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Experimental Passive Damping Concepts for Space Structures with Tubular Members

Zia Razzaq
Department of Civil & Environmental Engineering
Old Dominion University
Norfolk, Virginia 23529, USA
zrazzaq@odu.edu

Bassam S. Najjar
Department of Civil & Environmental Engineering
Old Dominion University
Norfolk, Virginia 23529, USA

***Abstract* — 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-in-chamber dampers installed inside the hollow space of the tubular member(s). It is found that the silly-putty-in-chamber and the wool swab dampers provide effective passive damping.**

Keywords—passive damping concepts and devices; outer space structures; tubular members; grillage structure; damping efficiency

I. 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. The primary aim of the present study is to develop passive damping devices and study their effectiveness in reducing vibrations in a long tubular aluminum alloy member and a two-bar grillage structure. One of the challenges dealt with is that of having to develop and install potential damping devices inside the hollow space of the tubular member(s).

A variety of passive damping materials and procedures have been utilized in the past to control structural vibrations. For example, Rao [1] presented an application of passive damping technology using viscoelastic materials to control noise and vibration in vehicles and commercial airplanes. Such damping materials are also used in the automotive and aerospace industry in a variety of applications to reduce noise and vibration and to improve interior sound quality. Brennan [2] has presented some examples of prototype adaptive tuned vibration absorbers including a control scheme to automatically tune such absorbers over a range of frequencies by incorporating a variable stiffness element that can be adjusted in real-time. Bassani et al [3] described the design optimization and fabrication of a hybrid composite material for the passive suppression of flexural vibrations in slender and light structures. The material is made from glass fiber/epoxy resin laminate, reinforced with two thin, fiber-laser patterned sheets of NiTiCu shape memory alloy. The thickness of the shape memory alloy layers and their pattern geometry are then optimized by numerical calculation of the first natural frequency and of the structural damping of the hybrid composite applied to beam-shaped prototypes. Such vibration damping materials are typically applied on *external or exposed* surfaces of the structures being damped. The prime objective of the present paper is to explore and develop novel passive damping devices which can be installed *inside* of tubular aluminum alloy slender structural members for possible use in future space stations. Another objective of the study is to quantify the relative efficiency of various damping approaches considered.

Razzaq et al [4-8] investigated both *external* viscoelastic as well as novel *internal* passive damping devices installed inside of 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. were 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 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 sub-assemblage taken from a proposed space station prototype model. The two bar assembly studied is in the horizontal plane while the vibration is induced vertically. The resulting sub-assemblage 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.

II. 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_i, and is subjected to natural or forced flexural vibration at its mid-span. 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 stiffness 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 CO of equal stiffness K = 19.85 lb/in. Natural or forced flexural vibration is introduced at mid-span B of member AC. The specific problem considered herein is to first identify an efficient passive damping concept for the member shown in Figure I, 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 8 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 4 and 7 for the case of natural vibration.

III. 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 cord which in turn is attached to the first damper. The nylon cord 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 cord 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 cord 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.

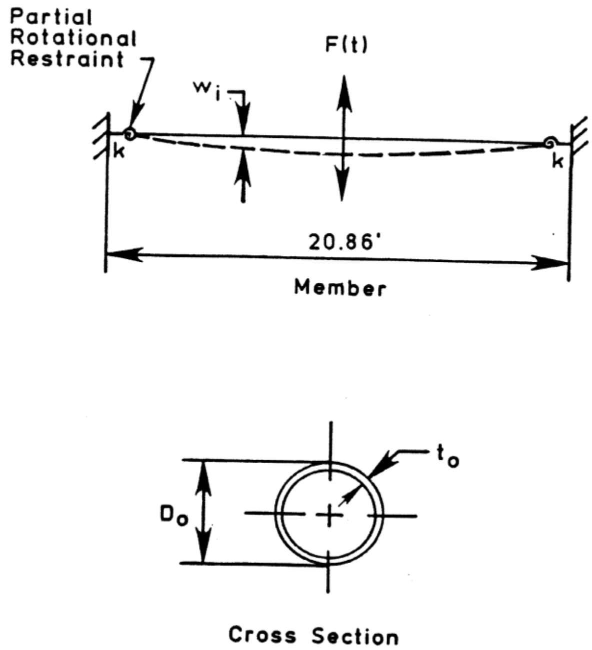

Figure 1. Schematic of tubular member

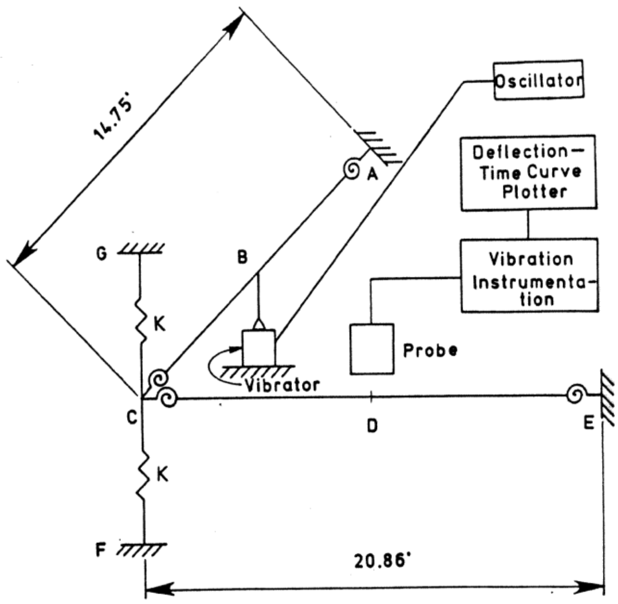

Figure 2. Schematic of grillage test setup

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 gm, 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 gm. 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 "Bristol 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 gm.

IV. 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 mid-span 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-dependent deflection at member mid-span 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:

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

In Equation 1, F_o = 4 lb., t = time, and Ω = frequency of the forcing function.

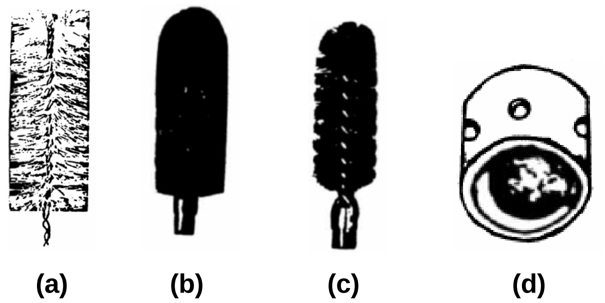

Figure 3. Passive dampers

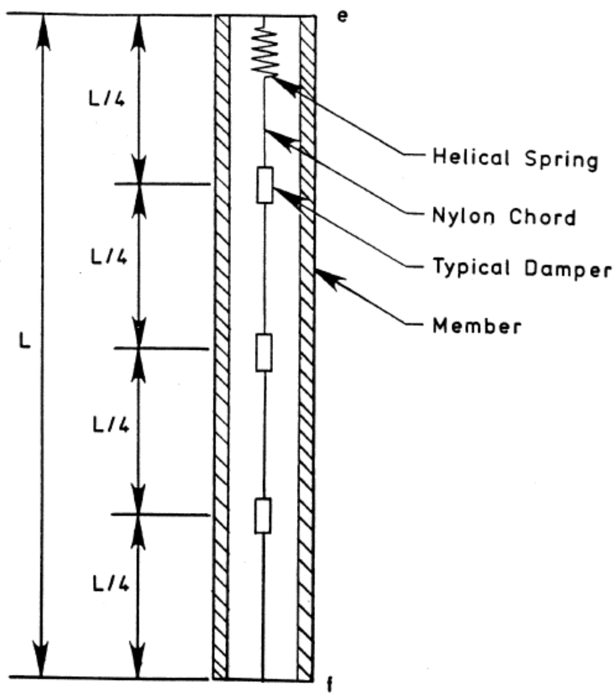

Figure 4. Typical spacing of passive dampers

To induce natural vibration in the grillage shown schematically in Figure 2, a weight W = 7.9 lb. attached at a point B of the member AC is released suddenly. The deflection-time response of the member CE is recorded at its mid-span D. For the forced vibration tests, the function F(t) is applied at B and the deflection point-time response is recorded at mid-span D of the member CE up to a certain time where-after 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 mid-span.

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 averages.

V. 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 mid-span deflection versus time relationships when five wool swab or five silly- putty-in-chamber dampers were used.

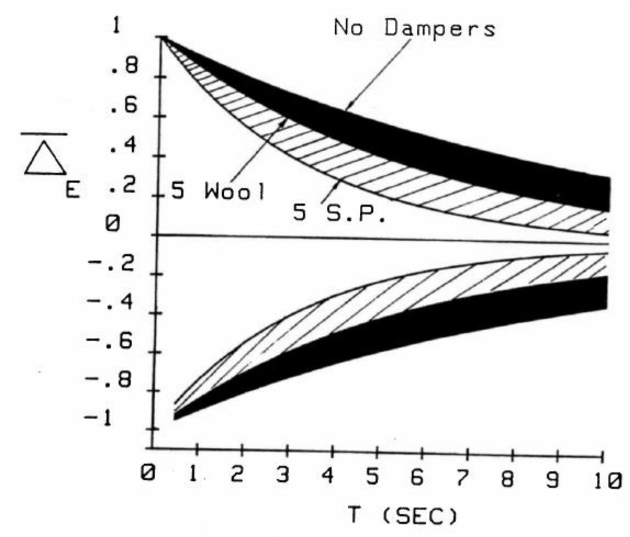

Figure 5. Member deflection-time envelopes

Figure 6 presents the damping ratio versus the number of dampers (N) relationships under natural vibration for the various damping concepts.

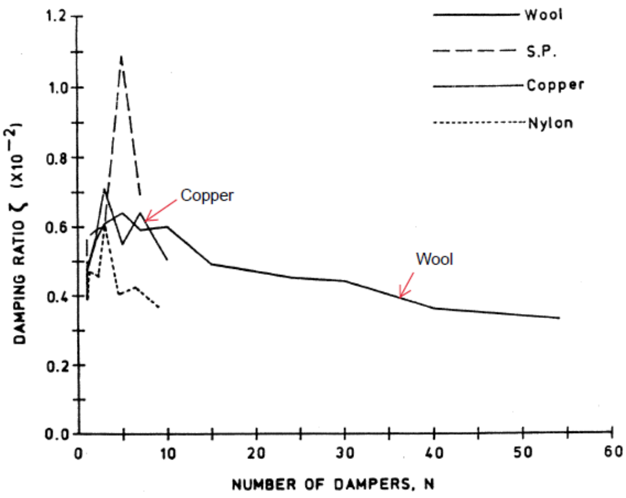

Figure 6. Damping ratio versus N

Figure 7 shows the passive damping efficiency index curves for these concepts. . The index is defined as follows:

$$\eta = \frac{\zeta - \zeta_o}{M_d} \tag{2}$$

In Equation 2, ζ_o is the damping ratio in the absence of any passive damping device, M_d is the mass of the passive damping device, and ζ 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 4 and 7 for member natural vibration were in excellent agreement with the tests and are documented in Reference 9.

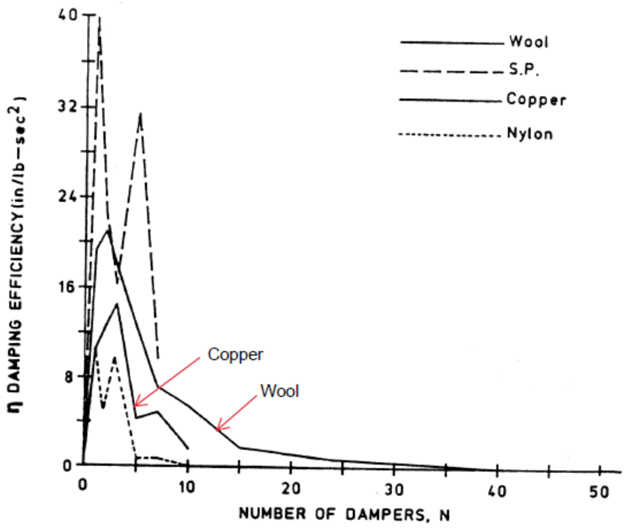

Figure 7. Damping efficiency versus N

For the grillage under natural vibration, the five wool swab damper configuration provided greater damping than that of the five silly-putty-in-chamber damper configuration. Under the forced vibration, the five 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.

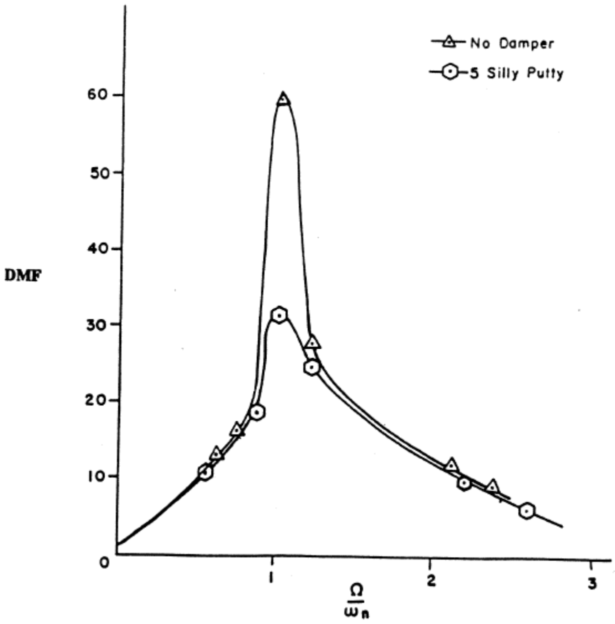

Figure 8. Dynamic magnification factor versus frequency ratio for grillage vibration

VI. CONCLUSIONS

Four passive damping concepts studied show that the member and grillage vibration amplitudes can be reduced to various degrees. The following key conclusions are drawn from the study conducted:

1. Under member natural vibration, it is found that the silly putty in chamber dampers provide considerably greater passive damping as compared to that of the nylon brush, wool swab, and copper brush dampers.

2. For the grillage under natural vibration, the five wool swab damper configuration provides greater damping than the five silly-putty-in-chamber damper configuration.

3. For the grillage under forced vibration, the five silly-putty-in-chamber configuration provides very effective passive damping only at and around the resonant frequency resulting in a 52 percent reduction in the dynamic magnification factor.

Future studies need to consider the effectiveness of the various passive damping concepts on reducing the vibration of three-dimensional structural sub-assemblages.

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