E-44

ARR No. E5G06

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WARTIME REPORT

ORIGINALLY ISSUED

August 1945 as
Advance Restricted Report E5G06

PISTON-RING VIBRATION AND BREAKAGE

By J. C. Nettles and André J. Meyer, Jr.

Aircraft Engine Research Laboratory
Cleveland, Ohio

NACA

WASHINGTON

E-44

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

ADVANCE RESTRICTED REPORT

PISTON-RING VIBRATION AND BREAKAGE

By J. C. Nettles and André J. Meyer, Jr.

SUMMARY

Tests were made to determine the stresses and bending moments required to break pieces 1 inch or less in length from the ends of cast-iron, keystone, compression piston rings. Piston rings were fatigue-tested at amplitudes related to ring clearance. With the piston ring vibrating in the free-free condition, strain gages were used to determine the stress pattern for various modes of vibration. The natural frequencies corresponding to these modes were calculated and frequencies that might break off short pieces were found to be very high. All information necessary to calculate the natural frequencies of piston-ring vibrations in the free condition is included. Observations and high-speed motion pictures were made of vibrating rings assembled on a piston and confined inside a Lucite cylinder.

Measurements made with strain gages verify that the maximum-stress points of vibrating piston rings are at the antinodes or points of maximum deflection. Attempts to break off short pieces of rings by resonant vibration were unsuccessful. The natural frequencies that could cause ring breakage 1 inch or less from the ring gap were found to be above 2000 cycles per second. A suggested source of frequencies of this magnitude in an engine is offered. Fatigue tests made on specimens in bending indicate from considerations of ring clearance that breakage about 1 inch from the end cannot be attributed to axial vibrations but can be the result of radial vibrations.

INTRODUCTION

Piston-ring breakage, especially in the vicinity of the ring gap, has been thought to be caused by vibration (references 1, 2, and 3). It is generally believed that a ring under some conditions of operation is deflected in a wave form within the ring groove as though resonance occurred. This theory is plausible but its effects have been apparent only through indirect evidence, such as uneven wear, discoloration, and increased blow-by. The interpretation of these factors is questionable. Little is known about the origin of ring vibrations and their relations to gas pressures, piston speed, and ring dimensions.

Direct observations have not been made of the motion of a ring while operating in an engine because of the high pressures and temperatures that exist and because of the difficulty of maintaining visibility and still properly lubricating the rapidly moving parts of the full-sized engine. Ring failure occurs primarily at high power outputs, which magnifies these problems. In order to study the flections of a piston ring, an apparatus to simulate vibratory conditions was assembled at the NACA Cleveland laboratory. A piston with a compression ring was placed in a Lucite cylinder and oscillated with limited stroke at various frequencies. In addition to this work, a knowledge of ring vibration was obtained by studying its motions when vibrated as a free body.

APPARATUS AND TEST PROCEDURE

Electronic Exciter

An electronic exciter was used to vibrate a keystone-type compression piston ring in the free state and when it was assembled on a section of a standard piston. The exciting apparatus consisted of a beat-frequency oscillator, a 500-watt amplifier, and an electric coil suspended in a steady magnetic field. A diagrammatic sketch of the test setup is shown in figure 1. The amplified output from the oscillator was passed through the coil. Because of its continually changing polarity, the coil was attracted and repelled; thus a means was provided for reciprocating the piston with ring in place at any frequency up to 1000 cycles per second. For frequencies below 100 cycles per second, the stroke was limited to 0.2 inch. For frequencies above 100 cycles per second, the stroke was limited to an amount corresponding to an acceleration of 250 times the acceleration of gravity. A polished Lucite cylinder was attached to the stationary part of the exciter to simulate the constrained conditions of a ring in an engine.

Free-Free Ring-Vibration Equipment

Two rings were mounted on small flexible springs to obtain free-free conditions. On one ring, strain gages were cemented close together on the ring face and on the other ring they were placed on the top side. The gages, 1/8 inch wide and 3/8 inch long, were of a bakelite-bonded, wire-wound construction especially designed for these tests. Connecting wires to the gages were small and flexible to avoid interference with vibrations. The strain signals after being amplified were recorded on a 12-channel oscillograph.

Equipment for Observing Piston-Ring Motion

Ring fluctuations were observed through the polished Lucite cylinder with a stroboscope and the motions that might cause breakage were photographed with a high-speed motion-picture camera. The camera was a continuous-film, rotating-prism type capable of taking pictures at the rate of 2500 frames per second. The light necessary for high-speed photography was supplied by an 8000-watt carbon arc light supplemented with flood lamps. Some of the high-speed pictures were taken with the cylinder and the exciter assembly mounted on a pivot, which rotated the assembly past the stationary camera to permit a 360° view of the ring.

Equipment for Simulating Engine Conditions

Air pressure above the piston was manually controlled at values varying from 0 to 15 pounds per square inch gage pressure. The power output obtainable from the exciter and the strength of the cylinder limited the amount of pressure that could be used. The Lucite cylinder was machined to give a piston side clearance of 0.015 inch between the piston lands and the cylinder wall, which corresponds to the estimated hot clearance of an actual engine in operation. Blow-by measurements were made with a gas meter in the air-supply line. Air pressure was used to force a thin oil into the cylinder for lubrication.

DISCUSSION AND RESULTS

The type of piston-ring failure that would cause breakage of the rings at approximately 1 inch from the end or gap location was investigated. Similar breakage has often been thought to be the result of ring vibration.

Preliminary Tests

In order to have a basis for comparison, rings were broken by statically loading them at various points. Breaks were made by clamping the ring and by applying concentrated loads at the free end either perpendicular or parallel to the axis of the ring.

Loading applied radially in the plane of the ring caused deflections like those resulting from what is known as radial or parallel-ring vibration. Loads acting on the ring perpendicular to its plane produced deflection curves similar to those formed by axial or transverse-ring vibrations (fig. 2). The fractures were compared with breaks that occurred in actual engine operation and it was found that all three fractures appeared to be the same. The moment required to break the ring in the axial direction averaged slightly less than 24 inch-pounds and in the radial-vibration manner it was approximately 48 inch-pounds. The breaking stress of the cast-iron ring material in bending was experimentally determined as 88,000 pounds per square inch. When the ring was so clamped as to form a 1-inch long cantilever, the axial deflection at the breaking point was about 0.025 inch, whereas the radial deflection was about 0.011 inch. It is feasible therefore to break rings by radial vibration because piston side clearance and ring back clearances would easily allow sufficient freedom.

Ring failure caused by axial vibrations is unlikely because without an extreme amount of piston distortion, the ring will not have sufficient clearance to vibrate at a frequency and amplitude that would cause a high stress point 1 inch from the end. In order to confirm this statement, a ring was clamped to form a 1-inch cantilever and was mechanically deflected more than 5,000,000 times at a rate of 3600 cycles per minute without failure. The deflection was 0.008 inch, which corresponds to an extreme side clearance between the ring and the piston lands. The deflection was increased to 0.016 inch and another 5,000,000 cycles were completed without ring failure. The factor of safety apparent from these tests was taken to indicate that investigation of the effect of combined stresses was not necessary. A ring clamped to form a 3/4-inch cantilever was deflected 0.008 inch and it also would not break. A ring clamped to form a 1/2-inch cantilever deflected the same amount did break after 500,000 cycles. A frequency that would cause high stress points 1/2 inch from the end would be beyond reason.

Free-Free Ring Vibrations

In an attempt to explain the location of ring failure, tests were made to determine the distribution of stresses and the maximum stress points of a vibrating ring. Equations for finding the natural frequencies of free-free circularly curved bars are given in reference 4. These equations, modified for use with English units, and the stresses indicated by strain gages applied to the piston ring were used to compare calculated and observed natural frequencies for the various modes. The method of calculation is explained in the appendix.

The natural frequencies of radial and axial vibrations for a keystone-type compression piston ring used in a 5.5-inch-bore cylinder are presented in the following table:

Mode of vibration	Calculated frequency (cps)	Observed frequency (cps)
Radial vibrations		
1	(a)	96
2	206	210
3	499	500
4	921	1140
5	1472	1610
6	2153	2370
Axial vibrations		
1	(a)	82
2	125	158
3	314	321
4	590	523
5	950	(b)

[a]The frequency equations given in the appendix are not applicable for the first mode of vibration.

[b]The fifth mode of vibration could not be excited with the equipment used.

The discrepancies between calculated and observed values can be explained by the increased mass and stiffness caused by the strain gages and by the variance between the method of suspension and actual free-free conditions.

An analysis of the strain-gage signals clearly indicated that the maximum stresses were located at the antinode points, or points of maximum displacement, with the exception of the antinode located at the extreme end of the ring. Figures 3 and 4 are plots of relative stresses along the length of the ring for radial-ring and axial-ring vibrations. All nodal points marked on these graphs were calculated from the tables in reference 4. The values in the tables were experimentally determined and were carried out only to the fifth and sixth modes for the axial and radial vibrations, respectively. For these highest modes, the closest antinode positions are still more than $1\frac{7}{8}$ inches from the end. From figure 5, in which the antinode position closest to the end of the ring is plotted against frequency, it can be readily seen that breakage 1 inch or closer to the end by resonant vibration would require an extremely high frequency.

Actually many rings were broken simply by exciting them with the electronic exciter at resonant frequencies of the ring but in all cases the break occurred one-half or one-third of the way around the ring. It was not possible to vibrate the ring in such a way as to cause breakage 1 inch from the end.

Ring Vibration in Lucite Cylinder

The tests were continued by confining the ring between standard piston lands and inside of the previously described Lucite cylinder. The piston was securely attached to the electronic-exciter coil and was moved up and down at various frequencies. It was soon noticed that, at a frequency of 2400 cycles per minute, the ring could be made to turn in either direction by slightly manipulating the frequency, the amplitude of vibration, or the air pressure. The rate of rotation was not constant but was usually in the order of 1/2 to 1 rpm.

High-speed motion pictures taken at the condition of 2400 cycles per minute reveal that at some instants the ring was merely moving up and down relative to the ring groove as a result of being simply pushed along by the piston. At another time one end of the ring was lying on the bottom of the ring groove throughout the complete cycle while the other end was vibrating up and down in the groove. At the same frequency, sometimes the end that had remained at rest relative to the groove began oscillating and the other end stopped its motion.

The motion between ends of the ring indicated that the ring was being stressed somewhere around its circumference. The exciter mounted on the pivot was rapidly turned while the ring was vibrating. Additional motion pictures were taken and a study of them showed that the amplitude of ring movement gradually decreased around the ring from the moving to the still end. Very low stresses were consequently being distributed throughout the ring and no maximum stress point that could break the ring existed. Other observations were made with the stroboscope at frequencies related to the natural frequencies of the ring as calculated and observed in the free-free condition. In all cases the ring seemed to be making normal excursions up and down in the cylinder and pictures taken with the high-speed camera confirmed that the ring was making normal excursions. It was impossible to excite the resonance of the ring, probably because considerable damping was supplied by the piston-ring material, the oil, the air pressure, and the confinement of the ring between the lands. It must be remembered that, with the manner of suspension, excitation was possible in only one direction: that leading to axial vibrations.

Analysis of Other Experiments Related to Ring Breakage

Some very interesting experiments concerned with ring breakage are reported in reference 5. In one test, rings of various widths and of rectangular cross sections were run in a small high-speed engine and it was found that the widest rings (0.093 in.) broke at and engine speed of 5000 rpm. A narrower ring (0.062 in.) broke at an engine speed of 6000 rpm. The narrowest ring (0.038 in.) did not break although a speed of 7000 rpm was reached. When the natural-frequency equation for radial or parallel vibrations (equation (1), appendix) is considered, a change in width equally affects the moment of inertia in the numerator and the mass in the denominator. The natural frequency consequently remains the same and would not explain the change in the speed at which the ring broke. For axial vibrations it has already been shown that the limited side clearance will not permit breakage caused by vibration. When it is assumed that sufficient clearance does exist, a careful analysis of the axial-frequency equation (equation (2), appendix) reveals that the natural frequencies for the higher modes of vibration are almost proportional to ring widths. Thus when the ring width is decreased, the natural frequency would decrease instead of increasing, as the previously mentioned speed variations seem to indicate.

A condition where the ends of the rings butt together because of low-mode radial vibrations has been suggested as the cause of ring failure. In the tests reported in reference 5, the ring gaps were made so large that the ring would bottom in its groove before butting. The rings broke, however, close to the ends. It was also found that with a back clearance of 0.034 inch between the ring and the groove, the length of the piece that broke off was 1 inch. When the back clearance was reduced to 0.028 inch, the broken piece was 1/4 inch long; when the back clearance was reduced to 0.014 inch, the piece was only 1/8 inch long. These facts are unexplained by vibration theory because 0.028 inch is more than sufficient clearance to allow the radial-vibration deflections required to break off a 1-inch piece and the same is true of the 0.014-inch back clearance and the 1/4-inch length.

Excessive blow-by was invariably experienced in the tests reported in reference 5 when breakage of piston rings occurred and it was discovered that the sudden increase in blow-by preceded the failure. In an attempt to correlate blow-by with ring vibration, measurements were made of the amount of blow-by at various piston frequencies. In some of the tests the pressure was held constant; in others the amplitude was maintained constant; and in still others, both pressure and amplitude were fixed at definite values. Results of these tests were neither significant nor consistent.

A ring vibrating in an eighth-mode axial manner would deflect in a wave form with 10 nodal points. (The frequency calculation is given in the appendix.) In this case, hot gases would flow by the ring, especially at the nodal points, because the deflection would prevent these points from properly seating against the piston land. The antinode points strike the lands very rapidly; consequently, the points are bright owing to wear and heat removal as well as to interruption of gas leakage at these points. This condition might satisfactorily explain the even spacing and the symmetrical location of the discoloration on the under side of the piston ring shown in figure 6, which indicates that vibration of the ring did exist. This piston ring was removed from a single-cylinder test engine after 50 hours' running time at constant speed and power. Such discoloration has been previously attributed to an uneven surface caused by grinding chatter during manufacture of the ring but such reasoning would not account for the perfect symmetry of the discoloration or the fact that rings operated at other powers seem to show a different number of discoloration points.

Two possible sources of excitation of these high natural frequencies are: the ring passing over the cylinder undulations or honing marks; and the combustion-chamber pressure causing the ring to act like the reed of a musical instrument. This second source would explain why ring breakage is more prevalent at high powers.

SUMMARY OF RESULTS

From the analyses and tests that were made in connection with the vibration and breakage of aircraft-engine keystone compression piston rings, the following results were obtained:

1. The measurements made with strain gages verify that the maximum-stress points of vibrating piston rings are at the antinodes or points of maximum deflection.

2. The natural frequencies that could cause ring breakage 1 inch or less from the end or gap location are higher than 2000 cycles per second. It is feasible that such a frequency can be excited by gas pressures.

3. Fatigue tests made on specimens in bending indicate that ring breakage about 1 inch from the end cannot be attributed to axial vibrations but can be the result of radial vibrations.

Aircraft Engine Research Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio.

APPENDIX

CALCULATION OF NATURAL FREQUENCIES OF FREE-FREE RINGS

The equations for determining the natural frequencies of free-free ring vibrations (reference 4) as modified for use with English units are:

For the radial or parallel vibrations:

$$f_n = \frac{S^2(n - q)^2 \, \theta^2 K_1}{L^2} \left(\frac{EI_y}{m}\right)^{1/2} C_1 \qquad (1)$$

where

f natural frequency, cycles/sec

$S, q, \theta^2 K_1$ constants dependent on central angle of ring (356° for a piston ring)

n mode of vibration

L length of ring at neutral axis, in.

E modulus of elasticity of ring material, lb/sq in.

m mass of ring per unit length, lb/in.

I_y moment of inertia on cross-sectional area of ring about an axis through neutral axis of ring and parallel to ring and cylinder axis, $in.^4$

C_1 constant necessary to adapt equation as developed in reference 4 from metric to English units

For the case of sixth-mode parallel vibrations, the values used in equation (1) are:

$$S = 0.797$$

$$q = 0.21$$

$$\theta^2 K_1 = 2.751$$

$$L = 16.67$$

$$E = 19.4 \times 10^{-6}$$

$$m = 0.0053$$

$$I_y = 74.39 \times 10^{-6}$$

$$C_1 = \sqrt{\frac{980.7}{2.54}} = 19.649$$

For sixth-mode parallel vibrations:

$$f_6 = \frac{0.797^2(6 - 0.21)^2\ 2.751}{16.67^2}\left(\frac{19.4 \times 74.39}{0.0053}\right)^{1/2} 19.649$$

f_6 = 2153 cycles/sec

= 129,180 cycles/min, or 43 order of engine speed at take-off rating

For axial or transverse vibrations:

$$f_n = \frac{Q^2(n - q)^2}{L^2}\,\frac{\theta^2\psi}{}\left(\frac{EI_x}{m}\right)^{1/2} C_1 \tag{2}$$

where

I_x moment of inertia of cross-sectional area of ring about an axis through neutral axis of ring and perpendicular to axis of ring or cylinder, in.^{4}

$Q,q,\theta^2\psi$ constants dependent upon value of $\frac{EI_x}{C}$ where C is torsional rigidity given by βbc^3G (See fig. 7, which is a plot of values given in reference 4.)

β constant dependent upon b/c (See fig. 8, which is a plot of values given in reference 7.)

b,c rectangular length and width substitution for the keystone cross section in accordance with reference 6, in.

G modulus of rigidity of ring material, lb/sq in.

The values and definitions of the symbols E, L, m, and C_1 are the same as for equation (1). For the case of fifth-mode transverse-ring vibrations, the values used in equation (2) are:

$$I_x = 16.58 \times 10^{-6}$$

$$Q = 0.729$$

$$q = -0.300$$

$$\theta^2\psi = 3.67$$

$$n = 5$$

and values for computing C are:

$$\beta = 0.241$$

$$b = 0.215$$

$$c = 0.096$$

$$G = 7.27 \times 10^6$$

The accuracy of the value of C, consequently the values of β, b, c, and G, are not very critical. In fact, a 20-percent error in the value of C results in only a 0.5-percent error in the final answer given by the frequency equation.

For the fifth-mode transverse-ring vibration:

$$f_5 = \frac{0.729^2(5 + 0.300)^2 \; 3.67}{16.67^2}\left(\frac{19.4 \times 16.58}{0.0053}\right)^{1/2} 19.649$$

$$f_5 = 950 \text{ cycles/sec}$$

= 57,000 cycles/min, or 19 order of 3000 rpm take-off speed

For the eighth-mode transverse-ring vibration:

$$f_8 = f_5\left(\frac{8 + 0.300}{5 + 0.300}\right)^2 = 2350 \text{ cycles/sec}$$

REFERENCES

1. Stern, N.: Piston Rings. The Theory Underlying the High Point Design. Proc. Inst. Auto. Eng. (London), vol. 28, no. 367, Jan. 1938, pp. 27-28.

2. Williams, G.: Piston Rings. Design, Material, and Casting Methods in Relation to Performance. Proc. Inst. Auto. Eng. (London), vol. 28, no. 374, Aug. 1938, pp. 305-308.

3. Williams, C. G., and Young, H. A.: Piston Ring Blow-by. Proc. Inst. Auto. Eng. (London), vol. 32, no. 425, July 1942, pp. 283-288.

4. Pliskin, W. A., Edwards, J. E., and Bundy, F. P.: The Vibration Characteristics of "Free-Free" Circularly Curved Bars. Jour. Appl. Phys., vol. 14, no. 8, Aug. 1943, pp. 410-417.

5. Williams, C. G., and Young, H. A.: Piston-Ring Blow-by on High-Speed Petrol Engines. Engineering, vol. 147, no. 3830, June 9, 1939, pp. 693-694; vol. 147, no. 3831, June 16, 1939, pp. 723-726.

6. Timoshenko, S.: Strength of Materials, Part II. D. Van Nostrand Co., Inc., 2d ed., 1941, p. 266.

7. Timoshenko, S.: Strength of Materials, Part I. D. Van Nostrand Co., Inc., 2d ed., 1940, p. 270.

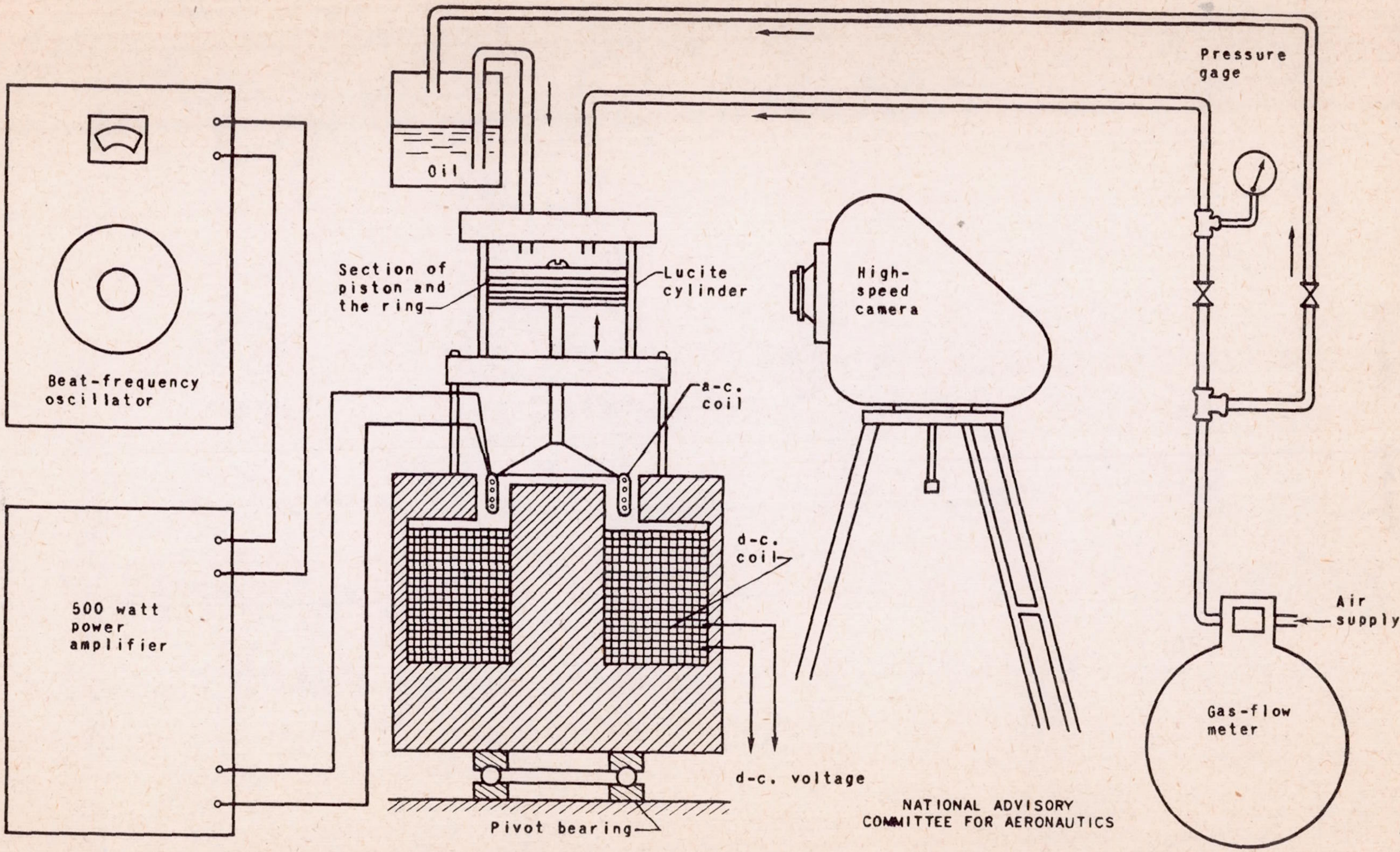
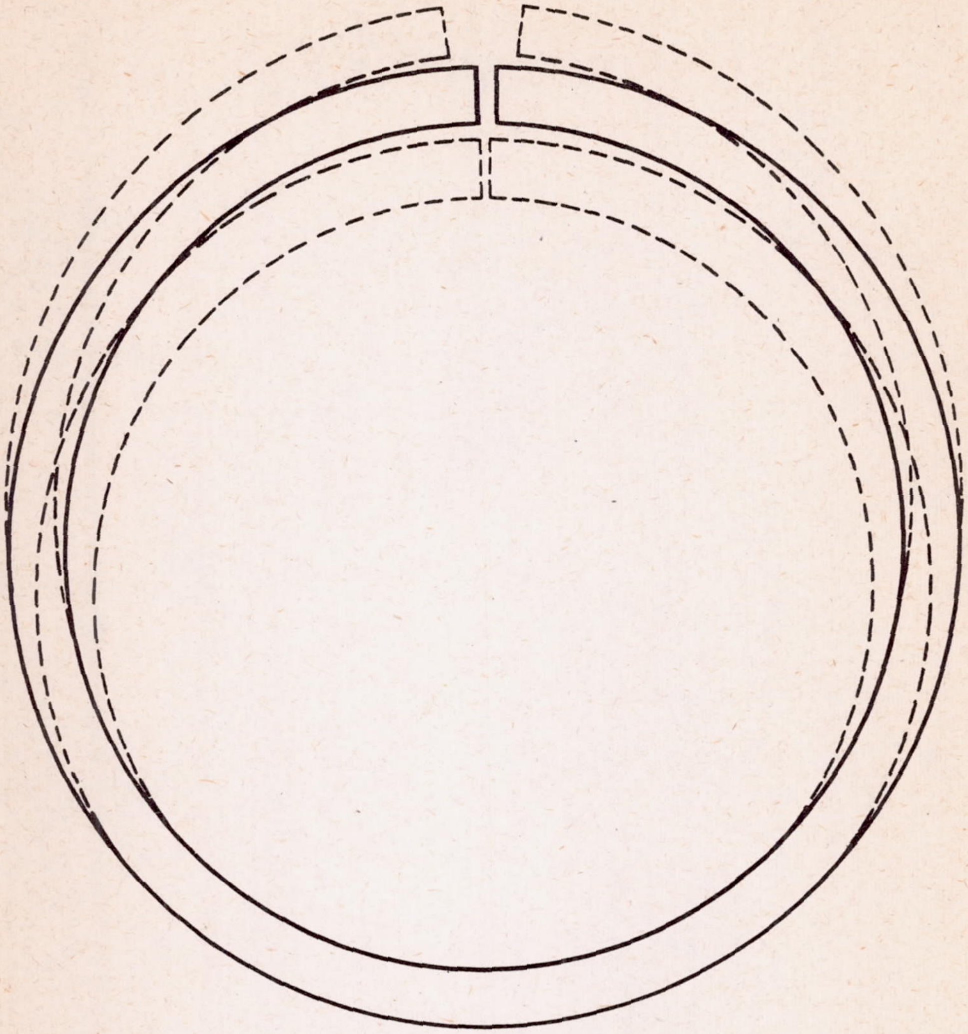
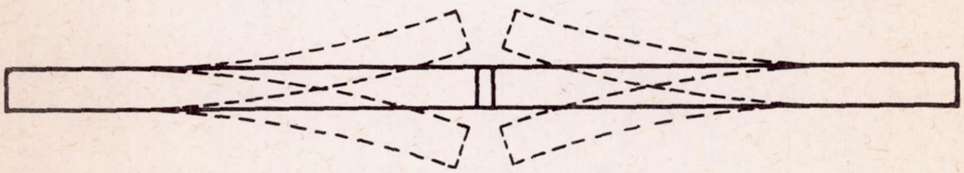


Figure 1. - Arrangement of test equipment used to oscillate and photograph a piston ring in a Lucite cylinder.



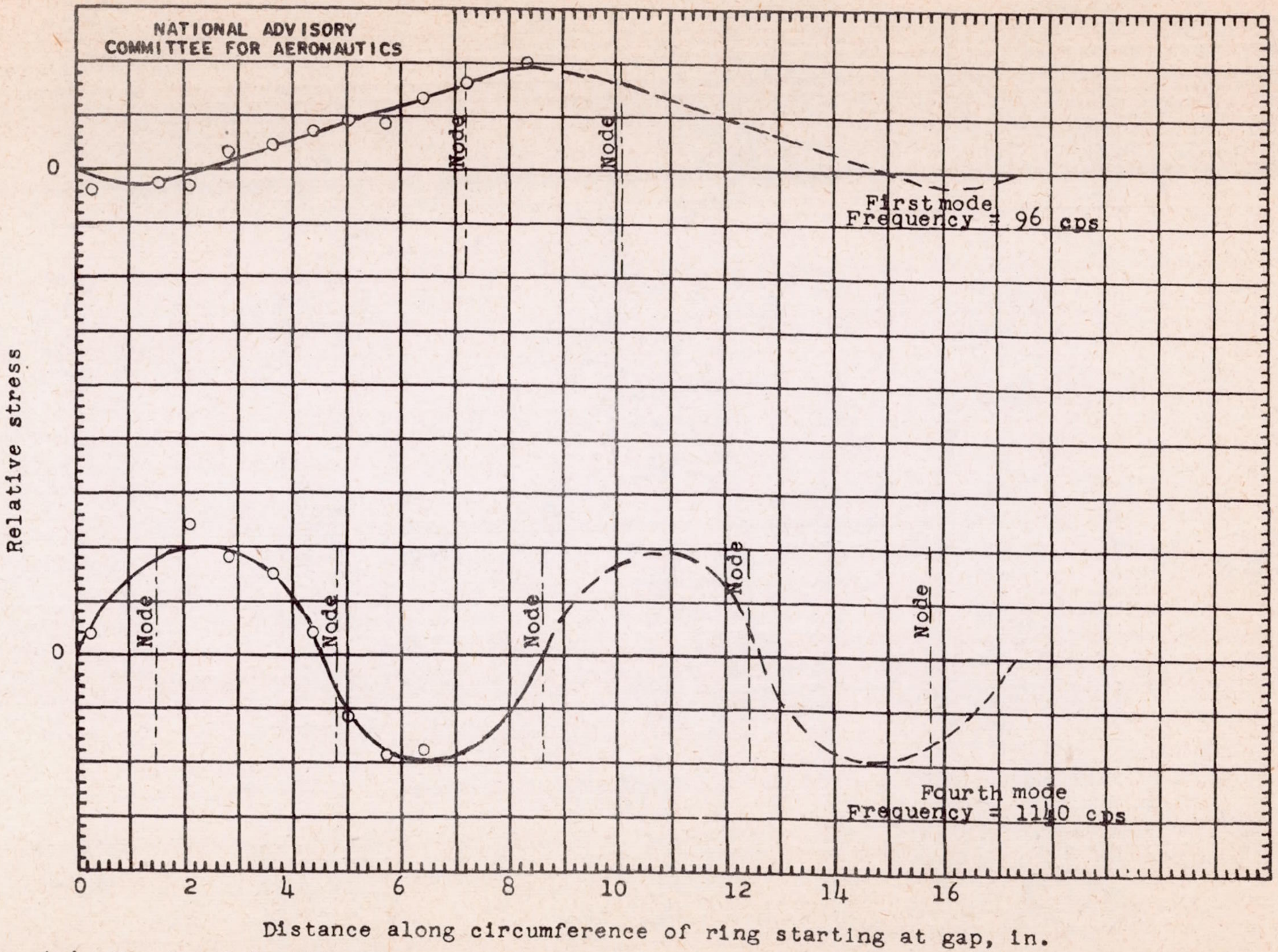
Radial or parallel vibration

NATIONAL ADVISORY
COMMITTEE FOR AERONAUTICS



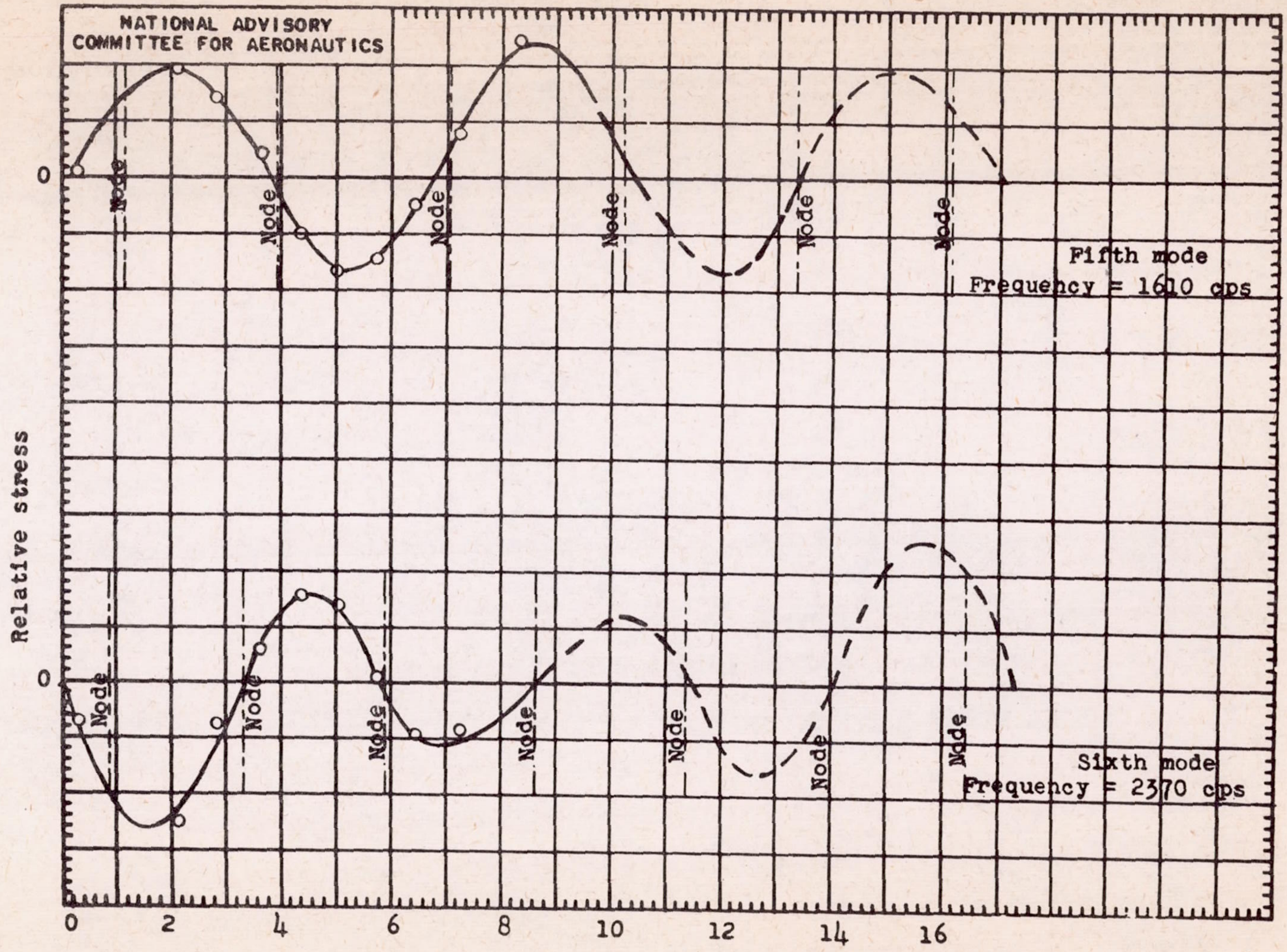
Axial or transverse vibration

Figure 2. - Types of motion associated with vibration of piston rings.



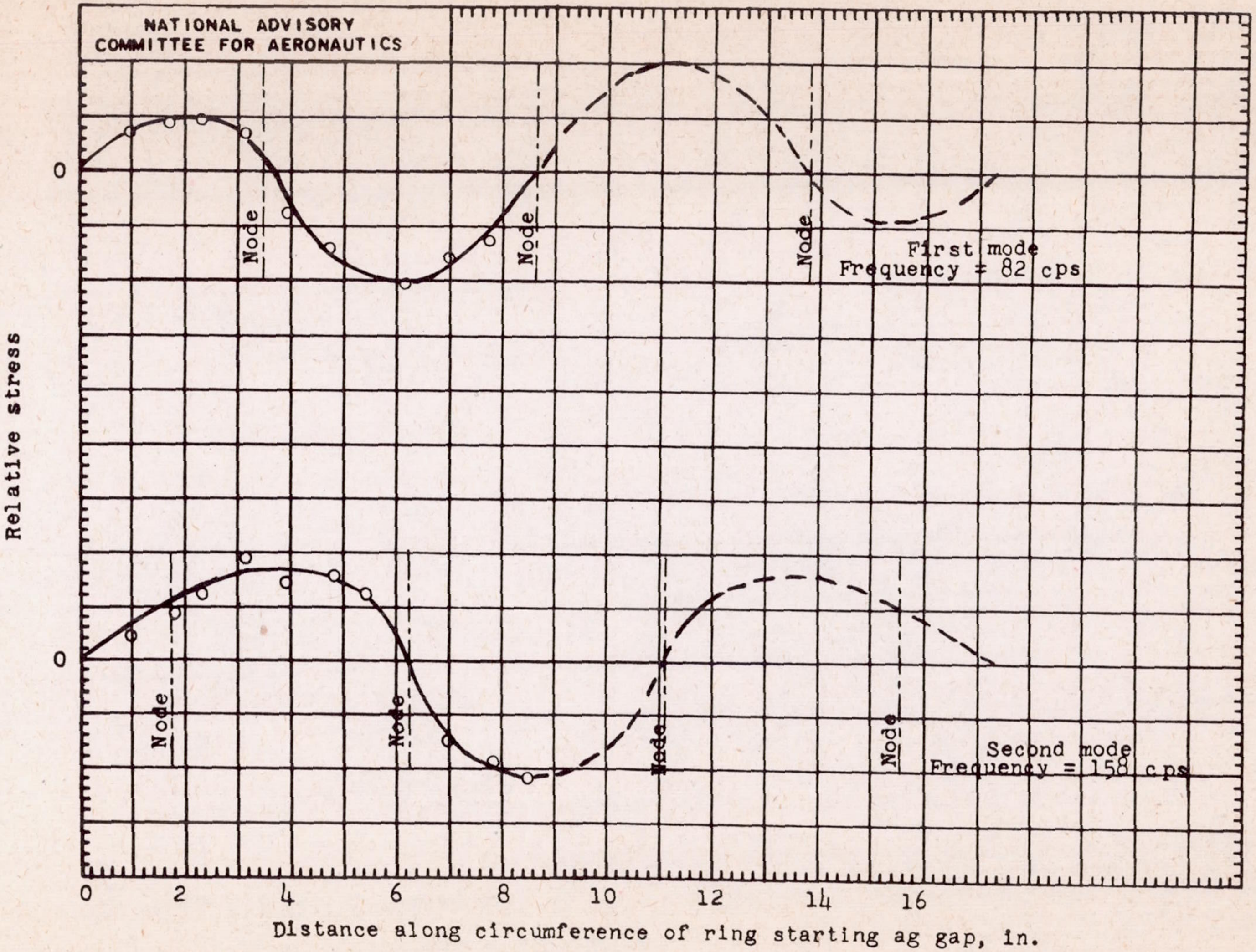
(a) First and fourth modes.

Figure 3. - Stress pattern of radial or parallel-ring vibrations as interpreted from strain gages on a compression piston ring of the keystone type. (Node points calculated from reference 4.)



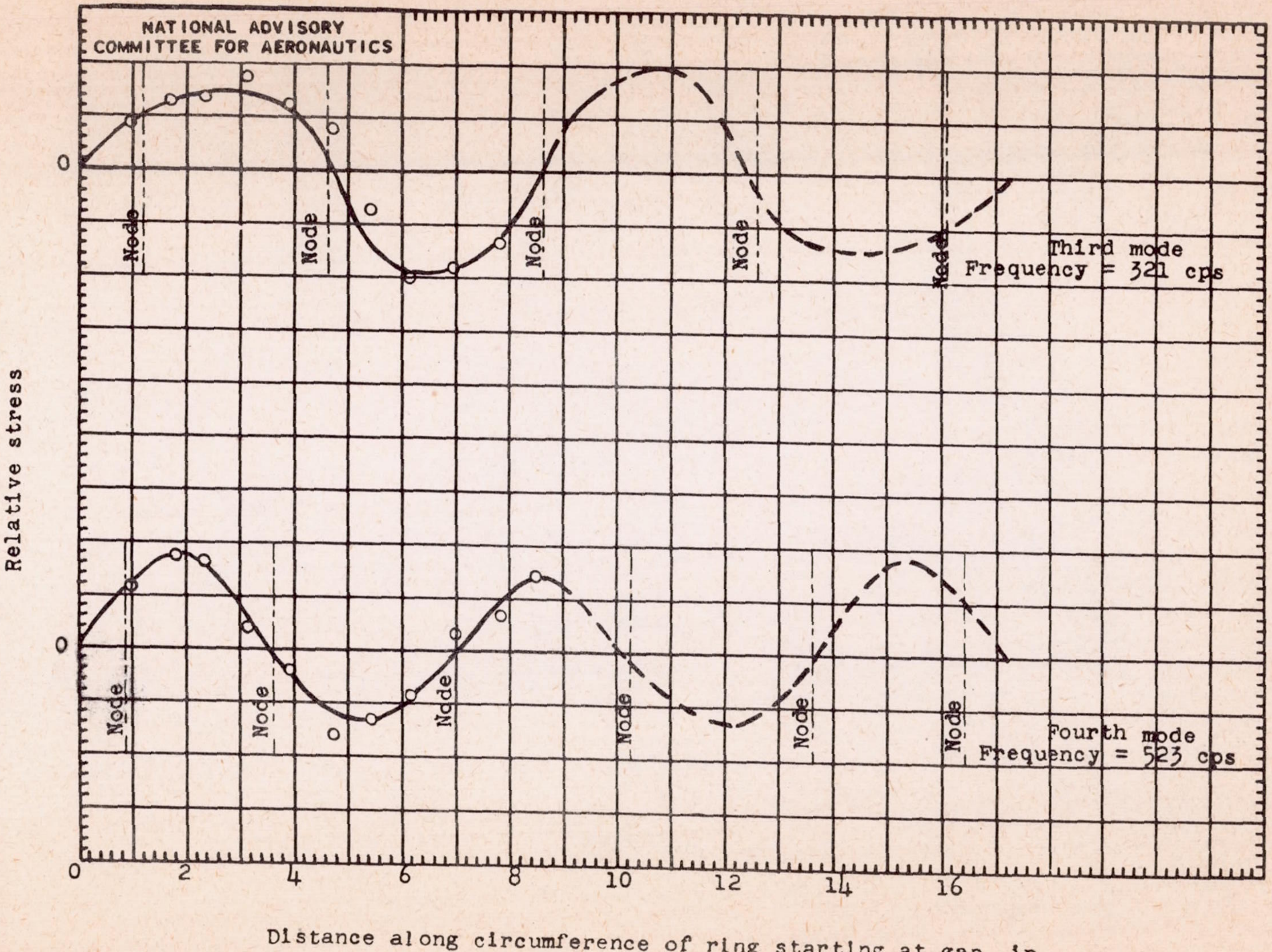
(b) Fifth and sixth modes.

Figure 3. - Concluded.



(a) First and second modes.

Figure 4. - Stress patterns of axial or transverse-ring vibration as shown by strain gages cemented on a compression piston ring of the keystone type. (Node points calculated from reference 4.)



(b) Third and fourth modes.

Figure 4. - Concluded.

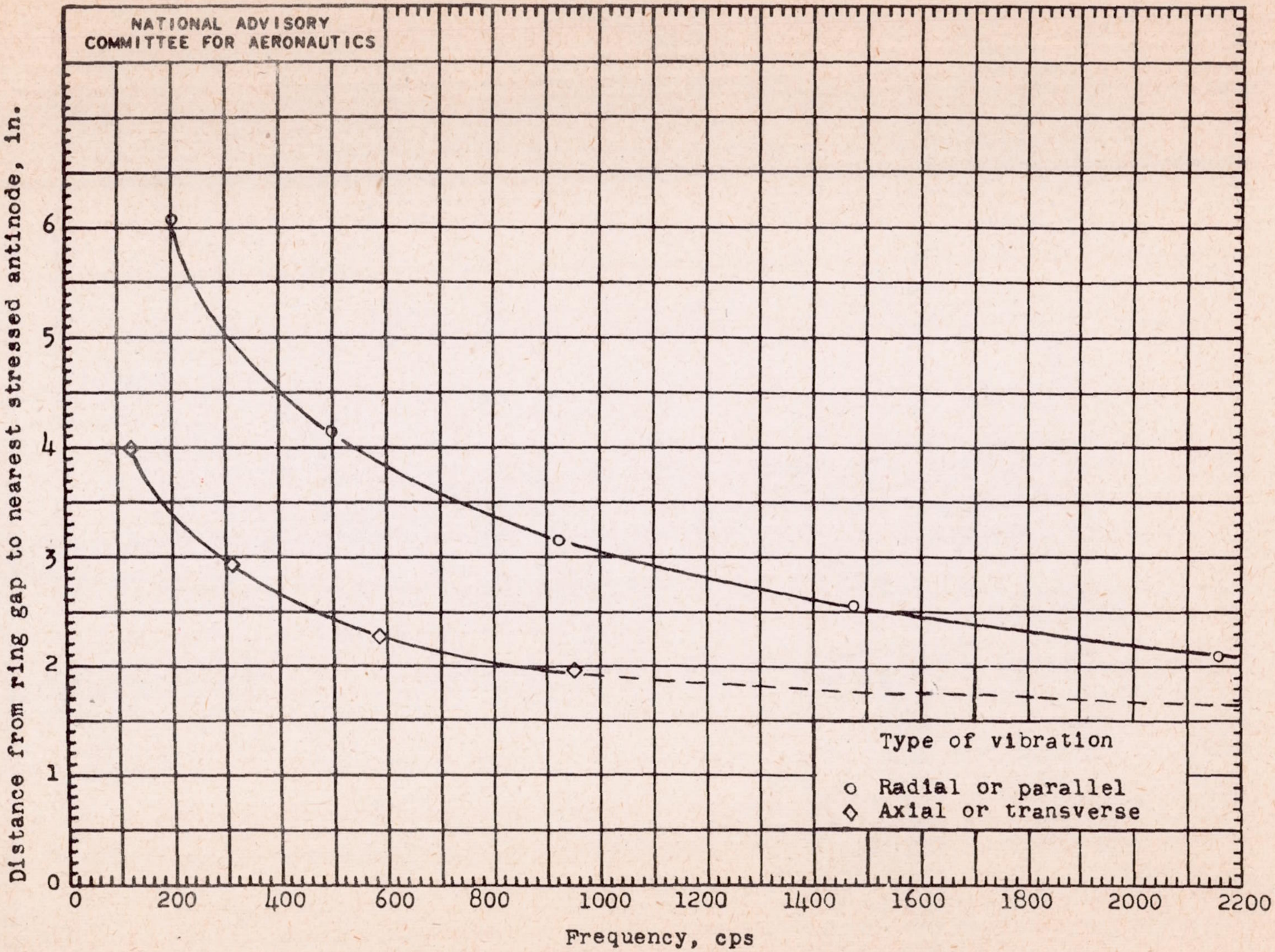
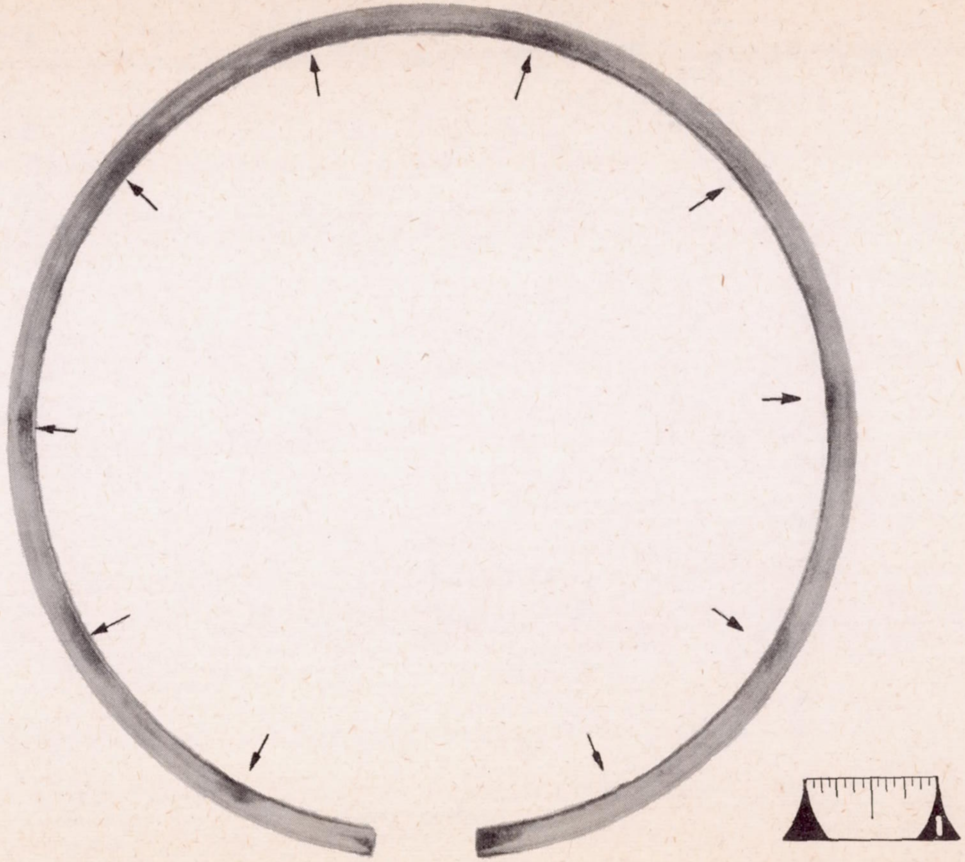


Figure 5. - A plot of natural frequency against the calculated distance from the gap of the ring to the nearest stressed antinode.



NACA
C-6230
8-23-45

Figure 6. - Piston ring showing symmetric and even spacing of discoloration believed to be the result of eighth-mode axial vibration.

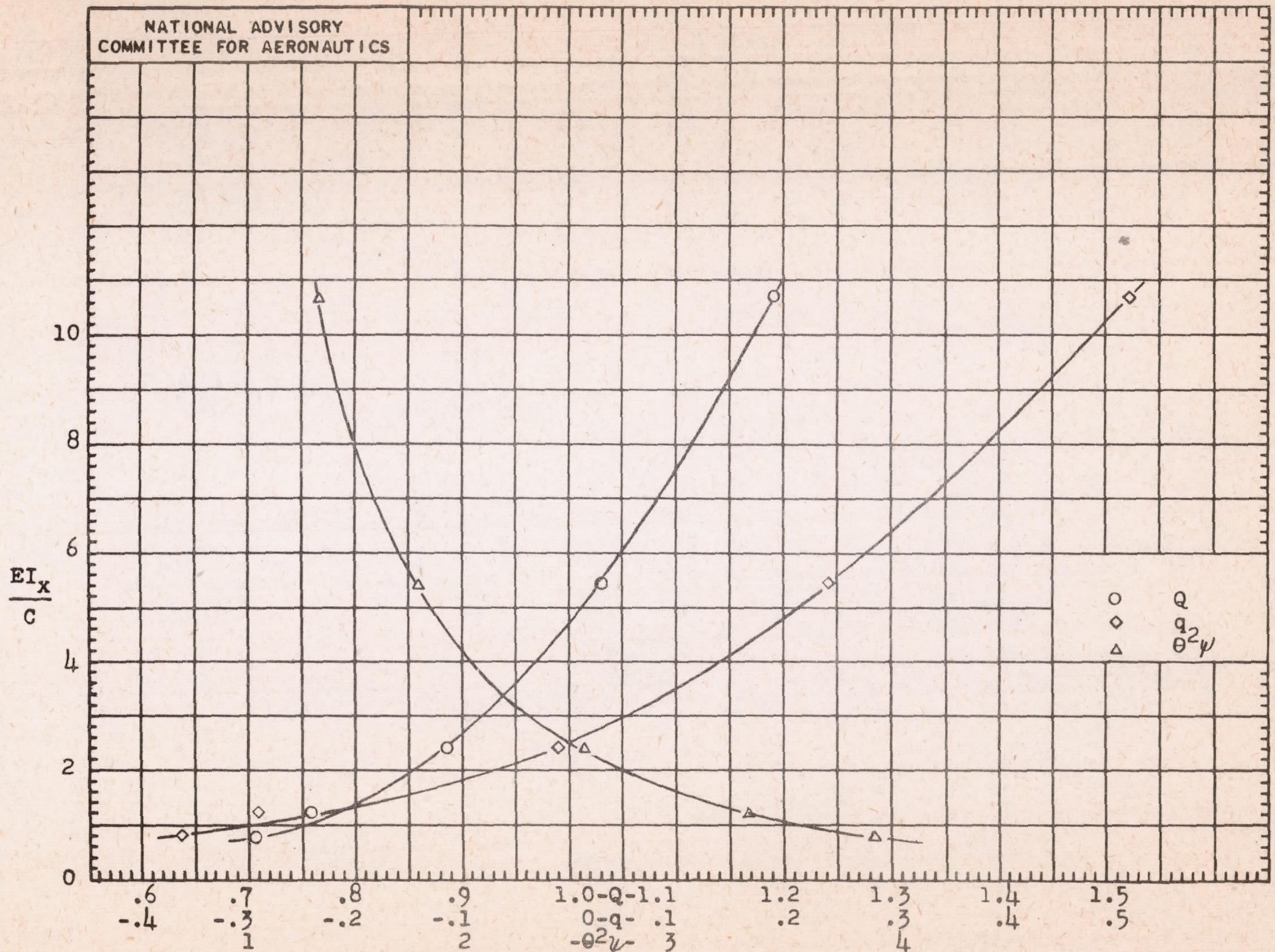


Figure 7. - Constants Q, q, and θ^2_ψ, which are used in the equation for the axial or transverse ring vibration frequency. (Values from reference 4.)

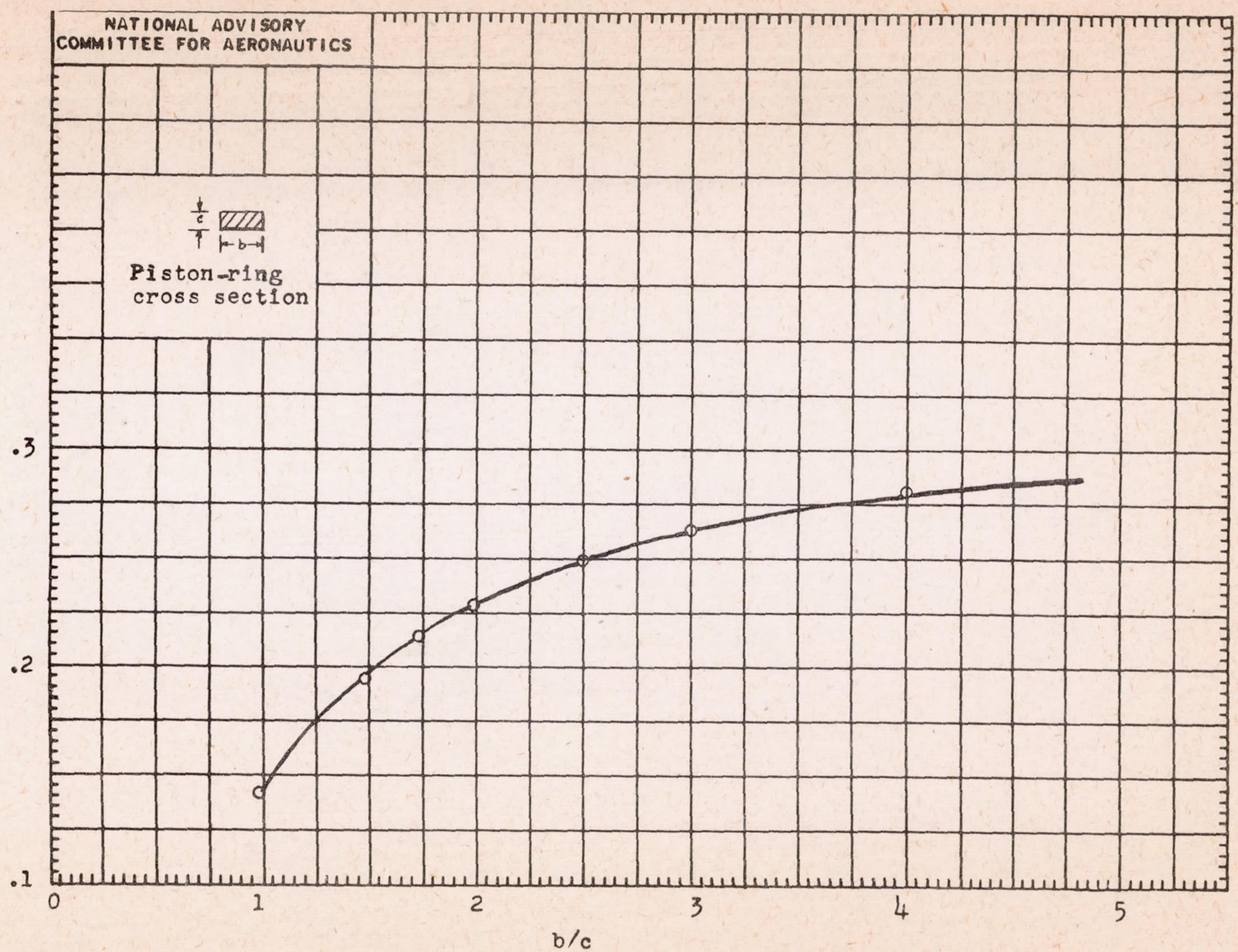


Figure 8. - Relation of the factor β to the length-width ratio b/c of piston-ring cross section. (Data from reference 7.)