R. & M. No. 2988
(17,132)
A.R.C. Technical Report

MINISTRY OF SUPPLY

AERONAUTICAL RESEARCH COUNCIL
REPORTS AND MEMORANDA

An Investigation of the High Alternating Stresses in the Blades of an Axial-flow Compressor

By

J. R. FORSHAW, M.Eng., A.M.I.Mech.E.

LONDON: HER MAJESTY'S STATIONERY OFFICE
1957
SIX SHILLINGS NET

An Investigation of the High Alternating Stresses in the Blades of an Axial-flow Compressor

By

J. R. Forshaw, M.Eng., A.M.I.Mech.E.

Communicated by the Principal Director of Scientific Research (Air), Ministry of Supply

Reports and Memoranda No. 2988*

June, 1954

Summary.—A vibration investigation of the 1st, 2nd and 3rd stage stator blades of an axial-flow compressor gave high values of the alternating stresses of the order of 10,000 lb/sq in. On three runs excessive values of the alternating stress of 24,000, 27,500 and 48,000 lb/sq in. were obtained, and for these and for some of the other maxima the incidence was estimated to equal the stalling incidence.

There is excitation at all speeds of low orders showing dissimilar flow in different parts of the stage and this excitation increases in a critical manner at or near stalling incidence indicating a partial stall which is localised in parts of the stage. At greater incidence the flow deteriorates, the contrast between various parts of the stage is reduced, and the level of the alternating stress falls to a value slightly higher than that at incidences lower than the partial stall.

This phenomenon occurs for critical operating conditions and is different from stalling flutter. Both phenomena could occur in the same compressor, the latter would be influenced by blade stiffness and could occur over wide ranges of operation provided the incidence and mass flow are greater than the lower boundaries.

1. *Introduction.*—There have been a number of failures of rotor and stator blades in the early stages of axial-flow compressors, and although it is known that the excitation is due to aerodynamic conditions dissimilar from those at operating conditions around the design point, further knowledge of the excitation is required. Usually it is not an isolated blade failure and a strain-gauge investigation gives relatively low values of alternating stress. The measuring techniques are well developed and any errors introduced should be slight. There is a scatter of ± 40 per cent or ± 50 per cent in alternating stress around a particular stage due to varying root damping and individual natural frequencies and the manner in which the mode of vibration of the complete stage is excited at the various frequencies. When all allowances have been made a factor of 3 or 4 is required to give a stress sufficiently large to cause a short-term failure.

High alternating stresses can occur in flutter investigations of cascades of constant-section blades representative of mean diameter and tip sections of axial-flow compressor blades. One zone of high alternating stress occurs for high positive values of incidence, one for operating values of incidence at Mach numbers higher than would occur in normal compressor operation and for some cascades a zone for high values of negative incidence. In the high incidence zones there is a peak in the alternating stress for increasing incidence with constant Mach number but for the higher incidences there is an increase in the stress for increase in Mach numbers and values of the stress higher than the peak value could be obtained.

A number of the 1st, 2nd and 3rd stage stator blades of an axial-flow compressor had failed in such a manner to indicate the blades had been excited in their fundamental mode of vibration.

* N.G.T.E. Rept. R.157, received 21st October, 1954.

An investigation of the stresses in the stator blades that had failed and adjacent blades was made with the compressor installed on the compressor test rig so that any desired condition of operation could be obtained.

The results of this investigation are analysed and sufficient readings have been obtained to give some indication of the conditions prevailing in these regions of high alternating stresses in compressors and comparisons are made with the conditions in the cascade.

The values of the incidence are estimated for the design point and an attempt is made to indicate the trends in the values of incidence for the part load operation of the compressor.

The excitation of these high alternating stresses is discussed.

2. *Design Details*.—2.1. *Details of the Compressor*.—The following values were taken in the design of the compressor:

Mass flow	= 72 lb/sec
r.p.m.	= 9,000
Number of stages	= 8
Pressure ratio	= 4·5 : 1.

The blade design assumed 50 per cent reaction at all diameters, axial velocity distribution constant and equal work done at all blade heights. Two sets of blades were used in the compressor, one set for stages 1 to 4 and one set for stages 5 to 8 with the stagger adjusted for each stage. The rotor blades were cropped at the tip and the stator blades cropped at the root. The blades had C.4 profile on a circular-arc camber-line, the camber at mean diameter was 26 deg and 36 deg respectively.

The blades were manufactured by casting the rotors from Vitallium and the stators from H.R. Crown Max the numbers of blades being as follow:

Stage	I.G.V.	1	2	3	4	5	6	7	8
Rotor		65	71	73	77	111	113	115	117
Stator	57	66	70	72	110	112	114	116	118

The following design stresses were used:

	1st stage		*5th stage*		
	Rotor	*Stator*	*Rotor*	*Stator*	
Gas bending stress ..	5·0	5·2	5·2	4·75	tons/sq in.
Centrifugal stress ..	12·6		6·4		tons/sq in.

The design was intended for fully machined blades but for the build for this investigation using cast blades the stress factors are representative of aircraft practice.

2.2. *Particulars of Failed Blades*.—The compressor was used for three series of aerodynamic investigations and inspection after the third series revealed cracks in following blades.

1st Stage: 1 blade	..	Trailing edge $\frac{7}{16}$ in. from root
2nd Stage: 22 blades	..	14 convex side between root and $\frac{1}{2}$ in. from root
		6 trailing edge between root and $\frac{5}{16}$ in. from root
		1 leading edge $\frac{5}{8}$ in. from root
		1 concave side $\frac{1}{8}$ in. from root
3rd Stage: 4 blades	..	Trailing edge between root and $\frac{7}{16}$ in. from root.

The casting of the blades introduces some scatter in the properties of the material and in the consistency of the material at a particular point. The blade lengths were 4·33, 3·74 and 3.15 in. for stages 1, 2 and 3 respectively and the number and position of above failures indicate that in each case the blades were being forced mainly in a fundamental flexural mode of vibration, the highest stresses occurring in the 2nd stage.

The different strengths of the materials chosen for rotor and stator blades explains the concentration of the failures in the stator blades but it is considered that running at certain compressor conditions would cause failure of the rotor blades.

3. *Description of Experiments.*—3.1. *Amplifiers and Recording Equipment.*—During previous strain-gauge investigations due to the limitations of recording cameras, combinations of single-channel amplifiers and recorders had been used with switching of strain-gauge signals. On some investigations a drum camera had been used with stationary rewinding of the film. Considerable difficulty had arisen in the correlation of signals particularly in the assessment of the operating conditions.

Three A.E.R.E. type 1008 amplifiers were available and driver amplifiers were constructed to deflect V.C.R.X.214 cathode-ray tubes. Two additional cathode-ray tubes were used for time and speed marking on the film. A drum camera with reloading mechanism and magnetic shutter was constructed and gave satisfactory operation in the investigation up to film speeds of 600 in./sec and subsequently up to film speeds of 1,000 in./sec.

The amplifiers and recording equipment are shown in Fig. 2a and the camera, drum and reloading mechanism shown in Fig. 2b.

The reloading mechanism comprises co-axial feed and take-up spools. The film passes from the take-up spool across one side of a sprocket around the drum across the other side of the sprocket and is guided in a spiral path by four inclined rollers, around another sprocket on the same spindle as previous sprocket to the feed spool. It was found necessary to have the axes of the drum, spools and sprockets parallel to each other, to have tight guide rollers on the sprockets and to have the direction of rotation of the feed spool opposite to the take-up spool to prevent loose lengths of film inside the drum. The rewinding of the film was started by applying a brake to a circular groove in the carrier of two planet wheels which when stationary caused the drum to rewind the take-up spool through a reduction ratio of $65/20 \times 20/64$. The rewind is stopped by a pin on a gear wheel driven by the sprockets tripping a lever which releases the brake when the film has moved one drum length. Twenty-eight records were obtained from 100-ft reel of film which facilitated the experiments and the three-channel recording simplified the analysis.

3.2. *Application of Strain-Gauges.*—Approximately 40 strain-gauges were used comprising some of each of the following types:

'British Thermostat' 200-ohm gauges with *Durofix* adhesive were used on inlet guide vanes and stage 1, 2, 3 and 4 stator blades

'Tinsley' 6H 50-ohm gauges with *Durofix* adhesive were used for 45-deg rosette on inlet guide vanes and 2nd stage stator blade

N.G.T.E. 240-ohm $\frac{5}{16}$-in. gauge length Bakelite gauges comprising three double loops of $0 \cdot 0006$-in. diameter nichrome wire with Bakelite adhesive were used on stage 8 stator blades.

The installation of the gauges in the compressor is shown in Fig. 1. The wires were soldered to the gauge leads and small clips were welded in position to fasten the leads to the blade platform to prevent loads in the vicinity of the welds of gauge wire to gauge leads. The wires were attached to the casing with protective tapes and adhesive and passed through small holes drilled in the compressor casing.

The gauge leads were passed from the compressor to a ganged switch so that sets of three gauges could be obtained in quick succession. On some of the runs records were obtained from two sets of gauges for each operating point of the compressor.

3.3. *Operating Conditions During the Experiments.*—Records were taken on four runs of the compressor and the aerodynamic conditions are plotted in Fig. 14. Particulars of the speeds and the gauges used are as follow:

(1) A variable-speed run using flexural gauges on inlet guide vanes and 1st and 2nd stage stator blades; and on 3rd, 4th and 8th stage stator blades at pressure ratios approximately 75 per cent of stalling pressure.

(2) A variable-speed run using flexural gauges on inlet guide vanes and 1st and 8th stage stator blades with a more comprehensive series of compressor speeds.

(3) A variable-speed run with 45-deg strain rosette on inlet guide vanes and 2nd stage stator blades.

(4) Six constant-speed runs at 4,000, 5,000, 6,000, 7,000, 8,000 and 9,000 r.p.m. with five equally spaced points on pressure characteristic.

4. *Analysis of Results.*—4.1. *Harmonic Analysis of Records.*—The strain readings have been analysed into their constituent harmonic components and after corrections have been made for gain settings, gauge factor and modulus of elasticity the predominant harmonic components of the stress have been plotted against compressor speed as follow:

Fig.	3	4	5	6	7	8	9	10	11	12	13
Stage	I.G.V.	1	2	3	4	8	I.G.V.	1	8	I.G.V.	2
Run	1	1	1	1	1	1	2	2	2	3	3

For the constant-speed runs the alternating stresses have been plotted on the pressure ratio *vs.* mass flow characteristic curves in Figs. 15 and 16 for the inlet guide vanes and 2nd stage stator blade respectively. The harmonic components are plotted in the form of circles the centre of the circle at the operating point the radius of the circle representing the amplitude of vibration and the number on the circumference representing the order of vibration.

The maximum values of the alternating stress for each run for a particular stage in lb/sq in. are as follow:

Run	*Stage*	*Stress*	*Order*	*Total Stress*	*Orders*	*r.p.m.*
1	I.G.V.	± 5,200	3	± 7,500	3 & 2	6,600
1	1	± 5,700	2	± 8,300	2 & 3	6,050
1*	2	±24,000	3	±40,000	3 & 2	6,550
1	3	± 5,800	4	± 8,400	4 & 5	5,650
1	4	± 4,600	5	± 5,700	4, 5 & 6	6,400
1	8	± 2,600	15	± 4,200	15 & 14	6,400
2†	I.G.V.	±27,500	2	±37,000	2 & 3	6,700
2	1	±10,000	2	±14,000	2 & 3	5,500
2	8	± 4,800	11	±12,000	11, 14 & 13	8,300
3	I.G.V.	±14,800	2	±16,800	2 & 3	6,700
3	2	±13,600	3	±16,000	3 & 2	6,700
Constant speed	I.G.V.	± 9,000	2	± 9,800	2 & 1	9,000
Constant speed*	2	±48,000	2	±55,900	2 & 3	7,000

Two of the peak values of the alternating stress for the second stage marked with an * and one value of the alternating stress for the inlet guide vanes marked with a † are of sufficient magnitude to cause a short-term failure. The remainder of the maximum values are representative of the high stress levels usually obtained in an investigation of this nature and would indicate a long-term failure.

For each blade there are a number of resonances involving combinations of natural frequency, order of vibration, r.p.m., stiffness of blade but the peak values are dependent on another parameter. The magnitude of the peak value recorded on a particular run is a function of the condition of operation of the compressor. The excitation of these peak values appears to be some critical condition in the aerodynamic flow in the stage.

4.2. *Estimation of Incidence at Design Point.*—Flutter experiments on cascades of blades in a wind tunnel indicate that the zones of high self-excited alternating stresses appear for particular values of the incidence. Previous investigations on compressors have shown that if incidences of approximately 7 to 10 deg occur in parts of a compressor when the compressor as a whole is operating at high efficiency high alternating stresses are excited.

It is extremely difficult to calculate the values of incidence at all blade heights for all conditions of operation of the compressor and trends in the changes of incidence at mean diameter for changes in conditions of compressor operation would be of interest for the stages in the compressor which are subjected to high alternating stresses.

The values of incidence at mean diameter for the first five stages of the compressor were estimated using Refs. 1, 2 and 3 and taking values for the mass flow and pressure ratio from the experimental curves.

4.3. *Estimation of Incidence at Part Load.*—The estimation of the part-load performance of a compressor is difficult and scatter appears in the results. It is difficult to draw general conclusions but one example giving well-defined results is given in Ref. 4, and also the points at which the compressor or a particular stage stalls or chokes are discussed.

In the experiments the running time was kept to a minimum because of the short life of the strain-gauges when subjected to high values of alternating strain, and failure of another part of the compressor did not permit full readings to be taken. The stage pressure readings were not obtained and an accurate stage performance estimation could not be attempted in this instance.

Approximate values of the incidence at mean diameter were calculated in a similar manner to that used at the design point for various speeds, and for various delivery pressures for stages 1 to 5 as listed in Fig. 17. The values obtained should be taken to indicate the trends in the changes in incidence the numerical values which are greater than stalling incidence should be discounted and also the values which are dependent on previous stages which have high values of incidence.

The factors having most influence on the calculations are:

- U the peripheral speed which can be obtained accurately
- V_a the axial velocity is dependent on the density—which can be estimated accurately for the early stages and the accuracy decreases with the number of stages
- δ the deviation which is difficult to estimate accurately but the errors will not have a major effect on the results
- η the efficiency which is difficult to estimate and on which further information is required.

4.4. *Trends in Incidence with Changes in Operating Conditions.*—The flow in multi-stage axial-flow compressors is complex and is dependent on the characteristics of the individual stages, inter-stage matching and the overall matching of the compressor. However, present compressors have similar design criteria and the following general trends in the values of incidence occur for changes in the operating conditions.

At reduced compressor speeds the mass flow falls to a greater extent than the peripheral speed and the values of U/V_a are large resulting in high positive values of incidence in the early stages of the compressor. These high values of incidence persist for a small number of blade rows and the resulting low efficiency reduces the work done per stage. The increase in density is not sufficient to maintain approximately constant axial velocity in the converging passage and there is an increase in the axial velocity with a tendency to reduce the incidence. This effect continues through the compressor resulting in negative incidence in the later stages sometimes leading to choking of the outlet stages.

At constant speed for increases in pressure ratio the density increases, V_a falls and there is a tendency for the incidence to increase. If the mass flow falls the effect is increased. For a decrease in the pressure and increase in the mass flow there is a tendency for decrease in the incidence.

The above are general simplified statements and combinations of changes and the conditions in a particular compressor can give rise to different variations.

5. *Discussion of Results*.—5.1. *Cascade Tests*.—Ref. 5 reports extensive tests to record the zones of flutter that occur in a cascade in a wind tunnel. The sections used were constant section from root to tip and were representative of mean diameter or tip sections of compressor blading. The stiffness of these blades is relatively less than that of a tapered compressor blade.

Two zones of high alternating stress have been observed for all sections, one at operating incidences but at higher Mach numbers than would occur in normal compressor operation, and one at high positive incidences. For some sections there is a third zone at high negative incidences.

For the zone at high positive incidence increasing the incidence of the air stream at constant Mach number produces a peak in the alternating stress, but for the incidence values higher than that at the peak if the Mach number is increased the alternating stresses are increased and values greater than the above peak value can be obtained. The fall in amplitude with increased incidence above the peak appears to be due to the fall in energy in the stream due to decrease in efficiency rather than a reduction in the excitation.

The values of incidence at the peak vary from + 6 deg to +12 deg for the sections tested.

5.2. *Correlation of Peak Stresses and Incidence*.—In Fig. 17 the peak stresses have been plotted on the approximate incidence values in the form of circles the centre of the circle at the operating condition, the radius representing the amplitude and the number on the circumference denoting the run.

The following are values of the stalling incidence for mean diameter extrapolated from curves given in Ref. 7:

Blade row ..	I.G.V.	1R	1S	2R	2S	3R	3S	4R	4S	5R	5S
Stalling incidence	2·5	7·5	7·5	7·5	7·5	7·0	7·0	7·0	7·0	7·0	7·0

The incidences have been estimated for mean diameter and the three-dimensional flow in the compressor will influence these values. The conditions at other blade sections will differ from those at mean diameter and it is thought that the root or tip sections will stall first.

Comparing the above values with those from Fig. 17 starting from the more accurate low values of incidence and moving to the higher values, the peak values of alternating stress occur at or near stalling incidence for the stage.

5.3. *Excitation of the Peak Values of the Alternating Stress*.—Three values were obtained for the peak alternating stresses in the 2nd stage stator blade as follow:

Run	*Stage*	*Stress*	*Order*	*Total Stress*	*Orders*	*r.p.m.*
1	2	±24,000	3	±40,000	3 & 2	6,550
3	2	±13,600	3	±16,000	3 & 2	6,700
Constant speed	2	±48,000	2	±55,900	2 & 3	7,000

Each set of readings was obtained from a strain-gauge attached to different blades in the stage and a scatter in amplitude and a slight difference in natural frequency is expected. The method of obtaining the readings could introduce slight differences in the peak values but the above differences in amplitude indicate differences in the excitation. The first two readings were obtained during variable-speed runs at approximately the same position on the pressure characteristic curves and the third reading was obtained at the high pressure point on a constant-speed run. For each case the incidence would be approximately the same value.

The thrust acting on a blade for order m of stage with N_R blades (Ref. 6),

$$\text{is } t_s \propto P_m \sum_{n'=1}^{n'=N_R} \cos m\left(\theta + \frac{2\pi n'}{N_R}\right)$$

which indicates that the first order with large amplitude will be N_R.

This is not the case in practice, low orders persist at all compressor speeds and the assumption of equal loads on all blades is not justified. Irregularities persist in the flow at all times and for the peak values the amplitude of the orders excited by these irregularities, increase. The first two peak values show an increase in the orders but the third peak shows a change from 3rd to 2nd order as the largest, indicating a complete change in the stage of a critical nature (Fig. 16). This is consistent with a partial stall in the stage.

A partial stall will have an effect similar to partial admission in a stage which produces high values of the lower harmonics.

As the incidence is increased beyond the value at the stall the flow in other parts of the stage deteriorates and the contrast between the flow in particular parts of the stage is reduced. The excitation is high relative to the conditions around the design point but is much smaller than that at the partial stall.

Fig. 17 shows that the excitation of the peak stresses is greater the greater the efficiency of the compressor as a whole at the critical condition.

5.4. *Comparison with Cascade Tests.*—Throughout the investigation the predominant harmonic components of alternating stress have been due to vibration forcing the blade in the fundamental flexural mode the frequency of which corresponds to the lower forcing orders 2 to 6. This excitation is due to dissimilar flow in individual blade passages which increases for a critical incidence zone around the stalling incidence for the stage.

The vibration in the cascade for the high incidence zones is stalling flutter exciting the fundamental flexural mode of the blade.

Both of these phenomena could occur in the same compressor and the lower incidence limit could be similar but the upper limit would not be so well defined for the case of stalling flutter.

The stalling flutter would be influenced by the blade stiffness and could occur over wide ranges of operation provided the incidence was greater than the lower limit around the stalling incidence and the mass flow was sufficiently high.

5.5. *Readings Obtained from Strain Rosettes.*—On the second run 45-deg strain rosettes were fitted to the inlet guide vanes and the 2nd stage stator blades. For the fundamental mode of vibration the principal planes were inclined from 3 to 6 deg to the planes including the blade section perpendicular to the line passing through the points of maximum thickness. The values of the maximum-principal stress differed less than 2 per cent from the value of the stress given by the strain gauge placed along the blade. For some of the higher modes there was a torsional component of the stress and the principal planes were not along the blade axes. These components were minor harmonics in the analysis and accurate values could not be obtained.

5.6. *Alternating Stresses from Other Sources of Excitation.*—The stresses corresponding to the blades vibrating in the fundamental mode of vibration predominated for all conditions of operation. The following Table gives the natural frequencies of a set of blades vibrated in the laboratory and the modes which could be detected on the strain records are underlined. In some cases the amplitude for the higher modes was 40 per cent of the amplitude for the fundamental mode.

Natural frequencies of stator blades

Stage	Flexural				Torsional				Complex	
	F1	F2	F3	F4	T1	T2	T3	T4	C1	C2
I.G.V.	276	1,134	2,650	4,898	1,519	3,440	5,590	7,830	1,819	6,209
1	230	1,143	2,873	5,280	1,390	3,262	5,543	7,770		8,692
2	359	1,374	3,590	6,555	1,466	3,695	6,115	9,038	1,770	
3	442	1,780	4,912	8,752	1,640	4,420	7,319	12,900		4,324
4	547	2,652	6,549		2,526	5,901	9,543	14,400		
5	667	3,157	7,320		2,477	5,889	9,510			
6	858	4,040	9,700	15,700	2,760	6,853				
7	1,095	5,430	13,800		2,957	7,370				8,040
8	1,435	6,923			3,340	8,520				

In Table 1 the resonant frequencies given by the strain records are listed and the complex nature of the mode of vibration of the complete stage is shown. For the different runs readings were obtained from gauges on different blades giving different groups of resonant frequencies. The scatter in frequencies is large which is a feature of cast blades.

5.7. *Alternating Stresses in Inlet Guide Vanes.*—The incidences at entry to the inlet guide vanes will be relatively low at all conditions of operation and the high alternating stresses recorded reflect high incidences at entry to the 1st stage rotor blades. There are two zones of high alternating stresses one on the variable-speed runs at 6,700 r.p.m. where the incidence to the 1st stage rotor blade will be approximately 8 deg and one on the constant-speed run at 9,000 r.p.m. the stresses increasing at the low pressure ratios where the incidence to the first stage will be negative and increasing in magnitude as the pressure ratio decreases.

5.8. *Suggested Methods to Reduce the Peak Values of the Alternating Stresses.*—The high alternating stresses in axial-flow compressors occur at part load when the particular stages are operating at high values of incidence. It has been shown in this investigation that the region of excessive alternating stresses occur for critical incidence values.

The high-incidence operation of particular stages of compressors at part load cannot be avoided for high pressure ratios using the usual design parameters. To reduce high-incidence operation at part load the regions of critical incidence should be arranged at low speed and during operation in the critical and high-incidence zones the load on the compressor should be kept to a minimum.

The following aerodynamic features can reduce the zones of high-incidence operation:

(*a*) Compounding

(*b*) Variable-incidence stators in the early stages

(*c*) Reducing the incidence in the early stages by increasing the mass flow in them by bleeding after say, stage 3 or 4 (*d*) Reducing the incidence in the early stages by increasing the axial velocity by changes in effective area

(*e*) Arranging the matching point away from the surge line or below the design speed will tend to reduce incidence a little at all conditions of operation, and hence will move the critical incidence zone to a lower compressor speed.

Mechanical features which can be used to limit stresses caused by large exciting forces will generally result in increased weight but since the high alternating stresses occur only in parts of the compressor the additional weight can be restricted. The following methods can be used to reduce the alternating stresses:

(i) Use of lower design gas bending stresses for stages 1, 2 and 3

(ii) Increased stiffness in the early stages using methods of construction to minimise increase in weight

(iii) Introduction of damping.

6. *Conclusions.*—The amplitudes of the alternating stresses were high at all conditions of operation and were of the order of 10,000 lb/sq in. at the resonance peaks. The predominant harmonic components excited the fundamental natural mode of the blade which corresponds to the frequencies of the lower forcing orders 2 to 6. On three particular runs excessive component stresses were recorded, 24,000 lb/sq in. and 48,000 lb/sq in. for the 2nd stage stator blade and 27,500 lb/sq in. for the inlet guide vane.

The results of this and other investigations show excitation from low orders at all speeds and conditions of operation. The forces on all blades in a stage are not equal indicating dissimilar flow in the individual blade passages. This excitation increases at or near stalling incidence in a critical manner indicating a partial stall in the stage. As the incidence is increased beyond this critical value the flow in other parts of the stage deteriorates and the contrast between the flow in particular parts of the stage is reduced. The excitation is high relative to the conditions around the design point but is much smaller than that at the partial stall.

The phenomenon investigated occurs for critical operating conditions and is different from stalling flutter. Both phenomena could occur in the same compressor. Stalling flutter would be influenced by the blade stiffnesses and could occur over wide ranges of operation provided the incidence and mass flow are greater than the lower boundaries.

REFERENCES

No.	*Author*	*Title, etc.*
1	A. R. Howell	The present basis of axial-flow compressor design. Part I: Cascade theory and performance. R. & M. 2095. June, 1942. Part II: Compressor theory and performance. R.A.E. Report E.3961. December, 1942.
2	A. D. S. Carter	The low-speed performance of related aerofoils in cascade. C.P.29. September, 1949.
3	A. R. Howell and R. P. Bonham	Overall and stage characteristics of axial-flow compressors. *Proc. I. Mech. E.*, Vol. 163, W.E.P. No. 60. 1950.
4	H. Constant	*Gas turbines and their problems.* Todd Reference Library, 1948.
5	D. A. Kilpatrick and J. Ritchie	Compressor cascade flutter tests. C.P.187. December, 1953.
6	J. R. Forshaw, H. Taylor and R. Chaplin.	Alternating pressures and blade stresses in an axial-flow compressor. R. & M. 2846. June, 1951.
7	R. A. Jeffs, A. F. Hounsell and R. G. Adams.	Further performance data for aerofoils having C.1, C.2, or C.4 base profiles on circular-arc camber-lines. N.G.T.E. Memo. M.139. A.R.C. 14,755. December, 1951.

TABLE 1

Resonant Frequencies from Strain-gauge Records

Stage	Run	Frequencies								
I.G.V.	1		217		267 276	293 295 298 304	330		398	
	2	190	220		255 270(2) 280(2)	285 295	320 333 334 340	365		
	3	187	225 237	250	270(2) 280	300	320			
1	1		199 200 201		238	253 260		297	315	
	2	183	198 200	214 217	240 243(2)	260 266	276 280	293 298		335
2	1	190	217		257	272(2) 286	294 295(2)	328	374	
	3		212	236		266 268 280	300	330 335	365	
3	1	300	368 369	377 380	400 420	433	460			
4	1	400	430	470	530(2)	550 560(3)	580			
8	1	1,390 1,400	1,440 1,450	1,460 1,470		1,560	1,580 1,600	1,630		
	2		1,450 1,460	1,470 1,490(2)	1,500 1,510(3) 1,520(3) 1,530(4) 1,540	1,560 1,570	1,590		1,820	1,960

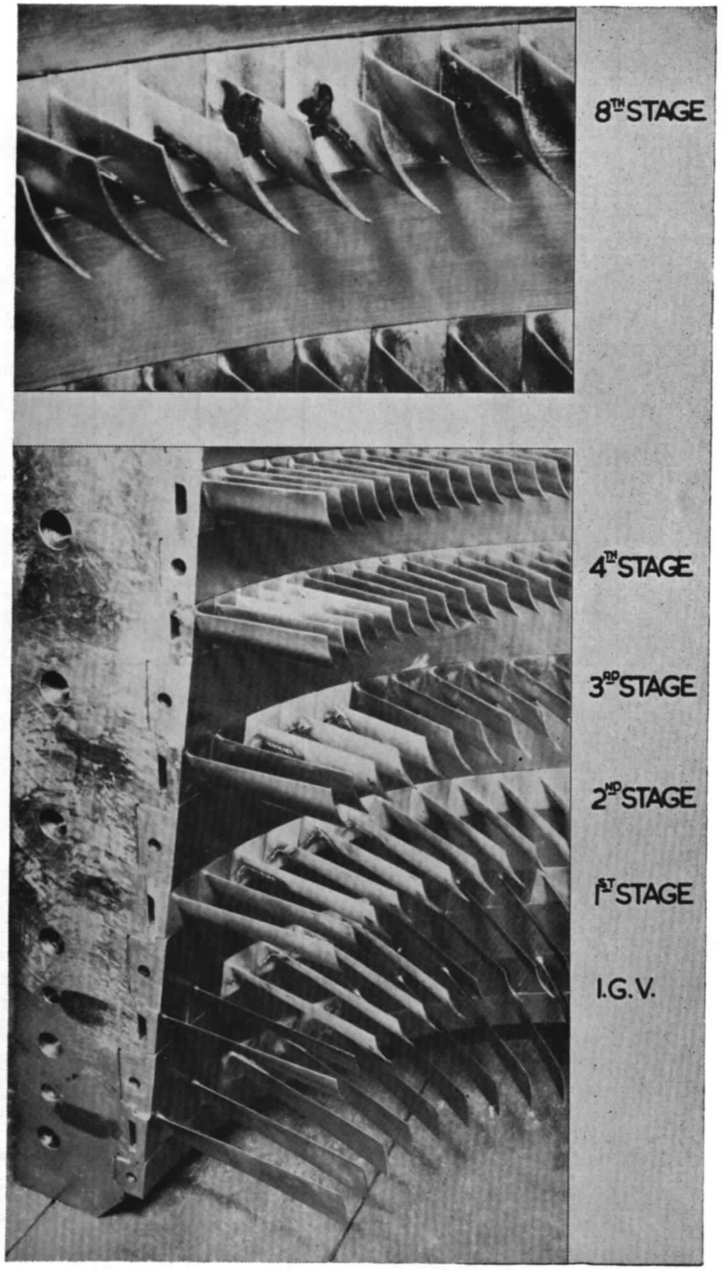

Fig. 1. Strain-gauges attached to stator blades.

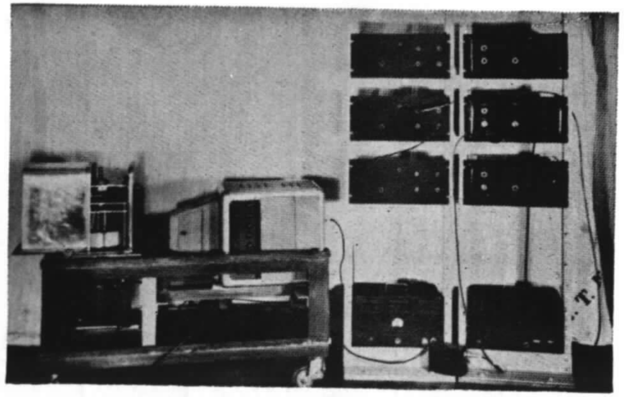

Fig. 2a. Amplifiers and recording equipment.

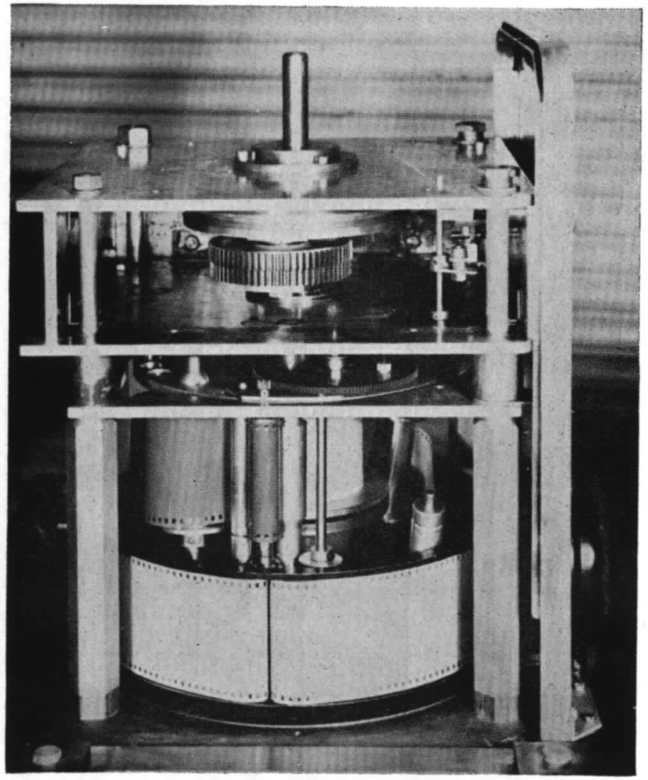

Fig. 2b. Drum camera showing reloading mechanism.

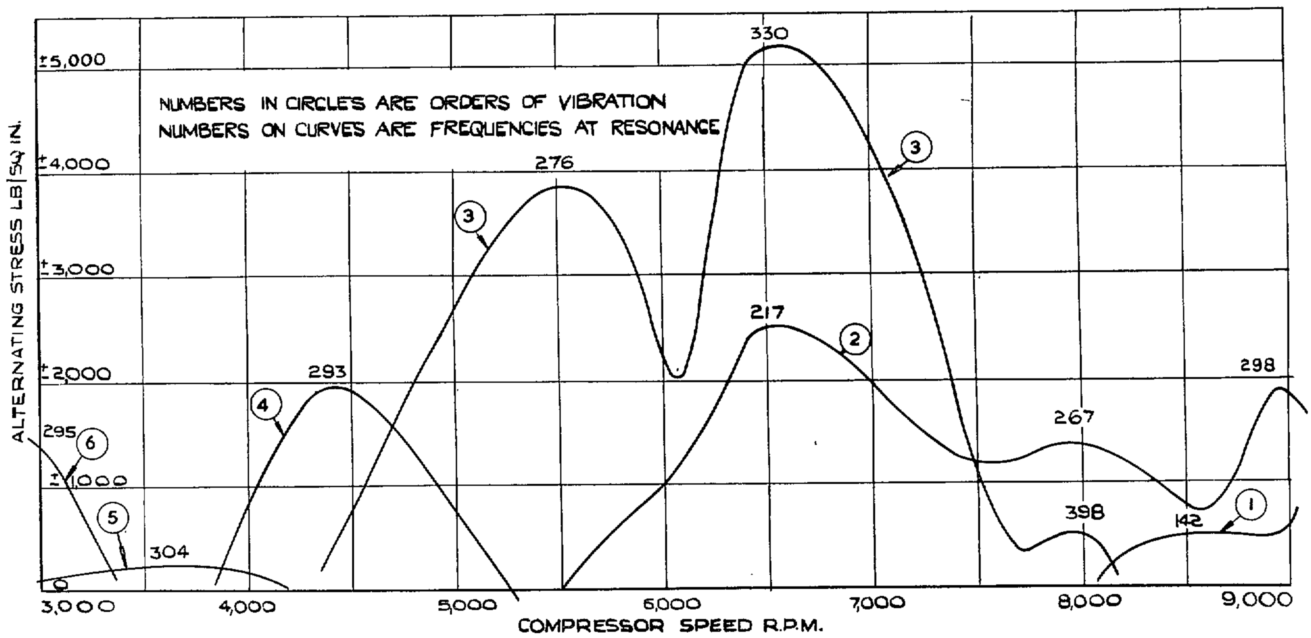

FIG. 3. Predominant harmonic components of stress at root of inlet guide vane. 1st run.

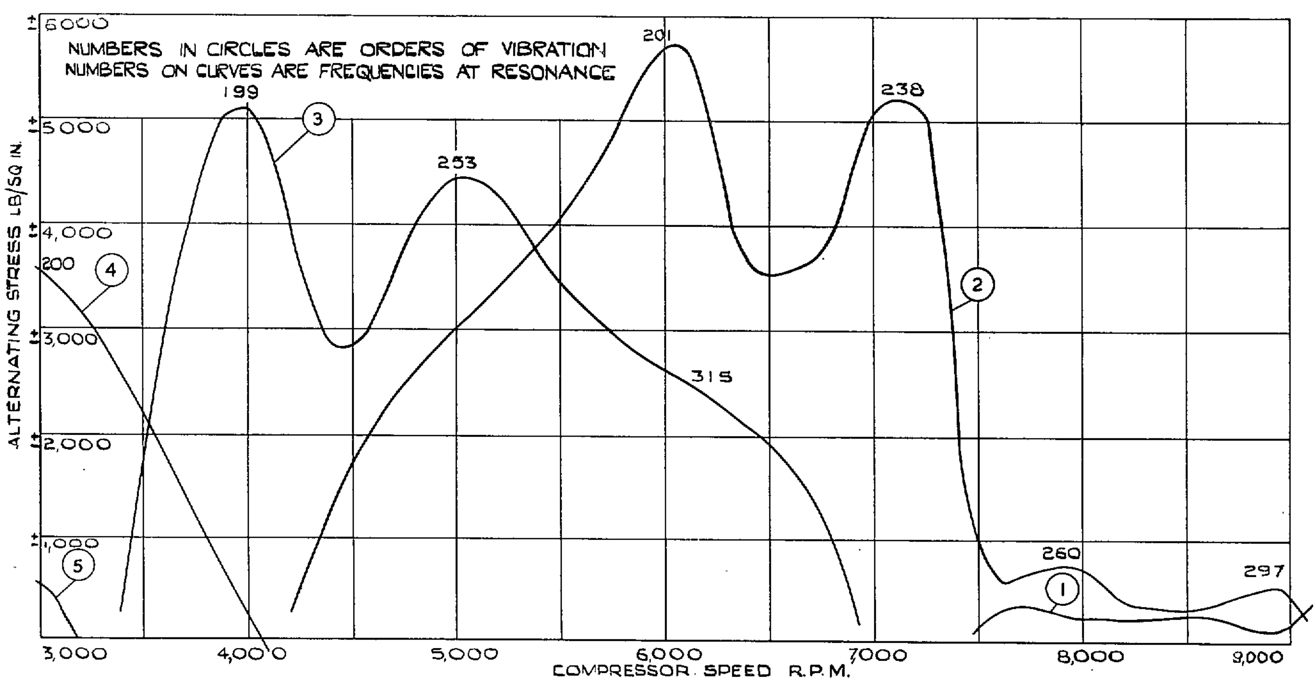

FIG. 4. Predominant harmonic components of stress at root of 1st stage stator blade. 1st run.

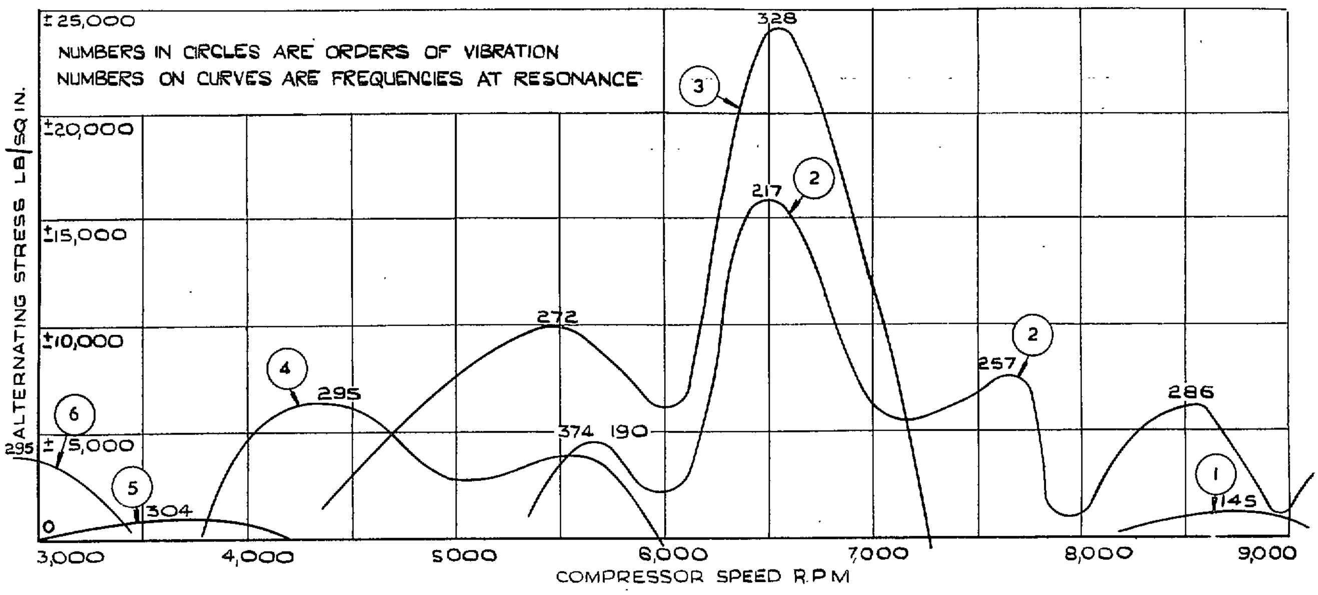

FIG. 5. Predominant harmonic components of stress at root of 2nd stage stator blade. 1st run.

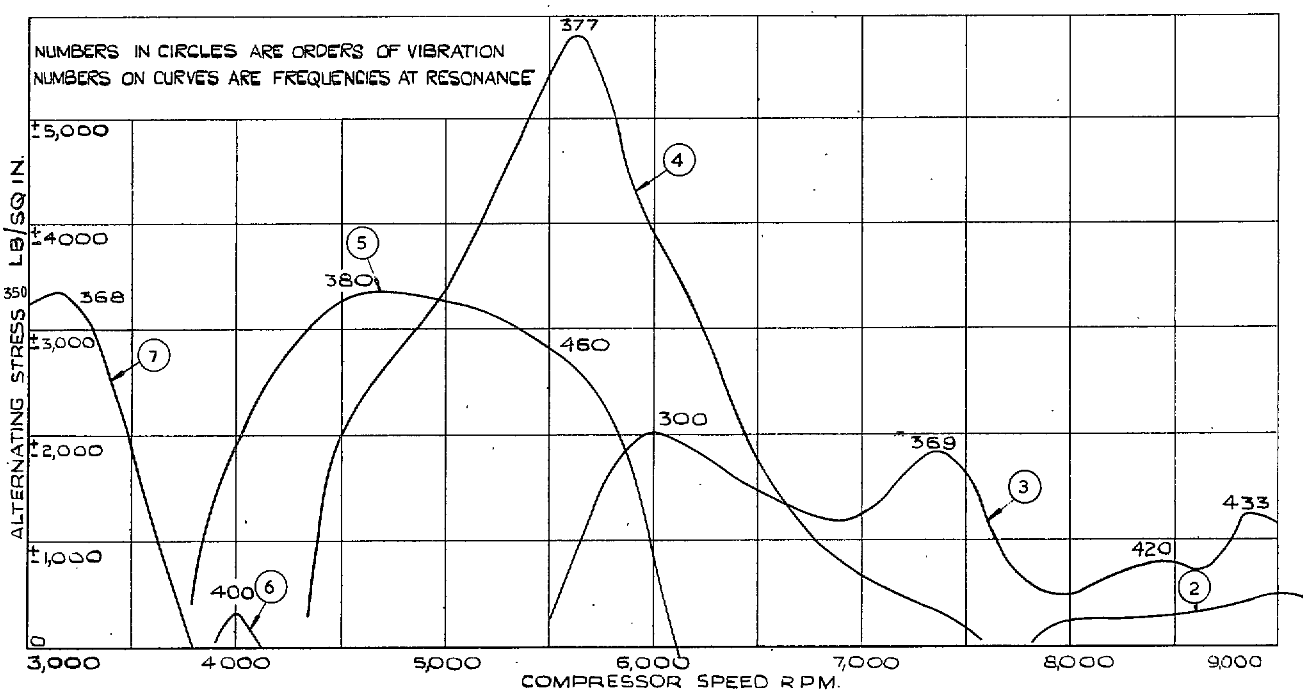

FIG. 6. Predominant harmonic components of stress at root of 3rd stage stator blade. 1st run.

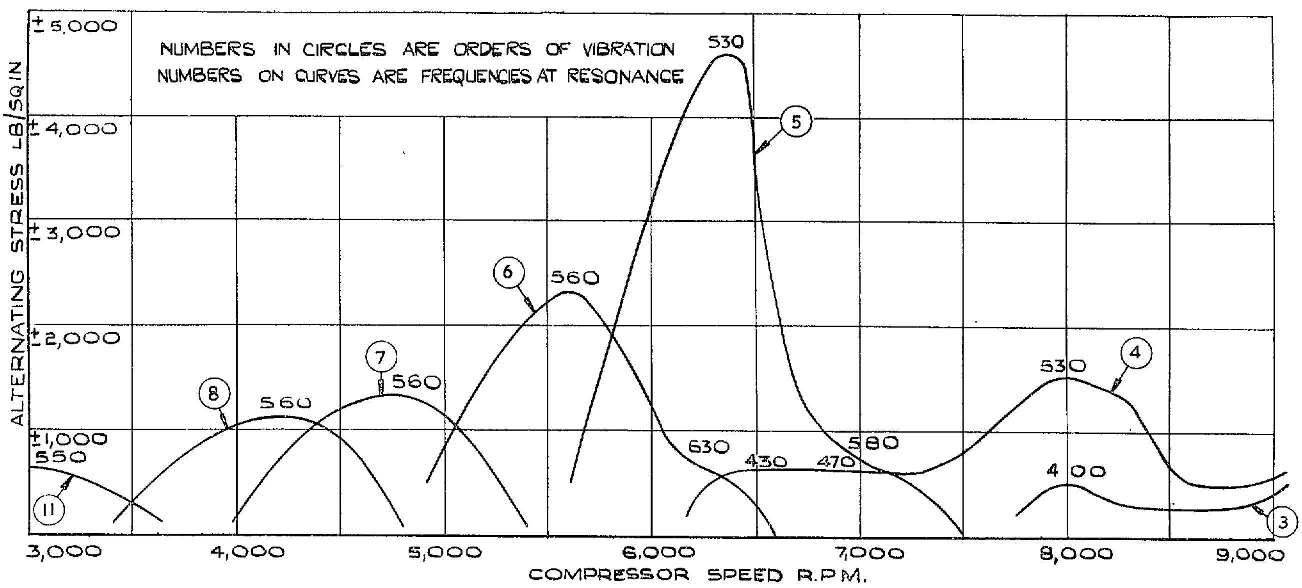

FIG. 7. Predominant harmonic components of stress at root of 4th stage stator blade. 1st run.

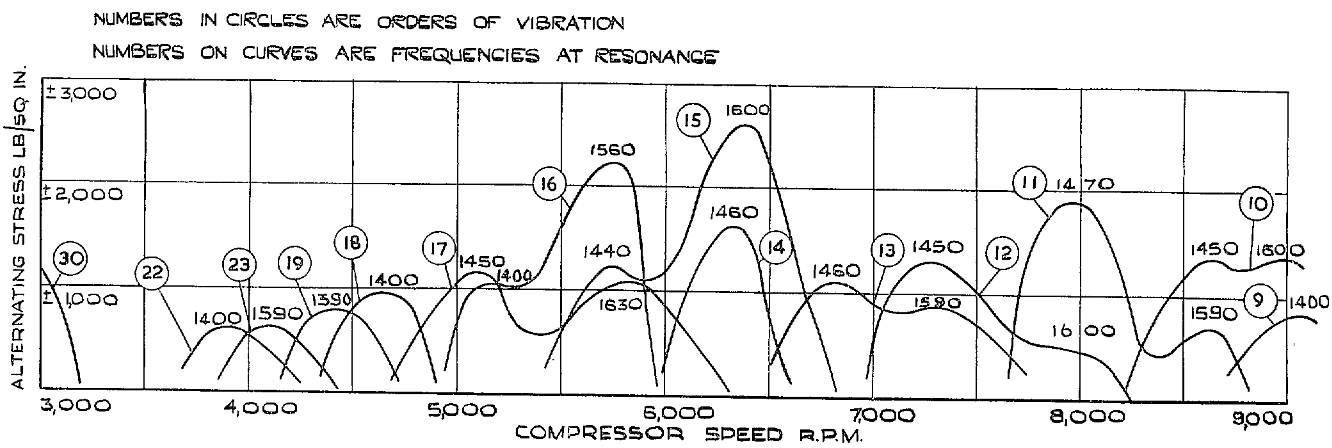

FIG. 8. Predominant harmonic components of stress at root of 8th stage stator blade. 1st run.

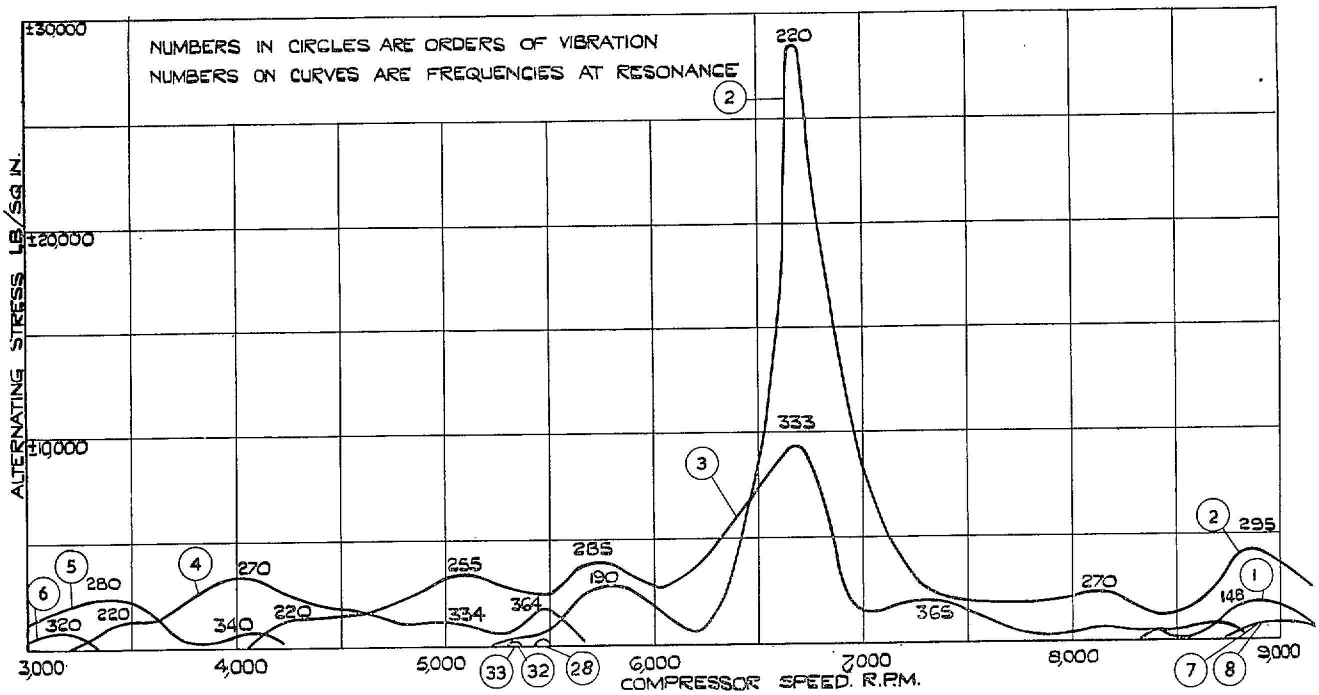

FIG. 9. Predominant harmonic components of stress at root of inlet guide vanes. 2nd run.

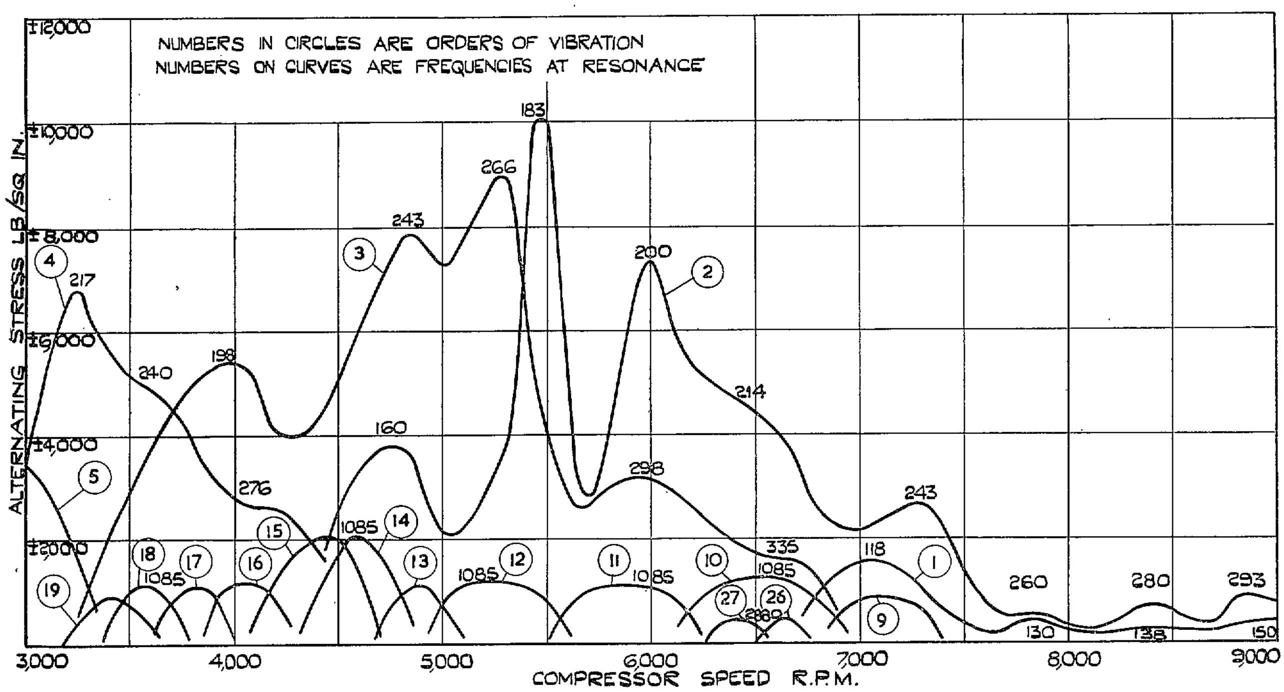

FIG. 10. Predominant harmonic components of stress at root of 1st stage stator blade. 2nd run.

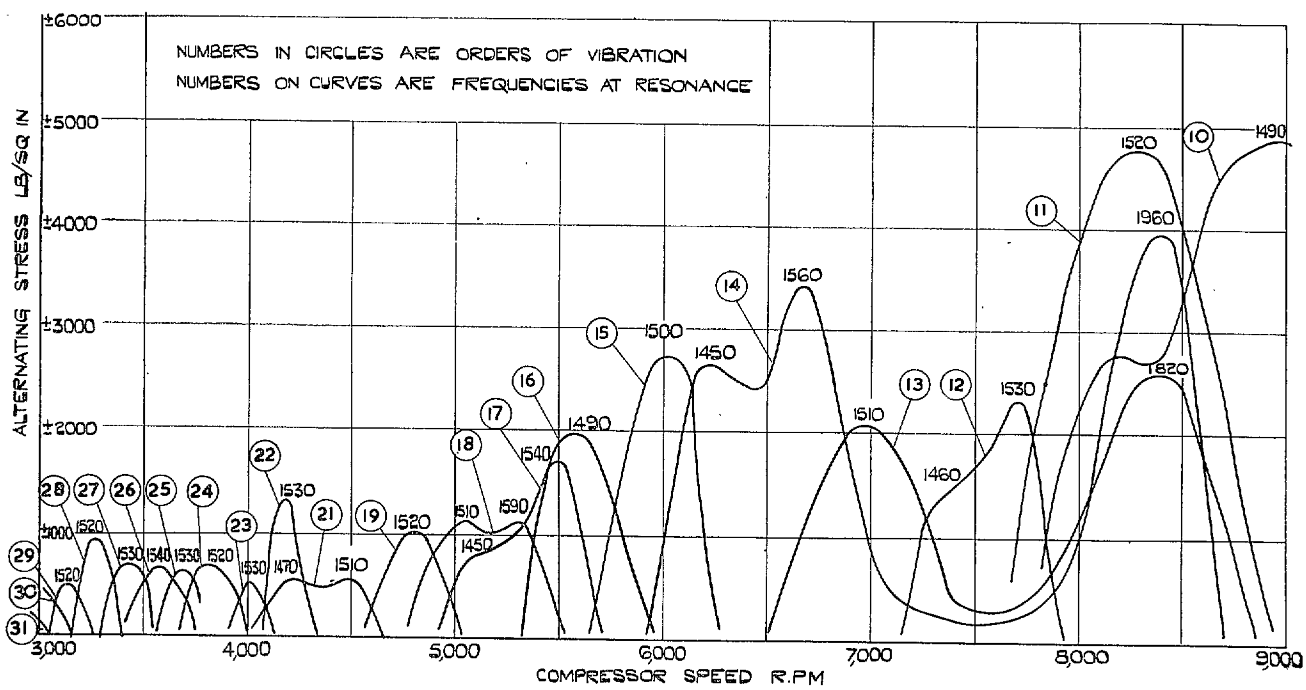

FIG. 11. Predominant harmonic components of stress at root of 8th stage stator blade. 2nd run.

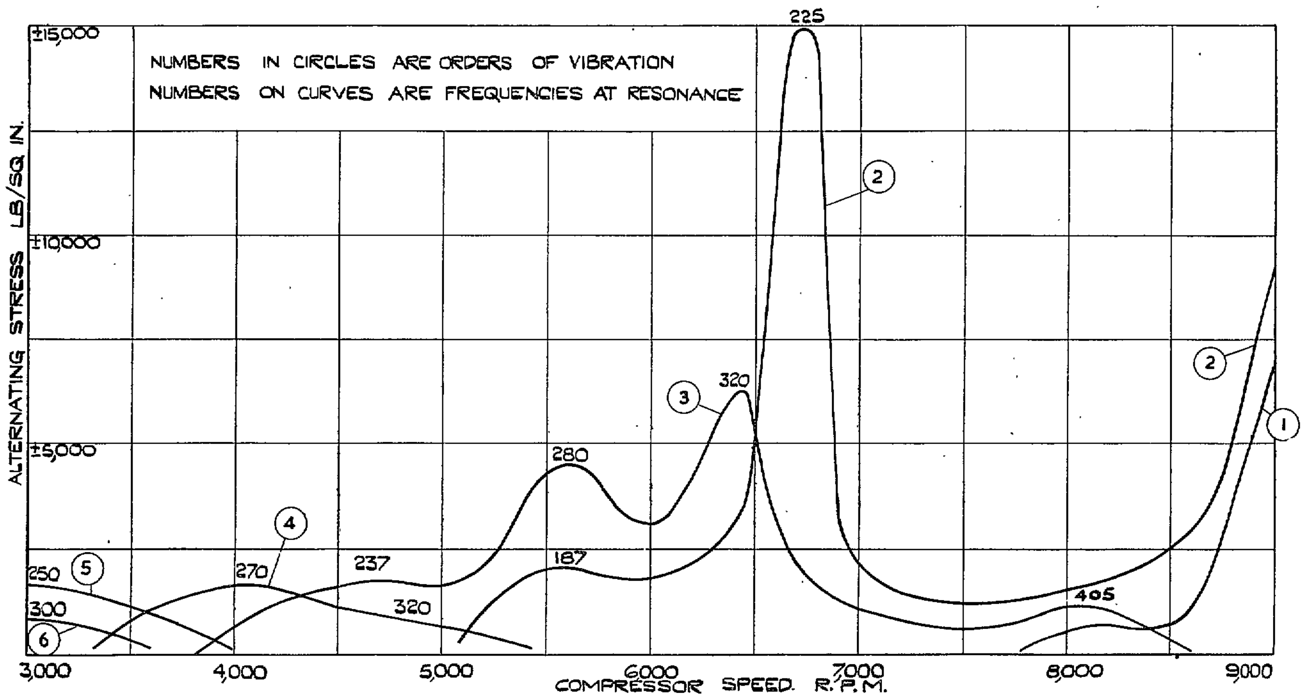

FIG. 12. Predominant harmonic components of stress at root of inlet guide vane. 3rd run.

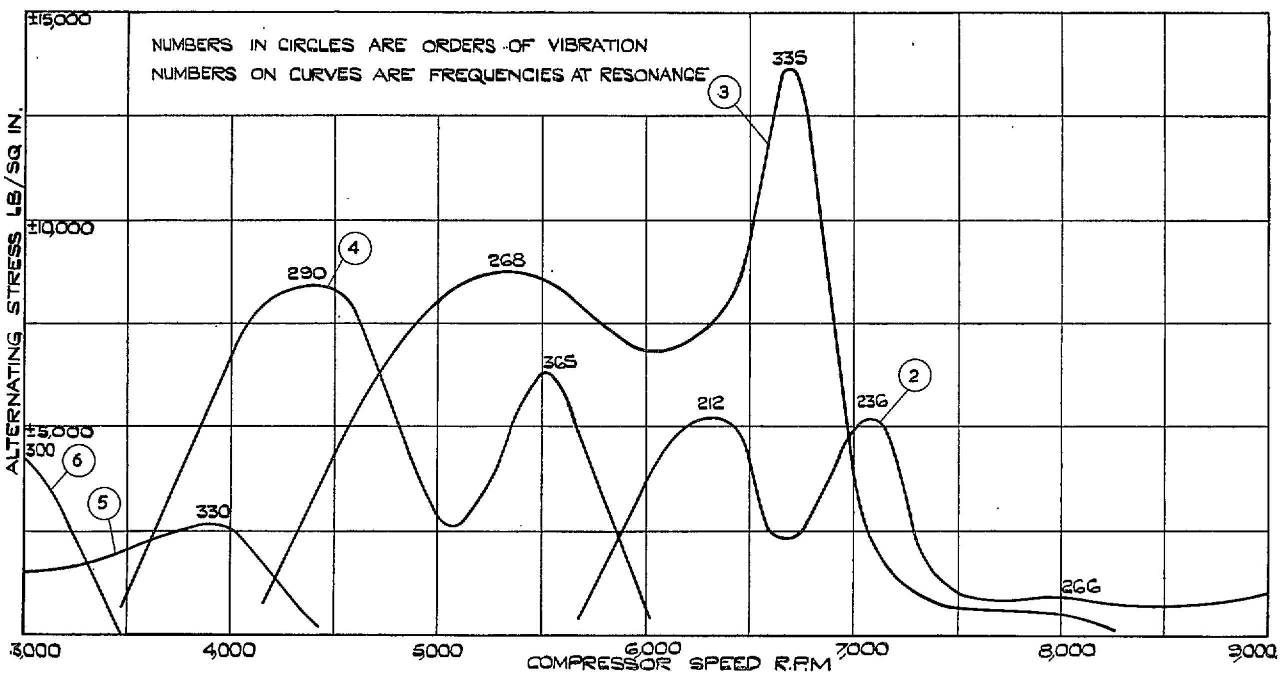

FIG. 13. Predominant harmonic components of stress at root of 2nd stage stator blade. 3rd run.

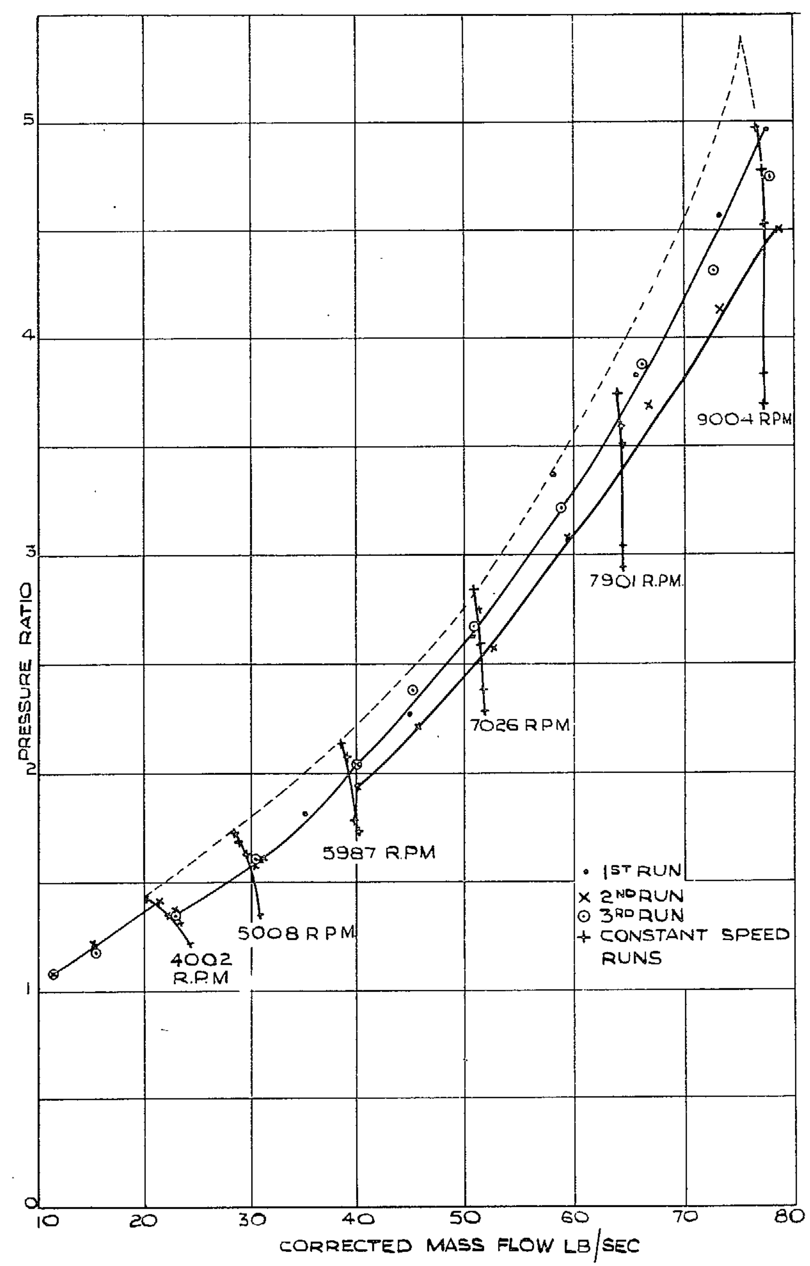

Fig. 14. Operating conditions for the various runs.

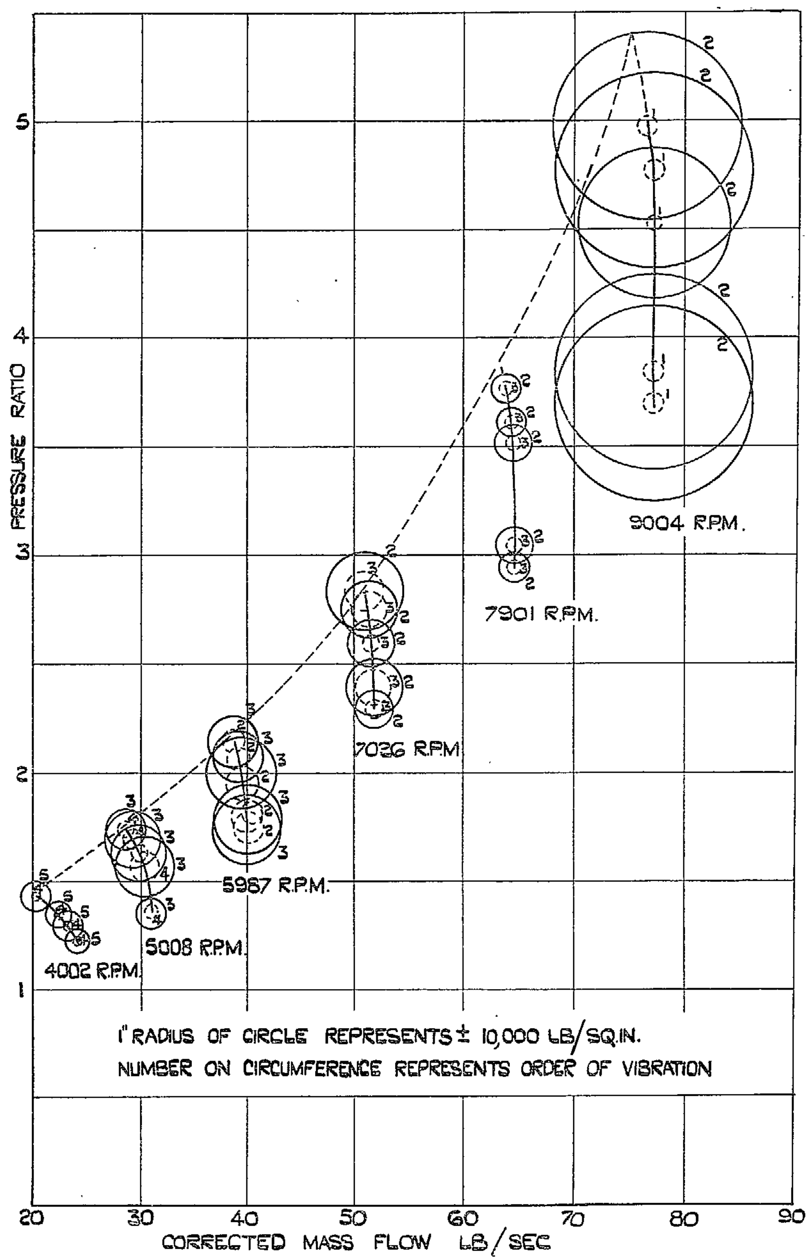

Fig. 15. Alternating stress at root of inlet guide vane plotted on characteristic curves. 4th run.

(4953) Wt. 20/9036 K.7 5/57 Hw.

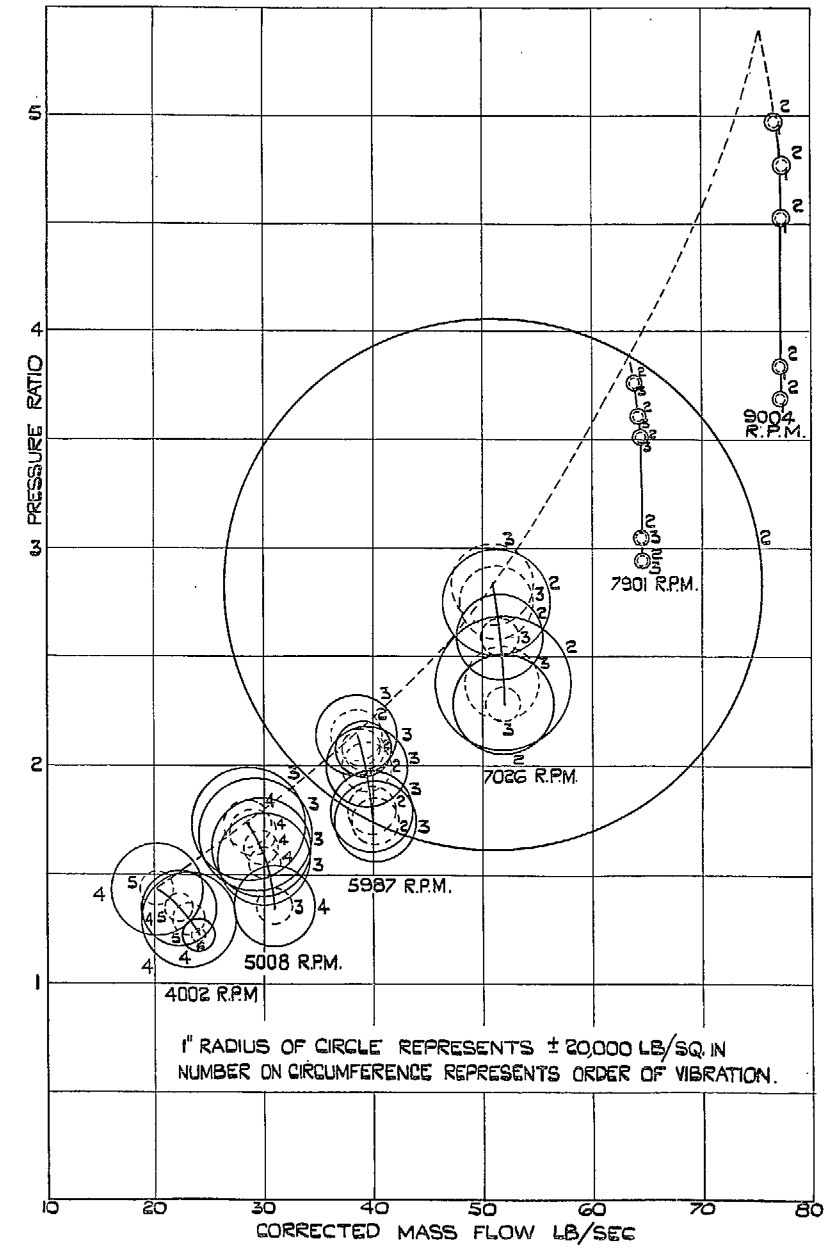

FIG. 16. Alternating stress at root of 2nd stage stator blade plotted on characteristic curves. 4th run.

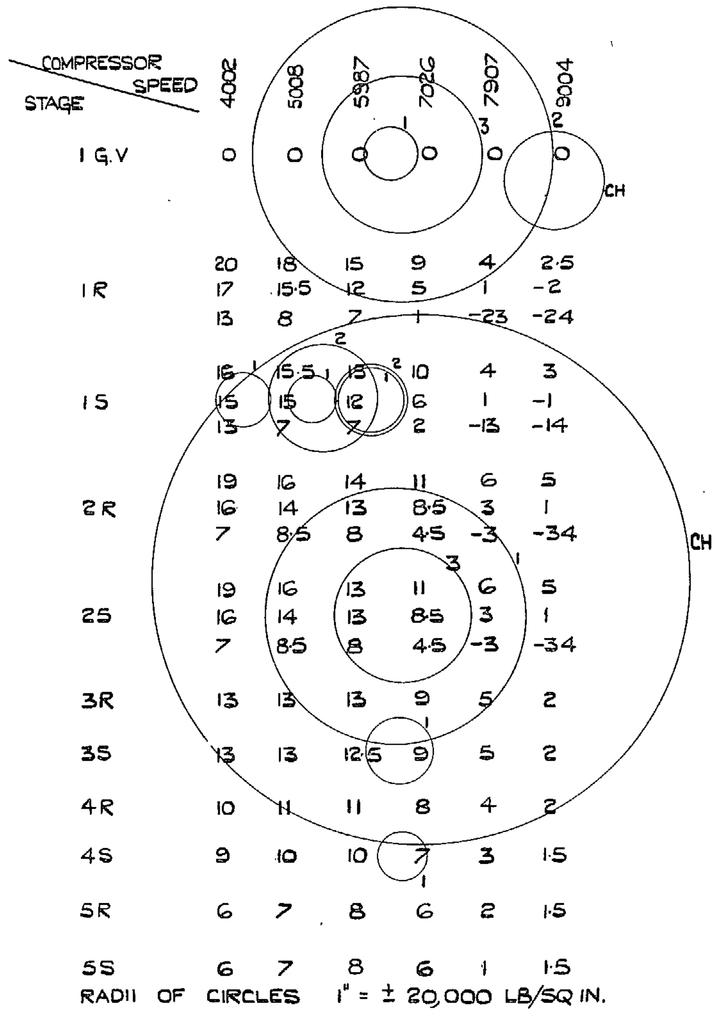

FIG. 17. Peak values of alternating stress plotted on approximate incidence values.

PRINTED IN GREAT BRITAIN

Publications of the Aeronautical Research Council

ANNUAL TECHNICAL REPORTS OF THE AERONAUTICAL RESEARCH COUNCIL (BOUND VOLUMES)

1939 Vol. I. Aerodynamics General, Performance, Airscrews, Engines. 50s. (51s. 9d.).
Vol. II. Stability and Control, Flutter and Vibration, Instruments, Structures, Seaplanes, etc. 63s. (64s. 9d.)

1940 Aero and Hydrodynamics, Aerofoils, Airscrews, Engines, Flutter, Icing, Stability and Control Structures, and a miscellaneous section. 50s. (51s. 9d.)

1941 Aero and Hydrodynamics, Aerofoils, Airscrews, Engines, Flutter, Stability and Control Structures. 63s. (64s. 9d.)

1942 Vol. I. Aero and Hydrodynamics, Aerofoils, Airscrews, Engines. 75s. (76s. 9d.)
Vol. II. Noise, Parachutes, Stability and Control, Structures, Vibration, Wind Tunnels. 47s. 6d. (49s. 3d.)

1943 Vol. I. Aerodynamics, Aerofoils, Airscrews. 80s. (81s. 9d.)
Vol. II. Engines, Flutter, Materials, Parachutes, Performance, Stability and Control, Structures. 90s. (92s. 6d.)

1944 Vol. I. Aero and Hydrodynamics, Aerofoils, Aircraft, Airscrews, Controls. 84s. (86s. 3d.)
Vol. II. Flutter and Vibration, Materials, Miscellaneous, Navigation, Parachutes, Performance, Plates and Panels, Stability, Structures, Test Equipment, Wind Tunnels. 84s. (86s. 3d.)

1945 Vol. I. Aero and Hydrodynamics, Aerofoils. 130s. (132s. 6d.)
Vol. II. Aircraft, Airscrews, Controls. 130s. (132s. 6d.)
Vol. III. Flutter and Vibration, Instruments, Miscellaneous, Parachutes, Plates and Panels, Propulsion. 130s. (132s. 3d.)
Vol. IV. Stability, Structures, Wind Tunnels, Wind Tunnel Technique. 130s. (132s. 3d.)

Annual Reports of the Aeronautical Research Council—

1937 2s. (2s. 2d.) 1938 1s. 6d. (1s. 8d.) 1939–48 3s. (3s. 3d.)

Index to all Reports and Memoranda published in the Annual Technical Reports, and separately—

April, 1950 - - - - R. & M. 2600 2s. 6d. (2s. 8d.)

Author Index to all Reports and Memoranda of the Aeronautical Research Council—

1909—January, 1954 R. & M. No. 2570 15s. (15s. 6d.)

Indexes to the Technical Reports of the Aeronautical Research Council—

December 1, 1936—June 30, 1939	R. & M. No. 1850	1s. 3d. (1s. 5d.)
July 1, 1939—June 30, 1945	R. & M. No. 1950	1s. (1s. 2d.)
July 1, 1945—June 30, 1946	R. & M. No. 2050	1s. (1s. 2d.)
July 1, 1946—December 31, 1946	R. & M. No. 2150	1s. 3d. (1s. 5d.)
January 1, 1947—June 30, 1947	R. & M. No. 2250	1s. 3d. (1s. 5d.)

Published Reports and Memoranda of the Aeronautical Research Council—

Between Nos. 2251–2349	R. & M. No. 2350	1s. 9d. (1s. 11d.)
Between Nos. 2351–2449	R. & M. No. 2450	2s. (2s. 2d.)
Between Nos. 2451–2549	R. & M. No. 2550	2s. 6d. (2s. 8d.)
Between Nos. 2551–2649	R. & M. No. 2650	2s. 6d. (2s. 8d.)

Prices in brackets include postage

HER MAJESTY'S STATIONERY OFFICE

York House, Kingsway, London W.C.2; 423 Oxford Street, London W.1 (Post Orders: P.O. Box 569, London S.E.1) 13a Castle Street, Edinburgh 2; 39 King Street, Manchester 2; 2 Edmund Street, Birmingham 3; 109 St. Mary Street, Cardiff; Tower Lane, Bristol, 1; 80 Chichester Street, Belfast, *or through any bookseller.*

S.O. Code No. 23-2988