TECHNICAL NOTE AST-278

NASA-CR-98385

TECHNICAL NOTE AST-278

FEASIBILITY STUDY OF EXPERIMENTAL METHODS FOR JOINT DAMPING ANALYSIS

by E. J. Rodgers
B. D. Liaw
R. G. Sturm

August 1968

RESEARCH LABORATORIES

BROWN ENGINEERING

A TELEDYNE COMPANY

Research Park • Huntsville, Alabama 35807

TECHNICAL NOTE AST-278

FEASIBILITY STUDY OF EXPERIMENTAL METHODS FOR JOINT DAMPING ANALYSIS

By

E. J. Rodgers, Ph. D.
B. D. Liaw, Ph. D.
R. G. Sturm, Ph. D.

August 1968

Prepared For

STRUCTURES DIVISION
PROPULSION AND VEHICLE ENGINEERING LABORATORY
GEORGE C. MARSHALL SPACE FLIGHT CENTER

Contract NAS8-20073

Prepared By

RESEARCH LABORATORIES
ADVANCED SYSTEMS AND TECHNOLOGIES GROUP
BROWN ENGINEERING COMPANY, INC.
HUNTSVILLE, ALABAMA

ABSTRACT

Results for the experimental feasibility investigation of a damping test apparatus using a structural bolted joint are presented. Experimental data as well as the test setup and the overall recording arrangement are described. Despite certain deficiencies that existed in the test machine, the results obtained seem adequate for answering some important questions set forth before the test program.

Recommendations for improvements in the equipment and the procedures for future test program are indicated and discussed.

Approved:

R C Watson Jr

R. C. Watson, Jr.
Vice President
Advanced Systems and Technologies Group

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

INTRODUCTION

Studies conducted by the Research Laboratories of the Brown Engineering Company, Inc., for the Structures Division of the Propulsion and Vehicle Engineering Laboratory, George C. Marshall Space Flight Center, indicated that in some cases the damping occurring at the joints and interstages of a vibrating missile-like structure was much larger than that occurring in the material portion of the structure (Refs. 1 and 2). Although many studies have been conducted on the damping capacity of various materials, little has been done in investigating the damping of structural joints (Ref. 1).

In order to assess the effects of damping on the vibrational characteristics of missile-like structures or to take advantage of the damping in the design of the structure to control the vibrational characteristics, a better understanding of joint damping is required. A program to obtain this understanding, which would also lead to advances in structural design, was outlined by Brown Engineering (Ref. 2). An essential part of the program was the experimental determination of joint damping characteristics.

This report discusses the results of feasibility tests, using a testing machine designed and built at Brown Engineering. These tests were aimed at determining whether or not experimental results could be obtained that could give data on the damping occurring at a joint as well as information as to the mechanism on which the damping is dependent. The joint tested was a double-cover-plate simple joint fastened with four 3/8-inch bolts.

DAMPING INVESTIGATION

PURPOSE

The purpose of this investigation is to study the feasibility of making measurement for determining the joint damping characteristics, to conduct tests on a typical bolted joint, and to investigate the behavior of structural joints subjected to cyclic loading conditions.

TEST SETUP

Test Machine

The test machine consists of two major parts; the steel frame and the loading mechanism, as shown in Figure 1. The frame was reconstructed from an old shaker frame carrying a large variable speed motor (50 HP) and belt drive arrangement. The modification consisted of extending the frame and installing transverse beams (not visible in Figure 1) whose resistance to bending could be modified by neoprene snubbers adjustable by the two bolts visible in Figure 1 on each side of the center.

The loading mechanism consists of an eccentric driven by the motor and belt drive. The eccentric in turn moves a slide-guided arm which is rigidly attached to the specimen assembly which in turn is also held and guided in a horizontal motion by adjustable gibbs. The specimen, shown in Figure 2, is moved back and forth by the eccentric when the motor is run. A threaded steel tail rod is attached to the other end of the specimen assembly and extends through holes in the transverse beams. Double nuts on each side of each beam are used for applying the load to the specimen. The amount of load can be controlled by either of the following:

- Changing the magnitude of the eccentric throw
- Using only one or both transverse beams
- Snubbing the beams by the neoprene snubbers
- Any combination of these adjustments.

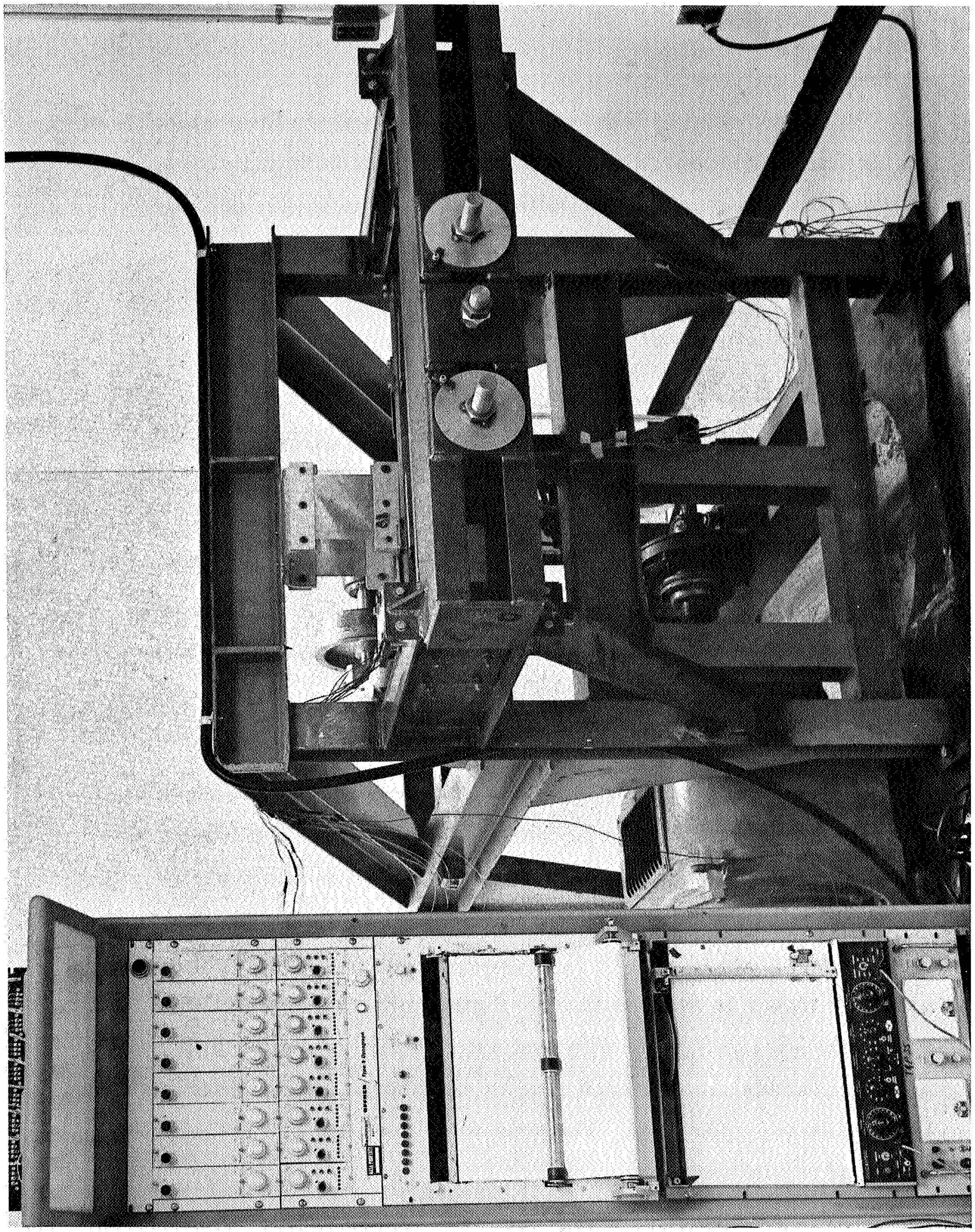

FIGURE 1. TEST MACHINE

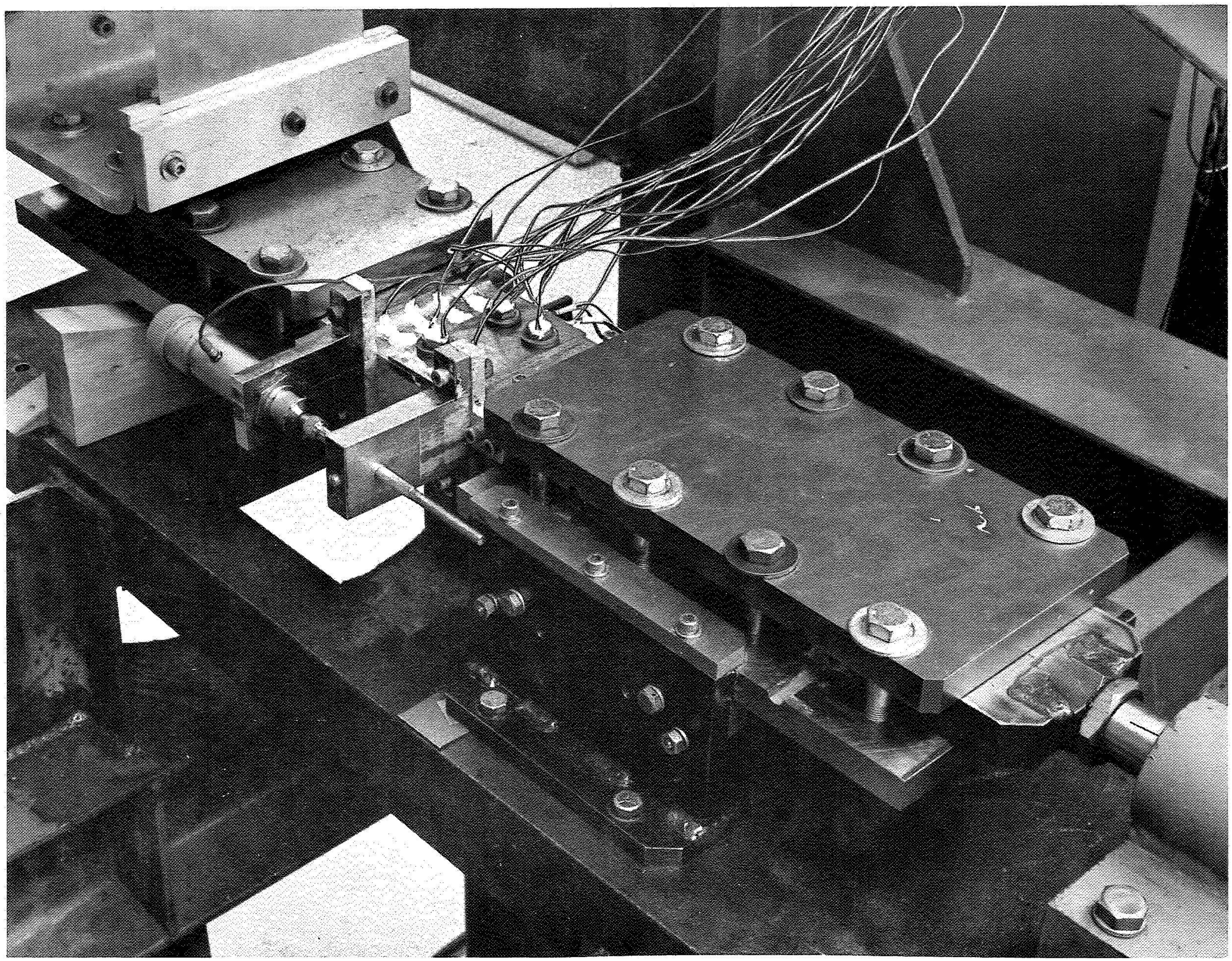

FIGURE 2. TEST SPECIMEN

While the motor had variable speed, the slowest speed actuated the specimen assembly at about 3 Hz. A different belt pulley combination can be used to operate at lower frequencies. The higher frequencies can be obtained by higher motor speeds.

Test Specimen

The specimen was a steel double-cover-plate butt joint with four 3/8-inch bolts. Strain gages were mounted in holes drilled in the bolts from the head end to a depth of 5/8 inch. The strain gages were located at 1/2 inch from the end or about 1/4 inch into the shank of the bolt. The loadings on the bolts were determined by the outputs of strain gages and were measured by a strain indicator. Conditions under three different bolt tensions were investigated; i.e., 25,500; 33,600; and 40,200 psi. Strain gages were mounted on the specimen and were arranged according to Figure 3. A Photocon proximity transducer (Model PT-5) was installed on one side of the specimen, as shown in Figure 2, for relative displacement measurements. Output data were recorded by an eight channel-strip recorder. One channel was reserved for recording the output from the proximity transducer while seven channels were used for recording the outputs of strain gages. No attempt has been made to measure the load transmission through the joint in the longitudinal direction at this time.

The machine eccentric double throw was set to give two different stroke lengths, namely, 13/64 and 9/32 of an inch, which are about equivalent to 3,750 and 5,000-pound loads, respectively, when only the solid beam was used. The motor speed was set for a frequency range of approximately 2.90 to 6.0 Hz.

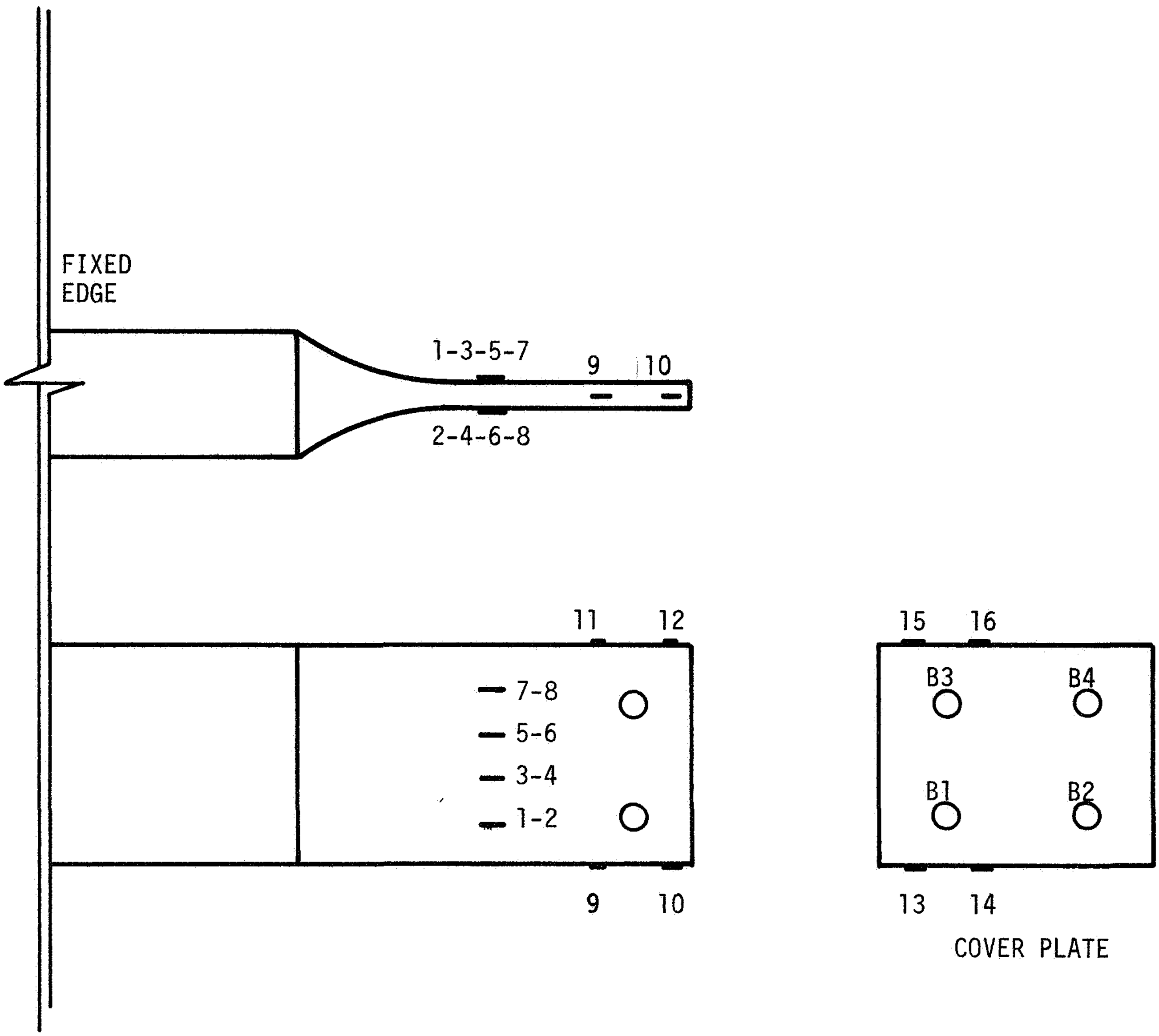

FIGURE 3. STRAIN GAGE ARRANGEMENT

DATA ACQUISITION

Recorder Outputs

A Beckman eight-channel strip recorder was used to record outputs from strain gages and the proximity transducer. The strain gages used were BLH SR-4 foil gages. The output from the strain gages were measured as the unbalance of a Wheatstone bridge, one for each gage, made up with fixed 350 ohm resistors. These resistors were not temperature compensated.

The proximity transducer has its own power supply unit, which is completely solid state to minimize the size, weight, and power consumption. The output from the proximity transducer was fed through an amplifier and into the first channel of the strip recorder.

In addition to the strip recorder, a Hewlett-Packard Model 7000 AR X-Y recorder was also installed to obtain the qualitative picture of the hysteresis loops. Parallel with the connection to the strip recorder, the outputs of the proximity transducer and one of the strain gages were fed into the X-Y recorder. The results for the 3, 750-pound load condition were shown qualitatively in Figure 4 for various frequencies as indicated beneath each set of loops. The first row was for the case with bolt tension of 25, 500 psi, the second row was for the case with bolt tension of 33, 600 psi, and the last row was for the case with 40, 200 psi bolt tension. It can be seen that the speed of the X-Y recorder was not high enough to follow the motion of the machine.

A sample of the strip recorder output is shown as Figure 5. The output on each channel is shown in Table 1. The traces reflect the type of loading to which the specimen assembly was subjected, and the trace of Channel 1 for the proximity transducer represents the response of the joint in terms of the relative displacement across the joint. Three variables were considered in the data acquisition program; namely,

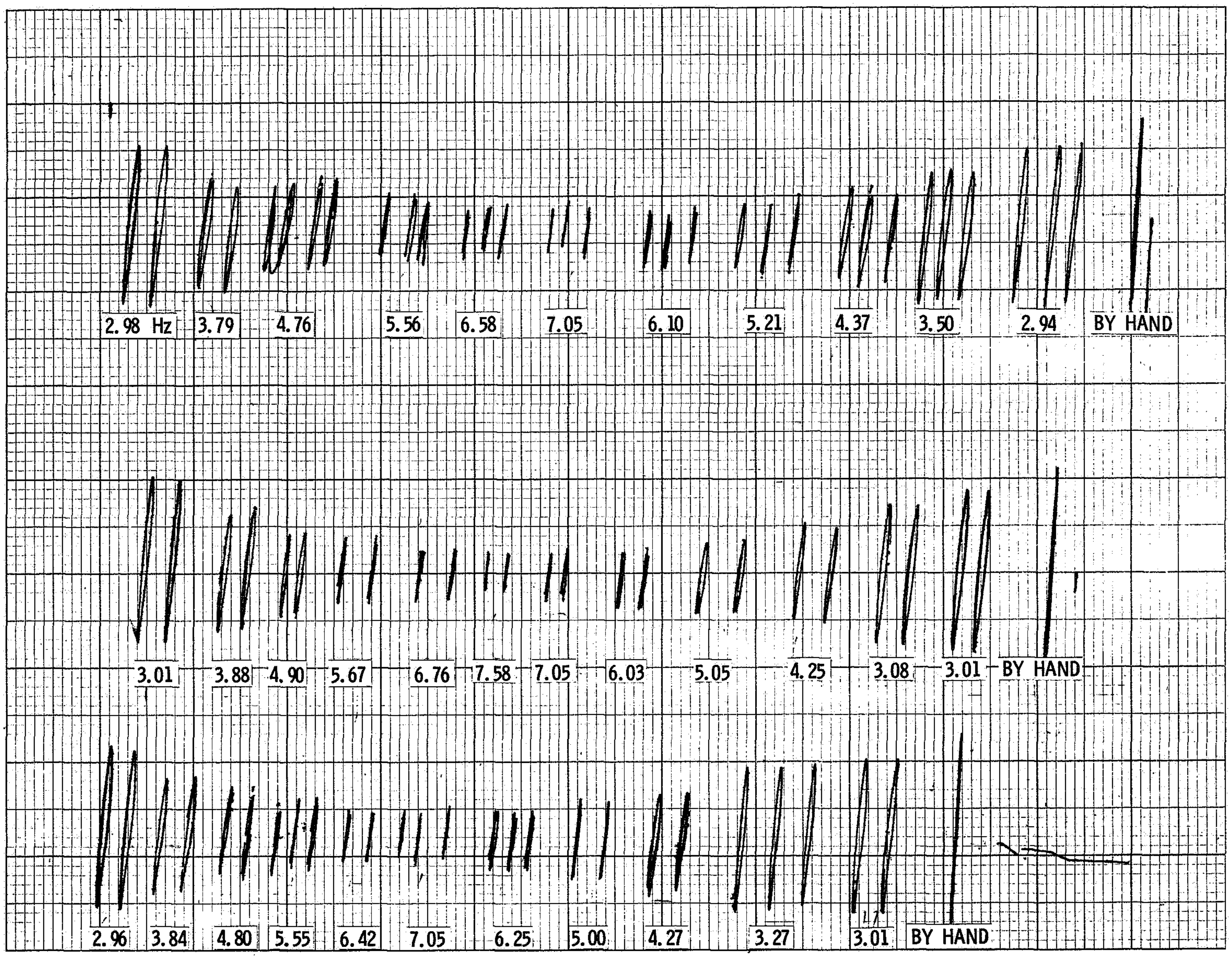

FIGURE 4. X-Y PLOTTER RESULTS

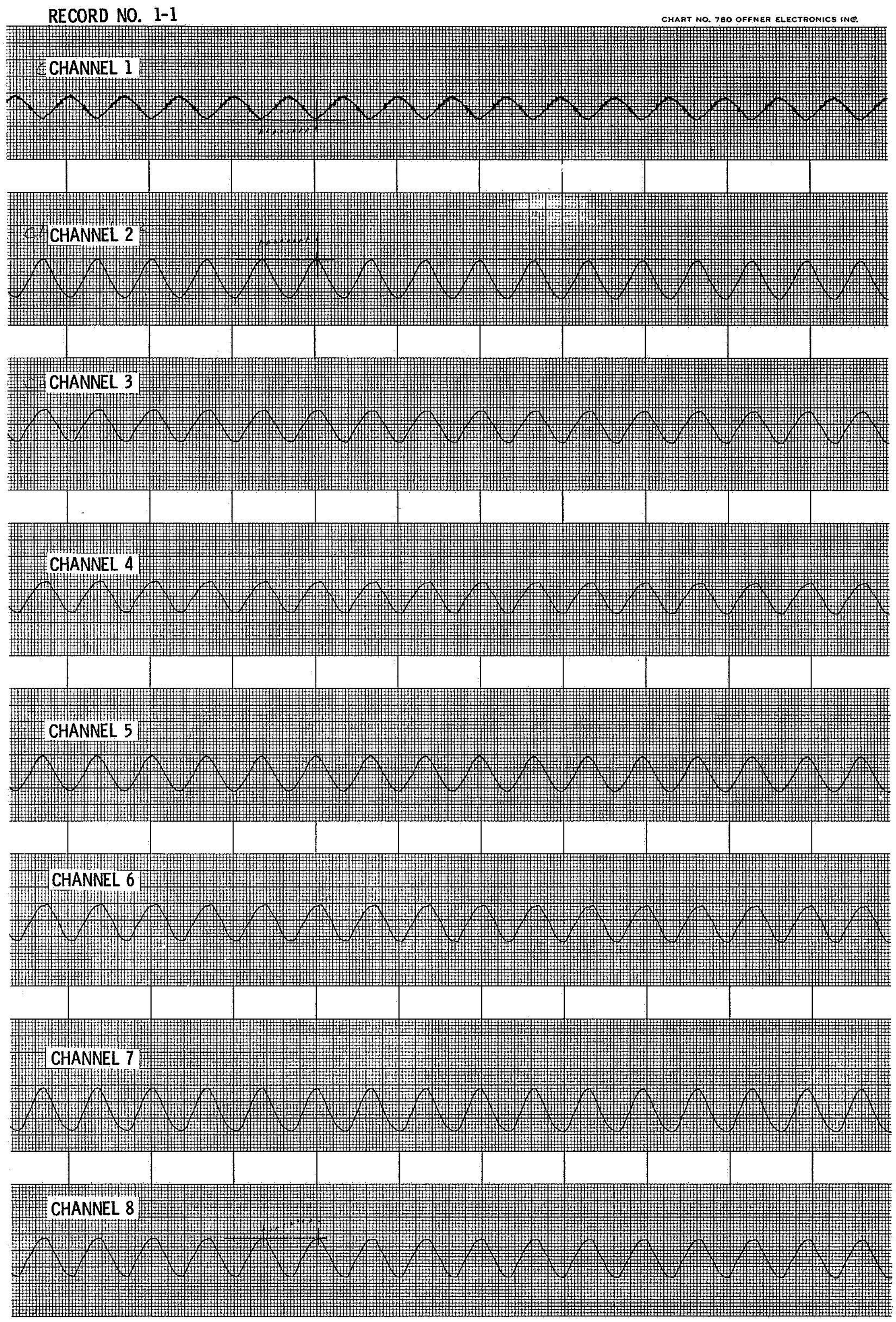

FIGURE 5. SAMPLE ORIGINAL DATA

TABLE 1. CHANNEL OUTPUT

Channel	Output
1	Proximity Gage
2	Strain Gage 4
3	Strain Gage 3
4	Strain Gage 5
5	Strain Gage 6
6	Strain Gage 7
7	Strain Gage 2
8	Strain Gage 1

motor speed, bolt tension, and the load. The load was given by different combinations of stroke length and the transverse beams. Thus, each data "point" is considered to represent the relative displacement across the joint and the corresponding stresses recorded by Strain Gages 1 through 7 at a certain frequency for a specified bolt tension and load.

Procedure

The procedure for the data acquisition can be described as follows:

- Adjust the eccentricity at the cam to obtain desired stroke length and pick the combination of transverse beams so that the operation can be obtained.
- Fasten the joint by tightening the bolts with the help of a strain indicator so that uniform bolt tension can be obtained.
- Run the machine at the lowest speed and then gradually increase the speed while the machine is running to get data at specified frequency ranges.
- Retighten the joint to increase bolt tension.
- Run the machine for the same frequency range.

The same process was repeated until data for all bolt tension desired at the particular load were obtained. The eccentricity at the cam was then adjusted to a new value of stroke length, or a different combination of transverse beams was used, so that a new loading condition was obtained. The same process was then repeated.

Every time the joint was reassembled, the relative position of the bolts, or the clearances between the bolts and holes, may have been changed. This fact was believed to play an important part in the joint response to the loads to which they were subjected. The detail influence will be discussed later by examining the hysteresis loops.

DATA REDUCTION

The first step in data reduction was to convert the electronic voltage outputs into real mechanical units. This was done through proper calibration of the recording devices. The original trace records were then read and tabulated, as illustrated in Table 2. Stresses were calculated in terms of psi and the relative displacements in terms of inches. These data were then plotted as shown in Figures 6a through 6d, which were typical among those obtained in this test program. The area enclosed between the loading and unloading branch of the stress-displacement curves represents the energy lost for the particular cycle. The area was measured by a planimeter and converted into mechanical units which in this case was inch-pound per squared inch of the cross section area of the specimen.

Loss Factor

In order to make comparison with existing data on damping, a certain standard energy loss unit was adapted. By carefully examining the loops obtained, certain analogies could be observed between these loops and those of material damping. For instance, the slopes of both loading and unloading branches were equal at zero stress level and could be identified as the "joint stiffness". This is analogous to the Young's modulus of elasticity. The increasing rate of displacement or "slip" at the high stress level is analogous to the phenomenon of "plastic deformation" in the material damping. There are additional characteristics at high stress levels, which are not observed in the material damping. These will be discussed later under the section on joint behavior. Nevertheless, these additional characteristics do not prevent one from attempting to make comparison between the joint damping energy loss and the material damping energy loss. Thus, the loss factor which has been commonly used in representing material damping energy loss is used here.

TABLE 2. A SAMPLE DATA SHEET (RECORD NO. 6-2-1)

Station	Stress at Gage 1 Reading in mm	Stress at Gage 1 psi	Relative Displacement Reading in mm	Relative Displacement 10^{-3} in.
1	0.00	0	0.00	0.000
2	2.50	3,180	2.40	0.874
3	5.50	6,980	4.10	1.490
4	8.10	10,300	6.00	2.180
5	11.00	14,000	7.20	2.620
6	13.15	16,700	8.50	3.090
7	14.10	17,900	9.10	3.320
8	13.60	17,300	8.80	3.200
9	10.90	13,850	7.80	2.840
10	7.60	9,650	6.00	2.180
11	4.90	6,230	4.30	1.565
12	2.00	2,540	3.00	1.090
13	6.50	6,350	1.20	0.437
14	0.00	0	0.00	0.000
15				
16				
17				
18				
19				
20				

NOTE:

Amplitude = 9/64 in.
Bolt tension = 40,200 psi
Frequency = 3.90 Hz
Stress conversion factor = 1,270 psi/mm
Displacement conversion factor = 0.364×10^{-3} in/mm

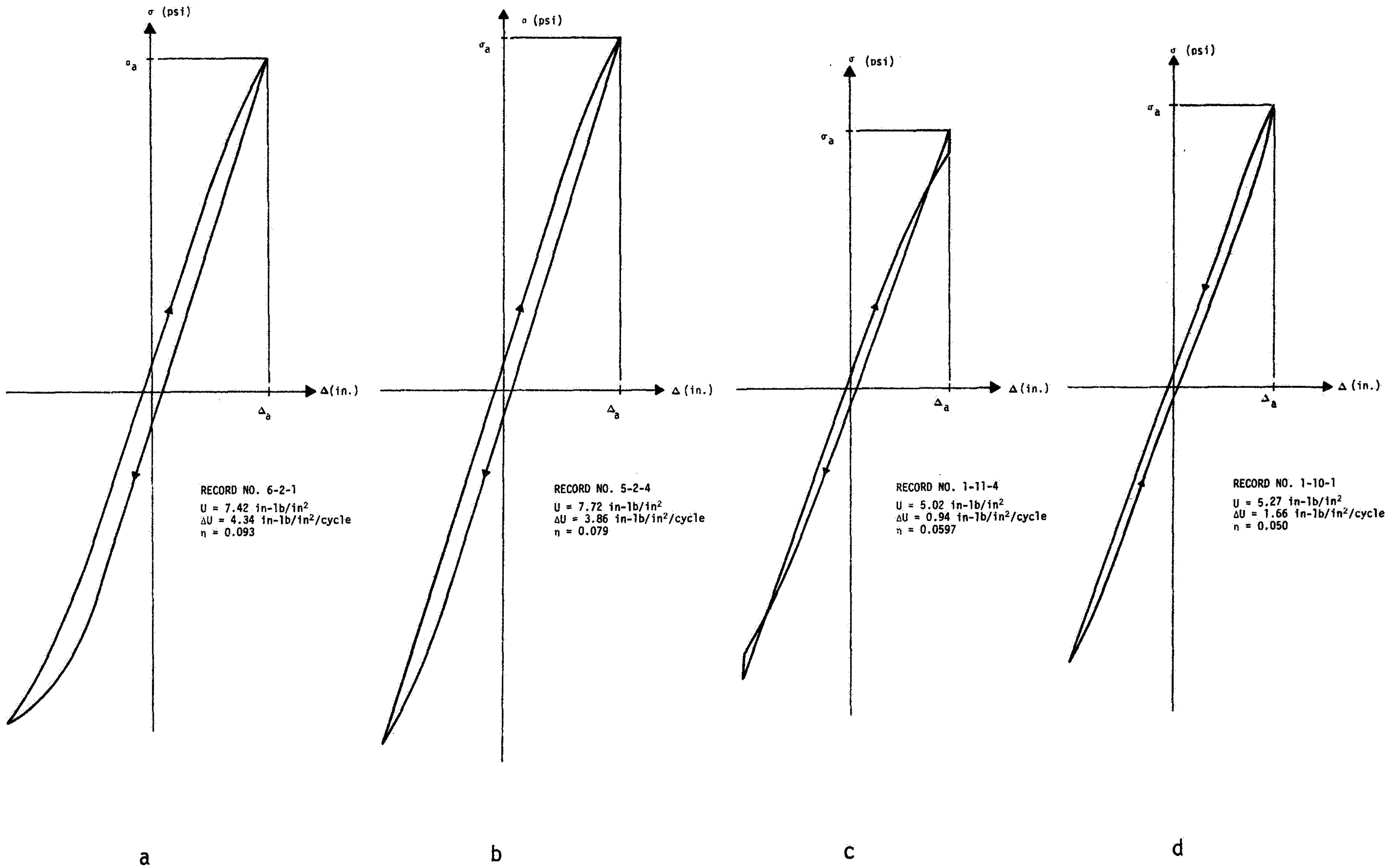

FIGURE 6. A TYPICAL HYSTERESIS LOOP

Let U be the peak strain energy for the material undergoing vibration with stress amplitude σ_a and strain amplitude ϵ_a. Assuming uniform stress distribution, the peak strain energy and the energy loss per cycle ΔU can be defined as

$$U = \frac{1}{2} \int_V \sigma_a \epsilon_a \, dV \qquad (1)$$

$$\Delta U = \int_V \int_\epsilon \sigma \, d\epsilon \, dV \qquad (2)$$

where

V	-	volume
σ	-	stress
ϵ	-	strain.

Let σ and ϵ be sinusoidal functions of time with ϵ lagging σ by the angle η, i.e.,

$$\sigma = \sigma_a \cos \omega t \qquad (3)$$

$$\epsilon = \epsilon_a \cos (\omega t - \eta) \; . \qquad (4)$$

Change Equation 2 to

$$\Delta U = \int_V \int_o^{2\pi/\omega} \sigma \frac{d\epsilon}{dt} \, dt \, dV \; . \qquad (5)$$

Substitution of Equations 3 and 4 into 5 and integrating with respect to time over the period $2\pi/\omega$ of a complete cycle yields

$$\Delta U = \int_V \sigma_a \epsilon_a \pi \sin \eta \, dV \; . \qquad (6)$$

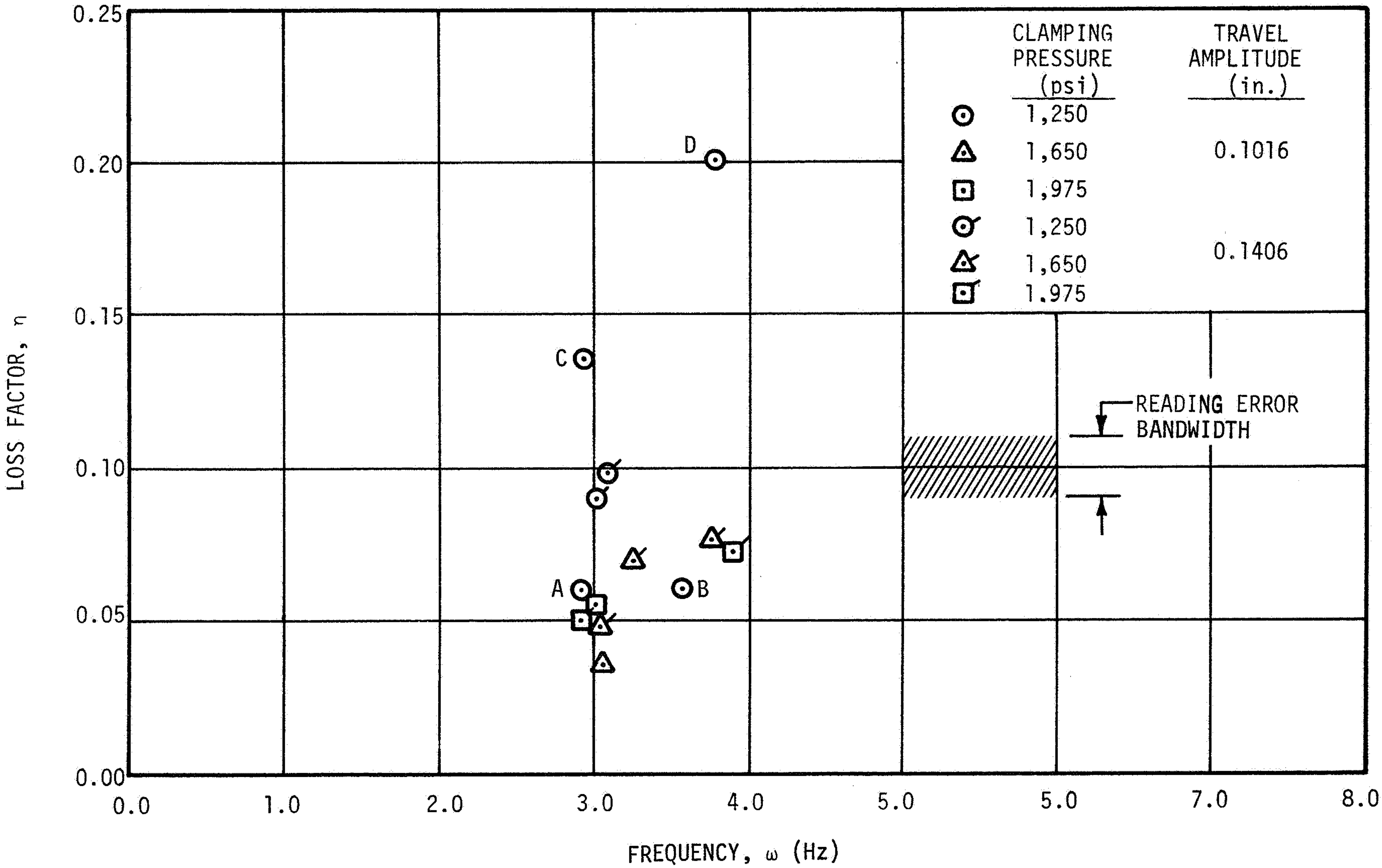

FIGURE 7. LOSS FACTOR AS A FUNCTION OF FREQUENCY

Then, the ratio of the energy loss per cycle to the peak strain during the cycle of vibration is

$$\frac{\Delta U}{U} = 2\pi \sin \eta \quad . \tag{7}$$

For small amounts of damping, the phase angles η are small and values of $\sin \eta$ are approximately equal to η, so that η can be obtained directly from Equation 7 as

$$\eta \simeq \frac{\Delta U}{2\pi U} \quad . \tag{8}$$

This quantity has been defined as the loss factor for the material damping. This definition is now applied to the test results of the current test program. The results for a small frequency range, 3 to 4 Hz, are presented as shown in Figure 7. Each data "point" is the average of the results of the outputs of two strain gages, Strain Gages 1 and 4 (as shown in Figure 3). Higher frequency runs were attempted, but due to certain difficulties occuring in the proximity transducer output device, the traces were not readable.

TEST RESULTS

The test results in terms of loss factor η were shown in Figure 7 for various frequencies. It was planned to obtain data for a frequency range of 1.0 to 8.0 Hz. But, due to the limitation on the motor speed at lower side of the frequency range and certain difficulties occurring in the proximity transducer recording device in higher frequency ranges, only those data, as presented in Figure 7, for a frequency range of 3.0 to 4.0 Hz were considered reliable. Therefore, it is difficult to draw a definite conclusion on the frequency dependency of the loss factor. However, the dependency of the loss factor on the bolt tightness, or rather on the "joint stiffness", is indicated if Points A and B are disregarded. Relation between the "joint stiffness" and the bolt tightness, in terms of clamping pressure, will be discussed later. Data Points A and C, and B and D were obtained approximately under same conditions except A and B were data taken after the high speed runs while C and D were before the high speed runs. The exact cause for this scattering is not known at this moment, but several interesting phenomena have been observed. First of all, the hysteresis loops for Points A and B indicate the presence of the effects of hole clearance, which will be discussed later. Secondly, the temperature was noticeably higher, although not measured, after the high speed runs than before the runs. Hence, Points A and B may be disregarded in considering the effects of joint stiffness on the loss factor.

ERROR ESTIMATION

Estimation of inherent errors in the test machine and electronics recording devices are not attempted at the present time. In the current test program, only the reading error is estimated. The trace outputs can be read within ±0.5 mm, which is equivalent to ±635 psi in stress and $\pm 0.4 \times 10^{-3}$ inches in the relative displacement. These give a possible error approximately equal to ± 0.01 in the loss factor η, based on a representative energy loss of four inch-pound per square inch of the cross section area of the specimen which is approximately equivalent to ± 0.6 degree in phase lag.

JOINT BEHAVIOR

Hysteresis Loops

By examining the hysteresis loops (some typical ones were shown in Figure 6a through 6d) obtained in these tests, certain important characteristics which are analogous to the loops for material hysteresis can be observed. These characteristics may be described as follows:

(1) At low stress level, particularly at the zero stress axis, both loading and unloading branches maintain the same slope which may be defined as the "joint stiffness". Average values of the joint stiffness of obtained loops for each particular value of clamping pressure were calculated and plotted versus clamping pressure as shown in Figure 8. As the clamping pressure increases, the value of the joint stiffness increases and rapidly approaches a point where additional clamping pressure will not change the joint stiffness significantly in practice. However, in theory, it should be asymptotic to a value equivalent to that of a mono-block. Although only one bolted joint specimen was tested, it can be surmised that joint stiffness tends to increase as the surface roughness increases.

(2) Some loops, as typified by Figure 6a, for higher stress levels show a gradual decreasing in rate of change of stress with respect to displacement. This behavior seems analogous to the plastic deformation in material hysteresis loops.

Should a mathematical formulation of the stress-displacement relation be attempted, these two analogies can be employed as basis for utilizing the familiar technique in describing the material hysteresis

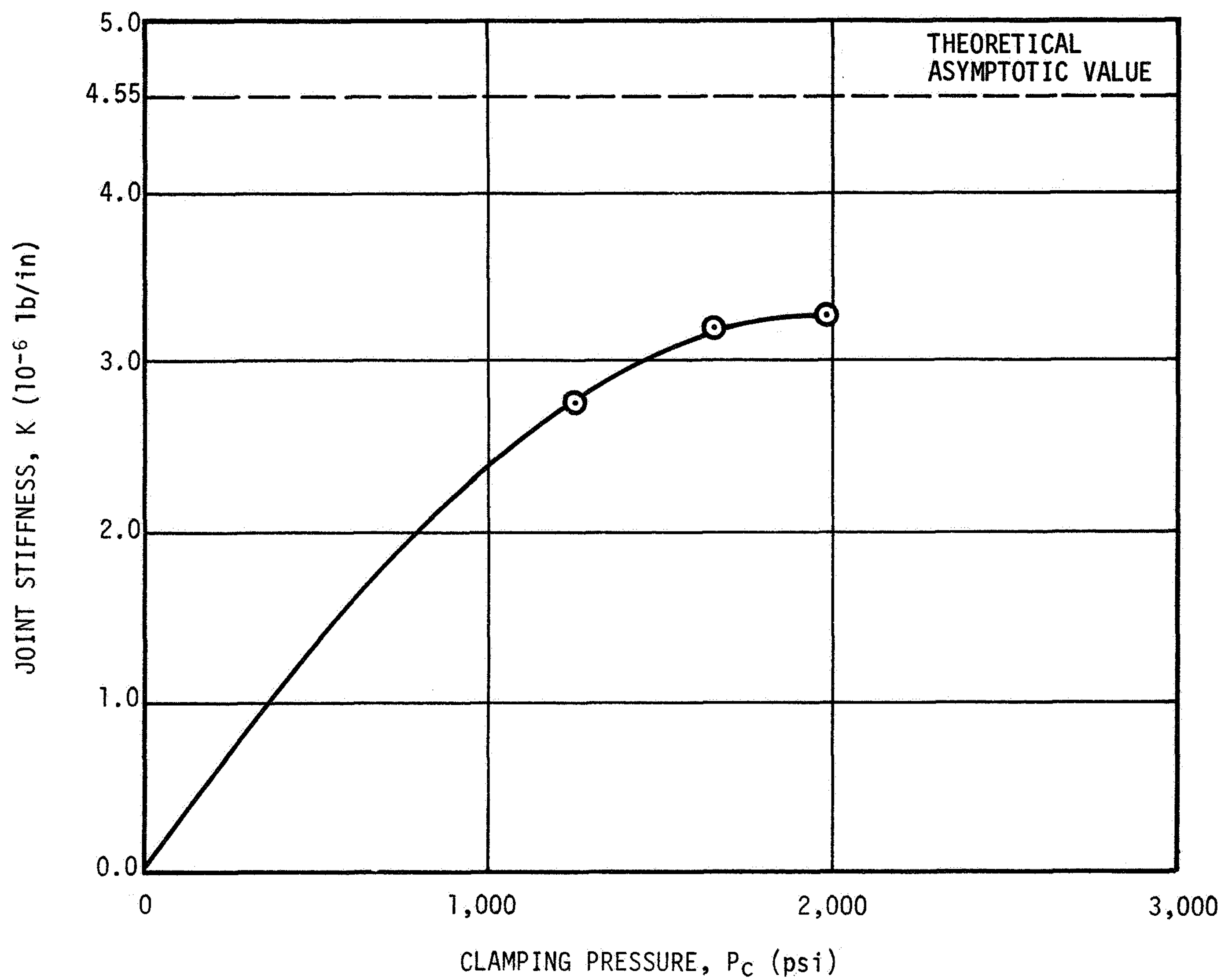

FIGURE 8. JOINT STIFFNESS AS A FUNCTION OF CLAMPING PRESSURE

loops mathematically, to construct the functional relationship between stress, and/or force, and relative displacement across the joint.

Effect of Bolt Clearance

In addition to the two important characteristics described above, certain deviations from the standard material hysteresis loops have been observed at the peak and/or near the peak stress level. The joint behavior at peak or near peak stress level was believed to be affected by the bolt clearance. Assuming all test conditions are maintained the same except the bolt clearance and the clearances at tension and compression sides are equal, the joint behavior described by their hysteresis loops are idealized and shown qualitatively, in Figure 9, in an alphabetical sequence. Figure 9a shows the case for which total displacement does not take up all clearance, or the clearance is largest among these four cases. The slope for the linear part represents the "joint stiffness". Figure 9b represents the case for which the total displacement could be slightly over the clearance and the stress was built up slightly due to the restraint of the hole or bolt. Cases for next larger and the largest amplitudes are shown as Figures 9c and 9d, respectively.

Points A through D shown in Figure 9 represent the following:

A - Starting point of the interface "plastic deformation" or "slip"

B - Bolt and hole edge make contact

C - From B to C represents the elastic deformation of the hole and/or bending of bolts

C - When stress is reversed initially, the elastic rebound is the main action, C to D, and then gradually the rate is reduced by the interface friction.

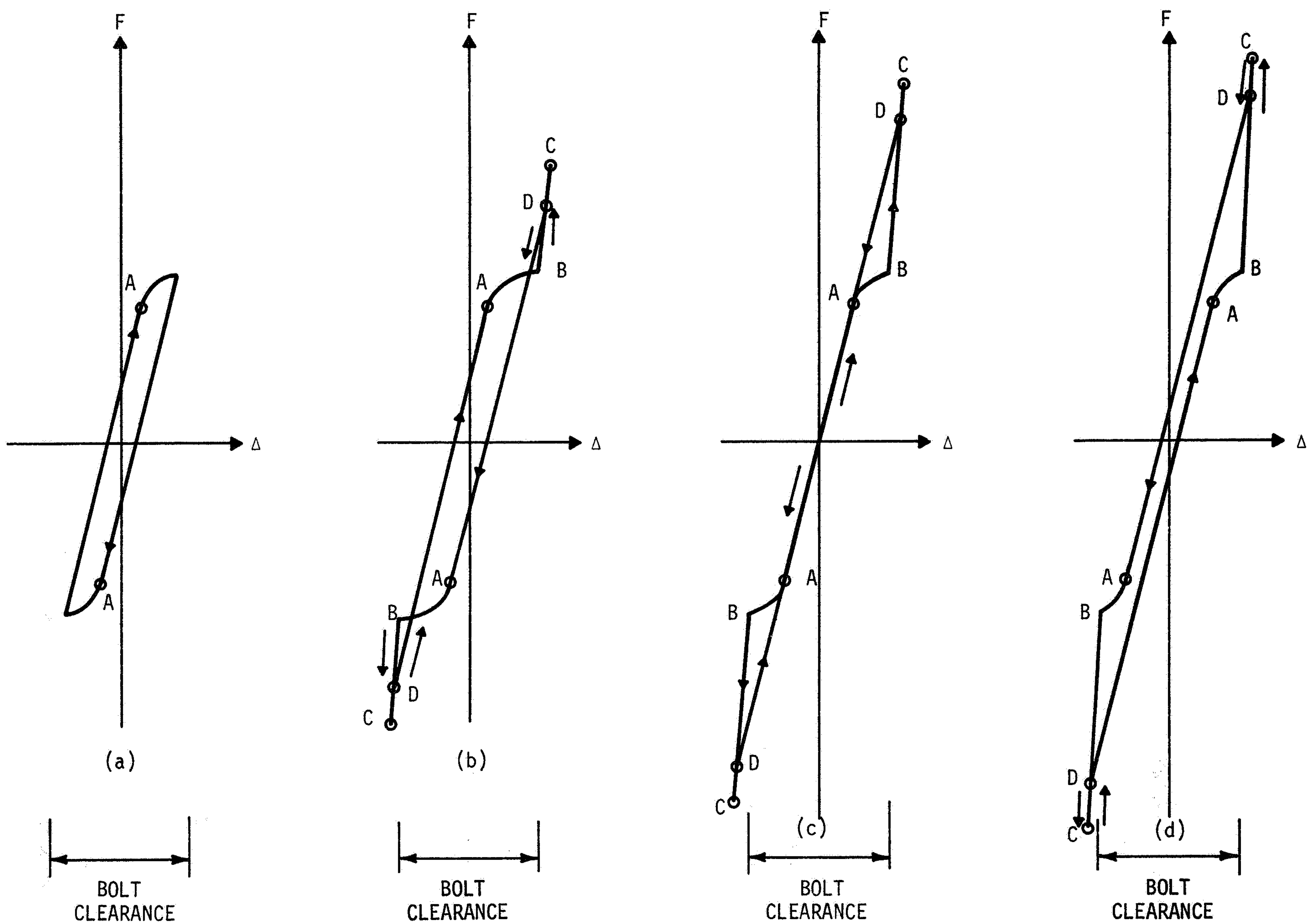

FIGURE 9. FOUR TYPES OF HYSTERESIS LOOPS FOR JOINTS

The case, Figure 9a, gives a loop shape which is most commonly obtained in test results and may be interpreted in the classical theory in structural and/or material damping. The other three cases in some instances represent "energy gain" instead of "energy loss". However, if one tries to locate the source of the energy, he will realize that this "energy gain" of the joint actually represents the "excessive energy" put into joint over and above what the joint was able to dissipate; consequently, this "excessive energy" has to feed back to the structure and builds up stress at the vicinity of the joint, which may in time cause local structural failure.

In addition to the error caused by the deficiency of the loading mechanism, the unevenness of the bolt clearances in the test specimen may be partially responsible for the nonuniform stress distribution across the width of the specimen, as shown in Figure 5. This type of error was not controlled and results can be analyzed only statistically.

PROBLEMS AND IMPROVEMENTS

The major problems encountered in the first damping tests as reporting in Reference 2 were the misalignments in the specimen and the wearing of the drive yoke mechanism. The aluminum block actuator of the drive yoke was replaced before current tests started. The misalignment problems continued to be the dominant trouble in the beginning of the current test program. The problem was so serious that an acceptable zero could not be obtained and maintained. It was found that some galling of the gibs guiding the clamping plates for the specimen has occurred. This was corrected by replacing the gibs by a double-plate fulcra using shim stock as the flex plates. Before each run, the joint was disassembled and the alignment across the joint was checked. Then, the joint was reassembled and bolts were tightened, using a strain indicator, to the desired bolt tension in each bolt. This effect resulted in obtaining the acceptable zero.

Once the acceptable zero was obtained and its precision could be maintained, the next step was trying to eliminate the misalignment of the loading shaft with the loading beams. By careful measurement, it was found that either the loading shaft was not perpendicular to the loading beam, or the loading beams were not properly assembled. It was further observed that only a few thousandths of an inch gap between the nuts and the beams could produce a bending stress at the specimen of the order of 1, 000 to 3, 000 psi. This problem could not be solved completely without reassembly of the beam loading mechanism. Thus, it was decided that only the solid beam would be used by shimming between the nuts and the beam with shim stocks to minimize the eccentricity of loading.

There were some problems encountered in the recording devices. First of all, the proximity transducer was installed originally at an unfavorable position where the effect of possible bending was magnified by the gage mounting. Since a second transducer was not available, it was then decided that the transducer be moved to one side of the specimen in order to minimize the possible bending effects. Secondly, the head plate available for the proximity transducer was 1/2-inch maximum diameter which did not give enough capacitance necessary for tuning the Dynagage (it needs 2 to 30 mmf for proper tuning) while still maintaining the proper linearity for calibration and recording. After certain adjustments were made, the proximity transducer worked properly at lower frequency range (below 4 Hz). Some noise, which looked like 120 cycle hum, was encountered on the recorded outputs. These were greater at higher frequencies. Data obtained for high frequency range, about 4 Hz, were not readable with acceptable reliability.

The lowest speed of the motor gave load cycle about 3.0 Hz so thât data for lower frequencies were not obtainable. The motor speed should be reduced to about 1 Hz for the frequencies of interest. An alternative would be to change the drive pulleys so the same motor speed would produce a lower frequency range of load cycle.

SUMMARY AND CONCLUSIONS

SUMMARY

The experimental feasibility investigation of a structural bolted joint is completed. Test results as well as the test setup and the overall recording arrangement are presented in this report. The content of this report may be summarized as follows.

The first part was the description of the overall test set-up , procedures of data acquisition, and the data reduction. The test machine was briefly described. The loading mechanism and the load range for the current tests, as well as the problems involved, were discussed. The photographs showing the machine setup and the specimen arrangement were also included. A Hewlett-Packard Model 7000 AT X Y recorder was installed to obtain the general picture of the hysteresis loops and sample output data were shown in Figure 4. The strain gage arrangement and the recording devices were described. The sample data traces were shown in Figure 5. The procedures of data acquisition and data reduction were discussed in detail. The results were then presented in terms of the loss factor, which was properly defined in Equations 7 and 8, for various frequencies.

The second major portion includes a brief description of the test results, shown in Figure 7, in terms of loss factor versus frequency, error estimation, joint behavior, effects of the bolt clearances, and a discussion of problems encountered during the tests. Due to the limitations on the motor speed at the lower side and the capability of the recorder for the proximity transducer at the higher side of the frequency range, only those data shown in Figure 7 were considered reliable. Estimation of inherent errors in the test machine and the recording process were not attempted. The error estimated due to trace reading is about ±0.01, or ±0.6 degree in phase lag. The joint behavior under the cyclic loads are discussed, based upon the hysteresis loops obtained. The slopes of both loading and unloading branches are found to be equal at zero stress level. This slope was defined as the joint stiffness. The dependency of the joint stiffness on the clamping pressure was discussed and shown in Figure 8. The effects of bolt clearance on the hysteresis loops were idealized and shown qualitatively in Figure 9. The problems encountered during the tests were discussed and improvements which had been made were also indicated.

CONCLUSIONS

Despite certain deficiencies in the test machine and the fact that only one solid loading beam was used, the data obtained in the tests seem to give results adequate for answering some important questions set forth before the test program. On the basis of those data obtained and the joint behavior observed, some conclusions may be drawn as follows:

- The test setup with suggested changes and modifications may be used to give reliable measurements necessary to understand the joint behavior under cyclic loading conditions.

- The results in terms of the energy loss factor as shown in Figure 7 indicate that the lower the clamping pressure, or rather the joint stiffness, the higher the loss factor tends to be for bolted joints.

- Clearance between the bolt and the hole plays a more important role than expected in affecting the amounts and ways of energy dissipation in the bolted joints and, consequently, must be considered in the designs of structural damping.

- A majority of the data obtained show that certain analogies can be observed between the hysteresis loops for bolts joint and material damping.

- A comparison with material damping in terms of loss factor shows that the amount of energy loss per cycle in the joint tested is about 10 times that for steel with the Young's modulus of elasticity of 30×10^{-6} psi (data for material damping were obtained from Reference 3).

RECOMMENDATIONS

Recommendations for improvements in the equipment and procedures for future test program are summarized and discussed as follows:

TEST SPECIMEN

Current test results show that the bolt clearance effects are important in determining the joint behavior. It is then logical to pose such questions as what is the behavior of body-bound bolted joints and bolted joints with large clearance. Also what is the effect of surface finish and different materials. The thicknesses of the specimen should be increased to make bending stiffness as large as feasible and to make room for mounting strain gages on the sides in order to study the force transmission patterns. A wider range of clamping pressure for the joints should also be studied.

In addition to the bolted joints other types of joints such as riveted joints should also be tested under the same conditions and considerations as those for the bolted joints. Perhaps, the riveted joints should be assembled independently for each specimen. The energy dissipation pattern for welded joint may also be tested by the same equipment. The welded joints should also be assembled independently for each joint.

TEST MACHINE

The flex beam loading mechanism needs to be reassembled and improved to such extent that the bending induced stresses will be negligibly small compared to the in-plane, force induced stresses. For large scale tests, the split beams with neoprene snubbers, in addition to the solid beam, should also be used. All of these require a complete reassembly of the flex beam loading mechanism and the drive yoke mechanism.

The motor speed should be reduced to about 1 Hz so that data for lower frequency range can be obtained. The alternative would be to change the drive pulleys so that the same motor speed would produce a lower frequency of load cycle.

RECORDING DEVICES

Noise sources should be detected and shielded out to improve the reliability of the outputs of the proximity transducer. It would be extremely helpful to have an additional proximity transducer so that the possible bending effect can be detected. The resistors used in the Wheatstone bridge network are too temperature sensitive. A temperature compensated Wheatstone bridge network should be used in future tests.

REFERENCES

1. Rodgers, E. J., R. G. Sturm, and J. D. Warrington, "Survey on Vibration Tests on Missile-like Structures and the Effects of Damping", Technical Note R-227, Brown Engineering Co., Inc., October 1967

2. Rodgers, E. J., R. G. Sturm, and B. D. Liaw, "Vehicle Damping - Review and Status", Summary Report PVEL-2888-1, Brown Engineering Co., Inc., January 1968

3. Ruzicka, I. E., "Structural Damping", Papers presented at ASME Annual meeting in Atlantic City, N. W., December 1959

DOCUMENT CONTROL DATA - R&D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Research Laboratories Brown Engineering Company, Inc. Huntsville, Alabama	2a. REPORT SECURITY CLASSIFICATION Unclassified 2b. GROUP N/A

3. REPORT TITLE

Feasibility Study of Experimental Methods for Joint Damping Analysis

4. DESCRIPTIVE NOTES (Type of report and inclusive dates)

Technical Note AST-278, August 1968

5. AUTHOR(S) (Last name, first name, initial)

E. J. Rodgers, Ph. D.
B. D. Liaw, Ph. D.
R. G. Sturm, Ph. D.

6. REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
August 1968	41	3

8a. CONTRACT OR GRANT NO. NAS8-20073 b. PROJECT NO. c. d.	9a. ORIGINATOR'S REPORT NUMBER(S) TN AST-278 9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report) N/A

10. AVAILABILITY/LIMITATION NOTICES

None

11. SUPPLEMENTARY NOTES	12. SPONSORING MILITARY ACTIVITY
None	Propulsion and Vehicle Engineering Lab George C. Marshall Space Flight Center Redstone Arsenal, Alabama

13. ABSTRACT

Results for the experimental feasibility investigation of a damping test apparatus using a structural bolted joint are presented. Experimental data as well as the test setup and the overall recording arrangement are described. Despite certain deficiencies that existed in the test machine, the results obtained seem adequate for answering some important questions set forth before the test program.

Recommendations for improvements in the equipment and the procedures for future test program are indicated and discussed.

14. KEY WORDS

Structures
Joints
Damping Capacity
Damping Tests
Joint Damping