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DYNAMIC MODELLING AND INPUT SHAPING CONTROL FOR POSITIONING STAGE

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Abstract

This paper presents the dynamic analysis and input shaping control of a positioning stage. Vibration characteristics of positioning stages are affected not only by the structural dynamics but by the servo actuators that consist of the mechanism, driving motor and controller. This paper proposes an integrated dynamic model to accommodate both the structural dynamics and the servo actuators. Theoretical modal analysis with a commercial finite element code is carried out to investigate the dynamic characteristics of the experimental positioning stage. Experiments are performed to validate the theoretical modal analysis and estimate the equivalent stiffness due to the servo actuators. This paper deals with an input shaping scheme to suppress vibration of the positioning stage. Input shapers are systematically implemented for the positioning stage in consideration of its dynamics. The effects of servo control gain are also investigated. The experiments show that input shaping effectively removes residual vibrations and then improves the performance of positioning stage.

1. INTRODUCTION

The increasing demand placed on positioning stages is higher speed and more precise movement than ever. This has led to the necessity of dynamic analysis and control for positioning stages [1,2]. This paper is concerned with the dynamic analysis and input shaping control of positioning stages. To this end, we established an experimental system consisting of a 3-axis positioning stage based on Cartesian coordinates and the associated servo actuators.

Vibration characteristics of positioning stages are affected not only by the structural dynamics but by the servo actuators that consist of the mechanism, driving motor and controller. Thus the accurate prediction of positioning stage dynamics requires good models of the structural dynamics as well as servo actuators [3]. This paper proposes an integrated dynamic model to accommodate both the structural dynamics and the servo actuators. Theoretical modal analysis with a commercial finite element code is carried out to investigate the dynamic characteristics of the experimental stage. The theoretical model is refined by including the dynamics of servo actuators consisting of the servo motor, driving mechanism and controller. An equivalent spring constant is employed to take account of the servo actuator dynamics. Experiments are performed to validate the theoretical model and to estimate the equivalent spring constant.

Many researchers have investigated various control methods to suppress vibrations caused by unbalanced movement in flexible structures. Among others, input shaping is known to be a very effective tool for suppressing residual vibration arising from maneuvering flexible structures, without employing any complicated systems [4-6]. This paper deals with an input shaping scheme to suppress vibration of the positioning stage. One of the goals of this paper is to develop a systematic method to implement input shaping schemes for positioning stages in consideration of the dynamics of positioning stages. Two kinds of input shapers such as ZV(zero vibration) and ZVD(zero vibration and derivative) [4,6] are implemented: they were designed based on the information from the dynamic analysis. The effects of servo control gain change are also investigated.

Through a series of theoretical and experimental studies, it is proved that the systematic approach based on the accurate dynamic analysis and input shaper implementation can effectively reduce the vibration of high speed stages.

2. DYNAMIC MODELLING OF 3-AXIS POSITIONING SYSTEM

2.1 The Experimental System

Figure 1 shows the experimental setup which consists of a bridge type 3-axis positioning stage and the servo actuators. This system is designed to be used for either a coordinate measurement machine or an industrial crane. However, this study is just concerned with the 3-axis positioning system. AC servo motors, which are adapted for positioning all 3 axes, are controlled by a computer equipped with a motion controller. The control is based on a semi-closed loop. Figure 2 shows the schematic diagram of the system and the coordinates system for this positioning system.

2.2 Dynamic Modelling and Analysis

A finite element analysis is carried out to investigate the dynamic characteristics of the system by using a commercial finite element code ANSYS®. Vibration characteristics of positioning stages are affected not only by the structural dynamics but by the servo actuators that consist of the mechanism, driving motor and controller. In particular, the compliance due to the servo motors is a very crucial factor to be included in the dynamic model. This study introduces an equivalent spring constant which takes account of the mechanism stiffness and the servo stiffness that is obviously a function of control gain. To evaluate the effects of the equivalent

Figure 1 Experimental setup.

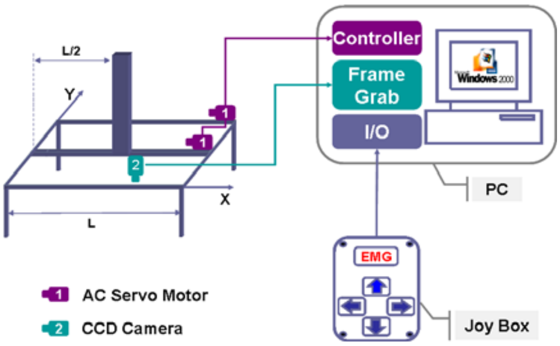
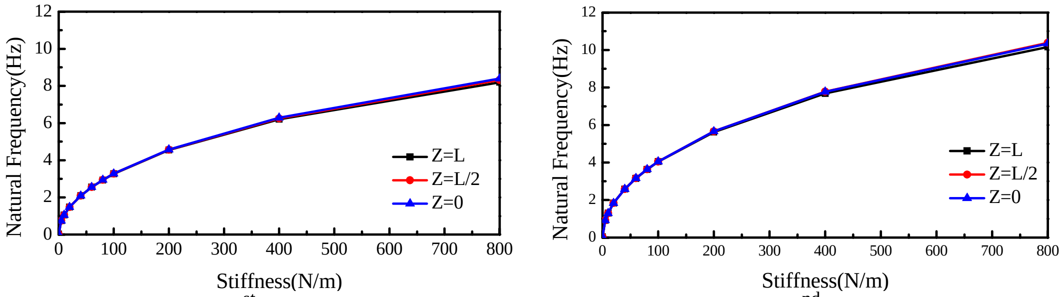
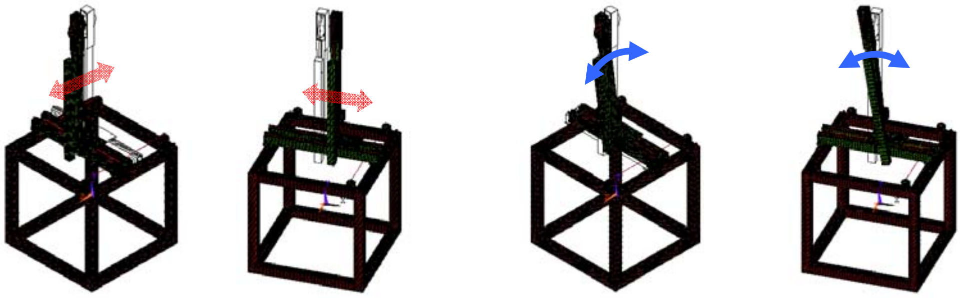


Figure 2 Schematic diagram of system and the coordinates system.



(a) 1st mode (b) 2nd mode

Figure 3 Comparison of natural frequencies with the equivalent stiffness varied.



1st Mode 2nd Mode 3rd Mode 4th Mode

Figure 4 The first four mode shapes.

stiffness coefficient, natural frequencies are computed with the equivalent stiffness varied from 0.01 N/m to 800 N/m.

Figure 3 presents the first two natural frequencies with the equivalent stiffness changed. The Z axis arm location is also changed to investigate the effects. The maximum stroke of the Z axis arm (L) is 600 mm and the natural frequencies are computed at three different configurations of the Z axis arm at 0, L/2 and L. Figure 3 reveals that the Z axis arm location is insignificant but that the equivalent stiffness due to the servo actuator can significantly change the first two modes, which mostly contribute to residual vibration.

Figure 4 shows the first four mode shapes. Y-directional translation motion of the Z axis arm is dominant in the first mode shape, while X-directional translation of the Z axis arm dominates the second mode. Since X or Y directional rigid body motion is dominant in the first two modes, the model may be simplified by two one degree-of-freedom models which are subjected to X and Y directional motions, respectively. On the other hand, the third and higher modes are affected mostly by the flexibility of the frame, little by the equivalent stiffness. Figure 4 also shows the 3rd and 4th modes, which appear to be depending on the deflections of the Z axis arm. The third mode is dominated by Y-directional deflection of the Z axis arm, the fourth mode by X-directional deflection of the Z axis arm. The 3rd and 4th natural frequencies are computed and shown in Fig. 5 by the finite element code with the equivalent stiffness and the Z axis arm location varied. Compared to the first and second modes, the 3rd and 4th modes are affected more significantly by the Z axis arm location. However, these two modes are less important than the first two modes and will be disregarded in the next step.

2.3 Experimental Validation and Identification of Equivalent Stiffness

Figure 6 illustrates the experimental setup to measure the natural frequencies of the system. Since this study is concerned only with the first two modes which are either X or Y direction dominant, experiments are performed to measure one mode in each of X and Y directions. The Z axis arm location is fixed at L/2 because of its insignificance on the first two modes as already described in the previous section.

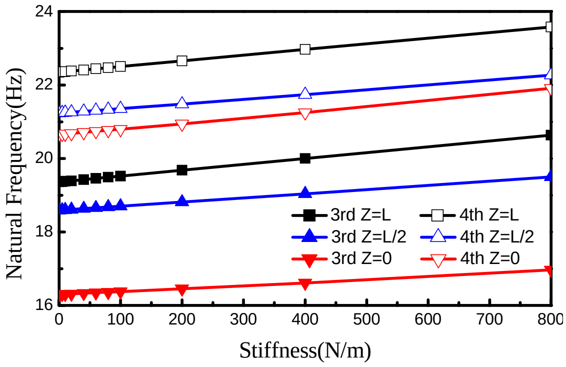


Figure 5 Comparison of natural frequencies with the Z axis arm positioned at 0, L/2, L.

The equivalent stiffness is evaluated by comparing the measured natural frequency with the computed one. Table 1 presents the first natural frequency and equivalent stiffness with the servo gain parameter varied. The relationship between the servo gain and the equivalent stiffness is almost linear as shown in Figure 7. This is a reasonable result because the servo gain K_v here is a velocity loop gain which comes from the controller, $K(s) = K_v(1+1/T_i s)$, T_i being the integration time constant. However, the equivalent stiffness is influenced not only by the servo gain but also by the timing belts and reducers implemented to the stage. The equivalent stiffness may be represented by a linear regression, $K = 257.68 + 3.10K_v$.

Figure 8 shows the first four natural frequencies computed with the identified stiffness used for computation. As far as the first two modes are concerned, the frame structure is rigid enough to ignore in the further analysis while the servo gain is significant enough to dominate the lower natural frequencies. Thus, the first two natural frequencies may be described as a function of the servo gain, though the entire system is obviously affected by the structure, the mechanism and the servo gains. We will discuss how to suppress the residual vibration caused by non-uniform motions of the system by using input shaping schemes, which necessitate the information regarding natural frequencies. The result in this section is used to implement input shaping schemes to the system.

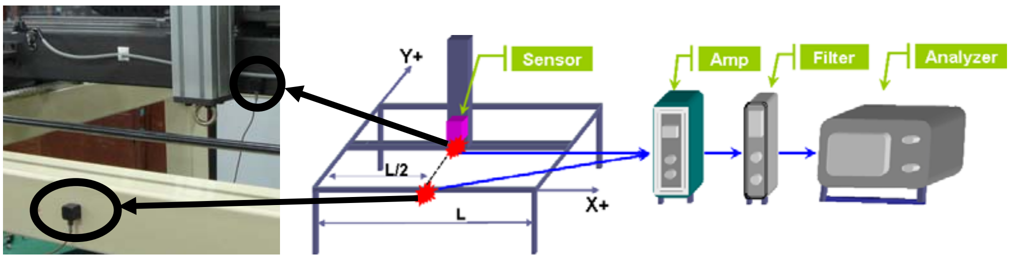


Figure 6 Schematic diagram of measuring mechanism.

Table 1. Measured fundamental natural frequencies with the servo gain changed and the associated equivalent stiffnesses.

Servo Gain	Natural Frequency(Hz)	Estimated Stiffness(N/m)
8	6.50	274
10	6.70	293
20	7.00	323
30	7.33	358
40	7.47	375

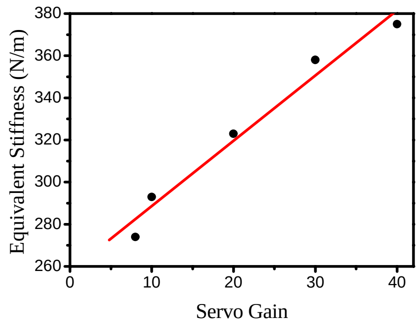


Figure 7 Equivalent stiffness vs. servo gain.

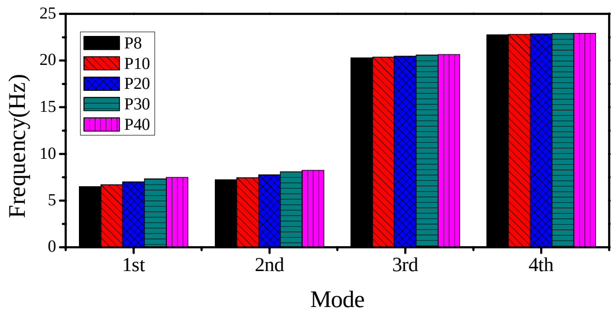


Figure 8 Comparison of natural frequencies with the servo gain.

3. APPLICATION OF INPUT SHAPING CONTROL

3.1 Concept of Input Shaping to Suppress Residual Vibration

Figure 9 illustrates a block diagram for typical control systems. The command in Figure 9 means a command generator, for example, a pendent in cranes and a joystick in coordinate measuring machines. The command is motivated by operators or machines but may be modified or re-generated inside of the control system to meet with certain requirements. Input shaper is a kind of command modifier to suppress the residual vibration possibly caused by non-uniform input commands to move machines.

This study deals with two kinds of input shapers, ZV(Zero Vibration) and ZVD(Zero Vibration and Derivative), proposed by Singhose et al. [4,6]. The natural frequency and damping ratio of the system should be known *a priori* to implement both ZV and ZVD. However, ZVD is more robust than ZV in respect to the accuracy of the natural frequency and damping ratio information.

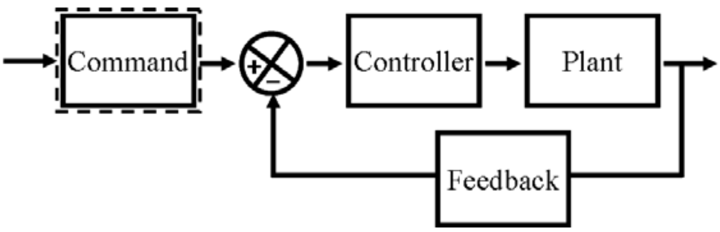


Figure 9 Block diagram of a generic control system.

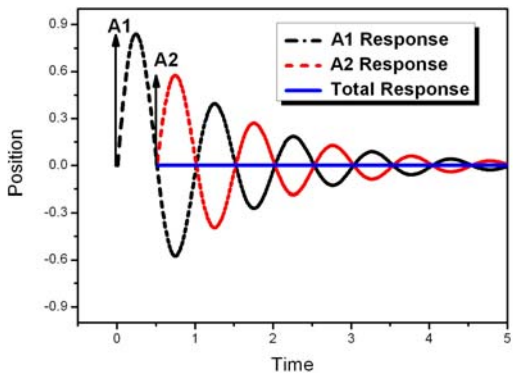
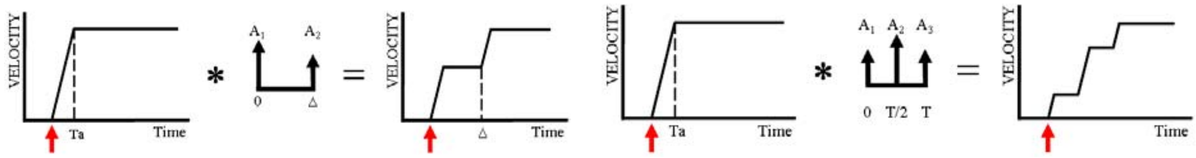
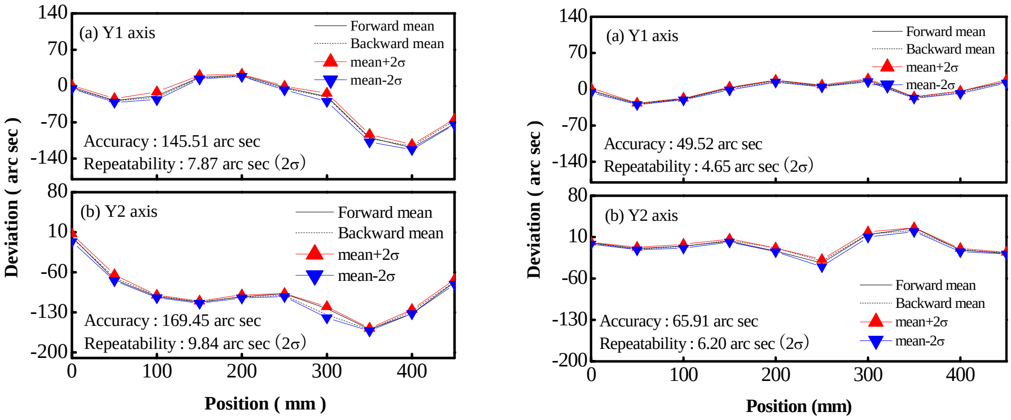


Figure 10 Vibration elimination with input shaping [4].



(a) ZV (b) ZVD

Figure 11 ZV and ZVD input shaping based on convolution.



(a) Before compensation (b) After compensation

Figure 12 Straightness errors in Y axis railways before and after error compensation.

Figure 10 illustrates the concept of ZV vibration suppression: the vibration caused by the first impulses is cancelled out by the vibration caused by the second impulse of which location and amplitude are determined at half a period and a relation based on the damping ratio, respectively. In order to implement input shaping to actual input commands, the input commands may be modified via convolution processes as shown in Figures 11(a) and (b). ZV contains two impulses while ZVD contains three impulses and requires a longer time.

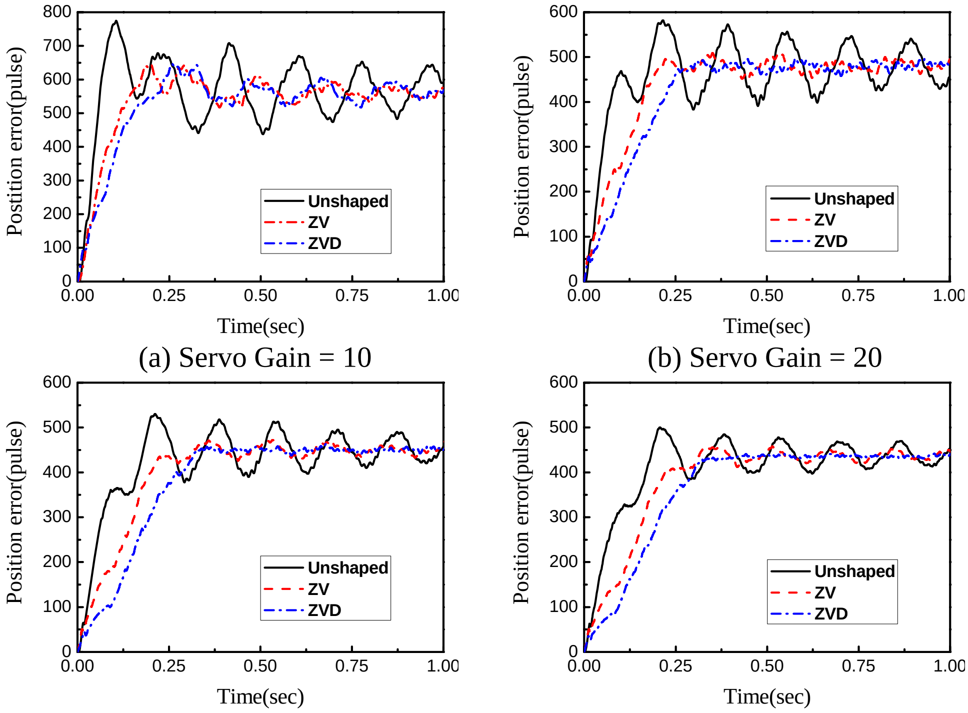
3.2 Precision Enhancement for Positioning System

The accuracy of railways implemented in the positioning system influences the vibration characteristics of the system as the vibration influences the accuracy of the system. In order to improve the vibration characteristics as well as the precision of the system, straightness error compensation is made for the system. Figure 12 shows the straightness measurements, with a laser interferometer, before and after error compensation. The compensation process is carried our by adjusting the assembly of X and Y railways. The results show that the straightness is improved by 80%.

3.3 Input Shaping Application

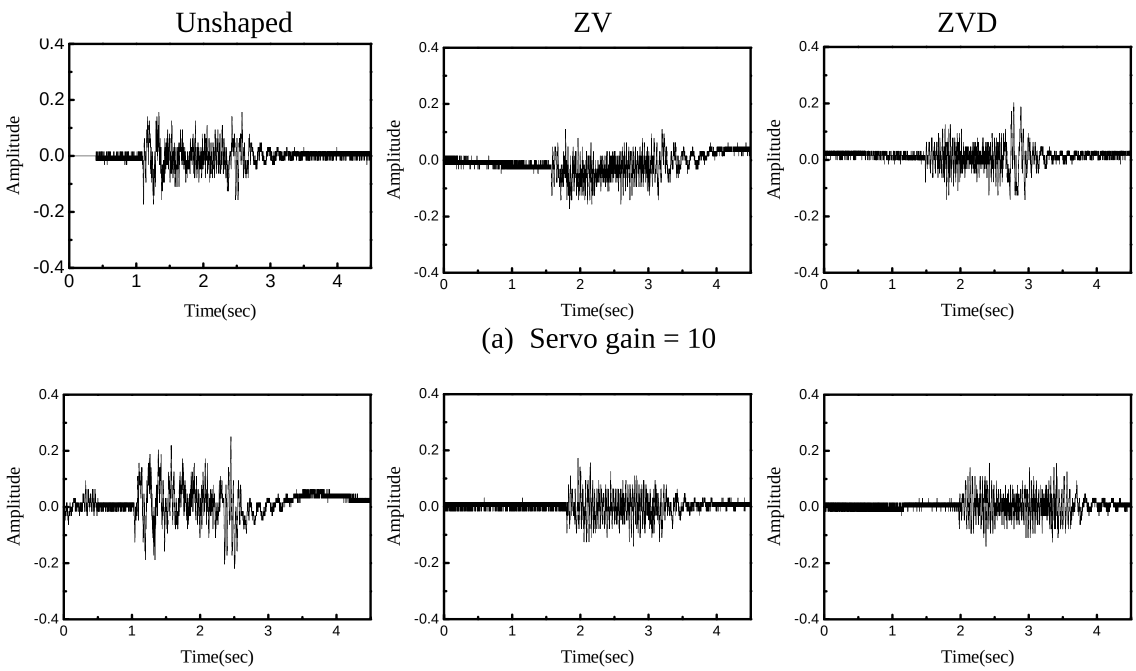
ZV and ZVD input shapers are implemented to the experimental positioning system. In order to show the effects of input shaping, we test the system with and without applying input shaping. Accelerations are measured at the frame as shown in Figure 6. Position errors which are defined as the difference between position command and actual position are retrieved from the motion controller to compare the results. Real-time inputs are generated from a push button type pendent and then convolved by ZV or ZVD shaper which is designed based on the dynamic analysis. In this kind of real-time input shaping process, the time spacing and duration time for servo motors are parameters to be assigned. Here both parameters are set 200ms. The maximum speed of XY axes is approximately 300mm/s.

Figure 13 shows the position errors with and without input shaping. In general, the position errors are getting les significant as the input servo gain increases. The reason is that the servo gain increases the natural frequency and damping ratio. Figure 14 is a comparison of acceleration signals which are acquired from the accelerometer at the frame. The experimental results definitely show that the input shaping effectively reduces the residual vibrations. Since the input shaping here takes care of only low frequency modes, high frequency components still reside in the response even after input shaping. Particularly, these high frequency components prevail in the frame acceleration signals, but they give little effects on the accuracy of the system.



(a) Servo Gain = 10
(b) Servo Gain = 20
(c) Servo Gain = 30
(d) Servo Gain = 40

Figure 13 Comparison of the position errors with and without input shaping.



(a) Servo gain = 10
(b) Servo gain = 20

Figure 14 Comparison of the accelerations at the frame with and without input shaping.

4. CONCLUSIONS

This paper presents the dynamic analysis and input shaping control of a 3-axis positioning stage and its peripherals. The dynamic analysis is carried out with a finite element model which contains not only the structural dynamics but also an equivalent spring for servo actuators. The dynamic model is validated experimentally and the associated equivalent stiffness due to the servo actuators is estimated. The dynamic analysis and experiments show that the first two natural frequencies are dominated by the equivalent stiffness. Two input shaping schemes, ZV and ZVD, are implemented to suppress residual vibrations of the positioning stage. The dynamic model is utilized to design the shapers. The experiments on input shaping control show that residual vibrations can be effectively eliminated by input shaping.

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