THE DAMPING CAPACITY OF CORRUGATED PAPERBOARD AS A FUNCTION OF STRESS AND FREQUENCY

Thesis for the Degree of Ph. D.

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James Winthrop Goff

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This is to certify that the

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Alvin J. Panshin
Major professor

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By

JAMES WINTHROP GOFF

AN ABSTRACT

Submitted to the School of Advanced Graduate Studies of Michigan State University of Agriculture and Applied Science in partial fulfillment of the requirements for the degree of

DOCTOR OF PHILOSOPHY

Department of Forest Products

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ABSTRACT

This study was undertaken to determine the damping capacity of corrugated paperboard and to ascertain the effect of stress amplitude, static load and frequency upon that property of the material.

Two methods, utilizing circular specimens similar to those used for the standard flat-crush tests of corrugated board, were developed. The first method, which produced load deflection diagrams for complete cycles at rates up to one-quarter cycle per second, made use of a Baldwin FGT-SR-4 universal testing machine equipped with a low range load cell and cycling controls. The ranges of loading used were 5 to 35 pounds, 35 to 65 pounds, and 5 to 65 pounds on a specimen which was two and one-half inches in diameter.

A special cyclic loading fixture was designed for use in the second method which involved loading in the same ranges, but at frequencies of five and seven and one-half cycles per second.

The material investigated was A-flute corrugated board, manufactured with two 42 pound kraft liners and a semi-chemical corrugating medium. All board used was supplied by the American Boxboard Company of Grand Rapids, Michigan. Specimens were cut, conditioned at 73 degrees Fahrenheit and 50 percent relative humidity for one week, selected for thickness, and samples of five were drawn at random. Each sample of five was tested at a single frequency and within a single loading range.

The total energy and energy dissipation per cycle were measured for each load deflection curve using a polar planimeter. The damping capacity and actual energy loss per cycle were calculated from these data.

Test results indicated that the damping capacity of the board under investigation lay in the range of 0.009 to 0.026 as determined at the 100th cycle of loading. The damping capacity was found to decrease with increased duration of cyclic loading.

The damping capacity and the actual energy loss per cycle appeared to be somewhat dependent on the frequency of vibration but no definite conclusions were drawn because of the lack of continuity of data in the overall frequency range studied.

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I. INTRODUCTION

The damping capacity of a material is defined as the ratio of the energy loss or dissipation per loading cycle to the total vibrational energy. Damping capacity is an important dynamic property of materials used for shock and vibration isolation in packaging and other areas. In packaging materials damping serves not only the function of reducing the amplitude of motion of a cushioned article at near-resonant frequencies, but is also important in that it may affect the magnitude of shock transmitted to the cushioned article as a result of instantaneous changes in velocity. For isolating near-resonant frequencies of vibration it would be desireable to have a maximum amount of damping, but from the standpoint of shock isolation too much damping can greatly increase the peak acceleration imposed on the article (12).

Corrugated paperboard in its several forms has long been used as a package cushioning material and it seems strange that only occasional mention is made of its damping characteristics (13).

This investigation was concerned with the development of methods for measuring and the measurement of the damping capacity of corrugated paperboard, with the variation in damping as a function of frequency, and with the variation in damping as a function of stress. Test frequencies studied were purposely selected to represent those possible of attainment on available cyclic testing equipment (one-eighth, one-sixth, and one-quarter cycle per second) and those which represent the continuous vibration spectrum imposed on packaged goods in rail-freight movement (five and seven and one-half cycles per second) (5). Three loading ranges were used as follows: five to sixty-five pounds, five to thirty-five pounds, and thirty-five to sixty-five pounds. The maximum stress in any range was well below that which would cause the first flat crush failure, consisting of buckling in the corrugating medium adjacent to the single-face liner, to occur. In general, no measurable permanent deflection of the board was anticipated, and the loads were confined to those which might reasonably be expected to occur in use under normal handling.

The material used for the investigation was A-flute board manufactured with two 42 pound kraft paperboard liners and an aspen semi-chemical corrugating medium. The board was fabricated by the American Boxboard Company in its Grand Rapids, Michigan plant.

II. PREVIOUS WORK

Methods of measuring damping. - The most direct method of measuring the damping capacity of a material involves the measurement of the total energy and the energy loss for a given cycle. This method has been described by von Heydekampf (18) who explained its use in connection with measurements of damping in metals at a maximum cycling rate of five cycles per minute. Bishop (2) treated the method mathematically for the hysteretic damping of an anchored spring but did not verify his treatment experimentally. No original report of actual damping measurements conducted by this method could be found. Apparently the difficulty in obtaining accurate load and deflection measurements at the rates of cycling involved had precluded the use of this procedure.

Damping capacity is approximately equal to twice the logarithmic decrement of decay of a damped free vibration when the logarthmic decrement is small, as shown by Zener (21). Kimball (7) describes the mathematical procedure involved in the determination of the logarithmic decrement, and Fusfeld (4), Pattison (14) and Smith (16) have developed rapid and accurate electronic instruments for determining this quantity.

Another method which has been widely used involves the measurement of the magnification factor Q which can be obtained from the band width of the frequency-amplitude curve between the half-resonant-amplitude values on either side of point of resonance. Zener (21), Wegel and Walther (19), Kruger and Rohloff (9), Horio and Onogi (6), and Seve and Perrin (15) have made use of this method with various materials.

A fourth method proposed and verified experimentally for a single material by Weissman, Pao, and Marin (20) involves the prediction of damping from static creep data for the material in question.

Damping measurements on wood and paper.- Kruger and Rohloff (9) using the method involving the measurement of the quantity Q found damping capacities for wood in bending parallel to the grain varying from 0.050 to 0.074 depending on species. Damping capacities perpendicular to the grain for the same group of species varied from 0.140 to 0.23. These investigators also found that damping capacity was dependent on the amplitude of vibration and independent of the frequency of vibration when measured in flexure between 10 and 700 cycles per second. Barducci and Pasqualini (1) have also investigated the damping capacity of wood and their results agree substantially with those of Kruger and Rohloff.

Horio and Onogi (6) have conducted damping measurements on wrapping paper using the resonance method involving the measurement of Q and report a value for the tangent of the mechanical loss angle of approximately 0.047. Zener (21) defines the mechanical loss angle as the angle by which deformation lags behind applied force in a periodic oscillation and Meredith (11) gives the proportionality factor between the tangent of this angle and the damping capacity as 6.28 or 2π, resulting in a damping capacity for the wrapping paper mentioned above of 0.30. These investigators also found that the damping capacity was essentially independent of the frequency of vibration.

III. EXPERIMENTAL PROCEDURE

Sampling and conditioning.- Specimens were cut from two sheets of board representing the same corrugator run and with identical liners and corrugating medium. The specimen cutting tool, which produced two and one-half inch diameter circular sections of board, was made according to specifications furnished by Hinde and Dauch, Inc. of Sandusky, Ohio.

The specimens were coded so that the location of the sample in the board from which it was cut could be later determined if necessary for proper analysis of the test results. A series of four numbers was used: (1) the first digit represented the board sampled, (2) the second digit represented the a length of sixteen inches in the direction of the flutes, (3) the third number, one through eleven, represented a length of five inches in the machine direction, (4) the fourth digit represented the specimen number within each five by sixteen inch piece of board. The extensive coding system was thought desirable in order to reduce the board to sections small enough to handle in the actual specimen cutting operation.

The specimens were conditioned for at least one week following the cutting at seventy-three degrees

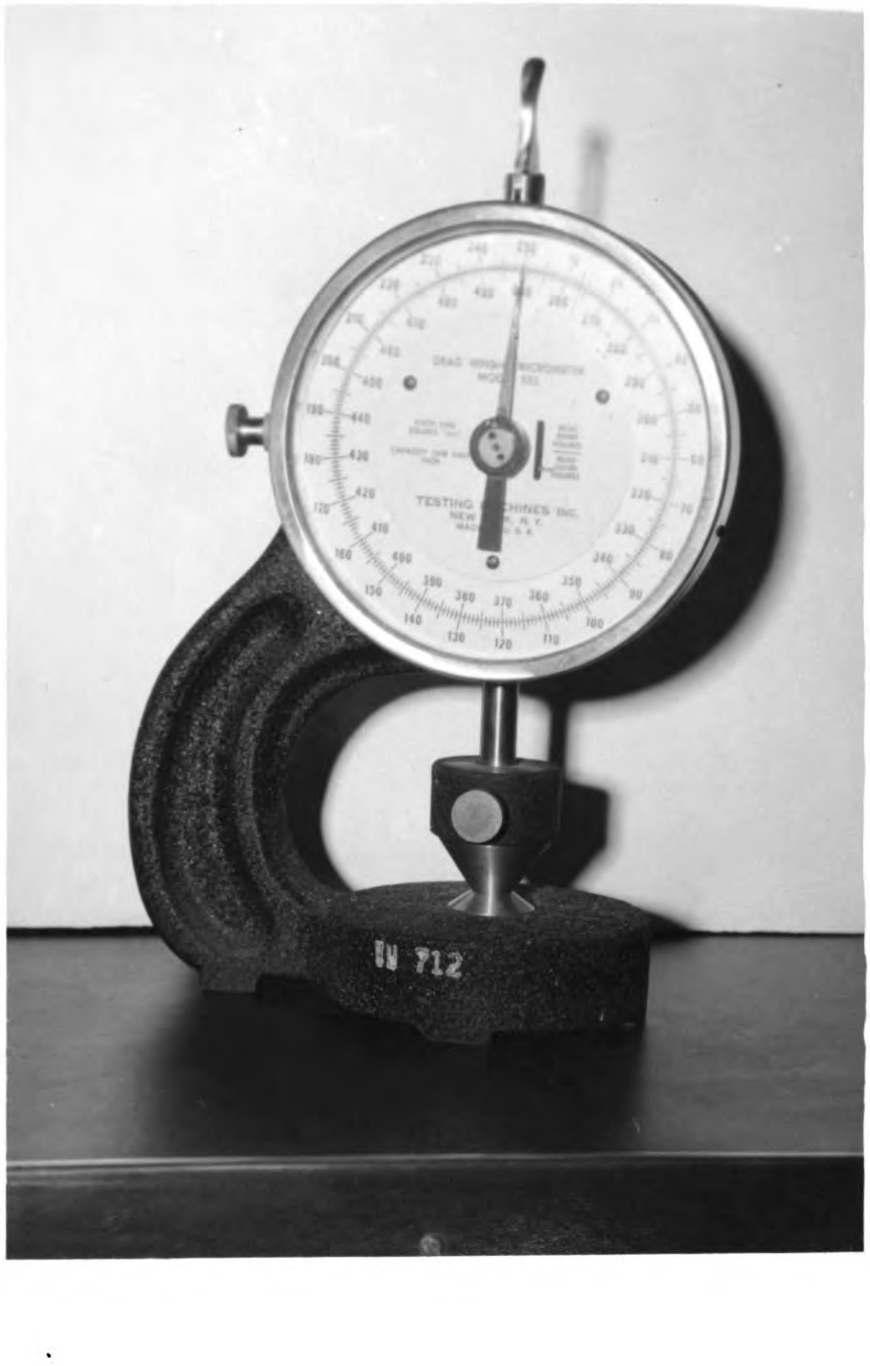
Fahrenheit and fifty per-cent relative humidity. Selected samples from the group were subsequently oven-dried and found to be at an average moisture content of eight and two-tenths per-cent on the basis of the dry weight of the board.

Moisture content of the specimens was maintained throughout the testing program within one-tenth of one per-cent through the use of a moisture vapor proof bag and a moisture barrier surrounding the platens for protecting the specimens en-route to and in the testing machine. A number of preliminary tests conducted during a time when the atmosphere surrounding the machine was extremely dry indicated that this procedure was highly effective in maintaining a uniform moisture content in the material under test.

Following the conditioning period all specimens were checked for overall caliper (thickness) by means of a dead-weight board micrometer as shown in Figure 1. Random samples of five specimens each were selected from those measuring 0.217 and 0.218 inches. These two thickness groups were the most highly populated in the thickness distribution and the mean thickness lay in the interval between the two values. A total of twenty such samples were drawn for the subsequent tests.

<u>Testing procedures</u>.- Each specimen was tested individually in a manner similar to that employed for the

Fig. 1.- Dead weight board micrometer.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standard flat crush test of corrugated board. Specially designed loading platens were used to restrict lateral movement of the liners and thus reduce the possibility of leaning-flute failures inherent in the test method. A drawing of one of the platens will be found in appendix A. Two different test procedures were developed and used as required to gain the desired information.

Test method A.- This method involved the use of a Baldwin, 50,000 pound capacity, FGT-SR-4, universal testing machine equipped with an auxiliary 500 pound load cell, load cycling controls, and a differential transformer type autographic recorder for load and deflection. The load was applied through the regular machine drive and controlled by the cycling system within the desired limits. The one-hundred pound load range provided by the load cell was used on all tests.

Deflection of the specimen was measured by means of a Baldwin model PD-1M deflectometer modified to provide a 400 to 1 magnification when used with the recorder set on its high range. Details of the modification are shown in appendix A. Figure 2 shows the machine as used for the tests, while Figure 3 provides a close-up view of the arrangement of apparatus on the machine platen.

The switching and acceleration limitations of the load mechanism and cycling controls as well as the

Fig. 2. Baldwin, 50,000 pound, FGT-SR-4, universal testing machine equipped with an auxiliary 500 pound load cell, load cycling controls, and a differential transformer type autographic recorder as used in test method A.

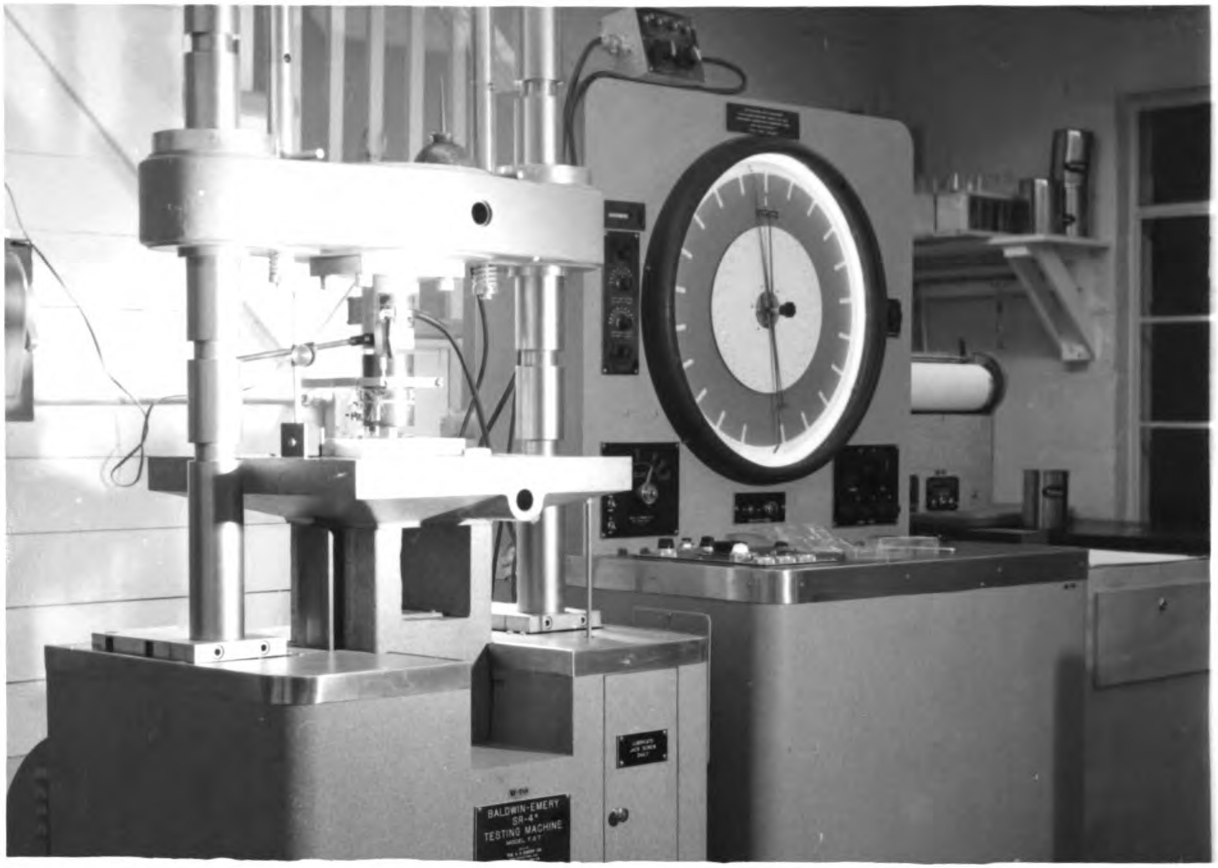
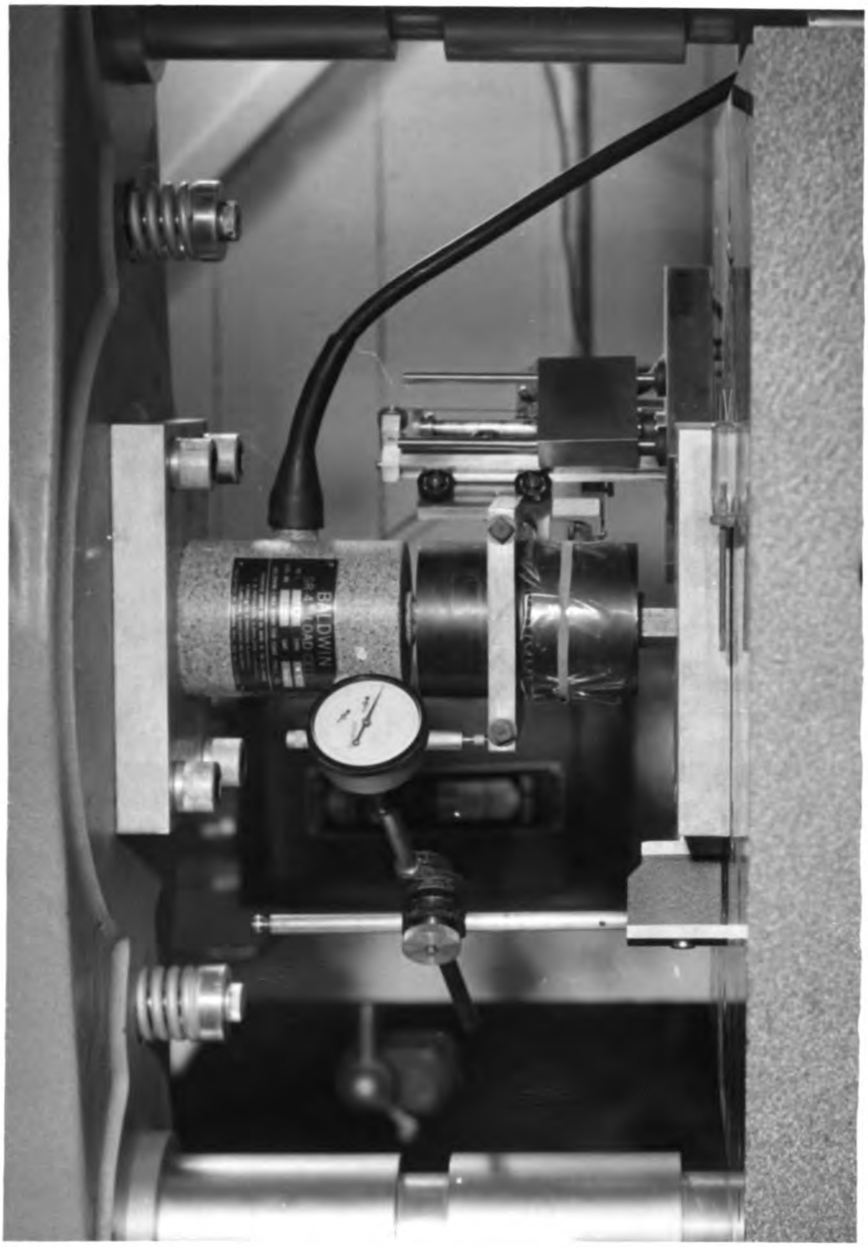

BALDWIN-EMERY
SR-4
TESTING MACHINE

Fig. 3.- Arrangement of apparatus on testing machine platen for test method A.


BALDWIN

response limitations of both the load and deflection recording systems restricted the test frequencies available with this test procedure to a maximum of one-quarter cycle per second. In the case of the maximum loading range used the greatest frequency obtainable was one-eighth cycle per second.

Test method B.- The frequency limitations of the first method made it necessary to seek another way of loading the specimens and of recording their behavior at higher frequencies. This second procedure utilized the Baldwin testing machine as a loading frame, as a load indicator, and as a variable speed drive for the loading system. The same platens were used as was the 500 pound load cell.

The lower platen was attached to a reciprocating mechanism rigidly constrained to move in a vertical direction only. This mechanism provided an adjustable stroke through the use of a four-jaw lathe chuck which held the crank pin driving the platen support. The loading system is shown in Figure 4 and is more completely described in Appendix A.

An oscillographic recording system was used in this series of tests to provide the necessary response characteristics. The output of the load cell was connected to a Tectronix Type 122 Low Level Preamplifier through a phase shifting device to the horizontal axis

Fig. 4. Loading mechanism, load cell, and deflectometer arrangement used in test method B.

of a DuMont Type 304A cathode-ray oscillograph. Deflection was measured by a specially built cantilever beam fitted with resistance strain gages. The output of this deflection sensing beam was connected to an Ellis Type BA-2 Bridge-Amplifier and then to the vertical axis of the oscillograph. A DuMont Type 302 Oscillographic Record Camera provided a means of recording the patterns appearing on the face of the instrument.

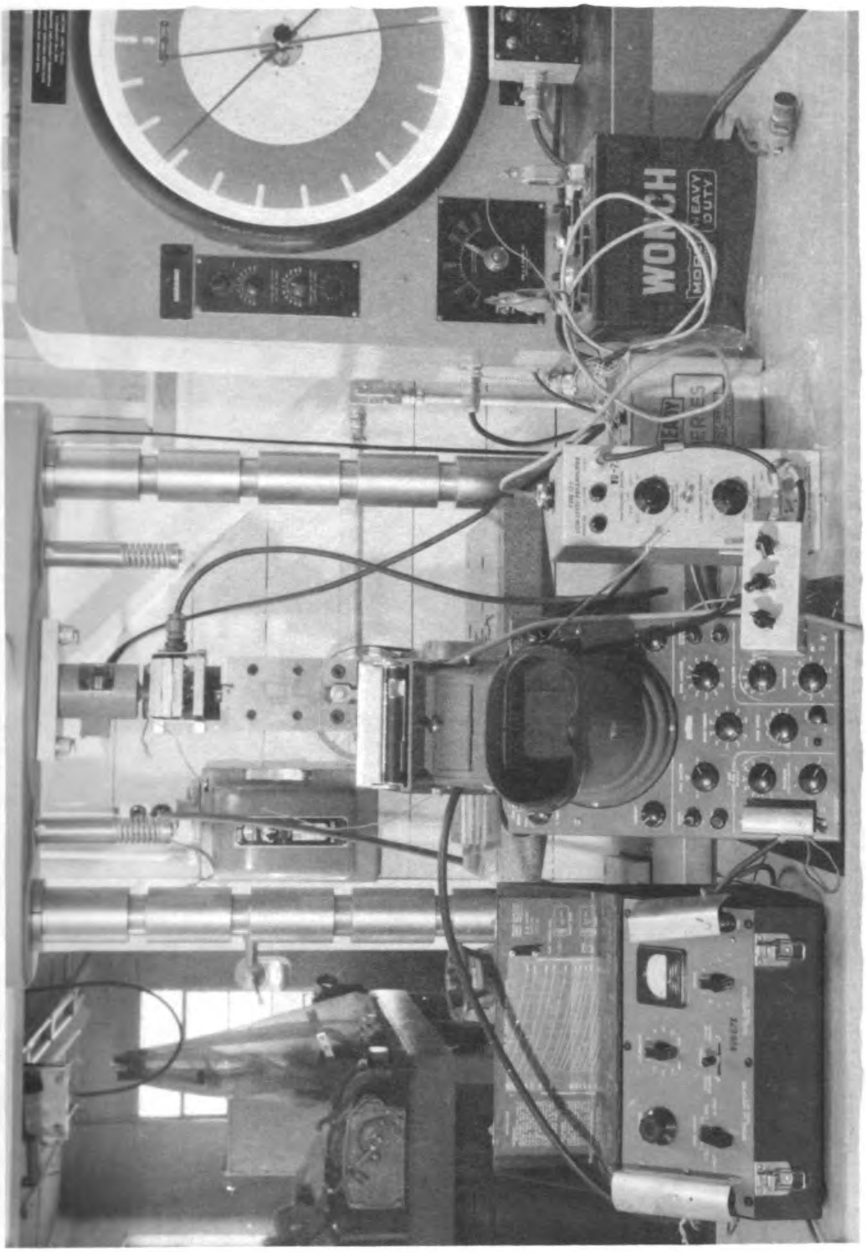
The phase shifting device mentioned in the preceding paragraph was designed to overcome the difficulty caused by the variable phase shift inherent in both of the external amplifiers used. In using the phase shifter, the signal from the load cell at the desired test frequency was applied to the input of both amplifiers through the use of a specially designed switching box. Since the internal amplifiers provided in the oscillograph were of the direct coupled variety, the resulting pattern on the oscillograph screen represented the phase difference due to the external amplifiers. This phase difference was adjusted to zero by the shifting circuit. The load signal was again connected to the horizontal channel only, and the deflection signal was restored to the vertical axis. The system was then ready to use.

The test frequencies used in this procedure were too high to determine by the simple expedient of timing a single cycle. For this reason the frequencies were adjusted by the use of an electronic counter. The output of the load cell as amplified by the preamplifier was connected to the input of a Hewlett-Packard Model 521A Electronic Counter equipped with a crystal controlled time base. The counter was manually gated for periods of ten seconds while the motor speed control was adjusted to give the desired frequency. The speed dial settings were noted and recorded for use in the subsequent tests.

All instrumentation is shown in Figure 5 and is further described in detail in Appendix A.

Operational Details.- The actual operating technique for both test procedures and a description of the difficulties and shortcomings of the apparatus used are given in Appendix B.

Fig. 5. Instrumentation used in method B.



IV. THEORETICAL ANALYSIS OF THE PROBLEM

A theoretical consideration of the question of damping in corrugated board can best be approached

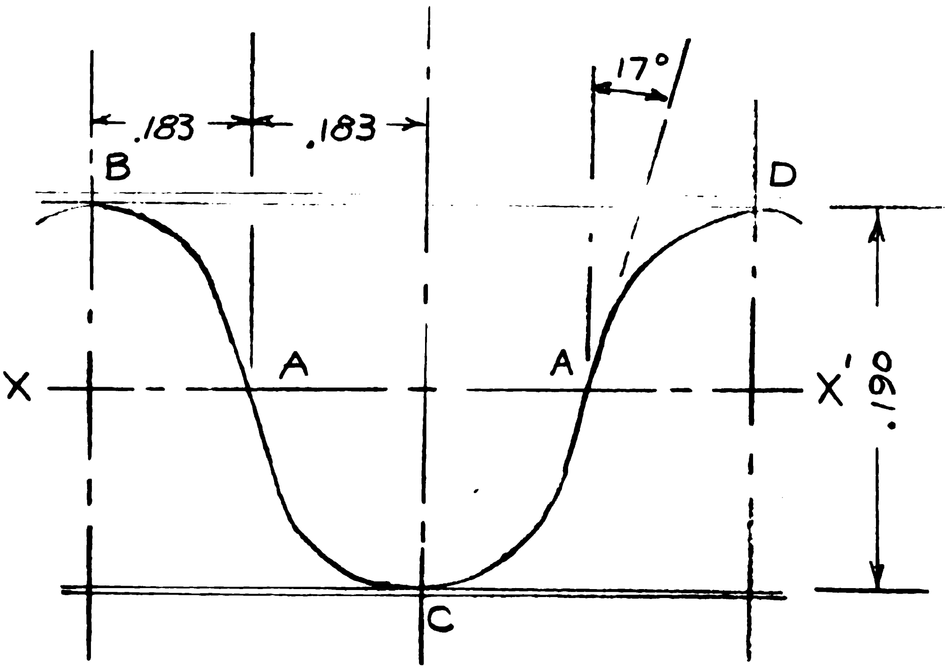
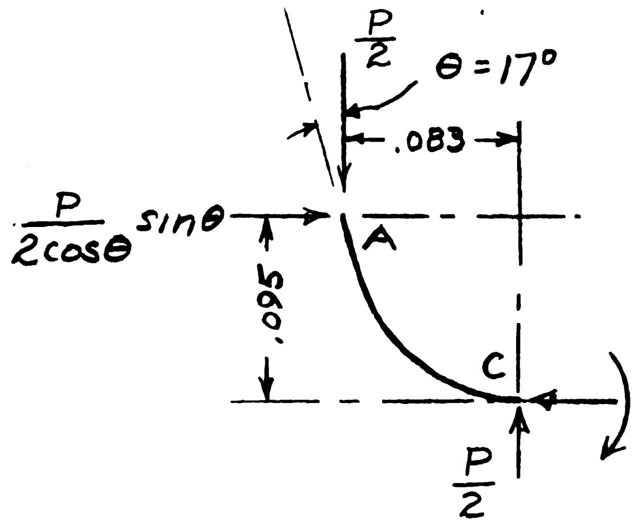


Fig. 6. Schematic cross-section, A-flute corrugated board.

by an analysis of the behavior of the material when it is subjected to a compressive load. Figure 6 above is a line representation of the cross-section of the A-flute board used in this investigation. The dimensions shown are based on the assumption of 36 flutes per foot, a corrugator flute depth of 0.181 inches and a corrugating medium thickness of 0.009 inches.

The deflection of such a material under load will cause either a sidewise collapse of the flutes in a horizontal direction or a bending of the individual flutes adjacent to the liners to a sharper radius. The former cannot occur because each flute tip is fixed to the board liner by means of an adhesive. As a result deflection of the board is primarily a function of the bending of the corrugating medium immediately adjacent to its point of adhesion to the liners. This bending can be better analyzed by taking a single quadrant of the flute from one of the points A on the mid-axis of the board to point C where the flute is rigidly bonded to the liner. A consideration of the moments acting about point A resulting from a load P applied to the faces of the section of board in Figure 6 will bring one to a realization that the bending moment in the corrugating medium at this point is zero. As a result the loads acting on a quadrant are as shown in Figure 7 below.

Fig. 7. Free body diagram of a flute quadrant.



The corrugating medium, adhered to the liner at point 6, can be considered as fixed at this point.

Initially, prior to any appreciable deflection of the board, the bending moment at point C is given by the expression

$$(1) \quad M = (0.083)\frac{P}{2} - (0.095)\frac{P}{2\cos\theta}\sin\theta$$

As deflection proceeds, the quadrant, fixed at point C and restrained from any horizontal movement at point A by the geometry of the situation, rotates about point A. This action results in a change in the angle θ and the bending moment at point C becomes

$$(2) \quad M = (0.083)\frac{P}{2} - (0.095)\frac{P}{2\cos(\theta-\Delta\theta)}\sin(\theta-\Delta\theta)$$

resulting in an increase in the bending moment at point C. Rotation of the medium about point A also effects a reduction in the direct compressive stress exerted on the portion of the quadrant which becomes more vertical and an increase in the direct stress on the portion which approaches the horizontal. Similarly, the rotation affects the shear stresses acting in the web.

From the above analysis it can be seen that small deflections of corrugated board sections bring into play rather complex combinations of direct stress, bending stresses, and shear. The damping inherent in the material, by definition related to the angle by which the strain in the material differs from the stress applied, can be expected to be a highly complex function. It would seem, however, that the strains caused by the bending would be of major importance and for this reason the damping capacity should approach that of paper materials in flexure.

V. ANALYSIS OF TEST DATA

Areas representing the total energy and energy loss per cycle were taken by means of a polar planimeter from each of the load deflection curves in test method A. The photographs of the load deflection curves obtained in test method B were photographically enlarged and the total energy and energy loss data were obtained in a similar way. The damping capacity was computed directly from the areas involved in each case while the actual energy loss was calculated from the hysteresis loop area using a scale factor determined from the known loading range and known deflection of the specimen.

Damping capacities and the actual energy losses during the 100th loading cycle are shown in Tables I through V for both methods of testing and for all frequencies and loading ranges. The actual energy losses shown are in inch-pounds per two and one-half inch diameter specimen.

Comparison of actual energy loss and damping capacity with the loading range shows that both are directly dependent on the amplitude of motion at all frequencies.

TABLE I

TEST DATA - METHOD A - 1/8 CPS - 100TH CYCLE

Loading Range	Specimen	Energy Loss Inch-Pounds per spec.	Damping Capacity	Mean Energy Loss Inch-Pounds per spec.	Mean Damping Capacity
5-35	463	.0067	.19		
	321	.0070	.21		
	391	.0065	.22		
	361	.0072	.18		
	3115	.0067	.20	.0068	.20
35-65	454	.0037	.14		
	4102	.0038	.14		
	295	.0030	.12		
	492	.0028	.11		
	385	.0027	.12	.0032	.13
5-65	2433	.0197	.17		
	2475	.0225	.15		
	2293	.0315	.21		
	2254	.0623	.26		
	2234	.0125	.11	.0297	.18

TABLE II

TEST DATA - METHOD A - 1/6 CPS - 100TH CYCLE

Loading Range	Specimen	Energy Loss Inch-Pounds per spec.	Damping Capacity	Mean Energy Loss Inch-Pounds per spec.	Mean Damping Capacity
5-35	375	.0052	.16		
	275	.0065	.18		
	441	.0052	.15		
	424	.0045	.15		
	485	.0043	.14	.0051	.16
35-65	255	.0038	.14		
	423	.0033	.15		
	253	.0043	.15		
	443	.0045	.16		
	225	.0038	.14	.0039	.15

TABLE III

TEST DATA - METHOD A - 1/4 CPS - 100TH CYCLE

Loading Range	Specimen	Energy Loss Inch-Pounds per spec.	Damping Capacity	Mean Energy Loss Inch-Pounds per spec.	Mean Damping Capacity
5-35	2345	.0033	.12		
	2295	.0058	.15		
	2354	.0050	.15		
	2365	.0053	.15		
	2373	.0047	.14	.0048	.14
35-65	283	.0040	.14		
	284	.0033	.11		
	234	.0030	.14		
	4113	.0035	.11		
	481	.0023	.09	.0032	.12

TABLE IV

TEST DATA - METHOD B - 5 CPS - 100TH CYCLE

Loading Range	Specimen	Energy Loss Inch-Pounds per spec.	Damping Capacity	Mean Energy Loss Inch-Pounds per spec.	Mean Damping Capacity
5-35	431	.012	.17		
	474	.012	.16		
	333	.013	.16		
	311	.014	.20		
	493	.015	.19		
				.013	.18
35-65	3114	.009	.15		
	362	.011	.16		
	421	.011	.16		
	461	.010	.15		
	lost	----	---		
				.010	.15
5-65	324	.036	.20		
	3112	.039	.21		
	462	.039	.21		
	413	.038	.20		
	332	.036	.19		
				.038	.20

TABLE V

TEST DATA - METHOD B - 7.5 CPS - 100TH CYCLE

Loading Range	Specimen	Energy Loss Inch-Pounds per spec.	Damping Capacity	Mean Energy Loss Inch-Pounds per spec.	Mean Damping Capacity
5-35	2374	.006	.17		
	2311	.007	.18		
	2221	.007	.16		
	2454	.006	.16		
	2274	.007	.17	.007	.17
35-65	2115	.007	.13		
	2445	.007	.13		
	22103	.008	.13		
	2362	.006	.14		
	2341	.007	.13	.007	.13
5-65	3103	.035	.19		
	451	.030	.17		
	363	.034	.19		
	392	.039	.19		
	394	.037	.20	.035	.19

The effect of the static load level is shown by a comparison of the damping capacities and the actual energy losses in the 5 to 35 pound range with those in the 35 to 65 pound loading range. In all except one case the actual energy loss decreases with an increase in static load, and in every case the damping capacity decreases with an increase in static load.

Both damping capacity and energy loss per cycle appear to be somewhat dependent on the frequency of the dynamic load, although the lack of information on these quantities in the frequency range $\frac{1}{4}$ to 5 cycles per second precludes any statement regarding this dependence.

Table VI is a compilation of Tables I through III for the 100th cycle with the addition of similar data for the 10th cycle of loading with the same specimen. The damping capacity and the energy loss per cycle are shown to be inversely related to the number of cycles of loading to which they are subjected. Table VII shows an identical but less pronounced trend for loading at 5 and 7.5 cycles per second.

Figure 8 shows, left to right, load deflection curves at 1,800, 3,600, 5,400, and 10,800 cycles of loading for a single specimen with a loading range of 5 to 75 pounds and a deflection of 0.0088 inches at a

TABLE VI

MEAN ENERGY LOSS AND MEAN DAMPING CAPACITY FOR SELECTED CYCLES

	Cycle	1/8 CPS			1/6 CPS		1/4 CPS	
		5-35 Range	35-65 Range	5-65 Range	5-35 Range	35-65 Range	5-35 Range	35-65 Range
Mean Energy Loss Inch-Pounds per specimen	10	.0079	.0056	.037[a]	.0065	.0051	.0075	.0043
	100	.0068	.0032	.0297	.0051	.0039	.0048	.0032
Mean Damping Capacity	10	.22	.22	.28[a]	.19	.20	.22	.15
	100	.20	.13	.18	.16	.15	.14	.12

[a]These two values represent the second cycle of loading rather than the tenth cycle.

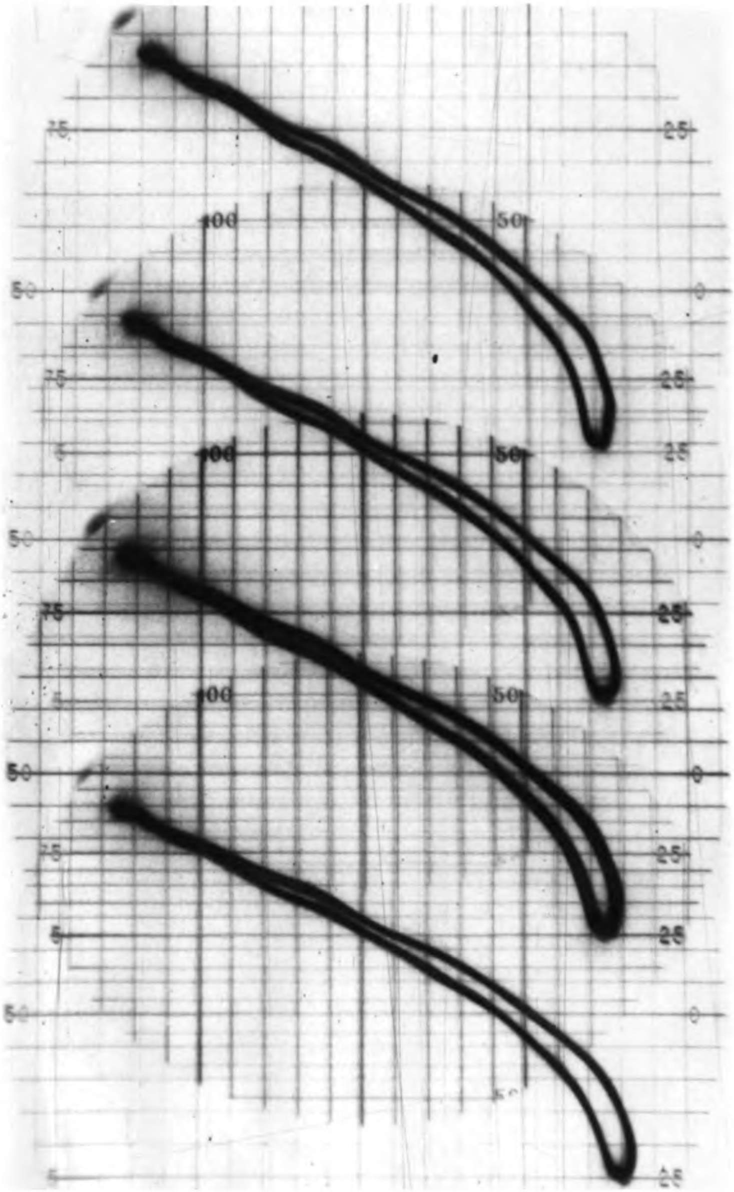
TABLE VII

EFFECT OF EXTENDED NUMBER OF LOADING CYCLES AT 7.5 CPS AND 5-65 POUND RANGE

Specimen	Property	75 th Cycle	4,500 th Cycle
2315	Energy Loss in.lb/spec.	.0440	.0379
	Damping Capacity	.26	.22
23102	Energy Loss in.lb/spec.	.0396	.0363
	Damping Capacity	.21	.19
2414	Energy Loss in.lb/spec.	.0408	.0408
	Damping Capacity	.20	.21
2225	Energy Loss in.lb/spec.	.0408	.0354
	Damping Capacity	.22	.20
2344	Energy Loss in.lb/spec.	.0374[a]	.0354
	Damping Capacity	.19[a]	.19

[a]These values represent the 1,500th cycle. Camera difficulties prevented the photographing of the 75th and 750th cycles.

Fig. 8. Load deflection curves for a single specimen with a load range of 5 to 75 pounds and a deflection of 0.0088 inches at a frequency of 7.5 CPS. Duration of loading, left to right, 1,800, 3,600, 5,400, and 10,800 cycles.



75
25
100
50
50
0
75
25
5
100
50
25
50
0
75
25
5
100
50
25
50
0
75
25
5
25
50
0
5
25

frequency of 7.5 cycles per second. Little difference can be discerned in energy loss per cycle between the first and the last trace, but careful examination of the coordinates will show that the area under the loading curve is increasing with each successive trace resulting in a decrease in damping capacity.

Direct examination of the individual photographs and of the testing machine records revealed the presence of creep during the dynamic tests and pointed up a basic difference between the two test methods. Since the cycling controls on the testing machine are controlled by the motion of the load indicator, load was the controlled function. The machine maintained the load between the prescribed limits regardless of deformation of the specimen. This resulted in a very slow continuous motion of the strain recorder, amounting to as much as 0.0005 inch actual deflection during the course of a 100 cycle run.

Test method B, on the other hand, made use of a loading device in which the deformation of the specimen was fixed by means of an eccentric and a constant deformation was applied to the specimen regardless of the load. The resulting photographs for some specimens show a slight relaxation of the load during the course of a 500 cycle run of as much as two to three pounds.

VI. CONCLUSIONS

It can be concluded that damping in corrugated paperboard is dependent on stress amplitude and is inversely dependent on the level of static stress. The exact nature of the dependence was not determinable on the basis of the experimental data.

No statement regarding the relationship between damping and the frequency of the disturbing force seems warranted in view of the discontinuity of the data in terms of the frequency spectrum and in view of the fact that different methods were used to obtain the data in the two frequency ranges studied.

Values of damping capacity were found to range between 0.13 and 0.20. These results compare favorably with the value of 0.30 as found by Horio and Onogi (6) for wrapping paper, and with those for wood as found by Kruger and Rohloff (9) when it is considered that the fibers in paper are oriented in a much more random manner than those in wood.

The investigation shows that for the range of frequencies studied the damping capacity tends to decrease with an increase in duration of cyclic loading, particularly at the lower frequencies. To the extent that the degradation may not be reversible with removal of the load the reuse of corrugated board and containers would seem to merit further consideration.

Practical use of the damping capacity values as determined in this investigation can be made in the design of container systems. Crede (3) and Mindlin (12) have treated this area extensively.

LIST OF REFERENCES

1. Barducci, J. and G. Pasqualini. Measurements of internal friction and elastic constants of wood. Nuovo Cimento, 5:416, 1945.

2. Bishop, R. E. D. The treatment of damping forces in vibration theory. Jour. Royal Aero. Soc. 59 (11): 783-742, Nov. 1955.

3. Crede, C. E. Vibration and Shock Isolation. John Wiley and Sons, New York, N.Y., 1951.

4. Fusfeld, H. T. Rapid measurement of internal friction. Review of Scientific Instruments 21: 612, 1950.

5. Guins, S. G. and J. A. Kell. Vibrations in railroad freight cars. A. S. T. M. Bulletin (173) :46, April 1951.

6. Horio M. and S. Onogi. Dynamic measurement of physical properties of pulp and paper by audio frequency sound. Journal of Applied Mechanics 22:971, 1951.

7. Kimball, A. L. Vibration problems, part IV, friction and damping in vibration. Journal of Applied Mechanics 8:A-37, 1941.

8. ______. Vibration problems, part V, friction and damping in vibration. Journal of Applied Mechanics 8:A-135, 1941.

9. Kruger, F. and E. Rohloff. Uber die innere Reibung von Holz. Zeitschrift Physik 110:58, 1938.

10. Lazan, B. J., J. Brown, A. Gansett, P. Kirmer, and J. Klumpp. Dynamic testing of materials and structures with a new resonance vibration exciter and controller. Proceedings A. S. T. M. 52: 858-876, 1952.

11. Meredith, R. (ed.) Mechanical Properties of Wood and Paper. Interscience Publishers, Inc., New York, N.Y., 1953.

12. Mindlin, R. D. Dynamics of package cushioning. Bell System Technical Journal 24:353-461, 1945.

13. Mustin, G. S. An introduction to shock and vibration for package designers. Proceedings of the Joint Industry Conference on Cushioning in Packing, Wayne University, 1953.

14. Pattison, J. R. An apparatus for the accurate measurement of internal friction. Review of Scientific Instruments 25:490, 1954.

15. Seve R. and H. Perrin. Mesure dynamique du module d'elastique des papiers. A. T. I. P. Bulletin No. 1, 1954.

16. Smith, A. D. N. An electronic instrument for measuring the damping capacity of materials. Jour. Scientific Instruments 28:106, 1951.

17. Strella, S. Vibrating reed tests for plastics. A. S. T. M. Bulletin (214):47-50, 1956.

18. von Heydekampf, G. S. Damping capacity of materials Proceedings A. S. T. M. 31(2):157, 1930.

19. Wegel, R. and H. Walther. Internal dissipation in solids for small cyclic strains. Physics 6:141, 1935.

20. Weissman, G. F., V. H. Pao, and J. Marin. Prediction of creep under fluctuating stress and damping from creep under constant stress. Proceedings Second U. S. National Congress of Applied Mechanics, June 1954.

21. Zener, C. Elasticity and Anelasticity of Metals. University of Chicago Press, Chicago, Illinois, 1948.

APPENDIX A. NOTES ON EQUIPMENT AND INSTRUMENTATION

Loading platens.- The specimens were loaded by means of identical upper and lower platens as detailed in Figure 9. After installation on the testing machine the platens were checked by means of feeler stock for parallelism and shimmed as necessary.

Test method A.- The only special equipment feature necessary in this test method was the modification of the Baldwin PD-1M deflectometer to provide a magnification of deformation of 400 from specimen to recorder. Without such modification the hysteresis loops were too small for area measurements of any degree of accuracy. The change made in the instrument consisted of providing an additional pivot bearing on the crank actuating the differential transformer and an extension bar to transfer the bearing point on the movable head to this new location. Figure 10 shows the instrument with the modifying extension bar attached while Figure 11 is a view with the bar removed. The modification in no way affects the normal operation of the deflectometer.

Test method B.- This method required the design and construction of a special loading device, a resistance wire strain gage deflectometer, a phase shifting

Fig. 9. Loading platen.

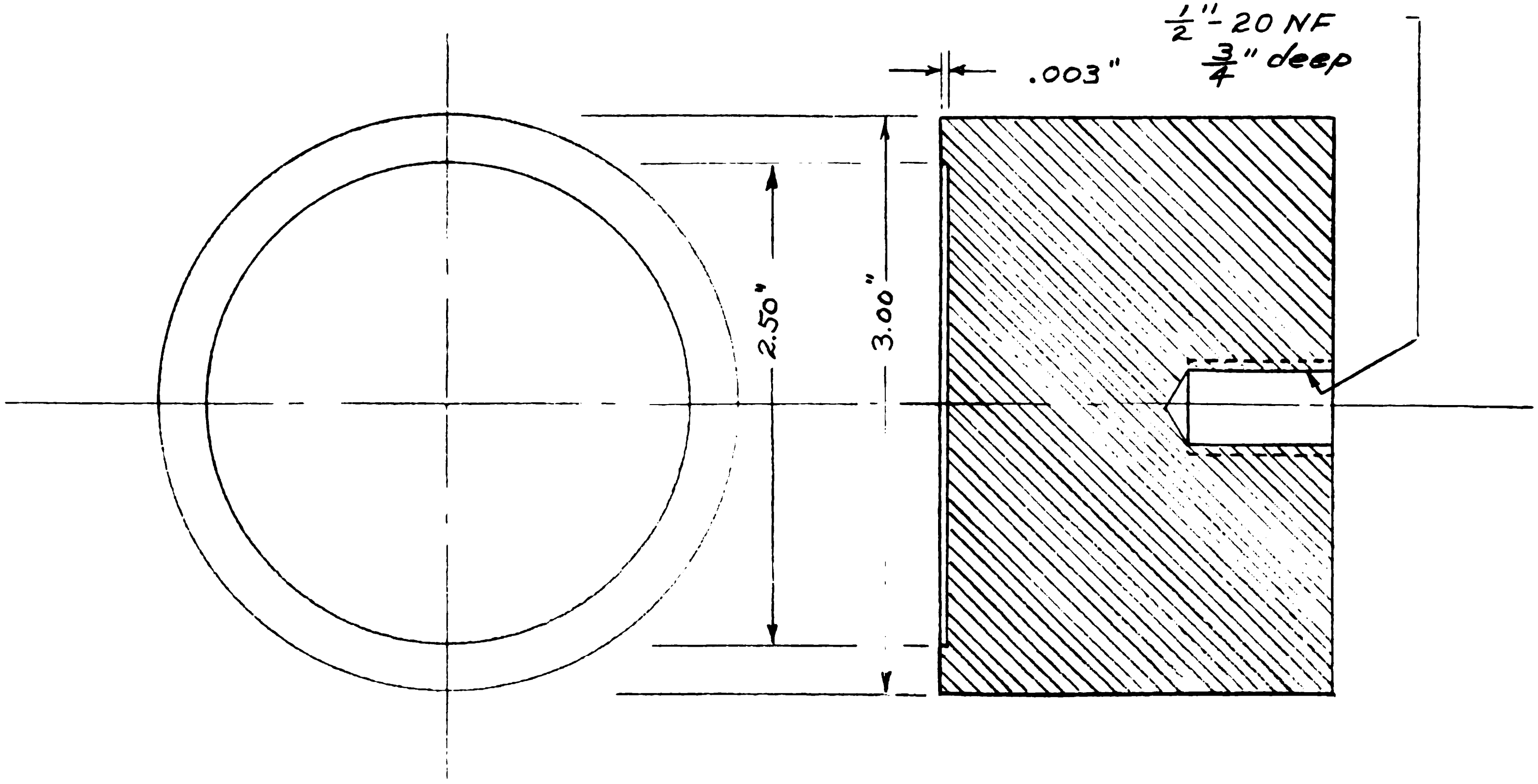


Fig. 10. Baldwin Model PD-1M deflectometer as modified for use in test method A.

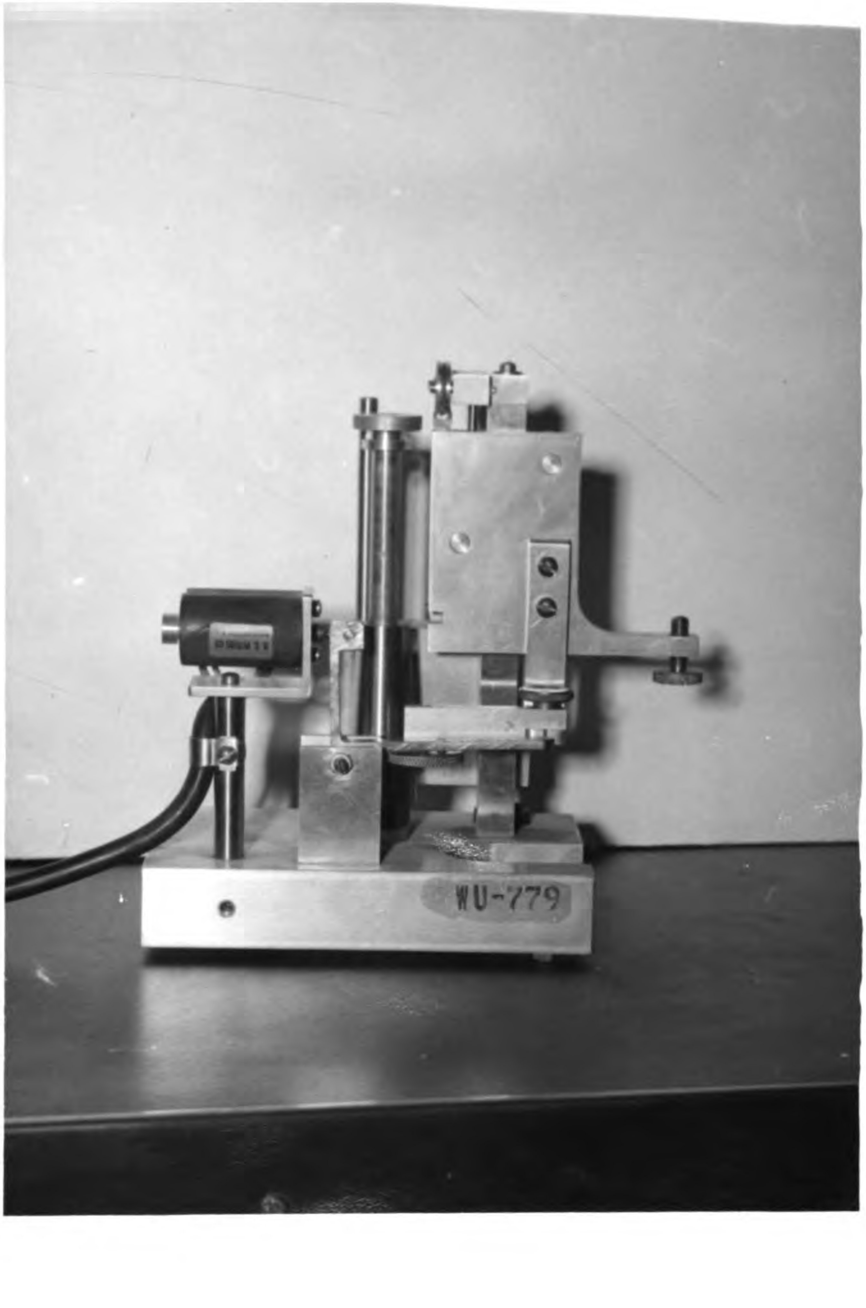
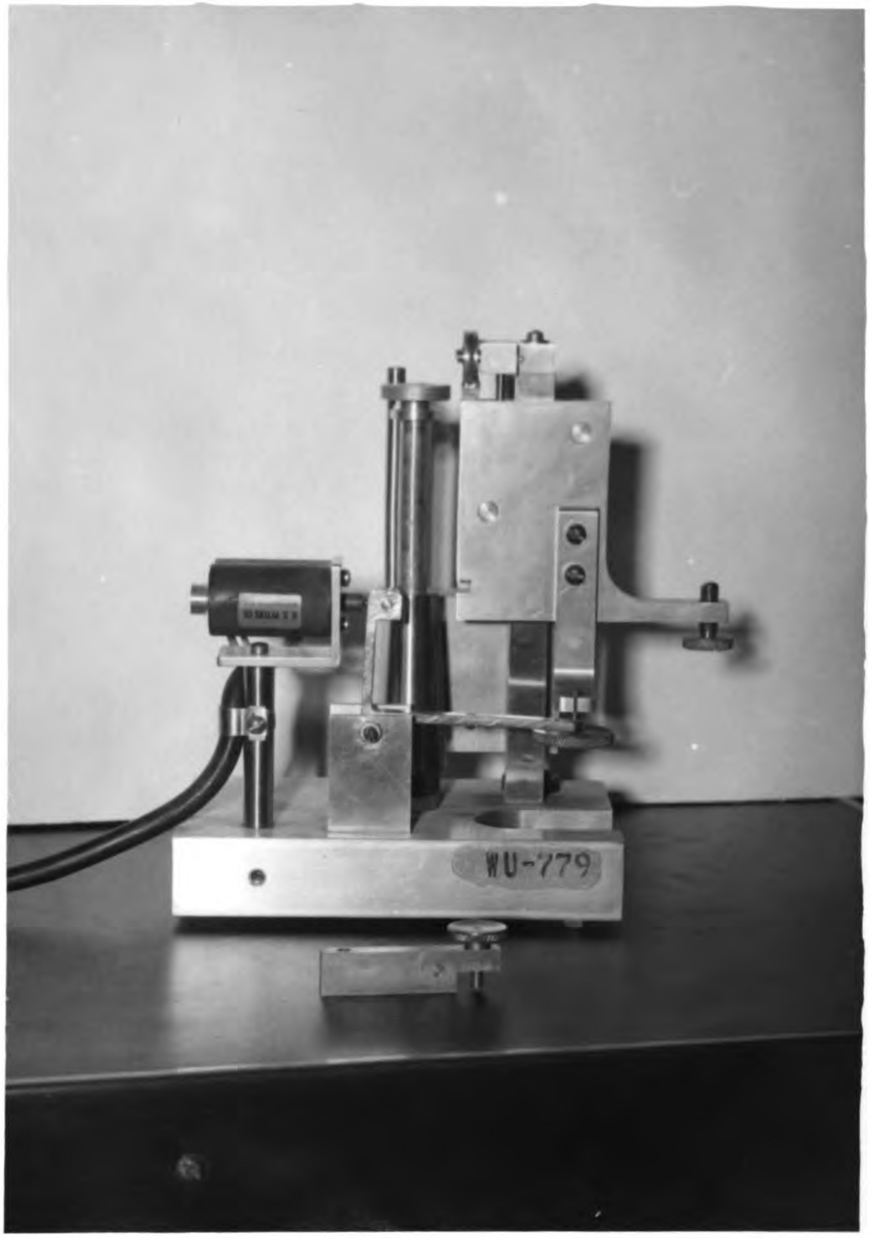

WU-779

Fig. 11. Deflectometer showing the modifying parts detached.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circuit and a switching system for making quick checks on phase relationships.

The loading device was described in the body of the thesis and is shown in Figures 4 and 12. Adjustment of the lathe chuck which controlled the eccentricity of the crank pin was accomplished with the aid of a dial indicator graduated in ten-thousandths of an inch.

The resistance wire strain gage deflectometer is shown in Figures 4 and 13. A load deflection curve for the instrument alone at the magnification used in the actual measurements is shown in Figure 14. The latter also indicates the absence of measureable hysteresis loss in both the load and deflection measuring systems.

The purpose of the phase shifting device was described previously. A schematic circuit diagram for the phase shifter is shown in Figure 15 along with the switching system designed to facilitate the use of the device.

Oscillograph traces for both in-phase and out-of-phase signals are shown in Figure 16.

A block diagram of the instrumentation used in test method B is shown in Figure 17. Dashed lines indicate the temporary connections made to the frequency counter for determining the speed controlled setting to provide the frequency desired and to the testing machine indicator for determining load range.

Fig. 12. Loading device for test method B showing V-belt drive from the variable speed motor on the Baldwin testing machine.

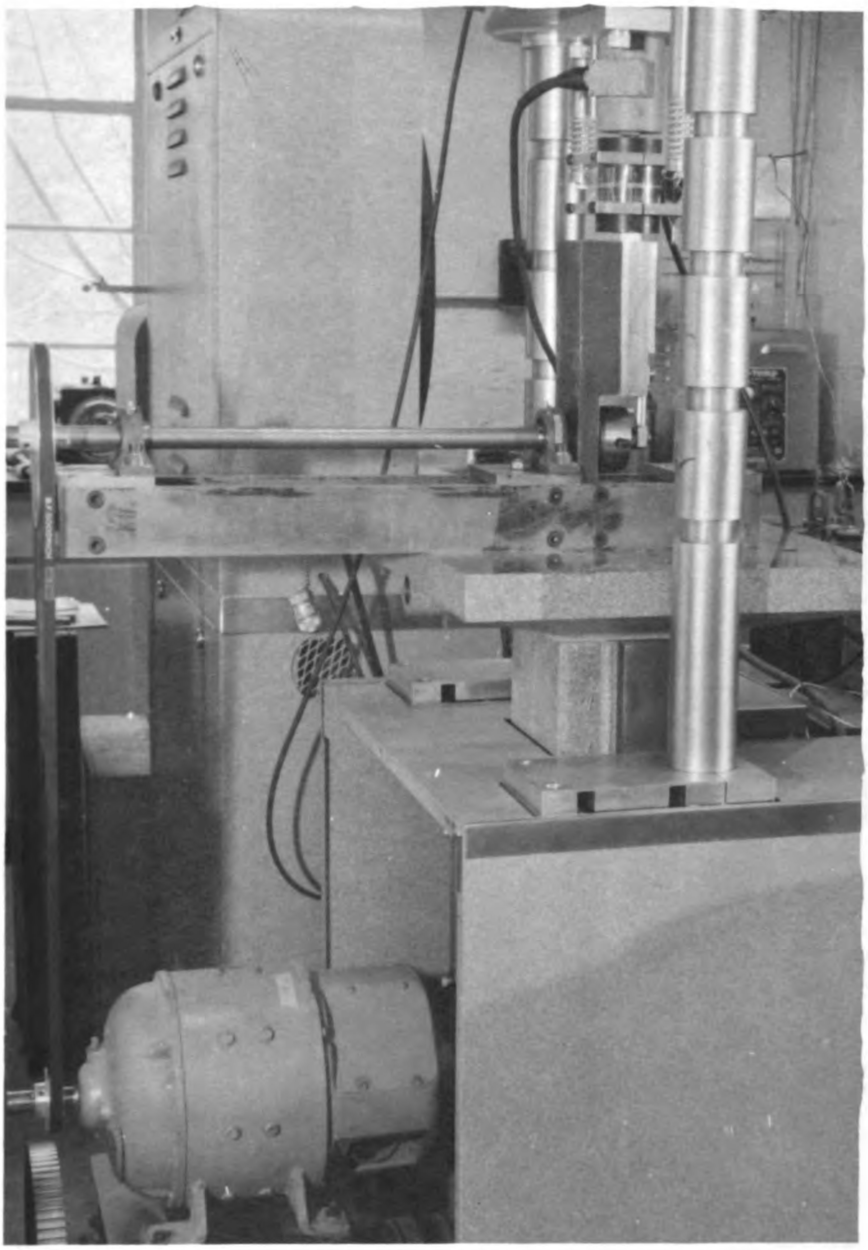


Fig. 13. Resistance wire strain gage deflectometer.

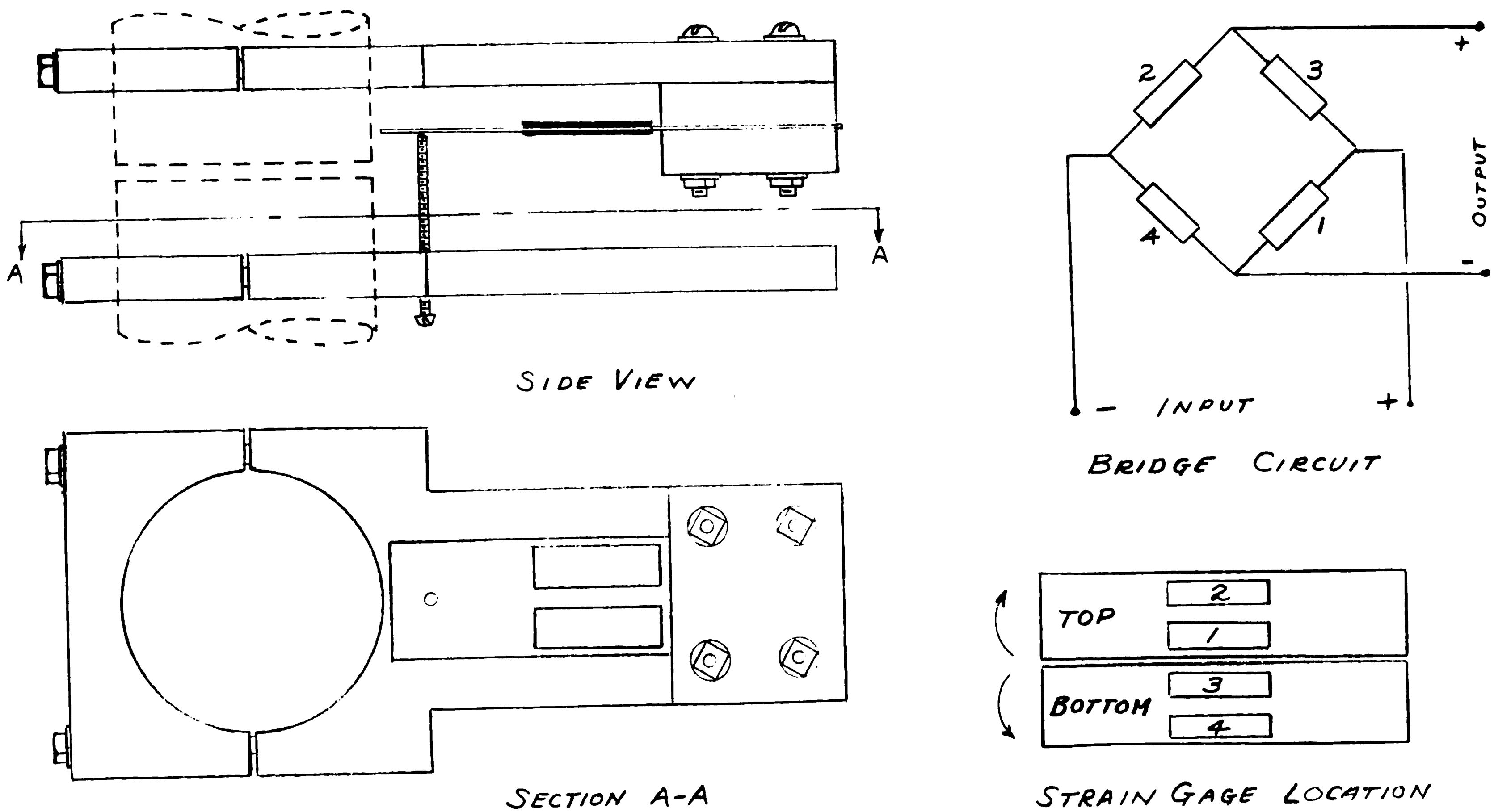


Fig. 14. Load deflection curves for the method B deflectometer without a specimen.

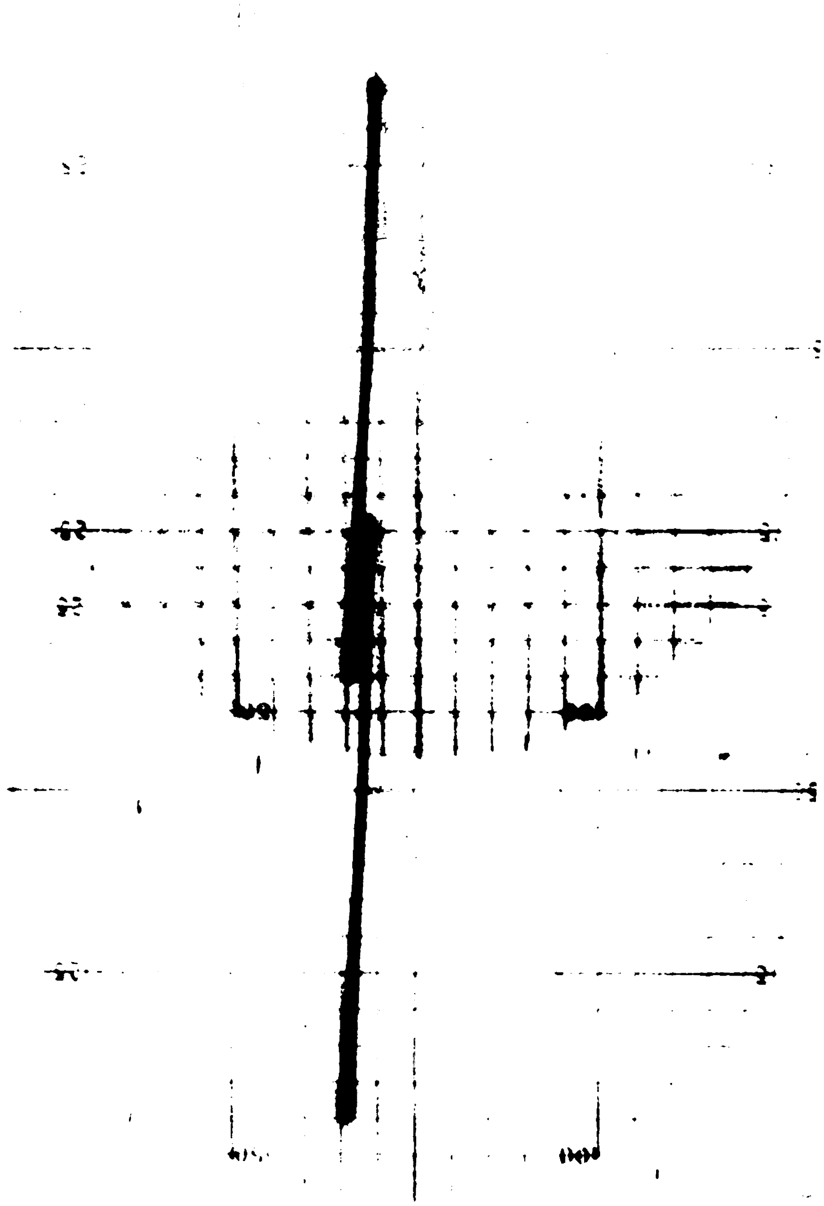
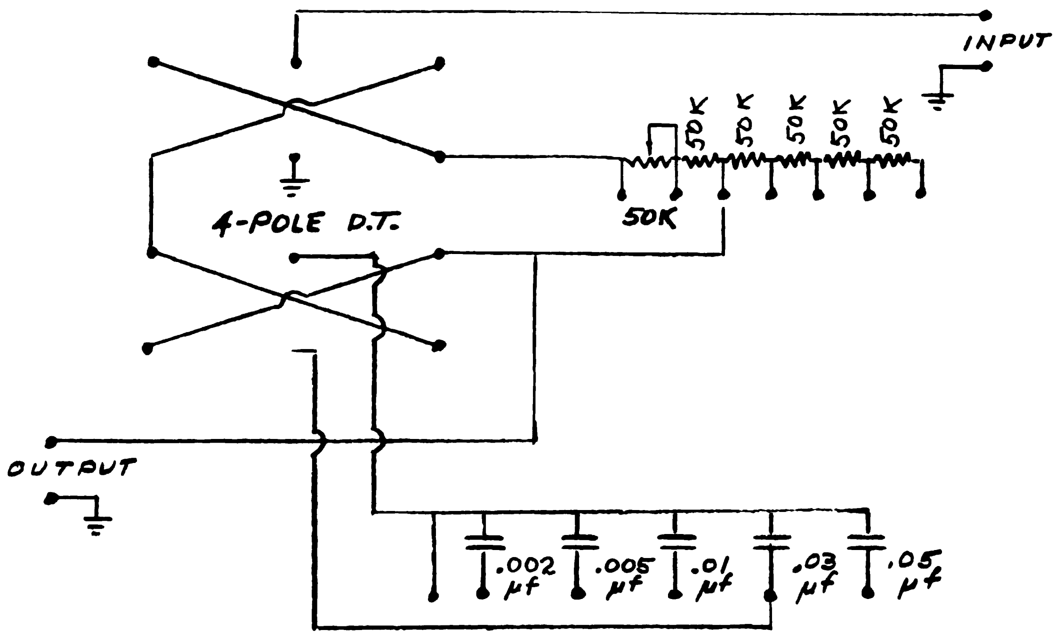
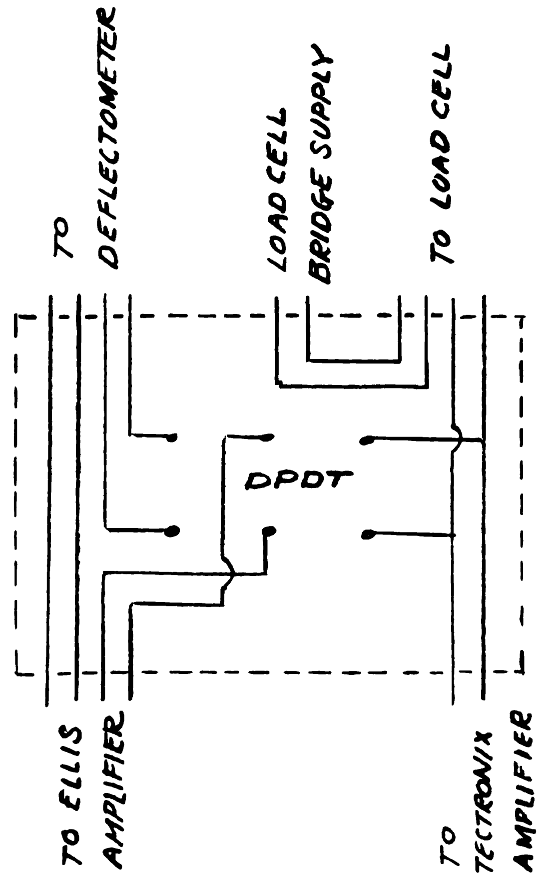


Fig. 15. Schematic diagrams for phase shifter and switching system.



PHASE SHIFTER



SWITCHING SYSTEM

Fig. 15. Schematic diagrams for phase shifter and switching system.

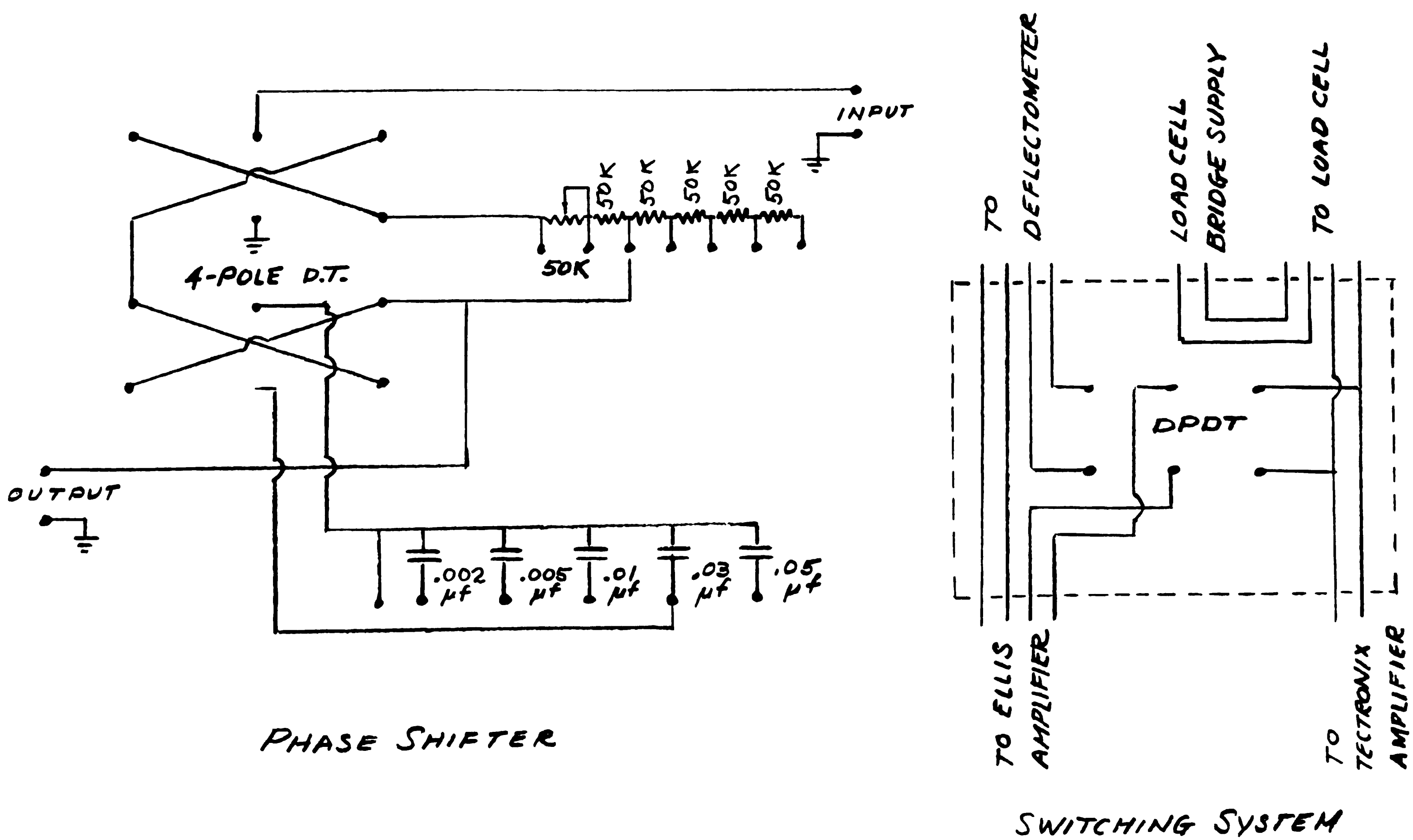


Fig. 16. Load deflection curves and corresponding phase shift traces.

Left to right:

1. Load deflection curve, relative phase shift adjusted to zero.
2. Load deflection curve, phase shift between amplifiers shown in 3.
3. Trace showing the phase difference for load signal through both amplifiers.
4. Trace showing the phase difference adjusted to zero. (the photograph is actually a negative which accounts for the negative slope)

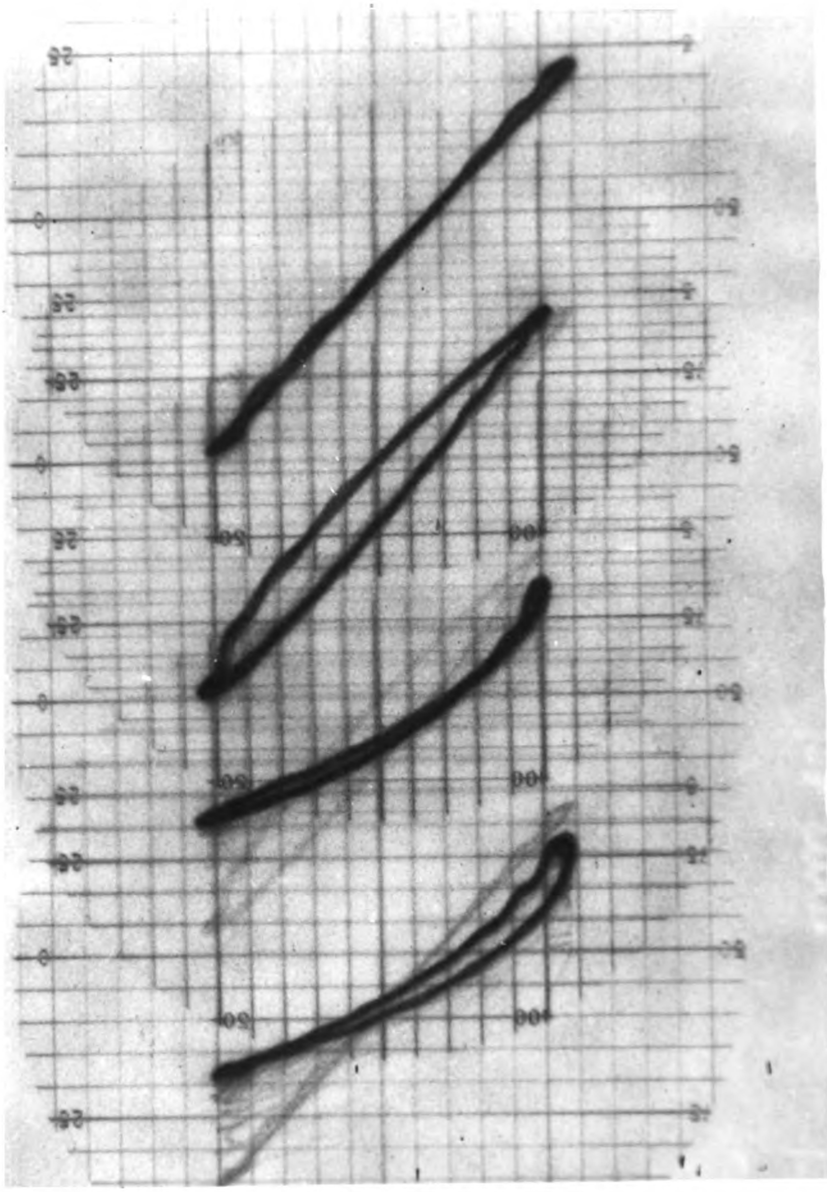
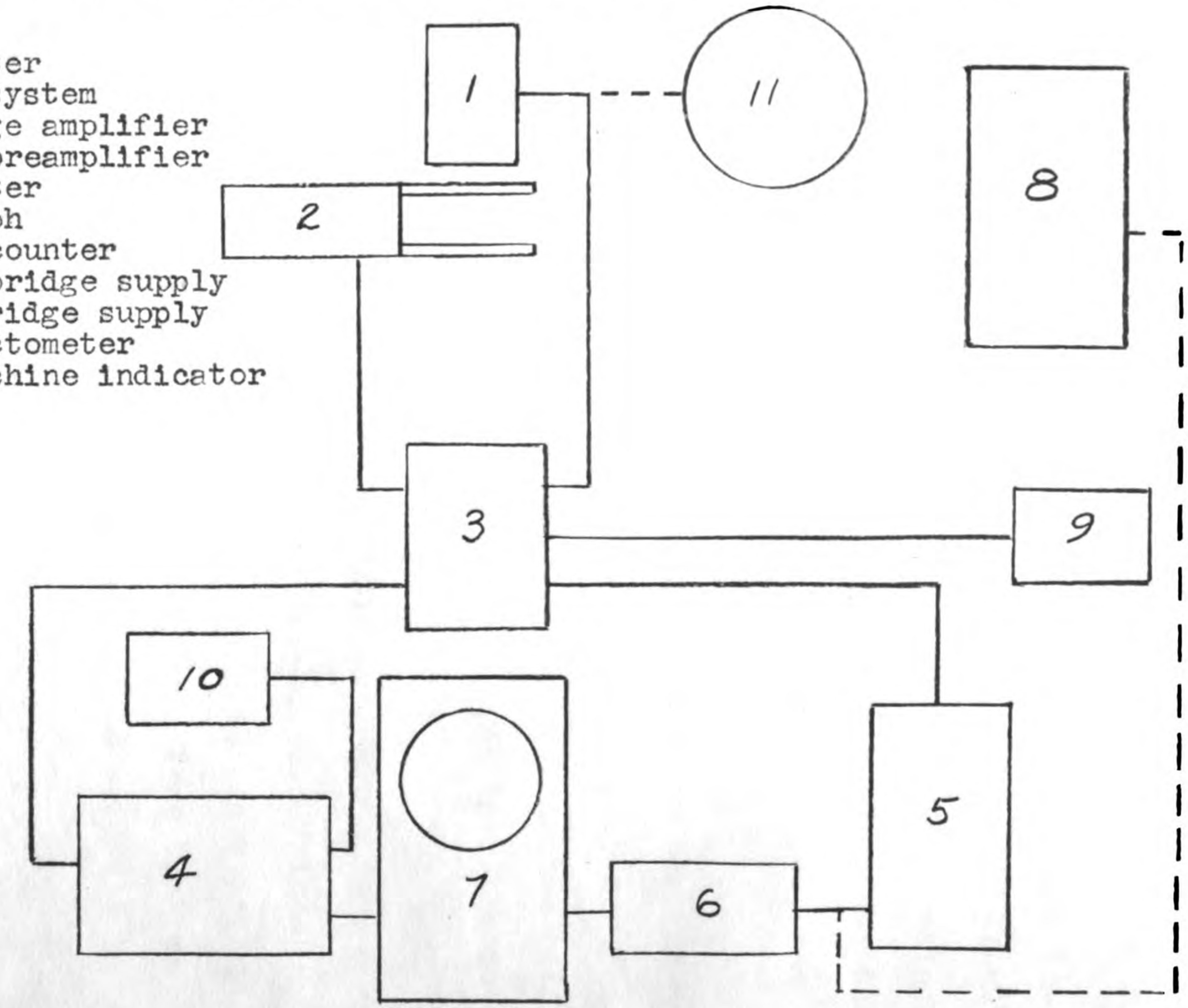


Fig. 17. Block diagram of instrumentation, method B.



1. Load cell
2. Deflectometer
3. Switching system
4. Ellis bridge amplifier
5. Tectronix preamplifier
6. Phase shifter
7. Oscillograph
8. Frequency counter
9. Load cell bridge supply
10. External bridge supply for deflectometer
11. Testing machine indicator

APPENDIX B. OPERATIONAL TECHNIQUES

Test method A.

1. Set up testing machine with the 500 pound load cell and the special loading platens. Adjust the platens for parallelism using shim stock under the mounting plates. Set the platen motion stops. Turn on power to the machine, the indicator and the recorder. Keep the recorder motor turned off.

2. Set the indicator range to 0-100 pounds, adjust the indicator to zero, set the recorder to zero load and fill the pen. Engage the recorder in the half scale position making certain that the pen does not move when this is done.

3. Connect the Baldwin PD-1M deflectometer and adjust it for the range of motion expected. Switch on the recorder motor and adjust the drum to its starting position at the proper platen position. (Keep the recorder motor turned off except when it is actually in use.)

4. Insert a trial specimen (not one of the sample specimens but of the same caliper) between the platens with the single-face liner up and the corrugations perpendicular to the front of the machine. Wrap a

narrow strip of saran film around the platens.

5. Start the machine using a slow rate of platen motion. Set the machine to "cycle" and adjust the cycling limit controls and the speed control simultaneously until the desired frequency of motion is obtained within the loading range wanted. (The speed and limit control settings were determined once for each range and frequency. This means that the frequency may vary slightly depending on the stiffness of the specimen particularly in the 5-35 pound loading range.) Record the limit control and speed control settings, stop the machine and remove the trial specimen.

6. Insert a sample specimen in the same manner as the trial specimen, set the cycle counter to zero, run the machine up to the low end of the load range, and set the machine to "cycle". (It must be taken out of the cycle position in order to stop it at the end of step five.) If either the speed controller or the limit controls have been moved be sure to return them to their proper settings. Turn on the recorder motor and set the recorder cylinder to its starting position by hand. Lower the pen and record the 10th cycle. Raise the pen and advance the drum by hand. Record the 25th, the 50th, and the 100th cycles in the same manner, stop the machine, remove the specimen and return it to the conditioning room.

7. Check the final caliper of the specimen and record it on the specimen if it differs from the initial thickness.

8. Repeat the above procedure for each specimen in the sample.

9. Measure the total area under each curve and the area within the hysteresis loop using a polar planimeter. Record the areas directly on the testing machine charts.

Test method B.

1. Adjust the upper testing machine platen to accomodate the special loading fixture, the load cell, and the two platens. Bolt the loading fixture to the lower platen and raise the platen to approximately the proper level. Remove the timing belt which drives the transmission, replace the timing sheave on the motor shaft with a three inch V pulley, and attach the V belt. (To assure the safety of the fixture, the loading limit controls should be adjusted at this time so that both of them are on the compression side. This will permit the motor to run only in the proper direction so that the chuck will not turn on the shaft and be unscrewed.)

2. Attach the resistance wire strain gage deflectometer to the platens and adjust the screw which contacts the sensitive beam.

3. Turn on the power to the machine, the indicator, and all instrumentation.

4. Adjust the eccentricity of the crank pin to the desired value using a dial indicator. Insert a trial specimen as in method A and raise the platen by means of the hand wheel on the transmission drive until the proper load is indicated on the testing machine load indicator. Check the loading range and readjust the eccentricity if necessary.

5. Disconnect the load cell from its span control box and connect the cell cable to the switching system. Connect the output of the Tectronix amplifier to the frequency counter and set the motor speed to give the desired frequency. The speed control settings for each frequency to be used should be recorded at this time.

6. Stop the machine, disconnect the frequency counter, and connect the Tectronix amplifier to the phase shifter. Using the switching system, connect the load cell output to both axes of the oscillograph through the external amplifiers. Start the machine and adjust the phase shift to zero for the frequency to be used. Remember that this adjustment must be made every time the frequency is changed.

7. Restore the deflection signal to the X axis by means of the switching system and check the size

and position of the signal on the screen. Adjust the signal trace, if necessary, and stop the machine. (Turn the brightness control on the oscilloscope down when the machine is off.) Remove the trial specimen.

8. Insert a sample specimen in the platens as before and connect the load cell output to its span box so that the loads can be read on the machine indicator. Adjust the load to the lower load limit by means of the hand wheel and check the loading range by rotating the shaft slowly by hand. Connect the load cell cable to the switching system again.

9. Check the setting of the speed control, start the machine, and record the 100th and the 500th cycles using the Polaroid Land camera. (A stopwatch was used to determine the 100th and 500th cycle since the frequency counter produced considerable noise in the oscilloscope trace.)

10. Enlarge the record photographs using the Land camera prints as paper negatives. Trace the enlarged curves on vellum paper and measure the areas as in test method A.