THE UNIVERSITY OF MICHIGAN
INDUSTRY PROGRAM OF THE COLLEGE OF ENGINEERING

FEATURES AND PERFORMANCE OF A GEAR TEST MACHINE

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INTRODUCTION

To help carry out a program to obtain design data on molded "Zytel" 101 as a gear material, five identical test machines were designed and built by the Engineering Research Institute of The University of Michigan, Ann Arbor. The basic principle employed in these machines is not new. The construction of the machines is quite unusual, however, and several unique features have been incorporated to make them into very useful research instruments.

The evaluation program is being sponsored by E. I. du Pont de Nemours and Co. and is still in progress. However, sufficient work has been completed to show that the test machines perform reliably and well.

PRINCIPLE OF OPERATION

The main function of the machines is to test gears under various conditions of load and speed to determine wear rates, fatigue life, friction losses in the teeth, and the effects of several variables upon these properties. Each machine tests two pairs of gears at one time, with one pair loaded against the other pair by applying a twisting couple to the shafts joining the gears. This is known as "back-to-back" or "4-square" testing and is commonly employed for gear work.

Figure 1 shows a schematic layout of the machine. Each gear is mounted on a splined hollow shaft supported in ball bearings from a common cast-iron base. The hollow shafts are connected by steel torsion bars. The twisting couple which loads the gear teeth is applied through the friction coupling at the right end. One half of this coupling is splined to the hollow gear shaft, while the other half is splined to the torsion bar. The two halves of the friction coupling are held together by four equally spaced clamping screws which extend through circular slots in the outer half of the coupling and are threaded into the inner half of the coupling. By twisting one half of the coupling relative to the other, and clamping the two halves together, a torque is imparted to the system, which puts equal loads on the teeth of all four gears.

These tooth loads remain essentially constant while the gears are rotated by a driving motor, which provides only enough power to overcome friction and windage losses. This eliminates any need to develop, absorb, and dissipate the power transmitted by the gears, a particularly worthwhile advantage on life or endurance tests.

Figure 2 shows one of the machines in operation, testing four identical gears of 2-1/2-inch pitch diameter. A sheet-metal cover, not shown in Fig. 2, is installed to cover the rotating parts when test work is in progress. The ball bearings are installed with caps to permit easy assembly, and this is clearly visible in Fig. 2.

LOADING THE GEAR TEETH

The friction coupling is twisted by a system of cables, pulleys, and weights, as shown in Fig. 3. One cable anchors the inner half of the coupling, while two other cables with buckets of lead shot attached to their ends twist the outer half of the coupling. This rather elaborate loading system is used to eliminate bending from those parts which support the two halves of the coupling. The cables are fastened to the coupling with small screws threaded radially into the coupling. After clamping the coupling halves together, the cables and their fastening screws are removed to permit rotation.

To prevent bending, the same weight of lead shot is placed in each bucket. Thus, the torque exerted on the gears is the product of the weight of one bucket and the diameter of the friction coupling. Any desired torque can be obtained by varying the weight of shot in the buckets.

CAPACITY OF MACHINE

Gears having pitch diameters of from one to four inches can be accommodated by moving the bearing supports to obtain the proper center distance. The four bearing supports of the rear, or right-hand, gears of Figs. 2 and 3 are movable as a unit, while the other supports are integral with the base.

Torsion bars having diameters of 1/8, 3/16, 1/4, and 5/16 inch are available for each machine. By the use of the proper torsion bars, a large angular displacement of the friction-coupling halves can be obtained with any torque load applied, thus virtually eliminating any change in load due to gear-tooth wear as the test progresses. The torsion bars are made of mild steel. Maximum torque capacity, using the 5/16-inch-diameter bars, is considered to be 180 lb-inches.

METHOD OF DRIVING

Each machine is driven by its own 1/2-horsepower, 1725-rpm motor through a speed variator. Operating speeds from 600 to 5000 rpm are thus available in infinitely variable ratios.

To permit investigation of friction losses in the gear teeth, a flexure-plate coupling was built into each machine. This is shown schematically in Fig. 1. It consists of a steel flexure plate clamped at its center to the shaft driven by the speed variator. An aluminum shroud containing two steel pins pressed into place 180° apart is splined onto the first gear shaft. Thus the shroud is driven by the flexure plate contacting the steel pins.

Bonded SR-4 strain gages mounted on the flexure plate indicate the strain, and hence the torque being supplied to overcome friction and windage in the test machine. Four active gages are used, connected to form the Wheatstone bridge as shown in Fig. 4. To facilitate replacement of the strain gages, the connecting is done at a fiber disk immediately adjacent to the flexure plate. The circuit to the strain-gage indicator is completed by four silver slip rings and carbon brushes. Details of construction of the flexure-plate drive are shown in Figs. 5 and 6.

It has been found that the flexure plate works very well provided the drive is smooth with little or no vibration. To obtain the required smoothness, however, several changes had to be made from the original method of supplying power to the test machines. The machines were at first assembled with 1/4-horsepower, 1725- and 3450-rpm motors driving through V-belts with adjustable pulleys to get the desired speed variation. This proved to be impractical, however, since at high speeds the entire power output of the motor was absorbed by vibrations of the belts.

The use of short V-belts with fixed pulleys reduced the vibration considerably and allowed the 1/4-horsepower motors to drive the machines without difficulty. However, small variations in the cross-section dimensions of the belts caused short but constantly recurring periods of acceleration and deceleration of the test machines. This condition causes no harm in most machines, but with the sensitive flexure plate these changes in speed and load caused the strain-gage indicator to fluctuate excessively.

The present drive of motor and speed variator, in line and coupled with 6-1/2-inch pieces of rubber hose, proved to be a good solution to the problem.

The variator contains a friction drive through cones and balls, and is very smooth. Even so, unless the hose couplings are used, alignment of all three units must be virtually perfect to allow proper functioning of the flexure plate. With the hose couplings, however, a reasonable amount of misalignment has no harmful effect.

The flexure plate is calibrated statically as shown in Fig. 7. While the shaft beyond the flexure plate is held with a wrench, weights are hung from the friction coupling to develop a known torque. Strain readings are plotted against torque to form a calibration curve. It is felt that this calibration is not affected by the speed of rotation of the flexure-plate assembly, and hence that no significant error is introduced by the use of this static calibration.

LUBRICATION

The test machines are lubricated by individual oil-mist systems employing Alemite Lubricators as shown in Fig. 8. Tubes carry the mist to each bearing and to the test gears. The quantity of lubrication can thus be controlled as desired. When the test gears are to be operated without lubrication, the mist system is shut off and the bearings are packed with grease.

BEARING TESTING

With but a slight change in assembly, this same machine is used to measure the friction torque in its own ball bearings under various conditions of load and speed. This is a necessary part of the investigation of gear-tooth friction.

To measure the friction torque of the eight bearings supporting the test gears, the machines are assembled as shown schematically in Fig. 9.

Bearings from the rear shafts are installed in place of the gears on the front gear shafts. Load is applied by spring scales to each of the bearing assemblies through the steel bearing rings shown. Thus, all four bearings in each assembly carry the same load. The torsion bar connects the two front gear shafts and allows the motor to rotate both shafts and their bearing assemblies. The flexure plate records the torque required to rotate this assembly. This is the torque required to overcome bearing friction and windage losses in the rotating assembly.

Figure 10 shows one of the machines set up for bearing-friction testing. The two load-applying scales are anchored in the floor at their lower ends.

GEAR TESTING

Before starting the gear testing, the center distance of the gear shafts must be carefully set to the desired value, and the backlash, or clearance, between the gear teeth measured. Figure 11 shows the way in which the center distance is set. A ground-steel bar having the same outside diameter as the ball bearings is installed in the rear bearing supports. The rear bearing supports are adjusted until the distance between the ground bar and the outside diameter of the front bearings is sufficient to provide the desired center distance. This distance between the bar and bearings is measured with gage blocks, and in this way the center distance can be readily set within 0.0005 inch of the desired value.

Backlash, or clearance, between the teeth is measured with the setup shown in Fig. 12. Because "Zytel" 101 has a low modulus compared to metals, it is difficult to establish the backlash by the common procedure of mounting a small indicator with its pointer on one tooth, holding the other gear, and rotating the first to determine the clearance. The teeth deflect readily, making it difficult to tell when the clearance has been taken up. The device shown anchors the left-hand gear and clamps an indicator on the right-hand gear. The weight of the dial indicator tends to rotate the right-hand gear clockwise, thus taking up the clearance between the teeth. A small weight hung on the cord over the pulley then exerts a counterclockwise torque on this same gear, rotating it to take up the clearance in the opposite direction. The change in indicator reading, multiplied by the ratio of the lever lengths involved, is taken as the backlash.

To permit investigation of dynamic action of the gear teeth and resulting variations in torque load on the torsion bars, provisions are made to mount strain gages on the short torsion bar and to bring the circuit out through rings and brushes. This assembly is shown in Fig. 13. This feature has not been used in the test program so far, however, since no dynamic investigations have been started.

During the endurance testings of the gears, flexure-plate strain readings are taken every half hour. Immediately after taking the reading the machine is shut down, the flexure plate freed of contact with the driven pins by rotating the shaft backwards a slight amount by hand, and a zero reading is taken. The difference between these two readings is used with the flexure-plate calibration to determine the torque being supplied to overcome windage and friction in the test machine. At the start and at the close of each day's operation, similar data are obtained with the machines running but with no load on the gear teeth. The no-load flexure-plate readings are used with the bearing-friction data to determine the friction torque of the gear teeth only.

TEST RESULTS

The gears are removed for inspection at intervals during the endurance tests, and in this way wear rates and other information are obtained. Only those test results which indicate the way in which the machines perform are mentioned here, however.

Figure 14 shows ball-bearing friction torque as measured by the machines. It will be noticed that this friction torque is not zero when the bearing load is zero. Instead, the zero-load friction torque increases with the speed of rotation. It is reasoned that this zero-load friction torque is caused by clamping forces exerted on the bearings by the bearing caps, by the balls sliding against their cages, and by windage. Hence this same friction is measured when the gears are operated without load at the start and at the close of each day's operation.

The increase in bearing-friction torque above the zero-load friction torque shown in Fig. 14 is considered to be caused by the bearing load and hence is not included in the readings taken when the gears are operated without load. Therefore, to determine the torque required to overcome friction in the gear teeth only, the torque required to drive the machine with no load on the gear teeth is subtracted from the torque required to drive the machine when the teeth are loaded. From this difference is subtracted the bearing-friction torque above zero-load friction torque to obtain the gear-tooth friction torque. The bearing-friction torque above zero-load friction torque is obtained by drawing curves similar to that shown at the bottom of Fig. 14. Drawn parallel to the 2000- and 3000-rpm curves, it is used in the determination of gear-tooth friction torque at 2500 rpm.

Typical results obtained with this procedure are shown in Fig. 15.

It will be noted that gear-tooth friction is shown to be zero when no torque is being transmitted. This would seem to verify the validity of the procedure used to determine the gear-tooth friction torque.

During the test work sets of gears have been interchanged among the machines to see if the various machines could reproduce the same gear-tooth friction torque results. Although the magnitudes of the flexure-plate strain readings varied with the five machines, essentially the same results were obtained with the same gears in different machines. This further illustrated the validity of the method and the reliability of the machines.

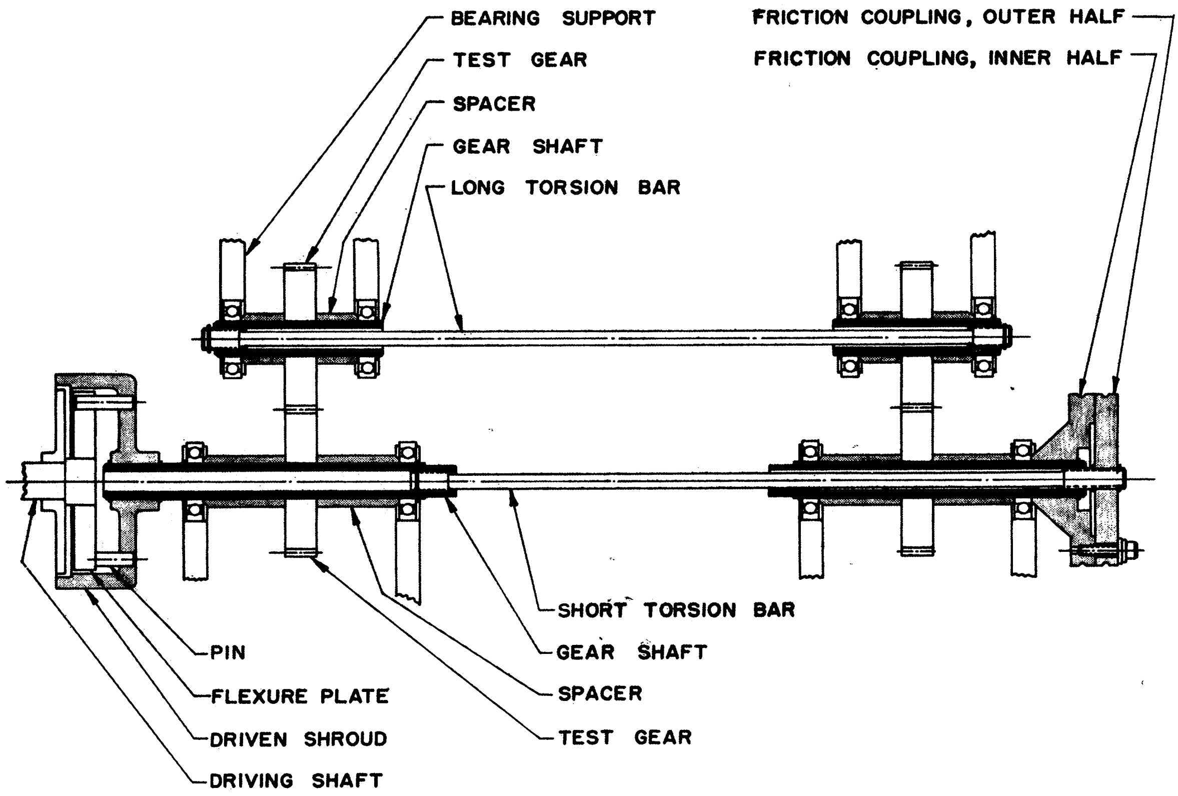

Fig. 1. Schematic layout of test machine.

Fig. 2. Test machine in operation.

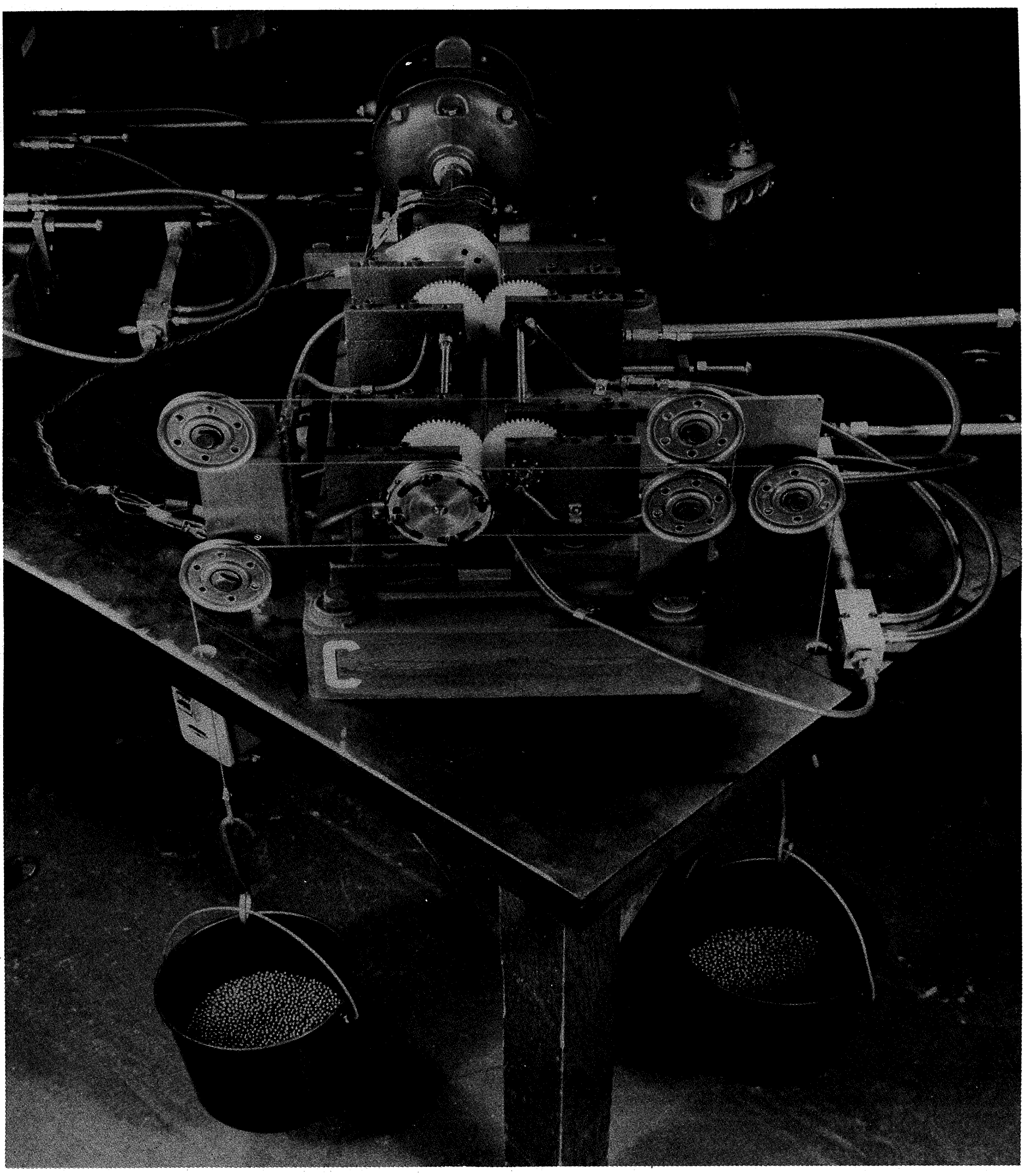

Fig. 3. Method of loading gear teeth.

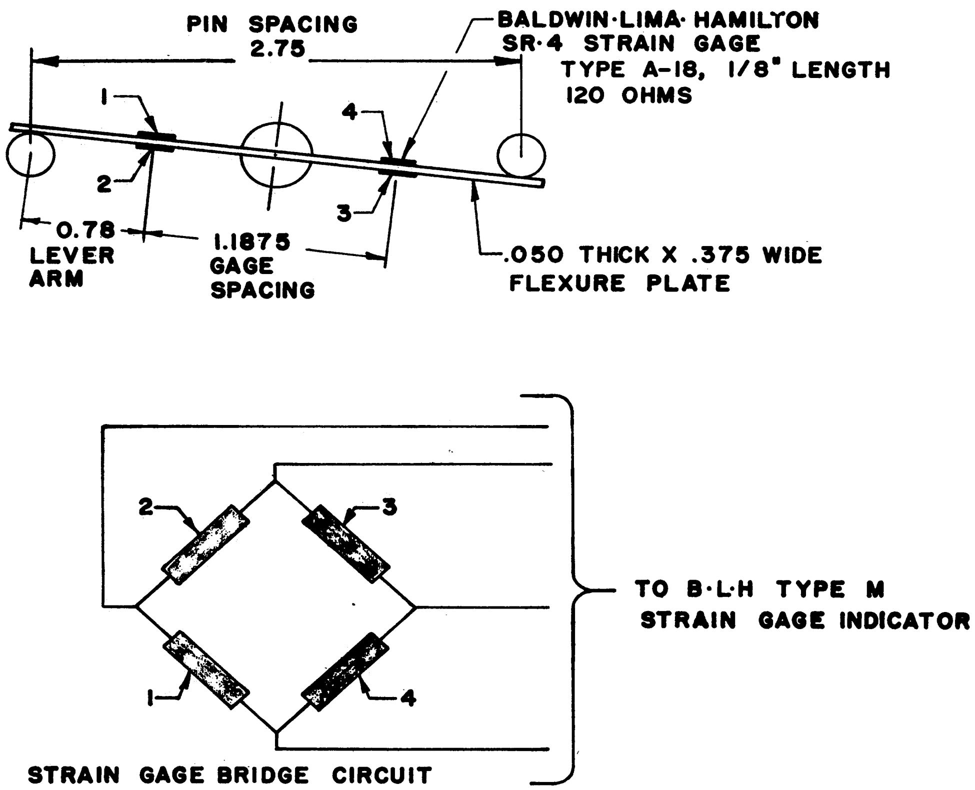

Fig. 4. Flexure-plate strain-gage circuit.

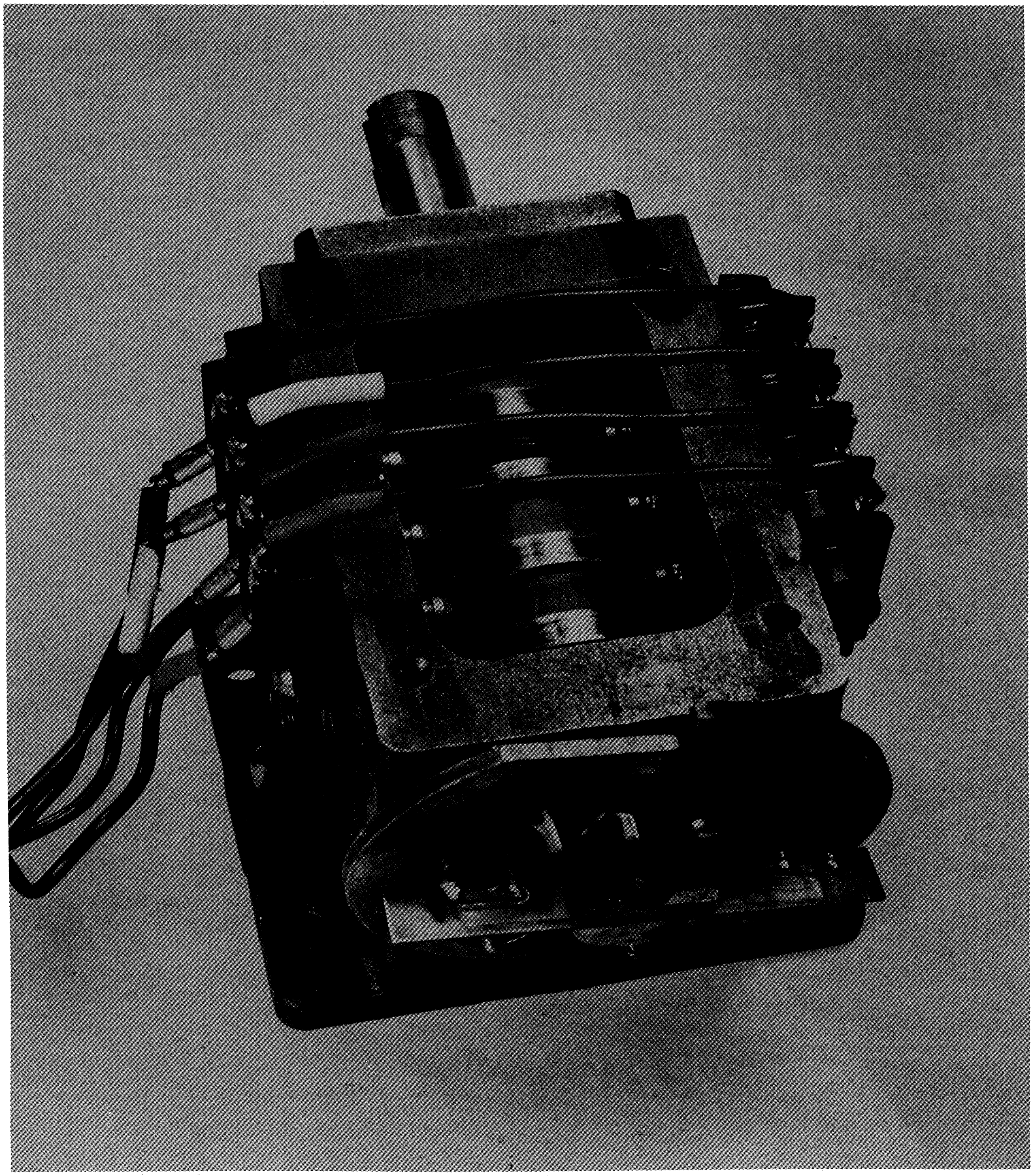

Fig. 5. Flexure-plate assembly, top view.

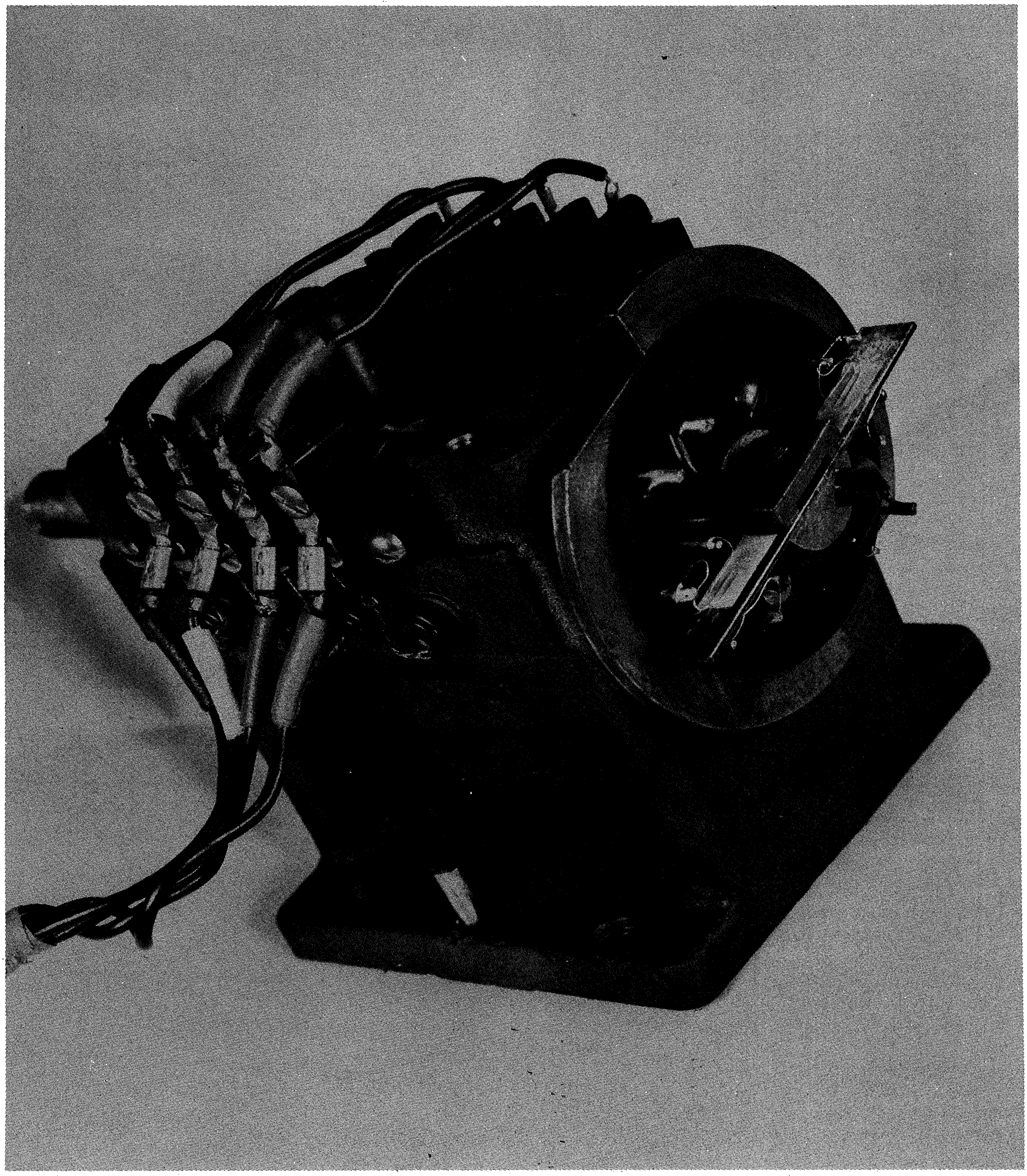

Fig. 6. Flexure-plate assembly, oblique view.

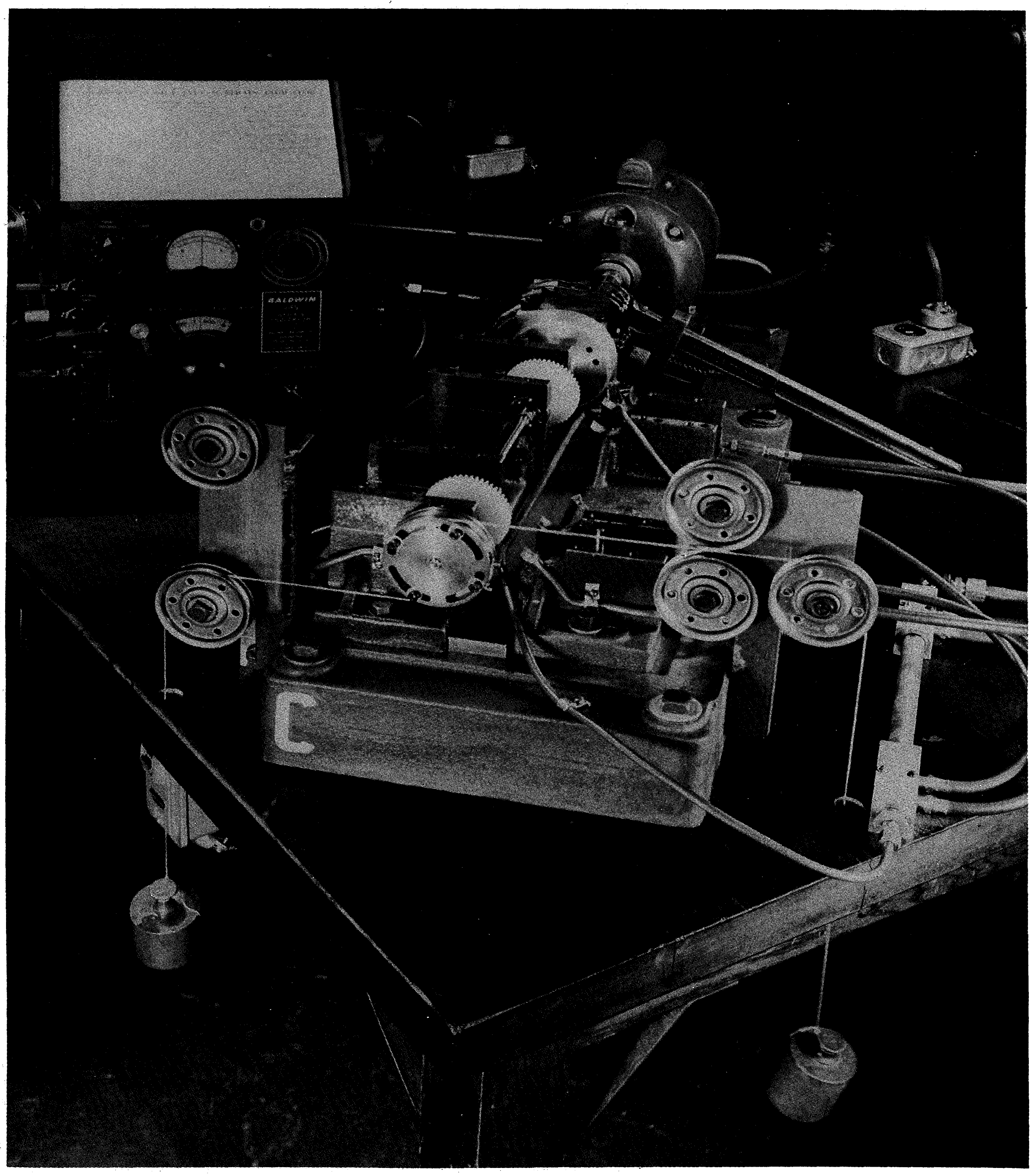

Fig. 7. Flexure-plate calibration.

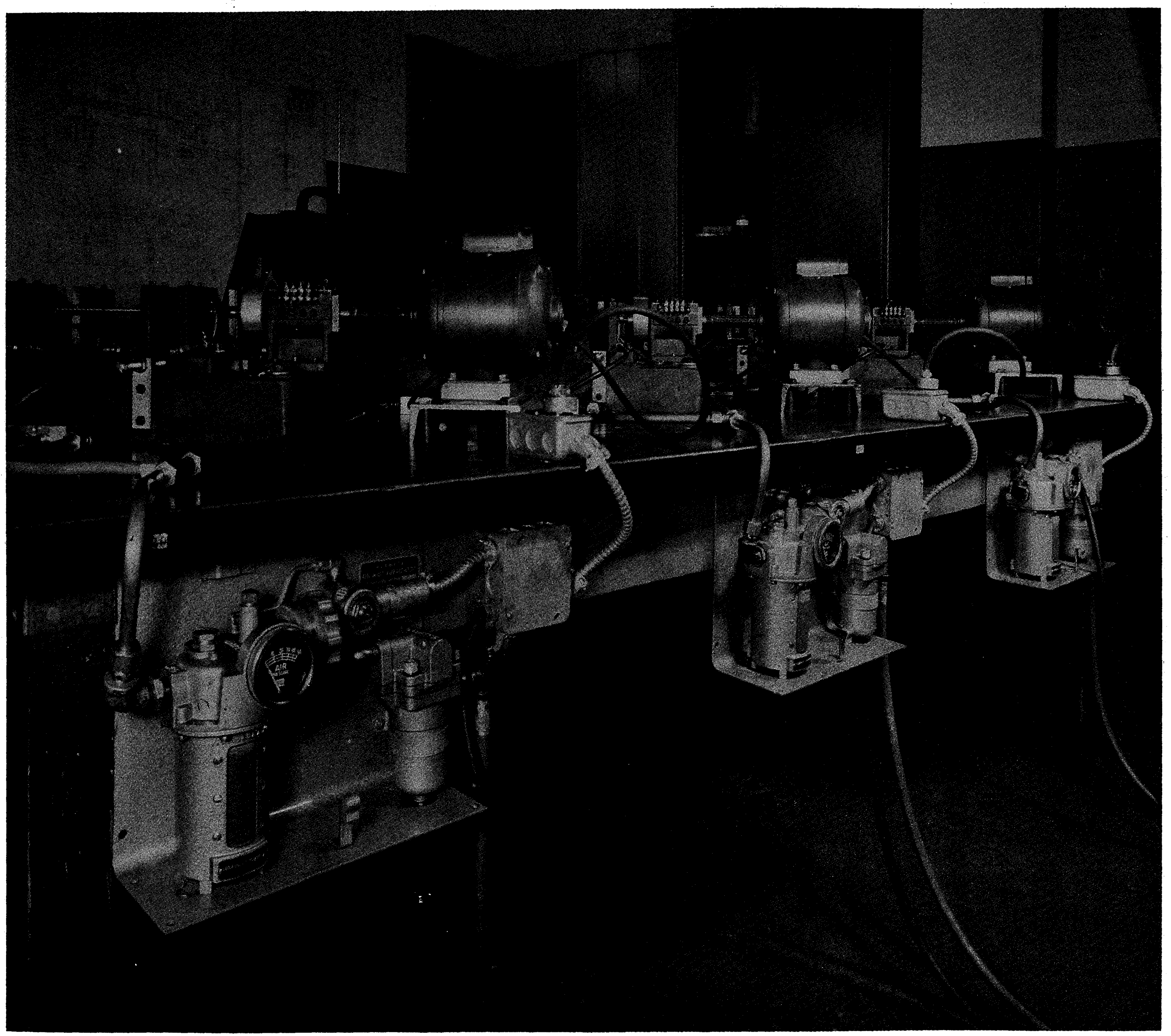

Fig. 8. Oil-mist lubricators.

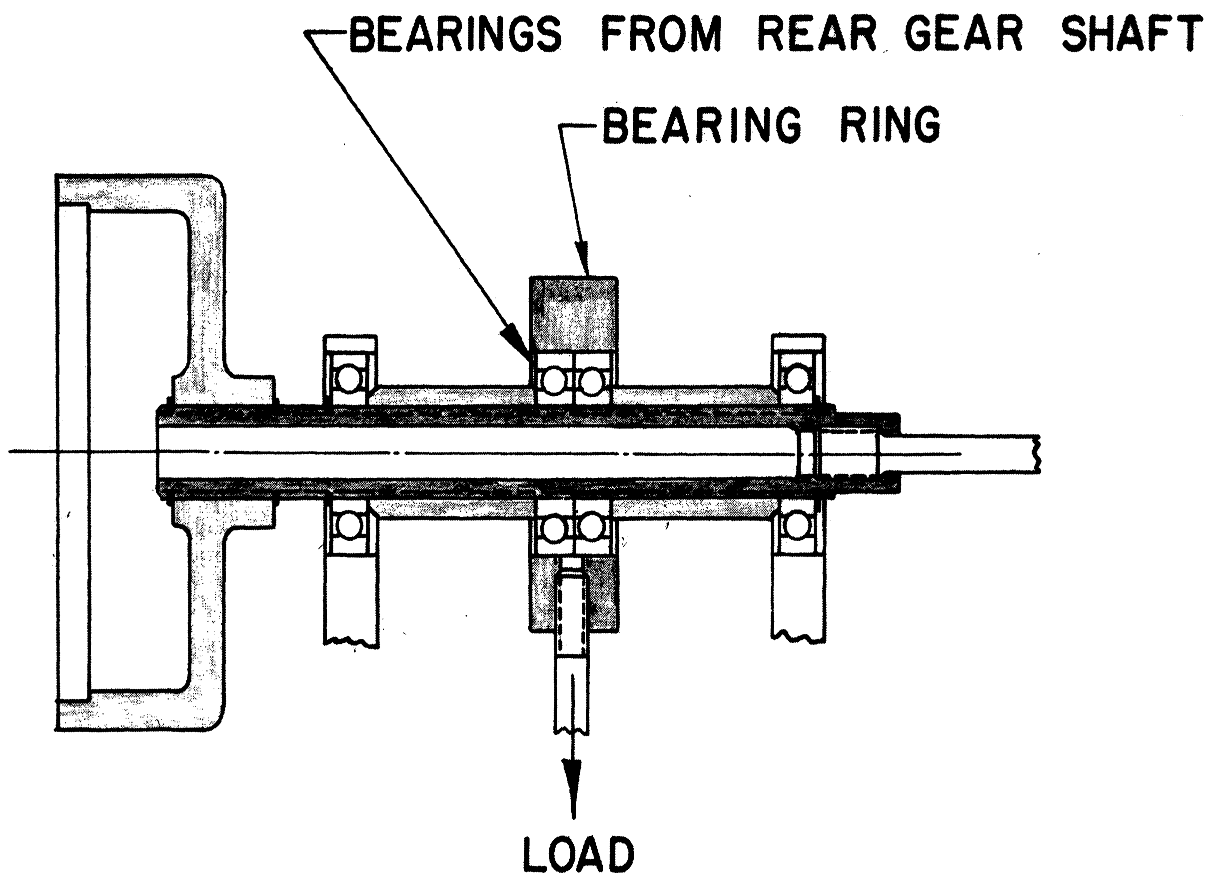

Fig. 9. Schematic layout of bearing-friction test setup.

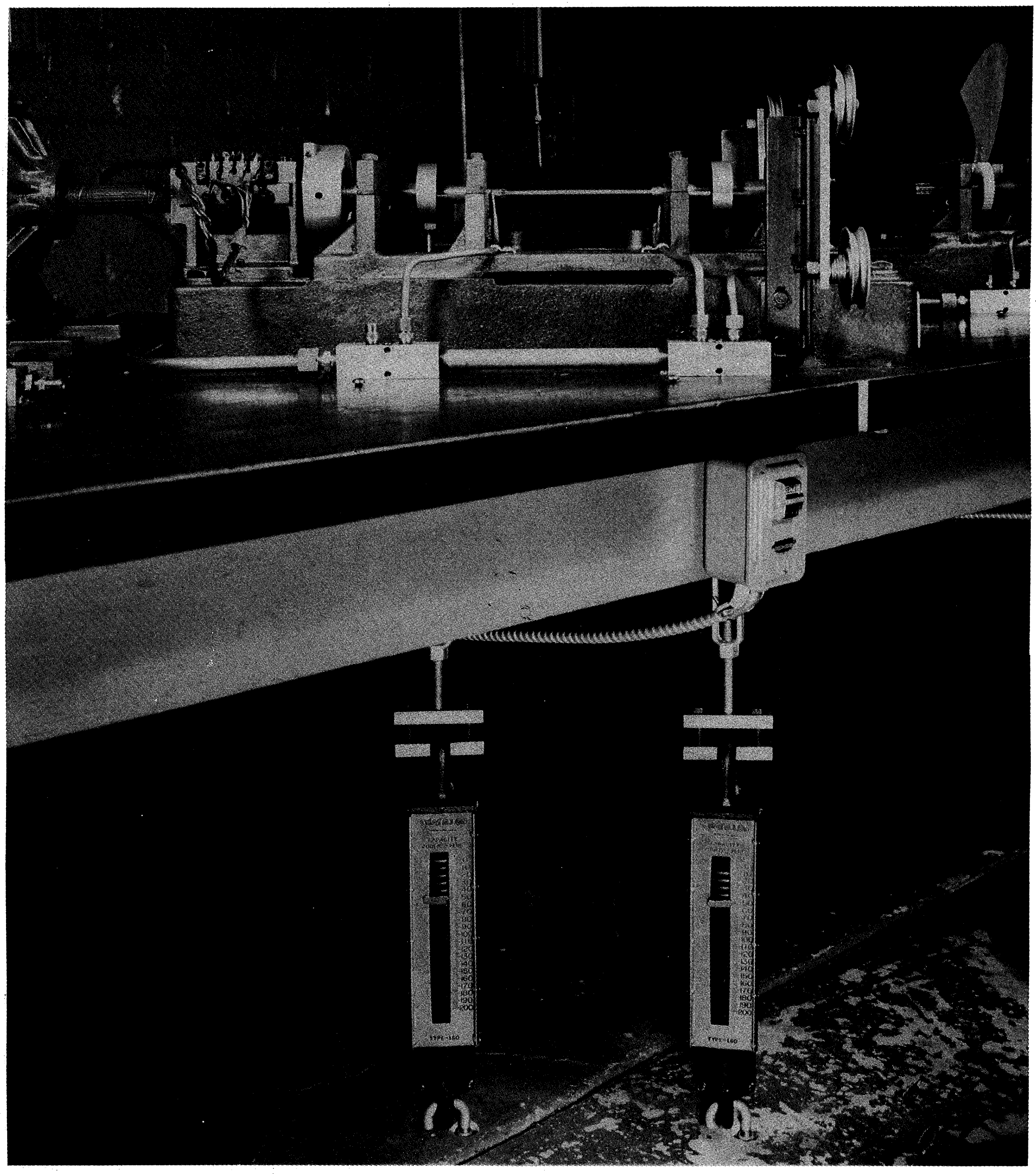

Fig. 10. Bearing-friction test set-up.

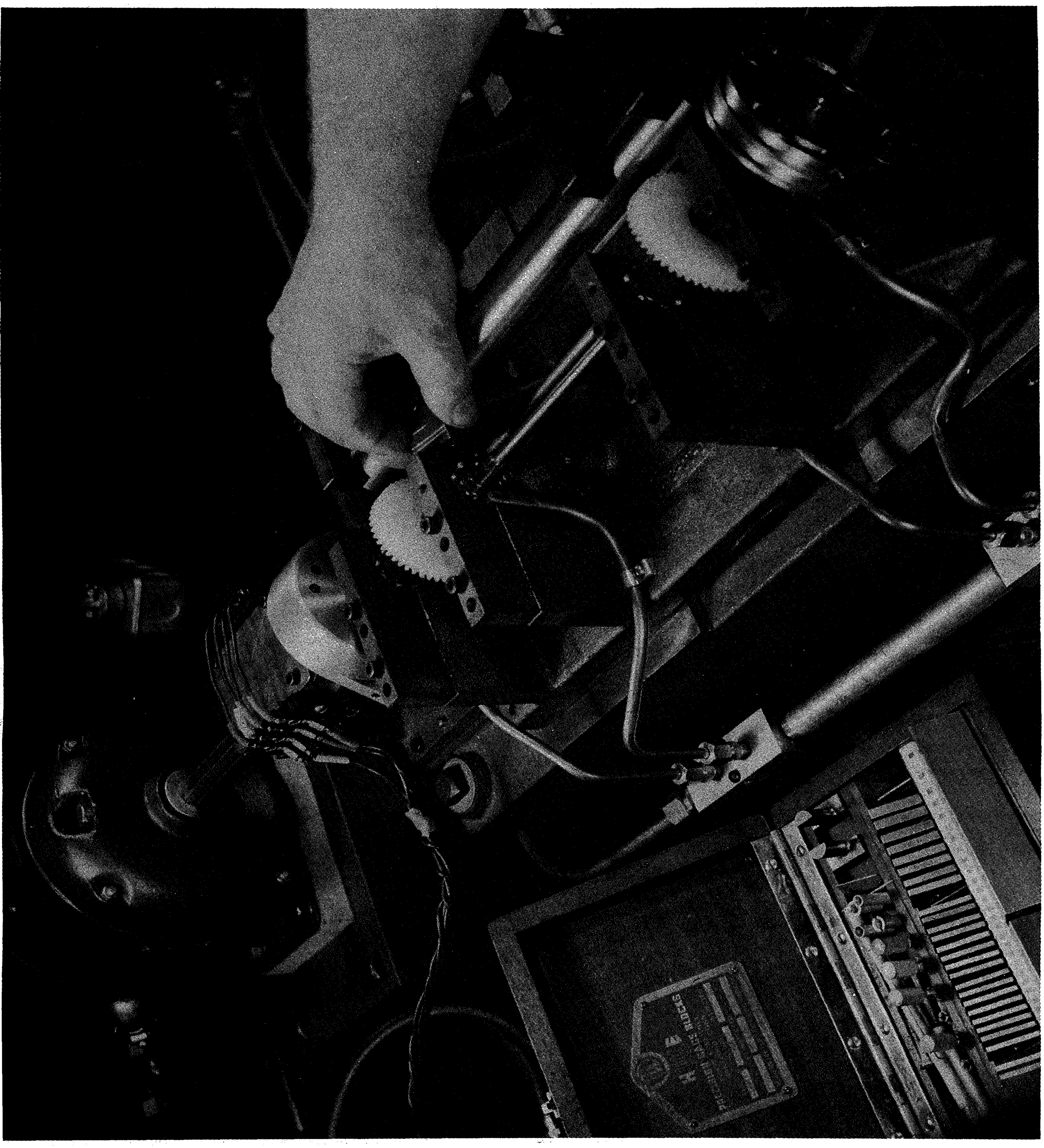

Fig. 11. Setting Center Distance.

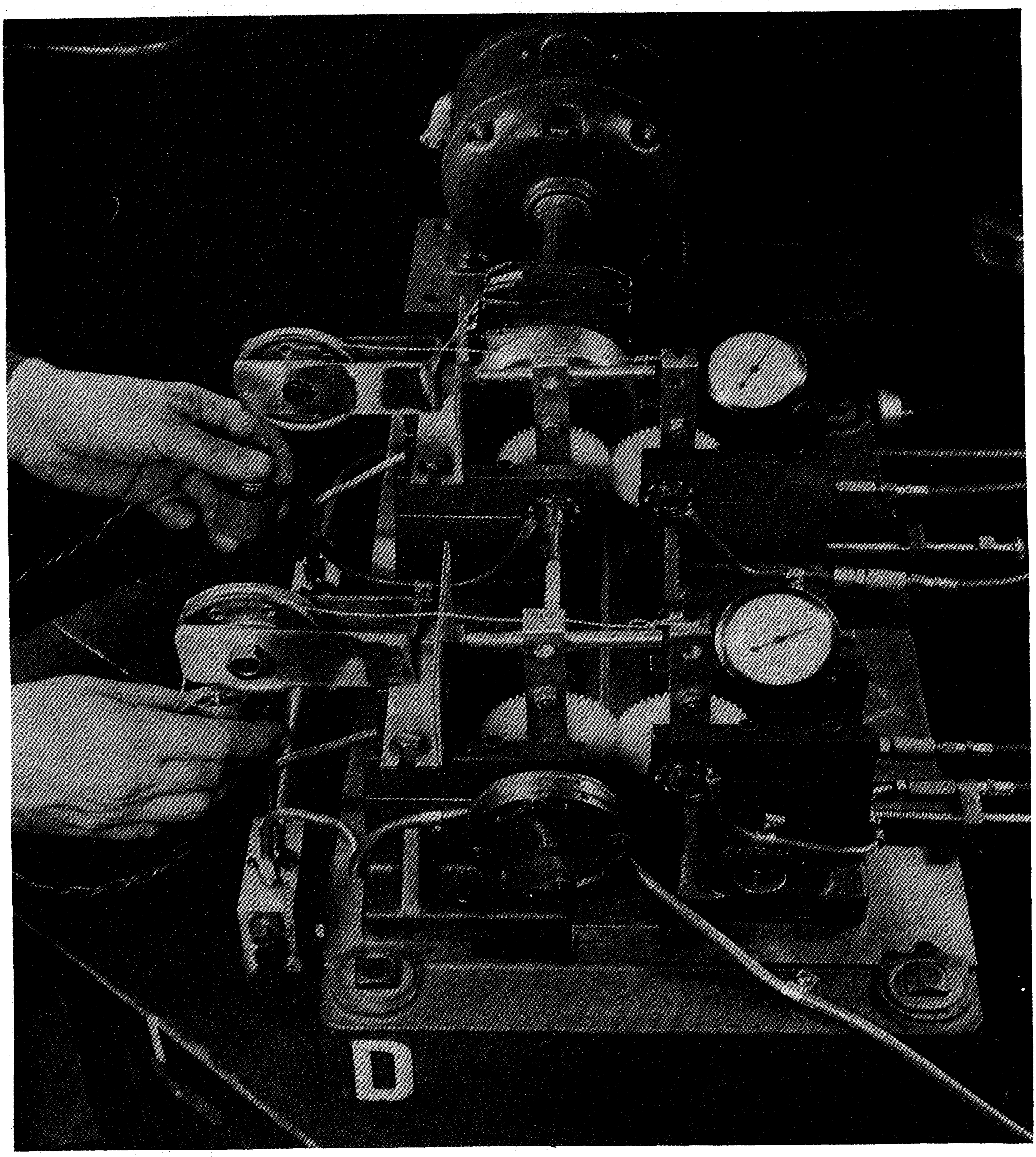

Fig. 12. Measuring backlash.

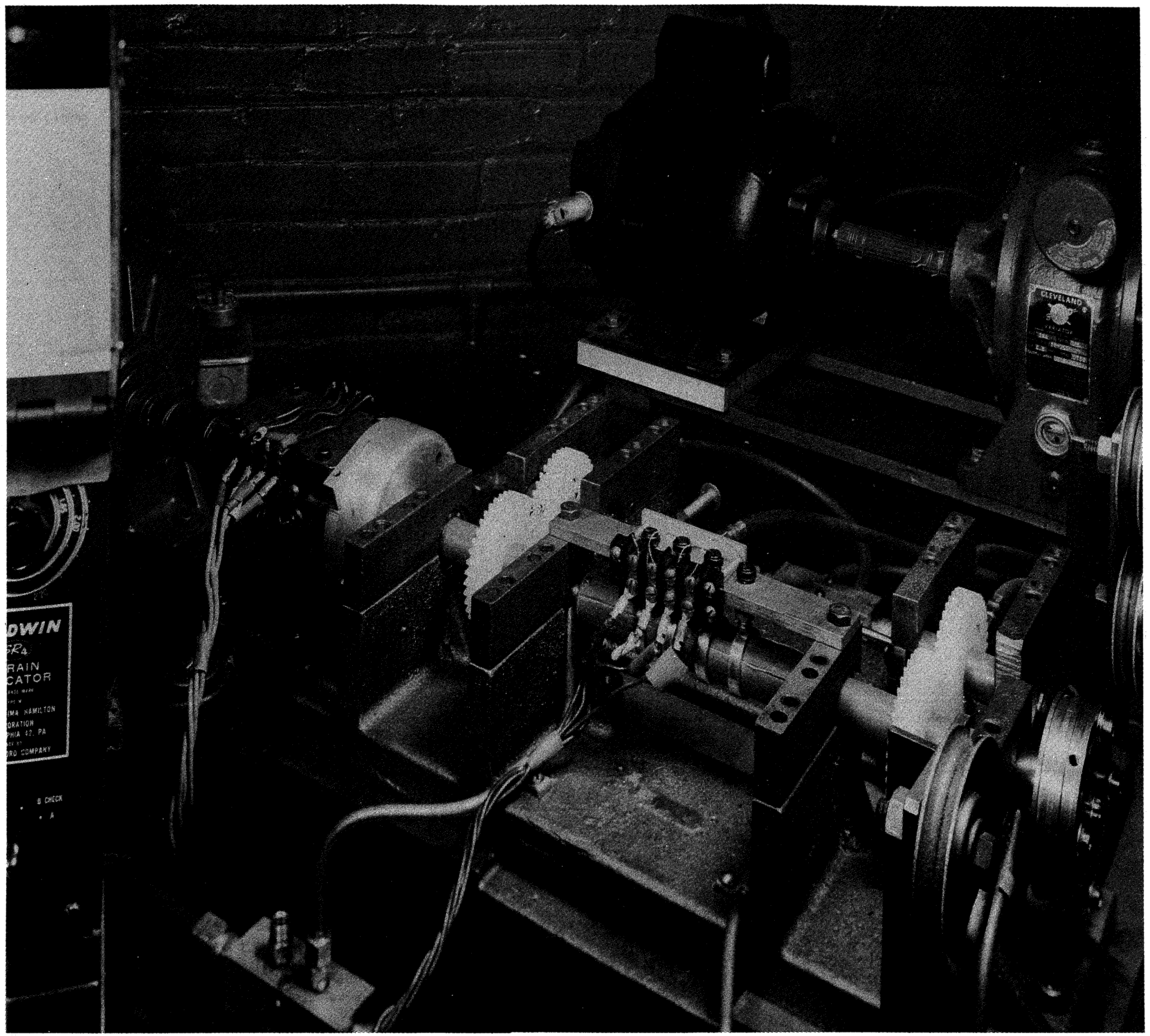

Fig. 13. Machine assembly with strain gages on short torsion bar.

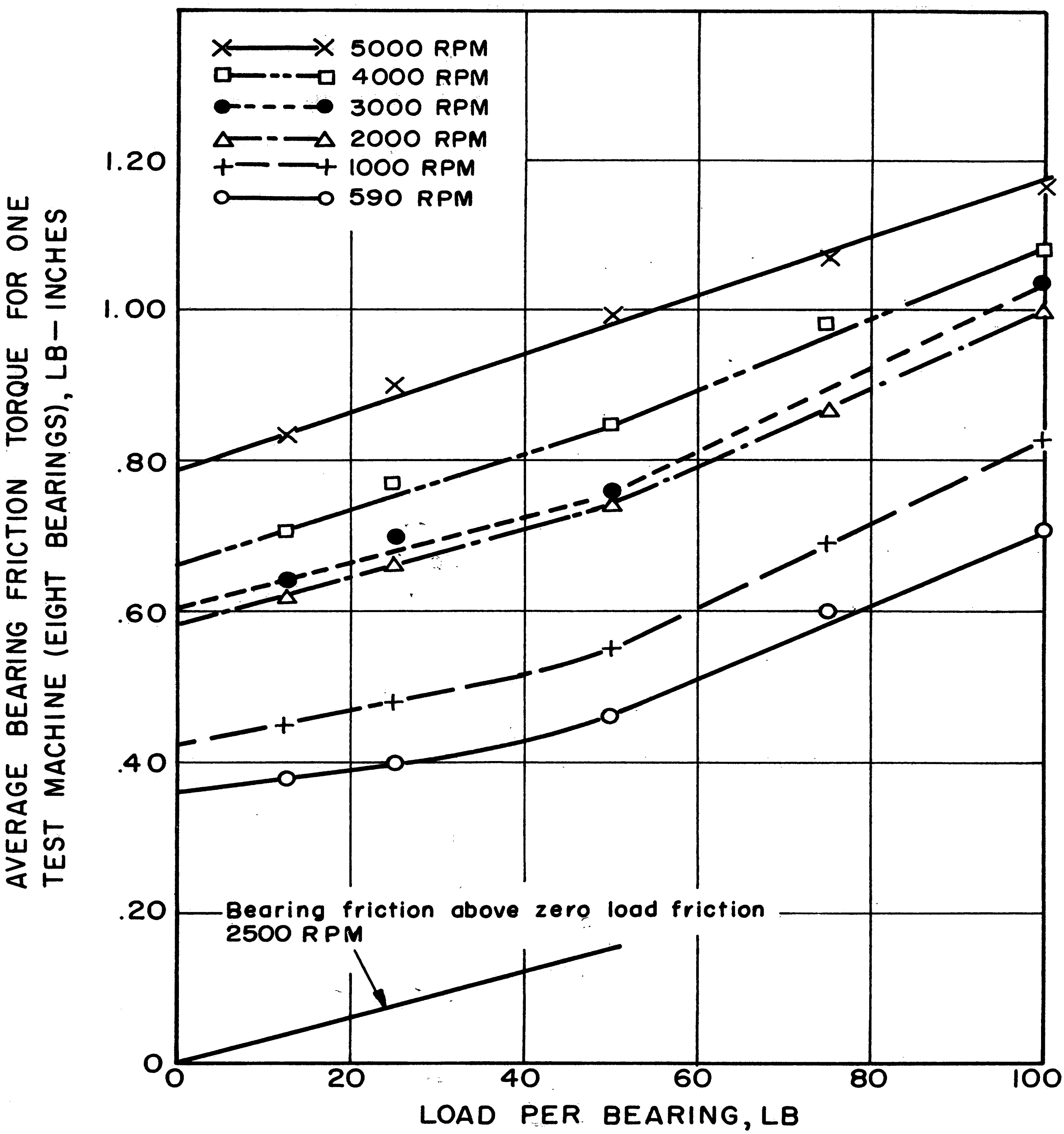

Fig. 14. Ball-bearing friction torque.