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Noninterference Technique for Measurement of Turbine Engine Compressor Blade Stress

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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. The noninterference technique uses blade-tip deflection measurements and special data-processing algorithms to infer local blade stress. A prototype of the noninterference technique equipped with a nonintegral blade vibration data-processing algorithm has been experimentally validated. The validation consisted of comparing the test results of the prototype noninterference system with those of a conventional strain-gage blade stress data system during an aeromechanical test of a turbine engine. Direct comparisons were made of amplitude and spectral results and real-time monitoring capabilities between the prototype noninterference and strain-gage systems for compressor instability and stall conditions. Expansion of the prototype noninterference processing algorithms to include the capability for inferring blade stress from blade vibrations integral to engine speed is planned for the near future. Longer term efforts will identify the necessary criteria for a multistage compressor noninterference stress measurement system for routine support of aeromechanical testing.

Introduction

DYNAMIC stresses on compressor blades during engine operations are measured exclusively with strain-gage systems. The strain-gage system is the only field-proven 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.

A noninterference technique for measuring stress in turbine engine compressor blades is being developed to alleviate disadvantages associated with conventional strain-gage measurement systems. The noninterference technique utilizes blade-tip deflection measurements and special data-processing algorithms to infer local blade stress. A prototype noninterference stress-measuring system has been assembled[1] for inferring blade stress from a single compressor stage from blade vibrations nonintegral to engine speed. (Blade vibrations integral to engine speed will require a more complex algorithm and possibly multiple tip deflection measurement stations.) The prototype system display of stress amplitude and spectral information is identical to conventional strain-gage system displays for the benefit of the stress analyst.

Validation of the prototype system was accomplished during an aeromechanical test of a turbine engine instrumented with both strain-gage and blade-tip deflection sensors. The instrumented engine was provided by the General Electric Company, Evendale, Ohio, and the test was conducted in an Arnold Engineering Development Center (AEDC) altitude test cell. The noninterference system results were compared with primary blade stress data obtained from the strain gages.

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The primary objectives of the validation were 1) to verify the ability of the noninterference system to extract both amplitude and spectral information from blade-tip vibrations, nonintegral to engine speed, using a single-periphery blade-tip deflection-measuring station and 2) to display this information, in a manner identical to strain-gage system displays, for the benefit of the stress analyst's online monitoring of compressor health and analysis of blade vibration characteristics.

A secondary objective was to obtain operational experience with the prototype noninterference system for use in developing criteria for a full-scale (multistage compressor) noninterference system for routine support of aeromechanical tests.

Prototype Noninterference Stress Measurement System

General Description

The noninterference stress measurement system uses blade-tip sensors located on the periphery of the compressor case to sense blade-tip deflections with respect to the blade platform[2-4] as depicted in the conceptual configuration of the noninterference technique in Fig. 1 and the simplified schematic in Fig. 2.

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 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.

Using the blade-tip deflection measurements and special processing algorithms, one can extract the amplitude and the frequency of blade vibrations nonintegral to engine speed and output them to conventional strain-gage-type analysis and display equipment. The noninterference measurement technique theory and a detailed description of the prototype system are contained in Ref. 1.

Functional Description

To aid the stress analyst in transitioning between strain-gage and noninterference stress measurement schemes, stress data analysis and display features of the prototype system

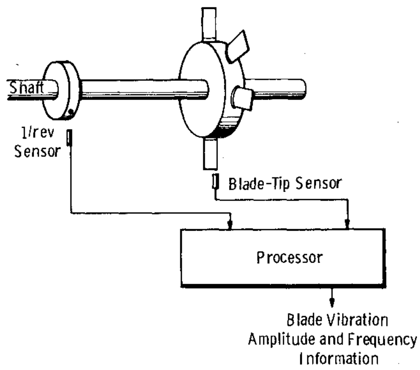


Fig. 1 Conceptual configuration of noninterference stress measurement system.

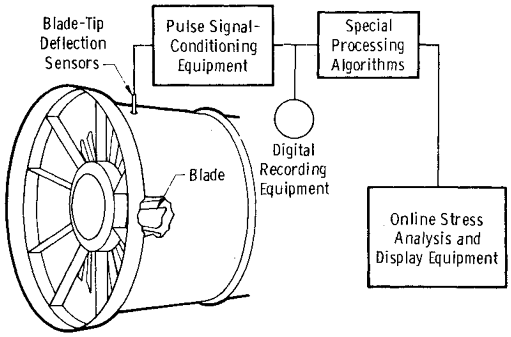


Fig. 2 Simplified schematic of noninterference stress measurement system.

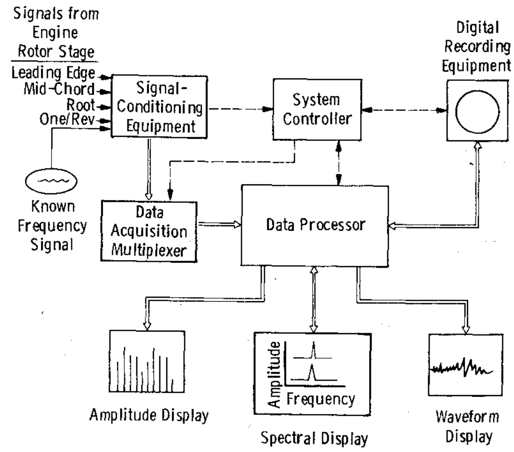


Fig. 3 Functional block diagram of prototype noninterference system.

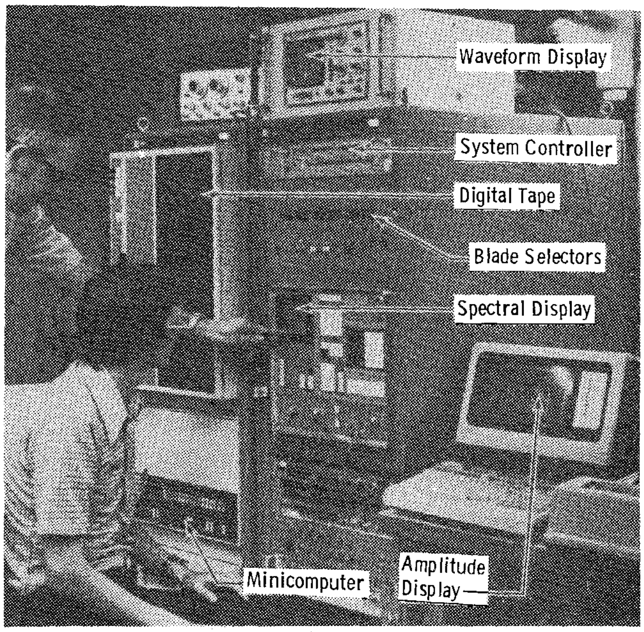


Fig. 4 Photograph of the prototype noninterference system.

were identical to the strain-gage system displays. The display features of the prototype noninterference stress measurement system are shown in the functional block diagram of Fig. 3. A photograph of the prototype system is shown in Fig. 4.

Since all blades on each compressor stage instrumented with a blade-tip sensor are visible to the sensor, individual blades of interest need not be predetermined, as is the case when strain gages are installed. The number of compressor stages and blades viewed simultaneously depends only on the capacity of the noninterference system.

The prototype noninterference stress system provides for real-time monitoring of data during the test and post-test playback and analysis of data. Instantaneous deflections from five blades, selected by the operator, are acquired each revolution of the rotor. The overall deflection amplitudes for all five blades are derived from multiple samples of each blade and are displayed on a CRT bar-graph. Amplitude/time history and spectral information from one blade are displayed on an oscilloscope and spectral analyzer, respectively.

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, the operator can select one blade for identification of vibratory mode. Synchronization information and deflections from all five selected blades are recorded on digital tape and during post-test analysis are displayed just as during real-time operations.

Engine Blade Stress Sensors

One compressor stage of the engine was equipped with six strain-gaged blades and a blade-tip sensor installed to detect the leading edges of the blades. A blade-tip midchord sensor is required to distinguish between torsional and bending components of blade vibrations,[1] but was not installed for this test. Therefore, for this test, stress comparisons were limited to blade bending modes only.

Blade-Tip Sensor

The blade-tip sensor was provided by the General Electric Co. and consisted of a light source, fiber-optic cables, an optic lens, and a photodetector as shown functionally in Fig. 5. A light beam is directed at the compressor blade row, and as the polished blade tips pass beneath the sensor, the reflected light is detected.

A one/rev signal for synchronizing the processor and the compressor stage was extracted from a gear attached to the rotor. Blade platform (root) reference signals were synthesized from this one/rev signal. Details of this synthesizing technique are included in Ref. 3. The tip sensor output pulse, in conjunction with the blade platform reference signal, is used to determine instantaneous blade deflections each revolution of the rotor.

As in the case for the strain-gage system, prior knowledge of blade characteristics is required to correlate blade-tip deflections with the maximum stress on the blade. Blade-tip deflection vs stress must be obtained from bench tests of representative blades in much the same manner as is currently done for strain-gage measurements.

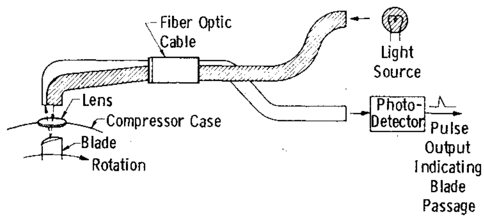


Fig. 5 Functional schematic of the blade-tip sensor.

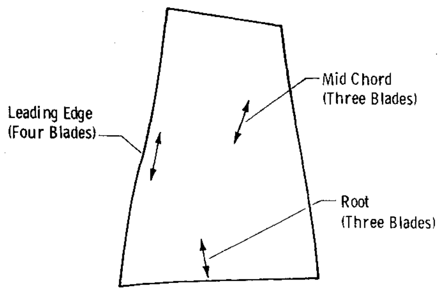


Fig. 6 Typical strain-gage locations on rotor blades.

Strain-Gage Sensors

A total of six blades on the second stage of the compressor were instrumented with two and three gages per blade. The location and orientation of the gages on the blades are shown in Fig. 6. All strain-gage data presented in this article are from the gages installed on the blade leading edges. The strain-gage leads were routed through a slip-ring assembly to the AEDC strain-gage conditioning, recording, display, and analysis system.

Test Installation

The engine was installled in the AEDC Propulsion Development Test Cell (J-2). The noninterference system (Fig. 4) was located approximately 100 ft from the engine and in the same room with the AEDC strain-gage online display and analysis equipment. The blade-tip sensor signal was routed through a line driver amplifier, located near the engine, to the prototype noninterference system.

The strain-gage signals exited the engine through a slip-ring assembly and were connected to the AEDC strain-gage system shown in Fig. 7.

Test Procedure

The test procedure consisted of establishing the desired simulated environment and engine operating conditions and then inducing an engine compressor instability or stall condition by controlling the engine exhaust nozzle area. During the instability or stall condition, both the strain-gage and prototype noninterference systems were monitored to detect the level of blade vibration activity. Data were recorded, and selected data channels were analyzed during these engine conditions from both strain-gage and noninterference systems. Comparisons of monitoring and analysis results were made between systems during the test, and post-test analysis results from representative engine conditions.

During one engine instability condition, the compressor speed was intentionally changed to verify the vibratory mode identification technique. This technique involves sampling a known frequency signal at the same rate as the blade vibrations and comparing resulting frequency spectra. Slight changes in rotor speed (sampling rate) yield the vibratory mode as described in detail in Ref. 1.

It was desired to obtain vibration amplitude and frequency distributions of all blades during an engine instability using

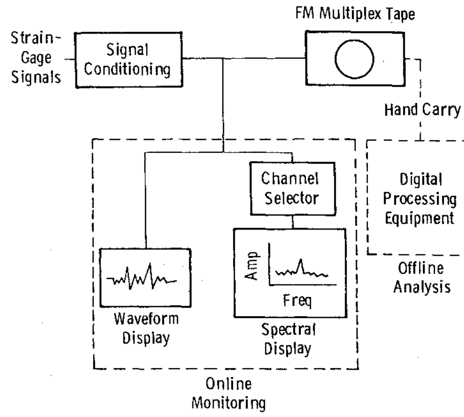


Fig. 7 Block diagram of strain-gage system.

the noninterference system; however, concern for compressor health did not permit maintaining the blade vibration condition for the approximate 1 min required to sequence through all 46 blades, five at a time. Typically, five of the blades were sampled during any given condition.

Test Results

The stress information output of the prototype noninterference measurement system was equivalent to the strain-gage system for blade vibrations nonintegral to engine speed. Both amplitude and spectral (vibration mode identification) information were satisfactorily extracted from blade-tip deflections using a single-tip measurement station on the compressor periphery. The character of the induced blade vibration condition was readily detectable from the noninterference system vibration amplitude/time displays. Real-time display and analysis of tip deflection data suitable for monitoring of engine health during blade stress boundary conditions were demonstrated.

Retention of identical display and analysis formats between the noninterference and strain-gage systems proved to be extremely beneficial. Immediate comparisons of the outputs of the two systems were possible during the blade stress condition.

The pulse output from the optical blade-tip sensor deteriorated only slightly during engine normal operations. On two occasions, however, severe core compressor stalls caused the lens of the sensor to be coated with an opaque material requiring the sensor to be removed and the lens wiped clean. Eight of the ten strain gages on the second stage of the compressor failed during the test program.

Blade Vibration Frequency Identification

The prototype noninterference system method for determination of blade vibration frequency requires a variation or change in rotor speed.[1] The speed change is equivalent to a blade deflection sampling rate change since each blade is sampled once per revolution. The speed change can be inherent variations in engine speed (approximately 0.5%) during operations or an intentional speed change. During the validation tests, inherent compressor speed variations were insufficient; therefore intentional speed changes were necessary.

An example of using the noninterference technique with an intentional change in rotor speed of approximately 50 rpm to identify blade vibration frequency is shown in Fig. 8. The spectrum obtained from the tip deflections of one of the rotor blades, with the spectral analyzer operated in a "peak hold" mode, is presented in Fig. 8a. The spectra of the reference frequency when set to 332 Hz, the predicted mode 1 blade vibration frequency; and 689 Hz, the predicted mode 2 blade

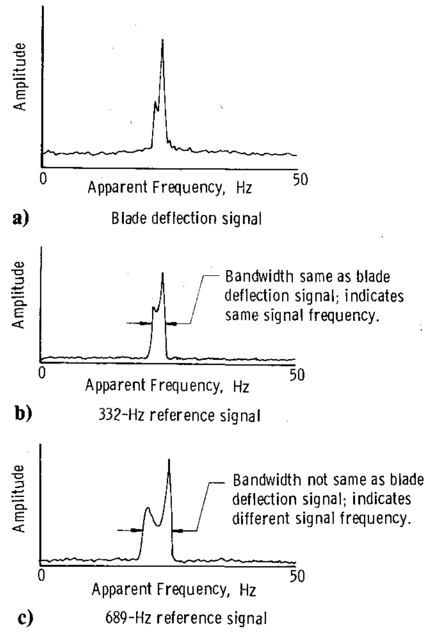


Fig. 8 Identification of blade vibration frequency.

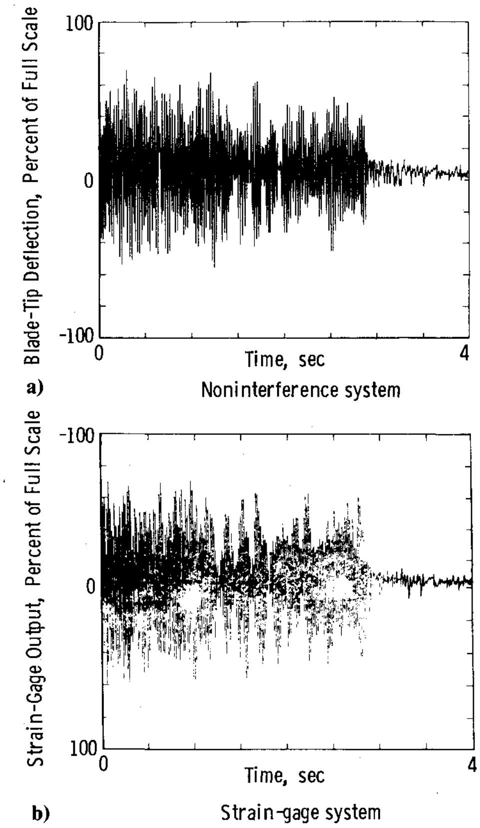


Fig. 9 Comparison of vibration amplitude/time characteristics during compressor instability.

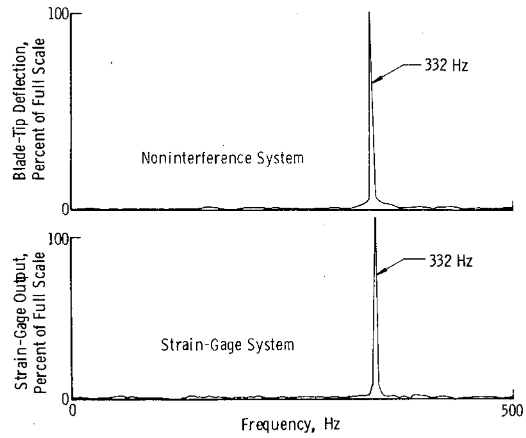


Fig. 10 Comparison of vibration amplitude/frequency characteristics during compressor instability.

vibration frequency, are presented in Figs. 8b and 8c, respectively. Spectra of the blade deflections and the reference signal frequency of 332 Hz are the same and thus indicate that the blade deflections are, in fact, mode 1 vibrations. The spectrum of the reference signal frequency of 689 Hz has a much wider bandwidth than the blade deflection signal, indicating that the blade vibrations are at a much lower frequency than 689 Hz and thus are not mode 2 vibrations. Conventional spectral analysis of the strain-gage data verified the blade was vibrating at 332 Hz (mode 1).

The blade vibration frequency can be identified using the prototype noninterference system within approximately 60 s when intentional rotor speed changes are made. When the rotor speed has inherent variations on the order of 0.5%, then the time to determine blade vibration frequency is estimated to be 20 s.

Comparison of Noninterference and Strain-Gage System Blade Vibration Measurement Results

Comparisons of strain-gage and noninterference measurement results were made to assess the ability of the noninterference system to characterize blade vibrations. Both amplitude/time history and spectral results are compared during compressor blade instabilities. Only amplitude/time histories are compared during compressor stalls as spectral information is normally not required for these events.

The onset of blade vibrations caused by compressor instability was readily detectable using the noninterference system display of signal amplitude vs time. Overall signal envelopes, obtained from strain-gage and blade-tip deflections as the compressor entered and then exited instability conditions, were the same (Fig. 9).

Comparable blade vibration spectral results were also obtained during compressor instabilities. A typical result, with both measurement systems indicating blade vibrations occurring at one predominant frequency, 332 Hz, is presented in Fig. 10.

The ability of the noninterference system to detect small tip deflections was checked by comparing results during low-amplitude vibrations which were just barely discernible on the strain-gage system. Typical amplitude/time results are shown in Fig. 11. In this case, the instability vibrations were also evident in the spectral displays of both systems.

The noninterference system was able to resolve closely spaced frequencies as anticipated. Figure 12 compares spectral results obtained from strain gage and the noninterference system when a blade was vibrating at two frequencies spaced approximately 6 Hz apart. The strain-gage processing equipment actually provided less frequency resolution in this case than the noninterference system. The noninterference technique shifts the blade vibrations to relatively low frequencies, thereby taking advantage of improved frequency resolution of spectral analyzers in the lower frequency ranges.

Compressor stall events were readily detectable online by the noninterference system display of blade-tip deflection vs time. These events were distinguishable from blade vibrations caused by compressor instabilities by the abrupt increase in blade deflection amplitudes during stall conditions. A typical result, comparing noninterference and strain-gage systems, is shown in Fig. 13. Both measurement systems indicate an abrupt onset of blade vibrations and rapid amplitude fluc-

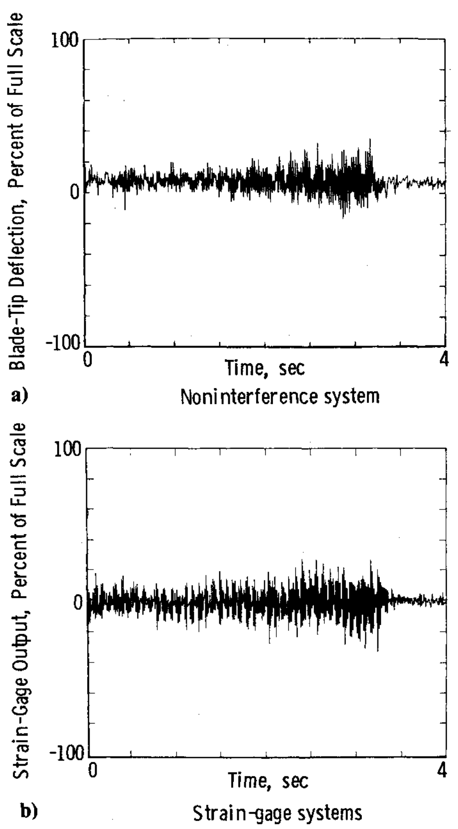


Fig. 11 Comparison of measurement system sensitivities during low-amplitude vibrations.

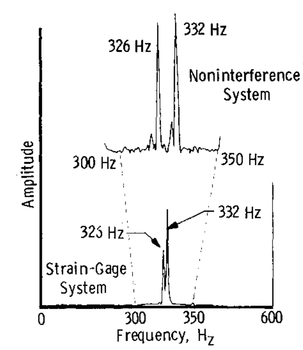


Fig. 12 Frequency resolution capability of the noninterference system.

tuations until the compressor recovers from the stall condition.

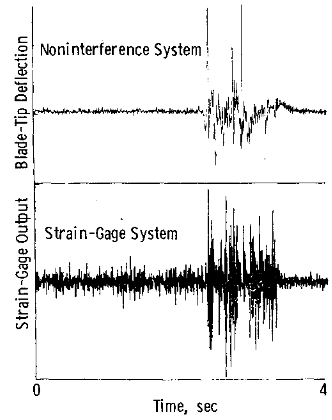
Comparison of Noninterference and Strain-Gage Online Blade Vibration Health-Monitoring Results

Presentations of stress data from the noninterference system are identical in data format and content to those typically used with strain-gage systems. Therefore the online monitoring features are the same. Displays provide information concerning overall blade deflection amplitudes, blade vibration frequencies, and deflection amplitude/time histories.

Overall vibrational amplitudes of five blades are displayed on a CRT in bar-graph form; typical results obtained prior to and during a blade instability condition are presented in Fig. 14. The bar-graph information was updated at approximately 2-s intervals and was used primarily to identify the most vibrationally active blades.

As is typical with strain-gage systems, spectral content of the vibrations from one selected blade could be viewed online. Additionally, instantaneous deflections from one blade could be displayed on an oscilloscope with the display updated each revolution of the rotor.

Fig. 13 Comparison of vibration amplitude/time characteristics during compressor stall.



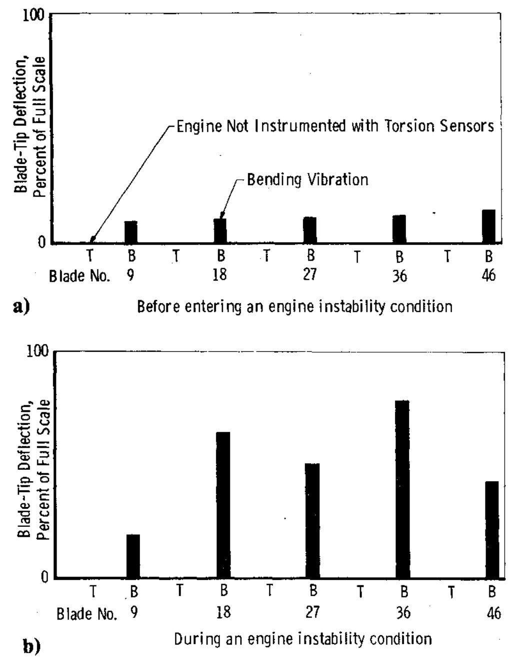


Fig. 14 Typical bar-graph display of blade vibrational amplitudes immediately before and during compressor instability.

As was the case with strain gages, the oscilloscope display proved to be the most effective real-time monitoring display for rapid detections of changes in blade vibrational amplitudes. Future noninterference systems, based on results with the prototype system, should contain three to five oscilloscope displays per rotor stage, which is consistent with typical strain-gage display systems.

Conclusions

A noninterference blade stress measurement technique has been validated by comparing noninterference results to strain-gage results during aeromechanical tests of a turbine engine. The validation tests were performed using a prototype noninterference system for blade vibrations nonintegral to engine speed and sized for a single compressor stage.

The prototype test results indicate that the noninterference technique is now a viable alternate to strain-gage systems for many engine stress test applications including compressor instability and stall investigations. Further development is needed, however, to achieve the capability for measuring blade vibrations which are integral with respect to engine speed. This additional capability would then extend the applications for the noninterference blade stress measurement technique to include assessment of compressor blade vibration characteristics over the engine flight envelope.

Acknowledgments

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