A High-Accuracy Car Level Compensation Device for Hydraulic Elevators
(Stabilization with Estimated Acceleration Feedback)

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ABSTRACT

Improved response and stability of a high-accuracy car level compensation device for hydraulic elevators is examined. The device adjusts the car position error caused by passengers getting on or off. The previous compensation device is used an electro-hydraulic position servo system. It is improved by introducing an observer to estimate the car acceleration and applying a pseudo-acceleration feedback control. The performance of the modified compensation device is examined by a stability analysis and experiments, and it is shown that the proposed control method halves the car position overshoot in comparison to the conventional device and the settling time is 0.7 s.

1. INTRODUCTION

Compared to an ordinary traction drive elevator, a hydraulic elevator has two major merits. One is flexibility in its machine room location; the machine room need not be placed on top of the building. The other is the low load given to a structure since the weight of a car is mainly supported by the pit floor. On the other hand, a hydraulic elevator involves the following problems.

(1) Sinking or floating of the car level caused by passengers getting on or off.
(2) Low frequency vertical vibration of a car while moving.
(3) Large power consumption (about three times as much as for a traction type).
(4) Unpleasant odors and contamination due to oil leaks.

Problem (1) is the most significant one, because passengers easily become aware of it. Furthermore, this problem becomes even greater in the case of a long stroke elevator.

A compensation device is ordinarily used to readjust the car level. This device does not work while the car level error is within some tolerance zone (usually ±20−30mm). When the car level error exceeds the limit, this compensation device supplies or drains some amount of oil the volume of which is equivalent to that of the car level error. This is called a non-servo type compensation device. Although this device is effective for constructing a stable system, the compensating speed is very slow and ride quality can not be enhanced.

Therefore, we developed a high-accuracy car level compensation device, which is based on the car position feedback servo system. However, this device did not achieve a satisfactory ride quality because of its low stability. Then, we examine how to improve the response and stability of our earlier compensation device.

2. CONFIGURATION OF CAR LEVEL COMPENSATION DEVICE

Figures 1 and 2 show schematics of the hydraulic elevator and the car level compensation device. This device consists of a pressure regulating system, a PWM amplifier, a hydraulic control valve and a controller. The controller receives the car position signal and the jack pressure signal, and calculates the control signal necessary to drive the control valve.

Figure 3(a) shows a block diagram of the ordinary position feedback controller. The dither signal is added to the control signal. This dither signal is introduced to cancel hysteresis and the dead zone of the control valve. Besides the dither signal, the previous controller has a position feedback system, so that stability fully depends on mechanical damping. Therefore, the position feedback gain K_f is limited by stability.

In order to improve stability, Shudou et al. [1] proposed a new control method shown in Figs.3(b) and (c). The former is the cylinder pressure feedback method, and the latter, the real acceleration feedback method.

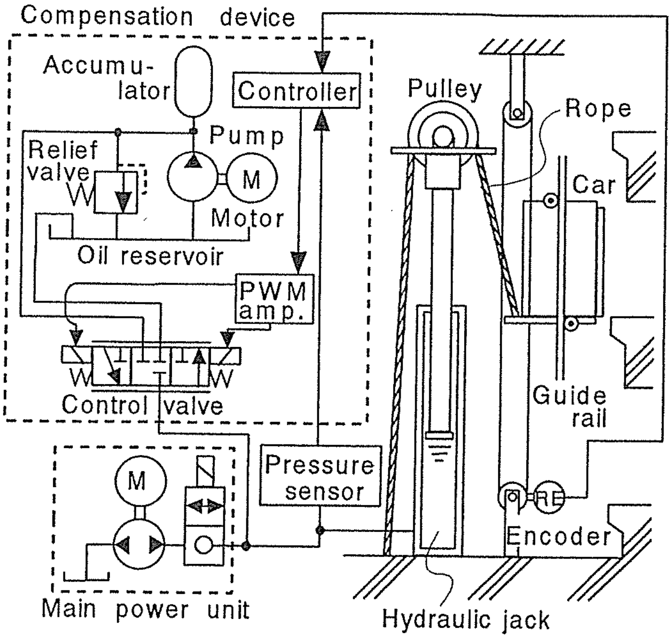


Fig.1 Configuration of hydraulic elevator system with compensation device.

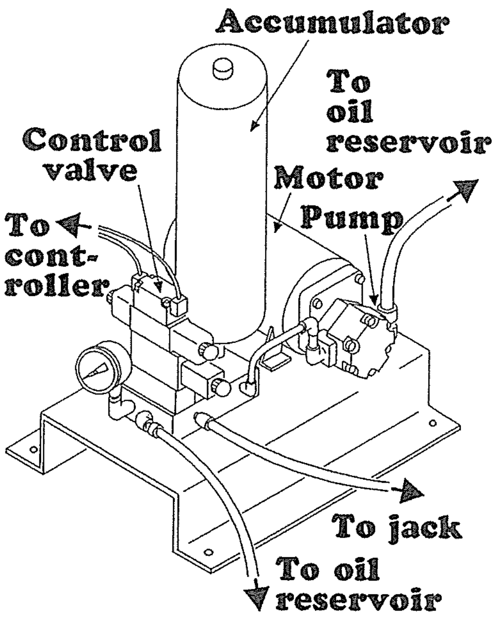
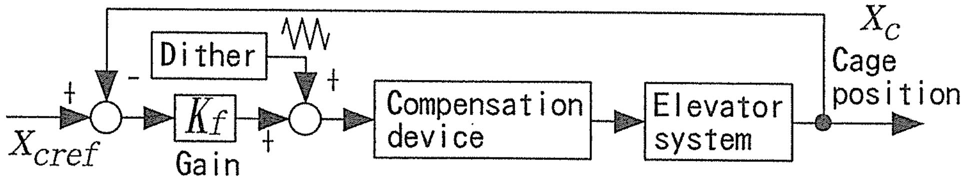


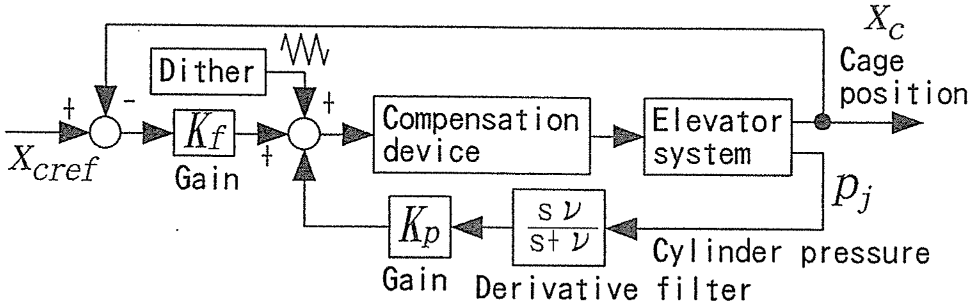
Fig.2 High-accuracy car level compensation device.

In the cylinder pressure feedback method, the pressure signal p_j is transferred by a pseudo-derivative filter. According to Shudou et al. (1), this method is effective for improving stability, but it enlarges the overshoot action of the car position, so that the total ride quality can not be enhanced.

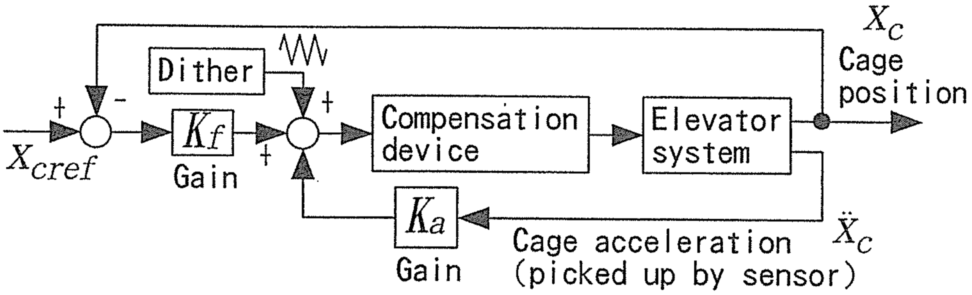
In the real acceleration feedback method, an acceleration feedback loop is applied, in which the car acceleration signal is selected by an additional sensor. This improves the stability and response, but it requires an acceleration sensor which increases the hardware cost, and it has lower total reliability.



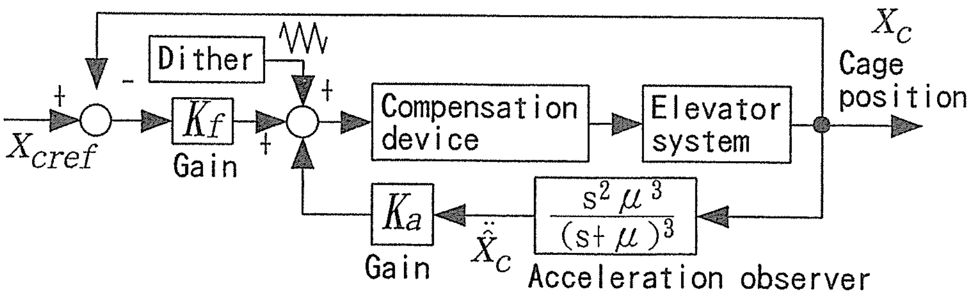
(a) Previous controller



(b) With cylinder pressure feedback



(c) With real acceleration feedback



(d) With estimated acceleration feedback

Fig.3 Configuration of controller.

Thus, we propose another control method, the estimated acceleration feedback method (Fig.3(d)). The method calculates the estimated acceleration signal as an observer. A similar system has been applied to suppress torque ripple of a motor (2).

Even thought the observer can be designated arbitrarily, the simplest way is to locate the observer poles at the same stable point on the real axis. Then, the system equation of the observer becomes as follows:

$$\frac{d}{dt}\begin{pmatrix}\hat{x}_c\\ \dot{\hat{x}}_c\\ \ddot{\hat{x}}_c\end{pmatrix}=\begin{pmatrix}0 & 1 & 0\\ 0 & 0 & 1\\ 0 & -3\mu^2 & -3\mu\end{pmatrix}\begin{pmatrix}\hat{x}_c\\ \dot{\hat{x}}_c\\ \ddot{\hat{x}}_c\end{pmatrix}+\mu^3(x_c-\hat{x}_c), \quad\cdots\cdots\cdots\cdots\cdots\cdots(1)$$

where a dot denotes a time derivative, μ is the angular frequency of the observer poles, x_c and $\hat{x}_c$ are the car position and its estimated signal, respectively. The transfer function of the observer $G_{obs}(s)$ can be derived as

$$G_{obs}(s)=\frac{s^2\mu^3}{(s+\mu)^3}. \quad\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots(2)$$

Figure 4 shows the frequency characteristics of $G_{obs}(s)$ when μ =126 rad/s(=20 Hz). In the low frequency range, the phase line comes close to the +180° line, so that the observer functions as somewhat like an acceleration estimator.

Usually, the estimated acceleration signal is distorted by noise, and sometimes it causes vibrations. When applied to the hydraulic elevator system, however, the estimated acceleration feedback works effectively for the following reasons.

(1) Since the first order natural frequency of an elevator mechanism is very low, it is possible to design the poles to get sufficiently small μ, so that the noise can be suppressed.

(2) The cut-off frequency of the control valve is very low (about 5−7 Hz), and the noise is mainly composed of relatively high frequency waves.

3. STABILIZATION ANALYSIS

3.1 Analytical model

In order to verify the efficiency of the proposed method, we analytically examine the stability of each control method. The analytical model is shown in Fig.5. The mechanism is replaced by a jack side mass m_j, a car side mass m_c and a hydraulic cylinder whose bulk modulus is B, volume is V_j, effective cylinder area is S_j and damping coefficient is c_j. The two masses are connected by a rope whose stiffness is k_r, and the displacement ratio of the two masses x_j / x_c is constrained to 1/4. The controller is an estimated acceleration feedback type, and the control valve is treated as a first order delay element whose cut-off frequency is λ. The model fundamental equations are as follows:

$$m_j\ddot{x}_j + c_j\dot{x}_j + 4k_r(4x_j - x_c) = S_j p_j, \quad \cdots\cdots(3)$$

$$m_c\ddot{x}_c - k_r(4x_j - x_c) = 0, \quad \cdots\cdots(4)$$

$$\dot{p}_j = \frac{B}{V_j}q_j - \frac{BS_j}{V_j}\dot{x}_j, \quad \cdots\cdots(5)$$

$$q_j = \frac{\lambda}{s+\lambda}\left[-K_f(x_c - x_{cref}) - K_a\frac{s^2\mu^3}{(s+\mu)^3}x_c\right]. \quad \cdots\cdots(6)$$

Table 1 lists model parameters, which are selected to make the natural frequency of the model equal to that of the experimental equipment used in the next section.

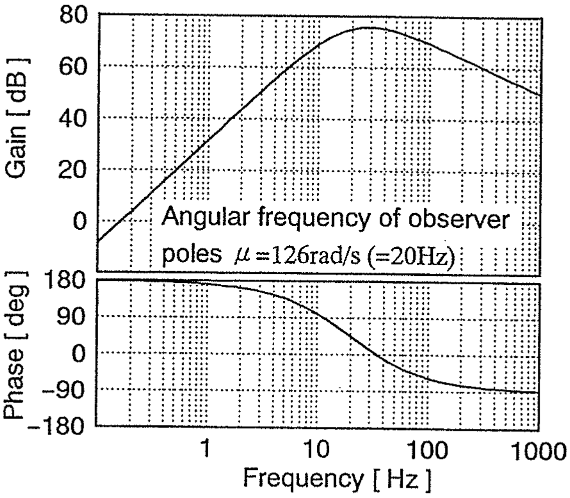


Fig.4 Frequency characteristics of acceleration observer.

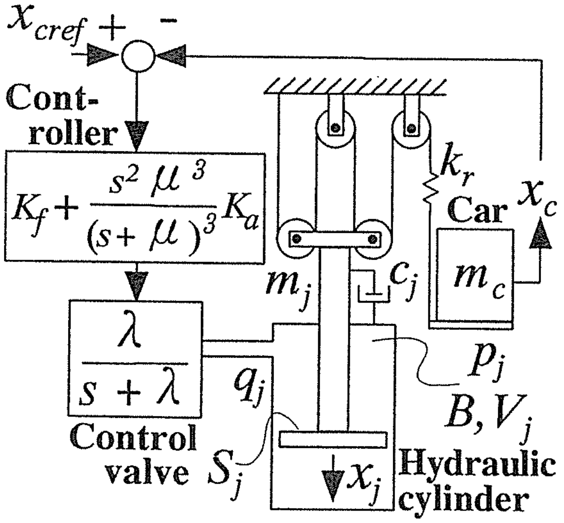


Fig.5 Model used in stability analysis.

Table 1 Parameters of model with rope elasticity.

Jack mass	m_j	112 kg
Car mass	m_c	780 kg
Damping coefficient	c_j	980 Ns/m
Rope stiffness	k_r	73333 N/m

Effective cylinder area	S_j	5.105×10^{-3} m^2
Bulk modulus	B	8.82×10^{8} N/m^2
Oil volume in cylinder	V_j	1.399×10^{-3} m^3

3.2 Role of acceleration feedback

Eliminating the variables p_j and q_j from Eqs.(3)－(6), and setting λ and μ as infinitely large, the following equation is derived:

$$m_c\dddot{x}_c + \left(c_j + \frac{BS_jK_a}{V_j}\right)\ddot{x}_c + \frac{BS_j{}^2}{V_j}\dot{x}_c + \frac{BS_jK_f}{V_j}(x_c - x_{cref}) = 0\,. \quad \cdots\cdots(7)$$

From the second term, it is clear that the acceleration feedback gain K_a works like the damping c_j. Thus, the acceleration feedback is an effective stabilization tool.

Generally speaking, a system with too much damping does not work well, because its response is too slow. In the case of the compensation device for the hydraulic elevator, however, the slow response provides a good ride quality. Therefore, the acceleration feedback is suited to the car level compensation.

3.3 Analysis results

Figure 6 shows the open-loop transfer function of the analytical model for K_a = 0 －2.0×10^{-3} m^2s. The mechanical resonance peak around 1.5 Hz is effectively stabilized as the acceleration feedback gain K_a increases. This result corresponds to the prediction of the last section.

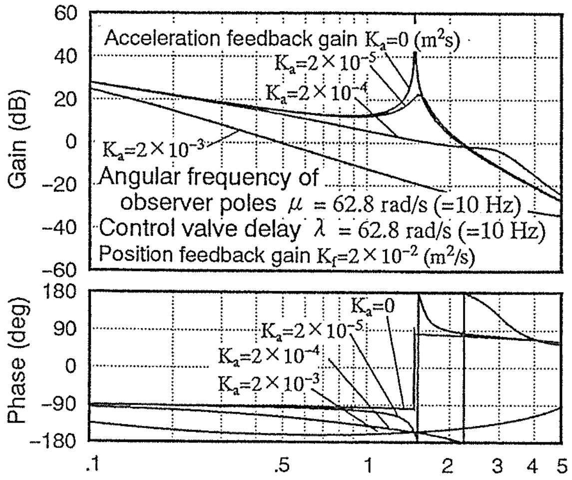


Fig.6 **Open-loop transfer function of analytical model.**
K_a = 0－2.0×10^{-3}(m^2s)

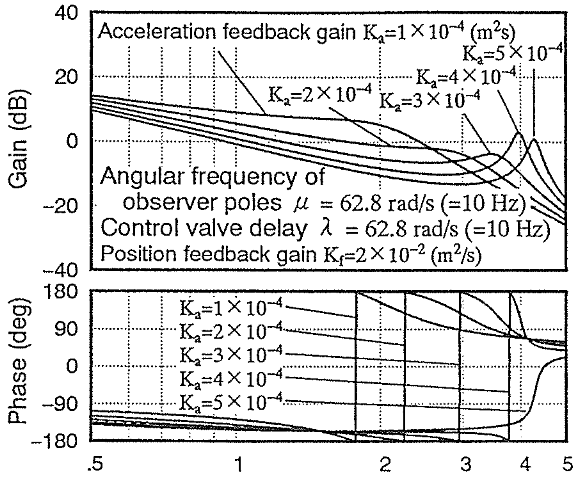


Fig.7 **Open-loop transfer function of analytical model.**
K_a = 1.0－5.0×10^{-4}(m^2s)

Next, the acceleration feedback gain K_a is trimmed around 2.0×10^{-4} m^{2}s and these results are shown in Fig.7. When the acceleration feedback gain K_a is small, the resonance peak is suppressed. As the gain K_a increases, however, the resonance peak significantly rises again. This means that if the gain K_a is set as too large, the system might be unstable. This phenomenon does not appear in the model without delays ($\lambda, \mu \to \infty$), so that it must be caused by the estimation and the control valve delay.

To summarize the above results, we note the acceleration feedback gain K_a should be chosen to maximize the stability (ex. gain margin or phase margin). For example, in the case of Fig.7, the acceleration feedback gain should be selected as $K_a = 2.0 \times 10^{-4}$ m^{2}s (gain margin = 3 dB and phase margin = 25 deg).

4. EXPERIMENT

4.1 Equipment and conditions

Next, we carry out experiments with actual size equipment in order to verify the efficiency of the stability analysis and the estimated acceleration feedback method. Figure 8 shows the configuration of the experimental equipment, and Table 2 lists its parameters. Unlike an ordinary hydraulic elevator, this equipment consists of a pull-type cylinder and four pulleys, which makes the car velocity four times as large as that of the cylinder. These modifications make the sinking and floating levels significantly larger than their usual values.

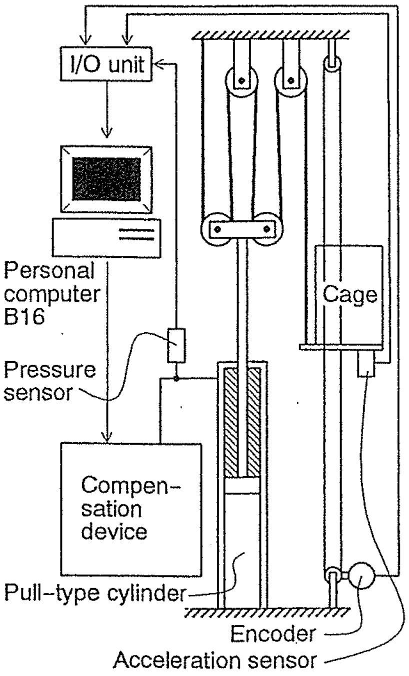


Fig.8 Configuration of experimental equipment.

Table 2 Parameters of experimental equipment.

Jack mass	m_j	112 kg
Car mass	m_c	780 kg
Mass of test weight	Δm_c	180 kg
Rope stiffness	k_r	73333 N/m
Effective cylinder area	S_j	5.105×10^{-3} m^2
Bulk modulus	B	8.82×10^{8} N/m^2
Oil volume in cylinder	V_j	1.399×10^{-3} m^3
Control valve delay	λ	44.0 rad/s
Angular frequency of observer poles	μ	125.6 rad/s
Elevator stroke	—	10 m
Sampling time of controller	—	0.010 sec

Then, four controllers (Fig.3) are compared when a 180 kg test mass is used to simulate passengers getting on or off.

4.2 Experimental results

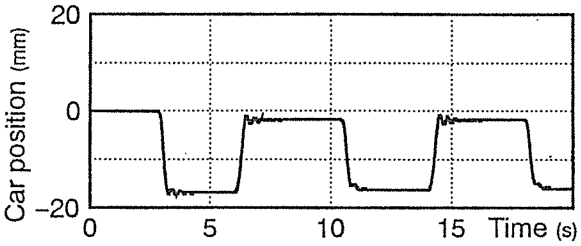
Figure 9 shows the results. The position feedback gain K_f is fixed at 1.6×10^{-2} m^2/s, while the acceleration and the pressure feedback gains K_a, K_p are adjusted to get better response through several trials.

(1) In the case of Fig.9(a), in which no compensation is applied, the car level varies according to ups and downs of the test weight. The cage level error is about 18 mm.

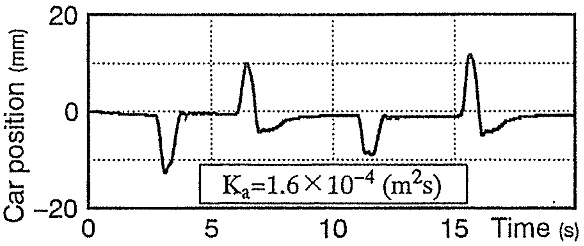
(2) In the case of Fig.9(b), in which only the position feedback is applied, the response becomes unstable.

(3) In the case of Fig.9(c), in which the cylinder pressure feedback is applied, the stability is improved. The overshoot, however, increases. Thus, this control method is not suited to the car level compensation.

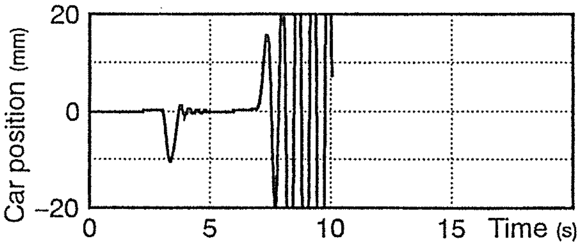
(4) In the case of Figs.9(d) and (e), in which the real and the estimated acceleration feedback is applied, respectively, both the stability and the overshoot are improved.



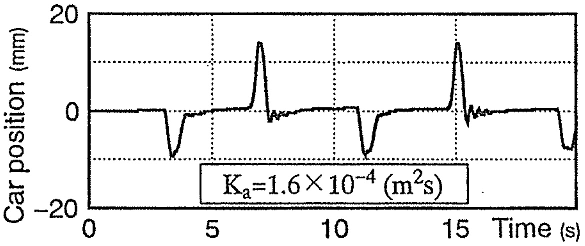
(a) Without compensation



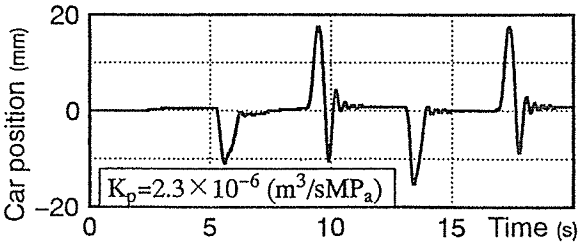
(d) With real acceleration feedback



(b) Previous control (only by position feedback)



(e) With estimated acceleration feedback



(c) With cylinder pressure feedback

Fig.9 Experimental results for comparison of controllers.

(5) The noise is not shown here, but its amplitude is about 1 m/s^2, and it mainly consists of around 100 Hz waves, which is the sampling frequency of the controller, so that the control valve does not response to the noise signal.

Next, the gains K_f and K_a of the estimated acceleration feedback are adjusted to make the system stable and to minimize the overshoot by trial and error. Figure 10 shows the response of this well-adjusted controller.

(6) The adjusted gains are almost equivalent to those of the stability analysis, which is $K_f = 2.0 \times 10^{-2}$ m^2/s, $K_a = 2.0 \times 10^{-4}$ m^{2}s. Thus, the proposed stability analysis is sufficiently effective to determine the gains K_f and K_a.

(7) The estimated acceleration feedback method realizes highly accurate compensation, in which the maximum overshoot is half as much as that of the case without compensation, and the settling time is about 0.7 s.

5. CONCLUSION

The stability of the car level compensation device was discussed. In order to improve the stability, the estimated acceleration feedback method was applied and examined in an analysis and experiments. The main points of this paper are summarized as follows.

(1) The estimated acceleration feedback method realizes a highly accurate compensation, in which the maximum overshoot is half as much as that of the case without compensation, and the settling time is about 0.7 s.

(2) Although the estimated acceleration feedback method may be generally affected by noise, the compensation device works well with this method.

(3) The proposed stability analysis method is sufficiently effective to determine the gains.

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(1) K. Shudou, et al., “Development of High-Accuracy Compensation Device for Hydraulic Elevator”, Proceedings of Japan Society of Mechanical Engineers, 910-62, pp.224-226 (1991) in Japanese.

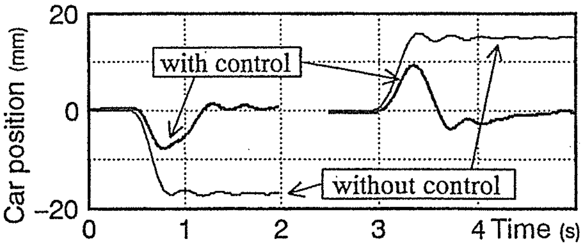

Fig.10 Experimental result of well-adjusted controller.
$K_f = 2.3 \times 10^{-2}$(m^2/s), $K_a = 2.3 \times 10^{-4}$(m^{2}s)

(2) K. Onishi, et al., “Microprocessor-Controlled DC Motor for Load-Insensitive Position Servo System”, IEEE Trans. on Ind. Elec. IE-34, pp.44-49 (1987).

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