中國機械工程學刊第四十二卷第一期第 1~10 頁(民國一百一十年)
Journal of the Chinese Society of Mechanical Engineers, Vol.42, No.1, pp 1~10 (2021)

Finite Element Based Input Shaping Design for Suppressing Motion-Induced Vibrations

Weiju Chen* and Kuo-Shen Chen **

Keywords: Vibration suppression, Input shaping, Finite element simulation, Dynamics

ABSTRACT

Input shaping is an effective method for suppressing motion-induced vibration to enhance the motion performance in point-to-point maneuverers. However, the success of shaper design depends on the accuracy of system dynamics but such an approach could not yield effective dynamic models with sufficient accuracy for systems with complex boundary conditions, motion constraints, and structural behaviors. In this work, it is proposed to hire finite element dynamic simulation directly in both trajectory planning and input shaping design. Two flexible motion systems are designed for serving as the test beds to ensure multiple mode excitations during transportation for evaluating the effectiveness in finite element simulation. The results confirm the effectiveness of using input shaping in vibration suppression and both the experimental and the simulation results agree to each other very well. Several case studies using finite element simulation are then followed for elucidating the possible future applications of the proposed approach in real engineering designs. It is believed that the proposed shaper design approach by hiring finite element simulation can handle many ad hoc issues such as nonholonomic constraints, time-varying and nonlinear structural dynamics, and possible power and bandwidth limitation in actuators.

INTRODUCTION

Point to point motion systems have been widely used in factory and cargo ship transportation applications [Abdel-Rahman et al, 2003, Singhose et al, 1996, Wang et al, 2014]. For those purposes, it is desired that the transportation process being as rapid as possible. These kinds of problems can usually be modelled as a gantry crane with hoisting cables and an end payload worked as a pendulum in linear translation and their effectiveness is often limited by both the transient swing and the residual oscillation of the payloads after reaching the destinations. Furthermore, any excessing swings and oscillations could also be dangerous to people working nearby. Therefore, devising a method to rapidly transport the payloads to follow a desired trajectory with minimum swings and residual vibrations after reaching the destination is a non-trivial task. Feedback control is traditionally used for solving this kind of problems [Lin et al, 2016]. However, such an approach would need the incorporation and integration of sensors, actuators, and controller. This implies a costly solution. In addition, it may have no suitable space for installing appropriate actuators. One possibility for solving the above-mentioned problem is to utilize command shaping methods for performing an efficient trajectory planning to minimize swing and suppress motion-induced residual vibrations.

For designing the corresponding command shapers, the system dynamics and responses must be modelled. Previously, the problems are usually treated as a pendulum consisted with rigid links mounted on a trolley with controlled movements [5] and only the rigid body modes such as swing are considered. As shown in Figure 1, cranes can be modelled either as single or double pendulums. By utilizing time optimization schemes, it is possible to find commands for trajectory optimization. Such an open-loop based input shaping scheme has been demonstrated to be effective for suppressing motion induced vibrations of flexible structures and swing of long range motion structures such as cranes [Singhose et al, 2008]. By input shaping, this problem has been successfully solved in many applications [Sun et al, 2012, Zhang et al, 2016, Ou et al, 2009, Ou and Chen, 2016, Daqaq et al, 2008].

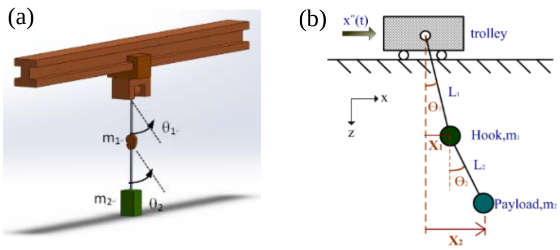

Fig. 1(a) Schematic plot of a gantry crane with

Paper Received April, 2020, Revised May, 2020 Accepted May, 2020, Author for Correspondence: Kuo-Shen Chen

** Graduate Student, Department of Mechanical Engineering, National Cheng-Kung University, Tainan, Taiwan, 70101, R.O.C.*

*** Professor, Department of Mechanical Engineering, National Cheng-Kung University, Tainan, Taiwan, 70101, R.O.C.*

payload and (b) its corresponding model.

Since the success of input shaping relies on accurate dynamic modelling, in many applications, modelling based on simple analytical rigid body dynamics may not be sufficient. For example, the compliance of the structures was not considered in those studies [Singhose et al, 2008, Sun et al, 2012, Zhang et al, 2016] and the associated structural vibrations cannot be estimated. This is not trivial since the structural vibration could also be excited due to the compliance existed in cable or structures. Meanwhile, there are also two major problems faced when employing analytical modelling. First, the problem (i.e., pendulum swing) is essentially nonlinear. To be solvable analytically, the system dynamics requires extensive linearization, which could lead to the loss of important information and cause the designed shapers to have inferior performance. Second, for realistic systems such as rolling contact systems, the nonholonomic nature makes the Lagrangian-based analytical modelling less effective. Consequently, it could be more difficult to solve the equations and the shaper design becomes more challenging. In addition, it is expected that once as the structure flexibility is considered, the dynamic equations would become more difficult to obtain and solve. From a practical perspective, the analytical modelling loses its advantages thus a numerical scheme is usually employed. To handle such systems, one may prefer to use a finite element (FE) dynamic analysis, which can handle both rigid body and extremely complicated structural modes with sufficient accuracy. It is also possible to characterize the necessary parameters for designing the associate input shapers such as natural frequencies and mode shapes via FE simulation directly. Furthermore, by adapting the viewpoint of force balance and energy conservation, it should be able to expand the traditional linear superposition based shaper design into nonlinear systems [Chen et al, 2006]. These are the major advantages of using FE for conducting input shaping design and this may eventually lead to a more efficient command shaper design and verification. Finally, with the aids of FE, other design concerns such as stress analysis may also be evaluated simultaneously.

Previously work [Dai and Chen, 2013] investigated the swing control of cranes using input shaping method. The crane hoist was either modelled as a travelled space pendulum analytically or by a FE beam model for evaluating the influence of the cable compliance. The experimental results and the associated FE analyses agreed with each other well. This study indicated that it is possible to use FE dynamic analyses for designing or optimizing the input shaper design. In this work, more general scenarios in crane transportation are considered. Namely, for suppressing both the bending and torsional vibration of flexible cable structures when they undergo high speed movement. The investigation flow is schematically shown in Figure 2. A simplified slender elastic structure is constructed for mimicking the behaviour of a long compliant robot arm in high speed operation. The associated FE simulations would then be performed to analyse the dynamic characteristics of the model for subsequent input shaping design. Finally, with the input shapers ready, the dynamic behaviour of the entire shaped system can then be simulated by FE method again. By comparing the simulation and the experimental results, the effectiveness of the FE-based input shaping design is examined first. After correlated with experimental data, the FE based input shaping design approach would then be used in systems with more complicated structural dynamics.

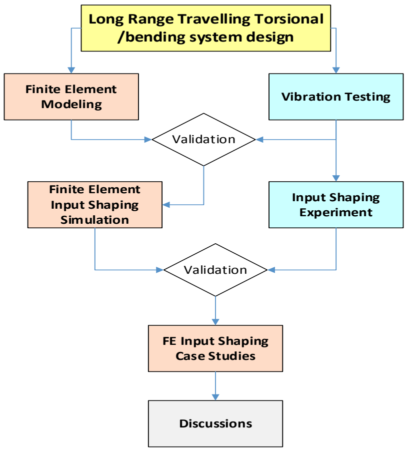

Fig. 2 The schematic flow of this work

Based on our previous conference presentation [Chen et al, 2019] with adding on more detail technical development and demonstrations, the contribution of this work is therefore mainly on proposing and validating the idea of using FE simulation in input shaper design and the subsequent dynamic simulation. Although the concept is straightforward, to the best of our knowledge, no previous works focused on numerical simulation in input shaper design. It is hope that the input shaper design can be more versatile with the FE dynamic analysis being incorporated through the effort of this work.

INPUT SHAPING SCHEMES

Input shaping has been proposed for many years [Chen et al, 2019, Singer and Seering, 1990, Yi et al, 2016, Shan et al, 2005]. It bases on the principle of linear superposition of responses caused by different inputs with similar form and proper time delays. Under certain circumstance, these responses would cancel each other to form a desired output. Various input shapers such as zero-vibration (ZV) and zero-vibration-and derivative (ZVD) [Singer and Seering, 1990], and many others, have been developed. In comparison with active controls, the input shaping schemes are cheaper and more flexible. In addition, it is also possible to integrate both input shaping and active control design for achieving a better performance [Chen et al, 2006]. However, due to its open loop nature, the success of input shaping relies on accurate dynamic modelling and the robustness itself.

Based on the same principle, input shapers for dealing with multiple DOF systems can also be developed. For example, for a 2-DoF system with two natural periods T_1 and T_2, the ZV shapers can be designed by using two impulse pairs with a time separation of $T_2/2$ between the two pairs while a time delay of $T_1/2$ is between two impulses. By convolving the original commands with this impulse train, vibration caused by the two vibration modes can be eliminated. By the same approach, the ZVD or other versions of multiple degrees of freedom vibration suppression can be achieved.

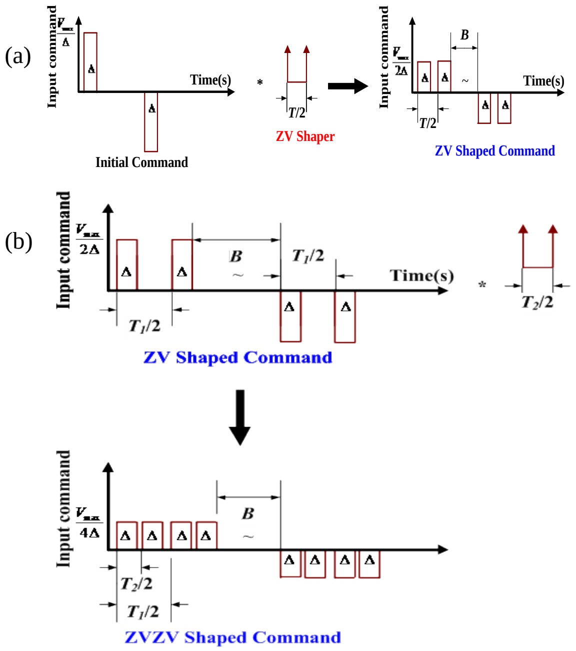

Fig. 3 Schematic plots of (a) ZV and (b) ZV-ZV shaping schemes in long range point to point maneuver

Finally, for long range transportation, ideally, bang-bang motions are usually planned for minimizing the travelling duration of rigid bodies. That is, the system undergoes the maximum acceleration for a period to attain the maximum speed and then use a maximum deceleration to stop it as the system reaches the destination in a time-optimal manner for rigid body translation. However, for flexible structures, although the traveling is the fastest, the motion-induced vibration would significantly increase the time for settling. By modifying the original bang-bang command with input shaper such as ZV, ZVD, and Half Zero Vibration (HZV) [Chen, 2011], the vibration and swing can be effectively suppressed under the new commands shown in Figure 3a (use ZV shaper for illustration). Furthermore, the motion command can be convolved the original bang-bang command with two impulse trains and methods such as ZVZV, ZVZVD, and Half Double Zero Vibration (HDZV) methods [Chen, 2011] for suppressing two vibration modes shown in Figure 3b (for ZV-ZV shapers). The waveforms of other shapers can be found from elsewhere [Chen, 2011].

However, the success of input shaping relies on accurate system dynamic modelling and the robustness of shapers themselves. This may represent a challenge if the structure is too complex or containing significant compliance, or if the system has non-holonomic constraints [Baruh, 1999]. On the other hand, by employing FE dynamic simulation, it is possible to obtain the modal stiffness and natural frequencies accurately. Presently, almost all input shaping design are based on analytical lumped model, which may deviate from the accurate dynamic behavior and could reduce the effectiveness of the performance of shapers. Therefore, properly use of FE simulation should be able to obtain more accurate dynamic parameters and thus a better shaping performance.

Figure 4 shows the proposed procedure of using FE simulation input shaper design and validation. First, the FE model should be constructed based on physical systems with certain reduction. A modal analysis should be performed for obtaining the natural frequencies and their associated vibration modes. The obtained information, in together with the required moving task planning (for example, moving with a straight path with a certain velocity), and the desired input shaping scheme (e.g., ZV-ZV) then generate a velocity boundary condition. In together with associated displacement boundary condition, a FE linear dynamics analysis is then performed. The analysis results would provide the motion history of all nodes of the machine model for subsequent judgement on the vibration and stress state of the system. Furthermore, additional design analyses such as stress and fatigue can be performed simultaneously.

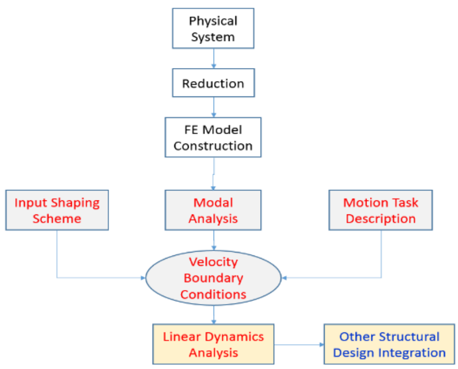

Fig. 4 The proposed input shaping design based

on finite element simulation

It is also interesting to compare the vibration suppression using both input shaping and feedback control. A system, shown in Figure 5, built by us previously [Lin et al, 2016] is used to conduct this comparison. The system consists of a double pendulum set mounted on a linear servo motor in which the dynamics is analogous to the above-mentioned crane transportation. By this design, the system has two swing and a rigid body motion degrees of freedoms. Laser displacement sensors were used for monitoring the vibration. Additional non-contact magnetic actuators and controllers were also designed for evaluating the performance of using feedback control. The detail design and realization can be found elsewhere [Lin et al, 2016].

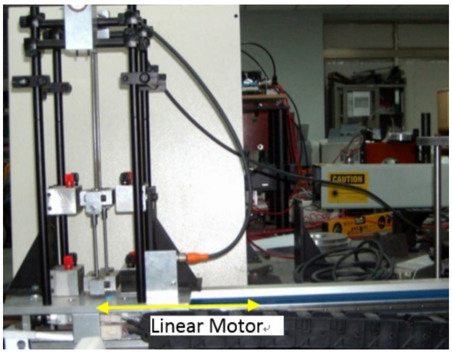

Fig. 5 A double pendulum-liked feedback control system moved along a linear servo motor

(a)

(b)

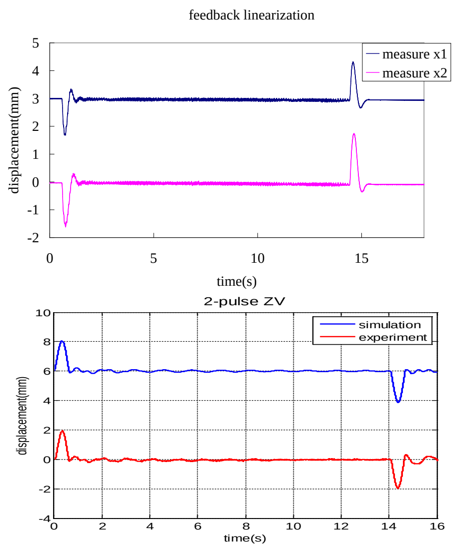

Fig. 6 Experimental results of the double pendulum during a long range transportation (a) with feedback linearization control and (b) with 2-pulse ZV shaping command.

Several feedback control schemes such as SDOF loop transmission design (aims to eliminate vibration of the bottom mass only) and 2-DOF full-state feedback (with state observer), as well as design with feedback linearization scheme, are conducted to examining the swing level and vibration suppression after a point-to-point maneuver. Figure 6a shows a typical control result of both upper and bottom masses using the nonlinear feedback linearization scheme [Chen et al, 2002]. The experimental results of other schemes have also been shown in our previous work [Lin et al, 2016]. Input shaping schemes are then applied in the same test system. Several types of input shapers, such as 2-pulse ZV, 3-pulse ZV [Shan et al, 2005], ZVD, and the 2x3 MISZVD [Shan et al, 2005], are realized to compare their performance in suppressing residual vibrations. The simulation and experimental data for both input shaping and feedback control approaches achieve similar results. As a result, input shaping scheme could be a preferred solution for cost-effective design. One approach to improve the performance and reduce uncertainty is through better dynamic modeling, in which FE simulation could play real contributions. To study this aspect, two test systems are designed for conducting both experimental and numerical input shaping studies and are addressed below.

EXPERIMENTAL SYSTEMS SETUP

The first issue in this research is to establish a test article for performing both experimental characterization and FE dynamic simulation. Such a system should have two close major vibration modes excited subjected to a long-range transportation. There are two systems designed for conducting this work. The first system, called as the *double pendulum system*, shown in Figure 7, is essentially a double pendulum made of flexible cables driven by a two-axis Yokogawa linear servomotor. The length and mass of each section are 0.5m and 0.43 Kg, respectively. The goal is to rapidly transport the system to the destination under a desired maze path without exciting exceed swing vibrations. By both structural test and FE analysis, the natural frequencies are 0.53 and 1.40 Hz for the first and the second modes, respectively.

The simplest mathematical model for the system can be treated as a double pendulum with rigid links shown in Figure 1b and the dynamics can be expressed as

$$\begin{aligned}&(m_1+m_2)l_1\ddot{\theta}_1+(m_1+m_2)g\theta_1+(m_1+m_2)\ddot{x}+m_2l_2\ddot{\theta}_2=0\\&m_2\left[l_2\ddot{\theta}_2+l_1\ddot{\theta}_1+\ddot{x}\right]+m_2g\theta_2=0\end{aligned}\qquad(1)$$

where $\ddot{x}$ is the controllable acceleration, which is assumed to be known by a prescribe displacement input. However, this model does not consider the distributed mass and the compliance of the links. Practically, it also has difficulty to monitor the vibration using non-contact displacement sensors, instead, two ADXRS614 MEMS gyroscopes are hired for monitoring the swing. A Yokogawa two-axis linear servomotor is used for driving the structure and associated vibration data collected by the gyroscopes are then processed and analysed by another computer. The motor is commanded to perform a long-range motion (e.g., 80cm) with speeds typically near 5-20 cm/s. Various input shaping schemes are used for evaluating the performance on vibration suppression. These test data are then used to compare with the results obtained from the associated FE simulation. After calibration, the FE model is then used for predicting the vibration under more general trajectories and other input shaper.

On the other hand, the second system, called as *the bending-torsion system* thereafter, is mainly for suppressing bending-torsion vibration during transportation. By mimicking a two-cantilever-arm type structure commonly existed in robots or machine tools (schematically shown in Figure 9a), a two-flexible beams under an eccentric configuration type structure is designed and is schematically shown in Figure 9b. Such a system exhibits both bending and torsional vibration during transportation and it is desired to suppress both the motion-induced bending and torsional vibrations.

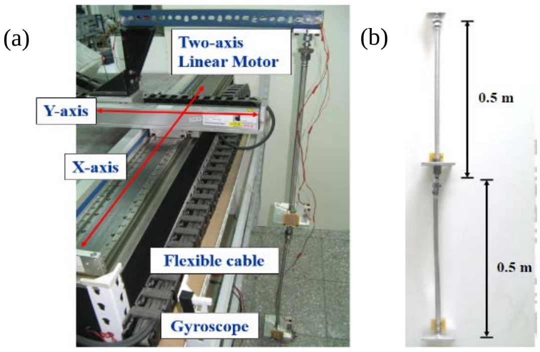

Fig. 7. The flexible double pendulm experimental system (a) the overall system setup and (b) the double pendulum

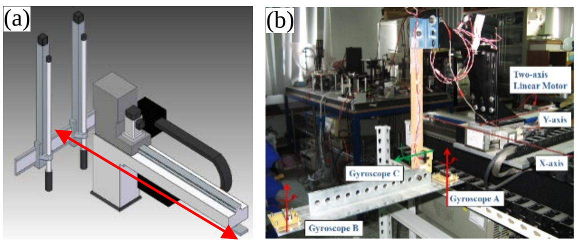

Fig. 9 (a) a high speed transverse robot for object transportation and (b) the corresponding bending-torsion experimental system.

For each beam segment, the bending vibration, *w(x,t)*, can be represented by the following partial differential equation,

$$\frac{\partial^2}{\partial x^2}\left[EI(x)\frac{\partial^2 w(x,t)}{\partial x^2}\right]+\rho A(x)\frac{\partial^2 w(x,t)}{\partial t^2}=f(x,t), \quad (2)$$

where E, $I(x)$, ρ, $A(x)$ are the Young's modulus, spatial dependent inertia, density, and spatial dependent cross-sectional area, respectively. $f(x,t)$ is the acted body force.

Meanwhile, the torsional vibration, $\theta(x,t)$, can be expressed as

$$GJ(x)\frac{\partial^2\theta(x,t)}{\partial x^2}+g(x,t)=I_0(x)\frac{\partial^2\theta}{\partial t^2}(x,t)\,. \quad (3)$$

Where G, $J(x)$, and $I_0(x)$ are the shear modulus, polar moment of inertia, and mass moment of inertia, respectively. $g(x,t)$ is the distributed torque acting on the structure. Eqs.(2) and (3) represents the simplest situation. In this test structure, the system is bending – torsion coupled and the structure geometry is too complex to be handled by these equations for yielding an effective analytical model. Practically, this system can be treated as a transverse robot or a machining tool with cantilevered arms as schematically shown in Figure 9a. By both structural test and FE modal analysis, the natural frequencies are 0.67 and 1.24 Hz for the torsional and the bending modes, respectively. In this case, three ADXRS614 gyros are hired for monitoring both vibration modes. The responses of gyroscopes A and B reveal that both the torsion and bending modes are excited and the output of gyroscope C mainly represents the bending mode.

EXPERIMENTAL RESULTS

The double-pendulum system

The linear motor is accelerated to 5cm/s with a constant acceleration of 1m/s^2 and an acceleration period of 50 ms. The travel time and distance is 6s and 30 cm. The double pendulum exhibits a swing with an overshoot/undershoot and a swing amplitude of ±20°/s and ±5°/s, respectively. After being stopped, the system continuous to swing with an amplitude of ±5°/s. The results are shown in Figure 10 for both the upper and the bottom masses.

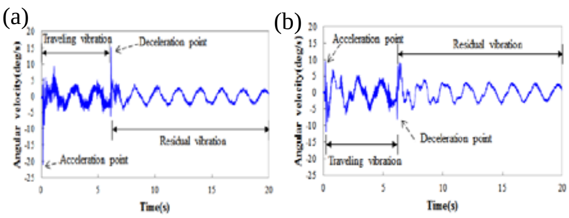

Fig. 10 Unshaped result (a) upper mass and (b) bottom mass

Meanwhile, with input shaping commands (i.e., ZVZV, HDZV, and ZVZVD), the results are different and are shown in Figure 11 for both the upper and the bottom masses. It can be seen that the travelling time is a little bit longer (6.5-7 seconds instead of the original 6 s). During traveling, there are ± 11°/s overshoot/undershoot and a ±0.25°/s swing amplitudes, respectively. After reaching the destination, the residual vibration is approximately ±0.05°/s. The vibration suppression level of each input shaping scheme are similar since the parametric uncertainties are not considered in this moment. Finally, the ZVZVD scheme results a slightly longer travelling time because it requires two extra half periods to finish the scheme.

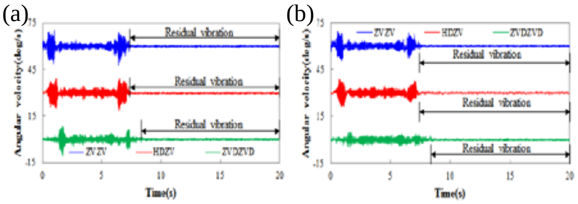

Fig. 11 Shaped results (a) upper mass and (b) bottom mass

The bending-torsion system

Based on modal analysis, input shaping schemes for the bending-torsion system are also implemented. The unshaped response using bang-bang command is shown in Figure 12 and the vibration level is severe under a simple bang-bang command and both the bending and torsional modes are excited. It takes over 20 seconds to settle from the motion-induced vibration. Such a multiple-mode behaviour is as expected in the platform design. Various input shaping schemes are designed but only ZVZV and HDZV are shown. Both methods can effectively suppress the motion induced vibrations. For example, as shown in Figure 13, the vibrational angular velocities observed from these gyroscopes are reduced from ± 100°/s to approximately ±2°/s. This result is not surprising but just to re-emphasizing the effectiveness of input shaping. Nevertheless, with the feasibility being validated, these responses then serve as the experimental data for validating the FE-based input shaping design. After this approach being validated, vibration suppression design for more complicated cases using FE analysis would then be followed.

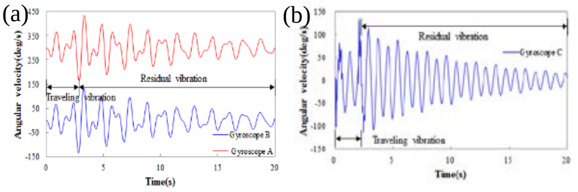

Fig. 12 The unshaped results from gyroscopes (a) A and B and (b) C

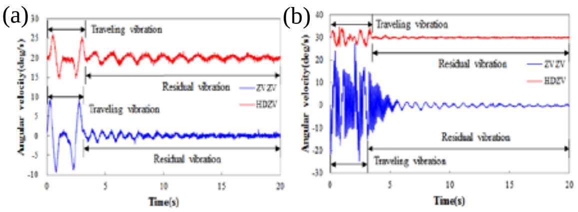

Fig. 13 The shaped results from gyroscopes (a) A and B and (b) C

FINITE ELEMENT VALIDATION

The corresponding FE dynamic simulation of the experimental structure are coded and performed by using Simulia Abaqus [Abaqus, 2009]. The model is constructed using S8R quadratic shell elements for cost-effective consideration. The experimental design shown in Sections III and IV is used for validating the FE model. Once the method is validated, the input shaping vibration reduction for other practical and complicate cases are then designed based on FE simulation.

The double-pendulum system

Figure 14 shows the schematic FE model. It consists of 80 B32 quadratic beam elements and two mass elements. The first two vibration modes and natural frequencies, 0.54 Hz and 1.31 Hz are very close to the experimental results (i.e., 0.53 Hz and 1.41 Hz). The unshaped simulation results are shown in Figure 15 and the simulation results agree with experimental data quite well. Notice that the experimental data are noisier due to other effects such as bearing friction or structural compliance, which are not modelled. Nevertheless, the major vibration characteristics are caught by the model to ensure the further applications in different loading situations. By input the loading sequence in the FE model according to the specific input shaping design, the simulation results of shaped response are obtained and plotted in Figure 16. It can be seen that the simulation data is cleaner due to lack of noise and unmodelled dynamics. Nevertheless, the major characteristics, such as the maximum overshoot/ undershoot, swing level, destination time, and residual vibration level, highly agree with that observed in experiment.

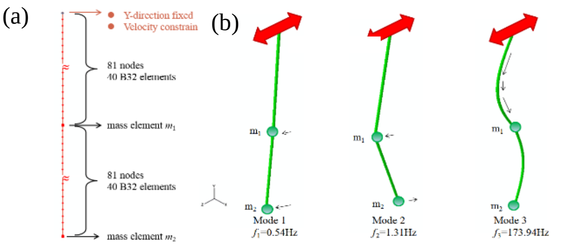

Fig. 14 (a) The FE model for the double pendulum system and (b) the associated mode shapes

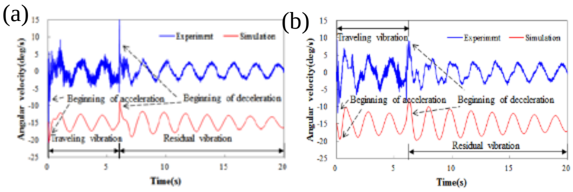

Fig. 15 The comparison between the unshaped experimental and simulation results (a) upper mass and (b) bottom mass

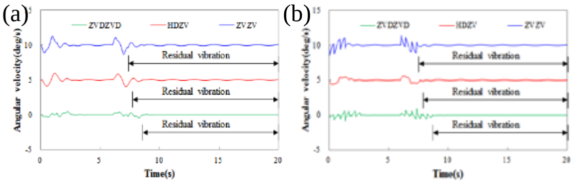

Fig. 16 FE simulation results using various shaping schemes (a) the upper mass and (b) the bottom mass.

The bending-torsion system

Figure 17 shows the simulated first 3 mode shapes of the bending-torsion structure designed. Where Modes 1 (0.7 Hz) and 2 (1.2 Hz) corresponding to the observed torsional and bending modes. The third mode is much higher (9.33 Hz) and this numerical system can essentially be treated as a 2-DoF system.

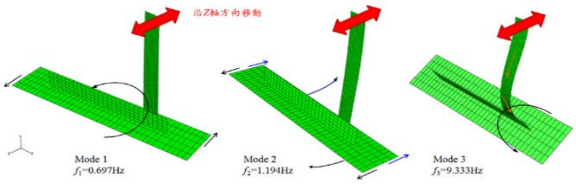

Fig. 17. The FE model for the double pendulum system and the first 3 mode shapes

Next, the same band-bang command is applied in the FE model and the simulated angular velocities at the locations of gyroscopes A, B, and C are plotted and shown in Figure 18a and b. It can also be seen that the FE simulation results agree with the experimental data very well. Both the vibration frequencies (0.7 Hz and 1.2 Hz) are excited in the model and the vibration amplitudes highly agree with those observed in experiments. Finally, both ZVZV and HDZV shaped inputs are applied in the model and the simulation results are then shown in Figure 18c and d. Again, it can be seen that the input shaping scheme reduce the vibration significantly and the results from the FE model agree with the major characteristics such as vibration amplitude and frequencies (0.7 Hz and 1.2 Hz). The minor difference is that the outputs of the FE model are cleaner since the possible sensor noises are not included in simulation. Meanwhile, the robustness of each input shaping schemes against possible parameter uncertainties are also investigated by constructing the sensitivity curves. Both the experimental and FE simulated results are shown in Figure 19 for ZVZV and HDZV shapers. In x-axis of Figure 19, ω_a and ω_n are is the estimated and the actual 1st natural frequencies, respectively and the variable ω_a/ω_n is a measure on the deviation in system dynamics due to errors in parameters identification. The results indicate that the residual vibration is minimum if no modeling error (i.e., ω_a/ω_n =1). With modeling error, the residual vibration amplitude would be increase. On the other hand, the HDZV scheme is less sensitive to modeling error. Finally, it can be seen that the FE simulated results agree with experimental data very well and the results fully validate the feasibility of FE dynamic analysis in input shaping design. Therefore, with the FE model being validated, it is then possible to use this approach to explore design in a more complicate senses.

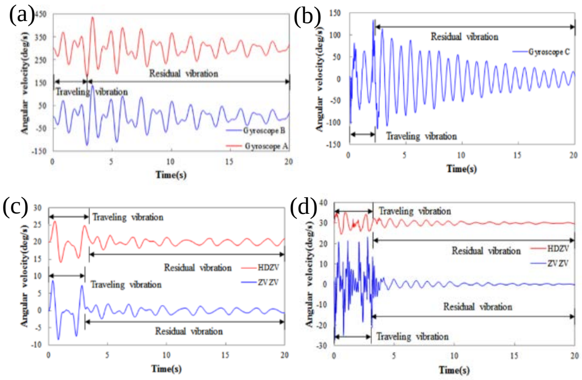

Fig. 18. FE simulation results on the unshaped behavior of gyroscopes (a) A and B and (b) C and the shaped behavior of gyroscopes (c) A/B and (d) C

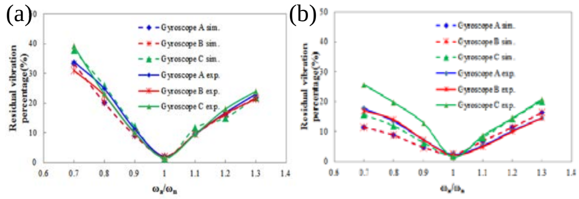

Fig. 19 The sensitivity study for evaluating the robustness of input shapers in the bending-torsion system: (a) ZV-ZV and (b) HDZV schemes.

FE CASE STUDIES

Two case studies in vibration suppressing for long range transportations are presented here using FE simulation for elucidating the proposed approach in input shaping design.

First, consider the loading/unloading of containers in cargo ship, schematically shown in Figure 20(a), the container is initially raised, then transmitted to above the ship, and finally moves down and released. Fast container transportation without exciting swing is an important issue and has attracted numerous investigations using both input shaping or control approaches [Sano et al, 2013, Hong et al, 2003, Yashida, 2002]. It is possible to design input shaper via FE simulation. Notice that the initial raising-up and finally move-down are not modelled at this moment since the continuously changed cable length induced time-varying dynamics actually causes additional challenges. Up to now, to the best of our knowledge, there are no efficient analytical procedures available and the corresponding shaping design relies on extensive trials [Chang and Park, 2005].

Nevertheless, this can potentially be performed via FE simulation. However, in this work, only the transmission phase is simulated.

A 2500 kg imbalanced container with size (6.06 x 2.44 x 2.59 m^3) is supported by four 8-m long steel cables with a cross-sectional area of 150 mm^2. The first two modes exhibit swing and torsional motions with natural frequencies of 0.175 and 0.386 Hz, respectively shown in Figure 20(b). Several input shapers are designed in FE and the results are shown in Figure 20(c)(d). It can be seen that the unshaped response exhibits a large amount of swing motion and it would take enormous duration for settle. On the other hand, with shaped input, the container can reach the destination with a much less settling time. From Figure 20(c)(d), it can be seen that the ZVDZVD method has the longest rising time since it requires more actions.

Second, as schematically shown in Figure 21, gantry cranes have been widely used in many factory transportation applications. For those purposes, it is desired that the process is as rapid as possible for working efficiency considerations, which is often limited by both the transient swing and the residual oscillation of the payloads after reaching their destinations. The system shown in Figure 21(a) is modelled as a 2-DOF steel cable system with a 500 Kg hook and an 810 Kg payload. The two natural frequencies are found as 0.24 and 0.68 Hz using FE analysis (Figure 21(b)). With the information ready, different shaping schemes such as ZVZV, HDZV, and ZVDZVD are then applied in this dynamic model for transportation with obstacle avoidance. As shown in Figure 21(c)(d), the unshaped trajectory could excite significant swing motion. On the other hand, the shaped commands could achieve a much smooth transition, faster settling, and safer for people nearby.

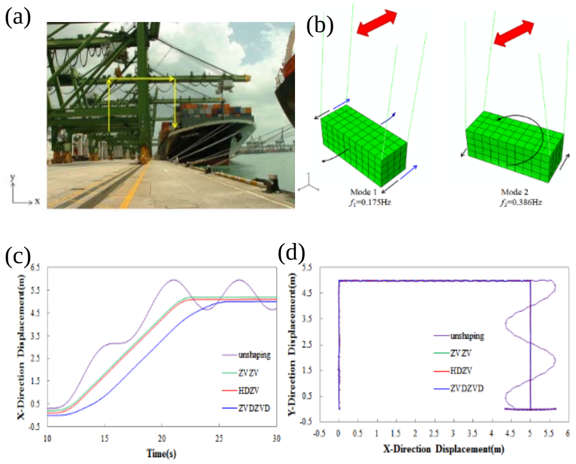

Fig. 20 (a) schematic of the cargo transportation, (b) the associated FE model to show the swing and torsional modes, (c) the moving history in X-direction, and (d) the swing behaviour

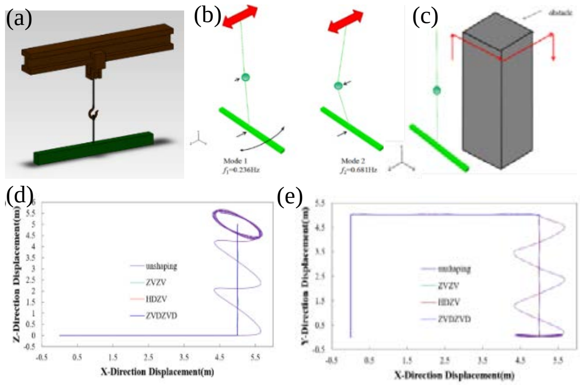

Figure 21 (a) schematic of the crane transportation, (b) the associated FE model to show the first two modes, (c) the schematic plot to show the moving path and (d) the moving histories

DISCUSSIONS

The performance and robustness of input shaper are largely depended on the accuracy of the plant dynamics. Here we employ FE dynamic simulation in input shaper design. The core of this work is to design two long range flexible motion systems as the test platform and their corresponding FE model for examining the effectiveness of input shaping schemes and for validating the FE dynamic simulations. The cases we chosen as the experiment are not trivial because they are actually structures commonly existing in machine tools and robot arms and both of them potentially suffer from motion-induced vibration in high speed maneuverer. The results clearly indicate the potential of input shaping in high speed point-to-point motions. Meanwhile, the experimental test structures are actually complex enough and it is hard to obtain the system dynamics via analytical dynamics. As a result, the agreement between the FE simulation and the experimental results suggests that it is possible to use FE simulation for input shaper design. In the future, more realistic problems can also be investigated.

Finally, although the FE-based design has been validated through the case studies, the true strengths of the proposed approach might not be fully revealed. A machine targeted for vibration control may have more complicate characteristics, such as nonholomonic constraints, limited actuator bandwidth [Danielson et al, 2008] or authorities, structural anisotropy, and even time-varying dynamics, could be taking into account in FE simulation for a success input shaper design. Finally, other design considerations such as stress, deformation, and fatigue, can also be considered simultaneously under a FE environment. Therefore, the FE simulation offers a more versatile manner to design the associated input shapers and we believe that the real strength of using FE simulation can be fully revealed in systems involving the above-mentioned characteristics.

CONCLUSION

Input shaping is an effective method for suppressing motion-induced vibration for flexible structures. However, the success of input shaper design largely depends on the accuracy of system dynamics models, which are usually created by analytical dynamics and this approach may be broken down for complex mechatronics systems. As a result, input shaping design based on FE dynamic simulation is proposed in this work. Two experimental systems are designed and the effectiveness of input shaping schemes are examined via experimental investigation and FE dynamic simulations. The successful results indicate that it is possible to use FE simulation for designing input shapers in complex mechanical systems such as systems with non-holonomic constraints. Several simulation examples are then further conducted to elucidate the concept of design and for promote the possible applications in the future. By this approach, other key design analyses can also be performed simultaneously under the same FE architecture.

ACKNOWLEDGEMENT

This work was supported by the Ministry of Science and Technology (MOST) of Taiwan under contact numbers MOST 105-2221-E-006-074-MY3, 107-2622-8-006-015, 108-2622-8-006-014

REFERENCES

ABAQUS Theory Manual, version 6.10, Simulia, 2009.

Abdel-Rahman, E., Nyfeah, A., and Masoud, Z. “Dynamics and control of cranes: a review." *Modal Analysis*, v. 9, pp. 863-908, 2003.

H. Baruh, *Analytical dynamics*, McGraw-Hill, 1999.

Chang, P. H., and Park, H. S., “Time-varying input shaping technique applied to vibration reduction of an industrial robot,” *Control Engineering Practice*, v. 13, pp. 121-130, 2005.

Chen, K.-S., Trumper, D. L., and Smith, S. T., “Design and Control of an Electromagnetically Driven X-Y-θ Stage,” *Precision Engineering*, v. 26, pp.355-369, 2002.

Chen, K.-S., Yang, T.-S., and Yin, J.-F., “Residual vibration suppression for duffing nonlinear systems with electromagnetically actuation using nonlinear command shaping techniques,” *ASME J. Vibrations and Acoustics*, v. 127, pp. 778-789, 2006.

Chen, W.-J., *Finite element simulations for input shaping design of flexible long range motion systems: analyses and applications*, MS Thesis, National Chen-Kung University, Taiwan, 2011.

Chen, W.-J., Chen, K.-S., and Tsai, M.-C., “Finite element input shaping design for vibration suppression of mechatronics systems,” *Proc. 2019 IEEE International Conference on Mechatronics (ICM)*, v. 1, pp., 37-42, 2019.

Dai, C.-X., and Chen, K.-S., “Residual vibration suppression of crane movement by input shaping: experimental investigation and finite element dynamics simulations,” *Journal of Chinese Society of Mechanical Engineers*, v. 34, pp. 497 – 506, 2013.

Danielson, J., Lawrence, J., and Singhose, W., “Command shaping for flexible Systems subject to constant acceleration Limits,” *J Dyn Syst T ASME*, v. 130, 051011, 2008.

Daqaq, M., Reddy, C., and Nayfeh, A.,” Input-shaping control of nonlinear MEMS,” *Nonlinear Dynamics*, v. 54, pp. 167-179, 2008.

Hong, K., Huh, C., and Hong, K., “Command shaping control for limiting the transient sway angle of crane systems,” *International Journal of Control, Automation, and Systems*, v.1, pp. 43-53, 2003.

Lin, W.-T., Chen, K.-.S, Chen, Y.-C., and Vechet, S., “Suppression of motion-induced vibration for a two-doF system using state feedback control,” *Proc. 2016 IEEE International Power Electronics and Motion Control Conference (PEMC)*, pp. 775-780, 2016.

Ou, K.-S., Chen, K.-S., Yang, T.-S., and Lee, S.-Y., "A command shaping approach to enhance the dynamic performance and longevity of contact switches," *Mechatronics*, v. 19, pp. 375-389, 2009.

Ou, K.-S., and Chen, K.-S., “Developing electromechanical coupling macro models for improving dynamic responses of MEMS elastic structures with command shaping techniques,” *IEEE T Device Mat Re.* v. 16, pp. 123-131, 2016.

Sano, H., Sato, K., Ohishi, K., and Miyazaki, T., “Robust design of vibration suppression control system for crane using sway angle observer considering friction disturbance,” *Electrical Engineering in Japan*, vol. 184, pp. 36-46, 2013.

Shan, J., Liu, H.-T., and Sun, D., “Modified input shaping for a rotating single-link flexible manipulator,” *J. Sound and Vibration*, v. 285, pp. 187-207, 2005.

Singer, N. and Seering, W., “Preshaping command inputs to reduce system vibration,” *J Dyn Syst T ASME*, v.112, pp. 76-82, 1990.

Singhose, W., Kim, D., and Kenison, M., “Input shaping control of double-pendulum bridge crane oscillations." *J Dyn Syst T ASME*, v. 130: 034504-1, 2008.

Singhose, W., Seering, W., and Singer, N., “Improving repeatability of coordinate measuring machines with shaped command signals,” *Precision Engineering*, v. 18, pp. 138-146, 1996.

Sun, N., Fang, Y., Zhang, Y., and Ma B., “A novel kinematic coupling-based trajectory planning method for overhead cranes,” *IEEE-ASME T Mech*, v. 17, pp. 166-173, 2012.

Wang, W.-C., Lee, J.-W., Chen, K.-S., and Liu, Y.-H., “Design and vibration control of a notch-based compliant stage for display panel inspection applications,” *Journal of Sound and Vibration*, v. 333, pp. 2701-2718, 2014.

Yi, J., Zhu, Q., Xiong, R., and Wu, J., “Vibration suppression based on input shaping for biped walking. In Humanoid Robots (Humanoids),” *Proc. 2016 IEEE-RAS 16th International Conference*, pp. 236-241, Nov. 2016.

Yoshida, Y., “Positioning and swing-rotation vibration suppression control of a suspended cargo container,” *Proc. IEEE 7th International Workshop on Advanced Motion Control*, pp. 556-560, 2002.

Zhang, M., Ma, X., Chai, H., Rong, X. Tian, X., and Li, Y., “A novel online motion planning method for double-pendulum overhead cranes,” *Nonlinear Dynamics*, v. 85, pp. 1079-1090, 2016.

有限元素分析於撓性長距離移動系統之輸入修正設計與應用

陳韋如　陳國聲

國立成功大學機械工程學系

摘要

長距離移動系統運作過程常因致動器加減速造成晃動量，並於完成作動後殘餘殘留振動量，降低系統工作精度與效率，並增加工作時間，甚至造成工安威脅。常見的振動抑制策略以回授控制的方式對系統進行振動抑制，此方法雖提供了系統對抗外界干擾的能力，具有良好的強健性，但須加裝額外的感測器與回授控制之致動器，大幅增加系統成本與複雜度。而輸入修正法提供了一個經濟且簡便的減振方式，然而傳統上輸入修正法主要依靠解析動力模型，但對結構與邊界條件複雜且具耦合之系統，解析解之獲得具有相當的挑戰性,且其形式也可能過分複雜，失去解析解簡潔的本意。本文在此提出以有限元素動力分析模擬撓性長距離移動架構，並在動態模型中設計輸入修正法對系統進行振動抑制，相較於解析解可更詳細描述系統之運動行為，且在多自由度系統的模擬上具有更佳的便利性。接著設計長距離移動系統中常見的天車系統與機械手臂系統之等效實驗模型，比較實驗與有限元素動態分析結果以驗證力學分析準確性以及所提想法之可行性。 最後我們將有限元素法的應用延伸至實際尺寸之長距離移動系統，討論各種輸入修正方法於系統之振動抑制情況。本研究將有助於長距離移動系統振動抑制之相關應用。