DISTRIBUTION OF STRESS IN A RECTANGULAR STEEL BEAM UNDER IMPACT LOADING

by

Roger Gerard Bowden

A Thesis Submitted to the Graduate Faculty for the Degree of

MASTER OF SCIENCE

Major Subject: Theoretical and Applied Mechanics

Iowa State College

1948

TABLE OF CONTENTS

INTRODUCTION

Objectives

The problem that has been treated in the investigation developed from the consideration of a problem in stress concentration. Initially the problem was a study of the standard A. S. T. M. Charpy or simple beam-type impact specimen. The beam was to be investigated for stress concentration in the neighborhood of the notch.

In order to conduct the investigation a model of the Charpy impact specimen was to be constructed of steel, geometrically similar to the specimen and with a scale of ten to one. Thus the model would have been ten times the size of the prototype in each dimension. This size was decided by the choice of the method for measuring strains, namely through the use of Baldwin Southwark SR-4 type resistance strain gages. Since the particular gages used had a gage length of 13/16 in., the model would necessarily be large. The model was to be loaded in impact as is the Charpy impact specimen; however, the load on the model would be applied by dropping a weight on it, while a swinging hammer is used for loading the Charpy specimen.

During the preliminary work it became apparent that for the information obtained to be of use in predicting the characteristics of the Charpy impact specimen it would be well to know something about the energy absorbed by the beam without the notch and also the stress distribution in the same beam. So little of the needed preliminary information proved to be available in the literature that it became apparent that the first step needed to be a study of stress distribution in the unnotched beam and it is with that phase that this thesis is concerned. The stress concentration at the notch was left as a future phase of the general problem.

The investigation reported in this thesis had two main objectives; first to determine the distribution of strain in a beam without a notch and second to determine the energy absorbed by the unnotched beam.

Review of the Literature

Timoshenko, St. Venant, Cox, Young, and others have all made theoretical analyses of impact problems, but until recently little experimental work had been done in this field. Hallam and Southwell (1) were concerned with a new type of impact testing machine which was designed to prevent any loss of the energy of the striker to the earth. The machine also made use of a new method of applying an impulsive load, so that a notched specimen would be subjected to only a bending moment. Finally a new type of impact specimen was designed. Loss of energy to the earth was prevented by supporting the anvil and the striker from wires in the manner of a ballistic pendulum. The results of tests using the new machine compared favorably with tests of standard Izod impact specimens.

In 1936 H. L. Mason (2) presented a paper containing both theoretical and experimental results for impact on beams. The theoretical solution was determined by the method of Timoshenko and the results of the solution compared with the experimental values.

Mason's experimental equipment consisted in part of an I-beam laid on its side and loaded at the center by a weight swinging on a V-loop of wire. Strain was measured by a single magnetic strain gage attached to the beam, a little off center to minimize the effect of local compressive stresses at the impact point. To record the strain the output of the gage was impressed on an oscillograph. Experimental results obtained in the investigation indicated that when the weight struck the beam there were several impacts and also that the stresses determined were about twice those indicated by the approximations used.

When two elastic bodies collide part of the initial kinetic energy is dissipated in starting elastic waves in the bodies. Most analytical methods hold only if the energy dissipated in the formation of elastic waves is a small portion of the original kinetic energy. Zener and Feschbach (3) have developed an approximate method that can be applied to problems where the energy dissipated is high. The error introduced by the approximate method was shown to be only 0.7% for energy dissipations as high as ninety percent of the initial kinetic energy.

Since the integral equation for the determination of stress and deflection for the horizontal impact of a mass on a beam developed by Timoshenko was tedious to apply, Lee (4) determined a shorter approximate equation for a single impact, but with provisions for repeated impacts. In the solution it was assumed that the time of contact was small in comparison with the period of vibration of the beam. Values determined from the equations were checked against the lengthier method of Timoshenko. The results for both the compressive forces and the motion agreed in some cases, but were definitely different in others. The author indicated that the deflection curves of the beam produced by impact differed widely from static deflection curves for the same beam.

Results of static tension tests for both notched and unnotched specimens were compared with Charpy notched-beam impact tests by

MacGregor and Fisher (5). A comparison was made by determining the energy absorbed by the tension specimen through the use of the true stress-strain curves and comparing this result with the energy absorbed per unit volume by a Charpy impact specimen, the latter values calculated after the method of Moser. The uniform tension specimens did not correlate as well with notched-beam impact tests as did the notched tension specimens. The authors concluded that while the static tension test of a notched specimen compared most closely with the Charpy impact test, the uniform tension test gave results that were definitely worthwhile. Although the static test of the notched specimen took longer to perform than the Charpy tests it had the advantage of supplying stress-strain data not available in the impact tests.

Fehr, Parker, and DeMicheal (6) have determined the tensile strength, the yield strength, and the breakage energy of cold rolled steel and of duralumimum, while being broken by a force applied at a great rate of speed. The values were found to be larger than the static strengths up to the maximum velocity used, in this case 100 feet per second. Measurements of strain were made using two SR-4 resistance strain gages, one on each side of the specimen to eliminate the effects of any flexural stresses that might occur.

A theoretical treatment of the phenomenon of impact presented by Welch and Quackenbos (7) indicates that the conditions for rupture under dynamic loading result from the same basis as for static or flexural failures. It was indicated that the theoretical treatment was not only applicable to plastics, but also to other materials of construction. The theoretical results were verified by experimental work using SR-4 strain gages for measuring strains in a beam loaded with a falling weight.

The authors concluded that the stress-strain diagrams for static and dynamic loading were essentially identical, that the dynamic breaking strength was greater than the static ultimate strength by ten percent due to the high rate of loading, that the modulus of elasticity was the same for static and impact tests, that the energy absorbed by the impact specimens was ten to fifty percent of the available energy, that the ability of the Charpy impact test to differentiate between materials alike in flexural and tensile properties was due to a large variation in the notch sensitivity of the materials tested, and finally that static flexural test techniques are sufficient to predict the service properties of certain plastics.

Fischer (8) has recently presented a solution of the general equation for lateral vibration of a uniform beam with hinged ends. The hinged ends with supports attached were displaced upward by a blow from a swinging hammer and then the ends were subjected to a complete reversal of motion. Experimental results were determined through the use of resistance strain gages, one at the quarter-point and one at the center of the beam. In conclusion the author stated that peak acceleration should not in general be a measure of the severity of a shock load.

A method for determining the strained state of a long wire or bar subjected to longitudinal impact of a finite duration has been treated by White and Griffis (9). The impact stress was constant and greater than the yield strength of the material. The velocity of the front and rear of the stress wave was a function of the local stress and strain, and the wave was calculated from the slope of the stress-strain diagram. It was shown that the final strains were greatest at the impact point, that these strains were followed by a region of constant strain due to the back of the wave overtaking the front, and that following the region of constant strain there existed a section of decreasing residual strain. The author discussed secondary wave effects and also indicated that a critical velocity of impact corresponding to the ultimate strength of the material could be determined.

Another theoretical treatment of impact is given in the article on standard procedure.

The objectives of the investigation differed from the work previously done in the field of impact loading in that the strain distribution in the beam was determined, as well as the energy absorbed by the beam.

Scope of the Investigation

The study was limited to a beam of structural steel 21.65 in. long, with a square cross section 3.94 in. on a side, and with a supported length of 15.74 in. The model was similar to a Charpy impact specimen except that it was left without the notch that occurs in the impact specimen.

Strains were measured at the quarterpoints of the beam, the center of the beam, and a single measurement was made on each of the supports. The magnitude of the weight used in loading the beam was 54.42 lb. and it was dropped onto the center of the beam from three distances above the beam; 5.03 in., 7.03 in., and 9.03 in. A check on the accuracy of the gages and the measuring devices was made by subjecting the beam to a given static load in a testing machine.

THEORY

Standard Procedure

A standard method for the solution of the problem of an impact load on a beam is given in most texts in mechanics of materials. In particular Seely (10) presents a method, known as the equivalent static load method, of determining the stresses and deflections in a member subjected to an impact or energy load. The maximum pressure or force exerted by the moving body on the member is determined and that force is considered as a static load.

Before a solution can be effected an estimate must be made of the proportion of the energy load effective in producing deformation of the member. The estimate is one of the weakest points, since it is really only a qualified guess. It is known that a portion of the energy is dissipated in deforming the weight, that more of the energy is transmitted through the supports to the earth, and finally that there are energy losses due to local deformations at the point of contact.

The problem of a simple beam with a central load produced by a falling weight can be solved by the method of equivalent static loads in the following manner. If Q is a static load that produces the same maximum deflection Δ as that caused by the energy load, then the work done by Q, which is $\frac{1}{2}Q\Delta$, will be equal to the energy supplied by the falling weight, which is $W(h+\Delta)$. W is the magnitude of the weight, h is the height from which the weight drops, and Δ is the maximum deflection of the beam, which occurs at the center. Then the work done by the falling weight may be equated to the work done by the static load.

Thus

$$W(h+\Delta) = \tfrac{1}{2}Q\Delta. \tag{1}$$

Since the weight of the beam is approximately twice the magnitude of the falling weight the stresses due to impact are reduced by local yielding of the member, in effect reducing the amount of energy that is useful in producing deflection. To account for this effect a factor, n, given by Seely (10) is introduced into Eq. 1 and is equal to $\frac{1+\frac{7}{15}q}{(1+\frac{5}{8}q)^2}$. In the factor n, q is equal to B/W, where B is the weight of the beam and W is the magnitude of the falling weight. The factor, n, does not account for all of the energy lost, but the values obtained from the solution can be compared with experimental results.

Introducing the factor, n, into Eq. 1, we obtain

$$\frac{(1+\frac{7}{15}q)\,W(h+\Delta)}{(1+\frac{5}{8}q)^2} = \frac{1}{2}Q\Delta. \tag{2}$$

The maximum deflection for a simple beam subjected to a static central load is equal to $\frac{QL^3}{48EI}$. Substituting the deflection into Eq. 2 and neglecting Δ in comparison to h in the term $(h+\Delta)$, since for a load, Q, of 40,000 lb. Δ is only about 0.005 in., while h varies from 5 to 9 in., Eq. 2 becomes

$$\frac{(1+\frac{7}{15}q)\,Wh}{(1+\frac{5}{8}q)^2} = \frac{Q^2L^3}{96EI}. \tag{3}$$

A more convenient form of the last equation can be found by solving it for Q.

$$Q = \sqrt{\frac{96\left(1+\frac{7}{12}q\right) W h E I}{\left(1+\frac{5}{8}q\right)^2 L^3}} \quad . \qquad (4)$$

In this investigation: B = 95.05 lb.

W = 54.42 lb.

q = B/W

h = 5.03, 7.03, and 9.03 in.

E = 30,000,000 psi.

I = 20.1 in.^{4}

L = 15.74 in.

and for h = 5.03 in., Q = 41,150 lb.

h = 7.03 in., Q = 48,600 lb.

h = 9.03 in., Q = 55,150 lb.

Calculating the static moment at the quarterpoint of the beam from the equation $M_s = \frac{PL}{8}$, we obtain for

h = 5.03 in., M = 81,000 in.-lb.

h = 7.03 in., M = 95,600 in.-lb.

h = 9.03 in., M = 108,600 in.-lb.

Assumptions

The method of Seely (10) depends on certain assumptions. First it was assumed that the stresses in the member were under the proportional limit of the material, second that a plane section before bending remains plane after bending, and third that the modulus of elasticity of the material remains the same for impact as for static loads. Also the energy dissipated through the formation of stress waves, by vibration, and by loss of energy to the earth through the supports was not taken into account. Finally the analysis does not account for any possible differences in the elastic curves of the beam under impact loading as opposed to static loading.

THE INVESTIGATION

Material, Specimen, and Testing Arrangement

The model tested was a steel beam of square cross section. The steel, a Bethlehem structural steel, had the following chemical composition: Carbon 0.18%, Manganese 0.44%, Phosphorus 0.011%, and Sulfur 0.025%. The beam was a model of the Charpy impact specimen except that it contained no notch. It had a length of 21.65 ± 0.01 in., a width and depth of 3.94 ± 0.001 in., and a supported length of 15.74 in.

The centrally loaded beam was supported on two standard steel beam supports bolted to a steel base plate. Two I-sections were bolted to the base plate at the center, forming the supports for a platform carrying an electromagnet. The platform was adjustable to permit a variation in the height of drop from five inches to approximately two feet.

The weight used was a standard 50-lb. weight to which was attached a rectangular steel block with a polished face to form the pole piece for the magnet. A one-half inch diameter, half-round steel rod was attached to the bottom of the weight to localize the application of the load.

The electromagnet had a core of hyperoyl steel wound with over one hundred turns of number 18 wire and with the current supplied by an Edison type storage cell. When the current was turned off the steel pole piece did not release immediately due to residual magnetism in the steel. To eliminate this difficulty a reversing switch was introduced into the circuit.

Two types of Baldwin Southwark SR-4 resistance type strain gages were used to measure strain in the beam. They were type A-1 gages with a resistance of 120 ± 0.2 ohms, and a gage factor of 2.04 ± 1%, and type AR-4-1 strain gage rosettes with a resistance of 120 ± 0.2 ohms, a gage factor of 2.04 ± 1%, and an auxiliary factor of 1/80. All the gages of each type were taken from one lot. The A-1 gage measures unit strain in a single direction, while the AR-4-1 rosette measures three unit strains in directions at 60 degrees to each other. The exact location of the gages on the beam is shown in Fig. 1.

A set of six gages of the same type and lot as those on the beam was placed on a steel plate to serve as a compensator. Each compensator gage and a gage on the beam (the active gage) formed two sides of a Wheatstone bridge. The other two sides were formed by two gages attached to a panel which had connections for six active and six compensator gages.

The input current was supplied to the gages by an oscillator with a frequency of 2000 cycles per sec. and the output current of the gages was put through an amplifier in conjunction with the oscillator. The unit containing six amplifier elements, so that six separate readings of strain in different gages could be made simultaneously, and the oscillator was a Hathaway type MRC-12 control unit. The oscillator had a variable voltage, and the amplifiers had various ranges of sensitivity in order that the deflection of the oscillograph for a given unit strain could be adjusted to any desired value.

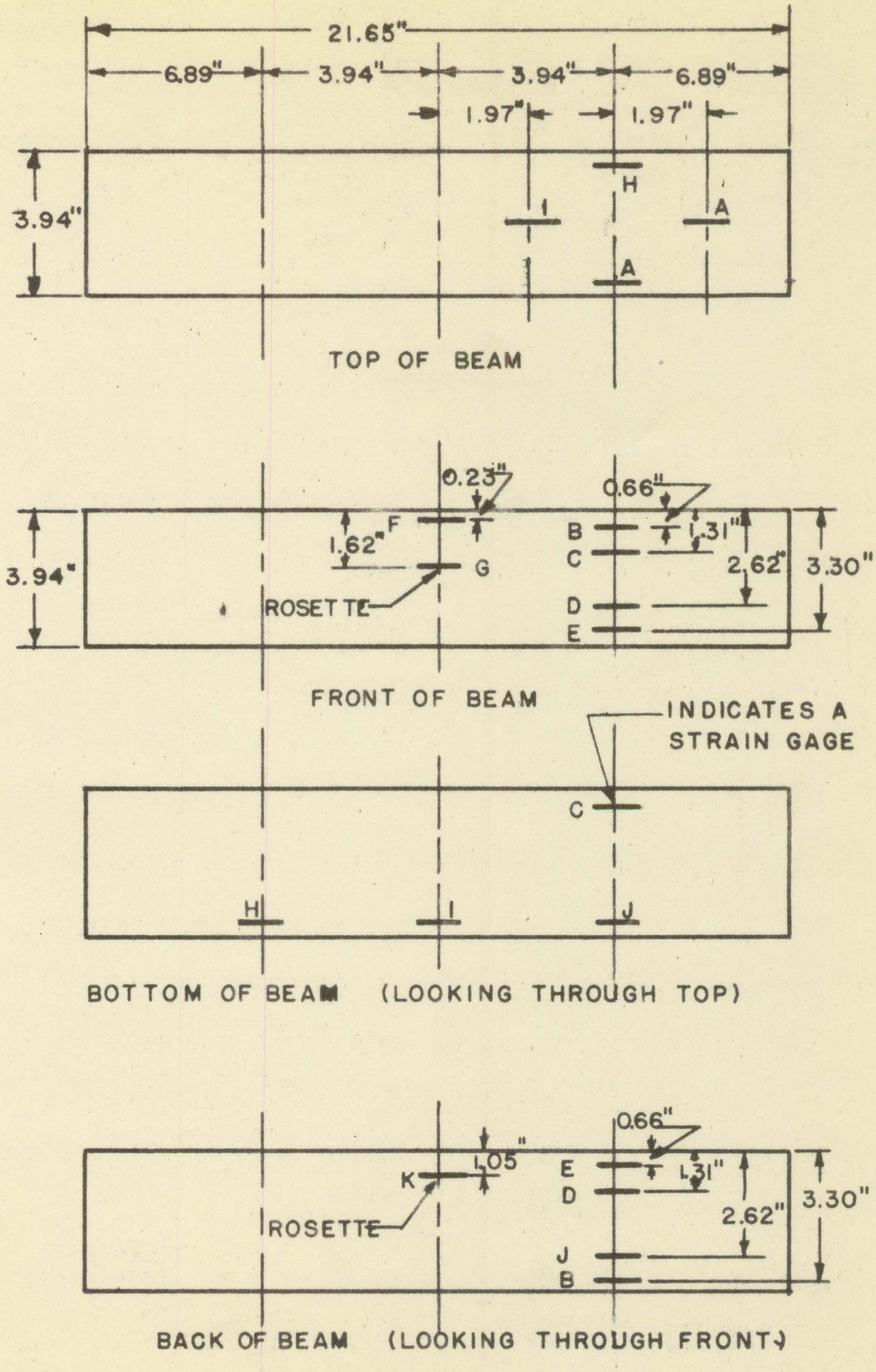

FIG. I. LOCATION OF THE STRAIN GAGES ON THE BEAM.

The output of the amplifiers was put on a Hathaway 1-S-14 galvanometer type oscillograph. The oscillograph contained six galvanometers, one for each of the channels (amplifiers) in the control unit. A mirror was attached to each galvanometer and a slit of light was focussed on each of the mirrors, the mirrors reflecting the beams of light onto a lens in the end of the oscillograph housing. A motor-driven film magazine containing a continuous roll of sensitized paper was attached to the end of the oscillograph. When the shutter was opened the six beams of light recorded as traces on the sensitized paper.

A timing wave was introduced into the circuit in certain of the runs by connecting a 60 cycle current into the oscillograph. The current input was kept at a very low value by putting a 150,000-ohm resistance in series with the lead bringing in the 60 cycle current.

The oscillograph was calibrated using an SR-4 standard calibrating box graduated in steps of 1000 micro-inches per inch strain.

The static tests were run in a Tinius Olsen 20,000-lb. capacity hand-operated testing machine.

Fig. 2. shows a general picture of the experimental equipment. The beam, the supports, the electromagnet, and the falling weight are shown on the left. The control unit is placed on the left side of the table while the oscillograph is on the right. Fig. 3. gives a closeup of the end of the beam, the electromagnet, the falling weight and the supports. In the photograph the weight is held by the electromagnet ready to be dropped on the beam. The strain gages shown in the figure are the gages at the quarterpoint of the beam.

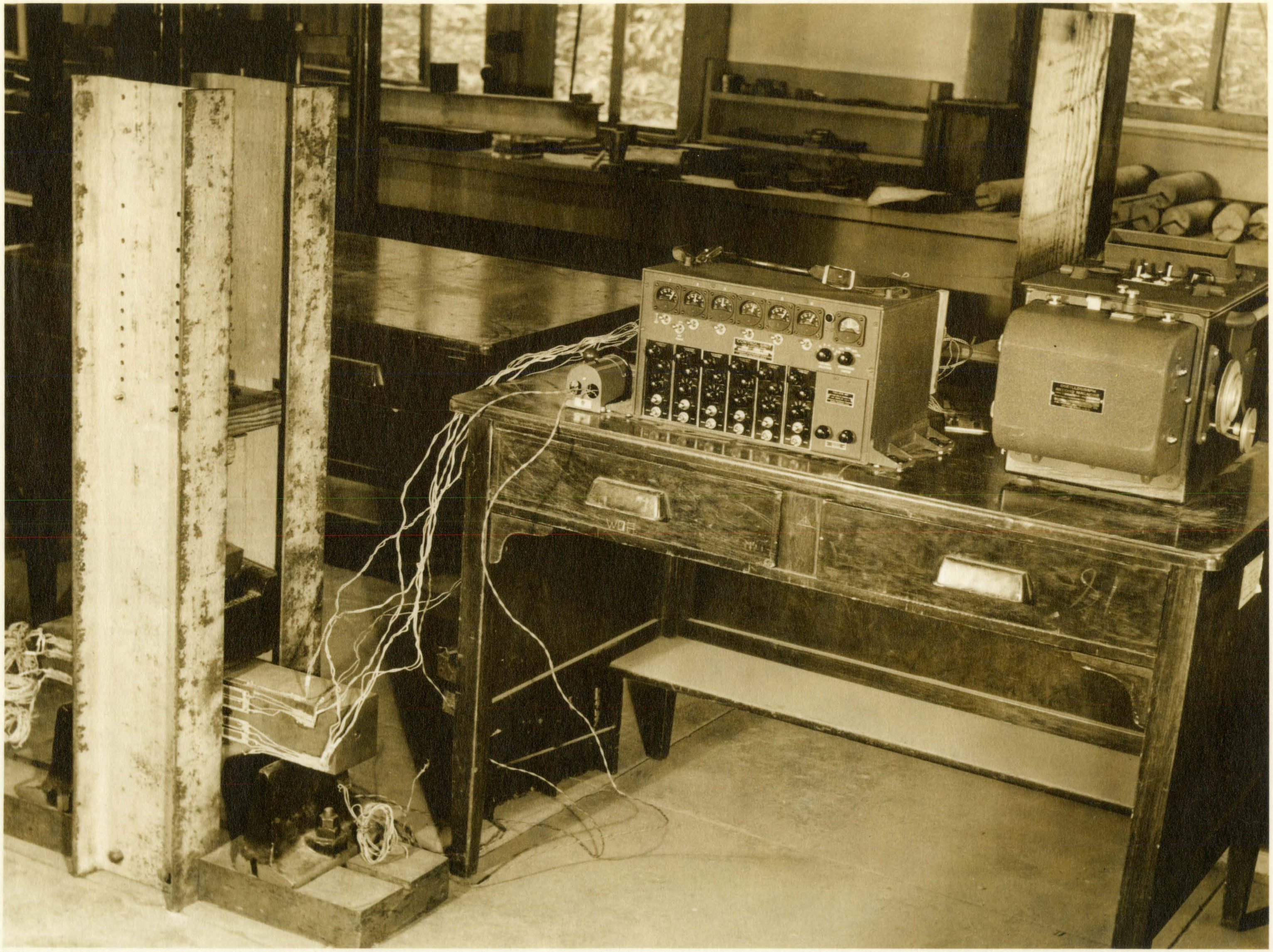

Fig. 2. General View of Equipment.

Fig. 3. Closeup of Beam and Supports.

Procedure

Before any actual tests were made the oscillograph was calibrated. Each channel was calibrated by putting the standard calibrating box in place of the active and compensator gages on the panel attached to the control unit. The gage voltage of the oscillator was adjusted to the desired value, the bridge balanced, and a short run was made with the calibrating box at zero strain and then at 1000 micro-in. per in. strain. The calibration was repeated for all six channels in the same manner. Then when the record was developed the calibration factor could be determined as the distance between the first and second positions of the galvanometer trace.

After the calibrations were made the beam was put in a standard static testing machine, a given load was applied at the center of the beam, and the strains in the beam at the given load were found from the oscillograph record. The stresses were then computed from the strains and compared with the results calculated by the flexure formula. The comparison served as a check on the accuracy of the SR-4 strain gages.

Then the actual impact tests were conducted at the three heights of drop, 5.03, 7.03, and 9.03 in. First the weight was attached to the electromagnet to remove all load from the beam. The amplifier and bridge circuits were balanced, so that there was no current flowing in the amplifiers or the bridges. The weight was then dropped and a record of the strains was made on the sensitized paper. The film in the magazine ran at a speed of about six inches per second to define the waves of strain clearly. After the run the film magazine was removed, the sensitized paper that had been exposed was developed, and the values of strain determined by scaling the distances from the film and then applying the calibration factors.

RESULTS

Discussion of Data

The static test run on the beam showed that the values of strain determined from the strain gages were no more than 20% from the values of strain determined by the flexure formula and in most cases the strains were within 10% of each other. The results of the test are shown in Fig. 4.

At the point of application of the load and also at the supports there was considerable local yielding of the beam, indicating that an appreciable amount of the energy of the falling weight was dissipated in producing local deformation of the beam. The local deformations tended to confirm the results as presented by Seely for the decrease in energy load due to local yielding.

The curves presented in Fig. 5-10. were plotted at the quarterpoint of the beam and show the strain distribution in the cross section at various times. The results show that a plane section before bending does not remain so after bending, since the distribution is not linear in any of the cases. The strain distribution, starting with the strain at a maximum in compression in the top of the beam at the quarterpoint, is of all the cross section in varying compression. Then referring to one-eighth and one-quarter of a cycle from the maximum in the top gage the distribution retains the same general shape, but moves toward tension. At three-eighths and

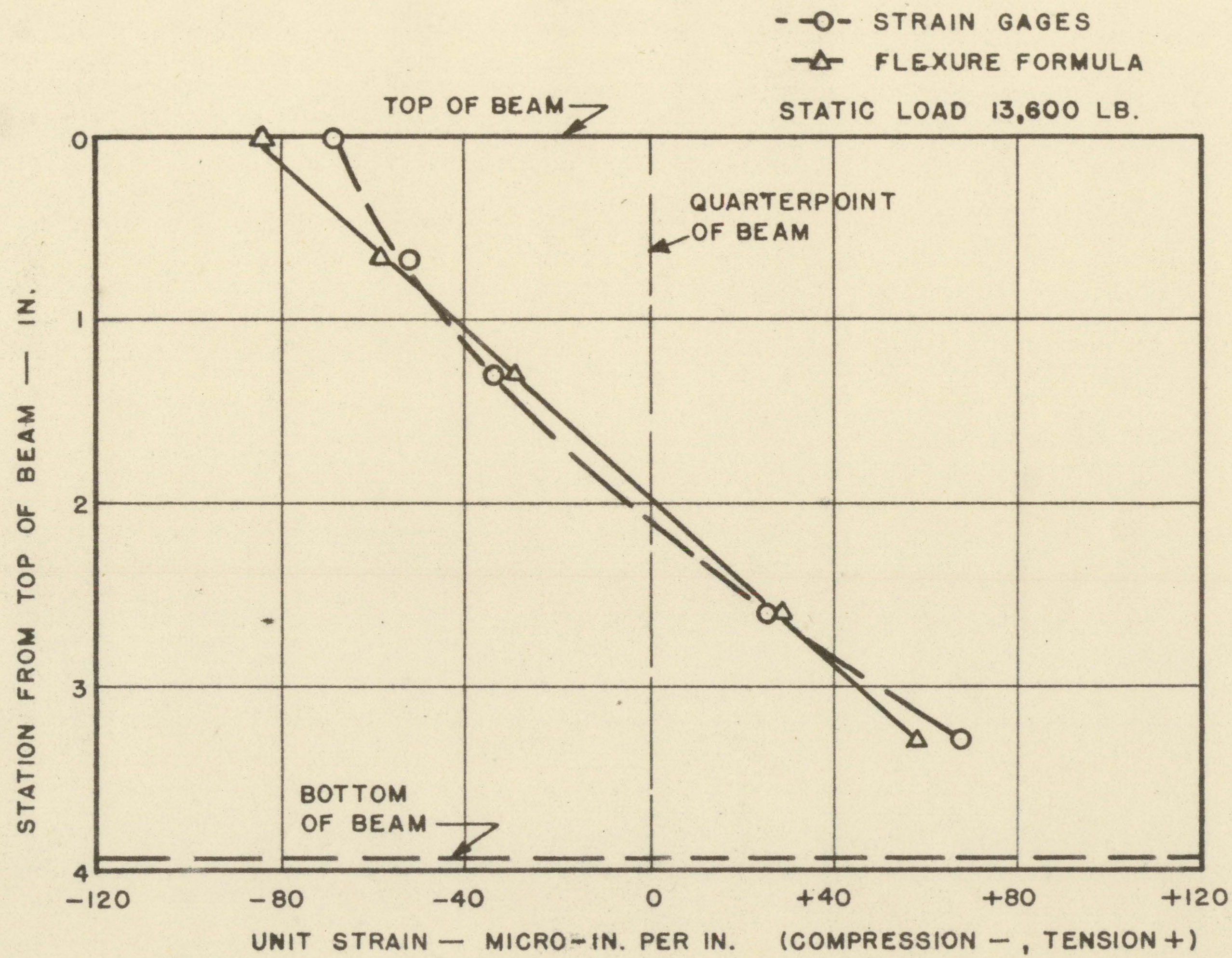

FIG. 4. COMPARISON OF STRAIN MEASUREMENT METHODS.

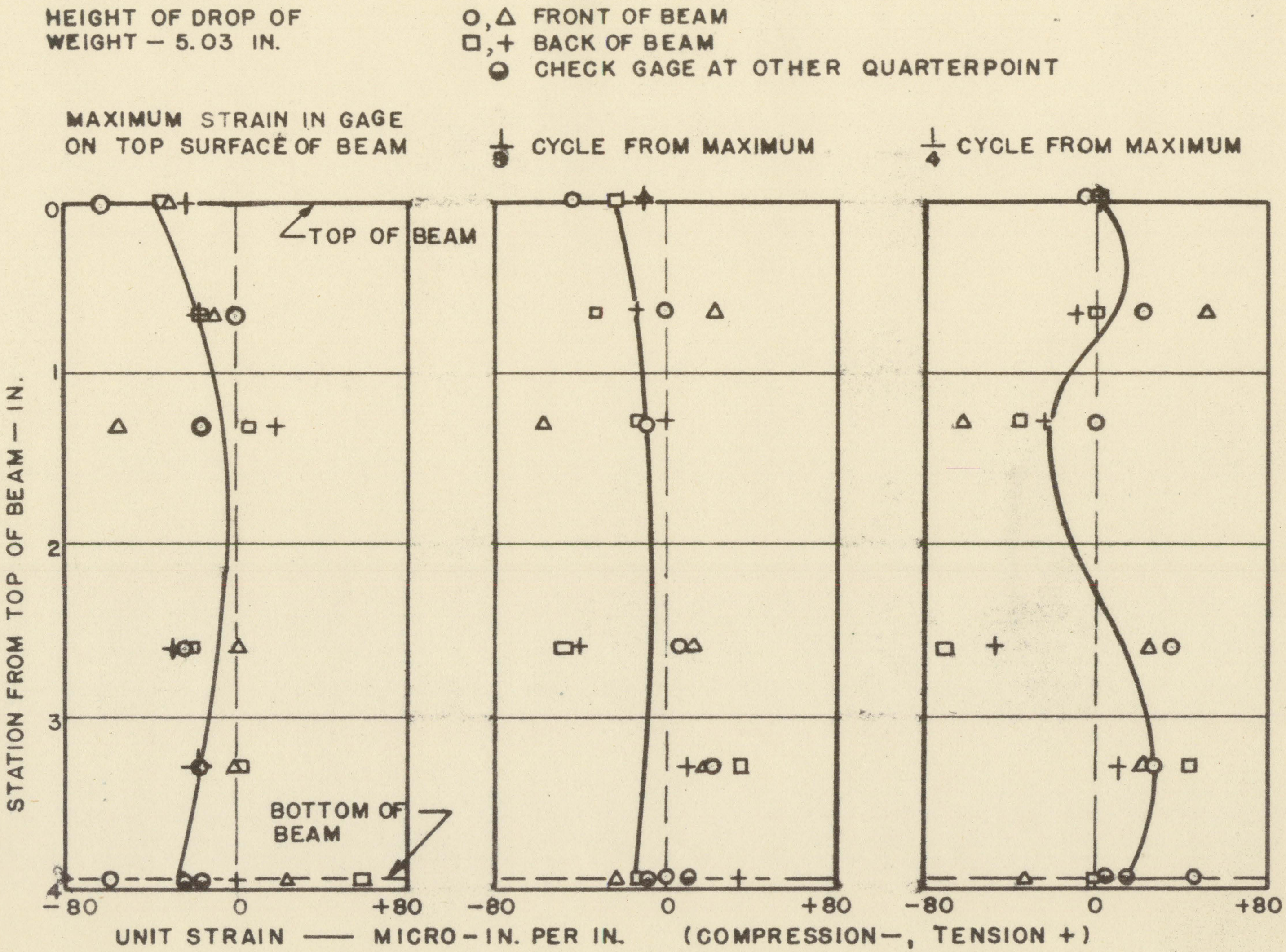

FIG. 5. DISTRIBUTION OF STRAIN AT THE QUARTERPOINT OF THE BEAM.

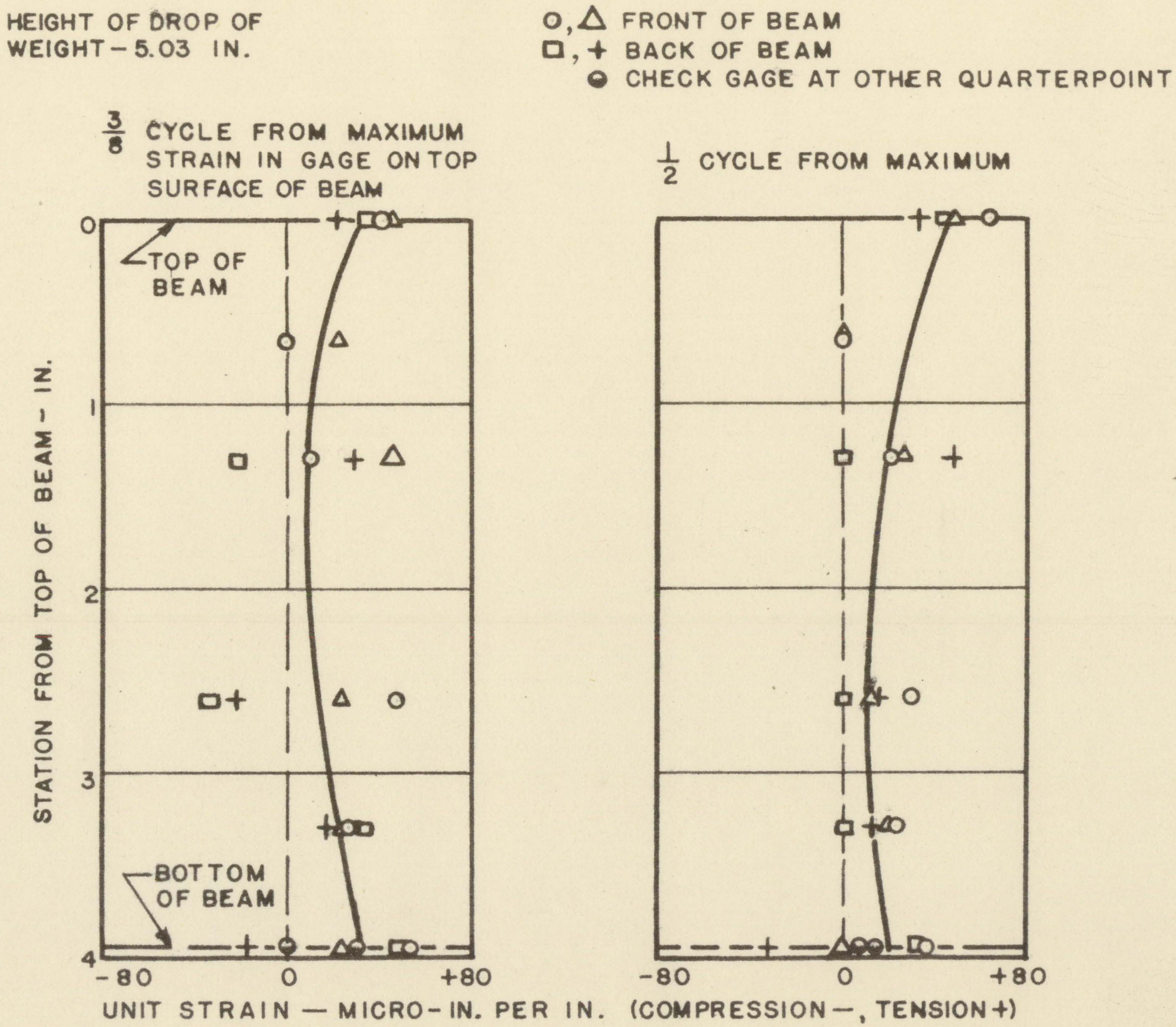

FIG. 6. DISTRIBUTION OF STRAIN AT THE QUARTERPOINT OF THE BEAM.

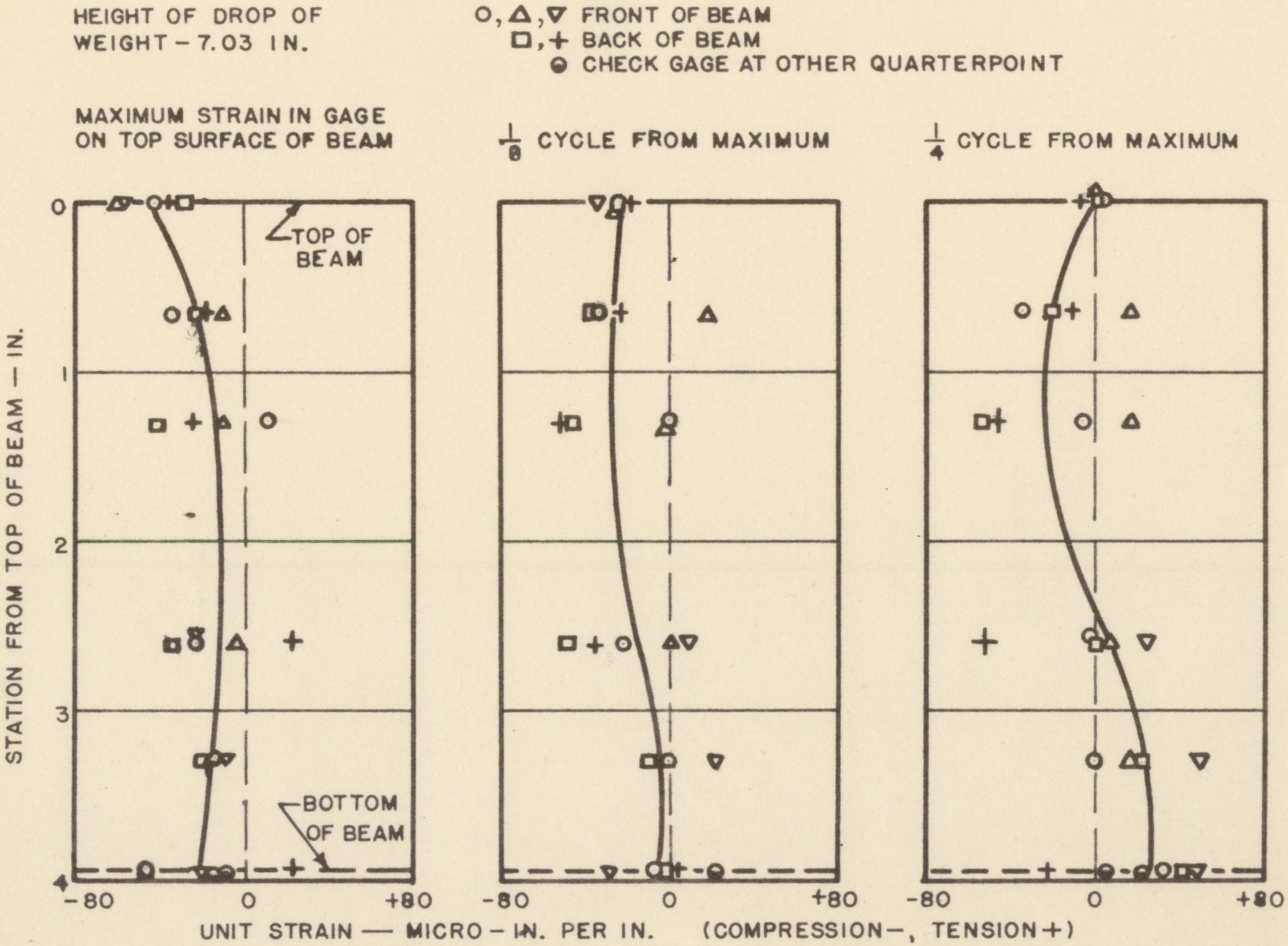

FIG. 7. DISTRIBUTION OF STRAIN AT THE QUARTERPOINT OF THE BEAM

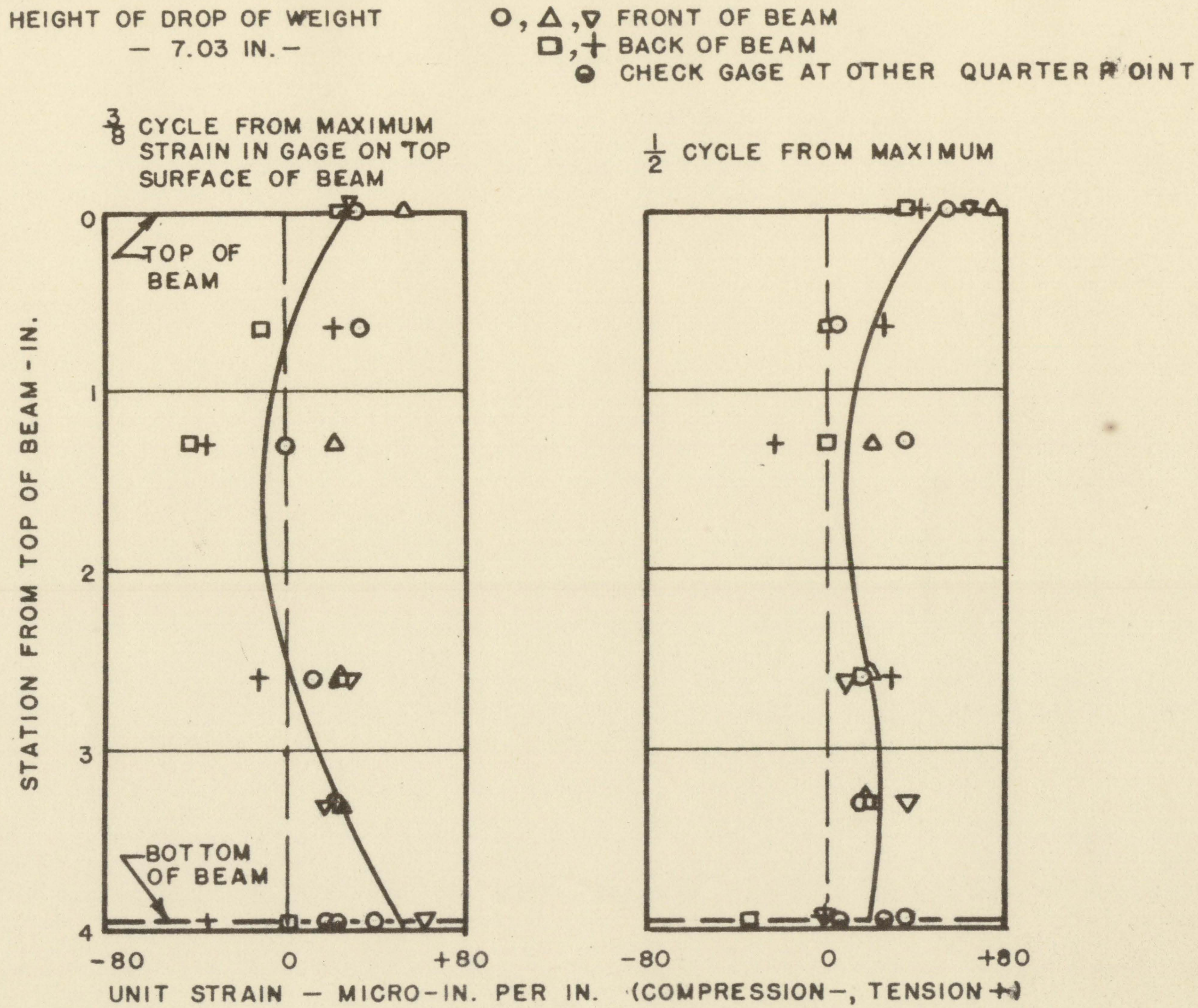

FIG. 8. DISTRIBUTION OF STRAIN AT THE QUARTER POINT OF THE BEAM.

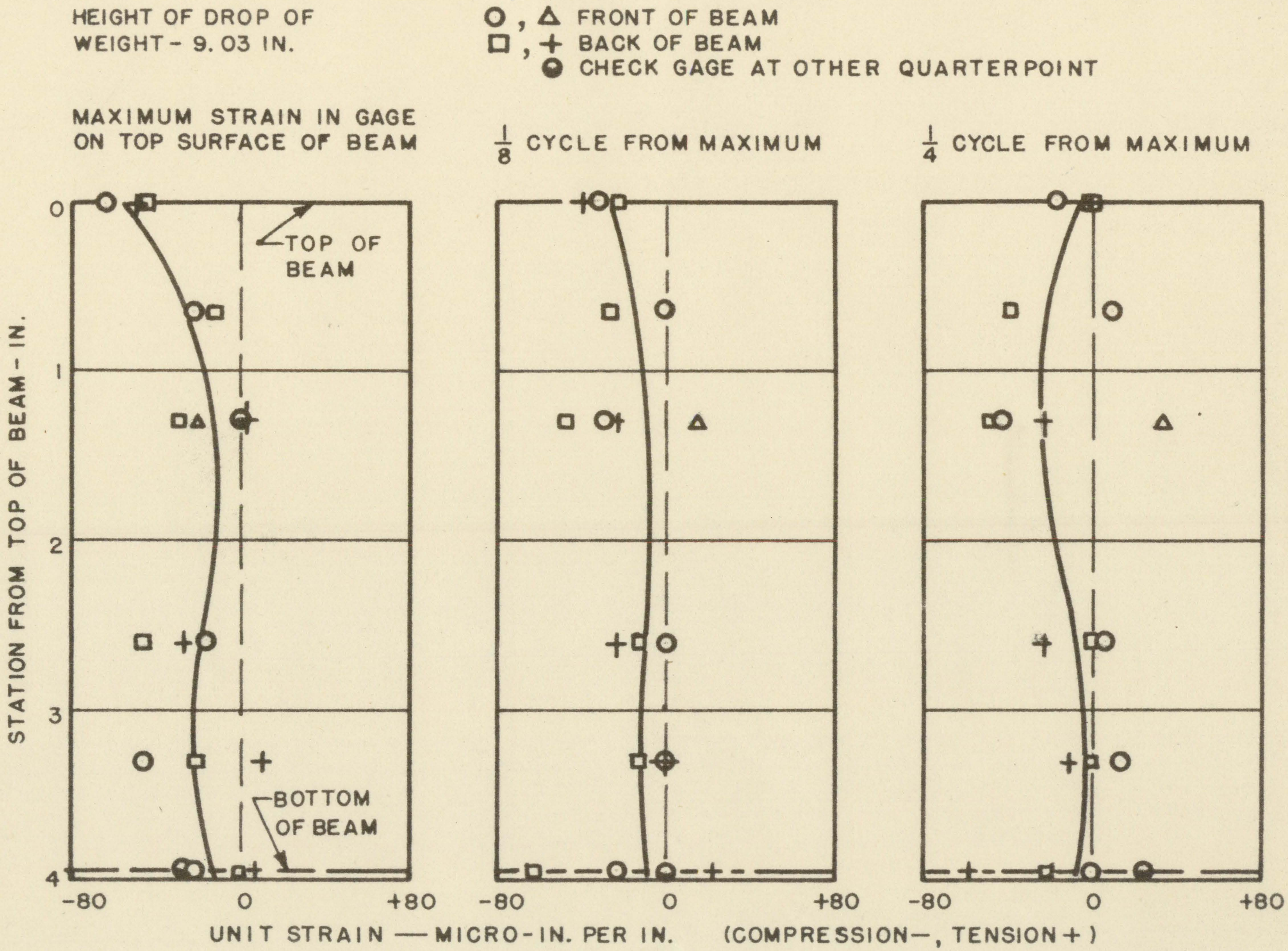

FIG. 9. DISTRIBUTION OF STRAIN AT THE QUARTERPOINT OF THE BEAM.

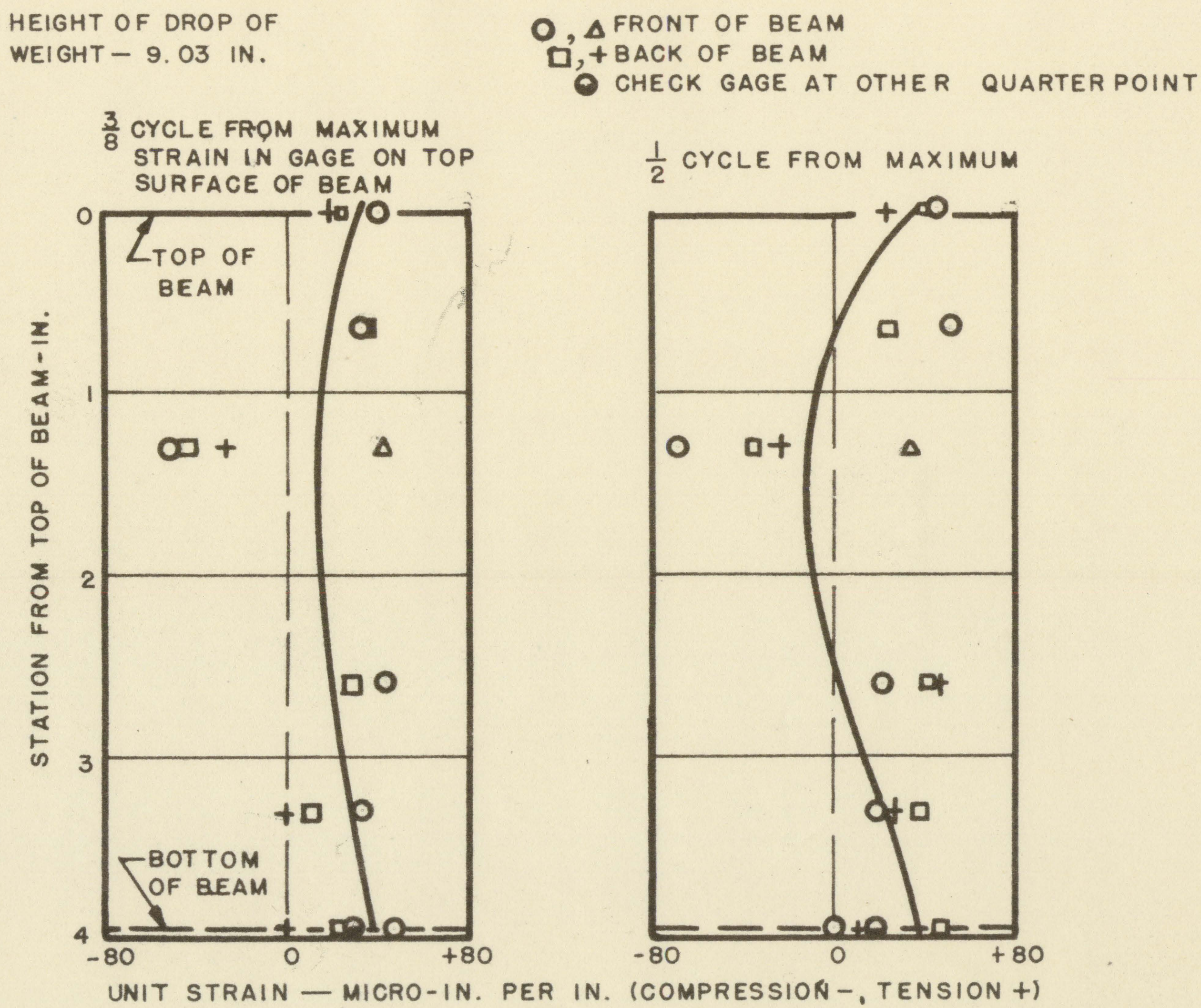

FIG. 10. DISTRIBUTION OF STRAIN AT THE QUARTERPOINT OF THE BEAM.

one-half of a cycle from the maximum in the gage on the top of the beam almost all of the gages show tension and reversals of curvature.

The values of strain determined at the quarterpoints on each side of the beam were somewhat different, indicating that the load was not exactly level when it struck the beam. In certain of the gages the values of strain were not at all consistent from one run to another. The waves of strain in the case of inconsistent results had frequencies that varied up to 100%, while the more consistent strains had much the same frequencies for all the runs. All of the frequencies are tabulated in Table 1. The value for the frequency of natural vibration of the beam computed from Timoshenko (11) was 1360 cycles per sec., while the waves of strain shown in the results of the gages had frequencies of about 30 cycles per sec. Thus the frequencies of the waves of strain were not the same as the natural frequency of the beam. The gages whose results were in question were sometimes difficult to balance in any channel, indicating that the gage may have had a poor connection. The external connections were checked carefully and there is a possibility that the difficulty was in the gage itself.

The strain determined in the investigation indicates that the beam does not vibrate laterally, but tends to vibrate longitudinally, with the formation of strain waves traveling to the end of the beam. Since the section is large in cross section in comparison with the length, the results may be applicable only to the type of beam considered here. The reversal of curvature of the strain distribution

Table I

Frequencies of the Waves of Strain

Front of Beam Gage*	Frequency (1) Cycles Per. sec.	Frequency (2) Cycles Per sec.	Back of Beam Gage*	Frequency (3) Cycles Per sec.	Frequency (4) Cycles Per sec.	Height of drop of weight in.
A	29.1	29.2	H	36.5	34.3	5.03
B	30.6	30.7	E	32.4	35.4	"
C	29.1	33.6	D	32.4	32.4	"
D	30.6	31.8	J	32.4	31.3	"
E	29.1	32.3	B	30.7	29.2	"
J	37.8	29.6	C	36.5	34.3	"
A	29.9	32.0	H	30.7	32.4	7.03
B	24.3	30.7	E	32.4	36.5	"
C	40.2	15.8	D	29.2	34.3	"
D	32.4	31.5	J	29.2	29.2	"
E	32.4	34.3	B	36.4	—	"
J	29.9	29.2	C	34.4	45.0	"
A	30.0	—	H	33.8	30.0	9.03
B	33.7	—	E	36.5	34.3	"
C	22.5	36.4	D	30.7	29.2	"
D	31.3	—	J	29.4	29.2	"
E	33.3	—	B	29.2	—	"
J	60.0	—	C	30.0	32.4	"

* See Fig. 1.

curves further indicates the presence of a strain wave traveling along the beam.

There is no assurance that the usual stress-strain relationships will hold for the results of the investigation. With this limitation in mind the amount of energy absorbed by the beam was calculated in two ways. First the moment at the quarterpoint of the beam was computed from the strain distribution curves and compared with the moment at the same point determined from the equivalent static load. The experimental results indicated that the relative amount of energy per unit length absorbed by the beam at the quarterpoint was 0.89%, 1.642%, and 1.542% of the amount predicted by the equivalent static load method. The three values are respectively for heights of drop of 5.03, 7.03, and 9.03 in. respectively. The second method was to calculate the maximum static equivalent stress at the quarterpoint for each height of drop of the weight. The flexure formula was used and the moments were found through the use of the equivalent static loads. The maximum stress, which occurred in the top surface of the beam, was then converted to strain using the usual relation that unit strain is equal to unit stress divided by the modulus of elasticity. The values of maximum strain from the experimental results were also found to be at the top of the beam allowing a direct comparison. For the three heights of drop in increasing order the ratios of the experimental maximum strain divided by the maximum strain computed from the equivalent static load were respectively 0.173, 0.166, and 0.157 or in terms of percentages

17.3%, 16.6%, and 15.7%. The values determined from the second method are more in line with the results of other investigators, and thus are probably more accurate, since the choice of a moment axis in the first method is arbitrary due to the absence of a neutral axis.

The strain gages on the supports did not show an appreciable amount of strain. Therefore no values are available for the amount of energy absorbed by the supports.

The investigation gives no certain indication as to what happens to the energy not absorbed by the beam; however, it appears that a great deal of the energy is transmitted through the supports to the floor.

Conclusions

Within the accuracy and completeness of the data the following conclusions seem justified.

(1) The values of strain determined from the gages in a static test differed from the results as computed by the flexure formula by no more than 20% and in most cases by no more than 10%. See Fig. 4.

(2) The strain distribution indicates that the beam tends to vibrate longitudinally and not laterally.

(3) A plane section before bending does not remain plane after bending. See Fig. 5-10.

(4) The relative amount of energy per unit length absorbed by the beam at the quarterpoint as determined in the investigation was 0.89% to 1.54% by the first method and 15.7% to 17.3% by the second method of the energy absorption predicted by the equivalent static load method. The first method of determining the energy absorbed is based on the assumption that the usual stress-strain relationships hold, but in the second case these relationships must only hold for static loading. It is clear that the experimental results show that the theory does not account for the major energy losses. In this investigation it appears that the major energy loss may be through the supports to the earth.

RECOMMENDATIONS FOR FURTHER STUDY

The investigation carried out in this thesis and the other investigations quoted in the review of the literature constitute a start in the field of impact loading. Many problems of great importance remain to be solved.

One of the problems remaining is to find out what actually happens to the modulus of elasticity of the material under impact loading. Another is to determine the amount of energy that is absorbed by the beam in shear. The problem originally suggested in this paper could also be carried out; that is the problem of the impact on a beam with a notch. The data on the stress concentration at the notch would be very valuable to a complete study of impact specimens. The beam supports might be mounted on springs in order to measure the energy lost through the beam supports.

While several theoretical solutions have been proposed for the impact of mass on a beam, there is still no solution that fits the experimental results as well as could be desired. Finally the problem that has been considered in this investigation could be carried further by the use of several weights of different magnitudes, and by the use of a series of beams of different cross sections.

LITERATURE CITED

(1) Hallam, H. and Southwell, R. V. Researches in Impact Testing Engineering. 138:689-690. 1934.

(2) Mason, H. L. Impact on Beams. Trans. Amer. Soc. Mech. Engrs. 58:A55-A61. 1936.

(3) Zener, C. and Feschbach, H. A Method of Calculating Energy Losses during Impact. Trans. Amer. Soc. Mech. Engrs. 61:A67-A70. 1939.

(4) Lee, E. H. The Impact of a Mass Striking a Beam. Trans. Amer. Soc. Mech. Engrs. 62:A129-A138. 1940.

(5) MacGregor, C. W. and Fisher, J. C. Relation between the Notched Beam Impact Test and the Static Tension Test. Trans. Amer. Soc. Mech. Engrs. 66:A28-A34. 1944.

(6) Fehr, R. O., Parker, E. R. and DeMicheal, D. J. Measurements of Dynamic Stress and Strain in Tensile Test Specimens. Trans. Amer. Soc. Mech. Engrs. 66:A65-A71. 1944.

(7) Welch, L. E. and Quackenbos, H. M. Relations between Impact and Flexural Tests for Molded Plastics. Trans. Amer. Soc. Mech. Engrs. 68:547-556. 1946.

(8) Fischer, E. G. Lateral Vibration and Stress in a Beam under Shock Machine Loading. Proc. Soc. Exp. Stress Anal. 1-5:78-89. 1947.

(9) White, M. P. and Griffis, LeVan. The Permanent Strain in a Uniform Bar due to Longitudinal Impact. J. Appl. Mech. 14:337-343. 1947.

(10) Seely, F. B. Resistance of Materials. 2d. ed. pp. 266, 267, 281, 282, 285, 288, 289. Wiley and Sons. 1935.

(11) Timoshenko, S. Vibration Problems in Engineering. D. Van Nostrand. 1937.

ACKNOWLEDGEMENTS

The author wishes to thank Dr. Glenn Murphy of the Department of Theoretical and Applied Mechanics, who directed the work, for the guidance and encouragement that he gave. His willingness to help at any time aided the progress of the investigation immeasurably.

To Professor H. J. Gilkey, Head of the Department of Theoretical and Applied Mechanics, the author would like to express his thanks for permission to use the facilities of the department, and also for his aid in acquiring the material used in the investigation.

Professor R. T. Othmer of the Department of Theoretical and Applied Mechanics assisted in the experimental work and for the help the author wishes to extend his thanks.

The author also wishes to thank Dr. H. L. Mason of the Mechanical Engineering Department for his suggestions regarding the investigation and for the references to literature in the field of impact loading that he suggested.

The Engineering Experiment Station of Iowa State College loaned the control unit and oscillograph and also furnished the machinist who did the machining of the beam. The author wishes to express his appreciation for this assistance.