Detection and Study of Compressor-blade Vibration

A device for measuring blade vibration—without contact—is described

by Rudolph Hohenberg

ABSTRACT—The many compressor blades in modern axially staged gas turbines are susceptible to vibration. The energy exciting the vibrations comes from many sources which are discussed. The vibratory stresses generated must be experimentally determined. A new device capable of detecting rotor-blade vibration without contact is described.

Introduction

Most modern gas-turbine engines use a large number of axially staged compressor blades to efficiently provide the high-volume, high-pressure air required for the combustion process. These blades are shaped like airplane wings—they have air-foil cross sections and are cantilevered beams. They rotate at high speeds. They are designed to perform an aerodynamic function, to work on the air to increase its density. The design must also consider the forces acting on the blades; the thoroughly known quasi-static forces due to centrifugal force and pressure loading, and also the unpredictable dynamic forces. Since the dynamic forces acting on the compressor blades are difficult to predict, they are studied experimentally. Before a rational experimental study can be made, it is useful to understand why the blades vibrate and the diverse sources of vibratory energy that will try to rattle the blades around.

Compressor-blade Vibration

Compressor-blade vibration is generally at a natural frequency. The blades can be excited in a variety of vibratory modes.

The first bending mode (Fig. 1 shows a model compressor blade in this mode) is of primary importance.

The first torsional mode (Fig. 2 is an end view of

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Paper was presented at 1966 SESA Spring Meeting held in Detroit, Mich., on May 4–6.

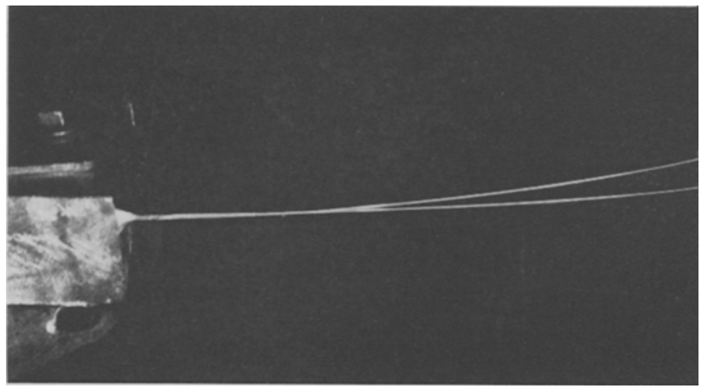


Fig. 1—Compressor-blade model in the first bending mode

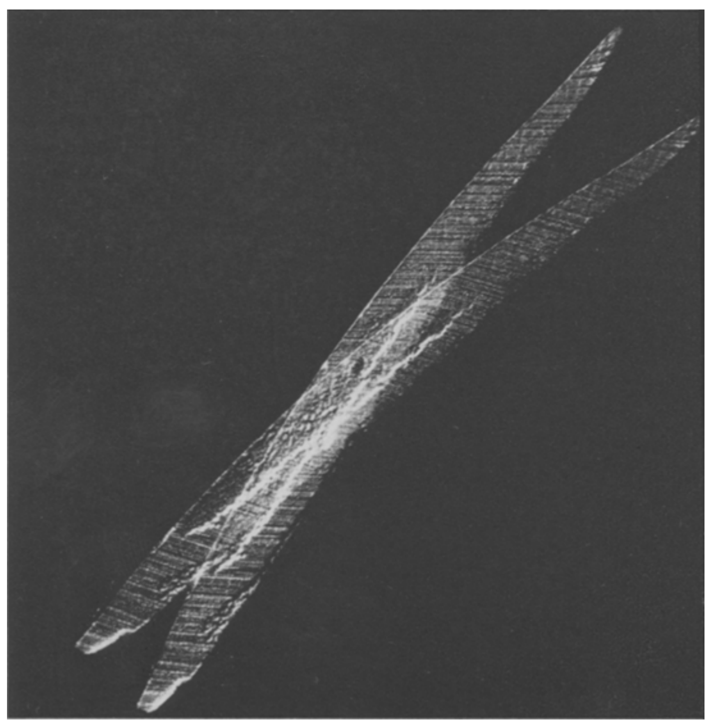


Fig. 2—The first torsion mode, end view

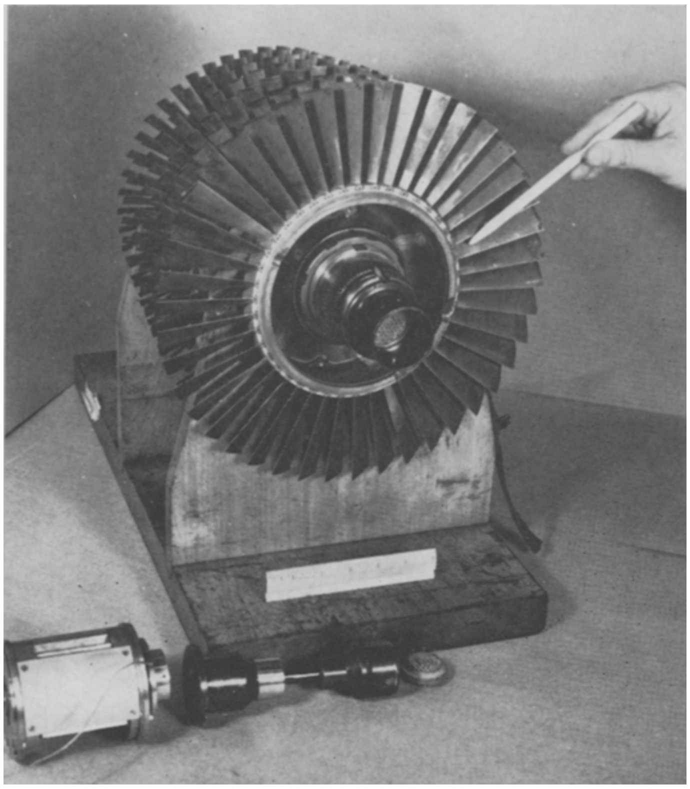


Fig. 3—Compressor rotor, instrumented with strain gages, with slip rings and drive shaft

a compressor blade in this mode) is sometimes significant.

Significant stresses in the second bending mode have rarely been experienced and are increasingly rare in the other still higher modes of vibration.

The natural frequency in each mode of the blades, in any one stage, also varies from blade to blade due to geometric differences from blade to blade.

TABLE 1—SOURCES OF VIBRATORY ENERGY

Excitation Sources		Frequency at Integer Multiple of Rotating Speed
In-flow distortion inlet duct struts stators	Local change in velocity or direction of flow is felt by the blade as it rotates as a change in pressure loading	Yes
Rotating stall	Aerodynamic phenomenon, not predictable	No
Surge	Causes shock loads, reduces aerodynamic damping	Yes and No
Flutter	Aero-elastic instability, frequently predictable	No
Bearing vibration	Anti-friction bearing cage unbalance and ball or roller passing	No

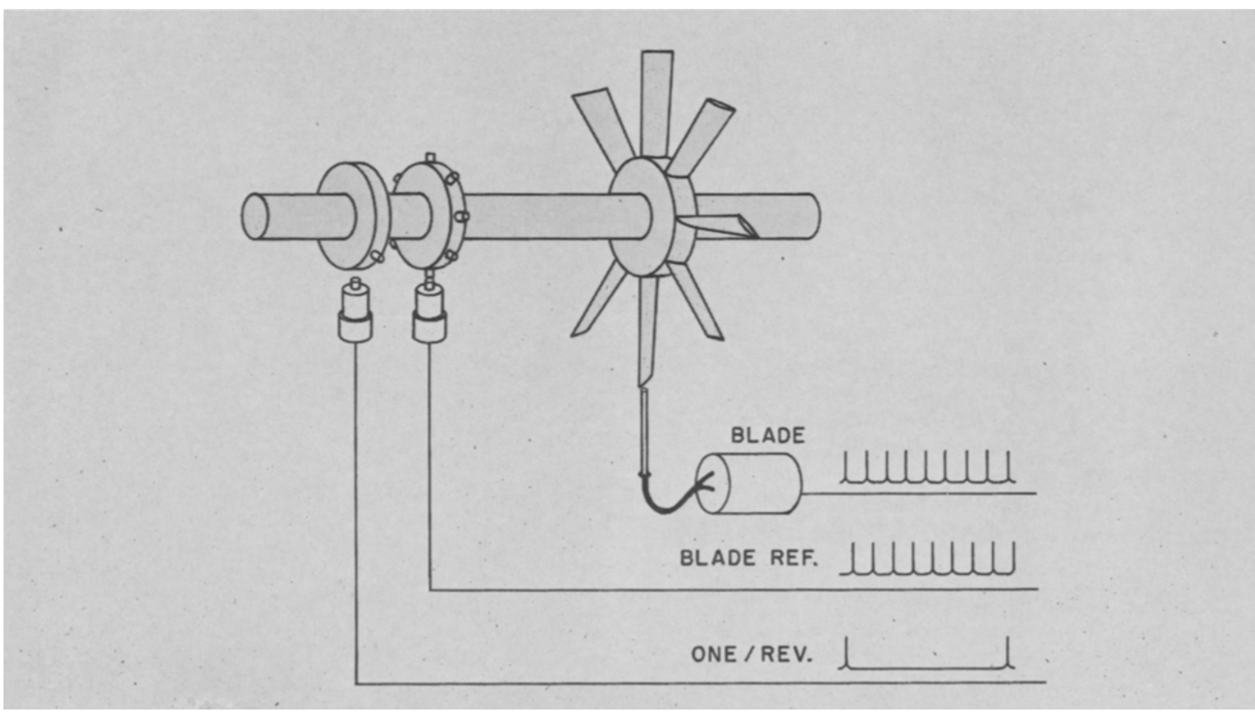


Fig. 4—Signals generated for non-contacting blade-vibration detector

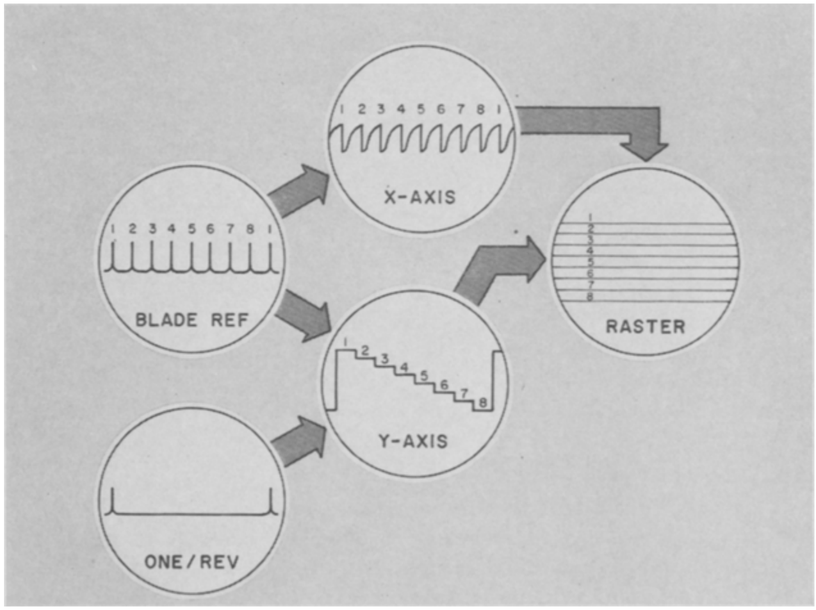


Fig. 5—Generating the frame of reference for the visual display of compressor-blade vibration

Then, too, the same mode of vibration occurs at substantially different frequencies in different stages. Sources of vibratory energy that have excited blades, in the experience of the author, are listed in Table 1.

The magnitude of the vibratory stresses generated when excitation frequency coincides with natural frequency is not analytically determined. Only one compressor blade need fail for immediate catastrophic failure to result. Therefore, measurements of the vibratory stresses felt by the compressor blades are made to predict the life span of a compressor. This is invariably true for aircraft engines because a pilot cannot get out and walk if an engine failure occurs.

Strain-gage Measurements

The most widely used method is to put strain gages on the compressor-rotor blades and connect them to slip rings. Figure 3 illustrates a multistage axially bladed compressor rotor with strain gages on a representative number of blades on each stage. They are connected through a flexible shaft to a 16-channel slip-ring assembly.

The strain-gage installation is exposed to centrifugal force approaching 100,000 g's, temperature as high as 600° F, and the erosive effects of a high-velocity air stream. The electrical noise generated by the slip rings is always a problem; but these problems have become routine in the development of gas-turbine engines and are accepted as a normal hazard of the game. The cost in terms of valuable experimental hardware and time for these necessary experimental studies is disconcerting to the management. This is especially true when a research compressor is first built to obtain aerodynamic information, and the parts are drilled for strain-gage lead wires which obviates later endurance testing. Non-contacting techniques to avoid this wanton destruction of test hardware has long been sought.

Lycoming Blade-vibration Detector

A non-contacting device has been developed at Avco-Lycoming to measure compressor-blade vibration. Figure 4 shows schematically how a pulse train may be generated by the compressor blade tips passing a stationary sensor. The sensor is a light source shining into the compressor through a fiber-optic and getting reflection to a photo diode whenever a compressor-blade tip passes. On a common shaft, a small wheel with rod magnets, equally spaced—one for each blade, generates another pulse train with a magnetic pickup. These pulses, identified as the blade reference pulses, are always in synchronism with the passing of the blade roots. (It is necessary to avoid any torsional looseness or torsional vibration between the blade roots and the wheel.) Still another small wheel with a single rod magnet generates one pulse per revolution to be used as a trigger.

It may be of interest that these 0.045-in. diameter by 0.150-in. long Alnico V magnets are used, instead of the more conventional toothed wheels, to generate sharp pulses which tend to be less dependent on wheel clearance and runout.

These pulse trains are used to compare the phase angle between the compressor-blade tips and compressor-blade roots passing a given nonrotating point. Any change in this phase angle is due to blade deflection and, if presumed to be in the first bending mode, can be expressed as a vibratory-bending stress. To effectively visualize this phase angle on an oscilloscope, a reference pattern is generated by television-circuit techniques, as shown in Fig. 5.

Each pulse of the blade-reference train generates a sawtooth pulse and also a step of a staircase wave-form. Each blade-reference pulse, sawtooth, and step of the staircase are phase coherent. The one-per-revolution pulse is used to reset the counter that makes the staircase wave-form. This provides synchronism between stair level and specific blades. The sawtooth wave-form is impressed on the horizontal axis of a medium persistence oscilloscope. The sweep across the face of the tube is maintained linear with respect to time by selecting a straight portion of the sawtooth. The staircase is applied to the vertical axis to space adjacent lines which represent adjacent compressor blades. The resulting array of horizontal lines is conditioned to represent always the same blade with the same line.

The pulse train generated by passage of the blade tips is now used to intensify a portion of the horizontal sweep lines which comprise the reference pattern (see Fig. 6). An equal time delay may be given each of the blade pulses to make them appear in the middle of the horizontal sweeps. Any variation in the position of the intensified point on the horizontal sweep must be due to a change in phase angle between the compressor-blade tip and the compressor-blade root when passing the blade sensor. This variation can only be due to bending deflection in the blade. The magnitude of the blade deflection is precisely proportional to the deflection of the intensified spot on the horizontal sweep line.

$$\frac{\text{Blade deflection}}{\text{Spot deflection}} = \frac{\text{rpm}}{60} \times \pi \times \frac{\text{Tip diameter}}{\text{Sweep velocity}} \quad (1)$$

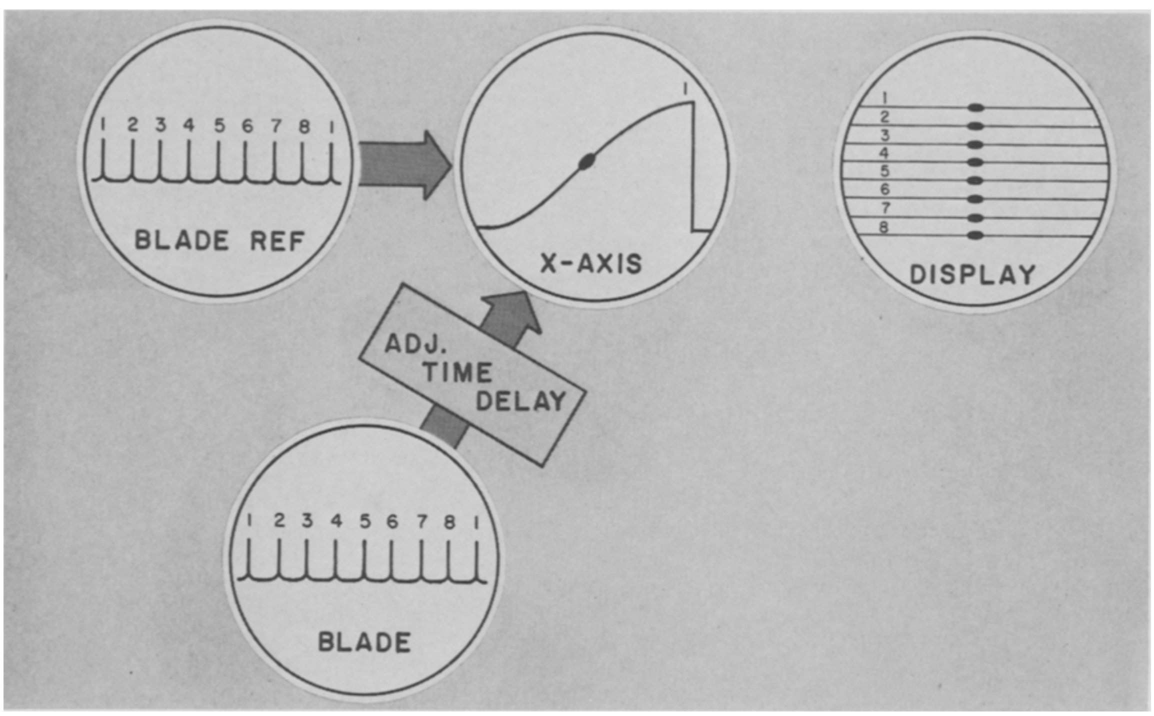


Fig. 6—Display of deflection of individual blades on the visual frame of reference which resembles a television raster

For a known blade geometry, the tip deflection can readily be converted into an equivalent stress—provided the vibratory mode is also known. In bending vibration, this is almost always the first bending mode.

Torsional vibration can be sensed in a like manner by generating the blade pulse train with the leading edge of the blade tips and the blade reference-pulse train with the trailing edge of the blade tips. Fiber-optics are then used to generate both pulse trains. When detecting bending vibration, the fiber-optics generating the blade pulse train should be positioned in line with the center of twist of the blades.

Vibration generated at a frequency which is an integer number times compressor rotation will cause a shift in the position of the intensified spot. A nonsynchronous vibration will cause a widening of the intensified spot on the horizontal line, due to persistence of the oscilloscope fluorescence. Table 1 lists commonly experienced sources of vibratory energy and differentiates those sources that will cause a shift of the spot from those causing a widening of the spot.

Figure 7 shows the optical-blade-tip sensing probe and the magnetic-blade-reference pickup. Figure 8 is the display equipment.

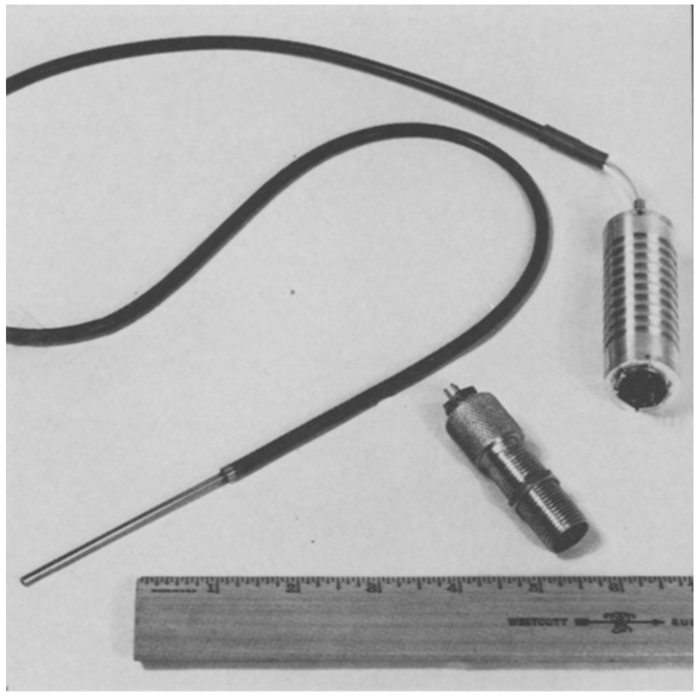


Fig. 7—Sensors used to obtain the pulse signals required for detecting blade vibration

Limitations of the Detector

A limitation, which has already been mentioned, is the inability to differentiate between the first mode in bending or torsion and the higher modes.

Blade vibratory deflection is detected for the blades of a selected compressor stage. Frequency of vibration cannot be measured.

It is possible that a blade vibrating an integer number of times per revolution is in its neutral position when passing the sensor. However, experience has shown that all the blades of a stage have discreetly different resonant frequencies; that is, they are not coupled. It is also known that there is a 180-deg phase shift in going through resonance.

Experience with this device has verified that the blades do not vibrate simultaneously. This problem does not exist when vibration is not an integer number times speed of rotation.

The equipment presently in use at Avco-Lycoming is capable of displaying 10, 12, 14, 16, 20, 24, 28, 32, 40, 48, 56, or 64 blades at any one time. If the actual number of blades in a stage differ from these available numbers, the next lowest number is used. The one-per-revolution trigger pulse assures that the top horizontal sweep on the screen will always be the same blade, that the next line will be the next adjacent blade, and so on. The vibration of almost all of the blades in a stage may be

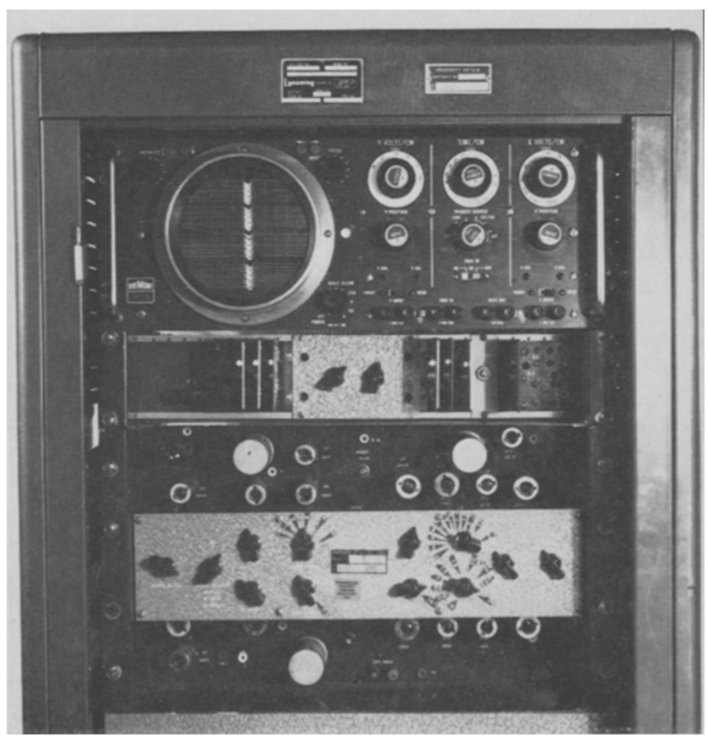
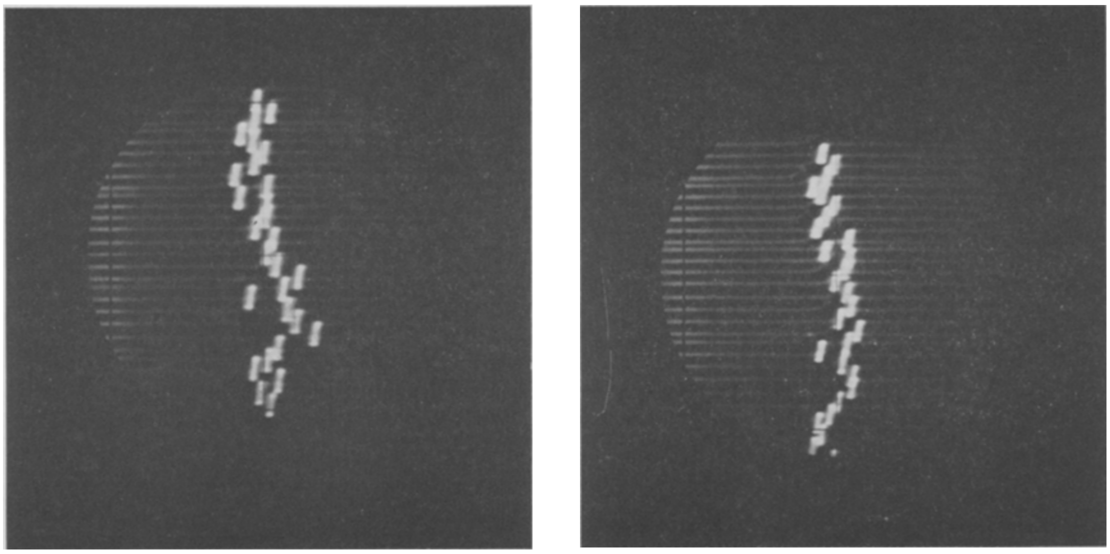


Fig. 8—Electronic equipment used to provide the display of compressor-blade vibration



Pattern with vibration — Pattern without vibration

Fig. 9—Display from a 28-blade compressor storage. Note especially the position of the intensified dots on lines 8, 10, 22 and 26 from the top

sampled—certainly more than the limited samples obtained through slip rings.

The equipment is capable of resolving blade tip deflection of ±.005 in. The measurement of vibratory blade deflection requires knowledge of:

Compressor speed, rpm
Compressor-blade-tip diameter, in.
Horizontal sweep-line velocity, screen divisions/sec
Spot deflection due to blade vibration, screen divisions.

The horizontal sweep-line velocity is obtained by shifting the intensified spot one screen division with the calibrated time-delay circuit—which is also used to put the spots in the middle of the screen. The required calculation of eq (1) can be expedited by use of a nomograph; refer to Fig. 10.

The optimum visualization of blade vibration on the oscilloscope screen is achieved by slowly traversing the desired speed range. Changes in the vertical pattern of intensified spots, caused by blade vibration, is readily apparent. The initial pattern is not a straight vertical line because of the manufacturing tolerances in blade spacing and the timing wheel; refer to Fig. 9.

$$\frac{\text{BLADE DEFLECTION}}{\text{SCREEN UNIT}} = \frac{\text{RPM}}{60}\,\pi\,\text{WHEEL DIA.}\,\frac{10^{-6}\text{ SEC.}}{\text{SCREEN UNITS}}$$

$$\text{EXAMPLE: } \frac{18000\text{ RPM}}{60} \times \pi\ 12\text{ INCHES}\ \frac{10^{-6}\text{ SEC.}}{.2\text{ UNIT}} = \frac{\pm .027\text{ INCHES}}{\text{UNIT}}$$

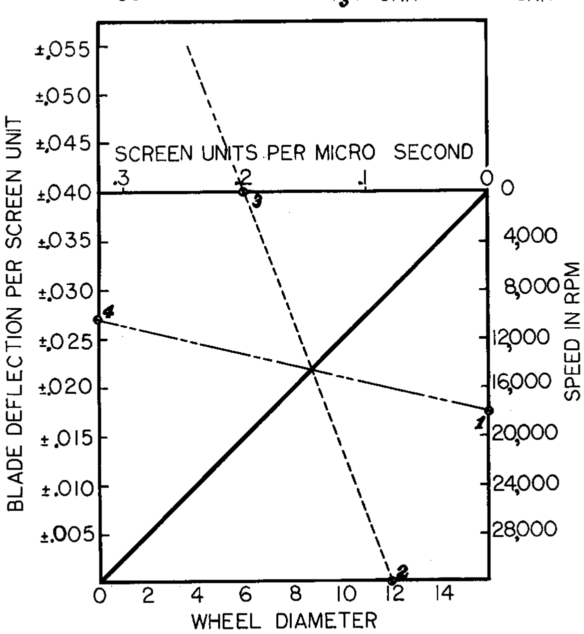


Fig. 10—Nomograph

Conclusion

The blade-vibration detector has found application during the development of research compressors and also in the course of evaluating field-revealed problems. This device will never replace strain gages—but it is unique in supplying data on essentially all of the blades of one stage. Since slip rings or telemetering are not necessary, it makes measurement of blade vibration on multi-spool machines much simpler.