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Semi-automatic control system for hydraulic shovel

Hirokazu Araya *, Masayuki Kagoshima

Mechanical Engineering Research Laboratory, Kobe Steel, Ltd., Nishi-ku, Kobe Hyogo 651 2271, Japan

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Abstract

A semi-automatic control system for a hydraulic shovel has been developed. Using this system, unskilled operators can operate a hydraulic shovel easily and accurately. A mathematical control model of a hydraulic shovel with a controller was constructed and a control algorithm was developed by simulation. This algorithm was applied to a hydraulic shovel and its effectiveness was evaluated. High control accuracy and high-stability performance were achieved by feedback plus feedforward control, nonlinear compensation, state feedback and gain scheduling according to the attitude. © 2001 Elsevier Science B.V. All rights reserved.

Keywords: Construction machinery; Hydraulic shovel; Feedforward; State feedback; Operation

1. Introduction

A hydraulic shovel is a construction machinery that can be regarded as a large articulated robot. Digging and loading operations using this machine require a high level of skill, and cause considerable fatigue even in skilled operators. On the other hand, operators grow older, and the number of skilled operators has thus decreased. The situation calls for hydraulic shovels, which can be operated easily by any person [1–5].

The reasons why hydraulic shovel requires a high level of skill are as follows.

1. More than two levers must be operated simultaneously and adjusted well in such operations.
2. The direction of lever operations is different from that of a shovel's attachment movement.

For example, in level crowding by a hydraulic shovel, we must operate three levers (arm, boom, bucket) simultaneously to move the top of a bucket along a level surface (Fig. 1). In this case, the lever operation indicates the direction of the actuator, but this direction differs from the working direction.

If an operator use only one lever and other freedoms are operated automatically, the operation becomes very easily. We call this system a semi-automatic control system.

When we develop this semi-automatic control system, these two technical problems must be solved.

1. We must use ordinary control valves for automatic control.
2. We must compensate dynamic characteristics of a hydraulic shovel to improve the precision of control.

* Corresponding author.
E-mail address: arayahrk@nedo.go.jp (H. Araya).

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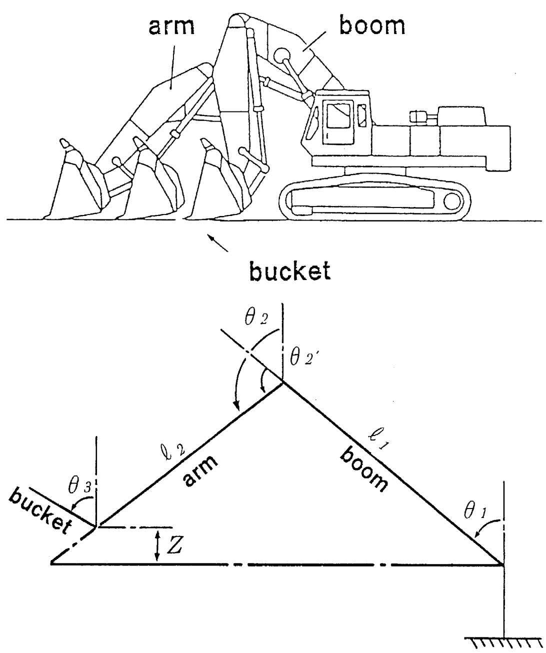


Fig. 1. Level crowding of an excavator and frame model of an excavator.

We have developed a control algorithm to solve these technical problems and confirm the effect of this control algorithm by experiments with actual hydraulic shovels. Using this control algorithm, we have completed a semi-automatic control system for hydraulic shovels. We then report these items.

2. Hydraulic shovel model

To study control algorithms, we have to analyze numerical models of a hydraulic shovel. The hydraulic shovel, whose boom, arm, and bucket joints are hydraulically driven, is modeled as shown in Fig. 2. The details of the model are described in the following.

2.1. Dynamic model [6]

Supposing that each attachment is a solid body, from Lagrange's equations of motion, the following expressions are obtained:

$$
\begin{aligned}
&J_{11}\ddot{\theta}_l + J_{12}\cos(\theta_1-\theta_2)\ddot{\theta}_2 + J_{13}\cos(\theta_1-\theta_3)\ddot{\theta}_3 + J_{12}\sin(\theta_1-\theta_2)\dot{\theta}_2^2 + J_{13}\sin(\theta_1-\theta_3)\dot{\theta}_3^2 - K_1\sin\theta_1 = \tau_1\\
&J_{12}\cos(\theta_1-\theta_2)\ddot{\theta}_1 + J_{22}\ddot{\theta}_2 + J_{23}\cos(\theta_2-\theta_3)\ddot{\theta}_3 - J_{12}\sin(\theta_1-\theta_2)\dot{\theta}_1^2 + J_{23}\sin(\theta_2-\theta_3)\dot{\theta}_3^2 - K_2\sin\theta_2 = \tau_2\\
&J_{13}\cos(\theta_1-\theta_3)\ddot{\theta}_1 + J_{23}\cos(\theta_2-\theta_3)\ddot{\theta}_2 + J_{33}\ddot{\theta}_3 - J_{13}\sin(\theta_1-\theta_3)\dot{\theta}_1^2 + J_{23}\sin(\theta_2-\theta_3)\dot{\theta}_3^2 - K_3\sin\theta_3 = \tau_3
\end{aligned}
\tag{1}
$$

where, $J_{11} = m_1 1_{g1}^2 + (m_2 + m_3)1_1^2 + I_1$; $J_{12} = m_2 1_1 1_{g2} + m_3 1_1 1_{g3}$; $J_{13} = m_3 1_1 1_{g3}$; $J_{22} = m_2 1_{g2}^2 + m_3 1_2^2 + I_2$; $J_{23} = m_3 1_2 1_{g3}$; $J_{33} = m_3 1_{g3}^2 + I_3$; $K_1 = (m_1 1_{g1} + m_2 1_1 + m_3 1_1)g$; $K_2 = (m_2 1_{g2} + m_3 1_3)g$; $K_3 = m_3 1_{g3} g$; and g = gravitational acceleration.

θ_i is the joint angle, τ_i is the supply torque, 1_i is the attachment length, 1_{gi} is the distance between the fulcrum and the center of gravity, m_i is the mass of the attachment, I_i is the moment of inertia around the center of gravity (subscripts i = 1–3, mean boom, arm, and bucket, respectively).

2.2. Hydraulic model

Each joint is driven by a hydraulic cylinder whose flow is controlled by a spool valve, as shown in Fig. 3. We can assume the following:

1. The open area of a valve is proportional to the spool displacement.
2. There is no oil leak.
3. No pressure drop occurs when oil flows through piping.

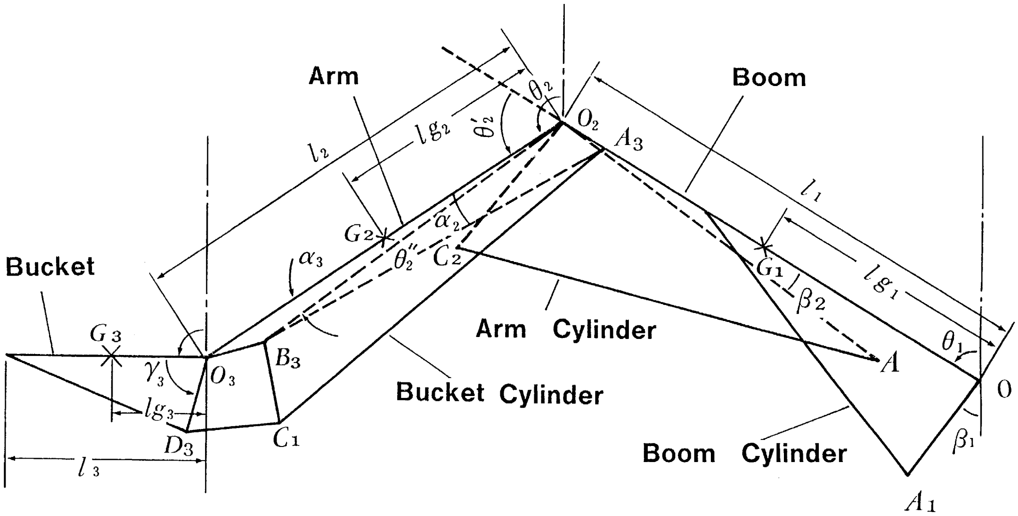


Fig. 2. Model of hydraulic shovel.

4. The effective sectional area of the cylinder is the same on both the head and the rod sides.

In this problem, for each joint, we have the following equation from the pressure flow characteristics of the cylinder:

$$A_i \dot{h}_i = K_{0i} X_i \sqrt{P_{si} - sgn(X_i) P_{1i}} - \frac{V_i}{K} \dot{P}_{1i} \qquad (2)$$

when,

$$K_{0i} = cB_i \sqrt{2/\gamma} \quad P_{1i} = P_{1i} - P_{2i}$$

where, A_i = effective cross-sectional area of cylinder; h_i = cylinder length; X_i = spool displacement; P_{si} = supply pressure; P_{1i} = cylinder head-side pressure; P_{2i} = cylinder rod-side pressure; V_i = oil volume in the cylinder and piping; B_i = spool width; γ = oil density; K = bulk modulus of oil; and c = flow coefficient.

2.3. Link relations

In the model shown in Fig. 1, the relation between the cylinder length change rate and the attachment rotational angular velocity is given as follows:(1) boom

$$f_1(\theta_1) = \frac{\dot{h}_1}{\dot{\theta}_1} = -\frac{\overline{OA_1}\,\overline{OC_1}\sin(\theta_1 + \beta_1)}{\sqrt{\overline{OA}_1^2 + \overline{OC}_1^2 + 2\overline{OA}_1\overline{OC}_1\cos(\theta_1 + \beta_1)}},$$

(2) arm

$$f_2(\theta_1,\theta_2) = \frac{\dot{h}_2}{\dot{\theta}_2 - \dot{\theta}_1} = -\frac{\overline{O_2A_2}\,\overline{O_2C_2}\sin(\theta_2 - \theta_1 + \beta_2 + \alpha_2)}{\sqrt{\overline{O_2A}_2^2 + \overline{O_2C}_2^2 + 2\overline{O_2A}_2\overline{O_2C}_2\cos(\theta_2 - \theta_1 + \beta_2 + \alpha_2)}},$$

(3) bucket

when $\overline{O_3D_3} = \overline{O_3B_3} = \overline{B_3C_3} = \overline{C_3D_3}$

$$f_3(\theta_2,\theta_3) = \frac{\dot{h}_3}{\dot{\theta}_3 - \dot{\theta}_2} = -\frac{\overline{A_3B_3}\,\overline{B_3C_3}\sin(\theta_3 - \theta_2 + \gamma_3 - \alpha_3 + \theta_2'')}{\sqrt{\overline{A_3B_3}^2 + \overline{B_3C_3}^2 + 2\overline{A_3B_3}\overline{B_3C_3}\cos(\theta_3 - \theta_2 + \gamma_3 - \alpha_3 + \theta_2'')}}. \qquad (3)$$

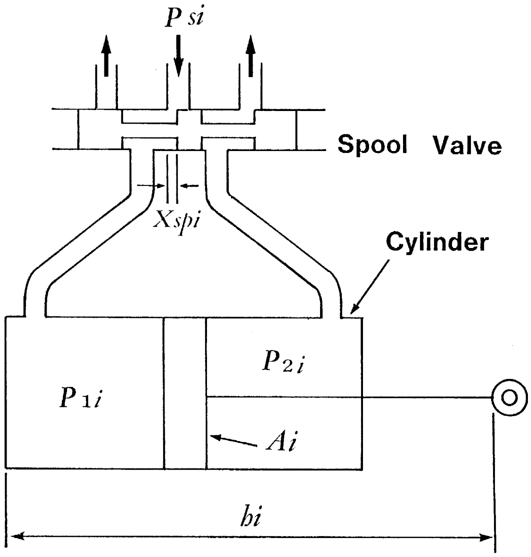


Fig. 3. Model of hydraulic cylinder and valve.

2.4. Torque relations

From the link relations of Section 2.3, the supply torque τ_i is given as follows, taking cylinder friction into consideration:

$$\tau_1 = -f_1(\theta_1) P1_1 A_1 + f_2(\theta_1,\theta_2) P1_2 A_2 + f_3(\theta_2,\theta_3) P1_3 A_3 - \left(C_{c1} f_1(\theta_1)\dot{\theta}_1 + sgn\left(\dot{\theta}_1\right) F_1\right) f_1(\theta_1) \quad (4)$$

$$\tau_2 = -f_2(\theta_1,\theta_2) P1_2 A_2 - \left\{C_{c2} f_2(\theta_1,\theta_2)\left(\dot{\theta}_2 - \dot{\theta}_1\right) + sgn\left(\dot{\theta}_2 - \dot{\theta}_1\right) F_2\right\} f_2(\theta_1,\theta_2)$$

$$\tau_3 = -f_3(\theta_2,\theta_3) P1_3 A_3 - \left\{C_{c3} f_3(\theta_2,\theta_3)\left(\dot{\theta}_3 - \dot{\theta}_2\right) + sgn\left(\dot{\theta}_3 - \dot{\theta}_2\right) F_3\right\} f_3(\theta_2,\theta_3).$$

Where, C_{ci} is the viscous friction coefficient and F_i is kinetic frictional force of a cylinder.

2.5. Response characteristics of the spool

Spool action has a great effect on control characteristics. Thus, we are assuming that the spool has the following first-order lag against the reference input.

$$X_i = \frac{1}{T_{spi}S + 1} X_i'. \quad (5)$$

Where, X_i' is the reference input of spool displacement and T_{spi} is a time constant.

3. Angle control system

As shown in Fig. 4, the angle θ is basically controlled to follow the reference angle θ_γ by position feedback. In order to obtain more accurate control, nonlinear compensation and state feedback are added to the position feedback. We will discuss details of these algorithms as follows.

3.1. Nonlinear compensation

In the ordinary automatic control systems, new control devices such as servo valves are used. In our semi-automatic system, in order to realize the coexistence of manual and automatic operations, we must use the main control valves, which are used in manual operation. In these valves, the relation between spool displacement and open area is nonlinear. Then, in automatic operation, using this relation, the spool displacement is inversely calculated from the required open area, and the nonlinearity is compensated (Fig. 5).

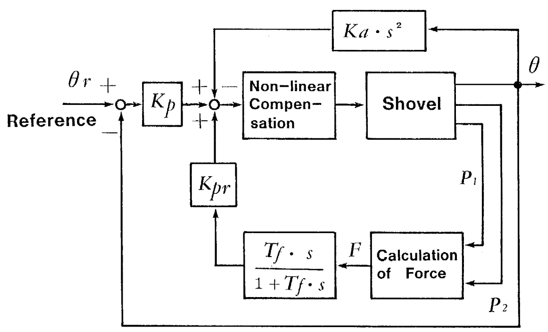
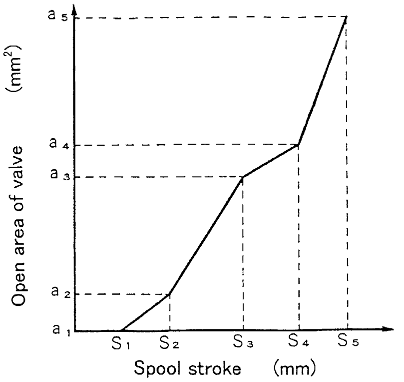
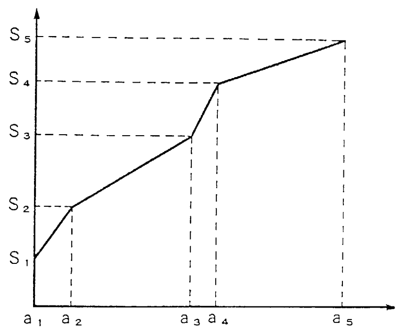


Fig. 4. Block diagram of control system (θ).



Relation between spool stroke and open area of valve



Function of compensater

Fig. 5. Nonlinear compensation.

3.2. State feedback

Based on the model discussed in Section 2, if the dynamic characteristics for boom angle control are linearized in the vicinity of a certain standard condition (spool displacement X_{10}, cylinder differential pressure P_{110}, and boom angle θ_{10}), the closed-loop transfer function can be expressed by

$$\theta_1 = \frac{K_p}{a_2 s^3 + a_1 s^2 + a_0 s + K_p} \theta\gamma_1 \qquad (6)$$

where, K_p is position feedback gain; and

$$a_0 = \frac{f_1(\theta_{10}) A_1}{K_{01}\sqrt{P_{s1} - P_{110}}} + \frac{C_{c1} f_1(\theta_{10})}{2 A_1 (P_{s1} - P_{110})}$$

$$a_1 = \frac{X_{10}\{J_{11} + J_{12}\cos\theta'_2 + J_{13}\cos(\theta'_2 + \theta'_3)\}}{2 A_1 f_1(\theta_{10})(P_{s1} - P_{110})} + \frac{C_{c1} f_1(\theta_{10}) V_1}{A_1 K K_{01}\sqrt{P_{s1} - P_{110}}}$$

$$a_2 = \frac{V_1\{J_{11} + J_{12}\cos\theta'_2 + J_{13}\cos(\theta'_2 + \theta'_3)\}}{A_1 f_1(\theta_{10}) K K_{01}\sqrt{P_{s1} - P_{110}}}.$$

This system has a comparatively small coefficient a_1, so the response is oscillatory. For instance, if in our large SK-16 hydraulic shovel, X_{10} is 0, the coefficients are given as $a_0 = 2.7 \times 10^{-2}$, $a_1 = 6.0 \times 10^{-6}$, $a_2 = 1.2 \times 10^{-3}$. Addingthe acceleration feedback of gain K_a, to this (the upper loop in Fig. 4), the closed loop transfer function is given as

$$\theta_1 = \frac{K_p}{a_2 s^3 + (a_1 + K_a) s^2 + a_0 s + K_p} \theta_{r1}. \qquad (7)$$

Adding this factor, the coefficient of s^2 becomes larger, thus, the system becomes stable. In this way, acceleration feedback is effective in improving the response characteristics.

However, it is generally difficult to detect acceleration accurately. To overcome this difficulty, cylinder force feedback was applied instead of acceleration feedback (the lower loop in Fig. 4). In this case, cylinder force is calculated from detected cylinder pressure and filtered in its lower-frequency portion [7,8]. This is called pressure feedback.

4. Servo control system

When one joint is manually operated and another joint is controlled automatically to follow the manual operation, a servo control system must be required. For example, as shown in Fig. 6, in the level crowding control, the boom is controlled to keep the arm end height Z (calculated from θ_1 and θ_2) to reference Zr. In order to obtain more accurate control, the following control actions are introduced.

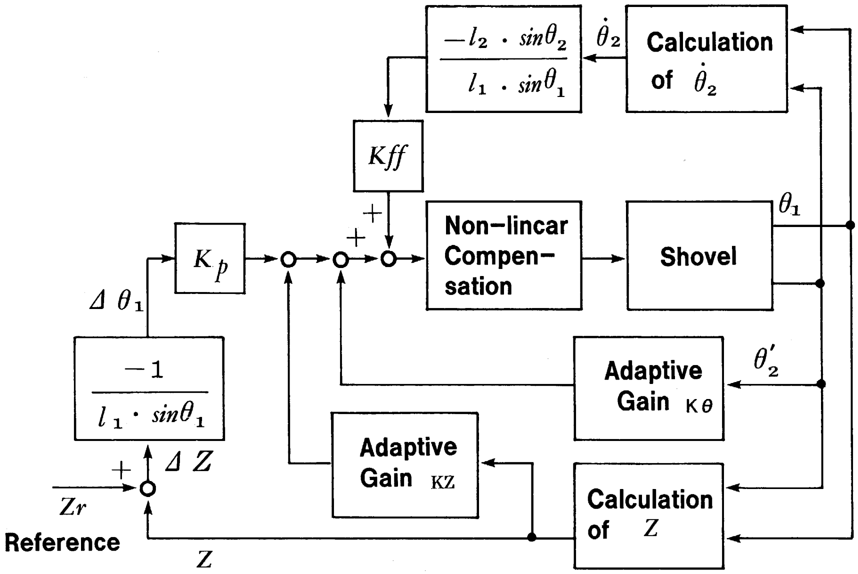


Fig. 6. Block diagram of control system (Z).

4.1. Feedforward control

Calculating Z from Fig. 1, we obtain

$$Z = 1_1 \cos\theta_1 + 1_2 \cos\theta_2. \tag{8}$$

Differentiating both sides of Eq. (8) with respect to time, we have the following relation,

$$\dot{\theta}_1 = -\frac{\dot{Z}}{1_1 \sin\theta_1} - \frac{1_2 \sin\theta_2}{1_1 \sin\theta_1}\dot{\theta}_2. \tag{9}$$

The first term of the right-hand side can be taken as the expression (feedback portion) to convert $\dot{Z}$ to $\dot{\theta}_1$, and the second term of the right-hand side is the expression (feedforward portion) to calculate how much θ_1 should be changed when θ_2 is changed manually.

Actually, $\dot{\theta}_2$ is determined using the difference value of $\Delta\theta_2$. To optimize the feedforward rate, feedforward gain K_{ff} is tunned.

There may be a method to detect and use the arm operating-lever condition (i.e. angle) instead of arm angular velocity, since the arm is driven at an angular velocity nearly proportional to this lever condition.

4.2. Adaptive gain scheduling according to the attitude

In articulated machines like hydraulic shovels, dynamic characteristics are greatly susceptible to the attitude. Therefore, it is difficult to control the machine stably at all attitudes with constant gain. To solve this problem, the adaptive gain scheduling according to the attitude is multiplied in the feedback loop (Fig. 6). As shown in Fig. 7, the adaptive gain (KZ or $K\theta$) is characterized as a function of two variables, θ'_2 and Z. θ'_2 means how the arm is extended, and Z means the height of the bucket.

5. Simulation results

The level crowding control was simulated by applying the control algorithm described in Section 4 to the hydraulic shovel model discussed in Section 2. (In the simulation, our large SK-16 hydraulic shovel was employed.) Fig. 8 shows one of the results. Five seconds after the control started, load disturbance

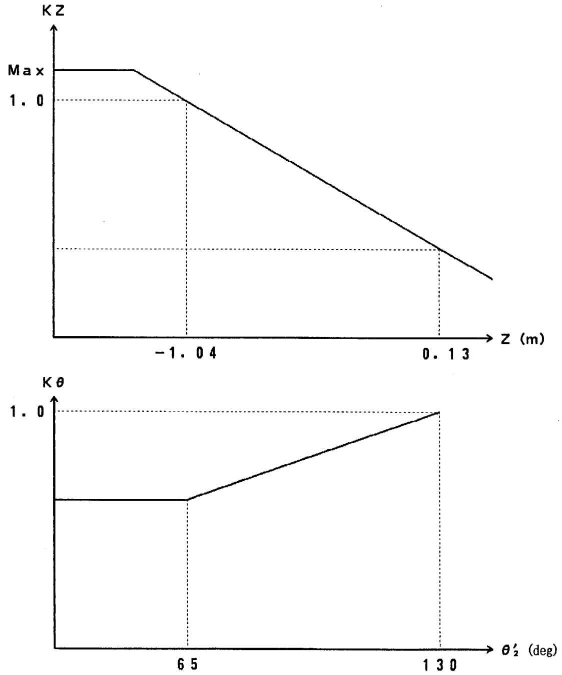


Fig. 7. Gain scheduling according to the attitude.

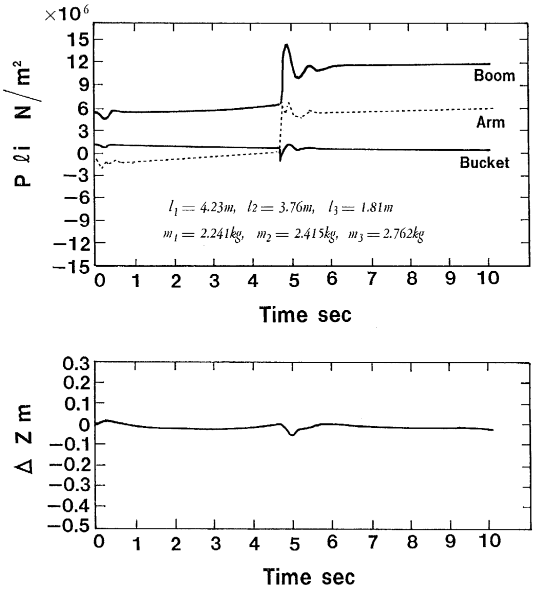


Fig. 8. Simulation result of level crowding.

was applied stepwise. Fig. 9 shows the use of feed-forward control can reduce control error.

6. Semi-automatic control system

Based on the simulation, a semi-automatic control system was manufactured for trial, and applied to the SK-16 shovel. Performance was then ascertained by field tests. This section will discuss the configuration and functions of the control system.

6.1. Configuration

As illustrated in Fig. 10, the control system consists of a controller, sensors, man–machine interface, and hydraulic control system.

The controller is based on a 16-bit microcomputer which receives angle input signals of the boom, arm, and bucket from the sensor; determines the condition of each control lever; selects control modes and calculates actuating variables; and outputs the results from the amplifier as electrical signals. The hydraulic control system generates hydraulic pressure proportional to the electrical signals from the electromagnetic proportional-reducing valve, positions the spool of the main control valve, and controls the flow rate to the hydraulic cylinder.

In order to realize high-speed, high-accuracy control, a numeric data processor is employed for the

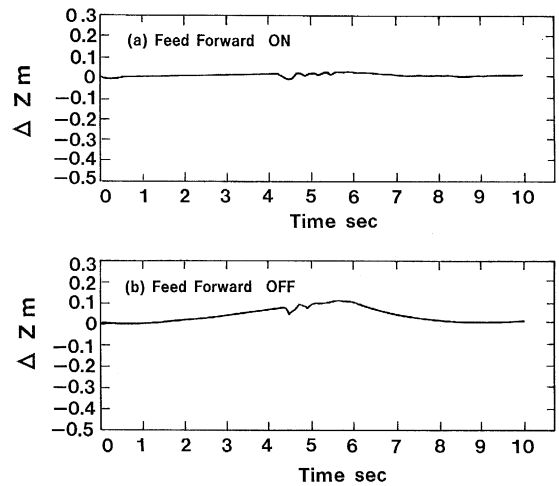


Fig. 9. Effect of feedforward control on control error of Z.

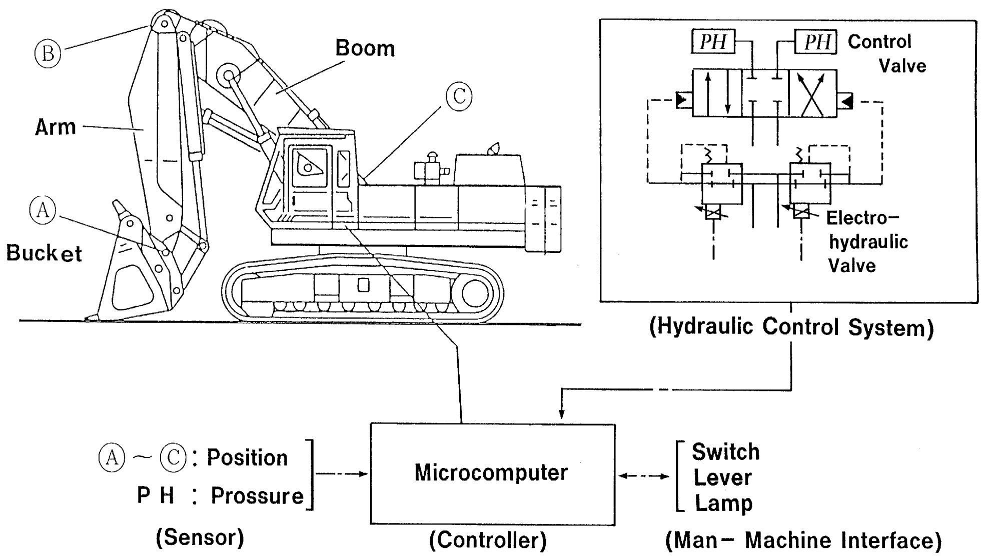


Fig. 10. Schema of control system.

controller, and a high-resolution magnetic encoder is used for the sensor. In addition to these, a pressure transducer is installed in each cylinder to achieve pressure feedback.

The measured data are stored up to the memory, and can be taken out from the communication port.

6.2. Control functions

This control system has three control modes, which are automatically switched in accordance with lever operation and selector switches. These functions are the following

(1) Level crowding mode: during the manual arm pushing operation with the level crowding switch, the system automatically controls the boom and holds the arm end movement level. In this case, the reference position is the height of the arm end from the ground when the arm lever began to be operated. Operation of the boom lever can interrupt automatic control temporarily, because priority is given to manual operation.

(2) Horizontal bucket lifting mode: during the manual boom raising operation with the horizontal bucket lifting switch, the system automatically controls the bucket. Keeping the bucket angle equal to that at the beginning of operation prevents material spillage from the bucket.

(3) Manual operation mode: when neither the level crowding switch nor the horizontal bucket lifting switch are selected, the boom, arm, and bucket are controlled by manual operation only.

The program realizing these functions is primarily written in C language, and has well-structured module to improve maintainability.

7. Results and analysis of field test

We put the field test with the system. We confirmed that the system worked correctly and the effects of the control algorithm described in Chaps. 3 and 4 were ascertained as follows.

7.1. Automatic control tests of individual attachments

For each attachment of the boom, arm, and bucket, the reference angle was changed $\pm 5°$ stepwise from the initial value, and the responses were measured; thus, the effects of the control algorithm described in Chap. 3 were ascertained.

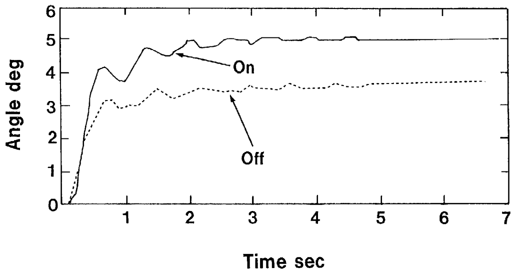


Fig. 11. Effect of nonlinear compensation on boom angle.

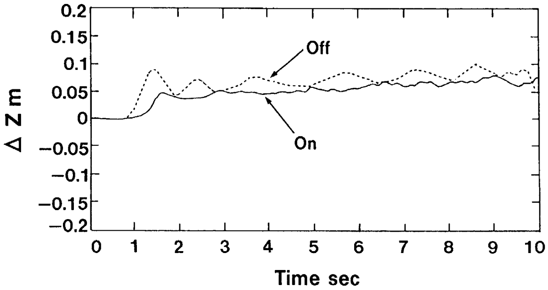


Fig. 13. Effect of feedforward control on control error of Z.

7.1.1. Effect of nonlinear compensation

Fig. 11 shows the test results of boom lowering. Because dead zones exist in the electro-hydraulic systems, steady-state error remains when simple position feedback without compensation is applied (OFF in the figure). Addition of nonlinear compensation (ON in the figure) can reduce this error.

7.1.2. Effect of state feedback control

For the arm and bucket, stable response can be obtained by position feedback only, but adding acceleration or pressure feedback can improve fast-response capability. As regards the boom, with only the position feedback, the response becomes oscillatory. Adding acceleration or pressure feedback made the response stable without impairing fast-response capability. As an example, Fig. 12 shows the test results when pressure feedback compensation was applied during boom lowering.

7.2. Level crowding control test

Control tests were conducted under various control and operating conditions to observe the control characteristics, and at the same time to determine the optimal control parameters (such as the control gains shown in Fig. 6).

7.2.1. Effects of feedforward control

In the case of position feedback only, increasing gain K_p to decrease error ΔZ causes oscillation due to the time delay in the system, as shown by "OFF" in Fig. 13. That is, K_p cannot be increased. Applying the feedforward of the arm lever value described in Section 4.1 can decrease error without increasing K_p as shown by "ON" in the figure.

7.2.2. Effects of compensation in attitude

Level crowding is apt to become oscillatory at the raised position or when crowding is almost completed. This oscillation can be prevented by changing gain K_p according to the attitude, as has been discussed in Section 4.2. The effect is shown in Fig. 14. This shows the result when the level crowding was done at around 2 m above ground. Compared to the case without the compensation, denoted by OFF in the figure, the ON case with the compensation provides stable response.

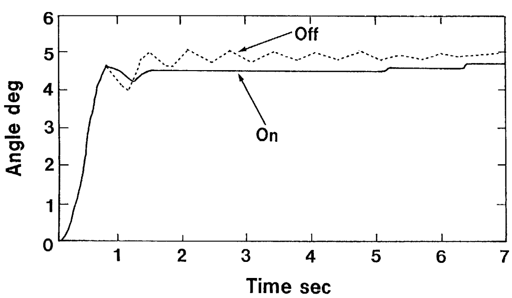


Fig. 12. Effect of pressure feedback control on boom angle.

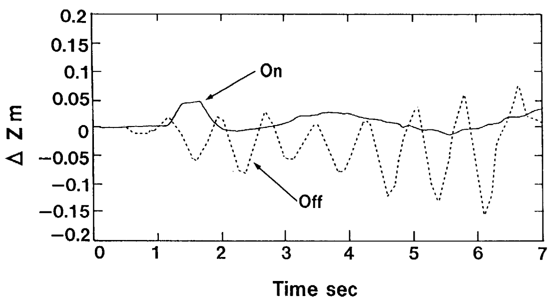


Fig. 14. Effect of adaptive gain control on control error of Z.

7.2.3. Effects of control interval

The effects of control interval on control performance were investigated. The results are:

1. when the control interval is set to more than 100 ms, oscillation becomes greater at attitudes with large moments of inertia; and
2. when the control interval is less than 50 ms, control performance cannot be improved so much.

Consequently, taking calculation accuracy into account, the control interval of 50 ms was selected for this control system.

7.2.4. Effects of load

A shovel with this control system carried out actual digging to investigate the effects of loading. No significant difference was found in control accuracy from that at no load.

8. Conclusions

This paper has shown that combining state feedback and feedforward controls makes it possible to accurately control the hydraulic shovel, and also showed that nonlinear compensation makes it possible to use ordinary control valves for automatic controls. The use of these control techniques allows even unskilled operators to operate hydraulic shovels easily and accurately.

We will apply these control techniques to other construction machinery such as crawler cranes, and improve the conventional construction machinery to the machines which can be operated easily by anyone.

References

[1] J. Chiba, T. Takeda, Automatic control in construction machines, Journal of SICE 21 (8) (1982) 40–46.
[2] H. Nakamura, A. Matsuzaki, Automation in construction machinery, Hitachi Review 57 (3) (1975) 55–62.
[3] T. Nakano et al., Development of large hydraulic excavator, Mitsubishi Heavy Industries Technical Review 22 (2) (1985) 42–51.
[4] T. Morita, Y. Sakawa, Modeling and control of power shovel, Transactions of SICE 22 (1) (1986) 69–75.
[5] H. Araya et al., Automatic control system for hydraulic excavator, R&D Kobe Steel Engineering Reports 37 (2) (1987) 74–78.
[6] P.K. Vaha, M.J. Skibniewski, Dynamic model of excavator, Journal of Aerospace Engineering 6 (2) (1990) April.
[7] H. Hanafusa, Design of electro-hydraulic servo system for articulated robot, Journal of the Japan Hydraulics and Pneumatics Society 13 (7) (1982) 1–8.
[8] H.B. Kuntze et al., On the model-based control of a hydraulic large range robot, IFAC Robot Control (1991) 207–212.