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UNITED STATES EXPERIMENTAL MODEL BASIN

NAVY YARD, WASHINGTON, D.C.

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STRAIN TESTS ON FLIGHT-DECK FRAMING OF USS YORKTOWN AND USS WASP

BY LIEUT. COMDR. W. P. ROOP, USN

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

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REPORT 471

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U.S.Experimental Model Basin
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INTRODUCTION

In the design of flight-deck supports for aircraft carriers the problems encountered are similar to those in shore hangars and other structures with a great span of unsupported roof. The solutions found satisfactory in shore practice are not directly available because of space limitations and because of the more severe loading that results from rolling of the ship. The flight-deck design of modern carriers, although satisfactory as to strength, is massive and heavy. It was considered that improvement might be obtained through experimental study, and special tests on the USS YORKTOWN were accordingly authorized (1).* Measurements of deflection and strain were made on the YORKTOWN in July 1938 under static vertical load. Application of horizontal load was found to be more difficult but was finally accomplished (2) in January 1940 on the USS WASP by means of a mechanical oscillator or vibration generator.

Since these tests differed widely in technique, they will be described separately. However, the results are comparable in nature and they will be combined for purposes of discussion.

GENERAL DESCRIPTION OF FLIGHT-DECK STRUCTURE

The structure is in general the same for both ships. The flight deck is made up of a series of sections placed end to end and separated by transverse expansion joints. Each section normally stands by itself, on rigid transverse frames called bents, without support from adjoining sections except for transverse horizontal reactions at the expansion joints.

The section tested on the WASP is shown diagramatically in Figure 1. Three bents, spaced at approximately 60 feet, support the section. Box girders beneath the deck edges, and a trussed girder along the centerline, extend over the length of the section and are supported by the bents. These longitudinal girders support intermediate transverse girders spaced at 16-foot intervals between bents. Resting on the transverse girders and on the bents are 12-inch longitudinal deck beams which support the deck plating to which the wood planking is attached.

USS YORKTOWN - TESTS UNDER STATIC VERTICAL LOAD

DESCRIPTION

Concrete blocks weighing about 5000 pounds each were placed on or near the centerline on the flight deck, over each of several bents. Measurements were

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* Numbers in parentheses indicate references at end of this report.

taken by increments of load, and they were plotted for increasing load, decreasing load, or both. In no case, however, was more than a single loading cycle applied.

Impact load equalling that of a plane landing on deck was applied by dropping a weighted airplane fuselage with landing gear, first from a height of 4 feet, then from a height of 8 feet.

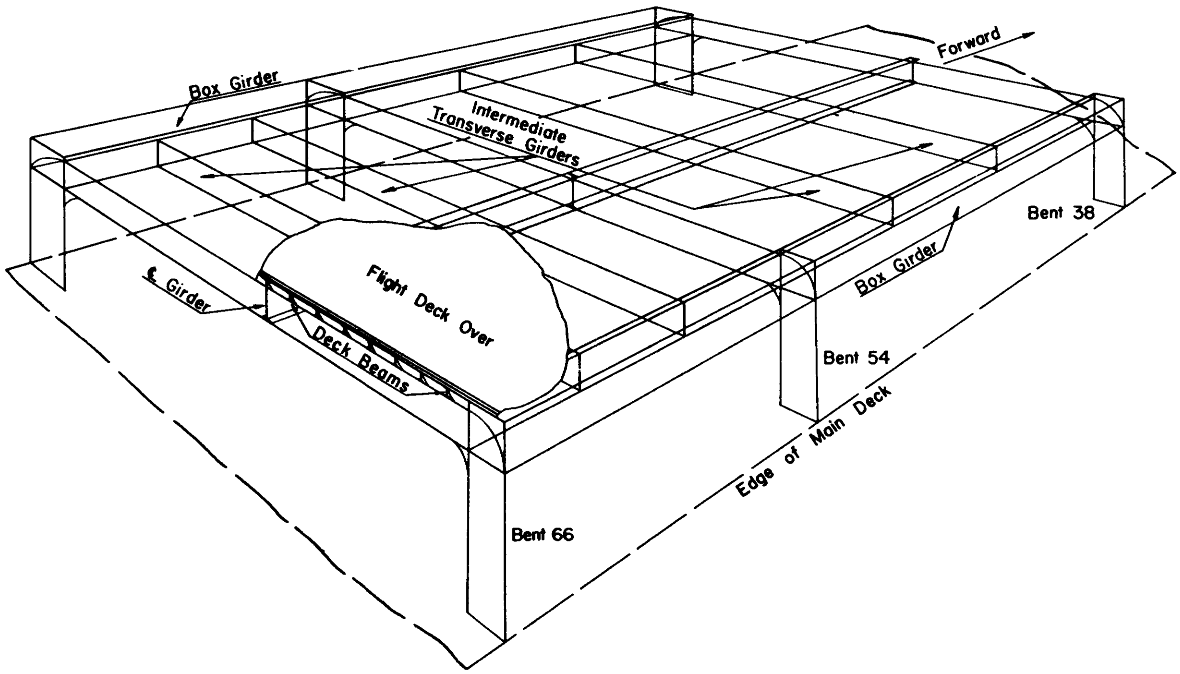

Figure 1 - Diagrammatic Outline of Flight-Deck Section IV

The structural response or deformation due to these loads was determined in various ways, as follows; see Figures 2 and 3:

(a) Vertical deflections of the loaded bent were taken with reference to the main deck by means of dial gages fitted with pipe extensions. No readings were taken on adjoining bents.

(b) Local strain measurements were made with Huggenberger gages.

(c) Lateral deflection of the lower flange of the transverse girder of the bent, resulting from torsional deformation, was observed through a transit telescope on the main deck. Estimated minimum observable deflection was 0.01 inch.

(d) Rotations in the vertical plane were taken by gunner's quadrant, with an additional quadrant placed on the main deck to obtain correction for trim.

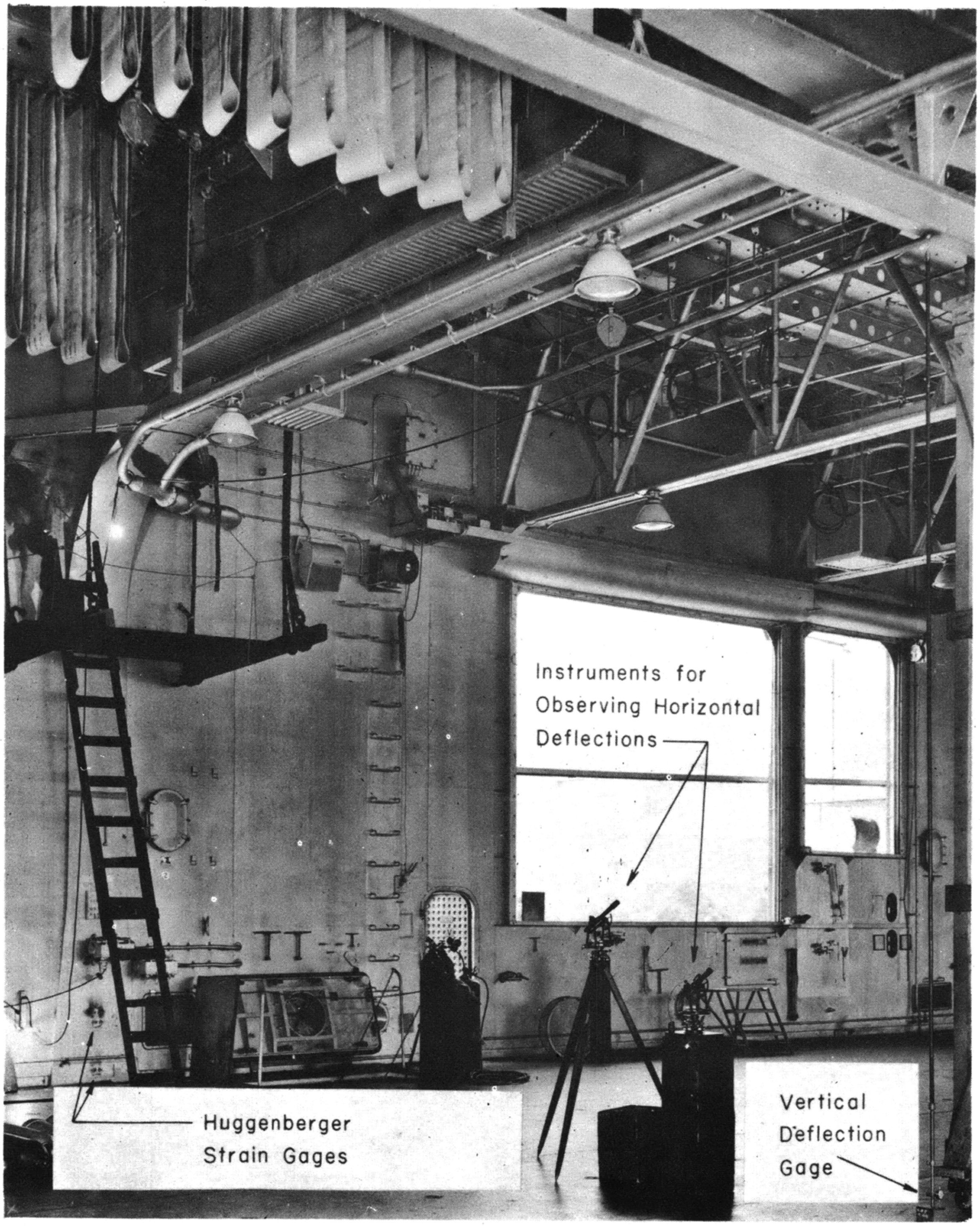

Figure 2 - USS YORKTOWN - Typical Arrangement of Instruments

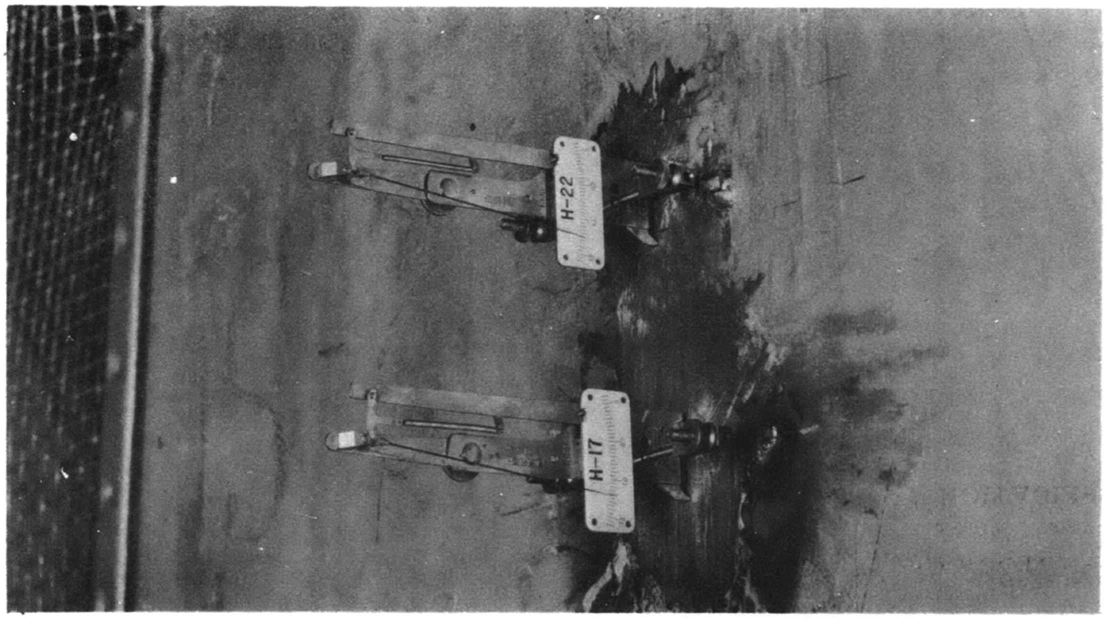

Figure 3 - USS YORKTOWN - Detail of Huggenberger Strain Gages in Place

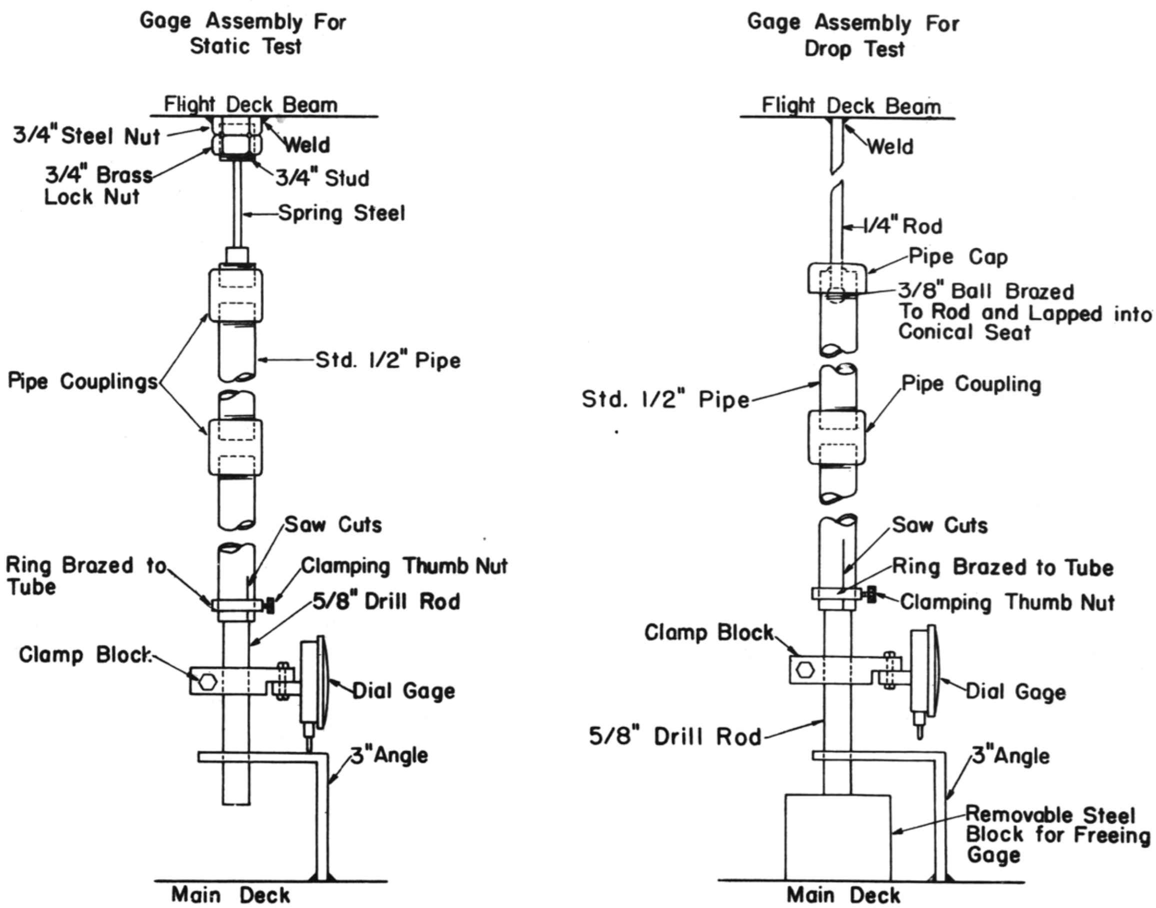

Figure 4 - USS YORKTOWN - Details of Vertical Deflection Gage

Observations by gunner's quadrant and by transit were not successful, as the actual deflections were too small to identify with certainty by these methods.

Special dispositions were necessary in connection with the deflection measurements because of the long reach (about 20 feet) to the reference datum level, the main deck. The problem was solved by hanging extension bars as shown in Figure 4, and placing micrometer dials at their lower ends. A stop at the bottom prevented pendular motion of each bar. Under the static conditions prevailing, good results were obtained, but owing to the slenderness of the extension bar, this device is not suitable where dynamic action is involved, even when the motion consists only of a slight rolling of the ship.

LAYOUT AND PROCEDURE

Tests under static load were made on Bents 142 (Figure 5), 54 (Figure 6), and 155 (Figure 7), and on an intermediate transverse girder at Frame 146 (Figure 8).

Similar static tests were made on the longitudinal centerline girder between Frames 157 and 173 (Figure 9), and on certain starboard longitudinal deck beams between Frames 161 and 165.

The impact load was applied at one station only, near Deck Beam 4, Frame 163, starboard.

Deflections were measured generally at the midpoints and quarter points of the members. Stresses were measured in both flanges near the main deck, on the lower flange at the centerline and at various other stations, as shown in Figures 5 to 9.

LOG OF TESTS

The YORKTOWN tests were conducted 18 July to 22 July 1938 at the Norfolk Navy Yard. The actual experiments were conducted during the evening to avoid interference by workmen on the ship. Sufficient stations were prepared during the day to carry out the tests planned for the evening following.

July 18. Stations were chosen on Bents 54 and 142 and preparations were made for mounting the gages. Bent 54 was loaded by concrete blocks grouped about the midpoint. Strains, vertical deflections, and horizontal displacements of the lower chord were measured. The bent was loaded by increments and readings were taken for loads up to 62,340 pounds. With load still on the bent, the gages were moved to Bent 142 which was then loaded as weights were moved from Bent 54.

July 19. Gages were mounted on intermediate transverse Girder 146 and vertical deflections and horizontal displacements of the lower chord were observed for increments of load applied near the quarterspan point on the port side. The maximum total load was 38,200 pounds. Midspan vertical deflections of starboard Deck Beams 1 to 8 between Frames 161 and 165 were next observed for loads up to 19,000 pounds applied near Frame 163 and Deck Beam 4. Gages were then attached to

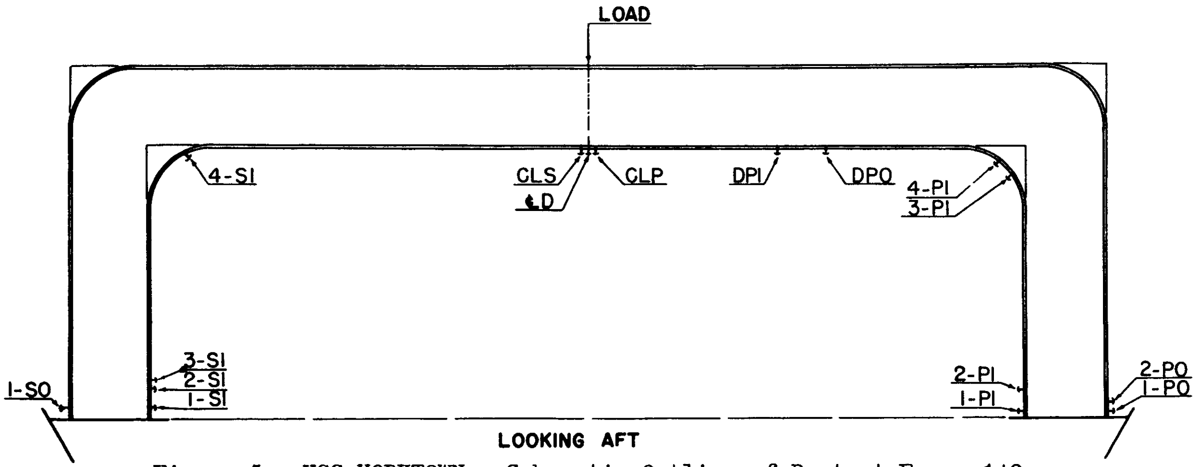

Figure 5 - USS YORKTOWN - Schematic Outline of Bent at Frame 142
Location of Observing Stations

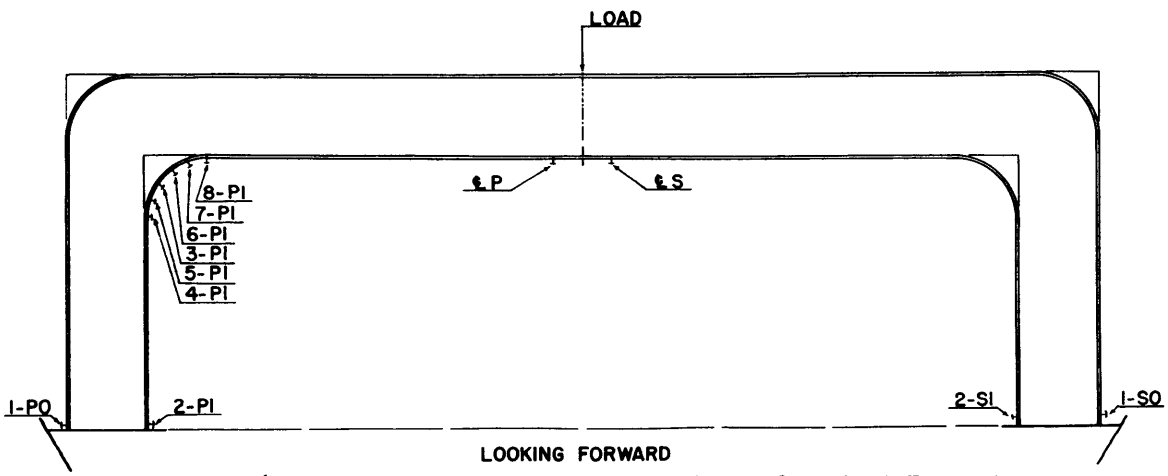

Figure 6 - USS YORKTOWN - Schematic Outline of Bent at Frame 54
Location of Observing Stations

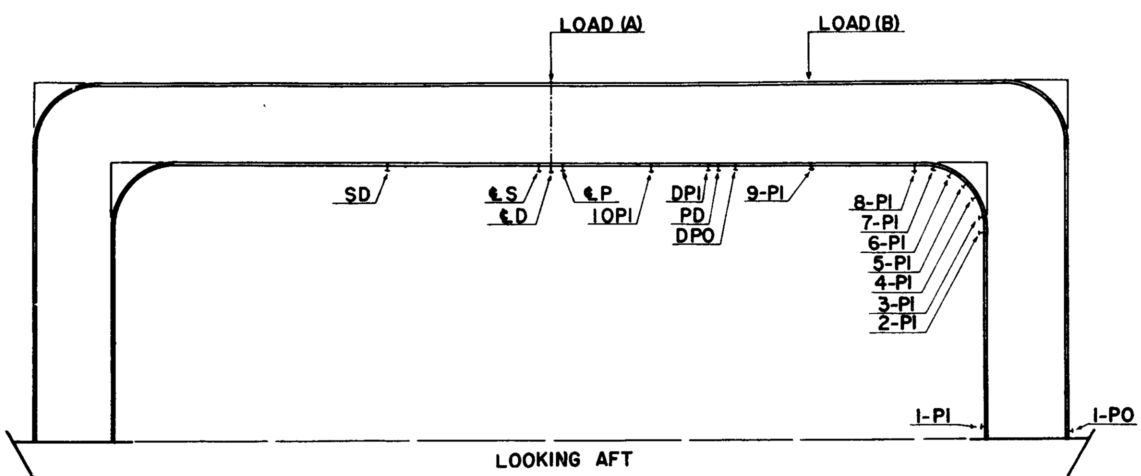

Figure 7 - USS YORKTOWN - Schematic Outline of Bent at Frame 155
Location of Observing Stations

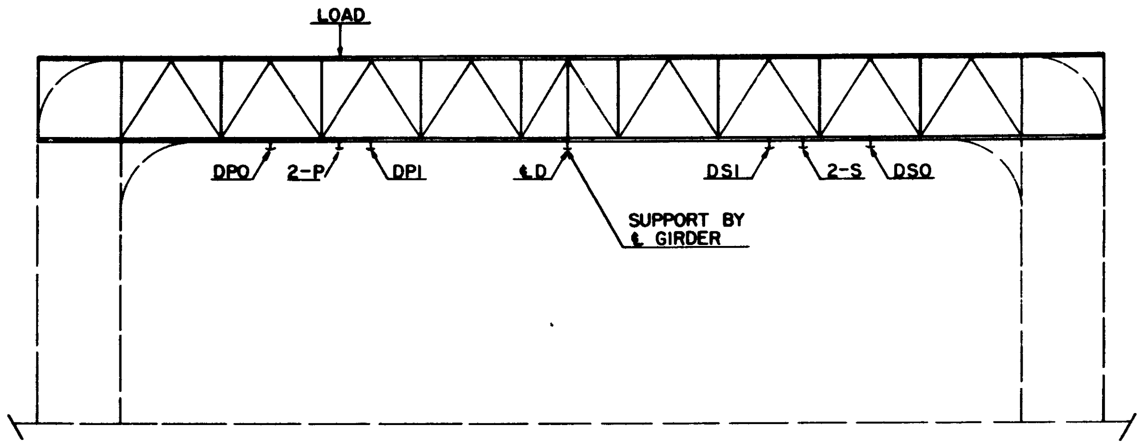

Figure 8 - USS YORKTOWN - Schematic Outline of Bent at Frame 146
Location of Observing Stations

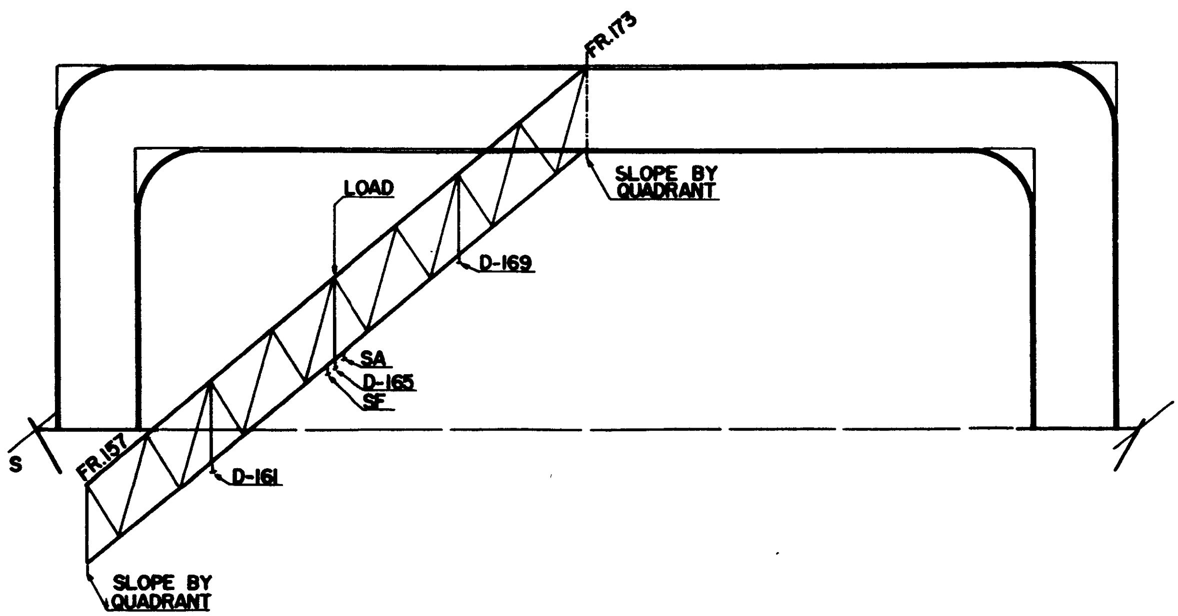

Figure 9 - USS YORKTOWN - Schematic Outline of Longitudinal Girder, Frames 157-173
Location of Observing Stations

In Figures 5 to 9, symbols beginning with a number indicate strain gage stations. The number indicates (with one exception on Frame 54) the order from the deck upward and inboard, the letter following denotes port or starboard, and the third symbol denotes inboard or outboard.

Symbols beginning with D indicate stations for horizontal transverse deflections (mostly found not observable).

Symbols ending with D denote stations for vertical deflections. The preceding symbol shows centerline, port, or starboard.

the centerline girder between Frames 157 and 173. A total load of 41,000 pounds was applied near the midspan by 5000-pound increments, and vertical deflections, strains, horizontal displacements of the lower chord, and rotations by gunner's quadrant were observed.

July 20. The longitudinal girder was unloaded by increments and a check of the previous measurements was made. A weighted airplane fuselage with landing gear was dropped freely near Frame 163 and starboard Beam 4, first from a height of 4 feet, then of 8 feet, and the permanent set produced in starboard Deck Beams 1 to 8 was measured at Frame 163. Special fittings permitted disengaging the dial gages during impact. The position of the deck beams with respect to the main deck was measured before and after impact but no strain measurements were made. Gages were next replaced on Bent 54. Base extensions were used on the Huggenberger instruments to give added sensitivity. Additional concrete blocks brought the total load to 80,440 pounds. Observations were made for each load increment while loading and while unloading.

July 21. Increments of load were applied approximately over the midspan of transverse Bent 155 until a total of 66,400 pounds was attained. Strains, deflections, and displacements were measured while loading and while unloading. Similar observations were made on this bent for a total load of 56,900 pounds over the port-side quarterspan point.

USS WASP - TESTS UNDER VIBRATORY HORIZONTAL LOAD

METHOD

The technique in these tests was a new departure in several important respects:

(a) Load was cyclic, depending on resonance to attain the desired magnitudes.

(b) Stresses, though of very low magnitude, were still measurable by the use of electric amplification.

(c) Phase differences at different stations were observed, in addition to stress amplitudes; these give important additional information.

(d) Local values of deflection amplitude were measured; these also give indications as to the mode of vibration and as to the variation of load with time.

The train of relationships between the quantities involved is quite different from that in a static test, in which load, strain, and deflection are directly measured and corresponding values are directly associated. In the cyclic or dynamic test, only the maxima and minima, or the peak values, are observed, or rather the ranges between them; in other words, only the double amplitudes of variation of the cyclic quantities are measured. A complete analysis would have to include exact knowledge of phase relationships between these quantities at all observation stations, whereas in the present case this was accomplished only to the point of distinguishing between in-phase and out-of-phase relationships.

External load (as in a static test) is represented in a cyclic test only by the alternating action of a mechanical oscillator or vibration generator exerting a force of known range or amplitude, frequency, direction and point of application. This acts on the structure as an exciting source of vibrations.

The responding vibrational motion of the structure, however, is affected by resonance, and so may greatly exceed the deflection which would be caused by a static load equal to the range or amplitude of the exciting load. The ratio by which the amplitudes of deflection and strain are magnified by resonance (known as gain or amplification in radio parlance) must be determined before comparisons can be made between load and stress as in a static test. To care for this situation the term *effective load amplitude* is introduced; this may be defined as the load value which would, if applied statically, cause deflections and strains equal to the observed amplitudes. This term is equal to the actual load amplitude multiplied by what may be called the gain factor.

In the present tests this effective amplitude or load value must be obtained by inference. From the observed value of natural frequency, combined with the known weights, the elastic rigidity, or load per unit deflection, may be calculated, since natural frequency is determined by the elastic rigidity and by the inertia in any elastic system. Thus a mass on a spring will vibrate at a frequency which is higher as the spring is stiffer, and lower as the mass is greater. The effective load value may then be obtained by multiplying the observed deflection amplitude by this calculated value of rigidity (see Figure 22).

The basic problem in the design of a structure of this kind is to so dispose the metal that a given load causes a stress not exceeding an assigned limit. It is the aim of the designer to obtain an assigned value of stress under a given load; this is a wholly separate task from load evaluation. This test affords a measure of the stress per unit load. The value of the ratio of stress to load given by the test is the ratio of observed stress to the effective load as described in the foregoing.

Pertinent information as to the characteristics of the structure is obtainable by this method, however, even without separate evaluation of load and stress, provided only that stress-amplitude values at different stations may be assumed to be in the same ratio to each other as under a given static load.

Details of design of the bents depend strongly on the assumed degree of fixation at the foot where it is built into the hull structure. Information on the fixation actually obtained may be drawn from the relative values of strain amplitude at different heights above the deck; in particular a point of inflection is marked by the zero value of bending moment, and hence of strain amplitude; see Figure 18, page 20. The position of this point of inflection may be determined on a basis of proportionality. Similarly, observation of the pattern or mode of vibration requires only that successive observations at different stations be made while the amplitude of response of the structure is maintained at a constant value.

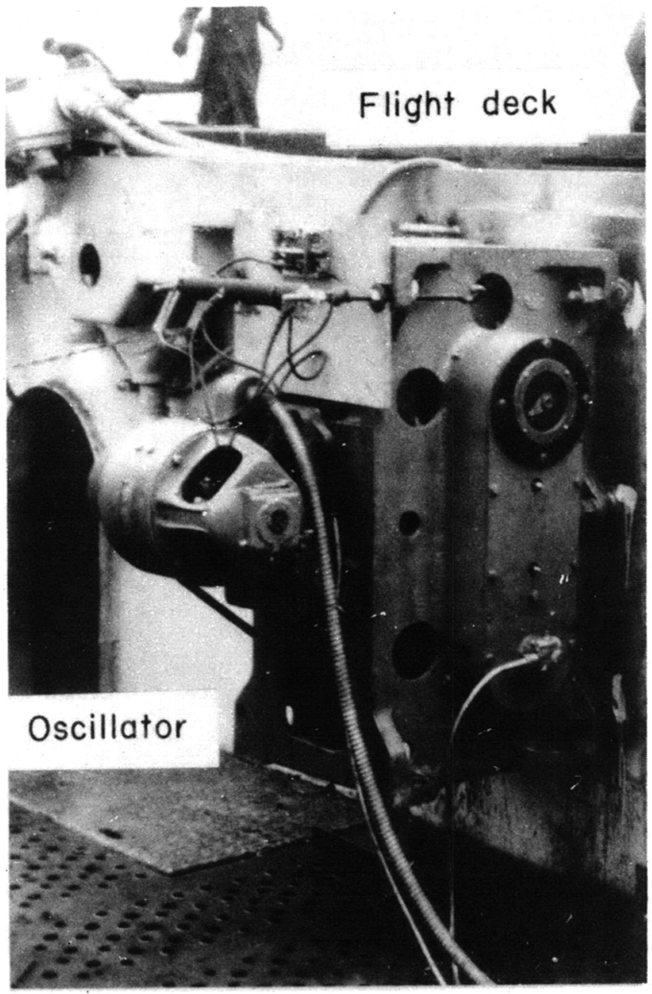

Figure 10 - USS WASP - Installation of Vibration Generator

INSTRUMENTAL EQUIPMENT

Mechanical Oscillator

The vibration generator used in these tests was a Baldwin-Southwark mechanical oscillator borrowed from the Rock Island Arsenal; see Figures 10 and 11. It consists of a frame carrying two parallel shafts permanently geared together so as to rotate in opposite directions. On each shaft is mounted an adjustable eccentric mass. The eccentricity of the center of mass can be varied from zero to 2-1/2 inches by eccentric rotation of the mass on its shaft from 0 to 180 degrees. The shafts are driven by a 2 HP, DC shunt motor through a system of pulleys and V-belts, permitting choice among several different speed ratios. The reactive components taken

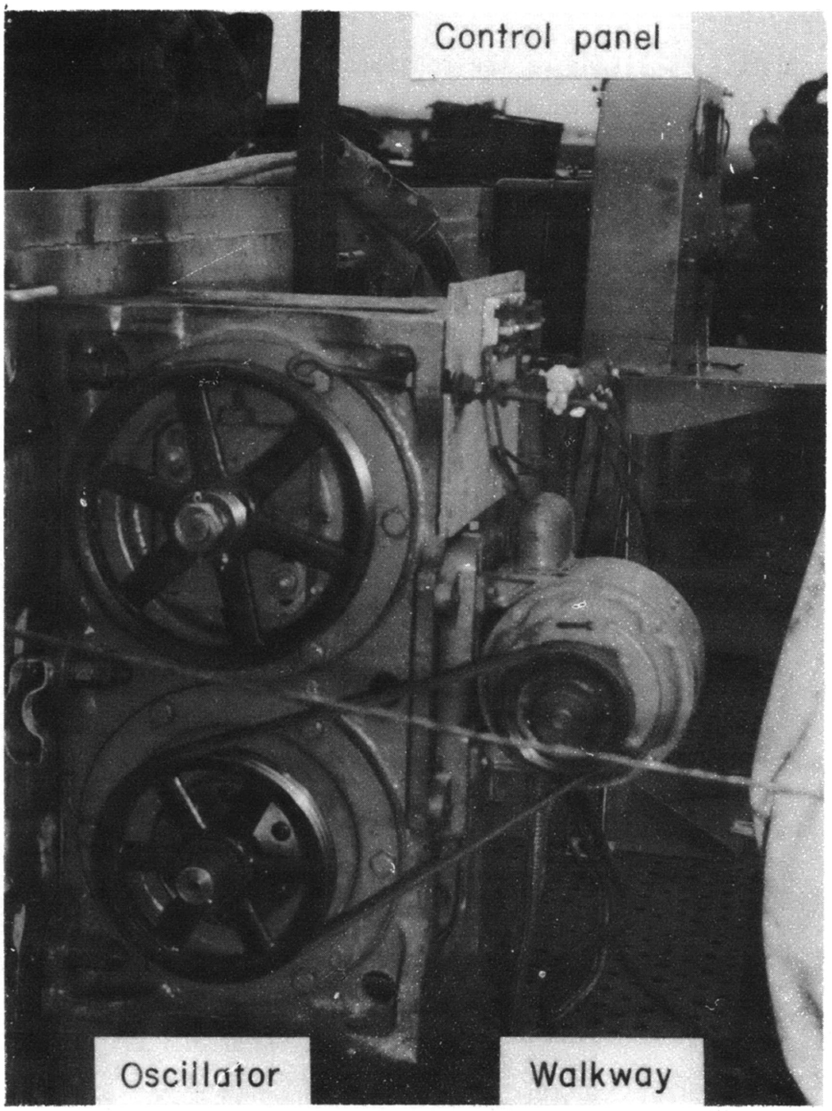

Figure 11 - USS WASP - Detail of Vibration Generator

parallel to the base of the machine are internally balanced against each other, while the components at right angles to the base are additive. Hence the generator sets up a sinusoidal force in one direction only.

The oscillator was mounted outboard just below the flight deck and in line with the bent at Frame 54. The method of mounting, as shown in Figures 10 and 11, produced an athwartship vibration of Section IV of the flight deck. The frame of the machine was held securely against the side plating by means of bolts welded to the plating. Heavy bracket plates were added between the inboard side of the plating and the upper face plate of the bent so as to insure transmitting the load from the oscillator directly to the web of the bent. This reinforcement is shown in Figure 12.

The double amplitude of the reaction of this machine on the structure is 27,800 pounds at resonant speed of 500 cycles per minute.

Vibrometers

Two instruments were used for measuring vibration amplitude, a horizontal-component pallograph developed at the Experimental Model Basin in connection with hull vibration studies, and a Karelitz-Type vibrometer manufactured by the Vibration Specialty Company of Philadelphia. The pallograph, shown in Figure 13, gives a record on waxed paper which may be analyzed for frequency as well as amplitude; a time record is obtained by connecting the timing magnet to a chronometer. The vibrometer

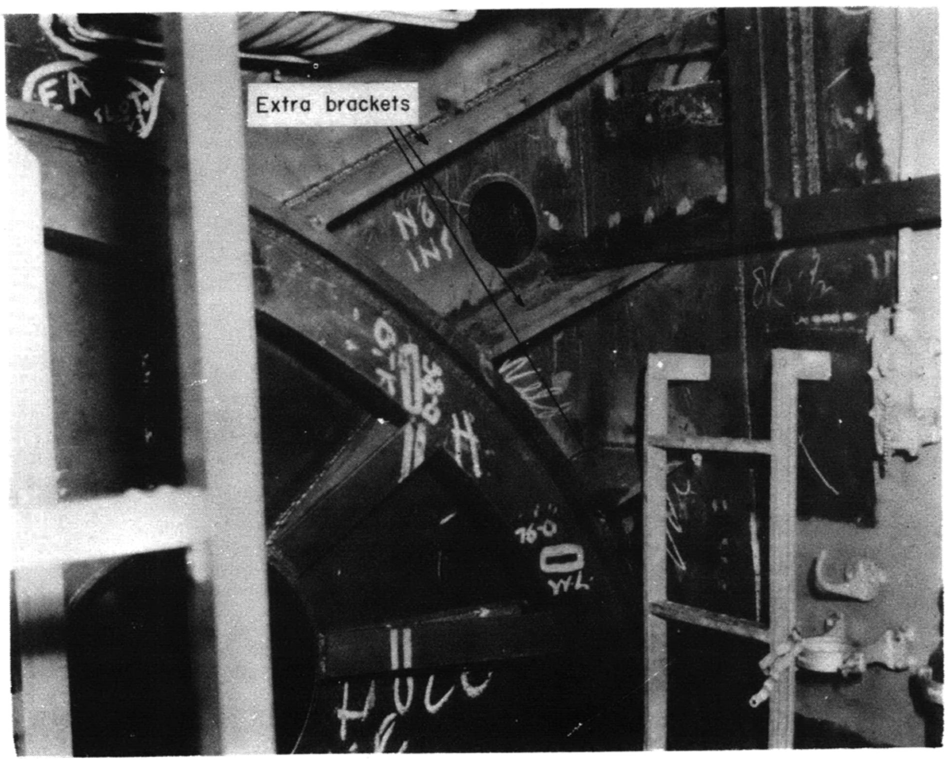

Figure 12 - USS WASP - Structure at Outer Flange of Knee with extra Brackets in way of Oscillator

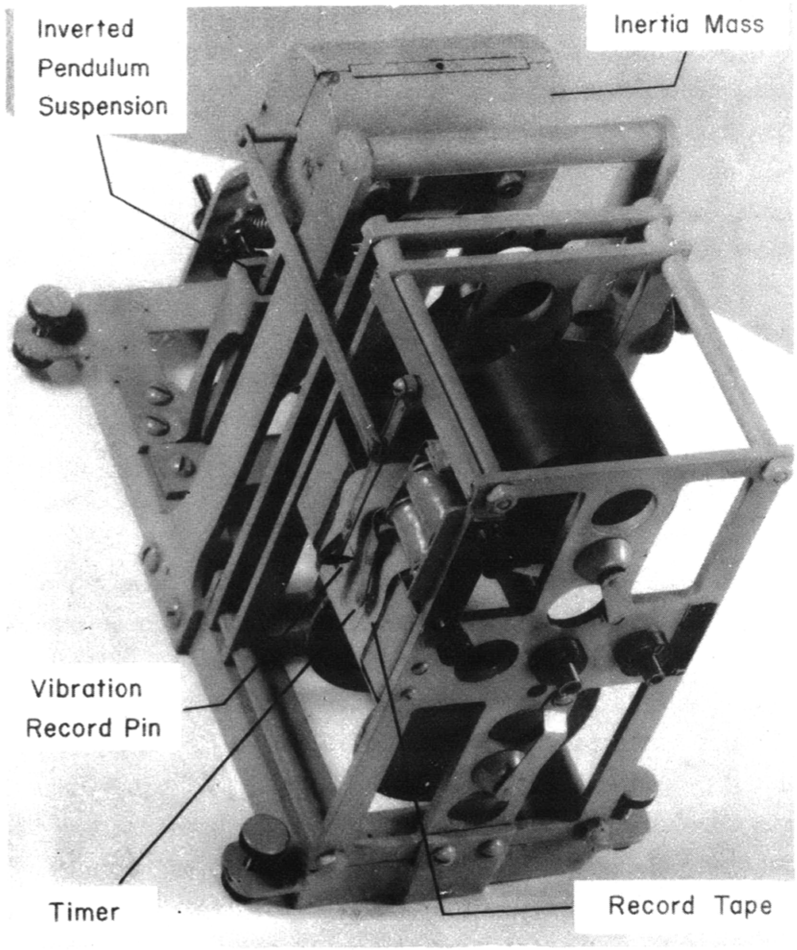

Figure 13 - EMB Pallograph

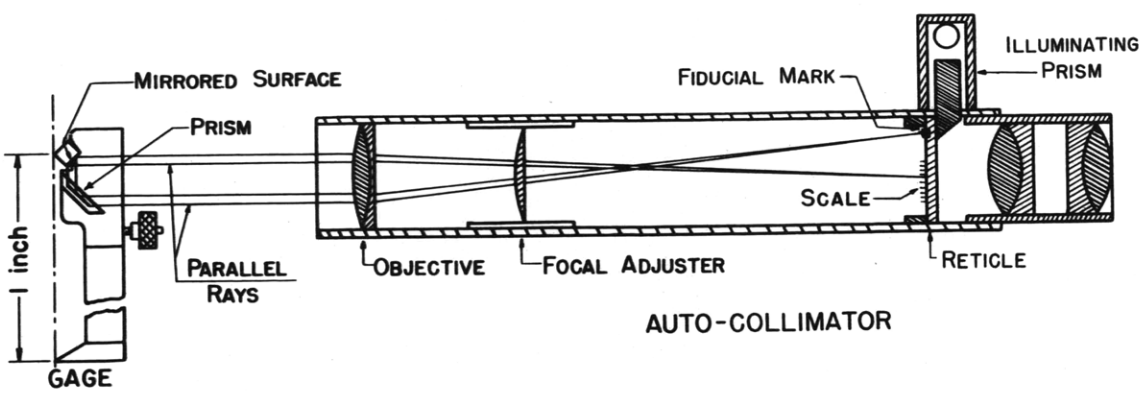

Figure 14 - Tuckerman Optical Strain Gage
Diagrammatic Arrangement of Gage and Collimator

is a visual instrument consisting of a spring-suspended seismic element and two dial micrometers, the stems of which bear against the element. The observer notes the limits of travel of each pointer, which gives the double amplitude of vibration. With this instrument two components of vibration can be read simultaneously, one vertical and one horizontal.

Observations of amplitude of vibration were also made by electrical means, but these will be described in connection with strain measurements.

Mechanical Strain Gages

Strain measurements were made with Tuckerman optical strain gages. As shown in Figure 14, the two essential elements of this apparatus are the gage and the auto-collimator. The auto-collimator is an independent unit and can be used to read a number of gages in turn. It projects a beam of parallel light rays onto the gage, where a fixed prism coupled with a mirror reflects this light back onto a scale fixed within the auto-collimator. The mirror surface rotates with the relative motion between the gage points and so moves the indicator or "fiducial mark" across the fixed scale by an amount proportional to the strain to be observed.

The gages were fitted with extensions so that they could be used on an 8-inch base. Rubber bands fastened to clips welded to the structure held the gages firmly against the metal surface, as shown in Figure 15. Since only two gages of this base length were available, the method of taking readings was to set up the gage at each station, observe the reading and the time, and then move the gage to the succeeding station and repeat. The readings were estimated to tenths of a scale unit after the movement of the image on the scale had been observed for about one minute. The fiducial mark used in the auto-collimator for these measurements produced a black-line image with light background on the scale. The amplitude of motion of this line was observed. The reading accuracy was about $\pm$ 0.1 scale division for readings up to 1.0 scale division, with greater possible errors where varying strain amplitudes were encountered. One scale division represents a stress of 150 pounds per square inch in steel.

Electric Strain Gages

Measurements of strain with electric gages were made by three parties; one from the Naval Aircraft Factory (NAF), one from the laboratory of Professor deForest of the Massachusetts Institute of Technology (MIT), and one from the Experimental Model Basin (EMB).

(a) NAF Gages

The equipment used by the NAF party to obtain strain data was developed under an NAF-RCA project. A description and photographs of this apparatus are available in the Bureau of Aeronautics.

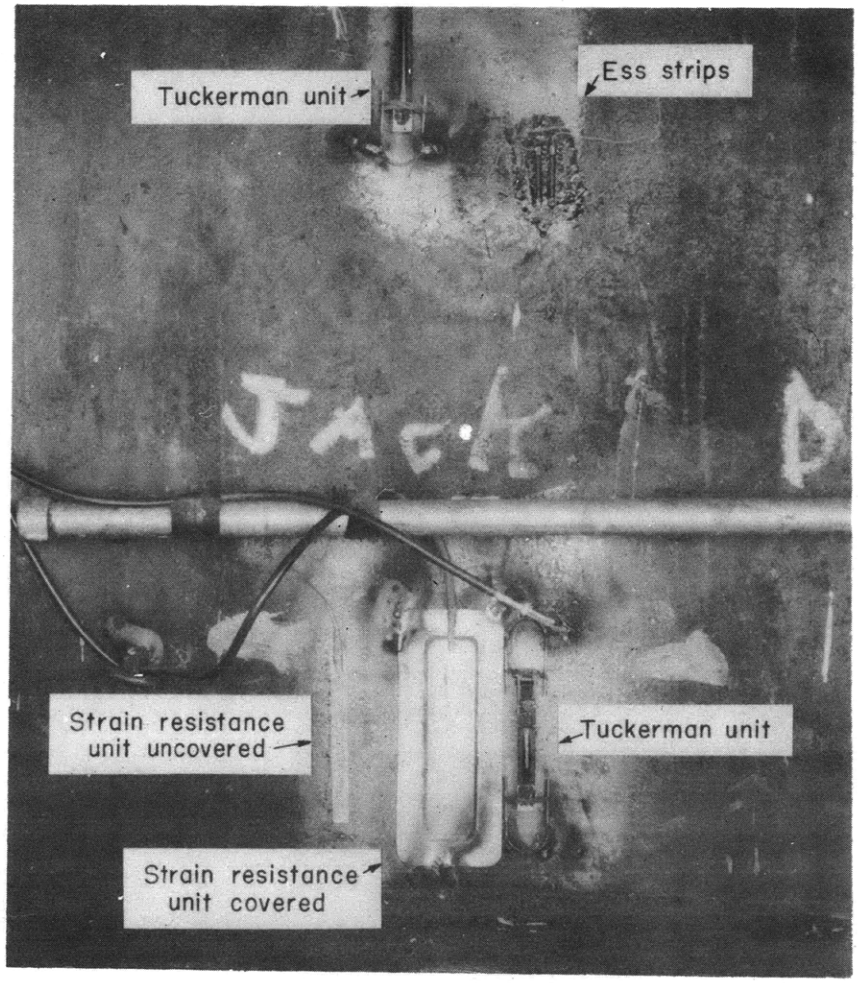

Figure 15 - USS WASP - Installation of Gages near Main Deck,
Frame 54 Starboard

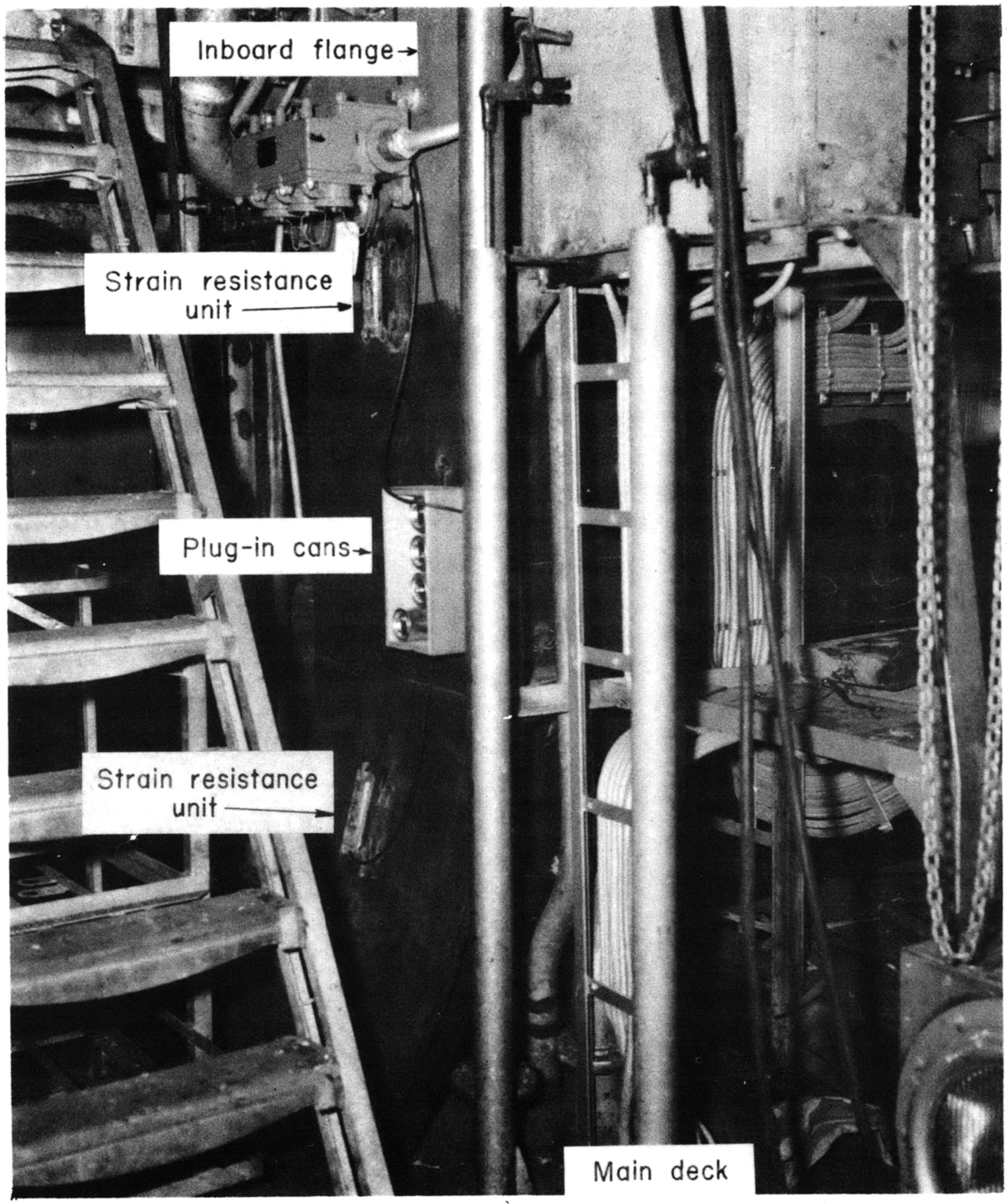

Figure 16 - USS WASP - Installation of Gages at Frame 38 Starboard on Inboard Flange of Flight-deck Bent

The strain pickup, installed as part of a bridge circuit, modulated a 3000-cycle per second carrier wave. The modulation was proportional to displacement or strain. This modulated wave was amplified and the carrier wave filtered out. The resulting signal was recorded on an RCA universal single-channel sound track recorder.

The NAF party also took data on amplitudes of vibrational motion, using instruments developed under a joint project of the Bureau of Aeronautics and the Massachusetts Institute of Technology, and known as the Sperry-MIT vibration measuring equipment. The pickup units operate on a seismograph principle. The relative velocity between the seismic element and the case, which is attached to the vibrating member, is transformed into an electrical signal by means of small electromagnetic generators incorporated in the pickup. This signal, which is proportional to velocity, is fed into an integrating amplifier, which gives an output signal proportional to displacement or amplitude of vibratory motion. This output is recorded on a photographic recording oscillograph. In the tests, two channels of a four-element oscillograph were employed.

(b) MIT Gages

Strain pickups of a resistance-sensitive type, known commercially as "Metalectric gages," were used by both the MIT and EMB parties. These gages are made of iso-elastic wire and have a resistance of 600 ohms. Sixteen inches of wire are laid down in W fashion over a gage length of four inches. The wire folds are secured temporarily to tissue paper for convenience in transferring to the structure, where they are finally secured by celluloid cement.

These gages were used in the basic circuit shown in Figure 17.

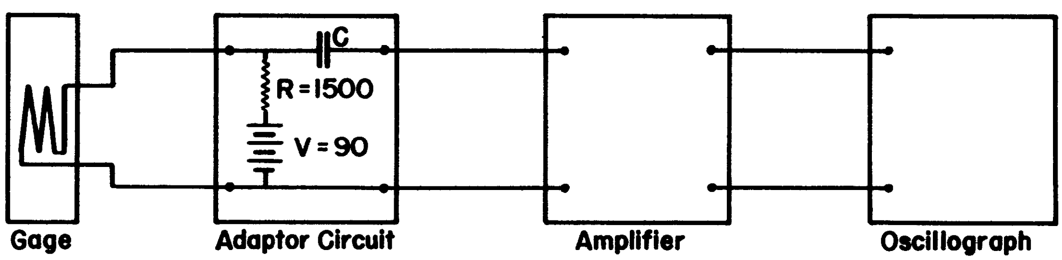

Figure 17 - Resistance-sensitive Strain Gages, Basic Circuit

As the periodic or cyclic load is applied to the structure, the strain is transferred to the gages. The resulting changes in the lengths of the wire folds of the gage change its resistance periodically, producing a variation in the voltage drop across the gage. This variation is transmitted as an AC signal by the blocking condenser C, and amplified by a battery-operated amplifier. The output of the amplifier is observed on an RCA-Type TMV-22B cathode-ray oscillograph. The double amplitude of the output signal is measured directly on the screen of the cathode-ray tube. This reading is proportional to the double amplitude of the strain.

By taking precautions with these instruments, a very low level of internal noise was attained. The matter of shielding required careful attention. Figure 15 shows one of the gage units uncovered, and adjoining it a cover in place over another unit. The covers were shallow-bumped shields of thin metal, welded in place. The gage leads were connected under the shield to shielded cables, and these cables for all the gages in a given group were brought together to a nest of "plug-in cans" as shown in Figure 16. These "plug-in cans" were telephone jacks mounted in grounded cylindrical shields. The oscillograph leads from the central shelter could thus be shifted at will from one gage to another without loss of shielding integrity. An additional advantage of this arrangement is that it permits interchangeable use of the gages with either the MIT or the EMB indicating instruments.

Phase measurements were made by applying the amplified output of one gage to the horizontal-deflection plates of the cathode-ray oscillograph and the amplified output of each of the other gages in turn to the vertical-deflection plates. The change from in-phase to out-of-phase was indicated by a shift of this diagonal line as in the figures of Lissajou.

(c) EMB Gages

The EMB party made use of the Metalectric strain pickups previously described. A circuit similar to that shown in Figure 17 was used, except for the following changes: A Ballantine vacuum tube voltmeter was used as the indicating instrument instead of an oscillograph; also, the values of R and V in the adaptor circuit were changed to approximately 600 ohms and 45 volts, respectively.

The root-mean-square value of the output signal voltage was read directly on the scale of the vacuum tube voltmeter. This reading was proportional to the maximum amplitude of the strain, and the constant of proportionality was determined by dynamic calibration.

CALIBRATION

The amplifiers and oscillographs for use with the MIT electrical resistance gages were checked at the time of the WASP tests by means of a portable calibrator. The calibrator consisted of a steel cantilever beam, 1/8 inch deep by 1/2 inch wide and 10-1/2 inches long, subjected to alternating bending loads by an eccentric driven by a variable speed electric motor. For this calibration the cantilever was vibrated at the same frequency as the deck. An electrical resistance gage was mounted near the fixed end of the beam. The static stress range induced at this point was first determined by a mechanical gage as pounds per square inch, and the dynamic range was then assumed to be the same. Later calibrations proved this assumption correct to within 5 per cent. Speeds were determined by a Strobotac.

A similar calibrator large enough to accommodate both mechanical and electrical gages at the same time was built. A cantilever beam 1/2 inch deep by 6 inches wide by 5 feet long was used. An eccentric with stroke variable from 0 to 1 inch induced any desired stress range in the beam from 0 to 10,000 pounds per square inch. A variable speed 1/4-horsepower motor drove the eccentric, and the frequency was obtained by counting the RPM of the motor. An electrical resistance gage and a Tuckerman gage were placed side by side near the fixed end of the beam and the simultaneous readings in Table 1 were taken.

TABLE 1

Calibration of Resistance-Sensitive Strain Gage

Reading Number	Tuckerman Gage Reading Stress Range in pounds per square inch	MIT Electrical Gage Reading Stress Range in pounds per square inch	Ratio of $\frac{\text{electrical gage reading}}{\text{mechanical gage reading}}$
1	320	420	1.31
2	600	610	1.02
3	6400	7150	1.12

Some discrepancies exist between these data. However, the strains measured on the ship were very small, and constancy in calibration ratios is of greater moment than the actual values. A closer agreement could no doubt be obtained with a more thorough calibration at larger stress ranges. The MIT party reported better calibration results than were actually obtained on the job.

The calibration constants for the NAF electrical strain gages were predetermined by checking against a Tuckerman gage on a similar calibrator. The NAF vibration pickups were calibrated on a linear vibration calibrator which was developed as part of the equipment.

TEST PROCEDURE

Preliminary runs were made with the vibration generator to determine the principal resonance points and the best pulley arrangement to cover the required speeds. Two principal natural frequencies of the structure were found, at about 450 CPM and 500 CPM. The eccentrics were then set at their maximum permissible offset and run continuously for about seven hours at or near 500 CPM, and then for about 3 hours at or near 450 CPM, while stress observations were made for each speed.

The pallograph was set on the hangar walkway in the vicinity of the oscillator and a continuous record was made during the test. From the analysis of these records a tabulation of frequencies and amplitudes was made at intervals of five minutes. As a record of time was made at every strain observation, this made possible a correlation between strain and amplitude.

Amplitude observations were made also on the flight deck and on the main and lower decks by means of the Karelitz vibrometer.

A shelter was provided in a central location on the main deck as operating space for oscillographs and as a general headquarters. Movable shielded leads were taken from this shelter to the various gage stations so that the oscillographs, which were few in number, could be connected in turn to the gages at the various stations. Readings from the different stations were thus successive, and as the work extended over a period of several hours, comparability was obtained only by special effort. Perfect uniformity in frequency and amplitude of vibration were not obtained, but the variations are known and can be allowed for. As an additional precaution onè gage was adopted as a standard for comparison, and readings of that gage were taken at frequent intervals throughout the series.

MODES OF VIBRATION

The observations on the WASP flight deck disclosed the presence of two different modes of vibration at the two natural frequencies mentioned. These have been distinguished as "table-top" and "bowstring" modes, as shown in Figure 18. In the table-top mode, the vertical component of vibration of the bent shows nodes at the sides and center of the deck. In the bowstring mode, the nodes are at the sides only, with a loop in the center. As for the horizontal component of vibration, there should be no nodes for the table-top mode and a node in the center for the bowstring mode, as the ends vibrate in opposite phase.

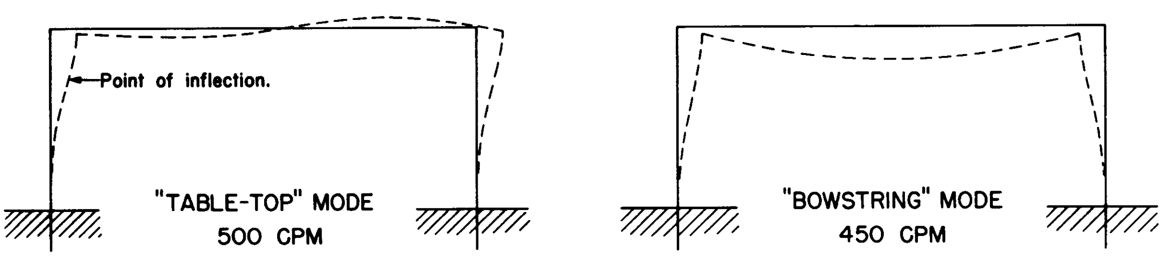

Figure 18 - Modes of Vibration of Flight Deck-Bent

While the presence of two different modes was clearly indicated, the nodes were indistinct. Two reasons for this are suggested. In the first place damping was considerable, as shown by the rather low resonance amplification discussed later. This meant that the resonance curves for both modes were quite broad, especially that at 500 CPM, and since the two frequencies were close together, there was overlap.* Moreover the table-top mode was more readily excited than the bowstring because of the position of the vibrator. Hence both modes were present at all times; the table-top was strongly predominant at 500 CPM, and the bowstring less so at 450 CPM. In the second place the bents are not perfectly free to vibrate in their natural modes because

* Response in one of the two modes under excitation at the frequency corresponding to the other mode.

of the restraining action of the wood flight deck. At 450 CPM, where the bowstring vibration would have required a node in the center for the horizontal component, the horizontal amplitudes were practically constant all the way across the flight deck at Frame 54, and in fact were fairly uniform over the whole section of the flight deck at this frequency, whereas at 500 CPM they were greater at Frame 37 than at Frame 66. The vertical component at 500 CPM was exceedingly small except directly over Bent 54 where there was a maximum of about 0.003-inch double amplitude with a distinct decrease in the center. Because of the absence of a horizontal node we must conclude that the table-top vibration figured largely also at 450 CPM.

At the ends of Section IV, measurements were made of the relative movement between this and the adjoining flight-deck sections. This was accomplished by placing a Whittemore strain gage so that one point rested on Section IV and the other on the adjoining section. The relative amplitude thus measured checked very well with the difference in the amplitudes measured on the two sides of the expansion joint by means of the vibrometer.

Vibration was evident on the main deck at 500 CPM in the vicinity of the bents, but the vibrometer indicated less than 0.001-inch amplitude either vertically or horizontally. Only a few ten thousandths of an inch amplitude could be detected in the vicinity of Bents 37 and 66.

LAYOUT OF GAGE STATIONS

Strains were measured at stations on the vertical legs of the bents at Frames 37, 54, and 66 (Figures 19, 20, 21). These three bents support the structure to which the oscillator was attached. A typical gage station is shown in Figure 15. The principal objects of the strain measurements were to determine the degree of fixity of the bents at the main deck by locating the points of inflection of the stresses in the legs of the bents, and to obtain an indication of the magnitude of these stresses for a given deflection of the flight deck. The stations were accordingly chosen as follows: On each of four of the legs, three stations were located on the inboard flange opposite the web, one as near the main deck as practicable, the others approximately 6 feet and 15 feet above the main deck. On the starboard legs of Bents 37 and 54, stations were located at approximately 2-foot intervals on the inboard flange opposite the web, and three stations were located on the inboard side of the outboard flange of Bent 54 about one foot forward of the web. This gave a more complete picture of the stress variations in these two legs than in the others, and also served as an aid in analyzing the observations on the remaining legs.

Station symbols of the following type were used: 54-is-6. The first numeral indicated the bent (Bent 54), the letters indicated the side of the leg and the side of the ship (inboard side of starboard leg), and the last numeral represented the height above the main deck to the nearest foot (6 feet above deck). The exact heights are tabulated.

TABLE 2

USS WASP - Layout of Strain-Gage Stations

Station	Height above main deck, feet and inches	Station	Height above main deck, feet and inches
54-is-0	0-4	38-is-2	2-1
54-is-0	0-5 (NAF)	38-is-2	1-11 (NAF)
54-is-2	2-2 (NAF)	38-is-4	4-2 (NAF)
54-is-2	2-3 (NAF)	38-is-6	5-8 (NAF)
54-is-3	3-2	38-is-6	6-0
54-is-4	4-2 (all gages)	38-is-7	7-0 (NAF)
54-is-6	5-8 (all gages)	38-is-9	9-0 (NAF)
54-is-7	7-2	38-is-11	11-5 (NAF)
54-is-7	7-1 (NAF)	38-is-17	16-10
54-is-10	9-9 (all gages)		
54-is-12	11-11	38-ip-3	2-7
54-is-12	11-6 (NAF)	38-ip-7	6-7
54-is-15	14-11 (all gages)	38-ip-17	16-11
54-os-2	2-2	66-is-1	0-8
54-os-6	5-8	66-is-6	5-10
54-os-14	14-3	66-is-16	16-1
54-ip-1	1-0	66-ip-1	0-7
54-ip-7	7-3	66-ip-6	5-11
54-ip-15	15-1	66-ip-15	14-10

(NAF) indicates Naval Aircraft locations.
No mark indicates Tuckerman and MIT gages only

LOG OF TESTS

The WASP tests were conducted 8 January to 16 January 1940, inclusive, at the plant of the Bethlehem Shipbuilding Corporation, Quincy, Mass.

January 8. The vibration generator was installed and preliminary runs were made to make sure the flight deck could be vibrated with this machine.

January 9. Further vibratory tests were performed with eccentricity settings of 40 and 90 degrees. Gage stations were chosen on the legs of the bents and preparation of the surfaces for mounting strain gages was begun.

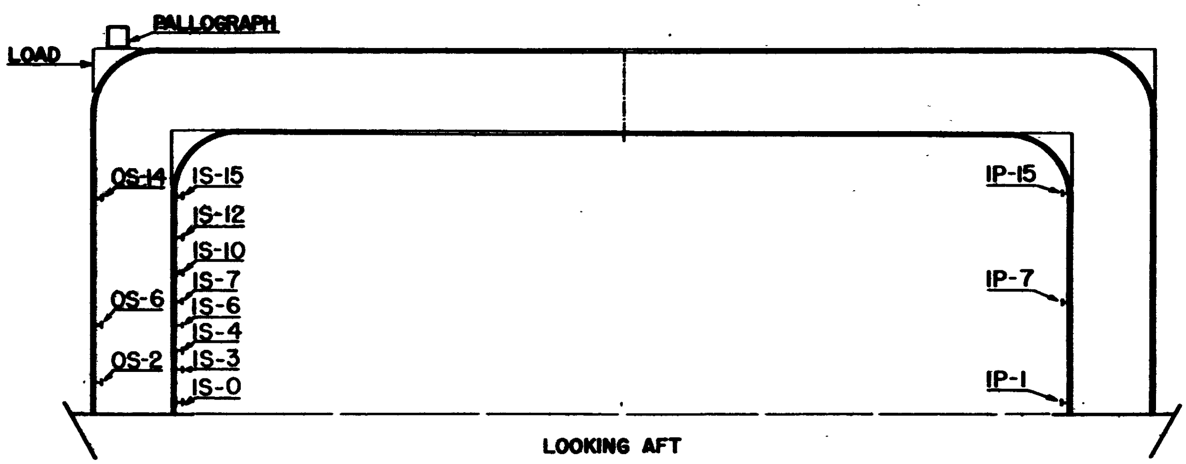

Figure 19 - USS WASP - Frame 54

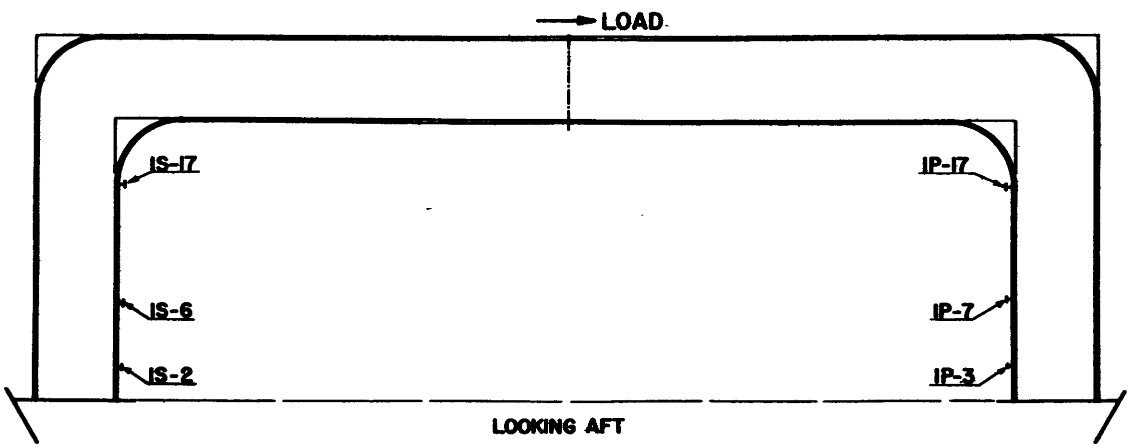

Figure 20 - USS WASP - Frame 38

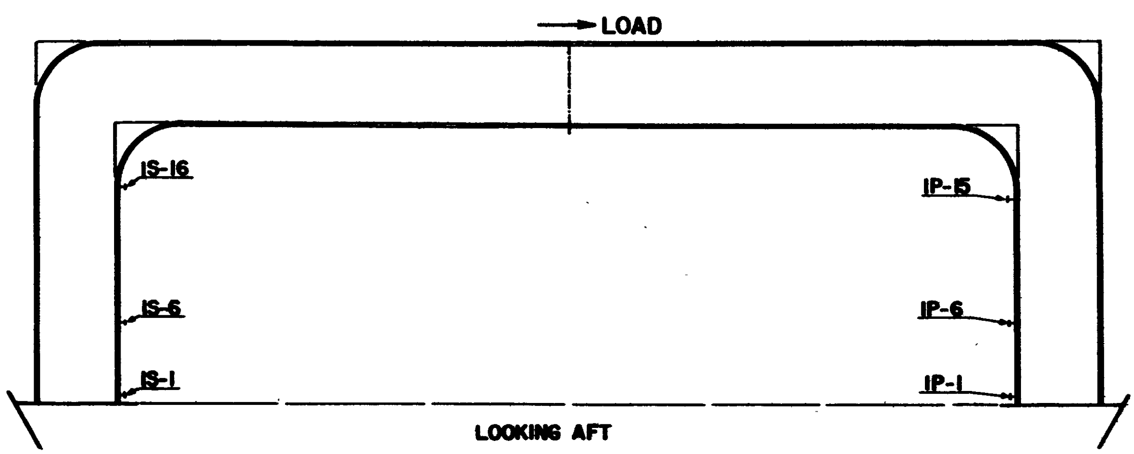

Figure 21 - USS WASP - Frame 66

Schematic Outline of Bents showing Location of Observing Stations.

January 10. Eccentricity on the vibration generator was increased to 120 degrees. On a two-hour run a survey of vibrations on the flight deck was made with a portable vibrometer. Preparation of strain-gage stations was continued. Assembly of electrical gage apparatus was begun.

January 11. Eccentricity of the vibration generator was increased to 180 degrees, the maximum possible, and a test of amplitude variation with frequency was made. The two distinct modes of vibration at 450 CPM and 500 CPM were established. Preparation of strain-gage stations was continued; clips for holding Tuckerman gages were welded to the structure, and the laying down of electrical resistance strain gages was begun.

January 12. Remaining strain-gage preparations were completed. NAF party arrived and their gages were mounted.

January 13 and 14. Tests by NAF personnel were begun. These included measurement of mode of vibration, and measurement of strains with RCA pickups.

January 15. NAF tests were completed. EMB and MIT final measurements were begun. The oscillator was operated continuously throughout the morning and afternoon at or near 500 CPM. A pallograph furnished a continuous record of the variation of vibration amplitudes with time. Strain measurements were taken with both Tuckerman gages and the electrical resistance gages and the time of each reading was noted. Readings of the electrical gages were taken on both the MIT cathode-ray oscillograph and on the EMB vacuum tube voltmeter. A survey of the vibrations of different parts of the flight-deck section was made by means of the portable vibrometer, and the relative motion of this section with respect to the two adjoining sections was made by means of a Whittemore strain gage. In the evening, the vibration frequency was changed to 450 CPM and the readings were repeated.

January 16. Tuckerman readings at 450 CPM were completed. Apparatus was dismantled and prepared for return shipment.

ORIGINAL DATA

1. USS YORKTOWN

Since all data on the YORKTOWN were taken by the agency responsible for their analysis, it is considered not necessary to record them in their original unaltered form, but only as reduced for analysis and tabulated in the section headed "DATA REDUCED FOR ANALYSIS."

2. USS WASP

The original strain data converted to stress units as reported are given in Tables 3 and 4 without change. NAF values refer to single amplitude, MIT and EMB values to double amplitude. All EMB data have been corrected for variation in amplitude and reduced to the equivalent for double amplitude of 0.010 inch at the pallograph. T indicates Tuckerman gage read by EMB personnel.

TABLE 3

USS WASP - Range of Stress derived from Strain Gage Observations

All values are in pounds per square inch

Station	Height above main deck, inches	Table Top Mode					Bowstring Mode	
		NAF*	MIT	EMB	TJ	TW	MIT	TJW
Frame 38 Starboard Inboard								
is2	24	150	288		219	170	128	90
is4	50	85						
is6	70	47	202	139	125	80	96	45
is7	84	67						
is9	108	3						
is11	137	3						
is17	203		–202	–191	–135	–140	–96	–60
Frame 38 Port Inboard								
ip3	31		–260	–195	–150	– 93	–96	–45
ip7	79		–144	–116	– 60	– 70	–64	30
ip17	203		+320	+210	+165	+175	+176	+45
Frame 54 Starboard Inboard								
is0	4	+100	+246	145	+140	+150	+248	150
is2	26	85			82	82		75
is4	50	50			55	40		45
is6	68	30	130	96 82	34	32	128	52
is7	86	45	116	73 64	34	40	104	45
is10	117	5	74	50	20	22	72	25
is12	142	+5	+ 14	0±40	+ 7	+ 15	+ 48	20
is15	179	– 70	–116		–30	– 55	– 88	0
Frame 54 Port Inboard								
ip1	12		–202		–156	140	–160	120
ip7	87		–116		– 30	40	– 72	30
ip15	181		+ 42		+ 15	+ 15	+ 72	+45
Frame 54 Starboard Outboard								
os2	26		–130			- 45	– 64	
os6	68					52		
os14	171		+ 74			+ 15	+ 24	
Frame 66 Starboard Inboard								
is1	8			111	112			
is6	70				45			45
is18	193				–30			–75
Frame 66 Port Inboard								
ip1	6			107	65	75		142
ip6	71			67	37	30		90
ip15	176			0±40	– 7			– 8

* Single amplitude.

TABLE 4
USS WASP - Phase of Stresses with respect to Station is0 on Bent 54, MIT data

Station	Phase relations at 500 CPM	Phase relations at 450 CPM
	Frame 38 Starboard Inboard	
is2	in phase	out of phase about 90^o
is6	in phase	out of phase about 90^o
is17	out of phase	out of phase about 90^o
	Frame 38 Port Inboard	
ip3	out of phase	out of phase about 90^o
ip7	out of phase	out of phase about 90^o
ip17	in phase	out of phase about 90^o
	Frame 54 Starboard Inboard	
is6	in phase	in phase
is7	in phase	in phase
is10	in phase	in phase
is12	in phase	in phase
is15	out of phase	out of phase
	Frame 54 Starboard Outboard	
os6	out of phase	out of phase
os14	in phase	in phase
	Frame 54 Port Inboard	
Lp1	out of phase	out of phase about 90^o
Lp7	out of phase	out of phase about 90^o
Lp15	in phase	out of phase about 90^o

TABLE 5
USS WASP - Phase of Horizontal Motion of Flight Deck referred to
Station over Starboard Column of Bent 54

Station Location	Phase Relations at 500 CPM	Phase Relations at 450 CPM
	Frame 38	
Over Starboard Column Over Port Column	In Phase In Phase	Out of Phase Out of Phase
	Frame 54	
Over Port Column	In Phase	In Phase
	Frame 66	
Over Starboard Column Over Port Column	In Phase	Out of Phase

Phase relationships on the flight deck, as observed over the six supporting columns by the NAF party are given in Table 5.

In tabulating and plotting stress data, the top station on each column has been assumed out of phase with the lower stations for both bowstring (450 CPM) and table-top (500 CPM) modes of vibration. This assumption seems justified since this relationship was indicated in all cases where phases were determined.

DATA REDUCED FOR ANALYSIS

LOAD, RIGIDITY, AND FREQUENCY

Load in the static YORKTOWN tests was directly observed and needs no comment.

In case of the vibratory WASP tests, evaluation of effective load needs special attention. Since the vibrational data on bowstring deflections are similar to the static data but not so good, no further use will be made of the data taken at 450 CPM.

Load is to be evaluated by first calculating the rigidity of the structure from its known mass and natural frequency, and then determining from this value of rigidity the static load which would produce a deflection equal to the observed amplitude of vibration. This will be known as the *equivalent static load.*

For a simplified first approach, the approximate frequency of maximum response, 500 CPM, is combined with the calculated effective mass of the deck section, 560,000 pounds, to give a value for spring constant (Figure 22)

$$k = 4\pi^2 \times \text{mass} \times \text{frequency}^2$$

$$= 39.5 \times \frac{560{,}000}{386} \times \left(\frac{500}{60}\right)^2$$

$$= 3.9 \times 10^6 \text{ pounds per inch}$$

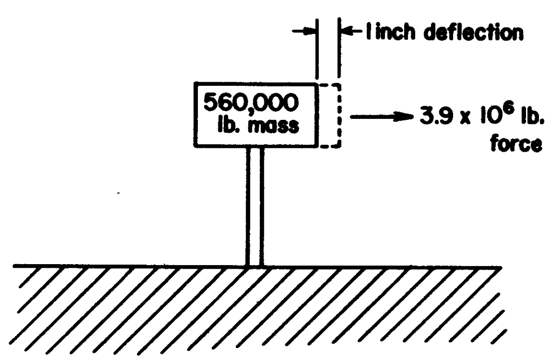

Figure 22 - Assumed Equivalent Simple System

The observed double amplitude, 0.0118 inch, if considered as a static deflection, would thus require a transverse horizontal load of 48,000 pounds, applied where the vibration generator was attached. The load exerted by the oscillator had a double amplitude of 27,800 pounds.

These figures must be given close scrutiny with a view to correction to get from them the best possible version of the information just approximated.

Since it is necessary in developing a sufficient load to make use of the phenomenon of resonant gain, and since the value of resonant gain is affected by damping, attention must be given the matter of the actual value of damping, (see page 9).

Resonant action under ordinary conditions of highly elastic behavior in a structure is simple and well understood, but when the vibration is opposed by frictional resistances of more than moderate amount, complications arise, (Figure 23).

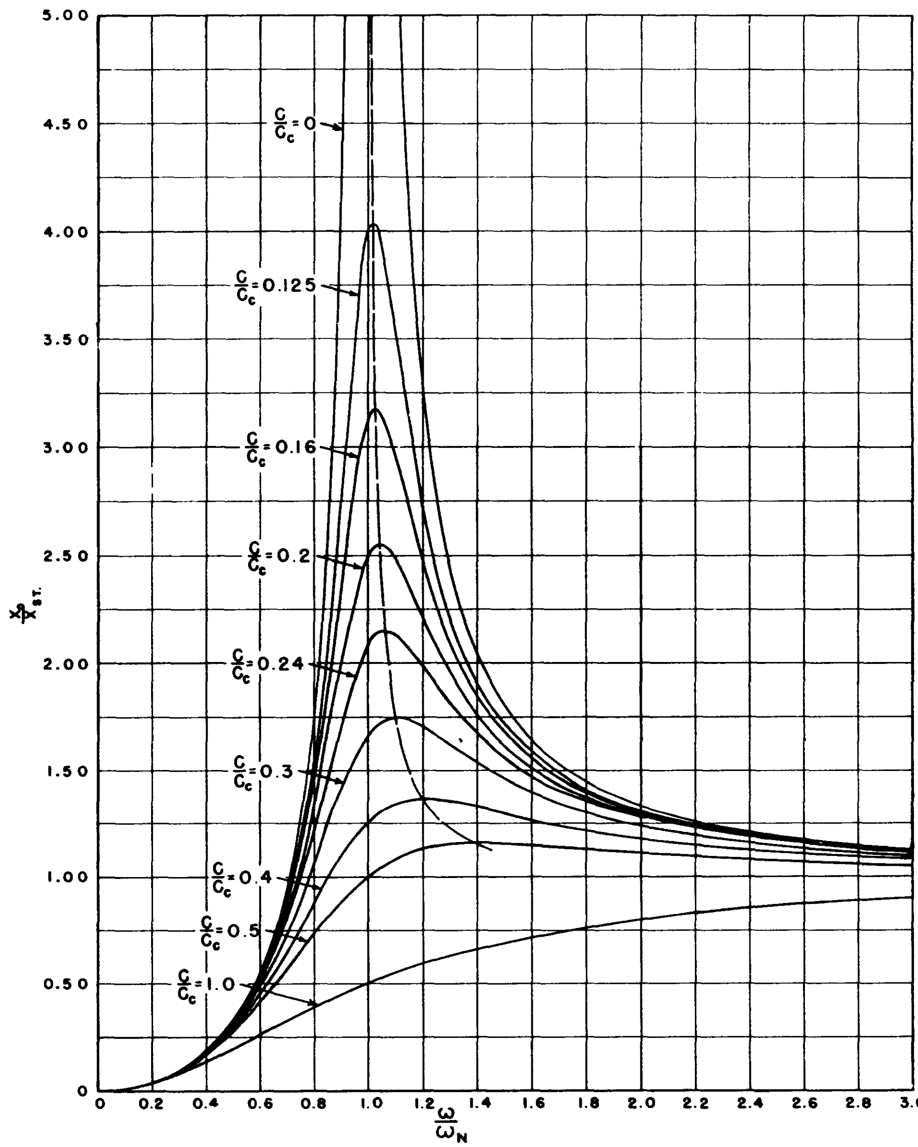

Figure 23 - Resonant Behavior under Excitation as Frequency Squared at Different Degrees of Damping

Single degree of freedom under sinusoidal excitation.

Ordinates are amplitudes of response expressed in terms of static deflection under given load.

Abscissas are frequencies expressed in terms of natural frequency or resonant frequency under zero damping.

Damping is expressed as a fraction of the critical value.

The effect of increased damping, in general, is to

1. Reduce the height of the resonant peak, and so the amount of resonant gain.

2. Broaden the relative spread of the resonant curve, so that the maximum is less definite.

3. Shift the position of the peak, so that the frequency of maximum response falls at a different value than the natural frequency of the elastic structure.

4. Reduce the abruptness with which phase relations are shifted as operating frequency varies about the value of natural frequency.

If the damping coefficient were definitely known, it would permit a definite correction to be applied to the frequency of maximum response to obtain the natural frequency. Fortunately the correction based on the value of this coefficient is small except for very high damping. This correction to frequency of maximum response required to obtain natural frequency becomes greater as damping increases, and if it exceeded 1/4 the critical value, the resonance curve would be so flat that the value of natural frequency would be ascertained only with difficulty. Such a situation seems not to exist in this instance, since the maximum response was well-marked, showing that the damping coefficient on the WASP has only a moderate value.

At the same time the resonant gain has an apparent value inferred from natural frequency of only 48000/27800 = 1.7, and this indicates a damping coefficient of $\frac{1}{2 \times 1.7}$ or 0.3, which must be considered rather high.

For a resolution of this dilemma an appropriate line of reasoning is proposed and developed in Appendix 1.

The rigidity of the flight-deck structure, derived by the method described in that Appendix, is taken to have a nominal value of 3.95×10^6 pounds per inch. Its actual value is higher by an indefinite amount. The double amplitude of nominal load causing the observed stress ranges is accordingly taken to be 47,000 pounds.

DEFLECTION AND STRESS

Strain-gage data are first presented in Tables 6 and 7a (YORKTOWN), and Tables 7b and 8 (WASP), in a reduced form preliminary to comparison with design calculations and other interpretative analysis. The only numerical manipulations involved in the data in Tables 6, 7, and 8 are the application of calibration factors and reduction of all results to terms of the same nominal load increment, 100,000 pounds. A minus sign indicates compression with load applied downward or to port.

In Table 6 are summarized all data on stresses and deflections on transverse beams in bowstring mode. These come entirely from static loadings on the

YORKTOWN, as the vibration data in bowstring mode on the WASP were not used. In three cases (Table 6c) the deflection data permit immediate inference of stress values, by assuming uniform curvature over the "net span" between port and starboard deflection stations; beside the observed value of curvature, this involves only depth of the beam, not its section modulus.

The gross deflections are taken with reference to the main deck. No deduction is made for the vertical deflections due to compression of the legs of the bents and to elastic actions in the hull.

Each entry in Table 6 is derived from a straight line or curve faired through about 10 points. The fairing was done both by subtraction and by graphic procedure.

TABLE 6

USS YORKTOWN - Flexure Under Vertical Static Load

a. Vertical Deflection Relative to Main Deck
inches per 100-kip load

Station	Frame 54	Frame 142	Frame 155		Frame 146
			Load Centerline	Load Port	
CL	0.110	0.114	0.161	0.087	0.115
P	0.066		0.106	0.105	0.255
S	0.062		0.103	0.043	0.048

b. Stress by Strain Gage
pounds per square inch per 100-kip load

Load	Frame 54	Frame 142	Frame 155		Frame 146
			Load Centerline	Load Port	
Increasing	1580*	870*	3660	510	
Decreasing	2810	1220*	3700	850	
Average		1045*	3680	680	

* Gage station on doubler plate

c. Stress Calculated from Local Deflections
pounds per square inch per 100-kip load

	Frame 54	Frame 142	Frame 155		Frame 146
			Load Centerline	Load Port	
	2270		4380	1000	

Tables 7a and 7b summarize all the stress data on both vessels bearing on the behavior of the columns and the knees at which they join the beams, with loads in both modes. The stations on different bents were not located in similar positions, so that data appearing in the same column in these tables are not directly comparable. There are also variations in the structure from bent to bent. Nevertheless the arrangement in the tables is in general similar for all bents. The order of stations on the ship from the main deck upward and inboard is followed in the tables downward for the port side and upward from the bottom for the starboard side.

Table 7a, dealing with the bowstring mode (YORKTOWN), shows stress at specified stations, location of which is shown in Figures 5 to 8, under a load of 100,000 pounds applied vertically to the deck at the frame named.

TABLE 7a

USS YORKTOWN - Stresses in Bent Frames at Increasing Distances from Main Deck

Bowstring Mode

Expressed in pounds per square inch per 100-kip load

Order Port	Frame 54		Frame 142			Frame 155	
	Station	Stress	Station	Stress	Station	Load Centerline	Load Port
1	4pi	-1390	2pi	-410	2pi	-180	0
2	5pi	- 920	3pi	-2180	3pi	-440	0
3	3pi	- 880	4pi	-1750	4pi	-2550	3070
4	6pi	- 300			5pi	-2670	3500
5	7pi	-1760			6pi	-3060	-3900
6	8pi	-1950			7pi	-3110	-3530
7					8pi	-1330	+ 340
8					9pi	- 680	+1590
9					10pi	+2430	+2100
10					clp	3320	+ 730
11					cl5	4030	600
	CL	2810	CL	1045			
3			4si	-650			
2			3si	-290			
1			2si	-470			
Order Starboard							
For gage station locations, see Figures 5, 6, 7, and 8.							

In Table 7b (WASP) a similar reduction has been made to an assumed load of 100,000 pounds, applied horizontally at the deck edge. The exciting load was actually applied at Frame 54 starboard, but it appears that the deflections were substantially the same as though the load had been distributed longitudinally over the whole of the section between expansion joints.

The stress values calculated from the data obtained by different observing parties are tabulated separately and may be directly compared. The stations at which the different pickups were located were at the heights above the deck as shown in Figures 19, 20, and 21.

Table 8 presents for the WASP a summary of the data in Table 7b, in more suitable form for further analysis. A measure of dispersion of values from different sources is also added.

TABLE 7b

USS WASP - Stresses in Bent Frames at Increasing Distances from Main Deck

Table-top Mode

Pounds per square inch for 100-kip load on whole section

Load applied horizontally in a transverse direction at flight-deck level and on Frame 54

Order Port	Frame 54							Frame 38					Frame 66	
	Inboard Flange					Outboard Flange								
	NAF	MIT	EMB	TJ	TW	MIT	TW	NAF	MIT	EMB	TJ	TW	EMB	TJ
1		-430		-390	-350									
2		-250		- 80	-100				-550	-490	-380	-230	-270	-160
3		+ 90		+ 40	+ 40				-310	-290	-150	-180	-170	- 90
4									+680	+530	+410	+440	0	+ 20
8	-300	-250		- 80	-140	+160	+ 40		-430	-480	-340	-350		- 80
7	+ 20	+ 30	0	+ 20	+ 40			10						
6	20	160	130	50	60			10						
5	190	250	170	90	100			280						
4	130	280	220	90	80		-130	200	+430	+350	+310	+200		+110
3	210			140	100			360						
2	360			210	210	-280	-110	640	+610		+550	+430		
1	+430	+530	360	+350	380								280	+280
Order Starboard														

TABLE 8

USS WASP - Average Observed Stress in Columns and Dispersion of Values

Table-top Mode

100 kip-load applied horizontally at deck edge in a transverse direction at Frame 54

Station and Height above Deck	Number of Observers	Average Stress lb/in²	Mean Error lb/in²
Frame 54, 4	5	+410	56
Starboard 27	3	260	67
50	3	150	40
68	5	~~860~~ 160	72
84	5	~~860~~ 160	52
117	5	84	49
142	5	+ 22	10
179	4	−192	82
Frame 54 13	3	−390	27
Port 86	3	−143	71
181	3	57	22
Frame 38 25	5	+558	75
Starboard 50	1	360	–
70	5	298	78
84	1	280	–
108	1	10	–
137	1	+ 10	–
179	4	−400	55
Frame 37 31	4	+412	108
Port 79	4	232	68
203	4	−515	90
Frame 66 8	2	280	0
Starboard			
70	1	110	–
193	2	*	–
Frame 66 6	2	215	55
Port 71	2	130	40
176	2	− 10	10

* Obviously erroneous readings omitted.

ANALYSIS AND INTERPRETATION

NOMINAL FRAMEWORK

In order to bring the results of the various measurements into a reasonable degree of comparability, it is necessary to adopt a nominal or schematic representation which will embody the essential features of the structure under test without confusing issues by details on which the tests are not expected to give any information. Only after the basic resemblance of the nominal to the actual structure is established can the correspondences between calculated and observed quantities be identified. The actual differences between them then form the basis for a critique of the design in the light of the experimental data. The first question is: What simplified schematic structure would behave in essential features as the actual structure is observed to do? When an answer to that question is found, comparison in detail of the actual structure with its nominal equivalent is sure to lead to fruitful suggestions.

The schematic framework tentatively adopted is shown in Figures 24 and 25. This picture of the structure is grossly simplified, but represents the simplified concept understood to have been used previously in design.

It represents a rectangular structure with three members; two equal vertical columns and one horizontal beam connecting them across the top. Two degrees of fixation of the column bases are considered; the first is equivalent to a pin connection incapable of transmitting a moment, while the second is equivalent to "complete" fixation, in which the moment transmitted is that which will wholly prevent rotational deformation at this point.

The members are all considered slender in proportion to their length, and their transverse dimensions are ignored. Nevertheless, they are so firmly joined at the corners that the rotations at the ends of the two members meeting there are taken to be equal.

The variables involved in this scheme are (1) the lengths of the members, (2) their moments of inertia (assumed to be uniform), (3) the fixation at column bases, and (4) the orientation and application of the loads. When these variables are known, or assumed, the formulas associated with these frameworks give nominal bending moments and axial loads from which, by use of section modulus values, nominal stresses are found. These stress values are given in Table 9. The calculations, as may be seen by comparison with observed values, fall rather wide of the mark.

PARAMETERS FOR NOMINAL ANALYSIS

Three basic design parameters, with respect to which assumptions must be made in course of design of flight-deck bents, are subject to uncertainty which the present tests are designed to alleviate. These are:

(a) Fixation of the columns at the main deck.

(b) Effective values of sectional moment of inertia.

(c) Effective values of beam length and column height.

A fourth question is related to these, but can not well be segregated for separate consideration in connection with the present tests. It relates to the elastic behavior of the column and beam members as affected by their depth, which is as much as a third of the length. This fourth uncertainty may be said to relate to errors of slender beam theory.

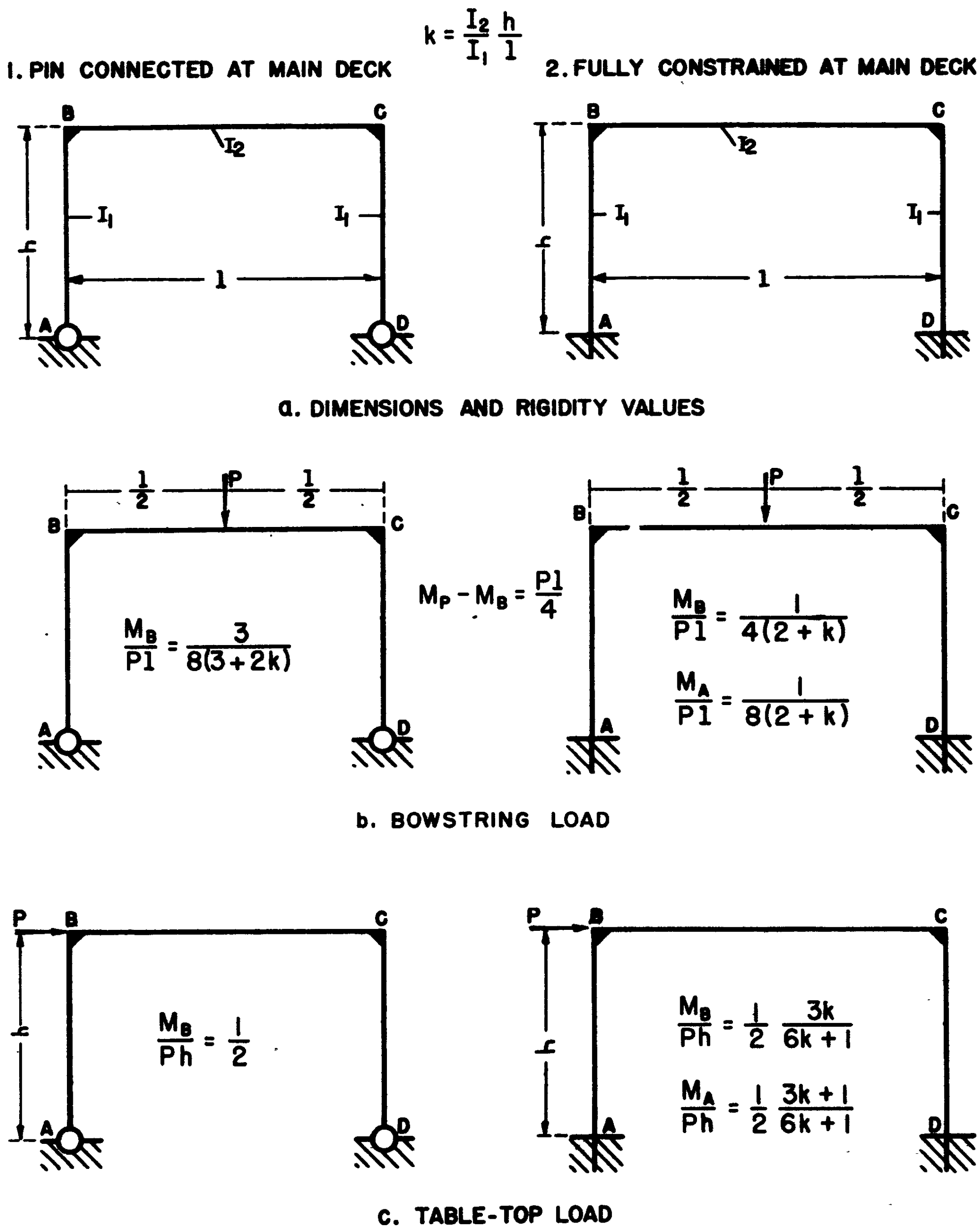

Figure 24 - USS WASP - Schematic Outline of Equivalent Bent

TABLE 9

Nominal Deflections and Stresses per 100-kip load

Comparison between Observed and Calculated Values

USS YORKTOWN

	Total Span Deflection in inches (vertical)			Deflection Center Section in inches (vertical)			Stress on Lower Flange at Midspan lbs. per sq. in.		
	Calculated		Obs.	Calculated		Obs.	Calculated		Obs.
Bent 54									
	Pinned at Deck	Fixed at Deck		Pinned at Deck	Fixed at Deck		Pinned at Deck	Fixed at Deck	
Flex.	0.120	0.112		.052	.050				
Shear	0.045	0.045		.023	.023		+4910	+4500	2810
Total	0.165	0.157	0.110	.075	.073	0.046			
Bent 142									
Flex.	0.187	0.178							
Shear	0.061	0.061					+6150	+5660	+1045*
Total	0.248	0.239	0.114						
Bent 155									
Flex.	0.290	0.276		0.110	0.107				
Shear	0.066	0.066		0.030	0.030		+9410	+8760	+3680
Total	0.356	0.342	0.161	0.140	0.137	0.057			

USS WASP

	Horizontal Deflection of Flight Deck			Stress at Main Deck, lb. per sq. in.			
	Calculated		Obs.	Calculated			Observed
Bent 54							
	Pinned at Deck	Fixed at Deck		Pinned at Deck	Fixed at Deck	85 per cent Fixity	
Flex.	0.110	0.024		0	830	705	< 450
Shear	0.010	0.010					
Total	0.120	0.034	< 0.025				
Bent 38							
Flex.	0.139	0.016		0	1104	939	< 560
Shear	0.015	0.015					
Total	0.154	0.031	< 0.034				

Terms of Calculation: Lengths of horizontal beam taken to inner flanges of columns, height of vertical columns taken to lower flange of beam. Deflections by flexure formulas with end moments as found for equivalent bent.

*Measured on doubler plate.

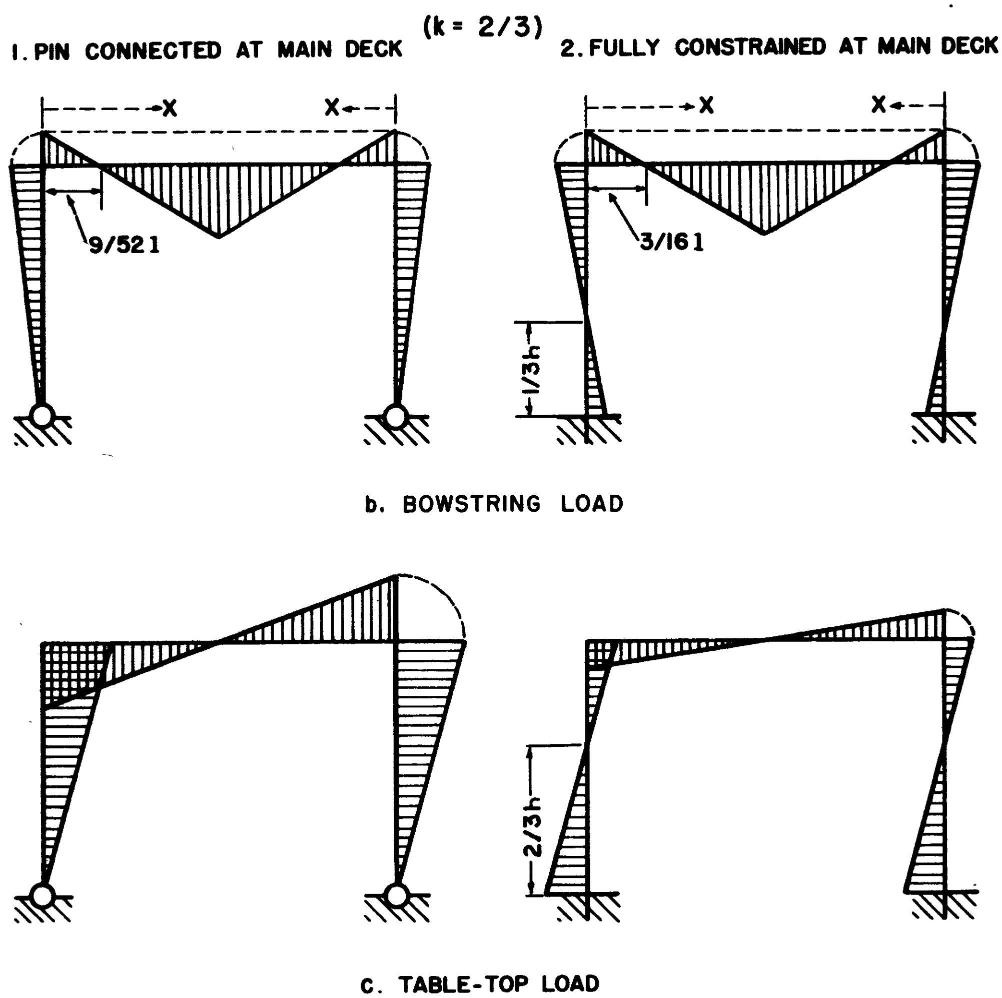

Figure 25 - USS WASP - Equivalent Bent, Moments

Beams and columns of the proportions that occur in the flight-deck bents are commonly analyzed as though depth were negligible, and if due allowance is made, as in Table 9, for shear deflections, slender beam theory should offer a reasonable basis for approximate calculations. Errors in stress will be localized, and resulting errors in deflection will be moderate, provided satisfactory usage with respect to the three design parameters mentioned can be established. It was at first hoped that the present tests might permit formulation of rules of thumb for the three parameters, which would serve reasonably well. For various reasons this has turned out not to be possible. Nevertheless the task of analysis will be approached from that angle.

(a) Fixation

With respect to the first of the three parameters, fixation of columns at the main deck, the data of the present test afford convincing evidence that a high degree of constraint has been achieved. This evidence has been reviewed and summarized in Appendix 2, to which reference should be made for details. The constraint consists of an elastic moment as shown in Figure 25, 2b and 2c.

Support, as distinguished from constraint, is afforded by the hull structure in the form of vertical and horizontal reactions to the load placed upon the hull by the bent. No satisfactory measure of the effectiveness of this support, or of the deflections of the hull under loads imposed by the bents, was obtained. Vibrometer observations on the WASP showed amplitudes of vertical and horizontal motion of the hull structure under the bents to be small but decidedly perceptible.

Observed features of the motion of Bent 54 on the WASP which are related to the elastic nature of support by the hull are discussed in Appendix 1.

Effective Values of (b) Sectional Data and (c) Lengths of Members

In view of the consistency of the data as taken by a variety of experimental methods, the discrepancies of Table 9 must be attributed to assumptions made in the calculations. The nominal framework affords only a gross approximation to the actual structure. Before proceeding to consider refinements which might be made in the nominal calculations, the nature of the sources of error must be identified. The practical means by which the designer may arrive at a closer approximation to actual values of stress per unit load may then be sought as a separate project.

The first task is to represent the facts fairly, to find expressions which fit the data, leaving it until later to put these expressions into form suitable for design use.

DISCUSSION OF DATA

(a) Bowstring Mode - YORKTOWN Data, Frame 155, Vertical Load at Centerline

For analysis of the YORKTOWN data, it is best to begin with Frame 155, since in this case alone a series of data are available for locating the section of zero moment on the horizontal beam. These consist of stress values on the lower flange of the beam at different distances from the centerline. They have been corrected for axial compression, multiplied by nominal section modulus, and plotted in Figure 26. They show an approximately linear distribution, with the section of zero moment about 265 inches from the centerline, and a moment of 5×10^6 inch-pounds at the centerline.

In addition, deflections relative to the main deck at the two quarter-length points and the midspan are available. The section is nearly uniform, with average nominal moment of inertia, $I = 39{,}000$ inch^4 and section modulus 1020 inch^3.

Three pieces of evidence agree in indicating that the strength and rigidity of the bent calculated by the nominal formulas of Figure 25 are too low. These are:

1. Absolute stress value at the centerline.

   The bending moment at the centerline is determined by load and span and amounts to over 10 x 10^6 inch-pounds per 100-kip load on nominal span, whereas the observed bending stress of 5000 pounds per square inch (after correction for axial load), gives a value of only 5 x 10^6 inch-pounds when multiplied by the nominal section modulus.

2. Deflection of the central part of the beam.

   The computed flexural deflection of the net span (346 inches) based on a graphical integration to allow for varying section, is 0.110 inch. When shear is also considered, the total deflection computed is 0.140 inch, compared with the observed deflection of 0.057 inch (see Table 9).

3. Variation of moment with distance from centerline.

   Moment obtained by combining strain-gage data with the nominal section modulus diminishes outboard at the rate of less than 20,000 inch-pounds per inch. The shear load is uniformly 50,000 pounds, whence the bending moment actually must diminish at 50,000 inch-pounds per inch.

Two explanations for these facts are suggested: the weight actually applied to the bent in the test may have been less than the total weight used, because of relief from load afforded by adjoining structure; and the moment of inertia at all sections of the beam may exceed the nominal values hitherto used.

Load relief cannot be determined exactly, but it appears from the original record that the weights overlapped the expansion joint by a small amount. Also there is a transverse truss at a distance of only 10 feet forward of Bent 155. The

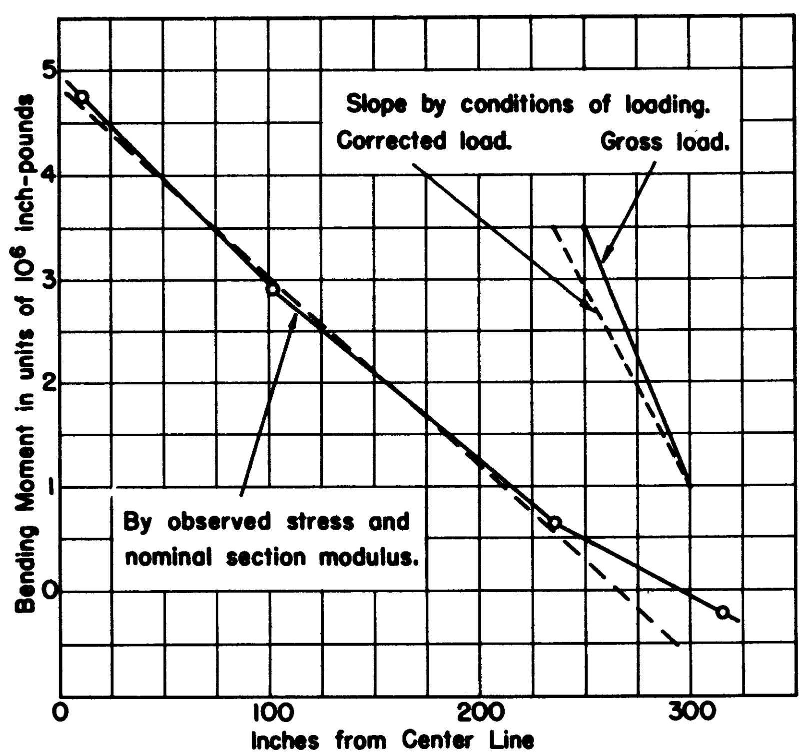

Figure 26 - USS YORKTOWN - Bending Moment in Beam, Frame 155

maximum possible deduction from load for such reasons is not over 25 per cent.

After a corrected load value is thus available, it becomes possible to estimate the excess of effective over nominal section modulus due to the presence of the flight deck and other supporting structure, piping, trolley tracks, etc. Basing the estimate on the experimental data shown in Figure 26, the effective section modulus is more than twice as great as its nominal value. The neglect of allowance for stiffening effect of structure adjacent to the bent itself thus seems to lead to large underestimate of strength in the beam.

Uncertainty in the choice of points to which beam length shall be measured enters into only the first of the preceding three items, and there merely to a minor degree. The conclusion that effective section considerably exceeds nominal section is not affected by uncertainty about length.

However, some information on the question of effective span is afforded by the overall deflections, considered in combination with the results just found for effective section modulus.

Calculations based on a revised value of moment at centerline, 9.4×10^6 inch-pounds, give deflections which vary with assumed span as follows:

TABLE 10

USS YORKTOWN - Overall Deflection in inches at Frame 155

Span between column centers is 840 inches

Assumed Span inches	Observed Deflection	Calculated Deflection	
		Nominal Section	Effective Section
624	0.161	0.177	0.099
672		.187	.104
720		.194	.108
768		.199	.111
816		.202	.113
840		.204	.114

It is apparent from these results that introduction of effective section data as indicated by the stress measurements reduces calculated deflections well below observed deflections even with high values of assumed span. The explanation is believed to lie in additional deflections that do not affect the flange stress. Thus shear in the deep webs is estimated to add 0.04 to 0.05 inch, depending on the span assumed. Other items are direct compression in the columns and depression of the ship structure under the columns.

It is also apparent that it will not be possible to allow for the effect of the numerous variables involved, by introducing any simple convention as to measurement of span length into the nominal formulas of Figure 25.

(b) Table-Top Mode - WASP, Frame 54

The situation here is even more intricate than in case of the bowstring data, but again there is evidence of greater strength and rigidity in the bent than are found by the methods of Figure 25.

In the discussion of equivalent load, Appendix 1, the frequency response curve is found to indicate rigidity in the bent itself higher than the apparent value observed in the bent as supported by the elastic structure of the hull. Nevertheless the estimate of test load, which was affected by these considerations, was given the minimum value obtained by ignoring the frequency response data and assuming the hull deflections to be negligible.

The comparisons of Table 9 show again that actual strength and rigidity (as indicated by values of observed stress and deflection under standard load) are generally greater than the nominal values obtained from Figure 25.

If allowance is made for the fact that the test load was actually higher by an undetermined amount than the minimum estimate adopted, the stress and deflection per unit load will have still smaller values than those given in Table 9, and the strength and rigidity of the bent will be found to be still further undervalued by the nominal formulas.

The explanation of this underestimate, in terms of fixation and of effective values of sectional data and lengths, cannot be said to receive any further support from the WASP data beyond the point of confirming the need for such an explanation.

In addition to this confirmation, the WASP data afford information as to the action of a horizontal load, as distinguished from the vertical load at centerline used on the YORKTOWN. For this reason the WASP data are more informative with respect to the columns, just as are the YORKTOWN data with respect to the beams. Now the columns are shorter and deeper members than the beams and the shortcomings of slender beam theory are thus more pronounced in the WASP than in the YORKTOWN data. At the same time the comparisons in Table 9 do not go far enough to permit definite evaluation of corrections even like those obtained in the case of the YORKTOWN, Frame 155.

As an alternative, further analysis of the WASP data is undertaken in terms of distribution of stress in the columns, with respect to which the data for Frame 54 starboard are fairly complete. The details are presented in Appendix 3. This analysis shows that the frame may with benefit be broken down into separate elements, column, knee, and beam; but the complete development of this proposal is left until more complete experimental data are available.

ADDITIONAL DATA

Data for Frame 155 on the YORKTOWN and Frame 54 on the WASP are more extensive than on other frames. For completeness of the record, however, data on the other frames are reviewed in the light of the analysis on these two.

YORKTOWN, Frame 54

The nominal calculated bending moment at the centerline is 12.2×10^6 inch-pounds, whereas the bending moment (based on observed data and section modulus of girder only) is 7.3×10^6 inch-pounds.

The computed flexural deflection of the 36-foot center section, based on the nominal bending moment at the centerline (12.2×10^6 inch-pounds) and graphical integration to allow for varying beam section, is 0.071 inch. This deflection plus the calculated shear deflection gives a total of 0.094 inch compared with the observed value of 0.045 inch.

The nominal calculations here are probably less reliable than for Bent 155 because of the tapered sections in both the beam and columns of the bent. The computed moment at the centerline is based on averaged moments of inertia. However, a calculation taking account of varying sections checked the foregoing value within about 10 per cent.

YORKTOWN, Frame 142

This is a central bent similar to that at Frame 54, except that the columns are built in at the main deck, the scantlings are lighter and the tapers less. The nominal computed moment at the centerline is 10.8×10^6 inch-pounds. The moment obtained from observed data and nominal section modulus (girder only) is 3.05×10^6. In this case, strain was measured on a doubler plate and is thus lower than if measured on the girder flange. Though the data are less complete and reliable at this bent than the others, qualitative confirmation of the results from the other bents is afforded.

In this case, as for Frame 54, deflections for the whole span are affected by the uncertainty regarding assumed lengths necessary to correct for action at the knees of the bent. Conventional measurement of length to the inner flanges has been adopted for nominal calculations of full-span deflections, central-span deflections, and stresses in Table 9. The results indicate that correction in length alone is not sufficient.

Effect of Off-Center Loads

Bent 155 was loaded near the port quarterspan point in addition to the centerline loading already discussed. The schematic diagram for this case is shown in

Figure 7. Calculated total deflection under standard load is 0.19 inch, of which 0.14 inch results from flexure and 0.05 inch from shear. The observed value of 0.106 inch substantiates the previous evidence that nominal calculations give too high values of deflection.

Rather extensive strain data are also available, so that the consequences of off-center loading can be studied in somewhat greater detail. The problem will be somewhat simplified, however, by making calculations only on the basis of full fixation at the deck, points A and B.

Decentering the load naturally throws the moments into an unsymmetrical pattern. At the knee nearest the load the moment is increased, while at the farther knee the moment is decreased. At the main deck the opposite is true of the moments in the columns.

The moment obtained by multiplying calculated section modulus by stress by strain gage is compared with calculated moment in Table 11. On the beam the results are consistent but observed values are only about one-half the calculated values. On the column, the observed value at Station 1pi, at the main deck, agrees with calculation quite directly.

The low moment values on the beam again indicate that the effective section modulus is about double the calculated value. In these calculations the contributions of the deck and all other supporting structure except the web and flanges were ignored. The good agreement at the foot of the column indicates that the effective section modulus at that point is about as calculated.

TABLE 11

USS YORKTOWN - Bending Moment, Frame 155, Off-Center Load

Units of 10^6 inch-pounds

Gage	M Calculated	M Observed
CLS	2.07	1.3
CLP	2.57	1.4
10pi	4.19	1.9
9pi	4.33	2.1
8pi	1.25	1.1
1pi	1.05	1.0

The stability of the lower flange of the bent was checked by observing its fore-and-aft movement directly under the load. No noticeable deflection of this sort was observable by the means used under loads up to 56,000 pounds, the maximum test load.

Distribution of Stress on Girth

Strain-gage stations on the YORKTOWN were located at various points at which high stresses were expected, though these stations were not similarly disposed in the tests on different frames. Data bearing on girth distribution are assembled in a suitable arrangement in Tables 7a and 7b on pages 31 and 32.

The schematic outlines of Figure 25 are again used for guidance in this analysis. The moments are easily followed by reference to these sketches, and since with few exceptions all observations were made on the inboard and lower flange, the sign of the moment is referred to only in terms of tension or compression in this inner flange. The variation of stress will be found to follow roughly the patterns of Figure 25, but no comparison of observation with calculation is warranted.

The moment of inertia and, at the knees, also the depth of the bent section, vary widely from point to point in the girth. The depth is also rather large in proportion to the height and span of the bent. Local stress values may therefore be expected to depart rather widely from nominal values calculated by simple beam theory.

Longitudinal Members

Strains and deflections were measured on a longitudinal girder of the YORKTOWN as shown in Figure 9 on page 7. For a midspan load of 100 kips, observed stresses and deflection compare with computed values (based on section modulus of girder only) as follows:

Stress, lb/in^2		Deflection at Midspan, inches	
Observed	Computed	Observed	Computed
5600	18,300	0.28	0.82

The proportionately greater difference between observations and nominal calculations in the case of longitudinals is to be expected, since the deck no doubt adds more to the section modulus of such a member than to the heavier bent section.

Deflection data were also taken on the longitudinal deck beams but they are not suitable for quantitative analysis.

WASP, Frame 54, Port

The bent is, in general, symmetrical above the main deck, so that differences between the port and starboard sides are incidental. The differences appearing in Tables 12 and 13 on pages 54 and 55, Appendix 2, are not in fact great enough to justify special conclusions. Around the feet of the columns differences in construction exist, however, which may have an influence on the degree of fixation. These differences are not shown in Figure 21, but consist largely in adaptation to the form of the hull which has a greater flare on the port than on the starboard side. The nominal calculations, which ignore such differences, give the same values for moment at the deck and for height of point of inflection on the port as on the starboard side. The observed strains show a value of constraining moment at the base somewhat greater on the port than on the starboard side, and the point of inflection is higher, which suggests that the difference in flare is not the controlling feature, since that would cause a lower value of fixation on the port than the starboard side.

WASP, Frames 38 and 66

In extending the foregoing analysis to the bents at Frames 38 and 66, it is in order to inquire as to the division of load between the bents. The allotment of half the total horizontal load to Frame 54 was based on its intermediate position and on the sectional moments of inertia and section moduli, which in Frames 38 and 66 are each rather accurately equal to half the corresponding values in Frame 54.

The division of the remaining half between Frames 38 and 66, however, is not on an equal basis. Frame 54 is not at midlength between Frames 38 and 66, and on a static basis, Frame 66 should take the greater part. A contrary effect is actually observed, as though the greater distance of Frame 38 from the vibration generator caused a whip which increased the amplitudes forward.

Observations with the Karelitz vibrometer and Whittemore gage show that the horizontal amplitudes at the three frames are in the ratios 6 : 8 : 11 for Frames 66, 54, and 38. The more accurate timed records of the pallograph indicate amplitudes, reduced to terms of 100,000 pounds total load, of 18.8, 25.1, and 34.5 in units of 0.001 inch. In view of the substantially equal rigidities in Frames 38 and 66 the division of load is assumed to be 1/6 on Frame 66 and 1/3 on Frame 38. The standard load at the starboard column, Frame 38, is thus 16,700 pounds.

Data on Frame 38 are nearly as complete as on Frame 54. On the starboard side the section of zero moment is 139 inches above the main deck, whence by the same reasoning as on Frame 54,

$$M_B = 2.24 \times 10^6 \text{ inch-pounds}$$

$$M_A = 2.32 \times 10^6 \text{ inch-pounds}$$

The comparison of gage stresses with values calculated as on page 44 is then as follows:

Height above deck, inches	Stress, lb/in²	
	Calculated	Gage
24	479	542
50	452	361
70	389	302
84	332	287
108	193	41
137	15	41

For deflection, following the same procedure as on Frame 54, the calculated value is 0.019, compared with an observed deflection of 0.0345 inch. The high value of observed deflection is associated with a hull section which is far forward and quite slender in proportion. The influence of hull elasticity on bent deflections is thus confirmed.

Stresses at Frame 38 port are in good agreement with those on the starboard side. The point of inflection is a little lower and the moment at the deck, as shown in Tables 12 and 13, is smaller, both in consistency with the greater flare on the port side, which may thus be supposed to reduce the degree of fixation of the column at the deck.

Data for Frame 66 are not complete enough to justify a similar analysis. They are, however, wholly consistent as to trend, the lower stress values confirming the lower share of the load taken at this bent.

COMMENT

The primary objective of these tests was to obtain experimental data which would afford a check, independently of calculation, of the adequacy of the flight-deck structure on two aircraft carriers, and thus of the correctness of the design calculations.

Assurance as to the correctness of the experimental data is naturally necessary. For this assurance dependence is placed on the consistency with which the different instruments operate in calibration and in use, and especially on the agreement among tests by different methods of loading and different methods of observation.

Strain-gage measurements, especially at low stress intensities, can make no claim to high precision. The methods now used, however, are more sensitive than those used in the past, and it is no longer permissible to dismiss unexpected variations as vagaries of the gages. Even though such variations may not be completely consistent, they may serve the useful purpose of directing attention to circumstances ignored or assumed to be negligible, but not actually so. Disagreement with results derived from pure calculation can not be accepted as evidence of the incorrectness of observations. Even though the experimental results may appear paradoxical, and even though their actual precision is not great enough to serve as a foundation for a complete revision of methods of calculation, yet the burden of the discrepancies between nominal calculation and observation must be borne by the calculations.

The evidence of the experimental data points with complete consistency to higher values of actual strength and rigidity than calculations by the nominal formulas indicate.

When the methods of calculation are satisfactorily modified to give results more in accord with the test data it is hoped that improvements in design will result. Before this can occur, however, the procedure in design calculation must be brought into a form that is adapted to the practical requirements of the situation. And before that can be done, the nature of the errors involved in the nominal formulas must be clearly understood.

The section on ANALYSIS AND INTERPRETATION is devoted to this task. Special methods of calculation are used, not as tentative substitutes for the present usages, but in order to identify the nature and cause of the discrepancies.

It is believed that calculation in terms of separated elements or components will lead in the desired direction. The beam, the knee, the column, the sub-structure, each is a unit which will repay more detailed consideration than it has so far received. The nominal framework, considered as a whole, provides a satisfactory scheme for use in preliminary work, where questions of proportion of parts are predominant. But for final design the nominal framework as pictured in Figures 24 and 25 is inadequate.

The fact that the sub-structure, or the hull proper below the main deck, is an elastic structure can no longer be ignored. The columns or legs of the bents are not completely fixed at the main deck level, not so much because of lack of material or insufficiency of fastenings at this point, but because the sub-structure itself is not rigid. It is customary to place flight-deck bents over transverse bulkheads, which in most cases extend completely across the ship from one leg to the other and are two deck heights or more in depth. It seems probable, therefore, that future calculations and analyses, at least of flight-deck bents as separate components, must consider the bent as a closed ring, integral with the sub-structure below the main deck.

Finally, it should be noted that throughout this whole study no attention has been given the evaluation of service loads. Assumed loads for purposes of design are specified by rather intricate requirements. The standard comparison load of 100 kips, applied vertically as on the YORKTOWN, should be multiplied roughly by 2-1/2 to give the assumed design load. The stresses as tabulated for vertical load, multiplied by 2-1/2, would therefore give a rough indication of working stresses to be expected if the assumptions used in design are realized in service. Similarly, the standard comparison load of 100 kips applied horizontally as on the WASP should be multiplied roughly by 7 to give the horizontal load assumed for design purposes, so that stresses as tabulated should be multiplied by 7 to afford an estimate of working stresses for horizontal service load.

With respect to actual loads occurring in service these tests give no information whatever.

CONCLUSIONS

1. Values of stress under test load as observed by strain gage are consistently low as compared with nominal calculations.

2. In order to obtain calculated values of stress corresponding to strain-gage values, it is necessary to make allowances for:

(a) Axial loads in addition to bending loads on the members.

(b) Reduction of length of members available for flexure incidental to presence of large knees.

(c) Reinforcement of strength members by incidental structure.

(d) Errors of slender beam theory.

3. Development of revised standard procedure in calculation of stress on structures of this kind should be undertaken.

4. In calculating rigidity, deflection, and natural frequency, it is necessary to make allowance for the four effects mentioned in Conclusion 2 and in addition for shear deflection, which is a large item, especially in the columns.

5. Data on rigidity and frequency indicate that elastic deflections of the hull in way of loads applied through bents are not negligible. The complete solution of the design problem will include the hull structure with the bent frame to form a closed ring.

6. The testing of heavy, rigid structures under moderate vibrational loads is demonstrated to be feasible, especially in view of new techniques for measurement of low stress values under cyclic conditions. New questions are raised by analysis of data obtained by this method, resulting in new information and points of view regarding the dynamic behavior of ship structure.

RECOMMENDATION

For development of satisfactory processes of design calculation, theoretical analysis of rigid frames by separate elements of beam, knee, column, and substructure and treatment of all these components together in the form of a closed ring is recommended, with experimental verification in steel models on not less than 1/4-scale.

PERSONNEL

The test party engaged on this project consisted of Lieutenant Commander W. P. Roop, Lieutenant R. T. Sutherland, Jr., Dr. D. F. Windenburg, R. T. McGoldrick, E. E. Johnson, G. H. Curl, and F. B. Bryant.

REFERENCES

(1) Bureau of Construction and Repair letter CV5/S11-6(RP) to Officer in Charge Experimental Model Basin of 7 June 1938.

(2) Bureau of Construction and Repair letter CV7/S11-6(DCV) to Superintending Constructor, USN, Quincy, of 13 January 1940.

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APPENDIX 1

FREQUENCY-RESPONSE CURVE AND RIGIDITY, USS WASP

The value of rigidity found as indicated on page 27 is 48,000 pounds per inch. The low deflection is attributed to high damping. On the other hand, if the damping coefficient is as small as the frequency-response curve suggests, the low deflection must be attributed to a higher value of rigidity.

If the damping actually has a rather low value, corresponding to a well defined resonant peak and a high resonant gain, the observed value of the natural frequency, and consequently of the rigidity, is lower than these circumstances would indicate. This discrepancy suggests that the hull fails in some degree to provide a completely rigid foundation.

These suppositions may be clarified by an analogy (see Figure 27). A cantilever spring, vibrating with respect to a completely fixed base, is compared with the half-length of a free-free bar such as would be formed by the cantilever and its mirror image in the plane of its base.

The bending of the cantilever is similar to the bending of the half of the free-free bar, but the amplitude of motion of the end of the free-free bar is smaller than that of the cantilever.

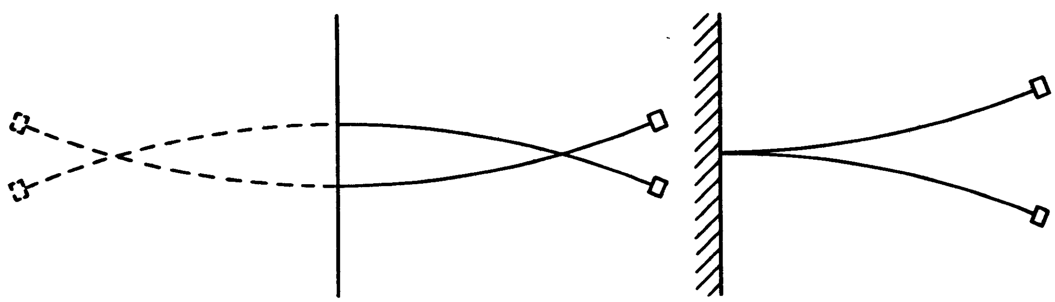

Figure 27 - Comparison of Cantilever with Free-free Bar

The main source of information bearing on this matter in these tests lies in the curve of distribution of amplitude of elastic response to various frequencies of excitation. The best data of this sort available are plotted in Figure 28.

The curve fitted to the observed spots in Figure 28 is not simply faired in, but is a calculated resonance curve, the parameters indicated having been evaluated by a process of trial. They show a damping coefficient of only 0.04, which is inconsistent with the value 0.30 obtained in the first approximation.

Resonant gain and damping coefficient are related with each other in a very simple way. In assuming different values of damping coefficient we are at the same time making assumptions as to the value of resonant gain. With a given exciting force amplitude, resonant gain is equal to half the reciprocal of the damping coefficient. In Figure 28, however, a different condition obtains; the exciting force is known, the amplitude of response is known, but it is the elastic rigidity and the damping coefficient which are unknown.

The form of the curve gives a clue as to the damping coefficient. A curve has been dotted in based on a different damping coefficient to show how impossible it is to reconcile the form of this curve with a damping coefficient much higher than the assumed value 0.04.

If this value of the damping coefficient were to be accepted as approximately correct, the resonant gain would have a value of about 12-1/2, which would indicate an elastic rigidity about seven times as high as the approximate value which formed the starting point of the discussion.

The explanation suggested on page 29 may now be restated as follows:

> The static rigidity of the bents alone is believed to be higher than the preliminary figure, given on page 29, and if the hull of the ship were in fact completely rigid, the natural frequency would be considerably higher than was observed. However, the hull itself is also an elastic structure, and though the fixation against local rotation of the columns at their bases is high, the whole ship is subjected to loads in way of the loaded bent by reason of the reactions applied to it by the columns. Deflections under these loads act to bring the natural frequency down to its observed value.

Since the hull deflections thus occurring are due to strains in a part of the structure with which these tests are not directly concerned, it is appropriate to consider the value of rigidity which the bent would have if it were actually mounted on a completely rigid base. The form of the frequency response curve of Figure 28 gives a value of 12-1/2 for the resonant gain. On this basis, the total load on the structure at its maximum deflection would be 12-1/2 times the applied load, instead of 1.7 times this load, as previously estimated. The rigidity of the bent would, according to this estimate, be 12-1/2 ÷ 1.7 or more than 7 times as great as the value taken directly from the observed natural frequency.

This high figure may well be much too high; too great dependence must not be placed upon the observed data on frequency response used for the construction of the curve of Figure 28. There seems, however, to be no doubt that a well defined and fairly narrow peak exists somewhat as shown in that figure. This narrow peak is not consistent with a high damping coefficient; in fact, the structure has no apparent features which would lead to the large frictional resistances necessary to cause higher damping. The low figure for resonant gain therefore seems beyond doubt too low.

The lower figure derived from the calculated rigidity has been adopted only because the evidence that it is too low is not specific enough to point to a definite higher value. Since the amount by which rigidity actually exceeds the low value is not determinable, the excess will be held as a reserve of indefinite value.

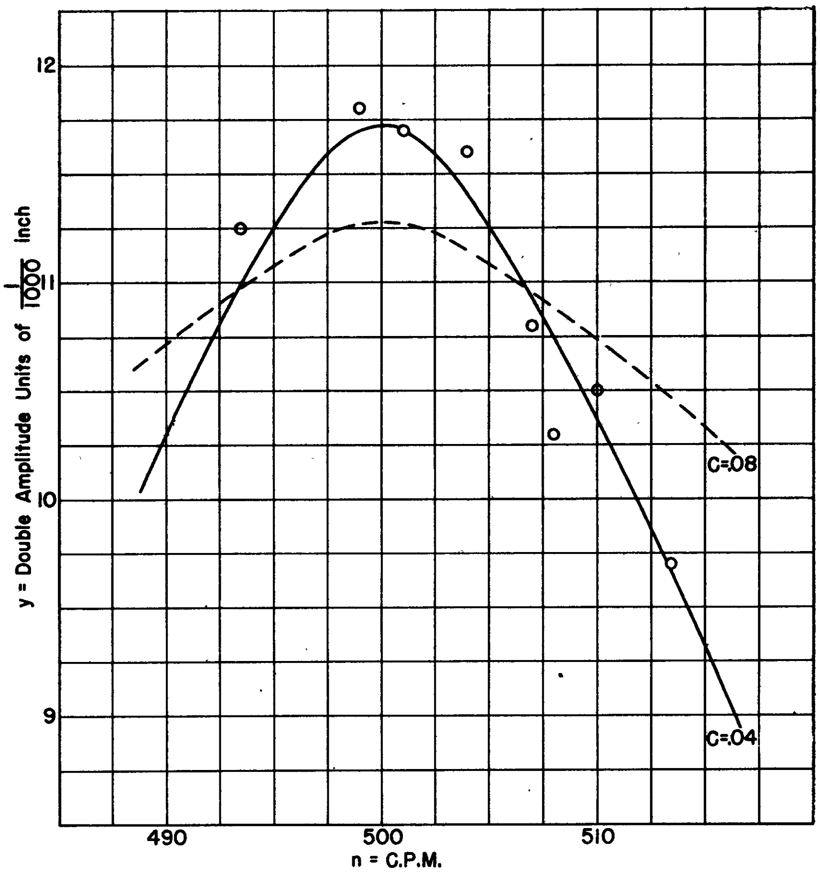

Figure 28 - USS WASP Flight Deck - Observed Frequency Response Curve

The solid line curve shown in Figure 28 has the equation

$$y = \frac{a}{\sqrt{\left[1 - \frac{n_0^2}{n^2}\right]^2 + 4c^2\frac{n_0^2}{n^2}}}$$

This is derived from the equation for forced vibration with viscous damping as given by den Hartog in "Mechanical Vibrations," page 56.

a represents the deflection which would be caused by a static load equal to the amplitude of the forcing load. y/a thus equals the resonant gain. n_0 is the natural frequency and c is the damping as a fraction of the critical value.

The curve as drawn corresponds to the following numerical evaluations:

n_0 = 500 CPM
a = 0.95 x 10^{-3} inch
c = 1/25 critical damping
y/a = 12-1/2 resonant gain at resonance

This curve takes account of variation with speed of amplitude of forcing load. A second calculated curve has been dotted in to show the departure from observed values caused by assuming c = 0.08, y/a = 6 at resonance.

APPENDIX 2

DATA ON COLUMN FIXATION

Fixation of the columns at the main deck is the first of the parameters, mentioned on page 34, which the nominal analysis of Figures 24 and 25 leaves open for consideration.

Fixation is of small concern in the YORKTOWN test, since it has little effect under a vertical load at the centerline. The beam has a partial fixation at the knees, depending on the relative rigidities of columns and beam, but the moments at the column bases are so far removed from the beam that their influence is reduced to a low order. The nominal formulas (see Figures 24 and 25) take full account of this fact. Uncertainty as to the degree of fixation does not affect analysis of the YORKTOWN data, and they can therefore yield no information about the actual degree of fixation.

In the case of horizontal load, however, as on the WASP, fixation at the main deck is a powerful source of support. The bases of the columns in the WASP received special attention in design, and the test load was well adapted to indicating the results of this procedure. The design was found to be highly successful in this respect.

The stress data from the WASP demonstrate high fixation in two different ways: (1) by observation of absolute value of stress at the base of the column, and (2) by determination of the height above the main deck of the section of zero stress.

The average stress value at the base of the column of Bent 54 starboard is nearly 500 pounds per square inch per 100-kip horizontal load. The calculated section modulus of the horizontal section of the column near its base is about 6400 inch^3. The moment obtained by combining these figures is 3×10^6 inch-pounds. Moments observed for the various frames are found by similar procedure to be as shown in Table 12.

Values for these moments are calculated from the nominal formulas of Figure 24 to be 5.4, 3.6, and 1.8×10^6 inch-pounds, for Frames 54, 38, and 66 respectively. Loads on these frames are assumed to be 1/2, 1/3, and 1/6 the whole, respectively. For further information on these nominal calculations, see Table 10.

The second method requires the consideration only of relative values of stress, and should therefore be free of errors arising in calibration and load evaluation. It is complicated, however, by the fact that the section of the column is not uniform, but has lower values of moment of inertia as the distance above the main deck increases. Linear variation of stress with height above deck could therefore not be expected. The variation in section is partly stepwise, due to butt-welded flange plates of different thicknesses, and similar dispositions, but by taking out values of section modulus on horizontal sections at the levels of the various stations and multiplying the observed stresses by them, values for actual moment in these sections are obtained. These values, as plotted in Figure 29, should vary in a linear way with height above the main deck.

TABLE 12

USS WASP - Loaded in Table-top Mode

Observed Moments at Main Deck in Inch-pounds x 10^6 per 100-kip Horizontal Load

Frame \ Gage	NAF	MIT	EMB Electric	EMB Tuckerman J	EMB Tuckerman W	Average	ME
54 Starboard							
Inboard	3.1	3.4	2.3	2.2	2.3	2.7	0.5
Outboard		2.5			0.9		
54 Port		3.2		3.0	2.8	3.0	0.1
38 Starboard	3.1	3.0	2.0	2.8	2.2	2.6	0.4
38 Port		2.5	2.2	1.8	1.0	1.9	0.5
66 Starboard				1.2			
66 Port			1.1	0.7	0.7	0.8	0.2

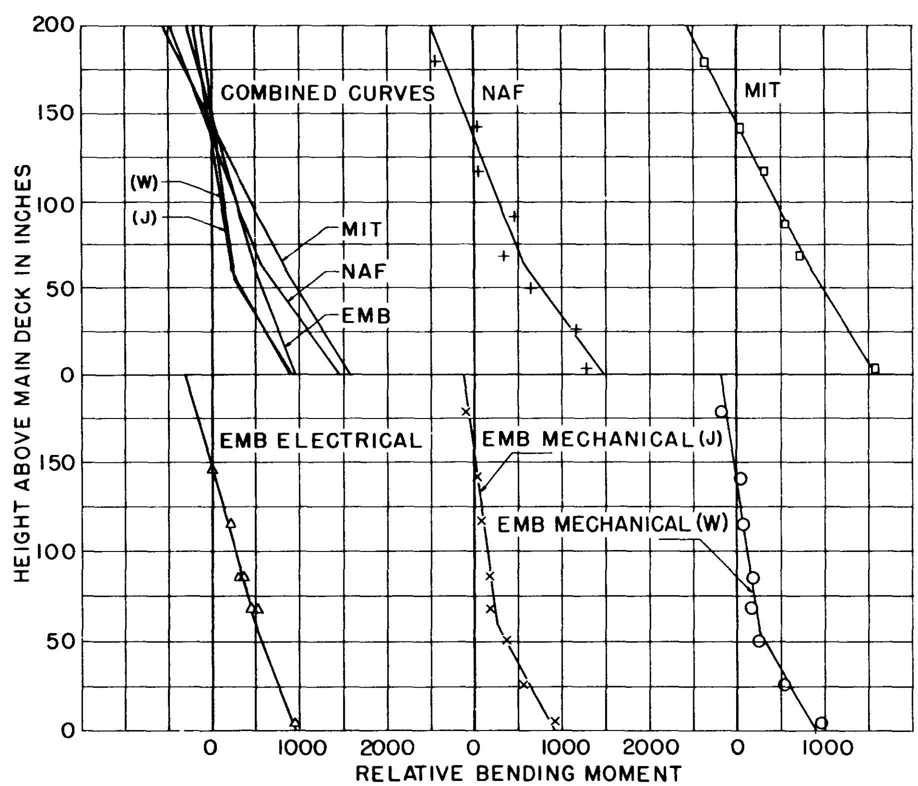

Figure 29 - USS WASP - Variation of Bending Moment with Height above Deck, Frame 54, Inboard Starboard

A graphical solution of this problem is offered for Bent 54 in Figure 29, in which relative moment (product of calculated section modulus by relative stress) is plotted against height of station above main deck. This is done separately for each observing party, and the combined results are also shown. The point of zero stress in Frame 54 starboard is indicated to be 144 inches above the main deck within a tolerance of not more than ± 7 inches.

Nominal calculation indicates the height of this point to be 165 inches in case of full fixation. The assumption of full fixation is thus a first approximation to the actual conditions.

Data on the outboard flange of the column at Frame 54 starboard are much less complete than on the inboard flange, as shown in Figure 30. They show two differences from the inner flange: lower intensities at a given level, and a higher position of the point of inflection. However, the indicated position of point of inflection, 160 inches above the main deck is still below the nominal calculated position, at 165 inches.

Axial load serves to explain at least in part such non-symmetrical stressing of inner and outer flanges. It seems to invalidate the simple multiplication of stress on the inner flange by a calculated value of section modulus to obtain moment. No other simple procedure is available, however, and the values given in the foregoing tables must be regarded as containing an unknown amount of error due to irregularity of stress distribution. The non-linearity of the curves in Figures 29 to 33 is probably associated with such irregularities of stress distribution. Figures 29 to 33 nevertheless afford consistent indications of a position of the section of zero moment which completely assures a high degree of fixation, even though some doubt as to the exact fraction of full fixation may remain.

Data from the other columns are plotted in Figures 31 to 33. The results are summarized in Table 13.

TABLE 13

USS WASP - Height of Section of Zero Moment in Columns, inches above Main Deck

Gage / Frame	NAF	MIT	EMB			Average	ME
			Electric	Tuckerman			
				J	W		
54 Starboard							
Inboard	136	144	147	155	140	144	5
Outboard					160	160	
54 Port		164		160	165	163	2
38 Starboard	132	149	137	147	130	139	7
38 Port		128	133	122	120	126	5
66 Starboard				153		153	
66 Port			173	160		166	7

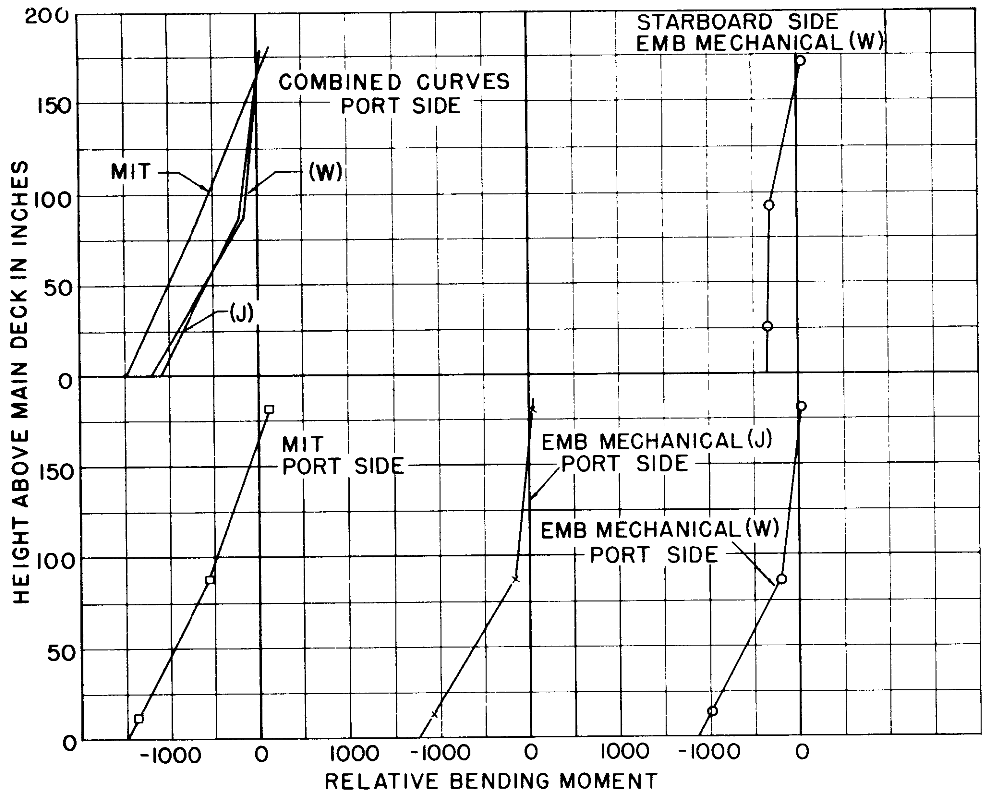

Figure 30 - USS WASP - Variation of Bending Moment with Height above Deck, Frame 54, Inboard Port and Outboard Starboard

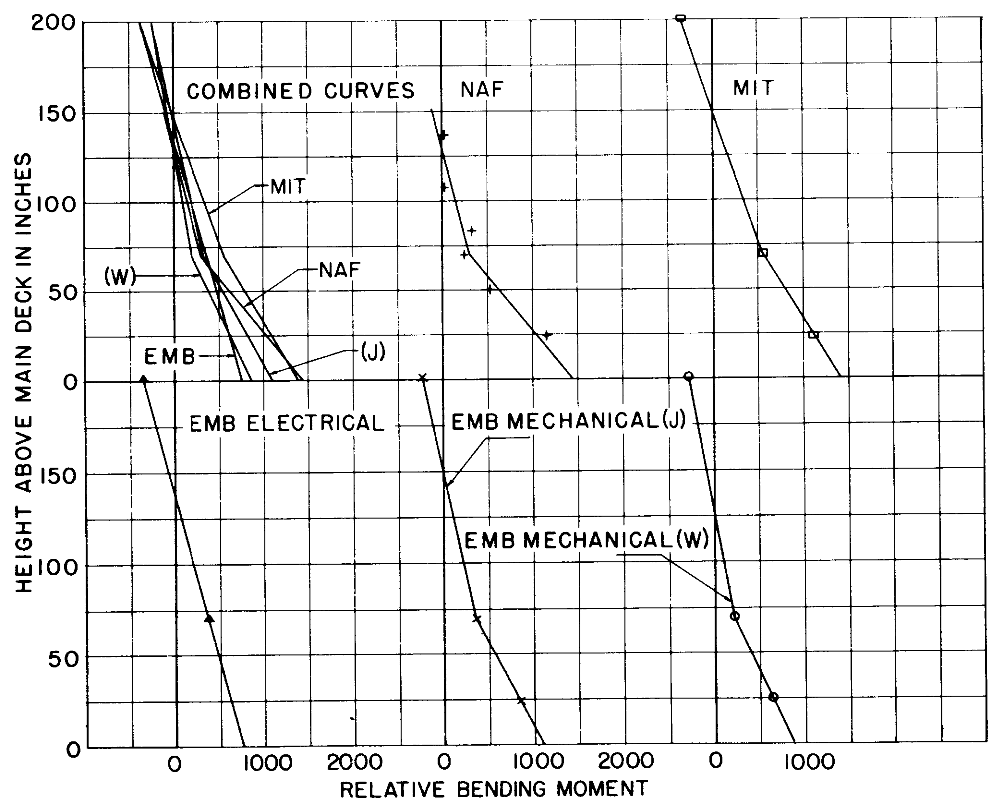

Figure 31 - USS WASP - Section of Zero Moment in Column, Frame 38, Inboard Starboard

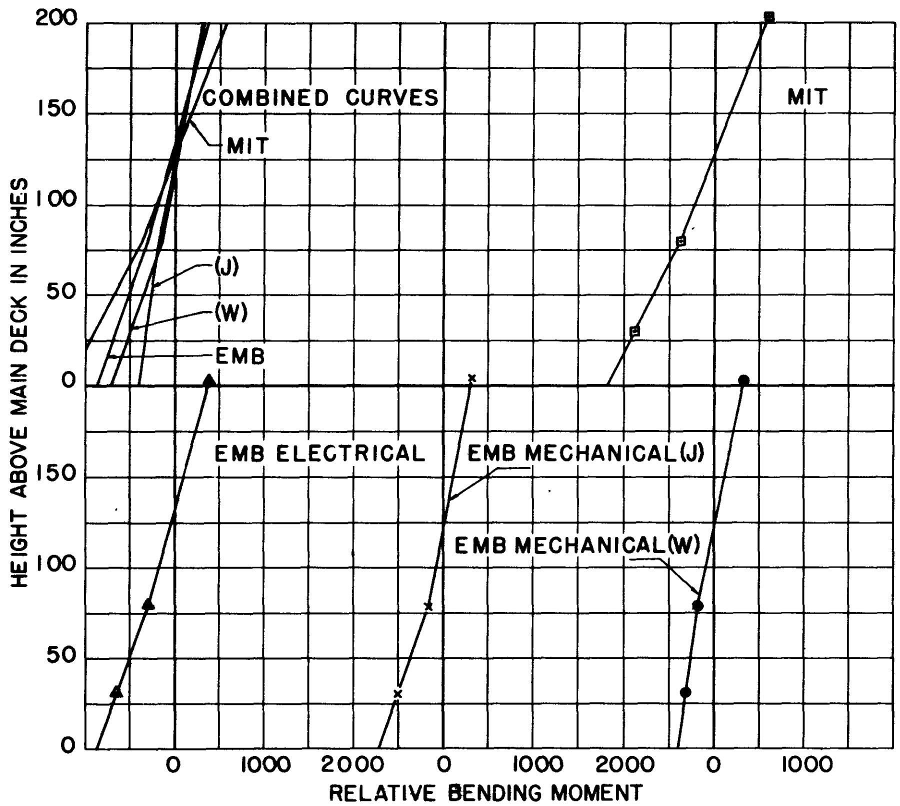

Figure 32 - USS WASP - Section of Zero Moment in Column, Frame 38, Inboard Port

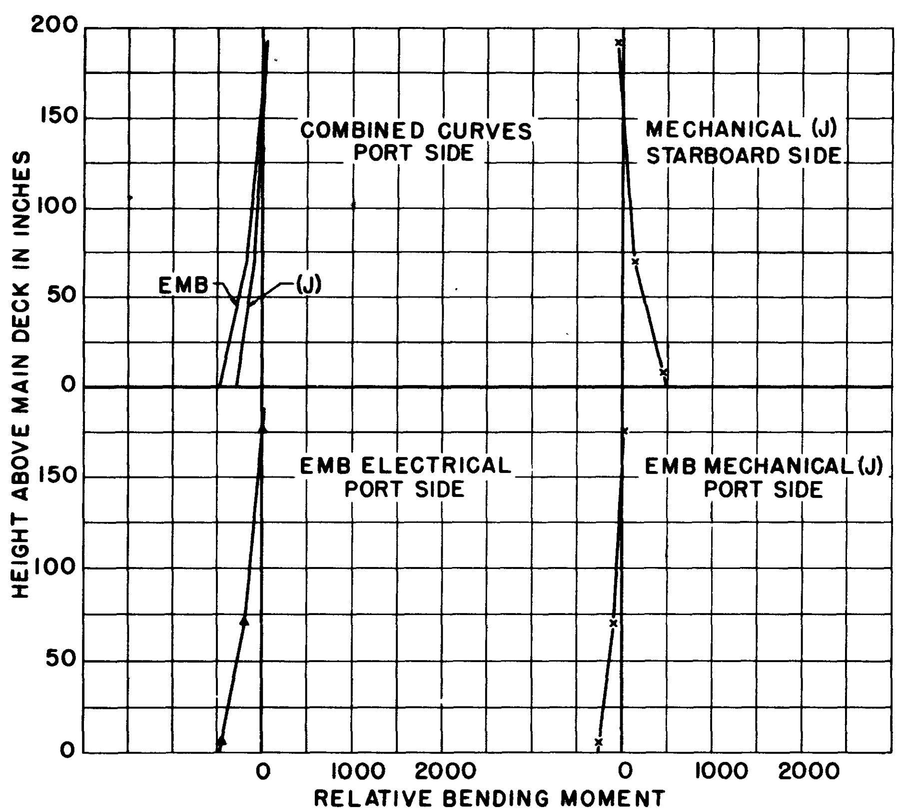

Figure 33 - USS WASP - Section of Zero Moment in Column, Frame 66 Inboard Port and Inboard Starboard

APPENDIX 3

ANALYSIS OF THE FRAME BY ELEMENTS

For purposes of analysis the knee is separated from the column and the beam. Consider the loads acting on the starboard column separately. These consist of a share of the external load, P, a moment M_B at the knee, a moment M_A at the deck, a horizontal reaction H_A at the deck, an axial compression V_A applied as shear from the beam through the knee, and a reaction at the deck, all as shown in Figure 34.

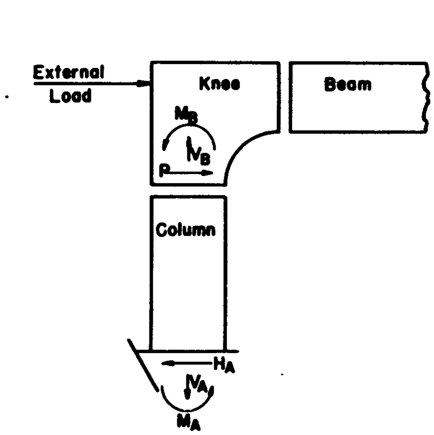

Figure 34 - Loads on Separated Column

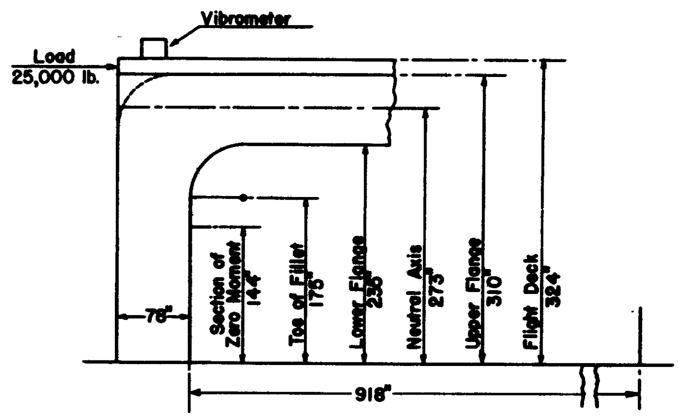

Figure 35 - USS WASP - Dimensions of Frame 54

The external load is applied at the centroid of the beam, 273 inches above the deck. The height of the column to the lower beam flange is 235 inches, but there is a fillet on the inner flange with a radius of 60 inches. Since there is no direct way of allowing for effective sectional constants within the limits of this fillet, it is convenient to consider the knee to extend to the toe of the fillet, in which case the length of the column effective for flexure would be 175 inches. The moment M_B is applied through the knee by the beam; its value for the present is undetermined; it is assumed to include whatever amount may arise from offset in opposed vertical loads.

The sum of the moments exerted by the knee M_B and by the main deck M_A on the column considered separately from the rest of the structure must equal the moment of the external load and its reaction at the main deck. The amount of M_A depends on the degree of fixation at the main deck; the greater this is, the less M_B will be. Since the position of the section of zero moment is known, the resulting solution is very simple: M_A equals P times the height of the section of zero moment. P, the estimated share of the standard load, is 1/4 of 100,000 pounds, so that M_A is 3.6×10^6 inch-pounds. The moment acting at the centroid of the beam is then 3.2×10^6 inch-pounds.

The axial load on the starboard column is tensile; it depends on flexibility of the beam and in this case amounts to about 7 x 10^3 pounds. Allowance for this leads to a correction on the moments. The section of zero observed net stress in the inboard flange lies 144 inches above the deck. The axial tension at this point, however, is 62 pounds per square inch, offsetting a bending compression of this amount. The section of zero bending moment is estimated to lie 9 inches higher, 153 inches above the main deck. The revised values of M_B and M_A are 3.0 x 10^6 and 3.8 x 10^6 inch-pounds.

Bending moments and stresses resulting from these calculations are shown in Table 14. Net stress values are obtained by allowing for axial load, which has a

TABLE 14

USS WASP - Calculated and Observed Stress in Starboard Column, Frame 54 under 100 kips Total Horizontal Load

Height above Deck, inches	Calculated			Observed Stress	Nominal Stress
	Bending Moment x 10^{-6} in. lb.	Stress, lb/in²			
		Gross	Net		
4	3.7	577	538	414	960
26	3.15	466	427	263	865
50	2.55	400	361	154	752
68	2.1	386	342	163	670
86	1.65	338	291	163	588
187	.87	208	155	87	448
142	.25	70	7	34	335
179	- .47	-149	-212	-166	+167

uniform value, though the varying area of section causes variation in axial stress. For comparison a column is added in the table showing nominal stress calculated by direct application of the formulas of Figure 24 under plausible assumptions as to equivalent parameters.

Deflection of the column has also been calculated for comparison. The observed rigidity is 4 x 10^6 pounds per inch applied to the whole section, 1 x 10^6 inch-pounds for the column in question, giving a deflection of 0.025 inch under 100,000 pounds load on the section.

The first calculation of deflection is based on separation of components due to cantilever action, uniform moment, shear, and rotation of the knee, all based on averaged sectional data substituted in the usual formulas. The second calculation consists of an integration extended over the variable values of M/I. Results are given in Table 15.

TABLE 15

USS WASP - Calculated and Observed Deflections per 100-kip Load on Section Column at Frame 54, Starboard. Units of 0.001 inch

	175 in. Cantilever			Knee Rotation	Total
	Concentrated Load	Uniform Moment	Shear		
Calculated I	7.06	-1.87	8.84	7.34	23.37
Calculated II	4.05		8.84	4.29	17.19
Observed					25.

The second calculation is considered more accurate than the first and the excess of observed over calculated deflection is considered to be due to incomplete rigidity in the knee and to deflection in the hull itself.

The purpose of these calculations of stress and of deflection of a column separately from the rest of the frame is to find whether by such analytical methods it may not be possible to obtain closer agreement with observation than by use of the nominal formulas for an equivalent bent as in Table 9. Tables 14 and 15 indicate that more detailed methods of calculation lead in the direction of closer agreement with observation.

The problem remains of devising a design procedure which can be used when definite information as to the position of the section of zero moment is not available. This task goes beyond that of analysis of experimental data, but plausible assumptions on this point made in the light of the present data would seem acceptable.