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Multimode Input Shaping Control of Flexible Robotic Manipulators Using Frequency-Modulation

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Abstract

Robotic manipulators used in heavy industries, such as the automotive industry, are generally bulky. Driving such manipulators requires large actuators. In many cases, manipulators hardly carry any significant load compared to their sizes, such as those used for performing spot welding. However, the manipulators are oversized to avoid vibrations caused by high input command profiles. Lighter flexible manipulators, on the other hand, are superior in terms of cost and energy consumption. However, size reduction comes at a price of slower performance in order to reduce inertia induced vibrations. In the present work, a command shaping strategy is developed to facilitate operating flexible manipulators at higher speeds while eliminating inertia excited vibrations. The strategy is based on input shaping techniques complemented with a multi-mode frequency modulation control system. The performance of the proposed strategy is demonstrated on a thin beam model of a robotic arm, using numerical and finite element simulations.

Keywords: *Frequency-modulation, input shaping, multimode, flexible manipulator, hanging beam*.

1. Introduction

In heavy industries, such as the automotive industry, robotic manipulators are generally heavy and bulky. Many of those manipulators perform tasks that do not involve carrying or moving large payloads. As a matter of fact, some manipulators do not carry any load. They just perform assembly tasks, such as robots used for welding, drilling, etc. Industrial robots are made bulky for one main purpose; to guarantee positioning accuracy of their end effectors. Geometrical oversizing of robotic manipulators boosts their stiffness and reduces vibrations that may compromise operations precision. As a result, industrial robots require large powerful actuators that consume excessive amounts of energy.

Although energy efficient, the structural flexibility of light weight manipulator is a major drawback. Light weight manipulators operate at lower speeds, compared to their heavy counterparts, to avoid inertia excited vibrations. Structural flexibility of light manipulators compromises operations precision [1]. Bearing in mind that operations in many heavy industries are continuous all year long, the amount of energy savings through the use of light manipulators is substantial. The slow operation speed penalty can be elevated by implementing control systems that eliminate vibrations in light manipulators operating at high speeds comparable to those achievable by heavy manipulators.

It is for such reasons that the dynamics and vibration control of beams have received large attention during the past few decades. Ample literature is published on the vibration and control of flexible structures [2, 3]. Many control strategies have been investigated for the control of flexible manipulators including optimal control, adaptive control, fuzzy logic control, neural networks, input shaping, and others [4-10]. Research included the implementation of dominant mode linear and nonlinear vibration control of flexible structures [6], and multimode simultaneous control for more flexible structures when the dominant mode approach is insufficient [7].

Open-loop control systems are ideal for the elimination of inertia excited vibrations. One of the most common and practical open-loop control systems is known as input shaping [11]. Inertia excited vibrations are eliminated by deriving command signals to actuators that mitigate their own excited vibration. Input shaping technique is based on

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convolving a sequence of impulses with a general input command. The impulses are precisely timed so that the convolved command eliminates its own excited vibration at the conclusion of the command. Essential concepts of input shaping were published and patented by Gimpel and Calvert in 1952 and by Calvert and Gimpel in 1957 [12, 13]. Because of its sensitivity to modeling uncertainty, input shaping was not widely used until it was made popular by the work of Singer and Seering [14] and Singer *et al*. [15] on robust input shaping methods. Since then, research on input shaping control surged to include different input shaping techniques for single mode and multimode flexible structures such as Zero-Vibration (ZV), Zero-Vibration-Derivative (ZVD), Zero Vibration Derivative (ZVDD), Specified Insensitivity (SI), and Extra-Insensitive (EI) input shapers [11, 16].

As flexible structures, robotic arm are multimode vibrating systems. Controlling these systems commands the use of multimode input shaping schemes. Multimode input shapers can be classified into two main groups; convolved shapers and simultaneous shapers. Convolved shapers involve convolving multiple sequences of impulses, each sequence targeting one vibration mode of a multimode system [17-20]. However, in some cases, the frequencies involved may be so high that convolved input shapers become practically inapplicable due to bandwidth limitations [18]. Simultaneous shapers may produce faster response, however, they generally exhibit lower robustness compared to convolved shapers [21-23]. Shapers may involve positive and negative impulses [24, 25]. Input shapers containing negative impulses have several advantages [26, 27].

Input shaping can be implemented as a standalone control strategy, or as a hybrid combination with other control strategies for improved robustness [4, 28]. Standalone multimode input shapers suffer an implementation drawback due to the large number of impulses involved in their designs. Depending on the control hardware used, mismatch between impulses and sampling rates may result in performance degradation [17, 21, 22]. To overcome this drawback, hybrid combinations of single-mode input shaping with other control strategies are utilized including combining single-mode shapers with different types of filters, such as notch, low-pass, band-stop, and time-delay filters [29-34]. An optimization technique taking into consideration hardware sampling rates was derived by Alghanim *et al*. [35] in an attempt to overcome this problem.

Pre-shaped command techniques are used to overcome excessive impulses in an input command. Continuous smooth commands may reduce or eliminate impulses in shaped commands. Erkorkmaz and Altintas [36] used quintic spline trajectory generation algorithm to generate continuous position, velocity, and acceleration profiles for high speed CNC systems. Xie *et al*. [37] introduced a method to reduce vibration in flexible systems by smoothing the original command. Wave-form commands were also used to reduce the number of impulses and jerks in shaped input commands [38-41].

To reduce the number of impulses in multimode input shapers, Singh and Heppler [18] showed that a single-mode shaper can eliminate vibrations at all frequencies that are odd-multiples of the design frequency of the single-mode shaper used. Their trials were based on finding a common frequency such that all modes of a multimode system are odd-multiples of this frequency. However, there were no guarantees that such a frequency existed in a multimode system. Later, single-mode pre-shaped commands were implemented on an approximate model of a two-mode system for which such a common frequency exists [28]. Virtual feedback system was used to match the response of the exact model of the system to the approximate model. Another attempt, known as frequency-modulation input shaping, was based on modifying the frequencies of a multimode system using model-based feedback to the point where such a common frequency exists [42, 43].

In the present study, a flexible Euler-Bernoulli hanging beam is used to model a robotic arm. Flexible beams exhibit broad spectrum of resonant frequencies. This wide spectrum makes the use of multimode impulse shapers impractical. This paper describes a multimode frequency-modulation input shaping strategy used to shape acceleration commands to the base of the robot arm model. Model-based feedback is used to modulate the frequencies of the beam so that all higher mode frequencies become odd-integer multiples of the fundamental frequency of the feedback model. Single-mode input shaping techniques are implemented to eliminate vibrations in all modes of the model simultaneously. Input commands to the plant of the feedback system are used as inputs to the base of the physical model. The main advantage of the frequency-modulation input shaping strategy is that only one single-mode input shaper is needed to eliminate vibrations in all modes of the system. Numerical simulations and finite element simulations, using Abaqus-v6.12 FEA package, demonstrate the effectiveness of the proposed strategy.

2. Single-Mode Input Shaping

Input shaping is a technique used to eliminate inertia excited vibrations in dynamic systems. A general command signal is convoluted with a sequence of impulses to produce a shaped command that results in zero residual vibrations. Although input shapers have been developed earlier for multimode systems, they are hard to implement due to the large number of fast input impulses involved. Single mode input shapers are easier to implement. Several single input shaping techniques have been developed over the past two decades. The most practical input shapers in the literature are the *Zero-Vibration* (ZV) input shaping, Fig. 1(a), and the robust *Zero-Vibration-Derivative* (ZVD) input shaping [16], Fig. 1(b). This is due to the reduced number of impulses involved.

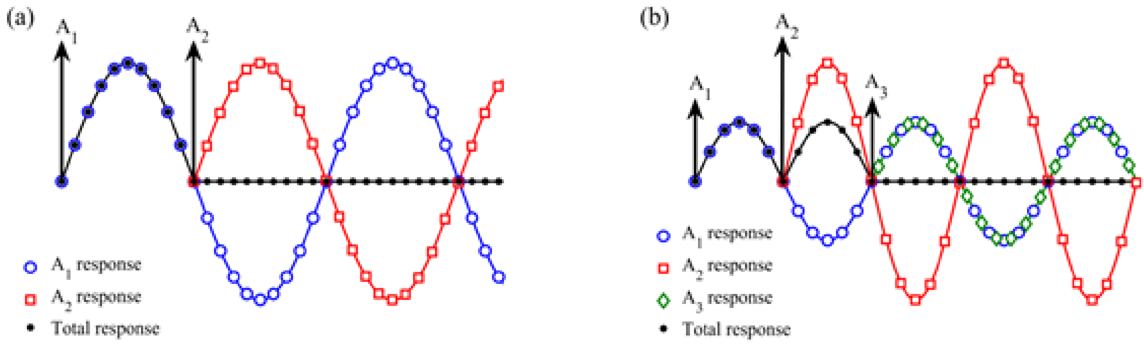

Figure 1. (a) Zero-Vibration (ZV) input shaping and (b) Zero-Vibration-Derivative (ZVD) input shaping.

Zero-Vibration input shaping is based on convoluting a general input command with a sequence of two impulses, where the response to the second impulse mitigates the vibration excited by the first impulse. The impulses are separated by half the vibration period of the system. The ZV input shaping matrix is

$$\mathrm{ZV} = \begin{bmatrix} A_i \\ t_i \end{bmatrix} = \begin{bmatrix} \frac{1}{K+1} & \frac{K}{K+1} \\ 0 & \frac{\tau_d}{2} \end{bmatrix} \quad (1)$$

where A_i and t_i are the i^{th} impulse amplitude and impulse time, respectively, τ_d is the damped period of the system, and

$$K = e^{-\zeta\pi/\sqrt{1-\zeta^2}} \quad (2)$$

where ζ is the damping ratio.

The ZVD input shaper includes three timed impulses separated by half the vibration period of the system. The ZVD matrix is

$$\mathrm{ZVD} = \begin{bmatrix} A_i \\ t_i \end{bmatrix} = \begin{bmatrix} \frac{1}{1+2K+K^2} & \frac{2K}{1+2K+K^2} & \frac{K^2}{1+2K+K^2} \\ 0 & \frac{\tau_d}{2} & \tau_d \end{bmatrix} \quad (3)$$

The ZV input shaper is a non-robust shaper, but it is the shortest in terms of its time duration. The ZVD input shaper is a robust shaper. The robustness of the ZVD shaper comes at a price of longer shaper duration. It is important here to emphasize that the aim of the proposed FM input shaping strategy is to be able to eliminate all modes of vibration of the system using one single-mode shaper. It is not a goal of the proposed FM input shaping strategy to enhance robustness nor is it a goal to increase the speed of the shaping technique used.

3. Multimode Frequency-Modulation Input Shaping

In multimode systems, a shaped command signal that eliminates vibrations of the first mode can eliminate vibrations in other modes provided that the frequencies of those modes are odd-integer multiples of the frequency of the first mode. Frequency-modulation [42, 43] can be used to modulate the frequencies of the system model to the point where the above odd-integer multiples condition is satisfied. In the present work, this task will be performed using model-based feedback strategy. Once the odd-integer multiple frequency condition is satisfied, virtually, any single-mode input shaping technique can be used to produce a shaped motion command for the multimode system. The command signal to the plant of the feedback system is used to drive the multimode system, Fig. 2.

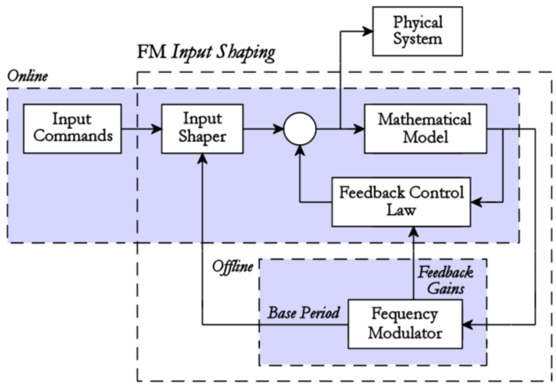

Figure 2. Schematic block diagram of frequency-modulation input shaping.

The FM input shaper, consists of two sequential stages; frequency-modulation, then input shaping. In the frequency-modulation stage, model-based feedback is used to satisfy the odd-multiple frequencies condition. The resonant frequencies of model-based feedback system are modulated to the point where all higher modes frequencies become odd-multiples of the first mode frequency. The single-mode input shaper used will be designed using the primary resonant frequency of the feedback model.

The main advantage of the Frequency-Modulation (FM) input shaping strategy is that only one single-mode input shaper is needed to eliminate vibrations in all modes of the system. This reduces the number of impulses involved in shaping process.

4. Illustrative Example

The performance of the FM input shaper is demonstrated on a thin hanging beam model of a robotic manipulator, Fig. 3. This type of manipulators is commonly used for pick and place maneuvers. A thin rectangular cross-section is used to magnify the vibration problem. The material of the beam is stainless steel. The material properties and geometric dimensions of the beam are; material density $\rho = 8030\ \mathrm{kg/m^3}$, Young's modulus of elasticity $E = 193$ GPa, beam length $l = 1$ m, thickness $h = 0.65$ mm, and width $w = 26$ mm. The slenderness ratio is $s = 5330$. Both numerical and finite-element simulations are used.

4.1. Mathematical Model

Consider a uniform flexible hanging beam, mounted on a horizontally sliding base. To derive the equation of motion for the lateral vibrations of the beam, the kinetic energy is assumed to be entirely due to translation. The governing partial differential equation of motion is [44-46].

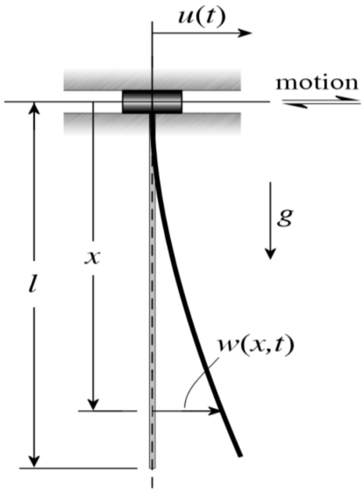

Figure 3. Hanging beam model.

$$EI\frac{\partial^4 w(x,t)}{\partial x^4}+m\frac{\partial^2 w(x,t)}{\partial t^2}-mg\frac{\partial}{\partial x}\left[(l-x)\frac{\partial w(x,t)}{\partial x}\right]=-m\frac{d^2u(t)}{dt^2} \tag{4}$$

where E is the Young's modulus of elasticity, I is the area moment of inertia of the beam's cross-section, m is the mass per unit length, l is the beam length, and g is the gravitational acceleration. The boundary conditions associated with the given beam support are [47]

$$\left.\begin{aligned} w(x,t)&=0\\ \frac{\partial w(x,t)}{\partial x}&=0\end{aligned}\right\} x=0 \tag{5}$$

$$\left.\begin{aligned} \frac{\partial^2 w(x,t)}{\partial x^2}&=0\\ \frac{\partial^3 w(x,t)}{\partial x^3}&=0\end{aligned}\right\} x=l \tag{6}$$

There is no closed-form solution to the differential equation of motion, Eqs. (4) – (6). However, the given system is a conservation self-adjoint system. We propose to derive an approximate solution in conjunction with Rayleigh-Ritz method. To this end, we assume a separable solution in x and t as

$$w(x,t)=\sum_{i=1}^{n}\phi_i(x)q_i(t)=\mathbf{\Phi}^T(x)\,\mathbf{q}(t) \tag{7}$$

in which $\mathbf{\Phi}(x)=[\phi_1(x),\ \phi_2(x),\ \dots,\ \phi_n(x)]$ is an n-vector of comparison functions of a complete set and $\mathbf{q}(t)=[q_1(t),\ q_2(t),\ \dots,\ q_n(t)]$ is an n-vector of generalized coordinates. We propose to use the eigenfunctions of a cantilever beam in free vibration as comparison functions in this analysis since they satisfy all the boundary conditions in Eqs. (5) and (6). Those eigenfunctions are

$$\phi_i(x)=\cosh(\beta_i x)-\cos(\beta_i x)-\frac{\cosh(\beta_i l)+\cos(\beta_i l)}{\sinh(\beta_i l)+\sin(\beta_i l)}[\sinh(\beta_i x)-\sin(\beta_i x)] \tag{8}$$

The values of β_i are determined by the characteristic equation

$$\cos(\beta_i l)\cosh(\beta_i l)=-1 \tag{9}$$

Substituting Eq. (7) into the equation of motion Eq. (4), pre-multiplying by $\mathbf{\Phi}(x)$, and integrating over the whole length of the beam, we obtain the spatially discretized equations of motion

$$M\ddot{\mathbf{q}}(t)+K\mathbf{q}(t)=\mathbf{f}\,\ddot{u}(t) \tag{10}$$

where

$$M=\int_0^l m\mathbf{\Phi}\mathbf{\Phi}^T\,dx$$
$$K=\int_0^l \mathbf{\Phi}L\mathbf{\Phi}^T\,dx \tag{11}$$
$$\mathbf{f}=-\int_0^l m\mathbf{\Phi}\,dx$$

are the mass and stiffness matrices, L is the *stiffness operator*, and **f** is the generalized force vector.

$$L=EI\frac{d^4}{dx^4}-mg\frac{d}{dx}\left[(l-x)\frac{d}{dx}\right] \tag{12}$$

The natural frequencies can be obtained by solving the eigenvalue problem

$$K\mathbf{p}_i=\lambda_i M\mathbf{p}_i \tag{13}$$

where $\lambda_i=\omega_i^2$ and $\mathbf{p}_i$ are the eigenvectors ($i=1,2,\dots,n$). The eigenvectors are orthogonal with respect to M and K and are normalized to satisfy

$$\mathbf{p}_j^T M\mathbf{p}_i=\delta_{ij}\,,\quad \mathbf{p}_j^T K\mathbf{p}_i=\lambda_i\delta_{ij}\,,\quad i,j=1,2,\dots,n \tag{14}$$

The discretized system of equations of motion Eq. (10) can further be decoupled. We consider a solution in the form

$$\mathbf{q}(t)=P\boldsymbol{\eta}(t) \tag{15}$$

in which $P=[\mathbf{p}_1,\mathbf{p}_2,\dots,\mathbf{p}_n,]$ is a eigenvectors matrix and $\boldsymbol{\eta}(t)$ is a vector of modal coordinates. Introducing Eq. (15), pre-multiplying by P^T and using the orthonormality relations, Eq. (14), we obtain the modal equation

$$\ddot{\boldsymbol{\eta}}(t)+\Lambda\boldsymbol{\eta}(t)=\mathbf{b}\ddot{u}(t) \tag{16}$$

where $\Lambda=\mathrm{diag}(\lambda_1,\lambda_2,\cdots,\lambda_n)$ and

$$\mathbf{b}=P^T\mathbf{f} \tag{17}$$

is the modal force vector. Equation (16) represents a system of n-number of independent equations of motion.

4.2. Frequency Modulation

The resonant frequencies of the model-based feedback system in Fig.2 are modulated to the point where all higher modes frequencies become odd-multiples of the first mode frequency as

$$\tilde{\omega}_i = r_i \omega_1 \qquad r_i = 2, 3, \dots, n \qquad r_i \in (2N+1) \tag{18}$$

Either one of the discretized models of the hanging beam in Eqs. (10) and (16) can be used as a model in the feedback system. However, The model in Eq. (10) requires a feedback of the generalized coordinates $\mathbf{q}(t)$. The response of any generalized coordinate may include components in all resonant frequencies of the hanging beam. Using such feedback may be undesirable due to its wideband of frequency content. Therefore, we will choose to work with the decoupled discretized model in Eq. (16) since this model gives us control over the frequency content in the feedback signal. Based on the hanging beam model in Eq. (16), a feedback law in the following form is used:

$$\ddot{u}(t) = \sum_{i=1}^{n} a_i \ddot{\eta}_i(t) = \boldsymbol{a}^T \ddot{\boldsymbol{\eta}}(t) \tag{19}$$

where $a = [a_1, a_2, \dots, a_{n-1}, 0]^T$ is the feedback gains vector. Note that the highest frequency mode is not included to minimize high frequency content in the feedback signal. Substituting Eq. (19) into Eq. (16) as

$$\ddot{\boldsymbol{\eta}}(t) + \Lambda \boldsymbol{\eta}(t) = \mathbf{b}\boldsymbol{a}^T \ddot{\boldsymbol{\eta}}(t) \tag{20}$$

the eigenvalue problem becomes

$$|\Lambda - \lambda(I - \mathbf{b}\boldsymbol{a}^T)| = 0 \tag{21}$$

where I is the $(n \times n)$ identity matrix. The modulated frequencies $\tilde{\omega}_i$ must satisfy the characteristic equation

$$(\lambda - \tilde{\omega}_1^2) \prod_{i=2}^{n} (\lambda - r_i^2 \tilde{\omega}_i^2) = 0 \tag{22}$$

where r_i are the odd-integer frequency ratios. Initial selection of the targeted frequency ratios are determined by rounding the exact frequency ratios of the hanging beam model to the nearest odd-integers as

$$r_i = 2 \text{ round}\left[\frac{1}{2}\left(\frac{\omega_i}{\omega_1} - 1\right)\right] + 1 \tag{23}$$

$i = 2, 3, \dots, n$ where $r_1 = 1$.

4.3. Numerical Simulations

To validate the performance of the FM input shaping strategy, simulations are performed using a 0.6 m maneuver with the maximum velocity set to 0.3 m/s and a maximum unshaped acceleration of 0.9 m/s^2. Time-Optimal Rigid-Body (TORB) acceleration command is used as a basic unshaped input command to the FM input shaper. Simulations are performed using both; ZV and ZVD primary input shapers.

Since the energy and mode participation of higher modes is minimal, a three-mode discretized model of the beam is used, which is a common practice in simulating transverse vibrations of beams. Given the beam properties described above, the inertia and stiffness matrices, M and K are

$$M = \begin{bmatrix} 0.1357 & 0 & 0 \\ 0 & 0.1357 & 0 \\ 0 & 0 & 0.1357 \end{bmatrix}$$

$$K = \begin{bmatrix} 3.510 & -0.5620 & -1.427 \\ -0.5620 & 67.26 & 2.515 \\ -1.427 & 2.515 & 470.3 \end{bmatrix}$$

The natural frequencies of the first three modes of the hanging beam, using Eq. (13) are

$$\begin{aligned} \omega_1 &= 5.079 \text{ rad/s} \\ \omega_2 &= 22.26 \text{ rad/s} \\ \omega_3 &= 58.87 \text{ rad/s} \end{aligned} \tag{24}$$

The associated modal matrix is

$$P = \begin{bmatrix} 1.000 & -0.008677 & -0.003064 \\ 0.008696 & 1.000 & 0.006244 \\ 0.003009 & -0.006271 & 1.000 \end{bmatrix} \tag{25}$$

According to Eq. (16), the three decoupled modal equations of motion of the beam are

$$\begin{aligned} \ddot{\boldsymbol{\eta}} + \begin{bmatrix} 25.80 & 0 & 0 \\ 0 & 495.6 & 0 \\ 0 & 0 & 3466 \end{bmatrix} \boldsymbol{\eta} \\ = \begin{bmatrix} -0.7875 \\ -0.4255 \\ -0.2546 \end{bmatrix} \ddot{u}(t) \end{aligned} \tag{26}$$

Based on the feedback control law of the frequency-modulation stage, Eq. (19), the feedback control law for the three-modes discrete model of the beam is

$$\ddot{u}(t) = a_1 \ddot{\eta}_1 + a_2 \ddot{\eta}_2 \tag{27}$$

Given the model frequencies Eq. (24), the frequency ratios are $\omega_2/\omega_1 = 4.383$ and $\omega_3/\omega_1 = 11.59$. According to the rounding scheme in Eq. (23), the target modulated frequency ratios are $r_2 = 5$ and $r_3 = 11$. Substituting the control law, Eq. (27), into the characteristic equation (21), and substituting the frequency ratios r_i into the characteristic equation of the modulated system Eq. (22), and solving both equations simultaneously, the first mode modulated frequency and the feedback gains are

$$\begin{aligned} \tilde{\omega}_1 &= 5.352 \text{ rad/s} \\ a_1 &= -0.1283 \\ a_2 &= -0.6479 \end{aligned} \tag{28}$$

Input shapers are designed for the first mode of the modulated feedback system $\tilde{\omega}_1$. Assuming undamped dynamic response, the ZV and ZVD input shapers, Eqs. (1) and (3) are

$$\text{ZV} = \begin{bmatrix} 0.5 & 0.5 \\ 0 & 0.5870 \end{bmatrix} \tag{29}$$

$$\text{ZVD} = \begin{bmatrix} 0.25 & 0.5 & 0.25 \\ 0 & 0.5870 & 1.174 \end{bmatrix} \tag{30}$$

Modal response of the discrete model in Figs. 4(b) and 5(b) demonstrate successful elimination of vibrations in all modes of the discrete model using both ZV and ZVD primary input shapers. However, due to the fact that the modulated second-mode frequency is five-times faster than the first-mode, high frequency content is included in the command signal of Eq. (27). This is reflected on the acceleration commands in the form of high command fluctuations in Figs. 4(a) and 5(a).

To overcome this problem, the modulated second mode frequency is brought closer to the modulated first mode frequency by setting the second-mode frequency ratio to $r_2 = 3$. The first mode modulated frequency and the control law gains become

$$\tilde{\omega}_1 = 5.352 \text{ rad/s} \quad a_1 = -0.1198 \quad a_2 = 1.940 \tag{31}$$

As expected, maintaining a fixed frequency ratio between the highest and the lowest modes of the system, in this case $r_3 = 11$, regardless of the intermediate frequency ratio, r_2, the modulated first mode frequency remains the same. Only the feedback gains of the control law change. Therefore, input shapers Eqs. (29) and (30) are applicable.

The frequency content and fluctuations in the acceleration commands in Figs. 6(a) and 7(a) dropdown significantly as a result of the lower frequency feedback command signal. To investigate further possible improvement in the command profile, the second mode frequency is set equal to the first mode frequency by setting the second-mode frequency ratio to $r_2 = 1$. The first mode modulated frequency and the control law gains become.

$$\tilde{\omega}_1 = 5.352 \text{ rad/s} \quad a_1 = -0.01323 \quad a_2 = 34.29 \tag{32}$$

Setting the second-mode frequency ratio to $r_2 = 1$ means that the feedback will have single frequency content, which is the lowest frequency of the modulated system. The effect of this frequency reduction is observed clearly in the much smoother acceleration commands in Figs. 8(a) and 9(a). However, this also means that the modulated system exhibits a double-root at the lowest frequency. The ZV input shaping technique fails in this case, Fig. 8. However, the ZVD input shaping technique continues to perform successfully due to its reduced sensitivity to double-roots in the system, Fig. 9.

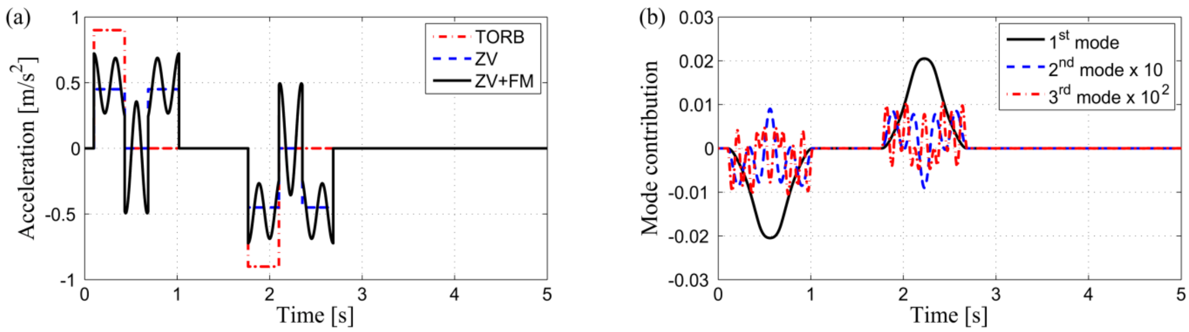

Figure 4. (a) Shaped acceleration and (b) modal response using a ZV primary shaper and a modulated second mode ratio of $r_2 = 5$.

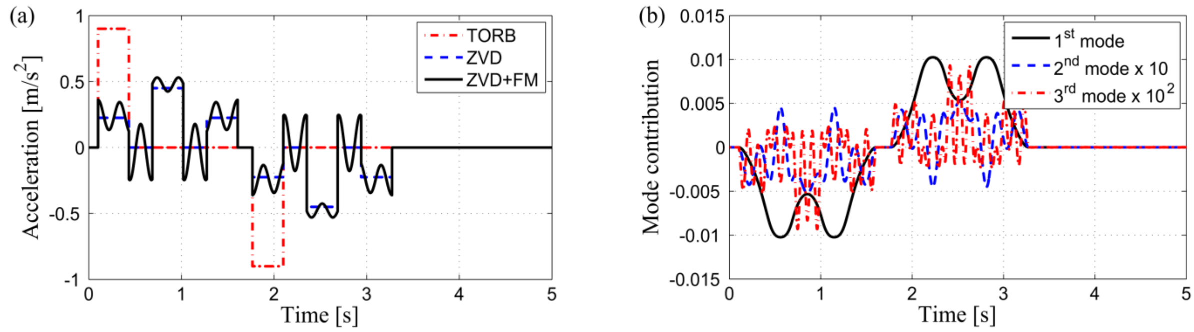

Figure 5. (a) Shaped acceleration and (b) modal response using a ZVD primary shaper and a modulated second mode ratio of $r_2 = 5$.

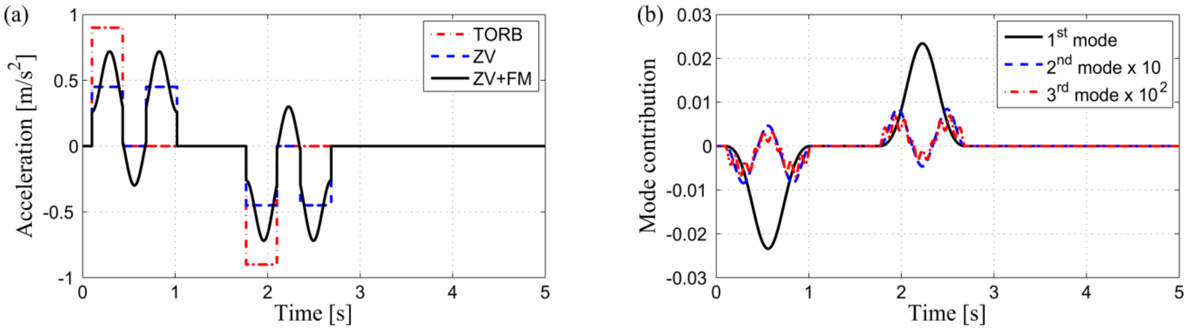

Figure 6. (a) Shaped acceleration and (b) modal response using a ZV primary shaper and a modulated second mode ratio of $r_2 = 3$.

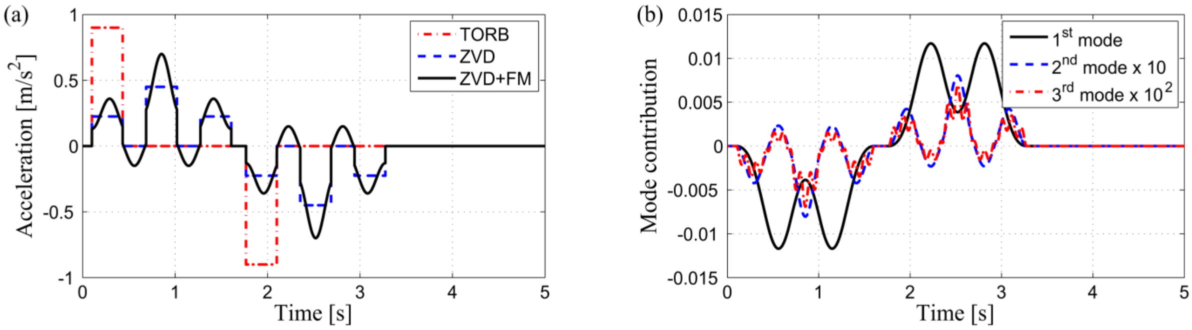

Figure 7. (a) Shaped acceleration and (b) modal response using a ZVD primary shaper and a modulated second mode ratio of $r_2 = 3$.

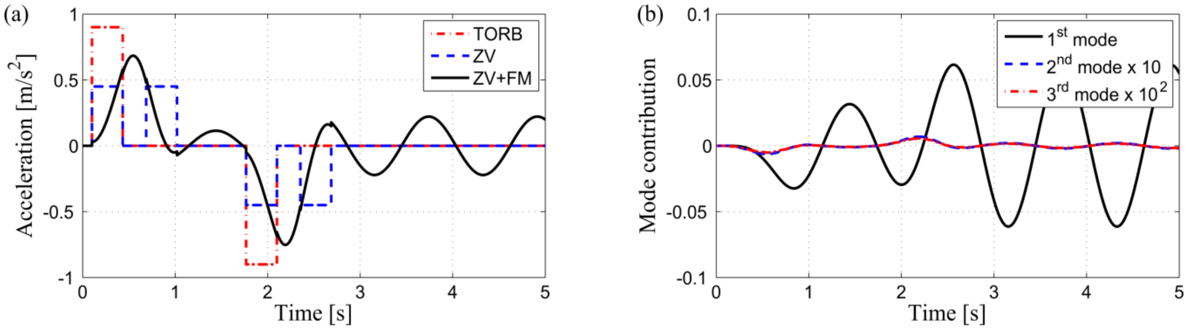

Figure 8. (a) Shaped acceleration and (b) modal response using a ZV primary shaper and a modulated second mode ratio of $r_2 = 1$

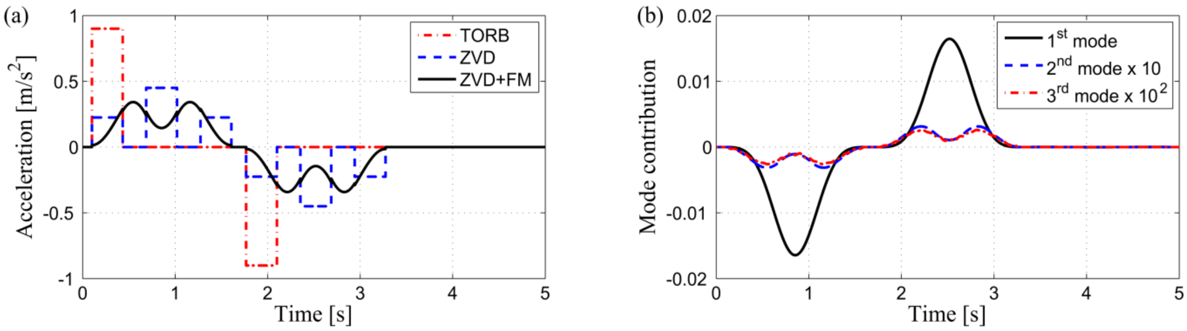

Figure 9. (a) Shaped acceleration and (b) modal response using a ZVD primary shaper and a modulated second mode ratio of $r_2 = 1$.

4.4. Finite Element Simulations

The finite element analysis simulations are performed using the commercial FEA software Abaqus, version 6.12. The analysis consists of two steps. In the first step, a linear perturbation procedure is performed to calculate the natural frequencies and corresponding mode shapes. In the second step, a transient modal dynamic linear perturbation analysis that utilizes modal superposition is conducted. Abaqus/Standard implicit integration technique is used in this work. The reason for choosing Abaqus/Standard instead of Abaqus/Explicit is due to its ability to model low speed dynamic events with high accuracy at reasonable computational cost. On the other hand, Abaqus/Explicit is particularly suitable for high speed dynamic events and applications where severe contact exists such as crash tests.

Two node planar beam elements that use linear interpolation with lumped mass formulation are utilized. The first three modes obtained are in Fig. 10. These elements use Timoshenko beam theory that allows for transverse shear deformation. A mesh convergence study is performed to ensure accurate results with a mesh that is sufficiently dense yet not overly computational expensive. As a result, the beam is modeled using 200 two node planar beam elements.

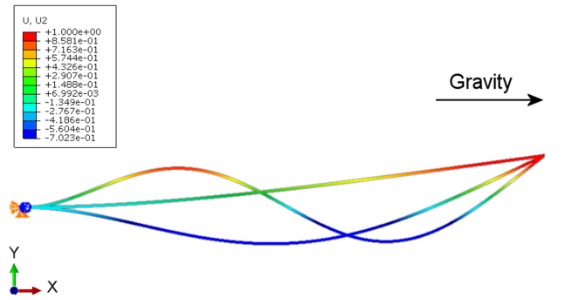

Figure 10. First three modes of the finite element model of a hanging beam

The default eigenvalue extraction method used in Abaqus is the Lanczos method. In this method, a set of Lanczos runs are performed. Each run consists of a number of iterations called steps. In each run, the spectral transformation, which allows rapid convergence to the desired eigenvalues, is applied. Further details about the Lanczos algorithm are available in [48].

The first three frequencies of the FEA model are $\omega_1 = 5.078\ \mathrm{rad/s}$, $\omega_2 = 22.25\ \mathrm{rad/s}$, and $\omega_3 = 58.85\ \mathrm{rad/s}$. These values are in excellent agreement with those obtained using the discretized model Eq. (13).

The simulation cases, performed using the discrete model of the hanging beam, are repeated using the finite element model of the beam. The beam-tip deflection in all cases is shown in Fig. 11. Results in Fig. 11 demonstrate excellent match between the discrete model and the finite element model simulations. The fact that the FM input shaper designed using a three-mode linear approximation of an Euler-Bernoulli beam retains its successful performance on a finite element Timoshenko model of the hanging beam demonstrates a strong robustness of the FM input shaping technique to modeling uncertainties.

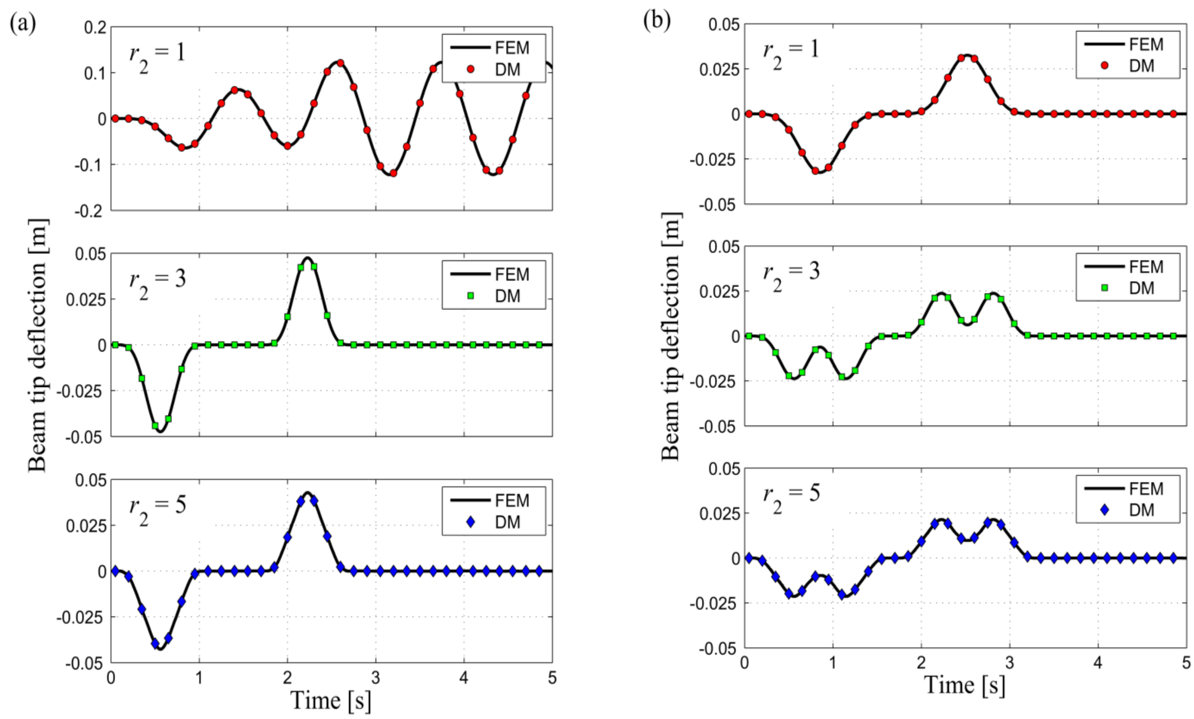

Figure 11. (a) Beam tip deflection of the finite element model (FEM) and the discrete model (DM) using frequency-modulation with (a) a primary ZV shaper and (b) a primary ZVD shaper for different modulated second mode frequency ratios.

5. Discussion and Conclusions

Multimode frequency-modulation input shaping is a control strategy developed to overcome one major drawback of multimode input shaping techniques. Multimode convolved input shaping techniques are derived from the convolution of several single-mode input shapers, each targeting a specific frequency. Convolved input shapers intended for high frequencies tend to be impractical due to the high rate of impulses required and the bandwidth limitations on systems' actuators. Inherent time delays in most multimode systems further compromise the performance of those high-frequency input shapers. Simultaneous multimode input shapers tend to be slower, and require high number of shaping impulses depending on the order of the system. These impulses result in practically undesirable jerky input commands.

Multimode FM input shaping strategy uses only one single-mode input shaper designed for the slower first mode frequency of the system. Using a single mode input shaper minimizes the number of impulses involved in the shaping process. Frequency-modulation facilitates eliminating all modes of vibrations of a multimode system simultaneously.

In this frequency-modulation strategy, the selection of the design set of frequency ratios that satisfy the odd-multiple frequencies condition is flexible. The ratios can be selected to meet certain design performance. In the case of the flexible manipulator presented, the design requirement was to reduce shaped input fluctuations, which is a desirable feature in most systems since smoother commands are easier to produce and follow. Smooth commands with minimum fluctuations can be achieved by imposing a double-root condition at the lowest frequency of the modulated system. However, it is imperative that a robust primary input shaping technique is used in conjunction with frequency-modulation.

It is important here to emphasize that the multimode FM input shaping may be implemented using different primary input shapers, and is not limited to the ZV and ZVD input shaping techniques. The proposed strategy is not intended for enhancing performance of input shaping techniques, rather it is intended to reduce the number of input shapers required to eliminate all modes of vibration of the system.

Simulation results show that the performance of the proposed strategy is as effective and as stable as the primary input shaping technique used in conjunction with the FM stage. Results show that the proposed multimode FM input shaping is not vulnerable to modeling uncertainties and omitted nonlinearities. This can be concluded by the excellent match between the simulation results using a linear Euler-Bernoulli model and the results obtained using a Timoshenko finite element model of a hanging beam.

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