table 1. Typical Results of Rotor Blade Bench Test

Strain-Gage Location	Vibration Mode	70-percent Engine Rotor Speed			100-percent Engine Rotor Speed		
		Frequency, Hz[1]	Gage Factor[2]	Stress Limit, KSIDA[3]	Frequency, Hz	Gage Factor	Stress Limit, KSIDA
Gage Mounted at Leading Edge, Convex Side, 0.75 in. from Base of Dovetail, Gage Axis Inclined 80 deg from Horizontal Plane	1st Flexural	832	1.55	52.3	984	1.60	47.6
	2nd Flexural	1,880	2.70	31.1	1,954	2.47	26.7
	3rd Flexural	2,810	1.33	57.8	2,989	1.75	37.1
	4th Flexural	3,253	3.91	19.2	3,489	3.21	17.9

[1]Frequency for mode of vibration

[2]Ratio of maximum stress on blade for given vibration mode to stress at gage location

[3]10^3 psi double amplitude

APPENDIX A
SIMPLIFIED SAMPLING PROCESS

The following is not intended as a rigorous treatment of the sampling process and is included only to convey the fundamental process of using a change in sampling rate to identify the predominant frequency component of an undersampled signal.

A signal reconstructed from discrete data samples will not only contain the original data frequency but also new frequencies at harmonics of the sampling frequency and at sums and differences of the sampling frequency and data frequency. Frequency components will occur at

$$Fi, Fs, Fs \pm Fi, 2Fs, 2Fs \pm Fi, \ldots \text{ etc.},$$

where

$$Fi = \text{original data frequency}$$

$$Fs = \text{sampling frequency}$$

If the sampling frequency is greater than twice the data frequency, all of the new frequencies will fall in the frequency spectra above the data frequency (Fig. A-1). If the sampling frequency is less than twice the data frequency, some of the new frequencies will fall in the spectra below the data frequency (Fig. A-2). In the latter case, one of the new frequencies which is the difference between a harmonic of the sampling frequency and the data frequency will always fall within a frequency band from zero to one-half the sampling frequency and is known as an aliased frequency. This aliased frequency, Fa, is the absolute difference between a harmonic multiple of sampling frequency and the data frequency as follows:

$$Fa = \left| K \times Fs - Fi \right|$$

where K is the integer harmonic multiple of the sampling frequency producing the aliased frequency.

If one limits observation of the spectrum of a sampled signal only to the frequency range from zero to one-half the sampling frequency, then the aliased frequency can be viewed and all other frequency components contained in the sampled signal can be excluded since they will fall outside the observed frequency range. The data frequency, Fi, can be derived mathematically using the above equation if both sampling frequency

Fs and the harmonic multiple of sampling frequency K producing the aliased frequency is known. The sampling rate is generally known; however, the harmonic multiple K must be determined.

From the above expression for Fa, it can be seen that if the sampling frequency is changed by ΔFs, then Fa will change by ΔFa as follows.

$$\Delta Fa = K x \Delta Fs$$

Thus,

$$K = \frac{\Delta Fa}{\Delta Fs}$$

therefore:

$$Fa = \left| \frac{\Delta Fa}{\Delta Fs} \ x \ Fs - Fi \right|$$

Further, since Fa is small in relation to blade vibration frequencies (Fi)

$$\left| \frac{\Delta Fa}{\Delta Fs} \ x \ Fs - Fi \right| \approx 0$$

Thus,

$$Fi \approx \frac{\Delta Fa}{\Delta Fs} \ x \ Fs$$

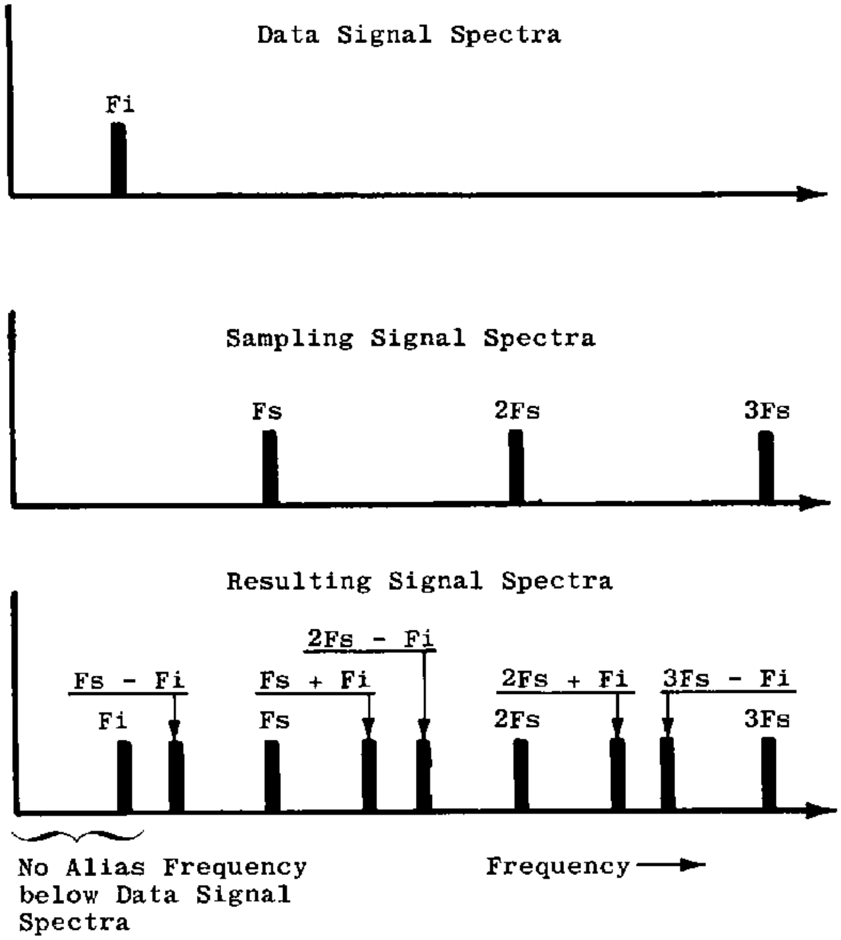


Figure A-1. Example of sampling data signal at more than two times per cycle.

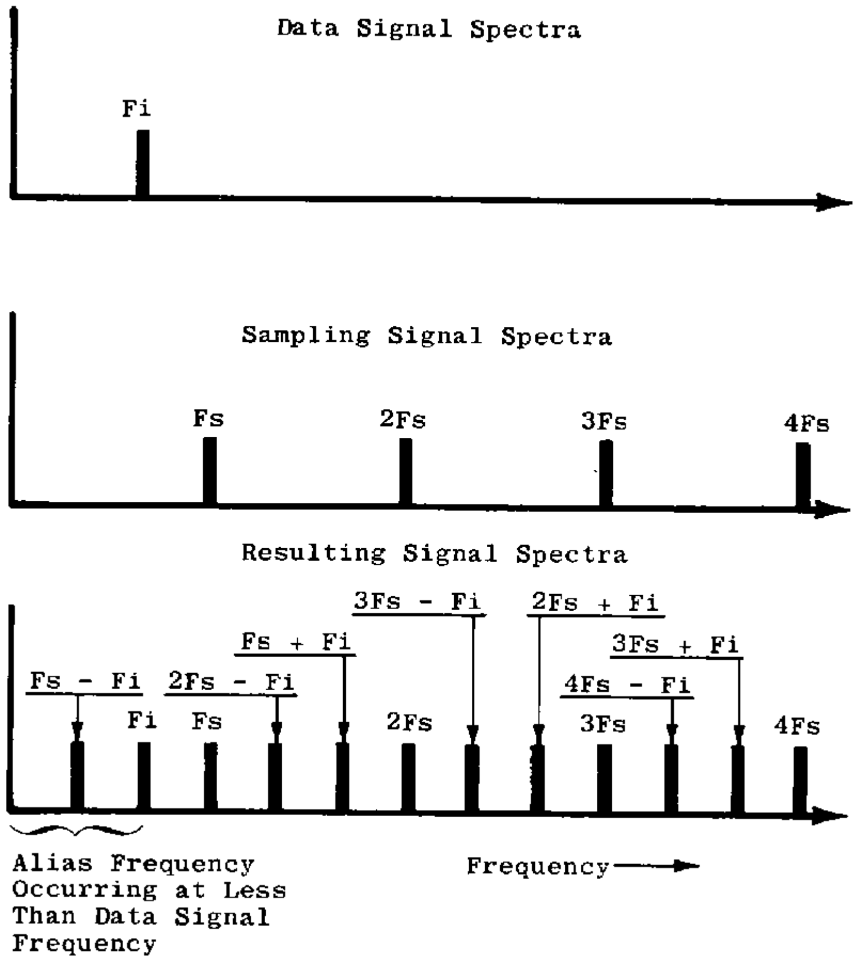


Figure A-2. Example of sampling data signal at less than two times per cycle.