On the Suppression of Ground Vibration by Active Force Controller *

(Suppression of Exciting Force by Feedforward Control)

By Nobuo TANAKA**

Yoshihiro KIKUSHIMA**

For the purpose of suppressing the ground vibration as public nuisance produced by vibrating machines such as a forging machine, a press machine, etc., this paper presents a new type active force controller. First, the principle of the active force controlling method is shown, and the system equations are derived. Secondly, the characteristics and the design parameters of the active force controller are presented. Thirdly, from a point of view of feedforward control method, three kinds of the dynamic compensators are designed so as to obtain sufficient effectiveness of suppressing the exciting force. Finally, in order to demonstrate the control effect, an experiment is carried out, and then the active force controller is proved to be applicable for suppressing the ground vibration as public nuisance.

1. Introduction

These days, realization of a technique to eliminate ground vibration as public nuisance is in great demand. Among the countermeasures proposed so far, supposedly the most effective method is regarded as the one to rule out the vibrating source itself. The countermeasure of this kind is mainly based on an elastic support method by bolstering a vibrating machine like a forging machine, a press machine, etc., with elastic materials such as coil spring, belleville spring, air suspension and so on. This method can be realized by simple mechanism; however, it contains undesirable aspects as well. For instance, as a result of applying the elastic support to a vibrating machine, the machine itself also vibrates, so that dimensional accuracy, machine life and productivity are not only deteriorated, but even the machine itself turns out to be an oscillating source which causes ground vibration as public nuisance. Furthermore, this method can never be applied to the case where a machine allows no movement.

Recently, some papers report on the vibration control technique of applying a dynamic damper to the foundation of a vibrating machine. Like the method described above, this idea can also be realized rather simply, so that it is applicable in a practical sense. However, before it can be sufficiently effective, this method carries many problems to be solved such as:

- i) A mass ratio of the damper to a control object must be large.

- ii) A frequency region to be controlled effectively is very limited.

- iii) Accurate damper parameters must be achieved.

In order to overcome the difficulties mentioned above, this paper proposes a new type vibration controller - an active force controller - which is used as a control element and makes it possible to support a vibrating machine rigidly. In other words, the machine no longer moves. This paper also aims at an active control of ground vibration as public nuisance. As for a design of control systems, this paper adopts a feedforward control method from a viewpoint of stability of the system. Furthermore, the result of the experiment of the active force controller is reported.

Nomenclature

M_d	: auxiliary mass
f_d	: inertia force
X_d	: displacement of auxiliary mass
f_{in}	: exciting force
A	: area of piston
D_0	: viscous damping coefficient between piston and cylinder
f_0	: piston force
g_m	: mutual conductance of servo valve
r_p	: inner impedance of servo valve
V	: volume of oil at pressure P
β	: oil bulk modulus
K_A	: servo amplifier gain
G_{eq}	: feedback gain of minor loop
X_s	: displacement of spool

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** Technical Official
Mechanical Engineering Laboratory,
1-2 Namiki Sakuramura Niiharigun
Ibaraki-ken

- q : oil flow rate
 p : oil pressure
 i_v : driving current
 K_f : sensitivity of force transducer
 G_v : transfer function of servo valve
 G_c : transfer function of dynamic compensator
 u : control input

2. Active Force Controlling Method

2.1 A principle of active force controlling method

By using a powerful vibrating machine such as a forging machine or a press machine as a control object, the principle of an active force controlling method is described. As shown in Fig.1, an exciting force f passes through a contact part between a machine structure and a foundation. Then, the exciting force works on the foundation as components, $f_1, 2, 3, \dots$. As a result, these components now excite the adjacent soil as well as the foundation. Then the vibration is propagated through the ground as a form of wave and finally causes a ground vibration as public nuisance. As is generally known, this is the typical pattern of the occurrence of the ground vibration as public nuisance.

The principle of the active force controlling method places its emphasis on the point that the exciting force causing the ground vibration must pass through the contact part between the structure and the foundation. When the flow of the force in the contact part is blocked before it reaches the foundation, the foundation is no longer excited, so that the ground vibration can be eliminated. Fig.2 shows a conceptual picture of the active force controller setup to control the force components in the very contact place. Namely, the principle of the active force controller is as follows; when an auxiliary mass of the active force controller is driven according to a control signal,

Fig.1 Flow of exciting force

an inertia force is produced. Using the inertia force, the active force controller exerts an active effect of suppressing the exciting force. As for the control signal, it is gained as the output of a dynamic compensator designed to make the active controller work efficiently. On the other hand, an input of the dynamic compensator is obtained from a force transducer attached to the contact part or is gained by detecting the exciting force directly.

Aiming at the suppression of a powerful exciting force produced by such a forging machine and a press machine, the active force controller adopts an electrohydraulic servo mechanism with a view to getting high-speed response and high-power gain.

2.2 Description of active force controller system

According to the figure shown in Fig.2, the equations of motion are derived as follows:

The equation of motion of the active force controller is given by

$$M_d \ddot{x}_d(t) = f_d(t) = AP(t) - D_0 \dot{x}_d(t) \quad \dots\dots\dots (1)$$

The relation between the oil pressure and the oil flow rate is given by

$$q(t) = g_n x_s(t) - \frac{1}{r_p} P(t) \quad \dots\dots\dots (2)$$

Taking into consideration the compressibility of oil, eq.(2) becomes

$$q(t) = A \dot{x}_d(t) + \frac{1}{2} V \beta \dot{P}(t) \quad \dots\dots\dots (3)$$

The piston force is given by

$$f_0(t) = AP(t) = f_d(t) + D_0 \dot{x}_d(t) \quad \dots\dots\dots (4)$$

Fig.2 Principle of active force controller

The relation between the spool displacement and the driving current of the servo valve is

$$x_v(s) = K_V/s; i_v(s) \quad (5)$$

where S means the Laplace operator. Using the input voltage, the driving current is given by

$$i_v(s) = K_A u(s) \quad (6)$$

The Laplace transform of eq.(2) becomes

$$q'(s) = K_A G_V/s; u(s) - \frac{1}{r_P} P; s; \quad (7)$$

where

$$G_V(s) = g_v K_V/s \quad (8)$$

The relation between the exciting force and the output voltage is given by

$$e(t) = K_f f_{in}(t) \quad (9)$$

Putting $G_c(s)$ as a transfer function of a dynamic compensator, the input-output relation of the compensator is given by

$$u(s) = G_c(s) e(s) \quad (10)$$

The position feedback from X_0 is needed to get an equilibrium between the piston and the cylinder of the controller and it is given by

$$\tilde{e}(t) = -G_{eq} x_d(t) \quad (11)$$

Using eq.(1) to (11) gives a block diagram of the active force controller system in Fig.3.

3. Design of the Active Force Controller System by means of Feedforward Control

A block diagram shown in Fig.4 gives a conceptual flow to eliminate the ground vibration by the active force controller

from a point of view of feedforward control; that is, the exciting force on the foundation is to be controlled by an output of the active force controller, f_d , before it reaches the foundation. Therefore, the purpose of this chapter is to design a dynamic compensator which gives the optimum control effect in the suppression of the exciting force.

3.1 Design of a dynamic compensator by phase shifter

As the first step to designing a dynamic compensator, this section shows a function of the active force controller driven by adjusting a phase and a gain of the exciting force arbitrarily. To satisfy this property, the dynamic compensator is designed by P-I action as shown in Fig. 5, which is well-known in classical control theory. Namely, the transfer function is given by

$$G_c(s) = \left(\beta_1 + \frac{\beta_0}{s + \omega_0} \right) \quad (12)$$

where ω_0 means a break point frequency of an approximate integrator, and it is set at 1 Hz in this paper. Since the phase of the fundamental signal is adjusted freely, the authors call this a phase shifter

Fig.4 Block diagram for suppressing the ground vibration by feedforward control

Fig.3 Block diagram of active force controller system

method. Optimum values of the parameters β_0 and β_1 are shown in Fig.6, giving the maximum control effect at each frequency. Fig.7 shows the frequency characteristics of the suppression of the exciting force depicted at every 5 Hz in the region between 5 and 30 Hz. As shown in the figure, more than 40 dB in the reduction is attained at the set frequency in any case. In addition, the controllable frequency range tends to be wider as the set frequency becomes higher. For instance, as for the frequency range where the control effect is obtained with more than 10 dB, it is very narrow in the case of the set frequency 5 Hz; that is, 4.2 ~ 5.7 Hz; however, in the case of 30 Hz, the range becomes wider and it is from 19.5 to 43 Hz. On the contrary, an increase of the gain in the higher frequency range is seen in any case. Especially in the case where the frequency is set at less than 10 Hz, this tendency becomes conspicuous. The reason for this phenomenon is considered to be in the use of P-action of the phase shifter, since it keeps feeding the control signal proportional to the fundamental one into the active force controller even when the phase of f_d and that of f_{in} come into phase in the higher frequency range.

Fig.5 Dynamic compensator by phase shifter method

Fig.6 Optimum parameters versus frequency

3.2 Experiment

3.2.1 Experimental setup

Fig.8 shows a picture of the experimental setup. As shown in the figure, an exciter with the maximum force 8 kN is placed in the middle of a bed whose dimension is 1 m x 1.5 m. On both sides of the exciter, a pair of active force controller with the maximum force 4 kN are set. An auxiliary mass M_d of the controllers is added up to 40 kg, each piece weighing 10 kg. Each corner of the bottom of the bed is fixed to the foundation with a force transducer which is capable of detecting the exciting force with the maximum measuring force 60 kN and the threshold 10 mN. Fig.9 shows the frequency properties of the ratio of the exciting force f_{in} to the exciting acceleration $\ddot{x}$. These data are obtained under the condition that the exciting force is tuned to 6 kN at 50 Hz. As shown in the diagram, a resonance caused by the bed and the supporting parts appears at 90 Hz. In the lower part of Fig.9 is shown the frequency response of the acceleration of the active force controller with an auxiliary mass 40 kg.

Fig.7 Suppression of exciting force by phase shifter

Fig.8 Experimental setup

3.2.2 Experiment of phase shifter method

In this section, the function of the active force controller driven by a phase shifter method is verified. As for the experimental condition, the exciter is operated so as to obtain 8 KN_{p-p}, 6 KN_{p-p}, and 4 KN_{p-p} under the frequency range between 10 and 50 Hz, respectively. Fig.10 shows the effectiveness of suppressing the exciting force by the active force controller. Black marks indicate the level of the exciting force, and by driving the active force controller, all of the forces in any case are suppressed under 40 N as shown with white marks. In Fig.11, the frequency characteristics of the control effect in terms of the phase shifter method are shown under the frequency range between 10 and 50 Hz. As seen in the analytical results, the phenomenon of the gain increase in the higher frequency range is recognized too, and this is particularly conspicuous in the case of the set frequency 10 Hz. As long as the amount of the gain increase is concerned, the experimental value is slightly larger than the analytical one; however, it can be said that both results show a sufficient degree of coincidence.

3.3 Performance index of control effect

For the purpose of designing the optimum dynamic compensator, this paper presents a performance index of control effect defined by taking into consideration the following aspects; that is,

- i) As long as the ground vibration as public nuisance is concerned, a lower frequency range is important because of the amplitude reduction with distance.

Fig.9 Frequency characteristics of exciter and active force controller

- ii) A man is most sensitive to the vertical vibration in the frequency range between 4 and 8 Hz.
- iii) The performance index should be simple and clear.

Therefore, this paper defines the performance index of suppressing the exciting force as

$$I = \int_{\omega_1}^{\omega_2} 20 \log \left| \frac{Af(j\omega)}{f_{in}(j\omega)} \right| \omega d\omega \dots\dots\dots (13)$$

Fig.10 Control effect by phase shifter

Fig.11 Control effect by phase shifter in frequency domain (experiment)

where ω_1 and ω_2 are 1 Hz and 20 Hz respectively. As shown in Figs.7 and 11, the notch effect with the amount of -50 ~ -60 dB is recognized in a very narrow range, so all the gain values under -30 dB are regarded as -30 dB, since the authors are aiming at the uniform control effect in a wide frequency range.

3.4 Design of dynamic compensator

The defect of the phase shifter method shown in the previous chapter is an increase of the gain in the higher frequency range and this is due to the P-action of the phase shifter method. To offset this defect, two kinds of dynamic compensators are presented in this section.

3.4.1 Dynamic compensator-type II

$$G_c(s) = \frac{\beta_1 s + \beta_0}{(s + \omega_0)^2} \quad (14)$$

Since the order of the denominator in eq.(14) is second, this compensator is called type II. As in the preceding section, ω_0 is set at 1 Hz. In Table 1, the performance indexes of the control effect

$$G_c = \frac{\beta_1 s + \beta_0}{(s + \omega_0)^2}$$

I	β_1	β_0
-54.2	0	0.1100
-85.7	0.0005	0.1100
-121.8	0.001	0.1000
-162.0	0.0015	0.1000
-198.2	0.002	0.0775
-150.6	0.0025	0.0600
-94.7	0.003	0.0650
-51.3	0.0035	0.0700
-16.4	0.004	0.0725
-164.3	0.0022	0

Table 1. Performance index (dynamic compensator type II)

Fig.12 Suppression of exciting force by dynamic compensator type II (analysis)

are shown with the parameters β_0 and β_1 . Fig.12 shows the suppression of the exciting force applied with the optimum parameters ($\beta_0 = 0.0775$, $\beta_1 = 0.002$). The performance index in this case is -198.2. For instance, in case the uniform suppression of the exciting force with -10 dB is obtained in the frequency range between 1 and 20 Hz, then the performance index is calculated to be -190. Considering this value, the control effect by the index I = -198.2 is found fairly high. In addition, dynamic compensator - type II is superior to the phase shifter in the controllable frequency range. As for the control effect in the frequency range between 4 and 8 Hz, -6 ~ -20 dB are gained. Fig.13 shows the experimental result obtained by using type II - dynamic compensator. In the process of the experiment, the combination of the parameters to obtain the best control effect turned out to be slightly different from that of the analytical result. In the experiment, the optimum parameters were found to be $\beta_1 = 0.0021$ and $\beta_0 = 0.0277$, and so the value of β_0 is different from the analytical one; however, as shown in Figs.12 and 13, the suppressions of the exciting force show a high degree of coincidence.

$$G_c = \frac{\beta_2 s^2 + \beta_1 s + \beta_0}{(s + \omega_0)^3}$$

I	β_2	β_1	β_0
-55.4	0	0.115	0
-85.8	0.0005	0.115	0.375
-121.3	0.001	0.115	0.750
-162.3	0.0015	0.115	1.125
-211.5	0.002	0.110	1.500
-257.5	0.0025	0.105	2.000
-212.7	0.003	0.110	2.500
-155.0	0.0022	0	0

Table 2. Performance index (dynamic compensator type III)

Fig.13 Suppression of exciting force by dynamic compensator type II (experiment)

3.4.2 Dynamic compensator - type III

$$G_c(s) = \frac{\beta_2 s^2 + \beta_1 s + \beta_0}{(s + \omega_0)^3} \quad (15)$$

The control effect of the dynamic compensator-type III is discussed here. Obviously this case has one more parameter than type II. Table 2 shows the performance index with the parameters β_0 , β_1 , and β_2 . As shown in the table, the combination of the optimum parameter is $\beta_2 = 0.0025$, $\beta_1 = 0.105$ and $\beta_0 = 2.0$. In this case, the performance index is -257.5 and this is much improvement on type II with -198.2. As an example, the relation of the performance index and the parameters β_0 , β_1 and β_2 is presented in Fig.14. Judging from this figure, it seems to be practical to apply these parameters to the compensator because of a slight change of the performance index in the vicinity of the minimum. Fig.15 shows the frequency characteristics of the optimum dynamic compensator-type III. Compared with the frequency characteristics of the phase shifter whose gain is flat in the higher frequency range, this one has a decrease of -20 dB/dec in the field of more

Fig.14 Parameters β_0 , β_1 , β_2 and performance index

Fig.15 Frequency characteristics of dynamic compensator type III

than 10 Hz, so that it is expected that the phenomenon of the gain increase in the higher frequency range will be controlled. Fig.16 shows the frequency characteristics of the suppression of the exciting force with the parameters $\beta_2 = 0.0025$, $\beta_0 = 2.0$ and $\beta_1 = 0.06 \sim 0.12$. Unlike type II, this one has two notches. As seen in the figure, when the parameter β_1 is set smaller, these two notches separate away and the control effect in the middle frequency range between the two notches becomes poor. On the contrary, by making β_1 larger, two notches are brought closer and finally collide at $\beta_1 = 0.12$, and the shape becomes like that of type II. As for the control effect in the middle frequency range, it becomes higher as the two notches approach each other and a minimum index exists in the process.

Fig.16 Suppression of exciting force by dynamic compensator type III (analysis)

Fig.17 Suppression of exciting force by optimum dynamic compensator type III (analysis)

Fig.17 shows the frequency characteristics of the suppression of exciting force with the optimum parameters. In this case, the notches are located at 3 Hz and 7.2 Hz, and the suppression of the exciting force by -20 ~ -40 dB is gained in the range between 4 and 8 Hz, so that the control effect in terms of the type III-dynamic compensator is satisfactory from a viewpoint of the suppression of the ground vibration as public nuisance. On the other hand, in this case a gain increase by 3.85 dB at 89 Hz appears; however, as long as the ground vibration as public nuisance is concerned, this phenomenon is not a big problem because the vibration level in such a high frequency decays rapidly in the propagation of the ground vibration.

Fig.18 Dynamic compensator type III simulated in analog computer

Next are shown the experimental results in terms of type III. The dynamic compensator is realized by an analog computer as simulated in Fig.18. Fig.19 shows the experimental results of the frequency characteristics of the control effect. As for the parameters set in the experiment, β_2 and β_0 are fixed at 0.002042 and 4.348 respectively and only β_1 is changed from 0.09 to 0.15. As shown in the figure, the phenomenon of the two notches which are characteristic to the dynamic compensator type III is verified experimentally. As shown in Fig.20, the maximum control effect gained in the experiment is attained by the combination of the parameters; that is, $\beta_2 = 0.002042$, $\beta_1 = 0.1382$, $\beta_0 = 4.348$, and these are different from those in the analysis. As shown in Fig.20, two notches are located at 5 Hz and 11 Hz, and these are a few hertz higher than those in the analysis.

Fig.20 Suppression of exciting force by optimum dynamic compensator type III (experiment)

Fig.19 Suppression of exciting force by dynamic compensator type III (experiment)

Fig.21 Exciting force being suppressed by active force controller

However, the depth of the notch and the control effect in the middle frequency range are the same both in experiment and in analysis. Furthermore, the fact that the two notches are brought closer by making β_1 smaller and finally come together at $\beta_1 = 0.15$ ($\beta_1 = 0.12$ in case of analysis) is demonstrated in the experiment. Fig. 21 shows the exciting force with an amplitude of 2450 Np-p being suppressed to 200 Np-p by the active force controller driven by the optimum compensator-type III. The difference of the optimum as well as the frequencies of the notches between the experiment and the analysis is considered to consist in the mathematical model used in the analysis, which carries an error with the actual characteristics. In addition, there seem to exist a non-linearity in the characteristics of a servo valve and a time lag. Furthermore, in the process of the analysis, a structure consisting of an exciter, active force controllers and a bed is assumed to be fixed to a foundation rigidly; however, as realized in Fig. 9, a resonance occurs at 90 Hz, so that strictly speaking, the assumption of the rigid support is not valid, and so an error seems to occur.

4. Conclusions

For the purpose of suppressing the ground vibration as public nuisance, this paper has presented an active force controlling method. By means of this method, the present paper has made it possible to replace the conventional spring support with a rigid support of the vibrating machine. The summary is as follows:

- 1) Based on the fact that the exciting force passes through a contact part between a structure and a foundation, an active force controlling method has been proposed which suppresses the force in the very part before it is fed into the foundation.
- 2) From a standpoint of feedforward control, a design of the active force controller system has been presented. As the first step to the optimum design of the control system, a phase shifter method has been proposed. This method consisting of an integrator and a few amplifiers is capable of controlling the exciting force in the range from 4000 to 8000 Np-p to from 10 to 40 Np-p; however, the frequency region that can be suppressed is very narrow. It has also been shown that the increase of the gain in a higher frequency range is conspicuous, especially when the set frequency is low.
- 3) With a view to designing the control system quantitatively, a performance index of suppressing the exciting force has been defined.
- 4) By using the performance index, a dynamic compensator-type II has been designed. As a result, this method has turned out to be superior to the phase shifter

method in the control effect.

5) A dynamic compensator-type III has been presented. Compared with the dynamic compensator-type II, this one has been shown to be superior in the control effect as well as in the controllable frequency range. By applying this compensator to the active force controller, two notches appear in the frequency characteristics, and the optimum parameters are determined under the influence of those frequencies.

6) To verify the control effect in terms of the dynamic compensator proposed in this paper, an experiment has been conducted, and the feasibility of the active force controlling method has been demonstrated.

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