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RESEARCH MEMORANDUM

VIBRATION OF TURBINE BLADES IN A TURBOJET ENGINE DURING OPERATION

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April 22, 1948

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RESEARCH MEMORANDUM

VIBRATION OF TURBINE BLADES IN A TURBOJET ENGINE DURING OPERATION

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SUMMARY

An experimental investigation was conducted to determine the vibration phenomena that occur in the turbine blades of a typical jet-propulsion engine during service operation; high-temperature strain gages were used to measure the turbine-blade vibrations. At turbine speeds within the cruising range, vibratory stresses of appreciable magnitude existed in the blades. Most of the vibrations observed at various turbine speeds occurred in the fundamental bending mode of the blades at a frequency of approximately 1200 cycles per second. Vibrations also occurred in the first torsional mode at a frequency of approximately 2000 cycles per second; and a high-frequency complex-mode vibration was observed within the limits of the cruising range of the engine. The frequencies of the principal vibrations were found to be related to the number of nozzle blades and combustion chambers.

INTRODUCTION

Among the important problems in the stress analysis of jet-propulsion engines is that of determining the vibratory phenomena that affect turbine blades during service operation. Vibrations may cause fatigue rupture of turbine blades. In an NACA investigation of turbine-blade failures, for example, most of the broken blades displayed the characteristic appearance of fatigue fracture, indicating the presence of critical vibrations. The development of the jet-propulsion engine in England was complicated by numerous turbine-blade vibration problems (reference 1). Little data are available, however, on the quantitative magnitudes of vibratory stresses present in typical gas-turbine blades under operating conditions.

Numerous potential sources of blade-vibration excitation are present in gas turbines. Most of the excitations are transmitted to the blades by means of the hot gases constituting the driving medium. Any influence that tends to produce a periodic variation in velocity about the blade tends to produce vibration. Front bearing supports, compressor blades, compressor-casing guide vanes, combustion tubes, and stationary nozzles affect the uniformity of the gas stream before it passes over the rotating blades. Even if the gas flow were perfectly uniform at the time of impulse in the turbine blades, the passage over the blades might induce aerodynamic conditions conducive to vibrations. Although the most serious sources of excitation are probably the combustion tubes and the stationary nozzle blades, no data are available for evaluating the various excitation influences.

An investigation including bench and dynamic tests was conducted at the NACA Cleveland laboratory to determine by means of high-temperature wire-resistance strain gages the actual vibrations existing in the blades of a turbojet engine in service operation and thereby to determine the importance of vibration as well as to evaluate, for the particular blades investigated, the relative order of importance of each of the vibration-excitation sources. The engine used was especially suited for this purpose because it has been under investigation at the NACA to determine related information on the blades, such as typical service life, creep rate, and temperature distribution during various operating conditions. Correlation of vibration frequency and turbine speed permitted a determination of the sources of excitation; the magnitude of the vibrational stress at each speed permitted an evaluation of the relative importance of the various vibrations.

EXPERIMENTAL EQUIPMENT AND PROCEDURE

The blades in the engine used in this investigation are of the unshrouded type with "fir-tree" attachment. Temperature distribution in the blades investigated is available from reference 2. The turbojet engine used is a 14-burner, straight-flow type with a centrifugal compressor.

Modes of vibration. - The nodal patterns in the various natural modes of vibration were determined by preliminary bench tests on a blade similar to those used in the dynamic investigations. In producing these patterns, the blade was inserted in a turbine disk with a tight fit in order to approximate the rigidity in a completely bladed wheel; vibration was excited by a speaker-type unit of variable frequency. In order to avoid mechanical damping or restraint of the blade, the excitation force was transmitted through a steel rod attached to a stub blade adjacent to the test blade in

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the turbine disk. The turbine disk was suspended in a sling above the exciter (fig. 1).

The excitation frequency was varied until resonance was observed by means of a crystal pickup. Location of the nodes was accomplished by moving the needle point of the pickup device along the lines of a grid drawn on the concave face of the blade until an oscilloscope signal indicated a minimum amplitude. Location of a sufficient number of such points was recorded to establish the location of the nodes for each of the resonant frequencies.

The bench tests on the blades showed that the restraint attributable to strain gages was negligible insofar as change in natural frequency was concerned.

<u>Instrument installation.</u> - High-temperature wire-resistance strain gages were used to obtain data from the turbine blades during service operation. The construction and the mounting of the strain gages were similar to those of the multiple-loop type described in reference 3. The strain-sensitive wire was a platinum-iridium alloy; Sauereisen No. 32 cement was used as the mounting medium. After the strain gage had been baked on a turbine blade (reference 3), the blade was placed in a high-temperature oven and slowly heated to 1700° F to stabilize the strain-sensitive characteristics of the strain gage and to improve the bonding of the cement. The strain gages were mounted near the blade bases along the trailing edges on the convex sides. This location was selected because it was satisfactory for vibration measurements and also afforded optimum temperature conditions for strain-gage life.

The instrumented blades were inserted in the turbine wheel, the lead wires cemented to the rear face of the turbine rotor, and the lead-wire cement baked by radiation from infrared lamps. The lead wires were connected to a circular terminal block at the center of the turbine wheel (fig. 2).

Several components of the engine were modified in order to provide passage of the lead wires from the strain gages to the slip rings necessary for transmittal of strain-gage signals to stationary observation and recording instrumentation. The principal modifications consisted of axial passages bored through the turbine wheel, the compressor, and the shafts connecting these components. A hollow auxiliary shaft, connected to the compressor, extended through the accessory housing on the forward end of the turbojet engine to provide a connection to the slip-ring unit on the accessory housing (fig. 3). A magnetic revolution counter, which also appears in figure 3, was used in order to provide a more accurate determination of turbine speed than could be obtained with the standard turbojet tachometer.

The slip-ring unit was of the cylindrical type with all rings of the same diameter. The strain-gage signals were transmitted through brushes mounted at an angle to the radii in the plane of each slip ring. The slip rings were monel metal and the brushes were silver graphite.

Two Wheatstone-bridge circuits, which permitted signals to be taken from two strain gages at the same time, were used. Removal of the exhaust cone of the outlet ducting permitted the terminal-plate connections to be so changed as to connect any two of the strain gages installed on the blades into the circuits. The three inactive arms of each Wheatstone-bridge circuit were strain gages mounted on a dynamically strain-free rotating part of the slip-ring assembly. This precaution (reference 3) served to minimize slip-ring interference effects.

The strain signals were transmitted to instrumentation consisting of amplifiers, oscilloscopes for visual study of the signals, a recording oscillograph, and a variable-frequency signal generator for determining vibration frequency. The oscillograph simultaneously recorded the two strain-gage signals, timing marks indicative of turbine speed, and another set of timing marks showing the oscillograph film speed.

The procedure consisted in operating the engine over the entire range of turbine speed at exhaust temperatures intended to duplicate service conditions. The speed ranged from idling (4000 rpm) to full turbine speed (11,500 rpm). As the speed was slowly increased, the strain-gage signals were under constant observation. At the appearance of signals indicative of vibration, the turbine speed was held constant and the signals were recorded. At the same time, the variable-frequency signal generator was used to determine the frequency of vibration.

RESULTS AND DISCUSSION

Modes of Vibration

Nodal patterns were obtained on a blade of the same type as those instrumented with strain gages to determine the approximate frequencies of resonant vibration of the blades. The nodal patterns of 10 vibration modes that were excited by means of the

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speaker-type exciter are shown in figure 4. The fundamental bending mode, not shown in the figure, occurred at a frequency of 1270 cycles per second. A point of interest was the similarity of the nodal pattern at 8500 cycles per second to that at 8800 cycles per second, a similarity apparently caused by the simultaneous occurrence of two modes mechanically coupled. Although similarity often exists among vibration nodal patterns at very high frequencies, the node patterns at 9450, 9625, and 10,700 cycles per second are dissimilar.

Centrifugal Stresses

The centrifugal stresses in the blade at different speeds of operation were first computed in order to assist in the interpretation of the significance of the vibratory stresses. The results are shown in figure 5. The centrifugal stresses at various locations along the length of the blade are shown for turbine speeds of 10,000, 11,000, and 11,500 rpm. Bending stresses due to gas loading are not included in these calculations because the stress values are to be used only qualitatively. Also plotted in this figure are the allowable stress values at each location along the blade, based upon the temperature distribution in the blade (reference 2) and the stress-rupture values for Vitallium at each temperature for durations of 100 and 1000 hours. For speeds below 11,000 rpm, a margin of safety exists between the operating centrifugal stress and the stress-rupture values. At or above 11,000 rpm, little margin of safety exists at any location in the blade and even small vibratory stresses may be sufficient to precipitate fatigue failure.

Vibration of Turbine Blades during Operation

An analysis of the vibration data obtained on two of the blades is shown in figure 6. These blades were standard turbine blades mounted diametrically opposite. Each of the data points represents an engine speed at which vibration was observed and the frequency of the blade vibration. Also shown are a series of solid lines labeled order. Each of these lines represents the locus of points at which the frequency in cycles per second is a definite multiple of the frequency of turbine rotation in revolutions per minute. Such lines therefore define, at any engine speed, the frequency of any exciting force that occurs at a definite multiple of engine speed. For example, because there are 14 combustion chambers in this engine, an excitation due to the velocity profile of these combustion chambers would occur 14 times for every engine revolution. The frequency of combustion-chamber excitation at any engine speed is therefore represented by the line for fourteenth order. A correlation between the observed points of blade vibration and the lines of multiple order is useful in ascertaining the sources of vibration excitation.

In addition to those points of sustained vibration shown in figure 6, several transient vibrations in the first bending mode at 1200 cycles per second were recorded near full-speed operation of the engine. These vibrations were recorded on the oscillograph film but were not visually identified on the oscilloscope screen. The duration of any of these vibrations was short, approximately 1/50 second; the cause of the vibrations was not ascertained. These vibrations could not be neglected, however, because of their occurrence at the speeds producing the greatest severity of temperature and centrifugal-stress conditions. Another point of interest not demonstrated in figure 6 was the absence of significant vibrations during a run in which the engine was accelerated from 4000 to 11,500 rpm within a period of 15 seconds while the oscillograph was continuously operated. Apparently sufficient damping existed to prevent the building up of vibration during rapid acceleration.

Inspection of figure 6 shows that the important orders of excitation are 7, 8, 11, 14, and 48. Excitations of the forty-eighth order are to be ascribed to the 48 nozzle blades in the turbine. In blade A, this source of excitation produced a stress range of 5200 pounds per square inch at a speed in the cruising range of the engine, where the temperatures are high and the margin of safety is low. The frequency of this vibration was 8600 cycles per second. Blade B was also excited by the forty-eighth order but at a different frequency, probably because of differences in the blade mounting. This vibration at a frequency of 5900 cycles per second did not occur within the cruising range.

The fourteenth-order excitation can be attributed to the 14 combustion chambers in the engine. This source produced first torsional vibrations at a frequency of 2000 cycles per second with a stress range of 2100 pounds per square inch in blade A and a stress range of 3100 pounds per square inch in blade B. Because the engine speed at which these vibrations occurred was below the cruising range of the engine, sustained vibrations due to this source are not likely and, moreover, they occur when the temperatures and the centrifugal stresses are relatively low so that greater vibrational stress is tolerable. The combustion chambers also produced fundamental bending vibrations at a frequency of 1170 cycles per second in both blades at a speed of approximately 5000 rpm but these vibrations are probably not of great importance because of the low speeds at which they occurred.

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Vibrations due to eighth-order excitations may be of importance. Although these vibrations in the fundamental bending mode at 1200 cycles per second occur at a speed below the cruising range, their relative amplitude is high. In blade B, the highest vibratory stress range observed of 6800 pounds per square inch was excited by an eighth-order effect.

The seventh-order excitations, which are important, produced vibrations in the fundamental bending mode at a frequency of 1200 cycles per second. The stress range for both blades produced by this excitation was relatively high (about 5500 lb/sq in.) and occurred at a speed of about 10,300 rpm, which is within the cruising range of the engine and is accompanied by high blade temperatures and centrifugal stresses. Several possibilities exist as the source of this order of excitation. One of the more probable causes may be that inequalities of mass flow from the various combustion chambers produce numerous harmonics of engine speed and the lowest order that can excite blade vibrations at any speed within the operating range of the turbine is the seventh. Another possibility is that the response of the blade to excitation is nonlinear and would therefore tend to introduce a one-half-order excitation of the combustion chambers (reference 4).

SUMMARY OF RESULTS

Through the use of high-temperature wire-resistance strain gages, vibratory phenomena in the blades of a typical gas-turbine engine have been observed and evaluated in terms of modes, frequencies, stress range, and probable sources of excitation. Most of the vibrations observed at various turbine speeds occurred in the fundamental bending mode of the blades at a frequency of approximately 1200 cycles per second. Vibrations also occurred in the first torsional mode at a frequency of approximately 2000 cycles per second; and a high-frequency complex-mode vibration was observed within the limits of the cruising range of the engine. The frequencies of the principal vibrations were found to be related to the number of nozzle blades and combustion chambers.

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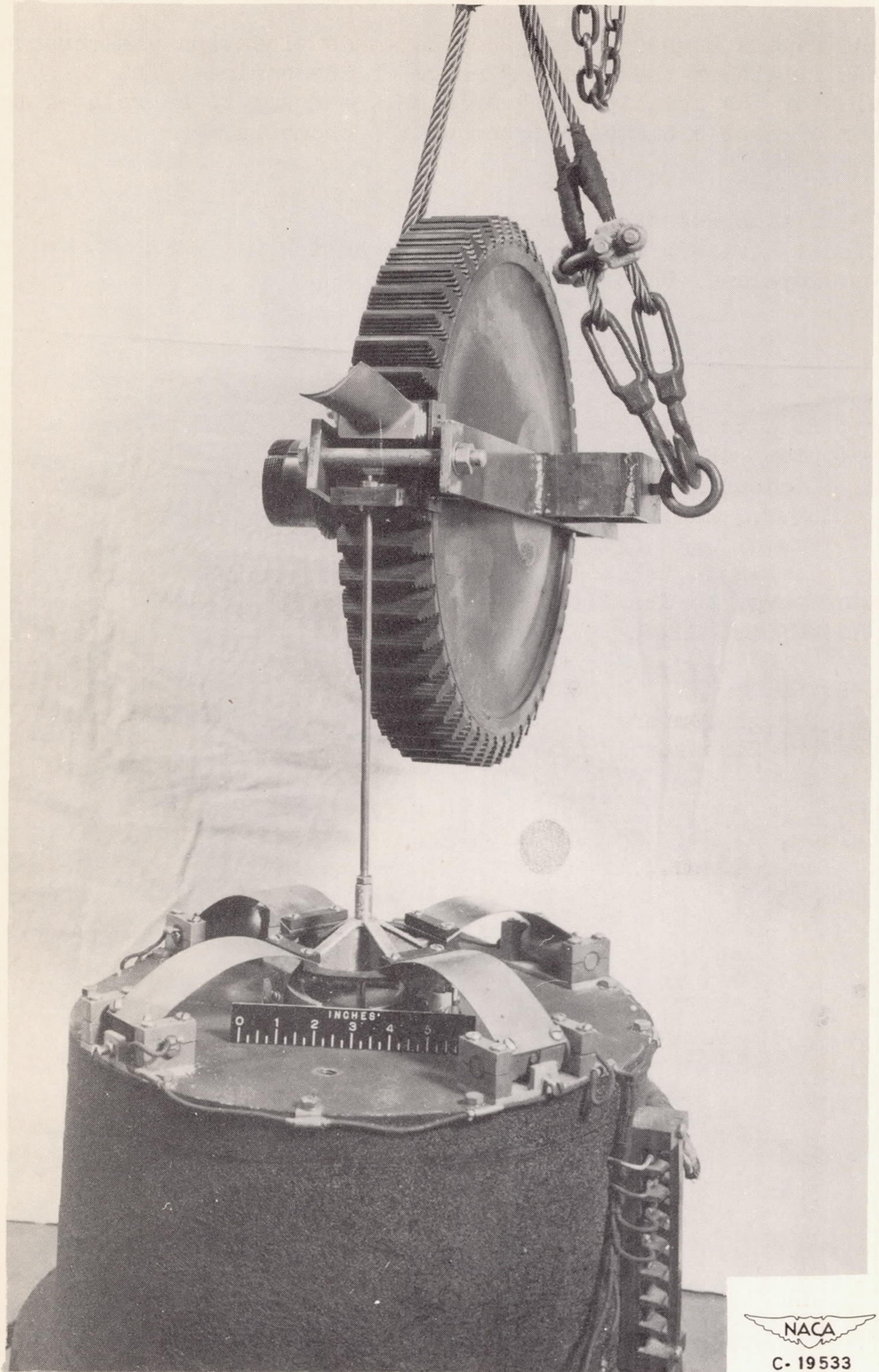

Figure 1. - Excitation equipment used in determination of nodes in turbine blade during various natural vibration modes.

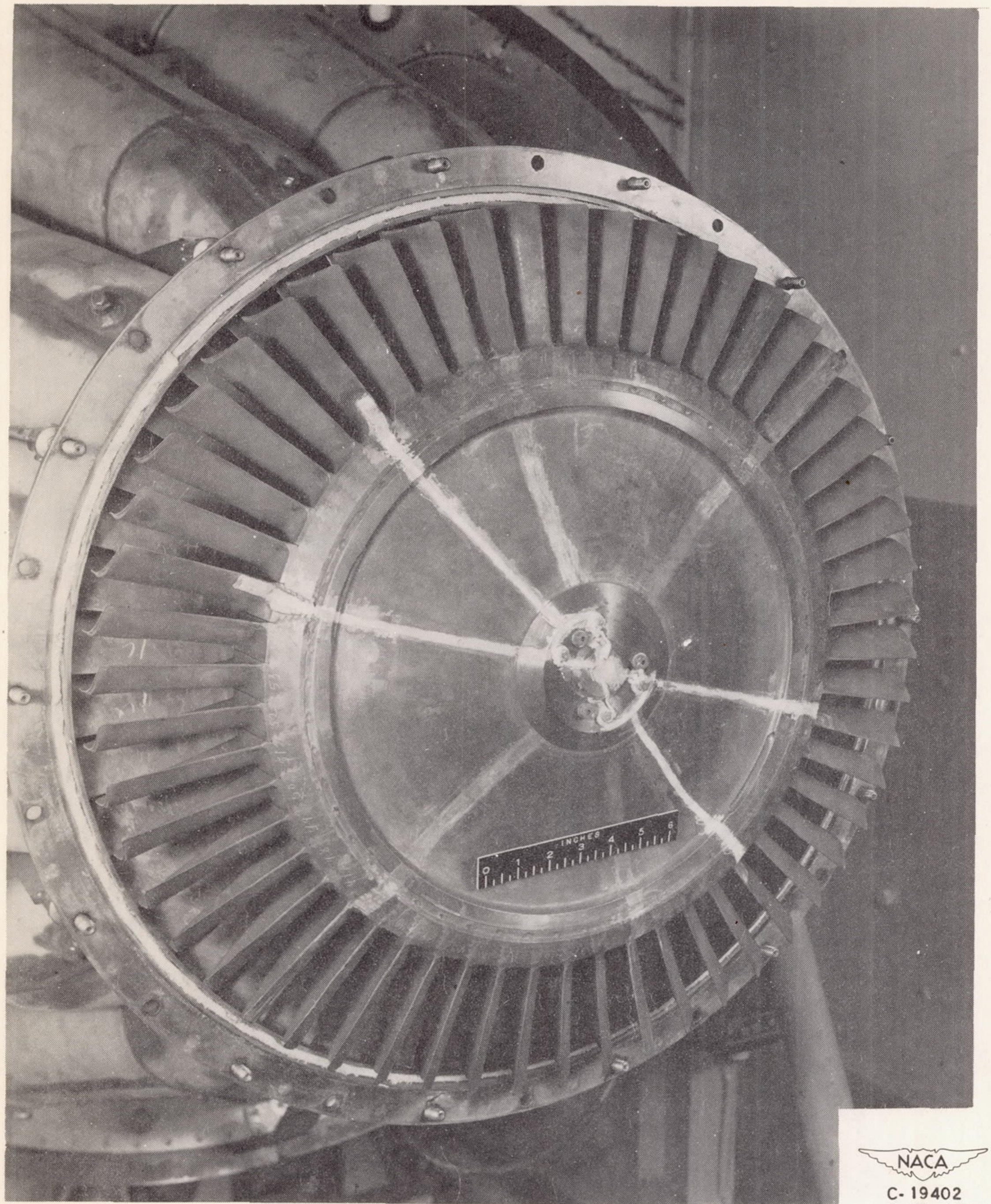

Figure 2. - Installation of high-temperature strain gages on turbine blades showing location of blades and lead wires.

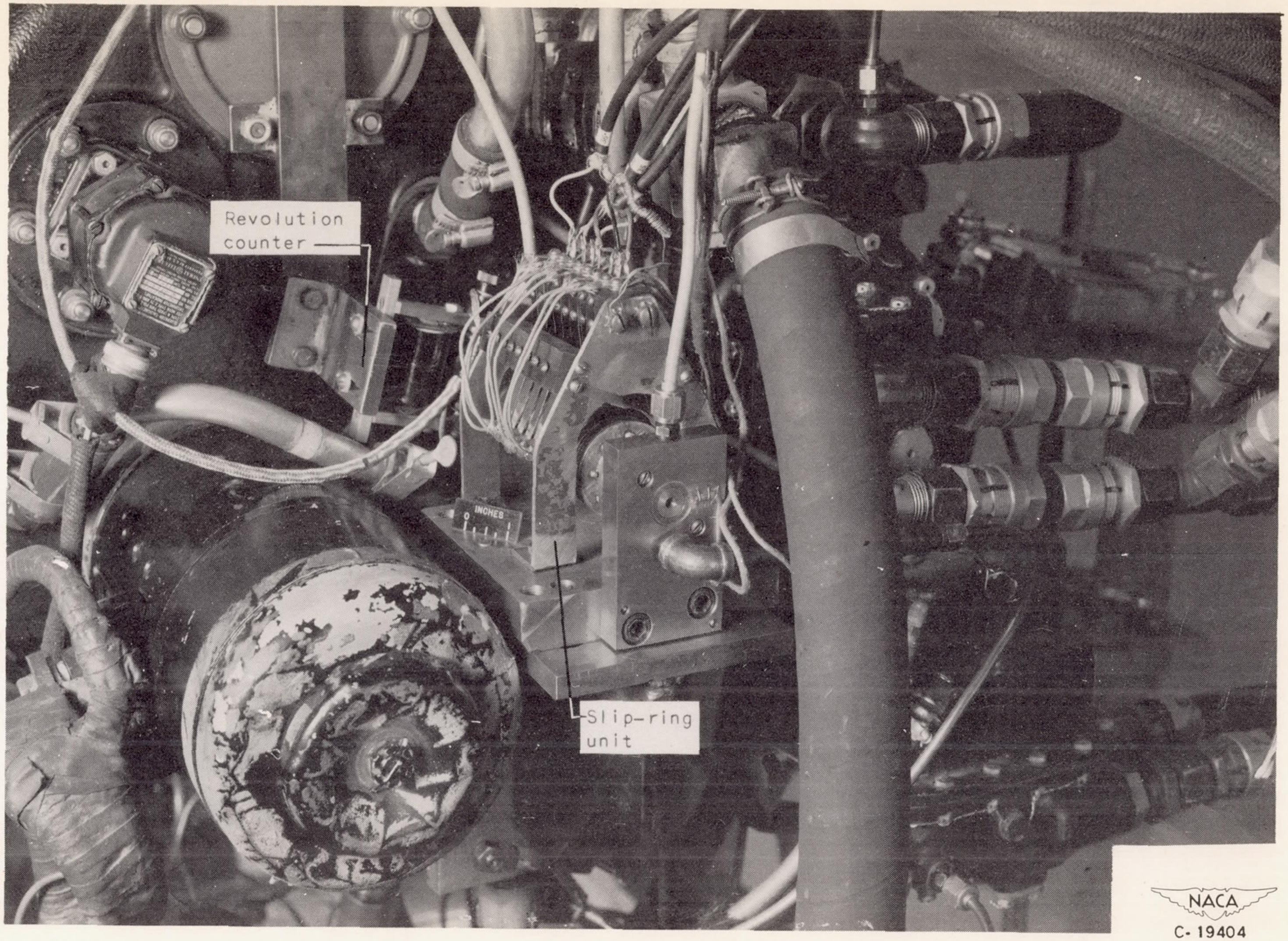

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Figure 3. - Slip-ring unit for transmittal of turbine-blade strain-gage signal to stationary instrumentation.

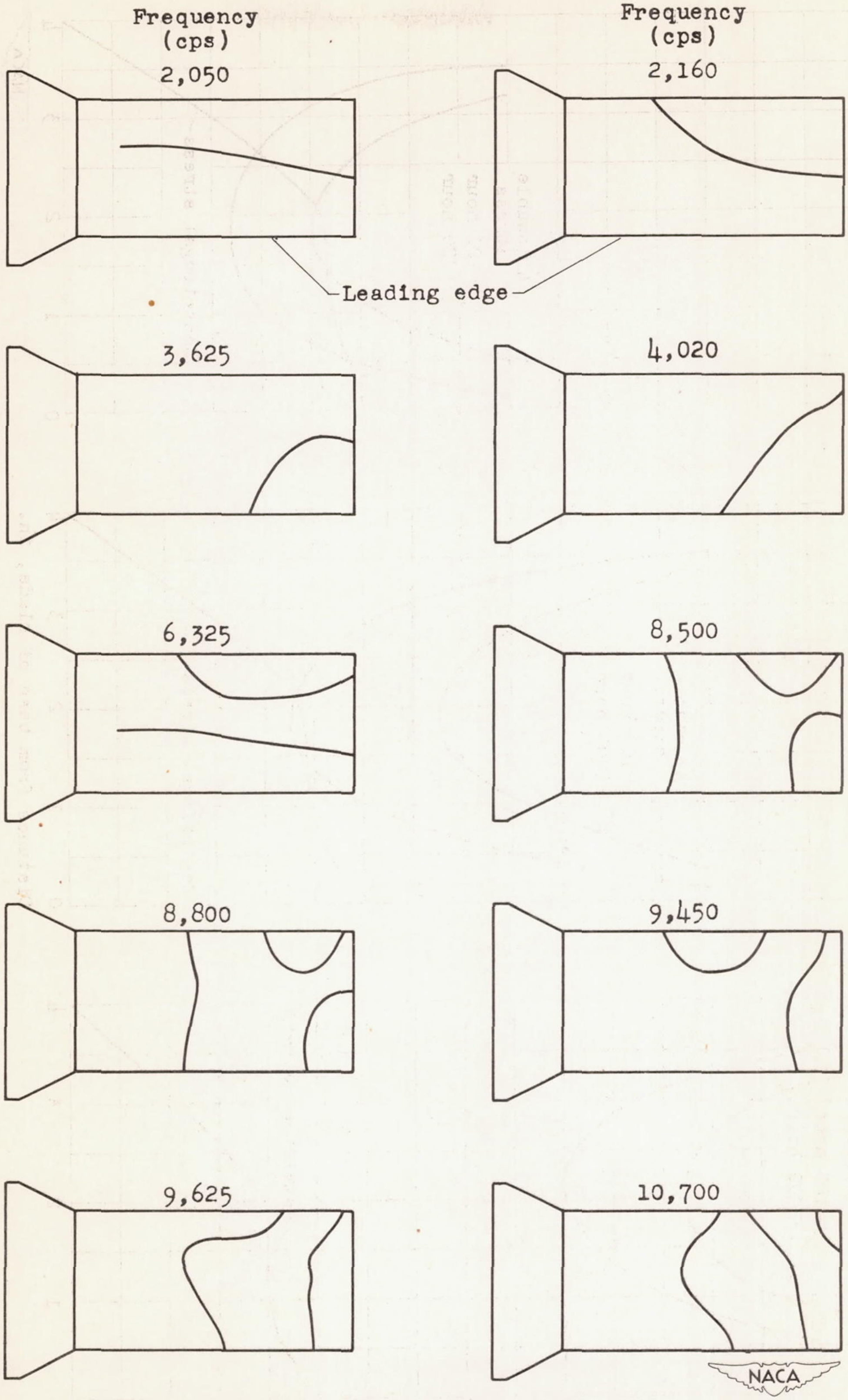
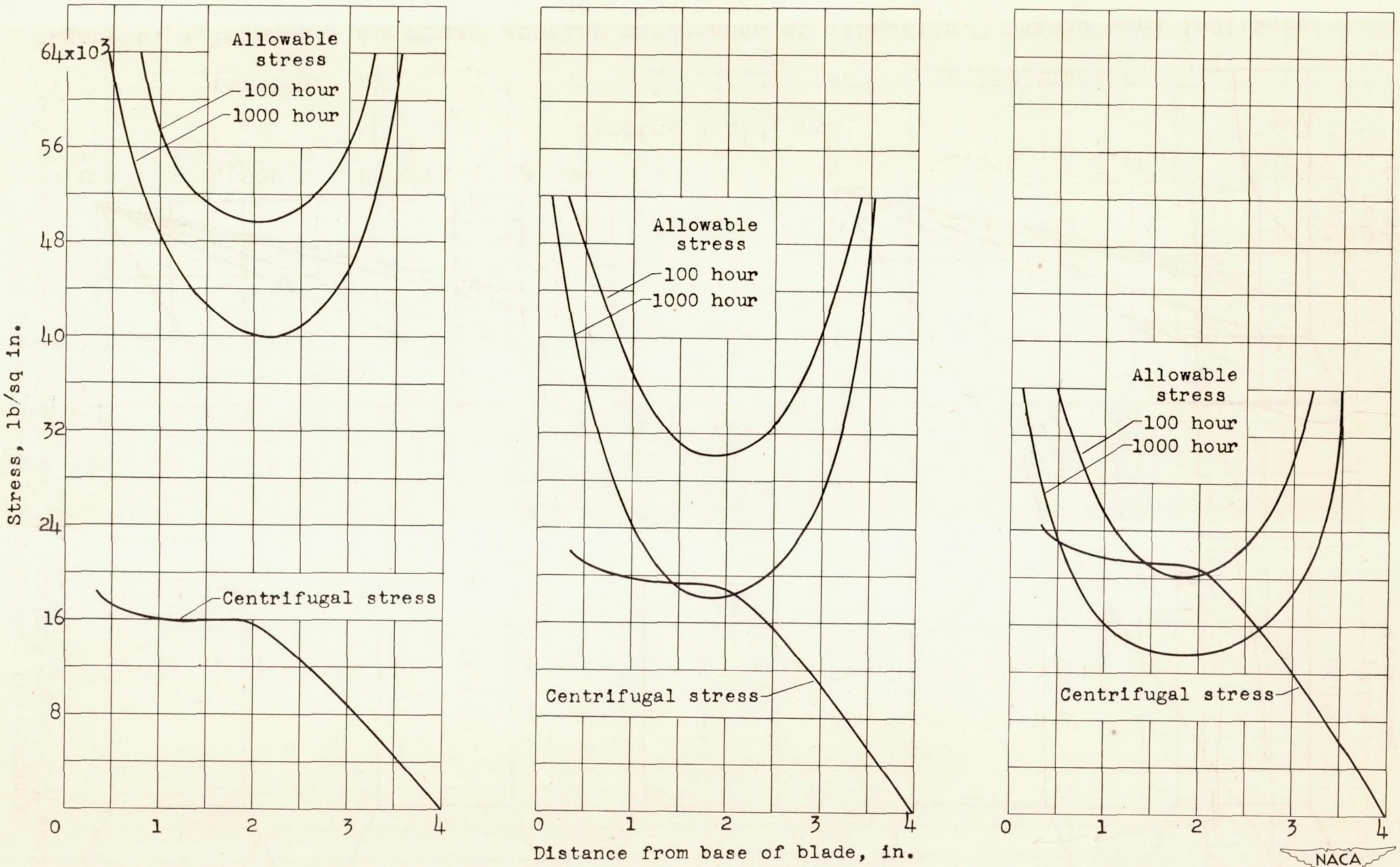
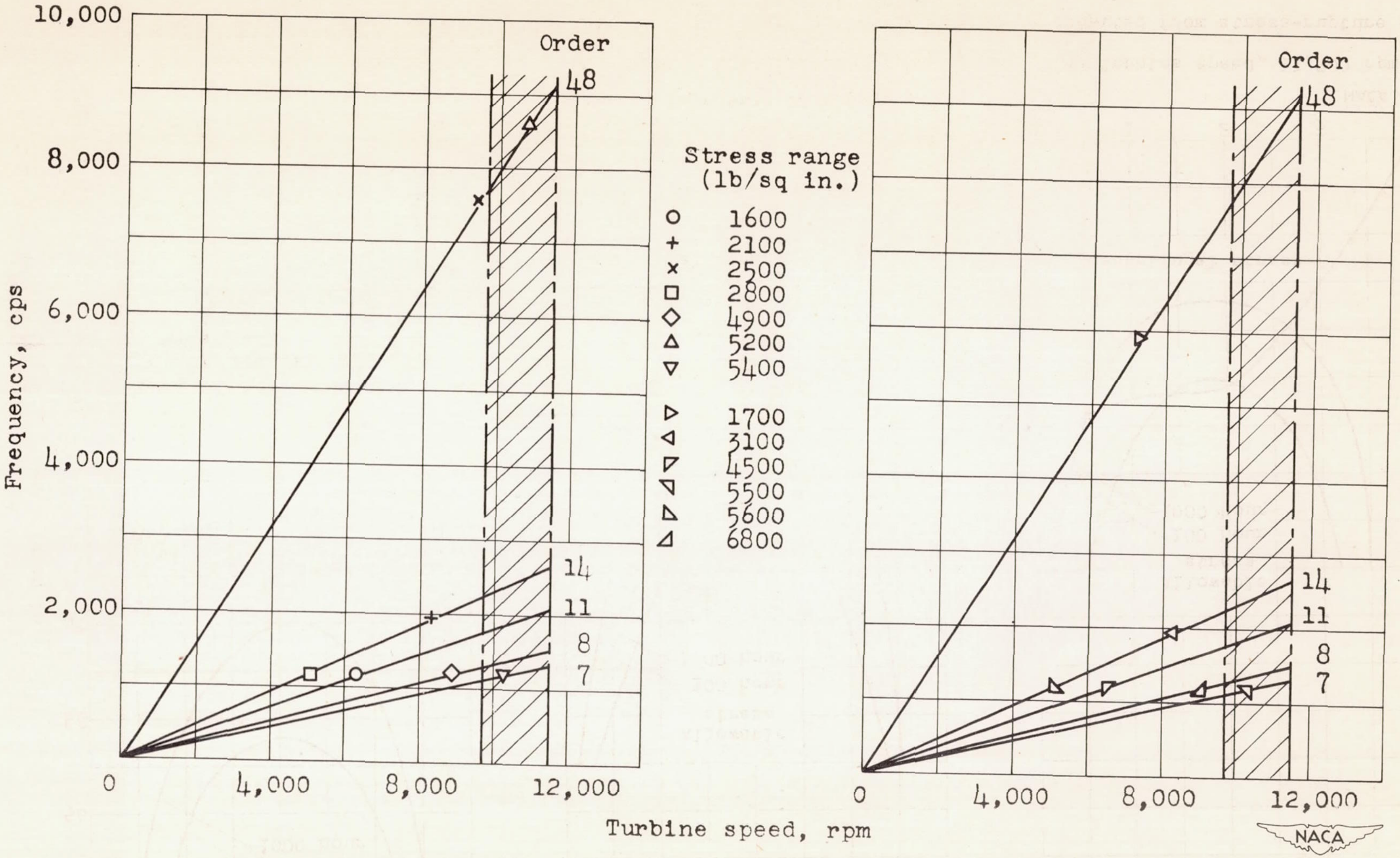


Figure 4. - Nodal patterns of 10 vibration modes determined. Fundamental frequency, 1270 cycles per second.



(a) Turbine speed, 10,000 rpm. (b) Turbine speed, 11,000 rpm. (c) Turbine speed, 11,500 rpm.

Figure 5. - Comparison between calculated centrifugal stress and allowable stress as computed from stress-rupture data for turbine blade at several turbine speeds.



(a) Blade A.

(B) Blade B.

Figure 6. - Resonance spectrums showing occurrence of vibration. Shaded area indicates cruising range.