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FATIGUE TESTING OF WING BEAMS BY THE RESONANCE METHOD

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National Bureau of Standards

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By William M. Bleakney

SUMMARY

Preliminary fatigue tests on two aluminum-alloy wing-beam specimens subjected to reversed axial loading are described. The method used consists in incorporating one or two reciprocating motors in a resonance system of which the specimen is the spring element. A description is given of the reciprocating motors, and of the method of assembling and adjusting the vibrating system. The results indicate that the method is well adapted to fatigue tests of not only uniform wing beams but also wing beams with asymmetrical local reinforcements.

I. INTRODUCTION

The present paper describes part of a research program which was requested by the National Advisory Committee for Aeronautics to obtain information on the fatigue strength of fabricated structural elements of aircraft. The need for such information has become increasingly apparent with the increased importance of vibration in modern high-speed and high-performance airplanes.

The strength of an airplane under steady flight loads can be computed with considerable accuracy from the flight loads and the strength of the airplane as determined by static loading on the ground. Little is known about the corresponding ability to withstand vibrational loads.

The determination of the ability of aircraft structures to withstand vibratory loads requires on the one hand a knowledge of the nature of the vibratory loads to which the airplane is subjected in flight, and on the other a knowledge of the strength of the structural parts of the airplane in withstanding vibratory loads of this nature. Information of the first type is being collected by the Navy Department in its extensive investigation on vibra-

tion of airplanes in flight and by the National Advisory Committee for Aeronautics in its study of dynamical loads under flight conditions in a flying boat. The work described in this paper is confined to information of the second type. It may be regarded as an extension to structural parts of airplanes of the resonance method of fatigue testing developed for airship girders by the Goodyear-Zeppelin Corporation (reference 1). The experience of the Goodyear-Zeppelin Corporation was drawn upon in planning the tests and in procuring the driving units.

The present report describes the application of the resonance method to wing-beam specimens having various types of discontinuities such as attachments, access holes, rivets, bolts, and sharp angles, which may affect the fatigue strength of the wing beam in the finished airplane. The presence of such discontinuities in the test specimen is considered particularly important since it is well known that they introduce stress concentrations which may lower the fatigue strength to a fraction of the value found in the absence of such concentrations. The effect of the discontinuities can be studied most conveniently by applying an alternating axial load to a specimen approximately uniform in section, so that, except in close proximity to the discontinuities, the stress is approximately uniform throughout the specimen.

It is realized that the vibratory stresses in wing beams in service will differ from those for this type of fatigue test in having a steady stress that is generally different from zero and in being due mainly to flexural rather than axial loads.

The effect on endurance of steady stresses superposed on alternating stresses has been the subject of considerable study in the case of structural materials, and to a less extent in the case of notched, riveted, and welded structures (reference 2), but no well-controlled experiments are known which show the effect of steady loads on the endurance of fabricated aluminum-alloy structures with local stress concentrations. It is hoped that an estimate of this effect may be obtained later in this investigation.

The fact that axial alternating stresses rather than flexural stresses are set up in the wing beam does not seriously detract from the value of results from axial fatigue tests since the direction of stress in both cases is the same. Thus the fatigue strength of the flanges of the

beam, which are the parts that may be expected to fail first in service since they are the most highly stressed, may be estimated from observations of flange failure in the axial load test. In addition the effect on fatigue strength of stress concentrations around small discontinuities in the web may be observed.

The natural frequency of flexural vibration of airplane wings is so high that the beams may in some cases be subjected to many millions of cycles of vibratory stress near the maximum range during their service life. The fatigue test in the laboratory should therefore be designed to provide loads alternating several million times within a reasonable test period. The most convenient method of accomplishing this is to make the specimen the elastic member of a resonant system of high natural frequency, and to drive this system at or very near this frequency. The resonance method, in addition to shortening the fatigue test, has the advantages of providing a large alternating load with a low amplitude of the driving force, and of loading only the specimen and its immediate attachments, so that the size of the loading mechanism may be reduced to a minimum.

A practical procedure for such a resonance fatigue test with alternating axial loads has been developed by the Goodyear-Zeppelin Corporation for their fatigue tests on airship girders and was successfully applied by them in tests to destruction of 18 aluminum-alloy girders of a number of different designs (references 1 and 3). In the present investigation the procedure used by the Goodyear-Zeppelin Corporation has been modified to a considerable extent; first, in order to make use of existing power equipment at the National Bureau of Standards and thereby simplify the set-up, and second, in order to adapt the method to wing beams in place of airship girders.

II. DESCRIPTION OF WING-BEAM SPECIMENS

The test specimens were supplied by the Navy Department in the form of complete wing cells containing 24ST aluminum-alloy I beams of typical construction. The beams were removed and cut into convenient lengths. This paper describes tests to destruction on two such specimens, cut from the undeformed portion of the front and rear beams of an upper wing cell which had been damaged in service (fig.

1). Care was taken to leave undisturbed all attachments, reinforcements, and rivet joints which might set up local stress concentrations in the assembled wing beams.

Specimen A was the central portion of the rear beam, extending out to the sections at which the beam began to taper. It was 96 inches in length, 5-3/4 inches in depth, and had a nominal cross-sectional area of 1.11 square inches. The web was 0.045 inch thick, the lower flange 1-3/4 inches wide and 0.15 inch thick, and the upper flange 1-3/4 inches wide and 0.234 inch thick.

Specimen B was an 88-inch length of the front beam, extending 18-1/2 inches to starboard of the center line and 69-1/2 inches to port, the remainder of the starboard end being damaged. The specimen tapered slightly near the port end. The beam was about 9-1/4 inches deep, with a nominal cross section of 1.19 square inches, including a narrow strip of skin material which was riveted to each flange. At the port end of the loaded portion of the specimen the taper reduced the section to 1.16 square inches. The web was 0.054 inch thick, and the flange dimensions were the same as those of the lower flange of the rear beam, 1-3/4 inches by 0.15 inch.

III. GENERAL DESCRIPTION OF THE RESONANCE METHOD

1. Arrangement for the First Wing Beam

The general arrangement of the resonant system employed for specimen A is shown in figures 2 and 3.

The Specimen S (fig. 2) was mounted vertically with a steel weight W attached to its upper end and a cylindrical electromagnet M attached to its lower end. The magnet winding C_1 was energized with direct current. Inserted in the annular air gap and free to move axially relative to the magnet was the ring-shaped armature A. This armature was connected to the upper weight W by means of the yoke Y and the two rods R, these connections being so rigid that the system moved nearly as a single unit. When alternating current was supplied to the armature winding C_2, alternating axial forces acted between the armature and the magnet in a manner similar to the push-pull action of a dynamic loudspeaker unit. When the frequency of the alternating current was adjusted to

coincide with the natural frequency determined by the ratio of the spring constant of the specimen S to the masses W and M and the parts rigidly attached thereto, these driving forces supplied the power which built up and maintained resonant vibration.

The system was isolated from the framework of the building by a tension spring S_1 at the top and four compression springs S_2 at the bottom (fig. 2). The static load on the specimen was eliminated by adjusting the height of the upper support until the length of the tension spring S_1 was equal to its length as measured before the specimen was connected to the weight W.

2. Arrangement for the Second Wing Beam

Two identical reciprocating motors were used to apply the load to specimen B, one motor being attached to each end of the specimen to form a symmetric arrangement. The method of attachment is shown in figure 4. The complete assembly is shown in figure 5.

The assembly was mounted horizontally on a bench B (fig. 4), and supported on small 1-inch rubber pads S, which effectively prevented vibration from traveling through the bench to the floor. The rigid unit comprising the armature A and the yoke Y was supported in the magnetic air gap by a system of 16 flexure plates designed to confine relative motion to the axial direction. (See F, fig. 8.)

At first the two yokes at opposite ends of the assembly were connected by pipes as in figure 5, so that this system, acted upon by vibratory forces which were oppositely directed at the two ends, remained stationary. The pipes, however, were later removed when it was found that the forced vibration set up in the freely floating armature and yoke was not appreciable at the frequency of the test.

The resulting simplified assembly showed a marked improvement in the flexibility of adjustment in centering the load. Other advantages of this assembly over the vertical set-up used in the first test were greater facility in mounting the specimen and in reading strains and amplitudes, and also the elimination of the rather noisy spring system used to support and isolate the first vibrating assembly.

3. Capacity of Existing Equipment

The length of specimen is limited by the available stroke of the armature relative to the magnet. In the two machines supplied for this investigation by the Goodyear Tire and Rubber Company the stroke was limited to 1/4 inch. The maximum change in specimen length in the arrangement of figure 2 is therefore 1/4 inch; in the arrangement of figure 4, 1/2 inch. Hence for an aluminum-alloy specimen with a stress amplitude of 10,000 pounds per square inch, the maximum length which can be tested is about 10 feet in the first case, and 20 feet in the second.

Within the limits of motion imposed by the 1/4-inch stroke, the maximum alternating load attainable is determined by the amplitude of the alternating driving force which can be attained, and by the total amount of damping in the system. For the same amount of damping, it is of course the same for the arrangement of figure 2 as for that of figure 4. For maximum safe currents supplied to the magnet and armature, the amplitude of the driving force was found to be about 480 pounds. The damping can be conveniently given in terms of the force amplification factor; that is, the ratio of the amplitude of the load variations to the amplitude of the driving force at resonance. In these tests this ratio ranged from 50 to 120, depending largely upon the temperature of the system. It may be possible to obtain values considerably higher than these in specific cases by reducing the damping in the attachments which connect the weights to the ends of the specimen.

IV. DRIVING MECHANISM

1. Power Supply

To drive the assembly at its resonant frequency, advantage was taken of the variable-frequency power available from the motor-generator set used in the study of propeller vibration (reference 4).* This set has an alter-

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*This is the main difference between the resonance method used here and that used by the Goodyear-Zeppelin Corporation (references 1 and 3). The latter method makes use of an inverter circuit controlled by a magnetic pick-up attached to the vibrating assembly to supply power at the requisite frequency.

nating current capacity of 55 amperes throughout the frequency range from 20 to 180 cycles per second, the voltage being adjustable. The sharpness of the resonance peak for the vibrating system required that the alternating current frequency be controlled with great precision in order that the amplitude variations may be kept within a few percent. This was accomplished by driving the variable frequency generator with a compound wound direct-current motor, the direct current current being supplied by a direct-current generator driven by a synchronous motor. The frequency was adjusted by altering the field current in the direct-current motor by means of field rheostats. It was found that the resonant frequency of the mechanical system changed somewhat more rapidly than the frequency of the power for a particular rheostat setting. (See section VII, 2.) Consequently, no further attempt was made to improve the power frequency control.

2. Design of Reciprocating Motors

As explained in section III, the power is transmitted to the vibrating assembly by means of one or two reciprocating motors which consist essentially of a cylindrical magnet M, a ring armature A (see fig. 2), and means for guiding the relative motion of these two parts. Two such reciprocating motors were designed and built for this project by the Aeronautical Department of the Goodyear Tire and Rubber Company. Although new in design, these motors have exhibited no trouble in service.

Figure 6 shows the field magnet of one of these machines, with the armature removed. The winding which encircles the core is not visible in the photograph. This part weighs about 580 pounds.

Figure 7 shows the armature which is welded directly to a tubular yoke designed to facilitate connections to members in the vibrating system. This unit weighs 66 pounds and contains, besides the armature winding, laminated iron parts designed to concentrate the magnetic flux to the best advantage.

Figure 8 shows the reciprocating motor completely assembled. The armature is centered in the magnetic air gap by the system of flexure plate guides F which effectively prevent contact between the armature and the magnet pole faces. The nominal clearance here is 1/32 inch. Ducts

are provided for air cooling. The photograph shows the air hose connected to the air inlet at the axis of the core. Exhaust ports are provided at the other end of the magnet.

Without artificial cooling the magnet winding will carry a continuous direct current of 4 amperes, and the armature a continuous alternating current of 15 amperes. With air cooling these currents may be maintained at 5 and 30 amperes, respectively.

3. Performance of Reciprocating Motors

Tests on the performance of these machines were carried out at Akron before shipment to the National Bureau of Standards. Some of the results on static load tests obtained with the set-up shown in figure 9 are plotted in figures 10, 11, and 12.

The forces shown in figure 10 as a function of displacement of the armature relative to the magnet arise from the spring action of the flexure plates and from the attractive forces between the armature and magnet-pole face on account of the iron in the armature. These forces do not contribute to the power available, but merely affect to a slight degree the natural frequency of the vibrating system to which the reciprocating motor is attached.

Figure 11 shows some typical static load-displacement curves. From curves such as these, taken for various field currents, data were collected on the load per ampere in the armature at the position of zero displacement. The results are plotted in figure 12.

The load given in figure 12 is the driving force available per ampere armature current for very low frequencies, since it was obtained from static tests. At higher frequencies the driving force for given armature and field currents may be expected to be somewhat less on account of eddy currents and hysteresis. A dynamic test was therefore carried out at this Bureau in the following manner.

The motor, in the position shown in figure 8, was energized with 3 amperes field current and armature currents of various frequencies and magnitudes, while the amplitudes of oscillation of the armature and yoke were measured. Precautions were taken to prevent strong vibration in the supports and in the guide system. The displacement

amplitude per ampere alternating current in the armature is plotted in figure 13. The values of driving force plotted in the same figure were obtained in the usual manner for undamped forced vibration. (See any text on vibration, e.g., Timoshenko: "Vibration Problems," or den Hartog: "Mechanical Vibrations.") For the purpose of comparison with static tests these forces are given on the basis of instantaneous armature current, instead of the usual root-mean-square value.

It will be seen from figure 13 that the drop in driving force for given armature and field currents was approximately a linear function of frequency within the range tested, and amounted to about 9 percent between 0 and 100 cycles per second. From these results it was estimated that the maximum mechanical power available at full stroke, using the maximum safe armature and field currents with air cooling as given above was about 1-3/4 horsepower at 60 cycles per second, and was nearly proportional to the frequency. The corresponding driving force amplitude is about 480 pounds. This is over three times the maximum value used to date in the wing-beam tests.

V. MOUNTING OF WING-BEAM SPECIMEN

1. Attachment of End Weights

The success of the resonance method of fatigue testing depends to a considerable extent on the means of attaching the weights to the specimen. An ideal attachment would meet the following five requirements:

(1) A resistance to fatigue greater than that of the specimen.

(2) Uniform load application to the end section of the specimen.

(3) Negligible damping.

(4) Ease of attachment without danger of weakening the specimen.

(5) Adaptability to specimens of different shapes of cross section.

The attachments used in the first test conformed to a sufficient degree with the first four requirements, but they failed to meet the fifth. Those used in the second test were designed to conform more closely to the fifth requirement.

The method consisted essentially in applying the load to a portion of the surface of the specimen near each end through a thin layer of soft alloy. Wood's metal was used because it could be cast in hot water, thus insuring that no weakening in the aluminum-alloy specimen would result from the heat supplied during the casting procedure. The form was made from heavy steel in two parts and sealed with thin rubber strips, so that after casting the two parts could be drawn together by means of bolts until a high hydrostatic pressure was attained.

Figure 14 is a sectional view of one of the terminal attachments used in the first test, and figure 15 shows a photograph of the attachment taken after removal from the specimen following the test. The attachment covers 9 inches of the specimen at the end. It could not be used in the second test because of the deeper beam section (9-1/4 inches compared to 5-3/4 inches for the first specimen). The design of the attachment used in the second test is shown schematically in figure 4 and photographically in figure 16. Here only a 5-inch length of specimen was covered because the thickest part of the front beam was only 0.15 inch as compared to 0.23 inch for the top flange of the rear beam. The average shear stress on the Wood's metal was of the order of 2 percent of the stress applied to the specimen, but the stress may be considerably higher in the Wood's metal farthest removed from the ends of the specimen. In fact, it is believed that the observed variations in damping (see section VII) were due almost entirely to variations in the temperature of the Wood's metal and their effect on the plastic strength of this soft alloy.

2. Centering of Load

It is important to load the specimen as nearly uniformly as practicable, in order to insure failure at the weakest points. Moreover, because of the dearth of dependable strain gages, it is not feasible to determine the load at the point or points of failure unless the load distribution over the specimen is known.

Nonuniform loading arises principally from the presence of flexural modes of vibration in the plane of the web. Only two such modes have sufficiently low frequencies to be important. The first mode involves rotation of the weights in opposite directions and subjects the specimen to a uniform bending moment. The second mode involves rotation of the weights in the same direction; this subjects the specimen to a uniform shearing force and causes it to vibrate with a node at the center.

In the first test neither mode caused any trouble. Care was taken to place the centers of gravity of the weights as accurately as possible along the centroidal axis of the specimen. The symmetry of the reciprocating motor then insures that the line of action of the driving force is also along the centroidal axis.

However, the same precautions were not sufficient in the second test because specimen B carried near its center a large reinforcement at the point where in the airplane the cabane support was attached to the front wing beam (see fig. 1), and the reinforcement extended a length of 25-1/2 inches along the upper flange but only 9-1/2 inches along the lower flange. Consequently, if the load was to be applied through the centroids of the end sections, provision had to be made for permitting these end sections and the weights attached thereto to rotate, in order to accommodate the greater change in length of the lower flange compared to that of the upper flange.

The position of the load line relative to the centroids of the end sections may be adjusted either by shifting the axes of the reciprocating motors relative to the specimen, or by attaching additional eccentric weights to the motors. The first method applies moments to the specimen by providing a coupling between the longitudinal mode and the two flexural modes mentioned above. The second method, in addition to providing such a coupling, changes the frequency of the axial as well as of the two flexural modes by changing the total mass and moment of inertia, and also provides driving moments, since the line of action of the driving force is shifted relative to the center of mass of the total weight. Ordinarily the effects on load distribution of changes in frequency and of driving moments are not important unless one of the flexural modes has a natural frequency very near the frequency of the applied load. The effect of the coupling also depends upon the difference between the natural frequency of the

axial mode and that of the two flexural modes. Because of the complexity of these effects and the lack of information regarding the flexural modes, it is usually more profitable to find by trial the direction and magnitude of a given effect than to try to estimate the effect in advance. In attaching additional weights to the assembly care must be taken to make the attachment so rigid that no new modes of vibration are introduced having frequencies in the neighborhood of that of the applied load. In practice this limits to very small values the shift in center of gravity which can be attained by this method.

In the test on specimen B the stress distribution was found to be very sensitive to shifts in the center of gravity of either weight. The observed strains indicated that the natural frequency of the second flexural mode was very near to the frequency of the applied load. There were also indications that the natural frequency of the first flexural mode was not greatly less. This is at first surprising since for a slender beam the two flexural frequencies are in the ratio of 1 to $\sqrt{3}$. In the present case, however, the beam is comparatively weak in shear, and this tends to decrease the frequency of the second mode, while the first mode is unaffected since no shear load is present. Thus the natural frequencies of the two flexural modes are not necessarily greatly different.

The two long studs shown in the foreground of figure 5 were used for attaching small weights to the assembly. With a similar arrangement at the other end the two weights were adjusted until, for a small amplitude of vibration, the strain amplitude at four stations was identical within ±2 percent. Of the four stations, two were located near each end of the specimen and at opposite points on the two flanges.

If the test frequency is well removed from all flexural frequencies so that the stress distribution is insensitive to changes in the eccentricity of the weights, shifting the specimen relative to the motors is ordinarily the most convenient method of centering the load. Provision was made in the design of the terminal attachments used for the second test to permit considerable relative motion for such adjustments.

3. Guiding of Specimen

In addition to flexural modes in the plane of the web,

nonuniform loading may arise from flexural vibration normal to the web, from torsional vibration, or from axial loads caused by the inertia of the specimen itself. Each specimen alone weighed less than 2 percent of the total weight of the assembly, so that the variation in axial load along the specimen was always less than 1 percent (harmonics involving axial motion were not to be expected because of their high natural frequencies, at least ten times that of the fundamental). Serious torsional modes of vibration were not observed, although coupling existed between torsional and flexural motion as a result of the various eccentric reinforcements and attachments on the specimens.

It is sufficient for limiting flexural vibration normal to the web to provide guides at various points along the specimen, since only vibration of large amplitudes in this direction will have an appreciable effect upon the load distribution. Thus the single guide used in the first test (fig. 3), consisting of two wooden bars clamped to the pipes connecting the upper weight to the armature at the bottom, with rubber pads separating them from the specimen, was all that was needed in this case. Without this guide there seemed to be no limit to the amplitude which might have been attained in this direction. The vibration was subharmonic, but ample energy was available for building it up since the peak compressive load was well over three times the critical static Euler load for the beam acting as a column without end restraints.

Another type of flexural vibration normal to the web, a vibration of the web itself as distinct from the flanges, appeared to be the source of nearly all the noise which emanated from the specimen, the noise level being high enough to be distinctly unpleasant to the operator. A rough computation confirmed the suspicion that this may have been buckling of the web near the peak compressive load for each cycle, the web acting approximately like a plate with "built-in" edges. It seems possible that periodic buckling of this kind may have had some influence on the endurance of the specimen, but if so, the failure was presumably "compressive" rather than "tensile," for the buckle would not affect the distribution of the peak tensile stress.

Although the second specimen was tested with roughly 30 percent less nominal stress than that applied to the first specimen, the greater beam depth again permitted

buckling in the web. This time a pair of wooden members extending the full length of the specimen and clamped to opposite sides of the web through thin rubber pads was tried, contact being made at several points along the specimen where it was found most effective in deadening the noise. This guide system added sufficient stiffness to the specimen to prevent objectionable lateral vibration and at the same time lowered the noise level below the threshold of discomfort to the operator.

VI. TEST PROCEDURE

In any fatigue test two quantities are required, the number of cycles to failure and the peak or turning point values of some quantity such as the stress, the strain, the load, or the deflection or relative displacement of some particular point on the specimen.

1. Measurememt of Number of Cycles

In the present tests the cycles were counted merely by noting the starting and stopping times to the nearest minute, and recording the frequency as indicated by an electromagnetic tachometer connected to the shaft of the generator. This tachometer was calibrated by a revolution counter and a stop watch.

2. Measurement of Amplitude

The amplitude of vibration of one of the weights is also an easily measurable quantity. The wedge-shaped scale shown in figure 17 was used for this purpose.* The wedges on this scale have a ratio of length to width of 10:1, and with the aid of a telescope or low-power microscope isolated from the vibrating system readings may be estimated to 1/5 division, which corresponds to 0.001 inch double amplitude. Somewhat higher sensitivity may be obtained by reducing the size of the scale and using greater magnification.

In figure 17 the amplitude scale is mounted within 2 inches of the axis of the system so that small rotations of the weight about its center of mass would have a negligible effect on the reading of the axial component ampli-

*A similar scale is described in reference 1.

tude. In the second test the scale was mounted on the axis at the armature end of the motor by attaching it to a flattened brass rod screwed into the air duct. (The air duct is shown connected to the hose line in figure 8 and also in the foreground of figure 5.)

3. Measurement of Strain

Strain measurements were made at various locations on the specimen during preliminary runs at low amplitude and also, in the case of the second test, at one location during the entire test. These measurements were made as a means of determining the load distribution over the specimen, and to provide as direct a correlation as possible between the test results and the vibratory strain to be measured under service conditions, as well as to assist in the determination of stress concentration factors.

For most strain gages there is considerable error in the readings if the gage is subjected to high acceleration. Probably the most reliable gage for this condition is the Tuckerman optical gage shown in figure 18. This special gage is lighter and more rigid than the regular Tuckerman gage (references 5 and 6). The gage length is 2 inches, the weight 0.8 ounce. The correction is small for longitudinal accelerations up to ten times gravity and is negligible for accelerations of this order of magnitude in any other direction.

Only two such gages have been constructed. To supplement these special light-weight gages regular 2-ounce gages were used in the second test. Attempts to obtain satisfactory performance of the heavier gages at the normal running amplitude of the test failed, even near the center of the specimen where the longitudinal motion is nearly zero, because of the lateral vibration present. However, with sufficient care relative measurements could be obtained with these gages during the preliminary runs at a reduced amplitude, so that they were useful in centering the load.

The gages were mounted in pairs at opposite points on the two flanges of the specimen, care being taken to set each gage as near the center of the flange as possible without getting too close to rivets or other abrupt changes in section. Several methods of attaching the gages to the specimen were tried. Figure 18 shows the method that was adopted for most of the strain measurements.

4. Determination of Stress

For the purpose of simplifying the discussion, it is convenient to express the results of these tests in terms of the "nominal stress amplitude" defined as the amplitude of the axial stress averaged over a cross section of the specimen free from rivets, open holes, and other irregularities.

Either the measured amplitude of one of the weights or the average strain amplitude as indicated by a pair of strain gages may be used as a basis for computing the nominal stress amplitude. The strains were converted into stresses by assuming a value of 10.5 by 10^6 pounds per square inch for the Young's modulus of the 24ST material. This value is taken from the primitive stress-strain graphs covering a large number of tensile and pack compression tests on this material and is believed to be accurate within ±1 percent over the range of stress covered in the fatigue tests. In the case of dynamic loading the adiabatic modulus should be used, but since this is only about 1/2 percent greater than the isothermal value (reference 7), the difference may be neglected. This method of computing the nominal stress amplitude does not take account of any phase difference between the two strain gages, which may be present as a result of flexural vibrations in the plane of the web. With such a phase difference the indicated amplitude of the average stress would always be in excess of the actual amplitude.

There are two methods by which the nominal stress amplitude may be deduced from the amplitude of vibration of the weight. The first was used by the Goodyear-Zeppelin Corporation (reference 1) to obtain a value for the stress amplitude in airship girders, and consists in taking the relative displacement between the two weights divided by the effective length of the specimen as the mean strain throughout the specimen, from which the corresponding stress is obtained by means of Young's modulus. This method was not practicable in the fatigue tests of wing beams because of the uncertainty in the effective length of the specimen. Both the presence of reinforcements of unknown stiffness along the specimen, and the fact that the depth of penetration of stress in the terminal attachments apparently varies with temperature and time, contribute to this uncertainty.

The second method consists in deducing the peak load

from the mass and acceleration of one of the weights and dividing by the nominal cross-sectional area to get the nominal stress amplitude.

If the axial displacement of the center of gravity of the weight is given by $B \sin 2\pi ft$ where B is the amplitude and f the frequency, the peak value of the acceleration will be $4\pi^2 f^2 B$, and the nominal stress amplitude will be $4\pi^2 f^2 BM/Ag$ where M is the mass, A the nominal area, and g the acceleration of gravity. The driving force does not enter into this computation since at resonance it is practically 90° out of phase from the inertia force, and therefore may be taken as zero at the instant the peak load is reached. A small correction, however, is required to take account of the magnetic attraction between the armature and the magnet as given in figure 10.

The principal assumptions made in this method are that the weight moves as a single unit and that the motion is sinusoidal. No evidence has been found for doubting the accuracy of the first assumption. Deviations from sinusoidal motion may arise either from harmonics or from lack of proportionality between load and extension. Harmonics involving axial motion are not to be expected because their frequencies are estimated to be greater than ten times that of the fundamental, and no adequate source of energy having such high frequencies is apparent.

Lack of proportionality between load and extension may be expected from the effect of frictional forces, such as friction in the vicinity of rivets and bolts, and particularly from the effect of slippage and plastic deformation in the Wood's metal at the terminals. Although the total damping is not large (see section III, 3), the effect of this type of damping on the peak load is difficult to estimate, occurring as it does near the peak load, in contrast to viscous damping which reaches its maximum as the load passes through zero.

Thus each of the two methods of determining the nominal stress amplitude is subject to some uncertainty, the first because it assumes all strains to be in phase, the second because it assumes sinusoidal motion. The observed discrepancy in the results (see section VII, 1 and 2) does not appear too large to be accounted for by failure of one or both of these assumptions. More data will be necessary to determine which of the two methods is the more trustworthy in the present case.

5. Observation of Failure

In fatigue tests such as these it is desirable that the number of cycles before the first appearance of a visible crack be determined, as well as the number of cycles before a major failure. The endurance based on the first criterion is probably the safer one to use insofar as it indicates a weakening of the structure against both static and dynamic loads. On the other hand, the determination of this endurance requires frequent and careful examinations of the specimen, which may be impractical in the presence of a guide system hiding a considerable portion of the specimen, such as that used for the second specimen.

Actually, in these tests, no cracks were observed until a sharp report and a sudden slight reduction in resonant frequency gave notice of a major fracture passing completely through one of the flanges. Examination then showed, in the case of each specimen, a number of additional cracks in various parts of the specimen and in varying stages of development, indicating the location of the weakest points in fatigue, and also indicating the sequence of formation of the larger cracks. In order to show up the smaller cracks more clearly and to reveal possible incipient cracks which had been overlooked, the specimens were vibrated at a reduced amplitude for some time after the major failure.

VII. RESULTS

1. Data on Specimen A

The data obtained from the fatigue test of the first wing beam are summarized in figure 19. The bottom curve gives the amplitude of the lower weight as read on the scale shown in figure 17. The second curve gives the frequency. The data from these two curves were used to compute the nominal stress amplitude given by the third curve, the calculation being made by the method described in section VI, 3.

During the preliminary runs covering the first 107 minutes strain amplitude readings were taken on the flanges near the center of the specimen with the two light-weight Tuckerman optical gages described in section VI, 3. The values of nominal stress amplitude deduced from these readings using for Young's modulus 10.5 by 10^6 pounds per square

inch (see section VI, 4), are given by the open circles in figure 19. The improvement in consistency toward the end of the preliminary runs is probably due to improved technique in the control of the amplitude. Only one strain reading was taken at full amplitude because of the difficulty of preventing creeping of the gage points as a consequence of the severe lateral vibration. The strain readings yield a value of stress amplitude on the average about 8 percent higher than that deduced from the amplitude of the weight.

From the results of both tests it is estimated that the first 304,000 cycles on the first specimen (fig. 19) is roughly 10 percent of the total endurance at the average amplitude of 0.0225 inch, and 44,000 cycles is about 5 percent of the endurance at 0.036 inch amplitude. Accordingly, taking account of these preliminary runs, the total endurance at the full amplitude of 0.040 inch should be about 15 percent higher than the 452,000 cycles actually observed, or 520,000 cycles. The corresponding average nominal stress amplitude is 8,300 pounds per square inch on the basis of the amplitude of the weight, 9,000 pounds per square inch on the basis of strain readings.

The top curve of figure 19 gives the armature current in the driving motor, which is directly proportional to the driving force and, at constant amplitude, very nearly proportional to the damping. At the break in this curve a recess of about two hours was taken. The tendency of the system during this period to recover its original low damping value is believed to be due to cooling in the Wood's metal which had been warmed and softened by the previous run.

2. Data on Specimen B

The data obtained from the fatigue test on the second wing beam are summarized in figure 20. The breaks in these curves represent recesses varying in length from a few minutes to several days. The effect of the preliminary runs covering the first hour and ten minutes on the endurance of the specimen is believed to be negligible. During these runs the load was centered as described above. (See section V, 2.) Subsequent strain readings were taken on only one gage. The average value of the nominal stress amplitude computed from these readings (see fig. 20) was 5,740 pounds per square inch, while the average deduced from the amplitude of the weight was 5,220 pounds per square inch,

leaving a discrepancy of 10 percent for the fatigue strength. The endurance was 2.86 by 10^6 cycles.

That the particular location of the strain gage was not stressed appreciably higher than the remainder of the specimen is clearly indicated by the fact that no cracks were found near the gage. It appears therefore that the conditions responsible for the discrepancy in the first test were equally serious in the second. (See section VI, 4.)

In both tests there was a marked tendency for the resonant frequency to fall during the course of the test. Such effects as changes in Young's modulus as a result of changes in temperature, progressive weakening of the material, and weakening of the structure as a result of loosening rivets and bolts, are considered inadequate to explain this behavior. The observed variations in frequency are believed to be caused by a combination of nonviscous damping and of changes in the effective free length of the specimen with variations in the depth of penetration of stresses in the terminal attachments. Both the damping and the change in effective length may be attributed to changes in the Wood's metal layer connecting the end weights to the specimen.

3. Location of Cracks

Cracks in various stages of development are shown on the photographs, figures 21 to 27 inclusive. The location of these photographs on the specimen is indicated by the corresponding number in figure 1. Crack No. 1, figure 21, was the main failure on the first specimen. The rib attachment under which it passed and the two rivets were removed after the test to show the crack more clearly. A total of eight cracks found on the first specimen are shown in figures 21, 22, and 23.

Crack No. 9, figure 24, is the main failure on the second specimen, and the additional six cracks found on this specimen are shown in figures 24 to 27 inclusive. The grouping of all the cracks but one, No. 15, near one end indicates a certain amount of overloading of the top flange near this end and to that extent reflects the difficulties encountered in centering the load on this specimen. One crack, No. 12, was found even inside the terminal attachment. And yet the occurrence of No. 15 near the other end

of the same flange shows that the second flexural mode in the plane of the web, which was the most difficult to control, did not predominate during the test for it would load the opposite flange the heavier at that end.

VIII. CONCLUSIONS

Preliminary fatigue tests of two wing-beam specimens have shown that:

1. The reciprocating motor in conjunction with a resonant system is a convenient means of applying rapid axial load reversals to large specimens such as wing beams for fatigue measurements;

2. Specimens with local asymmetries in section such as that portion of the front wing beam which includes the cabane-support attachment may be loaded nearly uniformly;

3. The load may be applied directly to the specimen through a layer of Wood's metal without the necessity of "building up" the ends to prevent failure at the terminals;

4. The problem of determining the stress applied to the specimen is not a simple one but will require further development; and

5. The beams tested had no outstanding weak points in fatigue but rather exhibit approximately the same resistance to fatigue around the many rivet holes and other discontinuities distributed throughout their length.

National Bureau of Standards,
Washington, D. C., July 1938.

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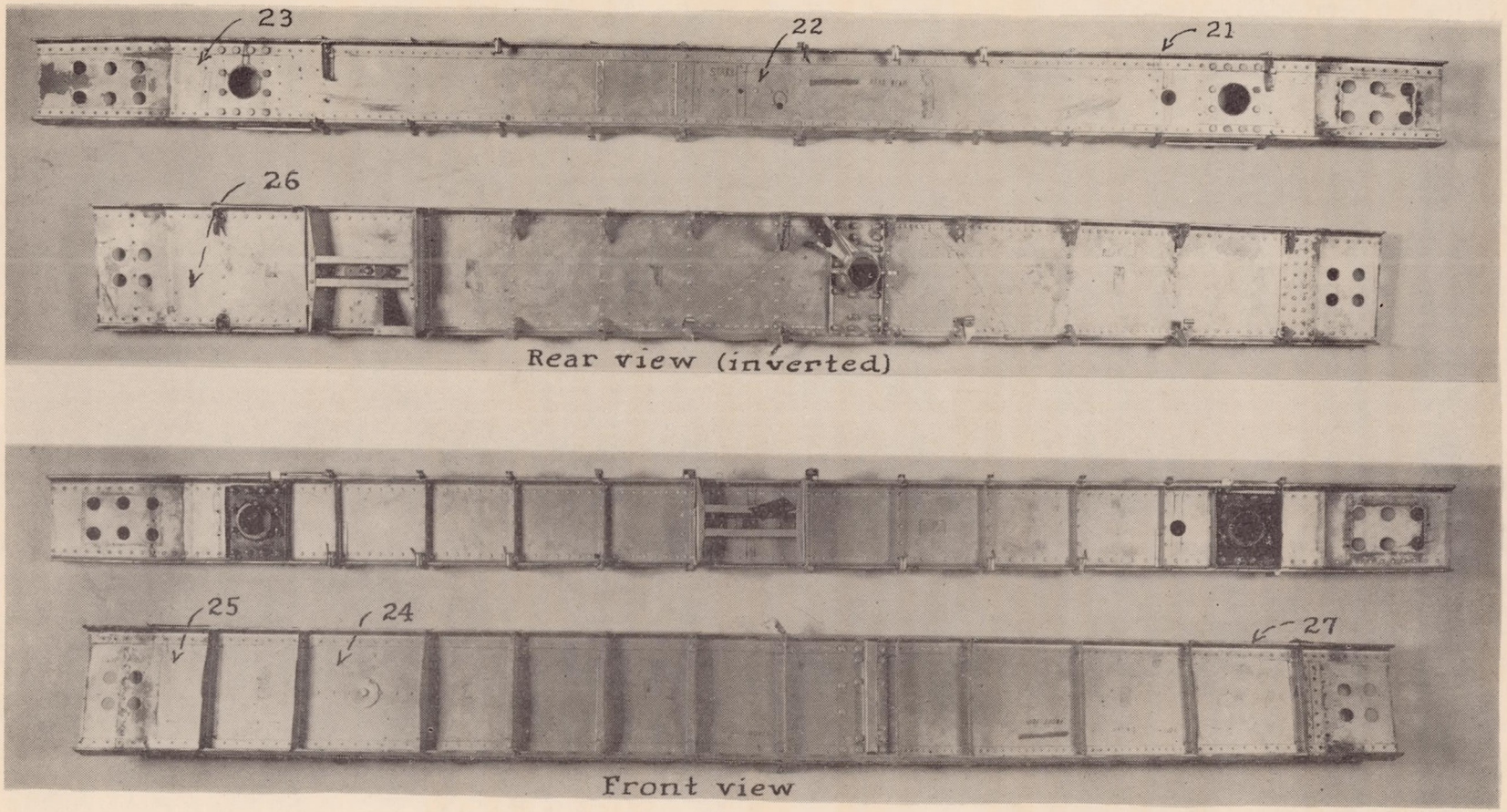


Figure 1.- Fatigue specimens of aluminum-alloy wing beams after tests to failure. The numbers indicate the location of crack photographs with corresponding figure numbers.

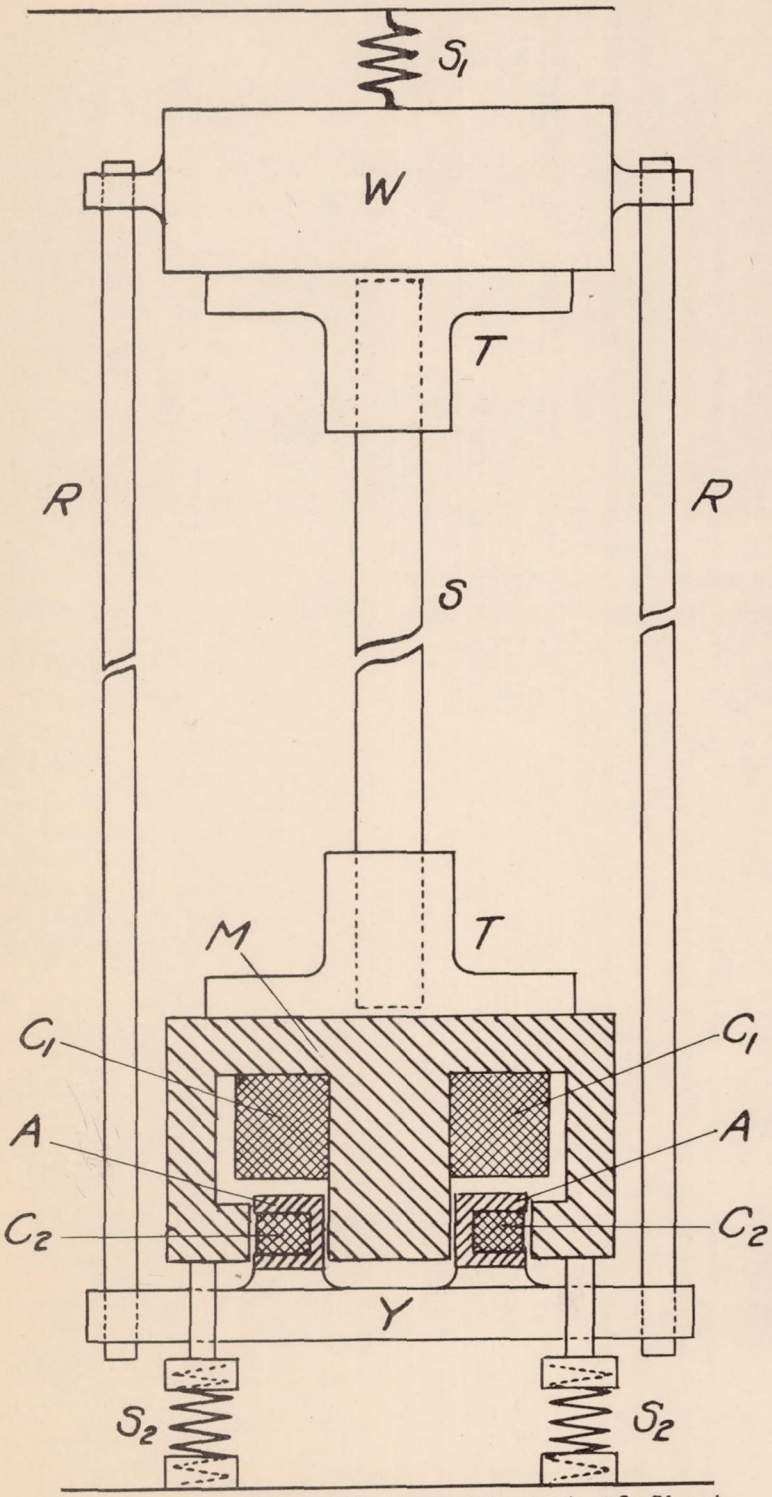


Figure 2.- Schematic arrangement of first fatigue test assembly.

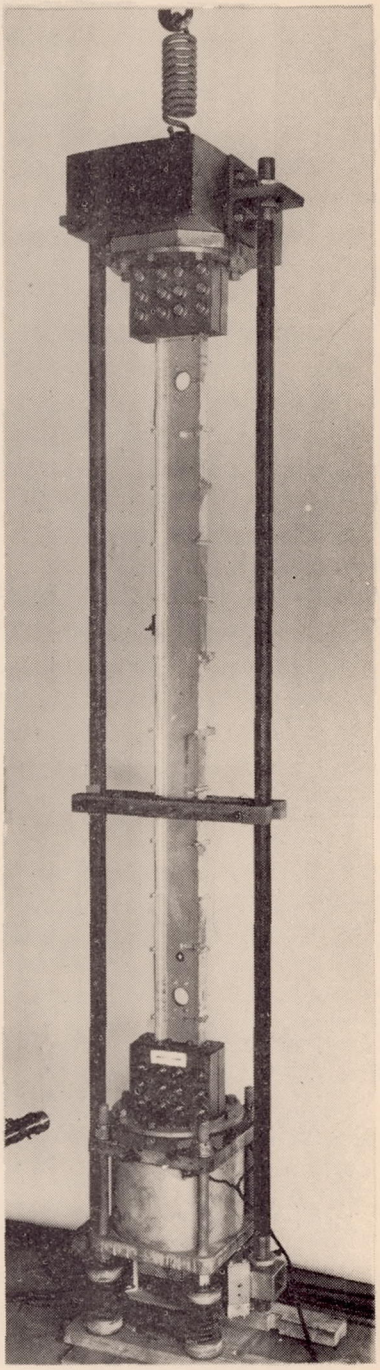


Figure 3.- First fatigue test assembly before final run to failure.

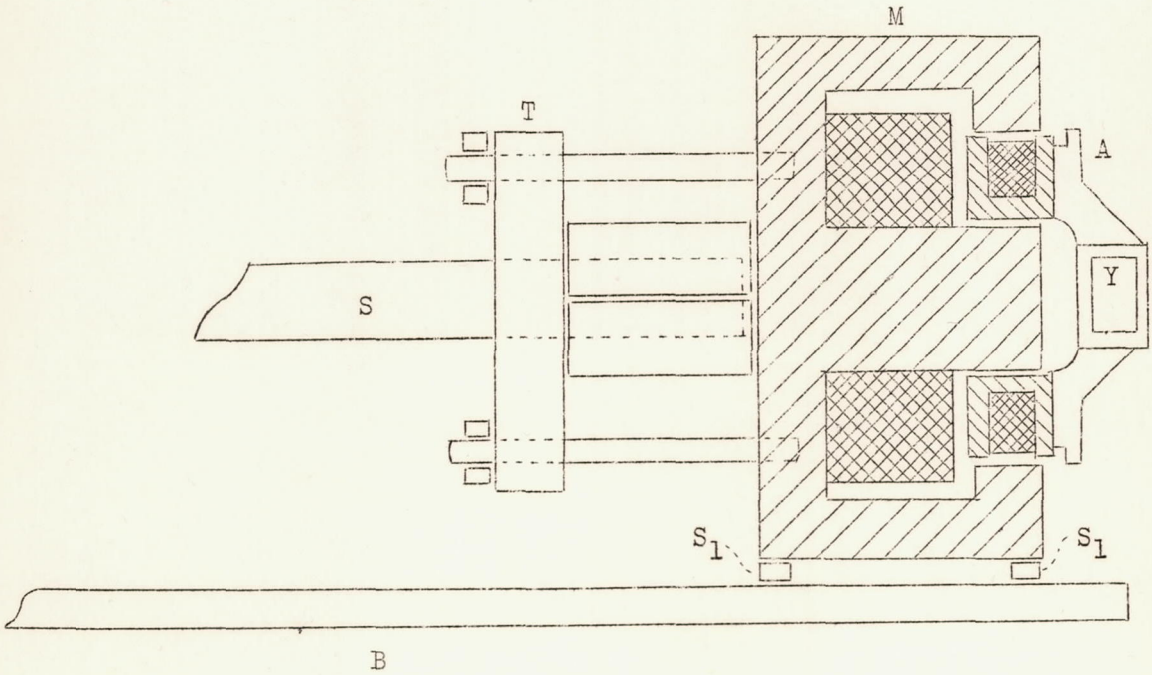


Figure 4.- Schematic arrangement of one end of second fatigue test assembly, the other end being identical to this.

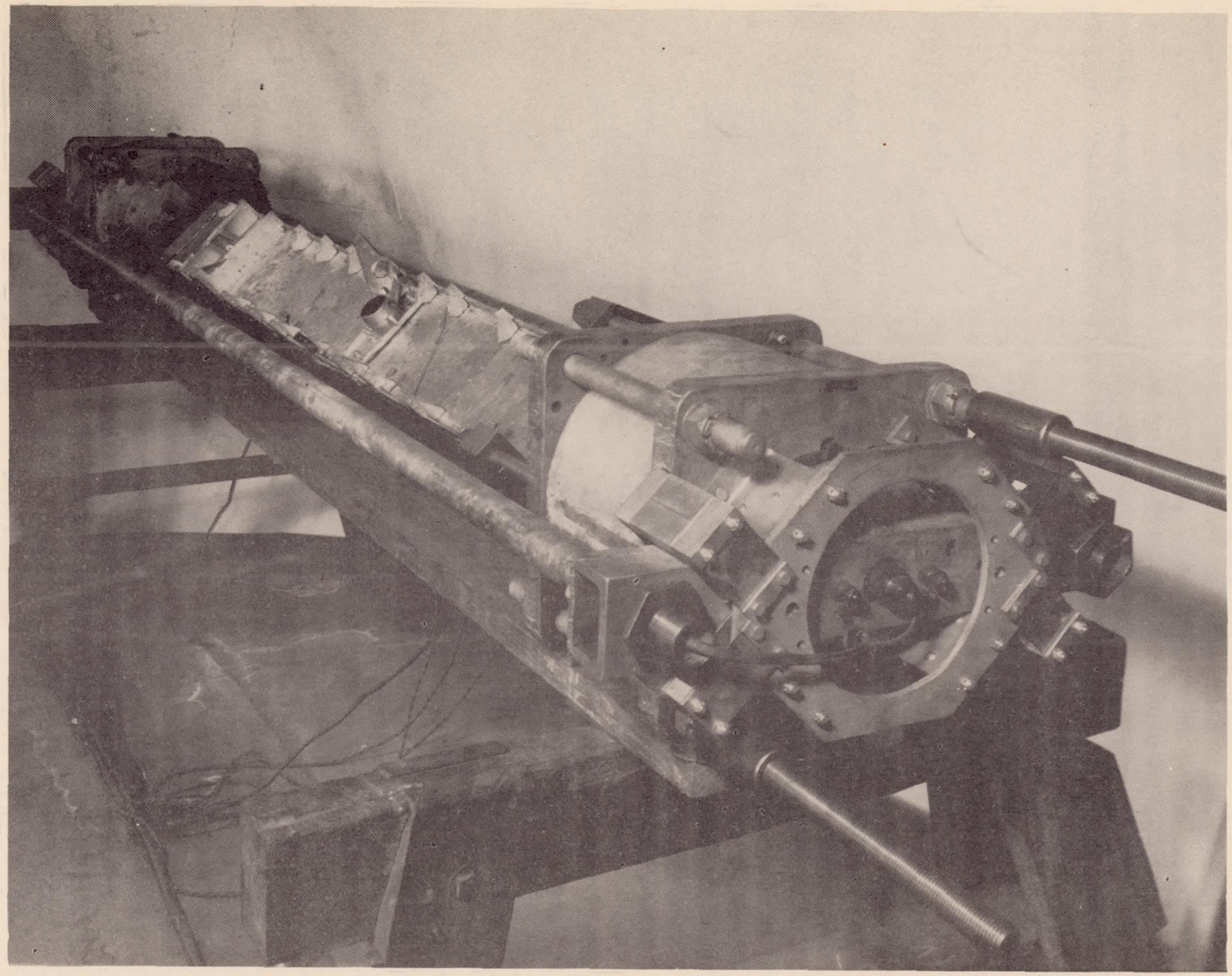


Figure 5.- Second fatigue test assembly with the exception of the guide system and load-centering weights. The symmetric arrangement reduces the mechanical difficulties and provides greater flexibility of adjustment.

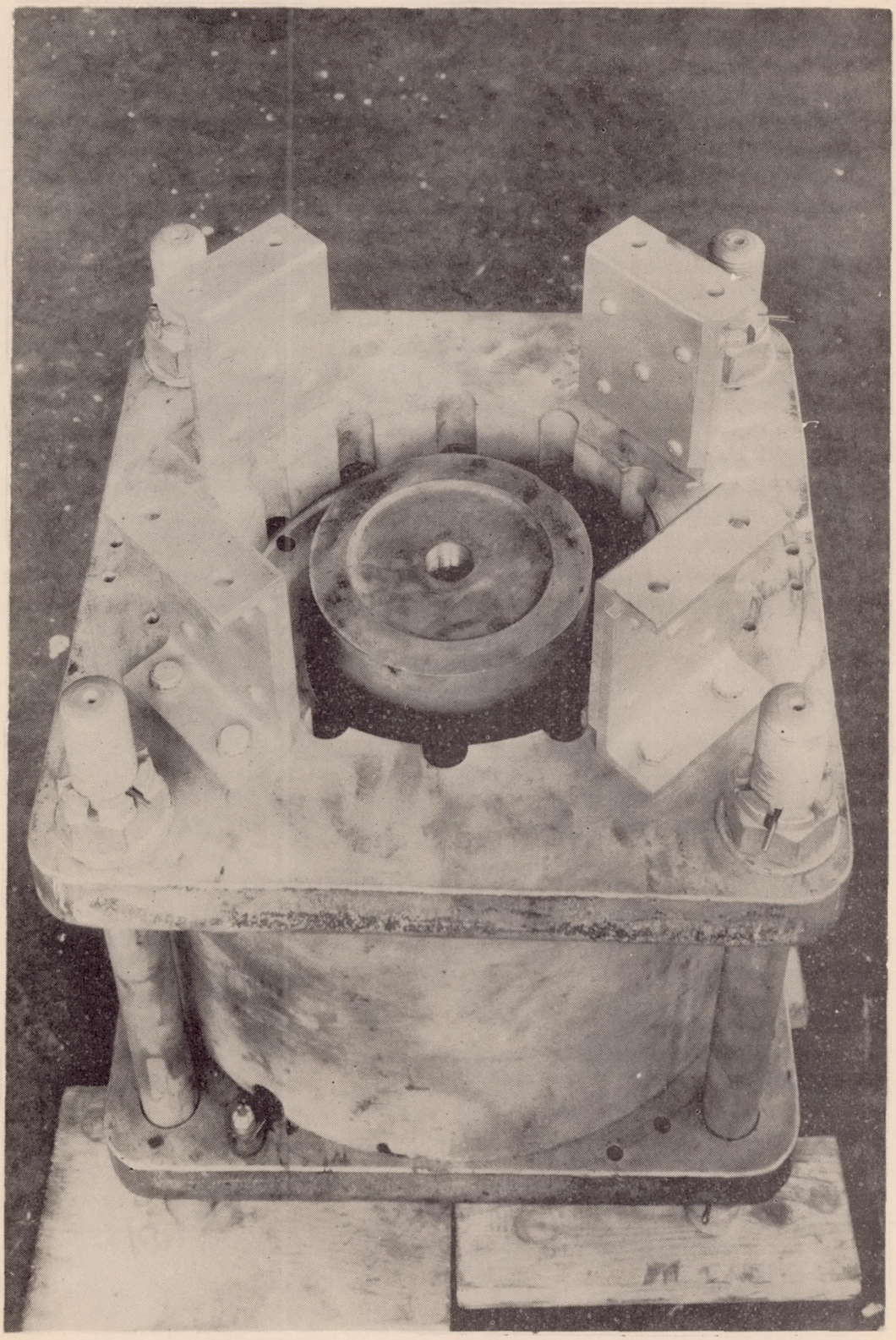


Figure 6.- Electromagnet of the Goodyear reciprocating motor.

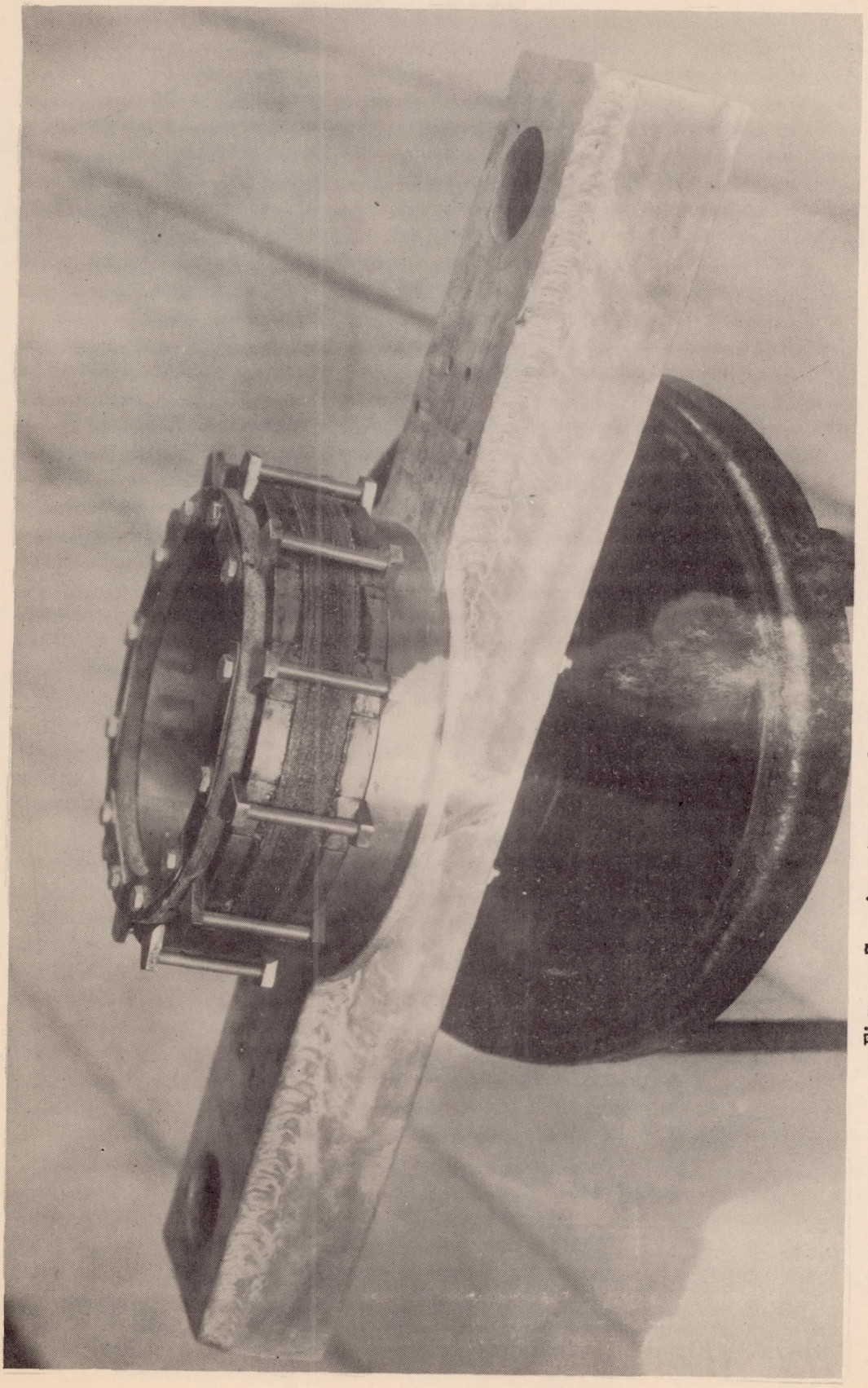


Figure 7.- Armature and yoke of reciprocating motor.

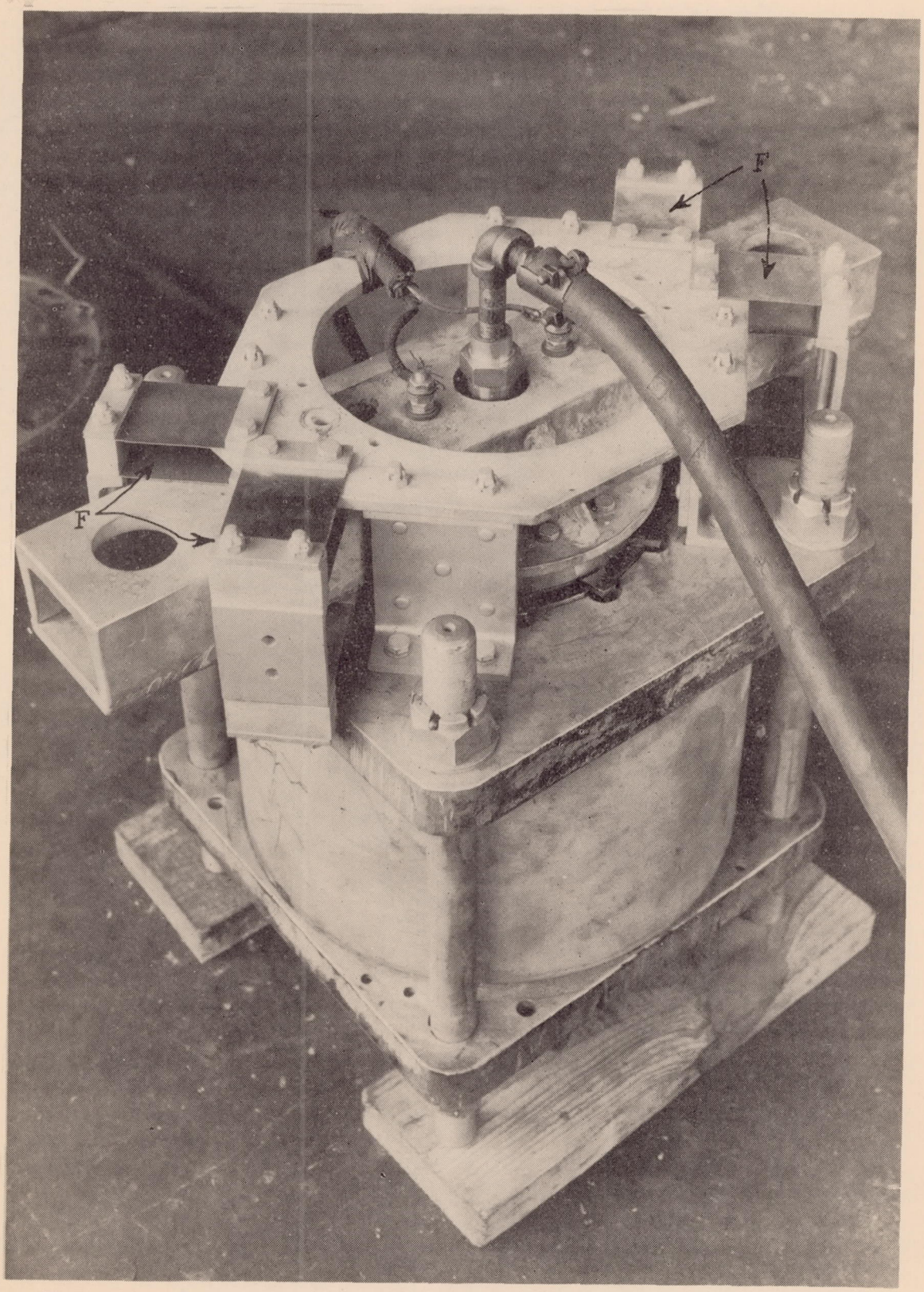


Figure 8.- Reciprocating motor complete with vents for air cooling.

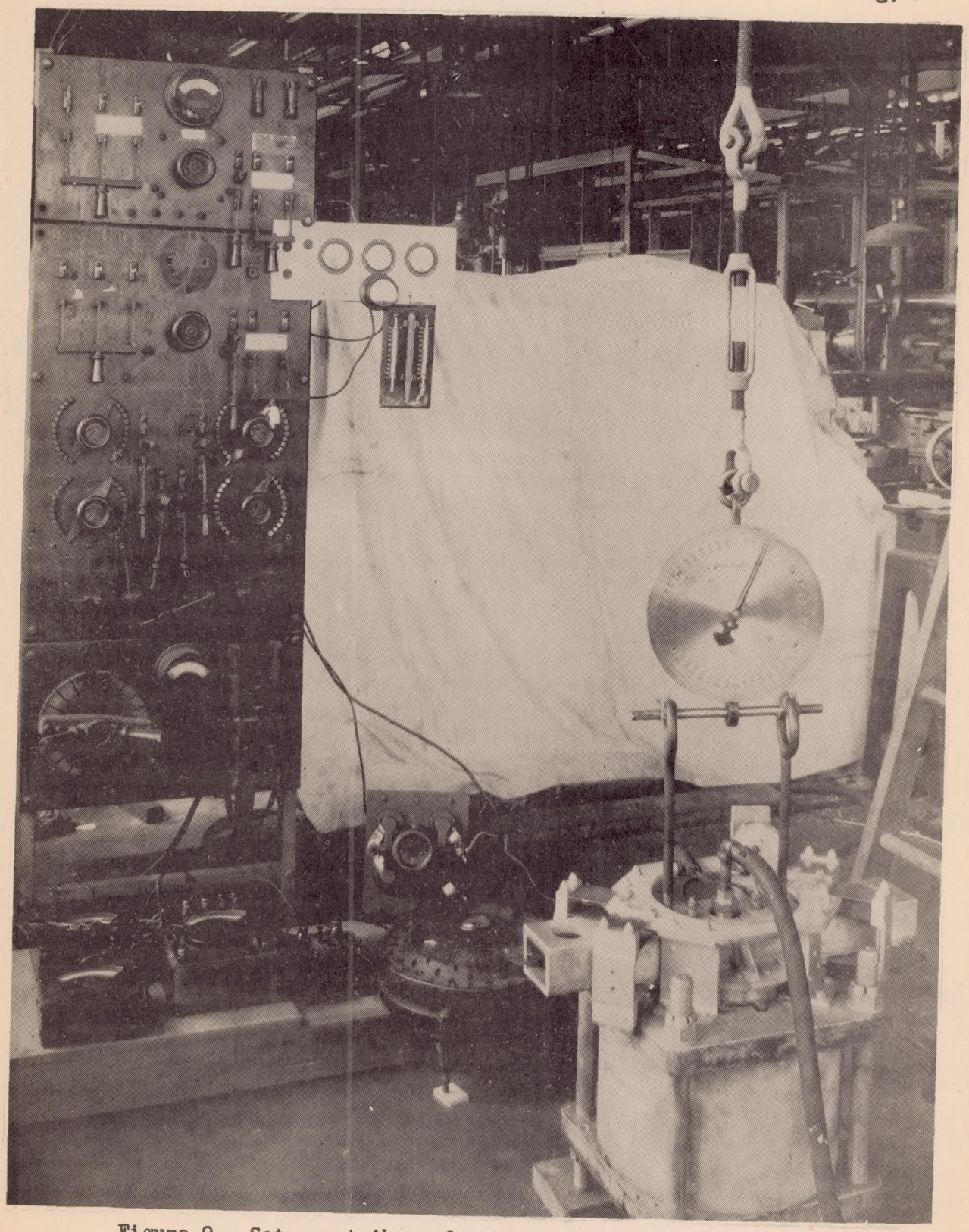
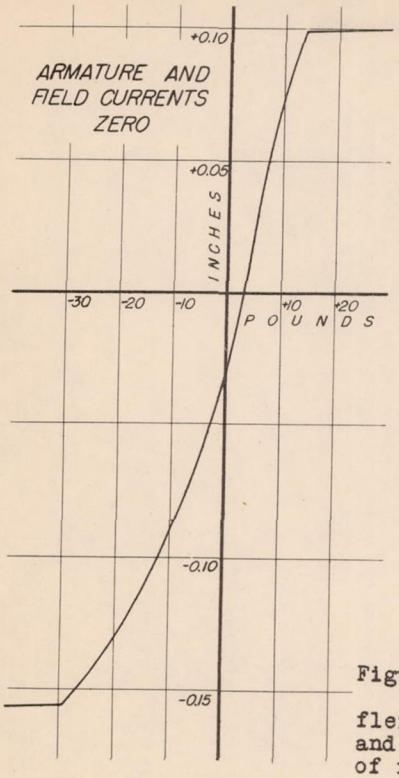


Figure 9.- Set-up at Akron for static load tests on the first reciprocating motor.



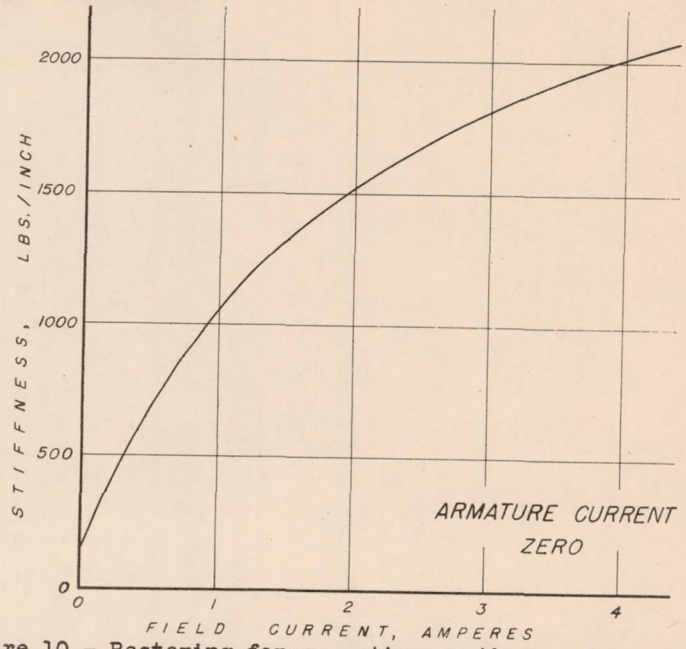


Figure 10.- Restoring forces acting on the armature in the absence of armature current. The small flexure spring force is shown on the left; total spring and magnetic force per inch displacement as a function of field current is shown on the right.

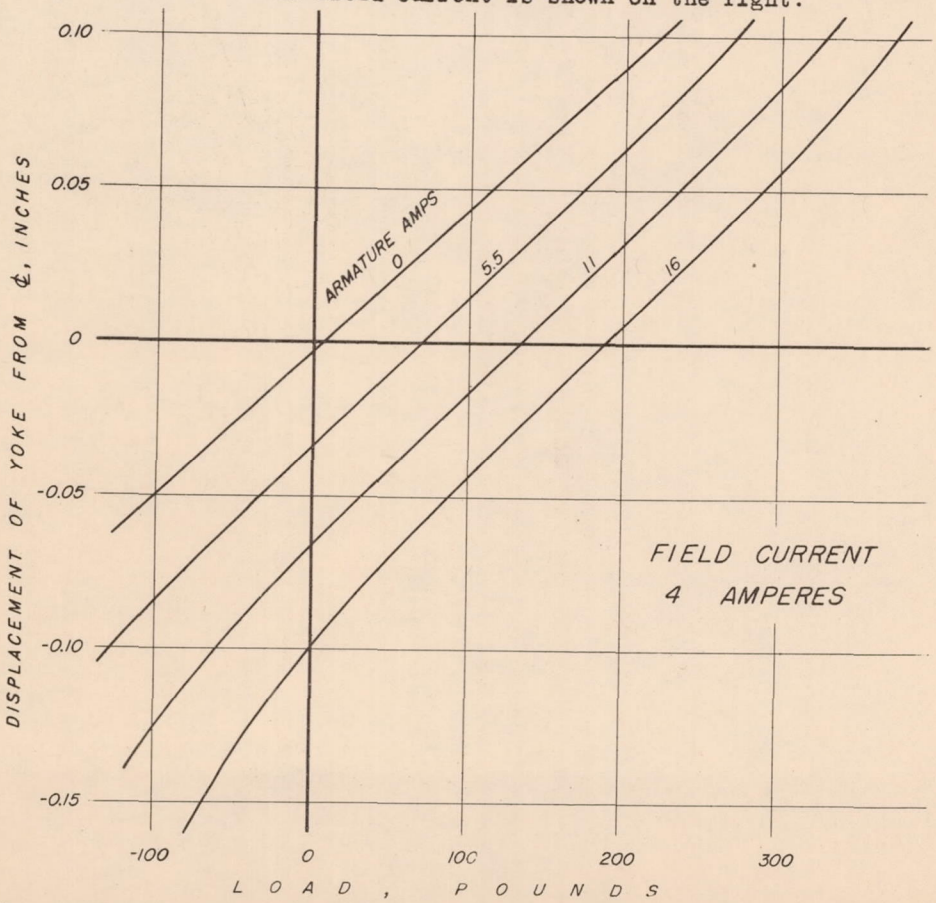


Figure 11.- A number of static load-displacement curves for the motor.

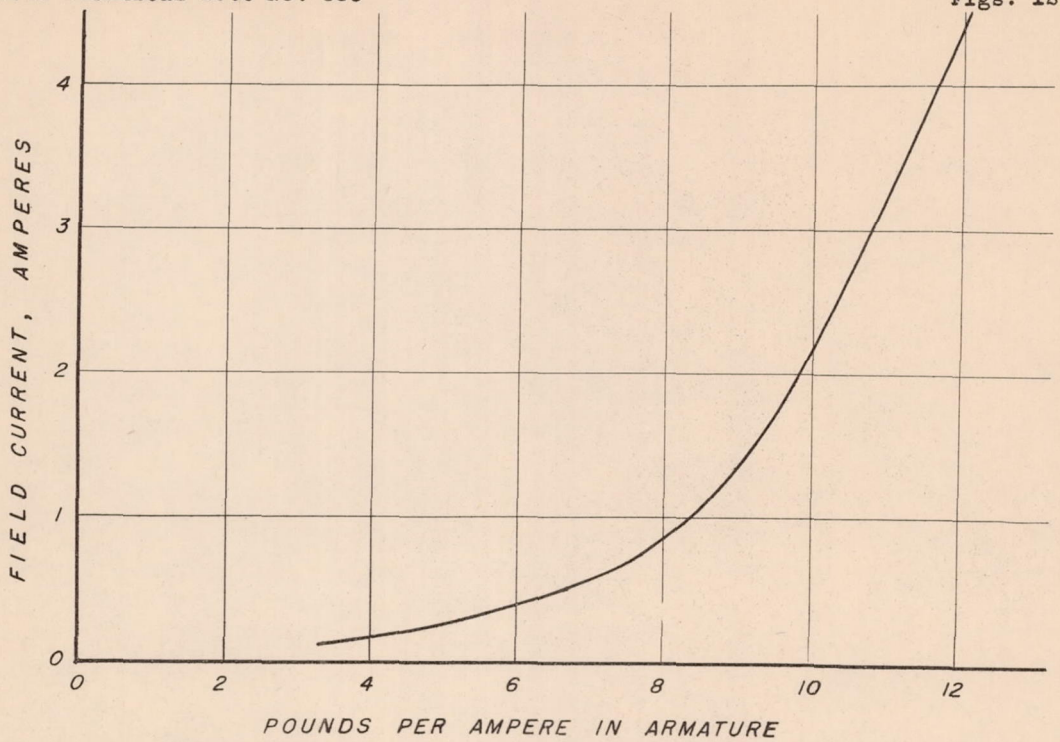


Figure 12.- This curve combines the static load test results to give the force acting on the armature for zero displacement. This is the driving force available for supplying power.

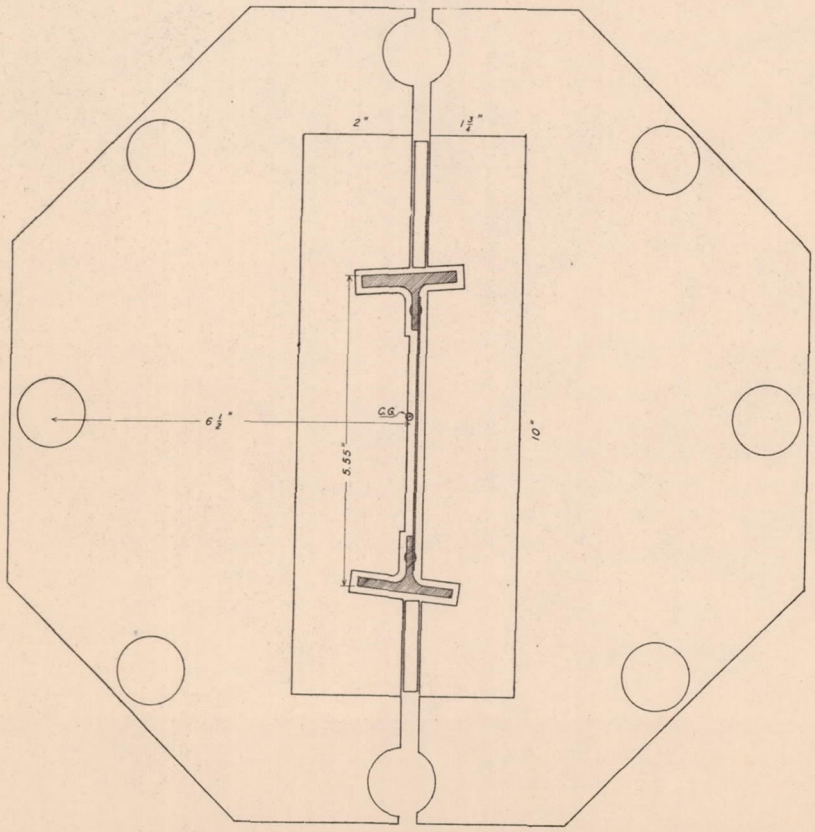


Figure 14.- Design of terminal attachment for rear wing beam.

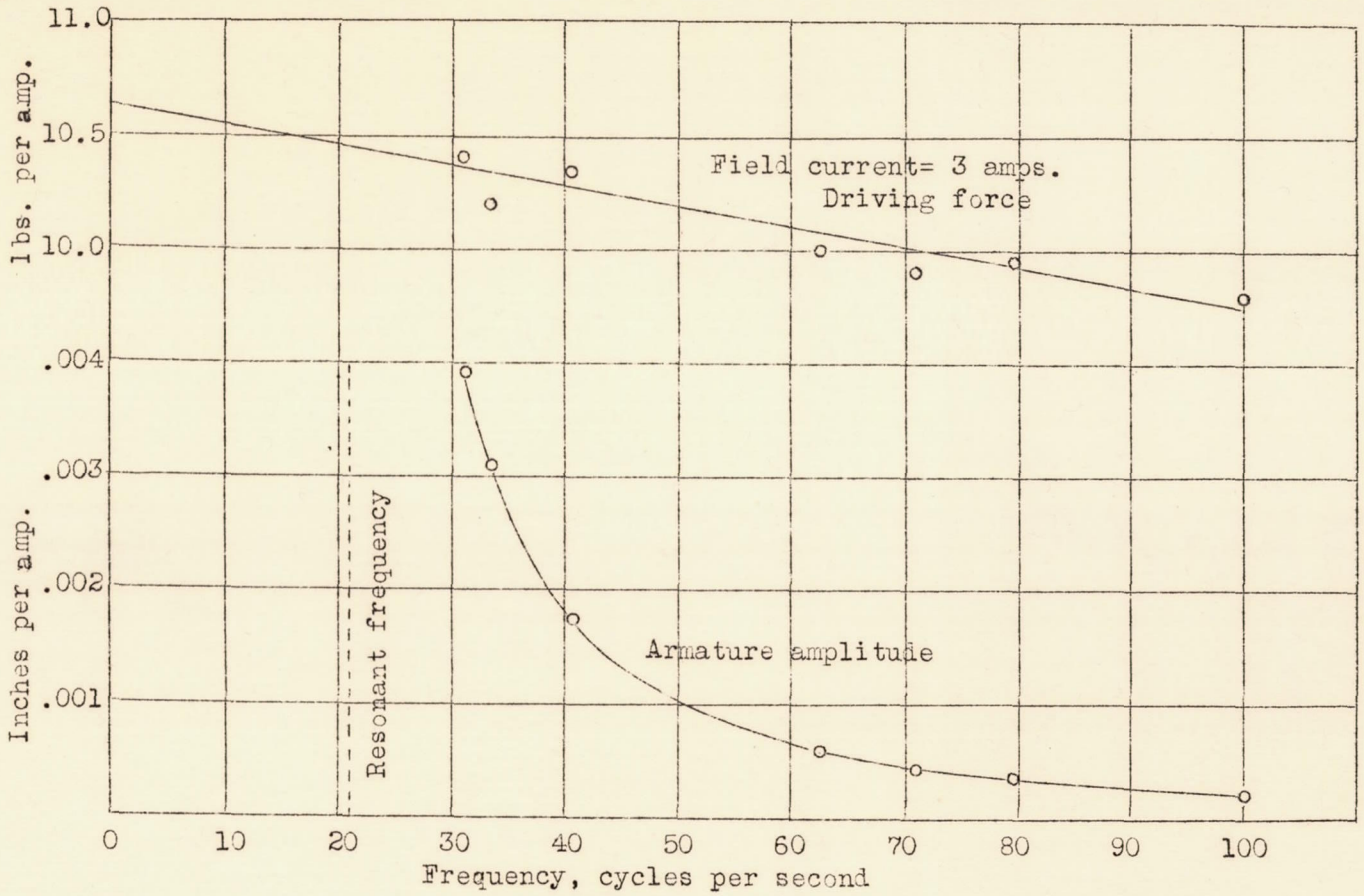


Figure 13.- Driving force and armature amplitude as functions of frequency.

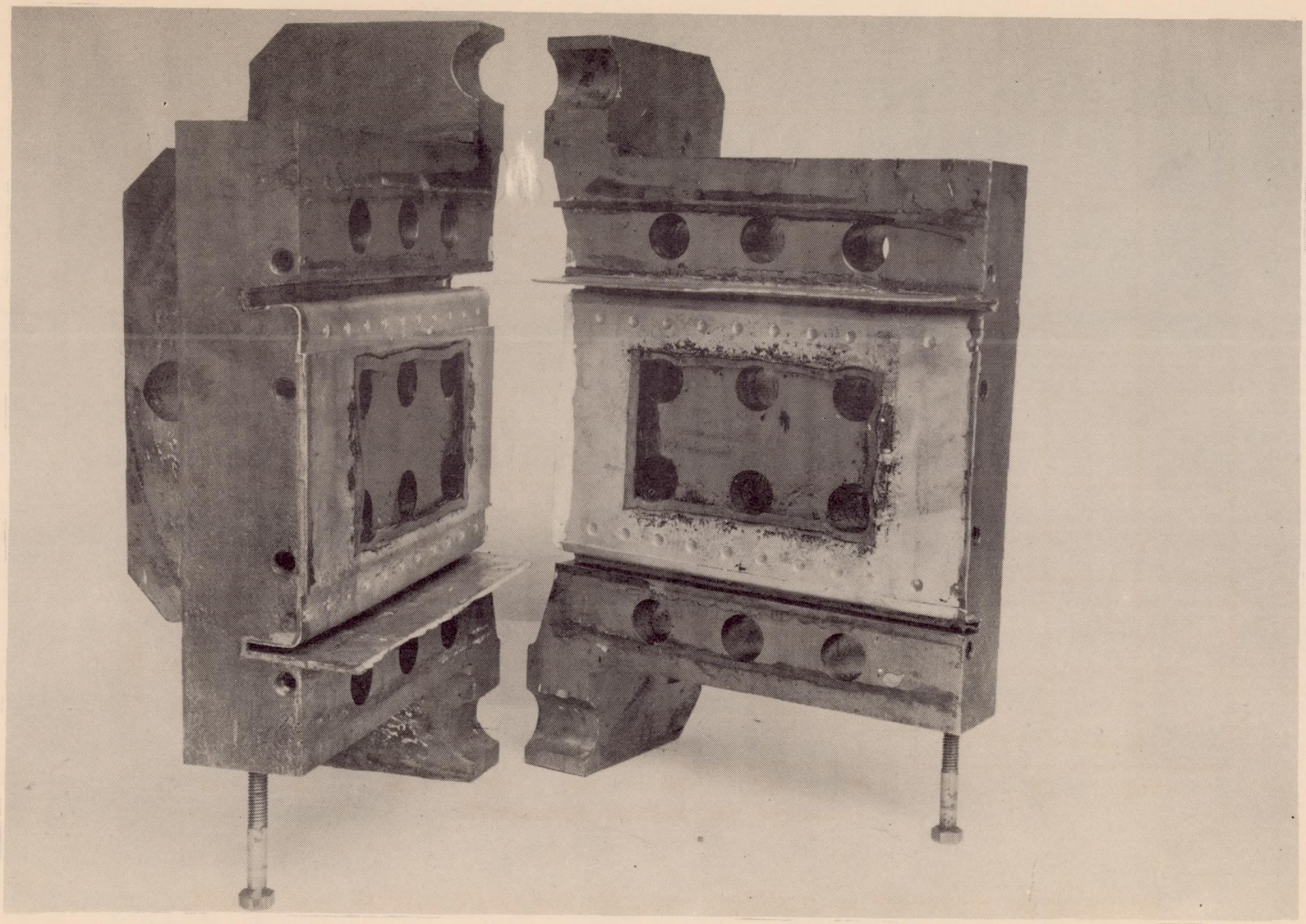


Figure 15.- Rear beam terminal attachment after removal from specimen following the first fatigue test.

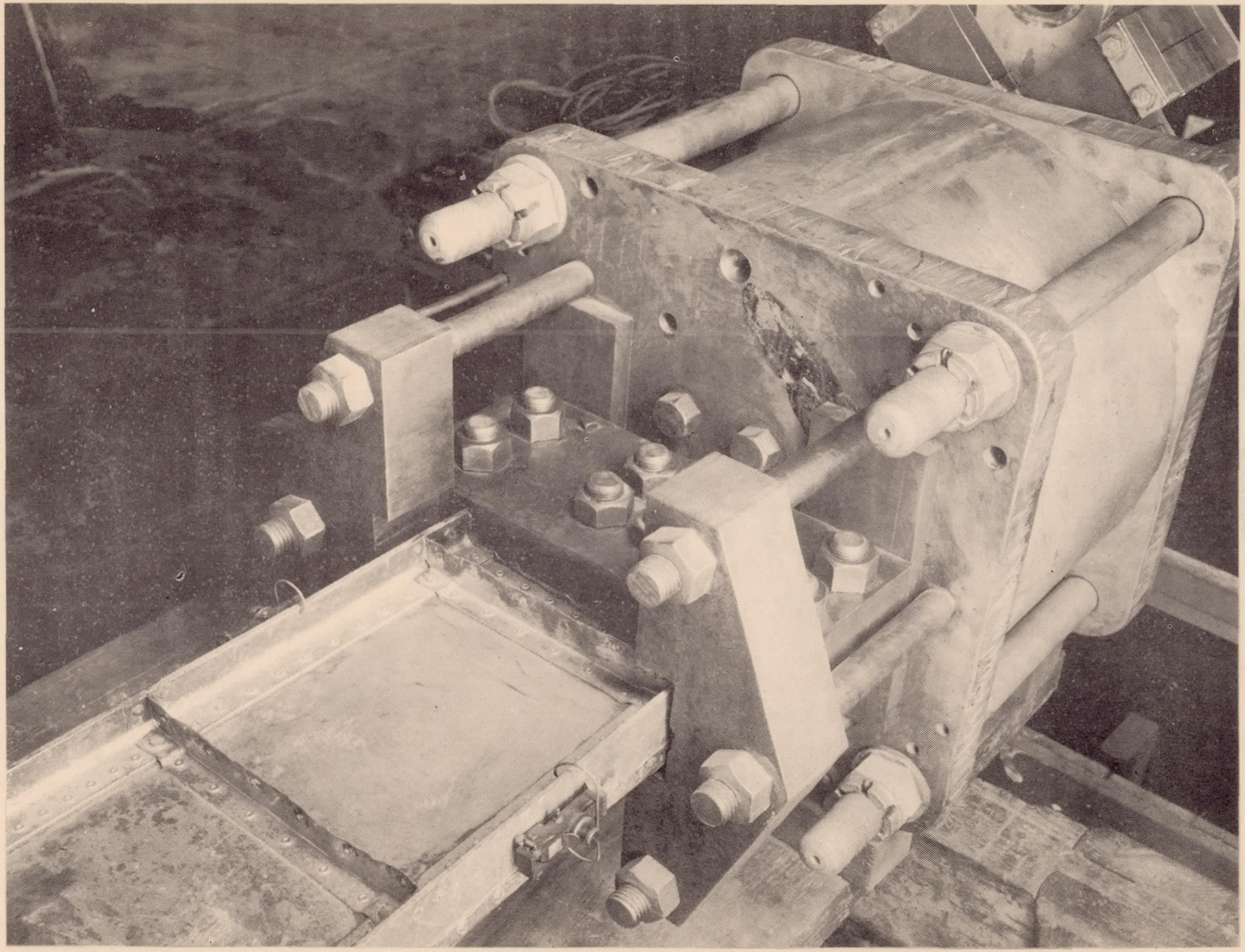


Figure 16.- End portion of front beam with terminal attachment in place.

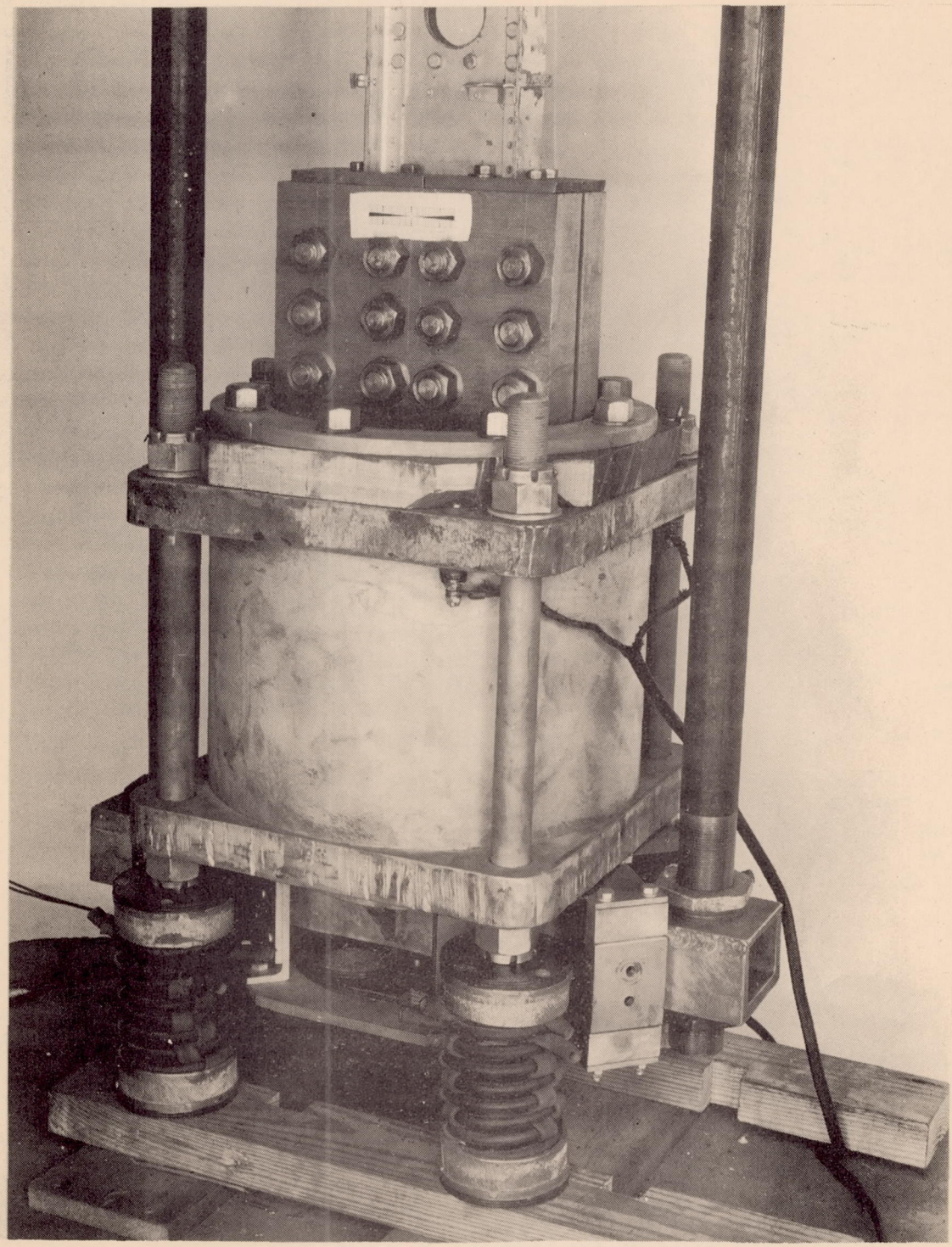


Figure 17.- Detail view of lower end of first fatigue test assembly, showing double wedge for measuring amplitude.

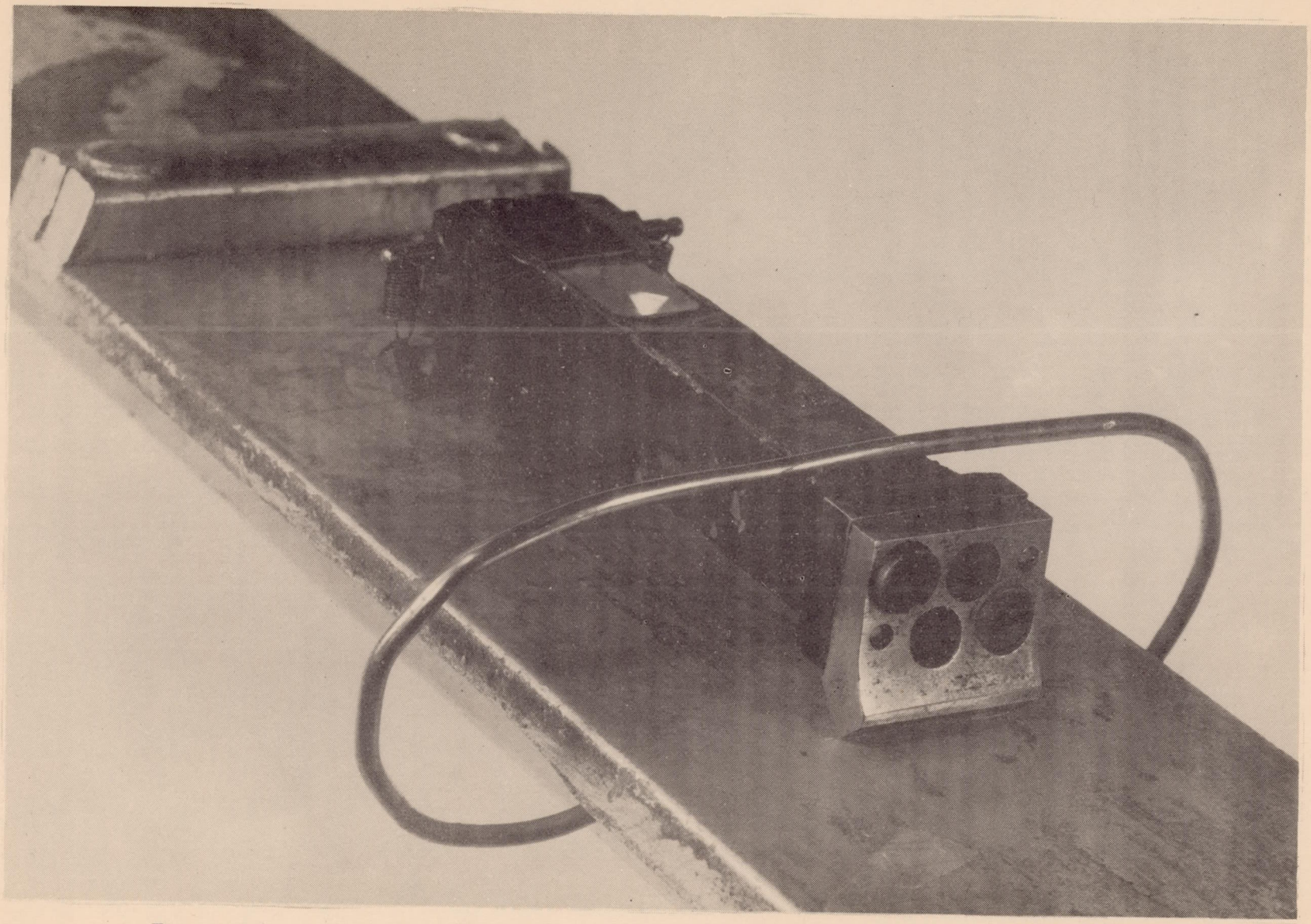


Figure 18.- Light weight Tuckerman optical strain gage clamped to the flange of a wing beam.

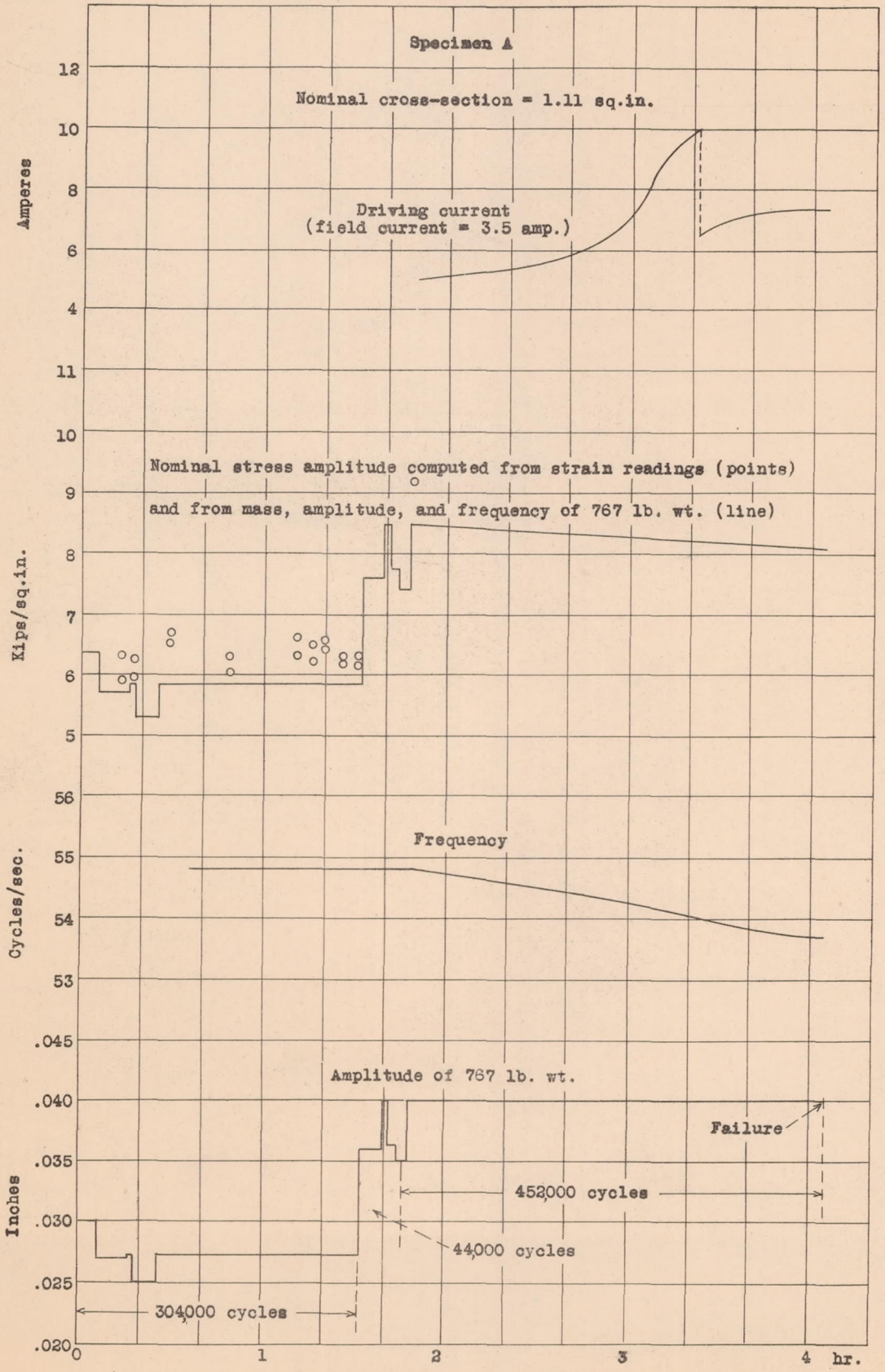


Figure 19. - Log of test data, specimen A

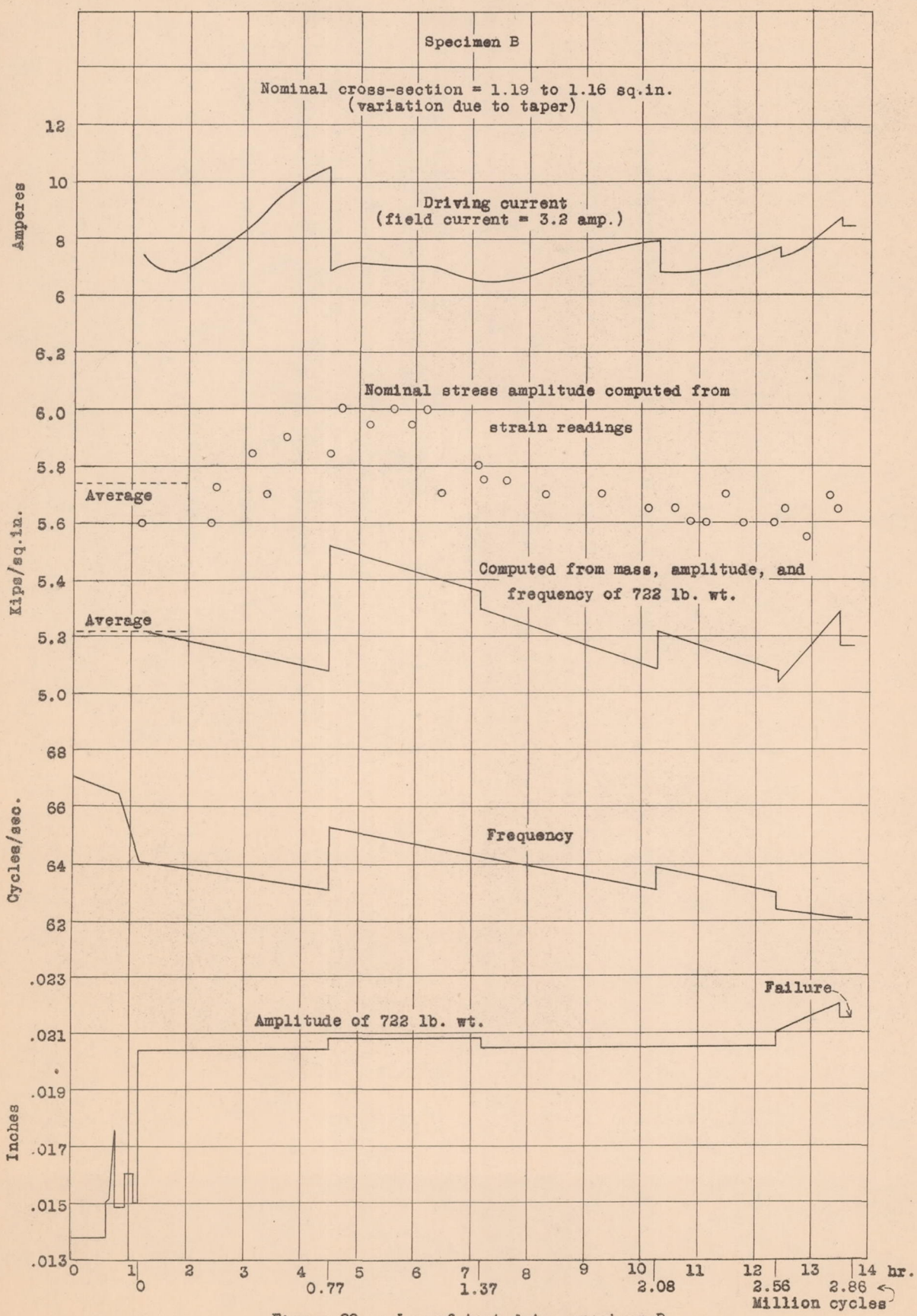


Figure 20. - Log of test data, specimen B.

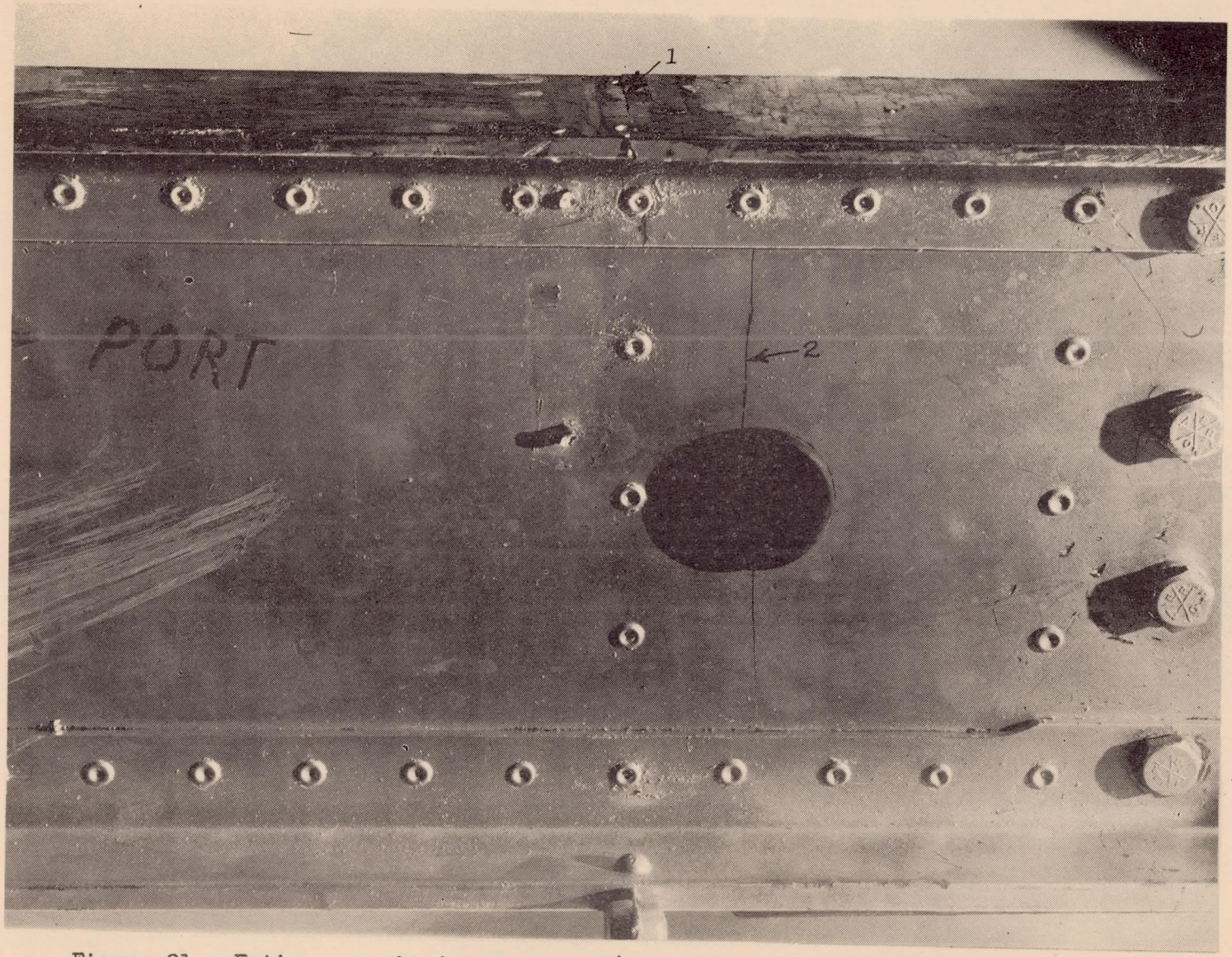


Figure 21.- Fatigue cracks in specimen A(for location on specimen see figure 1).

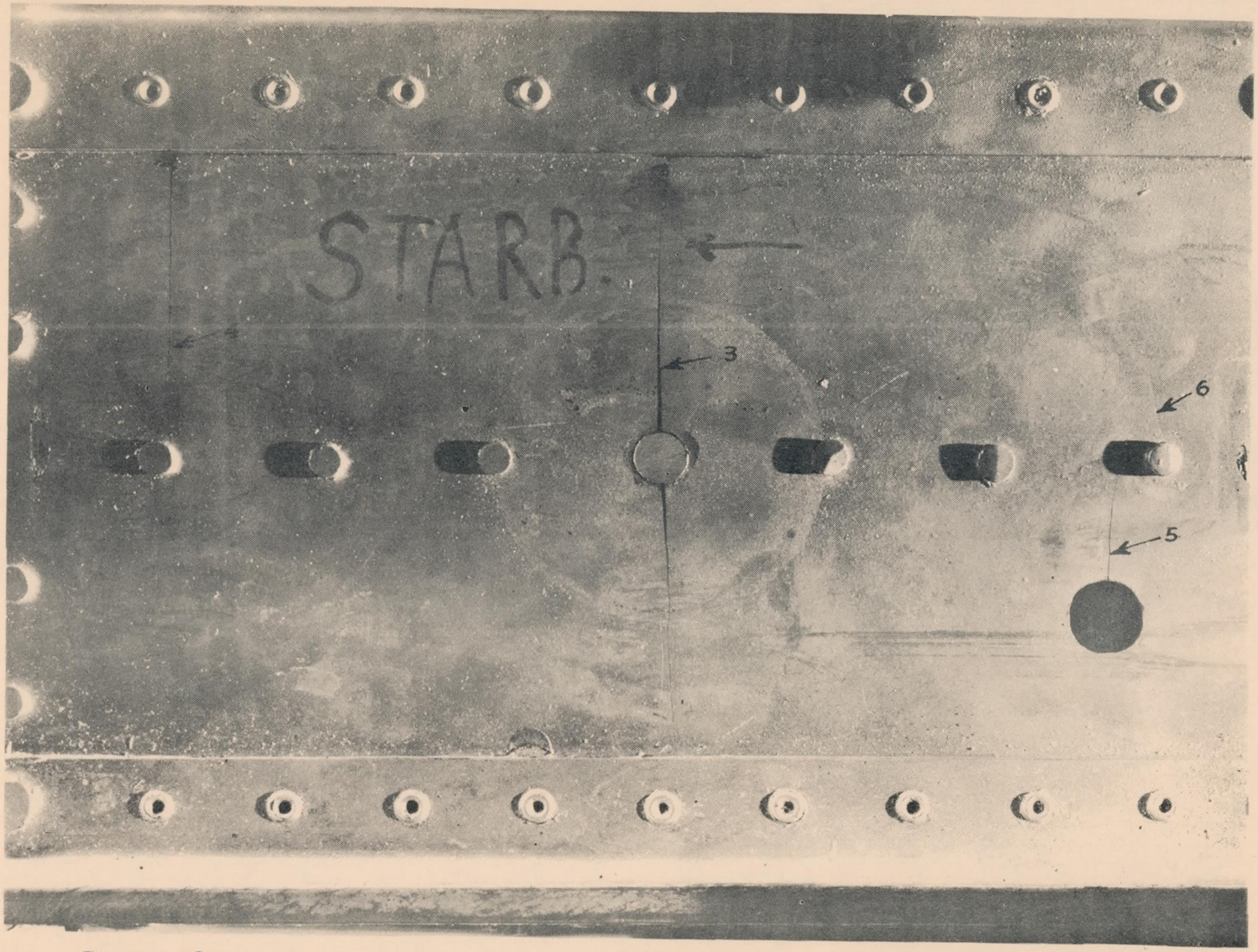


Figure 22.- Fatigue cracks in specimen A(for location on specimen see figure 1).

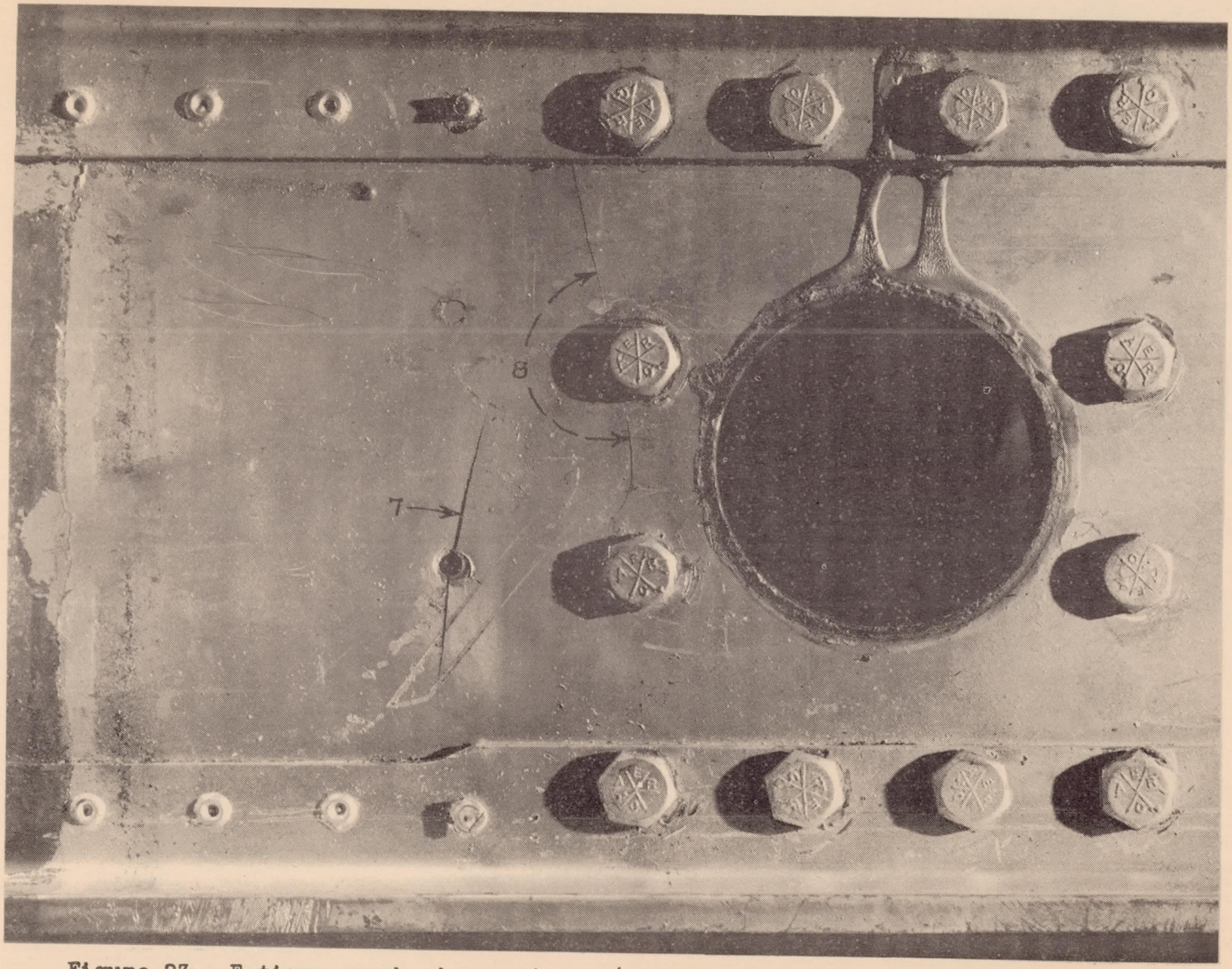


Figure 23.- Fatigue cracks in specimen A(for location on specimen see figure 1).

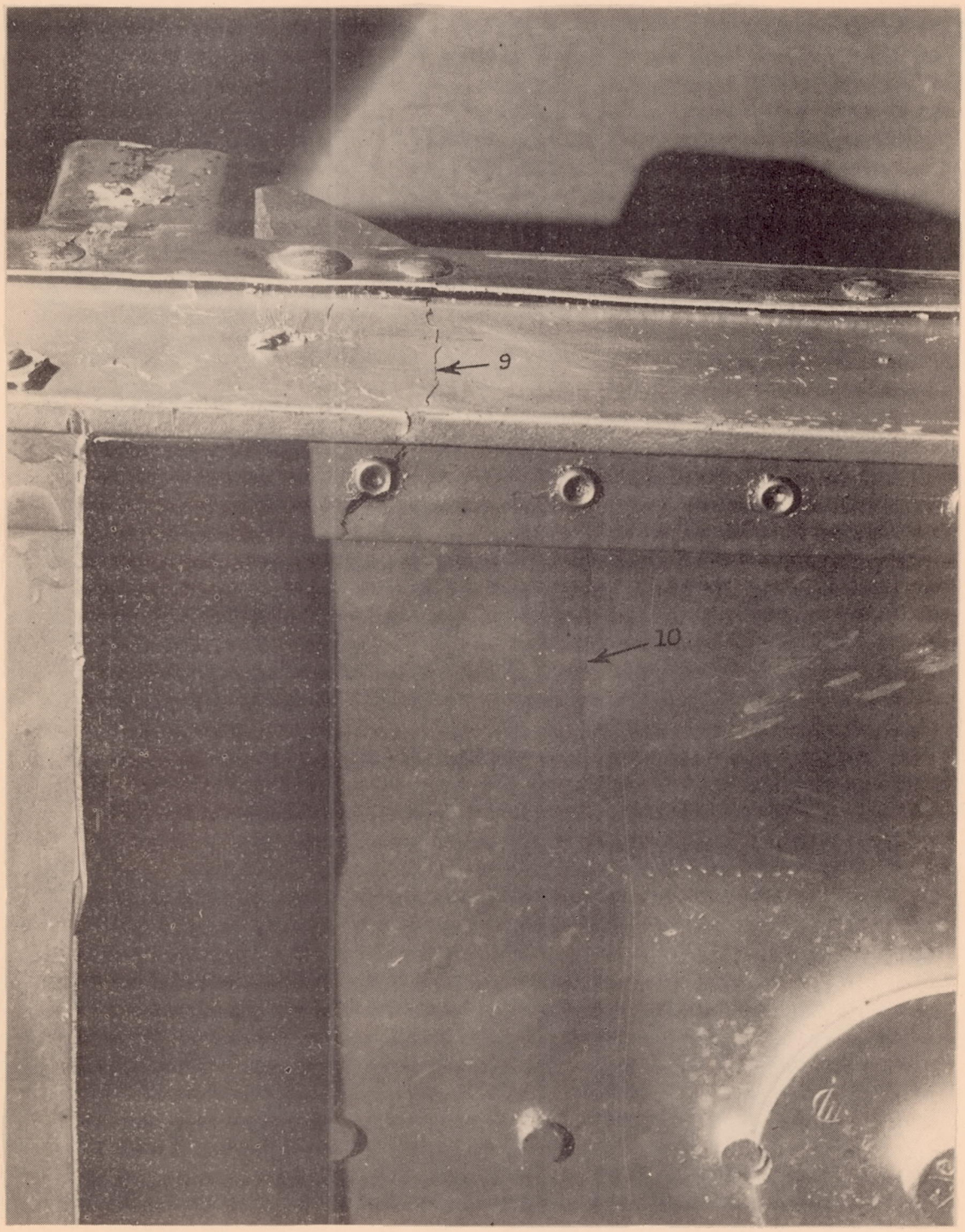


Figure 24.- Fatigue cracks in specimen B(for location on specimen see figure 1).

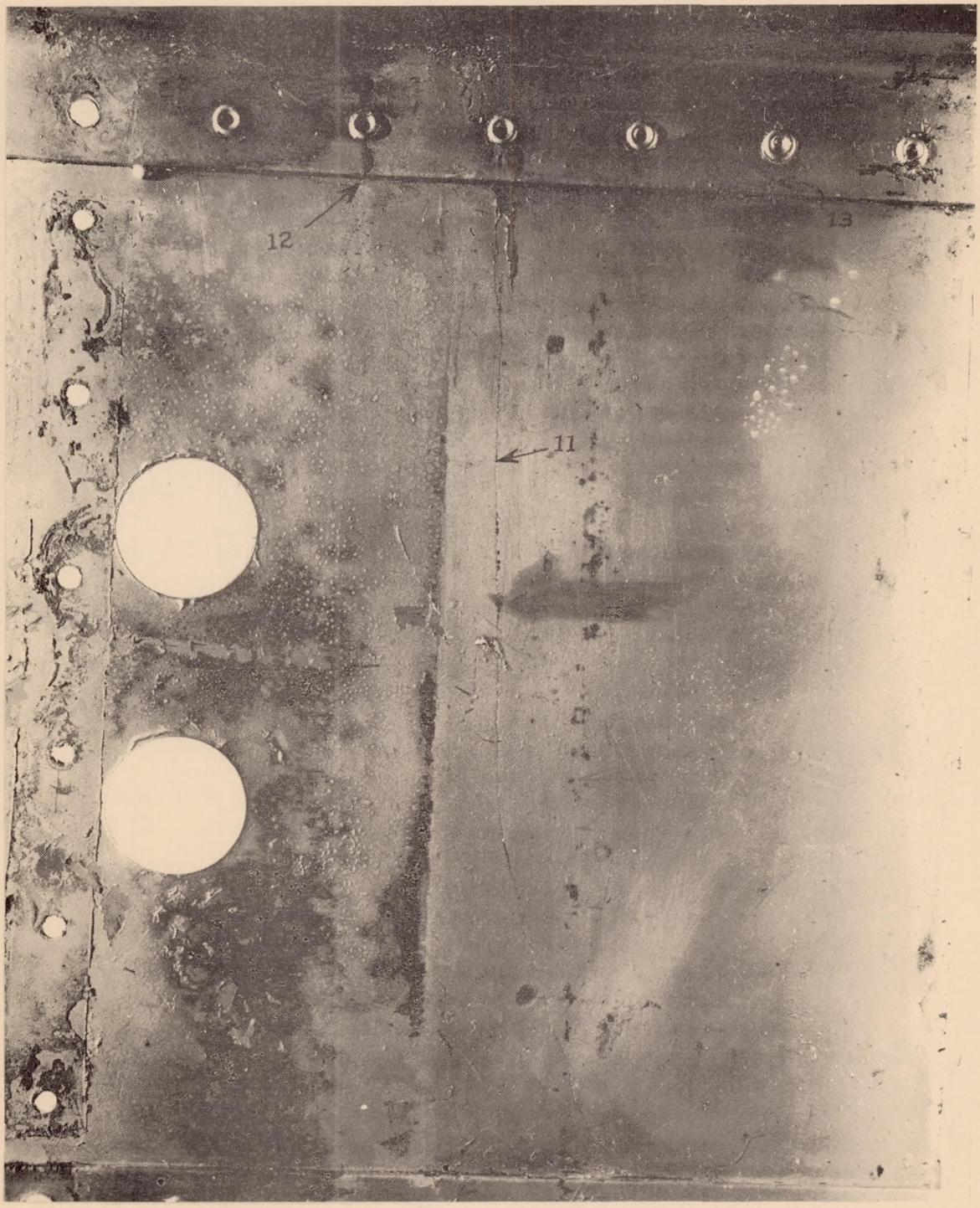


Figure 25.- Fatigue cracks in specimen B(for location on specimen see figure 1).

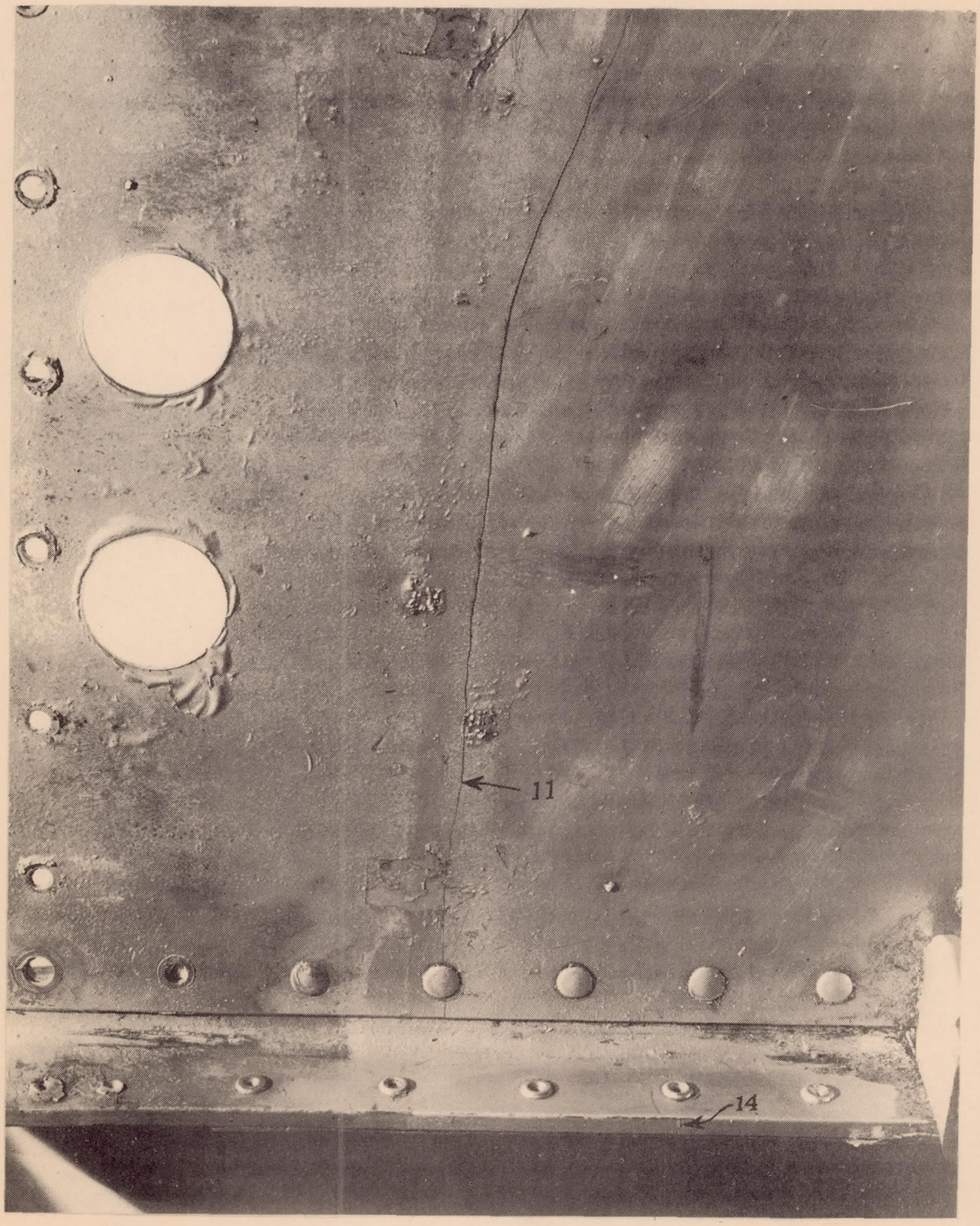


Figure 26.- Fatigue cracks in specimen A(for location on specimen see figure 1).

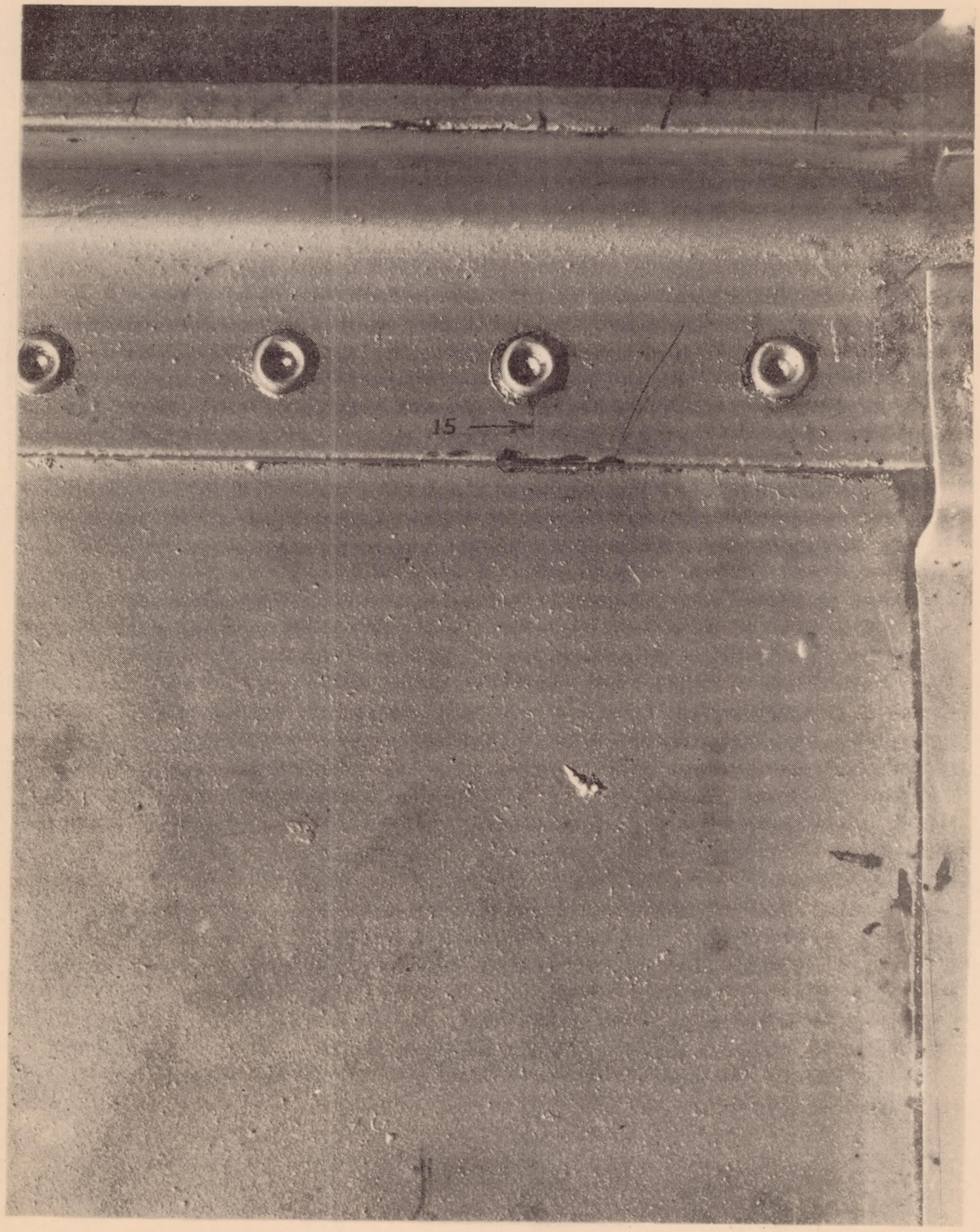


Figure 27.- Fatigue cracks in specimen B(for location on specimen see figure 1).