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Analytical Investigation of Damping Enhancement Using Active and Passive Structural Joints

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The low inherent damping of large space structures has prompted considerable research into active and passive damping augmentation. This paper discusses the development and analysis of improved joints for large space structures. These joints are able to give space structures higher levels of passive damping without significantly increasing the structure's weight or complexity. Two types of joint designs will be considered: passive joints and active joints. In each case, the force normal to a frictional interface is varied yielding a connecting joint with increased damping performance. A single-degree-of-freedom joint and a system consisting of two elastic beams connected by a single active/passive joint are considered. It is shown that these new types of joints enhance the energy dissipation from interconnected flexible structures in a relatively simple and robust way. Numerical simulation results are presented and discussed.

I. Introduction

ONE of the major problems remaining in the development of large space structures (LSS) is the anticipated low level of passive damping. This low level of damping affects the feasibility of placing large flexible space structures in orbit for a number of reasons. Perhaps the most important reason is that it is difficult to design attitude and shape controllers for lightly damped flexible structures. Since the open-loop system has low relative stability, it is quite possible that perturbations to the control scheme, such as observation spillover or plant uncertainty, can drive the closed-loop system unstable.[1] Many researchers have aimed to circumvent this problem by designing better (often more complicated and sophisticated) control systems. A review of the literature up until 1984 can be found in Ref. 2; the text by Joshi[3] provides a more recent overview of research in this area. An alternate approach is to design structures to have a greater passive damping capacity. It has been shown that the addition of passive damping to a flexible structure can greatly facilitate the model reduction and control design of flexible structures.[4] Many researchers have proposed techniques for adding to the passive damping level of structural systems. These techniques have included conventional approaches such as the addition of viscous dashpots,[5] constrained layer damping,[6] and passive vibration absorbers[7] to more exotic approaches involving impact damping.[8] This paper addresses the possibility of increasing the passive damping of trusslike structures by enhancing the energy-dissipating capability of connecting joints.

Connecting joints of traditional trusslike structures are already known to be important providers of passive damping.[9,10] One of the major dissipative mechanisms in joints is dry (Coulombic) friction. The analysis of structures with dry friction has received considerable attention, especially with regard to turbine and compressor bladed disks.[11-13] Although measurements of dry friction within vibratory systems is difficult to perform, dry friction has proven to be a reliable and essential component in built-up structures. Furthermore, it should be pointed out that, although uncertainty may be introduced into the system due to the presence of dry friction, the possible tradeoff between uncertainty and the added passive damping can be addressed with the use of robust control design such as H_∞.[14]

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In most of the prior theoretical work concerning dry friction damped structures, it has been assumed that the force normal to the sliding interface is constant. This may be termed the "classic" dry friction damped case. In the few studies where the normal force was allowed to vary with the relative slip amplitude, it was found that the system exhibited a viscouslike damping characteristic; see, for example, Hertz and Crawley,[9] and Ferri.[15-17] In the experimental study performed by Folkman and Redd,[10] the free response of a truss structure exhibits exponential envelopes of decay even though the primary source of damping is dry friction in the connecting joints. The viscouslike damping property suggests that some structural systems can be improved by configuring frictional interfaces in ways that allow the normal forces to vary with displacement.

It should be noted that viscous damping augmentation, especially in the form of viscoelastic materials such as those used in constrained layer damping, are subject to problems of "outgassing" in space environments.[7] This causes the materials properties of the viscoelastic material to change with time, resulting in a degradation of the effectiveness of the damping treatment. An active or passive joint design that is based on energy dissipation from dry friction could provide a viscouslike damping but still be "space realizable."

The outline of the remainder of this paper follows. Section II discusses background material related to the dynamics of a dry friction damped system and introduces the design of passive and active joints as a method of damping augmentation. Section III first describes the model development for single-degree-of-freedom joint systems. These models are then incorporated into a two-span flexible beam system. Sticking conditions for the joints are also obtained. Numerical studies are presented in Sec. IV, and Sec. V contains concluding remarks.

II. Background Material

As mentioned earlier, one of the major dissipative mechanisms in joints is dry (Coulombic) friction. It has been shown that dry friction can cause some significantly nonlinear behavior in an otherwise linear structure. Some of the dynamic characteristics are well known, for example "unbounded response at resonance" and "stick-slip" motion. "Unbounded response at resonance" refers to the fact that a structure damped only by classic dry friction can exhibit unbounded response when forced at a system natural frequency. This is due to the inverse dependence of effective damping on the amplitude of response in classic dry friction damped systems. Thus large levels of slip will decrease the damping contribution from the friction interface. Stick-slip motion refers to the fact that the sliding interface may stick and slip intermittently. One consequence of this is a net decrease in the amount of damping from the frictional interface because, of course, a frictional interface must undergo relative slip to dissipate energy.

It appears that the undesirable nonlinear effects of classic dry friction can be greatly reduced by the introduction of either an active or a passive mechanism by which the normal forces (and, hence, the frictional forces) can be allowed to vary. At the same time, the beneficial contribution of dry friction to the overall damping of the structure can be retained or enhanced. This paper presents two types of joints that meet these objectives. One proposed joint design is passive whereas the other is active, incorporating collocated feedback. A short description of the two joint types as well as a discussion of their relative advantages is presented next.

Passive Joint: One approach to improving the linearity and energy dissipation from a truss structure is by means of a passive connecting joint that takes full advantage of amplitude-dependent friction forces. Although many types of truss connecting joints possess this property to varying degrees, the principle has not yet been fully exploited. An example that demonstrates the principle is the modified pin-type joint shown in Fig. 1a. As suggested by the figure, the joint will function in most respects like a conventional pin joint except that the curved contacting surfaces will allow the normal force to vary with the relative rotation angle. To reiterate, this type of design can give the overall structure a nearly viscous damping effect, without the use of viscoelastic materials or dashpots.

Active Joint: An active joint can be designed to control the normal force based on sensor feedback from collocated or distributed sensors so that the energy dissipation from the joint is optimized. A preliminary design is shown in Fig. 1b. Although the figure shows the clamping mechanism to be electromagnetic, piezoelectric actuator elements are also possible. Note that the active control system acts solely to enhance the overall damping of the joints. Unlike other damping schemes, the method is not based on momentum management techniques such as thruster rockets, control moment gyros, and reaction wheels. A general disadvantage of momentum management techniques is a weight penalty and in the case of thruster rockets, a contamination of the local environment of the spacecraft.

There are several tradeoffs between active and passive joint types. The passive joint appears to be much less expensive and less massive than the active joint, especially if one takes into account the total expense and weight associated with the power harness required for the active case. A disadvantage of the passive joint is that it is nonadjustable. A particular geometry and preload must be chosen a priori and cannot be varied to accommodate changes in frictional properties or changes in the structural vibration environment. The active joint, on the other hand, is easily modified in real time, is more robust due to feedback, and can even be designed to behave linearly like an ideal viscous damping element (see discussion below). The passive joint may be prone to steady-state error due to sticking whereas the active joint can be made asymptotically stable.

The active joint proposed here should not be confused with the active hinge studied previously by Cudney et al.[18] In that study, the hinge was "fully-active"; consequently, it could transmit an arbitrary moment between the two elastic beams that it connected. The active joint described in this paper may be termed "semiactive" since it can only apply a resistive moment between the two beams. It should also be mentioned that there is a similarity between the active joint and semiactive suspension elements for rail and road vehicles.[19,20] An example of semiactive suspension is one in which an orifice in a hydraulic dashpot is actively varied to produce different amounts of damping. Semiactive elements are popular because they typically have lower power requirements than fully active elements. Systems containing either a semiactive suspension element or an active joint are not fully controllable but are only controllable to the origin. Another similarity is that a "failure" of the control system can only result in a degradation of performance and not in instability since the controlled elements can only remove energy from a system. An important difference, however, is that the mathematical model of the active joint is nonsmooth, making traditional optimal control strategies difficult to apply.

III. Model Development

In this section, models are developed for the active and passive joint designs. Both designs are modifications of standard revolute (pin) joints similar to those introduced in Figs. 1a and 1b. First, however, it is instructive to consider a single-degree-of-freedom (SDOF) prismatic joint. Afterward, the active and passive SDOF joint designs will be modified to relative rotation rather than translation and incorporated into a two-span flexible beam system.

SDOF Joint

Figure 2a shows a schematic of an SDOF joint including a frictional interface. The two connecting bodies are represented by the moving mass and the stationary frame. Using the nomenclature defined in Fig. 2a, a force summation in the direction tangent to the sliding interface yields the following equation of motion for the SDOF system:

$$m\ddot{x} + c\dot{x} + kx + R(x,\dot{x}) = F(t) \tag{1}$$

where $F(t)$ is a disturbance force, and $R(x,\dot{x})$ is the nonlinear force that resists relative slip in the joint. In the case of the active joint, $R(x,\dot{x})$ will be assumed to have the form

$$R(x,\dot{x}) = \mu N_A(x,\dot{x})\mathrm{Sgn}(\dot{x}) \tag{2}$$

where

$$\mathrm{Sgn}(z) = \begin{cases} +1 & ; \quad z>0 \\ 0 & ; \quad z=0 \\ -1 & ; \quad z<0 \end{cases}$$

and $N_A(x,\dot{x})$ is the active force applied to the sliding interface and has the form

$$N_A(x,\dot{x}) = K_0 + K_1 \, |x| \, + \, K_2 \, |\dot{x}| \tag{3}$$

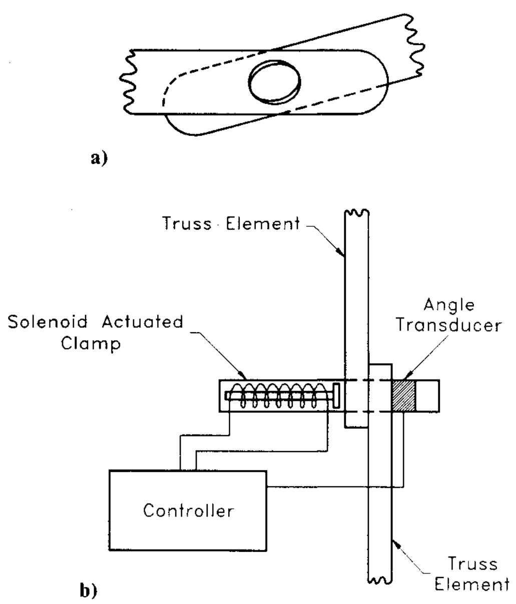

Fig. 1 Examples of revolute joints with amplitude and/or rate-dependent friction forces: a) passive joint, b) active joint.

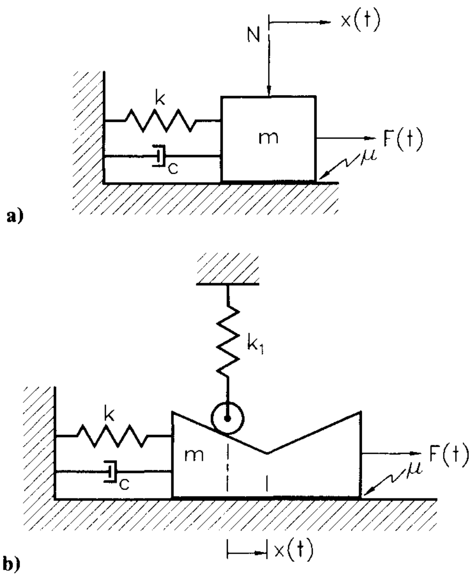

Fig. 2 Schematic representation of translational SDOF joint models: a) active joint, b) passive joint.

Although, in practice, nothing limits the functional form of N_A (other than the fact that $N_A \geq 0$), the form shown earlier was chosen due to its simplicity and ease of implementation. (As mentioned, the optimization of the joint control design is not trivial due to the nonsmooth nature of the system dynamics.) Note that the term

$$\mu K_2 |\dot{x}| \mathrm{Sgn}(\dot{x}) = \mu K_2 \dot{x} \tag{4}$$

can be interpreted as an additional *linear or viscous damping* term. Substituting Eqs. (2-4) into (1) yields the general form for the active SDOF joint:

$$m\ddot{x} + (c + \mu K_2)\dot{x} + kx + \mu(K_0 + K_1|x|)\mathrm{Sgn}(\dot{x}) = F(t) \tag{5}$$

A possible design of a passive SDOF joint is shown in Fig. 2b. In this case, variation in the normal force is achieved through contouring or profiling one of the two bodies and applying the normal force through an elastic spring element. A force summation in the direction tangent to the sliding interface yields the same equation as Eq. (1). In this case, however, the nonlinear resisting force is given by

$$R(x,\dot{x}) = \mu N_p(x)\mathrm{Sgn}(\dot{x}) + N_p(x)\tan\alpha \,\mathrm{Sgn}(x) \tag{6}$$

where the second term arises due to the component of the interaction force between the movable body and the roller in the direction tangent to the sliding interface. The $N_p(x)$ is the passively applied normal force given by

$$N_p(x) = k_0 + k_1 \tan\alpha |x| \tag{7}$$

where k_0 is the preload (force) present in the spring at the point $x = 0$. Substituting Eqs. (6) and (7) into Eq. (1) and recognizing that $|x|\mathrm{Sgn}(x) = x$ yields the general form for the passive SDOF joint:

$$m\ddot{x} + c\dot{x} + (k + k_1\tan^2\alpha)x + \mu(k_0 + k_1\tan\alpha|x|)\mathrm{Sgn}(\dot{x}) + k_0\tan\alpha\,\mathrm{Sgn}(x) = F(t) \tag{8}$$

Note that the term $k_1\tan\alpha$ in the passive joint model plays a role similar to K_1 in the active joint model.

Two-Beam Model

Consider the system shown in Fig. 3 consisting of two linear flexible beams connected with an active or passive revolute joint. The SDOF joint models discussed earlier were translational or prismatic in nature. However, it is clear that placing two prismatic joints symmetrically about a traditional pin joint will approximate either of the two configurations shown in Figs. 1a or 1b. The mathematical model for this system will be developed using a component mode approach. The flexural displacement of the two beams are denoted w_1 and w_2 and are expanded as

$$w_1(x_1,t) = \sum_{i=1}^{N} a_i(t)\phi_i(x_1); \qquad w_2(x_2,t) = \sum_{i=1}^{N} b_i(t)\psi_i(x_2) \tag{9}$$

where the $a_i(t)$ and the $b_i(t)$ can be thought of as generalized coordinates. The basis functions ϕ_i and ψ_i are chosen to be the normalized eigenfunctions for pinned-pinned beams of length L_1 and L_2, respectively:

$$\phi_n(x_1) = \left(\frac{2}{m_1 L_1}\right)^{1/2} \sin\left(\frac{n\pi x_1}{L_1}\right) \tag{10a}$$

$$\psi_n(x_2) = \left(\frac{2}{m_2 L_2}\right)^{1/2} \sin\left(\frac{n\pi x_2}{L_2}\right) \tag{10b}$$

where m_1 and m_2 are the mass/unit length for beams 1 and 2, respectively. Using these eigenfunctions as the basis functions yields the following equations of motion for the combined system:

$$\ddot{a}_i + 2\zeta_{1i}\omega_{1i}\dot{a}_i + \omega_{1i}^2 a_i = F_1(t)\phi(x_{F_1}) + M(t)\phi_i'(L_1), \qquad i = 1,N \tag{11}$$

$$\ddot{b}_i + 2\zeta_{2i}\omega_{2i}\dot{b}_i + \omega_{2i}^2 b_i = F_2(t)\phi_2(x_F) - M(t)\psi_i'(0), \qquad i = 1,N \tag{12}$$

$$M(t) = k_r\theta + rR(r\theta, r\dot{\theta}) \tag{13}$$

$$\theta(t) = \sum_{i=1}^{N} b_i(t)\psi_i'(0) - \sum_{j=1}^{N} a_i(t)\phi_i'(L_1) \tag{14}$$

$$\dot{\theta}(t) = \sum_{i=1}^{N} \dot{b}_i(t)\psi_i'(0) - \sum_{j=1}^{N} \dot{a}_i(t)\phi_i'(L_1) \tag{15}$$

$$' \equiv \frac{d}{dx_j}; \qquad j = 1 \text{ or } 2 \tag{16}$$

where $M(t)$ is the moment transmitted between the two beams through the joint and where $R(\cdot,\cdot)$ is defined for the active joint in Eq. (2) and for the passive joint in Eq. (6). The angle θ represents the relative angular displacement between the left end of the second beam and the right end of the first beam. Again, N_A and N_P are the forces applied normal to the plane of relative motion (N_A and N_P correspond to the active and the passive joints, respectively). The quantity r is an appropriate length dimension such that $\mu r N_A$ (or $\mu r N_P$) is the magnitude of

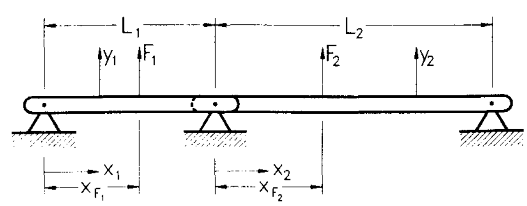

Fig. 3 Two-span flexible beam system connected through an active/passive revolute joint.

the frictional *moment* transmitted between the two beams. The total moment transmitted between the two beams is composed of the frictional moment, an elastic moment due to a restraining spring (having stiffness coefficient k_r), and for the passive joint an additional moment corresponding to the second term in Eq. (6). Note that ω_{ij} represents the jth natural frequency of the ith beam, without any coupling between the two beams. These are given by

$$\omega_{ij} = j^2\pi^2\left(\frac{EI_i}{m_i L_i^4}\right)^{1/2}, \qquad i = 1,2 \quad j = 1,N \tag{17}$$

Equations (11) and (12) can be placed in state-space form by defining the state vector to be

$$x = [a_1, a_2, \ldots, a_N, b_1, \ldots, b_N, \dot{a}_1, \ldots, \dot{a}_N, \dot{b}_1, \ldots, \dot{b}_N]^T \tag{18}$$

and defining

$$F(t) = [F_1(t), F_2(t)]^T \tag{19}$$

Equations (11) and (12) can be written

$$\dot{x} = Ax + GF(t) + BM(t) \tag{20}$$

where A, G, and B are easily identified.

The system outputs y_1 and y_2 are defined to be the flexural displacements of the first beam at $x_1 = 0.25L_1$ and of the second beam at $x_2 = 0.75L_2$, respectively:

$$y_1 = \sum_{i=1}^{N} a_i(t)\phi_i(0.25L_1); \qquad y_2 = \sum_{i=1}^{N} b_i(t)\psi_i(0.75L_2) \tag{21}$$

Defining $y = [y_1, y_2]^T$ and using Eq. (18), it is straightforward to relate y to x as

$$y = Cx \tag{22}$$

Finally, it is convenient to relate the relative angular displacement θ to the state vector x:

$$\theta = D_1 x; \qquad \dot{\theta} = D_1\dot{x} = D_2 x \tag{23}$$

where D_1 and D_2 are $1 \times 4N$ constant-valued row vectors that are easily identified using Eqs. (14), (15), and (18).

The two-beam system may be viewed as a general case of two linear, distributed elastic systems coupled with an active/passive joint. The pinned-pinned configuration is chosen solely for the resulting simple form of the modal expansions. The modal formulation is applicable to arbitrary coupled elastic systems by simply replacing the modal eigenfunctions and natural frequencies with those of the actual system.

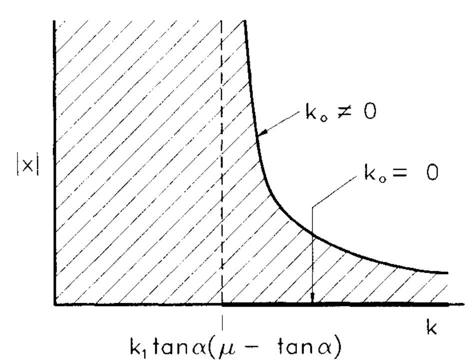

Fig. 4 Sticking values of x as a function of the elastic spring constant k for the case $\mu > \tan\alpha$. The shaded region is the region in which sticking will occur. Note that for $k < k_1\tan\alpha\ (\mu - \tan\alpha)$ sticking occurs for all values of x regardless of the value of k_0.

Sticking Regions

An important consideration in the design of an active or passive joint is joint lockup or sticking. Sticking degrades the system performance for two reasons. First, since the energy dissipation mechanism is dry friction, slipping is necessary for positive damping to occur. Intermittent sticking results in a degradation of energy dissipation; therefore it should be avoided. Second, permanent lockup is undesirable because it destroys asymptotic stability.

Sticking conditions can be obtained by examining the impending motion of the system when the relative slip velocity is zero. The body of work in the field of variable structure systems (VSS)[21] can be drawn upon to determine the sticking conditions and the resultant system dynamics when sticking occurs. (In the terminology of VSS, sticking of dry friction damped systems is referred to as *sliding*. In this paper, sticking and sliding will refer to the physical condition of the frictional interface.) For the SDOF system, sticking occurs when the following condition is met:

$$\dot{x}\,\ddot{x} < 0 \ \forall\ (x,\dot{x}) \text{ such that } |\dot{x}| < \epsilon \tag{24}$$

where ϵ is a small positive parameter. Substitution of $\ddot{x}$ from Eq. (1) into Eq. (24) yields specific conditions for sticking to occur in both the active SDOF system and the passive SDOF system.

Through application of Eq. (24), the sticking conditions for the active SDOF system with no external forcing are given as follows:

1) If $\mu K_1 \geq K$, then the system sticks for all x.
2) If $\mu K_1 < K$, then the system sticks for all x such that

$$|x| \leq \frac{\mu K_0}{K - \mu K_1} \tag{25}$$

The sticking conditions for the passive SDOF system with external forcing are given as follows:

1) If $\mu > \tan\alpha$ and $k \leq (\mu - \tan\alpha)k_1\tan\alpha$, then sticking occurs for all x.
2) If $\mu > \tan\alpha$ and $k > (\mu - \mu\tan\alpha)k_1\tan\alpha$, then sticking occurs for all x such that

$$|x| < \frac{(\mu - \tan\alpha)\,k_0}{k - (\mu - \tan\alpha)\,k_1\tan\alpha} \tag{26}$$

The sticking conditions for the passive joint are shown graphically in Fig. 4. As seen in the figure, when the elastic spring constant k is sufficiently small, sticking occurs for all values of x. For larger values of k, sticking occurs for smaller and smaller values of x. In other words, the sticking region grows smaller as the value of the elastic spring constant (or the value of α) increases. Also note that for the case $k_0 = 0$, no sticking occurs for $k > k_1\tan\alpha(\mu - \tan\alpha)$. These sticking conditions can be used to design a frictional interface so that it does not stick, or at least so that the response is ultimately bounded. Note that a sufficient condition for sticking not to occur is

$$\mu \leq \tan\alpha \tag{27}$$

When sticking occurs, an expression for the system dynamics can be obtained using the equivalent control method.[21] The expression for both the active and the passive SDOF system is given as follows:

$$m\ddot{x} = 0 \tag{28}$$

This system is linear and contains two zero eigenvalues resulting from the constraint that the velocity is zero during sticking. It should be noted that, for the unforced case $F(t) = 0$, once the system has stuck, it remains terminally stuck. Thus, while the system is stable in the sense of Lyapunov, it is not

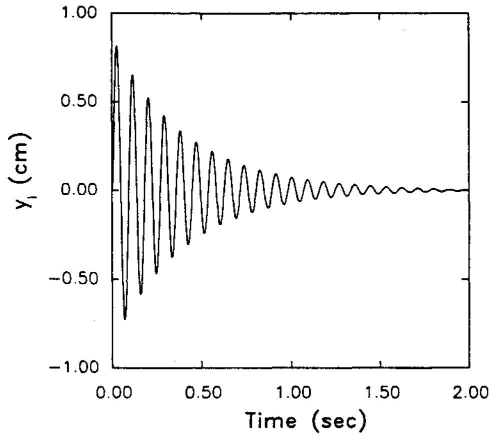

Fig. 5 Free response of y_1 for two-span beam system with passive joint ($k_0 = 0$, $k_1 = 200{,}000$, $\alpha = 10$ deg).

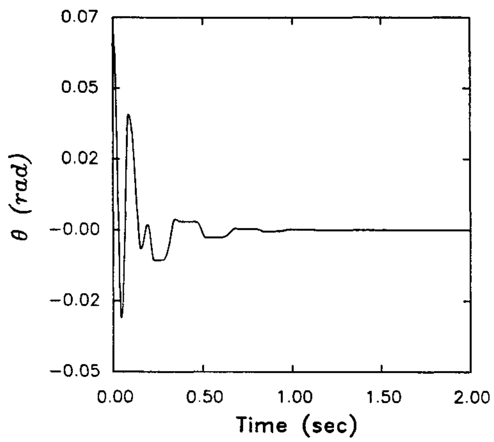

Fig. 6 Free response of θ for two-span beam system with passive joint ($k_0 = 0$, $k_1 = 1{,}000{,}000$, $\alpha = 10$ deg).

necessarily asymptotically stable. Also note that stability is preserved even under a control system failure.

The sticking conditions for the two-beam system with either the passive or the active joints are obtained in the same way as for the SDOF system. The relative slip displacement x must be replaced by the relative angular displacement θ. The first and second derivatives of θ with respect to time can be expressed in terms of the state vector $\boldsymbol{x}$ through use of Eqs. (20–23):

$$\dot{\theta}\,\ddot{\theta} = x^T D_2^T D_2 \dot{x} < 0 \ \forall\ (x, \dot{x}) \text{ such that } |\dot{\theta}| < \epsilon \tag{29}$$

where, again, ϵ is a small positive parameter. Substitution of Eq. (20) into Eq. (29) yields the sticking condition:

$$x^T D_2^T D_2 \,[Ax + GF(t) + BM(t)] < 0 \ \forall\ (x, \dot{x}) \quad \text{such that } |\dot{\theta}| < \epsilon \tag{30}$$

An expression for the dynamics of the system during sticking is again found using the equivalent control method.[21] Interestingly, the dynamics of the system during sticking are the same for both the passive and the active joints:

$$\dot{x} = [A - B(D_2B)^{-1}D_2A]x + [G - B(D_2B)^{-1}D_2G]F \tag{31}$$

As in the SDOF case, the dynamics of the system during sticking is linear with two zero eigenvalues corresponding to the constraint that the angular joint velocity must be zero for sticking to occur. Unlike the SDOF system, the unforced two-beam system can still experience motion when the frictional interface is stuck. Thus when sticking occurs, the damping of the system is reduced to that associated with flexural deformation of the beams. It is also interesting that the unforced system can break loose by itself so that stick-slip motion is possible.

IV. Numerical Results

A variety of simulation results are presented next to show the qualitative effect of the various joint design parameters on the impulse response of the two-beam model. Both active and passive joint configurations are considered. For each case presented, the properties of the A, B, C, D_1, and D_2 matrices are kept the same. A single beam mode is used to represent each beam. (Results not presented here show that numerical simulation results using two modes per beam and five modes per beam yield very similar results as those produced using one mode per beam.) The following numerical values are used for the system parameters: $m_1 = m_2 = 0.4$ kg/m^3, $L_1 = 1.0$ m, $L_2 = 1.8$ m, $EI_1 = EI_2 = 20$ N·m^2, $\mu = 0.5$, $r = 0.025$ m, and $\zeta_{11} = \zeta_{21} = 0$. Unless otherwise stated, the results shown next were generated using an impulsive force of amplitude 25 N and duration 0.01 s is applied to the first beam at $x_1 = 0.25L_1$.

Figures 5–9 pertain to the two-span beam system connected by the passive joint. Figures 5 and 6 show the relative effect of the transverse spring constant k_1. Figure 5 shows the free response of y_1 for $k_1 = 200{,}000$ N/m and $\alpha = 10$ deg. Note that the envelope of decay is exponential in shape, similar to the case of linear, viscous damping. It is important to note that the only source of damping in this system is dry friction since the beam structural damping has been set equal to zero. As the value of k_1 is increased, the response damps out faster,

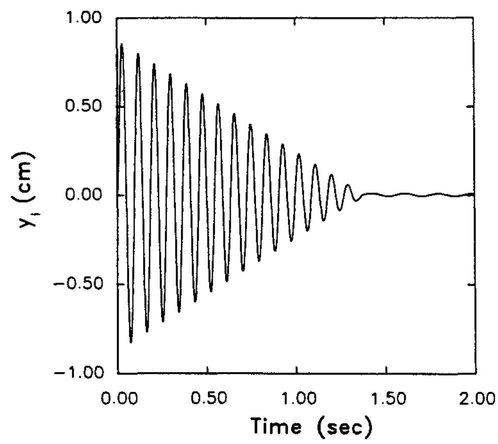

Fig. 7 Free response of y_1 for two-span beam system with passive joint ($k_0 = 5$, $k_1 = 0$, $\alpha = 0$ deg).

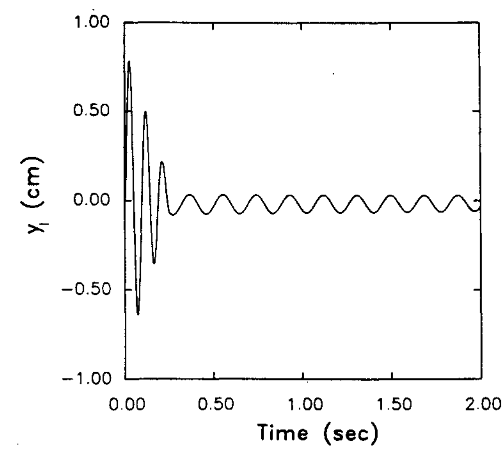

Fig. 8 Free response of y_1 for two-span beam system with passive joint ($k_0 = 25$, $k_1 = 0$, $\alpha = 0$ deg).

but the envelopes of decay remain exponential in shape for large values of displacement. Figure 6 shows the free response of the relative angular displacement in the joint θ for $k_1 = 1 \times 10^6$ N/m. (This result was generated using the initial condition $x = [0.01,0,0,0]^T$.) It is seen that stick-slip motion becomes prevalent as k_1 increases. The constant-valued portions of the curve in Fig. 6 correspond to sticking.

Figures 7 and 8 show the relative effect of k_0 on the free response. For these plots, $d = 0$ and $k_1 = 0$; therefore, this corresponds to the case of constant normal force, i.e., the

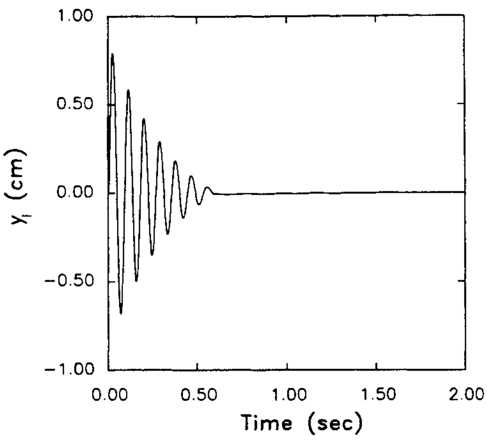

Fig. 9 Free response of y_1 for two-span beam system with passive joint ($k_0 = 5$, $k_1 = 200{,}000$, $\alpha = 10$ deg).

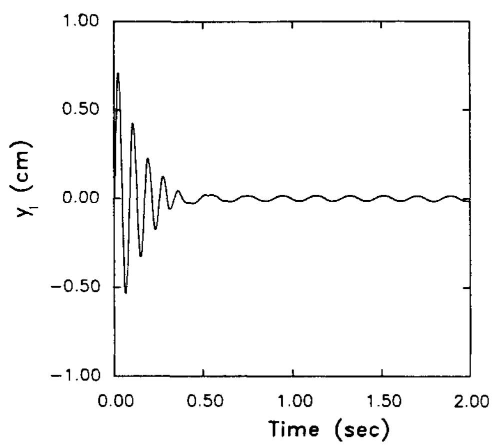

Fig. 10 Free response of y_1 for two-span beam system with passive joint ($k_0 = 5$, $k_1 = 200{,}000$, $\alpha = 20$ deg).

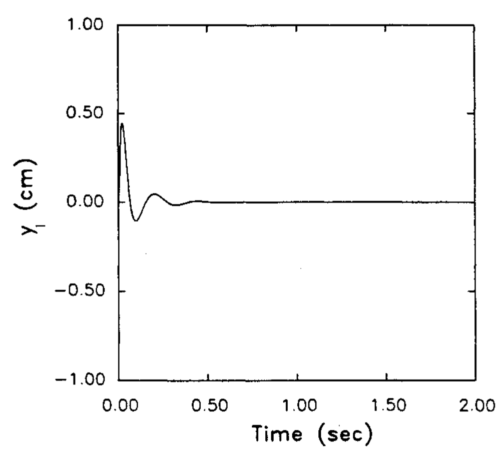

Fig. 11 Free response of y_1 for two-span beam system with active joint ($K_0 = 0$, $K_1 = 0$, $K_2 = 6000$).

classic dry friction case. Note that the envelopes of decay in these figures are linear like those traditionally associated with dry friction. As the value of k_0 is increased, the oscillations die out more rapidly; however, sticking becomes more noticeable. Observe that in Fig. 8, once permanent sticking has occurred ($t \simeq 0.25$ s), no further energy dissipation takes place and a steady-state oscillation results.

Figure 9 shows the free response of y_1 for the passive joint case with $k_0 = 5$, $k_1 = 200{,}000$, and $\alpha = 10$ deg. Note that the system exhibits both linear and exponential envelopes of decay. The envelope is more nearly exponential for large values of displacement and more nearly linear for small values of displacement.

The effect of α is seen by comparing Fig. 9 (where $\alpha = 10$ deg) with Fig. 10 (where $\alpha = 20$ deg). Essentially, increasing the value of α has two effects: it increases the natural frequency of the system by adding an apparent stiffness to the joint, and it increases the rate of decay of the free response.

The performance of the active joint is seen in Fig. 11. Figure 11 corresponds to the case $K_0 = K_1 = 0$, $K_2 = 6000$ N/(m/s). Recall that setting K_0 and K_1 both equal to zero results in the joint characteristics being equivalent to those of a linear viscous damping element. The value $K_2 = 6000$ N/(m/s) corresponds to the viscous damping element that gives the highest level of damping to the most lightly damped linear mode of vibration. As can be seen, excellent performance is obtained. An extensive study was conducted varying K_0, K_1, and K_2 for the system with the parameters just defined. It was found that using nonzero values of K_0 and K_1 in conjunction with $K_2 = 6000$ N/(m/s) resulted in the degradation of system performance over that shown in Fig. 11.

V. Conclusions

These results suggest that joints designed with amplitude or rate-dependent frictional forces can offer substantial improvements in performance over joints with constant normal forces. One advantage is that the overall damping characteristics can be made to resemble those of a linear viscous damping element. Thus, it may be possible to incorporate the joints in a global structural model as simple *linear* elements, greatly simplifying the analysis of the assembled structure. In addition, the modified joints are well suited to a space environment and do not present a significant weight penalty. It should be added that both the active and passive joint types can be used together in a single space structure. Although it may be unnecessary (or even impractical) to place active or passive joints at *every truss structure junction*, it may be possible to achieve good results simply by placing a few active or passive joints at strategic locations throughout the structure. The interaction of the active and passive joints with an overall attitude or pointing control strategy also needs to be addressed. This paper presents only one structure of feedback control law for the active joint. Many other schemes using both collocated and noncollocated feedback are possible. A particular feature that should be considered in future control laws is the ability to adapt or learn as the configuration of the space structure changes (for example, during construction) or as the frictional properties in the joint vary. Finally, this paper shows the feasibility of passive and active joints through numerical simulations. The actual performance of these types of joints has not yet been studied in a laboratory setting.

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