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EXPERIMENTAL STRESS ANALYSIS OF MODEL OF EMERGENCY FOREBODY RELEASE DEVICE USED IN DEEP DIVING RESEARCH SUBMARINES ALVIN, SEA CLIFF AND TURTLE

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TECHNICAL REPORT

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S. C. Daubin, Chairman
Department of Ocean Engineering

ABSTRACT

Tests were conducted on a full-scale model of the emergency forebody release used in the deep-diving submarines ALVIN, SEA CLIFF, and TURTLE. The model was machined from metal to the same dimensional tolerances as the prototype. Resistance strain gages, attached to the model, permitted measurement of forces on component parts of the device. Of primary concern was the bending stress which might be set up in the release operating shaft when the submarine is submerged in an inclined position. Tests were arranged to simulate three possible conditions of loading of the release device at a 30 degree vehicle list angle: case (1) righting moment of inclined forebody resisted by release components only; case (2) righting moment resisted by release with assistance from lower guides; and case (3) righting moment resisted by couple set up by release and rubber support ring. Test results show that shaft bending stresses (for ALVIN) are high (200,000 psi) for the case (1) condition, lower (40,000-90,000 psi) for case (2), and essentially zero for case (3). The conclusion is that the present forebody release design is adequate for all submarine attitudes encountered in normal operation, provided the vehicle has been assembled so that contact between sphere and rubber ring is assured at all times.

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I INTRODUCTION

During initial trials of the deep-diving research submarine ALVIN certain difficulties were experienced with the emergency forebody release. An effort to actuate the release during a test in shallow water resulted in galling of the contacting stainless steel parts. It was also recognized at that time that because the operating shaft projected as it did some 2-3 inches below the end of its sleeve bearing, any unbalanced side forces on the shaft could result in high shaft bending stresses. Normally, with the submarine in an upright position, the release dogs apply equal and opposite side forces to the end of the shaft. If the submarine is inclined while submerged the buoyant force of the forebody causes a righting moment which must be resisted at least in part by the emergency release device. It was thought that this resisting moment could set up a severe bending condition at the release.

In 1965 a new forebody release was designed and built with the objective of eliminating galling and reducing possible shaft bending stresses to a minimum (Ref. 1). It was felt that the shaft bending problem could be lessened in two ways: (a) by reducing the length of shaft which projects outside the hull, and (b) by increasing the diameter of the projecting portion of shaft. These two ideas were incorporated into the quarter-turn cam which is the basis of the present design. While no difficulties of any kind have arisen with the newly designed release installed in ALVIN, the need has been felt for a quantitative evaluation of the forces in the release mechanism. This feeling was intensified when, during the planning of the AUTEC submarines, the same forebody release design was adopted, but with a reduced shaft diameter.

Several attempts have been made to calculate the forces set up by unsymmetrical loading of the release. However, investigators found that they were forced to make certain assumptions of questionable validity. Because of the irregular shapes of the members, uncertainties were present regarding points of contact and force directions. A straightforward solution to the statically indeterminate problem appeared not to be possible, and those who attempted solutions agreed that results were generally unsatisfactory. Because of these difficulties the decision was made to test a full-scale model of the device.

II DESCRIPTION OF TESTS

The full scale model of the emergency sphere release was built from the drawings that were used to manufacture the actual ALVIN release. All dimensions, tolerances and surface finish requirements specified for the prototype were observed for the model. Model material was aluminum alloy 6061-T6. Pieces were cut from 3/4 inch thick plate and were brought to finished dimensions on a milling machine. Since the original release components are 1.5 inches thick, model thickness was one-half that of the actual device. However, since the device is essentially a plane "mechanism", it was considered sufficient to preserve full scale in the plane of motion. An exception to this was the shaft, which was made full scale in three dimensions.

Strain gages were mounted in three locations on the model (Fig. 16). Two gages were placed on the shaft diametrically opposite each other at section Y-Y. These gages, placed in adjacent arms of the strain indicator bridge, responded to bending strains at that section with double the sensitivity of a single gage. On dog No. 1 a portion was cut away below the contact face, and a 1/32 inch slot was milled parallel to the face and 3/16 inch away from it. This left a cantilever beam whose outer side was the dog contact surface. One strain gage was placed on this cantilever as close as practicable to the fixed end, to provide response to forces applied to the dog by the cam (see also Fig. 1). A single strain gage was placed on dog No. 2 at about mid-length on the inner face. In early tests this gage was found to have very low output and it was later discarded. The gages on the shaft and on dog No. 1 provided all necessary data for the complete force solution.

Calibration was done by separately mounting each strain-gaged component firmly to the laboratory bench and applying known loads by means of dead weights, either directly or through a lever arm on a knife-edge support. The loads were applied at the correct contact points after these points had been located in the assembled model. The system used for calibrating the gages on the shaft is shown in Figure 3. Curves were plotted of strain indicator difference readings against applied load. These curves were later used to determine internal forces in the assembled model, which resulted from the external test load.

Test loads were applied in such a way as to simulate loading of the actual device when the submarine is submerged in an inclined position. An inclination angle of 30 degrees was used since this was considered to be the greatest angle likely to be encountered in normal operation. Tests were arranged to provide three possible conditions of loading of the release: case (1) righting moment of inclined forebody resisted by release components only; case (2) righting moment resisted by release with assistance from lower guides; and case (3) righting moment resisted by couple set up by release and rubber support ring.

In all tests the model was supported in a position similar to that which it would occupy in the submarine, by means of a back-up plate bolted to the laboratory floor. A vertical arm was attached to the model and at the upper end of this arm the test loads were applied at the desired angle. The length of the arm was approximately equal to the height of the center of buoyancy of the forebody. Actually the gravity and buoyancy forces act at two distinct points although in this case these points are quite close to each other. For simplification in the tests a single point was selected at which a force representing the net difference of gravity and buoyancy forces would be applied. This point was located so that the force and moment reactions at the sphere release would bear the same relationship to each other in the model as they do in the actual submarine. Figures 4 and 5 show photographs of this arm as used in cases (1) and (2). For the case (2) loading a series of tests were done for different values of clearance between the lower guides and the release jaws. Clearance values, set by means of shims before each test, were 1/8 inch, 1/16 inch, .030 inch and .015 inch. For the case (3) tests additional wooden structure was required to represent that portion of the sphere which contacts the rubber support ring. This is shown in Figure 6. Case (3) testing was done in two ways: (a) where the rubber support ring is represented by a pivot or knife edge (Fig. 7), and (b) where a roller or simple support is used for this purpose (Fig. 8).

Since the model test loads were much smaller in magnitude than the loads encountered by the prototype device, it was necessary to balance out the weight of the model and associated structural parts. This was done by a system of counterweights. Cables from the weights were attached as nearly as possible to centers of gravity of the parts. By this means counter-balancing was achieved but no unwanted rotational moments were introduced.

Test loads were in the form of calibrated dead weights which were applied in one-half, one, or five pound increments, depending on the test. Readings were taken as the weights were added, and again as they were removed. In preliminary tests it was determined that strain gage readings were linear over a range of loads but began to fall off as this range was exceeded. This was probably caused by internal forces being great enough to distort slightly the original geometry of the linkage. Subsequent testing was done within the linear range of strain gage outputs.

III RESULTS

The numerical results obtained from the tests are shown in Table 1. Forces at the contact points and at the dog pivot pins are presented in dimensionless form as functions of the applied test load P (where P represents the net lift of the forebody D-W). Similarly, the bending moments and bending stresses at the section X-X of the release shaft are shown in terms of P. The actual bending stresses in the case of ALVIN are given in the final column. The stress calculations for ALVIN are based on data taken from the most recent weight-stability computations for that submarine (Ref. 2).

In computing the forces in the release device, readings of the strain gage on dog No. 1 permitted the determination of force F_B directly. A summation of moments on the shaft, using force F_B and readings of the gages on the shaft, provided a solution for force F_A. Finally, moment summations were done for the dogs separately, which yielded values for the remaining forces. With forces F_A and F_B known, it was possible to evaluate the bending moment and bending stress at section X-X of the shaft, where bending would naturally occur in the real device.

The directions of the forces at the dog pivot pins are the approximate ones necessary to satisfy static equilibrium for the system.

Contact points were located by direct examination. It was found that by placing a light source behind the model the exact locations of contact could be seen and photographed. An example of this technique is shown in Figure 2 .

Corresponding forces and stresses for SEA CLIFF and TURTLE would be higher for two reasons. The net forebody lift of these vehicles is approximately 4000 pounds against 1844 pounds for ALVIN. In addition, the release shaft diameter for the new submarines is 0.810 inch compared to 0.9375 inch for ALVIN, which would cause shaft bending stresses in SEA CLIFF and TURTLE to be approximately 55 percent greater than those for ALVIN.

Table 1 Test Results (Refer to Figs. 9,10,11,12)

Type of Loading	F_A/P	F_B/P	F_C/P	F_D/P	F_E/P	F_G/P	M_{xx}/P	For ALVIN σ_{xx}/P	σ_{xx}
Case(1) No Lower Guides	10.8	11.1	12.2	21.8	8.8	10	9.97	123	227,000
Case(2) With Lower Guides									
(a) Clearance 1/8 in.	1.55	3.94	3.69	5.19	2.14	3.70	3.88	48	88,400
(b) Clearance 1/16 in.	2.14	3.33	5.07	4.38	2.93	3.10	2.70	33.6	62,000
(c) Clearance .030 in.	2.41	3.27	5.73	4.33	3.32	3.03	2.47	30.4	56,000
(d) Clearance .015 in.	2.36	2.80	5.63	3.70	3.27	2.63	1.90	23.7	43,600
Case(3)									
(a) Pivot Support	0.7	0.7	1.20	1.00	0.5	0.7	0	0	0
(b) Roller Support	0.7	0.7	1.10	1.00	0.75	0.75	0	0	0

IV DISCUSSION

It can be seen that the loading of case(1) would produce excessively high bending stresses in the release shaft. However, it seems unlikely that this loading would be encountered in practise, because of the presence of the lower guides. These guides are permanently attached to the frame members of the submarine afterbody and (in the case of ALVIN) have been installed with a specified clearance of 0.030 inch. Thus, the loading of case(2) is probably the most severe that could be experienced if the pressure sphere should float free of the rubber support ring. The case(3) loading is that which is to be expected in normal operation of a correctly assembled vehicle of the ALVIN type. During assembly the air weight of the forebody will compress the rubber of the support ring slightly. In the water the buoyant forebody will tend to rise and relax the compressed rubber to an extent which will depend upon the vertical clearance in the release device and possible deflection of frame members to which the lower release components are attached. If the total possible vertical motion can be kept less than the initial compression of the rubber, the forebody will be held firmly against the rubber ring at all times.

In cases (1) and (2) motion of the forebody is, generally speaking, one of rotation about a center located at the release device. In case (1) the release alone must resist this rotation and forces on the release components can reach excessively high values. In case (2) a contact point is established at the lower guide on one side (Fig. 10). This tends to prevent further rotation and effectively reduces the bending tendency at the shaft. In case (3) the rubber support ring prevents forebody rotation about the release and, being a fixed point, acts as a pivot about which any possible rotation must occur. The reaction at the release is now simply a force, the direction of which depends upon the nature of the constraint at the rubber ring. Since the rubber ring is not attached to the sphere a simple support is suggested, but the presence of friction could cause behavior to approach that of a pivot. In the tests, both simple support and pivot were used in an effort to bracket the real situation. Tests showed that the two conditions produced results which did not differ markedly. In case (3a) (pivot support) the reaction R at the release was 2190 pounds and it made an angle α with the vertical of 34 degrees (Fig. 14). In case (3b) (simple support) the force R was 2480 pounds and the angle α was 5.5 degrees (Fig. 15). Since shaft bending was not present in either case, it appears that there is a range of values of the angle α for which the same symmetry of forces on the cam exists.

Time did not permit model testing at forebody list angles other than 30 degrees but some brief calculations were performed to determine the magnitude and direction of the reaction R at various other values of forebody inclination. The results of these graphical solutions are presented in Table 2 along with the values for 30 degrees. The values shown in Table 2 indicate that as far as case (3) loading (sphere in contact with rubber ring) is concerned, large angles of roll or list have relatively small effect on the reactive force at the sphere release. The maximum value of R is seen to be 2570 pounds at a list angle of 48 degrees; this is only 4 percent greater than the value of R at 30 degrees, the angle at which the testing was done.

The shaft bending stresses presented herein for ALVIN are based on linear extrapolation of the model test results. It is assumed that deformations are negligible and that original geometry is preserved. The relatively high loads in the actual device are probably sufficient to cause some deformation of release components resulting in a redistribution of internal forces. Since preliminary model tests indicated a falling off of internal forces as a result of redistribution this also could be true in the prototype. If so, predicted stresses are probably conservative.

It should be noted that the stresses reported in Table 1 are bending stresses only. The forces on the cam (in cases 1 and 2) are such that direct shear stresses are also present in the shaft at section X-X. A complete solution would include determination of these shear stresses, and, with the aid of a Mohr circle diagram, the determination of maximum stresses at the point in question. However, sample calculations of this type showed that the direct shear stresses had values which were only 3 to 7 percent of the bending stress values, and the resulting maximum stresses would be not over one percent greater than the bending stresses. Torsional shearing stresses would be present in the shaft only if the shaft were rotated for the purpose of actuating the release. The pressure of the dogs against the cam would set up friction forces which would resist rotation, and the torque required to actuate the release from within the sphere would be a function of these friction forces. Rough calculations indicate that the torque required to effect release would be approximately

100 foot pounds for the normal level submarine (ALVIN) and approximately 200 foot pounds for ALVIN inclined at 30 degrees (case 3 condition). The torsional shear stress in the ALVIN release shaft corresponding to a torque of 200 foot-pounds would be 14,900 psi, well below the torsional yield strength value of 62,000 psi for the Monel K-500 material of which the shaft is made.

The foregoing torque values are approximate only, being based on an estimated value for the coefficient of friction of 0.6. A detailed study of friction effects in the release device, relating both to the torque required to actuate, and the force reactions on the other parts of the device, has not been done. It is recognized that the effects of friction in the test model may be different from those in the actual device. The materials used and the presence of sea water environment in the case of the real mechanism make it practically impossible to model these effects. The intention in these tests therefore was to determine the essential force system due to the geometry of the release. It may be possible to collect reliable friction data for the metals in question and modify the test results accordingly, but this has not been included in the work covered by this report.

V REFERENCES

1. "Design and Manufacture of Emergency Sphere Release for ALVIN" by James W. Mavor and Arnold G. Sharp, W. H. O. I. Technical Report No. DS-20, August, 1965

2. "Weight, Buoyancy and Stability of DSRV ALVIN, 1967", by Arnold G. Sharp and Clifford L. Winget, W. H. O. I. Technical Report DS-26, January, 1968.

VI APPENDIX A - FIGURES

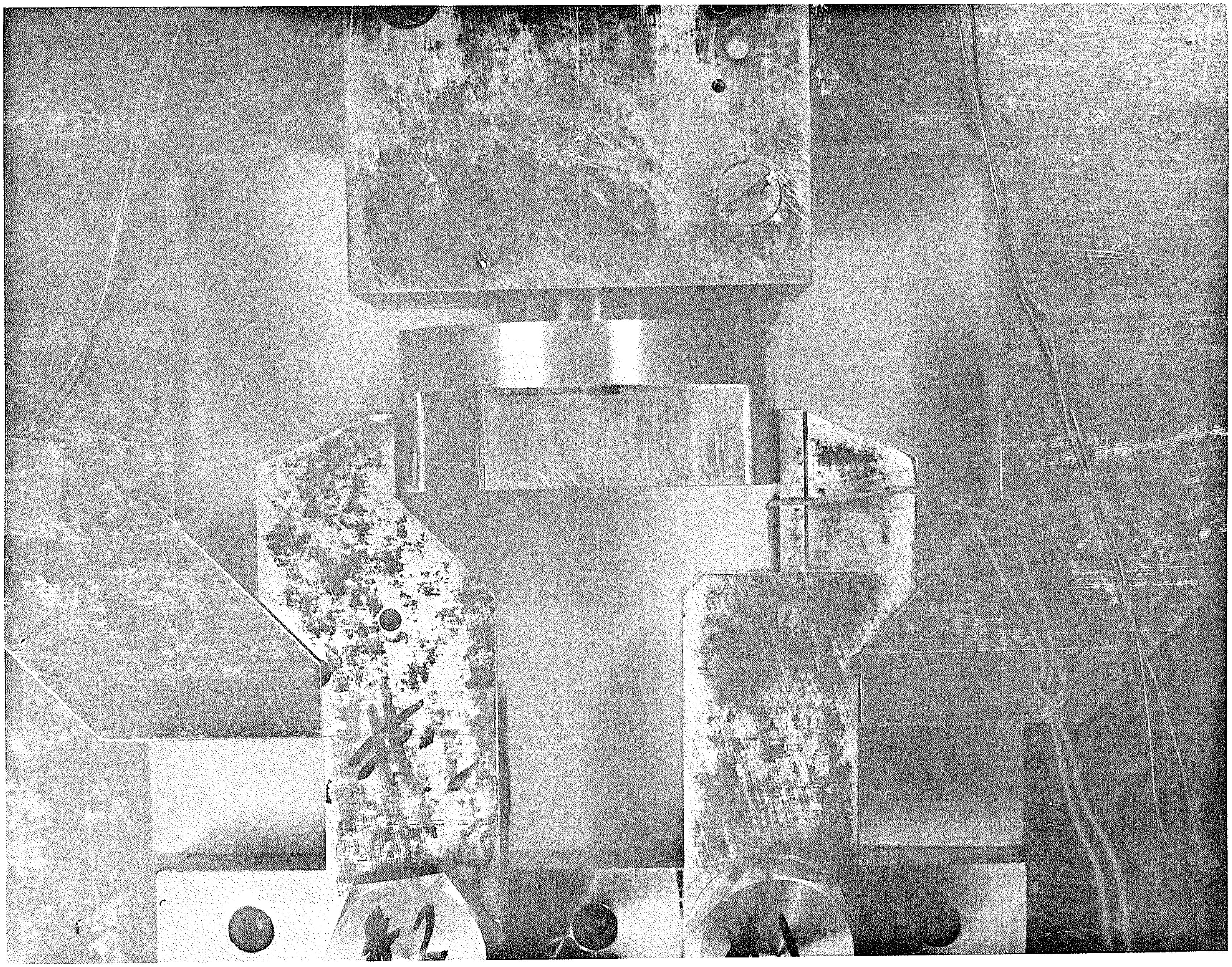


Fig. 1 Full scale model of ALVIN emergency forebody release

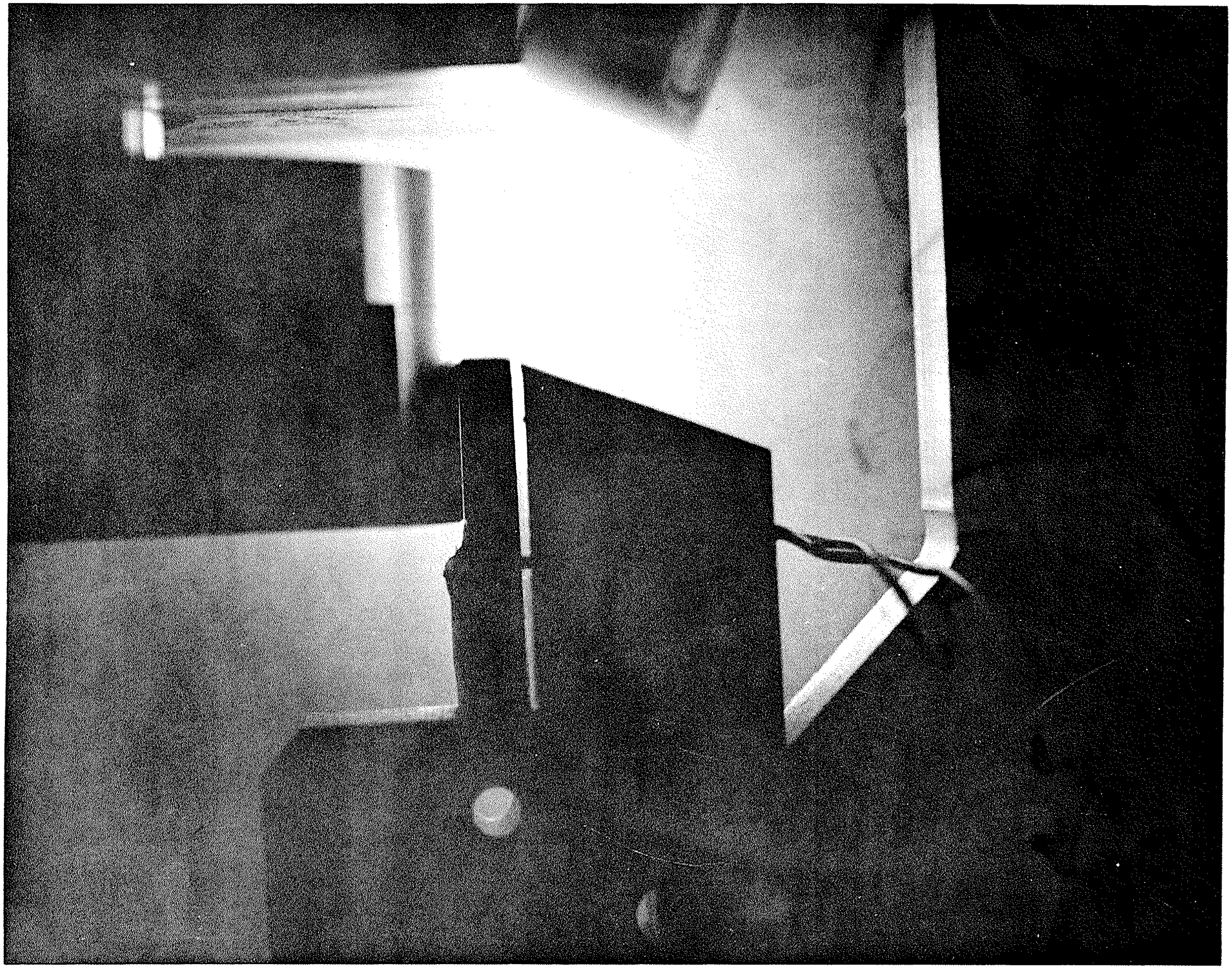


Fig. 2 Photograph showing how contact point locations were determined.

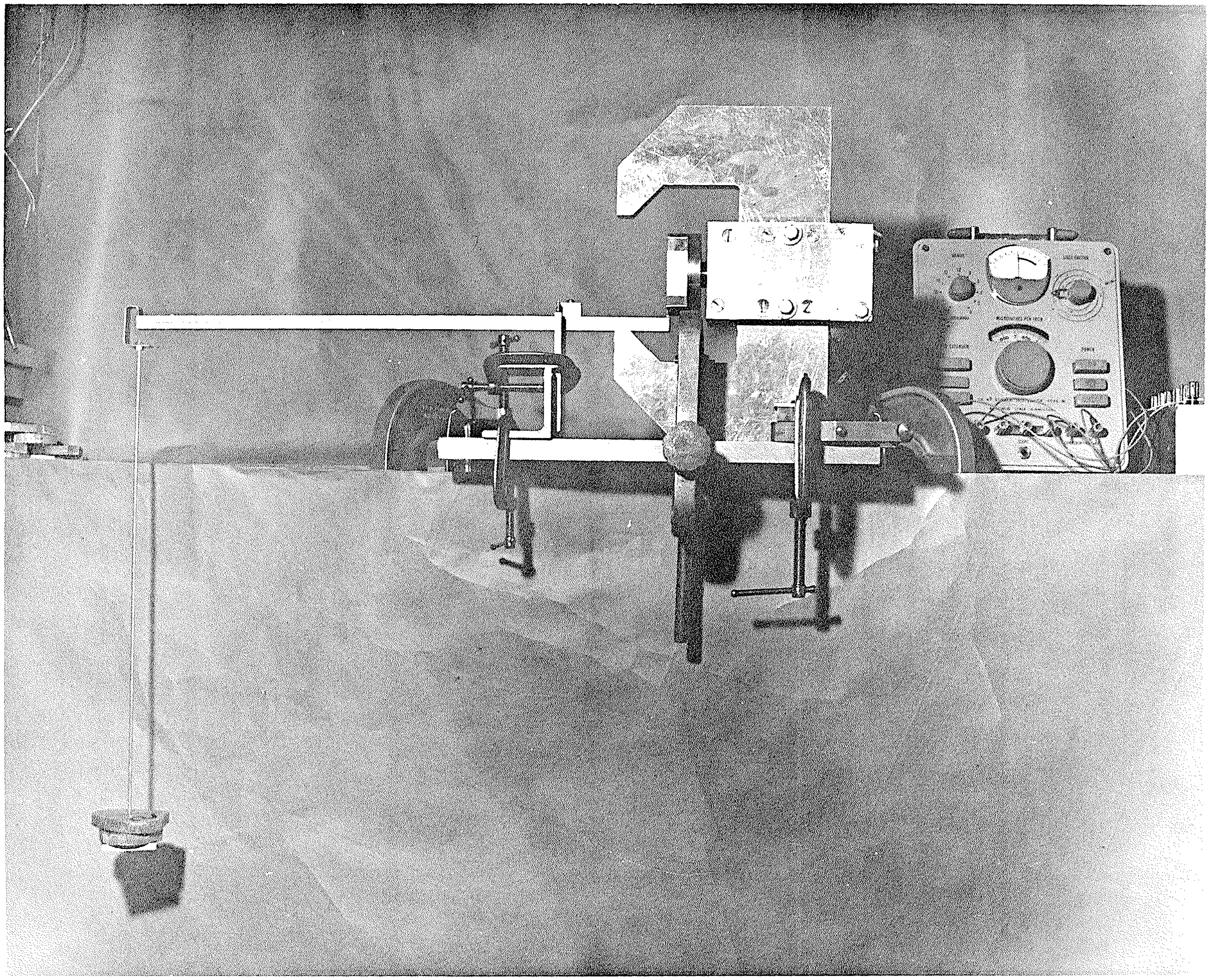


Fig. 3 Calibration of strain gages on release shaft.

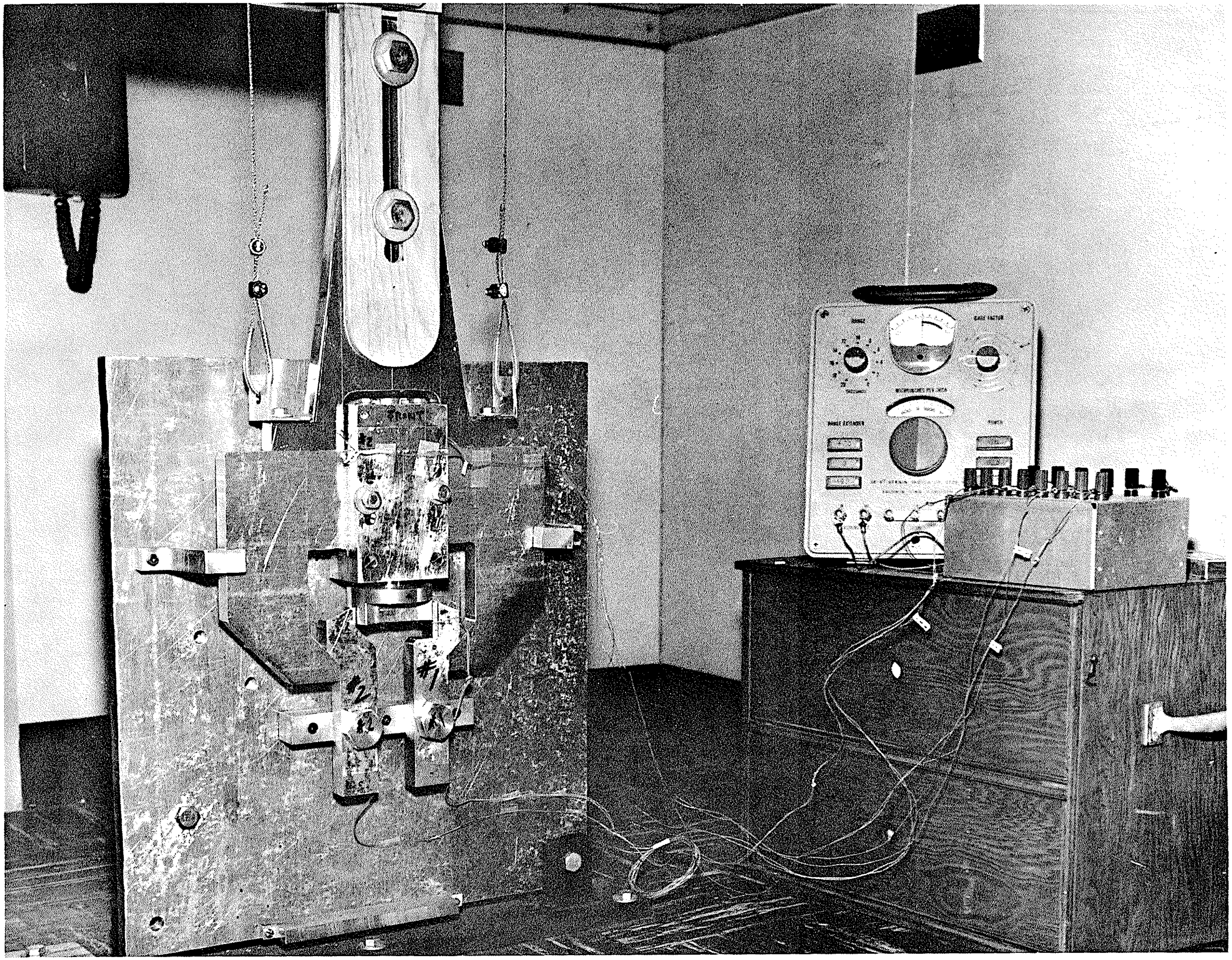


Fig. 4 Test apparatus for case 1 loading (without lower guides)

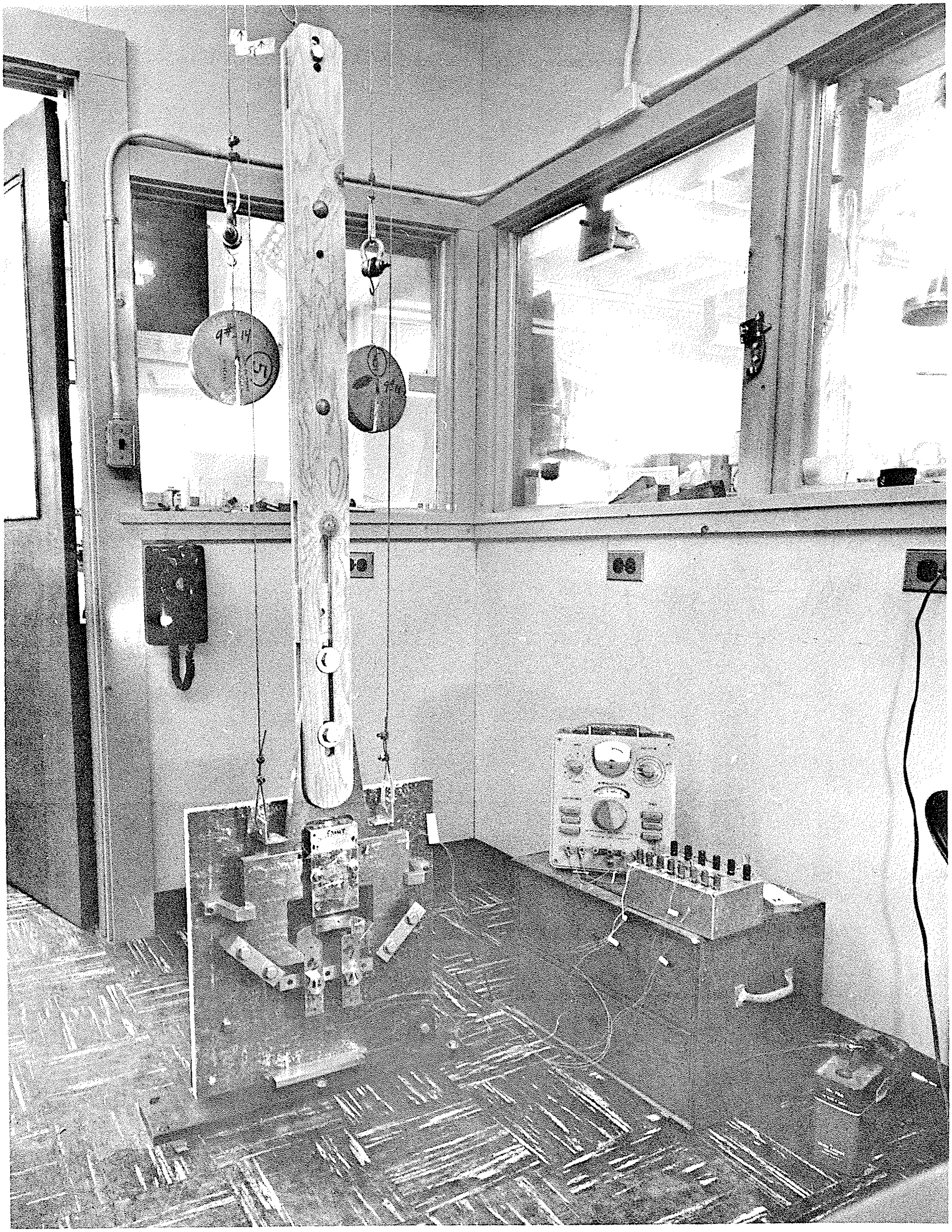


Fig. 5 Test apparatus for case 2 loading (lower guides in place)

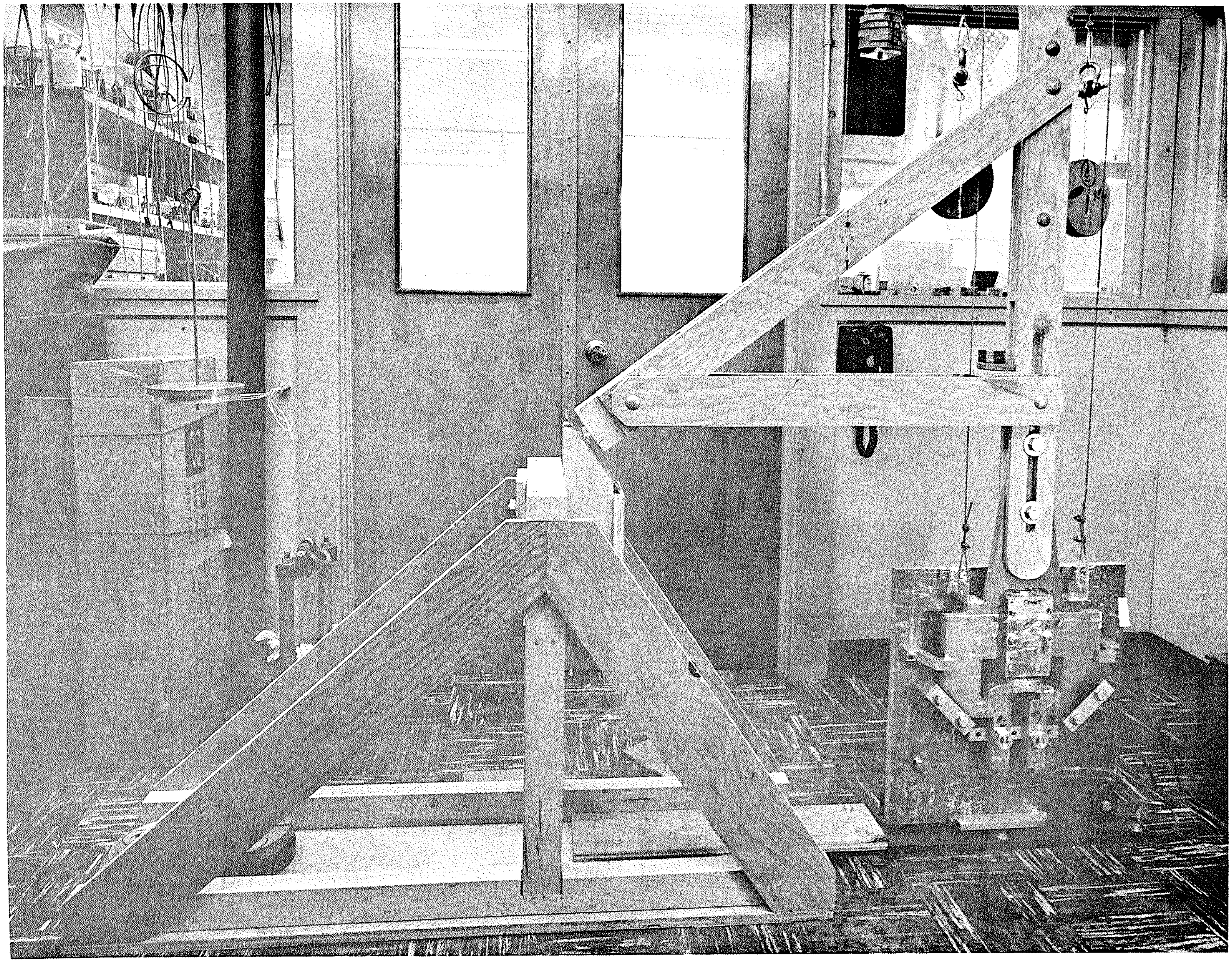


Fig. 6 Test apparatus for case 3 loading

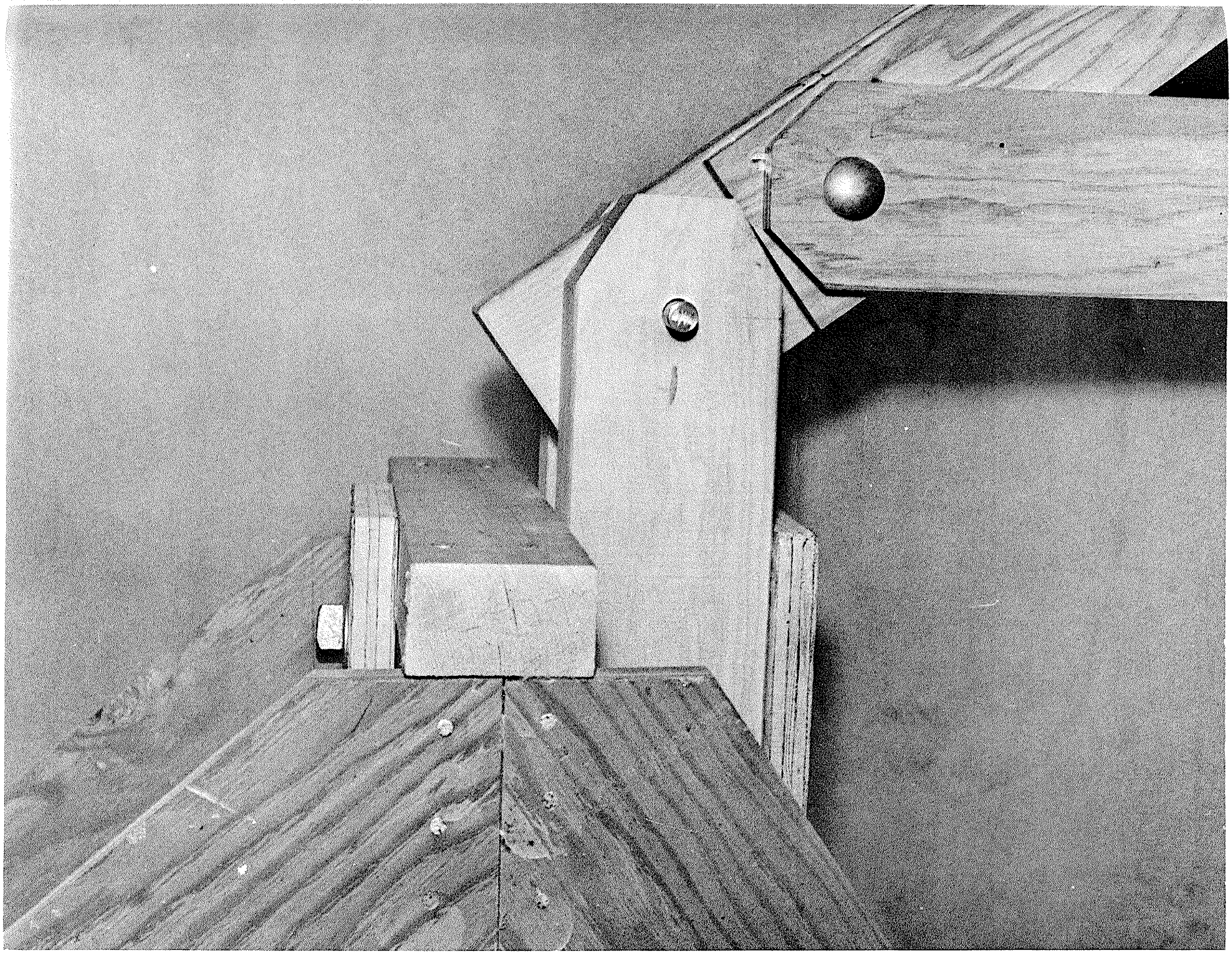


Fig. 7 Detail of case 3a loading (rubber support ring represented by pivot)

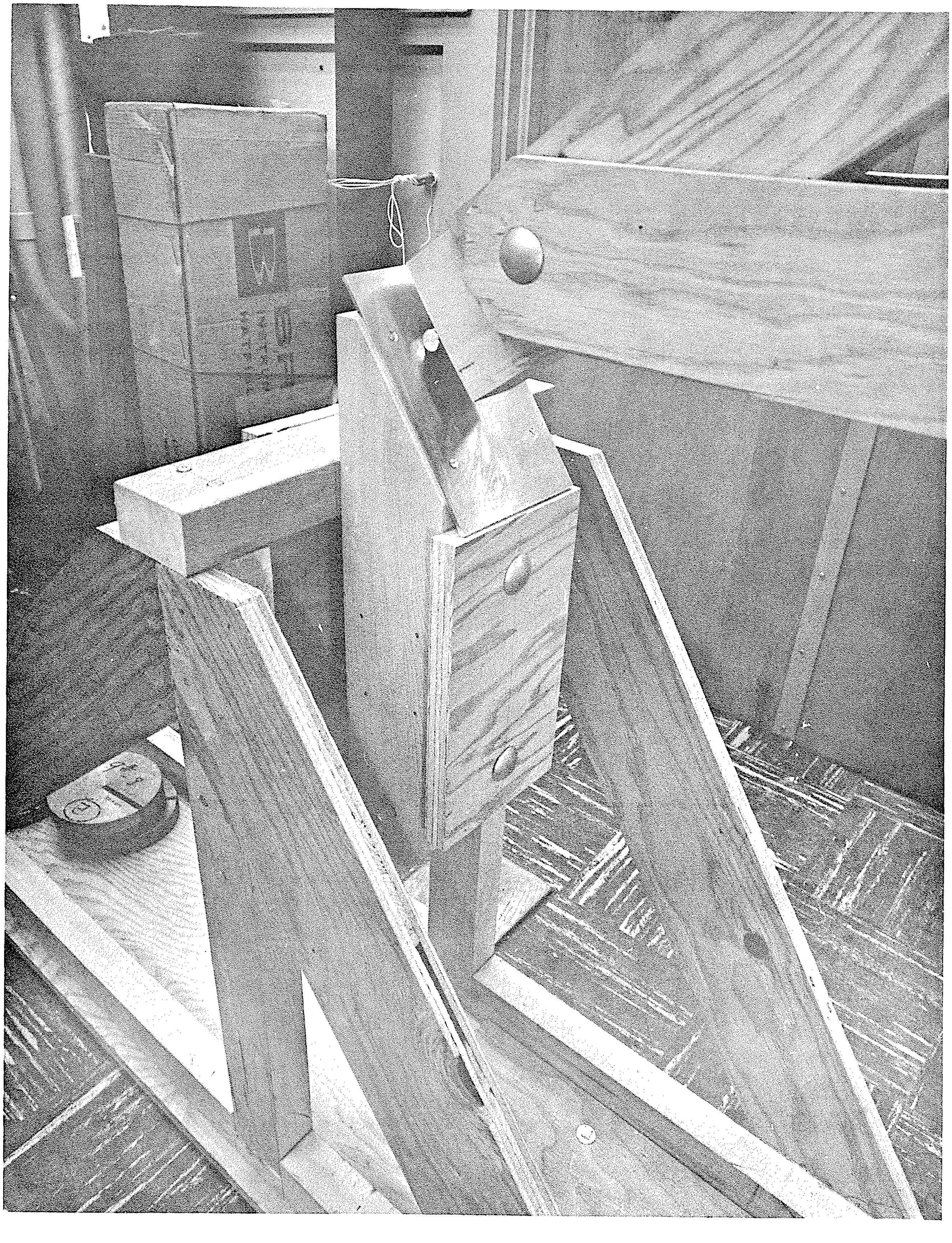


Fig. 8 Detail of case 3b loading (rubber support ring represented by roller support)

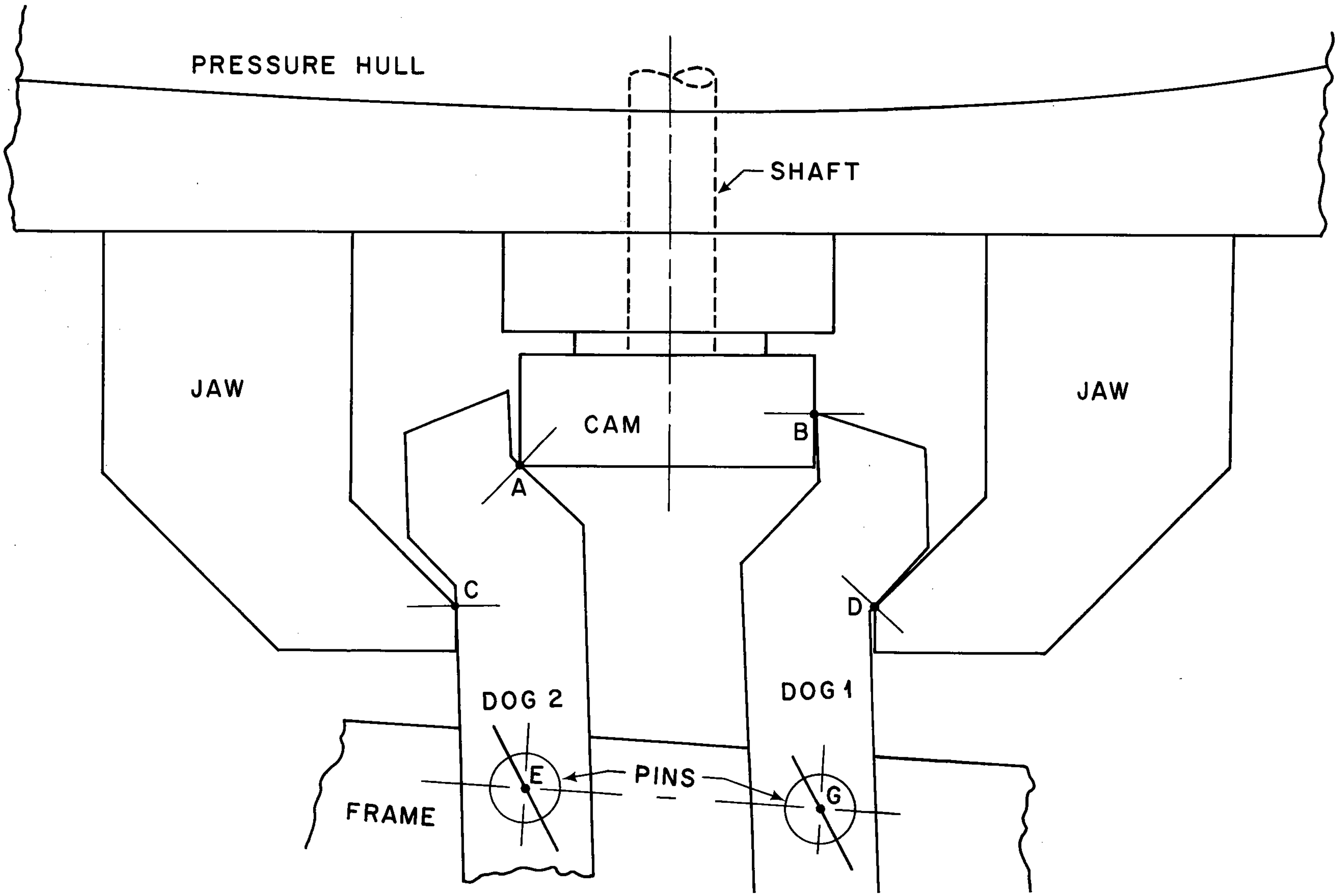


Fig. 9 Bearing Points and Force Directions for Case 1 Loading.

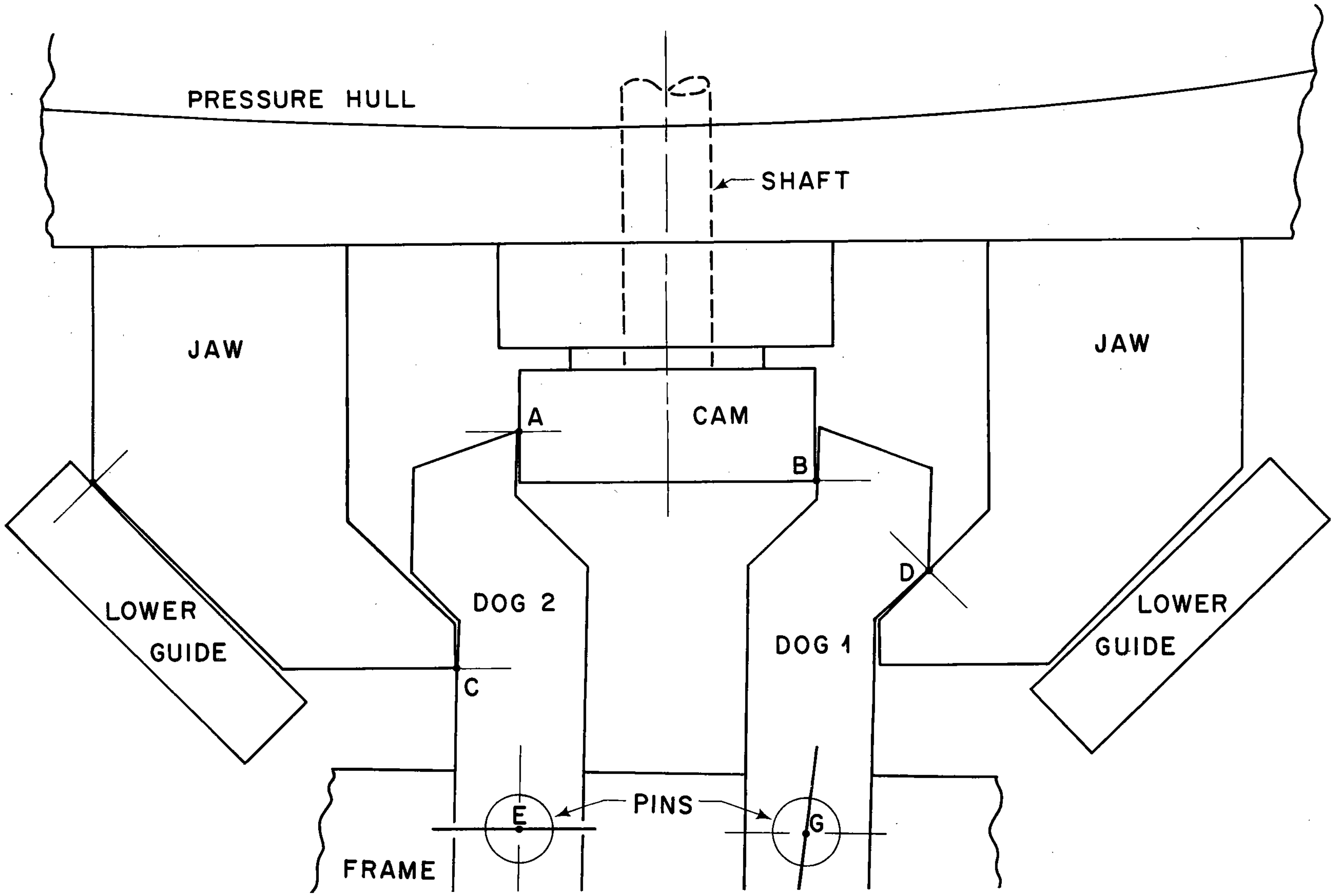


Fig. 10 Bearing Points and Force Directions for Case 2 Loading.

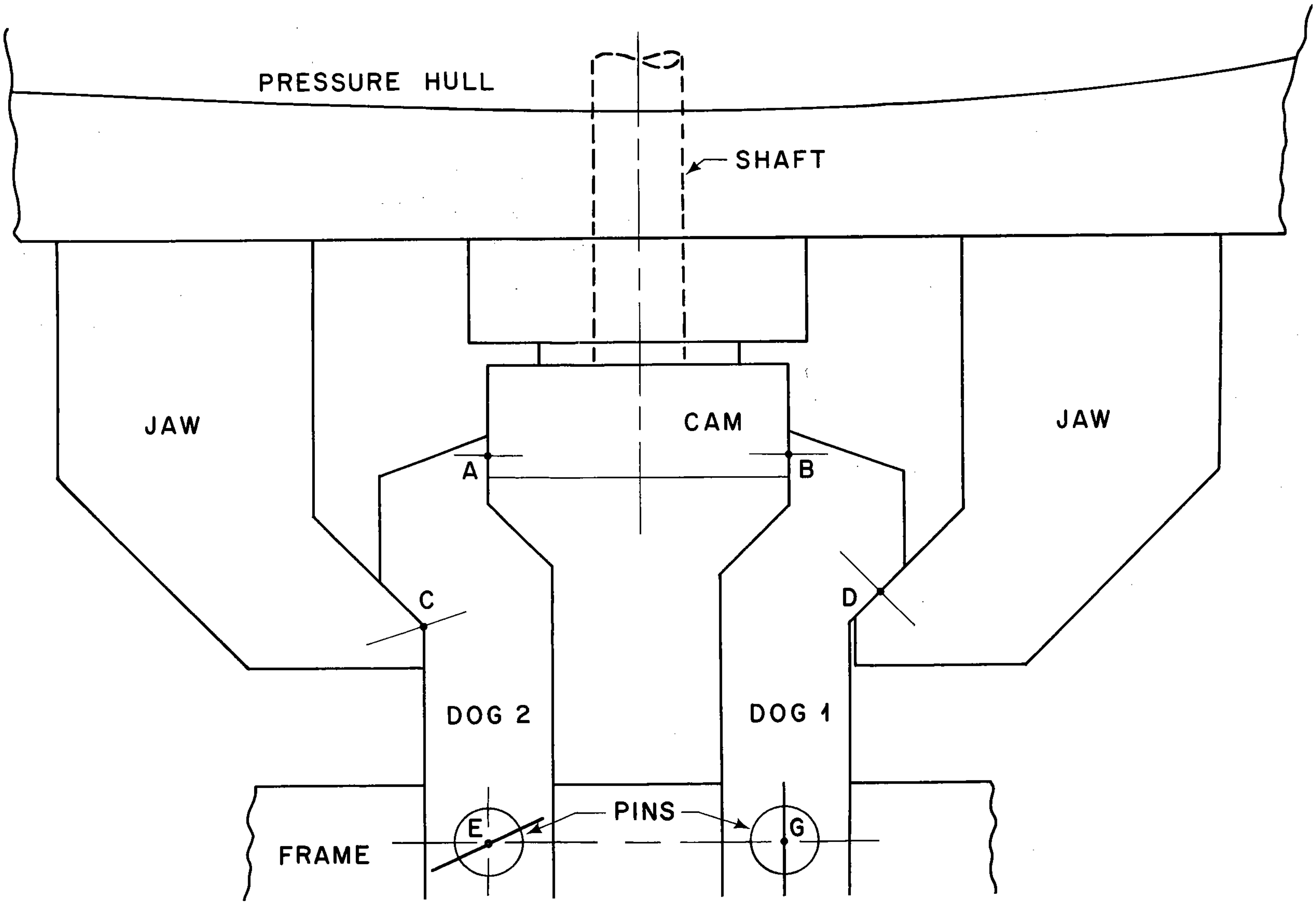


Fig. 11 Bearing Points and Force Directions for Case 3a Loading.

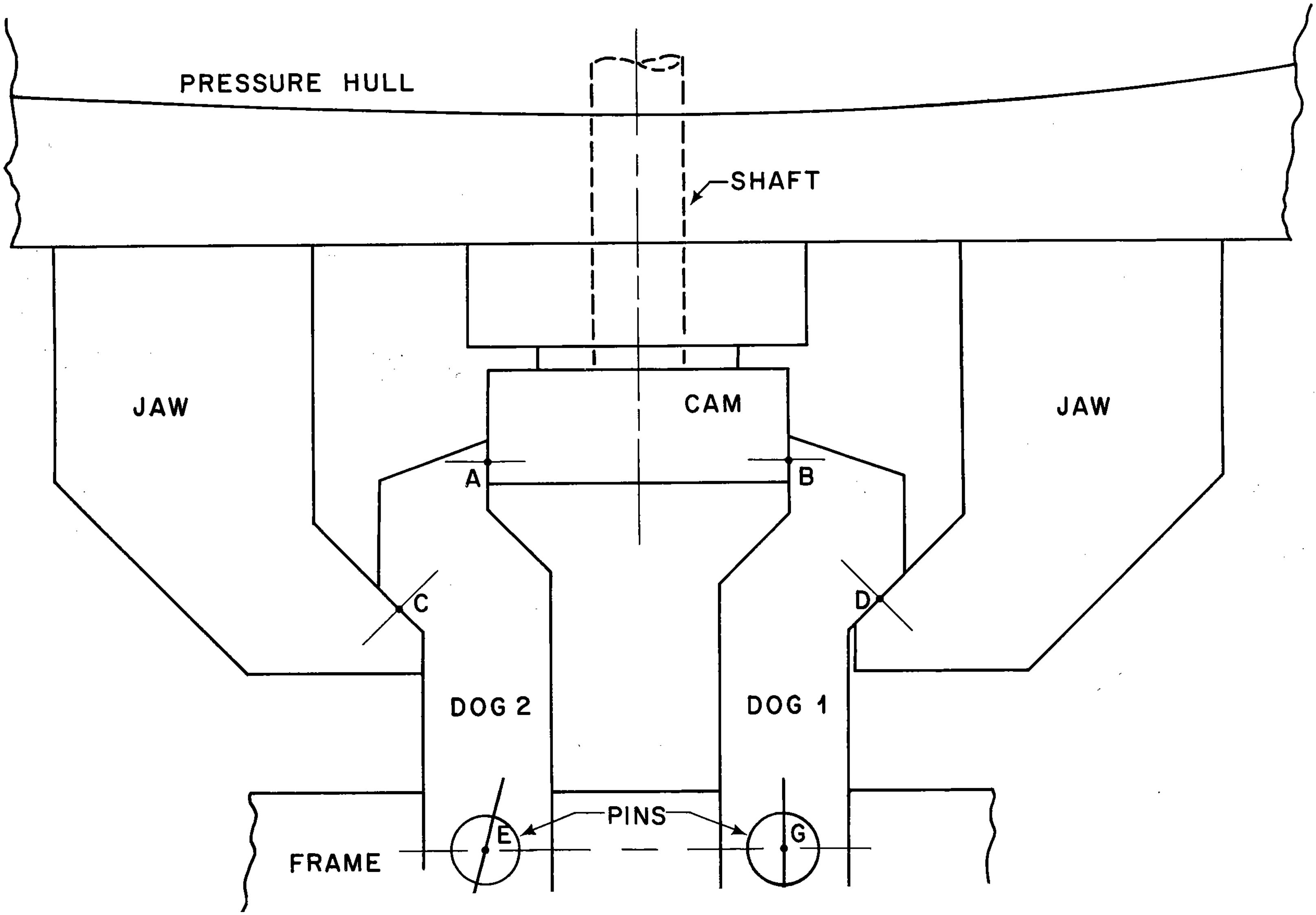


Fig. 12 Bearing Points and Force Directions for Case 3b Loading.

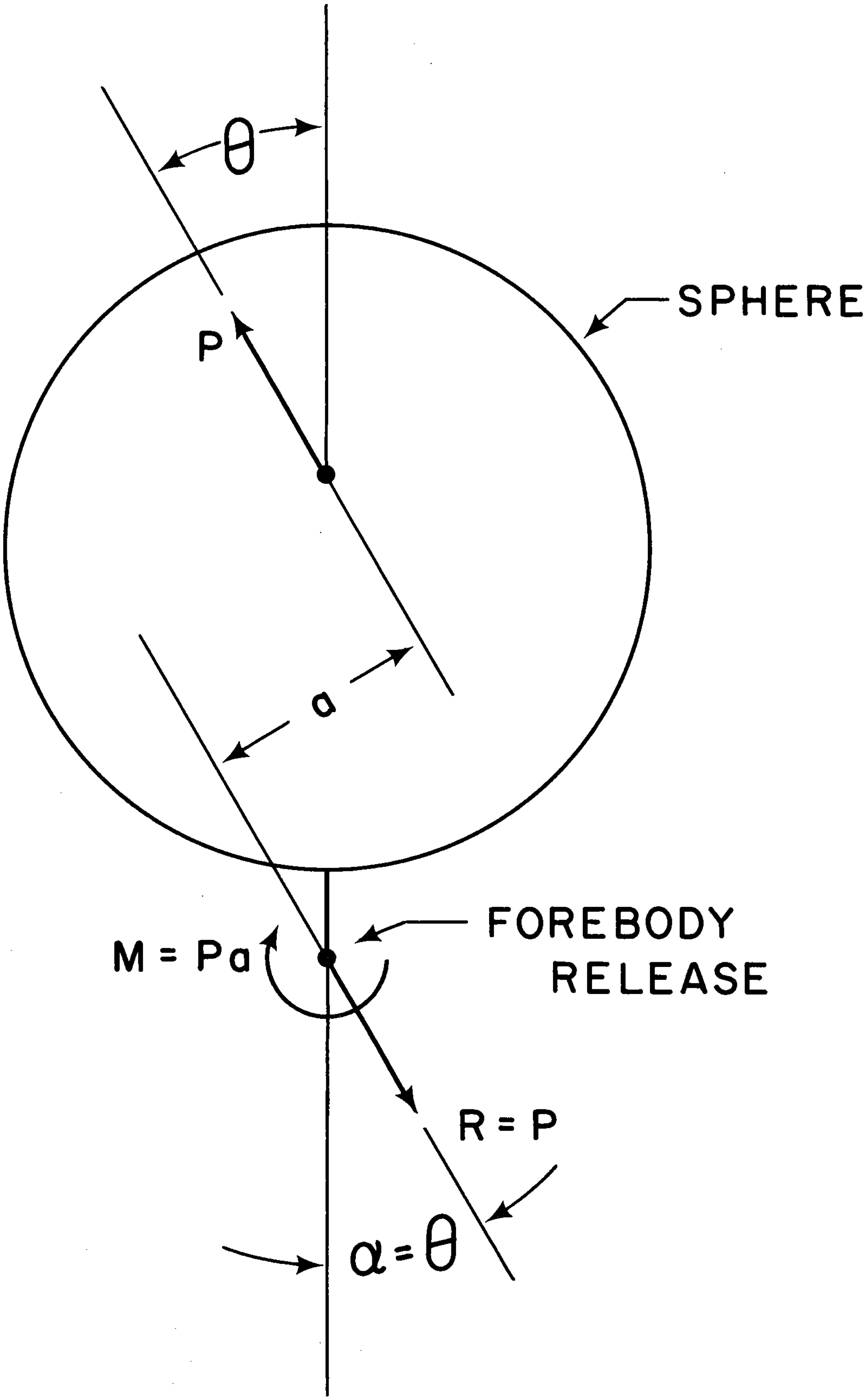


Fig. 13 Schematic of Loading for Case 1 and Case 2

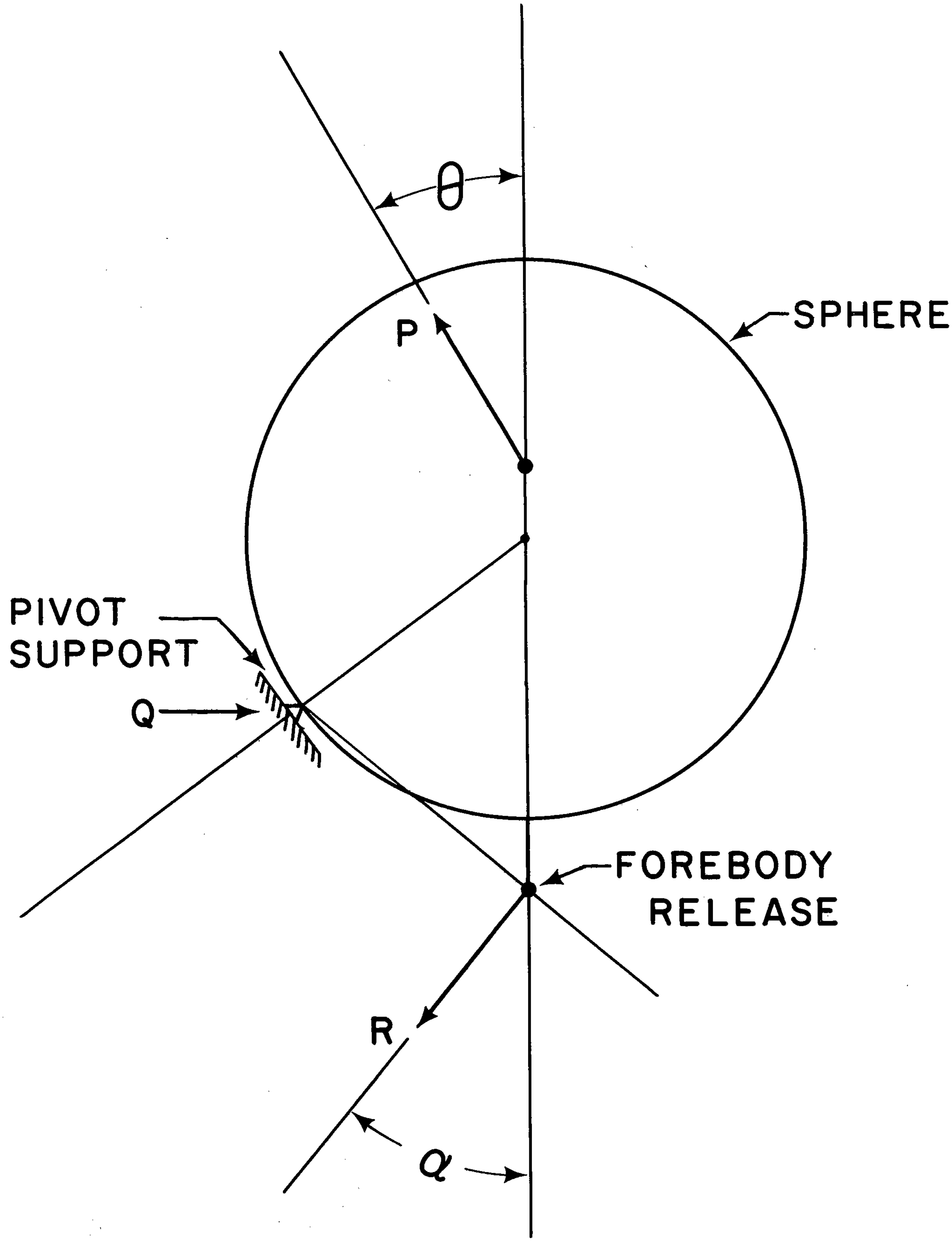


Fig. 14 Schematic of Case 3a Loading

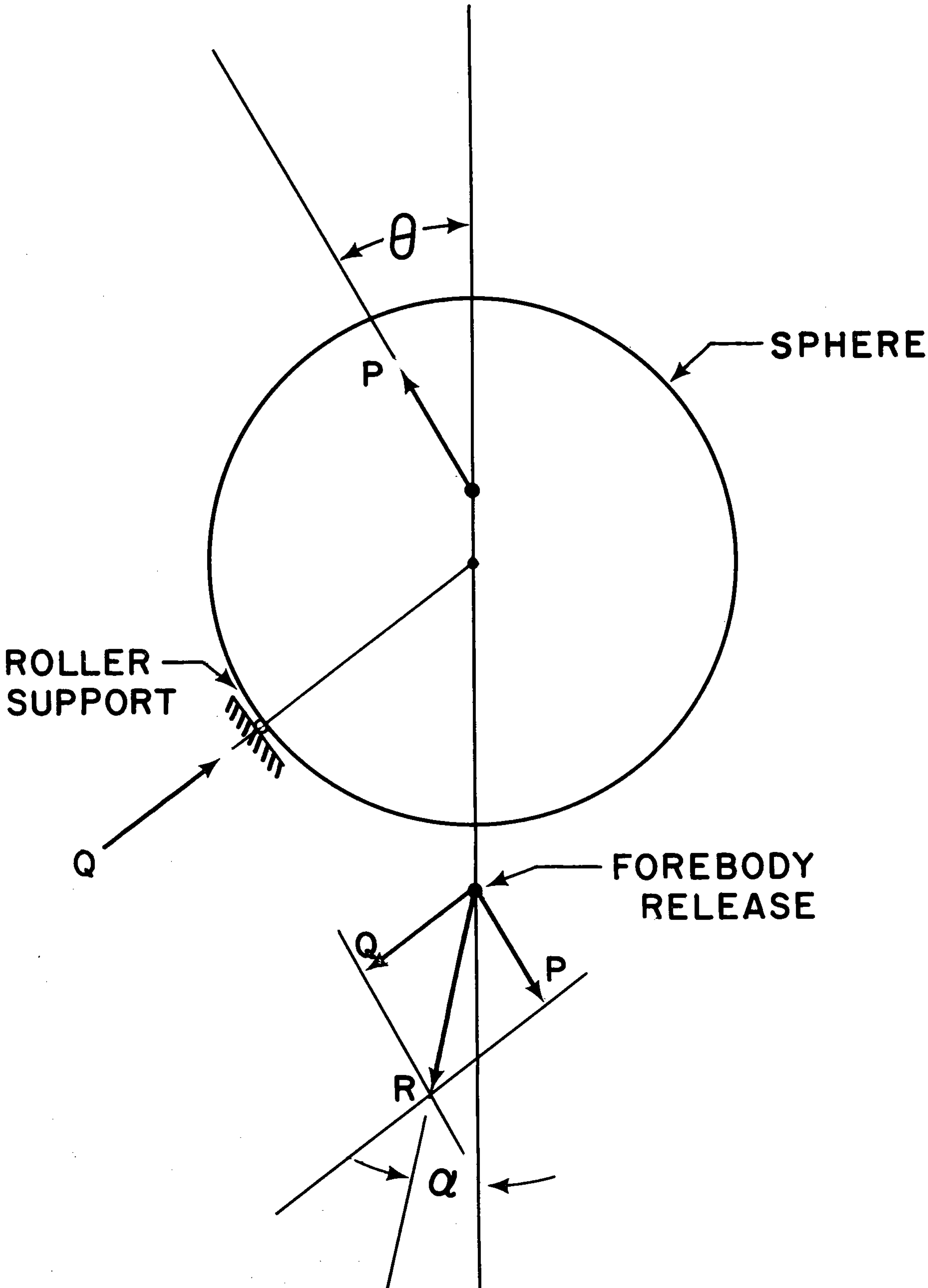


Fig. 15 Schematic of Case 3b Loading

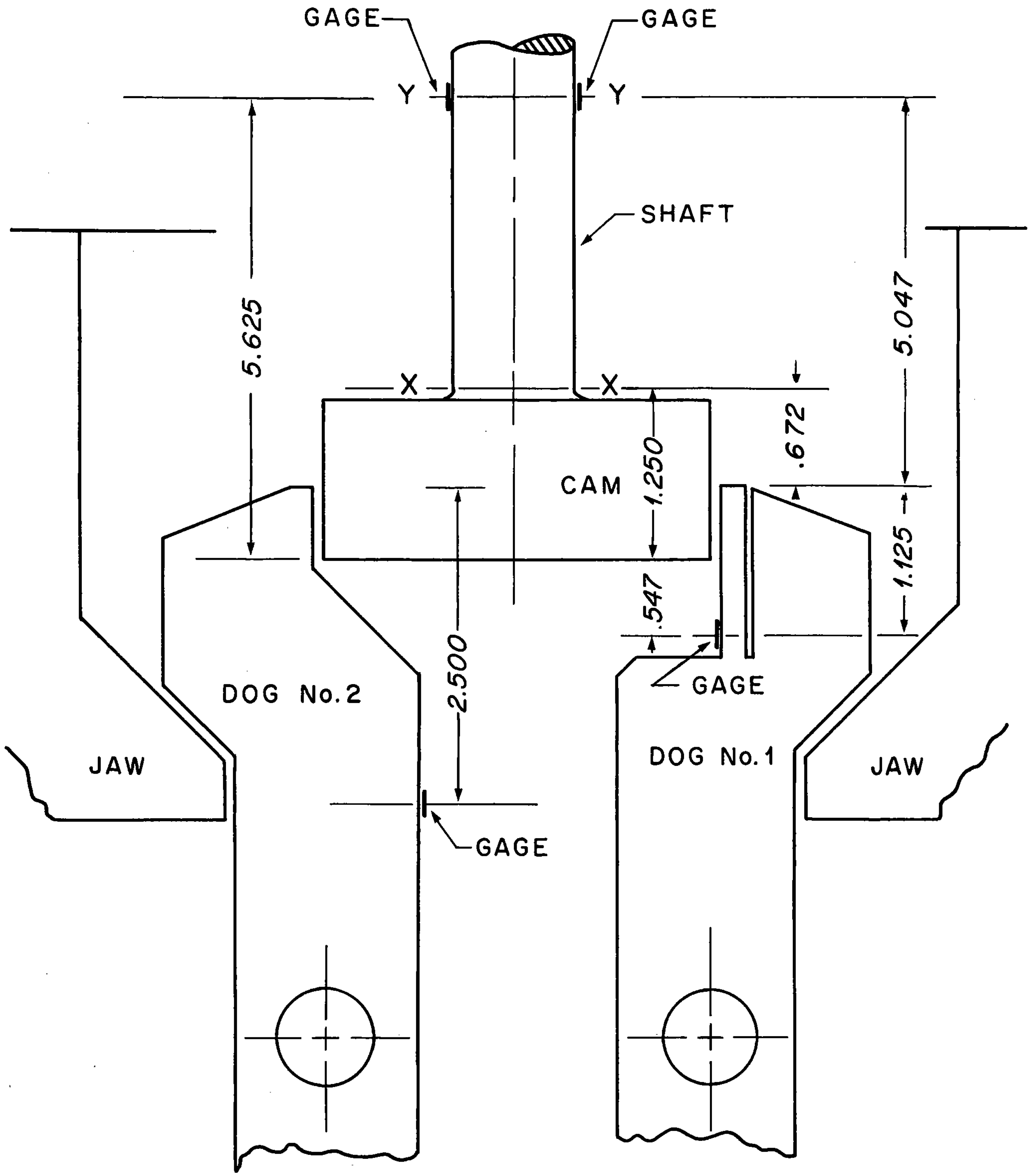
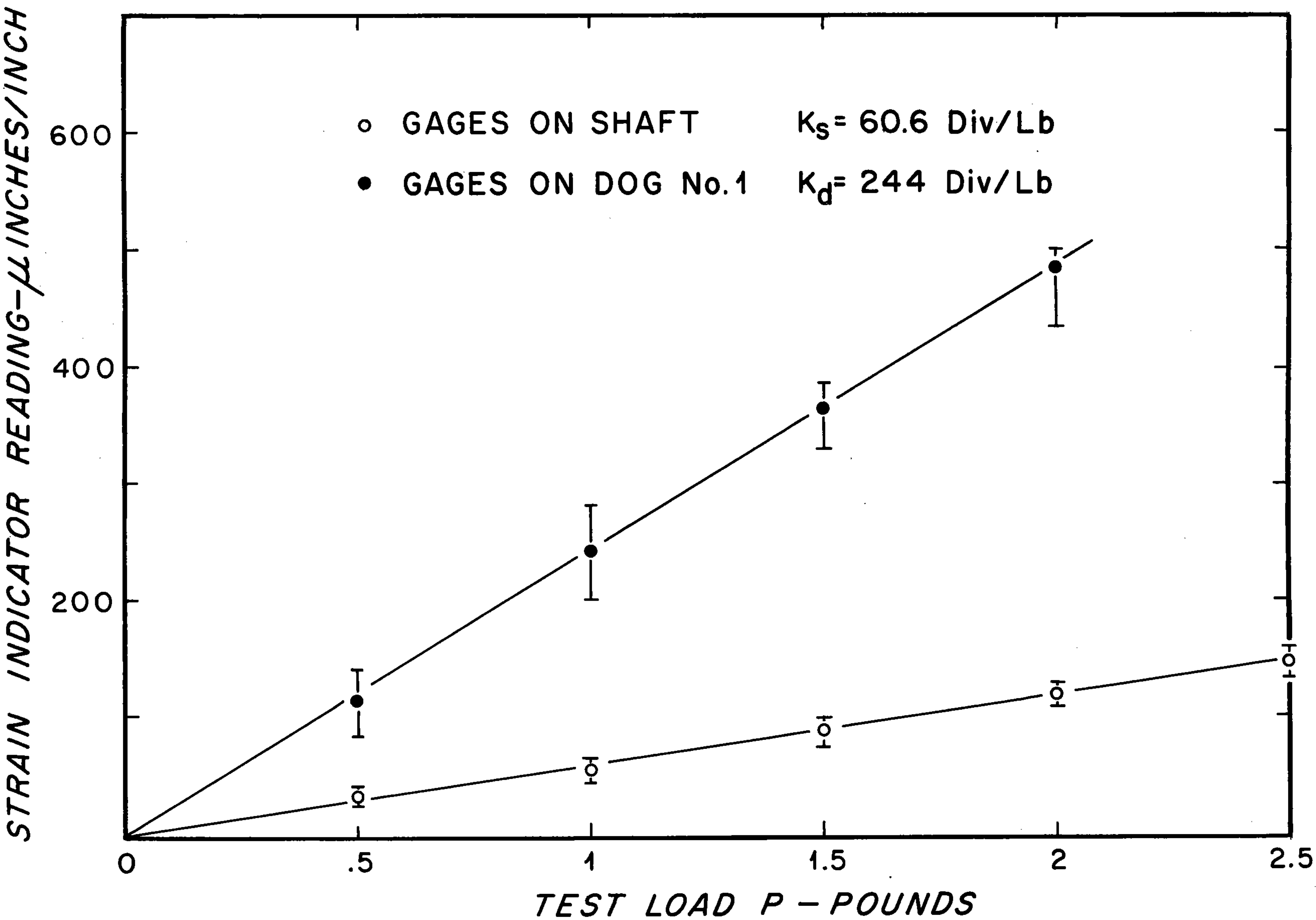
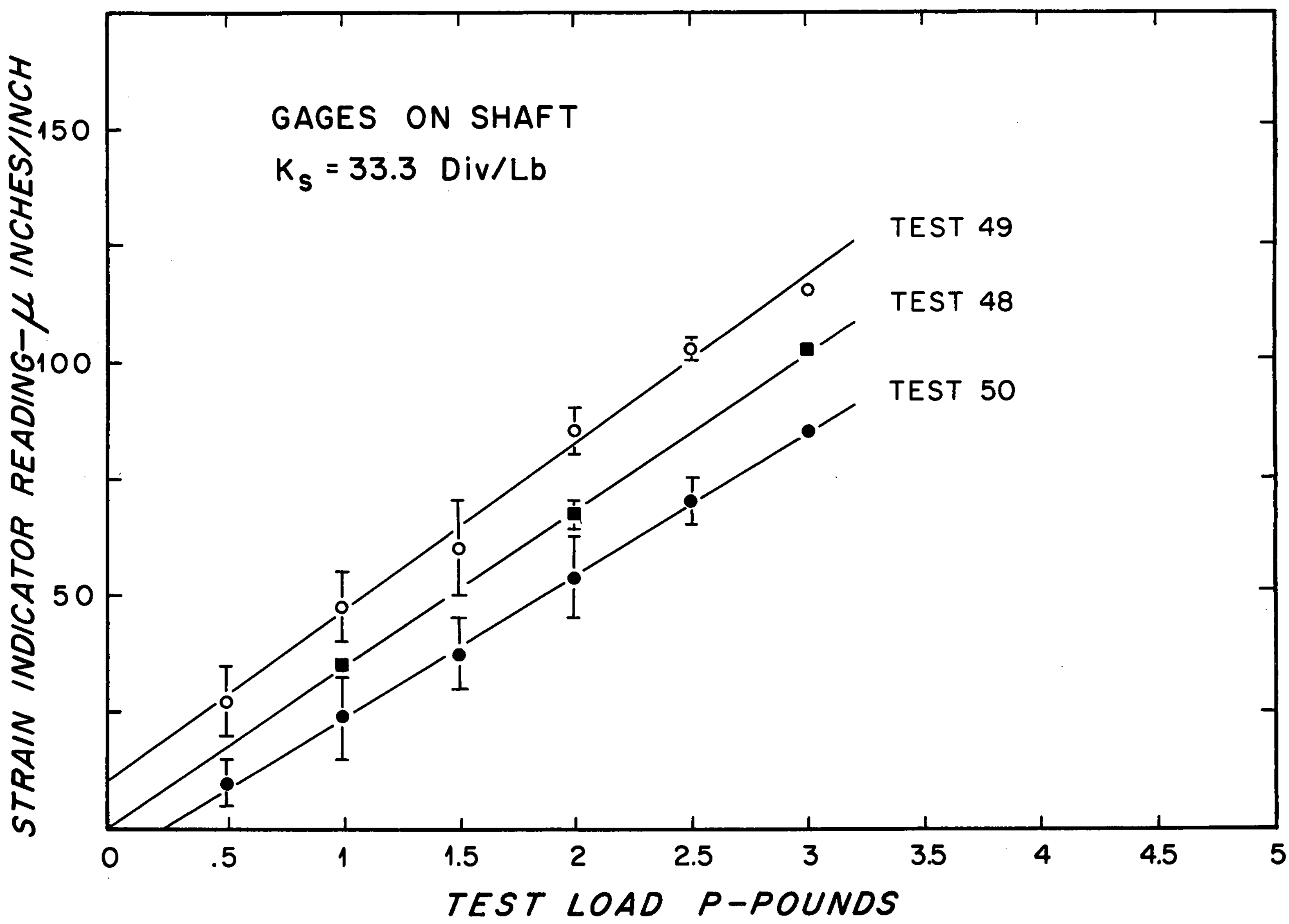


Fig. 16 Placement of Strain Gages on Model

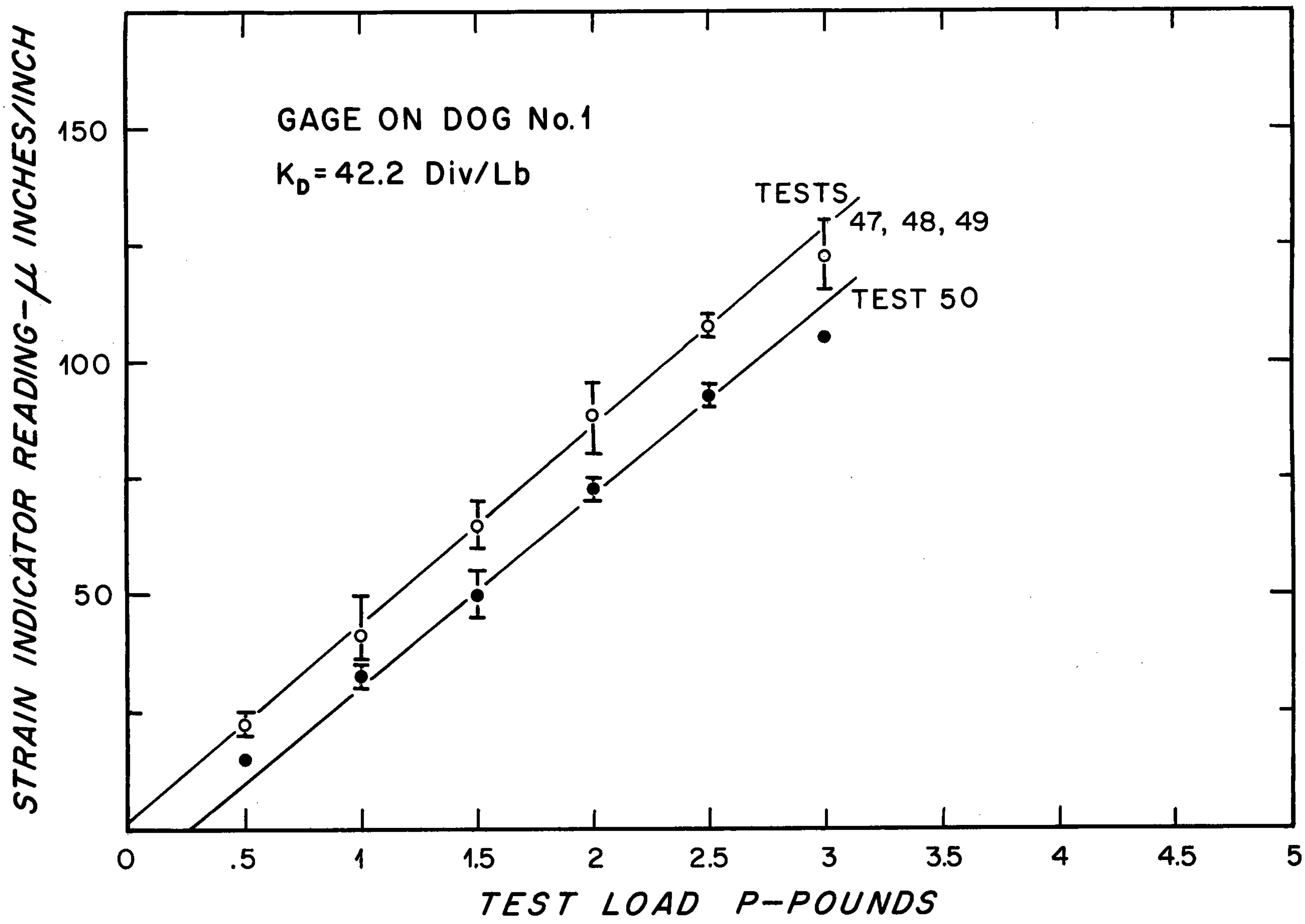
VII APPENDIX B - TEST DATA



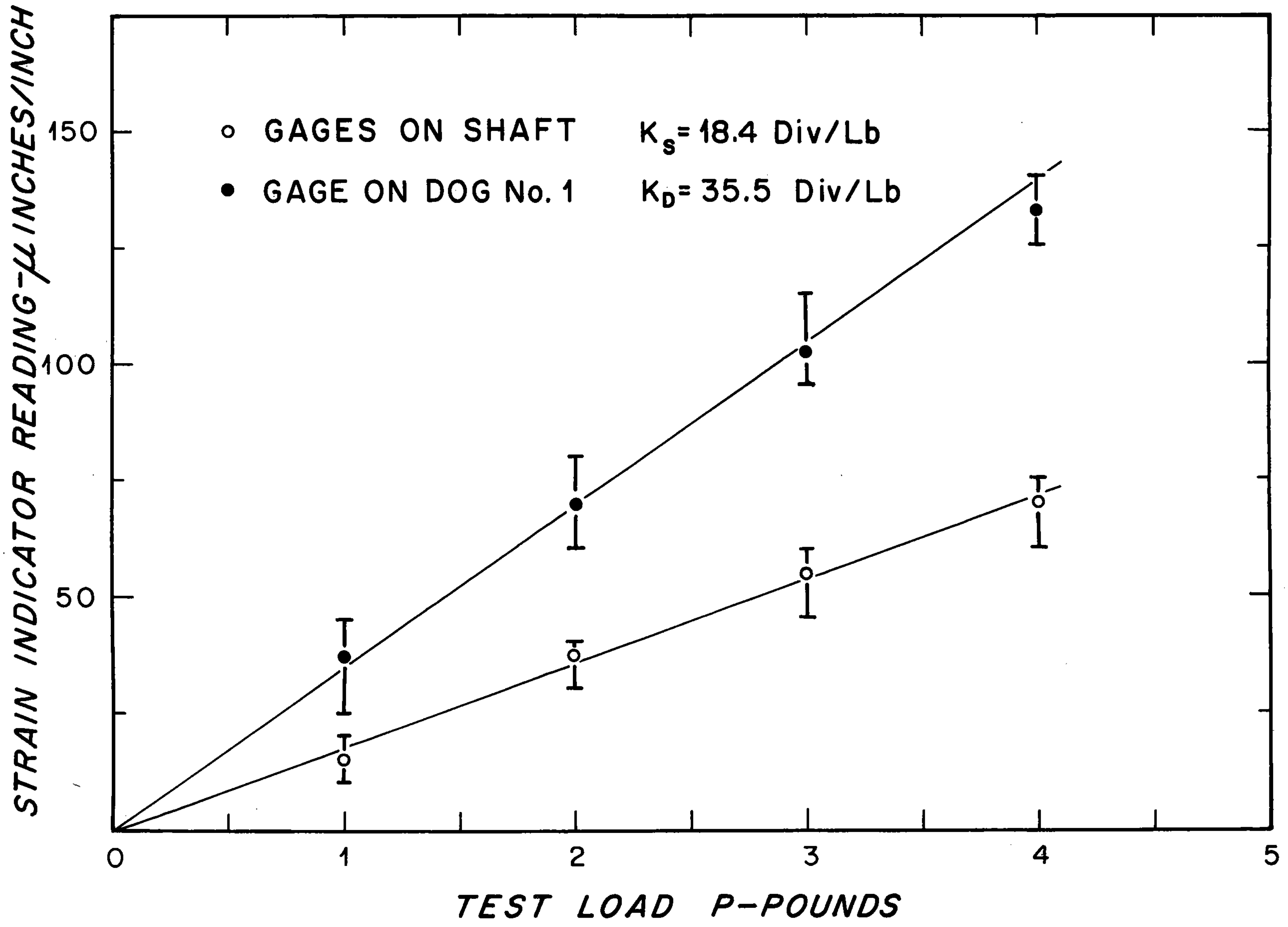
Test Data - Case (1)



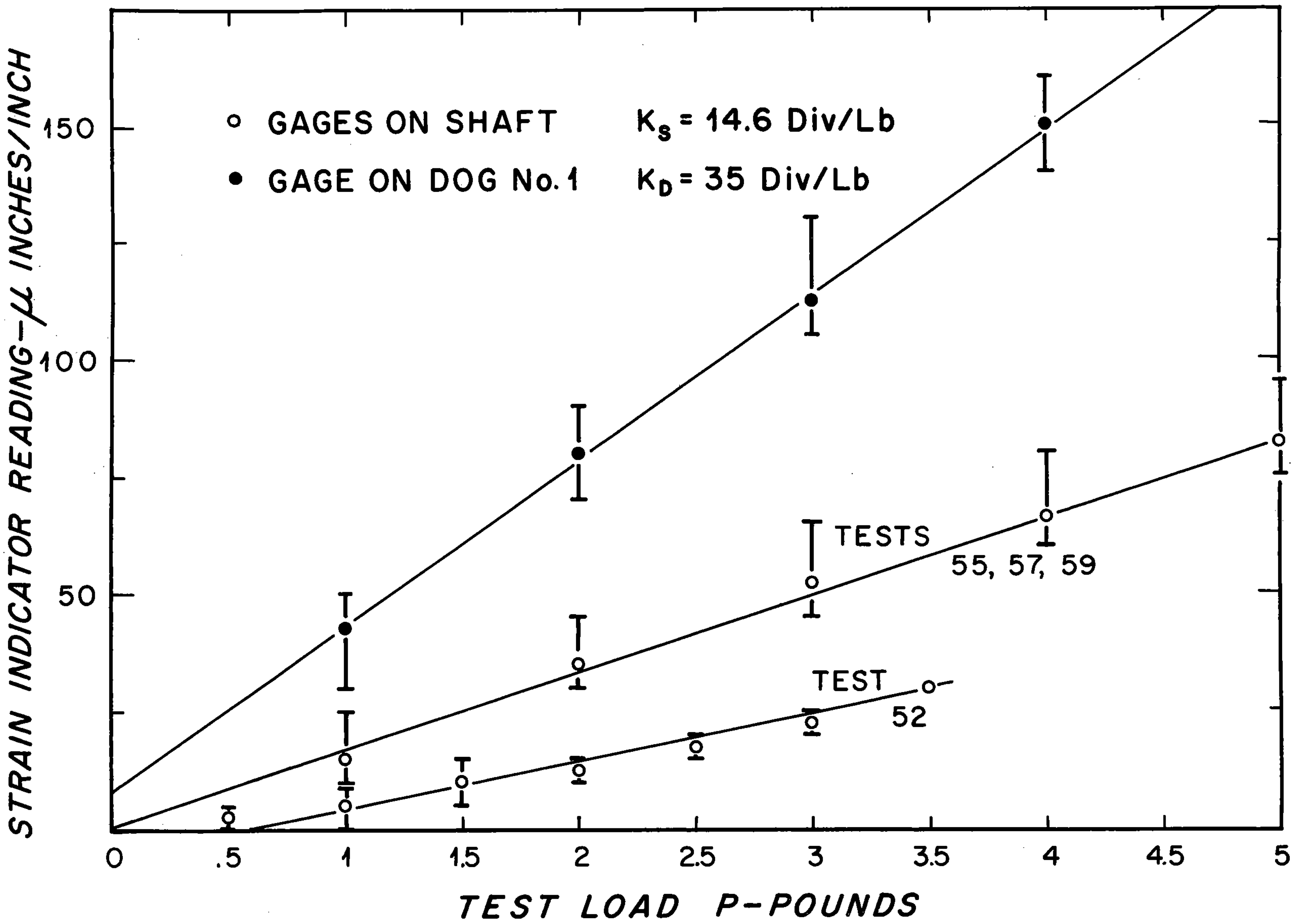
Test Data - Case (2); Clearance 1/8 inch; Shaft Only



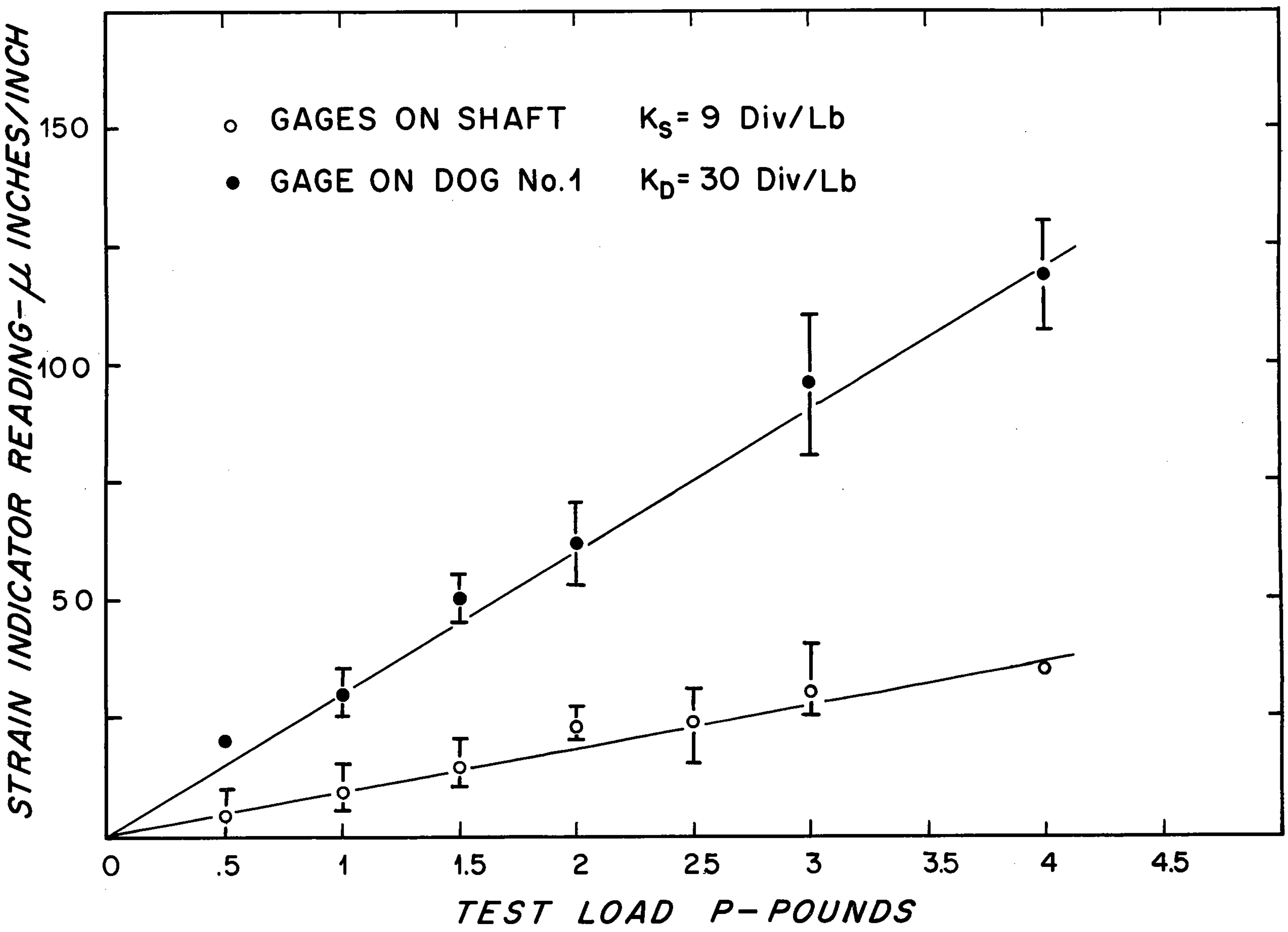
Test Data - Case (2); Clearance 1/8 inch ; Dog Only



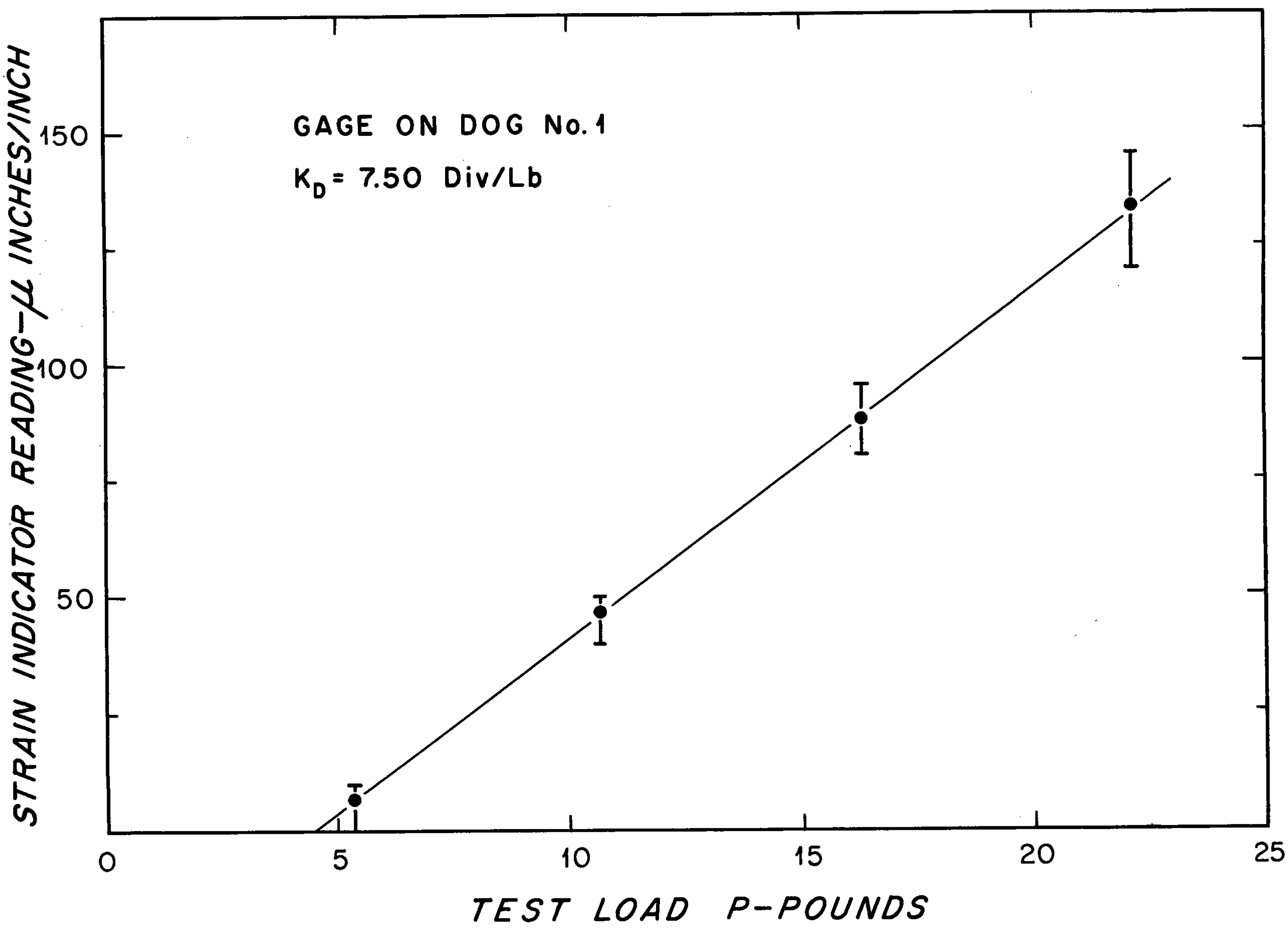
Test Data - Case (2) ; Clearance 1/16 inch



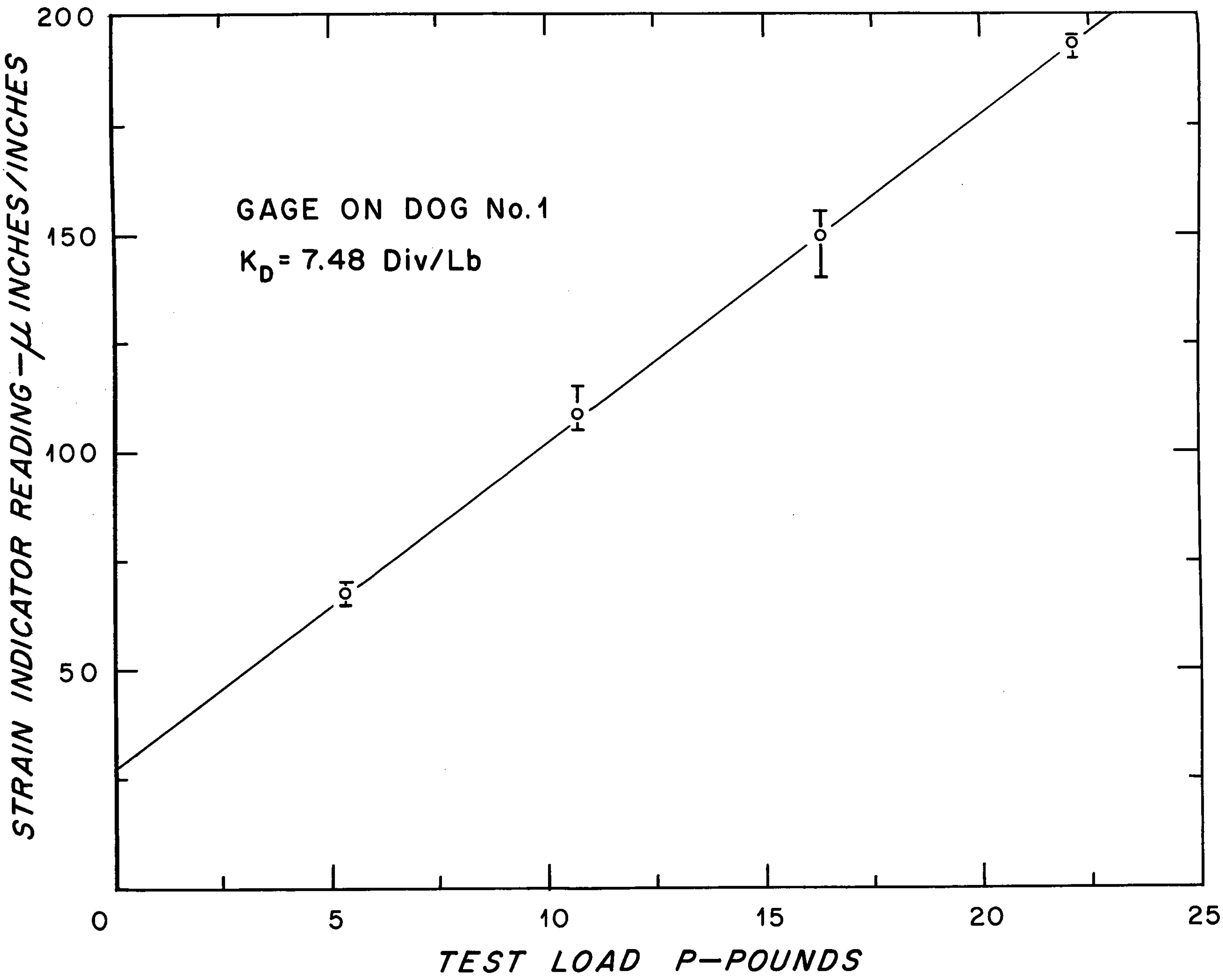
Test Data - Case (2); Clearance .030 inch



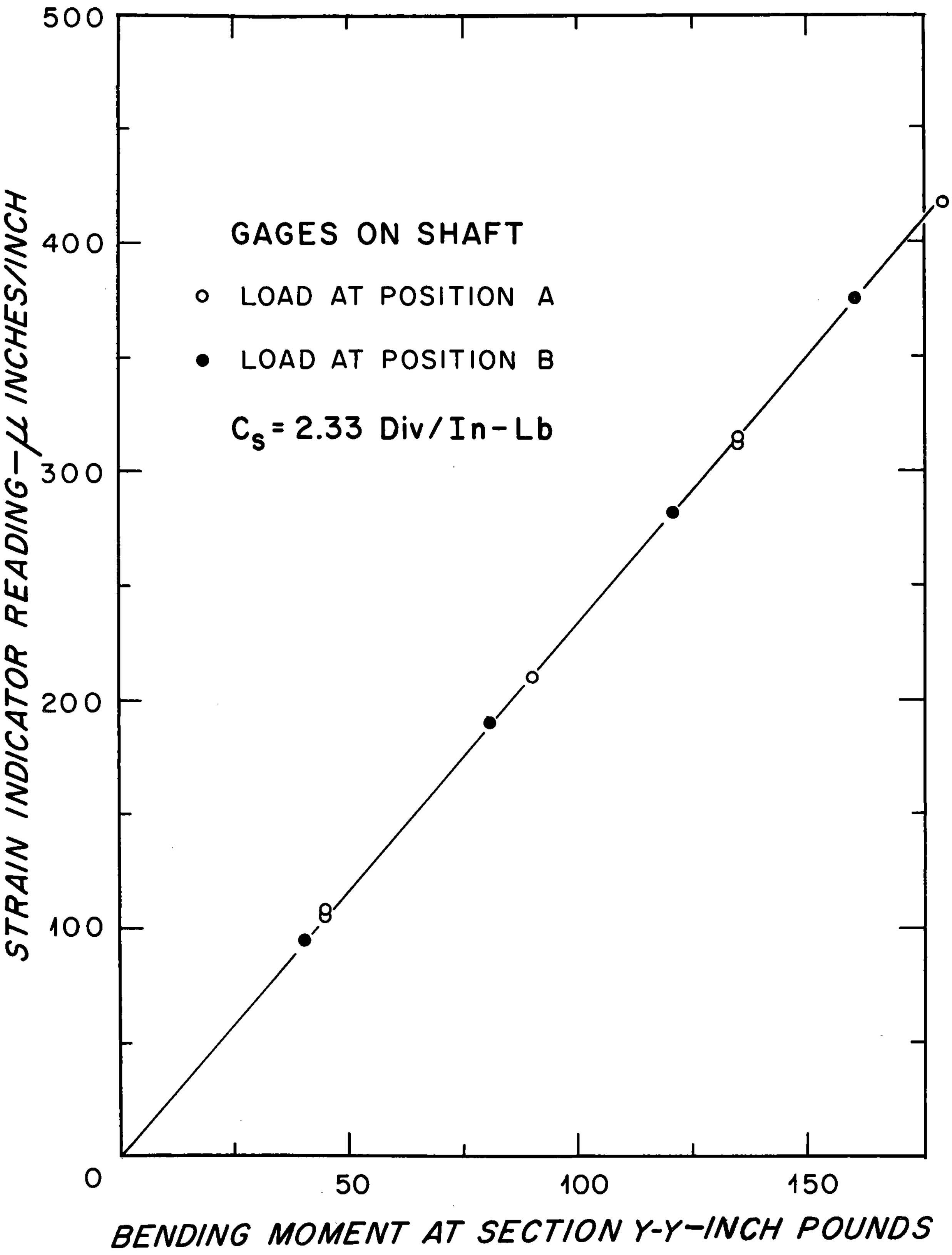
Test Data - Case (2); Clearance .015 inch



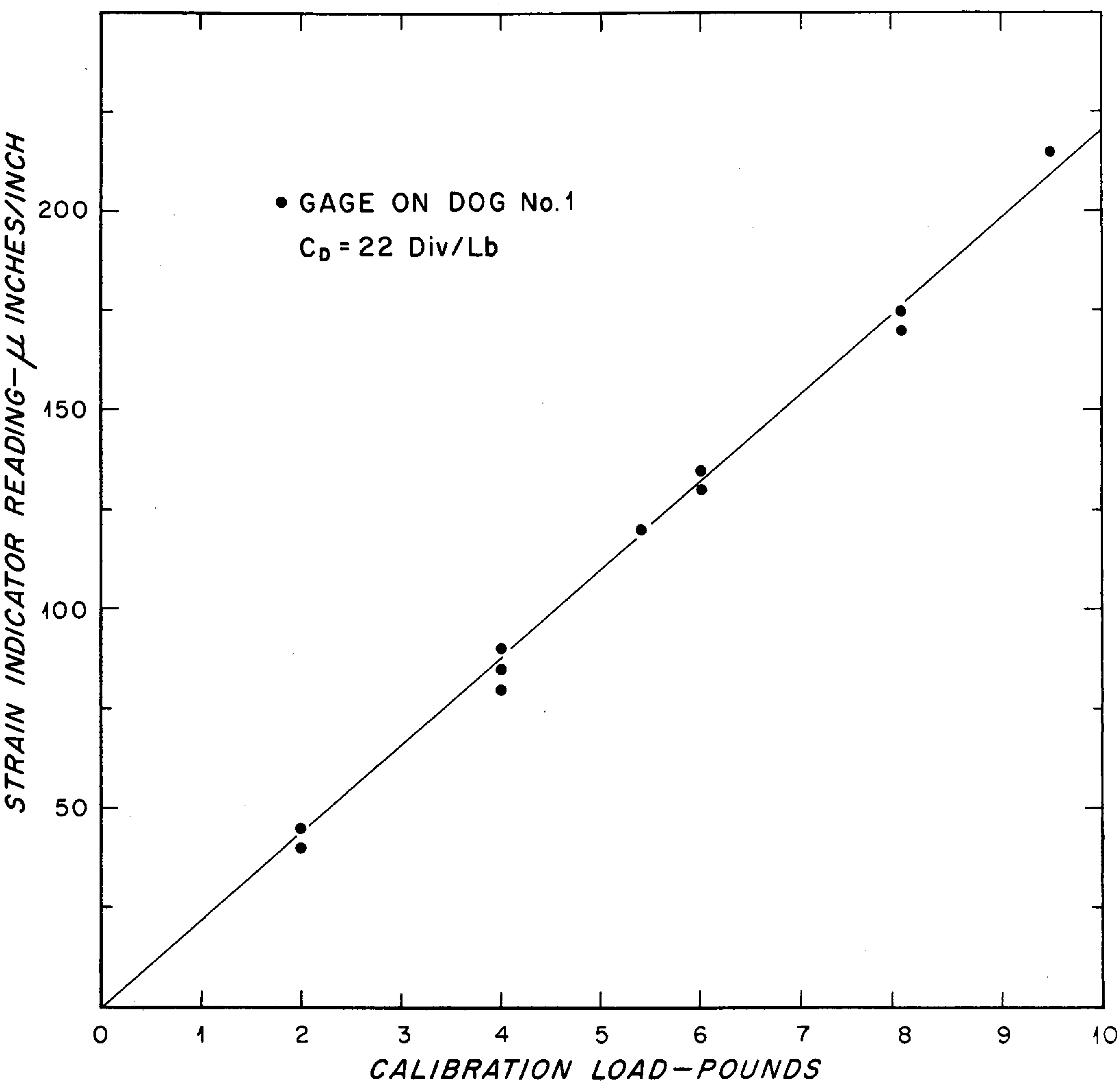
Test Data - Case (3a)



Test Data - Case (3b)



Strain Gage Calibration Curve for Shaft



Strain Gage Calibration Curve for Dog No. 1

VIII APPENDIX C - SAMPLE CALCULATION

Calculation of Shaft Bending Stress

Sample Calculation for Case (1)

1. Strain Gage Outputs

Shaft: $K_S = 60.6$ Div. per pound P

Dog No. 1: $K_D = 244$ Div. per pound P

2. Gage Calibration Constants

Shaft: $C_S = 2.33$ Div. per inch-pound

Dog No. 1: $C_D = 22$ Div. per pound

3. Bending Moment at Section Y-Y of Shaft

$$M_{YY} = \frac{60.6}{2.33} = 26 \text{ inch-pounds per pound P}$$

4. Force at B

$$F_B = \frac{244}{22} = 11.1 \text{ pounds per pound P}$$

5. Force at A (See Fig. 16)

a. Direction of F_A: 46.6^o with horizontal

Sin 46.6 = .727 Cos 46.6 = .687

b. Sum of Moments at Section Y-Y

$$11.1\ (5.047) - .687\ (5.625)\ F_A + .727\ (1.500)\ F_A = 26$$

From which $F_A = 10.8$ pounds per pound P

6. Bending Moment at Section X-X

$$M_{XX} = 11.1\ (.672) - 10.8\ (.687)\ (1.250) + 10.8\ (.727)\ (1.500)$$
$$= 9.97 \text{ inch-pounds per pound P}$$

7. Bending Stress at Section X-X (for ALVIN)

a. Moment of Inertia of Shaft

$$I = \frac{\pi}{64}(d)^4 = \frac{\pi}{64}(.9375)^4 = .0379 \text{ in.}^4$$

$$C = d/2 = .468 \text{ in.}$$

b. Bending Stress

$$\sigma_{xx} = \frac{MC}{I} = \frac{9.97\ (.468)}{.0379} = 123 \text{ psi per pound P}$$

For P = 1844 pounds

$$\sigma_{xx} = 123\ (1844) = 227{,}000 \text{ psi}$$

8. Sum of Moments on Dog No. 1

$$2.12\,F_D = 4.16\ F_B$$

$$F_D = \frac{4.16}{2.12}\ (11.1) = 21.8 \text{ pounds per pound P}$$

9. Sum of Moments on Dog No. 2

$$2\ F_C = 2.25\ F_A$$

$$F_C = \frac{2.25}{2}\ (10.8) = 12.2 \text{ pounds per pound P}$$

IX APPENDIX D - NOMENCLATURE

W - Weight in air of forebody of submersible vehicle, pounds.

D - Displacement of forebody of submerged vehicle, pounds of sea water.

P - Difference of displacement and air weight of forebody D-W; also value of test load applied to model, pounds.

θ - Angle of list or inclination of submerged vehicle, measured from vertical, degrees.

M - Righting moment of forebody of inclined submerged vehicle, inch-pounds.

R - Resultant reactive force at forebody release, pounds.

α - Angle of direction of reactive force R, measured from vertical centerline of vehicle, degrees.

Q - Resultant reactive force at rubber support ring due to righting moment of inclined forebody, pounds.

M_{xx}- Bending moment at section x-x of release shaft, inch-pounds.

M_{yy}- Bending moment at section y-y of release shaft, inch-pounds.

σ_{xx}- Maximum bending stress at section x-x of release shaft, psi.

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Tests were conducted on a full-scale model of the emergency forebody release used in the deep-diving submarines ALVIN, SEA CLIFF, and TURTLE. The model was machined from metal to the same dimensional tolerances as the prototype. Resistance strain gages, attached to the model, permitted measurement of forces on component parts of the device. Of primary concern was the bending stress which might be set up in the release operating shaft when the submarine is submerged in an inclined position. Tests were arranged to simulate three possible conditions of loading of the release device at a 30 degree vehicle list angle: case (1) righting moment of inclined forebody resisted by release components only; case (2) righting moment resisted by release with assistance from lower guides; and case (3) righting moment resisted by couple set up by release and rubber support ring. Test results show that shaft bending stresses (for ALVIN) are high (200,000 psi) for the case (1) condition, lower (40,000-90,000 psi) for case (2), and essentially zero for case (3). The conclusion is that the present forebody release design is adequate for all submarine attitudes encountered in normal operation, provided the vehicle has been assembled so that contact between sphere and rubber ring is assured at all times.

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