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# RESEARCH MEMORANDUM

## EFFECT OF INLET-GUIDE-VANE ANGLE ON BLADE VIBRATION AND ROTATING STALL OF 13-STAGE AXIAL-FLOW COMPRESSOR IN TURBOJET ENGINE

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# EFFECT OF INLET-GUIDE-VANE ANGLE ON BLADE VIBRATION AND ROTATING STALL OF 13-STAGE AXIAL-FLOW COMPRESSOR IN TURBOJET ENGINE

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## SUMMARY

A blade-vibration and rotating-stall survey was conducted on a modified version of a production turbojet engine incorporating a 13-stage axial-flow compressor with a design pressure ratio of approximately 7 and an air flow of 120 pounds per second. The engine was equipped with a variable-angle inlet-guide-vane assembly, and the guide-vane angles were varied within a range of 36°. Resistance-wire strain gages and constant-current hot-wire anemometers were used to measure the blade vibrations and rotating stall. The hot-wire anemometers were installed in radial survey devices located in the stator passages of the first six stages to determine the effect of the angular position of the inlet guide vanes on the radial position of the rotating stall.

A maximum vibratory stress was measured in the second stage at 5780 rpm with the inlet guide vanes opened 7° from the standard position. Closing the inlet guide vanes changed the rotating stall from a strong tip stall to a strong hub stall. With the inlet guide vanes closed 14°, the blade vibratory stresses were reduced approximately 51 percent.

## INTRODUCTION

Rotating stall is a major problem associated with both experimental and production axial-flow compressors. When a compressor is operated at speeds below approximately 75 percent of design speed, the rear stages tend to choke, and the front stages are forced to operate in a partially stalled condition. This partially stalled condition consists of low-flow or stalled zones spaced about the flow annulus (refs. 1 and 2). These stalled zones can be located anywhere along the span of the rotor blades. This stall phenomenon, known as rotating stall, rotates in the same direction as the rotor at approximately one-half the rotor speed. Rotating stall can excite serious compressor blade vibrations (ref. 2). Therefore, its control or elimination is of utmost importance. Variable-angle inlet guide vanes represent a possible means of controlling or eliminating this rotating-stall condition (ref. 3). The results of an investigation in which a 13-stage axial-flow compressor was operated as part of a turbojet engine equipped with a set of remotely controllable, variable-angle, inlet guide vanes are presented herein. These vanes were varied over a range of 36°. The compressor used in this investigation has a known history of rotating-stall operation (refs. 2 to 5).

This investigation was conducted at the NACA Lewis laboratory in a sea-level static test stand. The vibratory stresses were measured by the use of resistance-wire strain gages, and the flow fluctuations of rotating stall were detected by constant-current hot-wire anemometers mounted in radial survey devices. The engine was equipped with a variable-area exhaust nozzle to vary the pressure ratio and the air flow at a given speed.



## APPARATUS AND INSTRUMENTATION

### Turbojet Engine

The investigation was conducted on a turbojet engine incorporating a 13-stage axial-flow compressor with a design pressure ratio of approximately 7, a rated engine speed of 8300 rpm, and an air flow of approximately 120 pounds per second. The engine, which was installed and operated in a sea-level static test stand, was equipped with an adjustable exhaust nozzle to permit variations in pressure ratio and air flow at a given constant speed. When the exhaust nozzle was in the open position, the engine operated with rated engine temperature ratio at equivalent rated speed. The standard-production engines of the model used for this investigation are equipped with a 48-blade, fixed-angle, inlet-guide-vane assembly. The engine used for this investigation, however, was equipped with an available 72-blade, variable-angle, inlet-guide-vane assembly. The guide-vane angles were varied within a range of 36°, from -7° (open) to 29° (closed), by remote control. The zero vane angle represented the datum angle or the production-setting angle.

The rotor blades in the axial-flow compressor were fabricated from the following material:

| Stage | Material |
|---|---|
| 1, 2, 3 | Chrome steel, 13 percent |
| 4, 5, 6, 7 | Aluminum |
| 8, 9, 10, 11, 12, 13 | Chrome steel, 13 percent |

## Hot-Wire Anemometer

A constant-current hot-wire anemometer system as described in reference 6 was used to detect the rotating stall. Anemometer probes were installed in radial survey devices located in the first to sixth stage stator passages. The systems were adjusted so that the outputs of all hot wires could be directly compared. The dynamic component of the hot-wire signals was viewed on a dual-beam cathode-ray oscilloscope and directly recorded on a 24-channel recording oscillograph. The amplitude of these signals was also measured on a voltmeter, and the voltages were recorded. The method used to determine the number of stall zones per stall pattern was the same as that used in references 1 and 4.

## Instruments for Measuring Vibratory Stress

Commercial resistance-wire strain gages were cemented to four blades each in the first and second stages, and two blades each in the third to the twelfth stages. These strain gages were positioned approximately at midchord, close to the base of the airfoil. Lead wires were run from the strain gages to a 19-ring slip-ring assembly mounted on the front of the engine starter. The slip-ring assembly and strain-gage circuits were the same as reported in reference 7.

A 24-channel recording oscillograph was used to record the strain-gage and hot-wire-anemometer signals.

# PROCEDURE

The investigation was conducted in a sea-level static test stand with the compressor inlet and turbine exhaust open to the atmosphere. The engine was operated at various speeds up to approximately 80 percent of rated speed with the adjustable exhaust nozzle set at various positions. The engine was operated with various exhaust areas to force the compressor to operate with different pressure ratios at a constant speed. With the maximum area obtainable, the engine operated with rated engine temperature ratio at equivalent rated speed. The minimum area was limited by the maximum allowable tail-pipe temperature or the minimum area obtainable with the adjustable nozzle.

For the radial hot-wire surveys the engine was held at constant speeds of 55, 65, or 70 percent of rated speed as the radial survey devices were actuated to move the hot-wire element along the entire span of the compressor blades. The hot-wire and strain-gage data were recorded simultaneously at all conditions.

RESULTS AND DISCUSSION

The data obtained in this investigation are discussed in relation to the effect of inlet-guide-vane-setting angle on (1) rotating-stall patterns observed, (2) vibratory stresses in the first six rotor stages, (3) radial and axial extent of the rotating stall in the first to sixth stages, and (4) maximum speed at which the compressor operated with rotating stall.



Rotating-Stall Patterns

Relative stall frequency is plotted against actual engine speed in figure 1 to show the stall patterns observed with the four guide-vane angles investigated, that is, $-7^{\circ}$ (open), $0^{\circ}$ (standard production), $14^{\circ}$ (closed), and $29^{\circ}$ (closed). The relative stall frequency is defined as

$$f_s' = \frac{N\lambda}{60} - f_s$$

where

$f_s'$ relative stall frequency, cps

$N$ compressor speed, rpm

$\lambda$ number of zones in stall pattern

$f_s$ absolute frequency determined from hot-wire signals, cps

The four-zone stall pattern was the predominant stall pattern observed for this particular compressor. In previous investigations (refs. 2, 4, and 5), the three-zone pattern was the predominant stall pattern observed. It is assumed that the increased solidity associated with the 72-blade, variable-angle, guide-vane assembly altered the compressor stall characteristics so that the four-zone stall pattern became the most predominant (ref. 4).

The three- and four-zone patterns observed with vane angles of $-7^{\circ}$, $0^{\circ}$, and $14^{\circ}$ rotated at approximately 52.5 and 50 percent of the rotor speed, respectively (fig. 1). The relative frequency of the three- and four-zone pattern with the $29^{\circ}$ vane angle was lower than for the other angles, because the rotational speed of the rotating stall had increased. The five-zone stall was observed only at the $14^{\circ}$ inlet-guide-vane angle.

## Rotor-Blade Vibrations

Critical-speed diagrams showing the relation between the fundamental bending frequencies of blades in the first six stages and the harmonics of the three- and four-zone relative stall frequencies are shown in figure 2. The frequency of excitation due to rotating stall for a rotor blade is the stall frequency relative to the rotor blades.

The data points in figure 2 represent the actual measured frequencies for blades in the first six stages as determined from the oscillograph records. The solid lines (approximately horizontal) are drawn through these points to represent the vibrational frequencies of the blades over the speed range investigated, that is, the speed range in which the compressor operated with rotating stall. The dashed and dot-dashed lines represent the relative stall frequencies for the harmonics of the three- and four-zone stall patterns determined from figure 1.

The blade-frequency lines are intersected by the stall lines, and each of these intersections represent the source of a possible critical speed or critical vibration. These possible critical speeds are summarized in table I. For most intersections the blade frequencies tend to group near or follow the relative-stall-frequency line for a small range of speeds, such as the third stage at 4700 and 5300 rpm.

Plots of vibratory stress against engine speed for the first six stages are shown in figure 3 for compressor operation at the four inlet-guide-vane angles. The vibratory stress is presented as percent of the maximum stress measured with the inlet guide vanes at the standard position. The minimum vibratory stress reported herein is ±5000 psi. The maximum stress for the first stage (fig. 3(a)) was measured at approximately 5100 rpm with the standard vane angle. Figure 2(a) shows that the second-harmonic line of the three-zone stall intersects the first-stage frequency line at approximately 5200 rpm. Since the compressor was operating with a three-zone stall when this peak stress was recorded, the vibrations were excited by the second harmonic of the three-zone stall.

For the second stage (fig. 3(b)) the peak stress was recorded when resonance occurred between the natural frequency of the second stage and the second harmonic of the three-zone relative stall frequency at 5800 rpm (table I). This peak stress was measured with open ($-7^{\circ}$) inlet guide vanes. The maximum stress measured with the open ($-7^{\circ}$) vane position was 8 percent higher than that measured with the standard vane position. Figure 1 shows that the three-zone stall was only observed with the $-7^{\circ}$ and 0 vane angles, thus explaining why high stresses were recorded only with these two vane angles.

The highest stress measured in the third stage (fig. 3(c)) was at approximately 5300 rpm. High stresses were exhibited with both the standard and open vane angles. Table I shows that these peak stresses were caused by resonance of the blades with the second harmonic of the four-zone stall.

The highest stress measured in the fourth stage (fig. 3(d)) was at 6950 rpm, a speed above the rotating-stall speed range. This vibratory stress was a fourth-order excitation of the engine speed. Engine-order vibration was only observed with the vanes in the 29° position. The highest stress measured in the rotating-stall range was at approximately 4850 rpm. Figure 2(a) shows that the third harmonic of the four-zone stall and the fourth harmonic of the three-zone stall intersect the blade-frequency line at approximately 4800 rpm. Figure 1 indicates that only the four-zone stall was observed at this speed; therefore, the third harmonic of the four-zone stall excited the vibratory stress.

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The vibratory stresses for the fifth and sixth stages are shown in figures 3(e) and (f), respectively. The stresses in these stages were relatively low.

For the six stages, all the peak stresses measured in the rotating-stall speed range (4000 to 6300 rpm) and presented in figure 3 were excited by higher harmonics of the rotating stall, not the fundamental stall frequency. These data clearly indicate that the maximum stresses were measured with the standard and open vane angles, and that, in general, closing the guide vanes reduced the vibratory stresses. For example, in the second stage the stresses were reduced 61 percent when the guide vanes were closed 29°. Opening the guide vanes 7° increased the stresses 8.25 percent; closing the guide vanes 14° reduced the stresses 51 percent.

In figure 3 the vibratory stresses vary considerably for a given vane angle and speed. As the exhaust-nozzle area is reduced, the compressor is forced to operate at a higher pressure ratio or with higher loading on the blades. The periodic unloading and loading associated with rotating stall would then be higher with the reduced nozzle area. In figure 3(b) the stresses vary from 15 to 108 percent for operation with the -7° vane angle at approximately 5800 rpm. This stress range was associated with changes in the exhaust-nozzle area; that is, as the exhaust-nozzle area was reduced, the vibratory stresses increased.

## Radial and Axial Strength of Rotating Stall

The results of the radial surveys of the hot-wire anemometers at 55, 65, and 70 percent of rated equivalent speed are shown in figures 4(a), (b), and (c), respectively. The curves are plots of the recorded hot-wire output voltage against the passage height for each of the first six stages. Whenever a hot-wire anemometer broke during a survey, the curve was terminated. The upper curves are for rated nozzle operation, and the lower curves for operation with reduced nozzle area, that is, the compressor operating close to the stall line.

The data for 55 percent of rated equivalent speed (fig. 4(a)) show that the compressor operated with a strong tip stall when the inlet guide vanes were positioned at -7°, 0°, and 14°. When the guide vanes were closed 29°, the compressor operated with a tip stall in the first stage, tip and hub stall in the second stage, and a hub stall in the third to the sixth stages.

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At 65 percent of rated equivalent speed (fig. 4(b)), the compressor operated with a tip stall in all stages for guide-vane angles of 0° and -7°. The stall zones were larger and stronger with the reduced exhaust-nozzle area. Closing the guide vanes 14° caused the compressor to operate with tip stall in the first two stages and hub stall in stages three to six. Operation with the guide vanes closed 29° nearly eliminated the rotating stall in the first stage; however, a strong hub stall existed in the next five stages.

At 70 percent of rated equivalent speed and rated nozzle (fig. 4(c)) the stall was comparatively weak except for the hub stall in the third and fourth stages. When the exhaust-nozzle area was reduced, strong rotating stall was again encountered. With the 0° and -7° vane angles a strong tip stall was present in all six stages. With the 14° angle the compressor operated with a tip stall in the first stage and a hub stall in the third to the sixth; however, when the vanes were closed 29° the stall was nearly eliminated in the first two stages with a strong hub stall persisting in the third to the sixth stages.

Variable-angle inlet guide vanes had a great effect on the rotating stall. Adjustment to a higher angle nearly eliminated the stall in the first two stages at 65 and 70 percent of rated equivalent speed; however, the stall in the next four stages was changed from a tip stall to a hub stall. The hub stall had a relatively small effect on rotor-blade vibration because its center of force was near the root or fixed end of the blade.

## Maximum Rotating-Stall Speed

Inlet-guide-vane angle does affect the maximum speed at which the compressor operates with rotating stall. The following table summarizes these data:

| Maximum speed at which compressor operates with rotating stall, percent rated | Inlet-guide-vane angle, deg |
|---|---|
| 75 | -7 (open) |
| 70 | 0 (standard) |
| 67.5 | 14 (closed) |
| 71.5 | 29 (closed) |

## SUMMARY OF RESULTS

An investigation was conducted on a multistage axial-flow compressor to determine the effect of inlet-guide-vane angle on rotating stall and blade vibration. The results may be summarized as follows:

1. The four-zone stall pattern was the prevailing stall pattern; however, the three- and four-zone rotating-stall patterns were observed with all four inlet-guide-vane angles. A five-zone stall pattern was observed only with the guide vane at a 14° angle.

2. All the peak stresses measured below 80 percent of rated speed were excited by higher harmonics and not by the fundamental of the rotating-stall relative frequency. The maximum stress measured was in the second stage.

3. The maximum vibratory stress measured in a fourth-stage aluminum blade was at 84 percent of rated speed. This excitation was the fourth order of the engine rotational speed.

4. Closing the inlet guide vanes changed the stall from a strong tip stall to a strong hub stall.

5. Closing the inlet guide vanes made it possible to nearly eliminate the stall in the first two stages at 65 and 70 percent of rated speed.

6. Closing the inlet guide vanes 14° reduced the compressor blade vibratory stresses 49 percent.

7. Closing the inlet-guide-vane angle decreased, and opening the guide-vane angle increased, the maximum speed at which the compressor operated with rotating stall.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, November 4, 1955

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TABLE I. - CRITICAL SPEEDS IN FIRST SIX STAGES

| Compressor stage | Engine speed, rpm | Harmonic of - | |
|---|---|---|---|
| | | 3-Zone stall | 4-Zone stall |
| 1 | 5200 | 2nd | |
| 2 | 5800 | 2nd | |
| 3 | 4700 | 3rd | |
| | 5300 | | 2nd |
| 4 | 4850 | 4th | 3rd |
| 5 | 4750 | | 4th |
| | 4950 | 5th | |
| 6 | 4650 | | 5th |
| | 5000 | 6th | |
| | 5600 | | 4th |

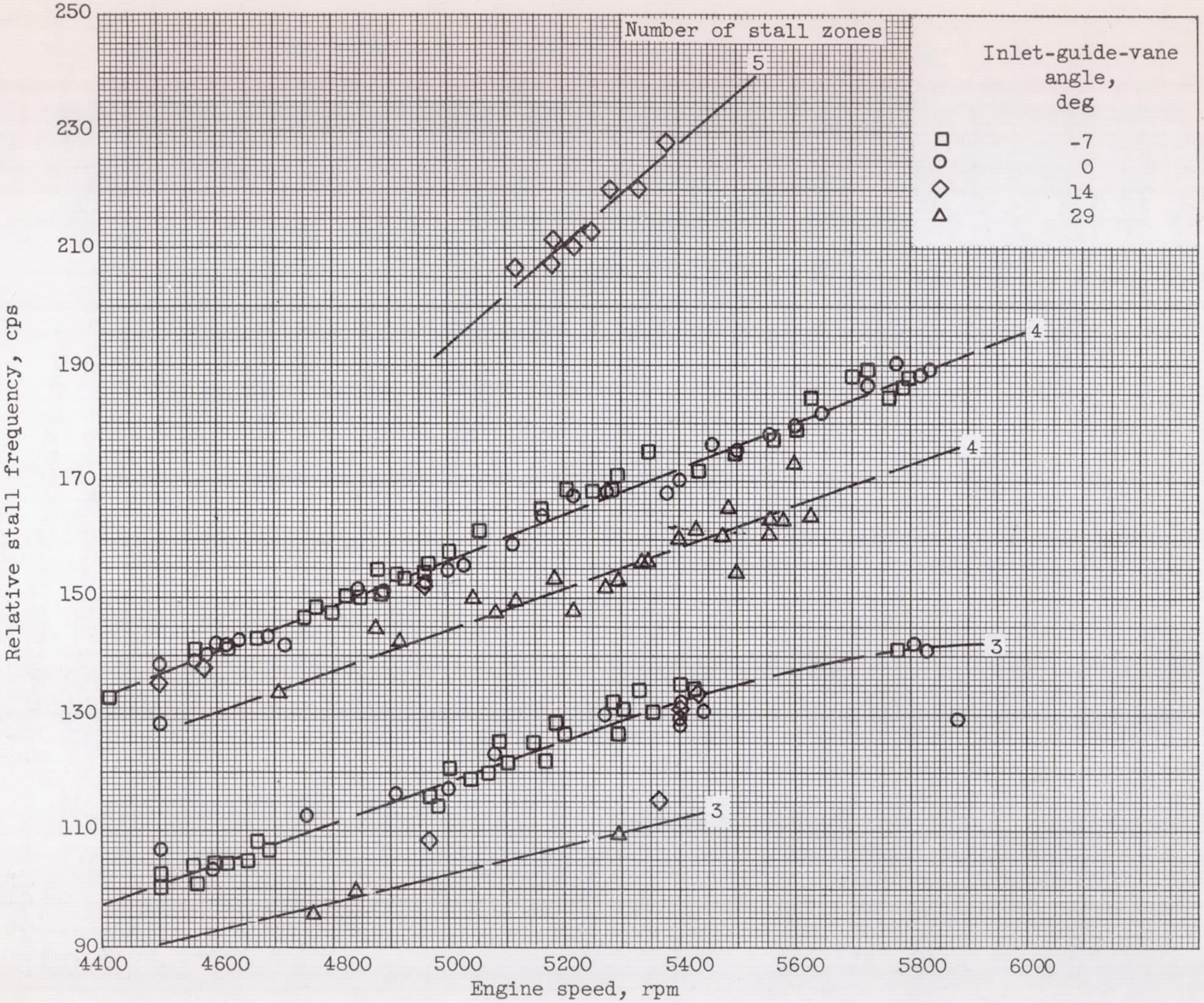


Figure 1. - Rotating-stall patterns observed with various inlet-guide-vane angles.

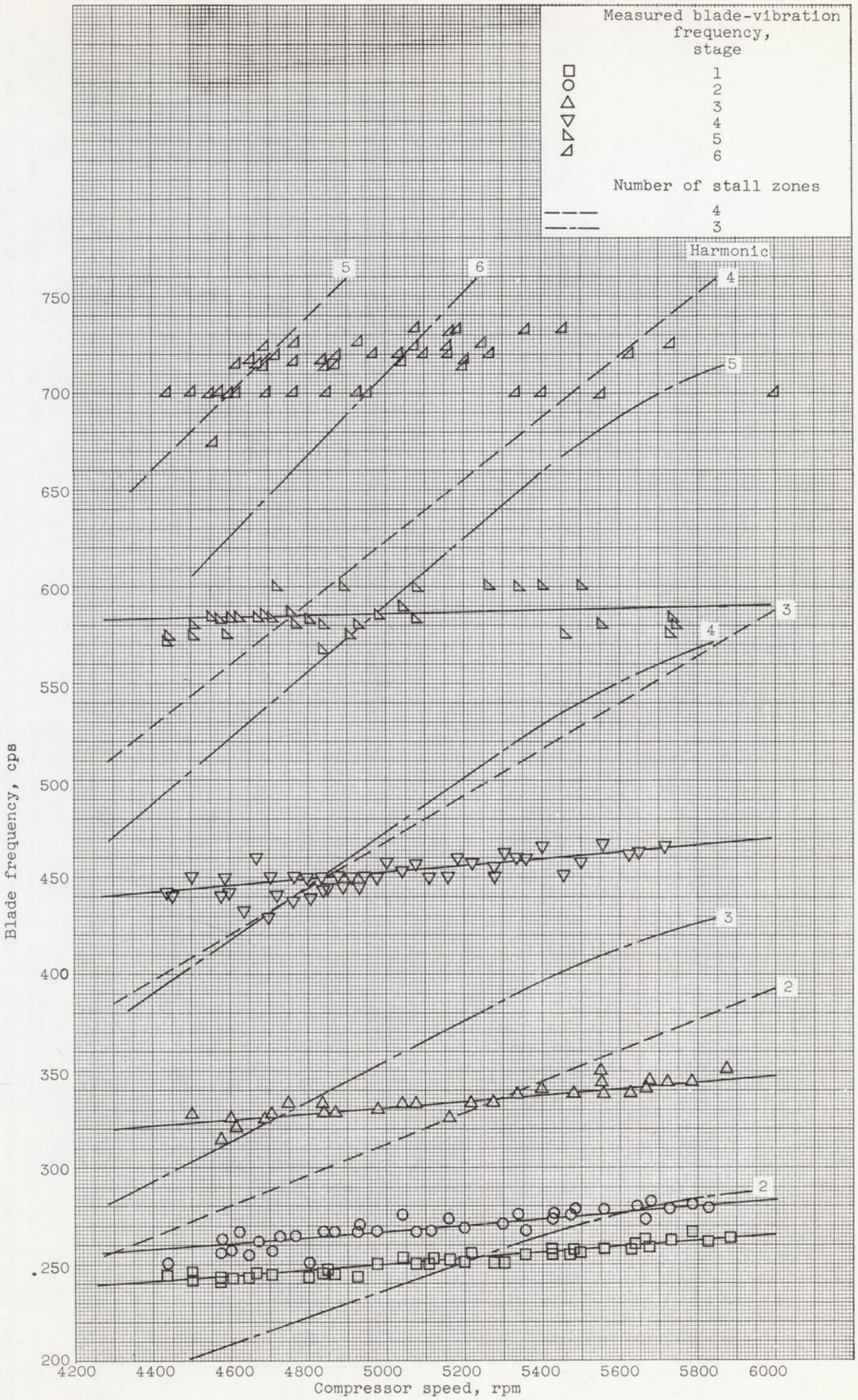


Figure 2. - Critical-speed diagrams showing correlation between rotor-blade and relative rotating-stall frequency for stages one to six.

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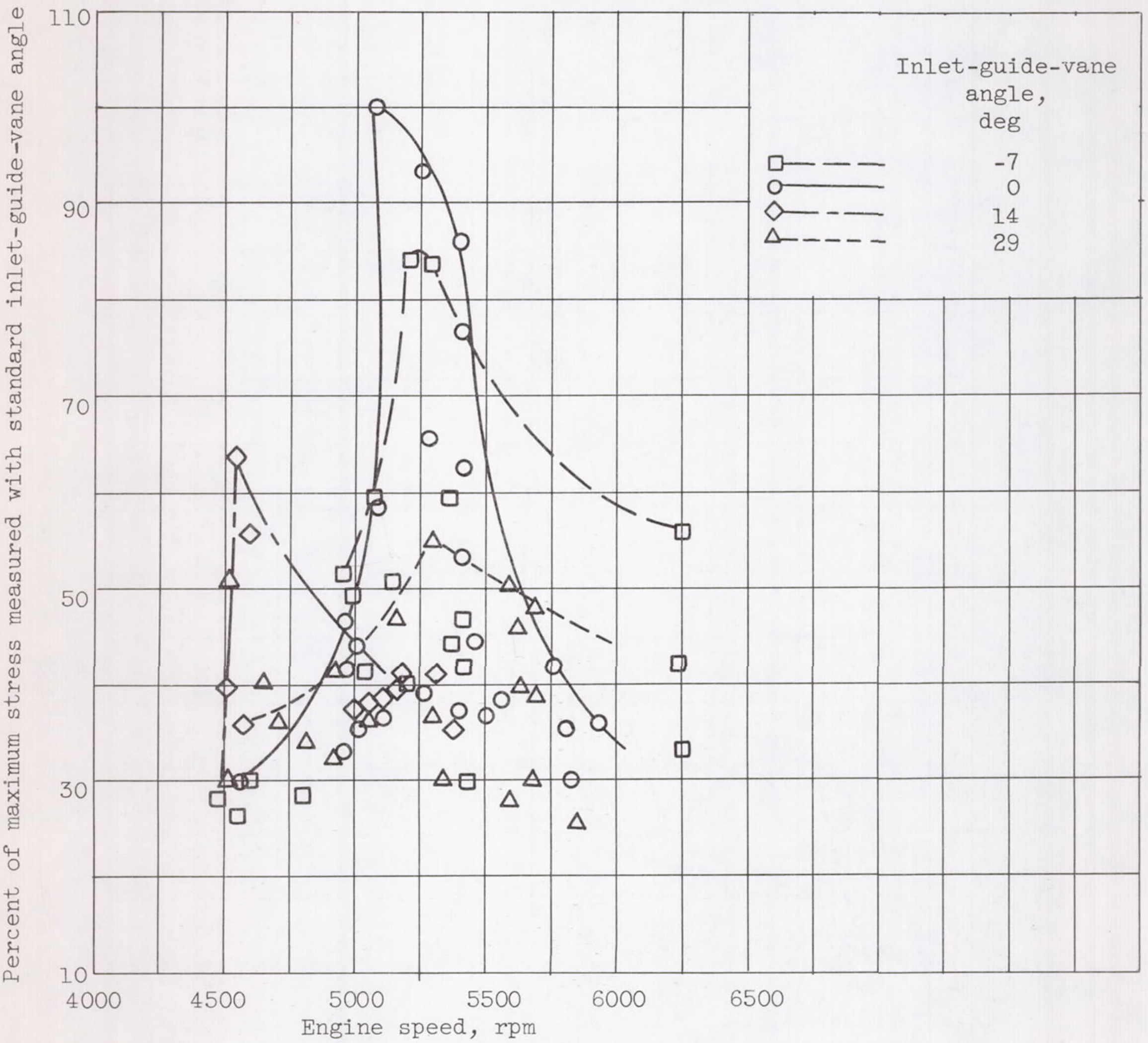
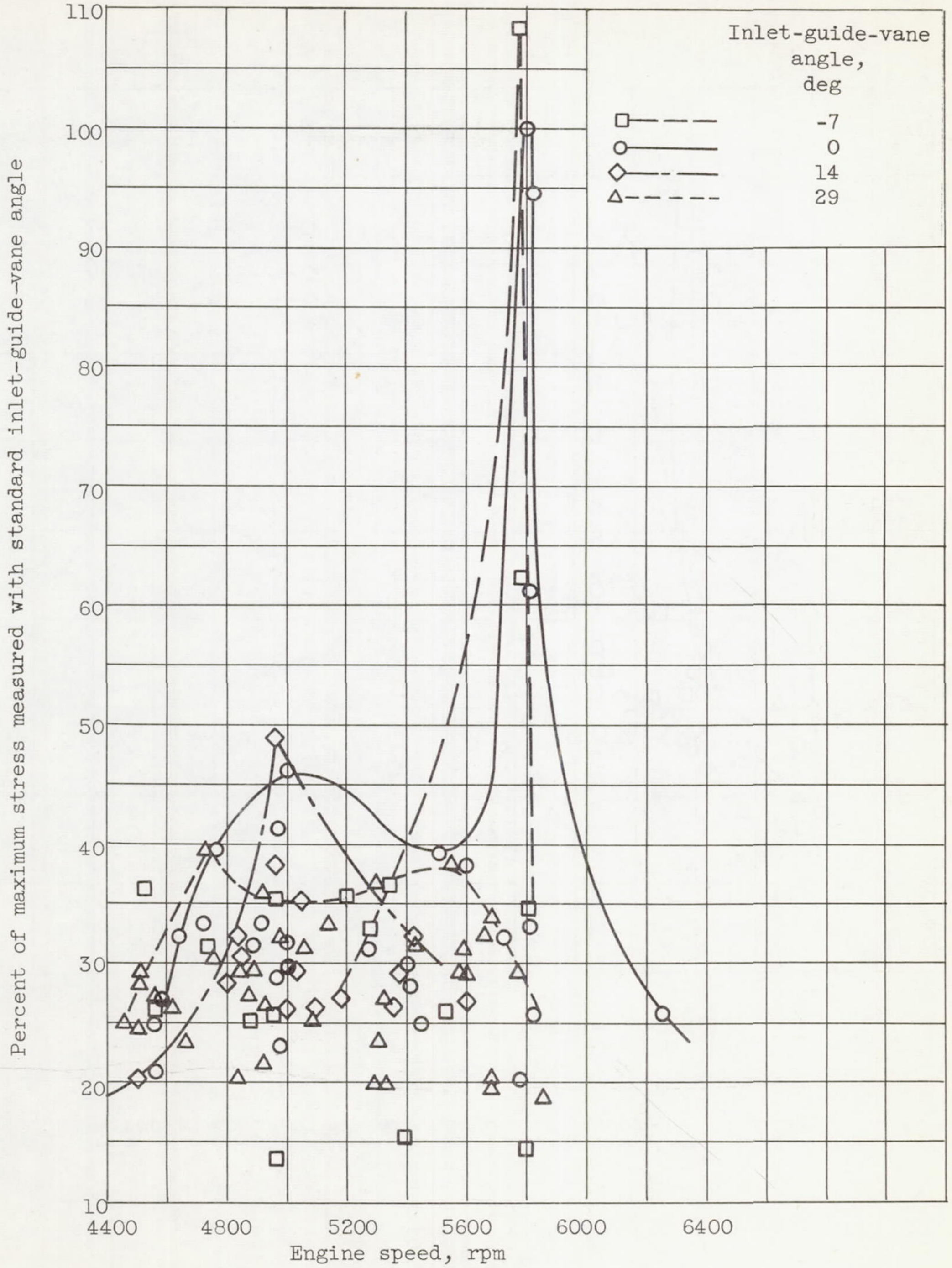



(a) First stage.

Figure 3. - Resonant vibratory stresses of rotor blades.




(b) Second stage.

Figure 3. - Continued. Resonant vibratory stresses of rotor blades.

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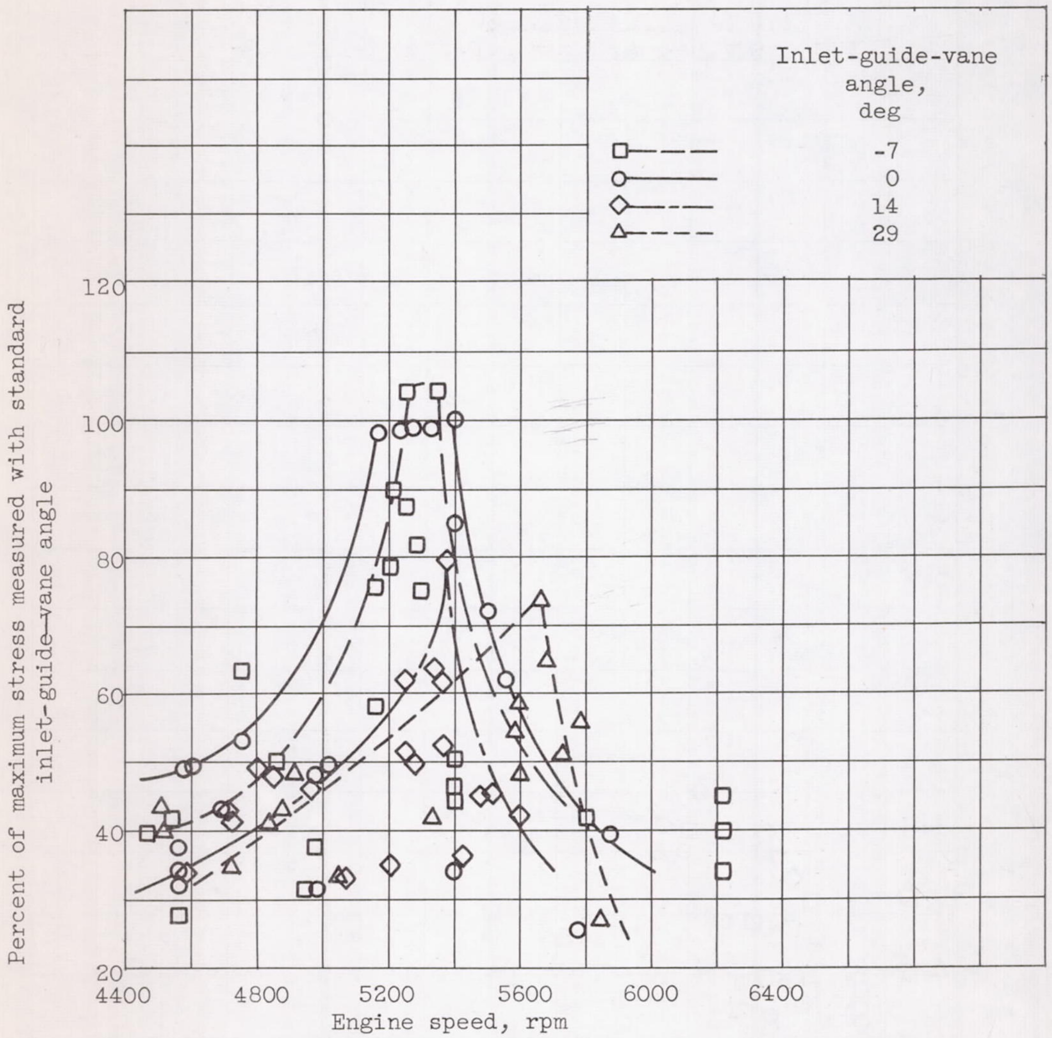


(c) Third stage.

Figure 3. - Continued. Resonant vibratory stresses of rotor blades.

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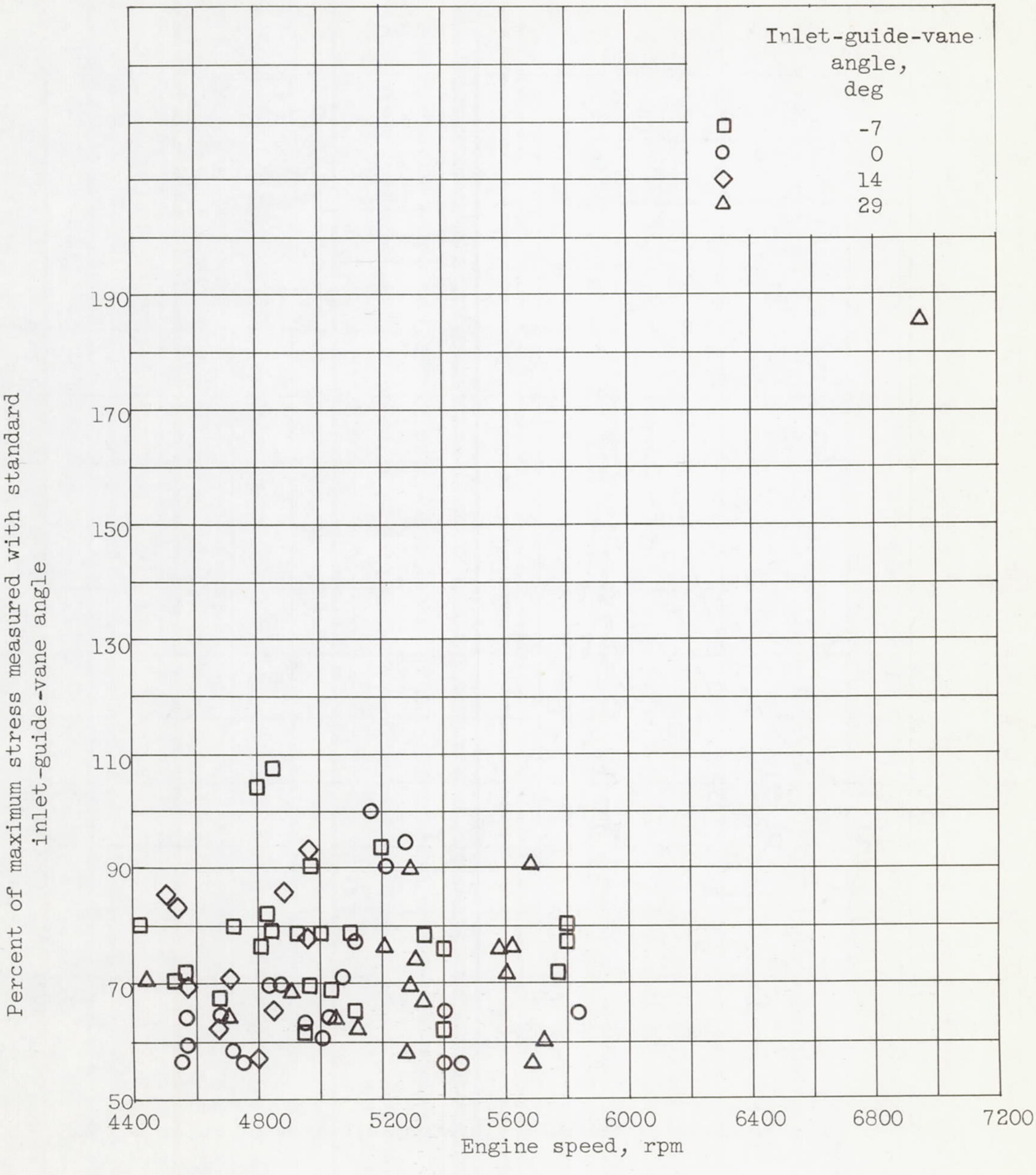


(d) Fourth stage.

Figure 3. - Continued. Resonant vibratory stresses of rotor blades.

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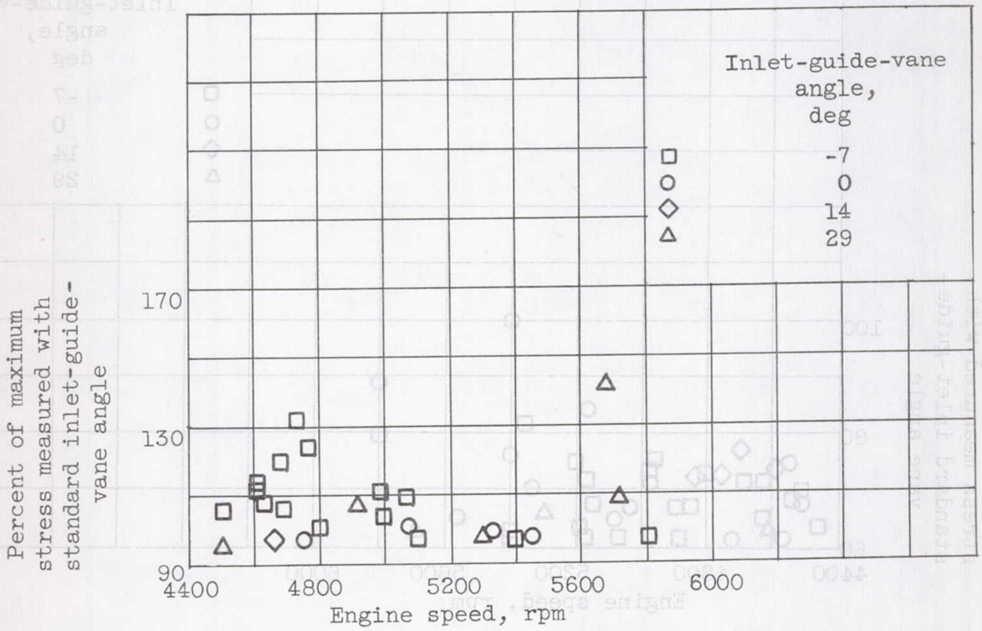
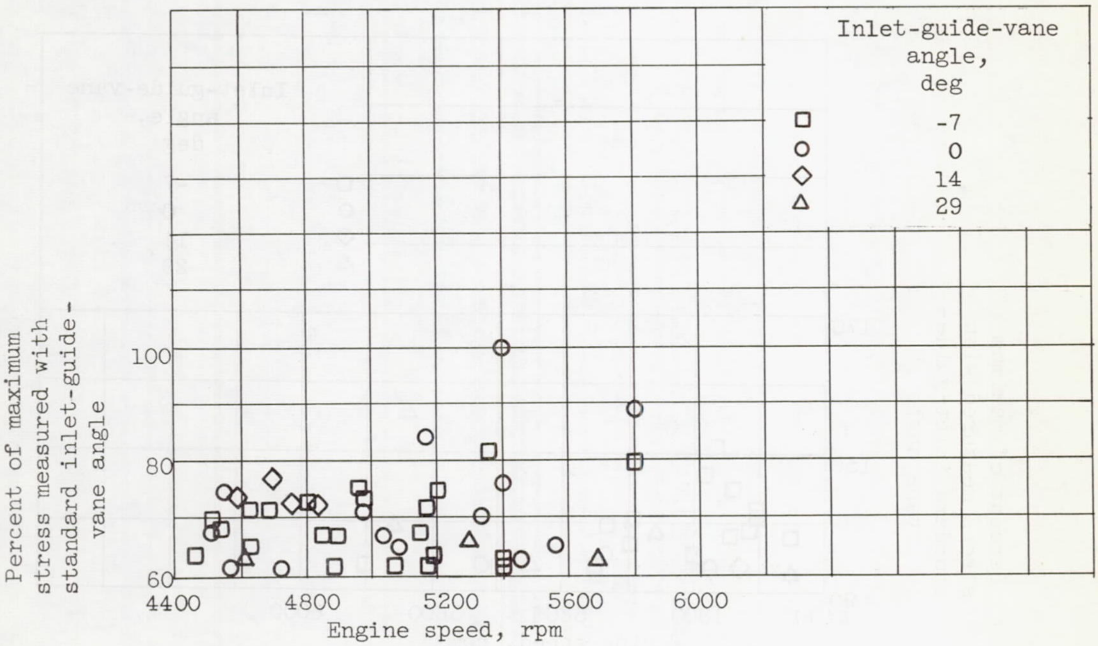


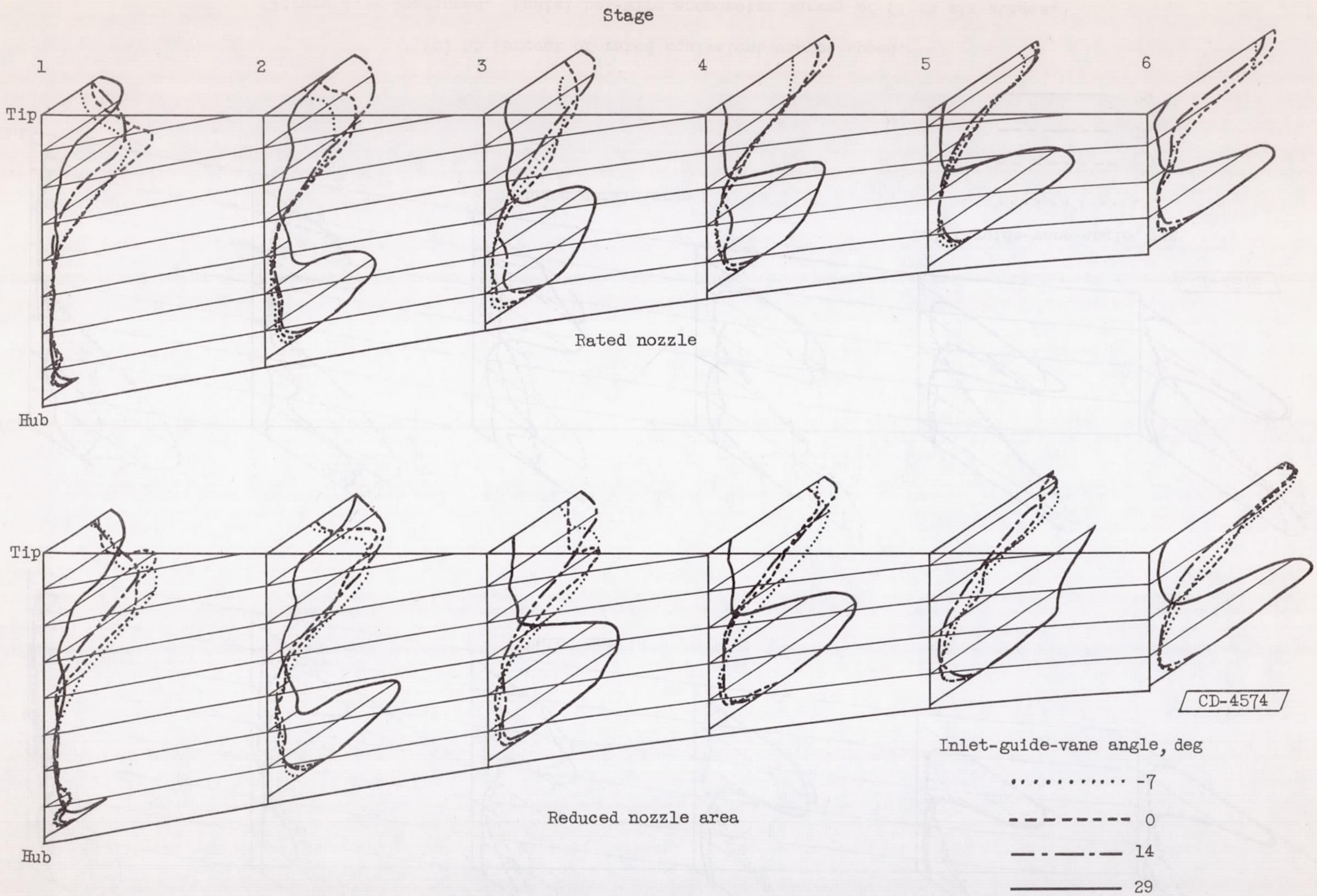

(e) Fifth stage.

Figure 3. - Continued. Resonant vibratory stresses of rotor blades.



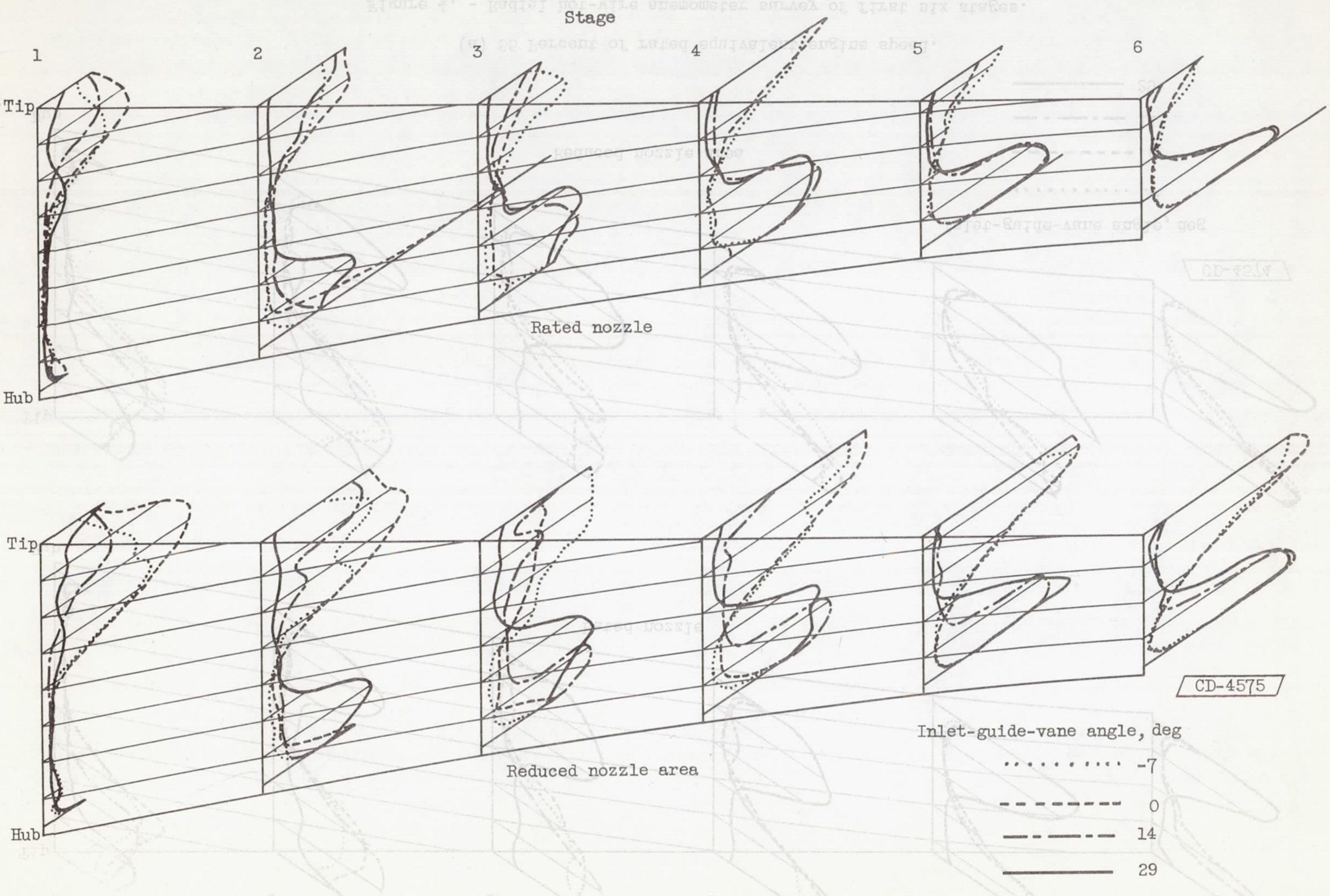

(f) Sixth stage.

Figure 3. - Concluded. Resonant vibratory stresses of rotor blades.



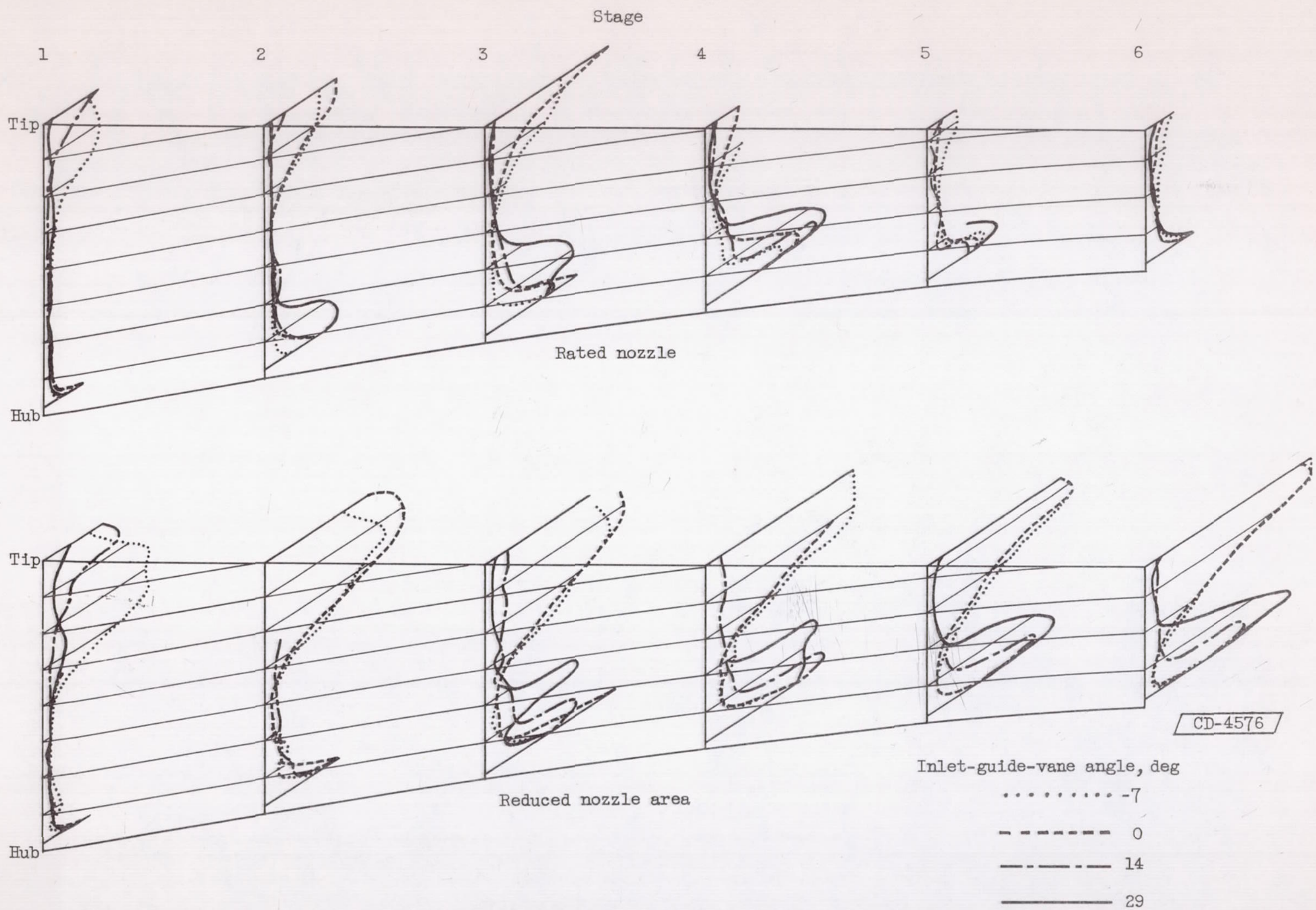

(a) 55 Percent of rated equivalent engine speed.

Figure 4. - Radial hot-wire anemometer survey of first six stages.




(b) 65 Percent of rated equivalent engine speed.

Figure 4. - Continued. Radial hot-wire anemometer survey of first six stages.




(c) 70 Percent of rated equivalent engine speed.

Figure 4. - Concluded. Radial hot-wire anemometer survey of first six stages.