

## **PASSIVE DAMPING CONCEPTS FOR SPACE STRUCTURES WITH TUBULAR MEMBERS**

**Zia Razzaq<sup>1</sup> and Bassam S. Najjar<sup>2</sup>**

**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 for possible use in the construction of future outer space stations. Four different passive damping concepts are studied under free and forced vibration and include nylon brush, wool swab, copper brush, and "silly putty" dampers. It is found that the silly putty, and the wool swab dampers provide good passive damping.

### **Introduction**

Aluminum tubular members may be used in the future to construct outer space structures. These members may be subjected to vibration induced by external disturbances. One practical problem is to identify a damping concept that will reduce the vibration of these members substantially. Recent investigations (References 1-5) have been conducted to study passive damping concepts 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: mass-string dampers; external viscoelastic tape; inner metal tube core (copper, aluminum; steel, brass); polyethylene tubing; chambers with oil, oil and discs, or sand; bright zinc chain; brushes for electrostatic and frictional damping; mass-string-whiskers assembly. Except for the viscoelastic tape, 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 paper summarizes 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 subassembly taken from a proposed space

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<sup>1</sup> Professor, Department of Civil Engineering, Old Dominion University, Norfolk, Virginia 23529, Ph. (804) 683-3753.

<sup>2</sup> Graduate Student, Department of Civil Engineering, Old Dominion University, Norfolk, Virginia 23529.

station prototype model. The two bar assembly studied is in the horizontal plane while the vibration is induced vertically. The resulting subassembly represents a basic space structure and provides a convenient means of testing passive damping concepts beyond the single-member level. The members possess moderate slenderness and are provided with semi-rigid connections.

### Domain of Investigation

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. The rotational stiffness provided by the connection is  $k$ . The member has an initial deflection  $w_o$ , and is subjected to natural or forced flexural vibration at its midspan. Also, Figure 2 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  $k_1 = 53.1$  kip-in/rad., and  $k_2 = 48.5$  kip-in/rad., respectively. At C, the junction of the two members is supported by a pair of vertical springs CF and CG of equal stiffness  $K = 19.85$  lb/in. Natural or forced flexural vibration is introduced at midspan B of member AC. The specific problem considered herein 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 2. At the member-level, the following four passive damping concepts were investigated:

- a. Nylon Brush Dampers
- b. Wool Swab Dampers
- c. Copper Brush Dampers
- d. Silly Putty in Chamber Dampers

The first three types of dampers were 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 was selected owing to the plastic deformation behavior of the silly putty when subjected to external forces. In addition to the experimental study, a theoretical analysis of the member in Figure 1 was conducted using the procedures given in References 1 and 4 for the case of natural vibration.

### Passive Dampers

Figure 3 shows the four types of passive dampers studied. From one to several dampers were provided inside the hollow space of the tubular member. Figure 4 shows a typical arrangement consisting of several parts. First, a helical spring, with a stiffness of 0.44 lb/in., is attached to the inside of the connection. To the other end of the spring is attached a nylon chord which in turn is attached to the first damper. The nylon chord used in this investigation had a 40 lb capacity. Thereafter, a series of nylon chords and dampers are attached along the member length until the other end of the member is reached. The end of the nylon chord at the other end 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 a slight tension in the helical spring, and subsequently tied to the connection externally. The resulting passive damping assembly is, therefore, aligned with the longitudinal axis of the tubular member. 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. The dampers are installed equidistantly between the

member ends. Tests are conducted with one or more dampers. A brief description of each of the dampers follows.

Figure 3(a) shows a nylon brush damper. It has a diameter of 1.75 in., a total length of 6.25 in., weighs 14.0 gms, and is manufactured by H. Hertzberg and Son, Inc., Middletown, N.Y. 10940. It has a plastic handle and twisted wires with which the nylon bristles are intertwined, and is commonly used for cleaning vegetables. The brush shown in Figure 3(a) 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.

Figure 3(b) 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. 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. It is commonly used in cleaning 12 in. gauge shotguns.

Figure 3(c) 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. It has a partly 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. This brush is also used in cleaning 12 in. gauge shotguns.

Figure 3(d) 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, Pennsylvania 18042. 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 diameter to 1.0625 in. through machining thus reducing its wall thickness. 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. 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.

### Test Procedure

The main instrumentation used in conducting the tests consists of a proximity probe, vibration instrumentation, and a deflection-time plotter. For natural member vibration tests, a weight  $W = 6.1$  lb. is first attached at the 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 which is connected to a deflection-time recorder. To induce 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:

$$F(t) = F_0 \sin \Omega t \quad (1)$$

in which  $F_0 = 4$  lb.,  $t =$  time, and  $\Omega =$  frequency of the forcing function.

Figure 2 shows a schematic of the grillage 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. For the forced vibration tests, the 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 to record the response for  $F(t) = 0$ .

The vibrator employed for the forced vibration tests allowed only a limited amount of travel. This meant that the deflection of the member at the location where the vibrator was attached was limited to what the vibrator could 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. Some of the results 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 under forced vibration conditions through tests on the individual member. However, the only way to evaluate the performance of the dampers under forced vibration was by allowing the member to develop dynamic deflections without direct restrictions imposed by the vibrator. The grillage test procedure provided this freedom. 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 also 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.

## Results

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 damping concepts. Figure 5 shows the dimensionless envelope of the midspan deflection versus time ( $\Delta_B-T$ ) relationships when 5 wool swab or 5 silly putty in chamber dampers were used. Figure 6 presents the damping ratio versus the number of dampers relationships for the various damping concepts. Figure 7 shows the passive damping efficiency index curves for these concepts. The index is defined as follows:

$$\eta = \frac{\zeta - \zeta_0}{M_d} \quad (2)$$

in which  $\zeta_0$  is the damping ratio in the absence of any passive damping device,  $M_d$  is the mass of the passive damping device, and  $\zeta$  is the damping ratio when dampers are provided. Also, as described earlier, due to the constrained motion imposed by the vibrator, the effectiveness of the passive dampers could not be adequately evaluated under forced conditions for the individual members. Furthermore, the theoretical results based on the procedures in References 1 and 4 for member natural vibration were in excellent agreement with the tests and are documented in Reference 6.

For the grillage under natural vibration, the 5 wool swab damper configuration provided greater damping than that of the 5 silly putty in chamber damper configuration. Under the forced vibration, the 5 silly putty in chamber damper configuration provided very effective passive damping at and around the resonant frequency. At resonance, these dampers resulted in a 52% reduction of the dynamic magnification factor, as indicated by the results plotted in Figure 8.

## **Conclusion**

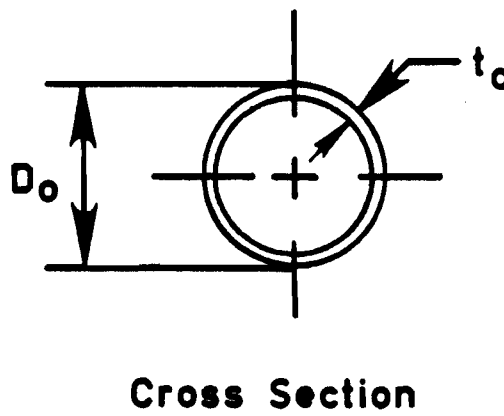
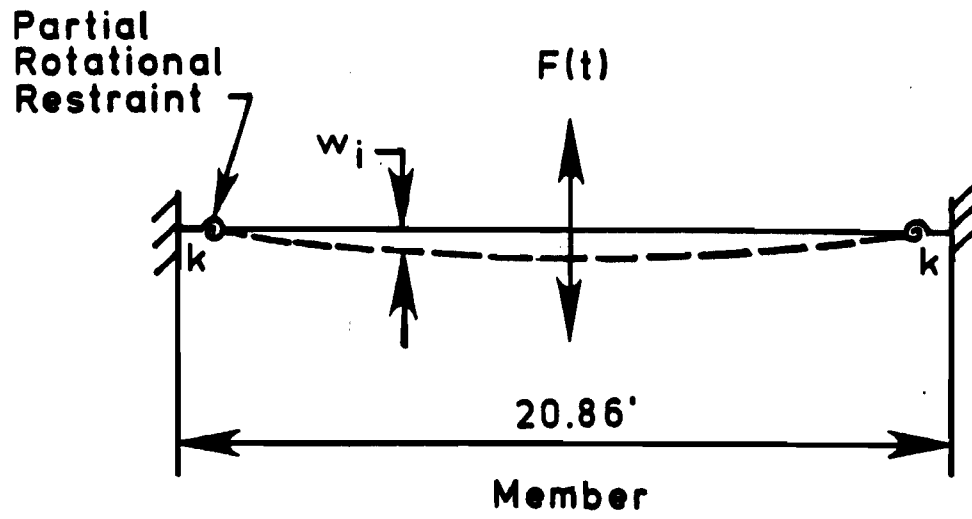
Four passive damping concepts studied show that the member and grillage vibration amplitudes can be reduced to various degrees. It appears that the silly putty in chamber dampers are generally more effective than the other three types of potential damping concepts studied.

## **Acknowledgements**

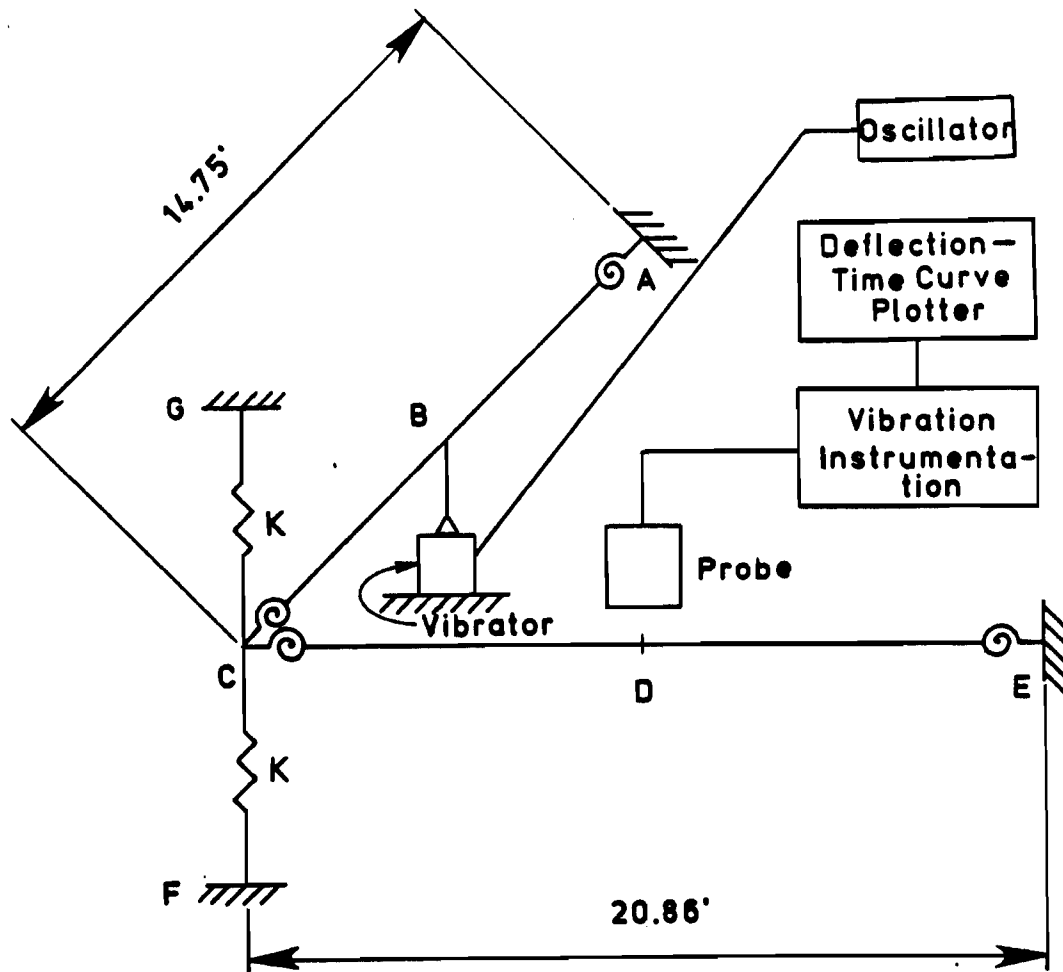
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**Figure 1. Schematic of tubular member**



**Figure 2. Schematic of grillage test setup**



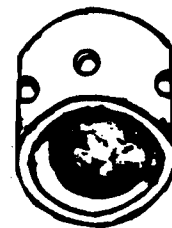
**(a) Nylon brush damper**



**(b) Wool Swab damper**



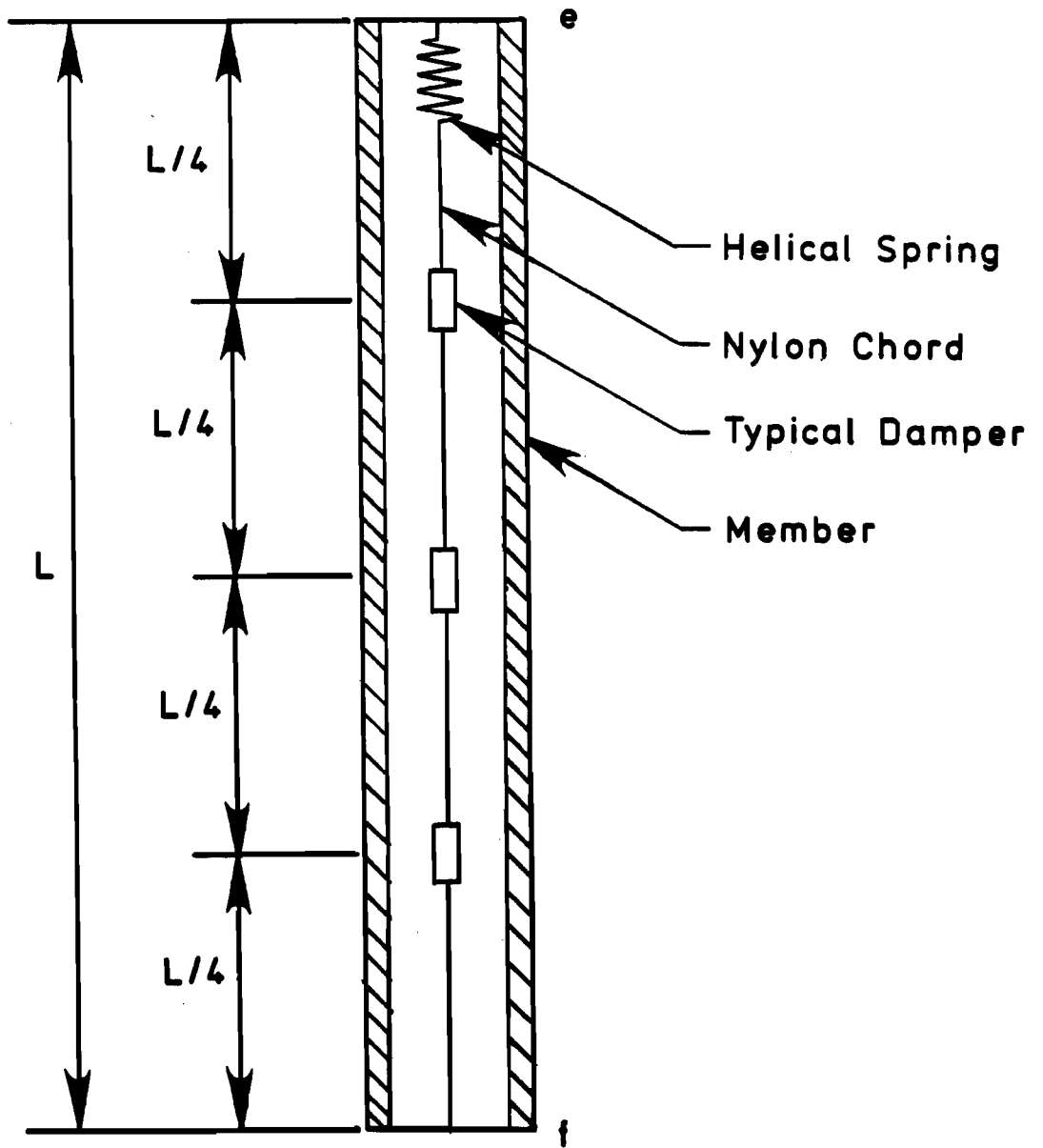
**(c) Copper brush damper**



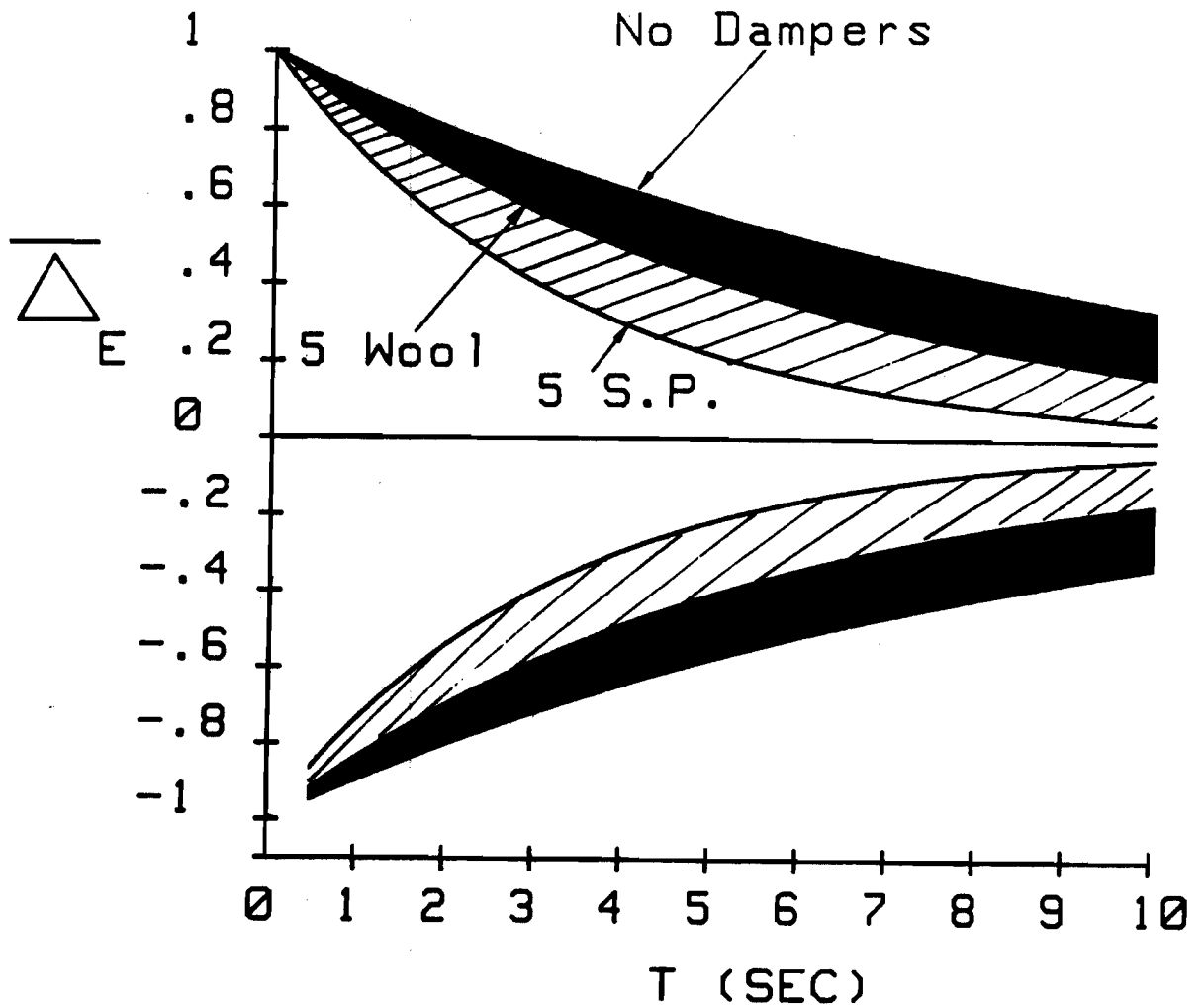
**(d) Silly putty in chamber damper**

**Figure 3. Passive dampers**

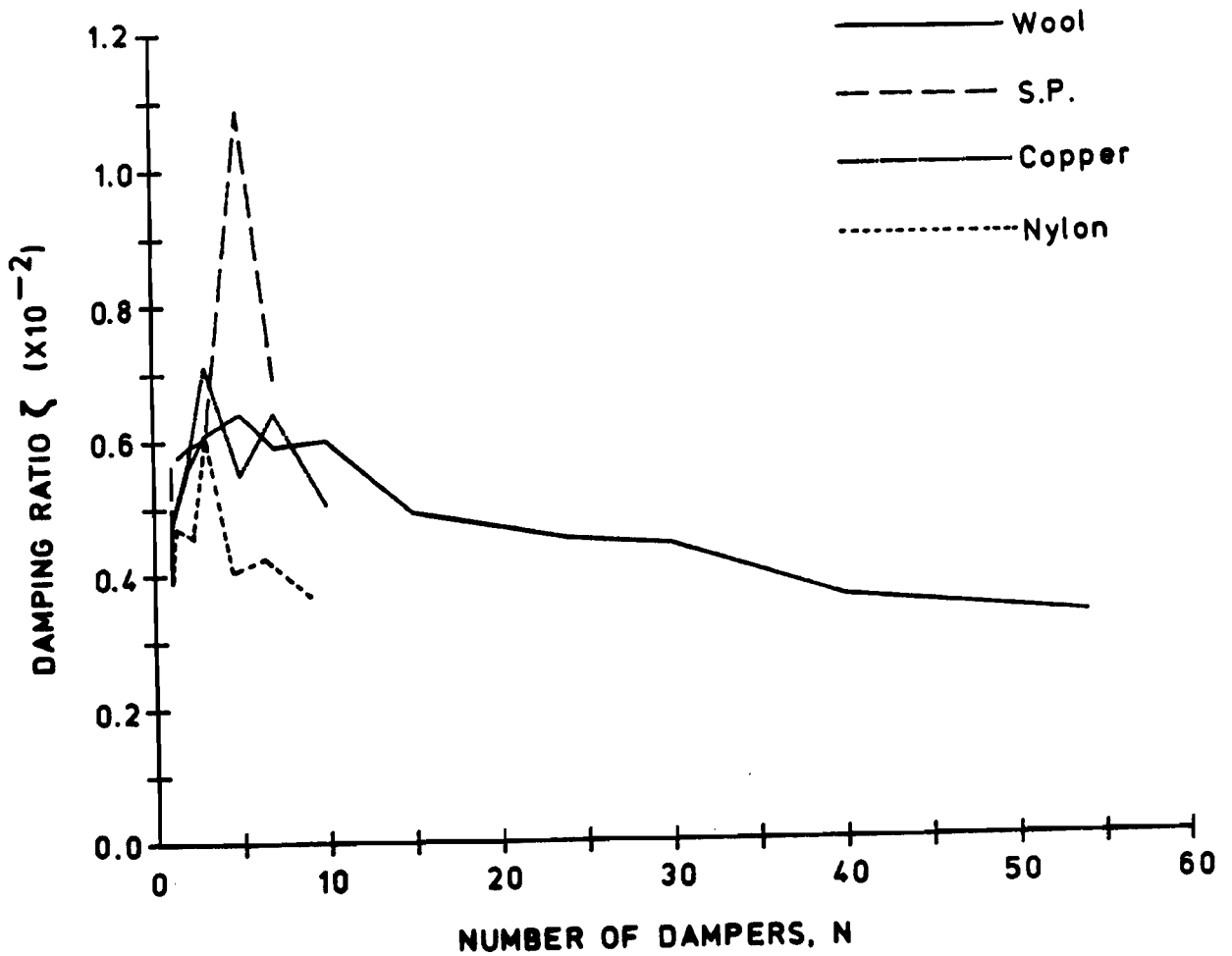




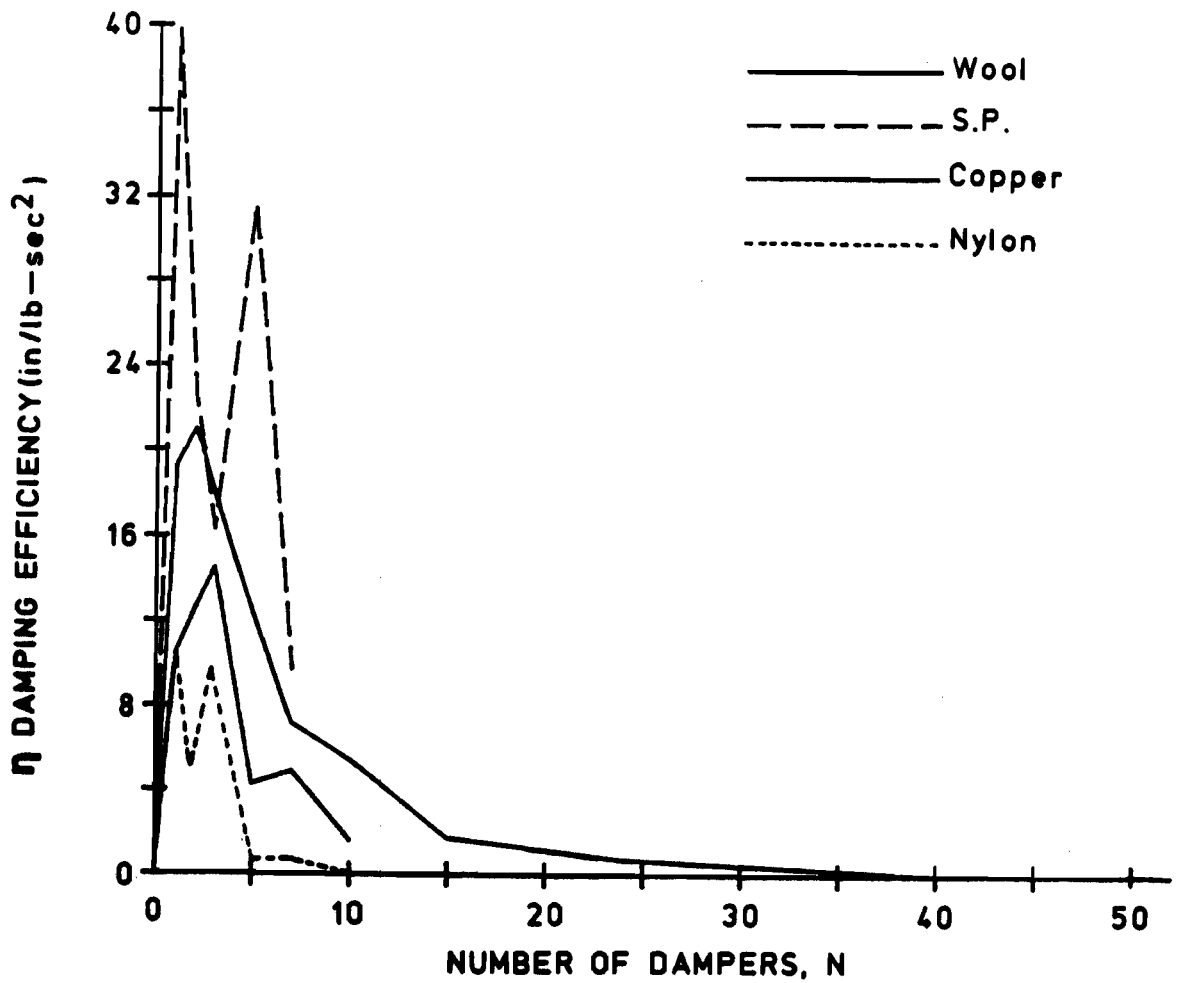
**Figure 4. Schematic for typical spacing of passive dampers**



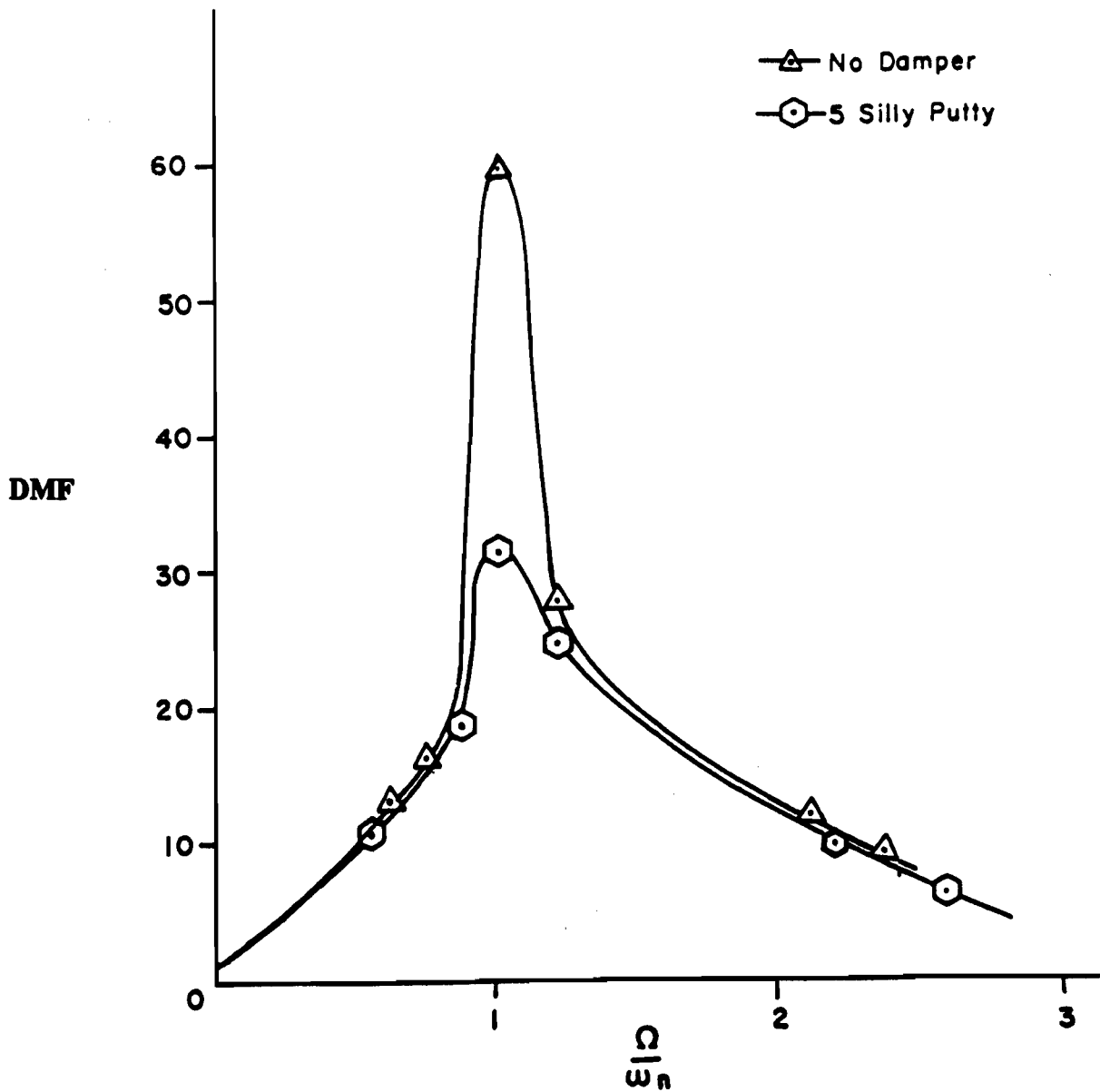
**Figure 5. Member dimensionless deflection-time envelopes**



**Figure 6. Damping ratio versus number of dampers for member natural vibration**



**Figure 7. Damping efficiency versus member of dampers for member natural vibration**



**Figure 8. Dynamic magnification factor versus frequency ratio for grillage vibration**