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FUNDAMENTAL STUDY OF ACTIVE MASS DAMPER FOR SLENDER HOUSES

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SUMMARY

In Japan, there are many 3-story houses built on limited space, due to the population concentration in the cities. Slender structures such as 3-story houses are susceptible to large amplitude vibration induced by external forces. This effect frequently creates an uncomfortable environment for habitation. Because houses often have a complex geometry, it is difficult to establish the vibration characteristics of a structure. Furthermore, the response characteristics of a house will change with environment, structural condition, additions and remodeling. An active mass damper was developed and tested to address some of these issues. The direct velocity feedback control is a vibration control which induces a proportional control force on the building at absolute velocity of the building, producing a skyhook effect. The direct velocity feedback control was used for the mass damper, because it did not need dynamic models to control the house and the mass damper. Also, stability is guaranteed when the mass damper and the sensor were located in the same position. Excitation tests were carried out for an experimental model of a 3-story house equipped with the active mass damper. The tests showed that the mass damper could dramatically improve performance across a wide frequency range. Of particular note was an increase of approximately 8% damping ratio in the first mode, and a reduction of about 5dB in the vibration acceleration level.

INTRODUCTION

The deregulation of detached housing construction in Japan has led to an increase in the building of three-story steel framed houses over recent years. With the continued concentration of Japan's population in major metropolitan cities, many of these three-story houses are built on extremely narrowly confined land, resulting in unusually slender structures for a detached house. With these types of detaches houses, concerns arise as to living comfort due to traffic vibrations caused by nearby major roads, freeways, rail lines, etc. Active control is one effective vibration control method for infinitesimal vibrations such as those caused by traffic vibration. In general, most detached houses feature a complex geometrical shape. As well, many non-structural building materials are used, making it difficult to assess the vibration properties of the house. It is also conceivable that the vibration characteristics of a structure can change

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due to factors such as building usage and additions/ remodeling. For our study, we selected DVFB Control (Direct Velocity Feedback Control) [1] as a control method expected to be stable and demonstrate vibration control effects with respect to vibration systems whose vibration properties cannot be identified. The DVFB Control used in this study is a mechanism that effects vibration control on a building using control force in proportion to the absolute velocity response of the building, enacting a so-called skyhook damper. Theoretically, DVFB Control using sensors and activators installed together comprises an absolutely stable system.

For an active mass damper to be practical for detached houses, it must be compact, easily installed, and low-cost. A DVFB control can consist of a one-sensor control system, and does not require complex operation processing. Therefore, DVFB control appears to be suited for use as a control method for an active mass damper in detached houses.

To conduct a fundamental study of active mass dampers for detached houses in this report, we built a small three-layer steel frame, and studied the vibration control effect of the active mass damper relative to small amplitude vibration inputs.

TEST DEVICES

The active mass damper ("AMD") used in this research is a mass damper that incorporates a ball screw and AC servomotor to drive mass, capable of exercising control in the horizontal direction only (Figure 1). As mentioned previously, AMD for houses must be adapted for buildings having undetermined vibration characteristics. As such, in order to achieve an equivalent control effect independent of the period characteristics of a given building, the mass damper should be constructed without a righting moment, and without a built-in period. Since there is no righting moment, it is conceivable that the moving mass may not be able to maintain its original position during AMD operation; however, we designed a control program to cancel control and forcibly return the mass to the original position once it has moved past a certain amount. Table 1 lists the AMD drive section specifications. The weight of the movable mass is 5kg. The distance of the stroke is ±136mm mechanically; however, a programmed limit of ±100mm was used. An acceleration sensor was installed at the top of the building, which served as the only sensor comprising the control system.

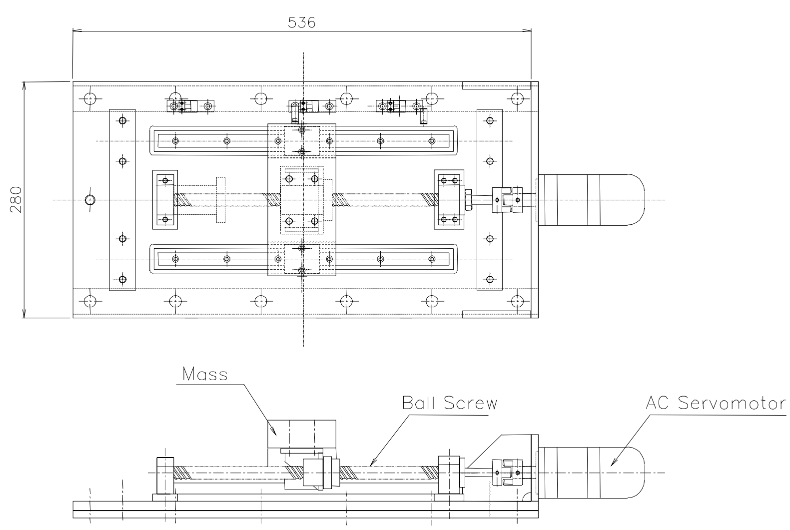


Fig.1:Active Mass Damper

Table.1:AMD Drive Specification

Weight	Weight	5 kg
	Max.Stroke	272 mm(200mm programmed limit)
Motor	Power	AC 100 V
	Output	400 W
	Max. Speed	0.5 m/s
Ball Screw	Diam.	15 mm
	Lead	5 mm

The frame model for tests simulated a three-story steel framed house. The structure was built with three layers of 1,000mm x 1,000mm H-section beams connected by four flat springs (Figure 2).

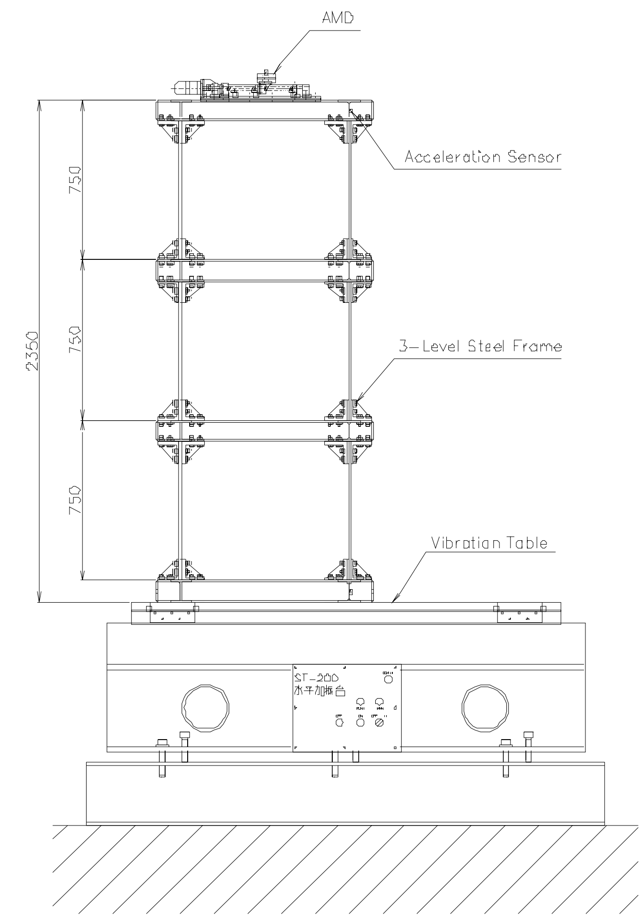


Fig.2:Test Device

Each level was 750mm in height, for a total of 2,350 mm, and the entire structure weighed approximately 700kg. Although it was not a built as a miniature model replicating a house in detail, we referenced the vibration properties of an actual steel-constructed three-story house[2)], and designed the flat springs for a primary characteristic frequency of around 4.5Hz.

Table.2:Test Model Modal Parameters

		1st Mode	2nd Mode	3rd Mode
Modal Mass (ton)		0.269	0.370	1.019
Modal Stiffness (kN/m)		222.2	2497	14864
Modal damping (%)		0.018	0.010	0.005
Natural Frequency (Hz)		4.57	13.1	19.2
Natural Period (s)		0.219	0.077	0.052
Modal Vector	3	1.00000	1.00000	1.00000
	2	0.71741	-0.36997	-1.69474
	1	0.42282	-1.05779	1.31247
Modal participation Factor		1.3160	-0.3612	0.0461

An AMD device was installed at the top of the test frame model, and placed the model on a horizontal vibration table, the vibration tests were performed according to random vibration. An overview diagram of the experimental system can be seen in Figure 3. Signals from the acceleration sensor at the top of the structure were captured by computer, which calculated control commands for moving the AC servomotor.

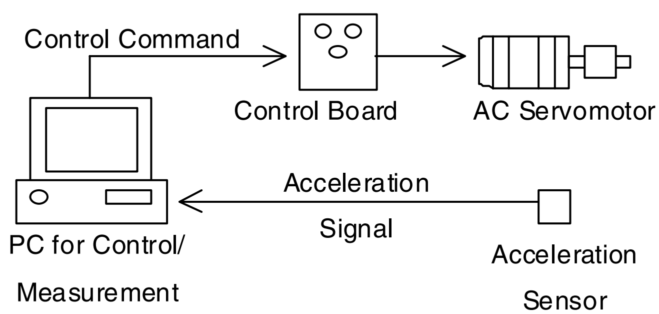


Fig.3:Active Control System

ANALYTICAL MODEL AND CONTROL METHOD

A three-layer test frame model was identified, and the vibration properties were searched for. The parameters for each mode are shown in the table below. The mass of the AMD movable mass was 5kg, which meant that the mass ratio of the first mode was approximately 1.8%. An analytical model was established based on this identified result. Figure 4 shows the transmission function between the input vibration and the acceleration response at the top of the structure for the test frame model and the analytical model.

The motion equation of the experimental model with the AMD installed on the third level of the structure is as follows:

$$[m][\ddot{x}]+[c][\dot{x}]+[k][x]=-[m][1]\ddot{z}+[s]u \qquad (1)$$

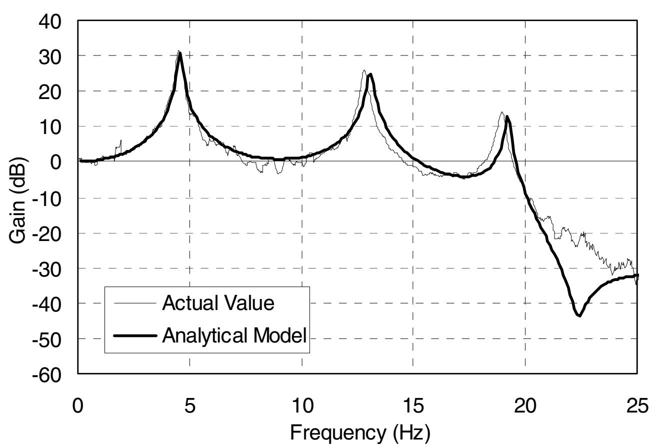
Here, [m], [c], [k] represent the respective model mass, attenuation coefficient, and rigidity matrix. represents the input disturbance, while u represents the control input, and [s] is equivalent to the vector [s]=[0 0 1]T , indicating the level on which the AMD is installed. As control input u is set proportionate to the absolute velocity of the level on which the AMD is installed, then if the absolute velocity response vector is,

$$u = Fg \cdot [s]^T [\dot{x}_a] \qquad (2)$$

Here, Fg represents the control gain. Meanwhile, if the mass of the movable mass is md , and displacement is xd , the motion equation is as follows:

$$m_d \ddot{x}_d = u \qquad (3)$$

The control system for the AMD employs only an acceleration sensor installed at the top of the structure. This integral calculus for this acceleration signal is calculated by computer, converting the result to an absolute velocity signal, which is used as a control signal, taking advantage of control gain Fg. If a drift component is included in the integrator, a divergence will occur; therefore, employing a Band Pass Filter (BPF) on the acceleration signal prevented drift and oscillation. The Band Pass Filter consisted of one secondary high pass filter of a 0.8Hz cut-off frequency, and one primary low pass filter of a 30Hz cut-off frequency. Each of the filters employed Butterworth characteristics. Control gain Fg was determined based on parameter studies. Theoretically, the larger the Fg is set for DVFB control, the greater the vibration control effect, generating absolute stability without spillover across all frequency ranges. In addition, the Fg should be able to be set to keep the stroke of the movable mass and output capacity for the servomotor within the permissible range. However, in actual practice, phase discrepancies due to control lag and filter processing caused deterioration in control performance.
Figure 5 shows the above-mentioned control methods compiled in a block diagram. Simulations using the analytical model were also conducted based on this block diagram.



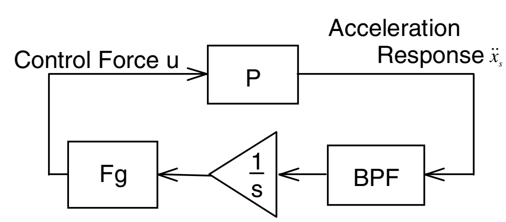


Fig.4:Frequency Response (Actual Value and Analytical Model) **Fig.5:Control System Block**

TEST RESULTS AND OBSERVATIONS

The input vibration was used to simulate traffic vibration, but in much the same way as with houses, there was an extremely large number of uncertain factors. In the experiments conducted in this research, we utilized white noise for purposes of testing the control effect related to vibration over a wide ranging period band. The input vibration wave is shown in Figure 6. The input command to the vibration table enacted a displacement wave, with Figure 6 showing the observed acceleration wave exerted directly on the vibration table. The r.m.s. value was 0.040 m/s^2.

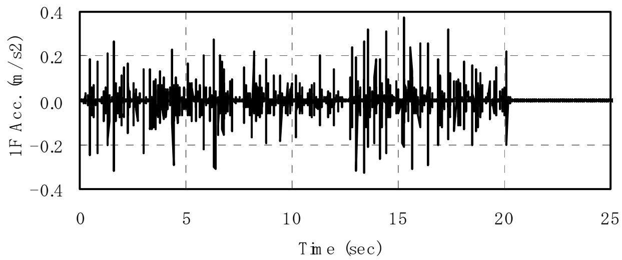


Fig.6:Input Acceleration

Vibration control Effect

The AMD vibration control effect was verified as feedback gain Fg=2.0 kN・s/m. Figures 7-1 through 7-3 show the respective time histories of the response acceleration for without control at the top of the structure during AMD non-operation, the response acceleration at the top of the structure with control during AMD operation, and the associated movable mass displacement. The response acceleration of the top of the structure during without control was 0.32 m/s2, with an acceleration response r.m.s. value of 0.094 m/s2. In contrast, the greatest response acceleration during with control was reduced to 0.16 m/s2, with an acceleration response r.m.s. value of 0.031 m/s2. At this time, the greatest displacement for the AMD movable mass was 16.2mm. Evaluating the transmission function between input vibration and response at the top of the structure (Figure 7-4) shows that in comparison to without control, values for the first mode, second mode and third mode showed attenuation of approximately 8%, 6% and 1%, respectively.

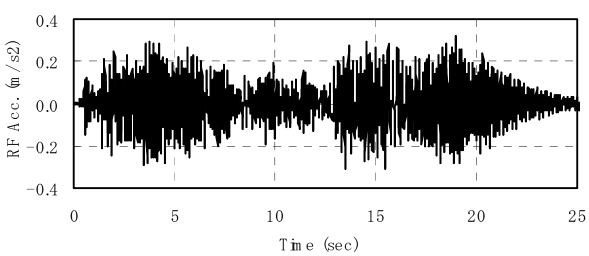


Fig.7-1:Response Acceleration Without Control

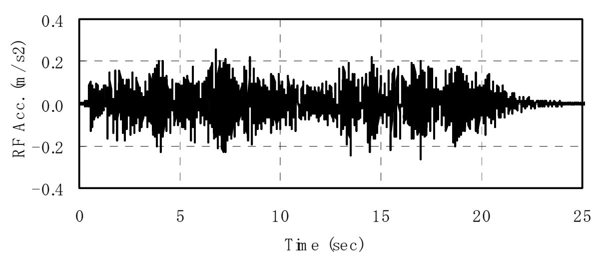


Fig.8-1:Response Acceleration Without Control (Simulation)

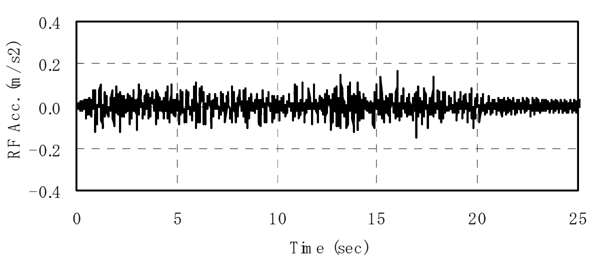


Fig.7-2:Response Acceleration With Control

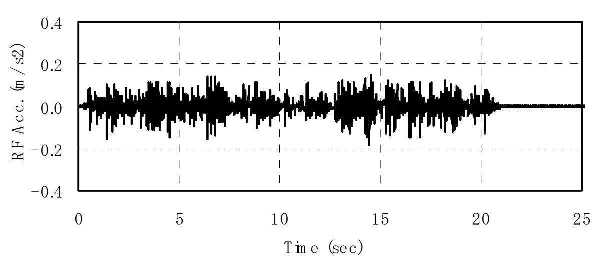


Fig.8-2:Response Acceleration With Control (Simulation)

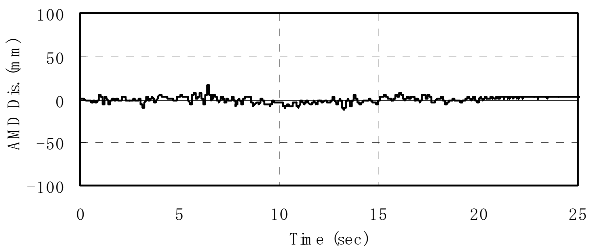


Fig.7-3:AMD Displacement With Control

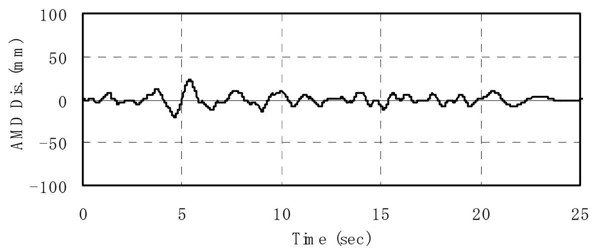


Fig.8-3:AMD Displacement With Control (Simulation)

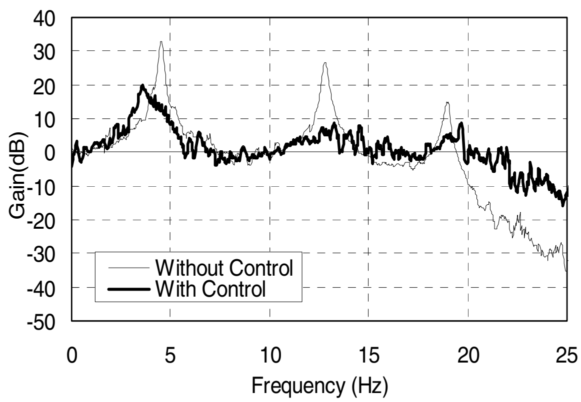


Fig.7-4:Transfer Function

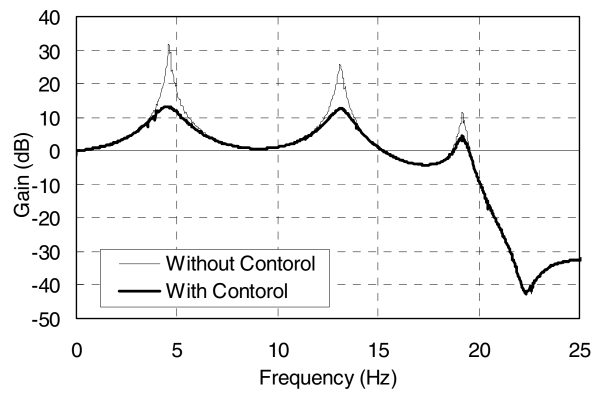


Fig.8-4:Transfer Function (Simulation)

Figures 8-1 through 8-4 show the results of conducting simulations with the analytical models under the same conditions. A comparison of time history responses leads us to conclude that values of both the response acceleration at the top of the structure and the AMD movable mass displacement virtually reproduce our test results exactly. Looking at the transmission function, the attenuation for each vibration mode is almost an exact match to that of our tests; however, there is an indication of differing characteristics in ranges above 20Hz. Under our simulations, the DVFB control executed correctly, representing a stable control system across all frequency ranges. However, during our experiments, we observed an amplification for ranges greater than approximately 20Hz in comparison to non-vibration control. This phenomenon is mentioned later.

Evaluations via Vibration Acceleration Level

Vibration acceleration level (in dB units) is the most common way to evaluate traffic and other environmental vibration. Vibration acceleration level L_a is

$L_a = 20 \log (A/A_0)$ [dB] (4)

A is the effective vibration acceleration value (r.m.s. value), and A_0 is the standard value of the vibration acceleration level, $A_0=10^{-5}$ m/s^2. Further, the human body's sense of vibration magnitude differs greatly based on the size of the vibration acceleration level, vibration frequency and vibration direction. The human body is most sensitive to horizontal vibration at a 1 to 1.6Hz range. Above this frequency, the sensitivity decreases. To evaluate vibration levels considering the sensitivity of the human body to vibrations, vibration sensation correction characteristics according to frequency band have been determined as shown in Figure 9[3),4)]. This research here deals only with horizontal vibration, and therefore we utilized vibration sensation correction characteristics with respect to horizontal vibration. If the vibration sensation correction is set C_R, the vibration acceleration level, taking vibration sensation into account, is represented by the following formula:

$$L=L_a+C_R \qquad (5)$$

Performing an evaluation of the experiment results shown in Figure 7 according to vibration acceleration level give results as shown in Figure 10. The overall vibration level is a sum of the vibration acceleration level for each frequency band, and is equivalent to the overall vibration acceleration level. Where the overall vibration level for non-vibration control was 72.7 dB, performing vibration control via AMD results in a level of 68.1 dB, which is a 4.6 dB comparative reduction.

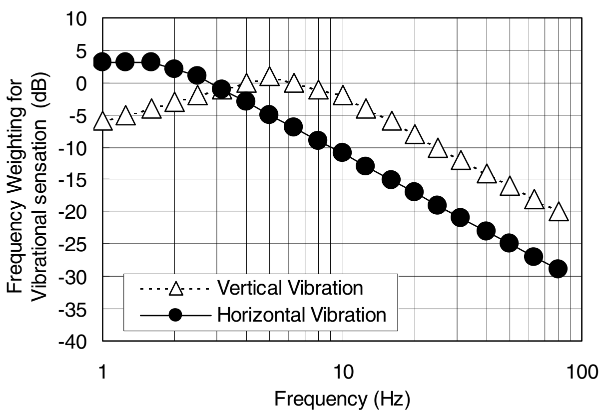


Fig.9: Frequency Weighting for Vibrational sensation

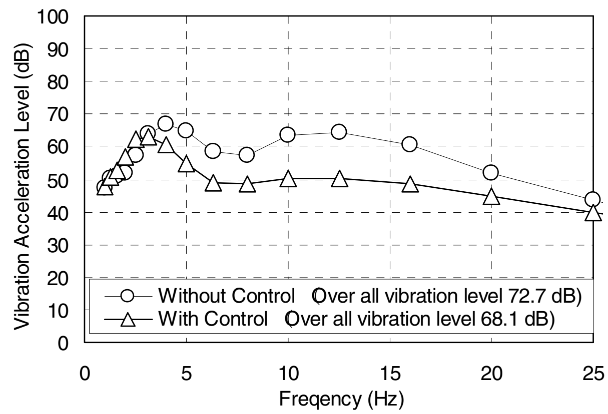


Fig.10: Vibration Acceleration Level

Parameter Study

We conducted a parameter study to test the vibration control effect/ characteristics for the AMD. The input vibration amplified the wave in Figure 6, and the r.m.s. values were approximately 0.04, 0.08, 0.12, 0.16 m/s^2. Input levels were set both weaker and stronger than the input wave amplitude(r.m.s. value of about 0.08 m/s^2) that resulted in a 70dB output at the vibration acceleration level at the top of the structure during without control. We designated input levels from 1-4 in order from smallest input wave amplitude to greatest. In addition, we changed the control gain Fg from 0, to 0.2, 1.0, and 2.0 kN・s/m. As mentioned earlier, the Fg amount was determined by trial and error, beginning by confirming smaller values first. Fg=0 represents non-vibration control.

Figures 11-1 to 11-4 plot the response acceleration at the top of the structure (first Y axis) and the maximum value of the AMD movable mass displacement (second Y axis) with respect to control gain Fg (X-axis) for input levels 1 through 4. The acceleration response was evaluated at the maximum value and the r.m.s. value. Each test condition was conducted vibration trials two times, and the maximum acceleration response and maximum AMD movable mass displacement values had the dispersion.

This means that the larger the Fg, the stronger control is exerted. When input vibration is at Level 1, the response accompanying increases in control gain Fg is held in check. When input is increased and control gain is raised, the response is held in check until the point where control is applied at a certain level of control gain, at which point we see a tendency for the response to increase. When the control gain is large, the AMD amplitude is also large, and moves rapidly. The load driving the AMD also becomes larger. As the AMD is driven forcefully, one may conclude that the AMD itself can become a source of vibration.

Figures 12-1 to 12-2 show the transmission function for the system changing the control gain. Within the frequency band of the first mode and second mode, strong control is exerted where Fg is equivalent or proportionally large. However, at ranges greater than about 20Hz, a larger Fg results in amplifying the response. From this, we believe that greater than 20Hz is a range for which control delays and hardware factors (mass driving vibrations) prevent control, acting as a negative influence on vibration control performance. However, evaluating Figure 12 with vibration acceleration levels results in a reduction of vibration acceleration level to less than 55dB for ranges in excess of 20Hz, as seen in Figure 13. A vibration acceleration level of 55dB is at the edge of the human body's ability to sense vibration. The human body cannot feel vibrations at levels under 55dB [3), 4)]. Considering that the input vibrations used in our experiments to understand AMD characteristics were at greater amplitudes than actual traffic vibration, the phenomenon occurring at ranges greater than 20Hz should not greatly affect AMD used to perform vibration control in houses.

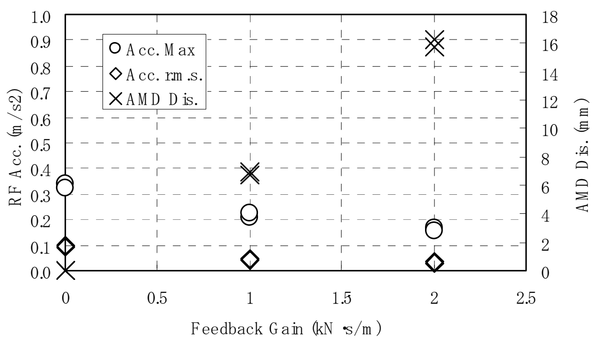


Fig.11-1:Input level 1(Acc r.m.s 0.04m/s2)

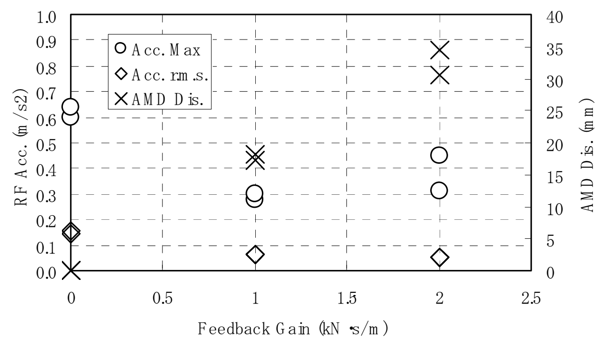


Fig.11-2:Input level 2(Acc r.m.s 0.08m/s2)

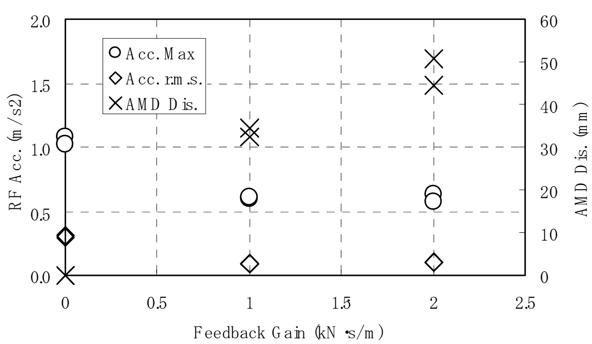


Fig.11-3:Input level 3(Acc r.m.s 0.12m/s2)

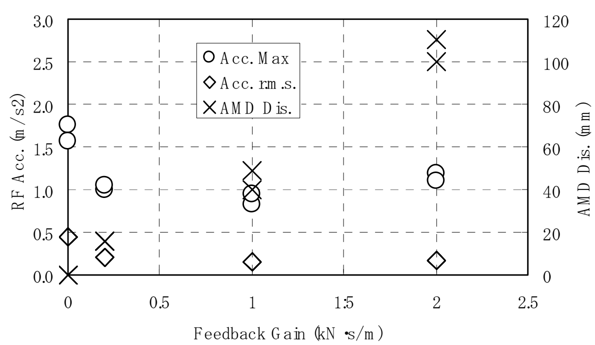


Fig.11-4:Input level 4(Acc r.m.s 0.16m/s2)

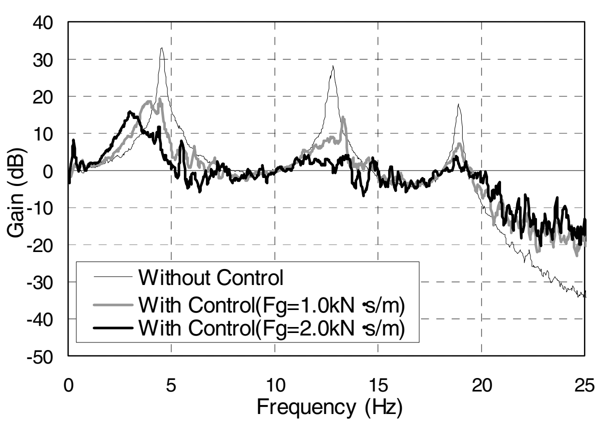


Fig.12-1: Transfer Function on Control Gain (Input level 1)

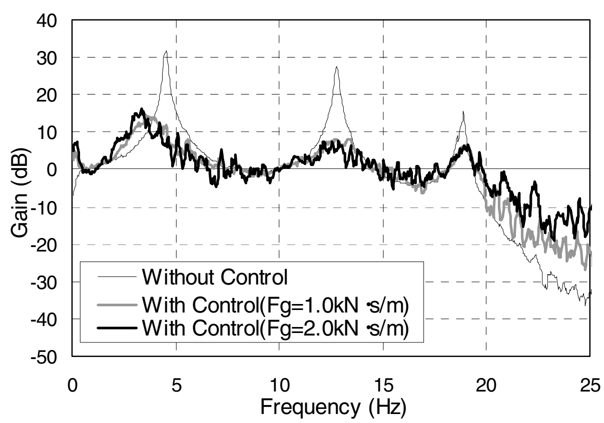


Fig.12-2: Transfer Function on Control Gain (Input level 2)

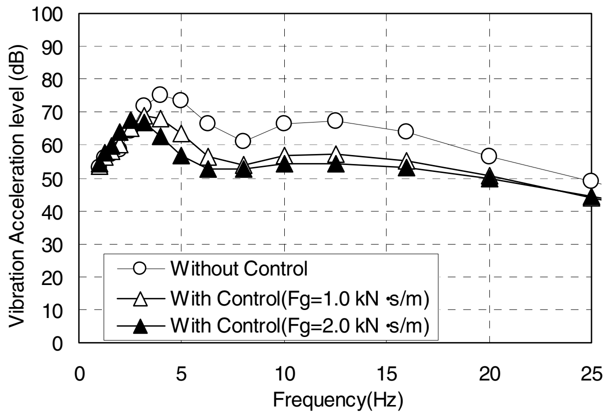


Fig.13-1: Vibration Acceleration Level on Control Gain(Input level 1)

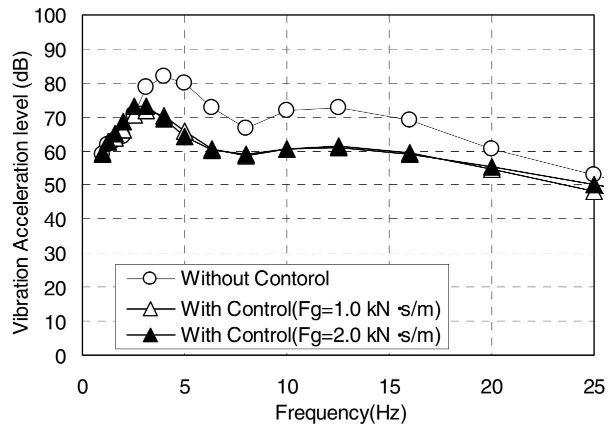


Fig.13-2: Vibration Acceleration Level on Control Gain(Input level 2)

CONCLUSION

Conclusions are obtained as follows:

DVFB-controlled AMD achieved a sufficient vibration control effect, even given its simplified control system. In addition, the control system was successfully configured with a single acceleration sensor.

A stable system across a wide frequency band was realized. More research is indicated for high-frequency band; however, this appears to have little effect on the performance goals for AMD designed for detached houses.

An AMD developed in the course of this research incorporating an AC servomotor and ball screw was adaptable for use in detached houses.

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