DVFC-Based Identification and Its Application to Robust Vibration Suppression Control of Flexible Structures

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This paper proposes direct velocity feedback control(DVFC)-based identification along with its application to robust vibration suppression control of flexible structures. At first, a flexible structure is modified by pseudo-passive control which is realized by DVFC. This DVFC behaves just like a mechanical damper if the sensor and actuator are collocated. Next, system identification is applied to the modified structure. Through this procedure, the accurate structure model for vibration suppression control is more easily derived without paying regard to a sampling time, an identification model and its order. Consequently, the uncertainty such as parameter errors and unmodeled dynamics can be expressed. Finally, the effectiveness of this identification method is verified through the experimental results primarily for a typical earthquake loading and additionally for vibration suppression control of a flexible structure.

Keywords: system identification, direct velocity feedback control(DVFC), uncertainty, robust control.

1. Introduction

A high-rise building consisting of light weight and low stiffness has been increasing. The vibration control of such a flexible structure subjected to an external disturbance, e.g., wind and earthquake loadings, has been intensively studied particularly after the earthquake in Kobe area, 1995. To meet research and development subjects involved in flexible structures, high-performance active control represented by $\mathcal{H}_\infty$ control and μ -analysis has been developed ^{(1)~(5)}.

In high-performance active control, it is a prerequisite to establish an accurate model of structure to avoid instability due to spillover or conservativeness of the controller gain of vibration suppression control systems ⁽⁶⁾. Modeling is now focussed on the system identification, which can be implemented only by using input and output data of a structure. The system identification for a flexible structure, which has plural vibration modes, however, is still on the way to development. Particularly, the research with regard to how to identify the flexible structure has not been reported much ⁽⁴⁾. Because it is complicated how to decide a sampling time, an identification model and its order. In addition, accurate modeling of the flexible structure over higher vibration modes by a conventional method such as sine sweep and modal analysis is also difficult due to its low damping factors.

In this paper, at first, introducing minor feedback control to the original structure modifies the damping factor of a flexible structure. This minor feedback control is realized by the well-known direct velocity feed-

back control(DVFC) using an active mass driver(AMD). Then, the system identification is applied to the modified structure. By this identification, an accurate model can be much more easily derived.

Taking the uncertainty between the nominal and the identified models into consideration, as an application of this identification method, a robust controller based on $\mathcal{H}_\infty$ control theory is designed to demonstrate the vibration suppression control capability for the accurate structure. The effectiveness of the proposed identification method is verified through the experimental results primarily for a typical earthquake(EL Centro NS) loading and additionally for vibration suppression control of a flexible structure.

2. DVFC-Based Identification

2.1 Modification of a Flexible Structure Before identifying an original structure, the damping factor of the structure that is closely related to accuracy of the identified model is optimized by the direct velocity feedback control(DVFC). DVFC is governed by the following equation:

$$f_c = f_v \dot{u} \quad \dots \dots \dots (1)$$

where f_c is control force to the structure, f_v is a DVFC gain, and $\dot{u}$ is velocity of the structure. In the proposed method, the same location of the measurement as the force input, i.e., collocation, is needed. Under the collocation, DVFC has the following characteristics:

- (1) The system is not to be instable even if the controller is designed using a reduced model for the structure.
- (2) The effect of DVFC is equivalent to that of the

Fig. 1. Diagram of system identification for original plant modified with DVFC.

mechanical damper mounted on the same location, that is, the effect of the passive control.

- (3) All damping factors of an observable vibration mode can be increased by DVFC.

From the viewpoint of designing the DVFC gain f_v , the minimum damping factor is maximized under the condition of stability. The characteristic equation G_{ce} for the N -story mechanical model compensated by DVFC is defined as:

$$G_{ce} = \prod_{n=1}^N (s^2 + 2\zeta_n(f_v)\omega_n(f_v)s + \omega_n^2(f_v)) \quad (2)$$

where ζ_n and ω_n are the damping factors and the natural frequencies, respectively. For the optimal DVFC gain f_{vo} , then, the following is given:

$$f_{vo} = \arg \max_{f_v} \zeta_{\min}(f_v) \quad (3)$$

In this design, nominal parameters derived by physical modeling or modal analysis are utilized even though these parameters may include an uncertainty.

2.2 DVFC-Based Identification In terms of the system identification, damping factors of a system is quite important to derive an accurate model easily. Therefore, the authors at first modify an original plant by DVFC and then apply the system identification to the modified plant with the optimal damping factor. The diagram of how system identification is implemented for an original plant modified by DVFC is shown in Fig. 1. In the system identification, the maximum-length linear shift register sequence is used for the input signal because of its wide-band characteristics. Defining a frequency range wanted to identify a plant accurately as ω_{id} , a sampling time is selected using the following relation between T_s and ω_{id} [rad/sec]:

$$\frac{2\pi}{100T_s} < \omega_{id} < \frac{2\pi}{5T_s} \quad (4)$$

Moreover, for the modified plant model, the authors use well-known *auto-regressive exogenous* (ARX) and *Box and Jenkins* (BJ) models defined as the following:

$$ARX : A_a(q)y(k) = B_a(q)u(k) + w(k) \quad (5)$$

$$BJ : y(k) = \frac{B_b(q)}{F_b(q)}u(k) + \frac{C_b(q)}{D_b(q)}w(k) \quad (6)$$

where q^{-1} is a shift operator and w is a white noise. Then, the accurate model of the plant can be identified

Fig. 2. Experimental setup consisting of the structure and its control unit.

not depending on the condition such as a sampling time, an identification model and its order. Moreover, the accurate model is derived without undesirable excitation of the plant owing to the improvement of the damping factor.

2.3 Design of Robust Control System It is expected that the conservativeness for the bandwidth of the control system can be improved if a multiplicative uncertainty can be implemented as correctly as possible. Therefore, the authors can derive the multiplicative uncertainty Δ_m using the nominal model P_n with DVFC, which is calculated by nominal parameters, and the accurately identified model P with DVFC by the following:

$$\Delta_m(s) = \frac{P(s) - P_n(s)}{P_n(s)} \quad (7)$$

The robust controller itself can be systematically designed by the $\mathcal{H}_\infty$ control theory.

3. Application of Proposed Identification

3.1 Structure of Application Model The experimental structure used in this study is the three-story building with an active mass driver (AMD) illustrated in Fig. 2. For simplicity, each story of the structure has one degree of freedom in the horizontal direction and an observable output of relative displacement is obtained only on the 3rd story. Vibration suppression control is performed by extinguishing the relative displacement against an external disturbance. The governing equation of motion of this system can be formulated by the Lagrange equation as the following:

$$M\ddot{u} + C\dot{u} + Ku = -m_1\ddot{u}_a - m_2\ddot{u}_g \quad (8)$$

where

$$M = \text{diag} [m_1 \quad m_2 \quad m_3]$$

$$C = \begin{bmatrix} c_1 + c_2 & -c_2 & 0 \\ -c_2 & c_2 + c_3 & -c_3 \\ 0 & -c_3 & c_3 \end{bmatrix}$$

Fig. 3. Gain plot of the original plant($f_c[V]$ to $u_3[V]$).

$$K = \begin{bmatrix} k_1 + k_2 & -k_2 & 0 \\ -k_2 & k_2 + k_3 & -k_3 \\ 0 & -k_3 & k_3 \end{bmatrix}$$

$$m_1 = [0 \ 0 \ m_a]^T$$

$$m_2 = [m_1 \ m_2 \ m_3]^T$$

$$u = [u_1 \ u_2 \ u_3]^T$$

M , C and K are the mass, damping and stiffness matrices of the structure, respectively. The first term on the right-hand side $m_1 \ddot{u}_a$, that is, $m_a \ddot{u}_a$ is the control force f_c generated by AMD using actively auxiliary mass. The second term $m_2 \ddot{u}_g$ is the force disturbance to the base such as earthquake loading. The nominal parameters that naturally include the uncertainty are shown in Table 1. Fig.3 shows the frequency response

Table 1. The nominal parameters.

m_1	1.74[kg]
m_2	1.65[kg]
m_3	2.34[kg]
m_a	0.047[kg]
c_1, c_2, c_3	2.2[N/m/s]
k_1, k_2, k_3	9233[N/m]
amp. of gap sensor	$\pm 2.5[V]/\pm 12.5[mm]$
transfer gain of motor driver	3[N]/5[V]

from the control force f_c to the relative displacement u_3 between the base and the 3rd story. It can be verified that the system has three vibration modes.

3.2 Application Example

3.2.1 Modification of the Structure Based on section 2.1, the minimum damping factor in the characteristic equation is maximized under the condition of stability:

$$G_{ce} = \prod_{n=1}^3 (s^2 + 2\zeta_n(f_v)\omega_n(f_v)s + \omega_n(f_v)^2) \quad (9)$$

In eq.(9), all the ζ_n and ω_n are functions of the DVFC gain f_v . The condition of stability of the system can be derived by the Hurwitz stability criterion. In this case, the modified structure consisting of nominal parameters shown in Table 1 can be stable when the negative feedback, namely, the feedback gain f_v is positive.

Figs.4 and 5 show the $f_v - \zeta_n$ plane and the root locus of the modified plant in the variation of the DVFC gain

Fig. 4. $f_v - \zeta$ plane.

Fig. 5. Root locus of the modified plant (u_3/f_c).

Fig. 6. Gain plots of the original and the modified plants(u_3/f_c).

f_v , respectively. From Fig.4, it is noticed that the optimal DVFC gain f_{vo} defined in eq.(3) is 2.0, which can maximize the minimal damping factor ζ_3 . Moreover, all the damping factors ζ_1, ζ_2 and ζ_3 are increased from 0.004, 0.01 and 0.016 to 1.27, 0.1 and 0.03, respectively.

Fig.6 shows the gain plot of the modified plant with f_{vo} and that of the nominal plant without DVFC. Both of them are calculated using the nominal parameters. From Fig.6, it is verified that all the resonance characteristics can be modified by DVFC. Therefore, it can be easier to identify the modified plant that permits much more accurate results than those obtained by identifying the original plant itself.

3.2.2 DVFC-Based Identification The system identification is applied to the experimental structure compensated by DVFC, which has the optimal damping factor. The output signal is the relative displacement u_3 between the base and the 3rd story. From eq.(4), the sampling time is selected at 0.005[sec] so

Fig. 7. Time response of input and output.

that ω_{id} can include the frequencies of three vibration modes. A portion of input and output data is shown in Fig. 7. For reference, a portion of output without DVFC driven by the same input is shown in the same figure. It is verified that the amplitude of u_3 with DVFC is smaller than that of u_3 without DVFC. Therefore, the proposed method requires no any excessive vibration to identify the system.

Regarding the system identification, two types of model and order are used to illustrate the merits of the proposed identification. Fig. 8(a) shows the gain plots when *auto-regressive exogenous*(ARX) and *Box and Jenkins*(BJ) models defined by eqs.(5) and (6) are utilized as the identified plant models without DVFC. Each gain plot is obtained with the same order of 30th. Fig. 8(b) shows the gain plots of the BJ model with the order of 20th and 30th, respectively.

To confirm the difference between with and without DVFC, on the other hand, the identification is also applied to the structure with DVFC. Figs. 9(a) and 9(b) are the counterparts of Figs. 8(a) and 8(b), respectively. From Figs. 8 and 9, the deviation between these two gain plots is smaller from 1st-mode to 3rd-mode of the model with DVFC than that of the model without DVFC. As a result, the identified model with DVFC can approach to a more real plant because of its high accuracy. Therefore, the deviation between Figs. 6 and 9 may be caused by an uncertainty of nominal parameters and unmodeled dynamics of the real structure. To evaluate the effectiveness of the proposed identification method, the authors apply the El Centro loading (decrease to Max: 0.336[N]), which was recorded at California in 1940, to the nominal model, the identified model and the experimental structure. Figs. 10(a) and (b) show the time responses with and without DVFC, respectively. To compare the time response with and without DVFC, it is confirmed that the identified model with DVFC well agrees with the experimental structure.

3.2.3 Design of Robust Control System The authors define the identified 20th-order BJ model with DVFC shown in Fig. 9(b) as the real plant P and the modified plant with DVFC composed by the nominal

(a) ARX and BJ models with 30th-order.

(b) BJ models with the 20th- and 30th-order.

Fig. 8. Gain plots of identified models without DVFC.

(a) ARX and BJ models with 30th-order.

(b) BJ models with the 20th- and 30th-order.

Fig. 9. Gain plots of identified models with DVFC.

parameters as the nominal plant P_n . Then, the multiplicative uncertainty Δ_m can be derived from Eq.(7). Fig. 11 shows the gain plot of this multiplicative uncertainty Δ_m .

The $\mathcal{H}_\infty$ controller is systematically designed for the nominal plant model P_n under the following conditions:

- (1) The control system should be stable even though unmodeled dynamics, i.e., Δ_m , exist in

(a) u_3 without DVFC

(b) u_3 with DVFC

Fig. 10. Experimental results of time response for El Centro NS loading.

Fig. 11. Gain plot of the multiplicative uncertainty Δ_m .

the minor feedback system.

- (2) The amplitude against the disturbance should be as low as possible.
- (3) The controller gain at low frequency range below the first vibration mode is sufficiently small because of limitation of the actuator stroke.

From the first condition, at first, the weighting function W_2 , which is for the complementary sensitivity function T , should satisfy the following:

$$|\Delta_m| < |W_2(j\omega)|, \quad \forall \omega \quad (10)$$

$$|T(j\omega)| < |W_2(j\omega)|^{-1}, \quad \forall \omega \quad (11)$$

Since $T = 1 - S$, S is the sensitivity function, the authors can make S small enough to satisfy the second

Fig. 12. Generalized plant for $\mathcal{H}_\infty$ control system.

Fig. 13. Gain plots of the $\mathcal{H}_\infty$ controller.

condition by setting T close to the uncertainty in which turn W_2 may approach to the uncertainty. In this case, the gain of the weighting function $W_2 = 5$ is larger than that of Δ_m at any frequency range. Moreover, considering the earthquake loading as an external disturbance, in this design, the vibration suppression control of u_3 against the loading can be realized.

From the third condition, the weighting function W_1 , which is for the control force, includes no integral factor, since its dynamics is directly reflected in the $\mathcal{H}_\infty$ controller. Therefore, the weighting function W_1 is recommended to select as high-pass characteristic. In this case, W_1 is determined as $(1+s)/(1+10s)$. The generalized plant including these weighting functions is illustrated in Fig. 12. As a result of these weighting functions, the $\mathcal{H}_\infty$ controller K of the 7th-order can be obtained. The gain plot of the $\mathcal{H}_\infty$ controller is shown in Fig. 13. It is verified that the gain below the first mode frequency is sufficiently small, and the gain at the third mode is big enough to improve the vibration suppression performance, which is not much improved by DVFC.

Fig. 14 shows the gain plot of the $\mathcal{H}_\infty$ control system from f_c to u_3 . Fig. 14 also shows the gain plots of original structure and the structure modified by DVFC. Excellent vibration suppression performance can be obtained by the $\mathcal{H}_\infty$ controller over all the frequency. Moreover, Figs. 15 (a), (b) and (c) show the gain plots for $u_1/\ddot{u}_g$, $u_2/\ddot{u}_g$ and $u_3/\ddot{u}_g$, respectively. In each figure, the thick line is obtained for the $\mathcal{H}_\infty$ control system, while the thin line for the original plant. From Fig. 15 (c), it is verified that the gain u_3 of the control system is smaller than that of the original plant over almost the

Fig. 15. Gain plots from $\ddot{u}_g$ to u_1, u_2 and u_3 . Fig. 14. Gain plots of the H_∞ control system.

frequency, in which DVFC is not effective well enough. From Figs.15 (a) and (b), the gain plots of the control system is higher than those of the original plant at anti-resonance frequency because of not considering u_1 and u_2 as the controlled value in the design of the H_∞ controller. However, the controller can suppress the gain of vibration modes.

4. Experiments

Fig.2 shows the experimental setup including the active control unit. The output of the gap sensor is introduced into a digital signal processor and the calculation of the controller is implemented. The controller designed in Section 3.2.3 is digitized by the zero-order hold method with the sampling time 0.001[sec]. The output of the controller is supplied to the linear motor through the driver.

In order to demonstrate an application of the proposed identification method to robust vibration suppression control of flexible structures, the authors apply the El Centro NS loading to the experimental structure which has been previously modified with this method. Fig.16(a) shows the time response u_3 of the modified structure without the H_∞ controller. Fig.16(b) shows the time response u_3 of the modified structure with the H_∞ controller. From these figures, it is understood that the robust vibration suppression control can be performed by the H_∞ controller not only to stabilize the system but also to improve the vibration suppression

(a) the modified structure.

(b) the modified structure with the H_∞ controller.

Fig. 16. Experimental results of time response for El Centro NS loading.

performance.

5. Conclusion

The authors verified the effectiveness of the proposed direct velocity feedback control(DVFC)-based identification by demonstrating the experimental results primarily for a typical earthquake(El Centro NS) loading and additionally for vibration suppression control of a flexible structure. The advantages of the proposed identification method were given as follows:

- (1) An original plant could be easily modified and stabilized by introducing the minor feedback control.
- (2) DVFC as minor feedback control was suitable for the identification approach of flexible structures.
- (3) The modified structure could be identified with higher accuracy.
- (4) The conservativeness for the bandwidth of the vibration suppression control could be improved.

- (5) Ideal vibration suppression control could be performed by the $\mathcal{H}_\infty$ control design with DVFC for flexible structures.

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References

- (1) H. Nishimura, K. Nonami and O. Nakada, " $\mathcal{H}_\infty$ Control with Pole Assignment in a Specified Region for Multi-Degree-of-Freedom Structure Using Active Dynamic Vibration Absorber," *Trans. JSME(C)* (in Japanese), Vol. 61, No. 584, pp. 87-94, 1995.
- (2) M. Shimakage, A. Moran, M. Nagai and M. Hayase, "Design of Vibration Suppression $\mathcal{H}_\infty$ Control System for High-Rise Buildings," *Proc. of International Power Electronics Conference (IPEC-Yokohama'95)*, vol. 1, pp. 464-469, 1995.
- (3) H. Amano and K. Nonami, " $\mathcal{H}_\infty$ Vibration Control of Flexible Truss Structure by Control Momentum Gyro," *Proc. of the Third International Conference on Motion and Vibration Control (MOVIC-Chiba'96)*, pp. 268-273, 1996.
- (4) H. Kagaya, T. Tamaki and Y. Suzuki, "Experimental Study on Active Seismic Response Control of Full-Scale Steel Structure Using $\mathcal{H}_\infty$ Control," *Proc. of the 5th Symposium on MOVIC*, pp. 78-81, 1997.
- (5) H. Nishimura, "Control Design for Active Vibration Suppression and Vibration Isolation of Structural Systems," *1998 National Convention Record IEEEJ* (in Japanese), S.24-1, 1998.
- (6) T. Kimura, S. Hara and T. Tomisaka, " $\mathcal{H}_\infty$ Control with Minor Feedback for a Pneumatic Actuator System," *Proc. of the Control and Decision Conference (CDC-Kobe'96)*, pp. 2365-2370, 1996.
- (7) L. Ljung and T. Glad, "Modeling of Dynamic Systems," PTR Prentice Hall, Englewood Cliffs, NJ, 1994.
- (8) W. Gawronski, "Balanced Control of Flexible Structures," Springer, 1996.
- (9) J. C. Doyle, B. A. Francis and A. R. Tannenbaum, "Feedback Control Theory," Macmillan Publishing Company, 1992.
- (10) S. Hashimoto, K. Hara, H. Funato and K. Kamiyama, "Robust Vibration Suppression Control using System Identification of Minor Control-Loop System," *Proc. of the Power Conversion Conference (PCC-Nagaoka'97)*, Vol. 2, pp. 607-612, 1997.

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