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DEPARTMENT OF CIVIL ENGINEERING
COLLEGE OF ENGINEERING AND TECHNOLOGY
OLD DOMINION UNIVERSITY
NORFOLK, VIRGINIA 23529

PASSIVE DAMPING CONCEPTS FOR FREE AND FORCED MEMBER AND GRILLAGE VIBRATION

By

Zia Razzaq, Principal Investigator

Bassam S. Najjar, Graduate Research Assistant

Progress Report
For the period ended June 30, 1988

Prepared for the
National Aeronautics and Space Administration
Langley Research Center
Hampton, Virginia 23665

Under
Research Grant NAG-1-336
Harold G. Bush, Technical Monitor
SDD-Structural Concepts Branch

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By

Zia Razzaq[1] and Bassam S. Najjar[2]

ABSTRACT

The performance of potential passive damping concepts is investigated for a long tubular aluminum alloy member, and a two-bar grillage structure. The members are restrained partially at the ends and are of the type being considered by NASA for possible use in the construction of a future space station. Four different passive damping concepts are studied and include nylon brush, wool swab, copper brush, and "silly putty" in chamber dampers. Both free and forced vibration tests are conducted. It is found that the silly putty in chamber damper concept provides considerably greater passive damping as compared to that of the other three concepts. For the grillage natural vibration, a five wool swab damper configuration provides greater damping than the five silly putty in chamber configuration. Due to the constrained motion imposed by the vibrator used in the tests, the effectiveness of the passive dampers could not be adequately evaluated for the individual member. However, it is found that for the grillage under forced vibration, the five silly putty in chamber damper configuration provides very effective passive damping although only at and around the resonant frequency. At resonance, these dampers provide a 52% reduction in the dynamic magnification factor for this case.

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NOMENCLATURE

C	Damping coefficient
C_c	Critical damping coefficient
E	Young's modulus
f	Undamped natural frequency
g	Acceleration due to gravity
I	Moment of inertia
K_1, K_2	Rotational stiffness
K	Spring constant
L	Member length
M_d	Mass of the passive damping device
P_y	Force in y-direction
m_z	moment about z axis
t	time
W	Applied static load
Δ_e	Ordinate of deflection-time envelope
Δ_o	initial deflection amplitude
Δ_s	Static deflection
Δ_D	Dynamic deflection
δ_y	deflection in y-direction
Δ_D^*	Constrained dynamic deflection
ζ	Damping ratio
ζ_o	Damping ratio without damping device
η	Efficiency index
ρ	Mass per unit weight
θ_z	Slope
ω, ω_n	Undamped circular frequency

Nomenclature-continued

ω_d	Damped circular frequency
Ω	Forcing function frequency
X, Y, Z	Space coordinates

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1. INTRODUCTION

1.1 Background and Overview

Aluminum tubular members are being considered by NASA for use in future space structures. When in space, these members may be subjected to vibration induced by external disturbances. One practical problem is to identify a damping device that will reduce the vibration of these members substantially. Recent investigations (References 1-5) have been conducted to study passive damping devices on very slender tubular steel members with various end conditions. For example, members with 0.5 in. outer diameter, a wall thickness of 0.065 in., and a length of up to 12 ft. have been tested. In these experiments, the following passive damping concepts were investigated in the presence of natural flexural vibration:

1. Mass-String Dampers
2. External Viscoelastic Tape
3. Inner Metal Tube Core (Cu, Al, Steel, Brass)
4. Polyethylene Tubing
5. Polyethylene Empty Chambers
6. Chambers-with-Oil
7. Chambers-with-Oil and Discs
8. Chambers-with-Sand
9. Bright Zinc Chain
10. Brushes for Electrostatic and Frictional Damping
11. Mass-String-Whiskers Assembly

These dampers were provided in the hollow space inside the members. The details are given in References 1-5. The natural vibration tests with these concepts indicated a wide range of damping efficiencies. The prime candidates for further study appeared to be the brushes for electrostatic and frictional damping as well as the mass-string-whiskers assembly.

The present research report contains the outcome of natural and forced flexural vibration studies conducted on a 20.86 ft. long tubular aluminum alloy member, and a two-bar grillage structure constructed from tubular members 14.75 ft. and 20.86 ft. long, with various passive damping concepts. A grillage structure is one which is subjected to loads or vibration at right angles to its own plane. The grillage structure used in the present study is obtained by retaining a typical side and a diagonal member of a cubical subassemblage taken from a space station prototype model. Under investigation, the two bar assembly is in the horizontal plane while the vibration is induced vertically. The resulting subassemblage represents a basic space structure and provides a convenient means of testing passive damping concepts beyond the single-member level. Such members are being considered for use in the construction of outer space stations. The members possess moderate slenderness, having an outer diameter of 2.0 in., a wall thickness of 0.125 in., and possess semi-rigid connections.

1.2 Problem Definition

Figure 1 shows schematically a hollow tubular aluminum member of length 20.86 ft. with an outer diameter D_o = 2.0 in. and a wall thickness of t_o = 0.125 in. Both ends of the members are partially restrained in the rotational sense. The partial rotational restraint is provided by semi-rigid connections. Figure 2 shows a typical connection. The rotational stiffness provided by the connection is k. The member has an initial deflection v_i, and is subjected to natural or forced flexural vibration at its midspan. Also, Figure 3 shows schematically a two member grillage structure. The member lengths are 14.75 ft. and 20.86 ft. and possess the same cross-sectional dimensions D_o and t_o. Members AC, and CE are partially restrained at the ends with rotational end stiffnesses of magnitude K_1, and K_2, respectively. At C, the end connection of both members is supported by two vertical springs CF and CG of equal stiffness K. Natural or forced flexural vibration is introduced at midspan B of member AC. The problem is to first identify an efficient passive damping concept for the member shown in Figure 1, and then study its effectiveness in damping member CE of the grillage shown in Figure 3.

1.3 Objective and Scope

At the member level, the following four passive damping concepts are investigated:

1. Nylon Brush Dampers
2. Wool Swab Dampers
3. Copper Brush Dampers
4. Silly Putty in Chamber Dampers

The first three types of dampers are selected for testing since the results given in Reference 1 showed that brush or whisker dampers were quite effective in reducing natural flexural vibration. The fourth type of damper is selected owing to the plastic deformation behavior of the silly putty when subjected to external forces.

Once the best of the four damping concepts is identified, vibration tests are conducted on the two-member grillage with dampers mounted only in the longer of the two members.

The primary objective of the research is to evaluate the damping efficiencies of the above-mentioned concepts. Also, the experimental deflection-time response of the member in Figure 1 is compared to theoretical response obtained using the finite-difference procedure reported in References 2 and 5. The experiment and the theory are compared for the natural vibration tests. For the theoretical analysis, the damping properties from the experiments are utilized. For the two-member grillage, a static structural analysis is conducted for use in plotting dimensionless experimental dynamic deflections versus the ratio between the applied forcing function frequency and the natural damped frequency.

1.4 Assumptions and Conditions

The following assumptions and conditions are used in this study:

1. The passive dampers are provided only in the hollow space inside the member.
2. The deflections are small.
3. The structures tested remain elastic.
4. Members are tested at room temperature and in a 1-g environment.
5. Axial effects are small and neglected.

2. EXPERIMENTAL STUDY

2.1 Tubular Member and Grillage

The experimental study consists of conducting natural and forced vibration tests on a tubular aluminum member as well as a two-member grillage structure. The dynamic response of the test specimens without passive dampers are studied first. Their deflection-time response is then measured in the presence of four different types of passive dampers. For the grillage structure, the dampers are mounted only in the longer of the two members while the vibration is induced by perturbing the shorter member at its midspan. A brief description of the test structures is given in Section 1.2 of this report. The details are summarized in the present section.

2.1.1 Connection Details

A typical end connection is shown in Figure 2. It is made of an aluminum alloy and has a total weight of 0.595 lb. excluding the fasteners. The volume of the connection is found to be 3.988 in^3. The connection has a total of nine clevis blades each of which can receive a pair of bolts as shown in Figure 2(a). Six of the clevis blades are in the horizontal plane, one is in the vertical direction and located at C, while the remaining two are at 45 degrees relative to the vertical at C and in the planes containing the two lower clevis blades shown in Figure 2(a). For convenience in explaining the member and grillage connection details, the fastener locations for the clevis blades in the horizontal plane are numbered 1 through 12.

2.1.1.1 Member Connection

For the member tests, fastener locations 1 and 2 shown in Figure 2(a) are used at each end of the member. Figure 4 shows one end of the member connection. Referring to Figure 2(a) again, fasteners at locations 5 through 11 are used to mount the connection to a fixed base plate. No fastener was installed at location 12 due to an interference problem with the support underneath the base plate. This, however, did not make any difference since the other fasteners provided sufficient fixity. Each fastener has a diameter of 0.25 in. and a length of 0.94 in. Washers are used at locations 1 and 2 only.

2.1.1.2 Grillage Connections

Referring to Figure 3, the connection at E is identical to that described in Section 2.1.1.1. The connection at A is similar to the connection at E except that the fasteners 3 and 4 are used instead of 1 and 2. The members CA and CE are joined at C through a connection of the type shown in Figure 2(a), with fasteners 1 and 2 used for member CE, and 3 and 4 used for member CA. In addition, a pair of helical translational (vertical) springs CF and CG are attached to the grillage at C. Figure 5 shows the connection detail for translational springs. As seen from this figure, the lower end of the upper spring is connected to the vertical clevis of the end connection by means of a horizontal bolt of 0.25 in. diameter. Though not shown in the figure, the upper end of the upper spring is attached to a cotton chord which in turn is attached to a crane hook. The upper end of the lower spring is attached at center of the end connection by means of a cotton chord wrapped around and tied firmly to the end connection itself. The lower end of the lower spring is attached to a floor beam. Before commencing the tests, the translational springs are first stretched elastically so as to keep the cotton chords in tension, allow the junction C to move up and down without losing the tension in these springs completely, and to align the grillage in the horizontal plane. It was observed during both static and dynamic tests on the grillage that the first two conditions were always met. The third condition need not be retained upon application of the vertical loads at point B shown in Figure 3. Figures 6 and 7 show additional details of the connection at C and the grillage.

The spring stiffness values K_1, K_2 and K indicated in Figure 3 are found to be 53.1 kip-in/rad., 48.5 kip-in/rad., and 19.85 lb/in., respectively. The procedure for determining these stiffnesses is summarized in Appendix A.

2.2 Passive Damping Concepts

Four different types of passive dampers referred to in Section 1.3, which are investigated in this study, are described in this section.

2.2.1 Nylon Brush Dampers

Figure 8 shows a nylon brush damper. It has a diameter of 1.75 in., a total length of 6.25 in., and weighs 14.0 gms. This "Vegetable Brush" is manufactured by H. Hertzberg and Son, Inc., Middletown, N.Y. 10940, registered under "Cleansbest" number 17201/12639 and an inventory control number 073315 17201. It has a plastic handle and twisted wires with which the nylon bristles are intertwined. It is commonly used for cleaning vegetables. The brush shown in Figure 8 is obtained by cutting the twisted wires about 1.5 in. away from where the brush starts. The twisted wires do not extend beyond the brush end. Figures 9 and 10 show schematically the attachments for the passive dampers and their spacing inside the tubular member. As shown in Figure 9, the passive damping assembly consists of several parts. First, a helical spring, with a stiffness of 0.44 lb/in., is attached to the inside of the connection through a hook at end e, as shown in Figure 11. To the other end of the spring is attached a nylon string which in turn is attached to the first nylon brush damper. The nylon chord ("Sportsfisher, Monofilament Line", manufactured by Kmart Corporation, Troy, Michigan 48084, 80-13-91 No. EPM-40, inventory control number 04528201391) used in this investigation has a 40 lb. capacity. Thereafter, a series of nylon chords and nylon brush dampers are attached along the member length until the end f of the member is reached. The end of the nylon chord at f is then passed through a hole and stretched from the outside by an amount of 2.0 inches in the longitudinal direction of the member to induce tension in the helical spring at e, and subsequently tied to the vertical clevis of the connection. The stretched helical spring is shown in Figure 12. The resulting passive damping assembly is, therefore, aligned with the longitudinal axis of the tubular member due to a small amount of axial tension in it. Since the nylon chord is fairly flexible, a significant portion of the 2.0 inches of stretching is due to the elongation of the chord itself, with the remaining portion of the stretching taking place in the helical spring at e. The dampers are installed equidistantly between the ends e and f. Figure 10 shows the spacing for three dampers. Tests are conducted with 1, 2, 3, 5, 7, and 10 nylon brush dampers.

2.2.2 Wool Swab Dampers

Figure 13 shows a wool swab damper with a 1.0 in. diameter, a total length of 3.0 in., and a total weight of 7.1 gms. The wool swab is manufactured by Omark Industries, Onalaska, Wisconsin 54650 with a U.S. Patent 415838, and an inventory control number 076683422187. It has a threaded aluminum piece at one end with a twisted wire attached to it to which the wool swab is attached. The aluminum piece is 0.75 in. long, while the swab itself has a length of 2.125 in., as shown in Figure 13. It is commonly used in cleaning 12 in. gauge shotguns. The dampers are mounted inside the tubular member as shown in Figure 10. Tests are carried out with 1, 2, 3, 5, 7, 10, 15, 24, 30, 40, and 54 equidistant wool swab dampers.

2.2.3 Copper Brush Dampers

Figure 14 shows a copper brush damper with a 0.8125 in. diameter, a total length of 3.125 in., and a total weight of 13.0 gms. The brush is manufactured by Omark Industries, Onalaska, Wisconsin 54650 with a U.S. Patent 41986 and an inventory control number 07668341989. It has practically threaded aluminum piece at one end with a twisted wire attached to it to which the copper bristles are attached. The aluminum piece is 1.0 in. long, while the brush itself has a length of 2.125 in., as shown in Figure 14. This brush is also used in cleaning 12 in. gauge shotguns. The dampers are mounted inside the tubular member as shown in Figure 10. Tests are conducted with 1, 2, 3, 5, 7, and 10 equidistant copper brush dampers.

2.2.4 Silly Putty in Chamber Dampers

Figure 15 shows a "Silly Putty in Chamber" damper. It consists of a silly putty ball of about 0.75 in. diameter placed inside a perforated hollow cylindrical chamber. The silly putty is manufactured by Binney and

Smith Inc., Easton, PA 18042, with an inventory control number 07166208006. The chamber is made from a "Bristole Pipe" (PVC-1120, Schedule 40, ASTM-D-1785) having an original outer diameter of 1.28 in., and a wall thickness of 0.14 in. The chamber is made by reducing the outer and the inner diameters to 1.0625 in. through machining thus resulting in a wall thickness of 0.125 in. Since the primary purpose of the chamber is to house a ball of silly putty, its weight is reduced by drilling a total of seven, 0.25 in. diameter, holes around its periphery halfway from its ends, in order to achieve better damping efficiency. The putty is kept inside the chamber by means of a plastic wrap taped around it with Scotch Tape. The putty is free to bounce around inside the chamber. The total weight of one damper including the silly putty, the chamber, and the taped wrap, is 7.4 gms. The dampers are mounted inside the tubular member as shown in Figure 10. Tests are conducted using 1, 2, 3, 5, and 7 equidistant silly putty in chamber dampers.

2.3 Test Setup and Procedure

The main instrumentation used in conducting the tests consists of a proximity probe, vibration instrumentation, and a deflection-time (Δ - t) plotter. Summarized in this section are the test setup and procedures for member and grillage vibration experiments.

Figure 16 shows a schematic of the member natural vibration test setup. A weight W = 6.1 lb. is first attached to the member at its midspan by means of a cotton chord. To induce natural vibration, the chord is then cut by a pair of scissors to release the member. The time-dependant deflection at member midspan is recorded by means of a proximity probe shown in Figure 17 which is connected to a deflection-time recorder.

Figure 18 shows the member forced vibration setup, a schematic of which is shown in Figure 19. To induce the forced flexural vibration, a vibrator (Model 203-25-DC) is used with an oscillator (Model TPO-25). The vibrator applies a forcing function of the type:

$$F(t) = F_o \sin \Omega\, t \qquad (1)$$

in which $F_o = 4$ lb., t = time, and Ω = frequency of the forcing function.

The forcing function F(t) is transmitted from the vibrator to the tubular member through a vibrator connector, as indicated in Figure 19. The vibrator connector details are shown in Figure 20. It consists of three main segments, namely, PQ, QR, and RU interconnected at Q and R by means of pins. The end P is connected to the vibrator. The end U is connected to the lower part of a metallic ring provided around the tubular member midspan, as may be seen in Figure 18. The parts QR and RU can be disengaged at R by pulling out the pin RS instantaneously in the R-S direction as indicated by means of an arrow at S. A string attached at S is used to pull out the pin. The vibrator connector in the engaged and the disengaged positions is shown in Figures 21(a), and 21(b), respectively. The disengagement process is used for recording the deflection-time response of the member once the forcing function F(t) is removed.

Figure 22 shows a schematic of the grillage natural vibration test setup. To induce natural vibration, a weight W = 7.9 lb. attached at point B of the member AC, is released suddenly. The deflection-time response of the member CE is recorded at its midspan D. Figure 23 shows a schematic of the grillage forced vibration test setup. The forcing function F(t) is applied at B and the deflection-time response is recorded at midspan D of the member CE up to a certain time whereafter the vibrator at B is disengaged (as described previously for the single member tests) to record the response for $F(t) = 0$.

2.4 Test Results and Discussion

In this section, the results from the member and grillage natural and forced vibration tests are presented and discussed. For a substantial duration of this study, nylon brush, wool swab, and copper brush dampers were considered for a detailed investigation. Initially, only natural vibration tests were conducted on the member to identify the most effective damping concept as well as the optimum number of dampers. It was found that a total of five wool swab dampers provided the maximum passive damping efficiency under natural vibration condition. The five wool swab damper configuration were then tested in the member under both natural and forced vibration. As mentioned earlier, only the larger of the two members in the grillage was provided with the dampers. During the final phases of this investigation, another passive damping concept based on the use of silly putty in PVC chambers under member natural vibration was found to be more effective than the wool swab dampers (Reference 7).

The vibrator employed for the forced vibration tests allows only a limited amount of travel. This means that the deflection of the member at the location where the vibrator is attached is limited to what the vibrator can allow. Nevertheless, forced vibration tests were conducted on the individual member since it was not known initially as to whether or not the dynamic deflections would exceed the vibrator capacity. The results presented later in this section indicated that the vibrator "constrained" the member deflection for a certain range of the forcing function frequencies including that which would otherwise have constituted a resonance condition. This limitation of the vibrator made the evaluation of the performance of the dampers difficult through tests on the individual member. However, the only way to evaluate the performance of the dampers under forced vibration is by allowing the member to develop dynamic deflections without direct restrictions imposed by the vibrator. As a result, the grillage test procedure was conceived and eventually adopted. Thus, while one member was being vibrated under the influence of a forcing function, the deflections of the other member were being recorded without any constraint at its midspan.

The forced vibration tests on the member and the grillage included the study of the wool swab and the silly putty in chamber dampers in the time domain past the discontinuation of the applied forcing function. Each test was repeated three times to obtain proper averages.

2.4.1 Performance of Dampers Under Natural Member Vibration

All passive damping concepts listed in Section 2.2 of this research report are tested with natural member vibration caused by releasing a weight at midspan as explained in Section 2.3. The initial midspan deflection, Δ_o, due to the suspended weight is 0.3125 in. To quantify the damping characteristics and efficiencies, the damping concepts were tested inside the member, as described in Section 2.2. Table 1 summarizes the test results for the 20.86 ft. member with no dampers, as well as with nylon brush, wool swab, copper brush, and silly putty in chamber dampers. The number of dampers, the weight of the damping assembly W_d , the natural frequency f, the damping ratio ζ, and the damping efficiency index η are listed for each passive damping assembly. The damping ratio is obtained using the logarithmic decrement method (Reference 6) and is defined as follows:

$$\zeta = C/C_c \tag{2}$$

in which:

C = damping coefficient,

C_c = critical damping coefficient = $2\ \rho\omega$,

ρ = member mass per unit length, and

$\omega = \sqrt{k/m}$

The efficiency index (References 1 and 2) is defined as:

$$\eta = \frac{\zeta - \zeta_o}{M_d} \tag{3}$$

in which ζ is the damping ratio in the absence of any passive damping device, and M_d is the mass of the passive damping device. The average values of ζ and f for the member with no dampers are 0.003884, and 4.5 Hz, respectively. Figure 24 represents the average Δ - t plot for this case. This plot is determined using the average ζ and f values from the experiments, and the following Δ - t relationship (Reference 6):

$$\Delta = \Delta_o\, e^{-\zeta\omega t} \left[\frac{\omega\zeta}{\omega_d} \sin \omega_d t + \cos \omega_d t \right] \tag{4}$$

Equation 4 represents the solution to the time-dependent ordinary differential equation obtained by the method of separation of variables applied to the governing partial differential equation for natural vibration of the member. The damped circular frequency, ω_d, is given by:

$$\omega_d = \omega \sqrt{1 - \zeta^2} \tag{5}$$

Additional details, including the listing of a computer program utilizing Equation 4 to plot the average Δ - t relationships are described in Reference 1.

For nylon brush dampers, the maximum $\zeta = 0.006109$ is obtained with an assembly of three dampers resulting into an $\eta = 8.34$ in/lb - sec^2, whereas the maximum $\eta = 10.68$ in/lb - sec^2 is obtained with a one-damper assembly corresponding to $\zeta = 0.004732$. The natural frequency, f, is in the range from 4.50 Hz to 4.45 Hz for one to ten dampers. Figure 25 represents the average Δ - t plot for the member with three nylon brush dampers. Figure 26 shows the effect of three nylon brush dampers on the deflection-time envelopes. The vertical ordinate in this figure is designated by Δ_E to indicate that the figure represents the envelopes rather than the complete Δ - t relationships. The damping ratio increases as the number of dampers is increased from one to three. A most surprising result is that the damping ratio decreases as the number of dampers is increased beyond three. A plausible explanation of this phenomenon may be as follows. The outer diameter of these dampers is practically the same as the inner diameter of the member. When the number of dampers is three or less, the frictional interaction between the dampers and the member inside surface takes place while allowing the dampers to move in the axial direction, thus absorbing a portion of the energy of vibration. When the number of dampers is increased to 10, the frictional resistance of the dampers is so large that no relative motion between the dampers and the member inside surface is possible, thus the dampers become inactive. The difference in the lengths of the deflected and the undeflected member may be taken by either the axial stretching of the nylon chord, and/or the helical spring to which the damping assembly is attached is nearly the same as that with no dampers. The small negative η value may be regarded as practically zero. For 5 and 7 dampers, only partial frictional interaction occurs.

For wool swab dampers, the maximum $\zeta = 0.006365$ is obtained with an assembly of five dampers resulting into an $\eta = 12.32$ in/lb-sec^2, whereas the maximum $\eta = 10.68$ in/lb-sec^2 is obtained with a two-damper assembly corresponding to $\zeta = 0.005574$. Figure 27 represents the average Δ - t plot for the member with five wool swab dampers. Figure 28 shows the effect of five wool swab dampers on the deflection-time envelopes. The damping ratio increases as the number of dampers is increased from one to five. Increasing the number of dampers beyond five results in a general decrease in f value. This trend is observed for up to 54 dampers. The η value decreases continuously for increasing number of dampers past the two-damper assembly. For 40 and 54 dampers, η is negative, implying that for these two cases the ζ values are less than ζ_o. This phenomenon may be explained as follows. The outer diameter of each wool swab damper is 0.75 in. less than the inner diameter of the pipe, therefore, the dampers can bounce back and forth freely during member vibration. There may be at least, two possible types of damping assembly motion inside the member. One type of motion may involve the impact of dampers against the member inside surface in a direction opposite to that of the member motion. When this occurs, the member vibration amplitude decreases resulting into ζ values greater than ζ_o . The other type of motion may involve the impact of dampers on the inside surface in the same direction as that of the member motion. Under such circumstances, the impacting dampers will amplify the member deflection rather than reduce it. The net result may be a ζ value less than ζ_o. Another noticeable effect of the number of dampers is on the member natural frequency which decreases form 4.50 Hz for one damper to 4.21 Hz for 54 dampers. The use of Equation 5 reveals that this variation in f cannot be attributed to the variation in ζ. Why is then f decreasing? A possible explanation may be as follows. The damper assembly (consisting of the nylon string, the dampers, and the helical spring attached at one end to keep a small amount of tension in the string) has its own natural frequencies and mode shapes varying with the number of dampers. During member vibration, its own frequency is influenced or altered by the dynamic behavior of the damping assembly. Thus, a measured f value does not simply represent the effect of the member vibration only but also of the motion of the damping assembly within the confines of the hollow space inside the member.

For copper brush dampers, the maximum $\zeta = 0.007082$ is obtained with an assembly of three dampers giving a maximum value of 14.46 in/lb-sec^2 for this damping concept. Figure 29 represents the average Δ -t plot for the member with three copper brush dampers. Figure 30 shows the effect of three copper brush dampers on the Δ_E - t relationship. Both ζ and f decrease with an increase in the number of dampers. Since the outer diameter of each copper brush damper is 0.9375 in. less than the inner diameter of the pipe, the member behavior with these dampers is also affected by the type and extent of their collision with the inside member surface.

For silly putty in chamber dampers, the maximum $\zeta = 0.010870$ is obtained with an assembly of five dampers resulting into an $\eta = 31.60$ in/lb-sec^2, whereas the maximum $\eta = 39.87$ in/lb-sec^2 is obtained with a one-damper assembly corresponding to $\zeta = 0.005648$. Figure 31 represents the average Δ - t plot for the member with five silly putty in chamber dampers, while Figure 32 shows their effect on the Δ_E - t relationship. For one to seven dampers, the natural member frequency remains practically unchanged.

2.4.2 Performance of Wool Swab Dampers under Forced and Free Member Vibration

Table 2 summarizes the member forced and free vibration test results with no dampers, and with 1, 2, 3, and 5 wool swab dampers. Forcing function frequencies of 2.0, 4.0, and 5.0, Hz are used resulting into Ω/ω_n ratios of 0.444, 0.888, and 1.111, respectively. Figures 33(a) through 33(c) shows the experimental deflection-time relationships for forced and free member vibration with forcing function frequency of 4.0 Hz, without dampers, with 3 wool swab dampers, and 5 wool swab dampers, respectively. Each of these three figures has a constrained forced and a free vibration part. The free vibration part of the deflection-time relationship is obtained by disengaging (or removing) the forcing function from the member midspan. The constrained dynamic deflection amplitude, Δ_D^*, and its dimensionless value, Δ_D^*/Δ_S, where Δ_S is the static member midspan deflection for a 4.0 lb. load, are listed in Table 2. The constrained deflection Δ_D^* is based on the initial "CONSTRAINED FORCED" part of the deflection-time curves shown in Figure 33. The data in Table 2 shows that the ratio Δ_D^*/Δ_S is nearly constant and around 2.0 for forcing function frequencies of 2.0 and 4.0 Hz with or without wool swab dampers. The constrained deflection-time response at member midspan was also monitored for several gradually increasing forcing function frequencies in the range 2.0-5.0 Hz. It was found that the constrained dynamic deflection amplitude remained nearly constant (around 0.4 in.) for forcing function frequencies in the range from 2.0 Hz to slightly over 4.5 Hz, whereafter a gradual decrease of the midspan amplitude resulted. For example, with 5 wool swab dampers, the dynamic deflection amplitude dropped from nearly 0.41 in., for the frequency range 2.0-4.0 Hz as tabulated and up to 4.5 Hz as monitored, to 0.38 in for 5.0 Hz. One serious consequence of the deflection constraint imposed by the vibrator is that no resonance phenomenon could be produced at the natural member frequency in the vicinity of 4.5 Hz.

The damping ratios obtained from the free vibration part of the deflection-time curves are listed in the last column of Table 2. As seen from this data and Figure 33, the 5 wool swab damper configuration provides a significant decrease in the free vibration amplitudes. Another important observation to be made here is that the ζ values in Table 2 are less than the corresponding ones given in Table 1. This is attributable to the dependence of ζ on the initial amplitude which is considerably greater for the results reported in Table 2 as compared to that used to obtain the results in Table 1, namely, with $\Delta_o = 0.3125$ in. This type of amplitude-dependence was also reported in Reference 1 for another type of tubular member.

2.4.3 Performance of Wool Swab and Silly Putty in Chamber Dampers under Natural Grillage Vibration

The grillage natural vibration tests are conducted by first suspending a weight W at midspan of member AC while monitoring the Δ - t response at midspan of member CE. The experimental static load-deflection relationship is found to be as follows:

$$W_B = 172\ \Delta_D \tag{6}$$

in which:

W_B = weight suspended at B, lb., and

Δ_D = vertical static deflection at D, in.

Natural vibration is induced in the grillage by first suspending a weight

$W_B = 7.9$ lb. by means of a cotton string, and then cutting the string to release the weight. Table 3 summarizes the grillage natural vibration test results with no dampers, with 1, 5, and 7 wool swab dampers, and with 5 silly putty in chamber dampers. The natural frequency f and the damping ratio ζ are listed for each passive damping assembly.

Figures 34(a) through 34(c) shows the experimental Δ - t relations for the grillage assembly with no dampers, 5 wool swab dampers, and 5 silly putty in chamber dampers. The highest damping ratio of 0.01115 is obtained for 5 wool swab dampers. The damping ratios are calculated using the gradually decaying part of the Δ - t curves, which commences nearly six seconds past the initiation of the experiment.

2.4.4 Performance of Silly Putty and Wool Swab Dampers Under Forced and Free Grillage Vibration

Table 4 lists the grillage forced and free vibration test results with no dampers, wool swab, and silly putty in chamber dampers. The number of dampers, the frequency of the forcing function, $\Omega / 2\pi$, the frequency ratio, the dynamic deflection amplitude, Δ_D, the dynamic magnification factor (DMF) given by Δ_D/Δ_S, and the average damping ratio ζ, are given for no dampers, 1, 5, and 7 wool swab, and 5 silly putty in chamber dampers. Figures 35 through 37 show the experimental Δ - t relationships for the grillage assembly with no dampers, 5 wool swab, and 5 silly putty in chamber dampers with $\Omega = 2$, 3, 4, 8, and 9 Hz., respectively.

Figures 38(a) through 38(c) present a comparison of the Δ - t curves with no dampers, 5 wool swab, and 5 silly putty in chamber dampers, respectively, with $\Omega = 4$ Hz. In the forced vibration region of the deflection-time curves, the amplitude of the forcing function is reduced when the dampers are used, as seen from the forced region in Figures 38(b) and 38(c). This is evident from the reduction of the amplitudes and the presence of the associated wave form as seen in these figures.

Figures 39(a) and 39(b) present the Δ - t curves at resonance with no dampers, and with 5 silly putty in chamber dampers. Figure 40 shows the DMF versus the frequency ratio for no dampers and 5 silly putty in chamber dampers. The results in this figure show that the silly putty dampers do not change the DMF appreciably for non-resonance frequency ratios. However, the dampers reduce the DMF by 52% at resonance.

2.5 Evaluation of Passive Damping Concepts

2.5.1 For Member

Figures 41 and 42 represent the damping ratio ζ and the efficiency index η versus the number of dampers with various damping concepts for the member natural vibration. Figure 41 shows that for all the dampers used the damping ratio ζ has a peak value between 0 and 10 dampers. This value decreases continuously with the increasing number of dampers.

Figure 42 compares the η - N curves for the wool, copper, nylon, and silly putty in chamber dampers. Clearly, the silly putty dampers provide the best of the four η - N curves. Figure 43 compares the dimensionless deflection-time ($\bar{\Delta}_E$ - t) envelopes for 20.86 ft. member with no dampers to those with 5 wool swab and 5 silly putty in chamber dampers. The superior performance of the silly putty dampers in reducing the vibration amplitudes can clearly be observed in comparison to the wool swab dampers. Furthermore, since the weight of each of the damping assembly is about 35 grams, and the weight of the member is 8,869 grams, the weight of the dampers is only 0.4% of the weight of the member being damped. Thus, a substantial increase in the damping ratio is achieved with a relatively small increase in the total mass of the member.

For the member forced vibration, only the wool swab dampers are tested. Due to the difficulties encountered in relation to the deflection constraint imposed by the vibrator on the member, a meaningful evaluation of the performance of the dampers could not be made at the individual member level under the action of the forcing function. The only useful data from these tests is the one that is obtained after the vibrator is disengaged from the member, that is, in the natural vibration domain. The corresponding ζ values in Table 2 for this range of member response, however, reflect the effect of initial amplitude on ζ after the disengagement of the vibrator. These ζ values, as mentioned earlier, are smaller than those in Table 1 which are obtained for a smaller initial amplitude under natural vibration conditions.

2.5.2 For Grillage

The results in Tables 3 and 4 show that the ζ values are amplitude-dependant when no dampers are used in the grillage. The type of amplitude-dependance observed previously for single member response is also found to occur for this case. However, there is practically little effect of the initial amplitude on ζ values in the case of the damped grillage. For the grillage under natural vibration, the 5 wool swab damper configuration provides greater damping than that of the 5 silly putty in chamber damper configuration. For the grillage under forced vibration, the 5 silly putty in chamber damper configuration provides effective damping only at and around the resonant frequency, which is nearly 3.50 Hz. At resonant frequency, the dynamic deflection amplitude is reduced by as much as 52% in comparison to the case with no dampers. The passive dampers, therefore, provide a dramatic reduction in the grillage deflection amplitudes at resonance.

3. THEORETICAL AND EXPERIMENTAL MEMBER FREE VIBRATION AND GRILLAGE STATIC DEFLECTION

3.1 Member Free Vibration

A comparison of the natural theoretical member natural vibration response based on the finite-difference procedure outlined in References 2 and 5 is presented in Figures 44 through 48. These figures show theoretical versus average experimental Δ - t plots for member free vibration with no dampers, 3 nylon, 5 wool swab, 3 copper, and 5 silly putty in chamber dampers, respectively. The corresponding ζ values have been listed previously in Table 1. A comparison of the theoretical and experimental curves shows clearly that the viscous damping model adopted in the theory is a good one.

The member average natural frequency based on the experimental results summarized in Table 1 is about 4.50 Hz. The frequency obtained from the finite-difference results in Figures 44 through 48 is nearly 4.35 Hz. Furthermore, a value of 4.68 Hz results for the frequency when the approximate formula given by Equation 28 of Reference 5 is used. Thus, the theoretical frequency values are in a reasonable agreement with those found experimentally.

3.2 Grillage Static Analysis

The governing first-order linear elastic load-deflection relationship for the grillage in Figure 3 can be expressed as follows:

$$\underline{P} = \underline{Q}\ \underline{\delta} \tag{7}$$

in which:

$\underline{P}$ = load vector,

$\underline{\delta}$ = deflection vector, and

$\underline{Q}$ = stiffness matrix.

In the expanded form, Equation 6 can be written as follows:

$$\begin{Bmatrix} \underline{p}_A \\ \underline{p}_B \\ \underline{p}_C \\ \underline{p}_D \\ \underline{p}_E \end{Bmatrix} = \begin{bmatrix} \underline{Q}_{11} & \underline{Q}_{12} & \underline{Q}_{13} & \underline{Q}_{14} & \underline{Q}_{15} \\ & \underline{Q}_{22} & \underline{Q}_{23} & \underline{Q}_{24} & \underline{Q}_{25} \\ & & \underline{Q}_{33} & \underline{Q}_{34} & \underline{Q}_{35} \\ & \text{Symmetric} & & \underline{Q}_{44} & \underline{Q}_{45} \\ & & & & \underline{Q}_{55} \end{bmatrix} \begin{Bmatrix} \underline{d}_A \\ \underline{d}_B \\ \underline{d}_C \\ \underline{d}_D \\ \underline{d}_E \end{Bmatrix} \tag{8}$$

in which the non-zero sub-matrices are defined as:

$$\underline{Q}_{11} = (\underline{K}'_{11})a \tag{9}$$

$$\underline{Q}_{12} = (\underline{K}'_{12})a \tag{10}$$

$$\underline{Q}_{22} = (\underline{K}'_{11})b + (\underline{K}'_{22})a \tag{11}$$

$$\underline{Q}_{23} = (\underline{K}'_{12})b \tag{12}$$

$$\underline{Q}_{33} = (\underline{K}'_{22})b + (\underline{K}'_{11})c \tag{13}$$

$$\underline{Q}_{34} = (\underline{K}'_{12})c \tag{14}$$

$$\underline{Q}_{44} = (\underline{K}'_{22})c + (\underline{K}'_{11})d \tag{15}$$

$$\underline{Q}_{45} = (\underline{K}'_{12})d \tag{16}$$

$$\underline{Q}_{55} = (\underline{K}'_{22})d \tag{17}$$

Also, since the two tubular members in the grillage are fairly rigid torsionally, the torsional deformations and member end-actions may be neglected, and the resulting typical load and deflection vectors in

Equation 8 would be of the form as given below:

$$\{\underline{P}\} = \left\{ \begin{matrix} P_y \\ m_z \end{matrix} \right\} \tag{18}$$

$$\{\underline{d}\} = \left\{ \begin{matrix} \delta_y \\ \theta_z \end{matrix} \right\} \tag{19}$$

The corresponding $\underline{Q}_{ij}$ matrices defined in Equations 8 to 16 will, therefore, have an order of 2 X 2.

To include the effect of the two vertical translational springs at C in Equation 8, the load vector $\underline{P}_C$ can be expressed in terms of the spring stiffness K as follows:

$$\underline{P}_C = \left\{ \begin{matrix} 2\ K\ \delta_C \\ \theta_C \end{matrix} \right\} \tag{20}$$

The experimentally obtained value of K is 0.01985 Kips/in.

The details of the individual $\underline{Q}_{ij}$ matrices are given in Appendix B. Before Equation 8 cans be solved, the left hand side of Equation 20 is transferred to the right hand side of the former equation for associating it with the stiffness terms in the $\underline{Q}$ matrix which form products with the δ_C deflection.

Equations 6 and B.9 represent the experimental and the theoretical static load-deflection relationships of the grillage shown in Figure 3, loaded as described in Section 2.4.3. The corresponding experimental and theoretical constants for the load-deflection relationship are 172 lb/in. and 190 lb/in., respectively.

4. CONCLUSIONS AND FUTURE RESEARCH

4.1 Conclusions

The following conclusions are drawn from the research conducted herein:

1. Under member natural vibration, the silly putty in chamber damper concept provides considerably greater passive damping as compared to that of the nylon brush, wool swab, and copper brush dampers.
2. Due to the constrained motion imposed by the vibrator, the effectiveness of the passive dampers could not be adequately evaluated using tests on the individual member.
3. The damping ratio for the member, with or without the passive dampers, is initial amplitude dependant.
4. For the grillage under natural vibration, the 5 wool swab damper configuration provides greater damping than the 5 silly putty in chamber damper configuration.
5. For the grillage under forced vibration, the 5 silly putty in chamber damper configuration provides very effective passive damping only at and around the resonant frequency. At resonance, these dampers result into a 52% reduction in the dynamic magnification factor.
6. For the grillage tested, the damping ratio is amplitude-dependant only in the absence of passive dampers. When dampers are used, such dependance practically disappears.
7. The natural vibration deflection-time relationships from the finite-difference analysis of the individual member is in excellent agreement with the experimental results.

8. The static deflection results for the grillage are in a reasonable agreement with the experimentally observed ones.

4.2 Future Research

The effectiveness of the passive damping concepts identified herein should be examined using the member and structural-subassemblage tests with boundary conditions which result into a combination of both flexural as well as overall translational motion of the supports. Further new concepts need to be evolved which would exhibit effectiveness at both resonant and non-resonant frequencies.

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TABLES

Table 1. Member natural vibration test results for nylon brush, wool swab, copper brush, and silly putty in chamber dampers (L = 20.86 ft.)

PASSIVE DAMPING CONCEPT	NUMBER OF DAMPERS	WEIGHT OF DAMPING ASSEMBLY W (gms.)	NATURAL FREQUENCY F (Hz)	AVERAGE DAMPING RATIO ξ	DAMPING EFFICIENCY INDEX (in/lb-sec^2)
No Dampers	0	0.00	4.50	0.003884	0.00
Nylon Brush Dampers	1	14.00	4.50	0.004732	10.68
	2	28.00	4.50	0.004659	4.88
	3	42.00	4.50	0.006109	8.34
	5	70.00	4.48	0.004094	0.53
	7	98.00	4.46	0.004279	0.71
	10	140.00	4.45	0.003861	-0.03
Wool Swab Dampers	1	7.10	4.50	0.004661	19.30
	2	14.20	4.50	0.005574	20.88
	3	21.30	4.50	0.006045	17.89
	5	35.50	4.50	0.006365	12.32
	7	45.70	4.49	0.005904	7.17
	10	71.00	4.48	0.006063	5.41
	15	106.50	4.47	0.004889	1.66
	24	171.40	4.44	0.004533	0.68
	30	213.00	4.39	0.004425	0.45
	40	284.00	4.38	0.003618	-0.17
	54	384.00	4.21	0.003313	-0.26
Copper Brush Dampers	1	13.00	4.50	0.004656	10.47
	2	26.00	4.50	0.005743	12.61
	3	39.00	4.50	0.007082	14.46
	5	65.00	4.47	0.005467	4.29
	7	91.00	4.45	0.006402	4.87
	10	130.00	4.44	0.005031	1.56
Silly Putty In Chamber Dampers	1	7.80	4.50	0.005648	39.87
	2	15.60	4.50	0.005842	22.13
	3	23.40	4.50	0.006038	16.23
	5	35.00	4.50	0.010870	31.60
	7	54.60	4.49	0.006910	9.70

Table 2. Member forced and free vibration test results for wool swab dampers (L = 20.86 ft.)

PASSIVE DAMPING CONCEPT	NUMBER OF DAMPERS	FORCING FUNCTION FREQUENCY $\Omega/2\pi$,Hz	$\frac{\Omega}{\omega_n}$	CONSTRAINED DYNAMIC DEFLECTION AMPLITUDE Δ^*_D, in.	$\frac{\Delta_D}{\Delta_S}$	AVERAGE DAMPING RATIO ξ
No Dampers	0	2.00 4.00 5.00	0.444 0.888 1.111	0.397 0.408 0.336	1.93 1.98 1.63	0.002512 0.002917 0.003990
Wool Swab	1	2.00 4.00 5.00	0.444 0.888 1.111	0.424 0.428 0.400	2.06 2.08 1.95	0.003303 0.003300 0.003833
	2	2.00 4.00 5.00	0.444 0.888 1.111	0.425 0.428 0.388	2.07 2.08 1.89	0.002948 0.003121 0.003543
	3	2.00 4.00 5.00	0.444 0.888 1.111	0.427 0.430 0.408	2.08 2.09 1.98	0.002840 0.003775 0.004303
	5	2.00 4.00 5.00	0.444 0.888 1.111	0.412 0.416 0.384	2.00 2.02 1.87	0.005494 0.005954 0.006070

Table 3. Grillage natural vibration test results for wool swab and silly putty in chamber dampers

PASSIVE DAMPING CONCEPT	NUMBER OF DAMPERS	NATURAL FREQUENCY F (Hz)	AVERAGE DAMPING RATIO ξ
No Dampers	0	3.75	0.004747
Wool Swab	1 5 7	3.67 3.67 3.67	0.003687 0.011500 0.008117
Silly Putty	5	3.50	0.008407

Table 4. Grillage forced and free vibration test results for wool swab and silly putty in chamber dampers

PASSIVE DAMPING CONCEPT	NUMBER OF DAMPERS	FORCING FUNCTION FREQUENCY $\Omega/2\pi$, Hz	$\frac{\Omega}{\omega_n}$	DYNAMIC DEFLECTION AMPLITUDE Δ_D, in.	$\frac{\Delta_D}{\Delta_S}$	AVERAGE DAMPING RATIO ξ
No Dampers	0	2.00	0.53	0.28	12.1	0.003082
		3.00	0.80	0.34	14.7	
		3.75	1.00	1.37	59.1	
		4.00	1.07	0.61	26.3	
		8.00	2.13	0.28	12.1	
		9.00	2.40	0.17	7.3	
Wool Swab	1	2.00	0.55	0.29	12.5	0.003694
		3.00	0.82	0.41	17.7	
		4.00	1.09	0.53	22.8	
		8.00	2.18	0.29	12.5	
		9.00	2.45	0.18	7.8	
	5	2.00	0.55	0.31	13.4	0.009453
		3.00	0.82	0.43	18.5	
		4.00	1.09	0.55	23.7	
		8.00	2.18	0.30	12.9	
		9.00	2.45	0.17	7.3	
	7	2.00	0.55	0.24	10.3	0.008821
		3.00	0.82	0.42	18.1	
		4.00	1.09	0.69	29.7	
		8.00	2.18	0.24	10.3	
		9.00	2.45	0.18	7.8	
Silly Putty	5	2.00	0.57	0.25	10.8	0.008700
		3.00	0.86	0.43	18.5	
		3.50	1.00	0.72	31.0	
		4.00	1.14	0.58	25.0	
		8.00	2.28	0.25	10.8	
		9.00	2.58	0.14	6.0	

FIGURES

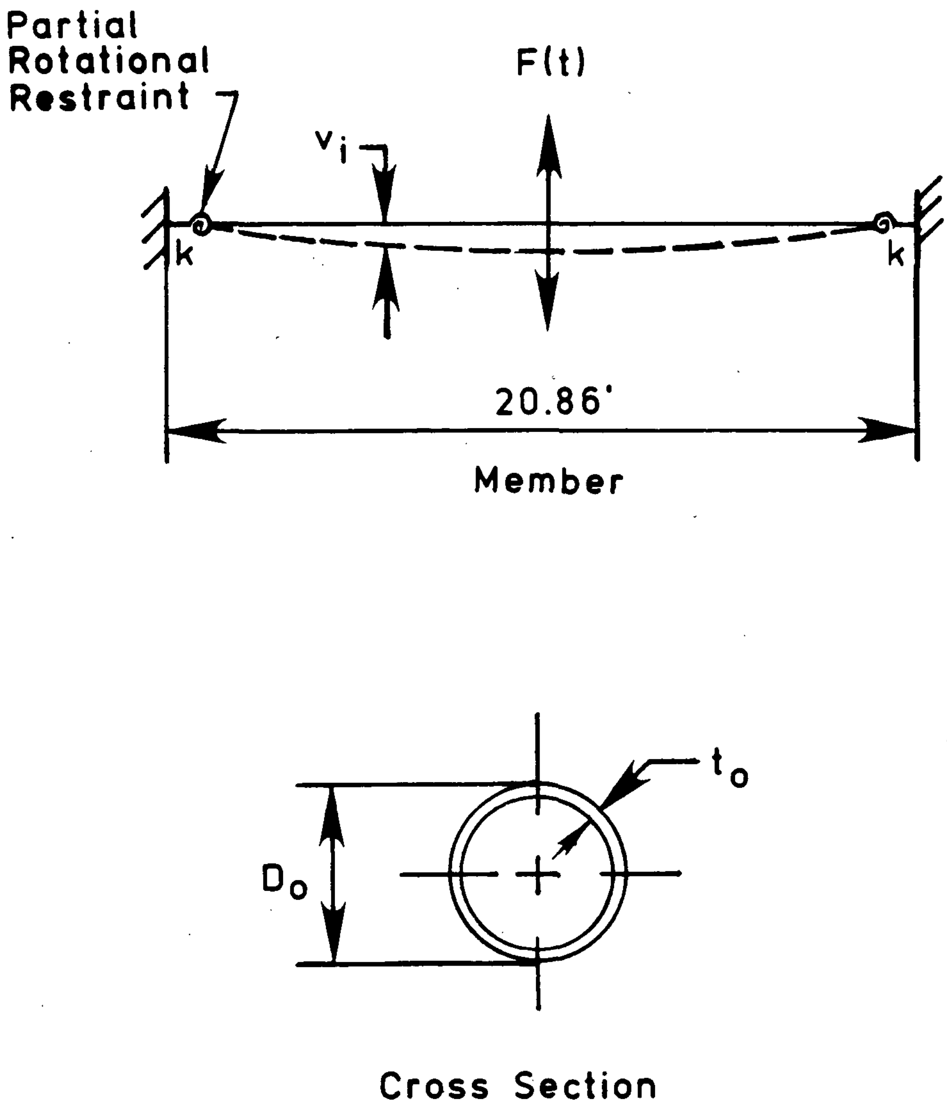

Figure 1. Schematic of tubular member

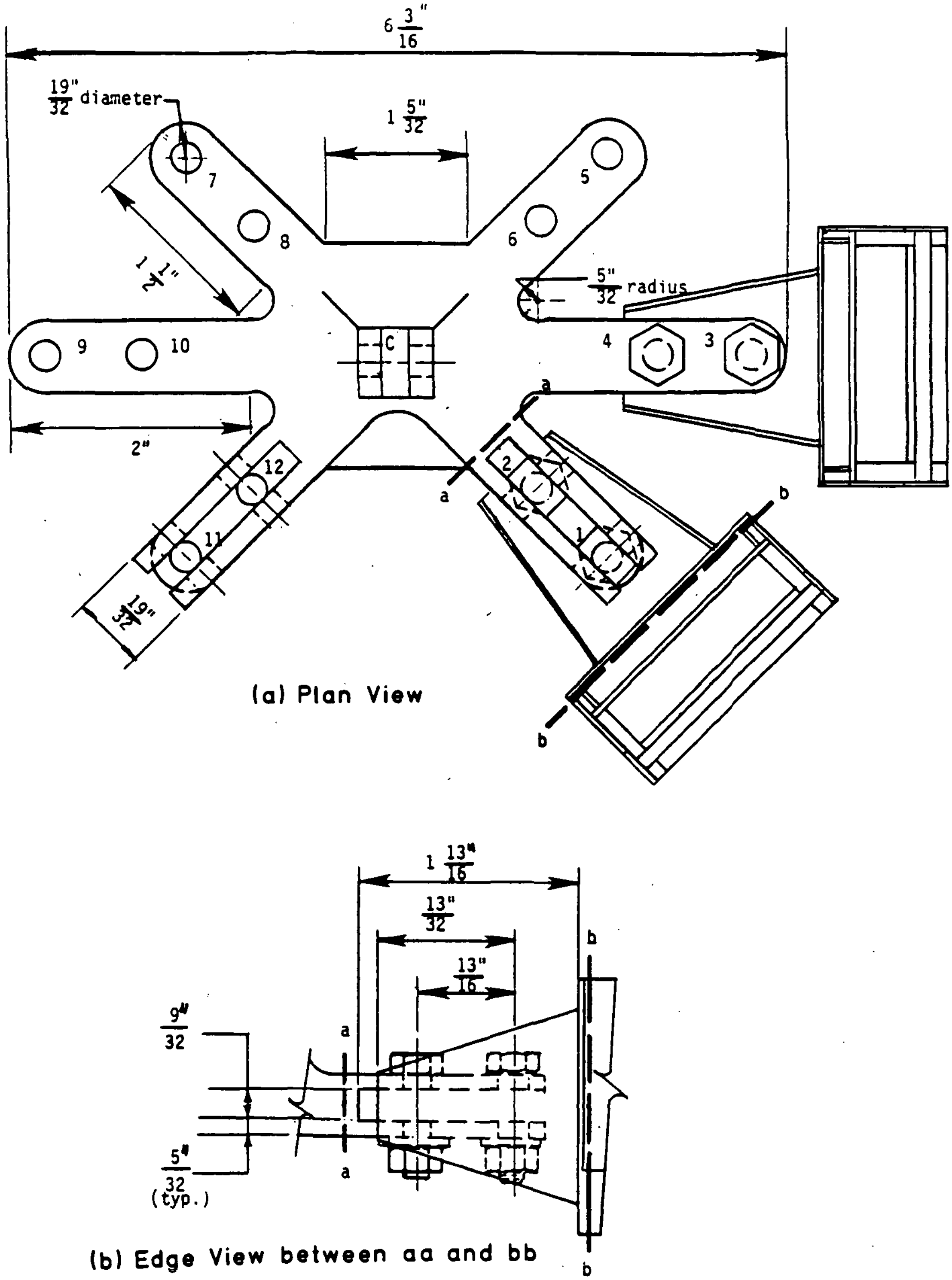

Figure 2. Some end connection details

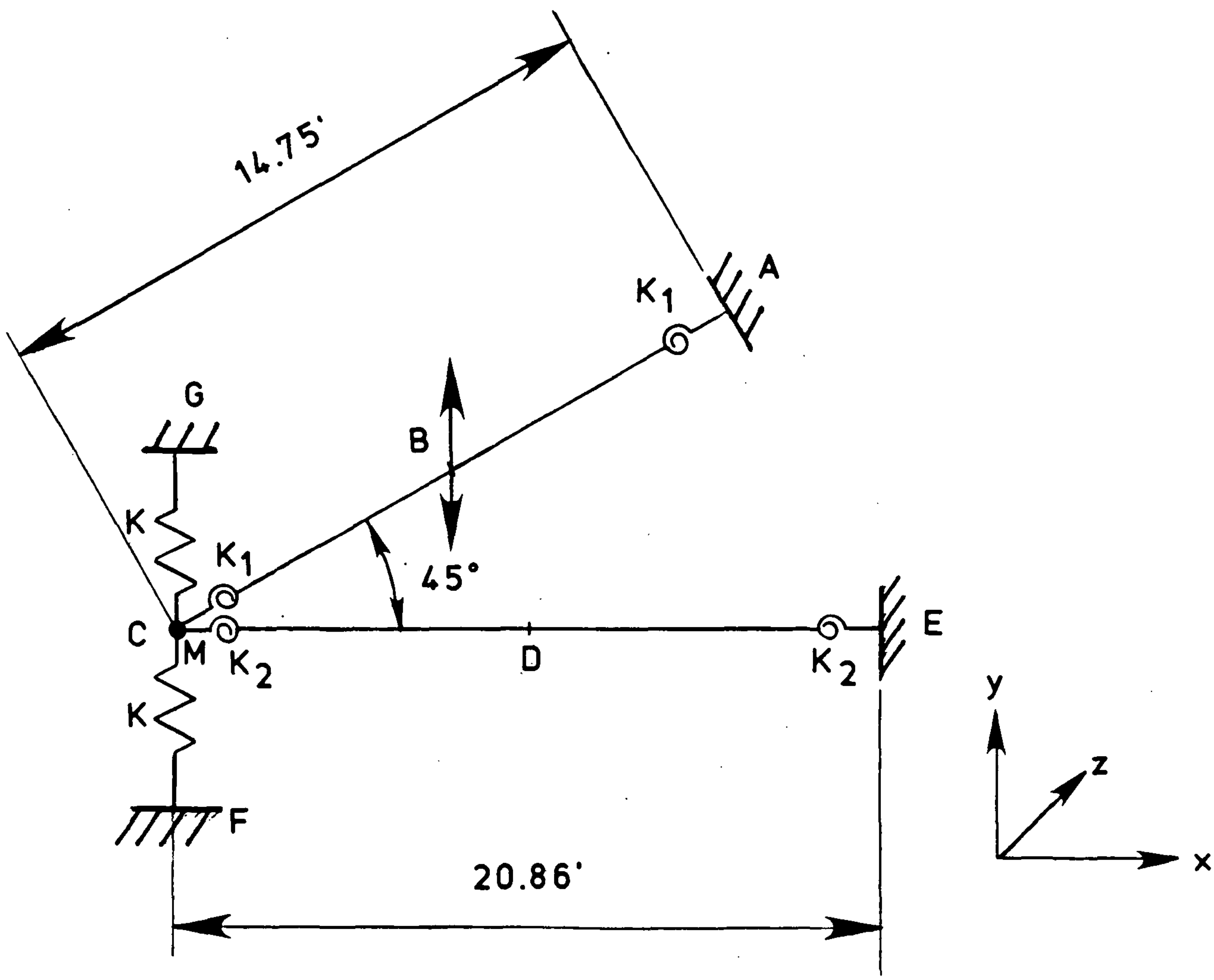

Figure 3. Schematic of two-member grillage

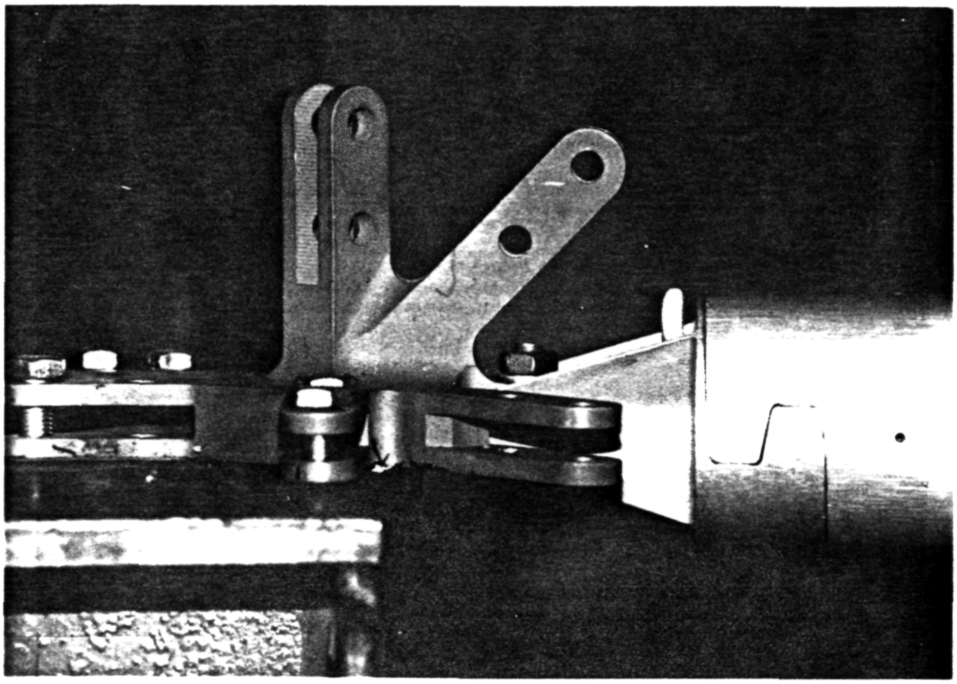

Figure 4. Member end connection

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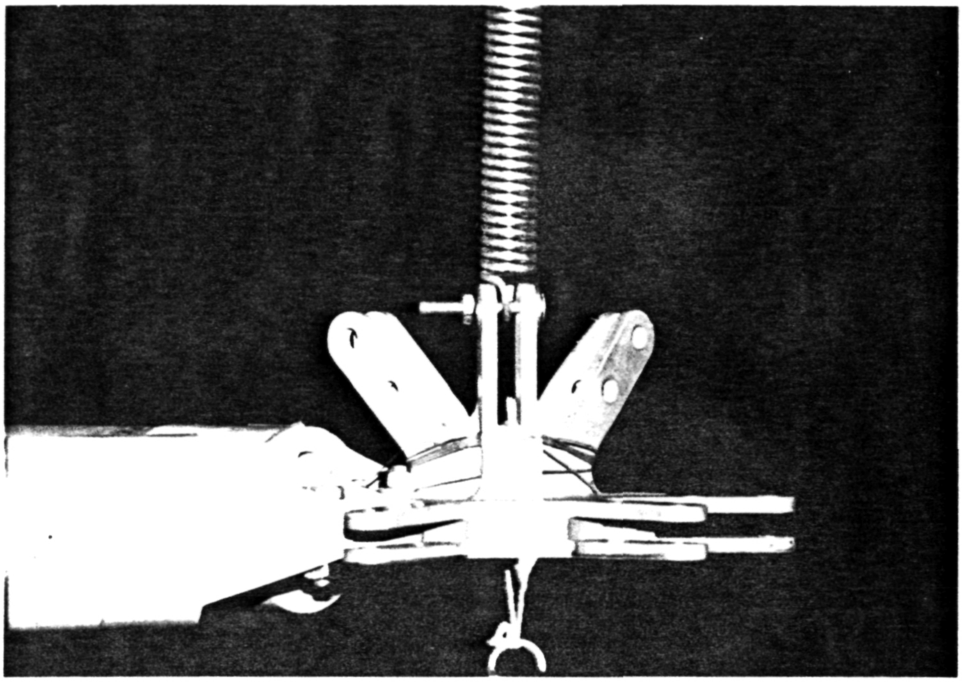

Figure 5. Connection detail at C for translational springs

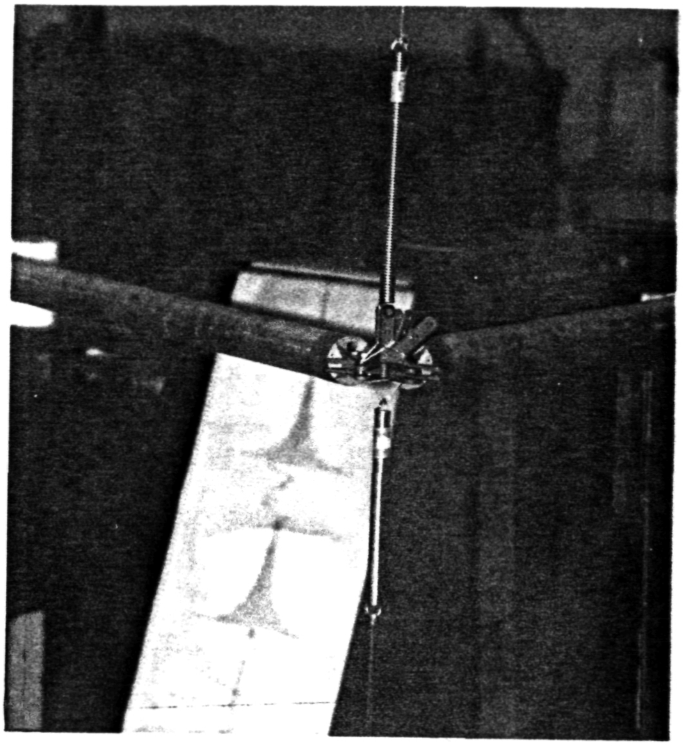

Figure 6. Detail of members CA and CE connection with two translational springs

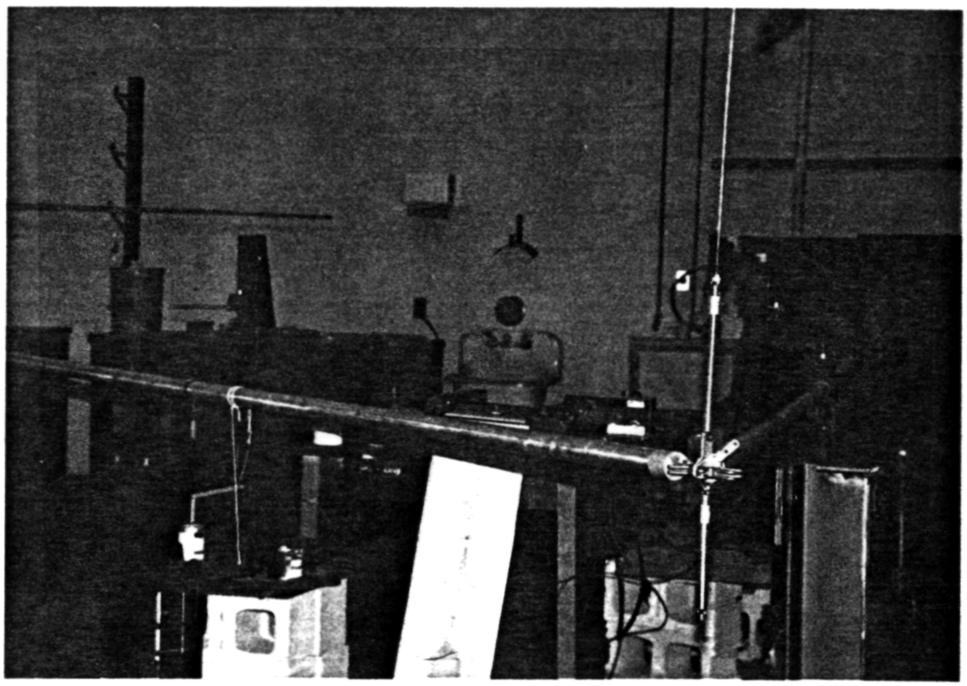

Figure 7. Grillage assembly

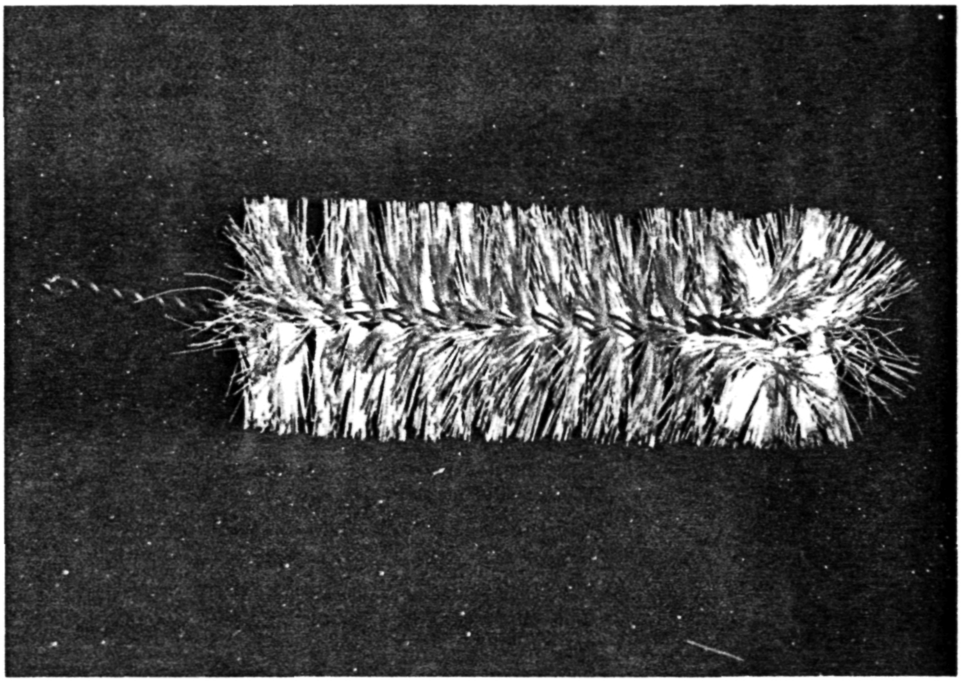

Figure 8. Nylon brush damper

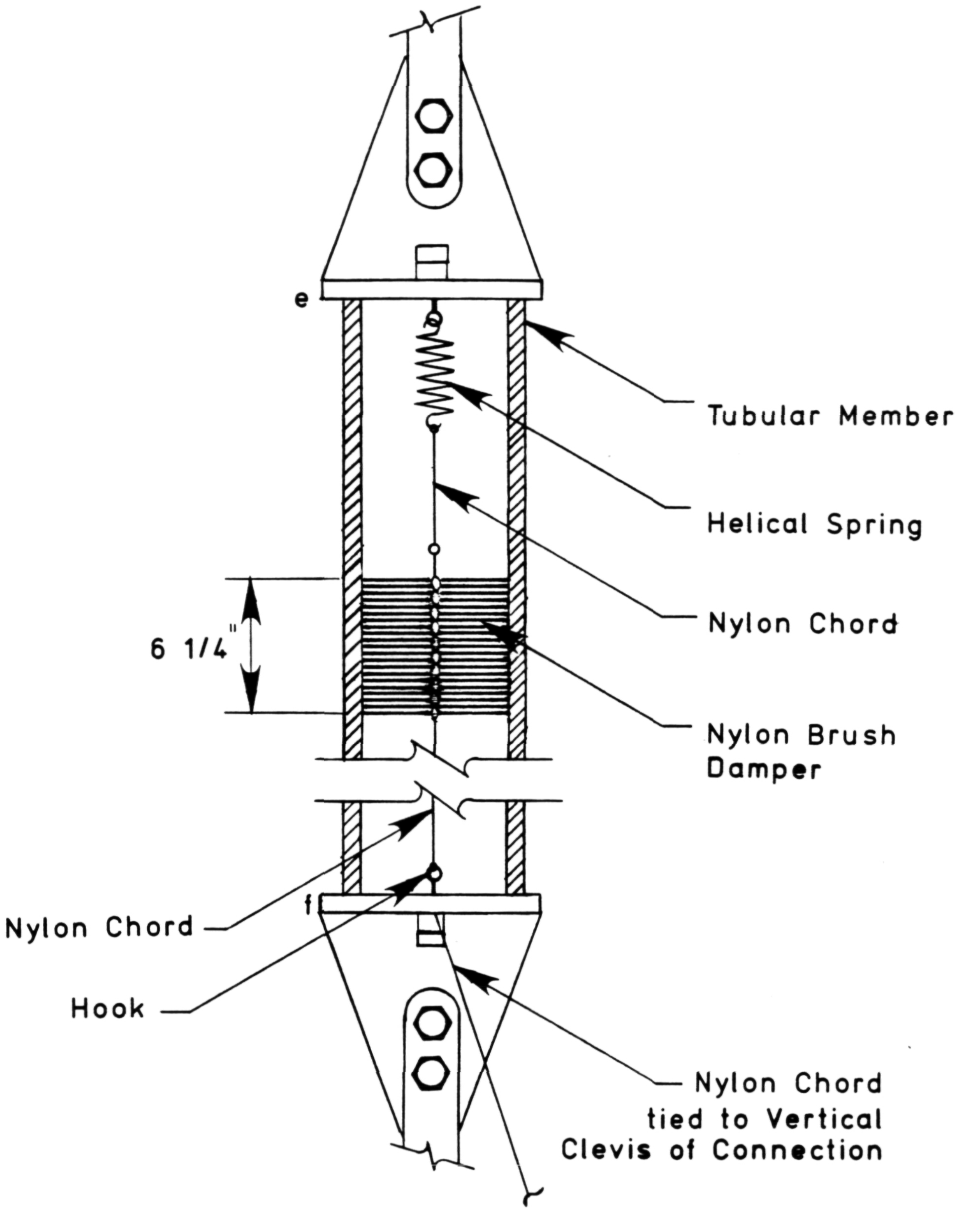

Figure 9. Schematic of attachments for passive damper inside tubular member

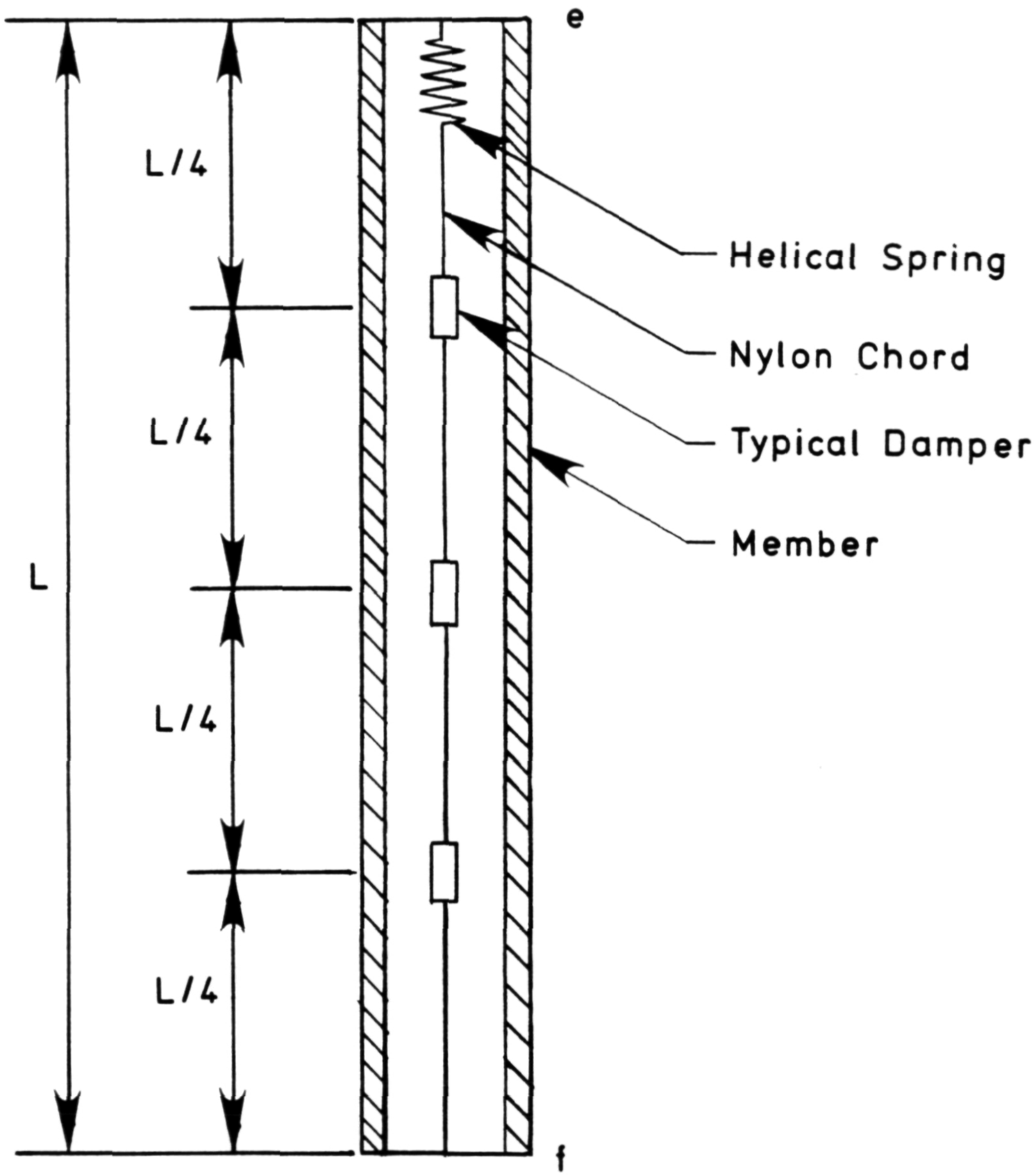

Figure 10. Schematic for spacing of passive dampers

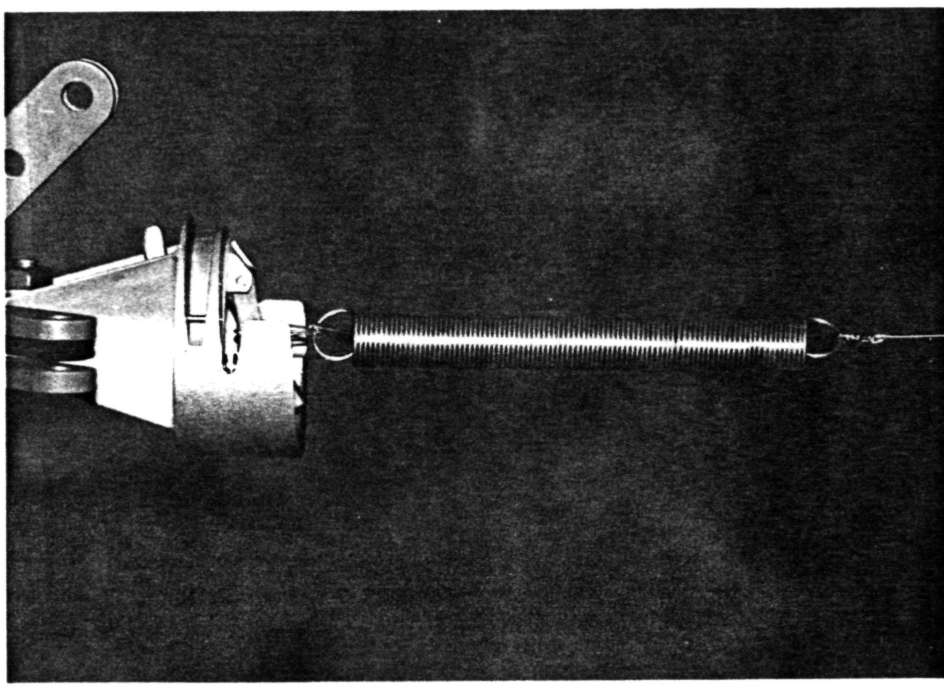

Figure 11. Helical spring attachment at end e

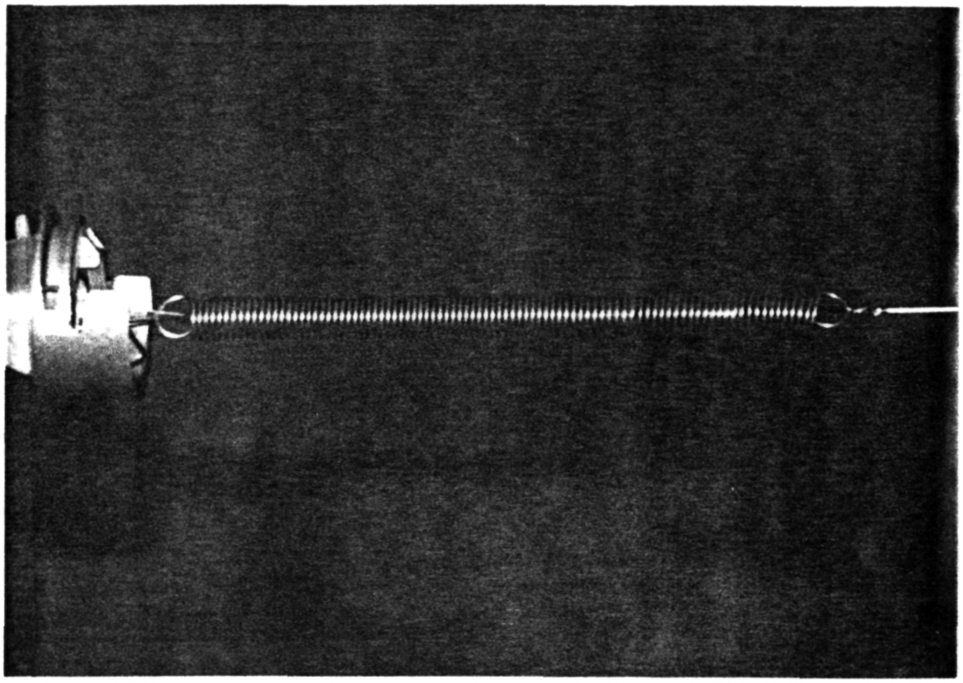

Figure 12. Stretched helical spring at end e

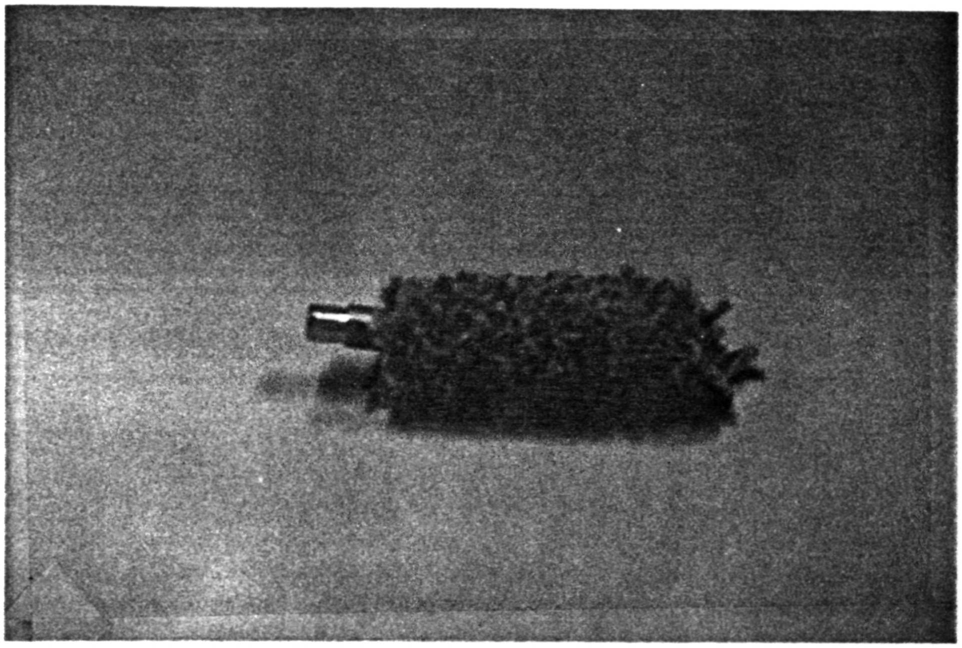

Figure 13. Wool swab damper

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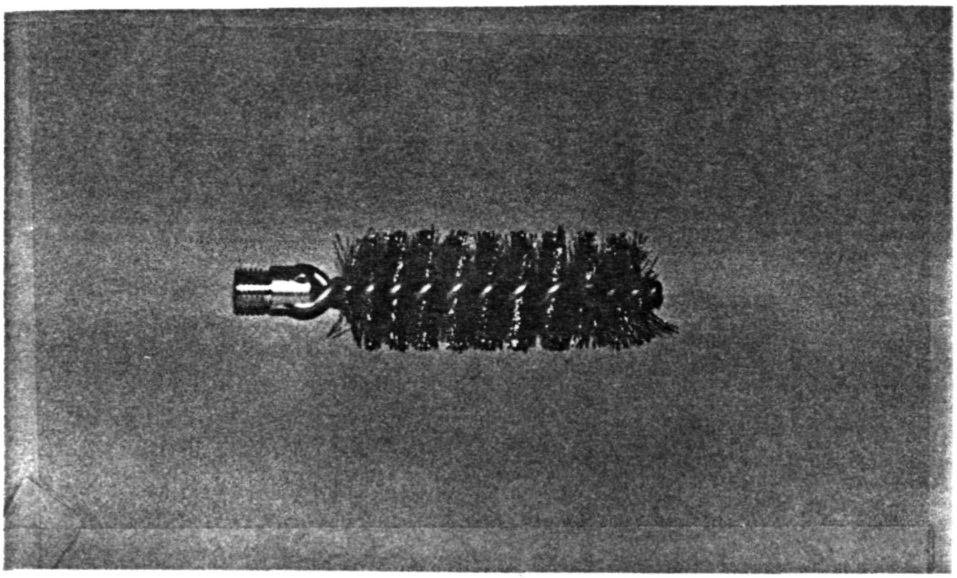

Figure 14. Copper brush damper

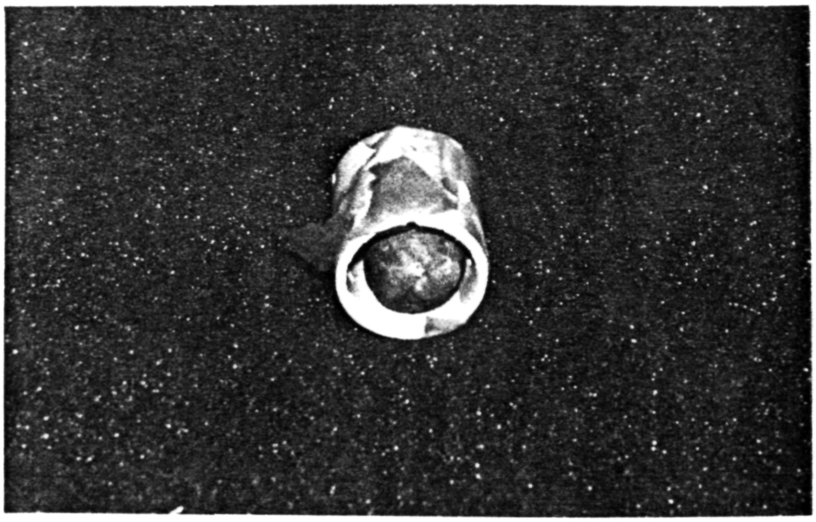

Figure 15. Silly putty in chamber damper

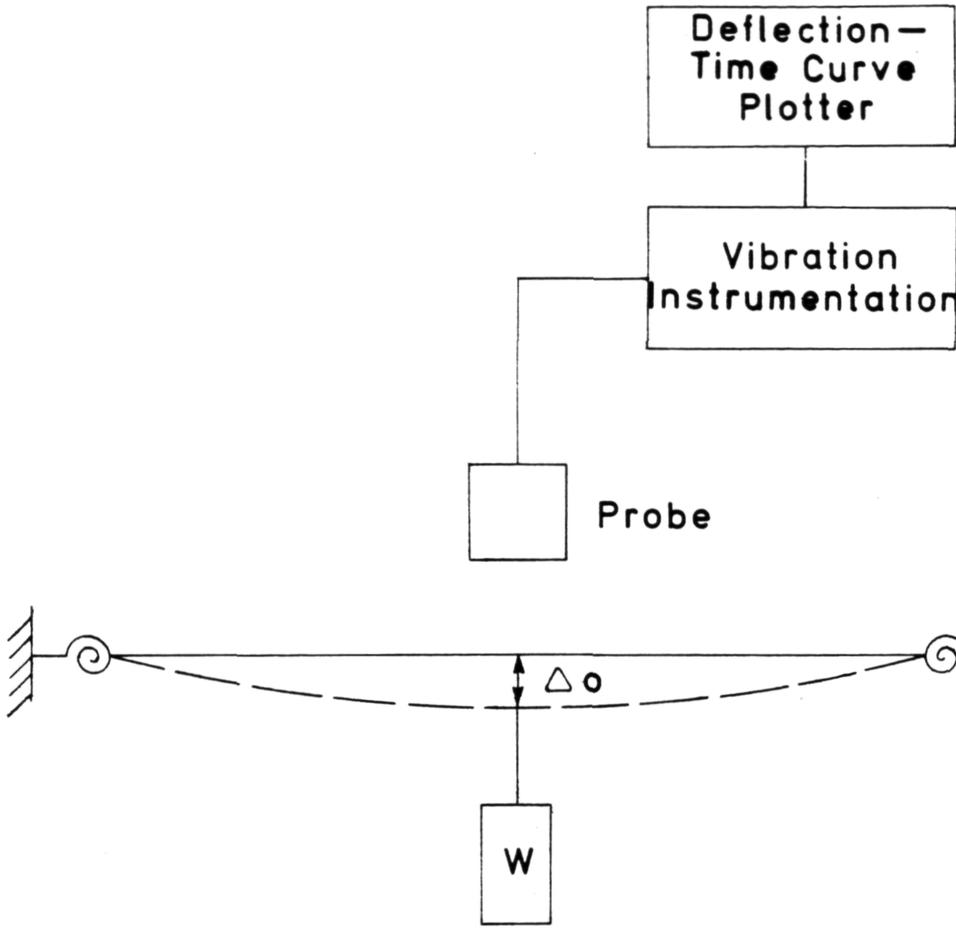

Figure 16. Schematic of member natural vibration setup

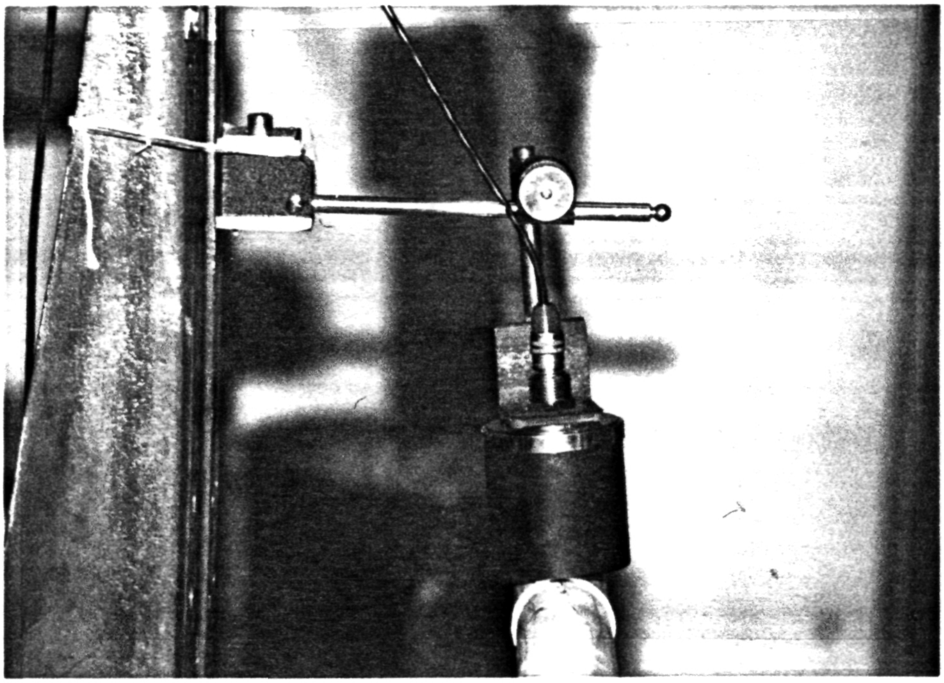

Figure 17. Proximity probe

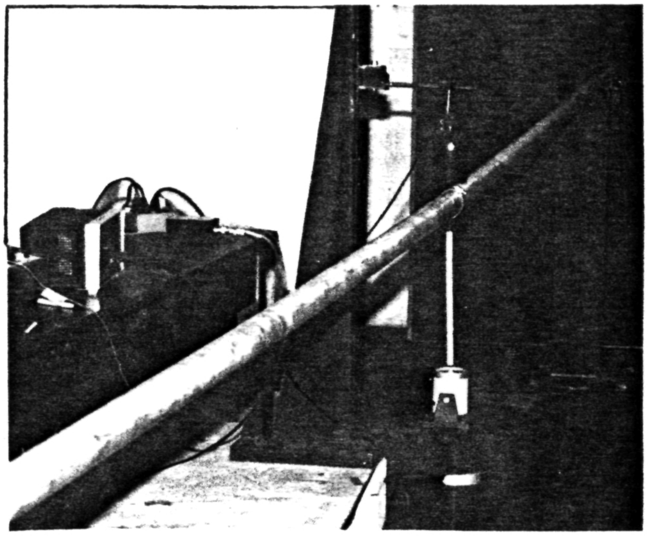

Figure 18. Member forced vibration setup

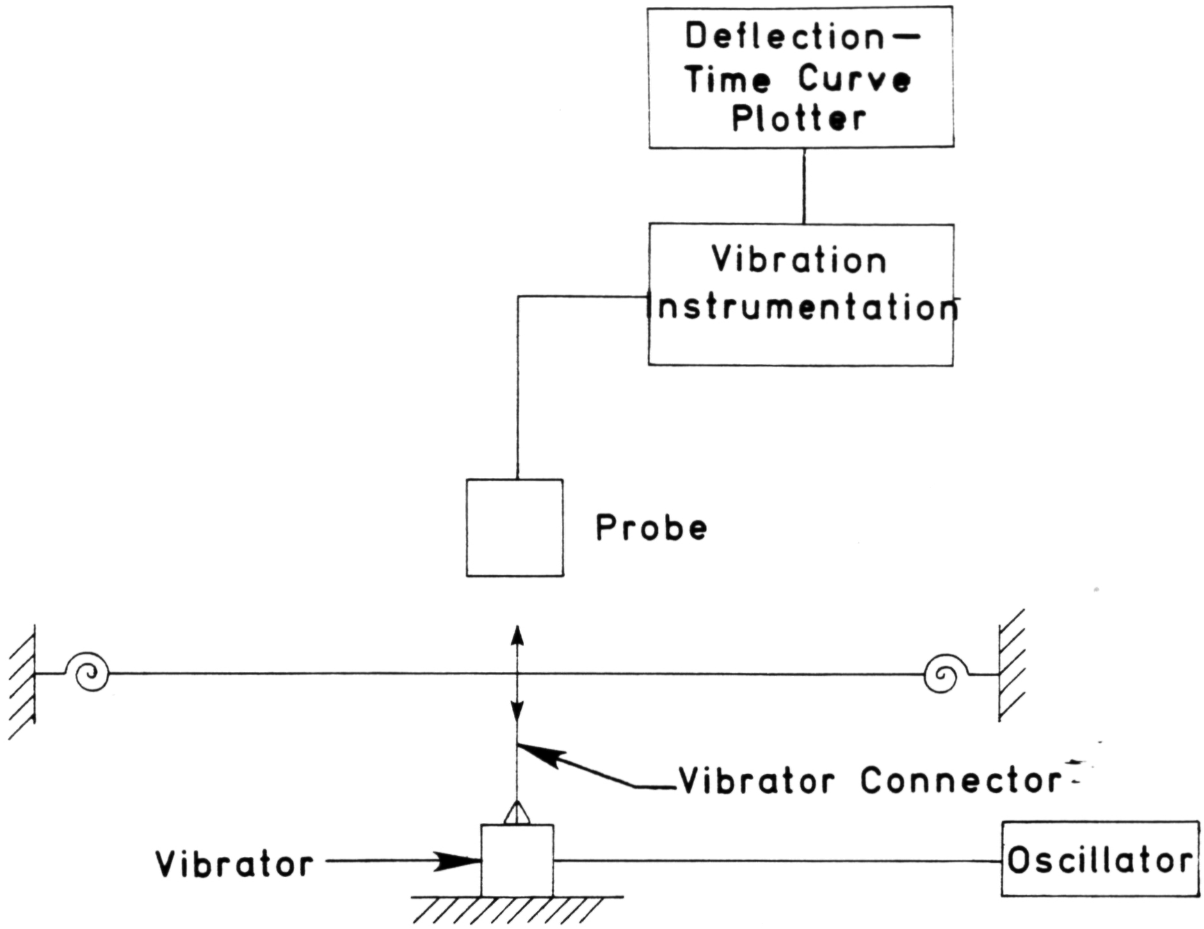

Figure 19. Schematic of member forced vibration setup

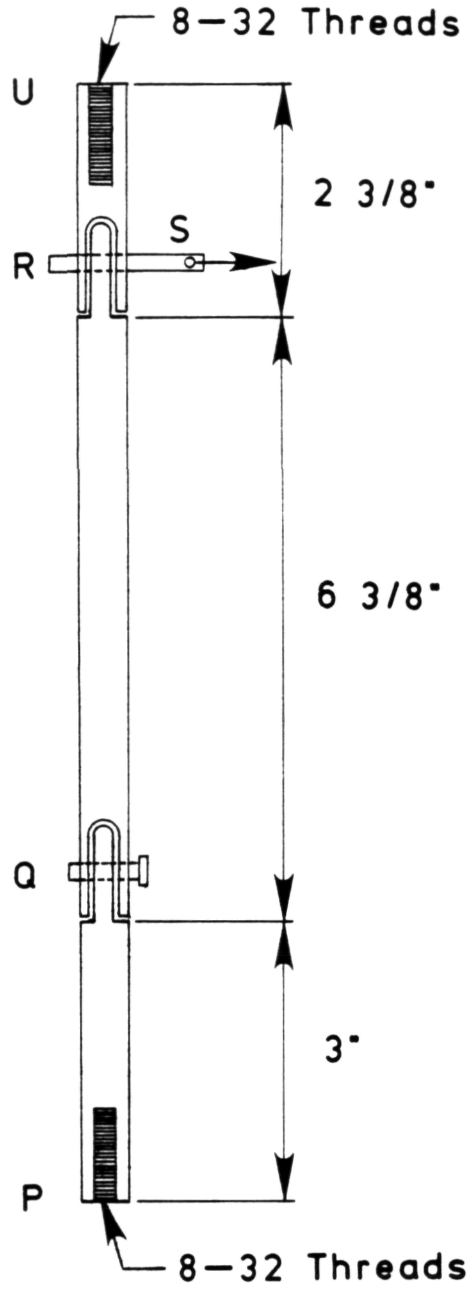

Figure 20. Vibrator connector details

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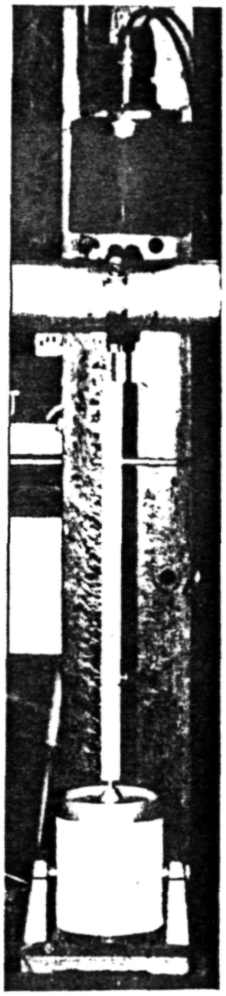
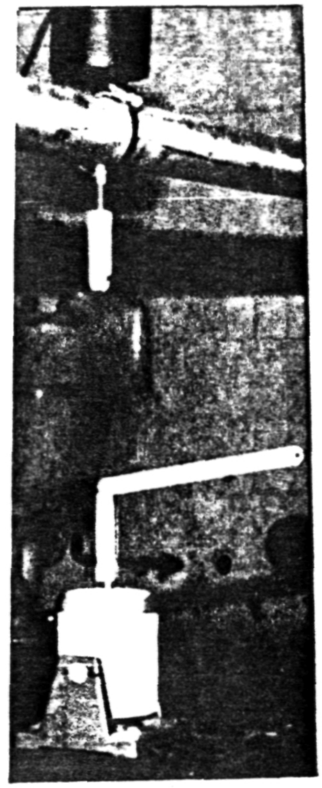


(a) Vibrator connector in engaged position



(b) Disengaged vibrator connector

Figure 21. Vibrator connector in engaged and disengaged positions

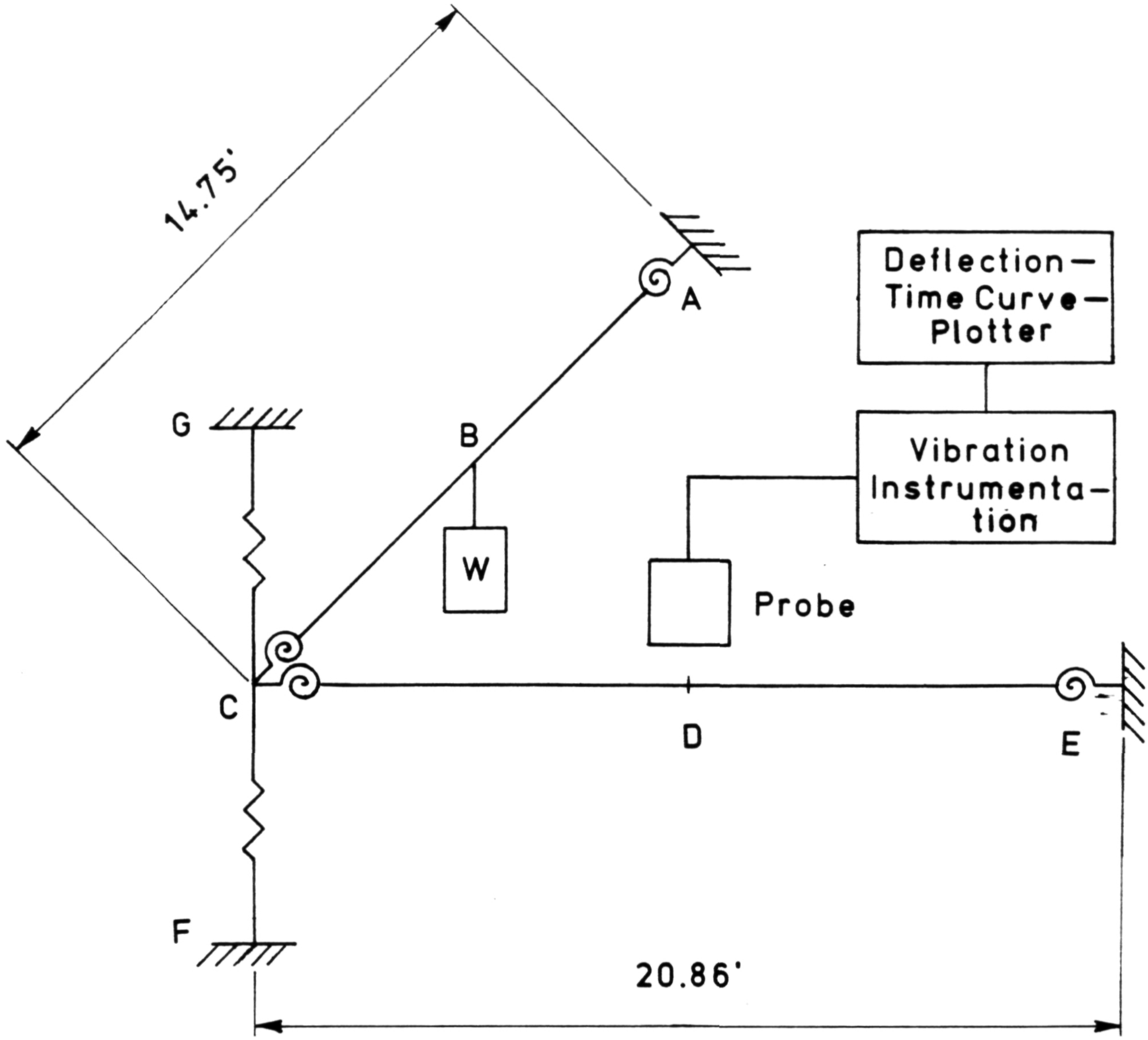


Figure 22. Schematic of grillage natural
vibration setup

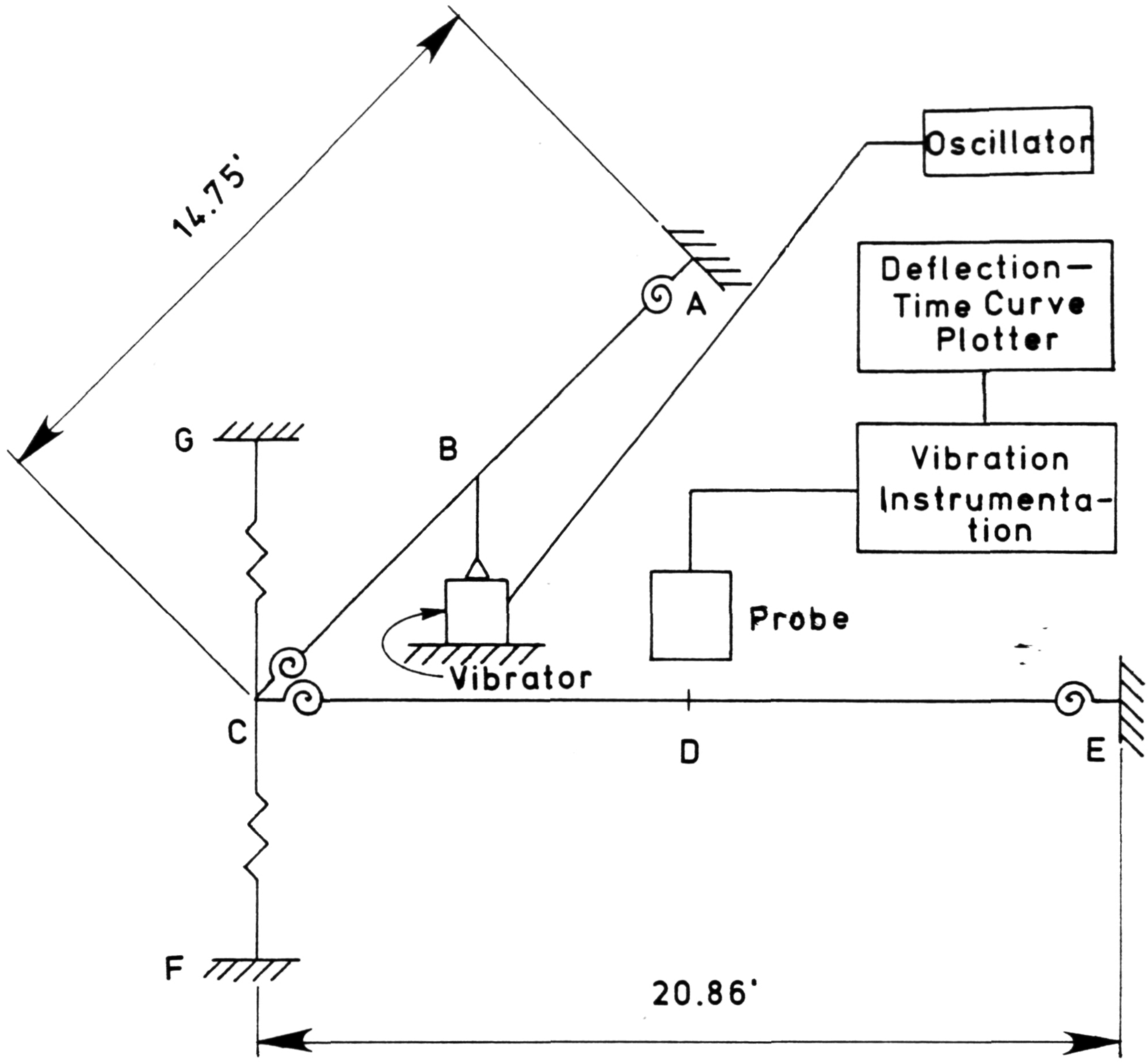


Figure 23. Schematic of grillage forced vibration setup

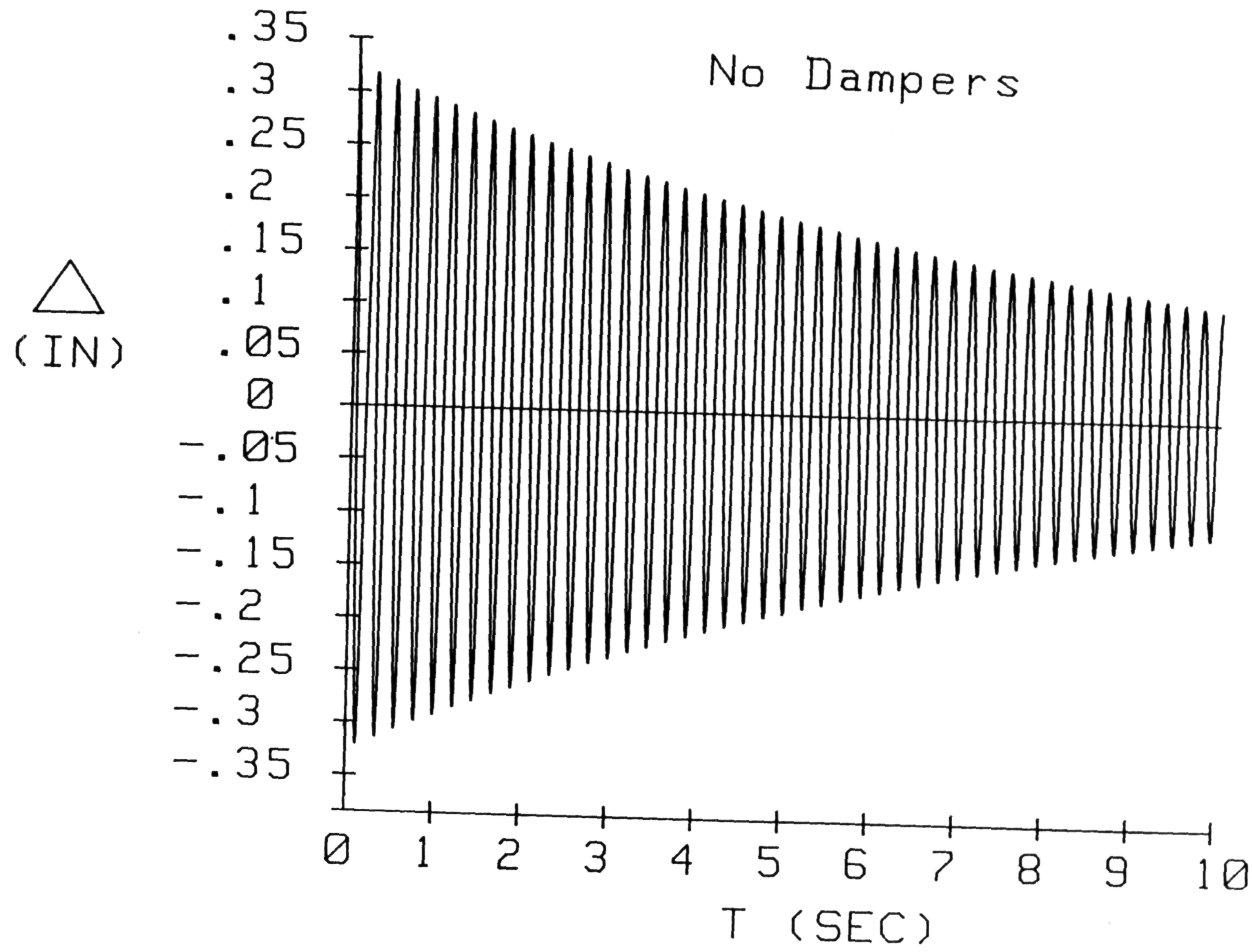


Figure 24. Average Δ - t plot for 20.86 ft. member with no dampers

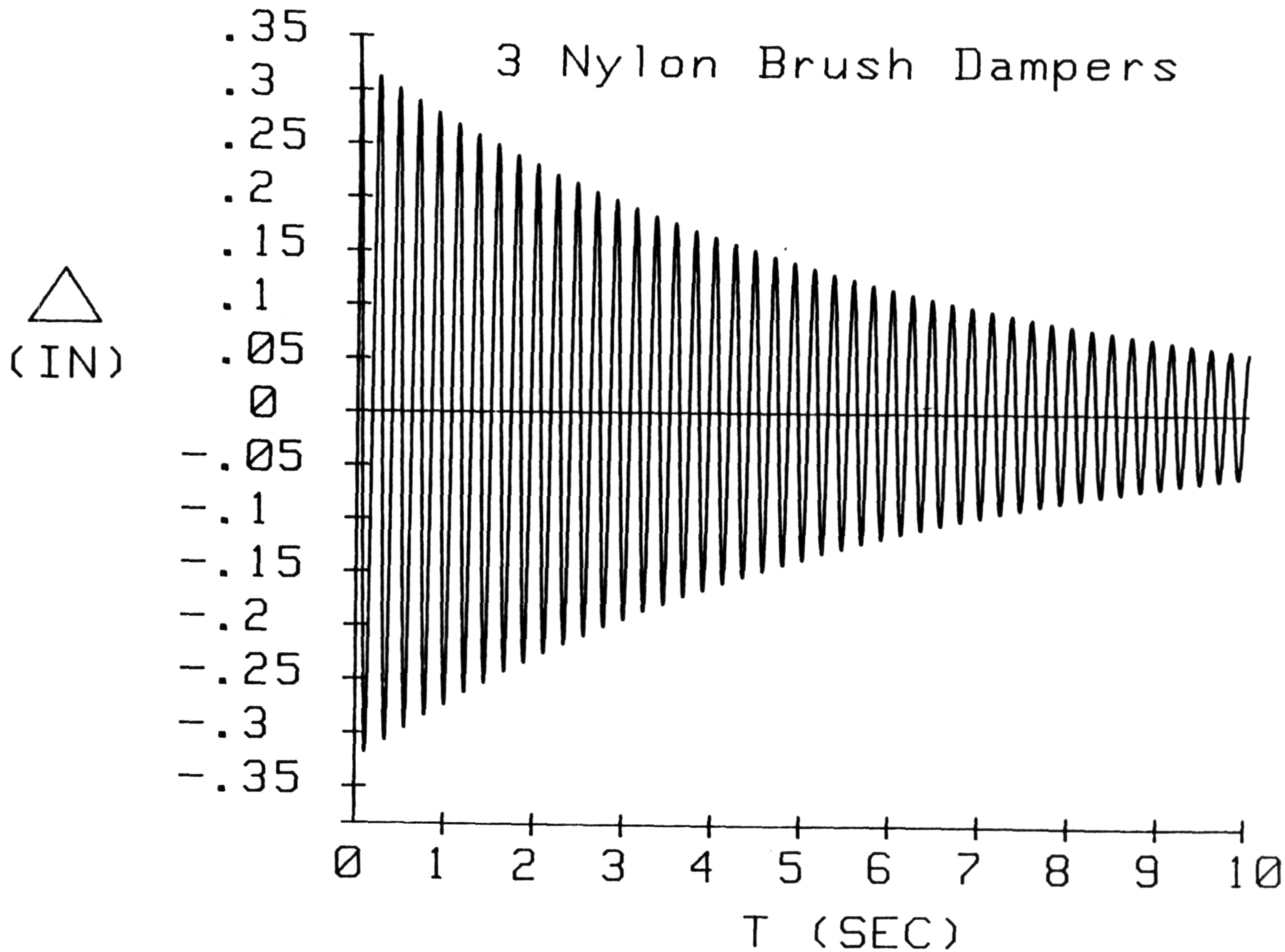


Figure 25. Average **Δ** - t plot for 20.86 ft. member with 3 nylon brush dampers

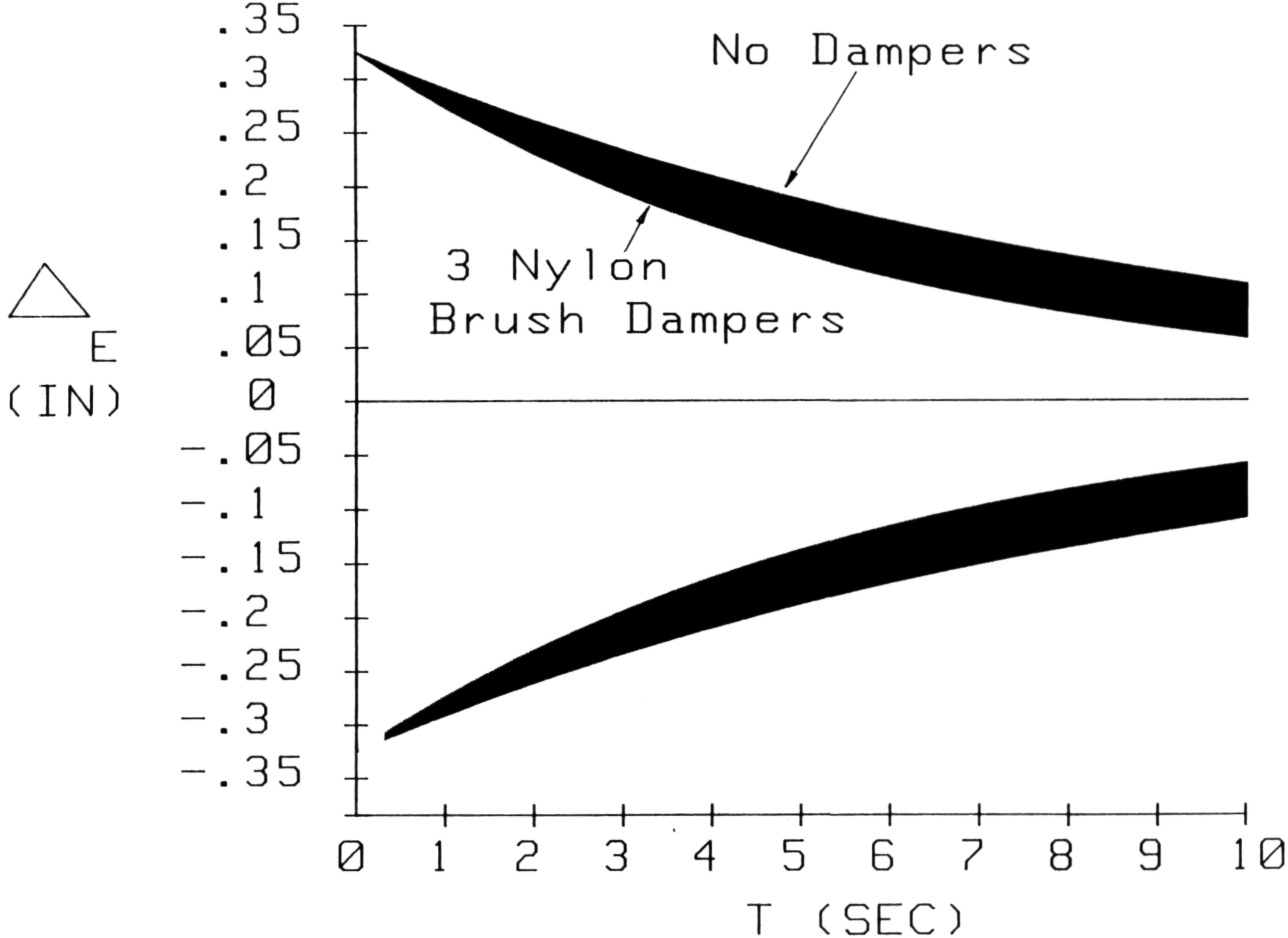


Figure 26. Effect of 3 nylon brush campers on deflection-time envelopes for 20.86 ft. member

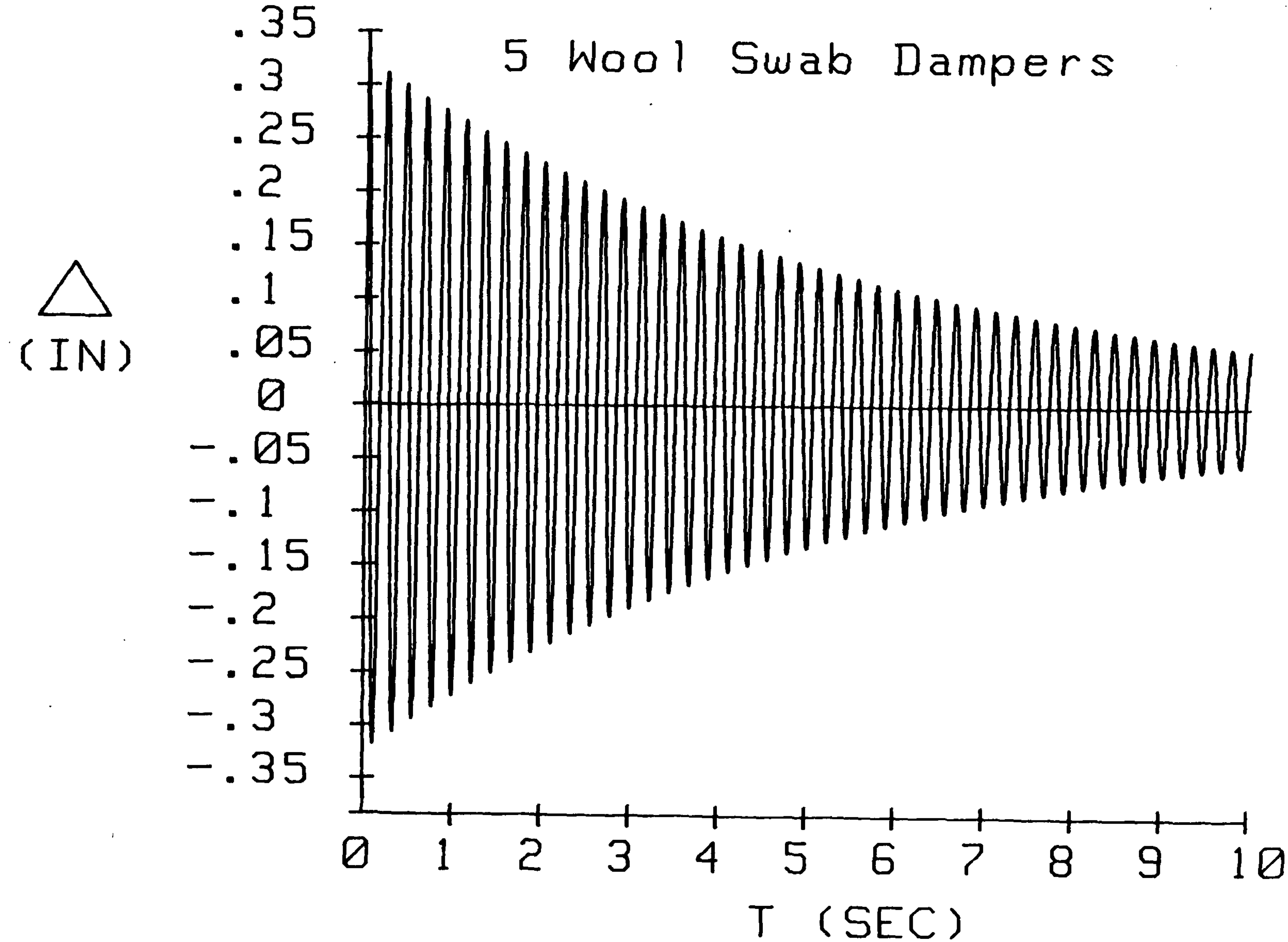


Figure 27. Average Δ - t plot for 20.86 ft. member with 5 wool swab dampers

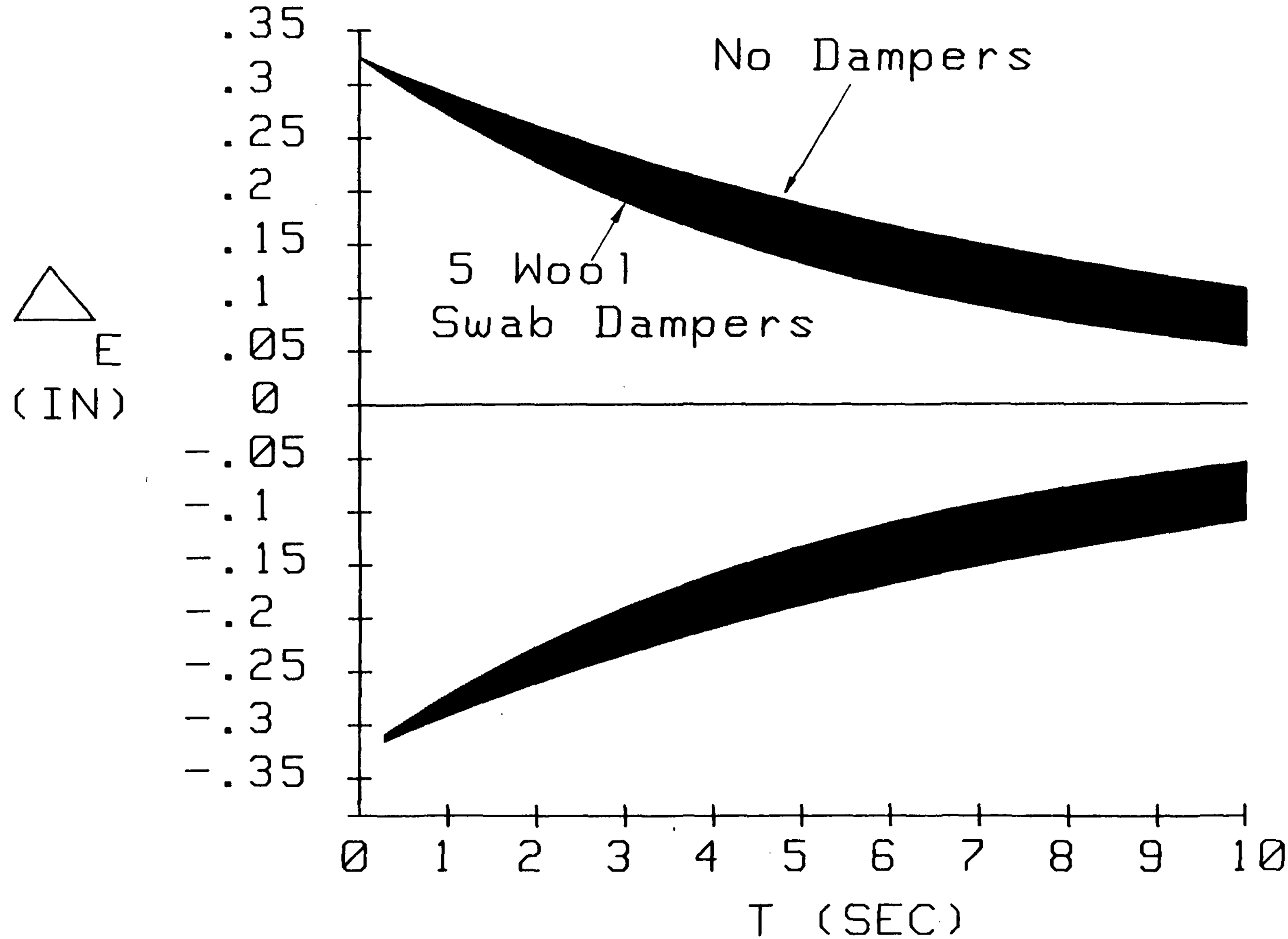


Figure 28. Effect of 5 wool swab dampers on deflection-time envelopes for 20.86 ft. member

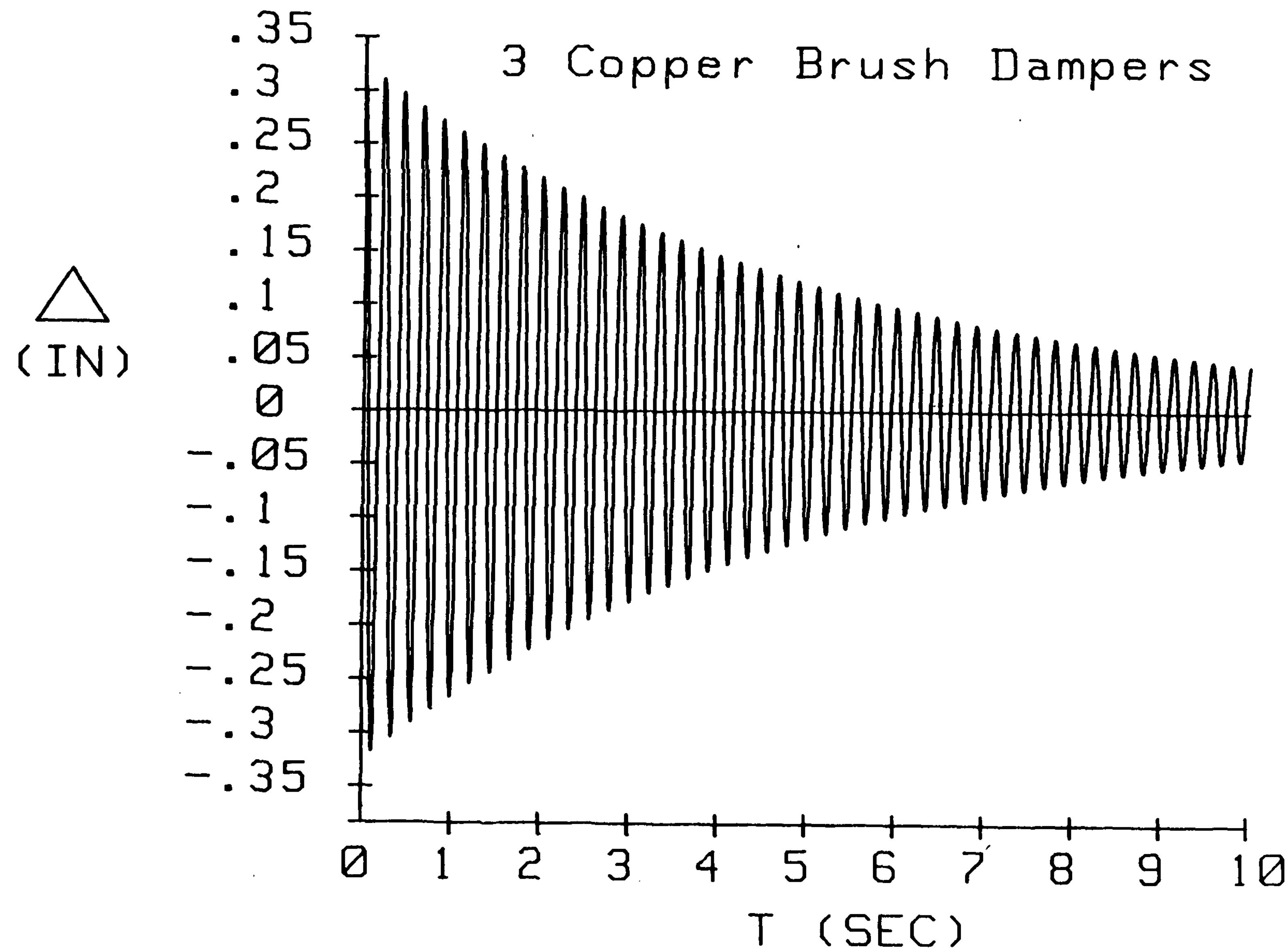


Figure 29. Average Δ - t plot for 20.86 ft. member with 3 copper brush dampers

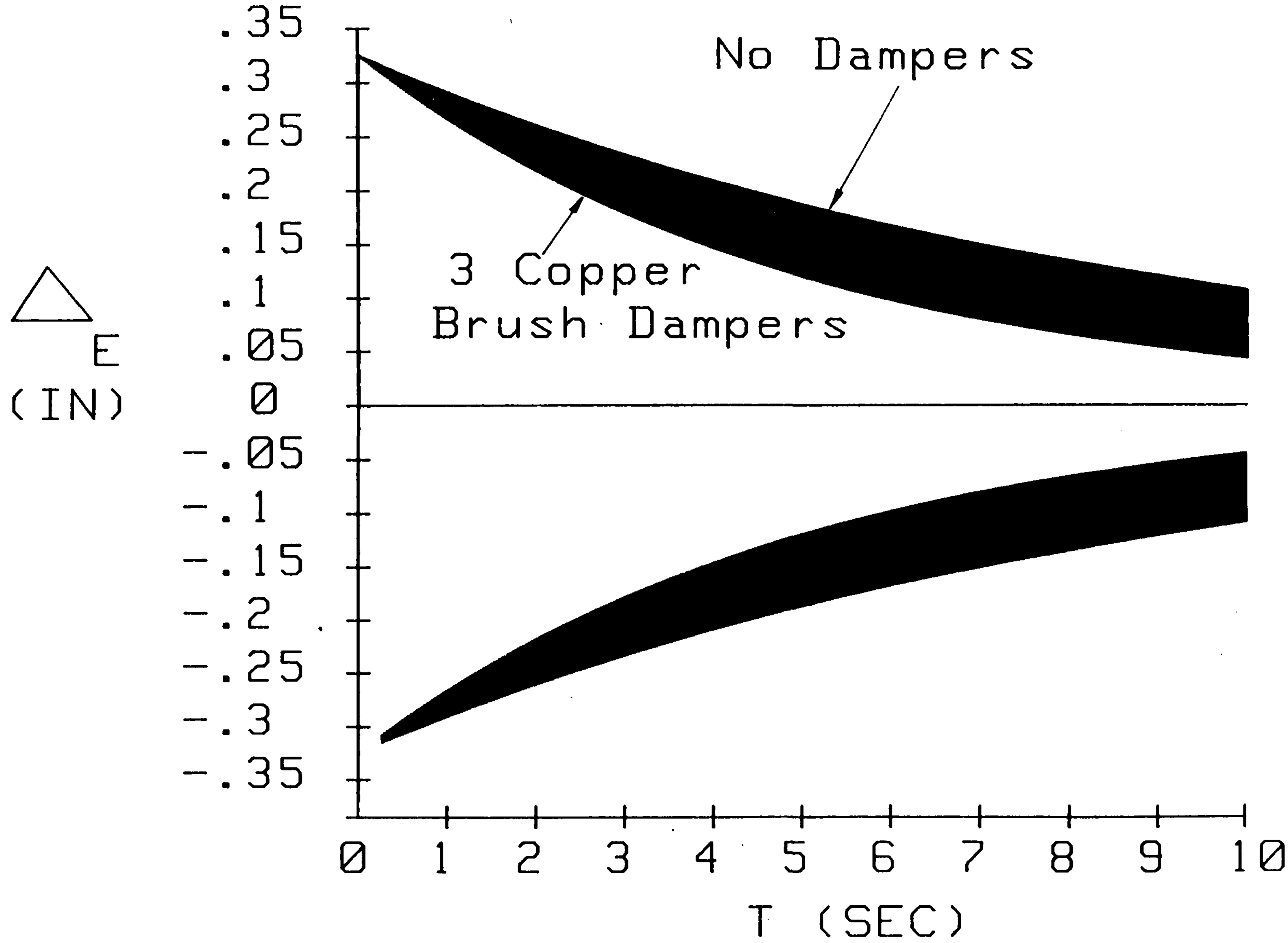


Figure 30. Effect of 3 copper brush dampers on deflection-time envelopes for 20.86 ft. member

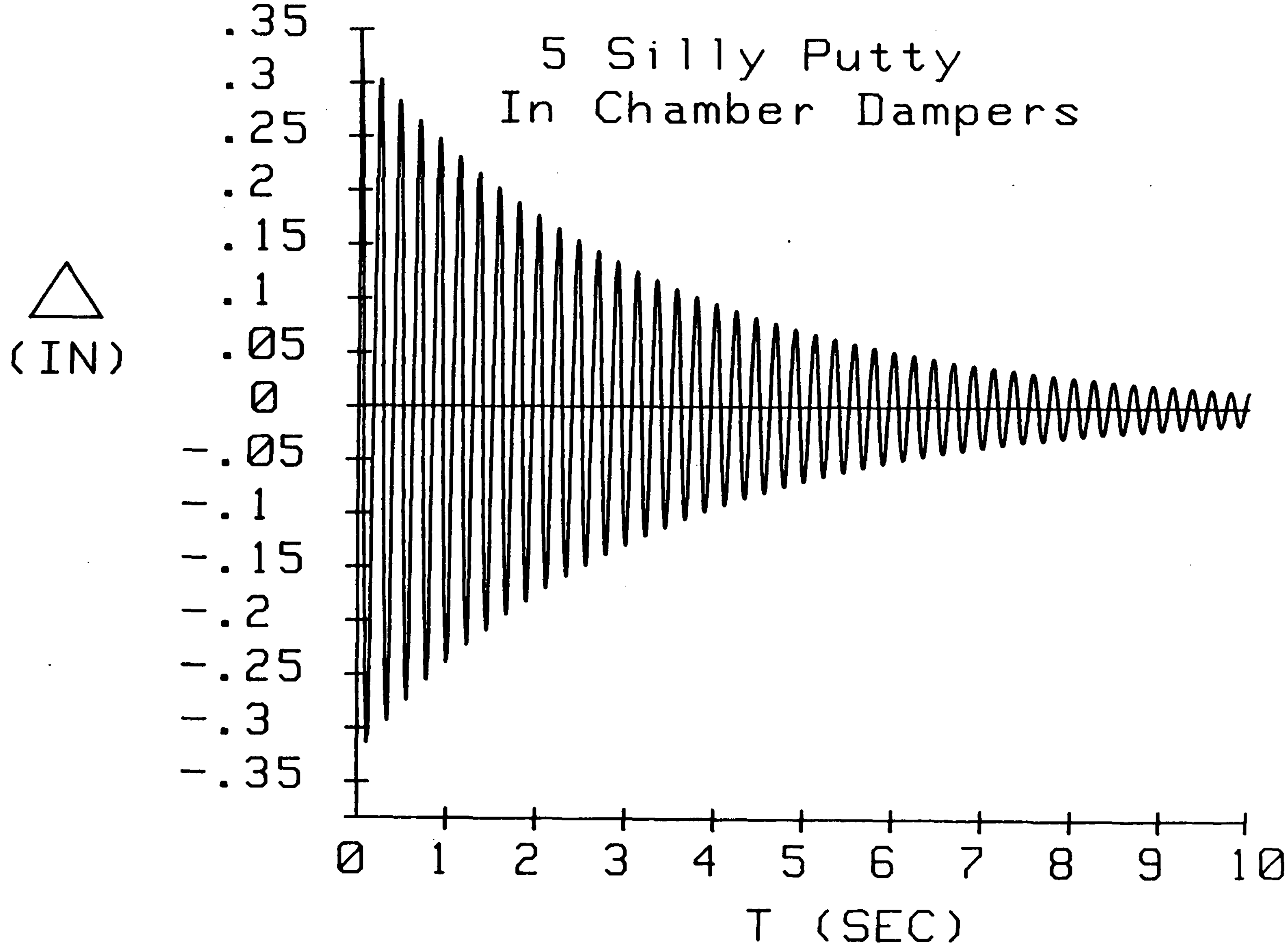


Figure 31. Average Δ - t plot for 20.86 ft. member with 5 silly putty in chamber dampers

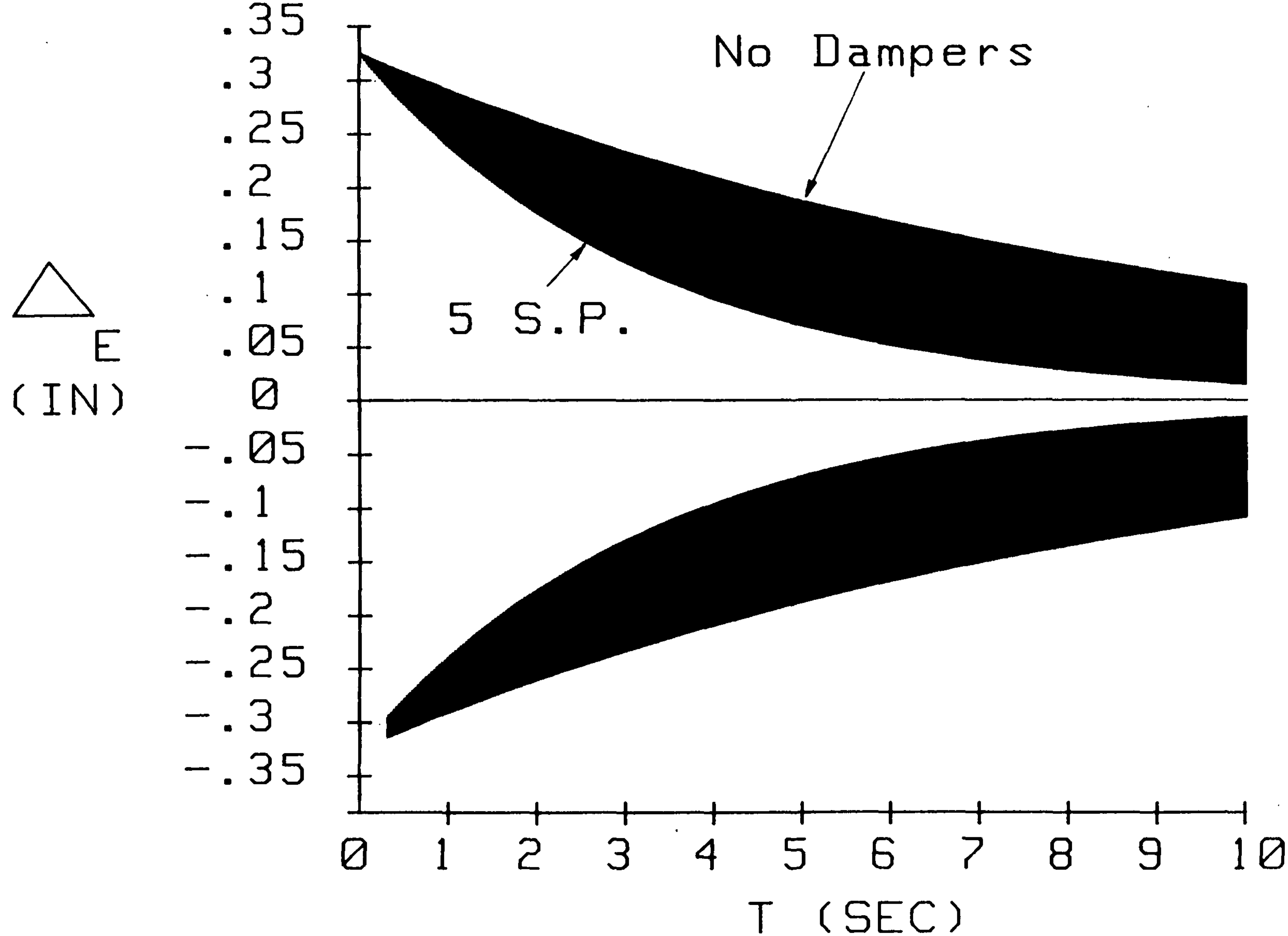


Figure 32. Effect of 5 silly putty in chamber dampers on deflection-time envelopes for 20.86 ft. member

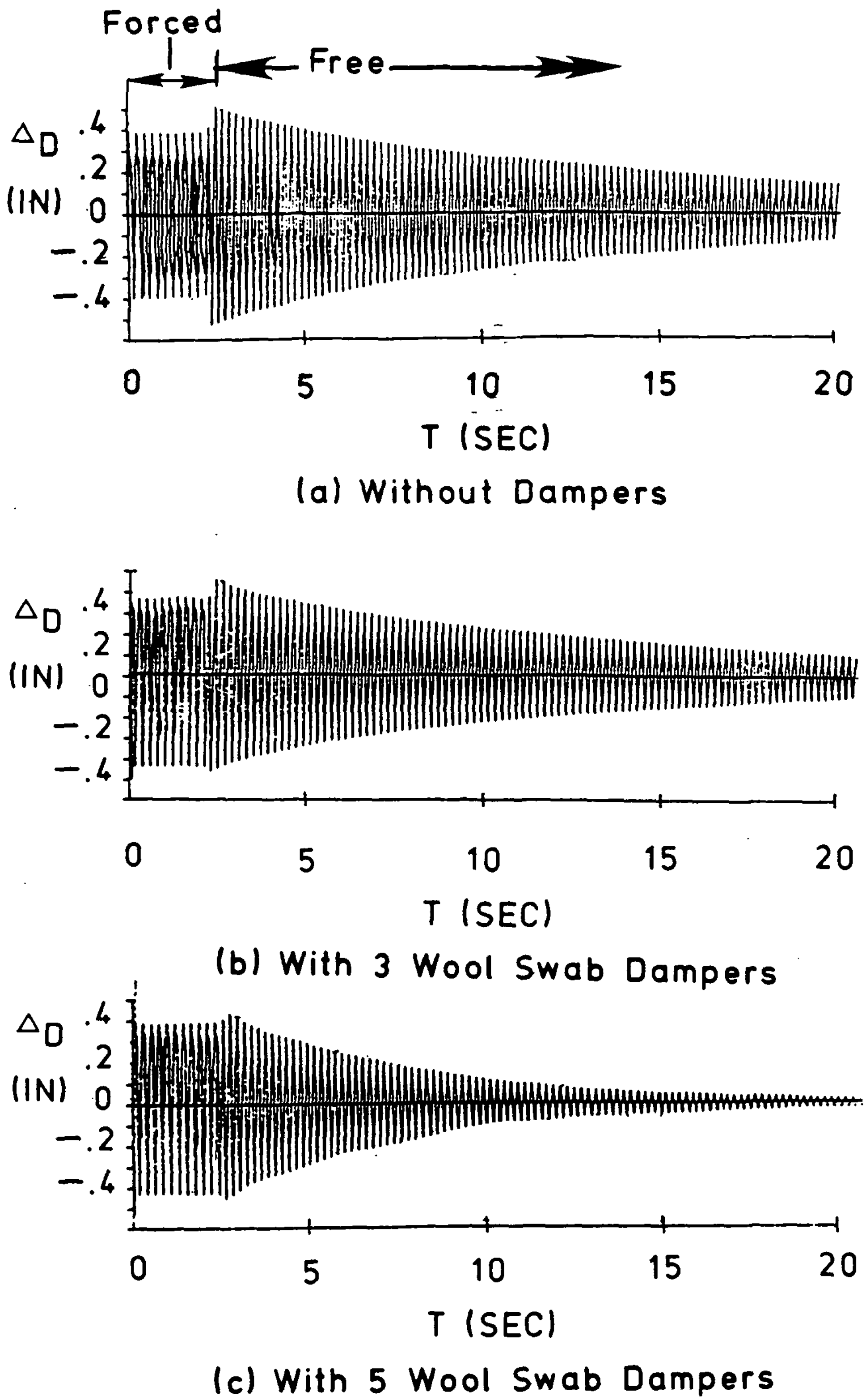
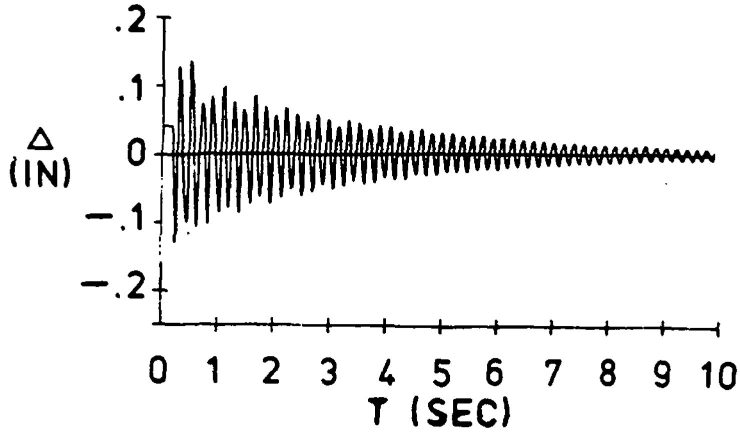
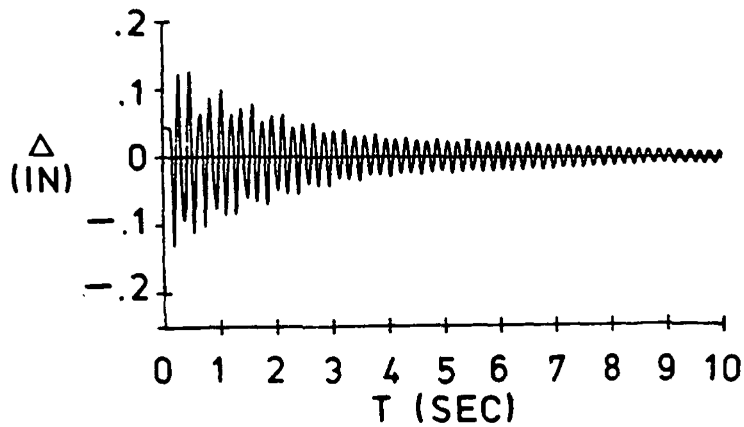


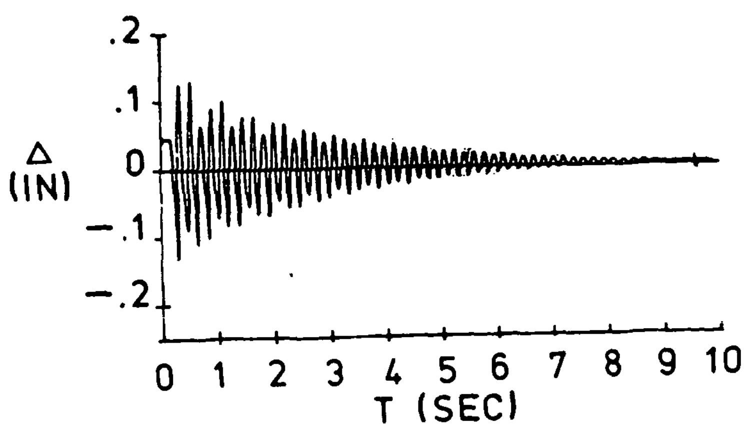
Figure 33. Experimental deflection-time relationships for member "constrained" forced and free vibration with forcing function frequency of 4.0 Hz



(a) Without Dampers

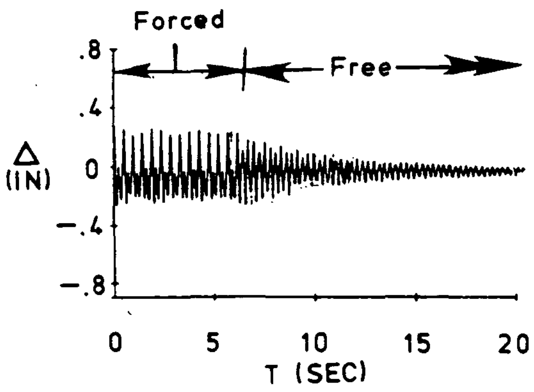


(b) With 5 Silly Putty

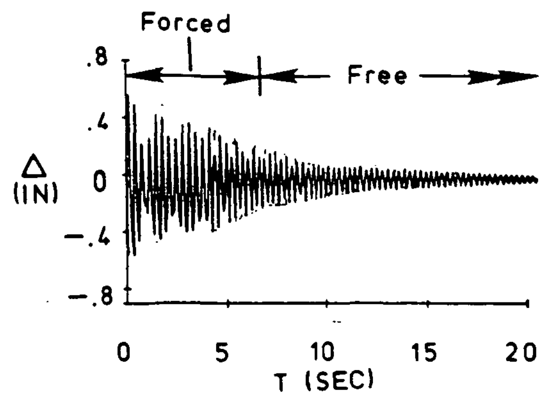


(c) With 5 Wool Swab

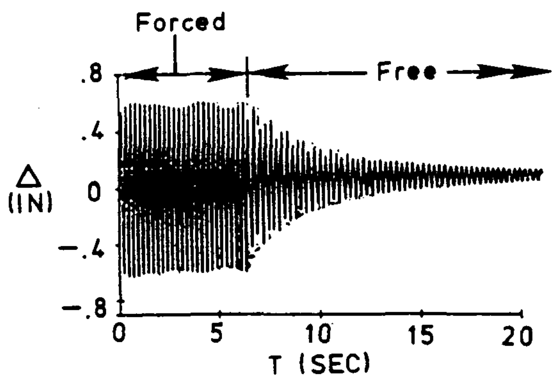
Figure 34. Experimental deflection-time relationship for natural grillage vibration with no dampers, 5 silly putty in chamber, and 5 wool swab dampers



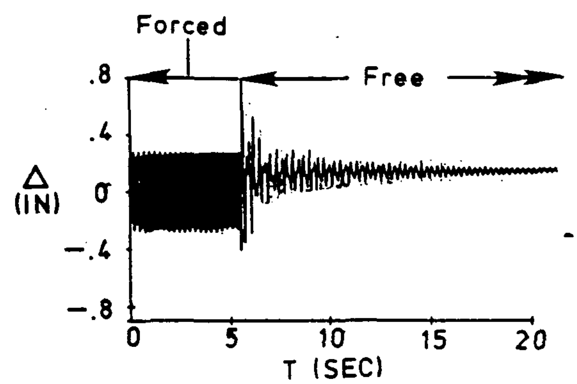
(a) 2 Hz.



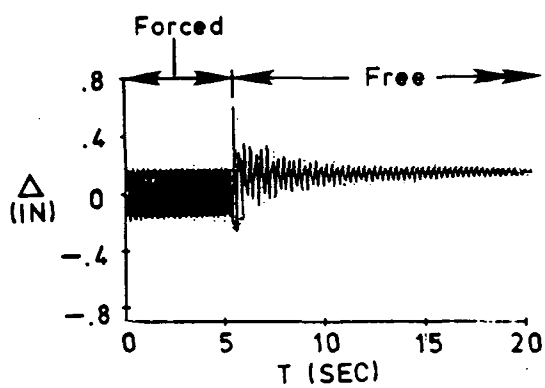
(b) 3 Hz.



(c) 4 Hz.



(d) 8 Hz.



(e) 9 Hz.

Figure 35. Experimental deflection-time relationships for grillage forced and free vibration with no dampers and with forcing function frequencies of 2, 3, 4, 8, and 9 Hz

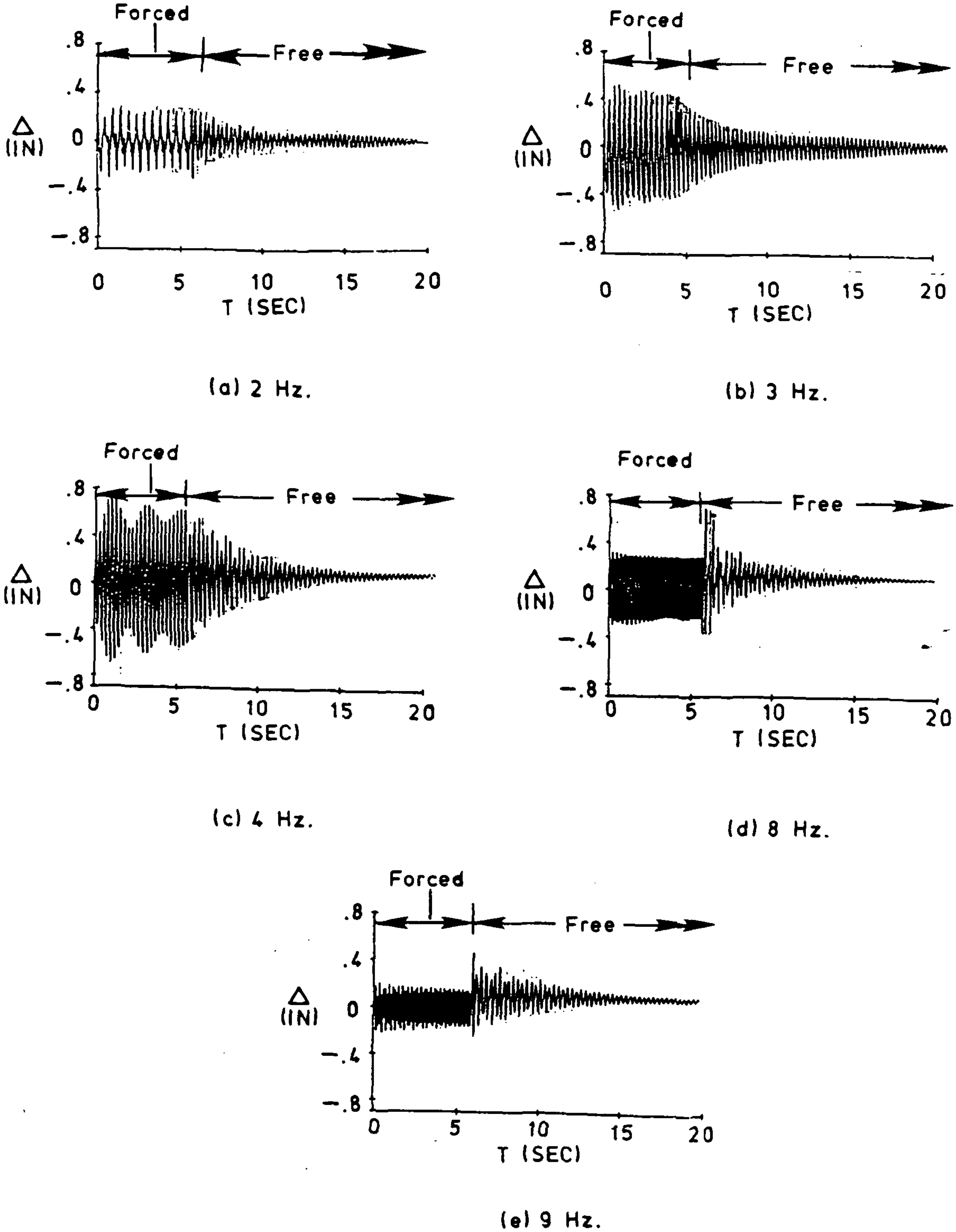
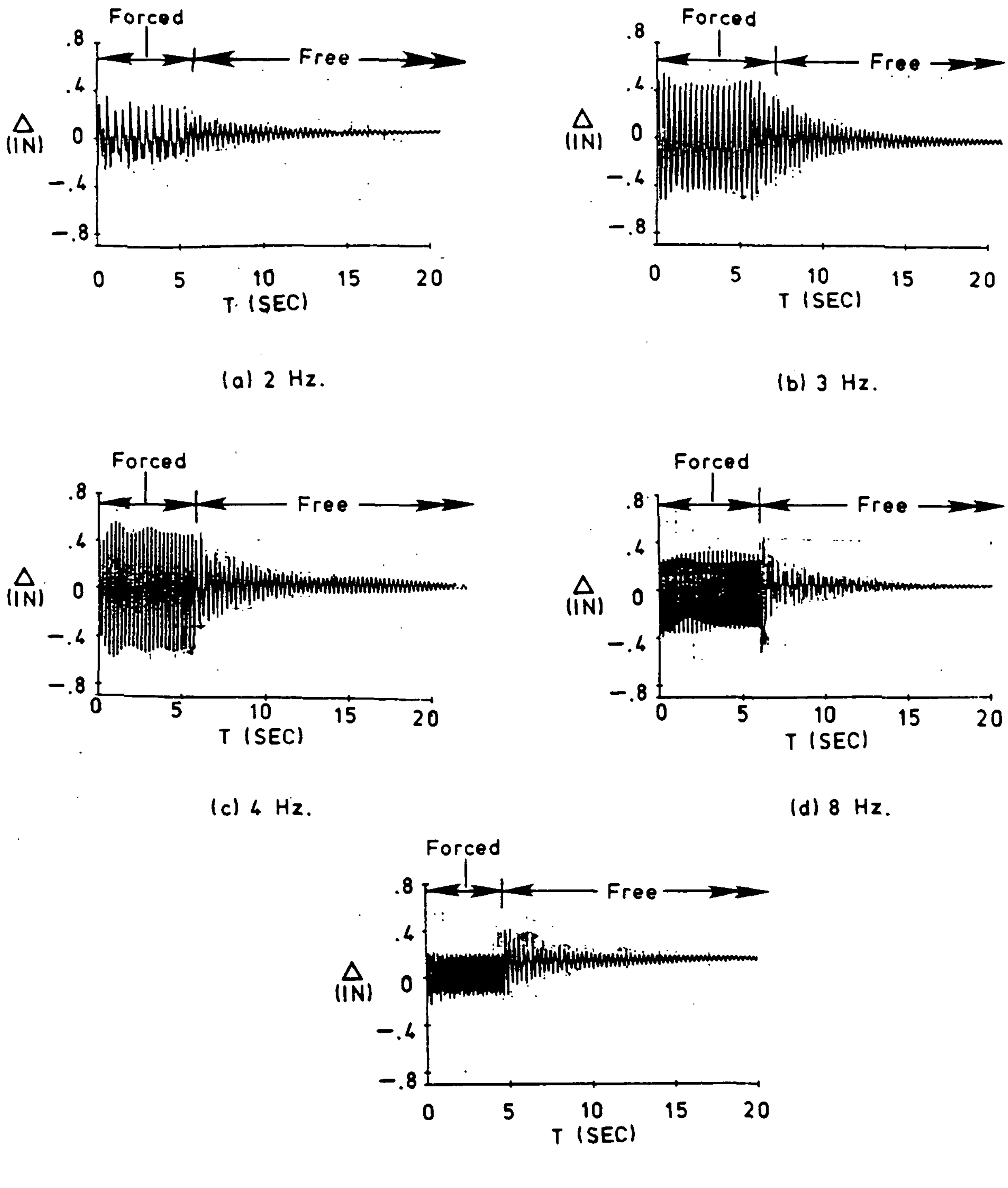


Figure 36. Experimental deflection-time relationships for grillage forced and free vibration with 5 silly putty in chamber dampers and with forcing function frequencies of 2, 3, 4, 8, and 9 Hz



(a) 2 Hz.

(b) 3 Hz.

(c) 4 Hz.

(d) 8 Hz.

(e) 9 Hz.

Figure 37. Experimental deflection-time relationships for grillage forced and free vibration with 5 wool swab dampers and with forcing function frequencies of 2, 3, 4, 8, and 9 Hz

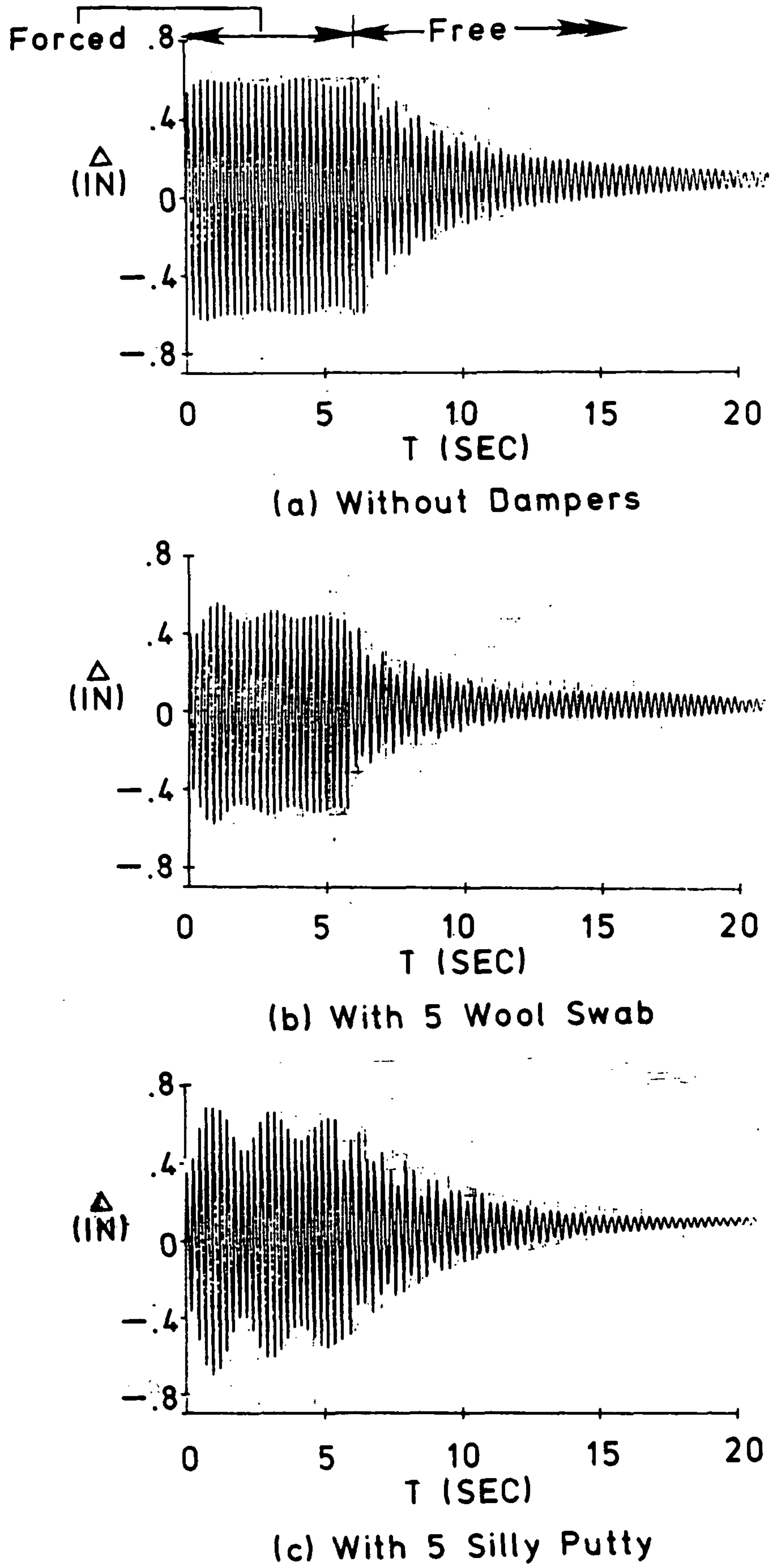
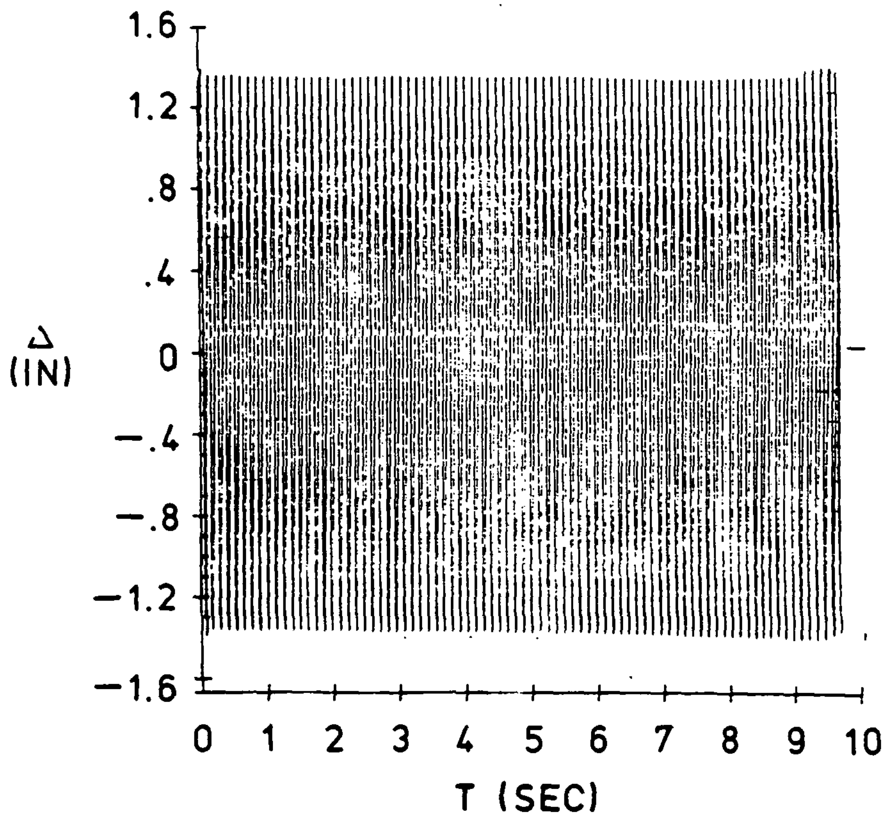


Figure 38. Comparison of the experimental deflection-time relationships with no dampers, 5 wool swab, and 5 silly putty in chamber dampers, and with forcing function frequency of 4.0 Hz. for grillage forced and free vibration



(a) Without Dampers

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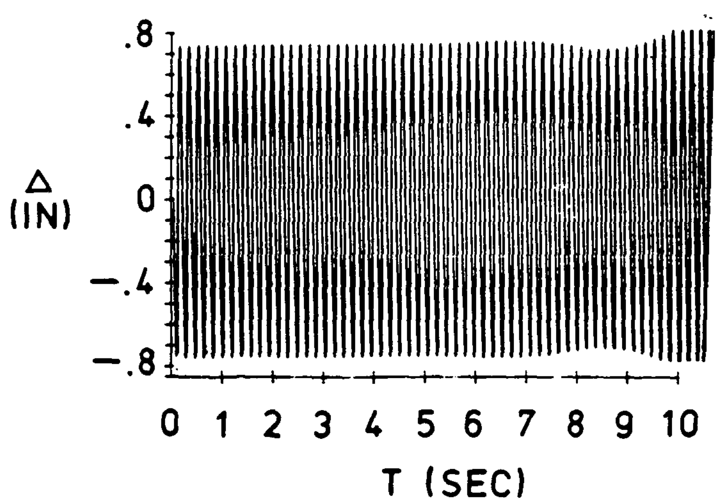

(b) With 5 Silly Putty

Figure 39. Experimental Δ - t relationships for grillage forced vibration at resonance with no dampers and with 5 silly putty in chamber dampers

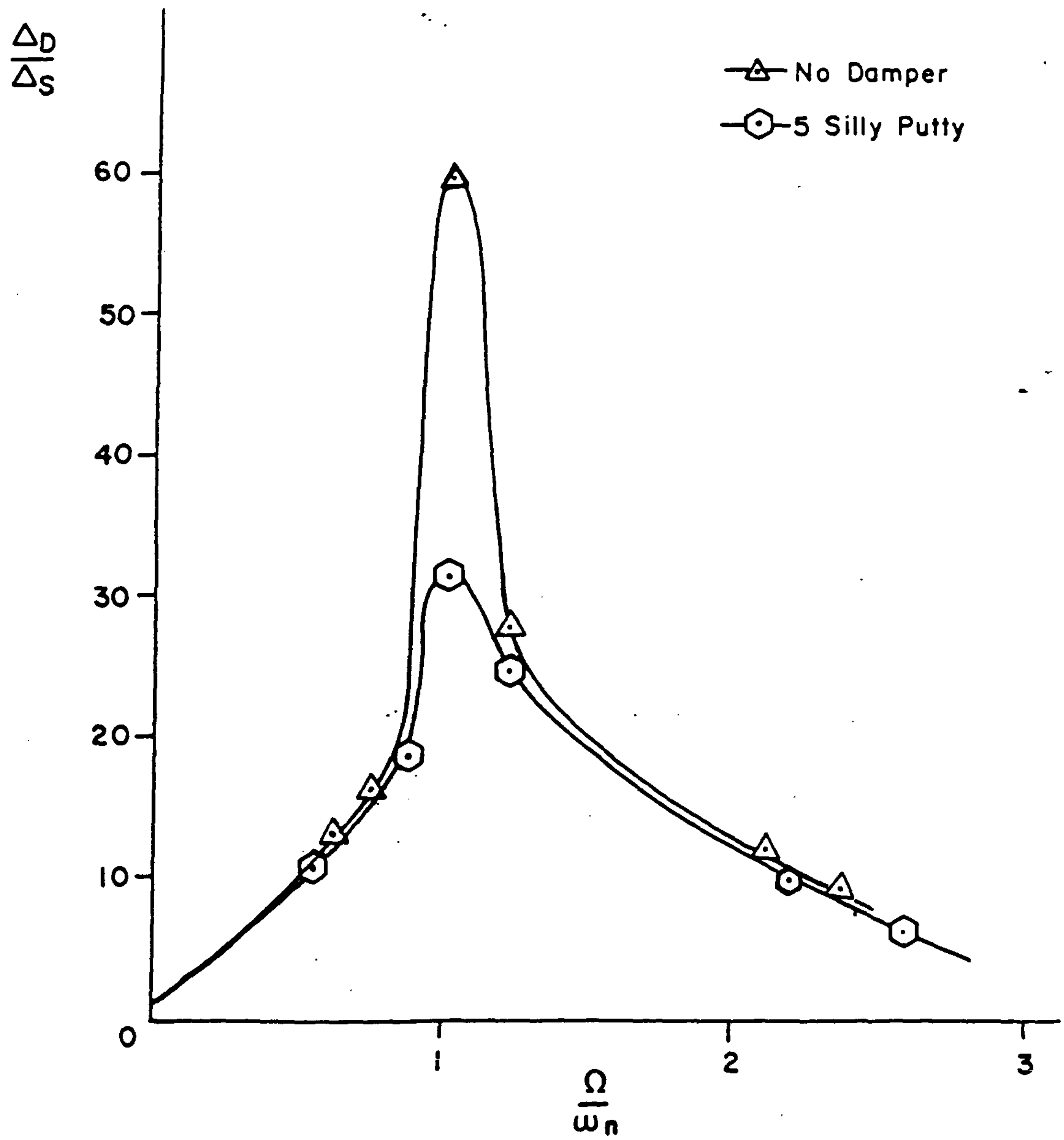

Figure 40. DMF versus frequency ration for no dampers and 5 silly putty in chamber dampers

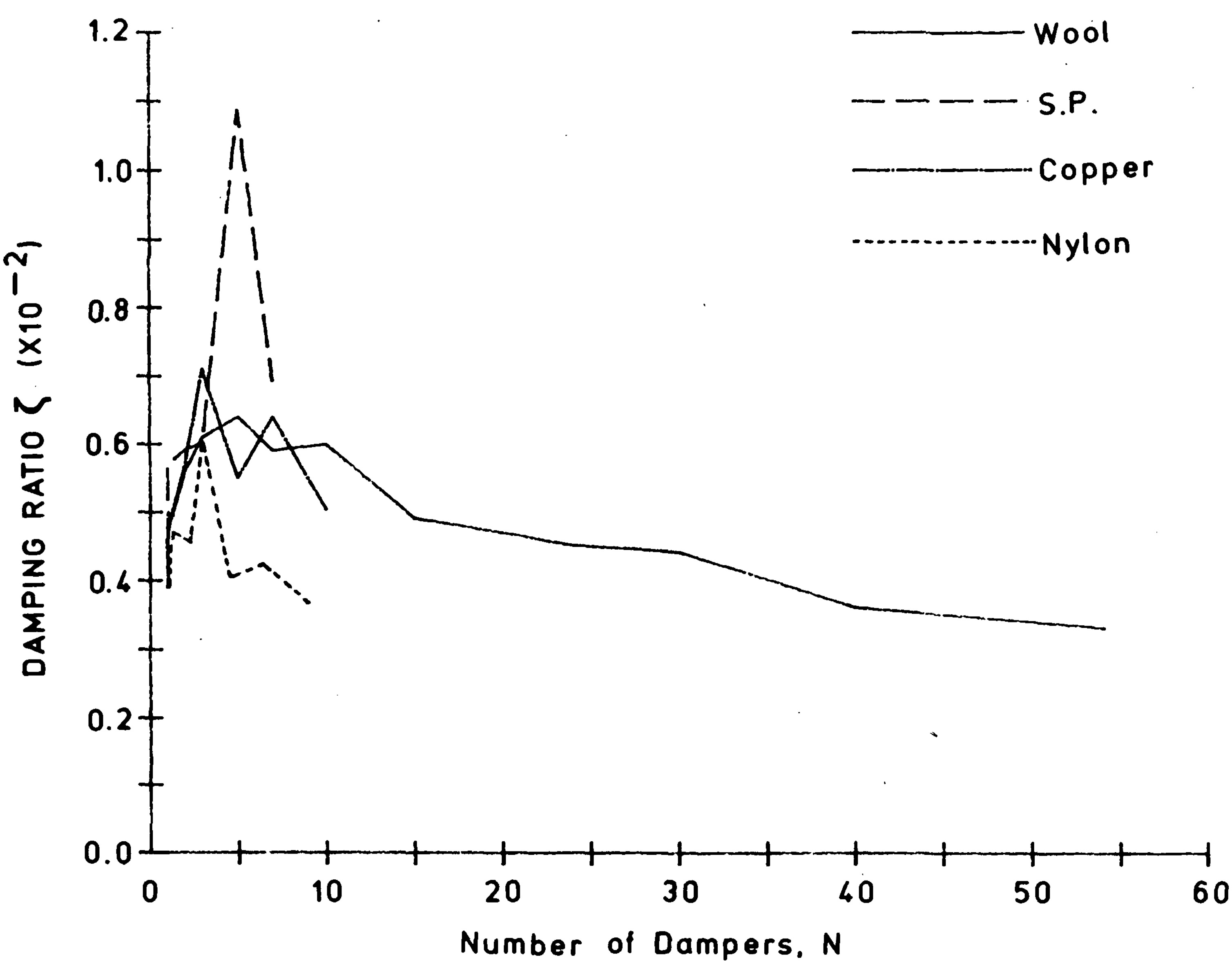

Figure 41. Damping ratio curves for 20.86 ft. member with wool swab, silly putty, copper brush, and nylon brush dampers

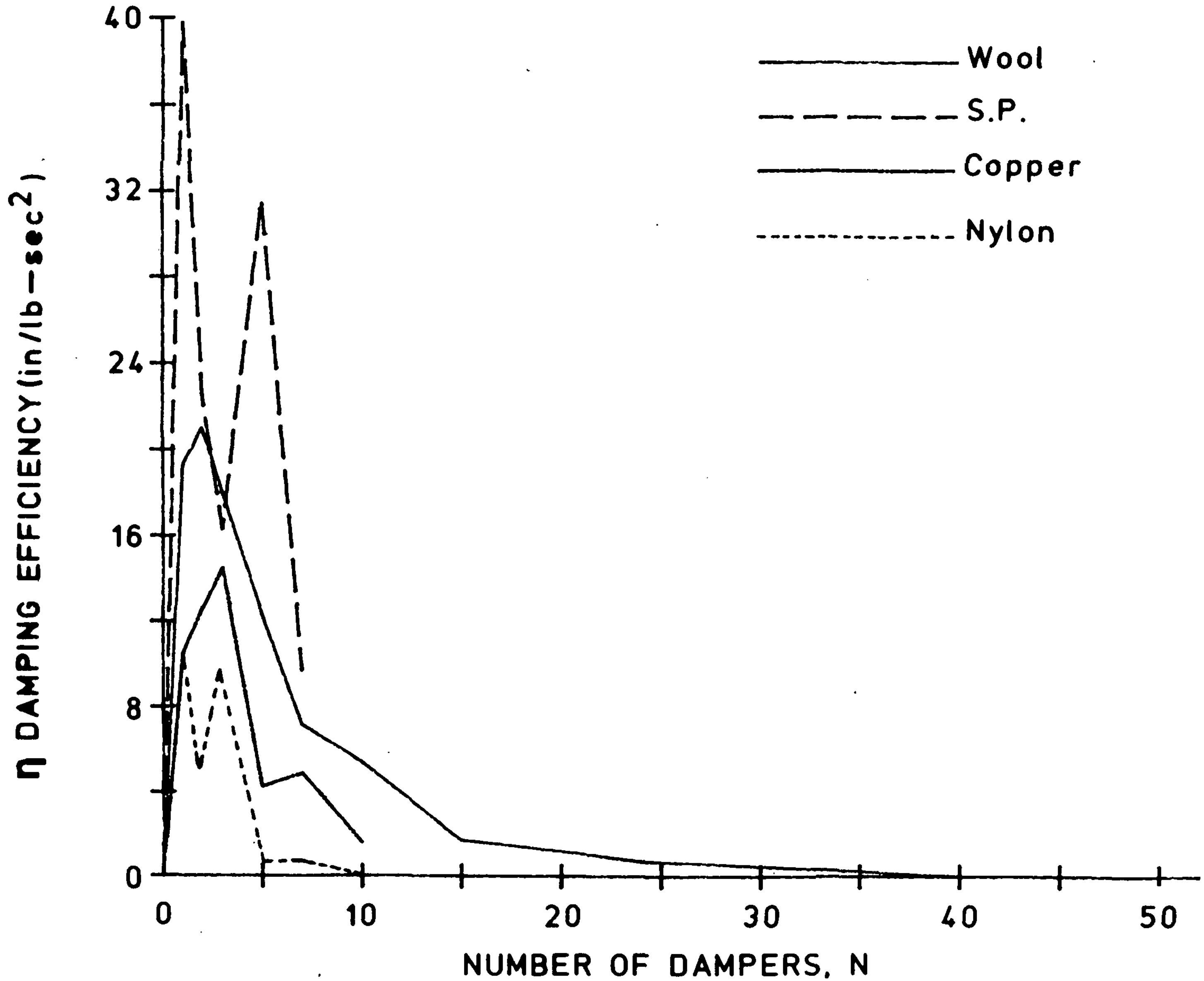

Figure 42. Damping efficiency index curves for 20.86 ft. member with wool swab, silly putty, copper brush, and nylon brush dampers

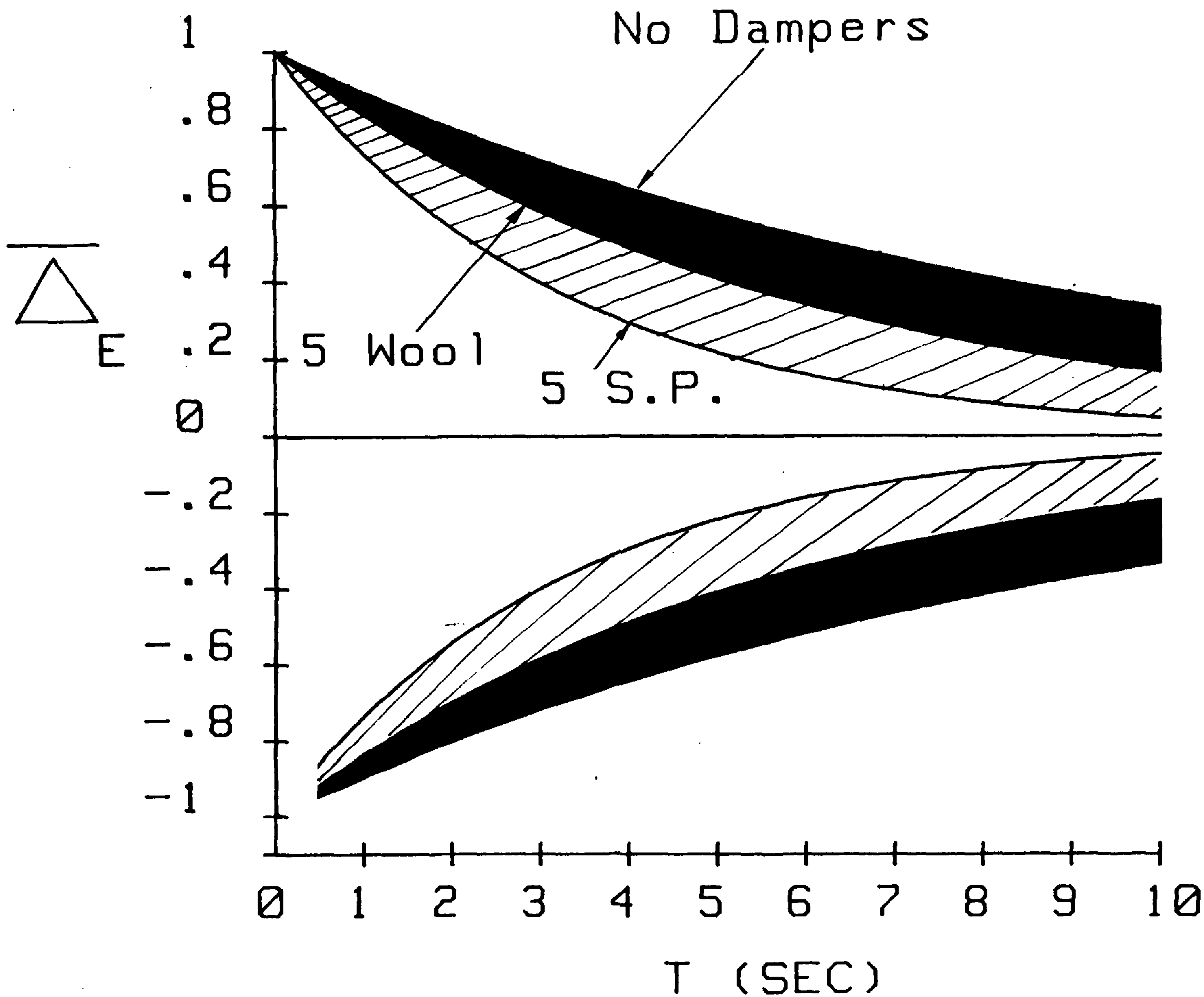

Figure 43. Comparison of dimensionless deflection-time envelopes for 20.86 ft. member with no dampers to those with 5 wool swab and 5 silly putty in chamber dampers

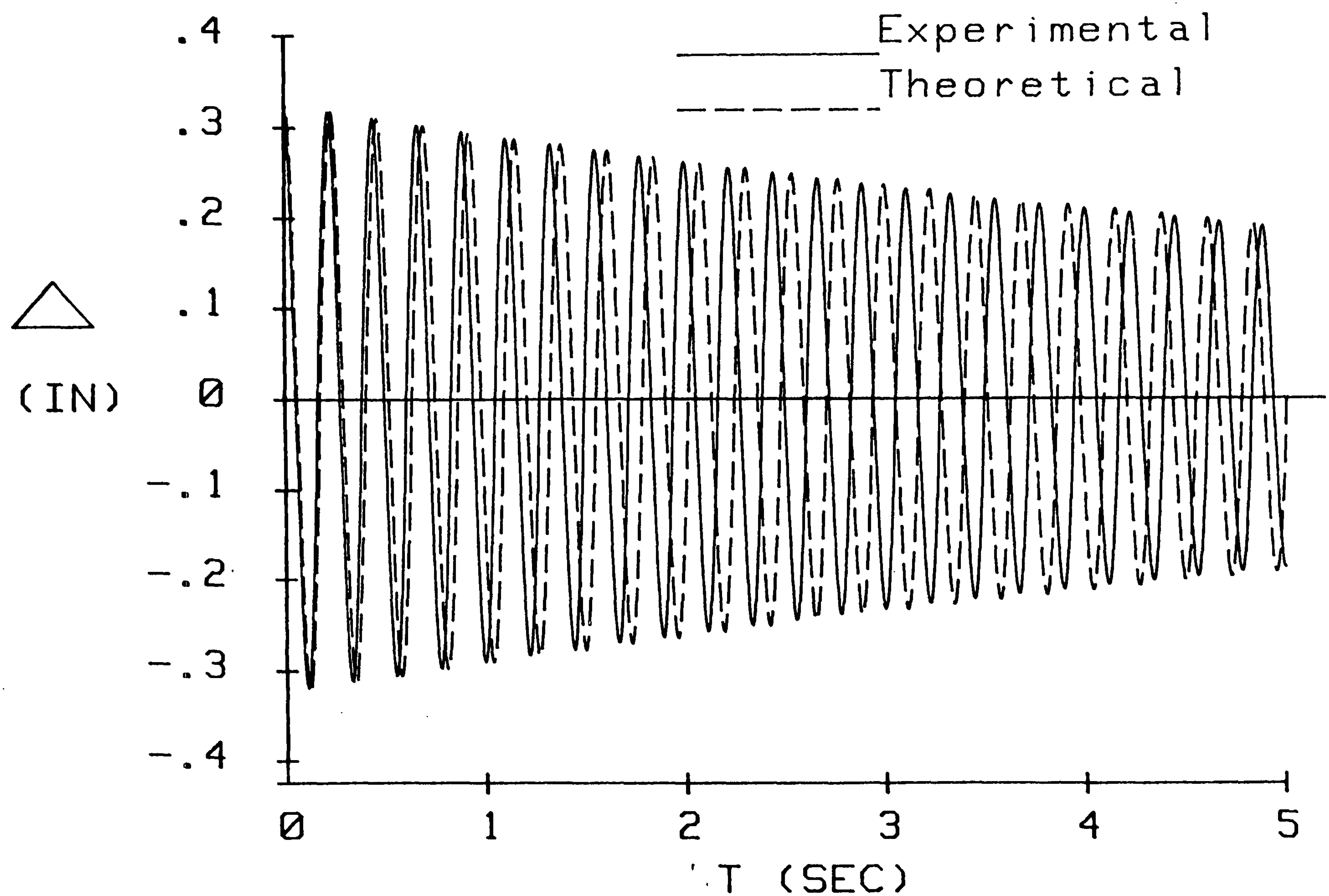

Figure 44. Theoretical versus average experimental Δ - t plots for member free vibration with no dampers

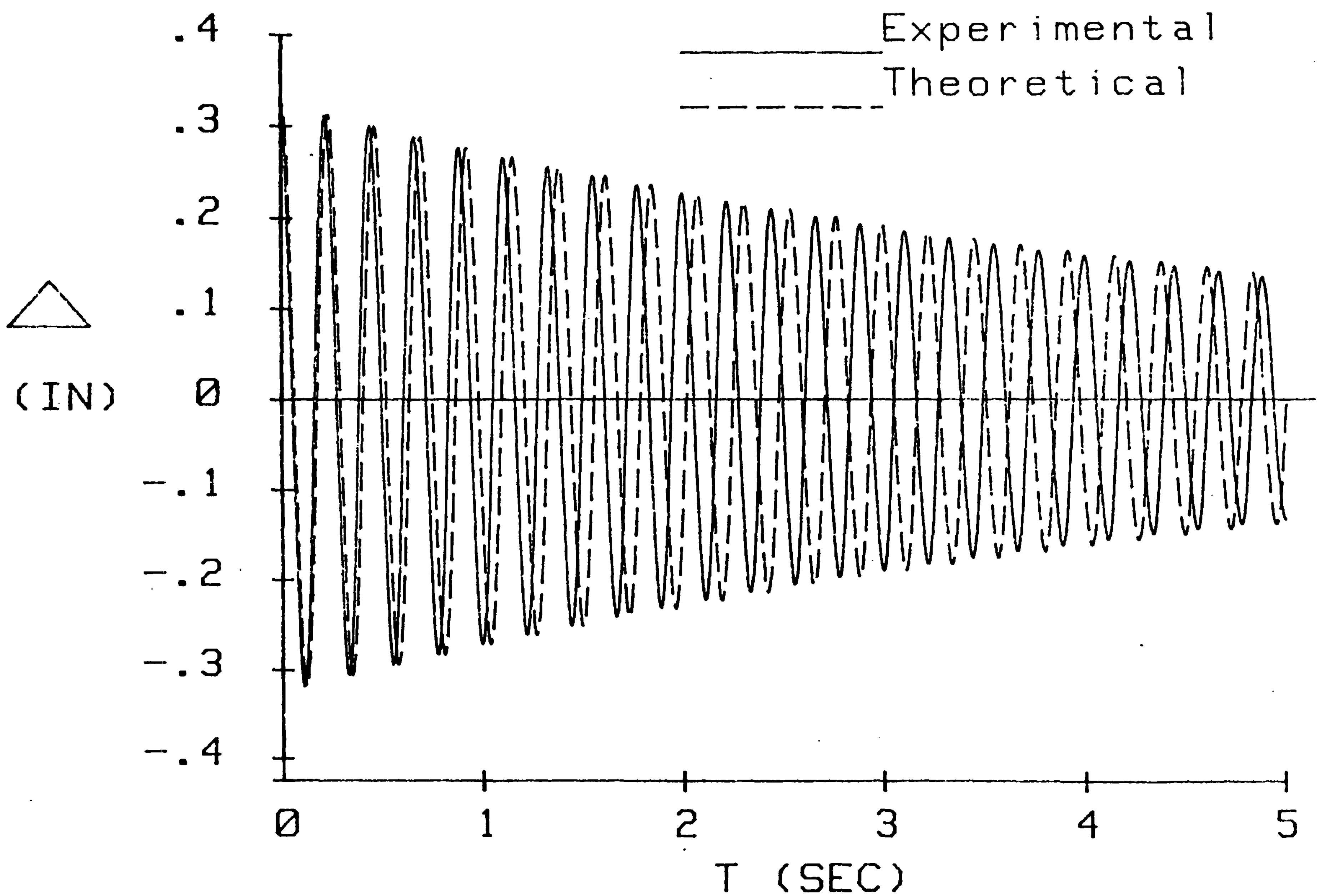

Figure 45. Theoretical versus average experimental Δ - t plots for member free vibration with 3 nylon brush dampers

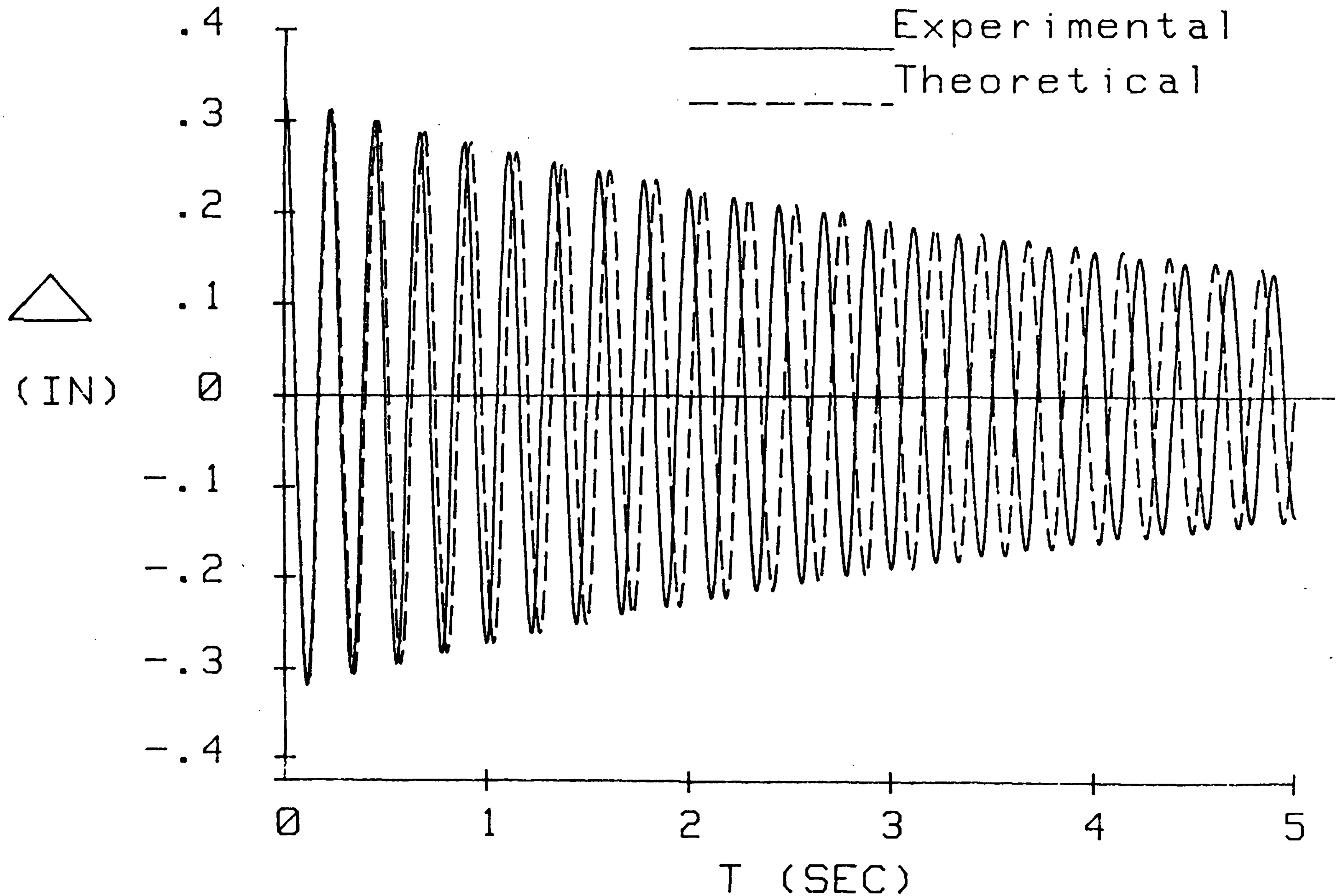

Figure 46. Theoretical versus average experimental Δ - t plots for member free vibration with 5 wool swab dampers

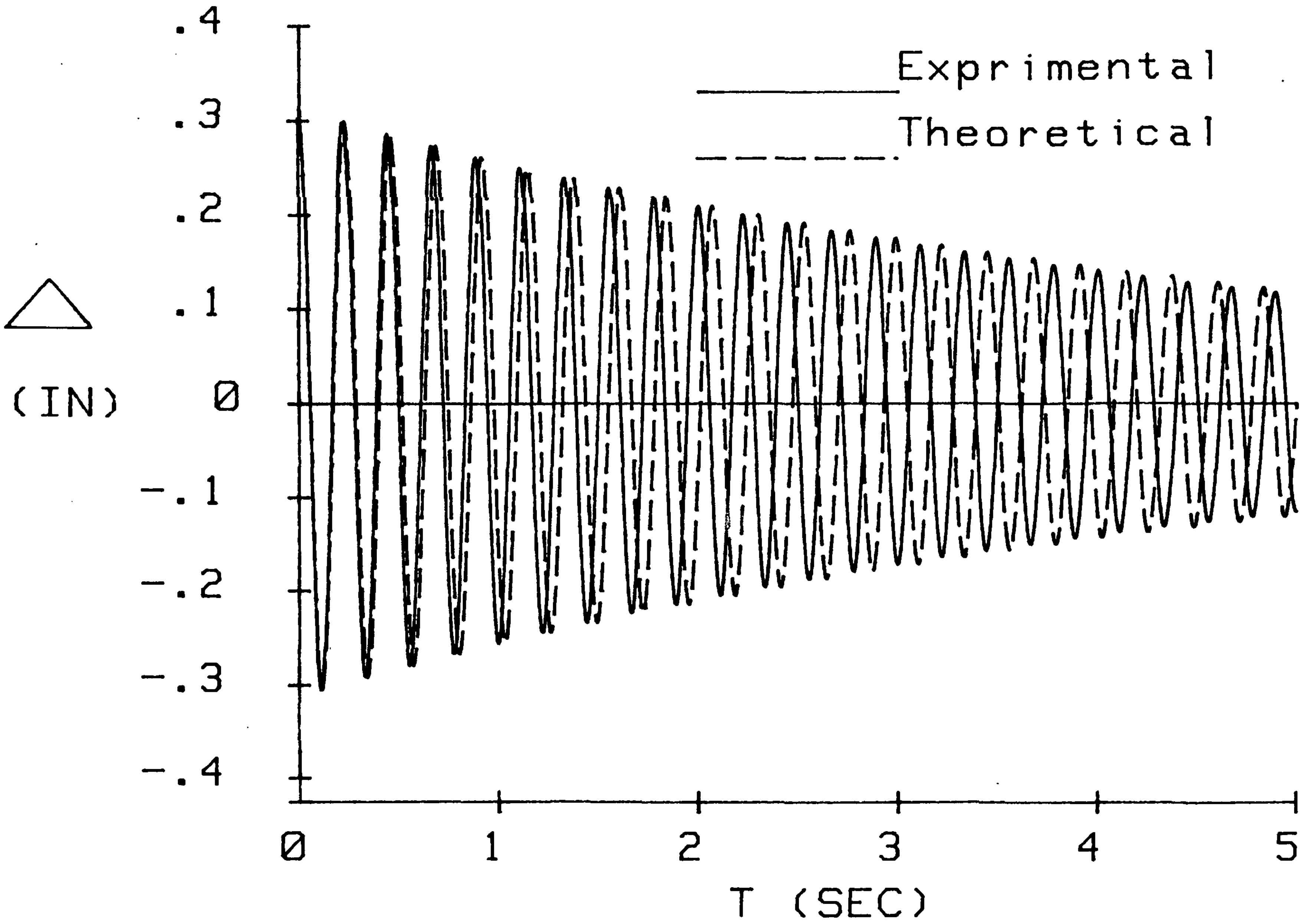

Figure 47. Theoretical versus average experimental Δ - t plots for member free vibration with 3 copper brush dampers

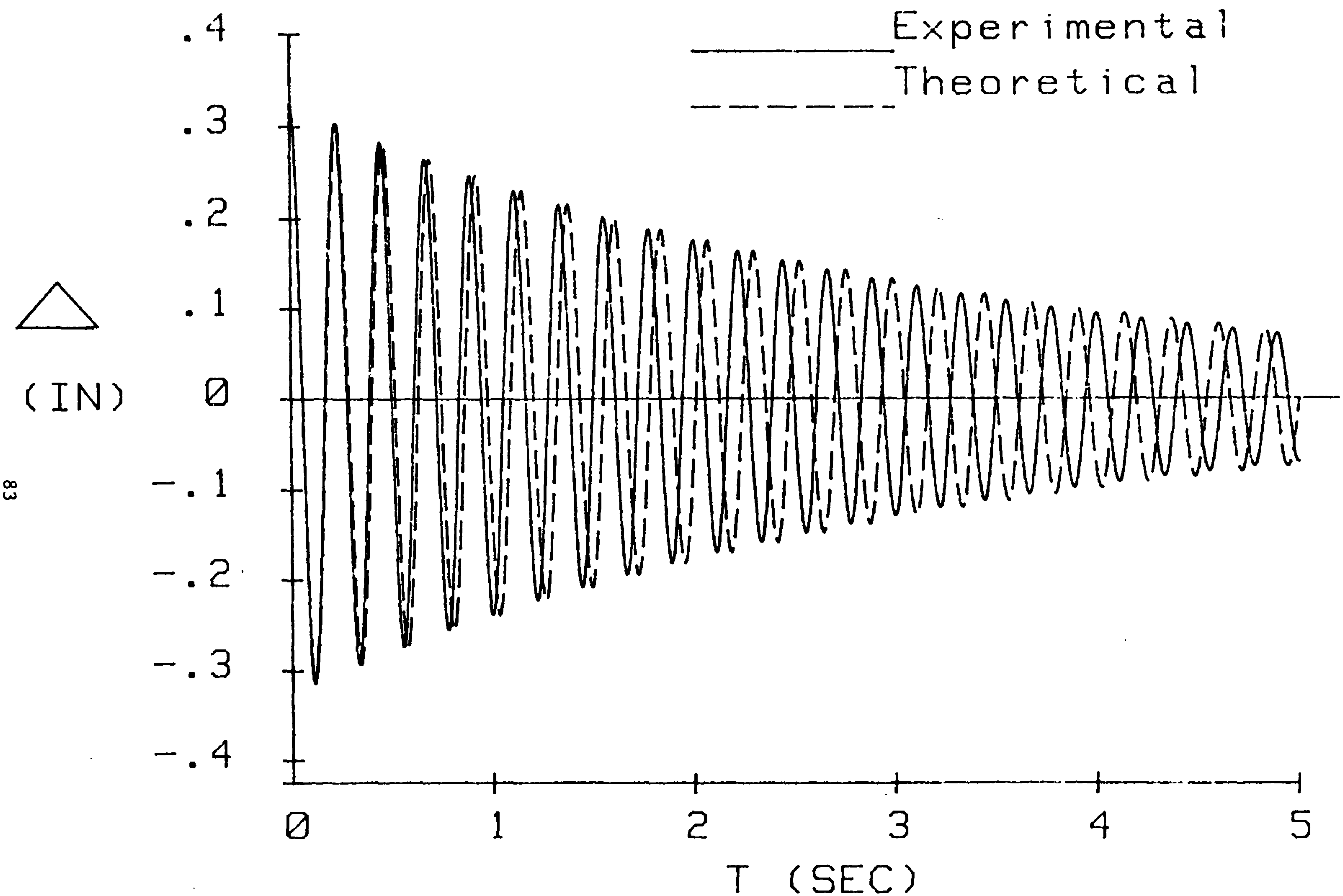

Figure 48. Theoretical versus average experimental Δ - t plots for member free vibration with 5 silly putty in chamber dampers

APPENDIX A

STATIC ANALYSIS OF MEMBER WITH PARTIAL RESTRAINTS

The partial rotational end restraint K of the 20.86 ft. member shown in Figure 1 has been determined using the elastic flexural equation:

$$EI \frac{d^2 \upsilon}{dz^2} = -M \qquad (A.1)$$

in which:

E = modulus of elasticity,

I = moment of inertia,

υ = deflection,

z = distance from the origin of coordinates, and

M = bending moment at a point.

To obtain the slope and deflection, this equation is integrated. Constants introduced during the integration are evaluated by observing the boundary conditions. Thereby, when the member of Figure 1 is subjected to a load W at its midspan, Equation A.1 can be written as:

$$EI\ \upsilon'' = -\frac{WZ}{2} + \frac{WL}{8}\left(\frac{1}{1 + 2/\bar{K}}\right) \qquad (A.2)$$

in which:

$$\bar{K} = \frac{KL}{EI} \qquad (A.3)$$

To obtain the member deflection (υ) the following boundary conditions are enforced:

$$\upsilon\ (o) = \upsilon' \left(\frac{L}{2}\right) = 0 \qquad (A.4)$$

Hence, solving Equation A.2 will yield the following deflection at any point along the member:

$$\upsilon(Z) = \frac{WL^3}{48EI}\left(\frac{Z}{L}\right)\left\{3(1 - \lambda) + 3\lambda\left(\frac{Z}{L}\right) - 4\left(\frac{Z}{L}\right)^2\right\} \tag{A.5}$$

in which:

$$\lambda = \frac{1}{1 + 2/\bar{K}} \tag{A.6}$$

Consequently, the deflection at midspan will be:

$$\upsilon\left(\frac{L}{2}\right) = \frac{WL^3}{96\ EI}\ [2 - 1.5\ \lambda] \tag{A.7}$$

Therefore, the rotational spring stiffness K is determined by solving Equations A.7, A.6, and A.3, respectively. Thus, substituting the experimentally measured deflection at mid-span $\upsilon(L/2) = 0.4063$ in. corresponding to an applied load W = 7.9 lb, the value of K turns out to be 48.5 kip-in/rad. Moreover, the value of K is verified using the moment-area method which yields a K value of 48.5 kip-in/rad.

APPENDIX B

GRILLAGE STATIC ANALYSIS CALCULATIONS

The basic $\underline{K}'_{ij}$ matrices for use in generating the $\underline{Q}_{ij}$ sub-matrices defined in Equations 7 and 8 are given by Reference 8:

$$\underline{K}_{11}' = \begin{bmatrix} C^2 \dfrac{12\ EI}{L^3} & C\ \dfrac{6\ EI}{L^2} \\ C\ \dfrac{6\ EI}{L^2} & \dfrac{4\ EI}{L} \end{bmatrix} \tag{B.1}$$

$$\underline{K}_{12}' = \begin{bmatrix} -C^2 \dfrac{12\ EI}{L^3} & C\ \dfrac{6\ EI}{L^2} \\ -\ C\ \dfrac{6\ EI}{L^2} & \dfrac{4\ EI}{L} \end{bmatrix} \tag{B.2}$$

$$\underline{K}_{21}' = \underline{K}_{12}'^{T} \tag{B.3}$$

$$\underline{K}_{22}' = \begin{bmatrix} C^2 \dfrac{12\ EI}{L^3} & -\ C\ \dfrac{6\ EI}{L^2} \\ -C\ \dfrac{6\ EI}{L^2} & \dfrac{4\ EI}{L} \end{bmatrix} \tag{B.4}$$

With E = 10,000 psi; I = 0.325 in^4; C = cos α; and the dimensions given in Figure 3, the relevant $\underline{Q}_{ij}$ matrices are found to be as follows:

$$\underline{Q}_{11} = \begin{bmatrix} 0.03 & -1.76 \\ -1.76 & 147 \end{bmatrix} \tag{B.5a}$$

C-2

$$Q_{12} = \begin{bmatrix} -0.03 & -1.76 \\ 1.76 & 73.5 \end{bmatrix} \quad \text{(B.5b)}$$

$$Q_{22} = \begin{bmatrix} 0.06 & 0 \\ 0 & 294 \end{bmatrix} \quad \text{(B.5c)}$$

$$Q_{23} = \begin{bmatrix} -0.03 & -1.76 \\ 1.76 & 73.5 \end{bmatrix} \quad \text{(B.5d)}$$

$$Q_{33} = \begin{bmatrix} 0.05 & 3.01 \\ 3.01 & 251 \end{bmatrix} \quad \text{(B.5e)}$$

$$Q_{34} = \begin{bmatrix} -0.02 & 1.25 \\ -1.25 & 52 \end{bmatrix} \quad \text{(B.5f)}$$

$$Q_{44} = \begin{bmatrix} 0.04 & 0 \\ 0 & 208 \end{bmatrix} \quad \text{(B.5g)}$$

$$Q_{45} = \begin{bmatrix} -0.02 & 1.25 \\ -1.25 & 52 \end{bmatrix} \quad \text{(B.5h)}$$

$$Q_{55} = \begin{bmatrix} 0.02 & -1.25 \\ -1.25 & 104 \end{bmatrix} \quad \text{(B.5i)}$$

Referring to Figure 3, the load vectors are:

$$P_A = \begin{Bmatrix} 0 \\ 27.02\ \theta_A \end{Bmatrix} \quad \text{(B.6a)}$$

$$\underline{P}_B = \begin{Bmatrix} -0.0040 \\ 0.0000 \end{Bmatrix} \tag{B.6b}$$

$$\underline{P}_C = \begin{Bmatrix} 0.0397\delta_C \\ -75.7\theta_C \end{Bmatrix} \tag{B.6c}$$

$$\underline{P}_D = \underline{0} \tag{B.6d}$$

$$\underline{P}_A = \begin{Bmatrix} 0 \\ 48.5\ \theta_E \end{Bmatrix} \tag{B.6e}$$

When these equations are substituted in equation 8, the following deflection vector results:

$$\{\underline{\delta}\}^T = \{-0.025,\ 0.0009,\ -0.165,\ 0.0023,\ -0.223,\ 0.0078,$$
$$-0.0210,\ 0.00002,\ -0.0007,\ 0.00001\} \tag{B.8}$$

Thus, using $\delta_D = 0.0210$in., the following simple elastic load deflection relationship can be written

$$W_{Bt} = 190\ \Delta_{Dt} \tag{B.9}$$

where B and D refer to the locations indicated in Figure 3, and t refers to the fact that this is a theoretically obtained expression. The units of W_{Bt} and Δ_{Dt} are lb., and in., respectively.