Valve Displacement Feedback Control Hydraulic Elevator

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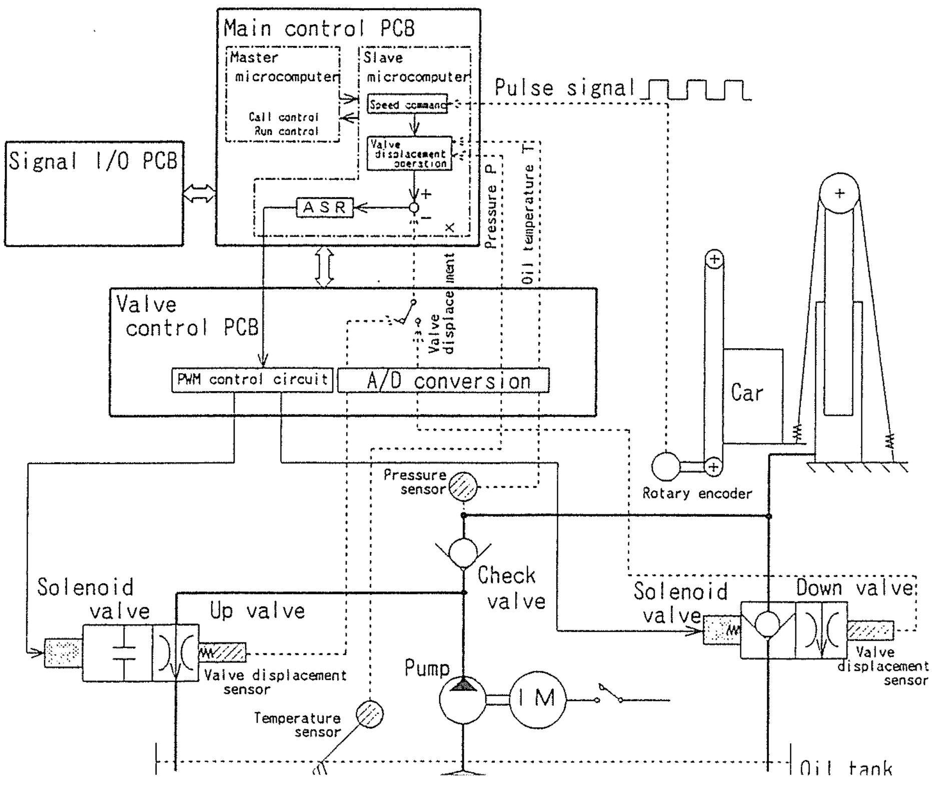
Abstract

In the hydraulic elevator,the cylinder capacity becomes large and the rope be long in proportion to the elevator stroke. We developed the valve displacement control system that feedbacks the valve displacement to the microcomputer. This system prevents the occurance of the low frequency vertical viblation,so the elavator can run smoothly even if the hydraulic oil temperature or the pressure change.

1. Introduction

In recent years , hydraulic elevator demanded has been increasing , because they do not require any machine room mounted at the upper part of the building which is in advantage to the oblique limitations, sunshade control, appearance design and etc. Presently , the number of hydraulic elevators being installed in Japan has occupied about 30% of newly installed elevators. Since hydraulic elevators either supply hydraulic oil into hydraulic cylinders or drain the hydraulic oil by a flow control valve. the running speed of the car may be deviated or the riding comfortability may change depending on passengers' load or the hydraulic oil temperature changes are the demerits (Fig.1) to hydraulic elevators. Hydraulic elevator with the valve displacement feedback control system introduced in this report detect the displacement (opening) of the flow control valve by means of a displacement sensor . The output of a valve displacement command is based on the signals from pressure sensors and oil temperature sensors , The speed command keeps the running speed of the car constant irrespective of the passengers load and hydraulic oil temperature change by the feedback control of the valve displacement command . As such a hydraulic elevator is being characterized as an elevator with excellent running efficiency .



2. Summary of valve displacement feedback control hydraulic elevator

2.1 Entire system configuration of valve displacement feedback control hydraulic elevator

Fig.1 shows the entire system configuration of the recently developed valve displacement feedback control hydraulic elevator. This control system is characterized with the following features.

(1) The hydraulic power unit is composed of oil tank, constant speed rotating motor, hydraulic pump, up valve, down valve, check valve, and many other components. The up and down valves are provided with pilot solenoid valves for controlling the valve displacement . Valve displacement sensor is for detecting the valve displacement and respectively.

(2) The control circuit is designed to the conventional function decentralized system using two microcomputers. The master microcomputer executes the processing of the car control generated on the halls or car as well as the running system . While the slave microcomputer executes processing of the speed command operation, valve displacement operation and valve displacement feedback control. Speed signals and various others safety device signals are input into these microcomputers for double monitoring so as to secure high safety as an elevator.

(3) The slave microcomputer calculates an optimum valve displacement command based on the position and speed of the car obtained by pulse signals being synchronized with the car movement, oil temperature obtained from the oil temperature sensor and the load pressure

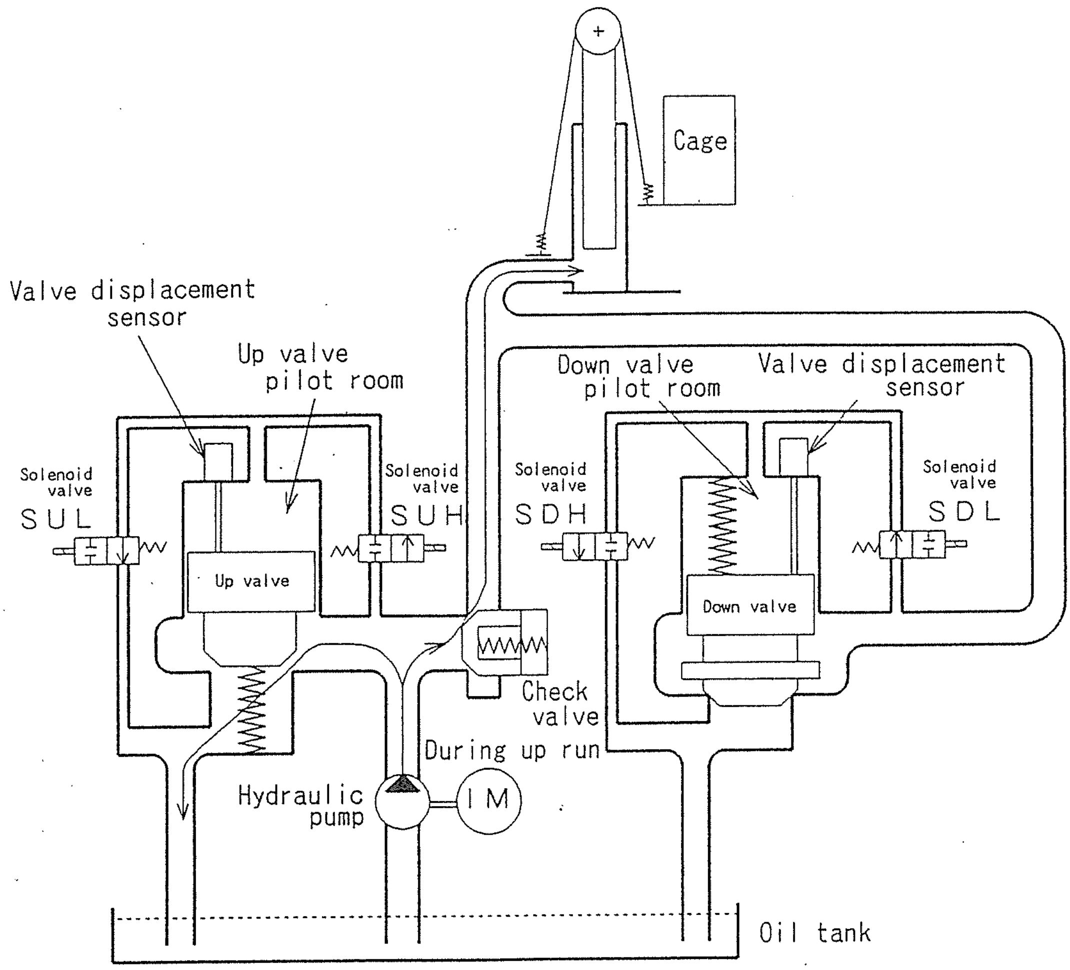


Fig.2 Construction of control valve

obtained from the pressure sensor. It calculates the deviation between the valve displacement command .The valve displacement detected by the valve displacement sensor sends the command for correcting this deviation to the PAM control circuit to drive the pilot solenoid valve to control the valve displacement. Thus the control valve executes the oil flow control to run the elevator smoothly.

2.2 Construction and functions of control valve

(1) Construction

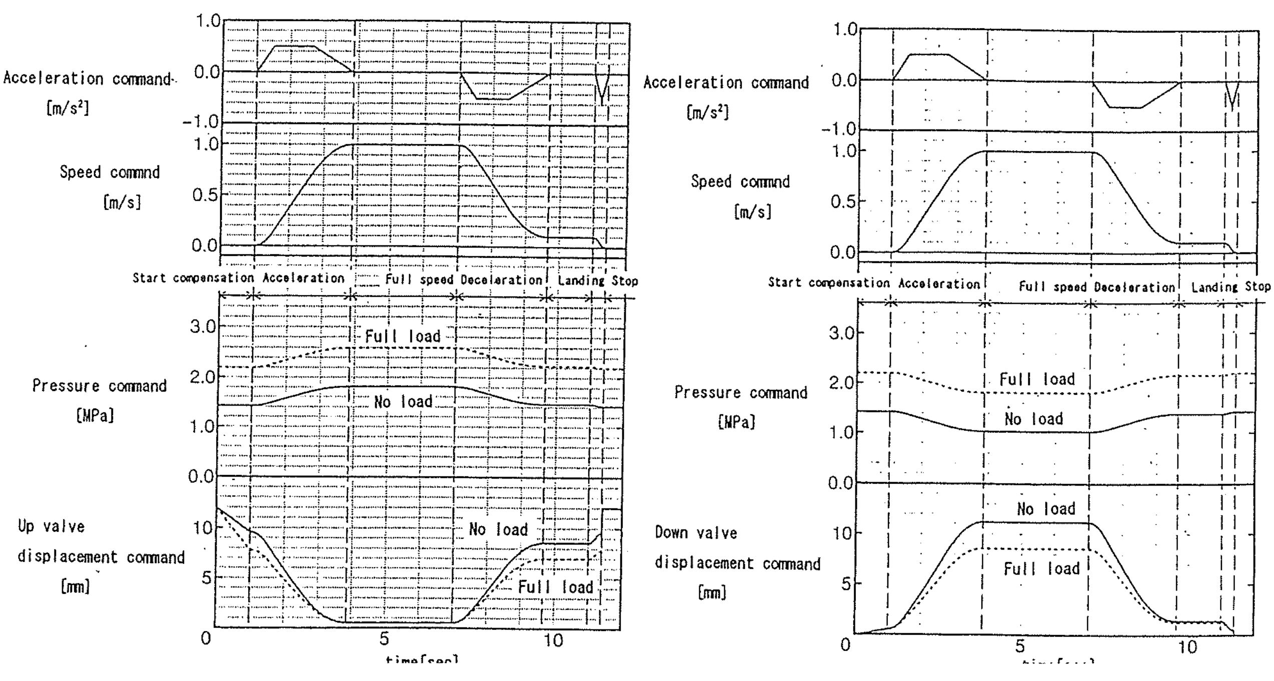
Fig.2 shows the construction of the control valve. The control valve consists of the up valve, down valve, check valve, and pilot relief valve. The up valve is composed of two high-speed solenoid valves (SUH, SUL) which served as main valve and pilot valve respectively. While the down valve is composed of two high-speed solenoid valves (SDH, SDL) which served as main valve and pilot valve respectively. The valve displacement sensor is used for detecting the displacement of the control valve . It is a highly reliable non-contact system using a magnetic resistance element and it is being provided with a built-in temperature compensation circuit.

(2) Functions of control valve

The control valve function in close direction when the hydraulic oil flows into the pilot room and function in open direction when the hydraulic oil flows out of the pilot room. The flow of the hydraulic oil into the pilot room of the up valve is controlled by SUH while the flow of the hydraulic oil out of the pilot room is controlled by SUL. In case of the down valve, the flow-out is controlled by SDH while the flow-in is controlled by SDL.

(1) Up-run

When an up command of an elevator is issued, the motor and hydraulic pump start running. The up control valve is fully open under this condition and all the pump flow is bled off to the tank. When a valve displacement command to the up control valve is issued as shown in Fig.3, the high-speed solenoid valve is driven so that the up control valve functions in the closing direction to increase the hydraulic pump pressure. As a result, the car starts running and accelerate.



When the valve displacement command reaches the high-speed run value, the up flow control valve is almost fully closed to be in the high-speed run status. When the car reaches the preset deceleration position, a deceleration command is issued to reduce the valve displacement command so that the up control valve functions in the opening direction. As a result, the bleed off flow increases to decelerate the car . When the valve displacement command reaches the low-speed run value, the car is set to the landing run status. Upon arrival at the stop position, a stop command is generated to fully open the up control valve so that all the hydraulic oil being discharged from the hydraulic pump is bled off into the tank causing the car to stop running.

(2) Down-run

In case of the down-run, the hydraulic oil being discharged from the hydraulic cylinder due to the net weight of the car is controlled by the down control valve without using any hydraulic pup. Its function is the same as the up - run mode.

3. Speed control

3.1 Technical problems of the speed control using the flow control valve

In case of the flow control system elevators, the oil viscosity changes and the volumetric efficiency of the hydraulic pump changes as the oil temperature and or the load pressure change in general. When the speed characteristic of the car is separated from the specified speed characteristic , the deceleration speed and landing run speed are also deviated as a result. The landing run time increases to cause a running efficiency drop and a dispersed landing level. A control system to compensate for the speed pattern according to the oil temperature and load pressure were offered as a remedial measure but the further improvement of the speed characteristed has been demanded.

3.2 Model systematization of controlled systems

The hydraulic oil flow must be controlled to keep the speed characteristic to be closed to the designed speed. The model systematization of the controlled systems by means of a mathematical expression model is executed by representing the relation of the elevator speed to the oil temperature, load pressure, manipulated variables of valves and flow by using mathematical expressions according to the physical laws. The control system is designed based on these mathematical expressions. Fig.4 shows the configuration of speed control circuit.

(1)Relation between the elevator speed and flow

The relation between the elevator speed(V)and the hydraulic oil flow(Qc)to the hydraulic cylinder is represented by the following formulas

$$V = 2\frac{Q_C}{S} \qquad \cdots(1)$$

$$Q_C = \frac{S}{2}V = \frac{\pi D_J^2}{8}V \qquad \cdots(2)$$

where, V:Speed command S:Sectional area of plunger Dj:Diameter of plunger

(2)Hydraulic pump discharge quantity correction operation

The actual hydraulic oil flow being fed from the hydraulic pump to the hydraulic cylinder is smaller than the theoretical discharge quantity of the hydraulic pump more or less. A differential flow from the theoretical discharge quantity that is leak flow changes according to the discharge pressure of the hydraulic pump, hydraulic oil temperature and rotating speed. Since the speed fluctuation is caused by a change of leak flow it cannot be compensated without correcting the above discharge quantity. The hydraulic pump discharge quantity is corrected by the following formula

$$Q_P = Q_0 - K_L \sqrt{\frac{P_P}{E}} \qquad \cdots(3)$$

3.3 Valve displacement command operation

From the orifice flow formula, we obtain the bleed off flow QB during up run by the following formula

$$Q_B = C \cdot A \sqrt{\frac{2 \cdot P_P}{\rho}} = C \cdot \sqrt{\frac{2}{\rho}} \cdot n \cdot d \cdot X \cdot \sqrt{P_P} \qquad \cdots(4)$$

Thus, the flow Qc flowing into the hydraulic cylinder during up run is expressed by the following formula

$$Q_C = C \cdot \sqrt{\frac{2}{\rho}} \cdot n \cdot d \cdot X \cdot \sqrt{P_C} \qquad \cdots(5)$$

In addition, the flow Qc flowing into the hydraulic cylinder during down run is represented by the following formula

$$Q_C = Q_P - Q_B \qquad \cdots(6)$$

From these formulas, the valve displacement command X refer to speed command V is represented by the following formulas.

During up run

$$X = \frac{Q_B}{C \cdot \sqrt{\frac{2}{\rho}} \cdot n \cdot d \cdot \sqrt{P_P}} \qquad \cdots(7)$$

During down run

$$X = \frac{Q_C}{C \cdot \sqrt{\frac{2}{\rho}} \cdot n \cdot d \cdot \sqrt{P_C}} \qquad \cdots(8)$$

where;V:Speed command Qc:Cylinder flow S:Sectional area of plunger Dj:Diameter of plunger Qp:Discharge flow of pump Qo:Theoretical discharge quantity of pump QB:Bleed off flow KL:Pump leak constant E:Engler viscosity Pp:Pump discharge pressure Pc:Cylinder pressure C:Flow coefficient A:Main valve opening area ρ :Hydraulic oil density n: of slits of main valve d:Slit width of main valve X:valve displacement command

Valve displacement command X is obtained from speed , Command V by the following method.

3.4 High-speed solenoid valve drive system

(1)PWM control

The PWM(Pulse Width Modulation)control system is also called electro-hydraulic control system. The up valve and down valve are controlled by the hydraulic pilot circuit and the high-speed solenoid pilot valves serve as the throttle in the hydraulic circuit. The up valve and down

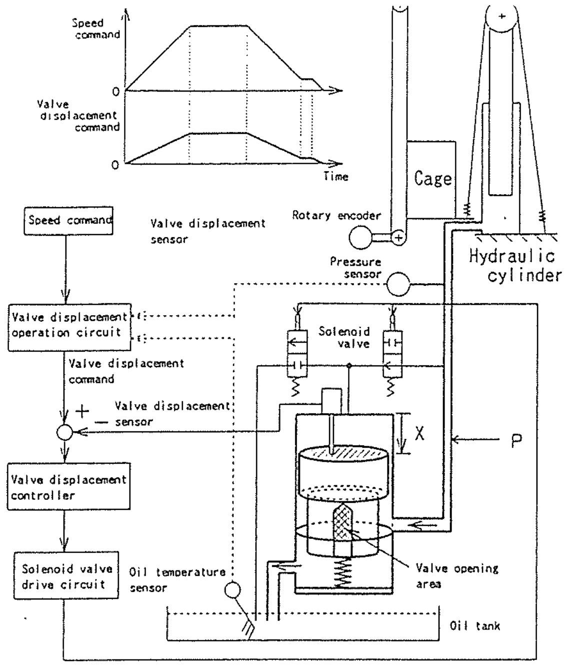


Fig.4 Configuration of speed control circuit

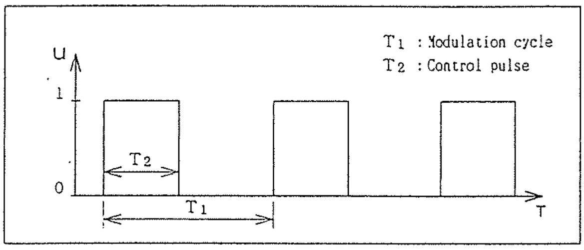


Fig.5 Solenoid valve drive timing chart

valve are provided with two high-speed pilot valves SUH, SUL and SDL, SDH respectively. The openings of flow control valves are controlled by controlling these high-speed pilot valves. The PWM control open and close these high-speed solenoid pilot valves SUH, SUL, SDL, SDH in the form of pulses and controls this opening or closing time so as to set the throttle equivalently proportional to the pulse width to control the hydraulic pressure of the pilot circuit of the up valve and down valve optionally. It can control the opening or closing degree of the flow control valves steplessly.

Fig.5 show a time chart of the high-speed solenoid pilot valves being turned on and off by applying a pulse signal to the up valve side high-speed solenoid pilot valves SUH, SUL during up - run or the down valve side high-speed solenoid pilot valves SDL, SDH during down - run. By changing the on-off time of these high-speed solenoid pilot valves, the hydraulic pressure of the pilot circuit is designed to control the opening or closing of the flow control valve. In Fig.5, T1 shows the modulation cycle while T2 shows the control pulse. Assume that the ratio ε (duty ratio)of the closing time of the high-speed solenoid pilot valves to pulse cycle T1 is expressed by $\varepsilon = T2/T1$, where T1:Pulse cycle and T2:Solenoid valve closing time, and we obtain the flow q flowing to the solenoid valves by the following formula.

$$q = C_0 \cdot A_0 \cdot \varepsilon \cdot \sqrt{\frac{2\Delta P}{\rho}} \qquad \cdots(9)$$

where;C0:Flow coefficient of solenoid pilot valves A0:Opening area of solenoid valves Thus, the pilot flow q can be optionally controlled by controlling the duty ratio ε. The PWM control is provided for controlling this duty ratio. Δ P:Differential pressure before and behind the solenoid valve,

(2)Same ratio command drive system

The pulse width pulse width Tp of a PWM signal is calculated by a deviation between the valve displacement sensor output and the valve displacement command to form the PWM signal which controls the high-speed solenoid valve. If an error signal is noticeable, a pulse having a long width is formed and the high-speed solenoid valve runs the maximum flow to cause the

main valve to be operated in high speed . If error=0, the main valve stop function.
Thus, a flow proportional to the valve displacement is obtained to control the elevator speed. The main valve displacement shows a displacement pattern corresponding to the speed command as shown in Fig. 5. The PWM modulation ratio command is given to the high-speed solenoid valves based on the same ratio command drive system giving the same modulation ratio to SUH, SUL (up run mode) or SDH, SDL (down run mode)

(3) Control system analysis

The valve displacement control system is analyzed by using the valve displacement feedback PWM control. Fig. 6 shows a simplified model of the hydraulic pilot circuit. For the hydraulic oil flow q of the high-speed solenoid pilot valve, the following formula holds true from the orifice flow equation,

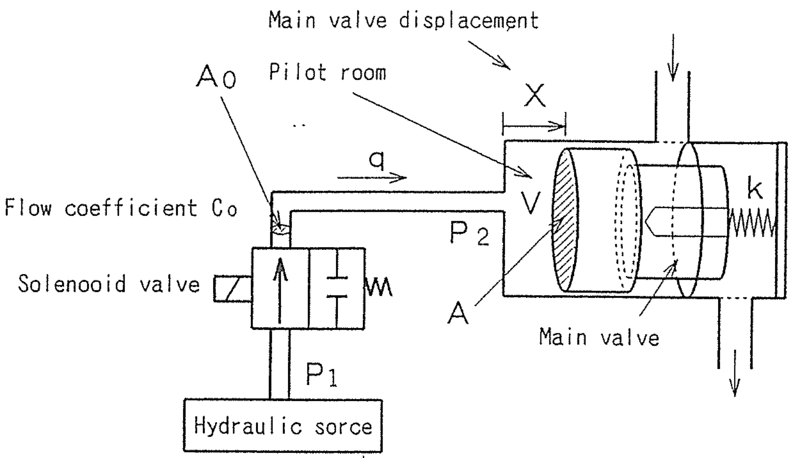


Fig.6 Hydraulic circuit model

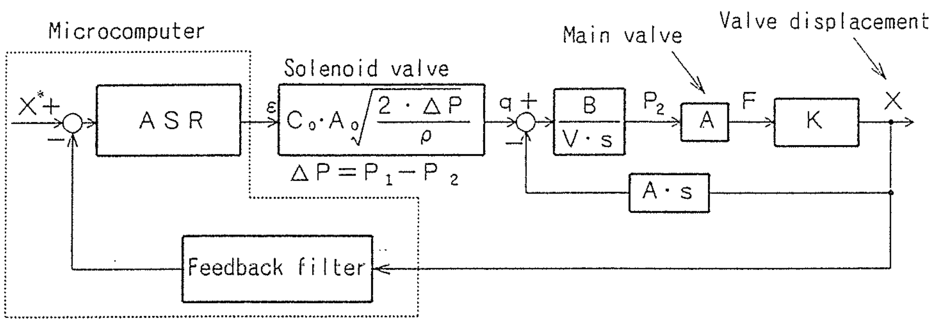


Fig.7 Block diagram of valve displacement control system

$$q = C_0 \cdot A_0 \cdot \varepsilon \cdot \sqrt{\frac{2(P_1 - P_2)}{\rho}} \qquad \cdots(10)$$

where C0 , flow coefficient of solenoid pilot valve and A0 , opening area of solenoid valve, ε , Duty command (P1-P2) :Differential pressure before and after the solenoid valve and ρ :Hydraulic oil density. Since the hydraulic oil being fed into the hydraulic pilot room pushes the control valve piston rightward while it is being compressed concurrently, the following formula holds true.

$$q = A \cdot \frac{dX}{dt} + \frac{V}{B} \cdot \frac{dP}{dt} \qquad \cdots(11)$$

Thus, the force F acting to the valve is represented by the following formula :

$$M \cdot \frac{d^2 X}{d^2 t} = A \cdot P_2 - k \cdot X \qquad \cdots(12)$$

The valve displacement control system is represented by a block diagram using the above formula as shown in Fig.7.

(4) Control responsibility

The speed control performance of an elevator is generally evaluated by the acceleration or deceleration characteristic of the car and the landing characteristic. The landing performance is judged by the control responsibility which takes the entire elevator system into account. Furthermore , the acceleration or deceleration of speed performance is determined by the configuration of the control system which will response according to the vibration mode of the hydraulic system, car and rope system . The control responsibility of hydraulic control units is generally desired to show the response time constant of 0.02-0.10sec. While the cross over angle frequency on the board diagram is desired to be about 10-50 rad/s. If the control response time constant is slower than 0.02-0.1sec, the follow up performance deteriorates to a load change to cause the landing performance to be not obtainable and if it is higher than the said value, the phase allowance is reduced to the natural vibration frequency of the plunger to cause frequent oscillations.

(5) Configuration of the control system by taking the natural vibration into account

Three modes shown in Table 1 generally come into question as the running vibrations of the elevator system.

Table 1 Vibration response modes of elevator system

No.	Items	Frequency bands	Remarks
1	Primary response	1-2Hz	Vibrations of car are controlling.
2	Secondary response	10-15Hz	Vertical vibrations of plunger controlling.
3	Resonance of each part of the car	30-40Hz	Vibrations of the car inside thimble rods controlling.

Alsomajor components of the control system, which induce these vibrations are shown in Table 2.

Table 2 Components of the control system which induce vibrations

No.	Items	Frequency bands Remarks
1	PWM solenoid valve drive	Caused by the frequency of pilot solenoid valve drive PWM signal and pulse cycle
2	Delay time of digital speed control system	Sampling cycle

By shortening the pulse interval of the pilot solenoid valve drive PWM output, the secondary resonance frequency having the highest response gain as running vibrations can be avoided. However, since the pilot solenoid valve contains a rise action delay, the increase of the solenoid valve drive frequency is restricted and the secondary resonance Frequency is not avoidable, but vibrations can be suppressed to be low by reducing the control system gain in the vicinity of the secondary resonance frequency within a possible range.

3.5 Hydraulic circuit with improved safety

The hydraulic circuit is constructed of top safeguard by taking the safety into account. The pump has stopped running and the car is supported by the check valve in the up - run mode, while the down valve is closed by shutting off the signals to the solenoid valves in the pilot circuit in the down - run mode. In addition to the speed control solenoid valves, a backup solenoid valve is added to the down hydraulic circuit in particular. Fig.8 shows the down hydraulic circuit.

SDL2 shows the backup solenoid valve. It is confirmed that by using this SDL2, the down valve is closed if the solenoid valves are wrongly energized during elevator stop or if the solenoid valve seizure occurs from the down run to the stop of the elevator.

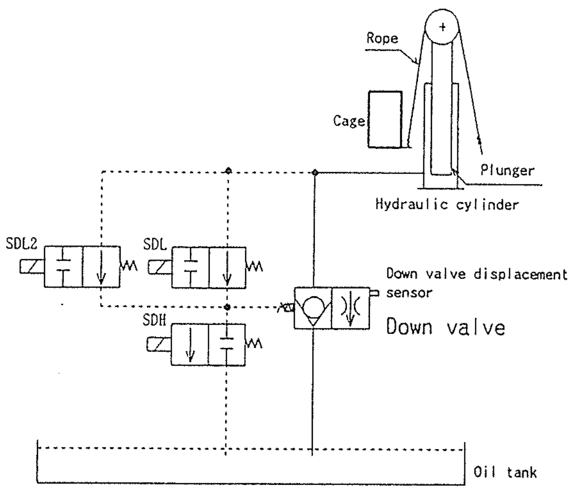


Fig. 8 Hydraulic circuit for down control valve

4. Experimental results

Performance verification tests were carried out with an actual elevator from the control panel being equipped with the developed valve displacement feedback PWM control system, and the test results were evaluated.

4.1 Acceleration/deceleration speed control characteristic

Fig.9 shows the no-load up-down characteristic of an elevator having a speed of 60m/min and a loading capacity of 750kg as an example of the speed control characteristic of the valve displacement feedback PWM controlled long-stroke hydraulic elevator. The speed curve from the start to the stop of the elevator changes smoothly to show good controllability.

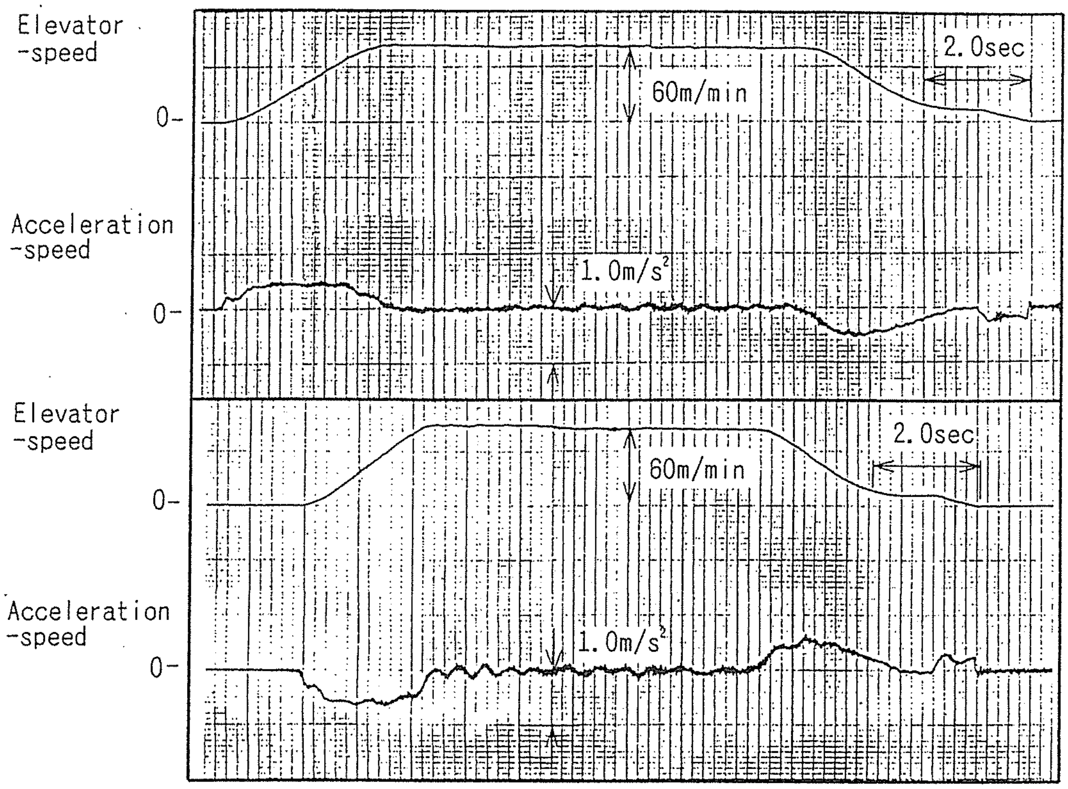


Fig.9 Running characteristics(up run /down run)

5. Conclusion

In this system, the speed control was done based on the mathematical expression model after systematizing this model for the hydraulic control valves as the controlled systems. As a result, the following improvement was done successfully.

(1) A stable running characteristic could be obtained without affecting any start-up vibrations, rated running speed, landing running speed, landing run time, and stop vibrations by the oil temperature and load pressure.

(2) The valve displacement command preparation system was established by taking the flow characteristic of the main valve into account.