07473-1-F

THE UNIVERSITY OF MICHIGAN

COLLEGE OF ENGINEERING
DEPARTMENT OF NAVAL ARCHITECTURE AND MARINE ENGINEERING
SHIP HYDRODYNAMICS LABORATORY

Study of Instrumentation for Rudder Torque Measurements

H. NOWACKI
Project Director: G. L. WEST, Jr.

For:

Bureau of Ships
Department of the Navy
Contract No. NObs-92533

Administered through:

November 1965

OFFICE OF RESEARCH ADMINISTRATION • ANN ARBOR

THE UNIVERSITY OF MICHIGAN

College of Engineering

Department of Naval Architecture and Marine Engineering

Ship Hydrodynamics Laboratory

STUDY OF INSTRUMENTATION FOR RUDDER TORQUE MEASUREMENTS

H. Nowacki

Project Director: G. L. West, Jr.

ORA Project 07473

For:

Bureau of Ships

Department of the Navy

Contract No. NObs-92533

Administered through:

Office of Research Administration

November 1965

Ann Arbor

TABLE OF CONTENTS

I. INTRODUCTION

A variety of possibilities exists for measuring the torque applied to the rudder of naval ships. In the past strain gage instrumentation of the rudder stock has been used with satisfactory success. The accuracy of recent strain gage measurements on the DLG 27 (Josephus Daniels) has been questioned, however, because the effects of bending may have been present in the results.

It is essential for reliable measurements that the torque depends on the stresses, or strains, in a well defined and, if possible, linear way. The effects of bending can only be eliminated by suitable gage arrangement and careful alignment. In order to design an instrumentation which minimizes such detrimental influences the stress pattern in the rudder stock and tiller must be known.

The purpose of this study is to analyze the stress situation in the system in order to discuss the previous measurements on DLG 27 and to investigate alternative methods for making the measurement. The alternatives include modifications of the present system such as altering the tiller shape or measuring angular deflection rather than strain. In principle any location within the system with stresses proportional to the torque may be appropriate for mounting a pickup. Most suitable are those spots where the greatest stresses occur for a specific torque. In addition the location must be accessible for mounting and maintaining the pickup. For these reasons the upper rudder stock and tiller are preferred locations.

II. TORQUE MEASUREMENT AT THE RUDDER STOCK

II-1. Force Situation

Diagram 1 shows a sketch of the rudder and rudder stock and curves of shear force, bending moment, and torque. These curves are based on the following assumptions:

a.) The resultant of the hydrodynamic pressure distribution has two components, one acting normal to the rudder (F_N) and one acting in the rudder central plane (F_T).

For (F_N) the information given in BuShips drawing number DLG 26/800/2,068,018 based on DTMB report number 933 was used for the present purpose. For example, the referenced drawing gives an (F_N) = 920,000 lbs. for a 40 degree rudder angle. The point of action of (F_N) is estimated to be 8 feet below the lower rudder stock bearing. The distributed character of the rudder load has no influence on the forces and moments in the rudder stock above the lower bearing.

The magnitude and location of the tangential force component (F_T) is not well known. It is usually much smaller than (F_N) and is ignored in the bending stress estimate. For similar reasons the moment produced by the weight and buoyancy forces on the rudder is also disregarded. The simple force picture thus obtained is assumed to be sufficient for making some qualitative conclusions.

b.) The steering gear and slide do not provide significant restraint. Therefore, it was assumed that the upper end of the rudder stock above the upper bearing can move laterally without restraint. Thus the upper end of the rudder stock is without shear forces and bending moments.

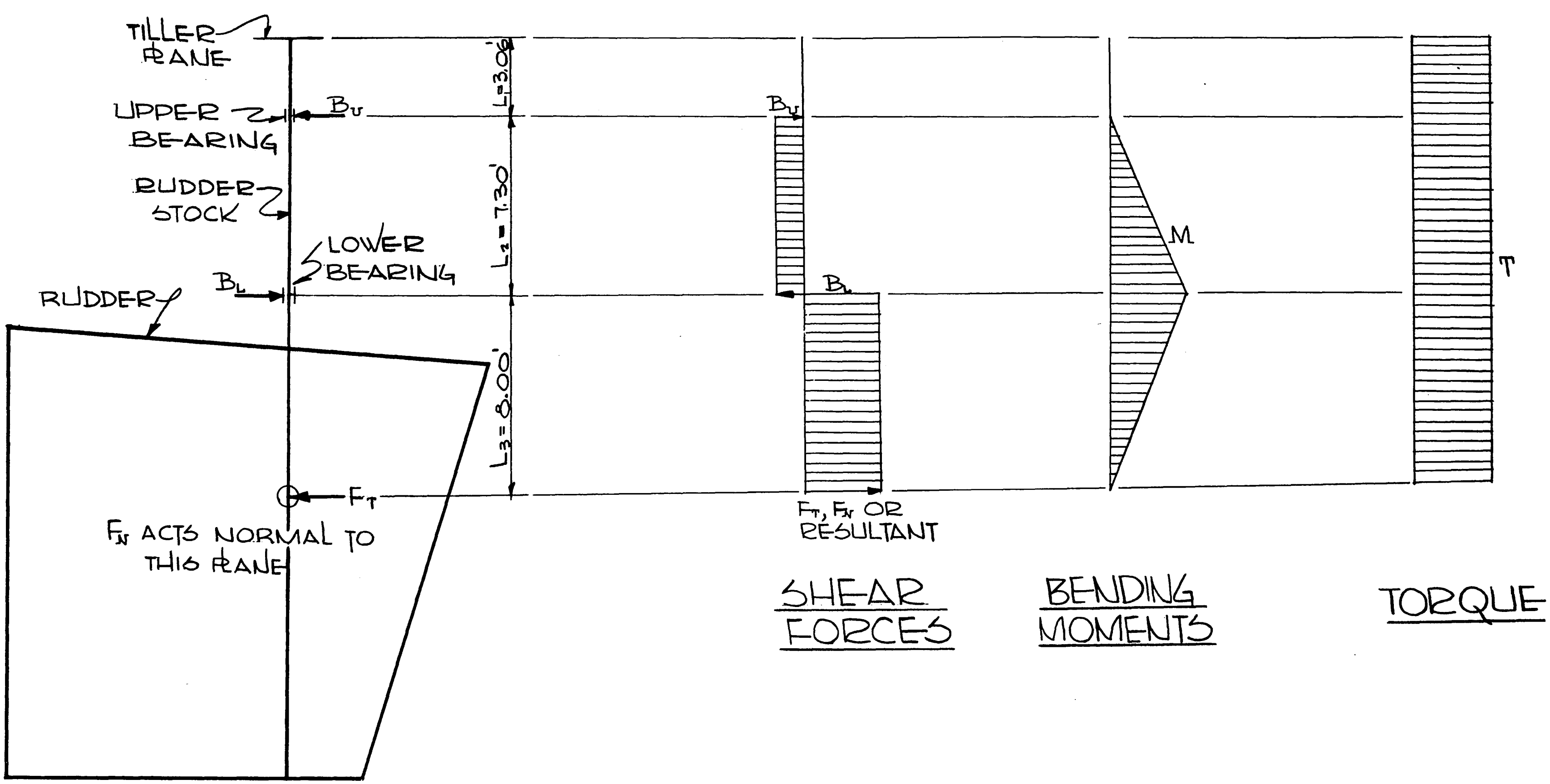


DIAGRAM #1

FORCE, BENDING MOMENT & TORQUE SITUATIONS

Only torsion is present.

c.) If it is assumed that the spherical roller bearings are perfectly aligned and mounted on a rigid foundation, the shear and moment diagrams of a simply supported beam are obtained, Diagram 1. In this situation the maximum bending moment occurs at the lower bearing

$$M_{max} = F_N L_3$$
$$= 920{,}000 \times 8$$
$$= 7{,}400{,}000 \text{ in. lb.}$$

It is likely that the bearing foundation is not completely rigid. An upper limit of the bending stresses is obtained if it is assumed that the lower bearing does not provide any bearing reaction while the upper bearing is rigid and carries the whole reaction. For this extreme case the maximum bending moment would occur at the upper bearing and have a value

$$M_{max} = F_N(L_2+L_3)$$
$$= 920{,}000 \times 15.3$$
$$= 14{,}100{,}000 \text{ in. lb.}$$

In this case the upper portion of the stock would have bending stresses.

d.) The frictional torque in the bearings is small; consequently, the rudder stock is assumed to carry a uniform torsional moment throughout its length. The bearings are of the anti-friction type, they have low frictional coefficients, especially when the shaft is moving. The torque applied by the tiller is with good approximation the torque experienced by the rudder. The maximum nominal torque of $T = 10.5 \times 10^6$ in.lb. at 40 degrees rudder angle was obtained from the previously mentioned drawing and used for the stress analysis.

II-2. Stresses in the Rudder Stock

The dimensions of the rudder stock are shown in the DTMB report, "Evaluation of Full-Scale Rudder Torque Measurements Made on Josephus Daniels (DLG 27)" by L.A. Becker. Diagram 2 is derived from the DTMB report. The following typical sections have been analyzed with respect to torsional and bending stresses:

Section A- just below the tiller in the thinnest portion of the rudder stock

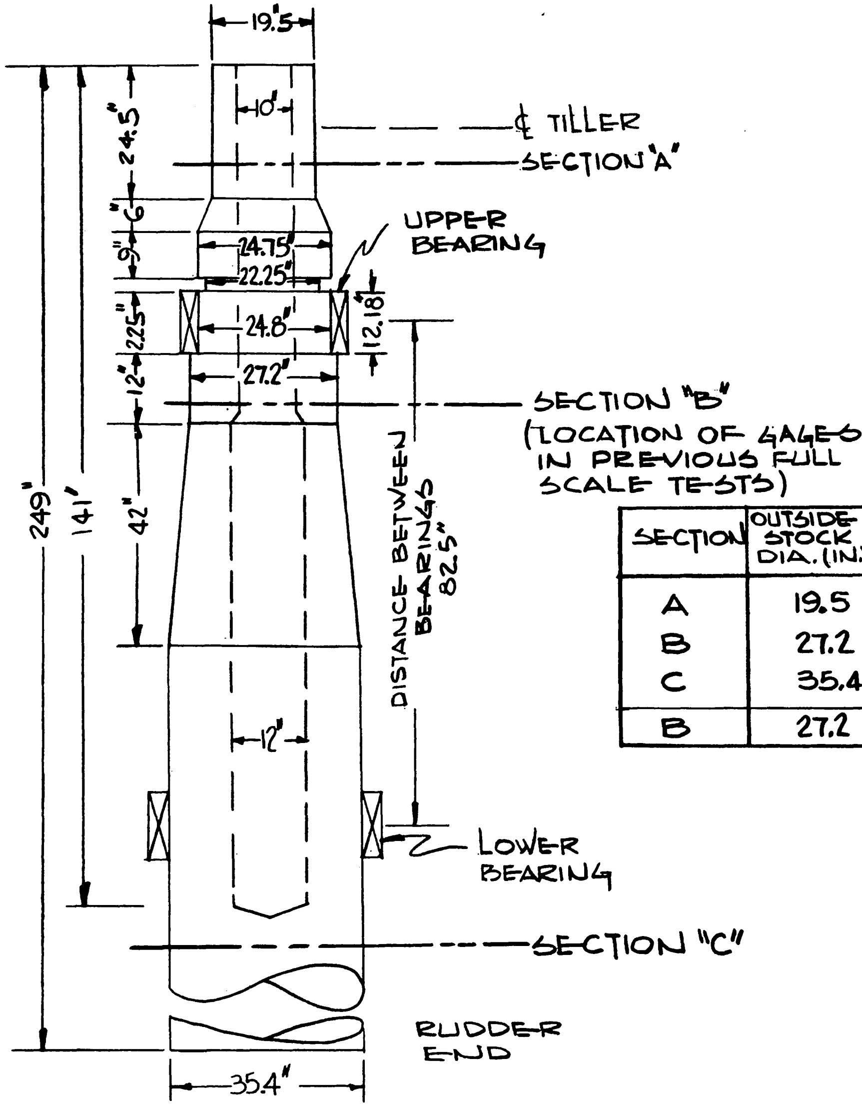
Section B- just below the upper bearing where the strain gages were located in the previous full-scale tests.

Section C- at or just below the lower bearing.

The results of the stress analysis are also given in Diagram 2. In the first three rows of the table rigid bearing supports have been assumed. The result of this assumption is that at location B the bending stresses are much smaller than the torsional stresses in the ratio of about 1:5. For the case of zero stiffness of the lower bearing support the bending stresses at point B are greater than the torsional stresses in the ratio of 2.7:1, row 4. If the upper bearing is also flexible laterally the ratio would be modified but the tendency for the bending stress to be great would remain.

II-3. Rudder Stock Instrumentation

In the previous tests on DLB 27 the strain gage arrangement, theoretically at least, should have compensated for bending. It is, however, difficult to obtain the ideal gage alignment of exactly 45 degrees between the shaft and gage axes. Also the two sets of gages should be precisely diametrically opposed in the same vertical plane. This is of particular importance if the bending strains are comparable in magnitude to the torsional strains. Since the rigidity of the lower bearing support on the DLG 27 is unknown, the bending



SECTION	OUTSIDE STOCK DIA. (IN.)	INSIDE STOCK DIA. (IN.)	POLAR MOMENT INERTIA (IN.^{4})	TORQUE (IN-LB)	BENDING MOMENT (IN-LB)	TORSIONAL STRESS (PSI)	BENDING STRESS (PSI)
A	19.5	10.0	13,200	10.5×10^6	–	7700	–
B	27.2	10.0	52,500	10.5×10^6	1.0×10^6	2720	518
C	35.4	12.0	152,000	10.5×10^6	7.4×10^6	1225	1725
B	27.2	10.0	52,500	10.5×10^6	14.1×10^6	2720	7300

DIAGRAM #2

and torsional strains may have been of the same magnitude during the first tests so that the measurements may have suffered from a combination of excessive bending and gage misalignment. Besides local stresses due to discontinuous variations in cross section at the upper bearing might have been detrimental.

The most obvious method of determining the magnitude of these influences is the calibration of the gages on the stock by applying a known force to the rudder while the ship is in dry dock. But since this direct approach may not be feasible other solutions must be sought. Among the improvements that would reduce, although not eliminate, the uncertainty in the results of the DLG 27 measurements are the following:

a.) Instead of four strain gage pairs around the stock arrange just two pairs on opposite sides of the stock in the neutral bending plane. This plane coincides approximately with the plane of the rudder.

b.) Select a gage location which is above the upper bearing and below the tiller.

These two modifications to the DLG 27 configuration would improve the measurement accuracy and provide the best data from an installation of strain gages on the rudder stock.

An alternative means for determining the torque is to measure deformation rather than strain. It was proposed that a long rod be inserted in the bore of the rudder stock. The rod would be fixed to the stock at the lower end, but remain free at the upper end. Thus the top of the stock would twist relative to the rod as torque was applied to the stock. The angle of twist is then given by

$$\varphi = \frac{T \cdot L}{G \cdot P}$$

T = torque

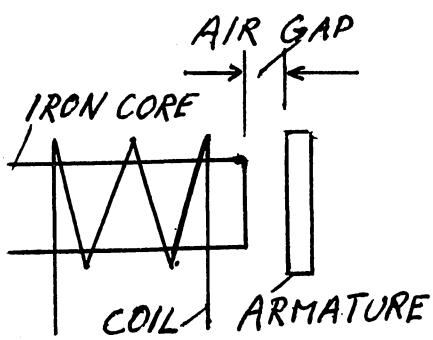
L = length of rod

G = torsional modulus of elasticity

P = polar moment of inertia

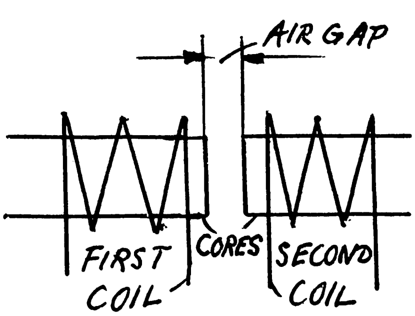
It turned out, however, that the twist angle under maximum torque is very small (about 0.1 degrees), so that sensitive transducers are required. Such transducers are available in the form of inductance type variable air gap gages. Their sensitivity is sufficient to operate with a base length of a few inches: hence the long center rod would not be necessary with these transducers. Two basic arrangements are possible for inductance pickups:

The <u>self-inductance type</u>, sketch 1, which consists of a coil with an iron core and an armature separated by an air gap. As the separation varies the magnetic flux through and the inductance of the coil change. The corresponding voltage variation in the coil circuit is measured.



<u>SKETCH 1</u>: Arrangement of self-inductance transducer.

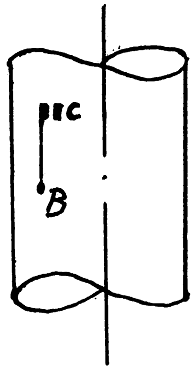
The <u>mutual inductance type</u>, sketch 2, is very similar, but instead of the armature a second coil faces the first one. The effective impedance of one coil is altered by changing the air gap and therefore the mutual inductance between the two coils.



<u>SKETCH 2</u>: Arrangement of mutual inductance transducer.

Both gages can be used for torque measurements if they are arranged on the rudder stock in the following way, sketch 3:

A pointer is attached to the stock at its base point B, but is not connected to the stock at any



other place. The pointer carries a coil at the upper or free end. This coil faces the other coil or the armature which is firmly attached to the stock at point C so that any twisting of the stock between the planes through B and C will cause a change in the gap length.

SKETCH 3: Inductance pickups on rudder stock for torque measurement.

Both types of inductance based transducers are very sensitive. For the self-inductance type the following estimate can be made: The gage measures the torsional deformation which is equal to the torsional strain multiplied by the gage base length. For the maximum torque $T = 10.5 \cdot 10^6$ lb. in. and for section A through the upper portion of the stock, Diagram 2, where the polar moment of inertia is $P = 13,200$ in.^{4} one obtains:

$$\text{Torsional strain} = \frac{T \cdot R_o}{G \cdot P} = 6.5 \cdot 10^{-4} = \sim 0.001 \text{ in./in.}$$

$$\text{Torsional deformation} = 0.001 \cdot \text{gage base length}$$

For an air gap of 0.001 in. and a pointer length of 1 in. for example, a strain of 0.001in./in. will change the length of the air gap by 10% and the impedance of the gage element by nearly the same amount. The voltage output per unit strain is in this case:

$$\frac{.1\,\Omega/\Omega}{0.001 \text{ in./in.}} = 100 \;\frac{\Omega/\Omega}{\text{in./in.}}$$

This figure is much higher than the sensitivity of a resistance strain gage which is about 2.1 Ω/Ω per in./in. Moreover it can be increased in linear proportion to the base length of the transdrcer. The mutual inductance transducer is of comparable sensitivity.

These advantages are achieved at the expense of some serious shortcomings:

a.) The inductance gages are not commercially available. A few standard sizes are usually at hand in model basins, but it is likely that a special gage must be designed for full scale rudder torque measurements.

b.) The gage characteristics are not quite linear. For this reason an air gap variation of 10% should not be exceeded. Careful calibration is important.

c.) The gages are susceptible to the influences of moisture and dust. To avoid such disturbances gages submerged in an oil bath have been applied successfully. There are also the problems of temperature compensation and zero stability. Generally speaking, it usually takes a period of careful laboratory tests to develop a reliable gage setup.

d.) To eliminate or reduce the influence of bending deformations special precautions are required similar to those for the resistance gages. The transducers ought to be arranged in the neutral bending plane and perhaps a compensating pair of pickups on opposite sides of the stock should be provided.

In summary, the principal advantage of the inductance gages, sensitivity, does not offset the many shortcomings. The resistance gage sensitivity seems to have been adequate for the present purpose. Thus we do not encourage the development of an inductance gage unless it is for the sake of checking the results of this new alternative against previous data.

III. TORQUE MEASUREMENT AT THE TILLER

The tiller, which is the element by which the torque is applied to the rudder stock, seems to be a promising gage location because of its good accessibility. Also for the present DLG 26 Hyde Windlass design the stresses are comparable to or greater than those in the rudderstock. It must however be realized at the outset that the stresses in the tiller boundaries are not independent of the rudder angle, i.e. of the location where the load is applied in the slotted tiller ares. This necessitates either a more complicated calibration and evaluation procedure or a more sophisticated instrumentation than on the rudder stock.

Despite these drawbacks it was considered worthwhile investigating the tiller alternative because it would remove all doubts about the type of support and bending influences. Also the tiller instrumentation can be calibrated conveniently in the laboratory whereas the rudder stock requires dry dock calibration.

III-1. Stress Analysis of Tiller

The DLG 26 Hyde Windlass tiller is reproduced in diagram 3. The load is applied by blocks driven by the steering gear plungers and sliding in the slotted tiller arms. The reaction torque is supplied by the rudder stock through keys and partially by the friction between tiller and stock.

Because of the complicated tiller shape and load situation there are no simple means of analytical stress prediction. It was therefore decided to evaluate the stress pattern by photoelastic experiments. For simplification the tiller was approximated by a two-dimensional configuration by collapsing the top and bottom arms and neglecting the cylindrical center portion of the tiller hub between the two arms.

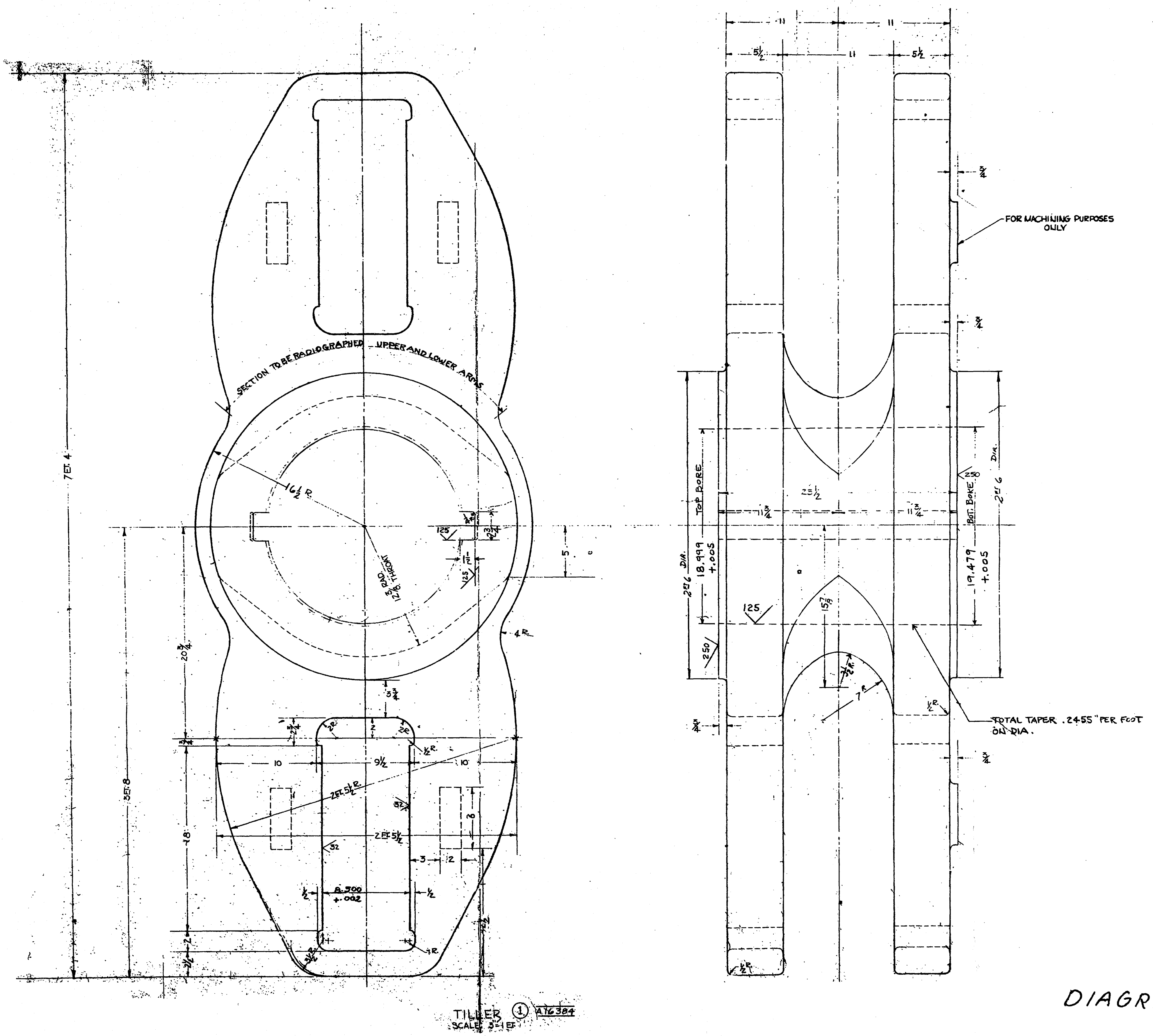
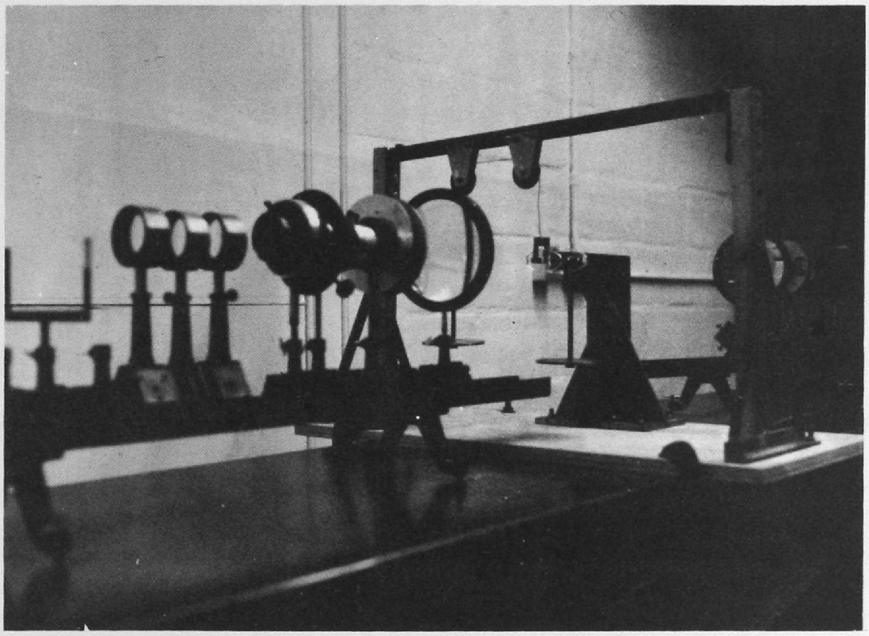

SECTION TO BE RADIOGRAPHED UPPER AND LOWER ARMS
FOR MACHINING PURPOSES ONLY
TOTAL TAPER .2455" PER FOOT ON DIA.
TOP BORE
BOT. BORE
18.999
+.005
19.479
+.005
TILLER (1) A16384
SCALE 3"=1 FT.

DIAGRAM 3

A model of 5.2 in. length (scale 1:17) was built of a transparent plastic CR-39 sheet, thickness 3/8 in. A special loading device was designed consisting of bars in the slots, which were loaded by weights suspended from a wire and pulley system; the reaction was taken up by a plate on which the model was sliding in its keyways. The frictional torque flow was neglected. Picture 1 shows model and loading device in the optical bench.



PICTURE I: Photoelastic model with loading device in optical bench.

In the tests the model was put under load with an optical arrangement that produced white circular polarized light. This results in a pattern of colored fringes or isochromatics in which on each fringe the difference between the two principal stresses σ_1, and σ_2 is a constant and proportional to the fringe order number. On the boundaries in particular one principal stress vanishes and the fringe gives the remaining normal stress directly. The patterns were visualized on a screen, sketched, and analyzed qualitatively. Photographs were taken for evaluation. The location of the load was varied in the slots from outside to inside. The pictures 2 through 7 show photographs taken with these load locations and loads of 15 and 25 lbs. The dark obstructions in way of the model are the supporting plate in the lower center part of the tiller and the shackles covering a strip of the tiller arms. In pictures 6 and 7 the inner ends of the slot are partly obstructed by screws. The bars in the slots can also be recognized.

The maximum number of fringes that can be obtained is limited by the yield stress of the model. Unfortunately, in this case the critical stresses occured in the keyways whereas the fringe order numbers encountered at the interesting boundaries of the tiller were relatively low. As a result the stress distributions were not so well resolved, but they ought to show the trends correctly.

Diagrams 4 through 7 show the model stresses evaluated for the outside and slot boundaries of the tiller. The stress pattern in the missing tiller half is cross-symmetric. Diagrams 8 and 9 depict the interpretation of the fringe patterns for two of these cases.

The most significant stresses are encountered in the slot corners. Along the outside boundary of the tiller the stress is relatively uniform with zeros in the plane of symmetry and somewhat below the load. There are also zero stresses at some points under the load bars in the slots. These must be due to deformations or bar misalignment.

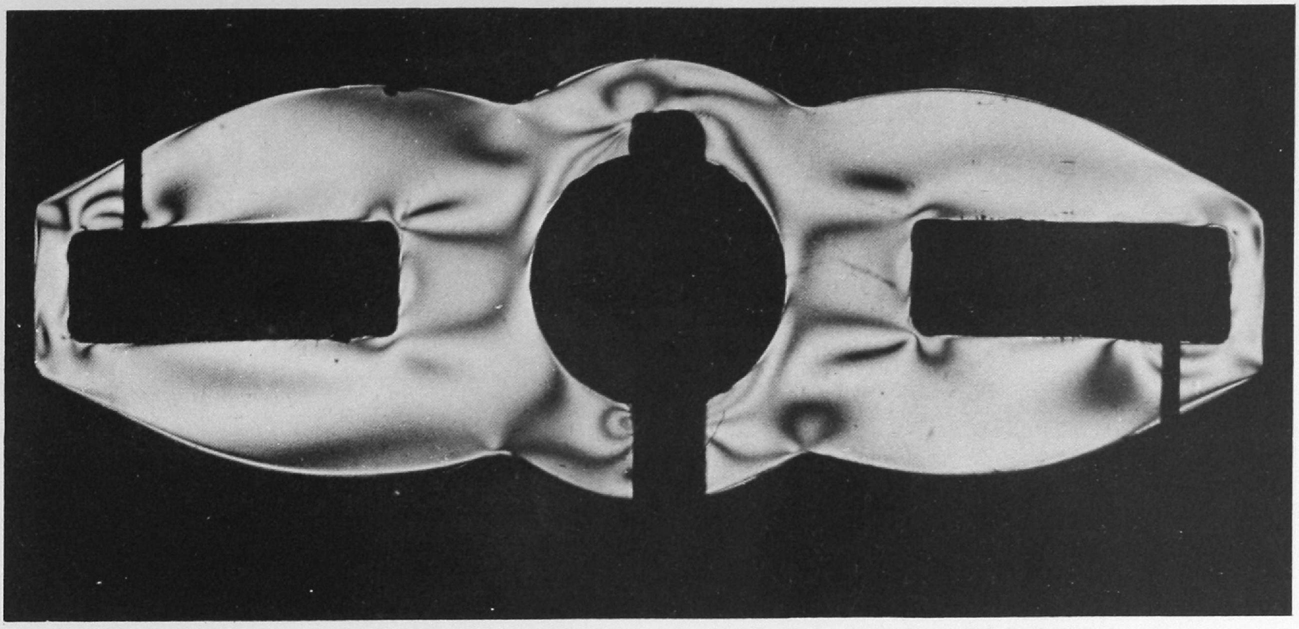
The model stress pattern can be converted to full scale for any load in the elastic range. Let us for example correlate the 25 lbs. load cases to the maximum design force normal to the slot which occurs at 40 degrees rudder angle:

Total force in block at 40^{o} angle:

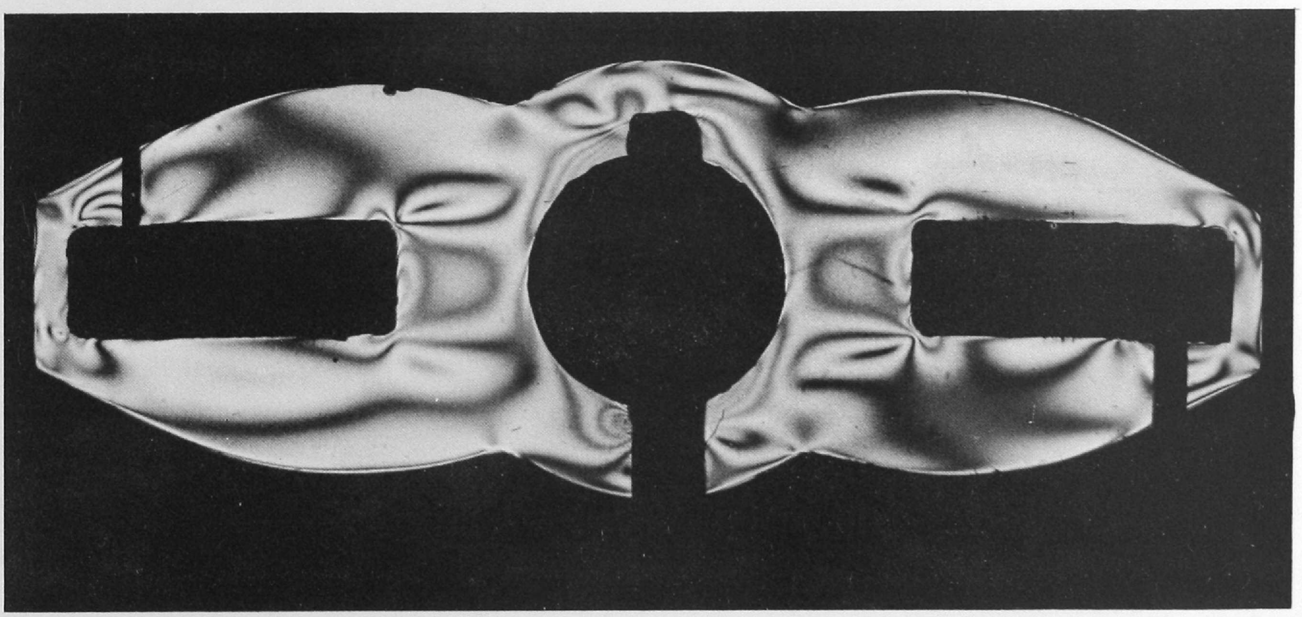
$$F = \frac{\text{design torque}}{2R_o} \cdot \cos 40^{o} = \frac{10.5 \cdot 10^{6}}{2 \cdot 26} \cdot 0.766 = 155{,}000 \text{ lbs.}$$

Adding about 25% for Raphson slide losses:

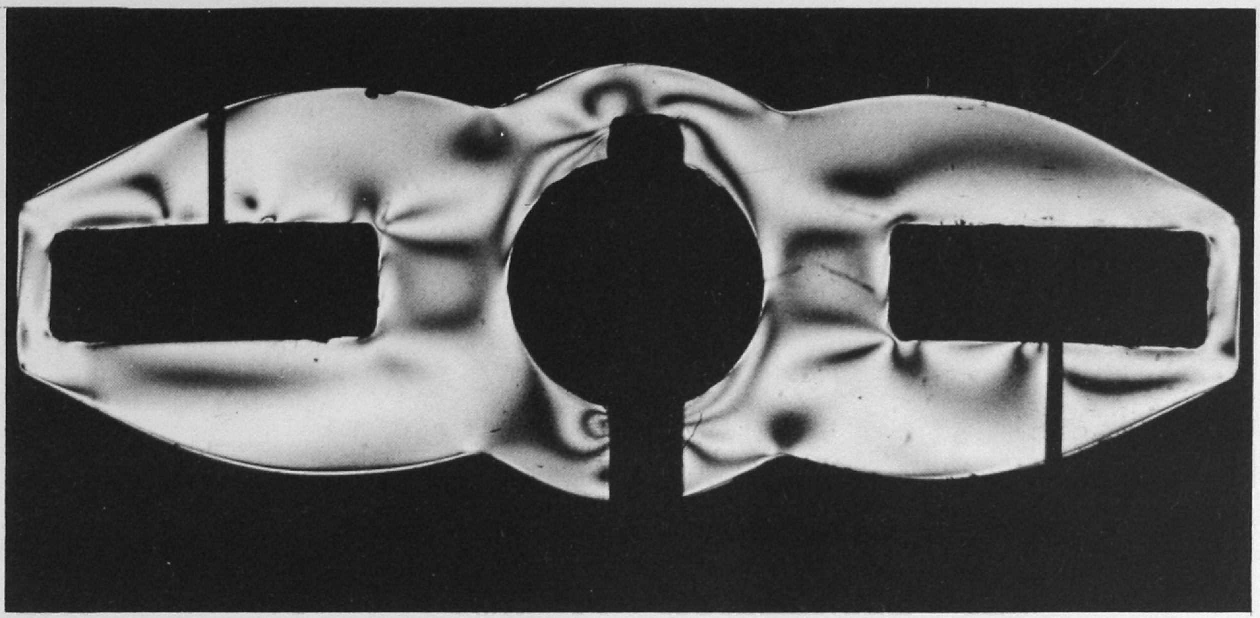
$F = 190{,}000$ lbs.



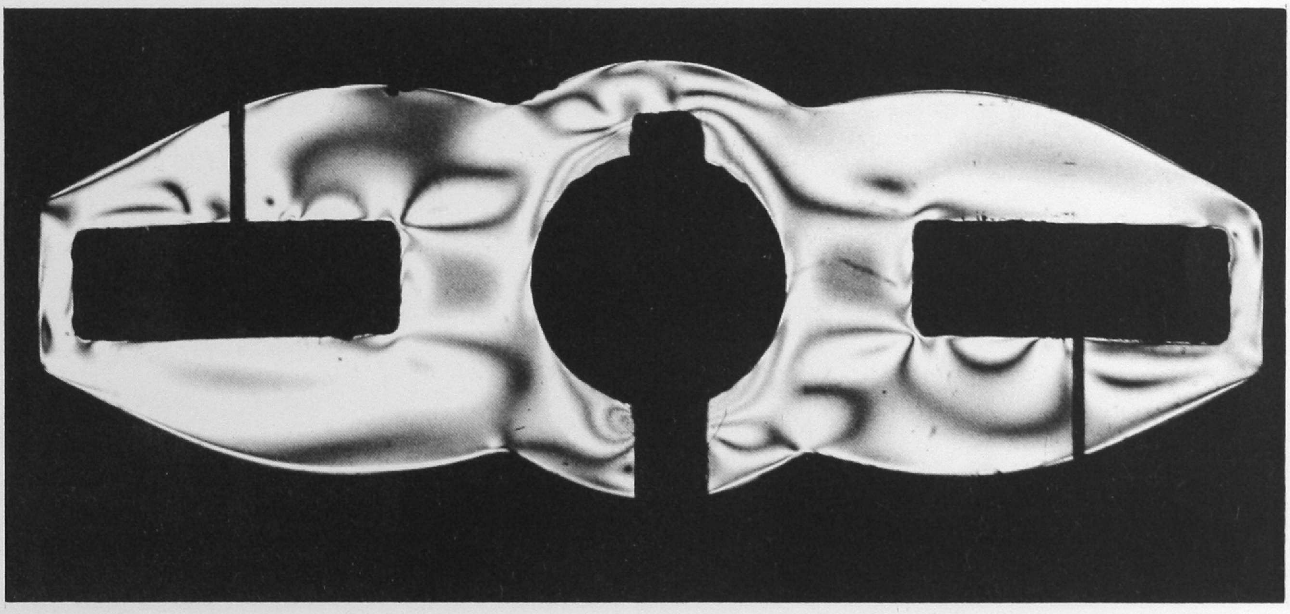
Picture 2: Isochromatics in model with torsional load of 15 lbs. in each arm, load applied at outer end of slot.



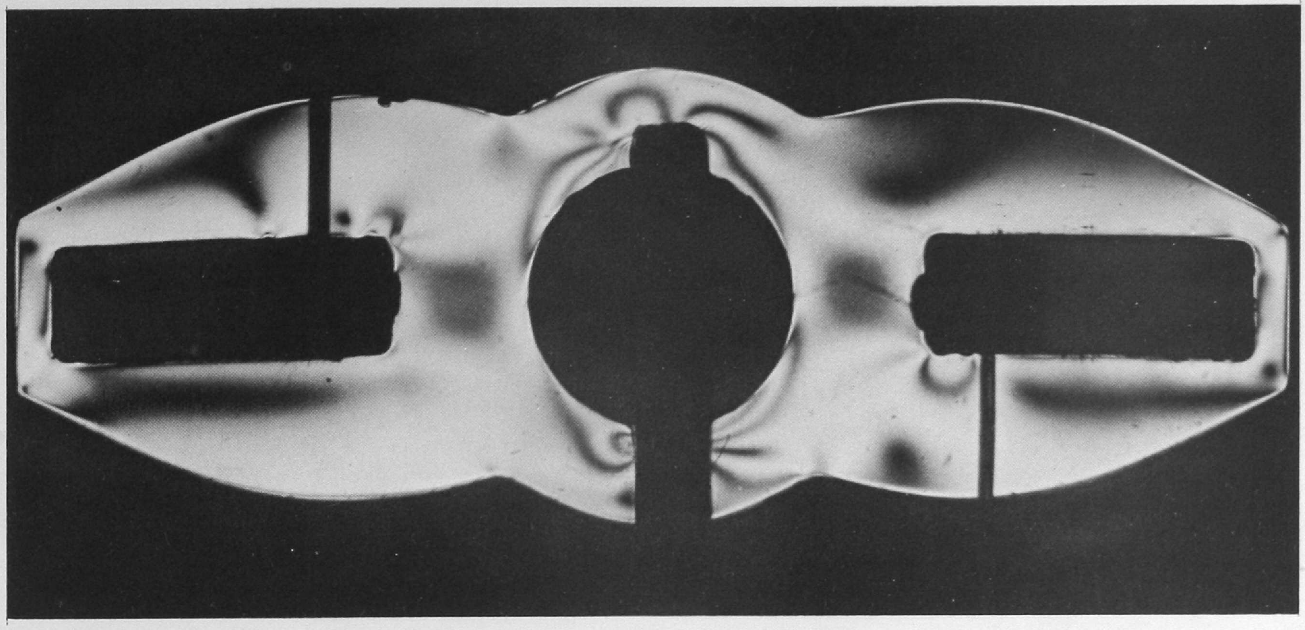
Picture 3: Isochromatics in model with torsional load of 25 lbs. in each arm, load applied at outer end of slot.



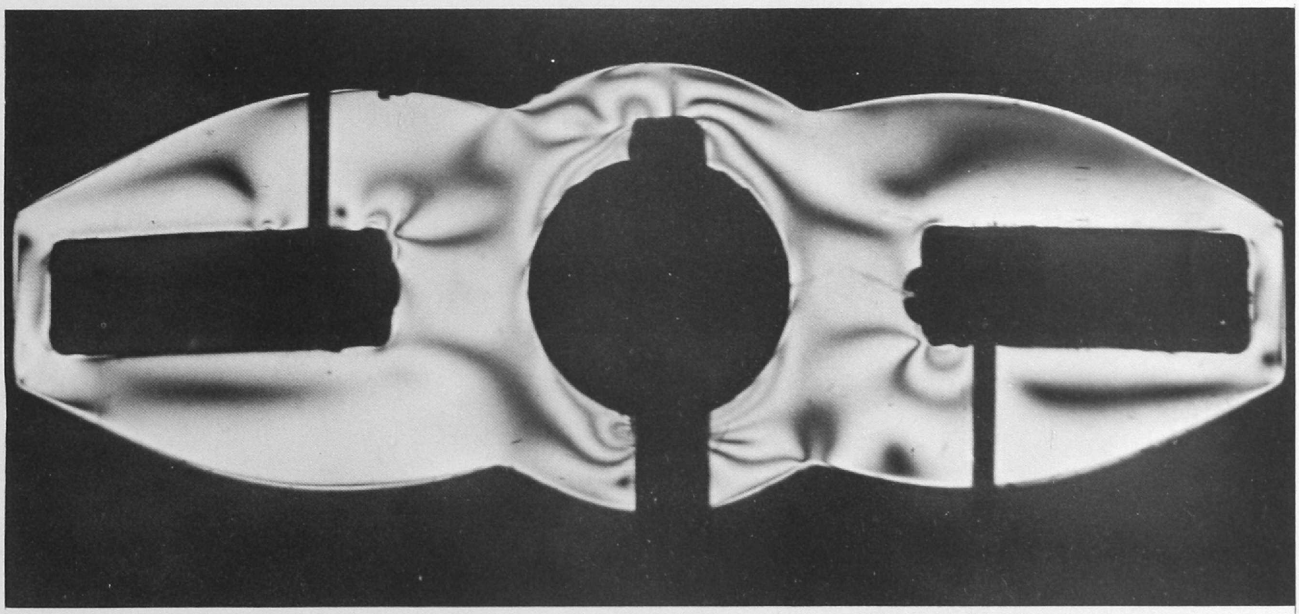
Picture 4: Isochromatics in model with torsional load of 15 lbs. in each arm, load applied at slot center.



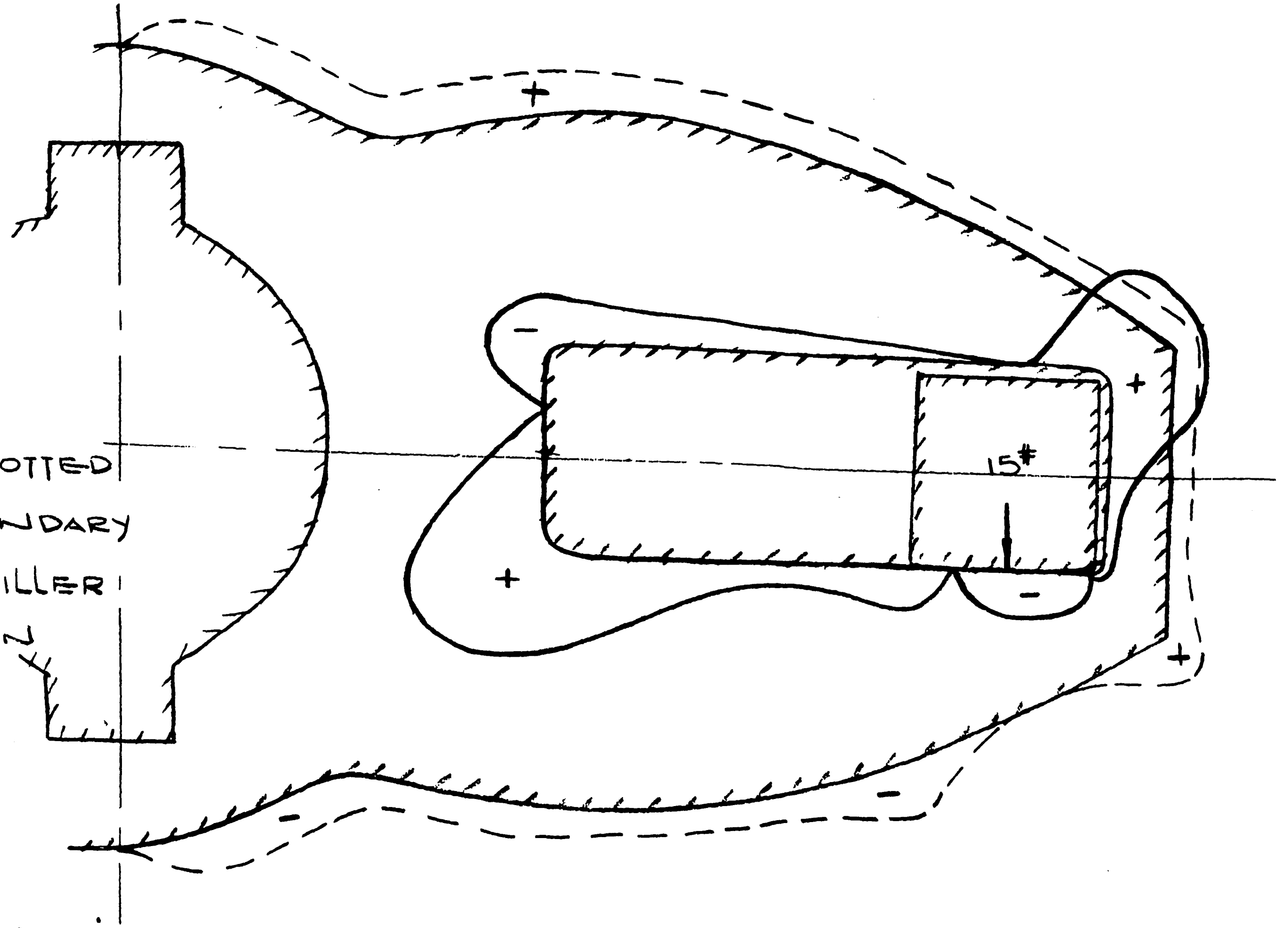
Picture 5: Isochromatics in model with torsional load of 25 lbs. in each arm, load applied at slot center.



Picture 6: Isochromatics in model with torsional load of 15 lbs. in each arm, load applied at inner end of slot.



Picture 7: Isochromatics in model with torsional load of 25 lbs. in each arm, load applied at inner end of slot.



NOTES:

1- ALL STRESSES PLOTTED NORMAL TO BOUNDARY

2- STRESSES ON TILLER SURFACE SHOWN AS ---

3- STRESSES ON SLOT SURFACE SHOWN AS —

SCALE: 1/4" = 250 psi

DIAGRAM #4 - STRESS DISTRIBUTION ON BOUNDARY FOR 15# LOAD APPLIED AT OUTSIDE OF SLOT

FOR NOTES SEE DIAGRAM Nº 3

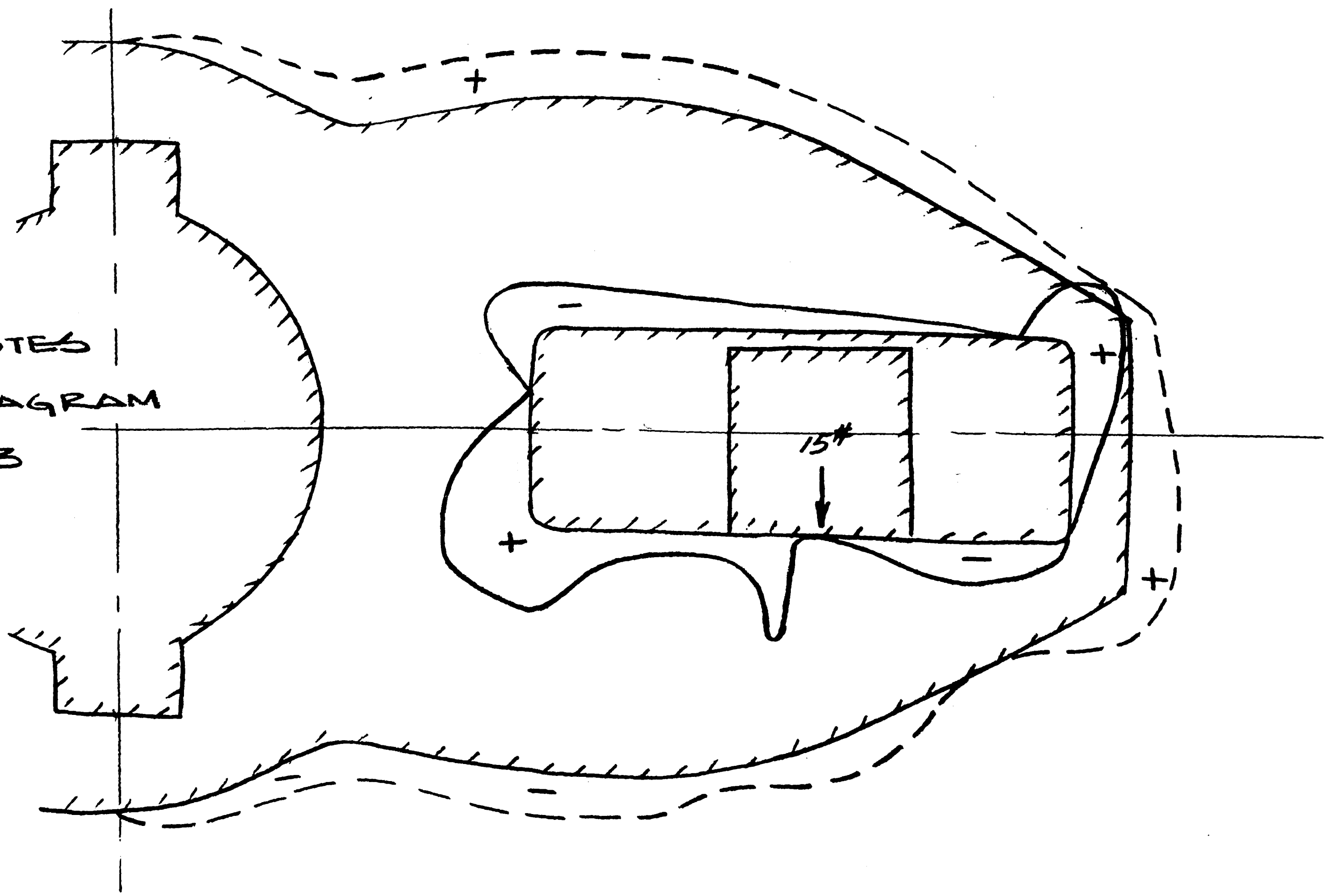
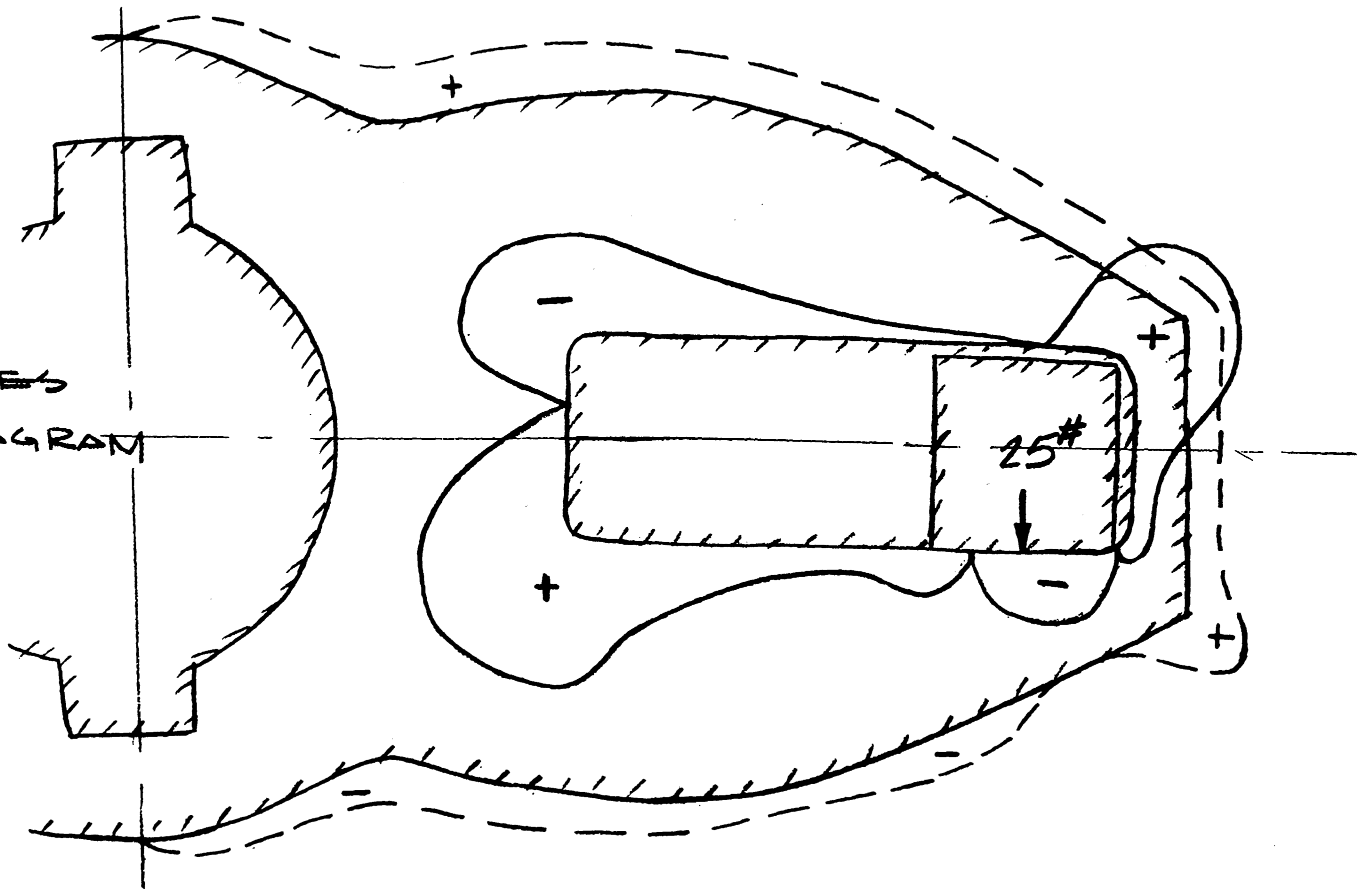


DIAGRAM #5 - STRESS DISTRIBUTION ON BOUNDARY FOR 15# LOAD APPLIED AT CENTER OF SLOT

SCALE: 1/4" = 250 psi

FOR NOTES SEE DIAGRAM No 3.



SCALE: 1/4" = 250 PSI

DIAGRAM #5. STRESS DISTRIBUTION ON BOUNDARY FOR 25# LOAD APPLIED AT OUTSIDE OF SLOT.

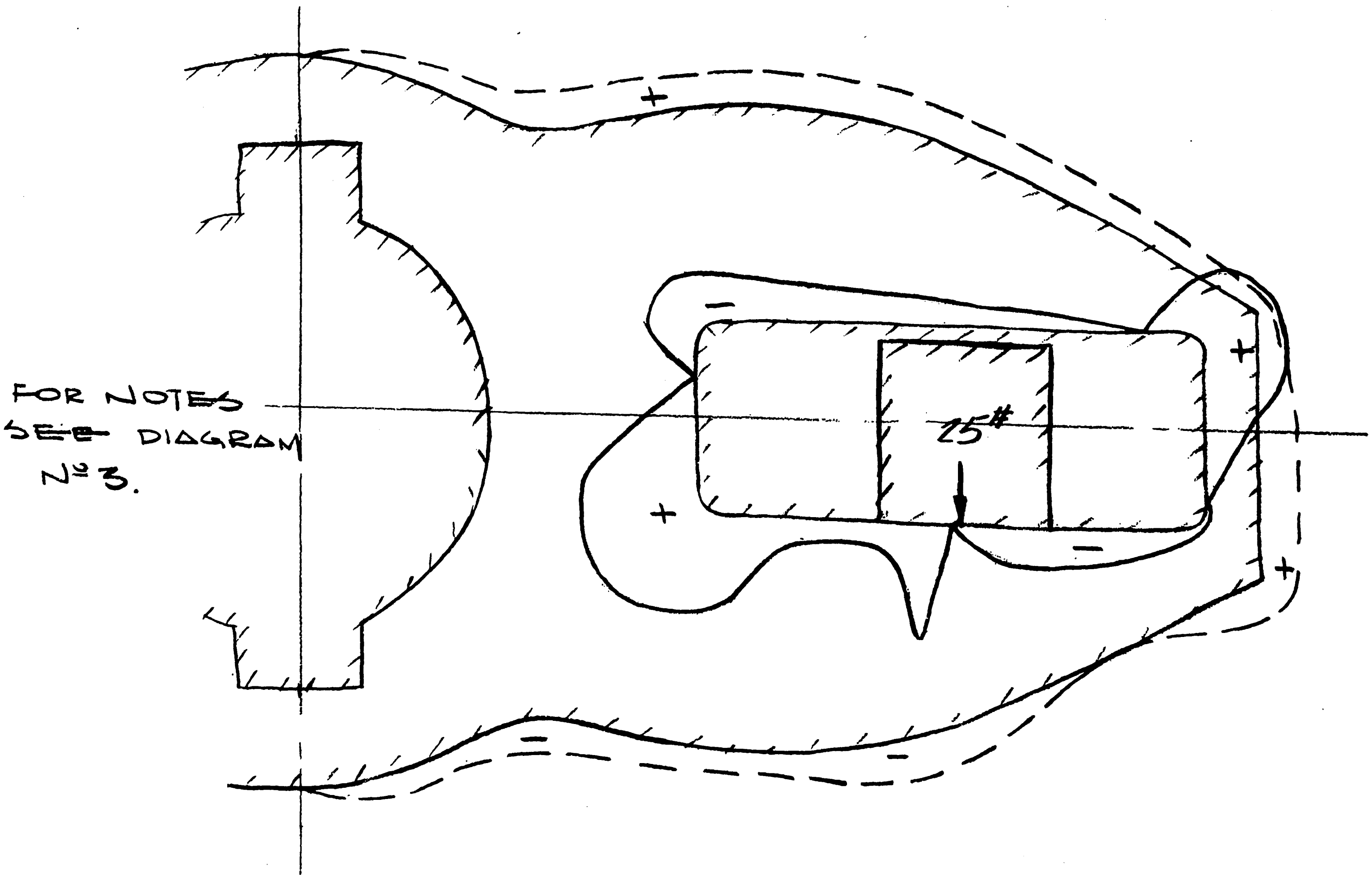


DIAGRAM #7 - STRESS DISTRIBUTION ON BOUNDARIES FOR 25# LOAD APPLIED AT CENTER OF SLOT.

SCALE: 1/4" = 250 psi

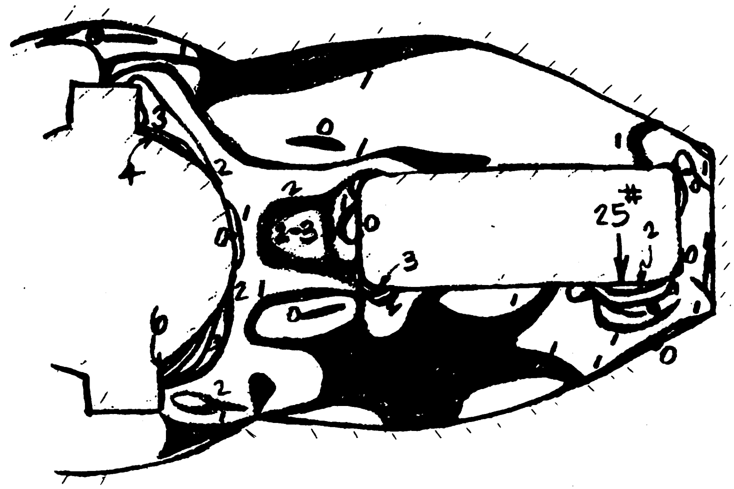


DIAGRAM #8 - INTERPRETATION OF PHOTOELASTIC STRESS PATTERN SHOWING FRINGE ORDERS FOR 25# LOAD APPLIED AT OUTSIDE OF SLOT.

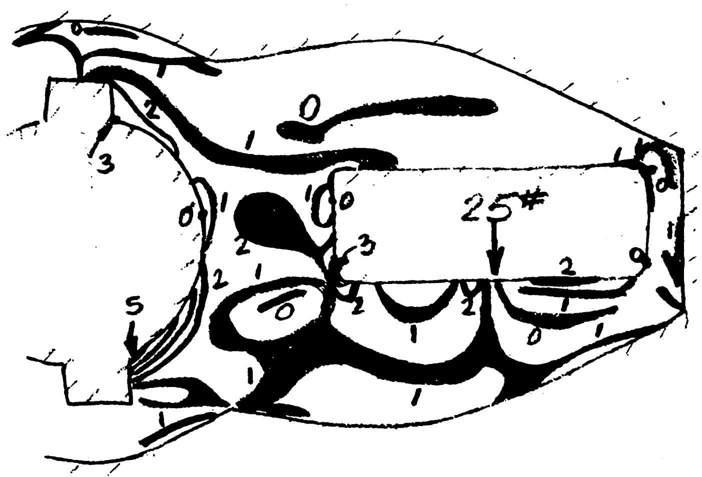


DIAGRAM #9 - INTERPRETATION OF PHOTOELASTIC STRESS PATTERN SHOWING FRINGE ORDERS FOR 25# LOAD APPLIED AT CENTER OF SLOT.

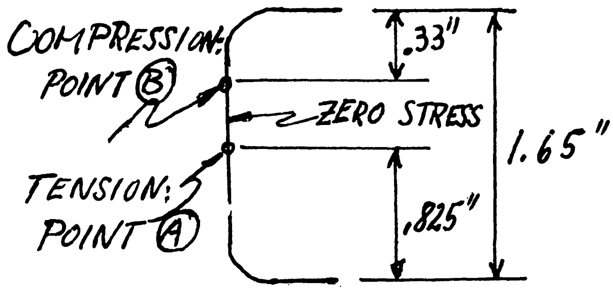
III-2. Tiller Instrumentation

The stress magnitude on the tiller boundary is in general sufficient to use resistance strain gages measuring the normal stresses in the tiller arms. It is necessary to calibrate the tiller in a laboratory test.

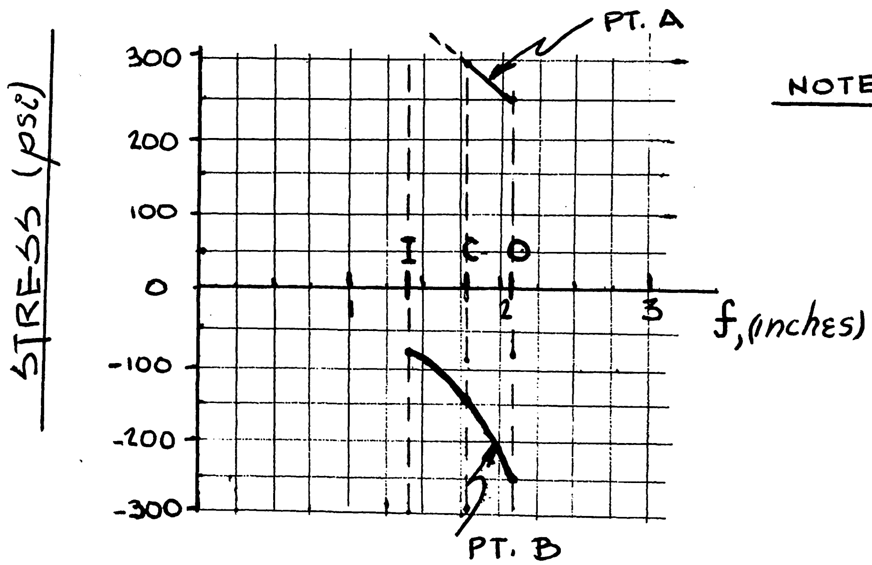
The greatest difficulty is, however, that the stresses at each single point depend on the load location. In order to reduce the labor of evaluating the test results it is reasonable to try to eliminate this influence by compensating stresses of opposite tendency. The idea is thus to add the signal caused by a stress increasing with rudder angle for a specific torque to that produced by a stress of decreasing tendency. On the basis of the photoelastic investigation a pair of such points can be sought at the inside face of the slot. Sketch 4 shows e.g. the two points A and B in the tension and compression domains respectively. In sketch 5 the stresses at these two points for a constant torque are plotted against the load location in the slot. At point A the tensile stress increases as the load moves in, at point B the compressive stress decreases in magnitude. Sketch 6 shows how the two active gages would be arranged in a bridge.

Because of the approximations in the photoelastic investigation no precise location for complete compensation can be predicted. But in practice suitable points can be found by trial and error. A series of gages can be mounted in the desired region and calibrated individually. Then it is easy to select an appropriate combination.

Promising regions are not only the neighborhood of points A and B, including the corners of the slot where the sensitivity is high, but also certain portions of the outside tiller boundary in way of the inner end of the slot, (combination of point C and D, see sketch 7). There the angular dependency seems to be relatively weak.



SKETCH 4: Location of points A and B at inner end of tiller slot



NOTES:

1- f DENOTES DISTANCE FROM ℄ OF MODEL TO POINT OF LOAD APPLICATION.

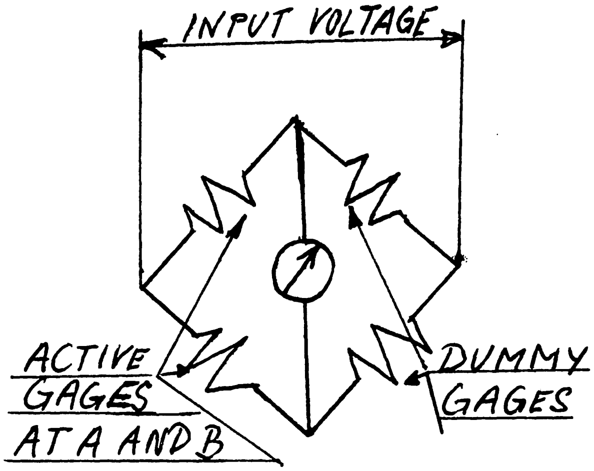
2- MODEL TORQUE IS EQUAL TO $25 \times 2 f_o$ (in-lb) = 108 in-#

3- I, C, O DENOTE LOAD APPLICATIONS AT INSIDE, CENTER, & OUTSIDE OF SLOT RESPECTIVELY.

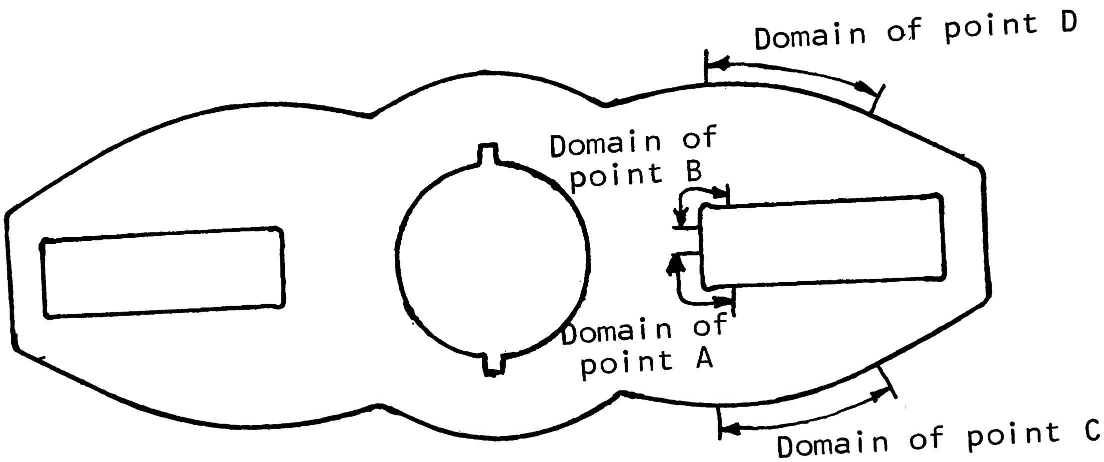
4- LEVER ARMS FOR INDICATED LOAD APPLICATIONS ARE AS FOLLOWS:

O: $f_o = 2.16"$
C: $f_c = 1.77"$
I: $f_i = 1.38"$

SKETCH 5: Variation of stresses in points A and B with point of load application for constant model torque of 108 in. lb.



SKETCH 6: Arrangement of gages in Wheatstone bridge.



SKETCH 7: Promising gage locations:
Either points in the A and B domains or points in the C and D domains.

IV. CONCLUSIONS AND RECOMMENDATIONS

Three instrumentations for rudder torque measurements have been investigated and appear to be feasible:

1. A new strain gage instrumentation of the rudder stock: In order to improve the previous resistance gage instrumentations, the following measures are recommended:
   a.) Arrange the gages above the upper bearing if the rudder stock is accessible there. This would increase the sensitivity (voltage output per unit torque) and reduce the significance of bending disturbances.
   b.) Use only one pair of active gages arranged in the neutral bending plane.
   c.) Exercise extreme care in aligning the gages in the same vertical plane on opposite sides of the stock, inclination of gage axes: 45 degrees.
   d.) To obtain absolute certainty about the correct alignment and the absence of local stress disturbances the instrumentation must be calibrated in dry dock.
2. An inductance gage instrumentation of the rudder stock: In the category of deformation measuring devices self-inductance and mutual inductance gages are the most promising ones. They are much more sensitive than resistance gages. In using them the following precautions are advisable:
   a.) Develop gages and circuits in laboratory for good zero stability and temperature compensation.
   b.) Use two gages on opposite sides of the stock above the upper bearing in the neutral bending plane.

c.) Align the gages carefully.

d.) After the above measures disturbing signals are not very likely to interfere, but the only way to make sure is also by calibration in dry dock.

3. Strain gage instrumentation of the tiller:

The stresses in many points of the tiller boundaries are more significant than those in the stock so that resistance gage pickups can be used successfully. The disadvantage is that the stresses are not directly proportional to the torque but depend on the rudder angle. Since it is not desirable to evaluate the tests on the basis of a complicated calibration curve we suggest using an arrangement that compensates against this influence. The following trial and error procedure is recommended to develop such an arrangement:

a.) Set up a tiller calibration device in laboratory.

b.) Distribute a number of gages in the interesting regions and calibrate these individually. Select a pair with output signals that add up to something approximately independent of the rudder angle.

c.) Calibrate the selected pair in a bridge circuit.

In summary, the three instrumentations which have been discussed will yield reliable results if carefully utilized. Calibration is necessary for all of them in order to check against disturbances. From this point of view the tiller is preferable because it can be calibrated in the laboratory instead of in dry dock. We also favor it in general since it is a true alternative so that it would satisfactorily check the data obtained in the previous tests by a rudder stock instrumentation.

V. ACKNOWLEDGEMENTS

The author wants to acknowledge the valuable help received from two students of the Department of Naval Architecture and Marine Engineering, The University of Michigan: Mr. Paris Genalis and Mr. Gerald M. Lazarek. He wishes to express his thanks to Mr. James Moss, head of the Ship Hydrodynamics Laboratory and his staff for their assistance during the model tests and in preparing the report.