Vibration damping and isolation systems using direct inertia force control

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Abstract

Active vibration control devices are extensively used in various industrial fields. These devices are categorized according to their mechanisms into the following two types: (A) The vibration is controlled by actuators whose ends are connected to the controlled object and to “a fixed floor or a reaction wall”. (B) The vibration is controlled by actuators whose ends are connected to the controlled object and to “movable mass”. Typical types (A) devices are active vibration isolation devices. The advantage of a type (A) device is its excellent vibration control performance. However, it is difficult to downsize these devices because the actuator has to support the controlled object. In contrast, typical type (B) devices are active mass dampers (AMD). They do not need to support the controlled object; therefore, it is possible to realize compact systems. However, the control system design tends to be complicated, especially for multi-axis plants. In this study, we propose a new vibration control system design concept called “Direct Inertia Force Control (DIFC)”. By using DIFC, we can achieve the above mentioned advantages of both types (A) and (B), as well as circumvent the disadvantages. Furthermore, the effectiveness of DIFC is verified via experiments on a newly designed single-degree-of-freedom active vibration control device.

Keywords : Vibration control, Active mass damper, Direct inertia force control

1. Introduction

Active vibration control devices have many uses, such as vibration isolation for precision devices, damping for building vibration, and improvement of vehicle comfort. These devices are categorized according to their mechanisms into the following two types. (A) The vibration is controlled by actuators whose ends are connected to the controlled object and to “a fixed floor or a reaction wall”. (B) The vibration is controlled by actuators whose ends are connected to the controlled object and to “movable mass” (Seto, 2006). This type of device can control the vibration with the inertia force generated by the movable mass and actuator. Typical types (A) devices are active vibration isolation devices. These are mainly used for the vibration isolation of the object devices which are installed on the floor. An example application of this type of device is the semi-conductor exposure apparatus. The advantage of type (A) devices is that they allow for the realization of excellent vibration control performance by using feed-forward control (Tagawa, et al., 1997). However, it is difficult to downsize these devices because the actuator has to support the controlled object. Typical types (B) devices are active mass dampers (AMDs). AMDs can improve the sharp resonance characteristics of

the vibration controlled object. Type (B) control devices are used in, for example, vibration damping for microscopes and buildings (Fujita, 2000, 1998). Since the actuator does not have to support the controlled object, it is possible to realize compact type (B) systems. However, it is difficult to achieve the same vibration control performance as with the type (A) devices. In addition, type (B) control system design tends to be complicated, especially for multi-axis plants. For these reasons, in this study, a control system design technique is proposed that has the advantages of both the type (A) and type (B) devices. In this design technique, an inertia force is used as a control input. Therefore, this design technique is classified as type (B). The difference with the AMD is that the vibration control force is generated by the inertia force control and that this force acts directly on the vibration controlled object. This technique, which controls the vibration by using the directly controlled inertia force, is called “Direct Inertia Force Control” (DIFC) by the authors. The details of DIFC are discussed in the next section. By using DIFC, the controller design of a multi-axis vibration system can be simplified. In addition, it can realize the advantages of type (A) devices and other useful vibration control techniques, such as vibration cancelling proposed by the authors (Tagami, et al., 2007).

In this paper, the basic concept of DIFC is described. Furthermore, to verify the effectiveness of DIFC, a single-degree-of-freedom (1-DOF) active vibration damping and isolation device was developed by using DIFC. The performance of this prototype was confirmed by experiment. In this study, vibration damping means suppression of the resonance characteristics of the vibration controlled object, and vibration isolation means suppression of the floor vibration transmitted to the vibration controlled object.

2. DIFC

2.1 Concept of DIFC

The concept of DIFC is described by way of the 1-DOF AMD system. In the case of vibration control with an AMD, the AMD is installed on the vibration controlled object as shown in Figure 1 (a). Note that even if the vibration controlled object is a 1-DOF system, the AMD controller must be designed with a 2-DOF model. On the other hand, in DIFC, the control force can be generated by the inertia force control, and vibration control is achieved by direct action of this force on the controlled object. For this reason, the Inertia Force Generator (IFG), which can generate the required inertia force quickly and accurately, is used instead of the AMD. The required hardware for the IFG is similar to that of the AMD. However, in order to control the movable mass acceleration, the IFG demands an accelerometer on the movable mass. The IFG can generate an arbitrary inertia force. This means that the IFG dynamics can be ignored in the design of the vibration control system. Thus, the vibration controller can be designed with a model in which the vibration control force acts directly on the controlled object. The advantages of DIFC are summarized as follows: (1) Although conventional controllers of the 1-DOF AMD are designed with a 2-DOF system model including the dynamics of the AMD, by using DIFC, the controller can be designed with a 1-DOF model as shown in Figure 1 (b). This simplification is particularly effective for the multi-axis vibration control system design as described in the next section, (2) When using DIFC, the dynamical model of the vibration controlled object is categorized as type (A) introduced in the previous section, though its hardware configuration is categorized as type (B). Therefore, by using DIFC, all the advantages of a type (A) device, such as the improvement of the vibration isolation performance by feed-forward control, can be realized.

Figure 1 Conceptual view of DIFC

2.2 Multi-axis vibration control with DIFC

The controller design for controlling the multi-axis vibration mode with AMDs often becomes a very complex process, because the model dimension increases with the number of DOFs of the AMDs (Figure 2 (a)) (Nogawa, et al., 2007). On the other hand, by using DIFC, the controller can be designed with a model in which the vibration control force acts directly on the controlled object (Figure 2 (b)). In addition, by combining with modal analysis, the controller design for the multi-axis system can be replaced to the several simpler 1-DOF controller designs.

3. One-axis Active Vibration damping and Isolation System with DIFC

3.1 Hardware

A prototype of the one-axis active vibration damping and isolation system was developed to confirm the

Figure 2 Multi-axis vibration control system

Figure 3 Overall view of the 1-DOF active vibration damping and isolation system

Figure 4 Picture of the 1-DOF active vibration damping and isolation system

Figure 5 Schematic view of the IFG

effectiveness of DIFC. Figures 3 and 4 show an overall view and a picture of the system, respectively. This system provides vibration damping and isolation for the vibration controlled object, which consists of the simple steel table ($400\text{mm} \times 500\text{mm}$, 19kg) supported by four springs and dampers. y_A in Figure 3 is the movable mass acceleration of the IFG and y_T is the table acceleration. y_F is the vibration acceleration of the device-installed plane. In a later discussion, y_F is called an environmental disturbance. Figure 5 shows a schematic view of the IFG, which is the main device for DIFC. The IFG consists of a movable mass supported by leaf springs, an accelerometer attached to the movable mass, and a voice coil motor with its driver circuit. u in Figure 5 is the command signal to the IFG driver.

3.2 Framework of the control system

The framework of the entire control system is shown in Figure 3. y_A can be controlled by the acceleration controller C_{Acc} . The control input to the vibration controlled object is the inertia force, which is obtained by the product of the controlled acceleration y_A and the movable mass. However, in practice, for many components such as the linear guide component, the masses of the springs and wires must be included in the movable mass. Accordingly, it is difficult to estimate the magnitude of the movable mass accurately. For this reason, the movable mass acceleration y_A was used as a control input to the vibration controlled object. This way, the movable mass of the IFG is not needed to estimate because it is included in the vibration controlled object. The vibration damping is achieved with the feedback controller C_{Dmp} whose input is y_T . Furthermore, vibration isolation is realized with the feed-forward controller C_{Iso} whose input y_F is measured with an accelerometer. The target acceleration of y_A is defined by C_{Dmp} and C_{Iso} , then C_{Acc} controls the IFG to track this target acceleration.

4. Control System Design

4.1 Design of the acceleration Controller C_{Acc}

The plant model of the IFG is derived as a first step in the design of the acceleration controller C_{Acc} . The plant model of the IFG is represented with two types of transfer functions, P_{Au} and P_{Ad} . P_{Au} governs the transfer characteristics from u to y_A and P_{Ad} governs the transfer characteristics from y_T to y_A (see Figure 5). These transfer functions are given as follows:

$$P_{Au} = \frac{k_A s^2}{s^2 + 2\zeta_A \omega_{nA} s + \omega_{nA}^2} \quad (1)$$

$$P_{Ad} = \frac{2\zeta_A \omega_{nA} s + \omega_{nA}^2}{s^2 + 2\zeta_A \omega_{nA} s + \omega_{nA}^2}, \quad (2)$$

where k_A is an amplifier gain, and ζ_A and ω_{nA} are the damping ratio and the natural angular frequency, respectively.

Figure 6 shows the measured frequency response of P_{Au} . From this result, k_A can be identified with the flat part gain in the high frequency area, and ζ_A and ω_{nA} can be identified with the peak gain and its frequency as follows;

Figure 6 Frequency response of P_{Au}

Figure 7 Block diagram of the acceleration control system for the IFG

Figure 8 Generalized plant for the IFG acceleration control system

$k_A = 4.97$, $\omega_{nA} = 75.37 \text{ rad/s}$, and $\zeta_A = 0.0125$. The simulation result of this frequency response with the above identified values is also shown in Figure 6. Figure 6 shows that the plant model of P_{Au} expresses the actual transfer characteristics from u to y_A accurately. Second, the design of the acceleration controller C_{Acc} is described. It is assumed that the effect of the y_T , which is considered as the disturbance for the IFG control, is significant, because the IFG is used on the vibrating table and also IFG has to generate the target acceleration under this vibration environment. In addition, the IFG is required a good tracking performance of the target acceleration for the upper stage controllers C_{Dmp} and C_{Iso} . Accordingly, C_{Acc} was framed by a 2-DOF controller, as shown in Figure 7. C_{fbA} in Figure 7 is a feedback controller, and F is the desired transfer characteristic from the reference value r to y_A . If C_{fbA} suppresses the disturbance and model error, the characteristics from r to y_A will match with F . C_{fbA} is designed according to the H infinity control theory with emphasis on the disturbance suppression performance. The generalized plant used for the controller design is shown in Figure 8. In Figure 8 W_S and W_T are weighting functions, z_S and z_T are controlled variables, v is sensor noise, ϵ is a constant value, and w_A is a disturbance. W_S and W_T were defined based on the following guidelines:

- To realize the robust stability.
- In consideration of the IFG's stroke restriction, the control gain at the low frequency region near the D.C. must be kept low.
- To suppress the sensor noise effects.
- To suppress the effect on y_A from y_T in the control band.

Based on the above, W_T covers the multiplicative error of P_{Au} for robust stability, and W_T was designed to have

high gain in both low frequency region near the D.C. and high frequency region where sensor signal is noisy. W_S was designed to have high gain in the control band for excellent disturbance suppression. From the above, the weighting functions were defined as follows:

$$\varepsilon = 0.002 \quad (3)$$

$$W_T = K_{WT} \frac{T_{T1}}{T_{T2}} \cdot \frac{1+T_{T2}s}{1+T_{T1}s} \cdot \frac{1+T_{T3}s}{1+T_{T4}s} \quad (4)$$

$$W_S = K_{WS} \frac{s^2}{s^2 + 2\zeta_{S1}\omega_{nS1}s + \omega_{nS1}^2} \cdot \frac{\omega_{nS2}^2}{s^2 + 2\zeta_{S2}\omega_{nS2}s + \omega_{nS2}^2} \quad (5)$$

where, $T_{T1} = 1/(2\pi \times 0.01)$ s, $T_{T2} = T_{T3} = 1/(2\pi \times 2.7)$ s, $T_{T4} = 1/(2\pi \times 4000)$ s, $K_{WT} = 24$, $K_{WS} = 400$, $\omega_{nS1} = 2\pi \times 0.5$ rad/s, $\zeta_{S1} = 0.8$, $\omega_{nS2} = 2\pi \times 40$ rad/s, and $\zeta_{S2} = 0.8$. The controller C_{fbA} was derived by using MATLAB and the above generalized plant. H infinity control theory shows that closed loop gain is always less than the inverse gain of the weighting function (Mita, 2000, Uchida, et al., 1996). In the above weighting functions were designed with trial and error by using this relationship. Figure 9 and Figure 10 show this relationship. T and R in these figures are closed loop transfer function. γ is a singular value defined by the solving process of the controller C_{fbA} . T and R are given as follows:

$$T = \frac{C_{fbA}P_{Au}}{1 + C_{fbA}P_{Au}} \quad (6)$$

Figure 9 Relationship between closed loop transfer function T and weight function W_T .

Figure 10 Relationship between closed loop transfer function and weight function W_S .

$$R = \frac{P_{Ad}}{1 + C_{fba}P_{Au}} \quad (7)$$

Figure 9 indicates the gain of T in high frequency region is shaped by W_T . Figure 10 also indicates the gain of R in the control band is shaped by W_S . Closed loop disturbance suppression performance with C_{fba} was compared with P_{Ad} by the frequency response. It was confirmed that the closed loop system shows an excellent disturbance suppression performance.

Next, the designs of the feed-forward controllers F and F/P_{Au} are described. The feed-forward controllers have to satisfy the following conditions:

- (1) Both feed-forward controllers must be stable.
- (2) Both feed-forward controllers do not have a D.C. gain.
- (3) The transfer characteristics from r to y_A is the band pass characteristics.

Since P_{Au} is given, F/P_{Au} can be written as follows:

$$\frac{F}{P_{Au}} = F \frac{s^2 + 2\zeta_A \omega_{nA} s + \omega_{nA}^2}{k_A s^2} \quad (8)$$

This equation shows that the unstable zero of P_{Au} is undesired. To satisfy the condition (1), s^2 of the denominator is needed to cancel. Considering condition (2), F also needs differential characteristics to remove the D.C. components. Therefore, the numerator of F needs s^3 as a factor. Additionally, F has to be a proper transfer function. Therefore, the denominator of F has to be at least of 3rd order. Finally, to satisfy condition (3), F has to combine the 3rd order high pass characteristics and at least the 1st order low pass characteristics. Accordingly, F is given as follows:

$$F = \frac{T_{FH} s^3}{(s^2 + 2\zeta_F \omega_{nF} s + \omega_{nF}^2)(1 + T_{FL} s)} \cdot \frac{1}{T_{FL} s + 1} \quad (9)$$

The gain and phase characteristics of F have to be flat in the acceleration control band. The required acceleration control band for this vibration control device is from several Hz to several dozen Hz. For this reason, the parameters of F were chosen as follows:

$T_{FH} = 1/(2\pi \times 0.5)$ s, $T_{FL} = 1/(2\pi \times 350)$ s, $\omega_{nF} = 2\pi \times 0.5$ rad/s, and $\zeta_F = 0.8$.

The designed acceleration controller C_{Acc} was implemented in the digital signal processor (DSP), and the reference tracking performance was evaluated. In the evaluation experiments, the frequency response from r to y_A (reference tracking performance) was measured with an FFT analyzer. The corresponding simulation was also performed. Both results are shown in Figure 11. Figure 11 shows that the experimental result matches the simulation result. This means that the desired acceleration control performance can be realized with C_{Acc} .

Figure 11 Reference tracking performance of IFG

4.2 Design of the Vibration damping Controller C_{Dmp}

In this section the vibration damping controller C_{Dmp} , which can suppress the resonance characteristics of the vibration controlled object, is described. The plant model needed for the design of C_{Dmp} is derived. The plant consists of the simple table, passive vibration isolation system (springs and dampers system), and the IFG as shown in Figure 3. From this figure, the plant model is given the following two types of transfer functions, P_{Tu} and P_{Td} , which represent the transfer characteristics from y_A to y_T and from y_F to y_T , respectively.

$$P_{Tu} = -\frac{k_T s^2}{s^2 + 2\zeta_T \omega_{nT} s + \omega_{nT}^2} \quad (10)$$

$$P_{Td} = \frac{2\zeta_T \omega_{nT} s + \omega_{nT}^2}{s^2 + 2\zeta_T \omega_{nT} s + \omega_{nT}^2}, \quad (11)$$

where k_T is the gain, and ω_{nT} and ζ_T are the natural angular frequency and damping ratio of the passive vibration isolation system, respectively. The parameters of these models were identified by the frequency response. The frequency response was measured with the experimental apparatus shown in Figure 3. However, only the C_{Acc} was used in this experiment. C_{Dmp} and C_{Iso} were not used. To measure the frequency response, the table was vibrated by the IFG with the acceleration controller C_{Acc} , and then, the excitation acceleration (the acceleration of the IFG's movable mass) and the acceleration on the table were measured. The frequency response was obtained with an FFT analyzer. The experimental results are shown in Figure 12. Using this result and the same method for identifying the IFG parameters in sub-section 4.1, $k_T = 0.03$, $\zeta_T = 0.149$, and $\omega_{nT} = 33.75$ rad/s were obtained. The frequency response was simulated with eq. (10) and the above-identified parameters. The simulation result is also shown in Figure 12. The simulation results match the experimental result well, and that indicates that a good plant model was obtained.

The damping controller C_{Dmp} was also designed via the H-infinity control theory. In designing the damping controller, the characteristics of the IFG were neglected, because they were considered ideal. In fact, it is considered that the IFG can provide the acceleration required from C_{Dmp} immediately. It is an advantage of DIFC that the IFG can be used as an ideal force generator. C_{Dmp} was designed in the same way as C_{fbA} . Therefore, the generalized plant used in the design is the same as in Figure 8. Definitions of the symbols used in the figure are the same, but P_{Au} and P_{Ad} are replaced with P_{Tu} and P_{Td} . The forms of the weighting functions are also the same as in eq. (3), eq. (4), and eq. (5). However, the parameters in the weighting functions were chosen as follows:

$K_{WT} = 3$, $T_{T1} = 1/(2\pi \times 2)$ s, $T_{T2} = 1/(2\pi \times 1000)$ s, $T_{T3} = 1/(2\pi)$ s, $T_{T4} = 1/(2\pi \times 0.005)$ s, $K_{WS} = 53$, $\omega_{ns1} = 2\pi \times 4.5$ rad/s, $\zeta_{s1} = 0.25$, $\omega_{ns2} = 2\pi \times 30$ rad/s, $\zeta_{s2} = 0.25$, and $\varepsilon = 0.001$. C_{Dmp} was derived by using the above generalized plant and weighting functions. The vibration damping performance is shown in section five.

Figure 12 Frequency response of P_{Tu}

Figure 13 Block diagram of the control system for the active vibration damping and isolation system

4.3 Design of the Vibration isolation Controller C_{Iso}

In this section, the design of the feed-forward controller C_{Iso} , which realizes the vibration isolation, is described. The purpose of C_{Iso} is to suppress the influence of the environmental disturbance y_F transmitted to the table by the IFG. To realize that, the control input for the vibration isolation is calculated from y_F . From Figure 3, eq. (10), and eq. (11), the block diagram of the vibration damping and isolation control system is given as shown in Figure 13. From this block diagram, following relation was obtained.

$$y_T = \left(\frac{P_{Td} + P_{Tu} C_{Iso}}{1 - P_{Tu} C_{Dmp}} \right) y_F \quad (12)$$

To make y_T zero, eq. (12) indicates that $C_{Iso} = -P_{Td}/P_{Tu}$ must be satisfied; however, P_{Tu} has an unstable zero. Therefore, this C_{Iso} is not preferred feed-forward controller. Hence, the following controller in which the denominator of C_{Iso} is swapped for the 2nd order pseudo-integrator was adopted as the vibration isolation controller C_{Iso} .

$$C_{Iso} = \frac{2\zeta_s \omega_{ns} s + \omega_{ns}^2}{K_T (s^2 + 2\zeta_s \omega_{ns} s + \omega_{ns}^2)} \quad (13)$$

where ζ_s and ω_{ns} are the damping ratio and the natural angular frequency of the pseudo-integrator, respectively. From the above, the final vibration isolation performance is obtained by substituting eq. (13) into eq. (12):

$$y_T = \frac{2\zeta_s \omega_{ns} s + \omega_{ns}^2}{s^2 + 2\zeta_s \omega_{ns} s + \omega_{ns}^2} \cdot \frac{P_{Td}}{1 - P_{Tu} C_{Dmp}} y_F \quad (14)$$

$P_{Td}/(1 - P_{Tu} C_{Dmp})$ in the above equation represents the original vibration damping characteristics. Eq. (14) indicates that the vibration isolation performance is given as the product of the low-pass characteristics and the original vibration damping characteristics. Therefore, the poles of the pseudo-integrator have a significant influence on the vibration isolation performance. For this reason, the poles of the pseudo-integrator should be assigned as much as possible near the imaginary axis. On the other hand, the IFG stroke is restricted. In consideration of both these conditions, $\omega_{ns} = 2\pi$ rad/s, and $\zeta_s = 0.8$ were chosen.

5. Performance evaluations

Each controller mentioned in Section 4 was implemented, and the vibration damping and vibration isolation performances were evaluated. The transfer characteristics from the environmental disturbance y_F to the table acceleration y_T were measured with an FFT analyzer and the experimental apparatus shown in Figure 3. Experiments were conducted with the following procedures:

- To create an artificial environmental disturbance, the work bench was vibrated by the shaker operating according to the command signal from a function generator.
- y_F and y_T were measured with the accelerometer.
- The frequency response was obtained by FFT using the measured y_F and y_T .

Figure 14 Vibration damping performance (condition (i) vs. (ii))

Figure 15 Vibration isolation performance (condition (ii) vs. (iii))

Figure 16 Effectiveness of the vibration isolation control (condition (i) vs (iii))

The above experiments were conducted according to the following three conditions. (i) The case of not using the control (original plant characteristics); (ii) The case of only using the vibration damping control by C_{Dmp} ; and (iii) The case of using both the vibration damping and the isolation control by C_{Dmp} and C_{Iso} . Figure 14 shows the performance comparison of condition (i) and condition (ii). Figure 15 shows the performance comparison of condition (ii) and condition (iii). Figure 14 shows that there is a 10dB resonance peak at around 5Hz, and that it can be suppressed quite well by the vibration damping controller C_{Dmp} . Figure 15 shows that the vibration isolation controller can improve the vibration isolation performance more around 10 dB. Figure 16 shows the comparison of condition (i) and the condition (iii). This figure indicates that C_{Dmp} and C_{Iso} achieved the totally 20 dB improvement of the vibration isolation. These figures also show that the simulation results match well with the corresponding experimental results. In experiments, from 12Hz to 15Hz, there are another resonance peaks. There are several possibilities for these resonance peaks. One is the effect of the unmodeled vibration mode. This unmodeled vibration mode is not taking account in the controller design. This problem can be solved by extending the vibration damping and isolation controllers to include a multitude of vibration modes. The other is the coupling effect which should be suppressed with IFG acceleration controller between vibration controlled object and IFG. For this problem, it is needed to clarify the relation between coupling effect and required disturbance suppression performance. This is an issue in a future.

6. Conclusions

The concept of DIFC was proposed as a new vibration control technique. By using DIFC, active vibration isolation devices can be designed simply and more systematically compared to conventional methods. To validate this new approach, a 1-DOF active vibration damping and isolation device was developed using DIFC, and the effectiveness of the device was confirmed with experiments.

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