Active and Regenerative Control of an Electrodynamic-Type Suspension*

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Active vibration dampers and isolators are capable of reducing vibration, and the results are remarkable and can be achieved adaptively. However, they require a large amount of energy to control the vibration. Since structural vibration has motional energy, a question arises as to how we can utilize this energy rather than just simply suppressing it. In this paper an energy regenerative type suspension is presented which utilized an electrodynamic actuator to regenerate electric power during the high-speed motion of the actuator. At low-speed motion, an active control algorithm is applied to the same actuator to provide good damping performance. In order to reduce energy loss in the control amplifier, active relay control is applied instead of linear control. The system is first simulated on a computer and followed by a simple experiment similar to a suspension system. The results are very encouraging and show that the system performed better than a passive suspension system and can regenerate vibration energy.

Key Words: Damping, Vibration Isolation, Vibration Control, Energy Regeneration, Control Device, Mechatronics, Simulation

1. Introduction

Active vibration control has been progressively used to suppress vibration because of its high damping capability⁽¹⁾. Some of the typical examples of active vibration control are in tall buildings⁽²⁾, large space structures⁽³⁾ and automobile suspensions⁽⁴⁾. In this paper a vehicle suspension system is considered and is shown schematically in Fig. 1. A passive system as shown in Fig. 1(a) is simple and does not require any external energy, but the performance is limited. While an active suspension system as shown in Fig. 1(b) has a good damping and vibration isolation performance. A major problem with active vibration control is its reliability; that is, active damper is apt to become unstable in certain range of operation and produces self-excited vibration. High cost associated

with its installation and maintenance are also the main problem for its application. In addition practical application of active control produces a new problem; high dissipation of energy. A typical active suspension of an automobile is said to spend about 5 HP of hydraulic power. Active vibration control of a building vibration requires large electric energy when excited by strong wind. Even though large electric energy is seldomly required, it needs expensive electric power contacts. Since the vibration is a kind of energy and the actuator is an energy converter⁽⁵⁾, they can be used to regenerate the energy. Then the idea of utilizing the vibrating energy and regenerating it to electric power can lead to high cost saving and

Fig. 1 Suspension system

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efficiency^{(6),(7)}.

This idea is applied to a vehicle suspension system which has a linear electrodynamic actuator and spring. Electric power is regenerated during high-speed motion of the actuator. At low speed motion, active control algorithm is applied to the same actuator to provide good damping performance. Similar research is carried out by Suda and Shiiba⁽⁸⁾, but they used two actuators; one for energy regeneration and the other for active control. In this paper only one actuator is used to provide both the functions of active control device and energy regeneration. Since regenerative control requires a highly efficient actuator, it can be very expensive to have multiple actuators in the control. The system is simulated on a computer to evaluate the vibration suppression property and energy regenerative capability. A single degree-of-freedom suspension system with a linear DC motor is made to evaluate the proposed idea and the results obtained are discussed in detail.

2. Energy Regenerative Suspension

Energy regenerative driving circuit is commonly used to control servo motors. However, this technique cannot be applied directly to active vibration control because the vibration amplitude is very small at high frequency motion. This paper proposes a new and efficient technique to overcome the above problem.

2.1 Energy regenerative control

Let us consider the regenerative operation of a linear DC motor as shown in Fig. 2. An ideal actuator has the following energy conversion property;

$$f = \phi i, \text{ and } e = \phi v \quad (1)$$

where f and v are the actuator force and velocity, e and i are the driving voltage and current, respectively, and ϕ [N/A=Vs/m] is the actuator constant.

When the actuator velocity v [m/s] is low and the back-electro-motive (BEM) voltage e [V] is lower than the voltage e_{fb} [V], which is the summation of the battery voltage e_b [V] and the diode voltage drop e_F [V], ($|e| = \phi v < e_{fb}$), then the actuator can not produce any regenerative current and damping force. On the contrary, if the BEM voltage $|e|$ is greater than

Fig. 2 Electrodynamic regenerative circuit

the summed voltage $e_{fb} (|v| > e_{fb}/\phi)$, then the actuator produces the following force and regenerative current,

$$f = \phi i = \frac{\phi}{R} (\phi v - e_{fb}) \quad (2)$$

From Eq. (2), the damping force characteristics has a dead zone when $\frac{e_{fb}}{\phi} > v > -\frac{e_{fb}}{\phi}$ which is shown by the solid line in Fig. 3.

Next, suppose the actuator is connected to the pure resistance R [Ω], then the current is $i = e/R$ and the actuator produces the following force,

$$f = \phi i = \frac{\phi^2}{R} v \quad (3)$$

Equation (3) is the same as passive damper with a damping coefficient of ϕ^2/R . This relation is shown by the dashed line in Fig. 3.

2.2 Suspension system

Let us consider a single degree-of-freedom suspension system as shown in Fig. 4. The system has the following equation of motion.

$$m\ddot{x} + k(x - x_0) = f \quad (4)$$

For high speed motion $|v| > e_{fb}/\phi$, regenerative control is applied to the actuator. While for low speed motion $|v| < e_{fb}/\phi$, active control algorithm is applied to the same actuator. Then we have the following force equations,

$$\begin{cases} f = \frac{\phi^2}{R} (\dot{x}_0 - \dot{x}) + \frac{\phi}{R} e_{fb} & \dot{x} - \dot{x}_0 > \frac{e_{fb}}{\phi} \\ f = -K_x x - K_v \dot{x} & \frac{e_{fb}}{\phi} > \dot{x} - \dot{x}_0 > -\frac{e_{fb}}{\phi} \\ f = \frac{\phi^2}{R} (\dot{x}_0 - \dot{x}) - \frac{\phi}{R} e_{fb} & \dot{x} - \dot{x}_0 < -\frac{e_{fb}}{\phi} \end{cases} \quad (5)$$

3. Simulation

The proposed system is highly nonlinear, there-

Fig. 3 Damping characteristics

Fig. 4 Suspension system

fore the system is simulated on a computer to confirm its capability. The system is simulated using Runge-Kutta 4th order with Borland-C program. The simulation is carried out using the coefficients of the experimental setup mentioned in the next chapter.

The results are summarized using the equivalent damping ratio (damping ratio of the system with the equivalent linear damping given by Eq. (3)),

$$\zeta_{eq} = \frac{\phi^2/R}{2\sqrt{mk}} \quad (6)$$

The foundation is excited by a sinusoidal signal $x_0(t) = X_0 \cos \omega t$ and the corresponding response $x(t)$ is calculated. The frequency responses are shown in Figs. 5 to 9, where the frequency is normalized by ω/ω_n ($\omega_n = 36.8 \text{ rad/s}$) and the excitation is normalized by $\bar{x}_0$ in Eq. (7).

$$\bar{x}_0 = x_0 / \left(\frac{e_{fb}}{\omega_n \psi} \right) \quad (7)$$

The dashed curve is the pure passive one with damping ratio of ζ_{eq} , while the dash-dot curve is the pure active one with $f = -K_x x - K_v \dot{x}$.

Fig. 5 Frequency response of simulation with regenerative and active linear control ($\zeta_{eq}=0.1$)

3.1 Regenerative and active control

In the first simulation, ψ is varied and the two cases of regenerative and linear active control are shown in Figs. 5 ($\zeta_{eq}=0.1$) and 6 ($\zeta_{eq}=0.5$). The top graph indicates the transmissibility from the input to the output displacement, while the bottom graph is the ratio of regenerative energy versus input energy. The minus energy means active control dissipates energy, while the plus means regenerative energy. The dead zone for the regenerative operation e_b/ψ is 0.1225 m/s. In Figs. 5 and 6 the dashed curves are the pure passive one with a damping ratio of $\zeta = \zeta_{eq}$ and the dash-dot lines are the pure active one with $K_x = 14.97$ and $K_v = 316.27$. The solid lines indicate the responses of the proposed operation with different input amplitudes: The input amplitude is derived from sinusoidal excitation $\bar{x}_0$ which ranged from 0.2 to 4.0, and the response is calculated as shown by the solid lines. For the low frequency range, the response is along the line of pure active case, and the regenerative energy is negative. For the high frequency, however, the response is close to the passive case, and the regenerative

Fig. 6 Frequency response of simulation with regenerative and active linear control ($\zeta_{eq}=0.5$)

energy becomes positive and is close to one. The response in Fig. 6 ($\zeta_{eq}=0.5$) shows a lower peak resonance and has better regenerative energy than that of Fig. 5 ($\zeta_{eq}=0.1$). This means that the equivalent damping ratio ζ_{eq} is an important factor, and requires the value to be big enough to produce good damping and regenerative property.

Previous simulation has shown the possibility of regenerating the vibration energy. However, it does not include the energy dissipation in the power amplifier. The efficiency of the power amplifier and the force saturation caused by the supply voltage can

Fig. 7 Relay control

cause big problem.

3.2 Regenerative and relay control

To reduce the energy consumption in the control circuit, a relay control is proposed and applied. If the actuating signal is greater than a certain value F , the supply voltage is fed directly into the actuator which produces a constant actuating force ($\pm F_{ac}$). For small feedback signal ($|K_p x + K_v \dot{x}| < F$), the actuating force is set zero. The actuating force property is shown in Fig. 7. This can be achieved by switching the relays and hence the energy loss in the controller can be neglected.

The simulated results are shown in Figs. 8 and 9, where the switching forces are normalized using the following equations.

$$\bar{F} = \frac{\omega_n \phi}{e_{fb} k} F \text{ and } \bar{F}_{ac} = \frac{\omega_n \phi}{e_{fb} k} F_{ac} \quad (8)$$

The damping ratio $\zeta_{eq}=0.5$ is used in this simulation, because it provides good results as shown in the previous simulation. The two figures differ only in the control dead zone, with $\bar{F}=0.08$ (Fig. 8) and $\bar{F}=0.4$ (Fig. 9). For low frequency and small excitation, the

Fig. 8 Frequency response of simulation with regenerative and relay control ($\zeta_{eq}=0.5$, $\bar{F}=0.08$, $\bar{F}_{ac}=0.83$)

Fig. 9 Frequency response of simulation with regenerative and relay control ($\zeta_{eq}=0.5$, $\bar{F}=0.4$, $\bar{F}_{ac}=0.83$)

transmissibility is lesser than the pure passive case. But it approaches to the pure passive case for large excitation. This is due to the force limitation of the constant acting force F_{ac} which is not big enough to reduce the large excitations. With a smaller dead zone $\bar{F}$ (Fig. 9), the equivalent damping property is increased even for small excitations. However, the relay control has a strong nonlinearity, hence the simulated response can be very complicated. For example, the range of small excitations, the equivalent gain of relay control becomes big as well. This leads to better damping characteristics than the linear active control. For the low frequency range, the transmissibility approaches to one, and for the high frequency range, it approaches to the passive case. For small excitation the energy dissipation is very small due to the control dead zone. This is recognized in Figs. 8 and 9 with the curves (a) and (b) dropping to zero for the lower frequency range of $\omega/\omega_n < 2$ to 5. The curves (c), (d) and (e) become negative for the frequency range ω/ω_n of 0.1 to 1, but also become zero as the frequency range reduces further. These results are considered very encouraging from the point of energy regeneration.

4. Experimental Results and Considerations

A practical suspension system normally consists of friction, parameter change and actuating delay. These factors adversely affect the system performance. To confirm the previous simulation results, an experiment is carried out.

4.1 Experimental setup

The test apparatus used is shown schematically in Fig. 10. The physical parameters of the setup are shown in Table 1. The actuator used is a commercially available linear DC motor (Sumitomo S. M. Co.,

Fig. 10 Schematic diagram of the experimental setup

LV-10). The actuator constant is $\psi = 9.8$ [N/A = Vs/m] and the coil resistance is $R = 3.7$ [Ω]. These values generate the equivalent damping ratio of $\zeta_{eq} = 0.233$. The sprung mass is the stator of the motor which is supported by two linear guides. While the moving-coil part of the actuator x_0 is excited sinusoidally by an eccentric rotating disc and a connecting rod. Two coil springs are used to provide the stiffness. One potentiometer and two moving coil velocity pickups are installed to detect the signals x , v and $v - v_0$, respectively. The measured signals are put into the computer to calculate the demand signals. The control circuit is shown in Fig. 11, with three relays used: relay 1 switches actuator to regenerative operation or relay control, relay 2 switches the control force ON-OFF and relay 3 switches the force to plus or minus F_{ac} . The constant force F_{ac} is produced by applying the constant voltage E , which is fixed at 3 [V]. The goal of this research is to use E as 2.6 [V] which is twice the regenerated voltage. Due to the weak regeneration, the supply voltage is chosen as 3 [V] instead of 2.6 [V].

4.2 Experimental results

The frequency response is shown in Fig. 12. The top graph indicates the transmissibility due to a

Table 1 The symbols and their values

m	: Sprung mass	1.58	[kg]
k	: Stiffness	2136.4	[N/m]
ω_n	: Natural frequency = $\sqrt{\frac{k}{m}}$	36.8	[rad/s]
ψ	: Actuator constant	9.8	[N/A = Vs/m]
R	: Resistance	3.5	[Ω]
r	: Resistance		[Ω]
e_b	: Regenerative voltage	1.2	[V]
E	: Supply voltage	3.0	[V]
F_{ac}	: Relay force = $\frac{\psi}{r}E$		[N]
e_F	: Voltage drop in diode	0.8	[V]
e_{Fb}	: $e_b + e_F$		[V]
K_x	: Position feedback gain	14.97	[N/m]
K_v	: Velocity feedback gain	316.27	[Ns/m]
x	: Displacement		[m]
x_0	: Input displacement		[m]

Fig. 11 Scheme of proposed active relay and regenerative controller

Fig. 12 Frequency response of experiment with regenerative and relay control ($\zeta_{eq}=0.24$)

sinusoidal excitation of 4.5[mm] to the sprung mass displacement. While the bottom graph shows the regenerated energy. The curves (a) to (d) are the simulated responses; (a) pure relay control, (b) energy regenerative and relay control, (c) pure regenerative control, and (d) pure passive control with the actuator terminals connected. For the low frequency range, regeneration is not in operation and the response of case (b) is the same of case (a). The symbols shown in the experimental results are: $\triangle$ means relay control, and $\bullet$ means energy regenerative and relay control. From the experimental results it can be seen that $\triangle$ and $\bullet$ are very close until 8 [Hz]. Above that frequency, the responses differ and the energy regeneration is achieved. However, the response $\bullet$ has a sudden jump. This is due to the mechanical delay of the relay controller. Hence the experimental results are not as good as the simulation results. However, the transmissibility is much better than the pure passive one.

The energy is calculated by the multiplication of

Fig. 13 Time response

the voltage and current measured at the actuator terminal. One example of the measured data at 8.76 Hz is shown in Fig. 13. The top graph is the measured current, the middle graph is the measured voltage and the bottom graph is the calculated energy from the above. The average energy in this case is +0.031 [W]. In the bottom curve, the minus sign means energy dissipation in the relay control, while the plus sign means energy regeneration. Energy is regenerated when the detected actuator speed is faster than e_{fb}/ψ . The peak noise in the curve is due to the switching of relay. This calculation is carried out for a range of frequency and the resulting energy is shown in the bottom curve of Fig. 12. Energy is dissipated for the low frequency range which is minus. For high frequency range over 8 [Hz] ($\omega/\omega_n=1.5$), energy is regenerated and has a value of about +0.7 [W] at 11 [Hz] ($\omega/\omega_n=1.95$). This is due to the regenerative energy which is greater than the dissipated energy in the relay control.

5. Conclusions

A new method of active and regenerative control of suspension system is proposed in this paper. At highspeed motion, the actuator regenerates the electric power. At lowspeed motion, active as well as relay control is applied to the same actuator to provide good damping effects. A simple experimental setup is made to confirm the proposed technique after a series of computer simulation tasks. The following results are obtained:

- The equivalent damping ratio of the actuator

$\zeta_{eq} = \frac{\psi^2}{2R\sqrt{mk}}$ is an important parameter for efficient energy regenerative control. This requires a high performance actuator which has $\zeta_{eq} > 0.5$.

- When the exciting frequency is greater than the resonant frequency, the regenerated energy becomes greater than the dissipated energy of the active control.

However, there are also problems associated with the system,

- For the low frequency range, the active control energy is always greater than the regenerated energy, hence energy is always dissipated.
- Experimental results on vibration isolation and energy regeneration properties does not agree with the simulation results.

To solve the first problem, a highly efficient actuator should be used. For the second problem, the delay in the mechanical relay should be improved. Further work is continuing to improve the system with the application of FET relay in the control circuit.

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