DYNAMIC FREQUENCY DE-TUNING USING CONTROLLABLE BEAM-COLUMN SEMI-RIGID CONNECTIONS

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ABSTRACT

The concept of frequency de-tuning and damping enhancement to avoid resonant frequencies and to suppress the vibration response of structures, respectively, has been known for passive vibration control systems since mid 1950's. Ideally the concept shifts the structural frequency away from resonant frequency and significantly reduces the structural response in the 'narrow-band' damping-controlled region. However in reality, the passive vibration control performance depends heavily on the dynamic characteristics of energy dissipation systems that are fixed and only applicable for a particular frequency band. Conversely, the Direct Current (DC) controllable beam-column semi-rigid connections using Magneto-rheological (MR) rotational fluid dampers, can act as real time adjustable passive control devices for any required frequency band. The supplied DC current level can adjust the device torque and rotational velocity characteristics that affect the structural frequency.

This paper presents a preliminary parametric study on vibration control performance of a single and a two storey prototype plane frame model due to combination of frequency de-tuning and its associated damping enhancement by selecting end beam connections as either hinged or fixed. The controllable semi-rigid connection characteristics are represented in terms of normalised frequency. And the control performance indicated by Amplitude Frequency Characteristics (AFC) of the models is studied numerically and presented graphically. The study yields an insight into the controllable connections prior to any experiments.

INTRODUCTION

Conventional steel frames with bolted connection systems are known to exhibit larger damping than those of welded connection systems. Several studies have indicated that the vibration response levels decrease due to the rotational friction mechanism at the connections during vibrations. It was believed that the passive control mechanism can change both structural stiffness and damping levels. The change of structural stiffness levels aims at moving the system frequencies away from the damping-controlled region, while the increased damping will reduce the structural response levels within the damping-controlled region. Passive vibration control systems, using some devices for frequency de-tuning (Mead D.J. 2000) and damping enhancement, have been extensively researched for more than two decades. The use of passive dissipative devices, tuned mass dampers (TMD), tuned liquid dampers, fluid dampers, vibration absorbers, metallic dampers and friction dampers cited in literature (Mead D.J 2000, Hartog, J.P 1956, Soong T.T & Dargush G.F 1997, Koronev BG and Reznikov LM 1993, Housner G.W et al. 1997) showed that some devices are superior in damping enhancement, while the others are better in frequency de-tuning. The only shortcoming of these passive control devices is their non-adjustable dynamic characteristics. However, the dynamic characteristics of the proposed controllable semi-rigid connections, using controllable MR rotational fluid dampers, can be actively adjusted in real time similar to shear dampers (Samali et al. 2004). It is expected that by using the proposed system, the vibration control performance can be markedly improved as shown in the simulation and further experimental results.

2. STRUCTURAL VIBRATION CONTROL

The dynamic model for structural systems with semi-rigid beam-column connections, treated as a linear system, can be expressed in modal coordinate system as

$$\ddot{x}+2\zeta_j\omega_j\dot{x}+\omega_j^2x=\phi_j^T M^{-1}F \qquad (1)$$

In Eq. (1), M_j, ζ_j, ω_j, ϕ_j, F and x are the j^{th} structural normalised modal mass, the j^{th} structural modal equivalent damping ratio, the j^{th} structural modal frequency, the j^{th} normalised mode shape vector, the excitation force matrix and the modal displacement, respectively. The study focuses on lower mode frequencies that are associated with lateral displacement responses. The higher mode frequencies that are associated with vertical and rotational displacement responses are eliminated using classic Guyan's condensation (Xu Y.L. and Zhang W.S. 2001). As a result, a single or a two-storey plane frame model can be represented as a single or a two lateral degree of freedom dynamic system. De-coupling the condensed dynamic system is carried-out using modal matrix to yield a set of single degree of freedom systems. For the purpose of parametric analysis, the structural system is further represented by its structural frequency in Eq. (1).

A simple technique to suppress vibration response is to avoid resonance region and to increase damping of the system. The use of passive or semi-active, active and hybrid energy dissipation systems is basically to passively or actively tune the system frequency away from the imposed frequency band and converting vibratory energy into other types of energy such as heat, sound, etc. Active frequency de-tuning of a frame model, having controllable beam-column semi-rigid connections, can be carried-out by adjusting the connections' dynamic characteristics. It can be seen in Figure 1 that by setting the connection types as either rigid or hinged conditions accordingly, the vibration response can be suppressed even with no damping enhancement. The frequency setting is similar to control pole selection in the pole placement technique. Moreover, the solid lines in Figure 1 indicate the control strategy by tuning, appropriately, using either rigid or hinged type connection.

For the multi-DOF systems, setting the system frequency to hinged condition should be carefully exercised as the second mode frequency is expected to be larger or equal to that for a rigid connection frame frequency ω_R. This frequency setting will be capable of suppressing the resulting vibrations in the frequency band between ω_R and ω_H. If this requirement cannot be achieved, then the controlled frequency band will be in the range of first mode to the second mode frequency of hinged connection frame system. Moreover, theoretically the controlled frequency band can be classified into three regions: region ***I*** is rigid connection controlled, region ***II*** is structural mass or stiffness controlled and region ***III*** is hinged connection controlled.

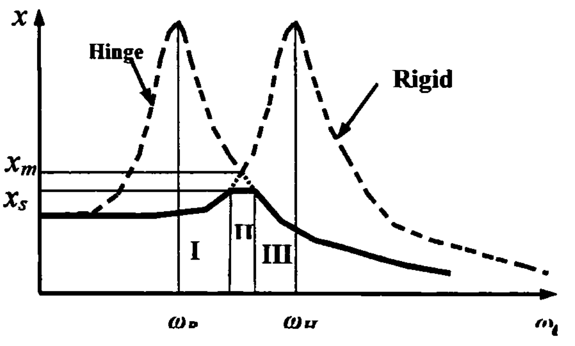

Figure 1: Typical vibration response versus model frequencies excited by forced frequency λ.

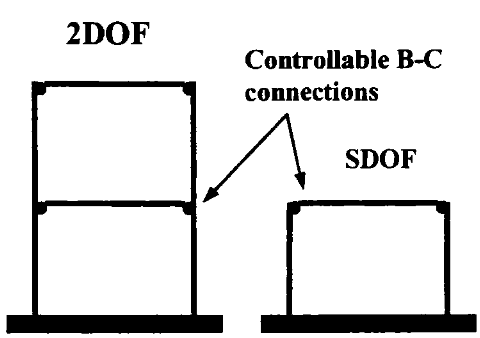

Figure 2: Typical one (SDOF) and two storey (2DOF) plane frame models.

The solid line in Figure. 1 indicates the displacement response constraint x_s due to structural member strength or serviceability criteria whichever is smaller. The use of damping to suppress vibration response x_m equal to or smaller than x_s is not effective, as region ***II*** is not in the resonant region for both hinged or rigid connection systems. The only way to overcome this problem is to increase the structural stiffness and structural damping simultaneously. To increase structural stiffness with small change in structural mass by redesigning the structural system may be effective in this regard.

In order to evaluate the performance of this dynamic frequency de-tuning and damping enhancement (Hartog J.P. 1956) for vibration control, the control performance is quantified in terms of amplitude ratio expressed as Amplitude Frequency Characteristics (AFC). The AFC as criteria can be amplitude ratios of displacement, velocity, acceleration or base shear response. To ease the formulation, periodic excitations with frequencies λ_k are selected as the forcing term in Eq. (1) and expressed in modal term as

$$\mathbf{F}_j = \phi_j^T \sum_{k=0}^{n} \mathbf{F}_k \exp(i\lambda_k t) \tag{2}$$

In Eq. (2), F_k is the kth harmonic force amplitude; subscript index k (0,1,2,....n) is index of the armonic or periodic excitations and n is the number of harmonic excitations for a certain duration of vibration. In view of linearity and principle of superposition for a single input harmonic force excitation, the AFC for displacement response or gain, denoted as A, can be represented in terms of the frequency ratio and the damping ratio ζ as

$$\boldsymbol{A} = \left\| \sum_{k=1}^{n} \left(\sum_{j=1}^{m} \left\{ (\phi_j \phi_j^T)^2 / \left((1-\lambda_k^2/\omega_j^2)^2 + (2\zeta_j \lambda_k / \omega_j)^2 \right) \right\} \right) \right\|^{1/2} \tag{3}$$

Eq. (3) shows AFC for displacement or dynamic amplification factor as Euclidean norm of matrix $\boldsymbol{A}$ that is a function of adjustable system frequency ω_j and subscript j representing the system mode number. The AFC can also be expressed in modal displacements (Biggs J.M. 1964).This expression holds for viscous damping treated as Rayleigh damping. In case of hysteretic damping, the above expression can be modified using 'modal loss factor' (Mead J.D. 2000). One can see in Eq. (3) that frequency de-tuning becomes ineffective for periodic excitations having a wide band frequency content, unless the system has an actively adjustable frequency ω_j and large damping ratio. The use of passive tuning frequency can be effective in suppressing the known or recorded human induced or man made equipment/machinery/vehicle vibrations. However, for unpredicted or unknown vibration characteristics, mostly due to environmental excitations (Hao H. 2002), Eq. (3) does not hold if used for control purposes. The use of Duhamel's (convolution) integral, to solve both non-deterministic environmental and deterministic excitations using the concept of impulse response, seems suitable for real time control purposes. To do so the forcing term is represented by a group of discrete step forces as

$$\boldsymbol{F}_{jl} = \boldsymbol{\phi}_j^T \boldsymbol{F}_k \delta(l-k) \qquad l \geq k = 0,1,2......p \tag{4}$$

In Eq. (4) $\delta(..)$ is Dirac delta function, the second subscript l is the considered discrete time step and p is the number of discrete time steps. As non-periodic excitations normally have broad frequency band, the AFC represented by the response ratio displacement response to root mean square (RMS) static displacement can be expressed as

$$\boldsymbol{A}_t = \left\| \left(\sum_{j=1}^{m} (\boldsymbol{\phi}_j \boldsymbol{\phi}_j^T) \frac{\sum_{k=0}^{l} \boldsymbol{F}_k T \left(\exp(-\boldsymbol{\zeta}\omega_j (l-k)T) \sin(\omega_{dj}(l-k)T) \right)}{\sqrt{1-\boldsymbol{\zeta}_j^2} \boldsymbol{\phi}_j^T \boldsymbol{K} \boldsymbol{\phi}_j} \right) \Big/ \left(\boldsymbol{K}^{-1} \left(\frac{1}{p+1} \sum_{k=1}^{p} \boldsymbol{F}_k^2 \right)^{1/2} \right) \right\|^{1/2} \tag{5}$$

Eq. (5), where $\boldsymbol{K}$ and T are the normalised structural lateral stiffness matrix and the time step or inverse of sampling frequency, respectively, shows that the AFC at any particular time t, can be minimized by dynamic tuning of system frequency ω and its associative damping ratio ζ. Furthermore, by varying the system frequency and taking the maximum response in the numerator of Eq. (5), one can observe, from the developed response spectra, the associated frequency ratio that leads to its minimum displacement response. This de-tuning technique is possible for vibration control if the frequency content of force spectrum is known beforehand. However, in view of Eq. (5), the AFC can be minimised if the exponential term is minimised, accordingly. This approach seems likely ineffective for real time control systems as it involves previous forcing terms. The other Duhamel's integral for recursion formula (Dempsey and Irvine 1978) that computes the displacement response at the considered time step using two displacements of previous steps can be synchronized with real time control signals. For dynamic systems which are at rest and employing a piecewise linear forcing function between two time stations or First Order Hold (FOH) sampling for signals, the AFC expressed in recursion relationship for the displacement response ratio can be derived as

$$A_k = \left\{ \sum_{j=1}^{m} \phi_j \phi_j^T \left(\frac{(B_1 F_{k-2,j} + (B_3 - B_2) F_{k-1,j} + B_4 F_{k,j})}{\phi_j^T M \phi_j} \right) + (B_5 x_{k-1} + B_6 x_{k-2}) \right\} / \left\{ K^{-1} \left(\frac{1}{p+1} \sum_{k=1}^{p} F_k^2 \right)^{1/2} \right\} \quad 6)$$

In Eq. (6), B_1, B_2, B_3, B_4, B_5, B_6 and F_j are recursion constants and root mean square of j^{th} modal force, respectively. For control purposes, the mean square force can be taken as the product of lateral static stiffness and x_s.

$$\begin{aligned}
B_{1,j} &= a\{(1+b)a - (b\cos(\omega_{d,j}T) + (1-c)\sin(\omega_{d,j}T)/(\omega_{d,j}T)\}/\omega_j^2 \\
B_{2,j} &= B_1 + a\{\cos(\omega_{dj}T) - a - \zeta_j \sin(\omega_{d,j}T)/\sqrt{1-\zeta_j^2}\}/\omega_j^2 \\
B_{3,j} &= b - a\{(1+b)\cos(\omega_{d,j}T) - ((1-c)/(\omega_{d,j}T) - \zeta_j/\sqrt{1-\zeta_j^2})\sin(\omega_{d,j}T)\}/\omega_j^2 \\
B_{4,j} &= -B_3 + \{1 - a(\cos(\omega_{d,j}T) + \zeta_j \sin(\omega_{d,j}T)/\sqrt{1-\zeta_j^2})\}/\omega_j^2 \qquad (7) \\
B_{5,j} &= 2a\cos(\omega_{d,j}T) \qquad B_{6,j} = -\exp(-2\zeta_j \omega_j T) \\
a &= \exp(-\zeta_j \omega_j T) \qquad b = 2\zeta_j/(\omega_j T) \\
c &= 2\zeta_j^2 \qquad \omega_{dj} = \omega_j \sqrt{1-\zeta_j^2}
\end{aligned}$$

Both Eqs. (5) and (7) indicate the role of exponential term and circular frequency in minimizing the displacement response at any time step. For the case of earthquake loading, the forcing term in Eq. (1) can be expressed as $F = -MHa_g$ (8)

In Eq. (8), H and a_g are the location vector (unity vector) and earthquake acceleration, respectively.

3. NUMERICAL STUDY

To demonstrate the concept of dynamic frequency de-tuning and damping enhancement of controllable semi-rigid connections, the study used both the single and the two storey plane frame models shown in Figure 2 having dimensionless parameters as in Table 1. In order to present two AFC curves in one plot, the normalised frequency is taken as λ/ω_0, where $(\omega_0)^2 = 2EI_0/(M_0L^3)$; E, I_0, M_0 and L are modulus of elasticity, any chosen moment of inertia, any chosen mass and beam span length, respectively.

Table 1. Model dimensionless parameters.

System	α	β	γ	δ	λ/ω_0	Force type
SDOF	1	0.01; 20	1	1	0.00 - 20	Harmonic
2DOF	[1]	0.01; 20	[1; 2; 4]	[1]	0.00 - 20	Harmonic, Random

In Table 1, α, β, γ and δ are ratio of column height to beam span, ratio of beam to column moment of inertia, ratio of lower to upper column moment of inertia and ratio of lumped mass to M_0, respectively, while notation [1] represents an identity matrix.

Maximum response ratios for a SDOF and a two-DOF system under a single harmonic frequency known as dynamic amplification factor is obtained using Eq. (5) and presented in Figures 3 and 4, respectively. For a random excitation such as El Centro earthquake, the response ratio using frequency de-tuning technique by switching beam end connections or dampers between hinged and rigid, is computed using Eq. (6) and presented in Figure. 5. The displacement response ratio due to damping enhancement-using equivalent damping ratio resulting from controllable dampers' friction and viscous characteristics, is presented in Figure 6.

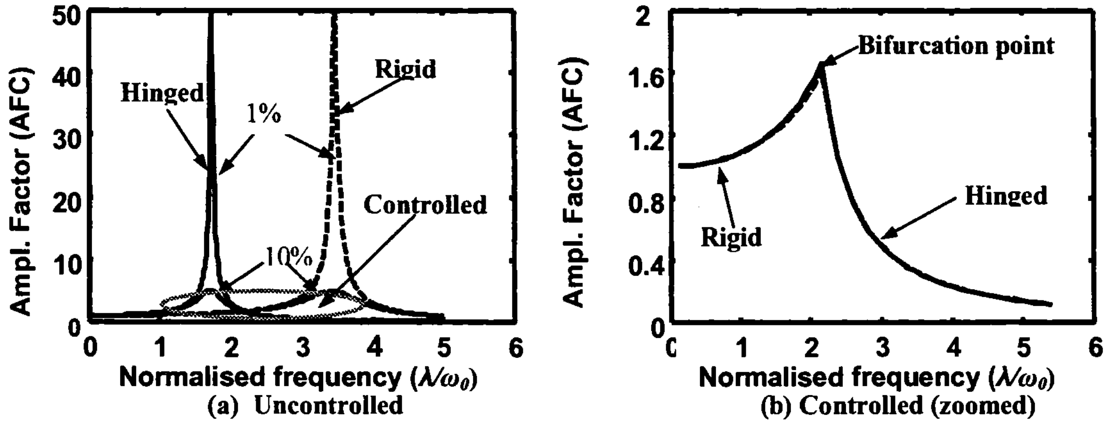

Figure 3: Dynamic Amplification Factor versus normalized frequency (SDOF system)

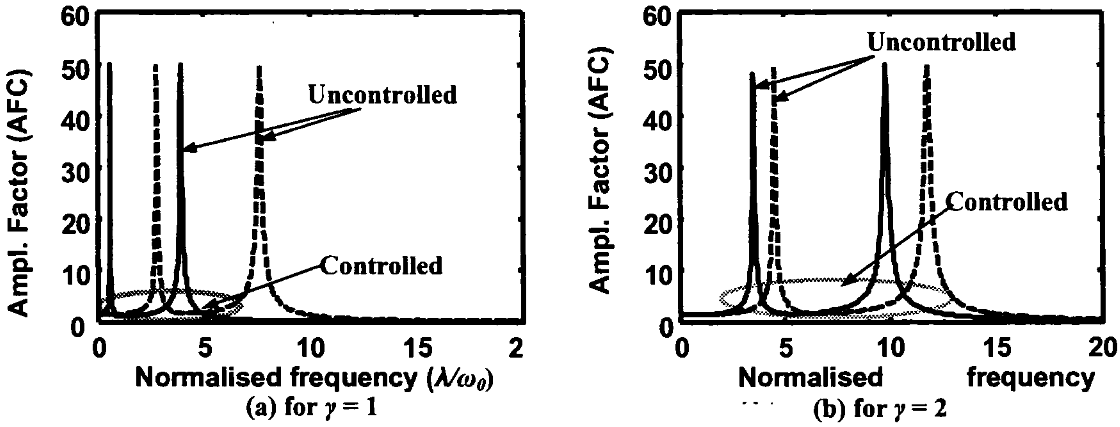

Figure 4: Dynamic Amplification Factor versus normalized frequency (two-DOF system) with damping ratio of 1 %.

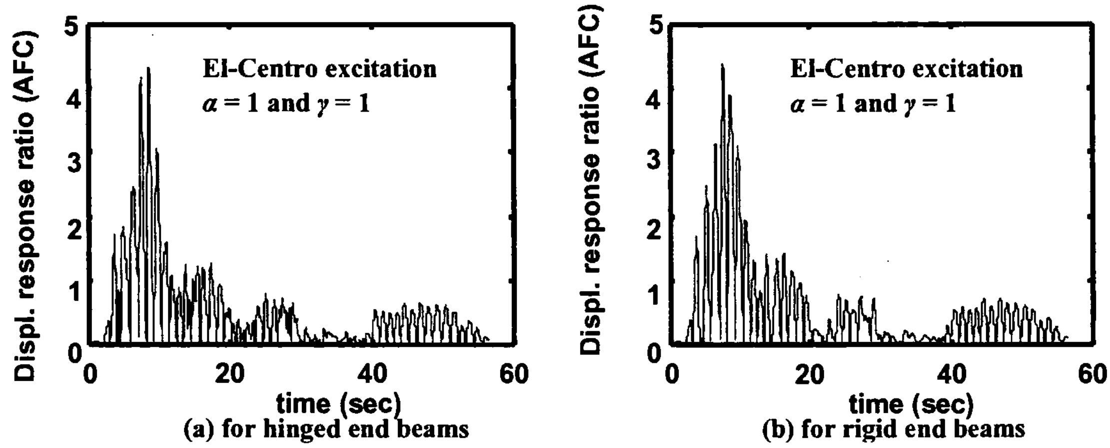

Figure 5: Displacement response ratio versus time (two-DOF system) with damping ratio of 5 %.

4. DISCUSSIONS AND CONCLUSIONS

The use of frequency de-tuning technique to suppress vibration response of dynamic systems is effective for harmonic excitations even for small damping as shown in Figs. 3 and 4. This is believed to be the case for periodic and narrow band-width random excitations as well. However, the technique of de-tuning structures by either hinged or rigid end beam conditions cannot guarantee good control performance of a structure when subjected to wide-band random excitations with frequency band of 0 to 5 Hz., such as El-Centro earthquake with 60% scaled peak ground acceleration intensity as shown in Figure 5. However, passive de-tuning, by setting a fixed structural frequency and applying a constant equivalent damping ratio, passive control performance can be achieved as seen in Figure 6. In this situation, controllable connections are supplied with constant DC power and act as passive dissipative energy devices. Dynamic frequency de-tuning by actively adjusting structural frequency and equivalent damping require appropriate control algorithms, controllable connection dynamic characteristics and left for future studies.

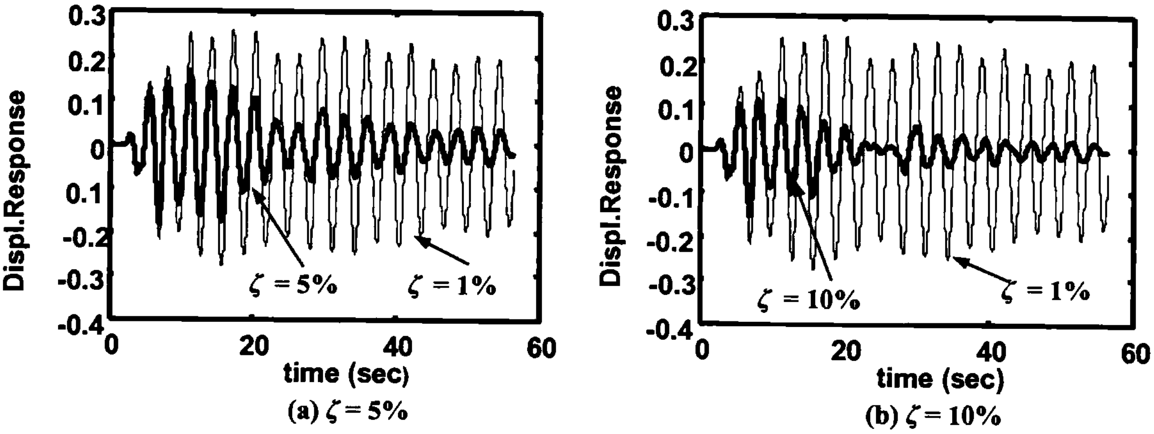

Figure 6: Typical displacement response versus time for top DOF with ω_o=1 and equivalent damping ratio of 5% and10% compared with initial structural damping ratio of 1%

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