A MINIMUM ENERGY CONTROL OF A LOAD-SENSING HYDRAULIC SERVO SYSTEM

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S.D. Kim and H.S. Cho

* Research Engineer, Division of Mechanical Engineering
** Professor, Department of Production Engineering
Korea Advanced Institute of Science and Technology,
P.O.BOX 150, Chongryangri, Seoul, Korea

Abstract

The dynamic characteristics of a load-sensing hydraulic servo system are complex and highly unstable. Another property of the system is that the setting value of pump compensator is closely related to energy efficiency as well as control performance of the system. This necessitates the development of an effective control algorithm which guarantees good control performance, stability and energy efficiency. This paper considers a suboptimal PID control for the velocity control problem of the load-sensing hydraulic servo system. The results of simulations studies and experiments show that the proposed suboptimal controller can produce much better control performance than nonoptimal controllers and give effective energy efficiency.

1. Introduction

The load-sensing hydraulic system utilizing a variable displacement pump is an energy saving system to minimize power loss by the load-sensing mechanism[1-3]. However, its dynamic characteristics are very complex and highly nonlinear. Furthermore, the stability characteristic is critically deteriorated compared with that of the conventional system due to addition of the load sensing mechanism[4]. This is because the interaction between dynamics of the pump part and that of the load part becomes stronger due to addition of the load-sensing mechanism. These features significantly add complexity to the controller design of the hydraulic servo system. For example, an advanced modern control such as a state feedback based upon a dynamic observer has many difficulties in actual implementation. In this paper, a simple proportional-plus-integral-plus-derivative (PID) control based upon output feedback is considered for easy implementation of the load-sensing hydraulic servo system.

There are several classical approaches to design the PID controller of servo systems. The root locus method is one of the classical controller design methods. In designing a linear control system, the root locus method is quite useful since it enables the designer to find the closed-loop poles from the open-loop poles and zeros by providing a graphic display, and decide a controller gain[5]. The classical tuning rule of the PID controller, which was proposed by Ziegler and Nichols [6, 7] have advantages to determine the PID controller gains at a time and produce a reasonable performance for open-loop stable systems. However, these classical approaches still have difficulties to be applied to the controller design of the load-sensing hydraulic servo system. The root locus method is not appropriate because this method is effective in tuning only one parameter of a linear control system. The Ziegler-Nichols rule, also, is not appropriate for highly unstable systems because this method can be applied to a system which is open-loop stable and responds to step change in a nonoscillatory manner. Therefore, it is difficult to obtain a satisfactory performance by employment of the Ziegler-Nichols rule in the highly unstable load-sensing system. Therefore, it is necessary to develop an effective design method of the PID controller suitable for the load-sensing hydraulic system.

There is an important property in addition to the above control problem of the load-sensing system, which is that a setting value of the pump compensator is closely related to energy efficiency as well as control performance of the system [1, 2]. In order to accomplish not only good control performance but also high energy efficiency of the system, decision of the setting pressure as well as the controller design should be carefully done in the design stage of the servo system.

The purpose of this paper is to suggest a suboptimal method to simultaneously determine the PID controller gains and the setting pressure, and to demonstrate effectiveness of this suboptimal method. The problem considered herein is to control the velocity of a load inertia operated by the load-sensing servo system. In order to deal with this control problem, the investigations were made in the following procedures. (1) A mathematical model was derived, which reasonably represents the complex dynamics of the load-sensing hydraulic system. (2) A subopitmal design method of the PID controller was presented. (3) Based upon the derived model and the design method, suboptimal PID gains and suboptimal setting pressure were obtained. The effectiveness of the suboptimal system was illustrated through computer simulation works and experiments. The suboptimal output responses were compared with nonoptimal output responses, and the energy efficiency of the suboptimal system was discussed in some details.

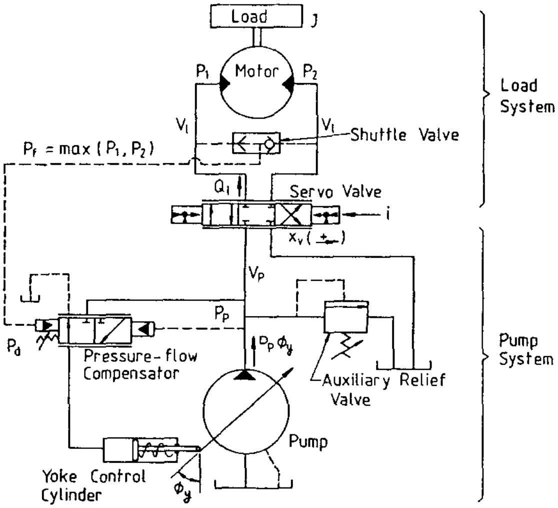


Fig. 1 Structure of the load-sensing hydraulic system

2. The Dynamic Model of the System

There are two types of the load-sensing mechanism in hydraulic systems: (1) load-sensing pump control, and (2) load-sensing valve control. The pump control type is more efficient from an aspect of the energy saving effect than the valve control type. In this paper, the load-sensing hydraulic system utilizing the pump control is treated. The overall structure of the system is shown in Fig.1. The structural difference of the load-sensing system from a conventional system utilizing a variable displacement pump is that the load pressure(P_l) is fed back to the pump compensator. A mathematical model to represent the load-sensing system can be formulated in the following manner.

The dynamic equations representing the load inertia and hydraulic motor consist of an equation of motion of the load inertia and a flow continuity equation of the line volume[8]. These equations are given by

$$J\ddot{\theta} + B\dot{\theta} + T_e = D_m P_l \tag{1}$$

$$Q_l = D_m\dot{\theta} + L_l P_l + (V_l/2\beta)\dot{P}_l \tag{2}$$

where $\dot{\theta}$ is the angular velocity of the load inertia, T_e is the external torque applied to the motor, P_l is the load pressure, and Q_l is the load flow of the servo valve. The model equation of the servo valve can be simplified as a first-order dynamic equation of the spool motion and a static equation for the control flow (Q_l) of the servo valve. They can be written as

$$T_v \dot{x}_v + x_v = K_v i \tag{3}$$

$$Q_l = \mathrm{sgn}(P_p - \mathrm{sgn}(x_v)P_l)K_t x_v \sqrt{|P_p - \mathrm{sgn}(x_v)P_l|} \tag{4}$$

where i is the input current of the servo valve, x_v is the spool displacement of the servo valve, T_v is an equivalent time constant, and P_p is the pump outlet pressure. The dynamic model of the load-sensing pump shown in Fig.1 can be simplified by several assumptions without significant error[9]; small volume of the yoke control cylinder, negligible inertia torque of the yoke and negligible friction torque of the yoke motion. Then, a simplified model of the pump can be given as

$$D_p\phi_y - L_p P_p - \mathrm{sgn}(x_v)Q_l = (V_p/\beta)\dot{P}_p \tag{5}$$

$$D_c\dot{\phi}_y - L_c P_c = K_p(\max(P_1, P_2) + P_d - P_p)\sqrt{WP_p - YP_c} \tag{6}$$

$$(G - R)\phi_y = D_b P_p - D_c P_c + G\phi_o \tag{7}$$

where P_p is the pump pressure, ϕ_o and ϕ_y is the initial angular position and angular displacement of the yoke respectively, P_c is the pressure of the yoke control cylinder, P_d is the setting pressure adjusted by a compensator spring, and the other parameters are spring constant, leakage coefficients and flow coefficients. In the equation (6), W and Y are the auxiliary variables to describe the nonlinearity of the pump compensator. The $\max(P_1, P_2)$ means a feedback pressure to the pump compensator, which is chosen as the larger pressure between two pressures of motor ports P_1 and P_2. The pressures of P_1 and P_2 can be described by the pump pressure (P_p) and the load pressure(P_l) from the symmetry condition of the servo valves' orifices[8]. These variables can be described as

$$P_1 = (P_p + P_l)/2, \quad \text{and} \quad P_2 = (P_p - P_l)/2 \tag{8}$$

$$\begin{aligned} W = 0, \ \text{and} \ Y = -1 \quad &\text{for } \max(P_1, P_2) + P_d \geq P_p \\ W = 1, \ \text{and} \ Y = 1 \quad &\text{for } \max(P_1, P_2) + P_d < P_p \end{aligned} \tag{9}$$

Based upon the mathematical model of equations (1)-(9), a suboptimal control method will be presented.

3. Effect of the Setting Pressure

The setting pressure represents a pressure drop of the servo valve and it is directly related to power loss which is dissipated into heat. The power loss of the system may be described as a steady loss since the power loss for transient response time is negligibly small in common hydraulic systems. Then, the power loss can be described as the following equation.

$$L = P_p(t_s)\, Q_p(t_s) - P_l(t_s)\, Q_l(t_s) \tag{10}$$

where t_s is the settling time, P_p is the pump outlet pressure, Q_p is the flowrate generated by the pump, P_l is the load pressure and Q_l is the load flow. The first term of the Eq. (10) denotes a power generated by the pump, while the second term denotes a power required in driving the load. Thus, the difference between the two terms is a wasted power.

If the leakage flows of L_pP_p and L_cP_c are assumed to be negligible in the equations of (5) and (6) and the system is in a steady state, the pump flow of $D_p\phi_y$ is same with the load flow (Q_l) and the pump pressure (P_p) is the value of the feedback pressure ($\max(P_1, P_2)$) plus the setting pressure (P_d). That is, the pump flow (Q_p) in the Eq. (10) is nearly same with the load flow (Q_l)

and the pressure difference between the pump pressure (P_p) and the load pressure (P_ℓ) is kept constant by the setting pressure (P_d) of the pump compensator. Thus, the power loss of the system is found to increase proportionally with the setting pressure. The setting pressure should be as low as possible in order to minimize the power loss of the system. However, an excessively small setting pressure deteriorates control performance of the servo system [1]. The low setting pressure reduces acceleration rate of the pump pressure and the load pressure and, thus, deteriorates the capacity to drive the load inertia. Therefore, the setting pressure affects the control performance as well as the power loss, and it should be determined as a trade-off value between the control performance and the power loss.

4. Suboptimal Controller Design

The suboptimal PID control problem of the load-sensing hydraulic servo system can be considered based upon its open-loop model which is formulated in the previous section. When the PID control is applied to the load-sensing hydraulic system, the schematic block diagram of the closed-loop velocity control system is shown in Fig. 2. The PID control may be described by

$$u = K_p e + K_i \int_0^t e \, d\tau + K_d \dot{e} \qquad (11)$$

where $e = \dot{\theta}_d - \dot{\theta}$. $\dot{\theta}_d$ is the desired load velocity and $\dot{\theta}$ means the output velocity, and u indicates the control current input (i) of the servo valve. The controller gain parameters K_p, K_i, and K_d are the proportional gain, integral gain, and derivative gain, respectively.

A suboptimal control problem to tune the PID controller gains for an arbitrarily chosen setting pressure (P_d) may be formulated as a standard servo problem. The control objective, then, is to minimize the cost of a performance index which represents the square of the error between the desired velocity and actual velocity, and the control input energy. The performance index J_c is described as

$$J_c = \int_{t_0}^{t_s} [\{\dot{\theta}_d - \dot{\theta}(t)\}^2 + \rho_1 u(t)^2] dt \qquad (12)$$

where ρ_1 is a weighting factor, and t_0 and t_s are the initial time and the settling time respectively.

The PID controller gains for a pre-chosen setting pressure are determined so that the cost of the performance index is minimized. However, the standard suboptimal system may not accomplish an effective energy efficiency of the system because the performance index of the Eq. (12) considers only the control performance without consideration of the energy efficiency. In order to achieve the effective energy efficiency, the PID controller should be designed by try-and-error method for various values of the setting pressure. This problem can be simply solved by adding the power loss of the Eq. (10) to the performance index of the Eq. (12), and regarding the setting pressure as another design parameter. Then, the new performance index J_e is described as

$$J_e = \rho_2 [P_p(t_s) Q_p(t_s) - P_\ell(t_s) Q_\ell(t_s)] + \int_{t_0}^{t_s} [\{\dot{\theta}_d - \dot{\theta}(t)\}^2 + \rho_1 u(t)^2] dt \qquad (13)$$

where ρ_1 and ρ_2 are weighting factors, and t_0 and t_s are the initial time and the settling time, respectively. The settling time is defined by a time at which the output response firstly reaches a steady state and stays within a specified small bound. The control input u(t) is constrained by a saturation value u_{sat}, which is given by

$$|u(t)| \leq u_{sat} \qquad (14)$$

Also, the control input has been indirectly constrained by including the input energy term in the performance index to avoid large current input.

The optimization algorithm used in this work was the Rosenbrock algorithm[10]. Because the Rosenbrock algorithm is based on the direct search method, the dynamic equations (1)-(9) and the controller equations of (11) and (14) should be solved to obtaine the system responses at each iterative operation of this algorithm. The system responses for a given initial condition were obtained by numerical integration via the fourth order Runge-Kutta algorithm. The numerical step size was 1 msec.

5. Simulations and Experimental Results

The suboptimal PID controller gains and suboptimal setting pressure have been obtained via the suboptimal design method. The values of the system parameters used in this work are listed in

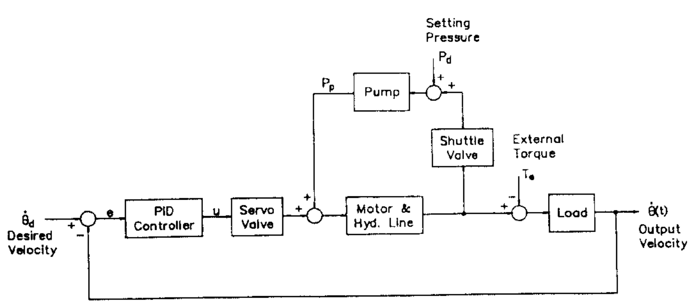


Fig. 2 Schematic block diagram of a load-sensing hydraulic servo system using a PID controller

Table 1. The system parameters

B = 0.03 Nm/(rad/sec)	D_b = 1.283×10^{-5} m^3/rad	D_c = 4.434×10^{-5} m^3/rad
D_m = 1.321×10^{-6} m^3/rad	D_P = 5.582×10^{-3} m^3/rad	G = 26.44 Nm/rad
i = 0 - 20 mA	J = 0.002 - 0.05 Kg-m^2	K_v = 2.5×10^{-5} m/mA
K_f = 7.45×10^{-2} (m^3/sec)/m	K_P = 1.76×10^{-14} $m^6/(N^{3/2}$ sec)	L_c = 3.51×10^{-6} (m^3/sec)/MPa
L_ℓ = 3.88×10^{-6} (m^3/sec)/MPa	L_P = 1.58×10^{-5} (m^3/sec)/MPa	P_d = 0.2 - 1.8 MPa
R = 18.92 Nm/rad	T_e = 0.0 Nm	T_v = 0.0071 sec
V_ℓ = 1.83×10^{-4} m^3	V_P = 1.7×10^{-3} m^3	β = 6.86x10 MPa
ϕ_o = 1.03 rad		

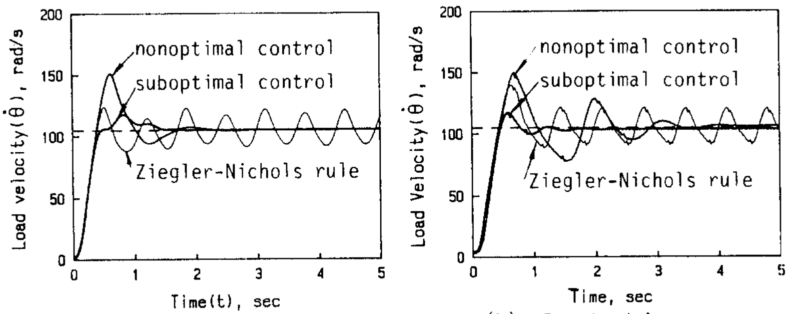
Table 1. The initial condition is in a state that the servo valve is fully closed and the load inertia is standing still. The desired output velocity is 105 rad/sec (1000 revolutions per minute). The weighting factors of ρ_1 and ρ_2 are chosen to be 0.75 and 0.2 respectively.

A series of computer simulations and experiments were performed in order to illustrate the effectiveness of the suboptimal design method. An experimental test rig was installed, which is identical to the load-sensing hydraulic servo system shown in the Fig. 1 and Fig. 2. The pump, the servo valve and the hydraulic motor used in this work are Sperry-Vickers PVB series 29 pump, Moog 73-102 servo valve and Nonzelli 40002-MC piston motor. Their geometrical and dynamic data are the same as the values listed in Table 1. The auxiliary relief valve shown in the Fig. 1 is set as 12 MPa (about 120 bar) for safety's sake and the pump outlet pressure is limited by this value. The output signal of the load velocity is measured by use of a tacho generator and this signal is filtered by a analog filter with cutoff frequency of 20 Hz. The computer simulations are similarly done with the experimental procedure, and the equations of (1)-(9) and (11) are solved via the Runge Kutta method under the input constraint of the Eq. (14). With aid of the results of computer simulations, various properties of the load-sensing hydraulic system can be effectively observed.

The computed suboptimal controller gains and setting pressure are obtained to be K_P=0.164 mA/(rad/s), K_i=0.912 mA/s(rad/s), K_d=0.044 mAs/(rad/s) and 0.83 MPa, respectively. In Fig. 3, the suboptimal response of the load velocity is compared with two nonoptimal responses. One of two nonoptimal responses plotted in this figure is obtained from a controller whose PID gains are computed via the Ziegler-Nichols rule for the suboptimal setting pressure of 0.83 MPa, and the other response is obtained from a controller whose PID gains are arbitrarily chosen for the setting pressure of 0.83 MPa. The PID controller gains based upon the Ziegler-Nichols rule are computed as K_P=0.110 mA/(rad/s), K_i=1.090 mA/s(rad/s), and K_d=0.0028 mAs/(rad/s), and the gains of the other nonoptimal controller are chosen as K_P=0.100 mA/(rad/s), K_i=1.500 mA/s(rad/s) and K_d=0.050 mA/s(rad/s). The Fig. 3 shows that the suboptimal response reaches the desired velocity within a reasonably short time and without an excessive overshoot, while the response based upon the Ziegler-Nichols rule is unstable, showing a limit cycle. The nonoptimal response whose controller gains are chosen in an arbitrary manner has an excessive overshoot and a relatively long settling time. This figure clearly shows that the suboptimal system has much better control performance than the nonoptimal control system.

From observation of the controller gain values of the three different systems, one remarkable point is that the system obtained via Ziegler-Nichols rule is unstable due to the low derivative action, while the suboptimal system and nonoptimal system with large K_d gain are stable. This fact shows that the highly unstable load-sensing system should have a large derivative action, and the Ziegler-Nichols rule is unsuitable for this kind of system.

Various responses of three different suboptimal systems, such as, the responses of the load velocity ($\dot{\theta}$), pump pressure responses (P_P) and load pressure (P_ℓ) are plotted in the Fig. 4. The system of the Fig. 4-(a) is the suboptimal system whose setting pressure and controller gains are determined as optimal values to minimize the performance cost of the Eq. (13). Whereas, the systems of the Fig. 4-(b) and Fig. 4-(c) are so-called standard suboptimal systems whose controller gains are determined as optimal values to minimize the performance cost of the Eq. (12) for the pre-chosen setting pressures. The effect of the setting pressure can be observed from the Fig. 4. When the setting pressure is low, i.e. P_d=0.3 MPa in the Fig. 4-(b), the load velocity response rises in a slower rate than the case of the Fig. 4-(a). Furthermore, the velocity response has a steady state offset because the setting pressure of the pump compensator is lower than a limit value at the maximum opening of the servo valve. It is, thus, observed that the low setting pressure deteriorates the control performance of the system. However, the power loss is very small due to the small pressure drop between the pump outlet pressure and the load pressure. On the contrary to the low setting pressure, when the setting pressure is much high, i.e. P_d=1.5 MPa in the Fig. 4-(c), the power loss of the system is much greater than that of the system with the setting pressure of 0.83 MPa, which is shown in the Fig. 4-(a). These results indicates that the setting pressure should be as low as possible to minimize the power loss but the higher setting pressure gives the faster response. From such a view point, the suboptimal system of the Fig. 4-(a), which has the suboptimal controller gains and the suboptimal setting pressure, may be concluded to satisfy the energy efficiency as well as the control performance of the system. Therefore, the proposed suboptimal design method can be concluded to be an effective method for the load-sensing hydraulic servo



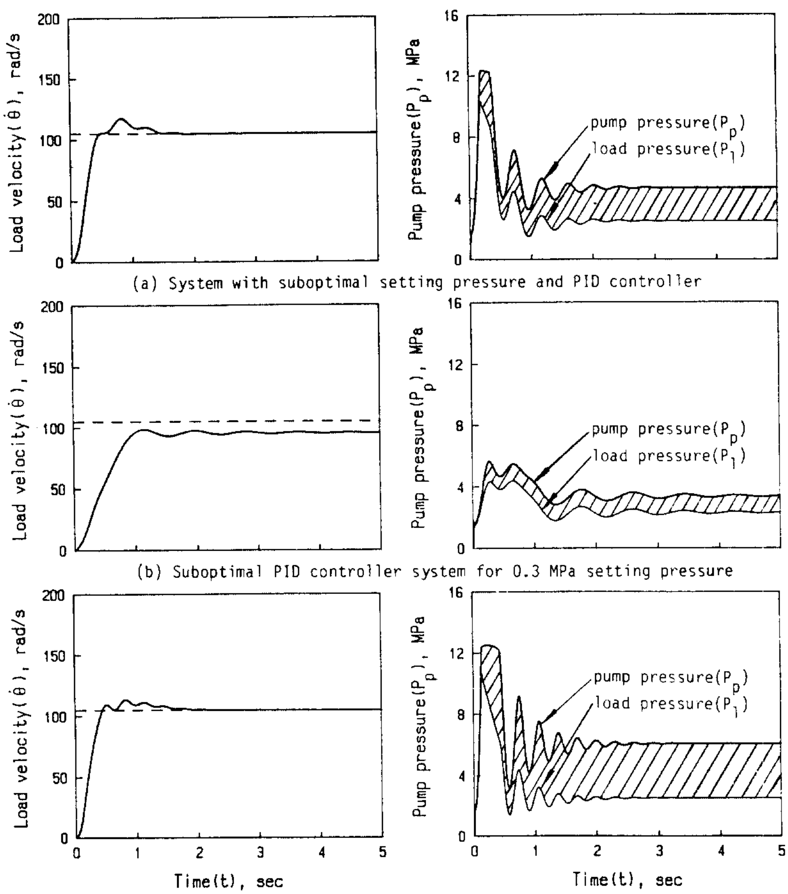
(a) Theoretical responses (b) Experimental responses

Fig. 3 The load velocity responses of the three different control systems

Suboptimal : K_p=0.164, K_i=0.912, K_d=0.044, P_d=0.83

Nonoptimal : K_p=0.100, K_i=1.500, K_d=0.050, P_d=0.83

Ziegler-Niccols : K_p=0.110, K_i=1.090, K_d=0.0028, P_d=0.83



(a) System with suboptimal setting pressure and PID controller

(b) Suboptimal PID controller system for 0.3 MPa setting pressure

(c) Suboptimal PID controller system for 1.5 MPa setting pressure

Fig. 4 The control performance and the power loss of the three different suboptimal systems : Simulation results

(a) K_p=0.164, K_i=0.912, K_d=0.044, P_d=0.83

(b) K_p=0.061, K_i=0.515, K_d=0.064, P_d=0.30

(c) K_p=0.250, K_i=0.763, K_d=0.047, P_d=1.50

system.

6. Conclusions

A mathematical model of a load-sensing hydraulic servo system has been derived. Based upon the model, a suboptimal design method to determine not only the gains of the PID controller but also the setting pressure of the pump compensator has been considered. Through a series of simulation studies and experiments, the effectiveness of the design method was illustrated by comparison with nonoptimal cases.

Based upon the results obtained from the simulations and the experiments, the following major conclusions can be made:

(1) The suboptimal control system has much better control performance than several nonoptimal control systems.

(2) The setting pressure is another important parameter which should be determined at the controller design stage, since it greatly influences the control performance and the energy efficiency of the system.

(3) The suboptimal design method system accomplish an effective trade-off between the control performance and the power loss of the system.

These results implicate that the proposed suboptimal method can be effectively applied on the load-sensing hydraulic servo system.

Acknowledgments

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References

[1] Oliver W. Johnson: Load Sensing Hydraulic Flow Control - A Versatile Controller, Society of Automotive Engineers, paper no. 780728 (1978).

[2] K. R. Lonnmo: A Method to Compare Control Losses for Various Mobile Hydraulic Systems, Society of Automotive Engineers, paper no. 831272 (1983).

[3] A. Qureshi: Load Sensing Techniques, The Journal of Basic Fluid Power Research program, pp. 351-362 (1982).

[4] S.D. Kim, H.S. Cho and C.O. Lee: Stability Analysis of a Load-sensing Hydraulic System, Proc. Institution of Mechanical Engineers, (submitted for publication).

[5] Katsuhiko Ogata: Modern Control Engineering, Prentice-Hall,Inc. Englewood Cliffs, N.J. Chapt.8, pp. 339-353 (1970).

[6] J.G. Ziegler and N.B. Nichols: Optimum Setting for Controllers, ISEJ, pp. 731-734 (1964).

[7] S. Thompson: Multivariable PID Controller for Unidentified Plant, ASME Trans. Journal of Dynamic Systems and Measurement and Control, Vol. 104, pp. 270-274 (1982).

[8] H.E. Merritt: Hydraulic Control Systems, John Wiely and Sons, Inc., Chapt.6, pp. 132-145 (1976).

[9] S.D. Kim, H.S. Cho and C.O.Lee: A Parameter Sensitivity Analysis for the Dynamic model of a Variable Displacement Axial Piston Pump, Proc. Institution of Mechanical Engineers, VOl. 201, No C3, (1987).

[10] James L. Kuester and Joe H. Mize: Optimization Techniques with Fortran, McGraw-Hill, New York, Chapt.9, pp. 320-330 (1984).

[11] P.J. Fleming and C.W. Jones: Determination of Optimal Constant Feedback Gains for Nonlinear Systems, Proc. IEE Vol. 126, No. 2, pp. 267-269, (1979).