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UNITED STATES EXPERIMENTAL MODEL BASIN

NAVY YARD, WASHINGTON, D.C.

ELASTIC CHARACTERISTICS OF FLEET OILERS

BY LIEUT. W. P. ROOP, (CC), U.S.N.

EXPERIMENTAL MODEL BASIN
ERECTED 1898
BUREAU OF
CONSTRUCTION AND REPAIR
NAVY DEPARTMENT

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JUNE 1930

REPORT NO. 260

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I. SUMMARY.

The fleet oilers, six in number, have acquired a reputation for weakness, based on the facts of leakage, vibration at critical speed, and buckling in the deck. The structural design of the vessels has been modified so as to increase their stiffness without curing the trouble, and conditions were so serious on one that she was withdrawn from service.

It now appears that the primary source of trouble lies in the unbalanced reciprocating engines: Evidence for this view is found in the facts that:

(a) The natural frequencies of the ships are entirely different from the critical engine speeds, in some cases higher, in others lower.

(b) The ship with the stiffest hull shows the same troubles as the others with leakage and buckling, and the ship that has had the least trouble is no stiffer than the one that has had the most.

(c) Strain gages show pulsating stresses in unison with engines which are superposed on the unavoidable stress due to cargo.

(d) Balancing the U.S.S. CUYAMA'S port engine has practically eliminated vibrations at all speeds when run with starboard engine stopped.

It is not consistent with the known facts to call these vessels weak. No major rupture has occurred in 9 to 14 years of service and no incipient rupture exists at present. Although the nominal designed stress lies in the upper range of what has been considered good practice, there is no evidence that the limit of strength is closely approached. The actual margin of strength existing will not, however, be definitely known until known bending moments sufficient to cause stresses exceeding nominal values are applied. Uncertainty exists also as to the bending loads actually occurring and it is unknown whether nominal values are actually exceeded in service.

Experimental studies have now been made whose course was mainly directed to the determination of three quantities: The effective moment of inertia of section, effective modulus of elasticity of the assembled structure, and effective loads occurring in a seaway. While high precision was not attained, the results tend to confirm the validity of the standard procedure for comparing hull designs through calculation of nominal stresses.

Returning, finally, to the question of strength, some qualitative considerations detailed below will serve to suggest how incomplete and obscure our information really is about failure of a complex structure through instability. Until more light is shed in this direction rational design of structure which must resist compression will remain impossible.

1. Stiffness.

On the U.S.S. CUYAMA, corresponding values of bending moment and deflection of the ship girder give values of the product of sectional moment of inertia and effective modulus, EI. Of 20 pairs of such values the average is 9.2×10^{12}, lb. ft.^{2}, with a probable error $\pm 1\frac{1}{2}$ percent. This gives a value of 3.29 for the dimensionless constant* in Schlick's formula:

$$n = \text{const}\sqrt{\frac{gEI}{DL^3}}$$

n being frequency of free vibration of the structure, D the displacement and L the length of the ship, and g the acceleration of gravity, all in absolute units. This value is lower than that used by Schlick, (3.80), which might be attributed to a weight curve flatter** than usual, the engines being aft in this type of vessel, or to the effective value of E being lower** than contemplated in German practice. The value for a uniform bar with 2 nodes is 3.56.

2. Effective Section Modulus.

Strain gage data were taken at about 150 stations and extensometer data at 3 stations covering 40 conditions of known bending moment. The average value of section modulus obtained is 21,000 in.^{2} ft. $\pm$ 1.2 percent which is available for comparison with calculations made in design. It is a little less than that obtained by standard procedure in the Bureau of C and R.

3. Effective Modulus of Elasticity.

As strain gage data were obtained only for the compression side of the ship girder the position of the neutral axis can not be inferred. Accepting the calculated position, the value of the modulus of elasticity is found to be 24×10^6 lb./in.^{2} which is somewhat greater than the accepted value for assembled structure, 22.4×10^6 lb./in.^{2}

4. Nature and Distribution of Buckling.

When continued under heavy sagging load and subjected also to the action of waves and of vibration due to the engines, the main deck developed wrinkles which

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*In English usage, this formula is written

$$N = c\sqrt{\frac{I}{D\,L^3}}\ .$$

c is thus not dimensionless, but includes under the radical the factors g and E. Accepting 32.2 ft./sec^2 for g, 22.4×10^6 #/in^2 for E, and expressing N in cycles per minute, I in ft.^{2}in.^{2}, D in tons, and L in feet, the conversion factor by which the dimensionless constant must be multiplied to obtain the English constant is 3.4×10^4.

**Frequency calculated by the German constant is higher than the observed value.

became progressively more pronounced until partially removed by hogging moments in the light condition. These wrinkles extended transversely, with nodes at frames, and plating bulging up and down in alternate frames. Figures 2a and 2b show the nature of this effect. In Figure 3 are indicated the areas where it is most pronounced. The buckling appears to originate in plating whose continuity is interrupted by hatches, to be limited by the stiffening effect of longitudinal members, but eventually to tend to spread into stringer plates. In particular the inboard after corner of the starboard stringer plate extending between frames 98½ and 110½ was found in static tests to show very little departure from uniformity in stress distribution, but four months later it was buckled as shown in Figure 4 and in the meantime tests under pulsating load at sea had shown local stress reversals, e.g., tensile stress occurring under compressive load. This is the only case in which increase in buckling with service was observed instrumentally but in other cases similar effects were found in spots where nothing had been noticed during the rather close inspection that occurred during the static tests.

5. Bending Load in a Seaway.

Continuous record of stress at 3 stations was obtained covering 50 days of steaming. Detailed reduction of these data has had to be deferred, but an approximate result obtained under the most severe conditions encountered shows compressive stresses of 13,000 lb./in^2 due to waves estimated to be 300 feet long and 15 feet high. This is superposed on the compressive stress in still water; this amounted to about 9,000 lb./in^2, the ship being in ballast at the time.

6. Conclusion.

Aside from the discomfort entailed by the excessive vibration of the ship, the only objectionable feature of her design for strength lies in the possibility that under emergency bending loads buckling might be induced in the stringer plates by adjacent plating. To materially alter the natural frequency of the hull by adding new material to the working section would be costly, might have an effect contrary to that desired, and is not recommended. If the vibration can be stopped at its origin, addition of a very moderate amount of intercostal stiffening locally would serve to prevent buckling in plating not forming part of the effective section, thus eliminating all danger. This alteration is recommended.

II. DETAILS OF THE TESTS.

1. Application of Bending Moments.

By filling tanks with water, bending moments ranging between -120,000 and +125,000 ft-tons were obtained. This is only a little over half the bending load the ship was designed to carry. Static tests were continued at every opportunity during cargo shifts, though the load increments thus obtainable were naturally even less. In all, 40 different conditions were thus obtained.

2. Flexure and Stress Measurements in Static Tests.

Flexure of the ship as a whole was observed by means of a telescope in a fixed position forward through which level-rod readings were taken at 7 points, symmetrically arranged about the 9th of 20 stations numbered from forward aft. Flexure stations were at frames 48, 60, 79, 91, 103, 123, 136, each lying over a transverse bulkhead. Such a series of observations was made on each side of the ship for each of 32 conditions of loading, combined for purposes of analysis into 20 pairs. The early data were all taken at close intervals, and increments were reckoned from the initial condition. Later flexure data consisted of increment pairs taken before and after important shifts of cargo. The temperature effects were eliminated as far as possible by choosing suitable time for the observation: On the whole the weather was favorable and check of readings under identical load but different weather indicates that temperature effects were not predominant. Nevertheless errors from this source undoubtedly act to produce a scattering of spots which makes precise work impossible.

At 16 conditions strain gage data were taken on the port and starboard stringer plates extending from Frame 98½ to 110½ and on the center line plate extending from Frame 93½ to 105½. Stations were arranged in three parallel fore and aft rows on each plate, at each frame and half frame, making a total of 216 stations. On the starboard plate a fairly full set of data were obtained with a Berry gage of the old type. On the other two plates the number of stations was reduced to 7 in each row and the port row on the center plate was omitted entirely. The port plate had a Whittemore gage and the center plate had a Whittemore and a new type Berry.

3. Extensometers and Flexuremeters.

On each of the 3 deck plates mentioned was mounted an extensometer, the general arrangement of which is shown in Figure 6. It consists of a 6-inch pipe rigidly secured to the plate near one end and suspended to permit free motion parallel to the plate at the other. The relative motion is measured by Belleview gages. Fittings are also provided for Ames dials giving visual data. The gage end of the extensometer is enclosed in a watertight box into which the pipe is admitted by means of a rubber diaphragm.

An attempt was also made to record data on flexure of the ship as a whole by fitting enclosed Belleview gages at the end of a lever rigidly secured to the bridge house and extending 70 feet aft, where relative vertical motion between the end of the lever and the deck occurred. The lever consisted of 3 light wire ropes converging toward the gage housing and placed under tension by means of a spring sufficient to assure a high degree of fixation with respect to the bridge structure of the point of convergence. A similar flexuremeter worked off the king posts aft which were suitably stayed to take the load. On account of its great length (180 feet) the after rig was deficient in rigidity. A third similar gage station was rigged between two wires so placed diagonally as to give relative vertical motion due to torsional deformation of the vessel.

There were thus in all six Belleview gage stations: Port, center, and starboard extensometers, forward and after flexuremeters, and torsion meter.

4. Belleview Gages.

These are a development of the "step by step" gage used in ballistic work, and consist simply of an enclosed coil of fine wire along which a contactor moves axially, picking off a potential intermediate to those established at the ends of the coil. Steps of about 5/1000 inch with resistance increments of about 1 ohm carrying current of .02 ampere on continuous duty were used. The contactor was carried by a plunger and fully waterproof enclosure was accomplished by means of a rubber diaphragm. The gages functioned perfectly without a single casualty. Details of these gages are exhibited in Figures 5a and 5b.

In this specific application the plunger was not actuated by a spring, but was held by friction in the furthest position to which it had been pushed, until the attached solenoid was energized, when the plunger returned for a new start. The gage thus measured the successive maxima occurring since the solenoid was last energized. At each station two such gages were used, one giving maximum motion in tension and the other in compression.

Solenoids were actuated at intervals of ten minutes for a period of 30 seconds. During the period when solenoids were energized the two paired gages at each station moved oppositely; the mean position thus established serves as a null point from which deformations in the two directions are measured. This null point will wander due to shifts of temperature and weights, but deformations taken from it give the net effects of the seaway, which are desired.

5. Recorders.

Data from the 12 Belleview gages, in the form of varying voltage, were automatically recorded by 12 recording voltmeters, six of which were furnished by the Brown Instrument Company and six by the Leeds and Northrup Company. As instruments of this type have not been extensively used at sea some details as to their arrangement and performance will be given.

The Brown recorder consists of a voltmeter of usual type with a moving coil supported by jewelled pivots. A flexible pointer swings over a strip of ruled paper which is fed at a given rate and at intervals of 7½ seconds the pointer is forced against the paper with an ink ribbon interposed so as to give a spot record of the position of the pointer. The instruments were specially constructed for this service, with high resistance coils to eliminate error from contact resistance, and to give high torque in the galvanometer coil, permitting use of heavy hair springs. In view of the service on a moving platform special precautions were taken in balancing the galvanometer system, and balancing was initially satisfactory as shown by uniform reading on open circuit. In course of service looseness developed in pivots but this was easily remedied. The paper feed functioned perfectly. Some objection may be made to the intermittent nature of the record which does not lend itself to quantities subject to quick fluctuations. For the maxima given by Belleview gages this system gives excellent records but doubt may exist as to the interpretation of the spots establishing the null position. As actually used, the recorders were paired to correspond with the gages. so that the spots from the two gages were made simultaneously. During the interval when solenoids were energized, the sum of the readings of pairs of spots should give the same value, as increase in one gage accompanied decrease in the other.

An enlarged copy of a record obtained from the Brown recorders is shown in Figure 7a. It will be seen that the detailed tracing of a null line from single spots is hardly practicable, nor is it considered necessary. Comparison for a short period of a null line taken from single spots with a fair curve drawn through the estimated centers of successive groups of scattered spots shows that the latter procedure will meet all practical requirements.

The Leeds and Northrup recorders are more elaborate, consisting of self-balancing potentiometers. The moving pointer is driven by a motor toward the point at which the internal potential is balanced against that which is to be measured, in steps two seconds long. But with a quickly fluctuating potential the pointer never catches up so that the establishment of the null curve encounters the same theoretical difficulties as in the Brown recorder. In fairness to both instruments it should be noted that they are not designed to replace oscillographs, and both probably follow electric fluctuations about as closely as is practicable without resort to photographic methods. The ribbon suspensions of the Leeds and Northrup recorders functioned perfectly without casualty and the balancing of the galvanometer system was satisfactory and permanent. This type of recorder is immune to error from contact resistance, though in this case there is no reason for suspecting trouble from this source. Paper feed caused some trouble and the capillary ink supply is inferior to the ribbon system.

Experience with these recorders demonstrates that both may be used satisfactorily at sea if given reasonable care and of the two the Brown recorder makes less demand on personnel for maintenance in effective operating condition.

6. Arrangement and Adjustment of Recording Gear.

Figure 8 shows in outline the electric connections for the gear required for a single station. Figure 9 gives the detailed wiring diagram for the complete installation. Adjustment is not complete until records from all stations can be taken continuously through a full cycle of loading conditions with due allowance of space on the charts for calibration and stresses due to seaway. The following adjustments were necessary:

(a) Set gages so as to cover the limits of motion to be recorded. Since the bending moments corresponding to different conditions were not known at the time the tests were made, this was effected only by trial and error, and as readjustments were necessary, this requirement was not met in early tests. In anticipation of this condition a pair of small rheostats R_1A_1, R_2A_2, was placed in series with each gage coil to assist in bringing recorders to desired position on scale and make it approximately direct reading. For various reasons this was not accomplished. In a more permanent installation adjusting coils for this purpose would be set once for all. In the present instance they were not touched during voyages; unfortunately changing one affects readings of all the gages which draw current from a single source.

(b) Zero of potential (ground) from which contactor voltage on gage coil is measured is obtained from a high resistance R_1R_2 (750 ohms) in parallel with the gage coils. Sliding the ground contactor on this coil will move all recorder pointers together but will not alter sensitiveness so as to require recalibration.

(c) An adjustable resistance is provided in series with the storage battery by means of which the terminal voltage on the gage coils can be controlled. The sensitiveness of the gages can thus be increased to obtain more open readings when desired, but this requires recalibration after each resetting.

(d) In calibrating, plugs of calipered thickness are inserted between gage plungers and pins on which they bear and the corresponding displacements noted on recorder. In this way all factors entering into the sensitiveness of the gages are covered directly and these need not be separately accounted for.

(e) Periodical resetting of the potentiometer current in the Leeds and Northrup recorders is accomplished by comparison with a standard cell included in the case with the recorder.

Complete interchangeability of units was provided for by bringing all connections through a central block. This proved to be an indispensible feature, but in less extensive and more permanent installation might well be omitted. Figure 10 shows arrangement of recorders. They were placed in the emergency cabin which was too small to permit photographing the entire layout on one plate.

7. Other Details.

Exploration of deck in buckled areas was made underway by two methods. Two strain gages were used in the first, not for estimating actual stresses, but for obtaining relative values which would indicate regions highly stressed. At operating speeds a strain gage applied almost anywhere on the ship would show pulsations in unison with the engines and these would often be rather constant in amplitude over long periods of time. By keeping one gage in a fixed location where it could be easily seen the second could be moved to a variety of nearby locations and the readings of the two roughly compared. Somewhat similar comparisons were made with respect to panting of the deck by means of 2 dial gages supported over adjoining mid-frame spaces by a wooden frame which was itself supported at the next mid-frames forward and aft, as shown in Figure 11. This arrangement was rather sensitive to the type of panting occurring. Motion of the two dials or of the two strain gages in opposite phase in adjoining frame spaces was taken as a positive indication of buckled condition. Although this was not very much more sensitive to buckling than the eye or the feet, it provided data independent of the sensations of the observer.

Visual inspection was made for evidence of structural deficiencies but none were found. On the other hand creaks could be heard, and felt through the feet, which strongly suggested slip in riveted joints and were easily distinguished from the numerous sounds due to slipping of pipes in brackets, and the like. Such creaks were heard even in static tests at the heaviest loads.

Observations of relative amplitudes of vibration at different engine speeds were made by means of a Sperry pallograph, which was also used for determining the natural frequency of vibration of the ship with engines stopped. Such vibrations were started by dropping an anchor through a scope of about four links and bringing it up sharply on a chain stopper. Observations of the state of the sea, force of the wind, etc., were taken from the ship's log. At the times of roughest sea independent estimates were also made and for several typical conditions motion pictures were obtained showing the state of the sea and the rolling and pitching of the ship.

III. ANALYSIS OF THE RESULTS.

1. Flexure.

Table I gives values of stiffness obtained from flexure under known bending moment. Moments are increments based on observed drafts and weights, reduced by the procedure outlined in Appendix B. Flexures were obtained as follows:

The seven stations at which level rod data were taken are symmetrically distributed about Frame 91, the outer pair being at a distance of 88 frames apart, the intermediate ones at 61, and the closest ones at 24 frames. Deflection is that usually considered in bending of beams, being the departure of the point at mid-length from the position occupied before application of the load. The amount of this deflection depends, of course, on the span considered; if the bending moment were uniform over the whole span the ratio of deflection to square of span would have the same value at all spans. Actually the bending moment departs considerably from uniformity with a span of 88 frames, more in some conditions of loading than in others. Deflections on a short span of 24 frames are so small that the measured values fluctuate rather irregularly. It was therefore necessary to determine the course of the bending moment curve over the middle half of the ship's length for each of the 40 conditions of loading. This is a large order, but by the use of approximations discussed in detail in Appendix B, it was accomplished. It is thus possible to associate with each measured value of flexure an average bending moment taken over the corresponding span. Assuming that the flexure is equal to that under a uniform moment equal to this average, we may now apply the formula

$$f = \frac{M}{EI}\frac{l^2}{8}$$

to the determination of the coefficient of stiffness EI.

2. Stress.

Measurements of stress under static load served two purposes: to calibrate the extensometers, so as to permit conversion of recorder data obtained at sea into equivalent loads imposed by a given seaway; and to separate the two factors E and I in the coefficient of stiffness. Incidentally the data shed some light on the uniformity of stress distribution over the three plates involved, but the detection of stress concentrations was not a major objective; the plates selected were chosen because they were thought to be relatively free from concentrations.

Comparison between stress data as given by strain gages and by extensometers is exhibited in Table II. As the load increment and other conditions are identical in each case, the observations being made simultaneously, errors due to temperature differences and the like are eliminated. The result shows that the extensometer data may be accepted as giving mean stress values within limits as close as the strain gages will work. Bending moment corresponding to unit stress is shown in Table III in which extensometer and strain gage data are shown separately. The ratio of these quantities is equal to the section modulus of the ship girder, the average value of which is 2.1×10^4 in.^{2} ft. When compared with the nominal value of 2.18×10^4 obtained in calculations made when the ship was designed, this indicates that in this case the choice of members for inclusion in the girder section was such as to account with approximate correctness for the various uncertainties involved.

By combining the value of section modulus with that of coefficient of stiffness EI, an effective value may be obtained for Young's modulus; this requires knowing the location of the neutral axis and as data obtained were confined to the main deck, only the calculated position is available.

In order to eliminate uncertainties as to bending moment values, temperature disturbances, and the like, the procedure is to combine stress and flexure data for the same load increments. From the formula for deflection under a uniform bending moment,

$$E = \frac{\frac{M}{I}}{8\frac{f}{l^2}}$$

By substituting for $\frac{M}{I}$ its value in terms of stress σ and distance from neutral axis y, we find

$$yE = \frac{\sigma}{8\frac{f}{l^2}}$$

Since the stress values all refer to the midship section an appropriate value of $\frac{f}{l^2}$ must be obtained. To accomplish this it is necessary to take the mean values referred to above for the ratio of deflection to mean bending moment taken over the span covered in the deflection measurement; this ratio multiplied by the local value of bending moment at the midship section gives a local value of $\frac{f}{l^2}$. It will be noted that only the ratio of local to mean bending moment is involved, and this is not affected by circumstances altering all bending moments alike.

Table IV gives values of yE obtained in this way, and by use of the value of y from original design a mean value for E is obtained which is somewhat above that commonly accepted for assembled structures. In spite of the buckled condition of part of the deck not directly sharing the bending load we may conclude that the strength members remain intact. The only risk is that of strength members ultimately being forced out of position by excessive buckling of adjoining plating.

Systematic variations in the stress data were sought by making the following comparisons:

(a) at frames and between frames.

(b) at odd and even half frames.

(c) at varying distance from longitudinal member, shear strake or CL bulkhead.

(d) at sides and center of deck.

Differences in (a) and (b) would indicate incipient buckling. (c) indicates lateral transmission of load through shear. (d) indicates concentration of load on the CL strake of deck plating, as high stress under a given bending moment corresponds to low section modulus.

These distribution data, which are summarized in detail in Table V, are on the whole rather inconclusive except as to (d) which shows definitely higher stress in the CL strake than in the stringer plates.

3. Load.

Data on recorded stresses and simultaneous log entries will be communicated separately. Their reduction is a rather long task for which opportunity has not yet occurred.

The problem of specifying load which a ship is to be designed to withstand cannot be fully solved, however, until an improved measure of state of the sea becomes available. Possibly the numerical measure of state of the sea may ultimately be put in terms of its effect on a calibrated ship as determined by stresses recorded somewhat as in the present tests.

4. Natural Frequency.

In Table VI are exhibited data obtained from four vessels on coefficient of stiffness. This was determined from natural frequency of vibration on the last three, by application of Schlick's formula, making use of the coefficient found by static flexure tests on the first. The weight distribution was practically identical in all four vessels so far as hull structure goes, but the value of the constant might be somewhat affected by the distribution of cargo. However the allowance for cargo made by the displacement term in the formula may be trusted to account for the major part of the effect and the values obtained for EI may be taken to show that three of the vessels have substantially equivalent stiffness while the U.S.S. NECHES has about 35 percent excess over the others. This is due to additional structure, not in the U.S.S. CUYAMA and U.S.S. BRAZOS, added for the purpose of curing the alleged weakness of the hull; for this purpose it was quite ineffective, though the effect on hull stiffness was very marked. We can only conclude that the weakness was only apparent and not due to structural deficiencies.

This is confirmed by the fact that the U.S.S. SALINAS has the least stiffness among the four ships. Vibration is reported to be about equally severe and buckling of the deck is observed to be equally marked in the first three vessels and almost wholly absent in the U.S.S. SALINAS. This can not be attributed to her longitudinal framing unless it be supposed that these members somehow prevent vibration and buckling without affecting natural frequency.

The critical speeds differ in each case from the observed natural frequencies, nor is any possibility apparent that frequencies other than the fundamental play a part. Critical speeds vary somewhat with loading of the ship. The first three vessels, all similar, report a critical range of about 8 turns, with maximum at 71 to 73 with full cargo and 76 to 78 in ballast. Variation in natural frequency with displacement is at least twice as great as this. Critical speed is believed to depend on immersion of screws and hence on trim rather than on displacement of the ship as a whole but this cannot at present be definitely established. The critical speeds given in Table VI are for the conditions at which the frequencies were determined.

5. Local Action.

In Figure 3 are outlined the areas of greatest buckling, including only those in which the deformation was apparent to the eye. Photographs appended show the nature of this effect on the U.S.S.CUYAMA and on other vessels as well. The most pronounced wrinkles occured near Station 9, frames 86 to 96, in line with the main cargo hatches. The longitudinal bulkhead smoothed them out locally for obvious reasons. The deep hatch coamings seemed to keep the wrinkles from approaching too closely forward and aft and their influence extended a long way transversely. Not so pronounced is the influence of the light stringer just outboard the summer tank hatches. Relative stresses due to pulsations from the engines are plotted in Figure 12 for the area surrounding the outboard hatch at Frames 90 to 92. Plus on this plot represents stresses in phase with the load, minus represents reversed phase, compressive stress accompanying tensile load and vice versa, as occurs on the convex face of a deep wrinkle. Motion pictures showing this reversal of phase were also obtained. No very definite conclusions can be drawn from these observations. It seems impossible to reconcile the facts with the usual point of view according to which a hatch is a soft spot. Plating adjoining a hatch forward and aft is far from refusing load entirely. And how can plating between hatches be buckled when that in the adjoining CL strake is not unless the hatches are effectively hard spots? In any event the concentration occurring in the CL strake is due to the lack of support from the deeply buckled plate immediately adjoining. The stringer plates are not equally affected because the outboard hatches are further away and more widely spaced. From this point of view the doublers fitted around the hatches on the U.S.S. NECHES were not a help but the contrary; perhaps they will explain in part why buckling is severe in spite of excessive stiffness of the ship as a whole.

If these hatches are really hard spots we ought to be able to take advantage of the situation; by providing enough local stiffening to stop buckling and make plating between hatches stand up to its load reduction in scantlings elsewhere should become possible. Even if these hatches were not provided with deep coamings, and as simple holes could be nothing else than soft spots, the load would still be carried around them by lateral transfer provided the plating were suitably stiffened against the shear and compression that must accompany such a lateral transfer of load.

IV. APPLICATION TO ELASTIC DESIGN.

We may regard this instance of buckling as having originated in actions of the sort just described due to the combination of high though acceptable working stresses and additional unforseen loads from unbalanced engines. The latter accentuated the wrinkling tendency by their highly repetitive action wherever local concentrations made wrinkling incipient. If the pulsating load had been absent or if exactly the right places had been suitably stiffened the wrinkling might never have started. But it did start and under the cumulative action of 60 to 90 impulses per minute rapidly spread until checked by the main strength members of the hull on which its action is much slower though still not zero. It is true that critical speeds were generally avoided, but it is also true that in this way only the maximum effect was avoided. At speeds differing by 11 turns per minute from critical, pulsating stresses were still well marked; the strain gage need only be brought into firm contact with plating anywhere to start off immediately like a clock, and watching it gave a vivid sense of the liveliness of the action.

The elimination of this unnecessary excess load seems obvious as the first step toward a cure of the trouble. The facts illustrate forcibly, however, our ignorance of the mode of action of typical ship structure under compressive load and subject to unstable failure. It is probable that by following up the clues obtained in this instance important savings can be effected in cases where compression is the controlling element. This will require extensive systematic studies of the action of plating under compression and while such studies have already been started, it will take a long time to bring them to definite conclusions.

More immediately applicable is the thought that the use of the word "strength" in elastic design is anomalous. We know as yet neither the loads which a ship must withstand nor the capacity of a given design for withstanding them. Our standardized process of design for "strength" is actually design for stiffness.

It is necessary to be perfectly clear as to the difference between strength and stiffness. Strength is resistance to rupture, stiffness is resistance to deformation. In beams under bending load, strength is expressed as section modulus, measuring load which can be carried on a given stress; stiffness is expressed as product of sectional moment of inertia and effective modulus of elasticity, measuring load which can be carried on a given deflection. Briefly:

$$\text{Section Modulus} = \frac{I}{y} = \frac{\text{Bending Moment}}{\text{Stress}}$$

$$\text{Coefficient of Stiffness} = EI = \frac{\text{Bending Moment}}{\text{Deflection}}$$

Reduction of a ship to a state of failure does not imply rupture of the strength members but excessive deflections such as lead to uncontrollable leakage. Resistance to failure depends upon the capacity of a ship for maintaining its form as a whole in spite of local damage due either to external agency or to concentrations incident to the detailed construction.

Sectional moment of inertia enters as a factor in both expressions for strength and stiffness, and the difference between the two lies in the other quantities concerned, depth of girder and effective modulus of elasticity.

Thus when emphasis is placed on strength, small depth of girder does not appear to be as detrimental as if stiffness were regarded as of primary significance. And design for strength alone leaves out of consideration altogether the factor of effective modulus of elasticity. Thus both welded structure, which has less slip at joints than riveted structure, and suitably stiffened plating in compression, which is more effective than unstiffened plating, present distinct advantages not adequately accounted for by conventional design for "strength".

The rational way to take cognizance of these effects is by adopting an appropriate value for modulus of elasticity. To attempt to make allowances by adjustment of working stresses is entirely false.

Strength is subject in practice only to calculation as no method has ever been found practicable for determining it experimentally. Evolution of ideas in elastic design has undoubtedly been retarded by the fact that their correctness has had only the indirect confirmation obtainable by a study of casualties. As these ideas are largely inherited from bridge builders whose structures differ radically from ships, it is small wonder that theory still serves shipbuilders very little and design remains deeply empirical. The stress concentrations which are so serious theoretically do not always cause trouble practically; yet we are still without a clear cut criterion by which to separate harmless concentrations from the others.

Stiffness, on the other hand, can be experimentally determined directly or by means of the natural frequency. This latter test is so simple that it should be made routine procedure on every ship when new, and at intervals during its life. In this way acceptable limits could be drawn from experience and made applicable to new design. It is probable that ways would thus be found of making large savings in weight by adopting for general use the lowest value found in satisfactory service.

Placing the primary emphasis on stiffness need not result, as it conceivably might, in a ship stiff enough but not strong enough. If it should happen that local overloading produced effects appearing dangerous, local use of high duty materials would increase the strength limit without altering stiffness or weight.

Finally, while the elimination of an exciter of vibrations may in this case cure the trouble, the complete removal of periodic loads from a screw-driven ship is impossible. While especial uncertainty attaches to permissible values of stress applied repetitively, the stiffness of the structure is exactly the feature which controls the response to such loads. Better information about actual stiffness is necessary before control of vibration can be secured.

"Elastic Design" should therefore replace the phrase "Design for Strength". EI should replace section modulus and deflection should replace stress as the primary elements in elastic design. And the accumulation of experimental data should be carried on continuously.

V. PROPOSALS FOR CONTINUED TESTS.

It is strongly recommended that systematic tests for natural frequency be made on all newly constructed vessels and a representative number of existing vessels of all types.

It is further recommended that as soon as methods and apparatus permit, tests for natural frequency be made a part of the routine of Material Inspections, in order to obtain a record of the elastic history of individual vessels.

In the pending elastic tests on two destroyers it is recommended that all available methods be utilized, including measurement of natural frequency, tracing of stress distribution under uniform pulsating load, measurement of stress and deflection under static loads carried to failure, and progressive replacement of material at regions of local failure.

To obtain data on loads occurring at sea a modified recording gear suitable for operation by ship's personnel should be developed.

Basic studies of elastic action of models embodying characteristic features of ship construction but not attempting to simulate all details of specific vessels have been begun at the Experimental Model Basin and should be energetically pursued.

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APPENDIX A

PREVIOUS EXPERIMENTAL STUDIES OF ELASTIC CHARACTERISTICS OF SHIPS

1. The prototype of all work in the field is that of Biles. His method consisted of measurement of strains and deflections under accurately known loads in drydock, with incidental measurements in a seaway to determine loads actually occurring.

Basic work in Germany was done by Pietzker, but no detailed account of his tests has ever been published. His ideas, however, are summarized in the 219 pages of "Festigkeit der Schiffe".

In some early work, strain gages were placed at selected stations, stress values due to action of sea inferred, with no other measurements. Readings give only range of strain and take no account of initial stresses in still water, whether due to unequal distribution of weight and buoyancy or to other causes. By such methods Howard found effects of slow load-shifts almost wholly masked by temperature effects. At sea he observed pulsations from engines to be of the same order as strains due to pitching. He also found the well-known concentration at the break of the bridge-house.

An extension of the method of simply trying on a strain gage here and there at sea is to secure the gage in a fixed location and obtain an automatic record from it. In this way maximum values may be obtained and slow temperature changes may be distinguished from sufficiently rapid load changes. If several recorders are used and synchronized, simultaneous momentary effects may be studied. Thus stresses during launching may be studied as well as transient effects in a seaway. If simultaneous observations are made at different heights above base, the neutral axis may be located.

2. By 1926 Siemann had perfected his telemeter type of recording gage and could get simultaneous records from six different stations. Range up to 17,000 lb. sq. in. was observed, divided between tension and compression in the ratio of 3 to 2 by a method undescribed. By guessing at the size of the waves and comparing calculated values for trochoids he arrived at the conclusion that stresses on the standard wave might be twice as great as those observed. Impact stresses of the order of $\frac{1}{4}$ to $\frac{1}{5}$ those due to pitching are shown in his records.

Siemann's 1928 paper marks the culmination of stress measurements at sea uncombined with static tests or deflection measurements. Autographic records were made of stresses, vertical accelerations due to pitching, and wave profiles.

This very ambitious scheme affords good evidence that more is to be accomplished by other methods. Siemann has been perfecting his instruments and procedure for 20 years and must still content himself with reference to what his methods may be expected to accomplish. It therefore does not seem unduly hasty to conclude that the case calls for something other than measurements at sea.

In 1926 the Hamburg Versuchs-Anstalt entered the field of study of elastic characteristics, and to them also measurements at sea seemed at first to offer the best prospect for conclusive data. But measurements of deflection as well as of stress were made, and some consideration was given to the matter of total stress, taken with reference to a null condition of zero stress. Initial stresses built into an assembled structure cannot be detected by measuring strains under applied load increments; but the condition of zero bending moment may be adopted as the starting point for the measurement of stresses which thus become absolute and no longer simply relative.

In the papers of 1926, continuous curves of deflection and stress extending over a long voyage are shown, but no analysis with summarized conclusions is given.

Lockwood Taylor on a series of four vessels measured stresses due to static shift of weights in still water and compared them with values calculated by the usual procedure. His measurements at sea consisted simply of observations of stress range with correlated observations of ship's motion, wind and sea. In spite of the abundance of numerical results obtained by Taylor, conclusions available for use in design are very meager.

3. After the various attempts to determine the elastic characteristics of ships by measurements at sea had, in part at least, shown the inadequacy of such measurements for solving the problem, more detailed consideration was given to just what kind of measurements are really pertinent. In 1928 Dahlmann overhauled his ideas. The problem, he said, consists in principle of a determination of the section modulus. In addition to stresses, this calls for bending moments, which may be determined only in still water. Young's modulus for assembled structures is found by combining flexure with strain measurements. Now comes the heart of the matter: "Static tests provide a calibration of the strain meter in terms of bending moment. Determination through observed stress of the bending loads occurring in a seaway is necessary because analytic development of dynamic bending moment loads is impossible and direct determination of bending loads at sea is mechanically impracticable".

4. Every problem in structural design has two aspects which must be borne clearly in mind as requiring separate determination; the load and the means of carrying it. By observations at sea, combined with calibrations in still water, accurate data on load may be obtained. How may the structure which must resist these loads be perfected? The answer is: Primarily by means of model studies, which a restricted number of full scale tests. The traditional procedure by which current practice has been evolved is that of design by judgment, with correction of defects indicated by casualties. This procedure is incapable of leading to refinements and reduction in weight except after slow, wasteful, and uncertain trials and errors.

Advantages of work with models lie in ease of repetition of tests, control of conditions, precision of measurements, and especially extension of test to failure. On the other side are difficulties of reproducing details on a small scale and uncertainties as to the law of comparison and scale effects.

Static similitude between two models of length ratio λ is obtained when force loads are proportioned to λ^2 and bending moments to λ^3; deflections are then proportional to λ, and stresses and slenderness ratios to λ^0; that is, unaffected by scale. Similitude of buoyant loads, however, would require a liquid of density proportional to λ^{-1}. Also bending loads due to weight of structure cannot be simulated except by adding weights to the model which take no part in the elastic action. It is thus clear that in model tests it is necessary to take means other than simply loading a floating model to obtain bending moments proportional to λ^3; in general the shearing action will then depart from the law of comparison. In cases where shear is of major importance special methods of distributing loads must be devised or corrections introduced. Similarly the effect of hydrostatic pressure on shell plating under compression must be duly accounted for. And objections have been made that dynamic loads cannot be simulated in a model.

Complete similitude is thus unobtainable as it is in propulsion tests also. But in the case of elastic tests, data on the actual values of load, including items arising from dynamic effects may be had by use of recording instruments on a calibrated ship at sea. Application of the corresponding load to the model will have identical elastic effects if the load is identical in magnitude and distribution, regardless of whether that load is purely static or originates in part in dynamic effects; for elastic properties of steel, especially within elastic limits, are not greatly affected by motion otherwise than through the action of inertia loads.

5. Lienau has a small laboratory at Danzig in which he has been making some careful studies of the fundamental characteristics of box girders bearing a general resemblance to a ship. He summarizes the situation as follows:

"In full scale tests there is great uncertainty both with respect to the external loads and to such conditions as end-fixation, which can neither be exactly known or controlled, especially when the loads are rapidly and continually varying, as in a seaway. The painstaking experiments of Siemann, Schafer, Dahlmann, and Kempf on ships at sea could therefore yield interesting items of information and data of value on instrumental technique but no valid conclusions as to distribution of stress and no theoretical results of scientific value".

6. Vibrations of ships have formed the subject of experimental and analytical studies for a long time and the papers covering these are well known and easily accessible. Those of 1893, -4, and -5, by Schlick in the TINA fully covered the case of resonance between unbalanced engines and fundamental vibrations of the hull in the vertical plane. It had been shown in 1884 that such vibrations "were in no way caused by weakness of the hull". By 1894 it was clear that "means for the

avoidance of such vibrations must be provided, not by the shipbuilders, but by the marine engineers", and "with the creation of the balanced reciprocation engine the first act in the solution of the vibration problem was in a manner brought to a conclusion".

These quotations are taken from Schlick's papers in TINA of 1911, in which he then goes on to discuss more obscure cases of vertical vibrations of higher order, transverse and torsional vibrations and the disappointing experiences in which it appeared that even turbine drive could not eliminate the unbalanced conditions occurring in case of a screw propeller, one of whose blades differs slightly in pitch from the others. Even perfect balancing will not eliminate asymmetry altogether, and there are cases in which the only resource is to take radical measures for breaking up the condition of resonance by altering the blades on one or both screws. A number of recent papers, such as these of Tobin, 1922, and Nicholls, 1924, both in TINA, and L. Taylor, NECI 1927–1928, are concerned with prediction of natural hull frequencies to permit keeping propeller speeds clear of them.

Another way in which an engine may cause hull vibrations in spite of perfect balance of reciprocating parts is through non-uniformity of the speed of rotation of propeller, such as is associated with torsional vibrations in the shaft. This would result in pulsating thrust which would communicate the effects to the hull. Although such action in shafts is known to occur and to be responsible for shaft breakage, no previous case is known in which the resulting forced vibrations in the hull reached amplitudes of importance. A paper on this subject by Gumbel appeared in TINA in 1912. Recent work in this connection has dealt more particularly with questions of engine design than with hull vibration.

APPENDIX B

CALCULATION OF LONGITUDINAL BENDING MOMENTS

All calculations were based on the trapezoidal rule applied to 10 stations forward and 10 aft.

For the initial condition a survey of temporary weights was made, and by combining this with returned weights of builder, inclining experiment data, and observed drafts, a zero condition was established which included only 105 tons of accretions not specified, including paint. This is not an unreasonable value for 12 years of service. The only weight distribution available was made before construction, and was found not to agree with observed trim. About $\frac{1}{3}$ the weight in the zero condition could be definitely located, the rest being included in the assumed distribution for "basic" weights. This distribution had to be altered considerably from that adopted in the calculations made during design, and this constitutes an important source of uncertainties in the ordinary process of design.

No attempt was made to split stations in locating weights; distribution within the station is assumed uniform, or concentrated at the station section. This is not quite equivalent to applying the trapezoidal rule to a continuous curve.

Buoyancy was taken from Bon Jean curves which were entered with observed draft. This gave a displacement differing irregularly up to 300 tons from the surveyed value. This effect was partly due to flexure of the ship which was estimated to account for as much as 100 tons. But effects of humidity on paper also figure and possibly other errors.

Buoyancy at each station was adjusted proportionately so as to force the total to agree with surveyed weights.

Combining the figures for weight and buoyancy gave a net load for each station. Multiplying this in each case by lever arm from section at which bending moment was desired gave the local contribution and single summation the total bending moment.

Values were thus determined for forward and after parts of the ship and the mean of the two values accepted. Differences were a little disappointing, as they ranged up to $\frac{1}{3}$ the total and in five cases even more. The higher moments were forward, with few exceptions. This difference is rather sensitive to slight changes of trim or to errors in weight distribution. From theory (to be detailed in a separate report), however, and from actual numerical trial, it is clear that the mean of the two values is practically unaffected even by very large discrepancies between them. Since the absolute values of the bending moments do not figure, but only differences between them, it is believed that nothing would be gained by the large expenditure of effort necessary to eliminate errors due to the procedure in numerical calculations.

Calculation of longitudinal bending moments is subject to errors in locating weights which may be magnified by the mathematical operations involved. A close scrutiny of the approximations involved in shortened calculations is being made, but the results of this study will be separately reported.

APPENDIX C

NOTE ON PRECISION OF THE MEASUREMENTS

Measurement of stress and deflection has nowhere reached the precision attainable in other fields of experiment and conditions on board a ship in full commission are not favorable for work of this nature. For this reason full data have been reproduced on which averages are based and a probable error has been appended to every such average. Instead of the rather long procedure involved in working out the root mean square error, the mean absolute error was taken and the probable errors quoted are 5/4 this value divided by the square root of the number of observations.

TABLE I

COEFFICIENT OF STIFFNESS FROM FLEXURE DATA
EI x 10^{-12} lb. ft^2

Condition	Span		
	88 Frames	61 Frames	24 Frames
1–5	8.1	7.7	6.9
1–6	8.6	8.3	9.9
1–7	8.9	8.4	11.5
1–8	9.2	9.5	7.2
1–10			7.1
1–16	9.7	9.3	7.9
1–19	8.4	7.8	9.0
11–20	8.1	7.5	
14–22	9.2	8.8	10.4
15–23	7.8	6.9	
27–28	10.5	10.7	12.1
29–30	8.1	9.3	10.6
31–32	10.6	8.9	8.2
32–33	8.9	8.1	
34–35	11.4	9.4	10.2
34–36	10.0	8.8	11.0
37–38	7.5	7.6	6.1
38–39	12.1	10.6	
40–41	9.2	9.1	13.5
40–42	11.5	10.8	
Average = 9.2 Probable error = 0.14			

TABLE II

RATIO OF STRESSES BY STRAIN GAGES AND BY EXTENSOMETERS

Condition	Starboard	Port	Center
1–2	.96	.94	.81
1–3	.98	1.19	.95
1–4	1.09	.94	.93
1–5	.95	.92	.93
1–6	1.05	.92	1.00
9–11	1.32	.78	.94
10–12	.73	.90	1.50
13–16	.80	1.00	1.23
Average = .99 Probable error = ±.02			

TABLE III

SECTION MODULUS IN FT. IN²

Condition	Range B.M. foot-tons	By Extensometer			By Strain Gage		
		Stbd.	Port	Cent.	Stbd.	Port	Cent.
1–2	51300	22600	17200	12400	23400	18300	15300
1–3	90900	22900	19000	15300	23300	16100	16100
1–4	170500	22200	20000	19200	20400	21300	20800
1–5	166000	22500	19700	18600	23700	21500	20000
1–6	193000	21300	18700	18800	20300	20200	19000
1–7	179000	21200	18800	18400			
9–11	120000		25100	23000	15800	26800	22200
10–12	80400	14000	21000	24700	19100	23400	16500
13–16	162000	21500	21100	20600	26800	23500	16800
14–19	189000	26800	25800	22300			
15–17	173000	23900	23600	24500			
11–18	217000	24900	24600	23000			
12–30	209000	26200	25600	24600			
13–21	121000	20000	18200	17900			
11–22	157000	22200	24600	19700			
11–23	154000	24100	23600	18500			
27–28	173000	20700	20100	20300			
31–32	106000	20300	18300	23700			
33–32	33800	21000	15800	16800			
35–34	199000	26800	27000	25900			
36–34	186000	25000	25000	24500			
37–38	87500	20900	21100	19800			
39–38	59600	24700	24300	23400			
41–40	139000	21900	21000	21400			
Average		22500	21600	20700			

By Strain Gage — Average: Stbd. 21600, Port 21400, Cent. 18300

Grand av. = 20400 ft. in²
Mean absolute error = 2650
Probable error = 456

Grand average = 21600 ft. in²
Mean absolute error = 2400
Probable error = 240

TABLE IV

EFFECTIVE MODULUS OF ELASTICITY

Condition	Av. x 10^6 $4f/l^2$ – B.M.	Average Sect. Modulus	yE x 10^{-6}
1–5	212	21000	358
1–6	179	19700	452
1–7	168	19500	487
27–28	144	20400	543
31–32	175	20800	438
33–32	189	17900	471
35–34	155	26600	387
37–38	229	20600	338
39–38	141	24100	469
36–34	162	24800	397
41–40	155	21400	480

Average value yE = 438 x 10^6
Probable error = $5\frac{1}{2}$ percent
Assuming y = 18.2, E = 24,000,000

TABLE V

STRESS DISTRIBUTION AT EXTENSOMETER STATIONS

Stations		Section Modulus x 10^{-2} ft. in^2					
		Halves	Wholes	All	Average	Even Halves	Odd Halves
Stbd.	Outbd.	218	220	219		212	223
	Inbd.	213	227	221	238	216	210
	Cent.	283	265	274		289	268
Port	Outbd.	230	227	229			
	Inbd.	224	232	229	234		
	Cent.	255	234	243			
Cent.	Cent.	189	194	191	187	185	192
	Stbd.	183	181	182		180	187

TABLE VI

COEFFICIENT OF STIFFNESS AND NATURAL FREQUENCY

Ship	Natural Frequency per min.	Displacement Tons	EI x 10^{-12} lb. ft.^{2}	Schlick's Constant (absolute)	Critical Speed R.P.M.
U.S.S. CUYAMA	60.26 60.43 60.35 av.	15,430	9.2*	3.29	73
U.S.S. BRAZOS	87.7 89.3 88.5 av.	7,600	10.0	3.29*	76
U.S.S. NECHES	81.4 80.0 81.4 81.1 av.	12,600	14.0	3.29*	74
U.S.S. SALINAS	74.7 74.3 72.4 73.5 av.	10,000	9.1	3.29*	59–61

* Value accepted for application of Schlick's formula.

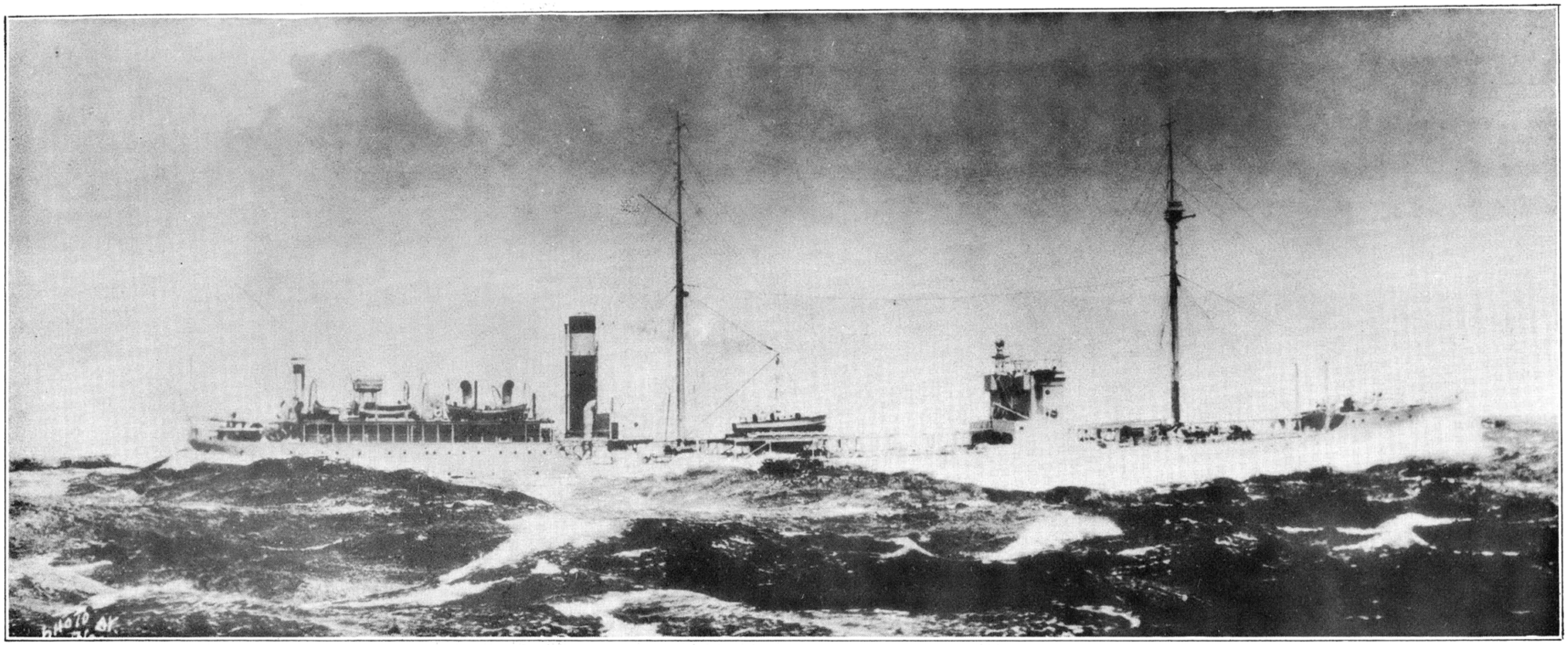


Figure 1. U.S.S. CUYAMA at Sea. Length 455 feet. Displacement 15,000 tons.

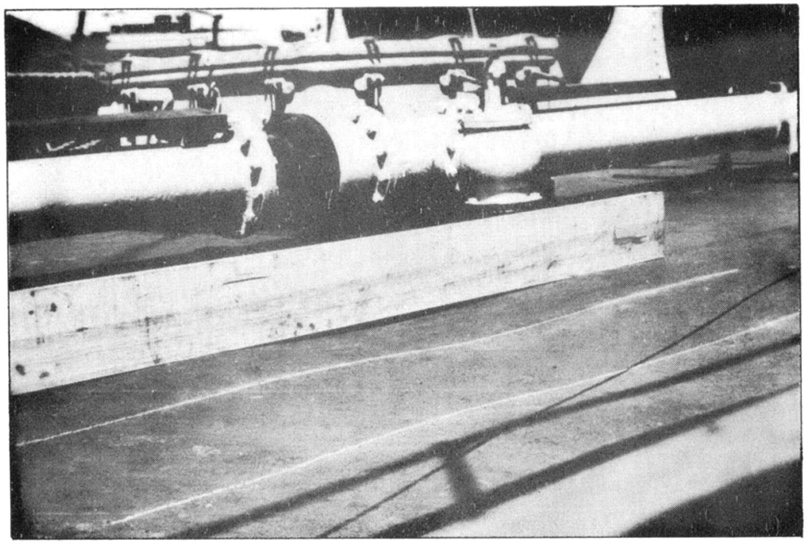


Figure 2b. Wrinkles, Portside Outboard.

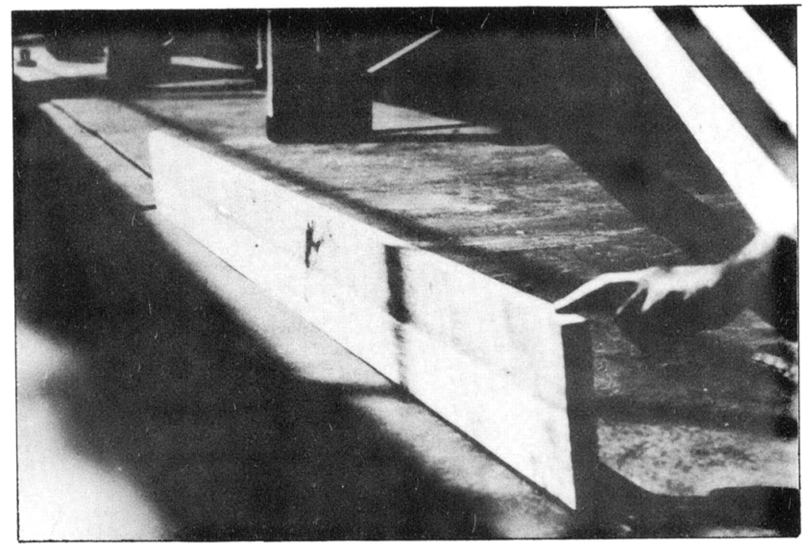


Figure 4. Starboard Stringer Plate.

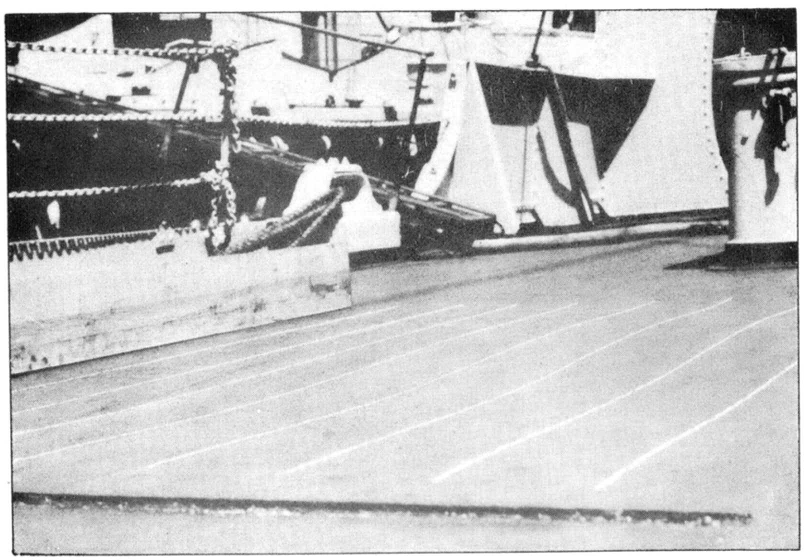


Figure 2a. The Worst Wrinkle Observed.

U.S.S. CUYAMA

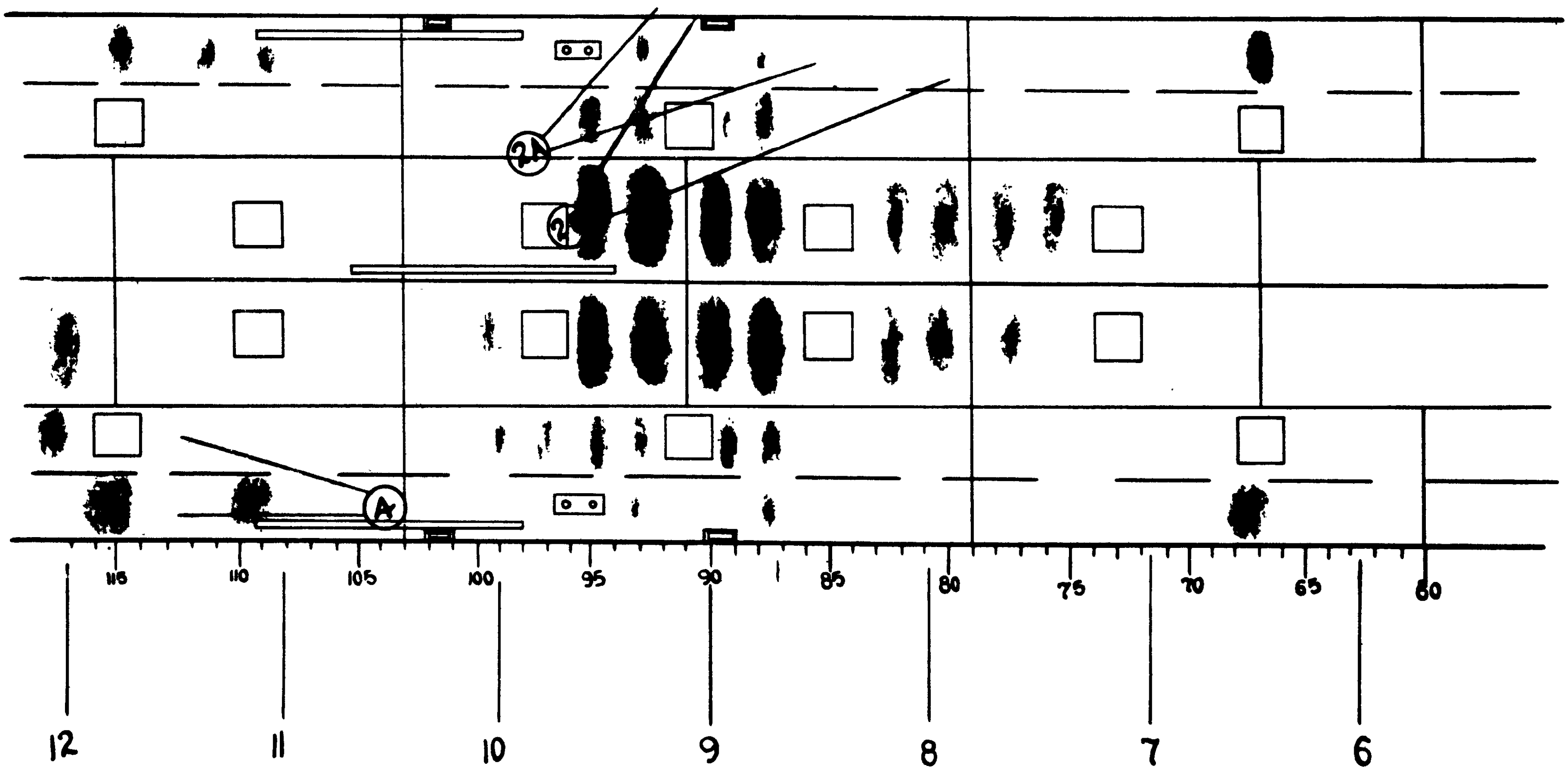


Figure 3. Areas of Pronounced Wrinkling.
Circled Numbers Refer to Photographs.

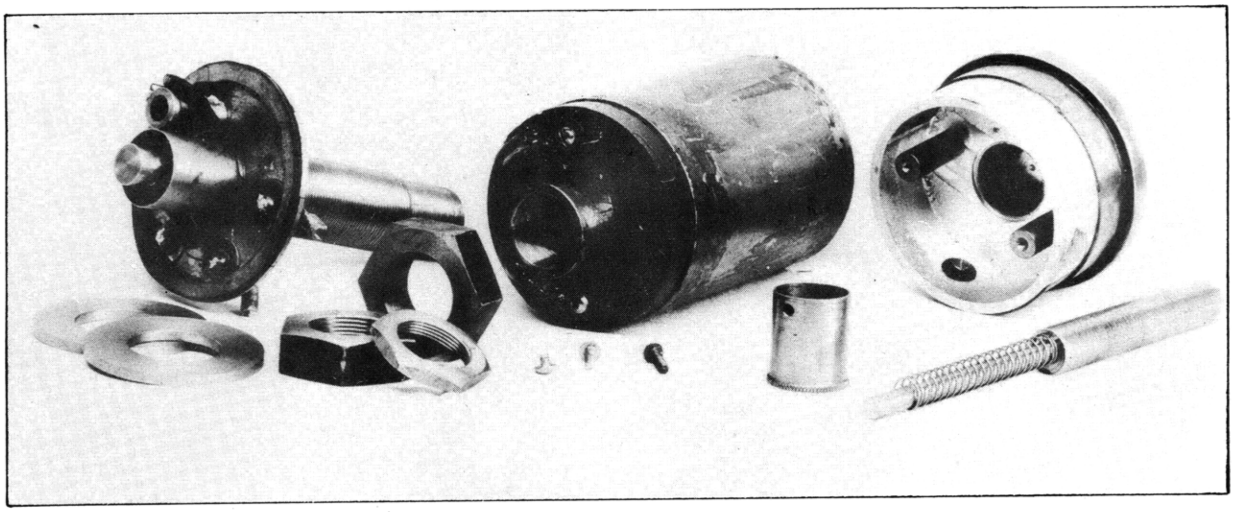


Figure 5b. Belleview Gage Details.

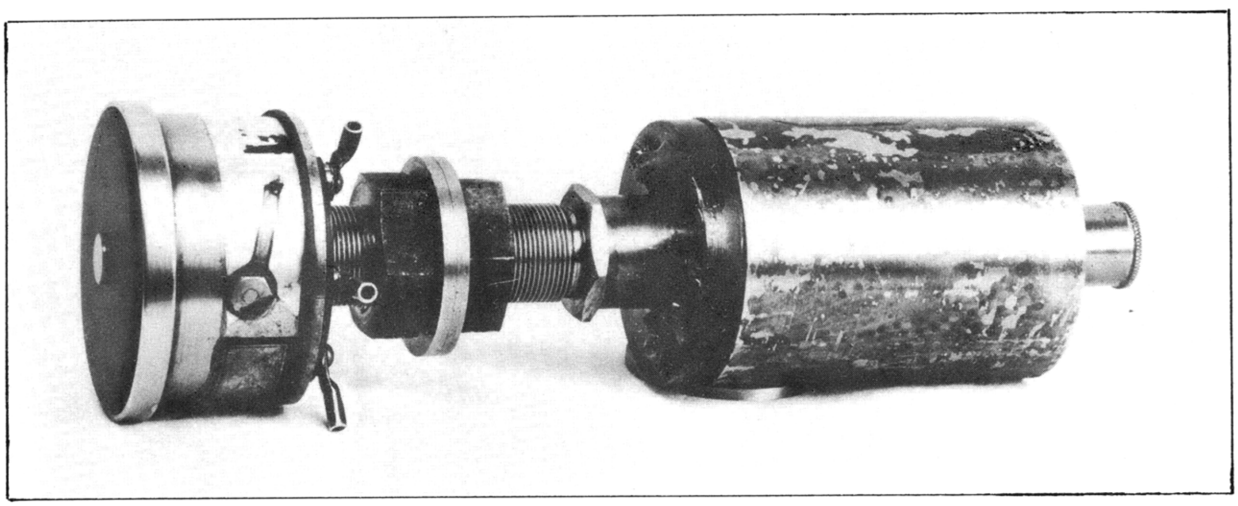


Figure 5a. Belleview Gage Assembled.

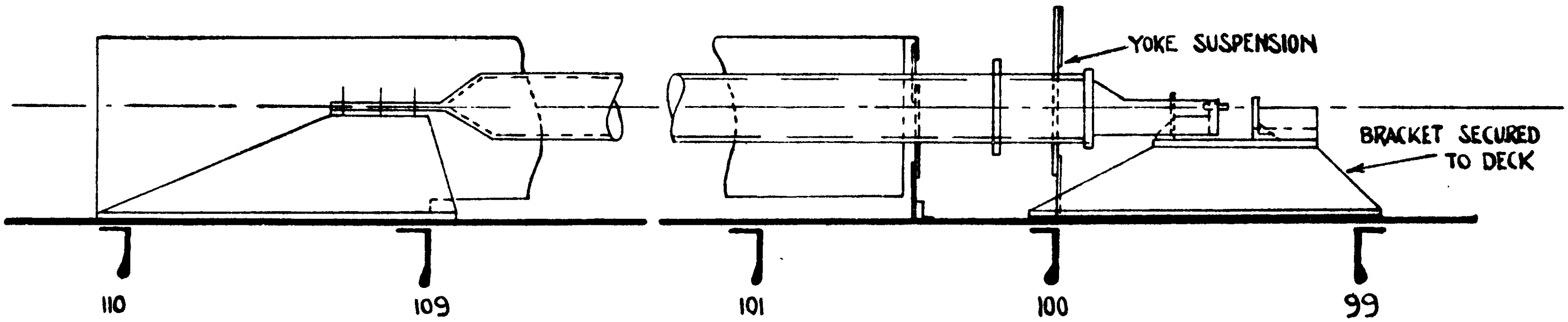


Figure 6a. Extensometer Assembly in Outboard Location.

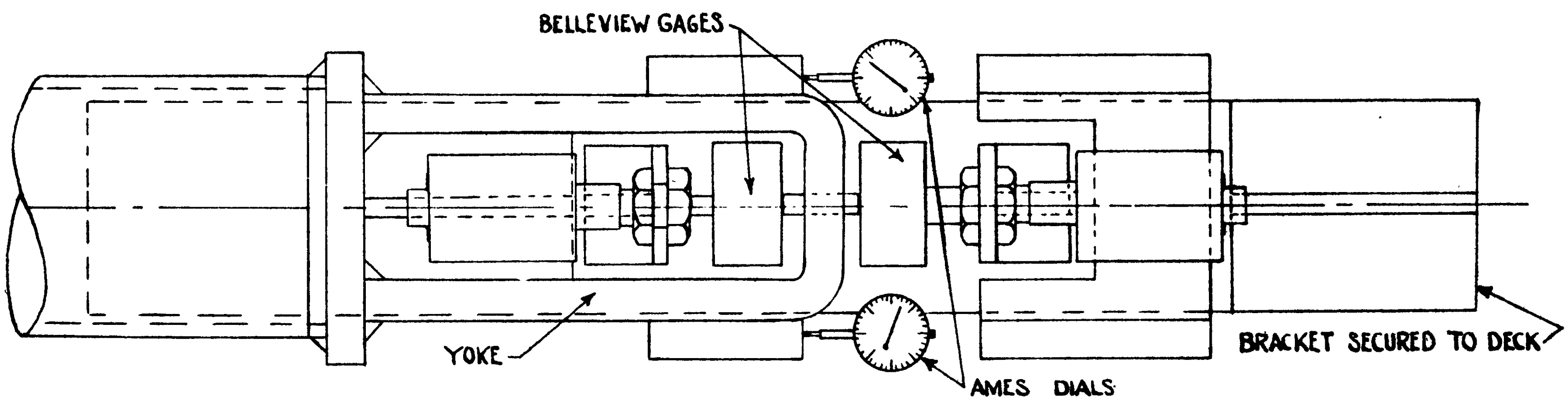


Figure 6b. Extensometer Gages Arrangement.

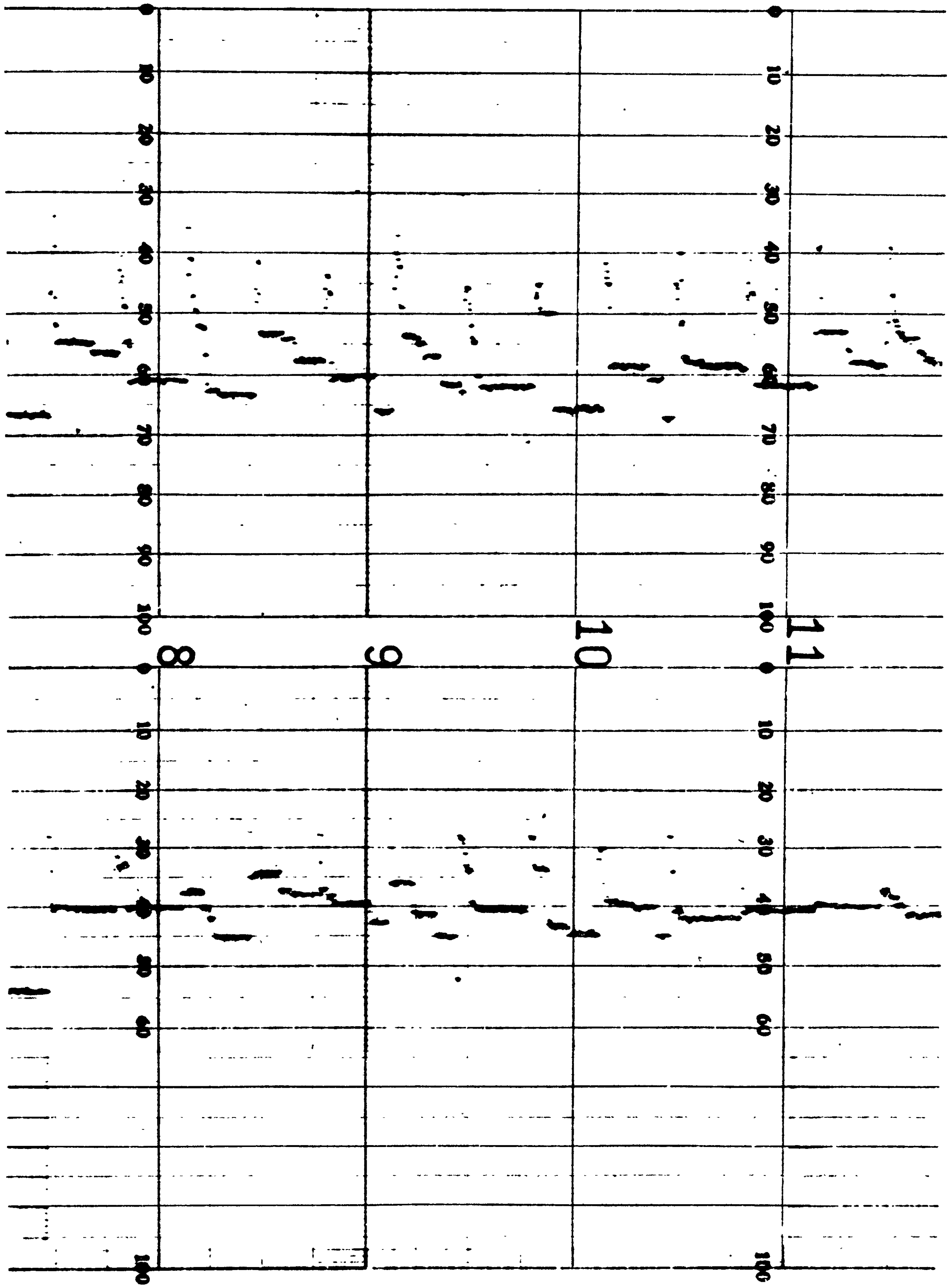


Figure 7a. Typical Charts from Brown Recorder.

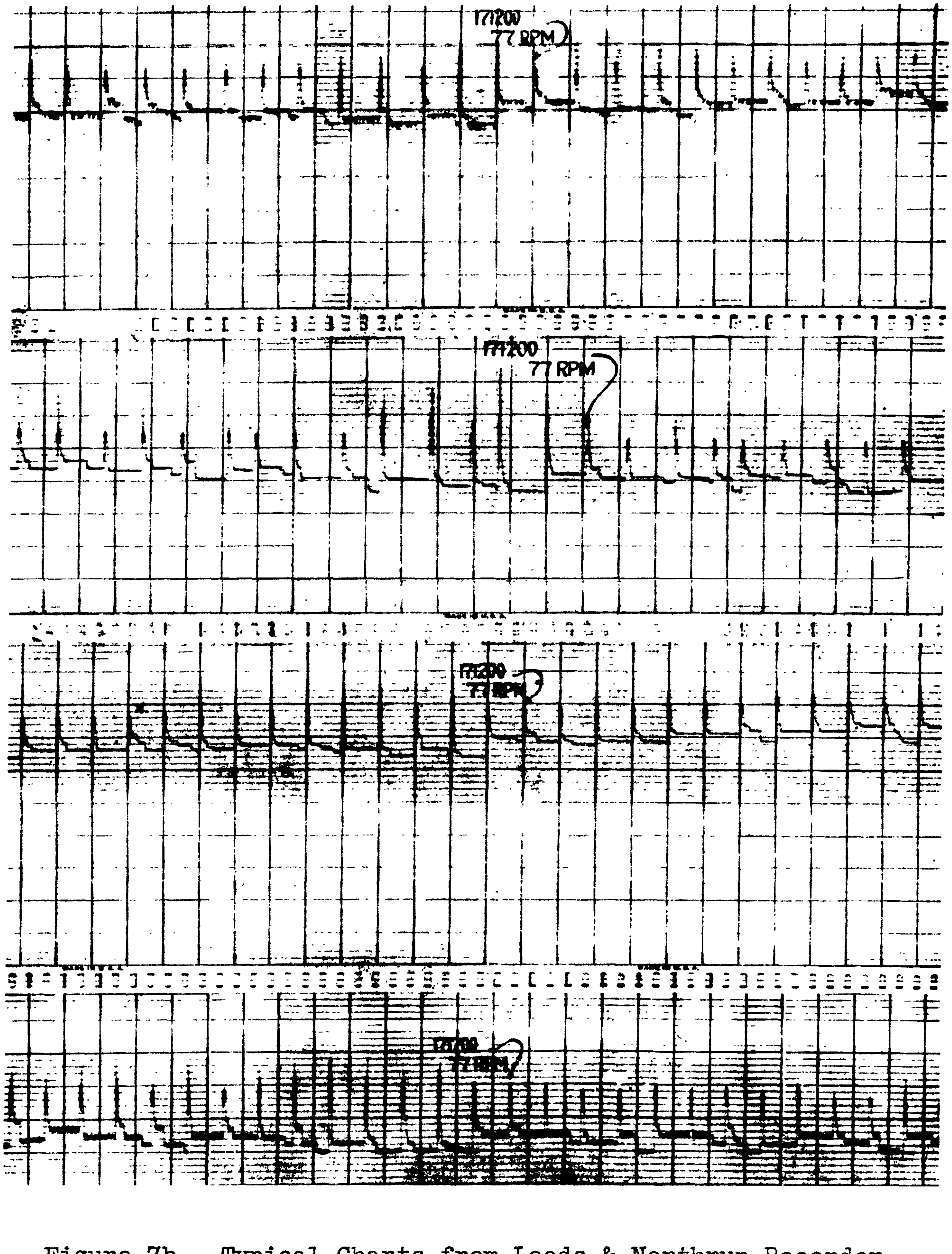


Figure 7b. Typical Charts from Leeds & Northrup Recorder.

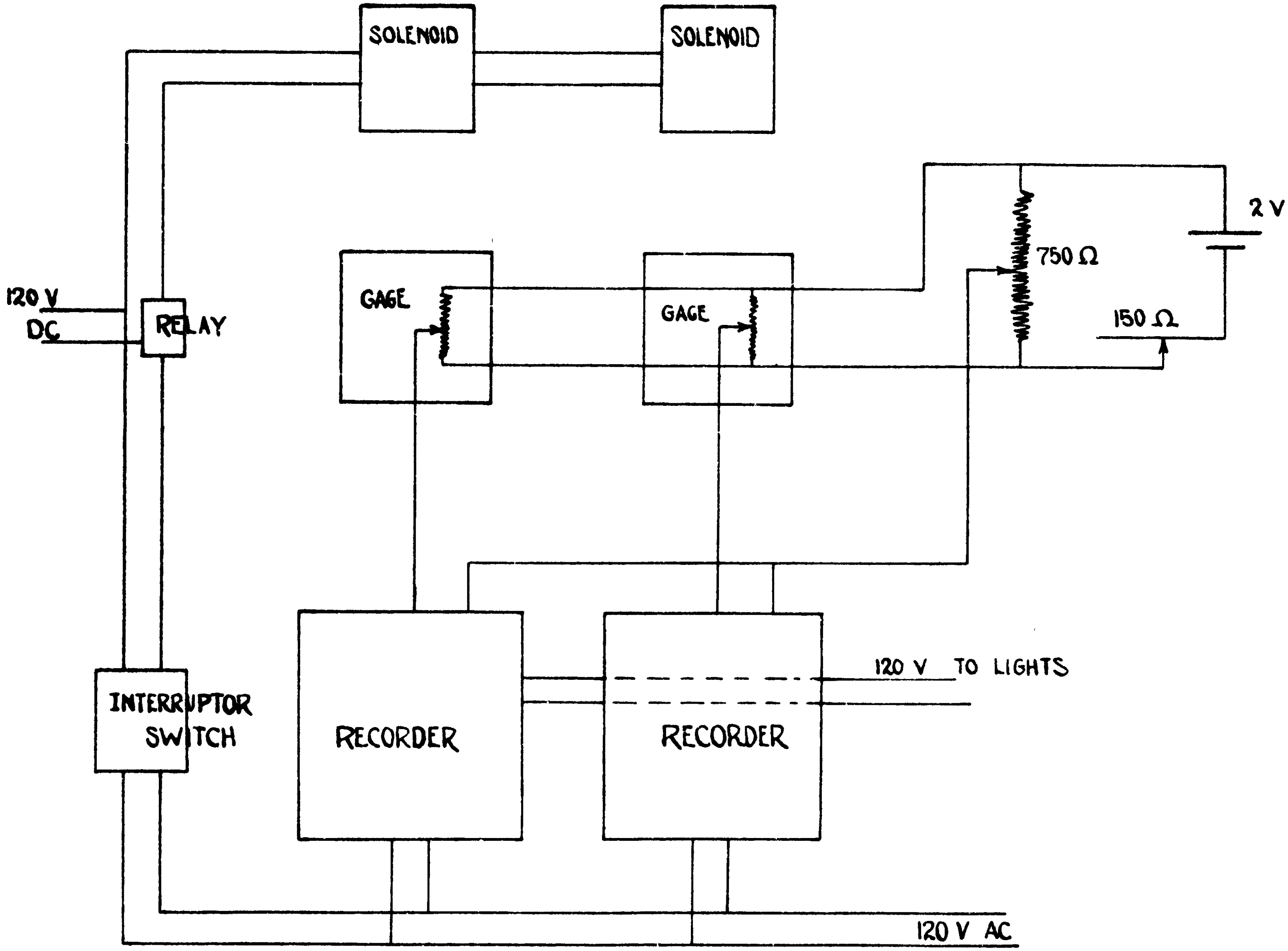


Figure 8. Diagram of Typical Connections for Stress Recorders.

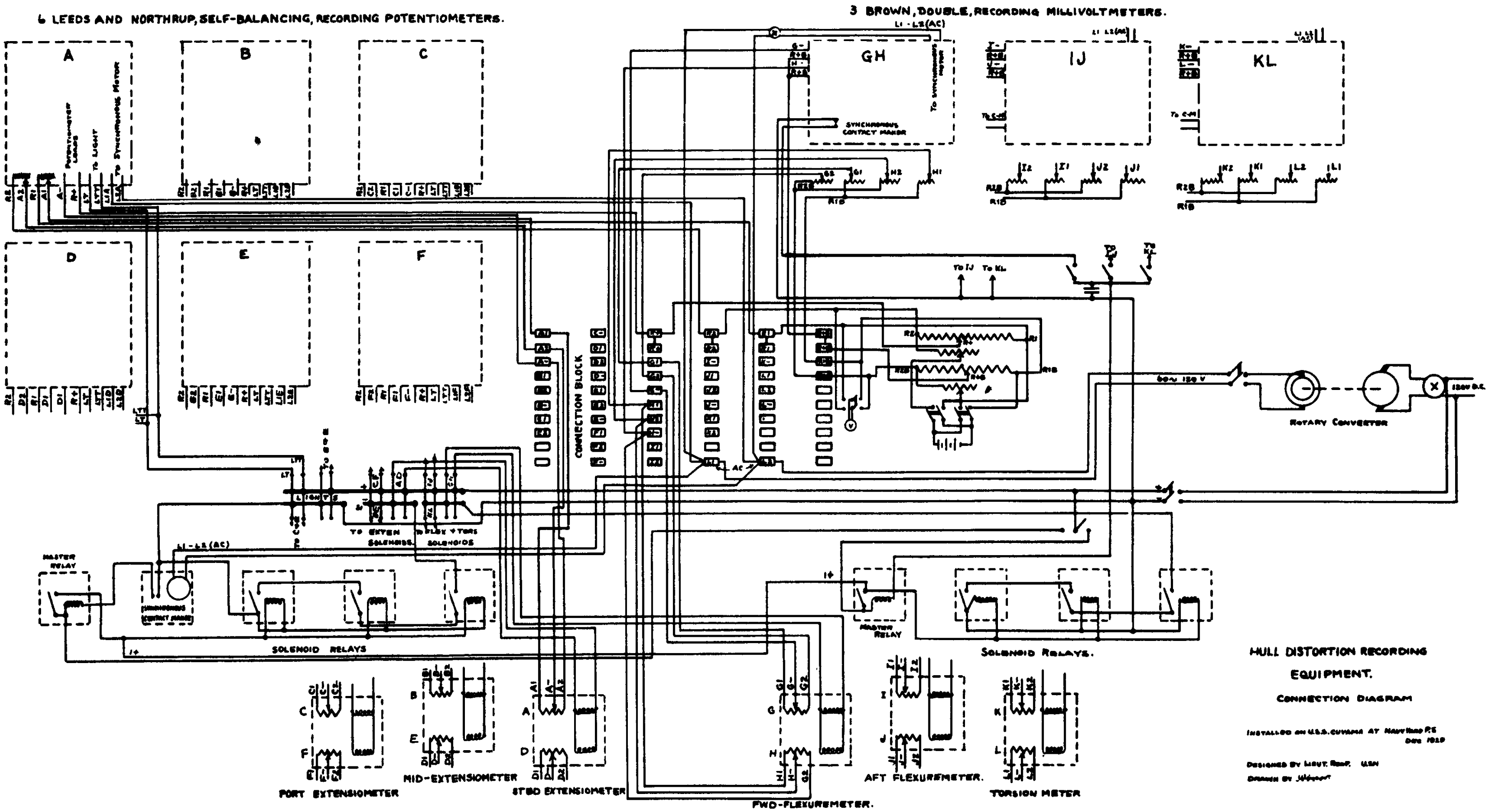
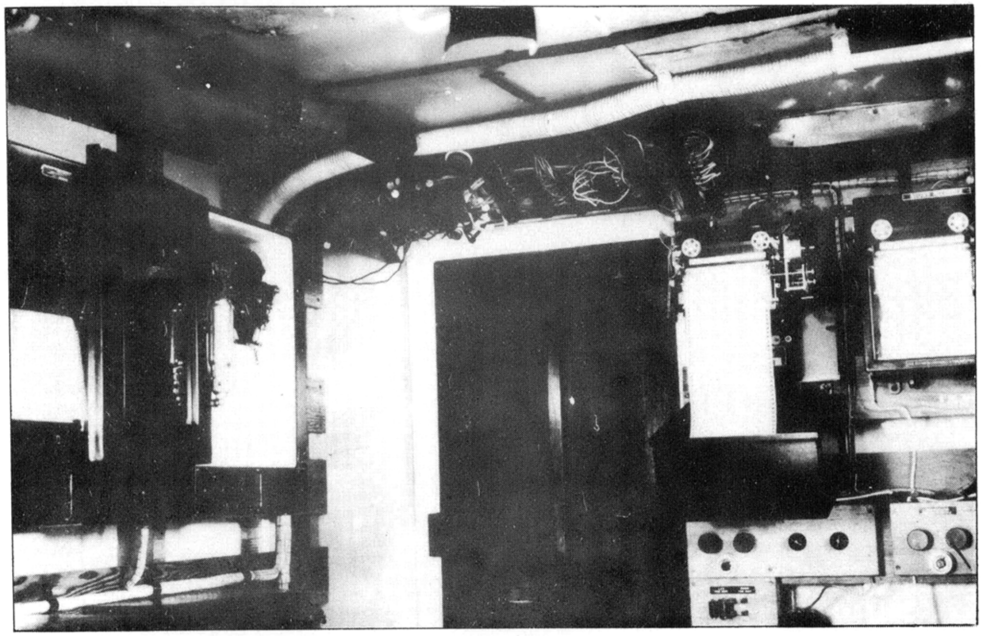
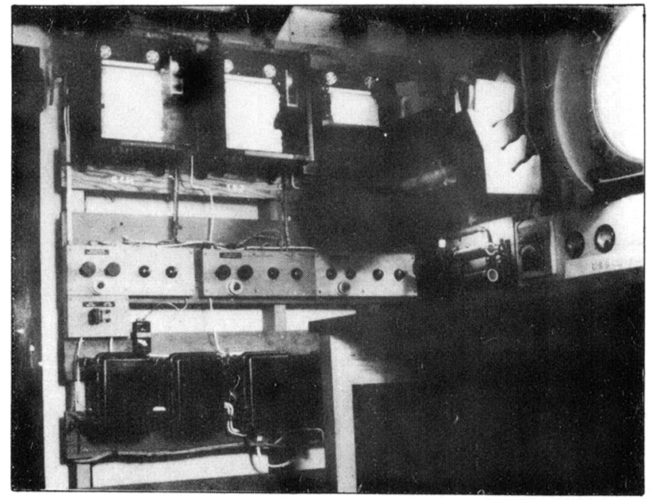


Figure 9. Detailed Wiring Diagram for Recording Gear.





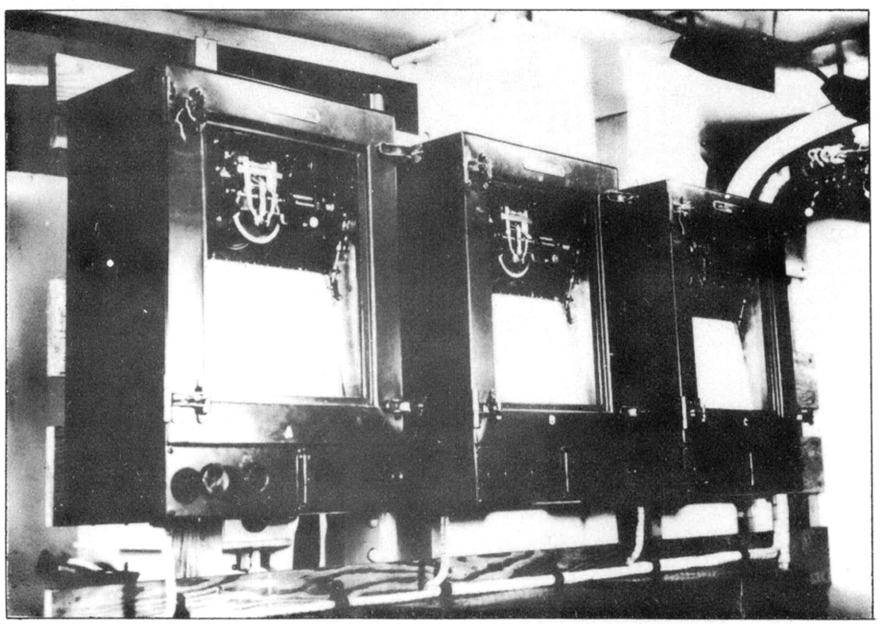


Figure 10. Recorders Installed in Emergency Cabin.

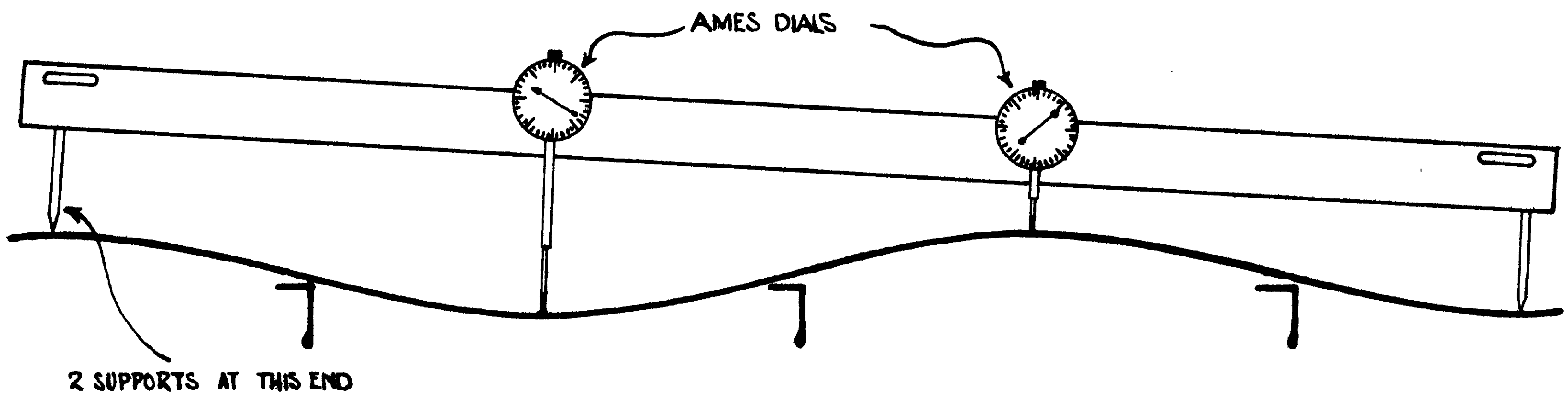


Figure 11. Arrangement of Dial Gages to Measure Panting.

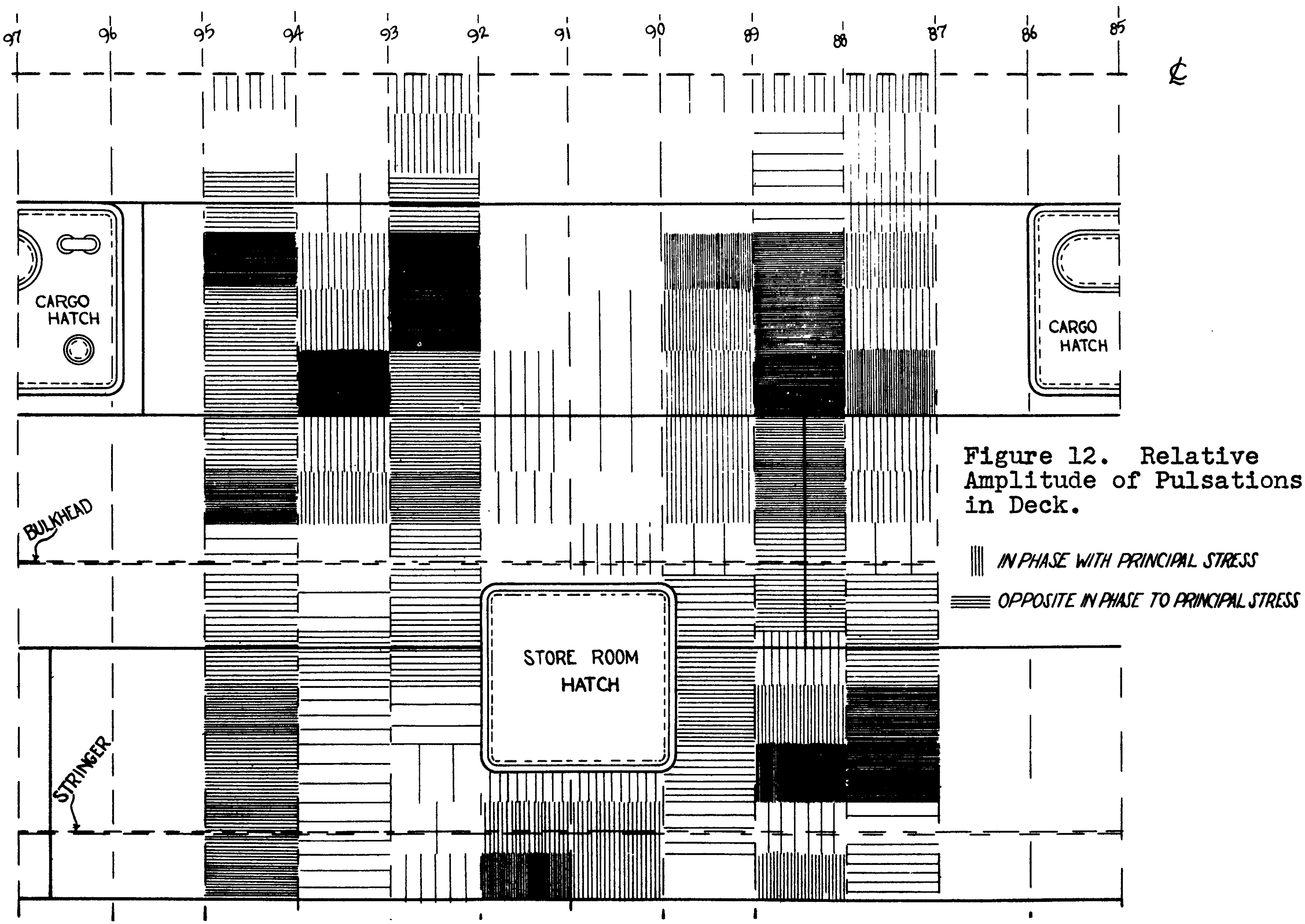
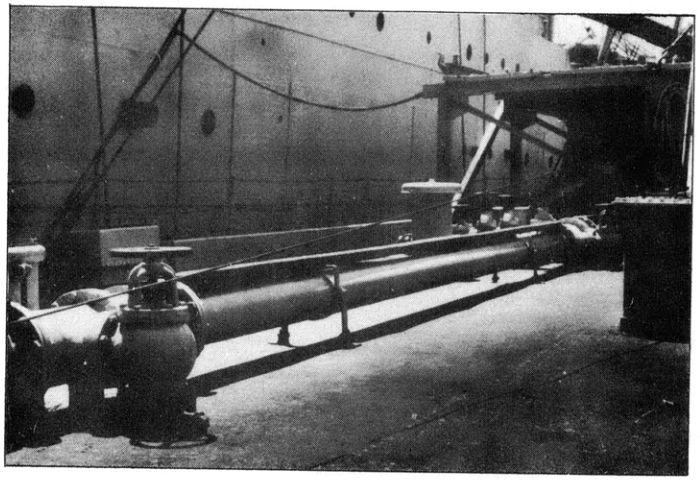
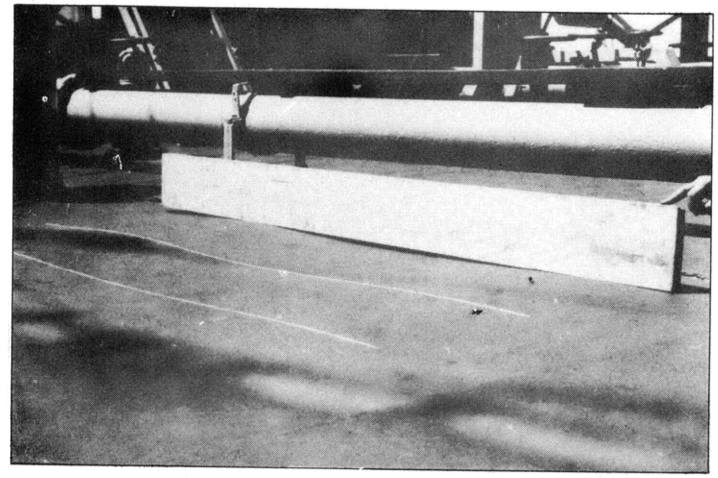


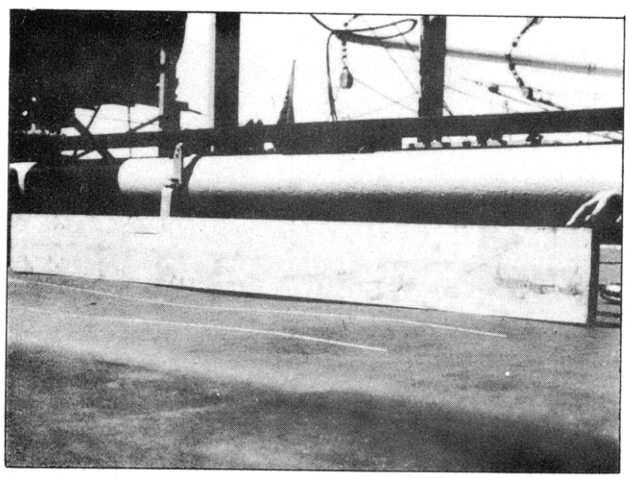
Figure 12. Relative Amplitude of Pulsations in Deck.



Frames 86-96 Starboard Between Cargo Hatches.



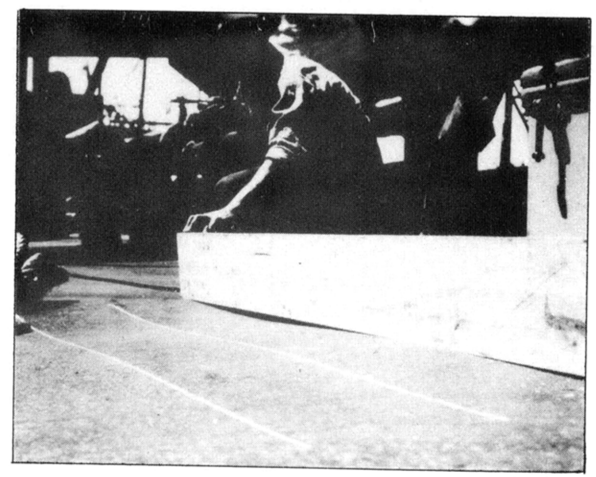
Frames 97-99 Starboard.



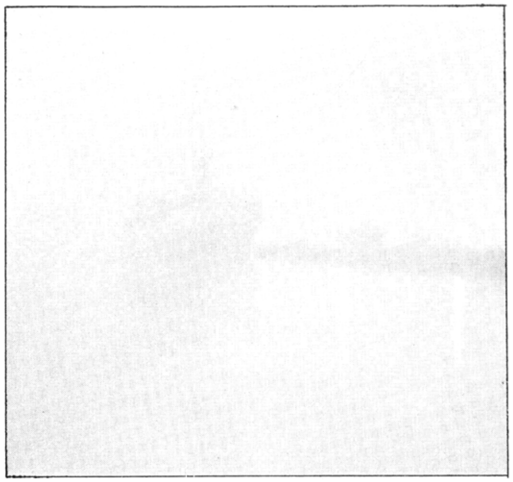
Frames 86-88 Starboard Forward of Storeroom Hatch.

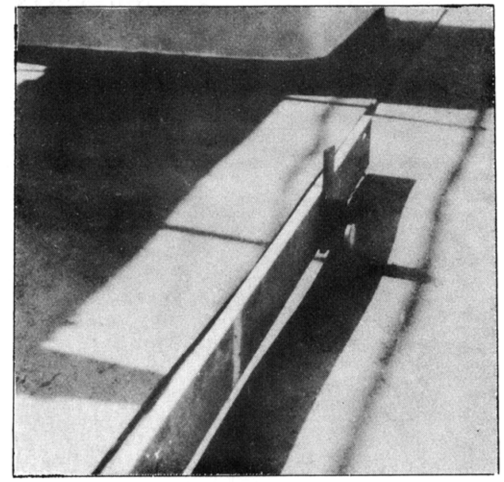
Plate 1

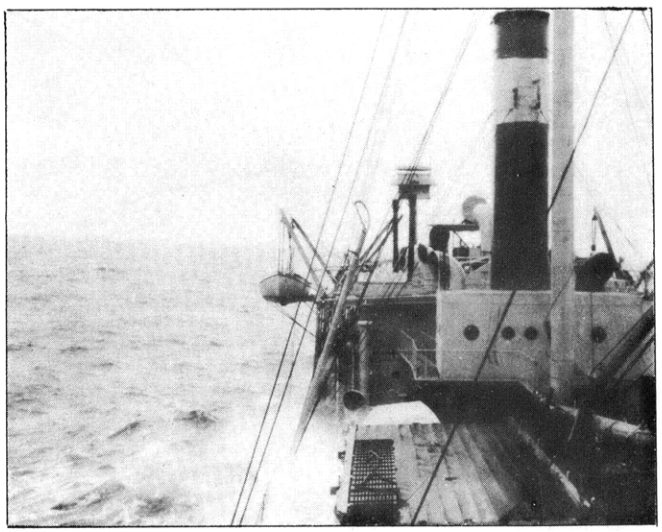
U.S.S. CUYAMA.



Frames 66-68 Stringer Plate Outboard Storeroom Hatch.







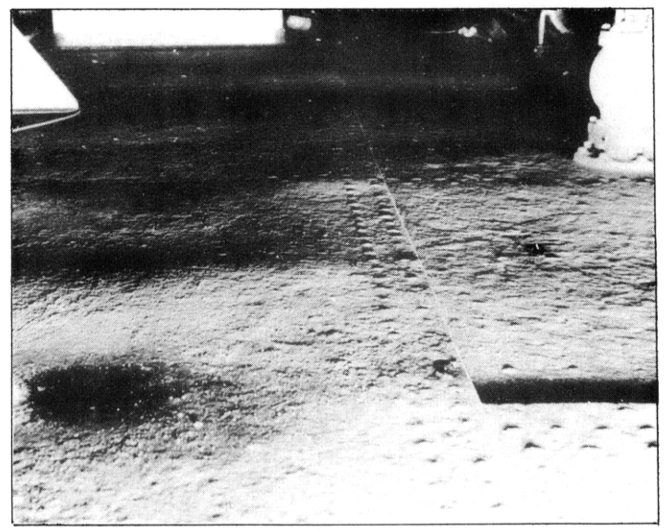
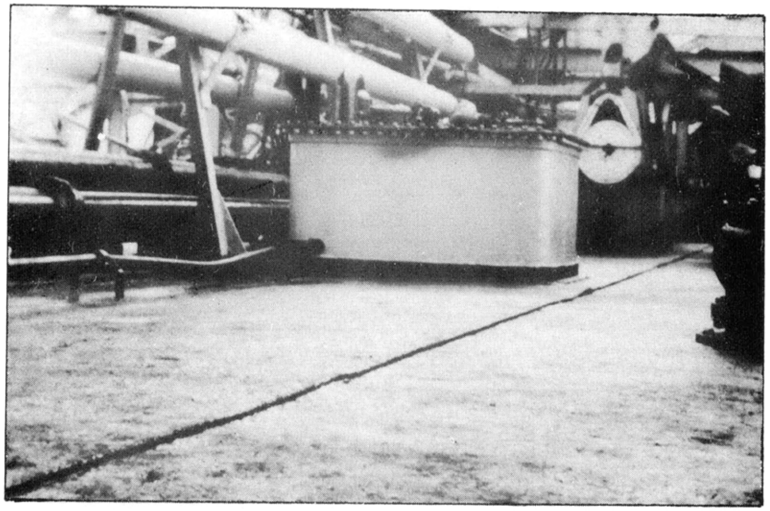
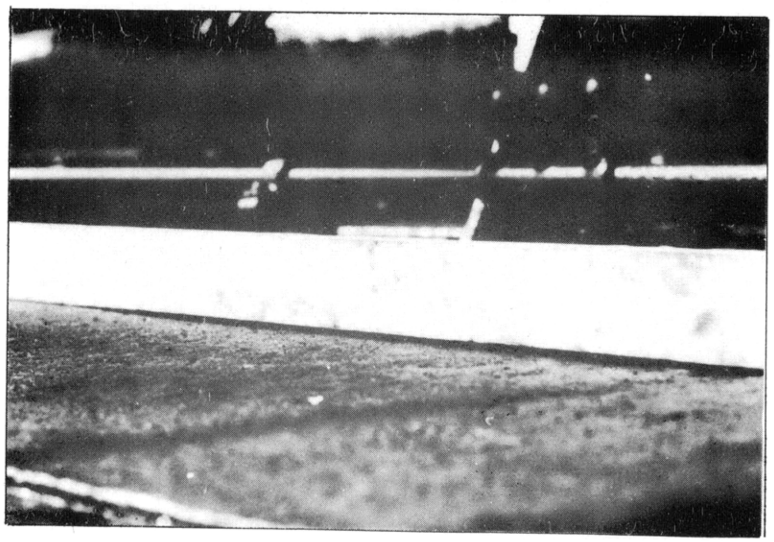


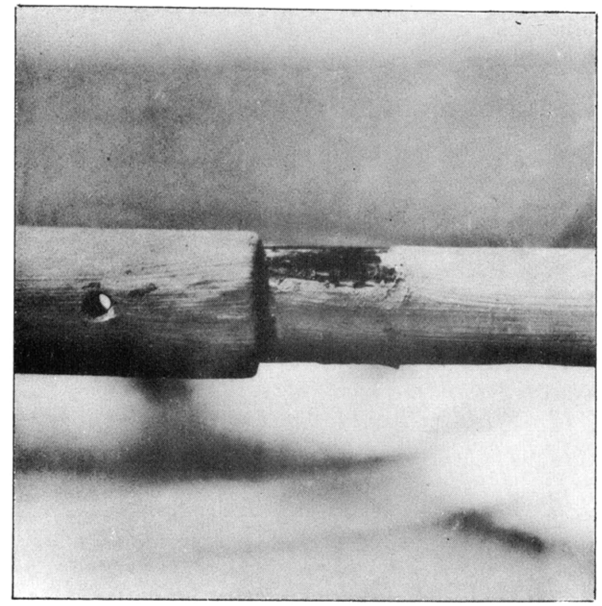
Plate la. U.S.S. CUYAMA.



Note Plate Edge and Bent Pipe.

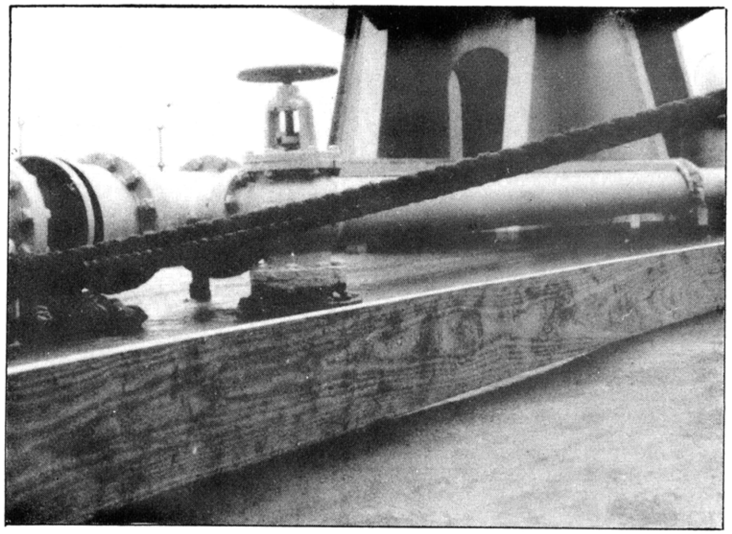


Buckled Plate.

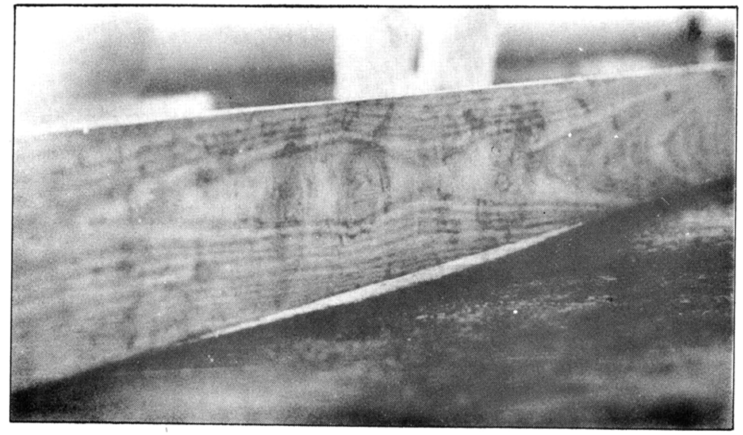


Slipjoint in Walkway Rail.

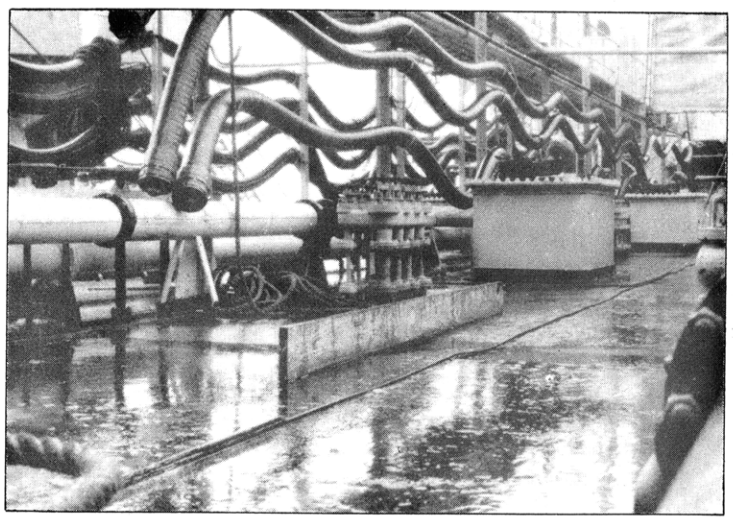
Plate 2. U.S.S. BRAZOS.



Buckled Plate.



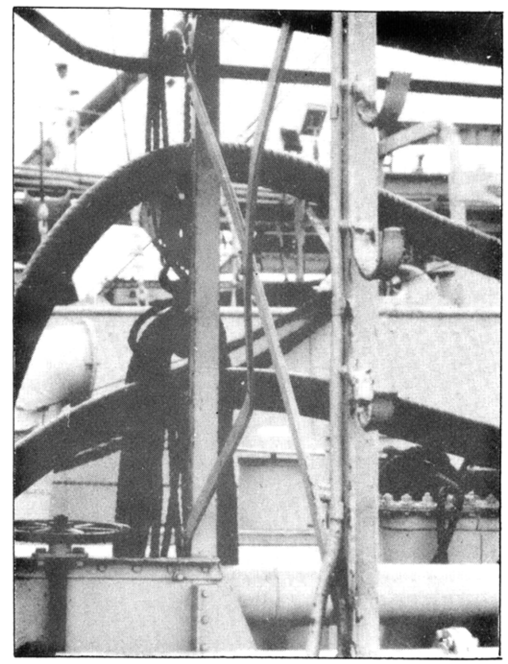
Buckled Plate.



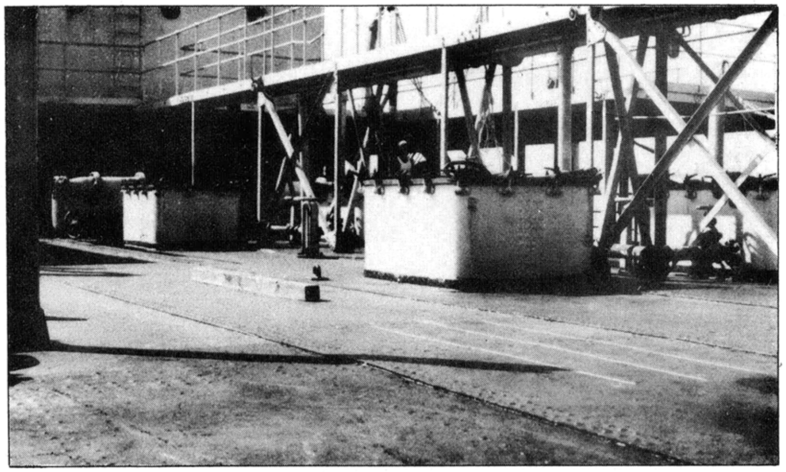
Note Seam and Doublers Around Hatches.

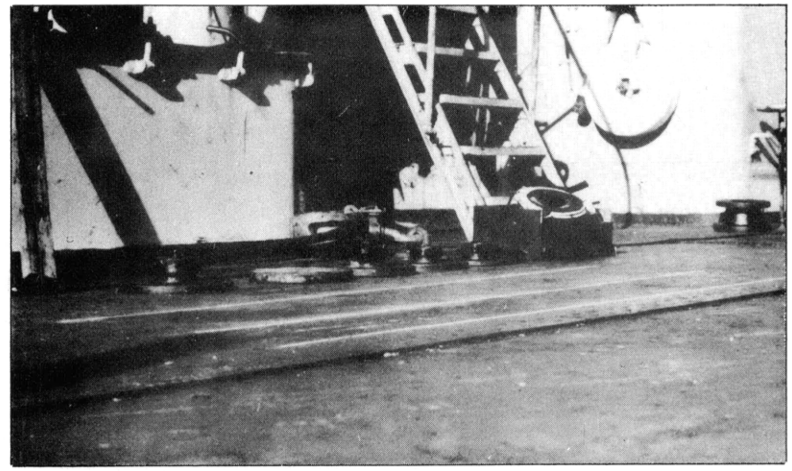
Plate 3.

U.S.S. NECHES.



Bent Brace at Break of Walkway.





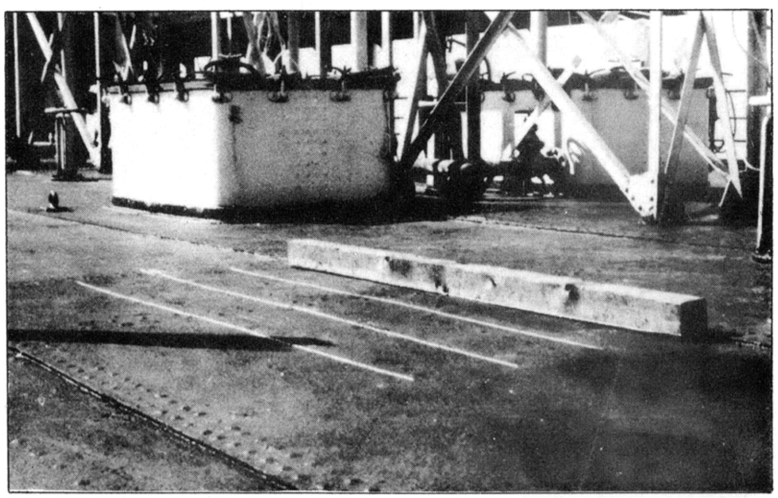


Plate 4. U.S.S. SALINAS

Buckling very Slight.