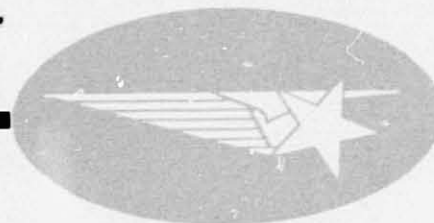


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NONLINEAR DYNAMIC ANALYSIS
OF STRUCTURES

Final Report

Volume II
STRUCTURAL JOINT DAMPING

June 1970

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by
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FOREWORD

This report presents results of work performed by Lockheed's Huntsville Research & Engineering Center for the Aero-Astronautics Laboratory of NASA-Marshall Space Flight Center under Contract NAS8-21435, "Nonlinear Dynamic Analysis of Structures." This work represents the Phase II effort entitled "Structural Joint Damping." Printed under separate cover is the Phase I effort, "Nonlinear Damping in Structures," LMSC/HREC D149171-I, dated July 1970.

The NASA-MSFC technical monitor for this contract is Mr. Larry A. Kiefling, S&E-AERO-DDS.

ACKNOWLEDGEMENT

The author is grateful to G. D. Robinson of Lockheed/Huntsville who contributed significantly to all experimental tasks of the project. He designed the test fixture, conducted tests and prepared Section 5 of the report.

SUMMARY

Results of detailed studies of simple joint damping are presented. In particular, the dissipation between two elastic plates pressed together by mechanical fasteners and experiencing relative dynamic deformations is discussed.

Experimental studies are employed to develop a working engineering theory of the contact damping problem.

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Section 1 INTRODUCTION

The principal type of damping in low-frequency vibrations of large, complicated structures has been identified to be joint damping (see, for example, Ref. 1). Associated with this recognition is also the acknowledged fact that joint damping is a highly complex phenomenon which has defied theoretical analyses for sometime. In a limited way, the objective of the present project is to take an initial step toward developing a workable engineering theory for analysis and prediction of joint damping. An ideal program to study joint damping would encompass the following tasks: (1) a laboratory investigation of the joint damping phenomenon in a simple configuration. Parameters which exhibit influence over damping are to be isolated; (2) development of an engineering theory based on physical insight thus obtained for the simple joint; (3) carrying out control experiments to establish the range of applicability of the developed engineering theory; (4) generalization of the theory of joint damping to more complicated structural joint configurations; and (5) final experimental verification of the approach on structures.

The investigations conducted during Phase II of the present project touched upon tasks (1) through (4) of the above list in various depths. Some of the results, as well as the techniques used to obtain them are considered new and significant to the long-range goal of understanding joint damping. No claim of completeness, however, is made. Indeed, the above outlined ideal program was not completely formulated until late in the current project.

In Section 2 of this report, joint damping is discussed in general to provide a background for subsequent discussions. In the order presented in the remainder of this document, the following studies are reported:

1. A parametric study of the area of contact between two elastic plates due to the clamping of a machine bolt.
2. A parametric study of plate deflections at remote locations with respect to the clamping bolt.
3. An investigation of damping of clamped plates in torsional vibrations.
4. A visual observation of the interaction of contact areas in a multiple-bolt lap joint.
5. Determination of the contribution to total damping of a single bolt in multiple-bolt joint specimens.

Section 2

GENERAL CONSIDERATIONS

On several occasions during the study of damping in Saturn launch vehicle models (full-scale dynamic test structures and reduced-scale models) it had been noted that the modal damping coefficients in many bending vibration modes vary with the modal response amplitude (Refs. 1 through 4). Furthermore, the measured damping values for such structures were usually greater (by an order of magnitude or more) than what can be attributed to other sources such as material damping and air damping combined (Ref. 1). Joints are, therefore, likely to be the major contributor of damping in complicated structures such as the Saturn launch vehicles. In a recent effort (Ref. 5), structural joint damping was measured for some of the typical joints of the Skylab orbiting cluster. Total modal damping coefficients were calculated, using the measured joint damping data and material damping properties available from the literature. It was shown in Ref. 5 that the latter contributes a negligible amount of damping in all vibration modes investigated for the structure. The need for more intensive investigation of joint damping was, once again, established.

The following is an order-of-magnitude type of estimate of joint damping which is included to reveal the (surprisingly large) magnitude of damping capacity of a structural joint. Consider two 1/8-in. plates clamped together by a steel bolt of 1/8-in. diameter, as shown in Fig. 1. The tensile strength of such a bolt is approximately 1000 lb. Yet, a 1/8-in. bolt can easily snap during structural assembly and, therefore, is not commonly used in practice. In other words, a single bolt clamping force of 1000 lb is not uncommon in actual structural joints. If it is assumed tentatively that uniform sliding is to occur across the two contacting surfaces in a sinusoidal motion with an amplitude of 10^{-3} in., the total energy dissipation per cycle may be computed by the expression

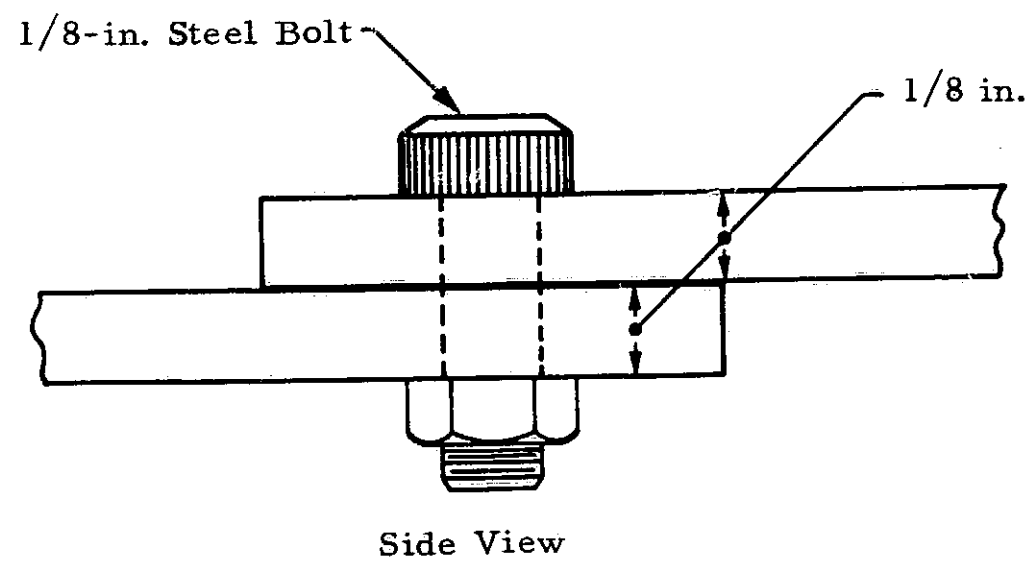
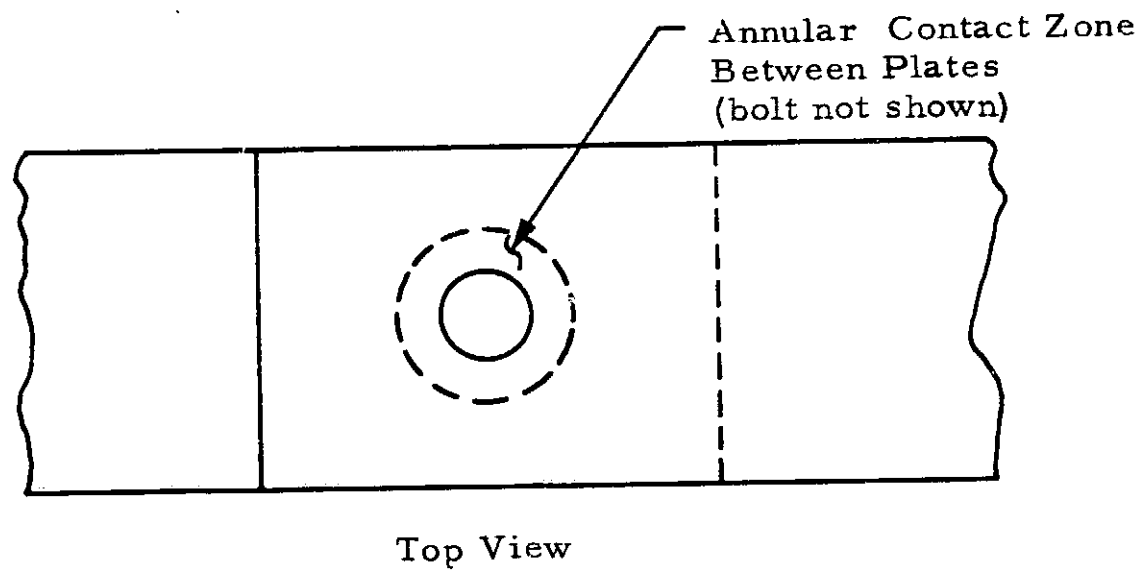


Fig. 1 - Single-Bolt Lap Joint

$$D = \mu \pi P x$$

where μ is the coefficient of friction and is assigned the value of 0.3, P is the total clamping force of 1000 lb, and x is the displacement amplitude of 10^{-3} in. The quantity D is, therefore, approximately 1.0 in.-lb/cycle. That this rate of energy dissipation is appreciable may be demonstrated with the following example.

Consider the aluminum beam of Fig. 2. A lap joint is used to cantilever the left end of the beam. The calculated dynamic characteristics of the rigidly cantilevered beam in the first mode are

$$\begin{aligned} \text{Natural frequency,} &= 62.5 \text{ rad/sec} \\ \text{Generalized mass*}, m &= 8.12 \times 10^{-5} \text{ lb-sec}^2/\text{in.} \end{aligned}$$

For a modal damping, ζ , of 1% of critical, the energy dissipation per cycle per inch of tip deflection amplitude is

$$D = 2\pi\zeta\omega^2 m \approx 2 \times 10^{-2} \text{ in.-lb/cycle}$$

Divided equally among the bolts, the energy dissipation at each of the four contact areas is only 5×10^{-3} in.-lb/cycle, about one two-hundredth of the 1.0 in.-lb/cycle value calculated previously. It is obvious, therefore, that the contacting plate surfaces do not totally slide with respect to each other under these conditions, and that the assumed 10^{-3} in. of sliding amplitude is too great. In short, joint damping for conventional structures can be appreciable without having gross sliding of one part with respect to another, and with (partial) relative slipping being of extremely small amplitudes.

*With the mode shape normalized to unity at the free end.

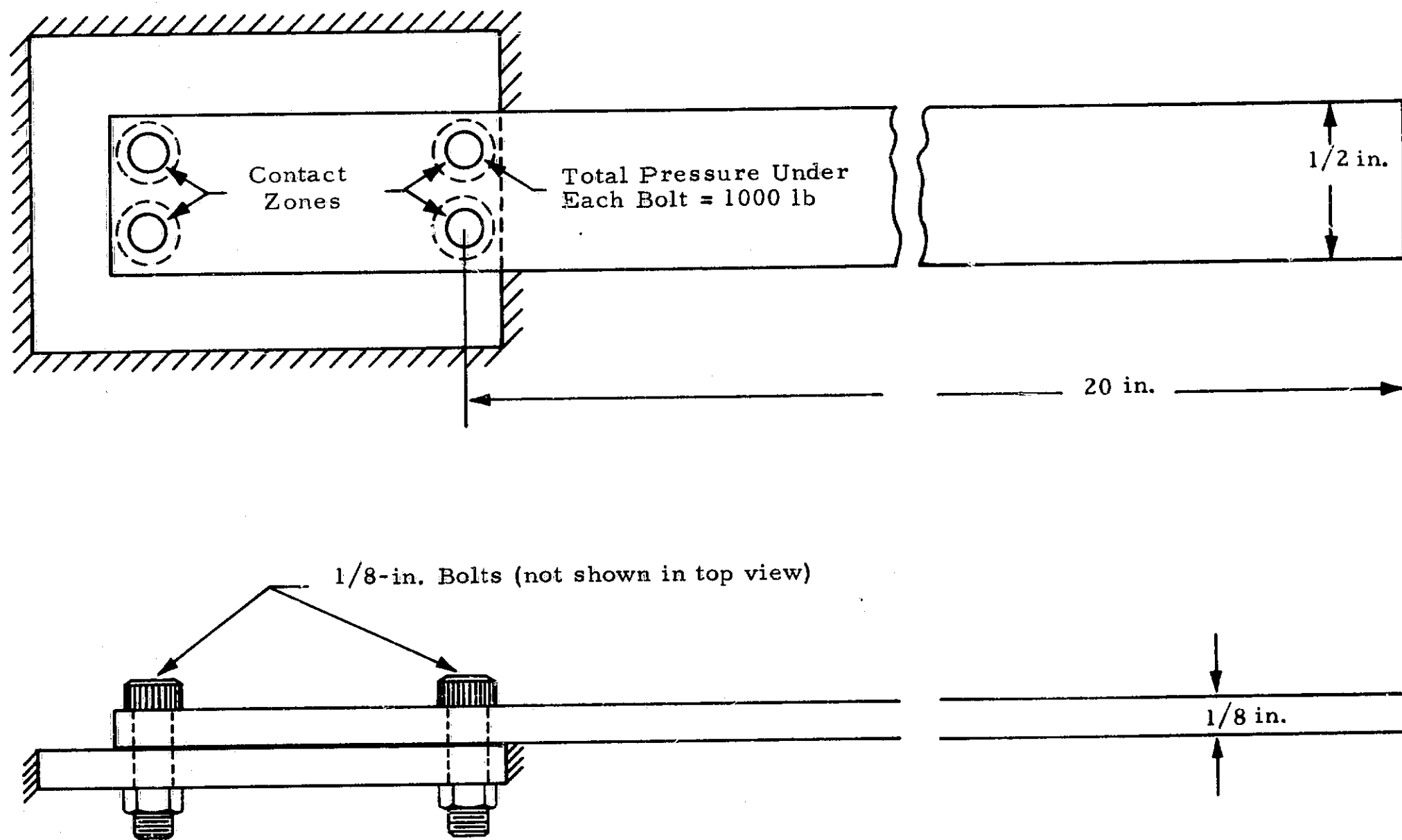


Fig. 2 - Cantilevered Aluminum Beam

The fact that under ordinary conditions gross slipping over an entire contact area is not likely to occur immediately increases the magnitude of the analytical task. The concept of partial slipping of contact areas indicates that, in order to compute joint damping, it is no longer sufficient to determine the total clamping force for a fastener only. Rather, the static contact stress distribution must be determined for a given joint clamping force. Under vibration conditions, this contact stress distribution at joint locations will undergo cyclic changes which must also be determined. To obtain an idea of what the analytical tasks involves, consider again the simple joint of Fig. 1. This joint is used in the construction of a beam with a single elastic-plus dissipative boundary condition, as shown in Fig. 3. In order to determine the steady-state response of the specimen under an applied couple as shown, the following tasks must be accomplished:

Determining the Static Preload Condition: This problem must be treated as a three-dimensional elasticity problem with radial symmetry because local deformations and stresses near the bolts are the needed solutions. One of the difficulties encountered in attempting to find an analytical solution to the problem is the fact that the boundary condition on the inner plate surfaces is a mixed one (zero normal displacement within the contact area*, and zero normal stress distribution on the outside), with the location of the boundary dividing the zero stress area and the zero displacement area being one of the unknowns of the problem. While similar difficulties have been overcome in the development of Hertz theory of contact stresses between solid bodies of revolution, the current problem is complicated by the presence of the bolt hole.

Determining the Steady State Vibration Conditions: After the static contact stress distribution over the area of contact has been determined, the steady-state vibration problems must then be formulated. One line of approach may, for example, proceed as follows. A distribution of the normal contact

*Also assuming symmetry about the interface

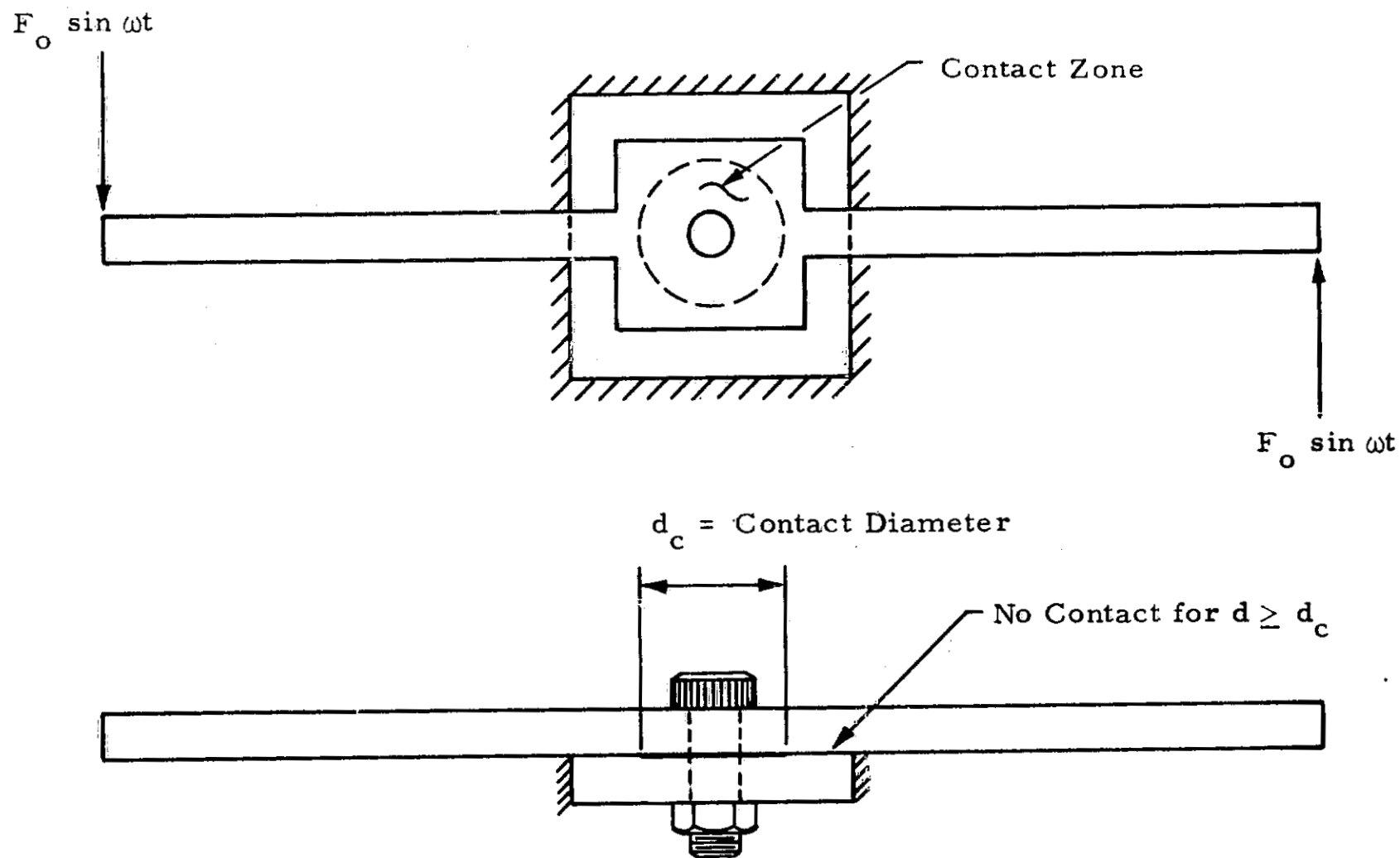


Fig. 3 - A Structure Representing the Simplest Case Where Joint Damping is Present

stress with a general shape indicated in Fig. 4 will be qualitatively assumed. For a given frequency and forcing function amplitude, the shear stress transmitted between the plates on the outer rim of the contact area will exceed the maximum friction stress first. Relative slipping will, therefore, occur there while the remainder of the contact area adheres. The extent of this "slipped region" is determined by balancing the energy dissipation per cycle over the slipped region and the rate of work being done by the excitation. Damping will likely be a function of the response amplitude, and the dynamic problem will accordingly be nonlinear. For small amplitudes and low damping, the technique developed in Ref. 1 may be used to advantage. The contact area between the plates can be represented by a nonlinear, dissipative spring, with the spring rate and the damping force analyzed in advance for static conditions.

For the more general case in which radial and plane symmetries are no longer preserved under dynamic conditions and where many fasteners are used, joint damping represents probably one of the most difficult problems in engineering. Engineering approaches, perhaps based on empirical concepts, must be developed to handle such problems.

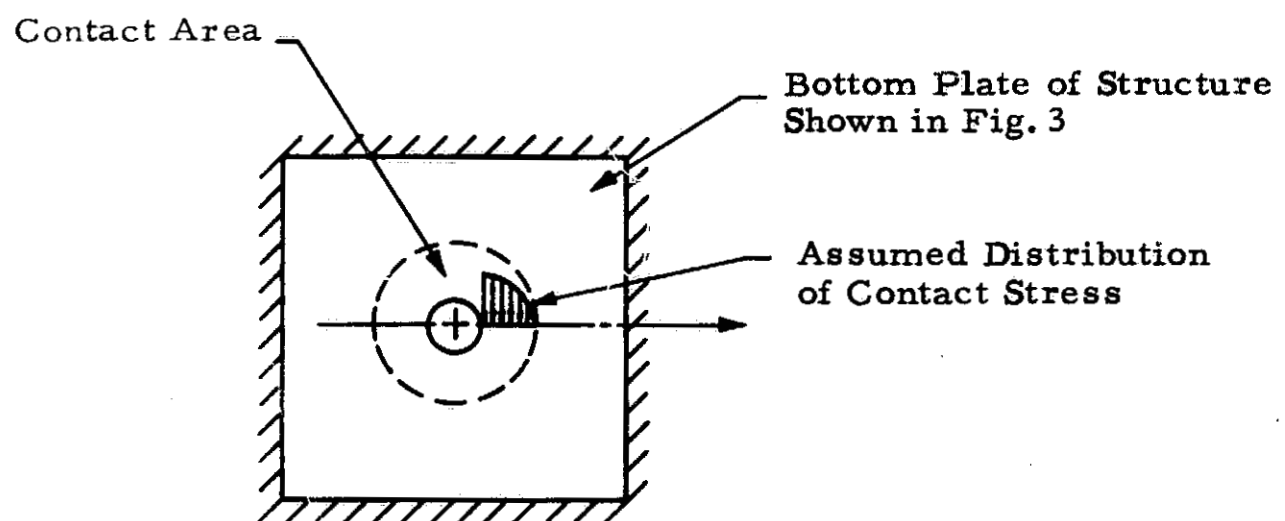


Fig. 4 - Assumed Contact Stress Distribution

Section 3

EXPERIMENTAL INVESTIGATION OF THE STATIC
CONTACT OF ELASTIC PLATES

According to previous discussions, the first needed task in a joint damping study obviously is to determine the static contact stress distribution. The simplest possible case is one where there are both radial symmetry about the bolt and plane symmetry about the contacting surfaces.

In view of the difficulties associated with seeking an analytical solution of the problem, a series of experimental investigations was conducted. A simple technique was developed for this purpose in which thick, Plexiglas plates were used as specimens. Matched pairs of circular or square Plexiglas plates with a centrally located hole ($3/8$ in. in diameter) are clamped together by a steel bolt through the hole. The clamping force is measured by a load washer as shown in the schematic drawing of the test setup in Fig. 5. Figure 6 is a photograph of the actual setup. The contact area is visible to the eye* under suitable lighting conditions and may be photographed for accurate measurements. Figure 7 is a sample of the test data. Figure 8 presents a closeup view of the specimen under test.

Four pairs of 6-in. diameter, circular plates and one pair of 6-in. square plates were used. The square plates are 1-in. thick, and the circular plate thicknesses are $3/4$ -, $1/2$ -, $1/4$ - and $1/8$ -in. Photographs representing test data for the 1-, $3/4$ - and $1/2$ -in. thick specimens are shown in Figs. 9, 10 and 11, respectively. The diameter of contact area for the $1/4$ - and $1/8$ -in. specimens were too small for measurement even for the maximum applied forces. Test results for the thicker specimens are summarized in Fig. 12, in which the

*A probable explanation of the reason why the area of contact between the plates is visible is that the neighboring surfaces simply cease to reflect light under hard contact.

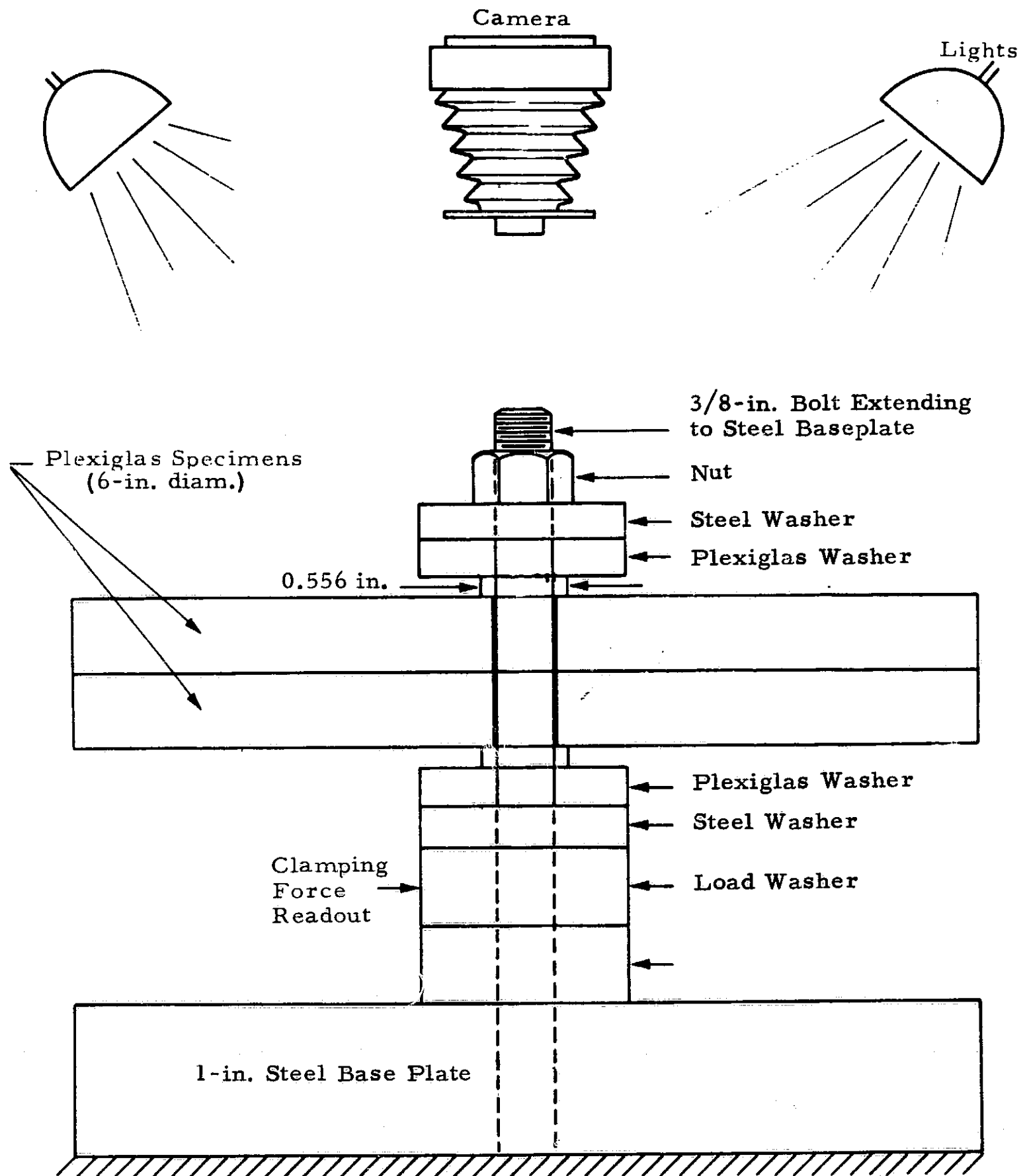


Fig. 5 - Test Setup to Measure Area of Contact Between Clamped Plexiglas Plate Specimens

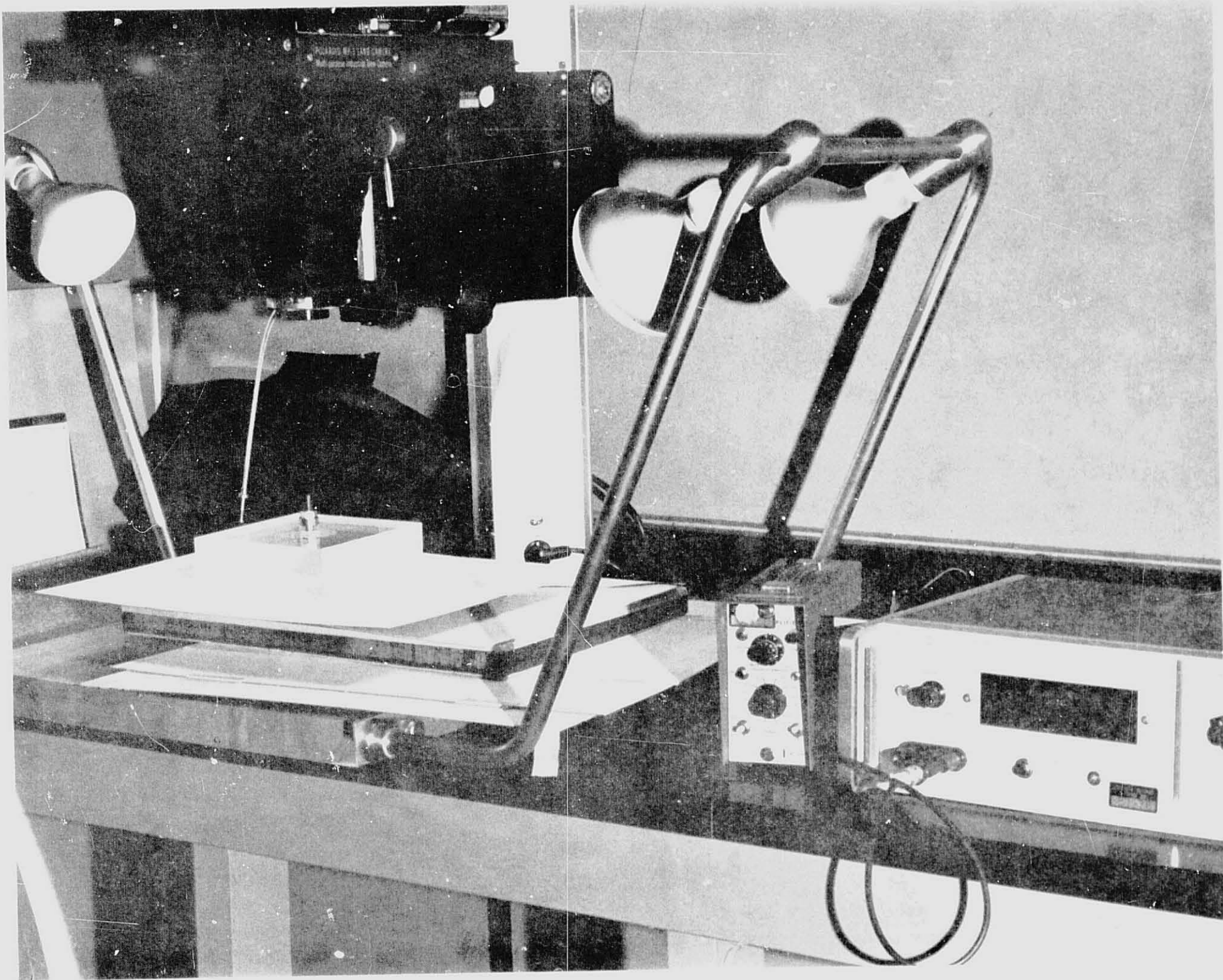
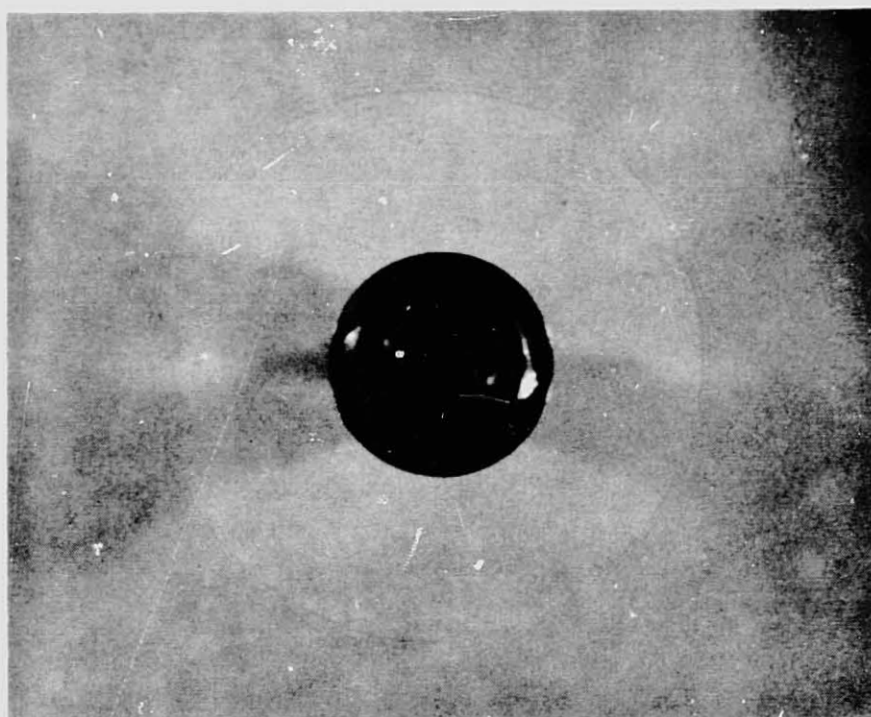


Fig. 6 - Photograph Showing Test Setup to Measure Area of Contact Between Clamped Plexiglas Plate Specimens



Actual Test Data

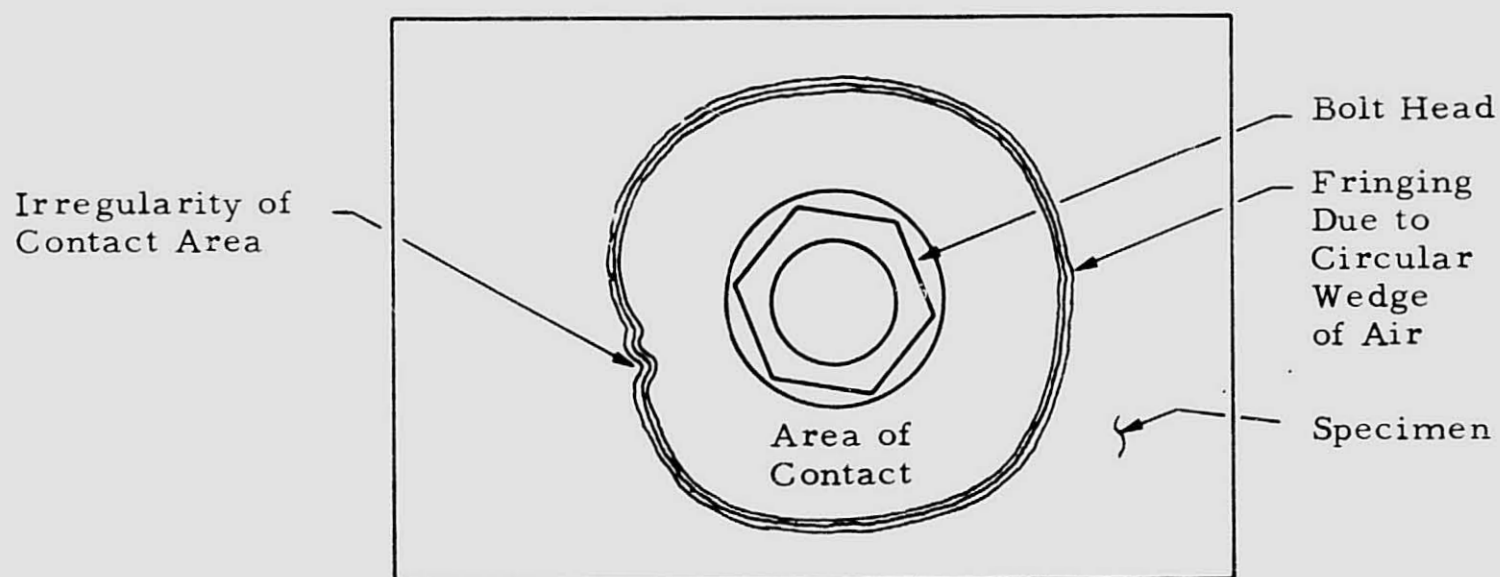


Fig. 7 - Sample Test Data - Contact Area of Plexiglas Plates Under Bolt Fastener

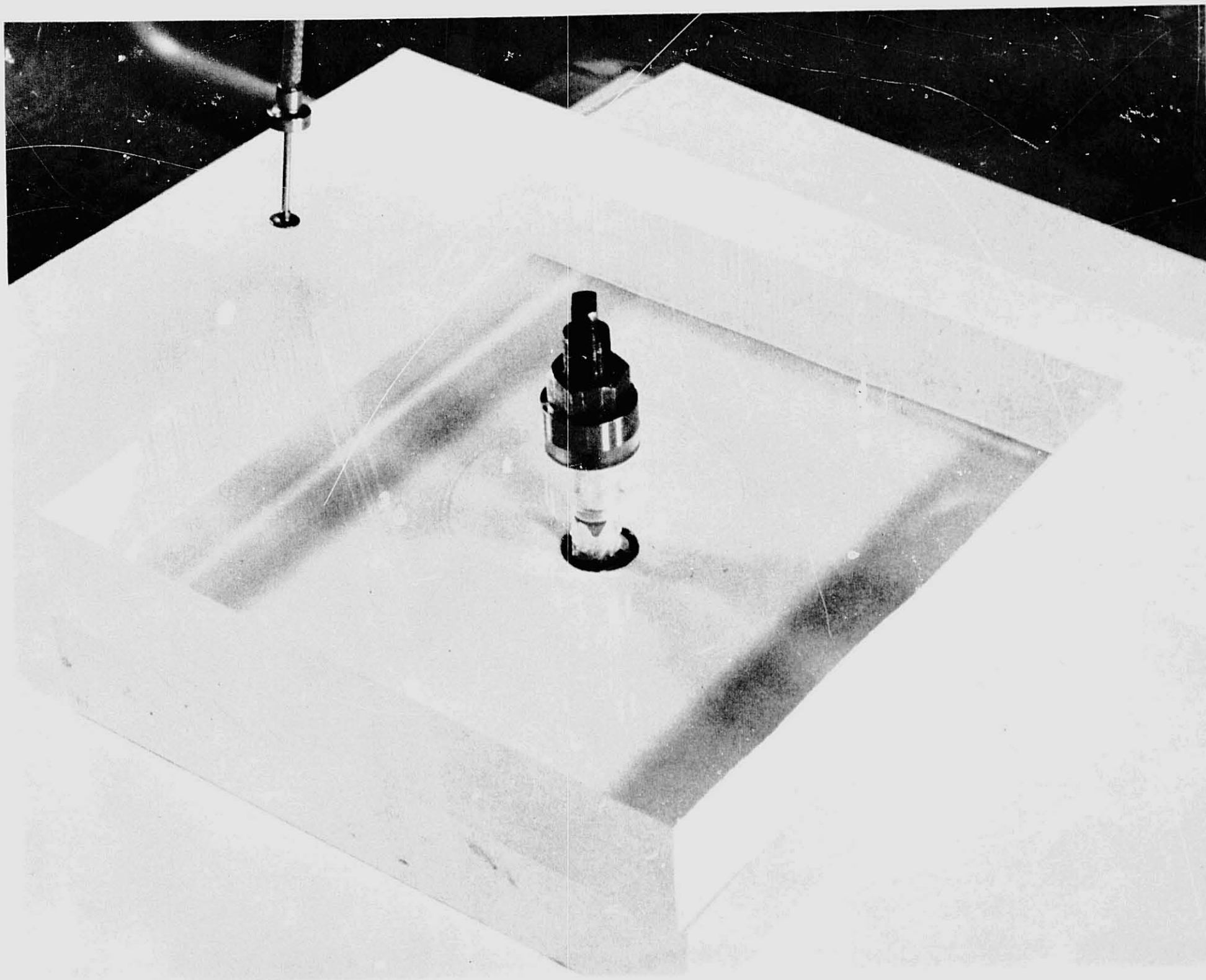
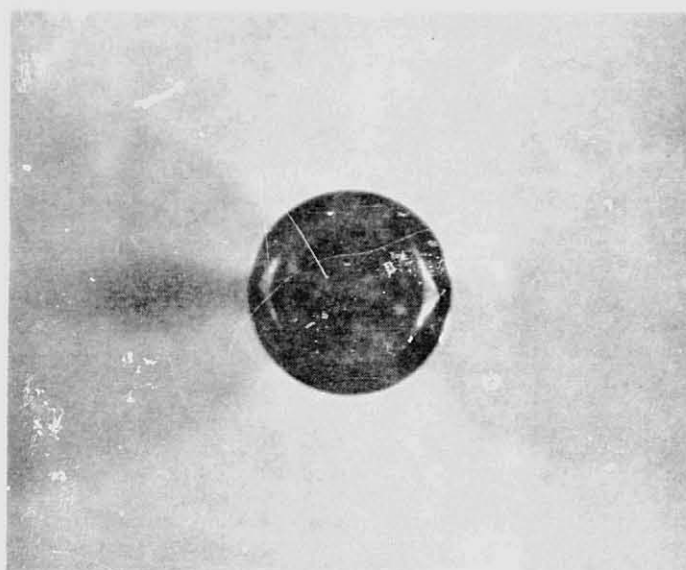
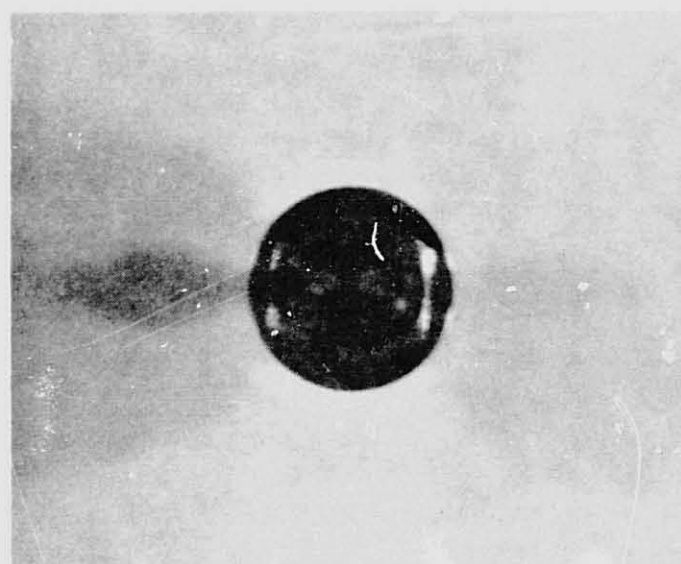


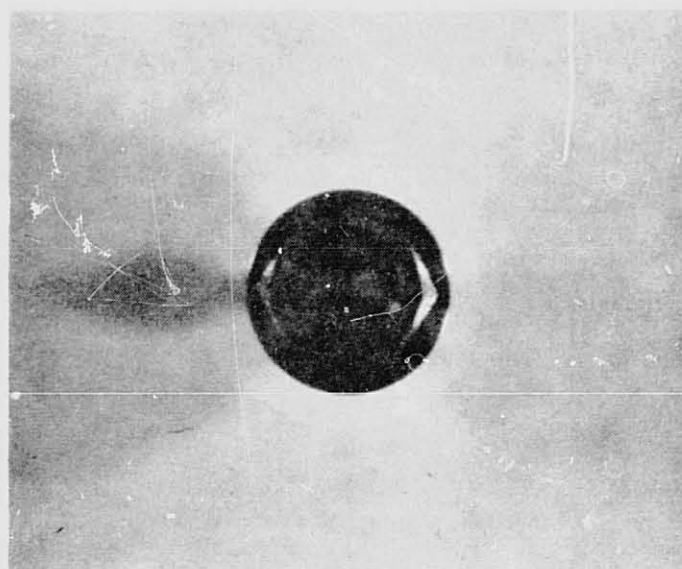
Fig. 8 - Close up View of Plexiglas Specimens Under Test
(Area of Contact is Visible at the Center of the Specimen)



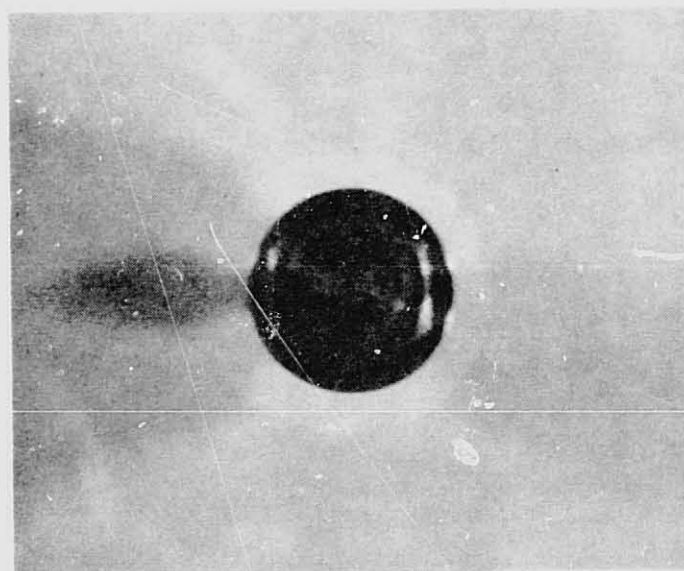
Clamping Force = 132.9 lb



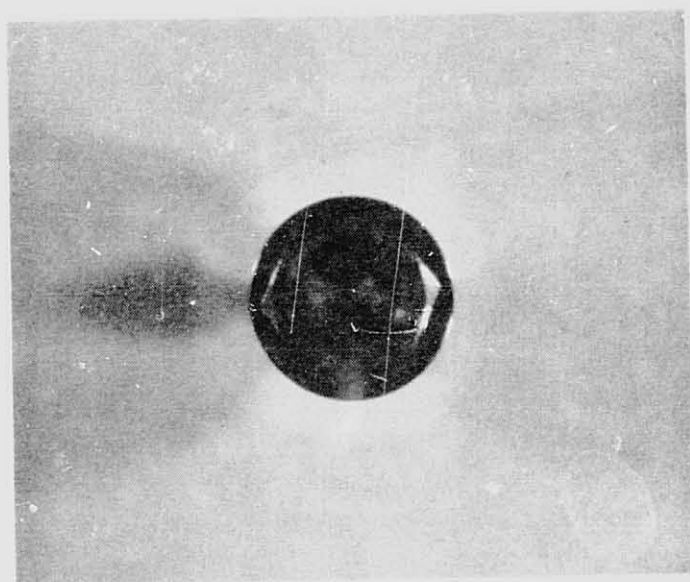
Clamping Force = 260.4 lb



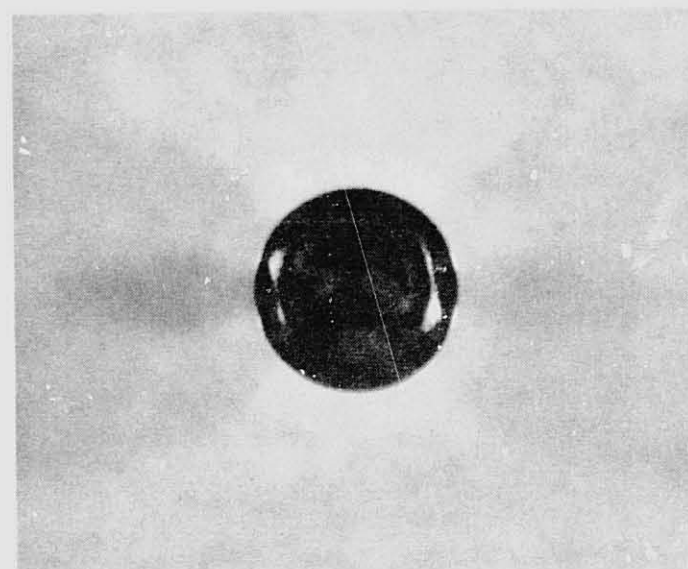
Clamping Force = 283.4 lb



Clamping Force = 510.9 lb



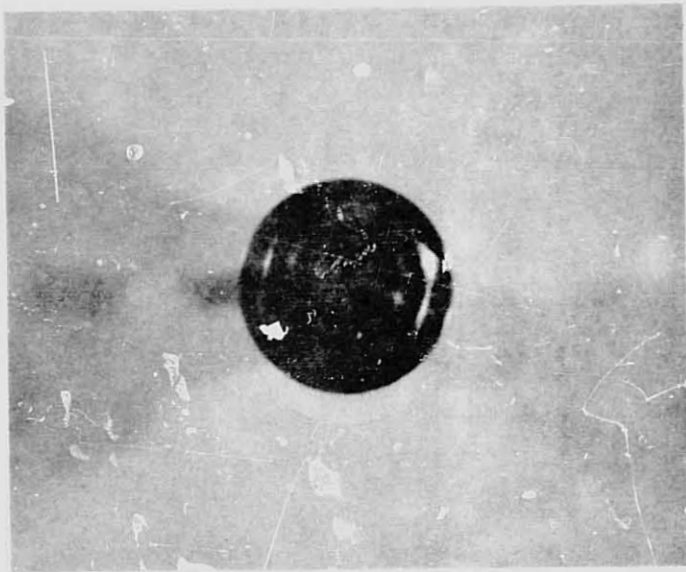
Clamping Force = 638.7 lb



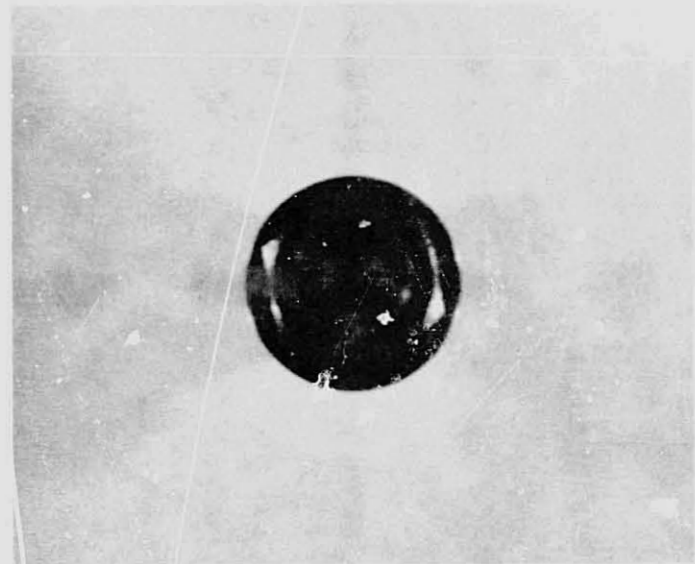
Clamping Force = 777.8 lb

Fig. 9 - Contact Area Test Data (Specimen: 1 in. Thick Plexiglas PCMES)

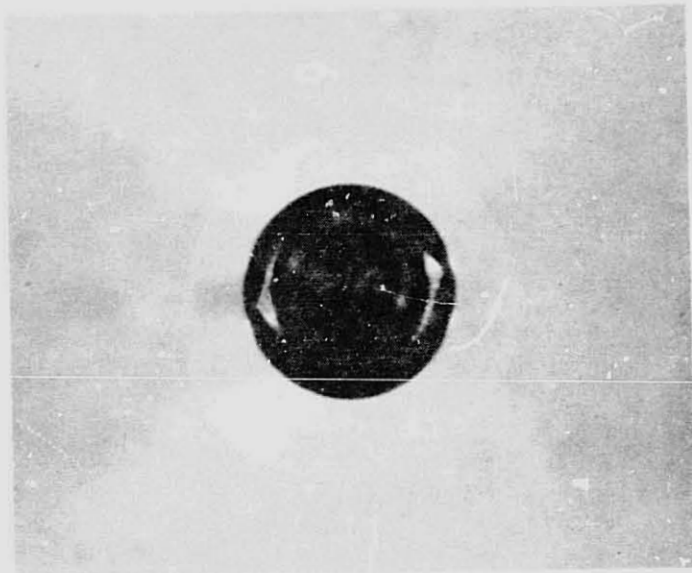
Scale (in.): 0 1
16



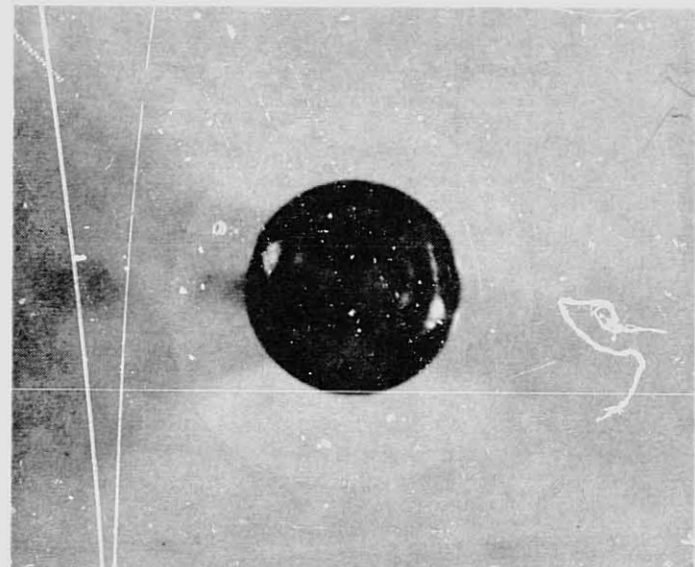
Clamping Force = 127.5 lb



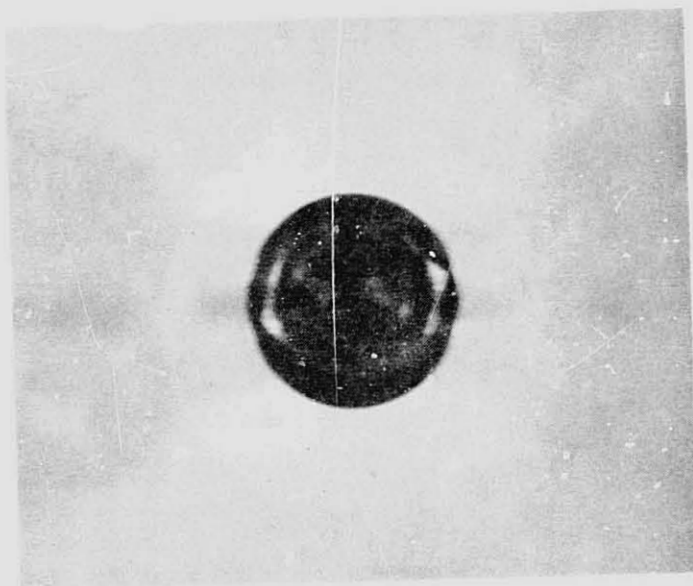
Clamping Force = 254.0 lb



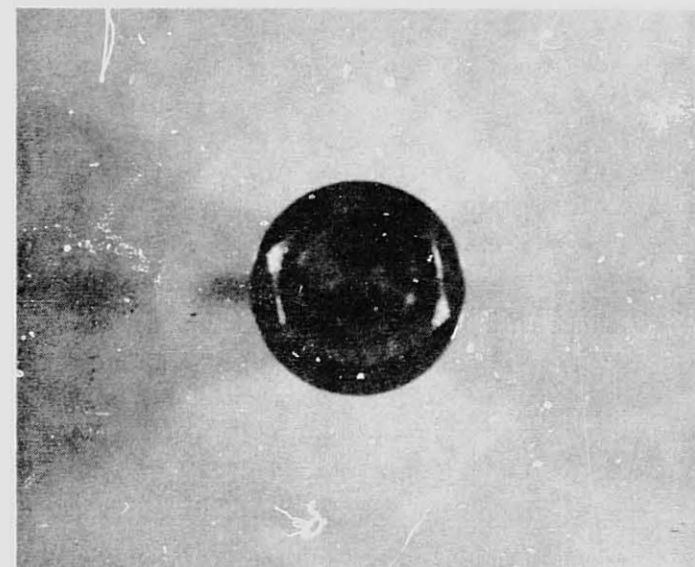
Clamping Force = 382.6 lb



Clamping Force = 515.0 lb



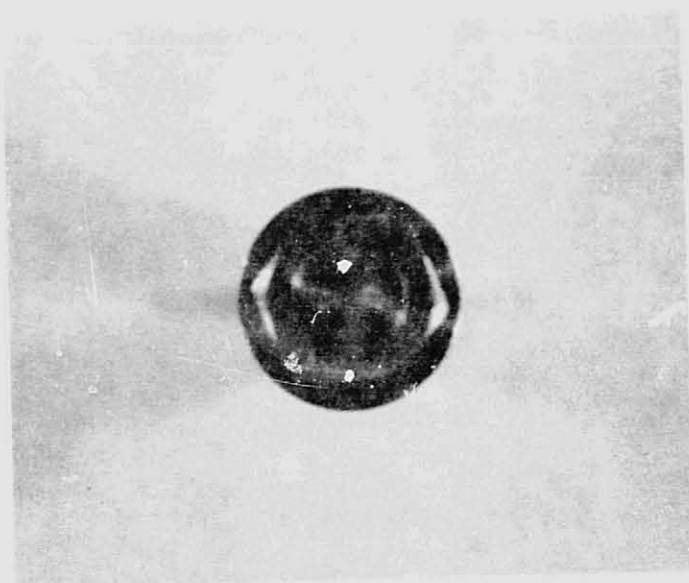
Clamping Force = 639.2 lb



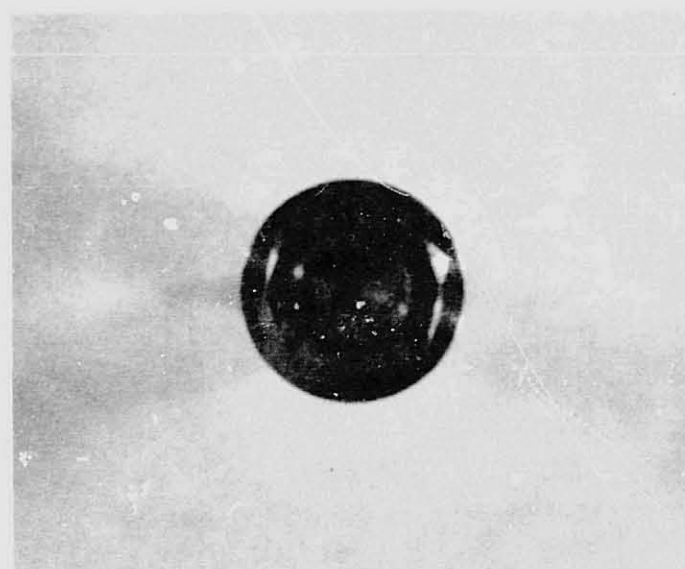
Clamping Force = 775.7 lb

Fig. 10 - Contact Area Test Data (Specimen: 3/4 in. Thick Plexiglas PCMES)

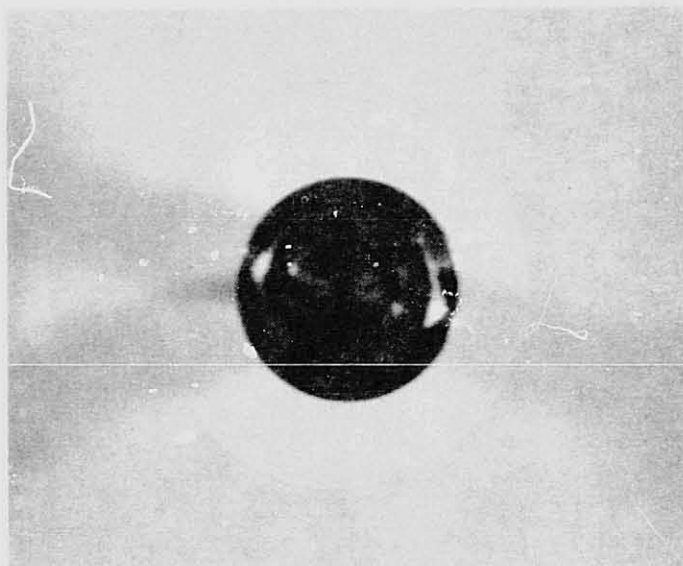
Scale (in.): 0  1



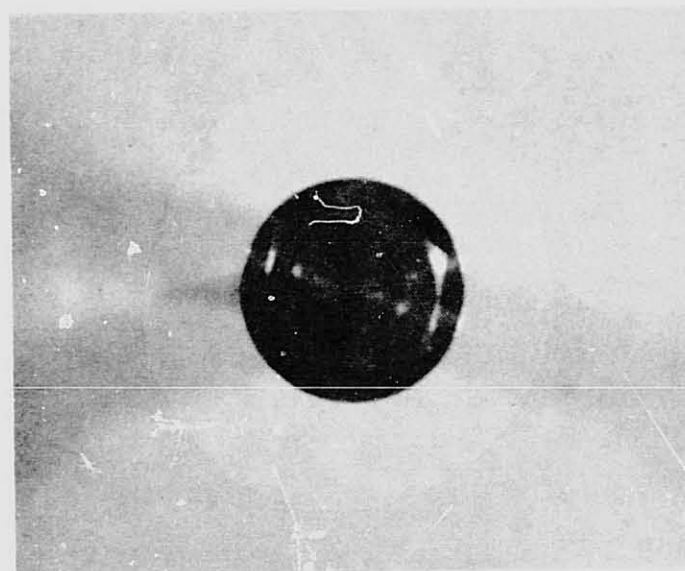
Clamping Force = 132.9 lb



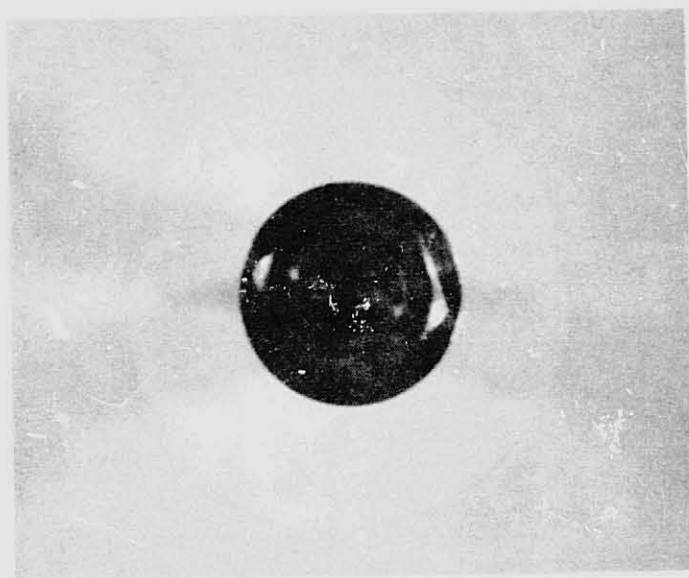
Clamping Force = 264.0 lb



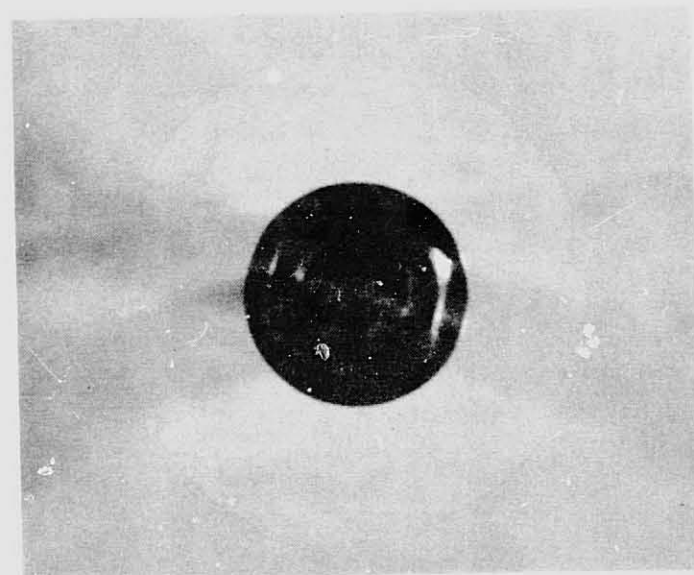
Clamping Force = 386.9 lb



Clamping Force = 514.5 lb



Clamping Force = 642.6 lb



Clamping Force = 768.1 lb

Fig. 11 - Contact Area Test Data (Specimen: 1/2 in. Thick Plexiglas PCMES)

Scale (in.): 0 1

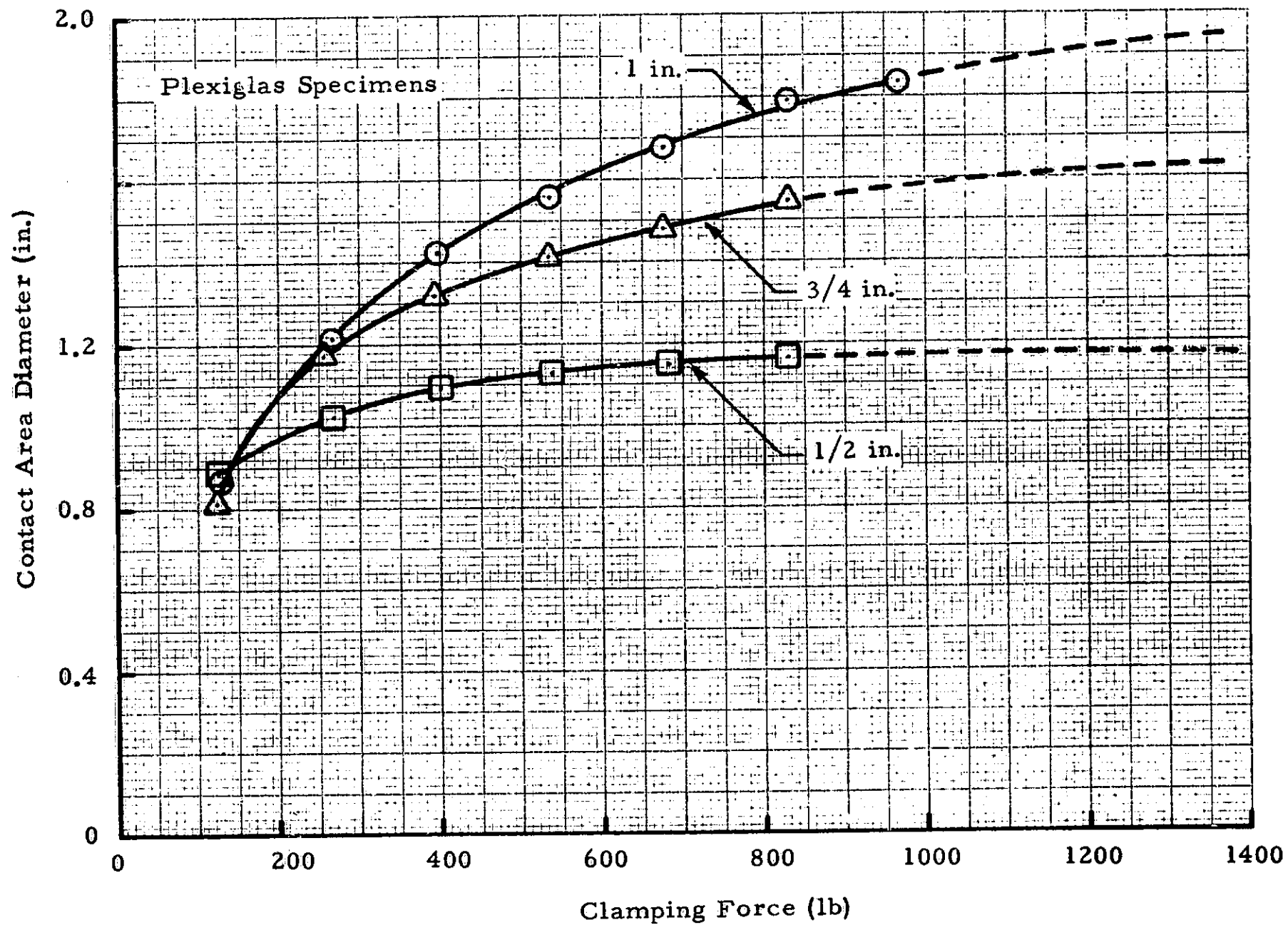


Fig. 12 - Summary of Test Results - Contact Area Diameter vs Clamping Force

average diameters of the near circular contact areas* are plotted against the clamping force. It is noted upon examination of test data that the outer diameter of the annular area of contact increases nonlinearly with the clamping force, the rate of increase being a decreasing function of the force. For different specimen thicknesses, the contact diameters appear to asymptotically approach (from below) a different limiting value for large values of the applied force. Of the five sets of specimens tested, the maximum contact diameter for the 1/8-, and 1/4-in. thick specimens is less than the diameter of the upper Plexiglas washer (0.558 in.) and could not be measured. Of the remaining three sets, the 1-in. thick specimen shows the greatest contact diameter for a given clamping force. The limiting diameter in the 1-in. specimens appears to be 1.96 in. for clamping forces greater than 1400 lb. The limiting diameter for the 3/4-in. specimens is 1.64 in. for forces greater than 1300 lb. For the 1/2-in. specimens, the limiting diameter is 1.17 in. for forces greater than 1000 lb.

To obtain an idea of the role that the modulus of elasticity plays in the solution of contact stress problem, a set of 1/2-in. glass plate specimens was tested to compare with the 1/2 Plexiglas specimens. Because it is difficult to drill holes in the glass specimen, clamping was not applied by a bolt. Rather, the specimens (without bolt holes) were clamped in a vise, as shown in Fig. 13, and the diameter of contact was estimated visually. The results of this test are summarized in Fig. 14. The most significant finding of this test is that the maximum contact area diameters for Plexiglas ($E = 532 \times 10^3$ psi) and glass ($E = 10 \times 10^6$) are not significantly different (1.05 in. versus 1.13 in.).

In the next series of experiments, the separation of the outer edges of the same specimens** under clamping were measured in addition to the local

* Initial tests showed irregularities of the contact area. The situation was drastically improved by removing all dust particles from the contacting surfaces.

** Actually, plate separations at a radius of 2.5 in. were measured.

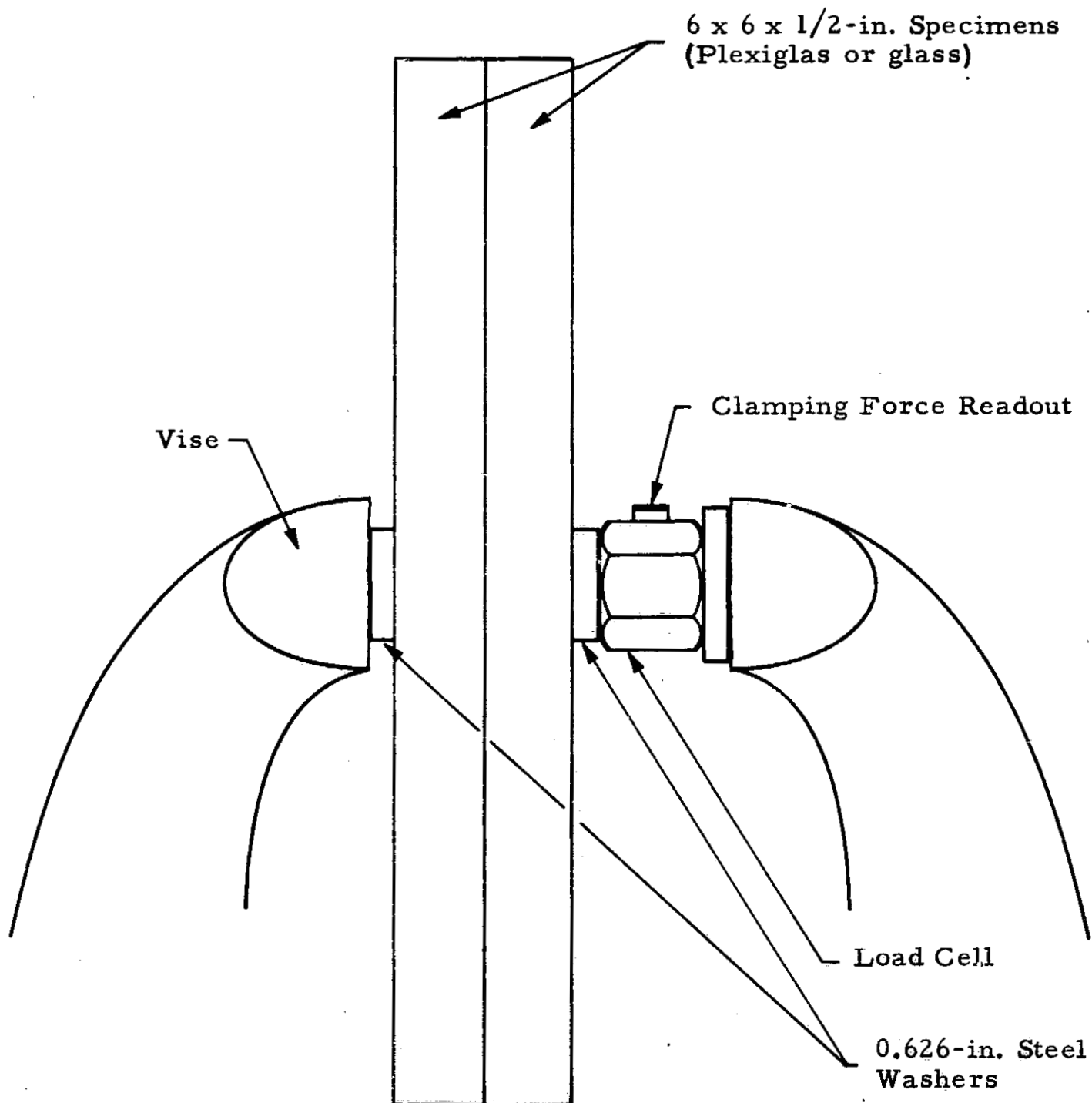


Fig. 13 - Test Setup for Investigation of Relationship Between Clamping Force, Contact Area and Specimen Moduli

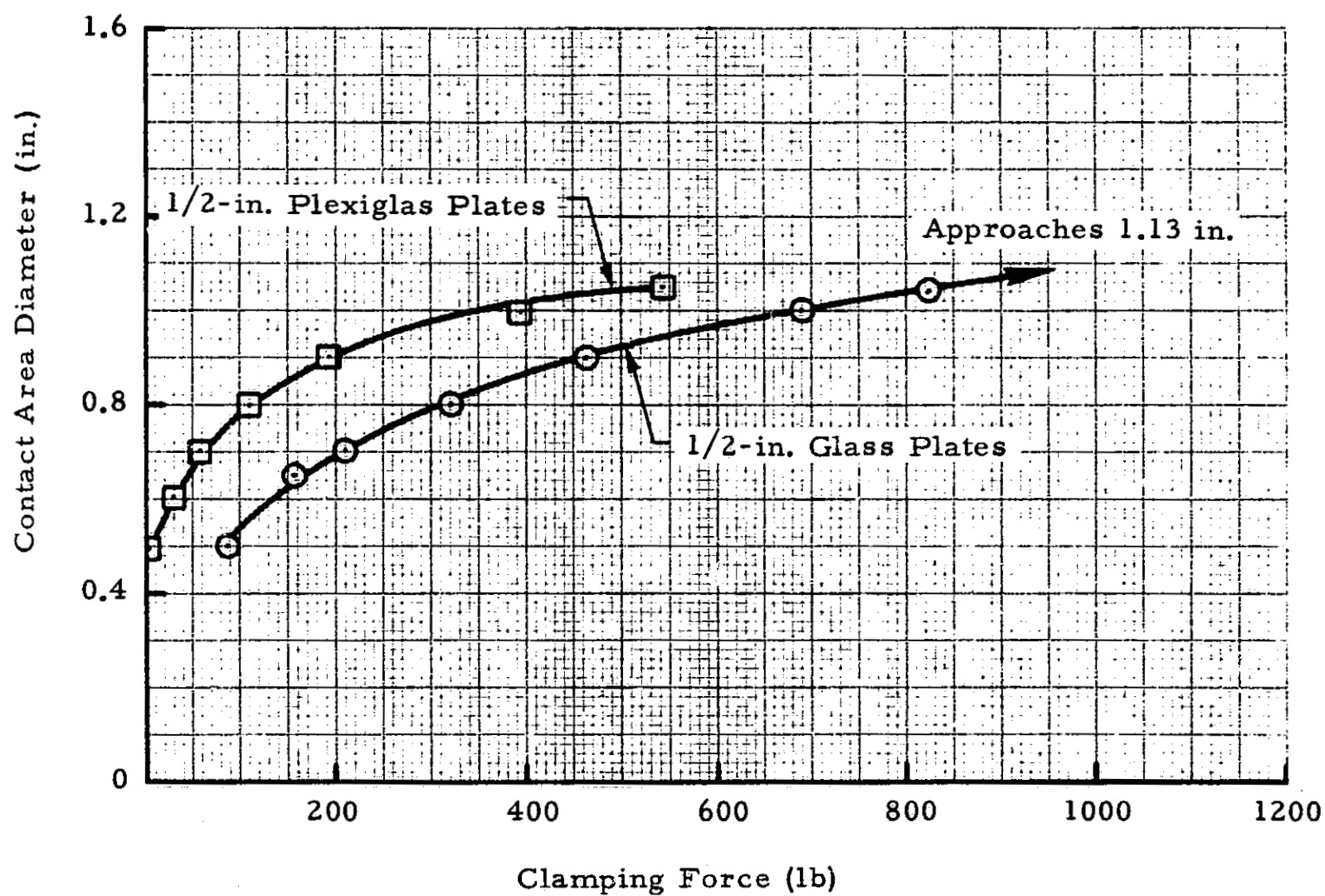


Fig. 14 - Comparison of Contact Zone Diameters Between Plexiglas and Glass Specimens

measurements described above. A linear-variable-differential-transformer (LVDT) was used to measure the total separation of the two top surfaces of the Plexiglas specimens as functions of the clamping force. The same test fixture as before was used; see Fig. 15. Test data were directly plotted on an x-y recorder and results are summarized in Fig. 16.* Edge-deflection for the 1/4-in. specimens is at least an order of magnitude greater than any other curve in Fig. 16 (note differences in vertical scales).

In appearance, each curve in Fig. 16 (corresponding to a given specimen thickness) seemed to have different characteristics. The differences are actually due to the method of measurement because the LVDT senses the total relative displacement of the upper surfaces of the specimen plates, which is the algebraic sum of the separation of the inner surfaces (sign: positive in Fig. 16) and the compression of the upper plate (sign: negative in Fig. 16) between the upper Plexiglas washer and the contact zone. The compression of the plate in the immediate vicinity of the bolt must be obtained via three-dimensional elasticity considerations. Several attempts along this line were made during the course of Phase II study without success. On the other hand, the separation of the inner surfaces at the edges of the specimens may be analyzed from plate bending theory if the stress distributions on both surfaces of the plate are known. For an approximate solution of plate edge separation, however, the distributed contact stress may be replaced by a circular line load centered about the bolt. The diameter of this circle is a function of the contact zone diameter. For small clamping forces, the contact zone is small. There is very little edge separation and a relatively large portion of the total LVDT output signal is due to the compression of the plate. For thick plates, this compression is quite easily identifiable by the negative-going, initial portions of the curves of Fig. 16. As the clamping force is increased, both the diameter and magnitude of the equivalent circular line load on the

*The 1-in. specimen showed practically zero edge separation and could not be measured with accuracy by the reported test setup.

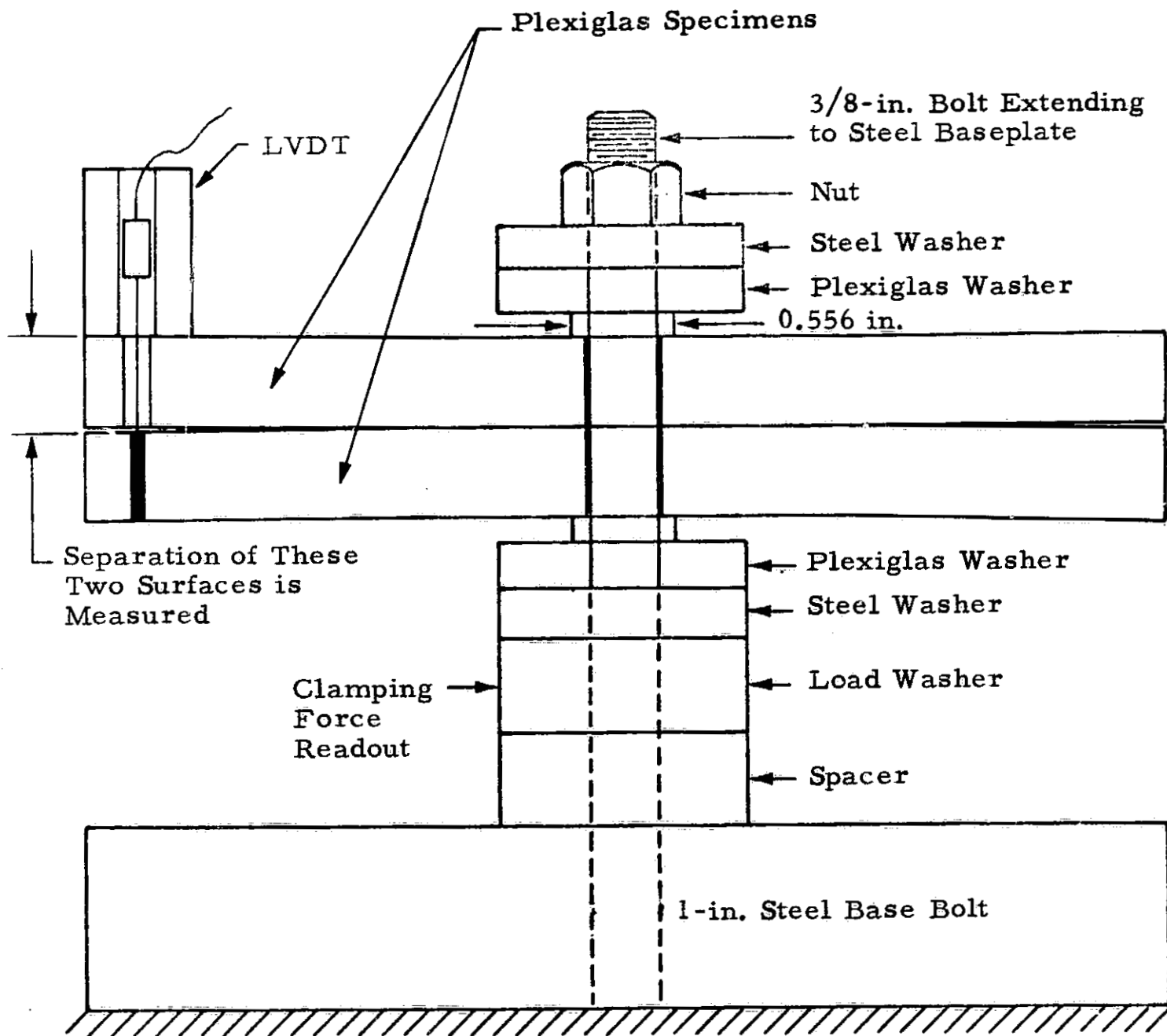


Fig. 15 - Test Setup to Measure Edge Deflection of Plexiglas Specimen Under Clamping

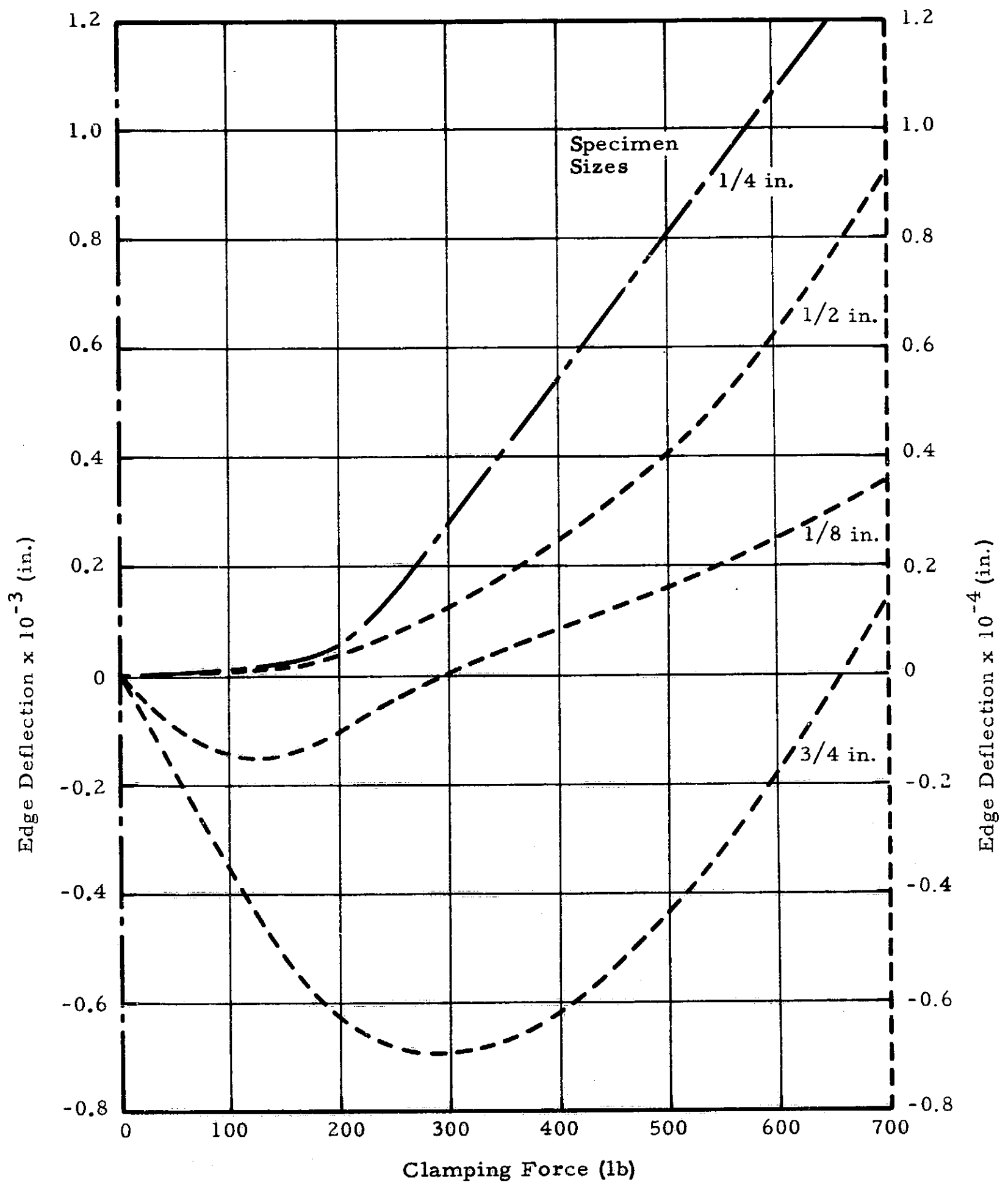


Fig. 16 - Edge Deflections for 6-Inch Circular Plexiglas Plate Specimens of Various Thicknesses

contact surface increase. The diameter of the equivalent line load between the upper Plexiglas washer and the plate cannot, however, be greater than the outside diameter of the washer. Plates, therefore, separate, and the slopes of the curves in Fig. 16 gradually increase. For very large clamping forces, the diameter of the equivalent line load on the contact surface approaches a limit, and both the compression and the separation components of the LVDT output turn into linear (incremental) functions of the clamping force. This phenomenon is most obviously demonstrated by the test data for the 1/4-in. specimens. It is interesting to note that for the 1/8-in. specimens, the measured deflections are of the same order of magnitude as the 1/2- and 3/4-in. specimens, while all of them are significantly less than the deflection of the 1/4-in. specimens for any given load. For the two thicker specimens, the large values of their flexural rigidities are the primary reason why the separations are small. For the 1/8-in. plates, the reason is that the maximum contact zone diameter is always small, and that edge separations due to compression and bending are of approximately the same magnitudes but with opposite signs. The net result is a small total measurement, as shown in Fig. 16.

Since in most engineering applications, the joints are designed to obtain maximum contact area and minimum plate separation, realistic studies of joint damping in the future must include the analysis of the very nonlinear portions of the force-deflection relationship.

Section 4

DAMPING OF SINGLE LAP JOINT IN TORSION

The objective of the series of experiments described in this section is twofold: (1) to make a thorough investigation of the single joint damping under controlled environments; and (2) to obtain an indirect measurement of the normal stress distribution at the outer edge of the contact zone through resonant frequency and damping measurements. The test series consists of resonant vibration tests^{*} at various excitation amplitudes and for different clamping forces.

The test setup is shown schematically in Fig. 17. The joint interface is formed between the two 1-in. thick Plexiglas plates and a special 3/8-in. machine bolt. An aluminum beam with a wide integral base flange is attached to the edge of the upper Plexiglas plate with a very thin layer of cement and two 1/4-in. bolts so that there was virtually no energy dissipation at this joint. The dimensions of the aluminum beam are 9.4 in. (L) x 1 in. (W) x 0.2 in. (T). This beam and the upper Plexiglas plate may then be considered as a beam having a stiff rotational base spring which is also dissipative (damping contributed by the joint interface) so that experimental techniques developed in Phase I of this study may be used to measure the spring stiffness and damping. The steel bolt is specially designed, see Fig. 18, so that it is very soft in torsion. Torsional stiffness of the base of the vibrating system is primarily obtained from the lower Plexiglas plate. Torque is transmitted from the top to the bottom plate via friction over the contact area between them. Relative partial slipping of the contact surfaces will be the primary source of damping since the bolt does not carry more than 1% of the total transmitted torque. Consequently, damping in the washers, threads between

* At frequencies where the response lags the excitation by 45, 90 and 135 degrees.

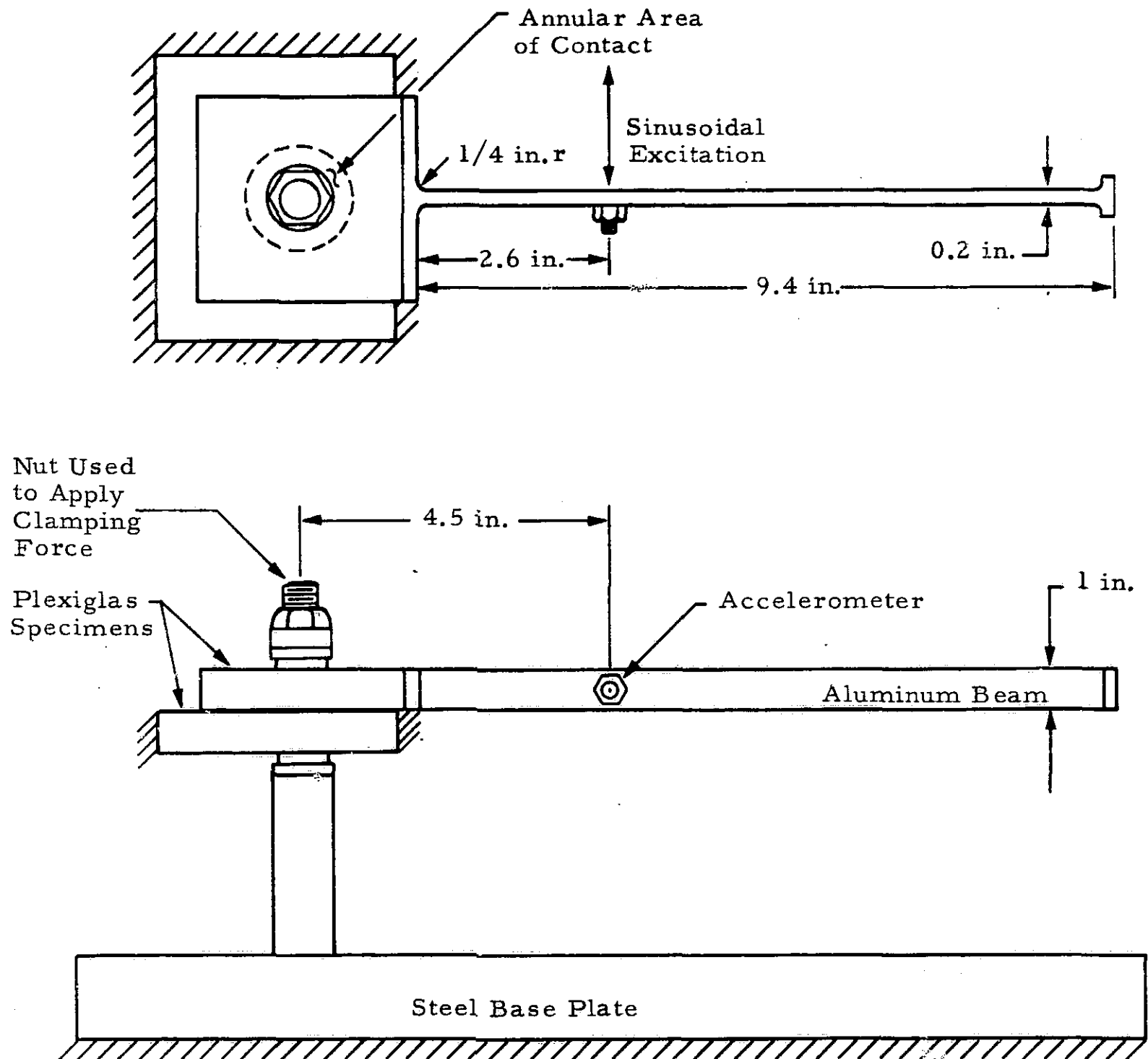


Fig. 17 - Schematic of Test Setup to Measure Damping in Torsion of Simple Lap Joint

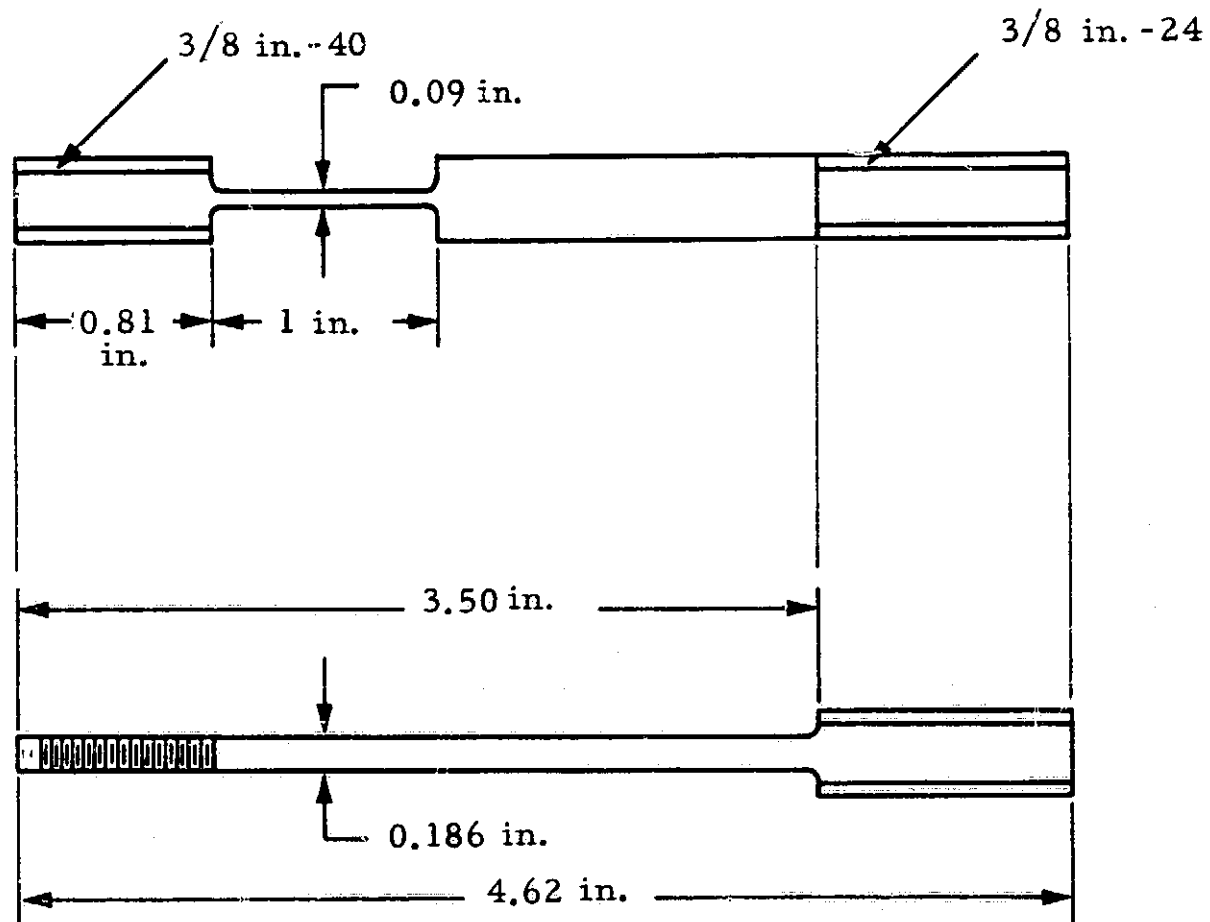


Fig. 18 - Specially Designed Steel Bolt to Minimize Torsional Stiffness and Contribution to Damping by Test Fixture

the bolt and the nut, and between the bottom threads of the bolt and the base plate are very small fractions of the total base damping, and may be easily calibrated and their effects eliminated.

Tests were conducted for various clamping forces which produced contact area diameters ranging from 0.65 to 2.05 in. For each clamping force, resonance tests in the second mode were conducted for a range of excitation and response amplitudes. Test data are summarized in Table 1. A quick examination of test results shows immediately that the resonant frequency in a given condition (clamping force) of the specimen does not vary by more than 0.2% over a significant dynamic response range. The restoring force of the system must, therefore, be relatively independent of the response amplitude. Consequently, modal damping has a well defined meaning and may be computed from the bandwidth. The energy dissipated per cycle is equated to the work done by the excitation force, and the generalized mass may be computed from the expression.

$$m = \frac{F}{2\zeta\omega_o^2 x}$$

where m is the generalized mass being sought, F is the excitation force amplitude, ζ is the fractional modal damping coefficient, ω_o is the resonant frequency, and x the response amplitude. The above computations carried out for all test data in Table 1 show that m does not vary by more than 3% and can be considered as a constant for the entire test series.

For the maximum applied clamping force condition (control zone diameter = 2.05 in.), the resonant frequency and bandwidth were found to be completely independent of response and excitation in amplitudes. Therefore, it may be assumed that the end conditions has approached a rigid cantilevered

* Where a wider dynamic range could be easily covered with the test setup and readout instrumentation than in the first mode.

Table 1
SINGLE LAP JOINT TEST DATA

Contact Diameter (in.)	Natural Frequency (Hz)	Exciting Force Amplitude (lb, 0-pk)	Resonant Response Amplitude (in., 0-pk)	45-degree Frequency (Hz)	45-degree Response Amplitude (in., 0-pk)	135-degree Frequency (Hz)	135-degree Response Amplitude (in., 0-pk)
0.6 ↓	312.5	28.02 x 10 ⁻³	11.23 x 10 ⁻⁴	311.4	7.953	313.3	8.681 x 10 ⁻⁴
	312.7	18.23	7.946	311.8	5.349	313.7	6.034
	313.0	12.50	5.645	312.1	3.621	314.0	4.238
	313.1	8.87	3.973	312.1	2.418	314.0	2.819
	313.1	6.52	2.795	311.9	1.783	314.1	1.999
	313.3	4.88	1.974	312.1	1.382	314.3	1.462
	313.3	3.40	1.400	312.3	1.024	314.3	1.036
	313.3	2.51	1.005	312.2	0.7342	314.3	0.7095
	313.2	1.67	0.6910	312.2	0.4997	314.2	0.4985
	313.3	1.12	0.4936	312.4	0.3516	314.3	0.3516
0.95 ↓	313.5	23.28	11.29	312.6	7.558	314.4	8.255
	313.7	16.26	7.953	312.8	5.417	314.6	5.941
	313.8	11.33	5.651	313.0	3.671	314.7	4.177
	314.2	7.63	3.961	313.2	2.591	314.9	2.869
	314.2	5.47	2.795	313.3	1.925	315.0	2.054
	314.3	3.79	1.980	313.4	1.406	315.0	1.431
	314.3	2.59	1.400	313.5	1.005	315.0	1.011
	314.3	1.80	1.005	313.5	0.7157	315.0	0.7157
	314.3	1.21	0.6910	313.6	0.4936	315.1	0.4873
	314.4	0.83	0.4874	313.6	0.3393	315.1	0.3395

(Continued)

Table 1 (Continued)

Contact Diameter (in.)	Natural Frequency (Hz)	Exciting Force Amplitude (lb, 0-pk)	Resonant Response Amplitude (in., 0-pk)	45-degree Frequency (Hz)	45-degree Response Amplitude (in., 0-pk)	135-degree Frequency (Hz)	135-degree Response Amplitude (in., 0-pk)
1.4 ↓	314.4	19.89	11.22×10^{-4}	313.6	7.860	315.1	8.483×10^{-4}
	314.5	13.55	7.946	313.8	5.293	315.2	5.799
	314.6	9.67	5.645	313.8	3.757	315.3	4.041
	314.6	6.89	3.973	313.8	2.751	315.3	2.918
	314.6	4.86	2.788	313.8	2.017	315.3	2.048
	314.6	3.32	1.974	313.9	1.400	315.3	1.437
	314.6	2.29	1.394	313.9	0.9872	315.3	0.9872
	314.7	1.67	1.005	313.9	0.6972	315.4	0.6910
	314.7	1.17	0.6910	313.9	0.4855	315.4	0.4966
	314.7	0.81	0.4936	314.0	0.3486	315.3	0.3461
1.4							
1.9	314.6	18.97	11.22	313.9	7.811	315.3	8.150
1.9	314.7	3.36	1.974	313.9	1.347	315.4	1.406
2.0	314.6	19.03	11.22				
2.05 ↓	315.0	15.83	11.22	314.3	7.872	315.5	7.891
	315.0	11.33	7.946	314.3	5.620	315.5	5.663
	315.0	8.00	5.645	314.4	3.967	315.6	4.004
	315.0	5.67	3.973	314.4	2.819	315.6	2.825
	315.0	3.96	2.795	314.4	1.961	315.6	2.005
	315.0	2.79	1.974	314.4	1.375	315.6	1.388
	315.0	1.98	1.400	314.4	0.968	315.6	0.999

(Continued)

Table 1 (Continued)

Contract Diameter (in.)	Natural Frequency (Hz)	Exciting Force Amplitude (lb, 0-pk)	Resonant Response Amplitude (in., 0-pk)	45-degrees Frequency (Hz)	45-degree Response Amplitude (in., 0-pk)	135-degree Frequency (Hz)	135-degree Response Amplitude (in., 0-pk)
 2.05	315.1	1.42	1.005×10^{-4}	314.4	0.678	315.6	0.729×10^{-4}
	315.1	0.97	0.6910	314.4	0.484	315.6	0.478
	315.1	0.70	0.5016	314.4	0.343	315.6	0.347

condition. Damping may, in this case, be assumed to be due to other sources such as material and air damping but not due to joint dissipation. In other words, damping for the maximum clamping force tests may be used as a reference for other test cases. The differences in the rate of energy dissipation between this and other cases of clamping forces may be attributed to the damping mechanism on the joint surface under investigation. These differences have been computed and results are tabulated in Table 2 and plotted in Fig. 19. Two remarkable features of the results are: (1) data scatter was not significant, especially in view of the consideration that the final results were obtained as the differences of two sets of large numbers. Associated with consistent data are, of course, increased confidence on the absolute values represented by them; and (2) the energy dissipation per cycle of vibration for all joint clamping forces are proportional to the square of the response amplitude so that linear damping is clearly indicated.

In summary, test data over a wide range of joint clamping forces and excitation and response amplitudes have shown joint damping to be: (1) of comparable magnitude as material and air damping combined; (2) linear, and can be measured with consistency. Neither of these features has been the distinguishing characteristics of joint damping as it is commonly known. It may, therefore, be concluded on the basis of discussions so far that in order to reach the goal of predicting structural damping, more complicated joint configurations must be considered. The commonly assumed nonlinear, high structural damping cannot be noticed from test data on a single, one-bolt joint with both radial and planar symmetries in dynamical conditions.

Measured linearity of joint damping serves to indicate that the original concept of partial slipping is correct, with the overall contact area determined under the static preload remaining constant during vibration, and with the normal stress distribution near the edge of the contact area approximately linear and continuous. As the torsional vibration amplitude is increased, an annular zone of dynamic slipping develops because the shear stress exceeds

Table 2
COMPUTED JOINT DISSIPATION RATE

Contact Diameter (in.)	Resonant Response Amplitude (in., 0-pk)	Work Done Cycle (in.-lb)	Damping $\frac{\delta \omega}{2 \omega_0}$ %	Joint Dissipation Per Cycle (in.-lb)
0.6 ↓	11.72×10^{-4}	$.09871 \times 10^{-3}$	.3040	$.04291 \times 10^{-3}$
	7.946	.04551	.3038	.01723
	5.645	.02214	.3035	.007953
	3.973	.01107	.3034	.003993
	2.795	.005725	.3513	.00225
	1.974	.003020	.3511	.001296
	1.400	.001496	.3191	.000625
	1.005	.000792	.3351	.0003442
	0.6910	.0003625	.3192	.000152
	0.5016	.0001793	.3032	.000069
0.95 ↓	11.22×10^{-4}	$.08155 \times 10^{-3}$	.2871	$.02575 \times 10^{-3}$
	7.946	.04056	.2869	.01228
	5.645	.02007	.2709	.005883
	3.973	.09553	.2705	.00242
	2.795	.00480	.2705	.00133
	1.974	.00234	.2545	.000614
	1.400	.00114	.2386	.000268
	1.005	.000568	.2386	.000120
	0.6910	.000263	.2386	.0000521
	0.5016	.000120	.2385	.0000097
1.4 ↓	11.22×10^{-4}	$.07011 \times 10^{-3}$	.2385	$.01431 \times 10^{-3}$
	7.946	.03382	.2385	.00554
	5.645	.017149	.2384	.00296
	3.973	.008599	.2384	.00152

(Continued)

Table 2 (Continued)

Contact Diameter (in.)	Resonant Response Amplitude (in., 0-pk)	Work Done Cycle (in.-lb)	Damping $\frac{\delta\omega}{2\omega_0}$ (%)	Joint Dissipation per Cycle (in.-lb)
1.4 ↓	2.795×10^{-4}	$.00423 \times 10^{-3}$	.2384	$.000760 \times 10^{-3}$
	1.974	.002059	.2225	.000329
	1.400	.000994	.2225	.0001736
	1.005	.0005272	.2224	.0000789
	0.6910	.0002540	.2224	.0000434
	0.5016	.0001216	.2065	
2.05 ↓	11.22×10^{-4}	$.055799 \times 10^{-3}$	.1905	0
	7.946	.028283	↓	↓
	5.645	.014187		
	3.973	.007077		
	2.795	.003477		
	1.974	.0017302		
	1.400	.0008708		
	1.005	.0004483		
	0.6910	.0002106		
	0.5016	.0001103	.1905	0

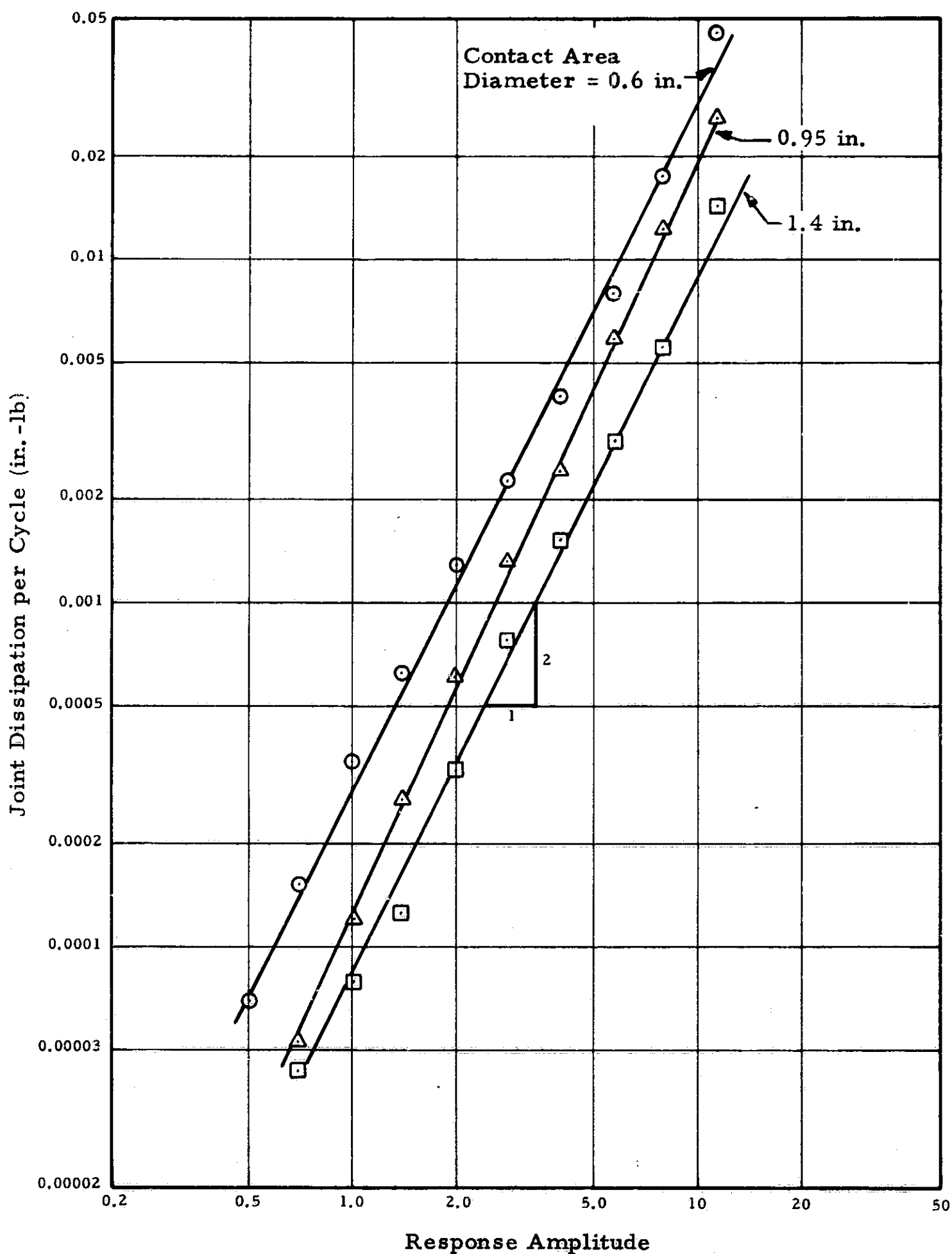


Fig. 19 - Joint Damping in Torsion

the product of the coefficient of friction and the local normal stress. The total frictional torque of this slipped zone is approximately proportional to the radial width because of the linear distribution of the normal stress. The energy dissipation per cycle is proportional to the product of the frictional torque and the angular displacement, and is approximately proportional to the square of the vibration amplitude if slipping is restricted to a very small annular area at the edge of the contact zone.

Section 5

CONTACT ZONES OF MULTIPLE-BOLT JOINTS

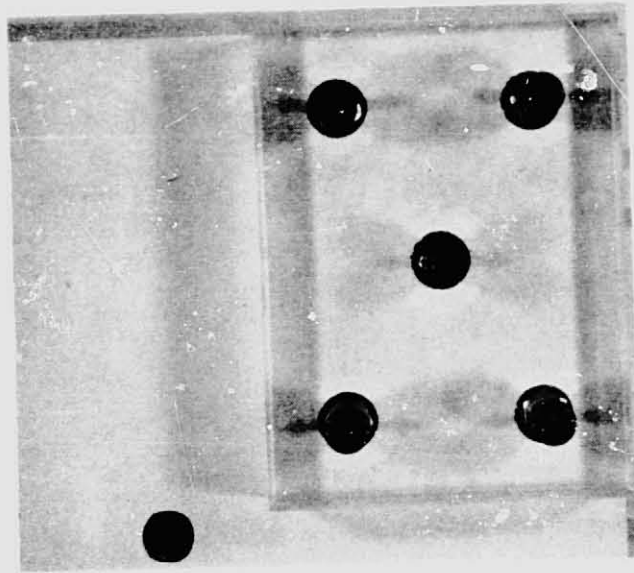
As a first step towards the study of damping in joints with more than one fastener, a visual investigation of the contact areas of such a joint was conducted using, again, Plexiglas plate specimens.

Two specimens were made of 1/2-in. Plexiglas with 3/8-in. bolts arranged to represent typical multiple fastener joints. A torque of up to 50 in.-lb was applied to each bolt in different sequences to study interactions of contact areas. One of the specimens has nine bolts arranged in a square matrix with 2 in.- pitch, Fig. 20a. The plates are 6-in. square. The contact diameter of the center bolt was increased by approximately 10% when surrounding bolts were tightened. The contact area remained approximately circular in shape.

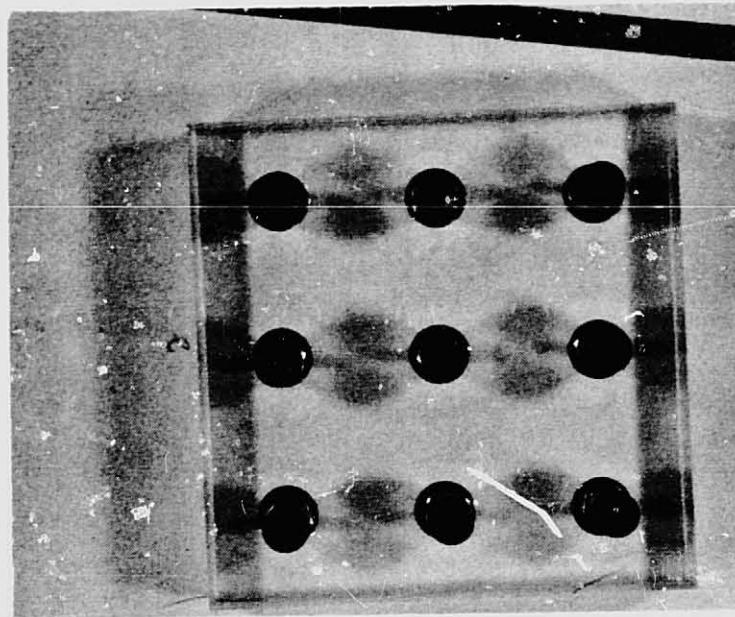
When one of the corner bolts was tightened first, and then the adjacent bolts, the contact areas under the corner bolt did not change noticeably in size, but shifted slightly toward the center of the plate. When one of the center side bolts was tightened first, then the adjacent ones, the contact area of the center side bolt showed a 10% increase in diameter of contact.

The other specimen has five bolts but they were arranged in a staggered manner, see Fig. 20b. The minimum bolt-to-bolt spacing was 2.8 in. When the center bolt was tightened first, then the remaining ones increase in contact area of the center bolt was very slight but noticeable. When a corner bolt was tightened first, then adjacent ones, there was no significant increase in contact area for the bolt.

The above results provide some of the necessary engineering data to extend contact diameter measurements obtained for a single bolt-specimen for actual application in multiple bolt joints.



(b)



(a)

Fig. 20 - Test Specimen for Multiple-Joint Damping Study

Section 6

DAMPING IN MULTIPLE-BOLT JOINT SPECIMENS

To obtain a set of controlled experimental data on damping in multiple-bolt joints, two sets of experiments were conducted on separate specimens.

The first of these specimens tested is shown in Fig. 21. The material used, again, was Plexiglas. The joint interfaces were machined surfaces and the coefficients of friction were found to be lower (0.407 against 0.48 for the static case, and 0.33 against 3.6 for the kinetic case). The joint configuration was selected such that: (1) it is a multiple-bolt joint; (2) the joint interface is planar; and (3) translation, rather than torsion, is the type of anticipated slipping of the joint interfaces. Therefore, the joint specimen is quite complicated as compared to the single-bolt joint in torsion. By testing the joint with all screws (3-48) in place and tightened first, and then removing selected screw(s), the damping due to each screw and the associated critical area can be isolated.

Resonant vibration tests were employed to measure joint damping. The following three joint configurations were tested:

Configuration	Description
P6	As shown in Fig. 21 all six 3-48 screws on
P5	Same as P6 but with one center screw removed
P4	Same as P5 but with the center screw on the other side also removed.

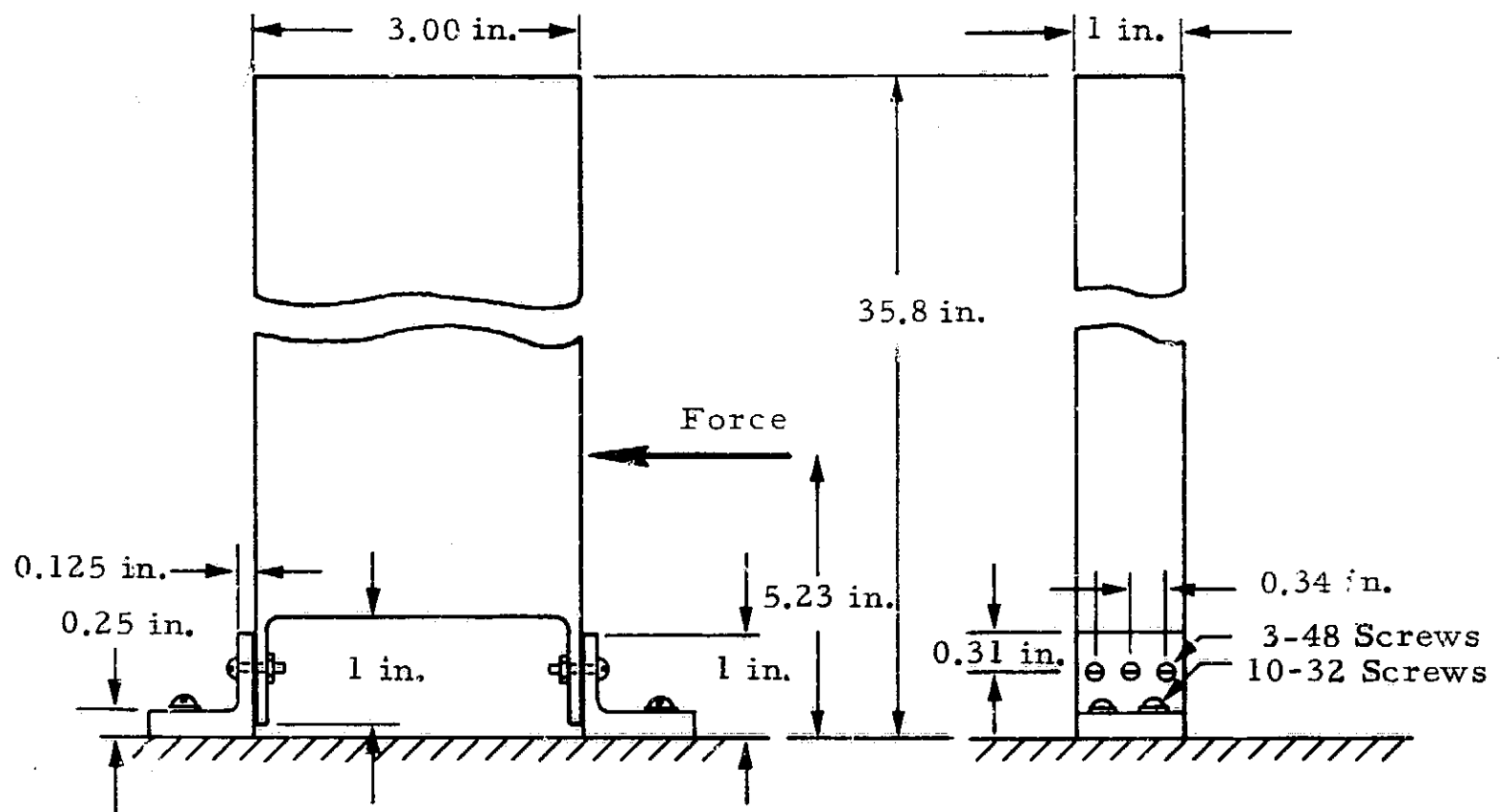


Fig. 21 - Test Specimen for Determination of Damping of One Bolt in Multiple-Bolt Planar Joint

Test data of these configurations are tabulated in Table 3(a), (b) and (c). The work done per cycle by the excitation for each test condition is computed for a common set of displacement amplitudes and are listed in Table 4. The ratios (dB differences) of energy dissipates at equal displacement amplitudes between Test P6 and P5, and between P5 and P4 represent the energy dissipation per cycle due to the center screws. These results are also listed in Table 4. It is seen that one of the screws had a much greater contribution to damping than the other. However, neither one contributed more than 0.1 dB (1%) to the total damping. Once again, when a bolt is singled out for damping studies, its contribution to damping is extremely small, even though the study was not performed on a single bolt, but on one of many bolts in a joint.

In a second series of tests, a multiple-bolt joint with a cylindrical interface was employed (Fig. 22). In diametrical cross-section, this joint is identical to the planar joint specimen. Also, some machining and assembly procedures were employed for both specimens, and the test equipment was the same.

Tests were conducted for the following joint configurations:

Configuration	Description	
C16	As shown in Fig. 22	} For test data, see Tables 5a through 5f
C14	C16 less screws 5 & 13	
C12	C14 less screws 1 & 9	
C10	C12 less screws 7 & 11	
C 8	C10 less screws 3 & 14.	

Energy dissipation per cycle of vibration due to each pair of screws is computed and tabulated in Table 6. It is seen that damping due to screws No. 1 and 9 is significantly greater than damping due to all other pairs. This situation is expected because maximum slipping occurs at these screw locations.

Table 3a

PLANAR MULTIPLE-SCREW JOINT DAMPING TEST DATA (CONFIGURATION P6)

Force Ampl. dB Re 8×10^{-4} lb	90° Displ. Ampl. dB Re 2×10^{-6} in	90° Period (T_{90}) ms	45° Period (T_{45}) ms	135° Period (T_{135}) ms	Damping % $\frac{(T_{45} - T_{135})T_{90}}{2 T_{45} \times T_{135}}$
50.00	44.79	6.4800	6.7632	6.2846	3.64
47.00	41.75	6.4757	6.7564	6.2817	3.62
44.00	38.70	6.4674	6.7487	6.2731	3.63
41.00	35.68	6.4639	6.7413	6.2660	3.63
38.00	32.75	6.4612	6.7381	6.2662	3.61
35.00	29.87	6.4581	6.7249	6.2632	3.53
32.00	26.99	6.4491	6.7062	6.2429	3.56
29.00	24.02	6.4373	6.6807	6.2440	3.36
26.01	21.08	6.4380	6.6806	6.2217	3.55
23.00	18.03	6.4380	6.7290	6.2236	3.88
19.98	15.04	6.4470	6.7200	6.2292	3.77

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Table 3b

PLANAR MULTIPLE-SCREW JOINT DAMPING TEST DATA (CONFIGURATION P5)

Force Ampl. dB Re 8×10^{-4} lb	90° Displ. Ampl. dB Re 2×10^{-6} in	90° Period (T_{90}) ms	45° Period (T_{45}) ms	135° Period (T_{135}) ms	Damping % $\frac{(T_{45} - T_{135})T_{90}}{2 T_{45} \times T_{135}}$
50.00 x 10^{-4}	44.98 x 10^{-6}	6.5322	6.8211	6.3317	0.0370
47.00	41.94	6.5271	6.8260	6.3306	0.0374
44.00	38.90	6.5233	6.8122	6.3235	0.0370
41.00	35.89	6.5190	6.8081	6.3162	0.0372
38.00	32.94	6.5151	6.8028	6.3156	0.0369
35.00	30.04	6.5090	6.7782	6.3026	0.0362
32.00	27.16	6.5062	6.7636	6.2939	0.0358
29.00	24.22	6.4966	6.7604	6.2910	0.0358
26.02	21.31	6.5055	6.7468	6.2934	0.0347
22.98	18.21	6.4969	6.7235	6.3048	0.0320
19.93	15.21	6.4937	6.7131	6.3046	0.0313

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Table 3c

PLANAR MULTIPLE-SCREW JOINT DAMPING TEST DATA (CONFIGURATION P4)

Force Ampl. dB Re 8×10^{-4} lb	90° Displ. Ampl. dB Re 2×10^{-6} in	90° Period (T_{90}) ms	45° Period (T_{45}) ms	135° Period (T_{135}) ms	Damping % $\frac{(T_{45} - T_{135})T_{90}}{2 T_{45} \times T_{135}}$
50.00 x 15^4	45.11 x 15^6	6.5887	6.8214	6.3833	0.0380
47.00	42.05	6.5842	6.8753	6.3783	0.0373
44.00	39.00	6.5819	6.8785	6.3719	0.0380
41.00	35.98	6.5715	6.8609	6.3658	0.0372
38.01	33.03	6.5638	6.8597	6.3581	0.0377
35.00	30.13	6.5574	6.8420	6.3500	0.0371
32.00	27.28	6.5526	6.8173	6.3416	0.0359
29.00	24.38	6.5484	6.7902	6.3300	0.0350
25.97	21.33	6.5428	6.8033	6.3277	0.0361
23.00	18.35	6.5367	6.8316	6.3548	0.0386
20.15	15.50	6.5298	6.8810	6.2992	0.0438

Table 4
COMPUTED DISSIPATION DUE TO ONE SCREW

90° Displ. Ampl. dB Re 2×10^{-6} in.	work done/cycle, dB, Re 5.03×10^{-9} in-lb			Dissipation/cycle, dB, Re 5.03×10^{-9} in.		
	Configuration			1 bolt		2 bolts
	P6	P5	P4	P6-P5	P5-P4	P6-P4
45	47.61	47.51	47.45	0.10	0.16	0.25
42	44.62	44.53	44.47	0.09	0.06	0.15
39	41.65	41.55	41.50	0.10	0.15	0.25
36	38.66	38.55	38.51	0.11	0.04	0.15
33	35.62	33.53	35.49	0.09	0.04	0.13
30	32.56	32.48	32.43	0.08	0.05	0.13
27	29.50	39.42	29.36	0.08	0.06	0.14
24	26.49	26.39	26.31	0.10	0.08	0.18
21	23.46	23.35	23.32	0.11	0.03	0.14
18	20.48	20.38	20.32	0.10	0.06	0.16
15	17.47	17.36	17.32	0.11	0.04	0.15
% Total				=	1%	0.6%
						1.6%

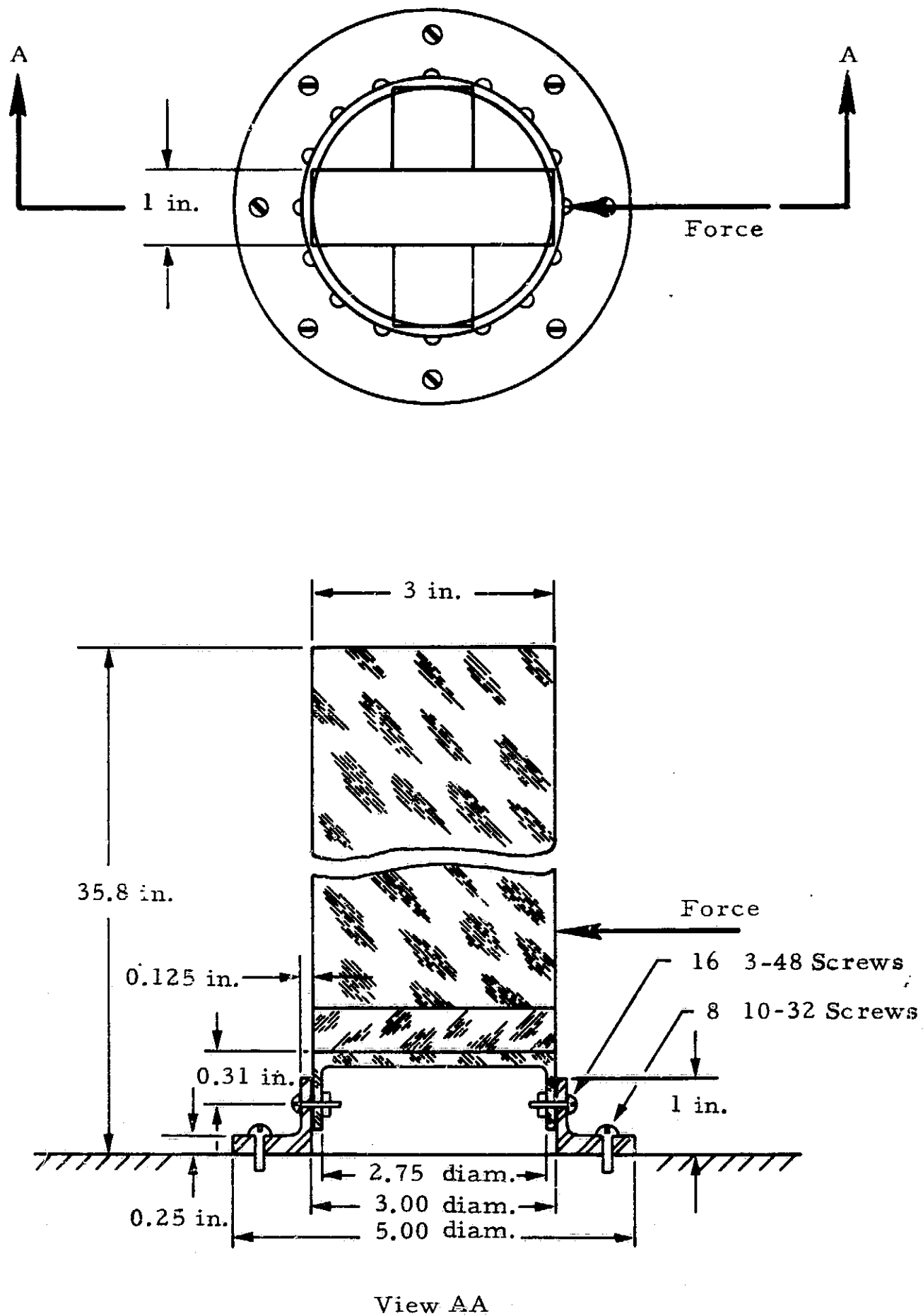


Fig. 22 - Test Specimen for Determination of Damping of One Bolt in Multiple Bolt, Cylindrical Joint

Table 5a

THREE-DIMENSIONAL JOINT DAMPING TEST DATA (ALL SCREWS ON)

Force Ampl. dB Re 8×10^{-4} lb	90° Displ. Ampl. dB Re 2×10^{-6} in	90° Period (T_{90}) ms	45° Period (T_{45}) ms	135° Period (T_{135}) ms	Damping % $\frac{(T_{45} - T_{135})T_{90}}{2 T_{45} \times T_{135}}$
62.00	38.49	6.9763	7.2835	6.6774	4.34
59.01	35.78	7.0033	7.2724	6.6514	4.49
56.00	33.15	7.0085	7.2647	6.6272	4.64
53.00	30.60	7.0090	7.2669	6.6018	4.85
50.00	28.20	7.0120	7.2521	6.5558	5.40
47.00	25.49	6.9998	7.2506	6.5747	4.96
44.00	22.81	6.9913	7.2135	6.7486	3.33
41.00	20.05	6.9814	7.2402	6.7183	3.74
38.00	17.31	6.9859	7.2030	6.7895	2.95
35.00	14.48	6.9860	7.1881	6.7962	2.80
32.00	11.68	6.9681	7.1844	6.8054	2.70

Table 5b

THREE-DIMENSIONAL JOINT DAMPING TEST DATA
(SCREWS, AT 90 AND 270 DEGREES, REMOVED)

Force Ampl. dB Re 8×10^{-4} lb	90° Displ. Ampl. dB Re 2×10^{-6} in	90° Period (T_{90}) ms	45° Period (T_{45}) ms	135° Period (T_{135}) ms	Damping % $\frac{(T_{45} - T_{135})T_{90}}{2 T_{45} \times T_{135}}$
62.00	38.37	6.9990	7.2864	6.6729	4.42
59.00	35.81	7.0267	7.2865	6.6534	4.58
56.00	33.36	7.0248	7.2659	6.6284	4.64
53.00	30.81	7.0203	7.2582	6.6106	4.73
50.00	28.21	7.0194	7.2634	6.5756	5.05
47.00	25.46	7.0117	7.2515	6.5814	4.96
44.00	22.86	7.0043	7.2437	6.5959	4.74
41.00	20.11	7.0009	7.2410	6.7753	3.32
38.00	17.32	6.9794	7.1991	6.7882	2.93

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Table 5c

THREE-DIMENSIONAL JOINT DAMPING TEST DATA (TWO MORE)
SCREWS, AT ZERO AND 180 DEGREES, REMOVED)

Force Ampl. dB Re 8×10^{-4} lb	90° Displ. Ampl. dB Re 2×10^{-6} in	90° Period (T_{90}) ms	45° Period (T_{45}) ms	135° Period (T_{135}) ms	Damping % $\frac{(T_{45} - T_{135})T_{90}}{2 T_{45} \times T_{135}}$
62.00	41.06	7.2046	7.4637	6.8379	4.41
59.00	38.26	7.1912	7.4452	6.8270	4.37
56.00	35.33	7.1804	7.4379	6.8974	3.78
53.00	32.47	7.1709	7.4235	6.9448	3.32
50.00	29.64	7.1672	7.4276	6.9591	3.24
47.00	26.84	7.1587	7.4115	6.9491	3.21
44.00	24.06	7.1522	7.3995	6.9527	3.10
41.00	21.89	7.1446	7.4064	6.9455	3.20
38.00	18.52	7.1401	7.4154	6.9481	3.23

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Table 5d

THREE-DIMENSIONAL JOINT DAMPING TEST DATA
(TWO MORE SCREWS, AT 45 AND 315 DEGREES, REMOVED)

Force Ampl. dB Re 8×10^{-4} lb	90° Displ. Ampl. dB Re 2×10^{-6} in	90° Period (T_{90}) ms	45° Period (T_{45}) ms	135° Period (T_{135}) ms	Damping % $\frac{(T_{45} - T_{135})T_{90}}{2 T_{45} \times T_{135}}$
62.00	41.72	7.2747	7.5349	6.9443	4.10
58.98	38.89	7.2612	7.5167	7.0149	3.45
56.01	35.84	7.2457	7.5121	7.0266	3.33
53.00	32.34	7.2356	7.4937	7.0275	3.20
50.00	30.05	7.2286	7.4978	7.0261	3.23
47.00	27.21	7.2198	7.4928	7.0135	3.29
44.00	24.37	7.2094	7.4693	7.0100	3.16
41.00	21.51	7.2076	7.4476	6.9900	3.16
38.00	18.71	7.1941	7.4207	7.0133	2.81

C

Table 5e

THREE-DIMENSIONAL JOINT DAMPING TEST DATA
(TWO MORE SCREWS, AT 45 AND 315 DEGREES, REMOVED)

	Force	90° Displ.	90° Period	45° Period	135° Period	Damping %
	Ampl. dB Re 8×10^{-4} lb	Ampl. dB Re 2×10^{-6} in.	(T_{90}) ms	(T_{45}) ms	(T_{135}) ms	$\frac{(T_{45} - T_{135})T_{90}}{2 \times T_{45} \times T_{135}}$
53	62.00	42.34	7.3639	7.6260	7.1310	3.35
	59.01	39.47	7.3475	7.6138	7.1226	3.32
	56.00	36.38	7.3333	7.5980	7.1183	3.25
	53.00	33.39	7.3154	7.5781	7.1127	3.15
	50.00	30.46	7.3005	7.5754	7.0960	3.25
	47.00	27.60	7.2926	7.5628	7.0927	3.19
	44.00	24.76	7.2847	7.5640	7.0779	3.30
	40.99	21.95	7.2817	7.5637	7.0733	3.33
	38.00	19.17	7.2739	7.5263	7.0705	3.11

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Table 5f

THREE-DIMENSIONAL JOINT DAMPING TEST DATA
(ALL SCREWS ON, REPEATABILITY CHECKED)

Force Ampl. dB Re 8×10^{-4} lb	90° Displ. Ampl. dB Re 2×10^{-6} in	90° Period (T_{90}) ms	45° Period (T_{45}) ms	135° Period (T_{135}) ms	Damping % $\frac{(T_{45} - T_{135})T_{90}}{2 \times T_{45} \times T_{135}}$
62.00	38.16	6.9711	7.2431	6.5938	4.73
59.01	35.76	6.9752	7.2331	6.5710	4.85
56.00	33.16	6.9753	7.2226	6.5385	5.05
53.01	30.59	6.9725	7.2074	6.5207	5.09
50.00	27.92	6.9650	7.1921	6.5016	5.14
47.00	25.26	6.9618	7.2076	6.4755	5.46
44.00	22.57	6.9617	7.1962	6.7287	3.36
41.00	19.84	6.9510	7.1773	6.6982	3.46
38.00	17.09	6.9478	7.1510	6.7364	2.98

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Table 6

COMPUTED DISSIPATIONS DUE TO VARIOUS SCREWS IN MULTIPLE-BOLT, CYLINDRICAL JOINT

	Response Amplitude, dB Re 2×10^{-6} in.	Total Work Done/Cycle dB Re 5.03×10^{-9} in-lb					Dissipation, dB, Re 5.03×10^{-9} in-lb Due to Screw Nos.			
		C16	C14	C12	C10	C8	5&13	1&9	7&11	3&14
55	38	49.75	49.82	51.47	51.14	51.83	+0.07	1.65	-0.33	0.69
	35	46.60	46.59	48.37	48.06	48.76	-0.01	1.78	-0.31	0.70
	33	44.42	44.32	45.33	45.07	45.81	-0.10	1.01	-0.26	0.74
	30	41.20	41.09	42.26	42.03	42.80	-0.11	1.17	-0.23	0.77
	27	38.90	38.89	39.18	39.97	39.77	+0.01	0.27	+0.79	-0.20
	24	35.75	35.77	36.08	36.89	36.70	-0.02	0.31	+0.81	-0.19
	21	32.59	32.57	33.97	33.81	33.62	-10.02	0.40	-0.16	-0.19
	18	30.47	30.45	30.55	30.74	30.53	+10.02	0.10	+0.19	-0.21
	15	27.34	27.34	27.74	27.64	28.41	+0.00	0.40	-0.10	+0.77

It is interesting to note also that, for the first time in the series of tests, data scatter was noticeable. This is rather significant because identical instrumentation and test procedures were used for the planar joint test where no noticeable scattering of data was observed. The causes must, therefore, be associated with the variation of local joint conditions introduced into the setup when an appreciable number of screws were used. The implication of this conclusion is that unless stringent assembly procedures (far exceeding those required to insure a sound structure) are used in a realistic structural joint, it will be rather difficult to predict overall joint damping from analyses (or test data) obtained for damping associated with a single fastener joint.

Section 7 CONCLUSIONS

Initial steps have been taken in a systematic study of joint contact damping. A new experimental technique has been developed to observe the area of contact between two elastic bodies. Since it does not depend on symmetry of the specimen about either the contacting surface or the axis through the center of contact, the optical method of measuring contact area is completely general and can be used in other applications such as in contact stress studies between non-spherical bodies, etc. It is anticipated that with suitable development, this approach can provide a simple approach toward obtaining engineering results related to the contact damping problem.

Plate interface separation tests indicate that local stress and strain conditions near a fastener cannot be efficiently determined by measurements at remote locations, as originally desired.

Experimental evidence indicates that local conditions which intimately govern the state of contact near each fastener in a multiple-fastener joint are extremely critical with regard to the ability to predict joint damping from analytical considerations which do not include second order effects, or from experimental results of single fasteners.

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