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NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

TECHNICAL NOTE 4100

VIBRATION SURVEY OF FOUR REPRESENTATIVE TYPES OF AIR-COOLED TURBINE BLADES

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SUMMARY

An investigation was conducted in a turbojet engine, mounted in a sea-level test stand, to determine the vibrational characteristics of four representative types of air-cooled turbine blades. Two of the types were standard-span (4 in.) blades for the engine used in this investigation, and the other two types were long-span ($6\frac{1}{4}$ in.) blades for high mass-flow turbines. Static fatigue tests were also conducted with these blades. The blades were all brazed assemblies of cast and sheet-metal components. The vibratory stresses were measured with NACA high-temperature strain gages.

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In all four blade types, the first-bending-mode frequencies were increased from 2 to 3 percent by a coolant-flow rate of approximately 8 percent. With the standard-span blades, cooling had no noticeable effect on the maximum vibratory stress measured; however, with the long-span blades, the vibratory stresses were reduced at high speeds by introducing cooling air. Also, the long span-blades vibrated at all speeds and the vibratory stresses increased rapidly with speed. The results of the long-span-blade tests may have been influenced by the testing conditions at the tip regions of the blades.

INTRODUCTION

The research on cooled turbine blades has experienced failures of experimental blades in engines (refs. 1 to 4). Some of these failures were attributed to fabrication and materials problems; however, other failures exhibited signs of fatigue caused by blade vibrations. The vibration problems associated with the hollow cooled turbine blades may be significantly different from those for solid uncooled blades. This vibratory stress problem in cooled turbine-rotor blades will become increasingly important as the high Mach number, high mass flow, and low pressure ratio turbojet engine is developed with inlet temperatures ranging from 2000° to 2500° F. To handle the high mass flow, these engines will have low turbine hub-tip radius ratios, and as the turbine blades become longer to permit higher and higher mass flows, blade stiffness will decrease, and the blades will be more susceptible to vibration. Also, the high turbine speeds associated with the use of the transonic compressor result in high centrifugal stresses. The anticipated combination of high levels of centrifugal and vibratory stresses establishes a need to determine the vibratory characteristics of representative air-cooled blades.

If the spanwise temperature and centrifugal stress profiles in the cooled blades are considered, the critical region with regard to stress rupture is approximately one-third span from the base. With engine operation at present inlet temperatures, premature blade failures frequently occurred in the region immediately above the junction of the blade base and airfoil, and not in the critical stress-rupture region. These failures of cooled experimental turbine blades in engines were probably caused by blade vibration. For this reason a program to determine the vibratory characteristics of air-cooled turbine blades was undertaken at the NACA Lewis laboratory. 4791

The investigation reported herein determined the vibrational characteristics of four representative blade types utilized in turbine-cooling research. These blades consisted of an assembly of cast and sheet-metal components which were brazed and welded together in a relatively rigid structure. Two of the blade types were those with spans equal to the standard blades for the engine used in this investigation and could be considered as representative types for engines designed for flight Mach numbers up to approximately 2. The other two types were long-span-type blades which could be considered as representative of blades in high mass flow and low compressor pressure engines for flight Mach numbers up to about 4. All blade types were operated in an engine both with and without cooling air to determine whether cooling airflow affected the vibrational characteristics of the blades. Static fatigue tests were also conducted with the test blades in an interruption-type air exciter. The turbine-blade vibrations for both the engine and static investigations were measured with strain gages mounted on the blade airfoils. In the engine investigation, the strain-gage signals were transmitted from the blades to a recording oscillograph by sliprings mounted at the front of the engine.

APPARATUS AND INSTRUMENTATION

Blades

A blade of each of the four types of air-cooled turbine blades investigated is shown in figure 1. The reference name for each blade is indicated. Table I shows the sketches of the tip cross sections, and gives the details of the materials used for the construction of blades reported herein. A shell-supported blade is one in which the outer shell is a main load-carrying member; a strut-supported blade is a blade in which an internal strut is the main member carrying the airfoil load. A corrugated-insert or tube-filled blade is a shell-supported blade in which corrugations or tubes have been inserted inside the shell to increase the heat-transfer area. The airfoil lengths of the short- or standard-span and the long-span blades were 4 and $6\frac{1}{4}$ inches, respectively. The chord length of the long-span blades was also greater than for the short-span blades. These blades were brazed structures utilizing cast bases and struts (if strut type), and sheet-metal shells and corrugations. All the blades in this investigation were new except the standard-span, tube-filled blades. These blades were used in a previous investigation and their prior running time was unknown.

Turbojet Engine and Installation

The engine investigation was conducted with a turbojet engine operated in a sea-level static test stand. The test engine had a centrifugal compressor and a single-stage turbine modified to supply cooling air to two test blades located 180° apart. The cooling air was supplied to the two test blades from the rear face of the turbine wheel through a special tailcone assembly shown in figure 2. The details of the modification of the turbine rotor and tailcone are given in reference 5.

The tailcone modifications reported in reference 5 were used only with the standard-span (4-in.) blades (standard production length for blades in the engine of this investigation). Additional modifications of the turbine shroud were made to accommodate the long-span ($6\frac{1}{4}$ in.) blades. Figure 3 schematically shows the special shroud that was used for this part of the investigation. The "nontest" blades were the same for the testing of all four types of blades. Therefore, the long-span blades extended radially approximately $2\frac{1}{4}$ inches beyond the tip of the other turbine blades. These portions of the test blades were partially shielded from the high-velocity working gas stream. The time and expense of providing a test facility in which a rotor, fully bladed with the long-span blades, could be tested was regarded as prohibitive. Therefore, the tailcone modifications described herein were utilized. The possible effect of this arrangement on the test results is discussed later.

Methods of Detecting and Measuring Vibratory Stresses

High temperature strain gages (fabricated by the Lewis laboratory and described in ref. 6) were installed on the test blades and used to measure the vibratory stresses and frequencies. Static laboratory tests were conducted to determine the optimum location for the strain gage. As a result of these tests, a strain gage was mounted on each blade at the trailing edge just above the pressure-face fillet. This location was not necessarily the region of maximum stress, but the region where the strain gages (1/4 in. wide and 3/8 in. long) produced the maximum signal for the first-bending-mode vibration. The maximum stress would be in the fillet region because of structural stress concentrations arising from the attachment of the shell to the root. The strain gage was used as the active part of a Wheatstone bridge circuit. The strain-gage signals were conducted from the engine through a slipring assembly (ref. 7) mounted on the front of the engine and then to the recording oscillograph. An electronic speed counter was used to keep an instantaneous check on the engine speed at all times.

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Static Fatigue Tests

The same test blades from the engine investigation were used for static fatigue tests. Commercial strain gages mounted at the same location as for the engine investigation were used to determine the frequency and to control the stress at predetermined levels. The blades were excited by an interruption-type air exciter such as described in reference 6. This type of exciter has a rotating disk that interrupts a high pressure stream of air directed at the tip of the test blade. This disk has 18 holes equally spaced around the disk periphery with the centers of the holes coinciding with the centerline of the airstream. Therefore, by controlling the speed of the disk, the excitation frequency of the force exerted by the airstream could be controlled to predetermined values.

PROCEDURE

Engine Investigation

Two instrumentated test blades of a given type were installed into the turbine rotor. The engine was then accelerated slowly from idle, 4000 rpm, to rated speed (table I) for the blades tested. Rated speed for the standard span blades was 11,500 rpm (rated speed for the engine used in this investigation), and rated speed for the long-span blades was that speed at which the centrifugal stress was equal to the design stress. Whenever a peak vibration was observed during this slow acceleration, the engine speed was adjusted for the maximum vibratory stress and the signal was recorded. The vibrational characteristics of the

blades were first determined over the complete speed range with maximum cooling airflow for this investigation. That is, the coolant-flow conditions were set so that, at rated speed for the test blades, the coolant flow rate per blade was approximately 0.08 of the hot gas flow between two adjacent turbine blades. After the data had been obtained with coolant-flow conditions, the engine speed was reduced to idle, the cooling air supply was shut off, and the test procedure repeated. This procedure was followed because the probability of a premature blade failure was much greater with air-cooled blades when they are operated without coolant
4791 flow. With the recording oscillograph and electronic speed counter, the data at each speed were obtained rapidly in order to keep the actual engine running time to a minimum.

Static Investigation

Commercial strain gages were mounted on the test blades after the engine investigation, and the blades were installed one at a time in the interruption-type air exciter. With an air-supply control and the strain gages with necessary instruments, the vibratory stress was maintained at the desired level. The excitation frequency was maintained at approximately the same value as was measured in the engine investigation by controlling the speed of the interruption-type air exciter.

ENGINE INVESTIGATION

The magnitude and the excitation frequency of the vibratory forces in turbojet engines vary considerably even between engines of the same model. However, the data presented herein from a specific engine will indicate some of the vibrational problems and characteristics of air-cooled turbine blades. For this investigation the blades were operated with the extreme conditions of maximum coolant-flow ratio of about 0.08, and no cooling airflow to determine if coolant flow had an appreciable effect on the blade vibration characteristics. As a result of the unusual tip conditions under which the long-span blades were investigated, the results of the standard-span and long-span blades are not directly comparable, and therefore will be discussed separately. Data are presented in plots of frequency and vibratory stress against engine speed for a given blade type (figs. 4 and 6 to 8). The frequency represented is that of the first-bending-mode vibration. The intersection of the data curves and the order of engine speed lines represents that frequency and engine speed at which the engine rotational frequency in cycles per second multiplied by the order of engine speed is equal to the blade-vibration frequency. Vibration occurring at this intersection is referred to as resonant vibration.

Standard-Span Blades (4-In. Span)

Tube-filled shell-supported blade. - Figure 4 presents vibratory frequency and stress data. These data include the number of peaks or critical vibrations, the change in vibration frequency with coolant flow, and the vibratory stress at each critical speed. It can be noted in figure 4(a) that the sixth- and seventh-order vibrations were observed only with coolant flow. Hence, this blade type was more susceptible to vibration with coolant flow than without; that is, more vibrations were observed with coolant flow. Also, with a coolant flow rate of about 8 percent the vibrational frequency was increased approximately $2\frac{1}{2}$ percent. For example, with the eighth-order vibration the frequency was increased from approximately 1035 to 1060 cycles per second. Calculations based on temperature measurements from previous heat-transfer investigations indicated that the frequency increase was due to the increase in the blade-material modulus of elasticity with a decrease in blade-material temperature. The fourteenth-order vibration was probably excited by the presence of 14 combustor cans.

The peak vibratory stresses reported in figure 4(b) varied between ±4000 to ±7500 psi; however, the peak stress measured either with or without coolant flow was approximately the same, indicating that coolant flow had little effect on peak vibratory stress. These peak vibrations occurred over a narrow speed range and the resonant peaks were sharp and often difficult to obtain. Figure 5(a) presents a typical oscillogram showing the resonant-type vibration signal obtained with the tube-filled-type, air-cooled turbine blade.

Corrugated-insert shell-supported blade. - The frequency and stress data are plotted in figure 6. The only vibration that was observed when the corrugated-insert shell-supported blade was operated with coolant flow was the fifth-order vibration. The fifth, sixth, seventh, and eighth orders of vibration were observed when the blade was operated without coolant flow. Therefore, this blade type was more susceptible to vibration when operated without coolant flow, or just the opposite result of that obtained with the tube-filled blade. The vibrational frequency was raised about 3 percent when the blade was operated with coolant flow.

The peak vibratory stresses measured were approximately ±3500 psi (fig. 6(b)) or approximately one-half as high as for the tube-filled blades. These peak vibrations also occurred over a narrow speed range. Also plotted in figure 6(b) are the vibratory stresses measured in another investigation and reported in figure 4 of reference 8. These data were obtained with high-temperature strain gages mounted on solid blades used in the same type engine as this investigation. The vibratory stresses were of approximately the same magnitude as those reported herein for the corrugated-insert shell-supported blade. The characteristic vibration signals for the corrugated-insert, shell-supported and solid blades were the same as for the tube-filled blade shown in figure 5.

Long-Span Blades ($6\frac{1}{4}$-In. Span)

The aerodynamic conditions around the tip region for the long-span test blades were abnormal (Turbojet Engine and Installation section of APPARATUS AND INSTRUMENTATION), and as a result the measured vibratory stresses could have been either decreased or increased by these testing environments. The vibration could have been decreased because the pulsating or exciting forces present in the hot gas stream would be more effective when striking the tip region rather than the lower or base region. During this investigation, the tip region was shielded from the hot gas stream. On the other hand, the vibrations could have been increased because of undesirable flow conditions, such as stall, acting at the tip region in the expanded turbine shroud section. Prior to this investigation, the long-span blades that were run in heat-transfer investigations failed before rated speed was reached.

Strut-supported blade. - The vibration data are presented in figure 7. The blade vibration frequency data in the critical speed diagram (fig. 7(a)) do not all correlate with the order of engine speed lines as did the data for the standard-span blades (figs. 4 and 6). The long-span blades were different from the short-span blades in that they vibrated at all speeds investigated, and not just at critical speeds or the intersection of the measured blade frequency and engine speed order lines. In fact, data could have been obtained at any speed between idle and rated, and for this reason a continuous curve was drawn through the peak stress points. At speeds between the third and fourth engine speed orders, many data points were obtained in order to define the shape of the stress curve. Blade coolant flow increased the blade vibrational frequency approximately 3 percent.

In figure 7(b) the maximum vibratory stress was approximately ±9680 psi at 9250 rpm without coolant flow. At speeds below 8700 rpm, the maximum stresses were measured with coolant flow. Figure 5(b) presents a typical oscillogram showing the characteristic vibration for a long-span strut-supported blade. Close examination of figure 5 shows that the amplitude of the long-span blade vibration is constantly changing. With the standard-span blade the amplitude was relatively constant. In figure 5(b) the stresses vary from approximately ±3280 to ±9680 psi, and the stresses varied in a like manner at all speeds. The excitation of these blades may have been associated with or influenced by the testing environment described in APPARATUS AND INSTRUMENTATION and shown in figure 3. The varying-amplitude-type vibration signal is considered to be associated with a stalling condition in axial-flow compressors. Since the tips of the long-span blades were rotating in a low-flow zone, it is possible that the tips were in a stalled condition and that this affected the character of the blade vibration.

Corrugated-insert shell-supported blades. The vibration data for the corrugated-insert shell-supported blades are reported in figure 8. 4791
The vibration frequency and character of the strain-gage signals were the same for both the strut-supported and corrugated-insert, shell-supported long-span blades. That is, the blades were vibrating at all speeds with an irregular amplitude. With the corrugated-insert, shell-supported blades, the vibration frequency was increased approximately 3 percent with coolant flow.

The maximum vibratory stress of ±12,000 psi (fig. 8(b)) was measured at 9250 rpm without coolant flow. Above 8000 rpm the stresses increased very rapidly with increased engine speed either with or without coolant flow.

STATIC FATIGUE TESTS

In an attempt to establish the relative importance of the vibratory stresses reported herein, static fatigue tests were conducted that utilized the test blades from the engine. An interruption-type air exciter was used for these fatigue tests. No attempt was made to duplicate the effect of centrifugal force, gas loading, or temperature on the fatigue life of the blades. In the engine investigation the blades were vibrating at their first-bending-mode frequencies. These fatigue tests were therefore, also operated at the first bending mode frequencies of the blades. Table II summarizes the data obtained in the fatigue tests conducted with the interruption-type air exciter.

Standard-Span Blades (4 In. Span)

Tube-filled shell-supported blade. - Figure 9 shows a typical tube-blade failure obtained during engine operation and reported in reference 1. Blades 1 and 2 of static investigation reported herein (table II) also exhibited fatigue failures at the same location. With a vibratory stress of ±10,000 psi, blade 2 failed in only $7\frac{1}{2}$ hours. Figure 4 shows that vibratory stresses as high as approximately ±7500 psi were measured during engine operation. A vibratory stress of ±7500 psi may appear to be too low to cause blade failure; however, because of stress concentrations at the fillet, the stress at the point of failure would be many times higher or high enough to cause blade failure.

The results summarized in table II of reference 1 are for the endurance testing of 69 tube-filled blades with various blade heat treatments and types of fillet at the base of the airfoil. A total of 23 of the failures were at the base of the airfoil and were of the fatigue type reported herein with the two static fatigue-tested blades. Of the failures, 15 occurred in less than 10 hours. It was concluded in reference 1 that the vibrational characteristics of the blades were not principal causes of the failures. However, the measurement of the vibratory stresses during engine operation reported herein indicates that the blade vibration could have been the principal cause of failure for two reasons: (1) the type of blade failure was reproduced in the laboratory with a static fatigue test; (2) even when the effects of centrifugal force, gas loading, and temperature were neglected the vibratory stress that would cause failures in $7\frac{1}{2}$ hours was close to that measured in the engine at a critical speed.

Corrugated-insert shell-supported blade. - The fatigue failures of the corrugated-insert blades occurred in the trailing-edge region near the tip (table II). Several blades of this design have been used for heat-transfer investigations in engines and have exhibited airfoil-tip failures similar to fatigue failures reported herein. When the fatigue stress was controlled to a level of only ±5000 psi, the airfoil cracked within 4 hours (table II). As with the tube-filled blades, blade vibration no doubt was a contributing factor to the failure of the air-cooled blades in the engine heat-transfer investigations.

Long-Span Blades

During the fatigue testing of the two types of long-span blades, no failures occurred in the airfoil (table II). All failures occurred in the serrations or root sections. In one phase of the investigation, the blade base was welded in position and the weld cracked. With the fatigue equipment available it was impossible to fatigue the airfoil of the long-span blades. Prior to this vibration investigation, the blades that were run in a heat-transfer investigation in an engine failed at the base of the airfoil before rated speed was reached. The investigation reported herein found that the peak vibratory stresses increased rapidly with engine speed. The rapidly increasing vibratory and centrifugal stresses in combination with the stress concentrations associated with fabrication could have been a major factor in the early failures of the long-span blade in earlier engine tests.

SUMMARY OF RESULTS

The results of an engine investigation to determine the vibration characteristics of two types of standard-span and two types of long-span air-cooled turbine blade can be summarized as follows:

Standard-Span Blade (4-In. Span)

1. A high coolant-flow rate of approximately 8 percent increased the measured first bending mode frequencies by 2 to 3 percent.

2. The vibratory stresses for corrugated-insert, shell-supported blades were approximately the same as for solid uncooled blades and the vibratory stresses for the tube-filled blade were approximately twice as high as for the solid blade.

3. Coolant flow had no noticeable effect on maximum vibratory stress measured.

4. The tube-filled blade had more critical speeds with coolant flow than without coolant flow, and the corrugated-insert shell-supported blade had more critical speeds without coolant flow.

5. All peak vibrations were engine-order vibrations.

6. Failures obtained in static-fatigue tests were similar to the failures (the time to failure and the type of failure) experienced in the engine and, therefore, vibration may have been the cause of failure.

Long-Span Blade ($6\frac{1}{4}$-In. Span)

(The following results may have been influenced by the testing conditions.)

1. The high coolant-flow rate of approximately 8 percent increased the first-bending-mode frequencies by about 3 percent.

2. The long-span blades were vibrating at all speeds. The vibration signals were similar to signals from stalled axial-flow compressor blades in that the vibration amplitude was continuously varying.

3. The vibratory stress increased rapidly with engine speed.

4. The vibratory stresses were reduced with coolant flow at engine speeds above 8000 and 8700 rpm for the corrugated-insert shell-supported and strut-supported blades, respectively.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, January 17, 1958

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TABLE I. - BLADE MATERIAL, RATED SPEEDS, AND CENTRIFUGAL STRESSES

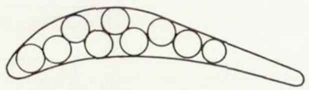
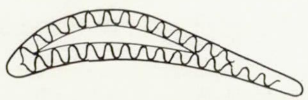
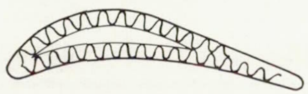
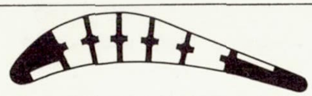
Blade type		Material					Rated speed, rpm	Design centrifugal stress at base, psi
		Base	Strut or tube	Shell	Corrugation	Braze		
Standard-span, tube-filled shell-supported		HS-21	1020	1722A-S		Nicro-braz	11,500	32,000
Standard-span, corrugated-insert shell-supported		HS-21		N-155	A-286	Nicro-braz	11,500	39,950
Long-span, corrugated-insert shell-supported		X-40		A-286	L-605	GE-81	9,550	43,100
Long-span strut-supported		X-40	X-40	L-605		GE-81	9,550	40,900

TABLE II. - FATIGUE TESTS CONDUCTED WITH INTERRUPTED AIR EXCITER

Blade type	Blade	Fatigue stresses, psi	Time to fatigue, hr	Time to fatigue, min	Location of failure
Standard-span tube-filled shell-supported	1	±20,000	1		Just above fillet
	2	±10,000	7	30	Just above fillet
Standard-span corrugated-insert shell-supported	3	±10,000		2	Airfoil cracked at tip near trailing edge
	4	±5,000	4		Airfoil cracked at tip near trailing edge
Long-span corrugated-insert shell-supported	5	±10,000	13	30	Blade serration teeth cracked
	6	±20,000	5	30	Blade serration teeth cracked
Long-span strut-supported	7	±20,000		30	Blade serration teeth cracked

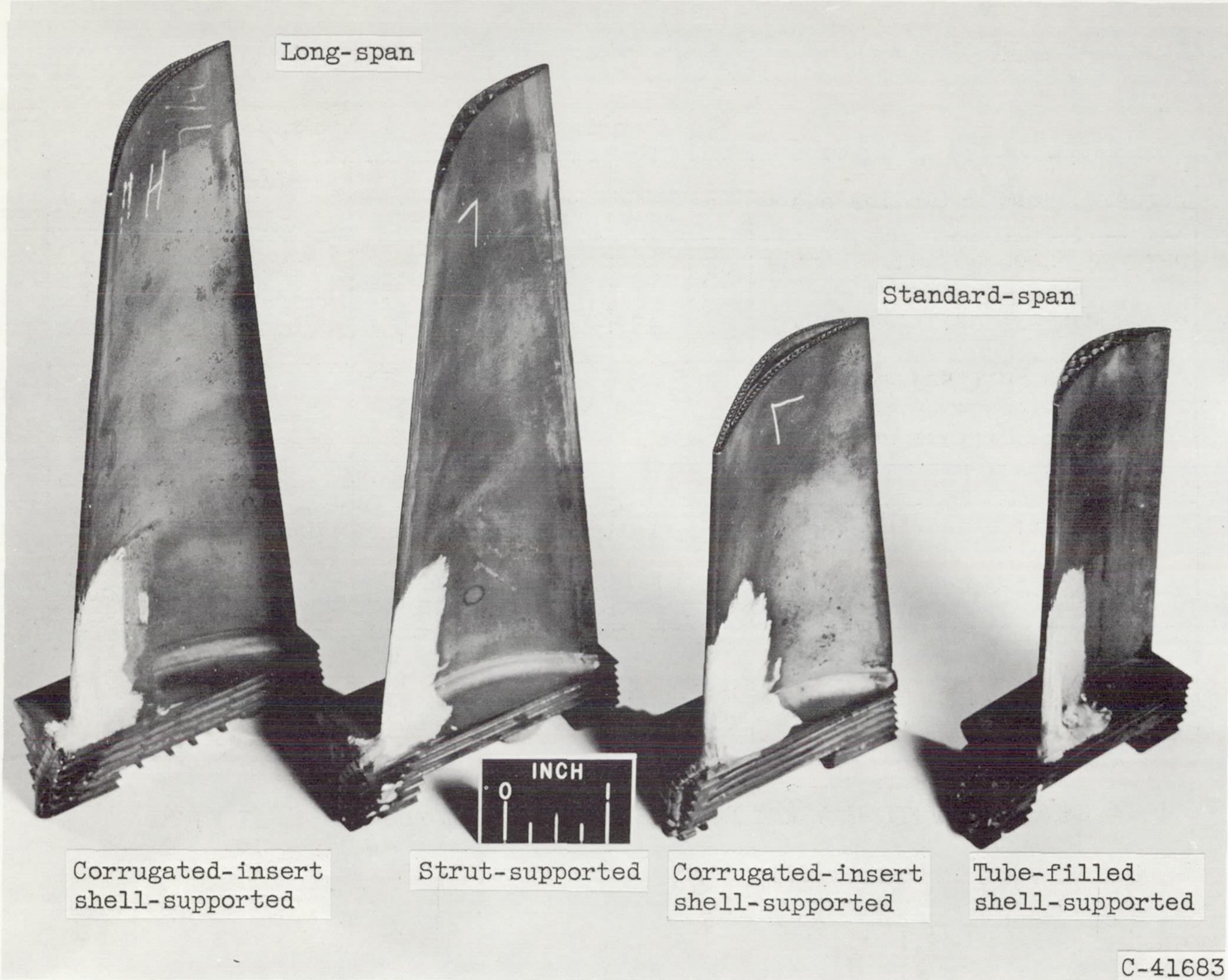


Figure 1. - Four blade types tested.

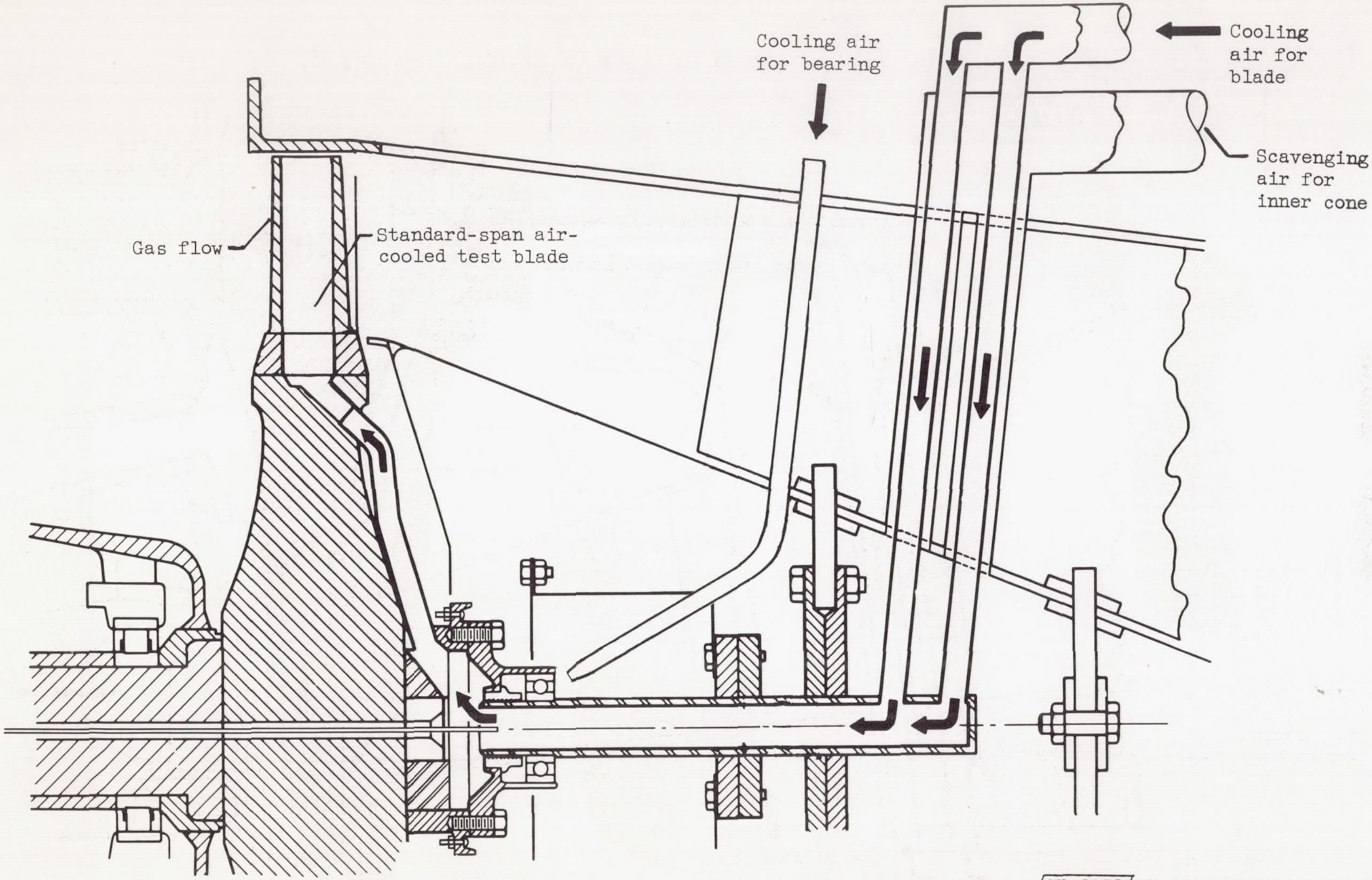


Figure 2. - Tailcone and air-supply system used for short-span blades.

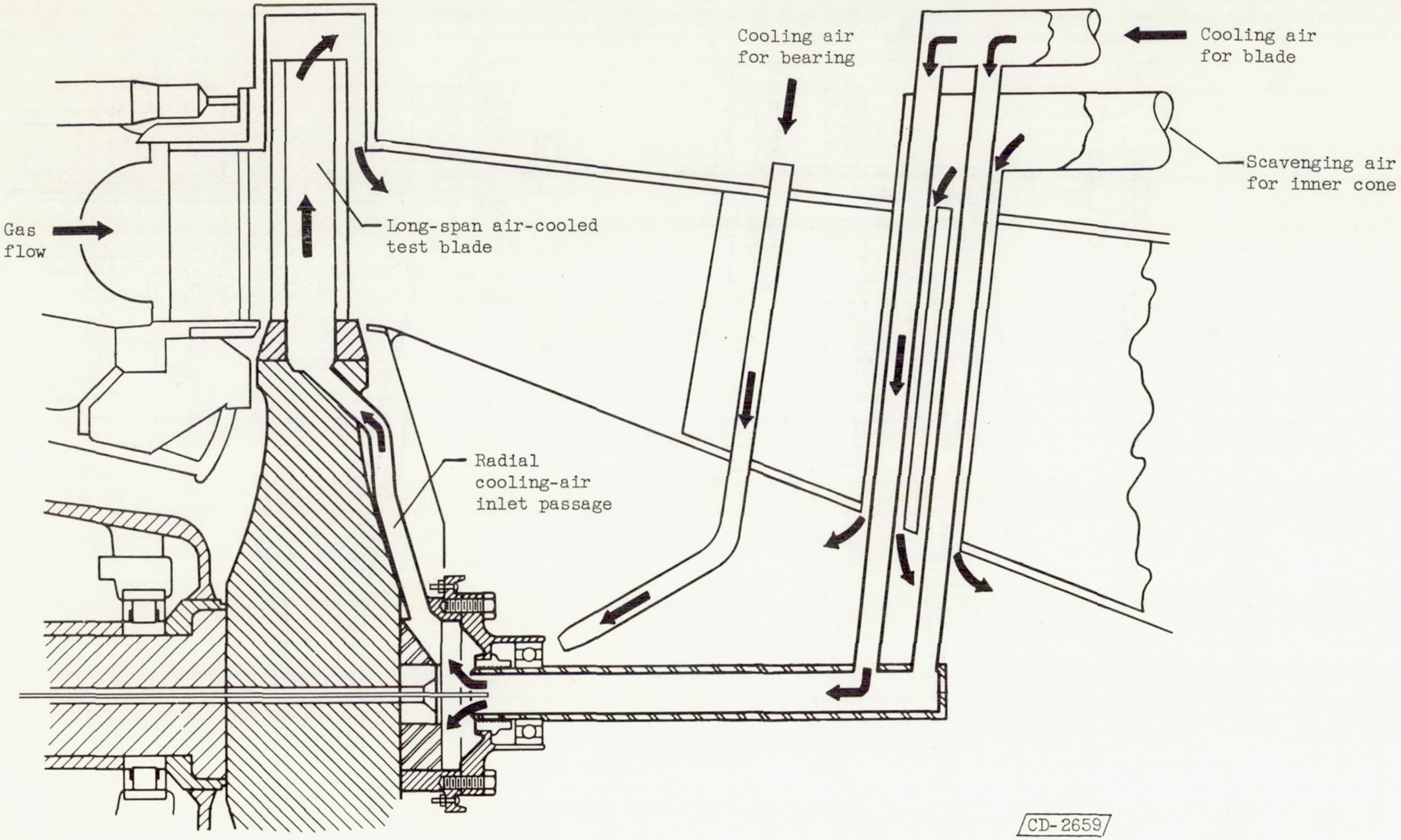
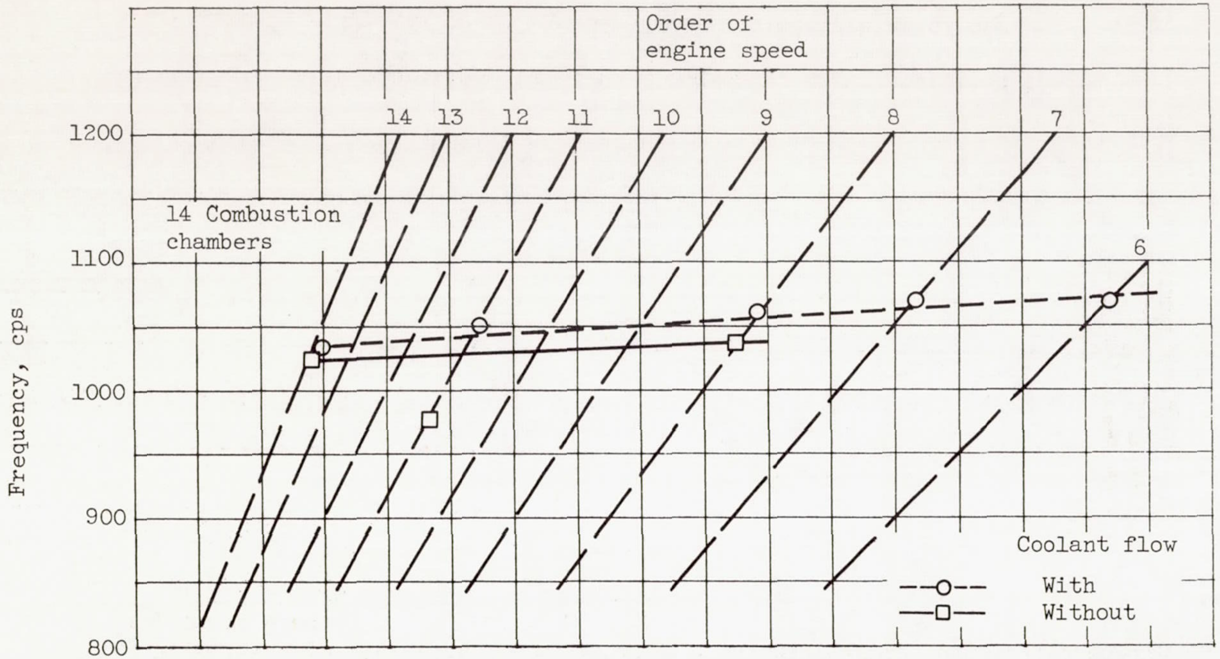
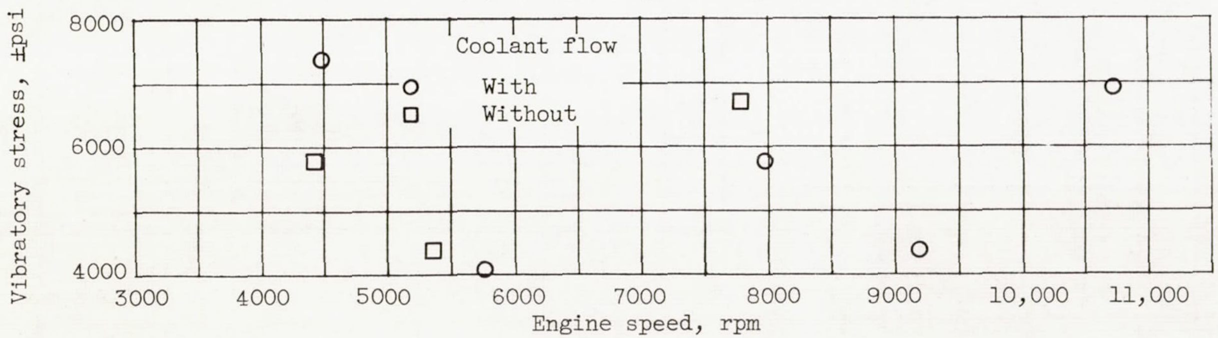


Figure 3. - Tailcone and air-supply system used for long-span blades.

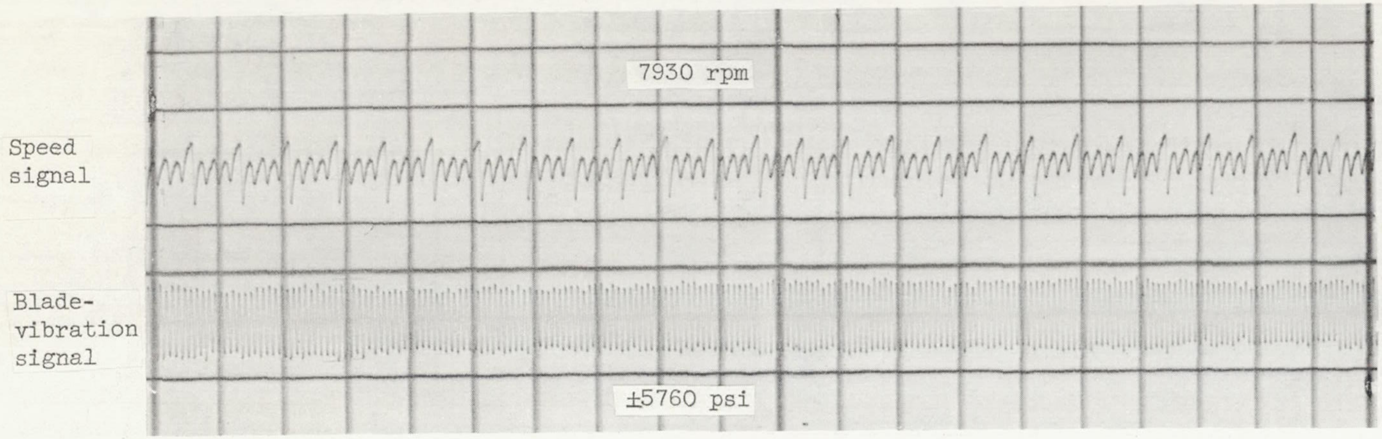


(a) Critical speed diagram.

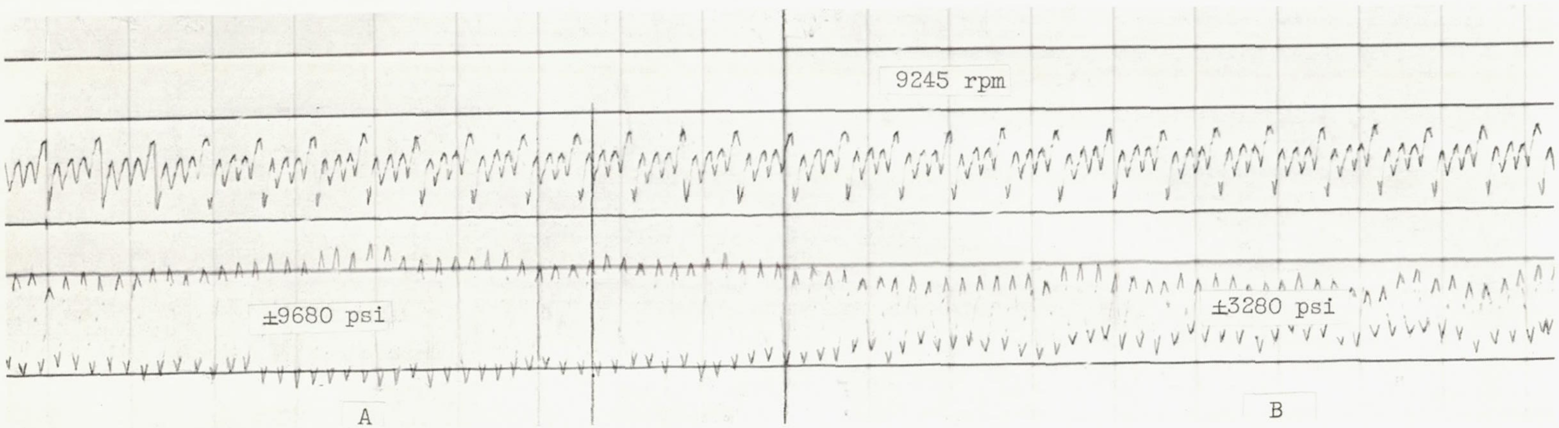


(b) Measured vibratory stress.

Figure 4. - Measured blade frequencies and vibratory stress of standard-span tube-filled shell-supported blades.

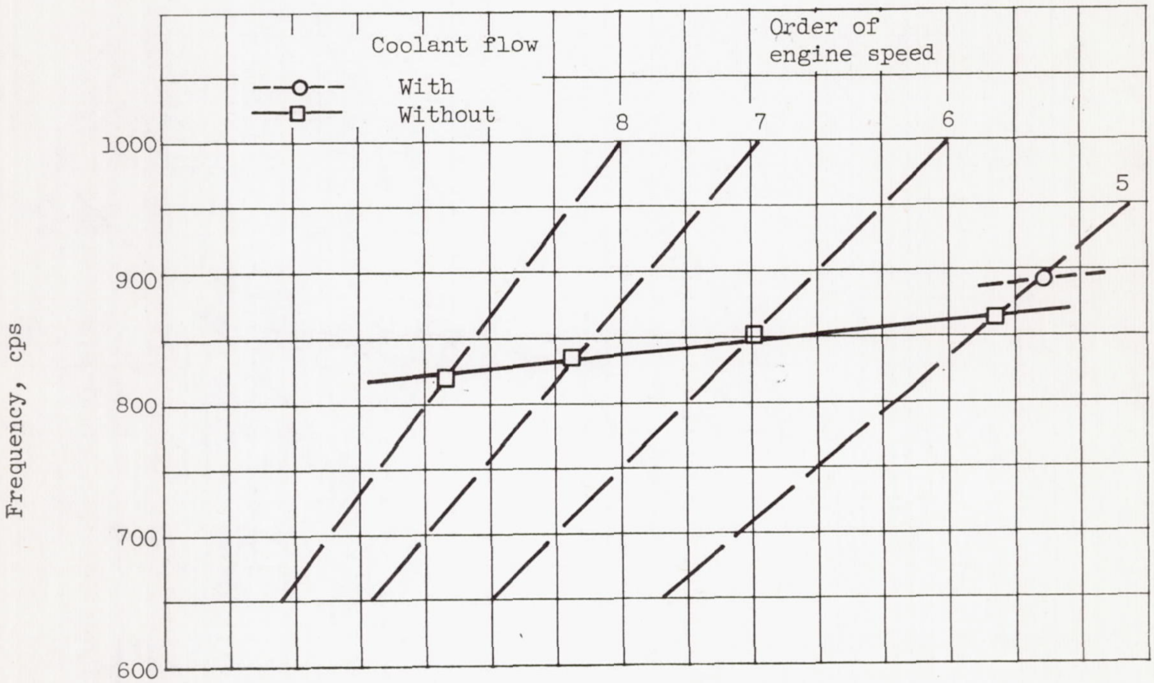


(a) Resonant-type vibration characteristic of standard-span tube-filled shell-supported blade.

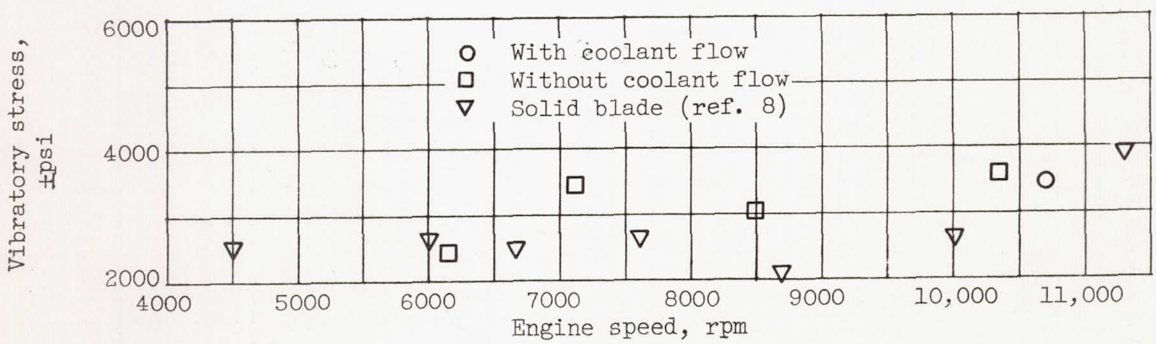


(b) Characteristic vibration signal for long-span, strut-supported blade.

Figure 5. - Typical oscillograms of air-cooled turbine blades.

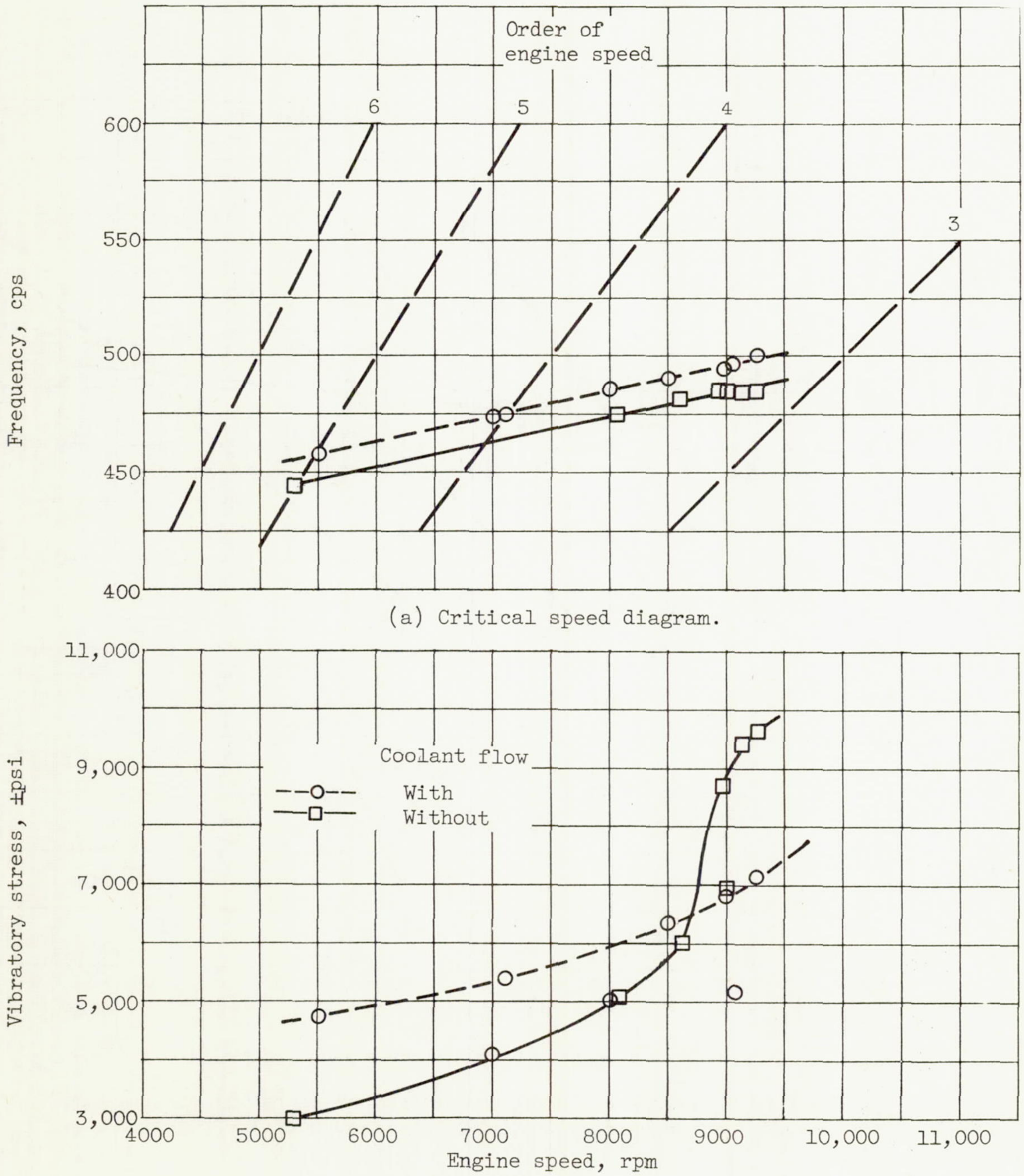


(a) Critical speed diagram.



(b) Measured vibratory stress.

Figure 6. - Measured blade frequencies and vibratory stresses in standard-span corrugated-insert shell-supported blade.



(b) Measured vibratory stress.

Figure 7. - Measured blade frequencies and vibratory stresses in long-span strut-supported blades.

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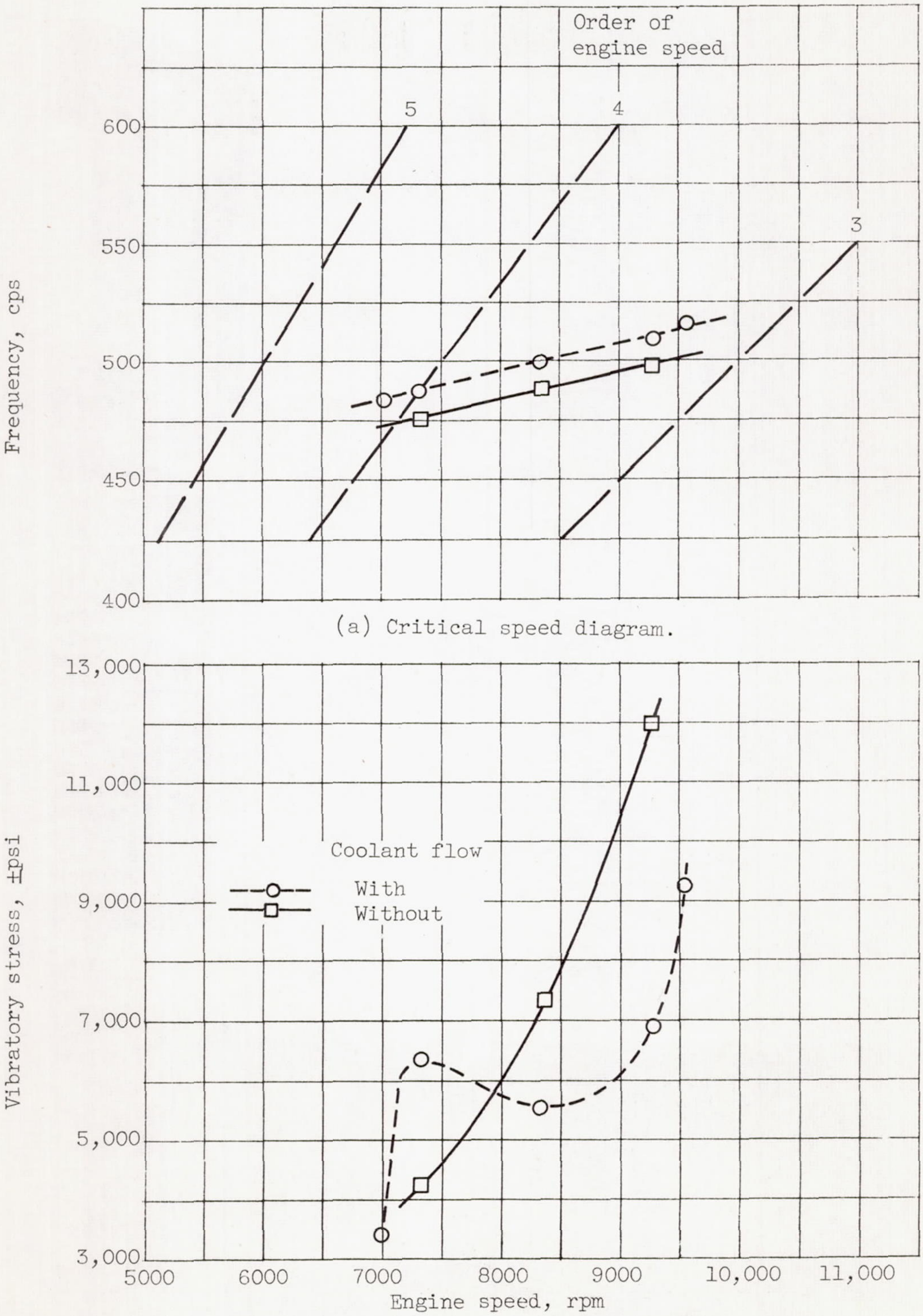

(a) Critical speed diagram.

(b) Measured vibratory stress.

Figure 8. - Measured blade frequencies and vibratory stresses in long-span corrugated-shell-supported blade.

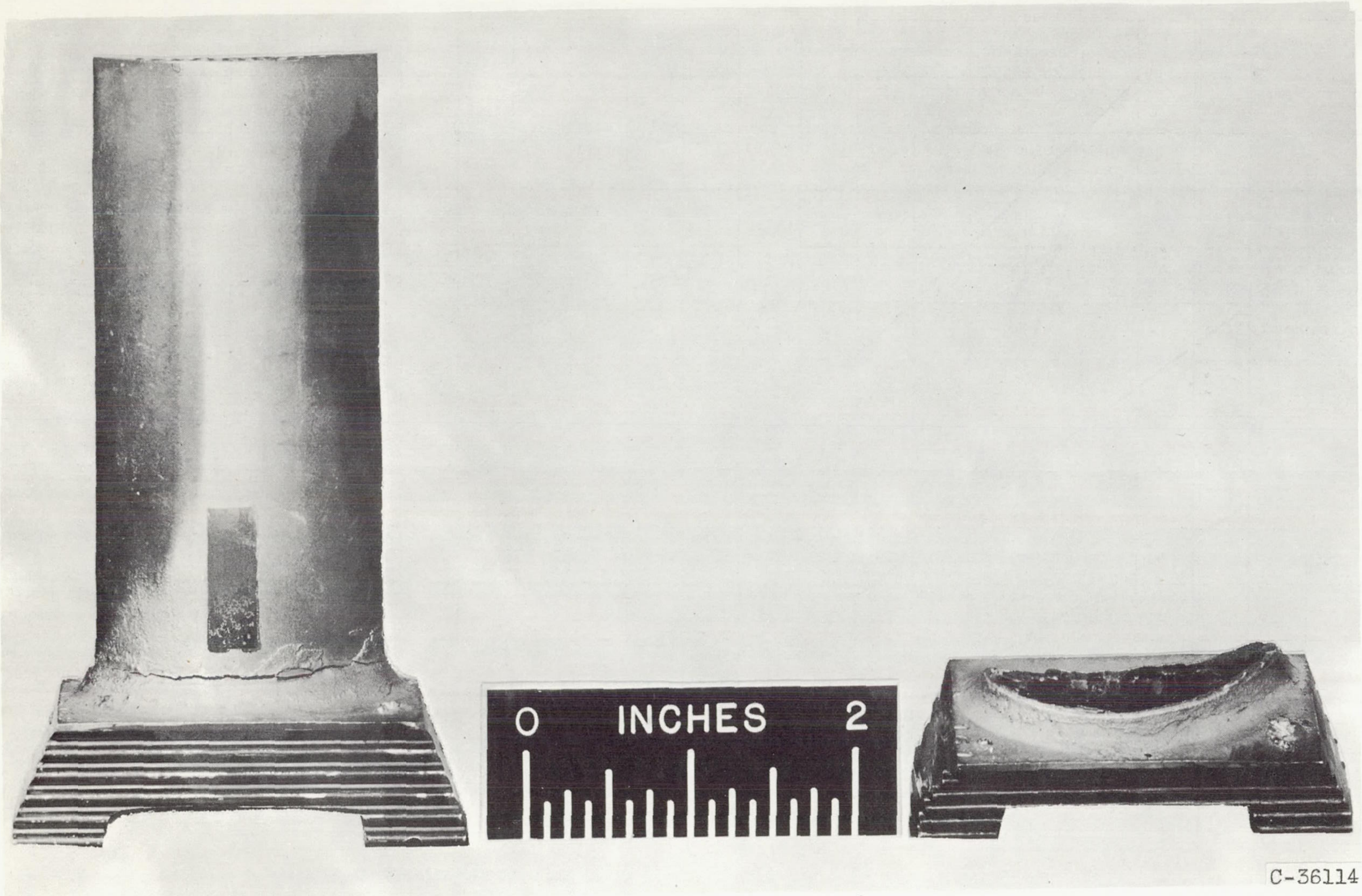

Figure 9. - Typical failures of short-span tube-filled shell-supported blades.