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DESIGN OF A NEW CONTROLLER FOR A HYDRAULIC SERVOMECHANISM

By

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ABSTRACT

This work deals with the investigation of the effect of a more recent, powerful control algorithm, called Pseudo Derivative Feedback, PDF, on the performance of a hydraulic servomechanism. The controlled system behavior is theoretically evaluated by developing a mathematical model for the controlled system and by using a computer simulation program. Also, the experimental measurement of the system dynamic performance is conducted. The developed simulation program is validated by comparing theoretical and experimental results of the system response. The simulation program is used to deduce a transfer function for the studied servomechanism. Once the transfer function of the controlled system is obtained, the control algorithm implementation is performed. To define the advantages of the proposed control algorithm, its output is compared to that of the already existing unity feedback system. Finally, A proposal for the design of the PDF controller is presented.

NOMENCLATURE

A	Restriction area of the directional control valve m^2.
A_p	Hydraulic cylinder Piston area, m^2.
A_c	Area of the piston of the integrating member, m^2.
C_d	Coefficient of discharge through restriction areas.
C_c	Flow gain of the spool valve of integrating member, m^2/s.
F	Externally applied load, N.
f	Viscous damping coefficient, N.sec/m.
G_r	Representative model transfer function.
K	Transfer function gain of representative model.
K_d	Pseudo first derivative coefficient.
K_i	Pseudo integrating coefficient.
K_p	Pseudo proportional coefficient.
K_c	Proportional controller coefficient.
L	Feed back link dimension, m.
M	Reduced mass of piston and moving parts, kg.

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P Pressure, N/m^2.
Q Flow rate, m^3/sec.
R_i Internal resistance to leakage flow, $N.sec/m^5$.
r_1&r_2 Radii of the controller spur gears. m.
T_r Rise time, s.
T_s Settling time, s.
V_o Initial volume of the hydraulic cylinder chamber,m^3.
x Spool displacement, m.
y Piston rod displacement, m.
y_1 Displacement of the controller piston, m.
y_2 Proportional element rack displacement, m.
y_i Displacement of the spool of the integrator, m.
Z Control rod displacement, m.
ΔP Pressure drop through restriction orifice, N/m^2.
ρ Fluid density, kg/m^3.
ξ Damping coefficient.
ω_n Natural frequency, Hz.

SUBSCRIPTS

a,b,c&d corresponding to the valve four restrictions.

ABBREVIATIONS

DCV Directional control valve
PDF Pseudo Derivative Feedback controller.
UFB Unity feedback system.
FCE Final control element.
SSE Steady state error.

INTRODUCTION

The hydraulic servo mechanisms are widely used in the aircraft control systems. They enable to control the position of the aircraft control surfaces subjected to great aerodynamic forces. These servomechanisms consist, in general, from a symmetrical hydraulic cylinder, a directional control valve and a proportional feedback mechanism. Glickman [1], presented a description of the common types of the hydraulic servomechanisms.

As a basic element of the aircraft control system, the hydraulic servomechanism, HSM, is actually operating in the transient conditions most of the time. Therefore, it is important to study its dynamic behavior and to evaluate the effect of the different constructional and operational parameters on its behavior.

The static and dynamic behavior of the hydraulic servomechanisms and their basic elements has been studied in numerous publications [2 to 9]. Rodden [2], gave a representative mathematical model of a high performance HSM. He discussed in detail its flow characteristics. Lambert and Davis [3], deduced the transfer function of a HSM by applying the small perturbation theory. They evaluated the damping effect of the coulomb friction and internal leakage. Martin [4], presented a study on the effect of unequal oil volumes, in the two chambers of hydraulic cylinder, on the stability and response of a HSM. He presented also an experimental and theoretical investigation of the displacement and pressure responses of a closed loop HSM to very large step inputs, where the flow rate saturates for substantial periods. He concluded that such a system does not cavitate during the transient period [5]. Martin and Keeting [6] evaluated theoretically the effect of the loading conditions and damping effect of the valve restrictions on the frequency response of a HSM. The effect of the internal leakage in the hydraulic cylinder on the HSM static and dynamic performance has been evaluated experimentally and theoretically by Rabie and Rashed [7]. Baz, [8&9] studied theoretically and experimentally the dynamic performance of a HSM equipped with a built in damper and determined the optimum diameter of the dampers orifice.

The usually used solid negative feedback mechanism has the advantage of simple design, small dimensions and mass and low production and maintenance costs. But it acts, continuously starting from the early part of the transient period, to reduce the throttling area of the valve restrictions. It reduces in this way the flow rate delivered to the hydraulic cylinder and increases, consequently, the response time. Therefore, when faster response is recommended, a new feedback mechanism should be developed.

Phelan [10], presented a new simple control algorithm; so called Pseudo Derivative Feedback, PDF. The behavior of this controller is compared with that of a simple proportional controller [11], and with a proportional plus integral controller [12]. These studies showed that the PDF controller presents shorter response time because of the greater rate of energy delivery during the early part of the response.

This paper deals with the theoretical and experimental study of the transient response of a HSM equipped with a proportional feedback mechanism and the deduction of a representative model for this HSM. It deals also with the improvement of the transient response of the HSM by designing a new PDF controller.

MODELLING OF THE STUDIED HYDRAULIC SERVOMECHANISM

This study is applied to the hydraulic servomechanism type ZL-3a. This unit consists of a 4/3 directional control valve, a symmetrical hydraulic cylinder and a solid negative feedback rod. The used directional control valve is of matched symmetrical type. A scheme of the studied HSM is given in Fig. 1.

The control of the hydraulic servomechanism is of servo type, that is, the distance, speed and direction of the piston movement are directly correlated to those of the spool of the DCV. The servo unit is equipped with a feedback lever, which acts to move the spool of the DCV to its neutral position when the piston displacement equals that of the control rod.

A mathematical model of the studied system is deduced, considering the following assumptions:

1- The DCV of the servomechanism is of matched symmetrical type;
$A_a(x) = A_b(-x)$ $\qquad$ $A_c(x) = A_d(-x)$
$A_a(x) = A_c(x).$ $\qquad$ $A_b(x) = A_d(x)$
2- The spool valve is of zero-lapped type.
3- The system has perfect external tightness.
4- The effect of elasticity of the cylinder walls is neglected, compared with that of the fluid compressibility.
5- The inner hydraulic lines are of negligible resistance.
6- The feedback lever is of negligible mass.

Taking into consideration these assumptions, the following mathematical relations describe the function of the studied HSM.

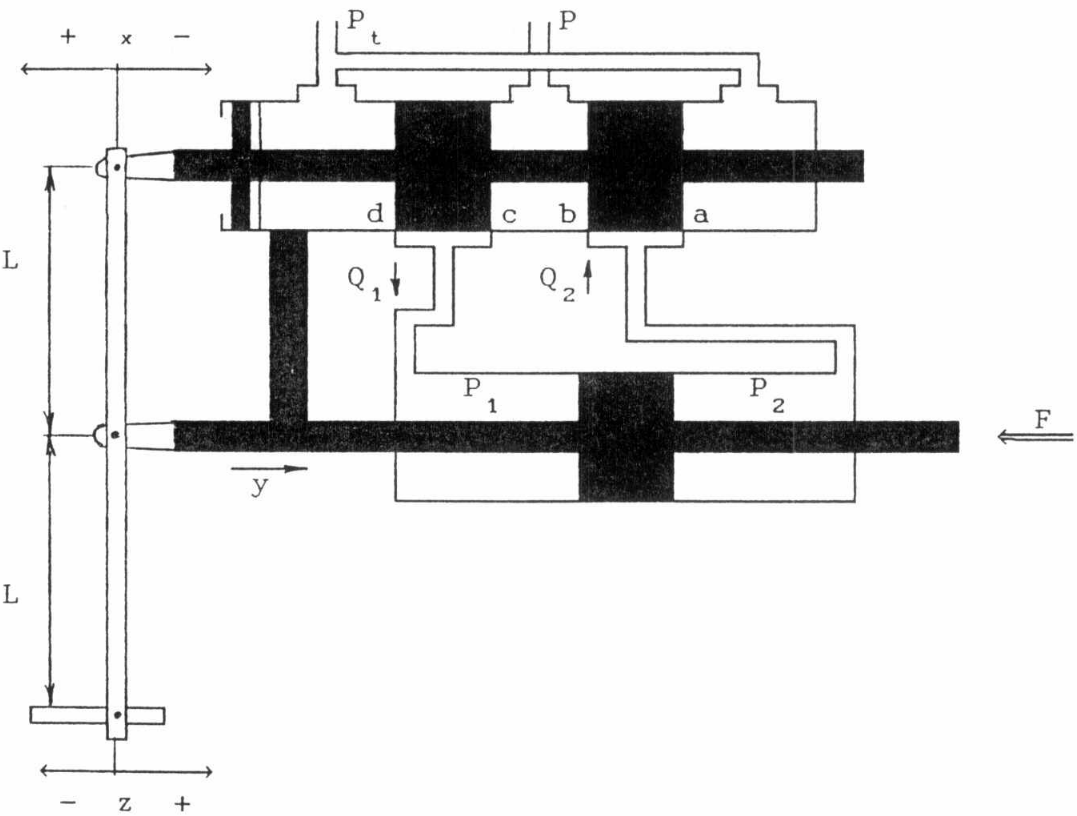


Fig.1 Scheme of the hydraulic servomechanism type ZL-3a

$$Q = C_d A(x) \sqrt{2 \Delta P/\rho} \tag{1}$$

$$Q_a = C_d A_a(x) \sqrt{2 (P_2-P_t)/\rho} \tag{2}$$

$$Q_b = C_d A_b(x) \sqrt{2 (P - P_2)/\rho} \tag{3}$$

$$Q_c = C_d\, A_c(x)\, \sqrt{2\,(P - P_1)/\rho} \qquad (4)$$

$$Q_d = C_d\, A_d(x)\, \sqrt{2\,(P_1 - P_t)/\rho} \qquad (5)$$

$$Q_1 = Q_c - Q_d \qquad (6)$$

$$Q_2 = Q_a - Q_b \qquad (7)$$

$$\left.\begin{aligned} A_a &= A_c = A(x) \\ A_b &= A_d = 0 \end{aligned}\right\} \text{For } x > 0. \qquad (8)$$

$$\left.\begin{aligned} A_b &= A_d = A(-x) \\ A_a &= A_c = 0 \end{aligned}\right\} \text{For } x \le 0. \qquad (9)$$

$$A_P\,(P_1 - P_2) = M\, d^2y/dt^2 + f\, dy/dt + F \qquad (10)$$

$$Q_1 - A_P\, dy/dt - Q_i - [(V_o + A_P y)/\, B]\, dP_1/dt = 0 \qquad (11)$$

$$A_P dy/dt + Q_i - Q_2 - [(V_o - A_P y)/\, B]\, dP_1/dt = 0 \qquad (12)$$

$$Q_i = (P_1 - P_2)/R_i \qquad (13)$$

$$x = Z - y \qquad (16)$$

The area of the directional control valve restriction, A(x), changes nonlinearly with the spool displacement. The variation of this area with the displacement x is calculated and plotted in Fig.2.

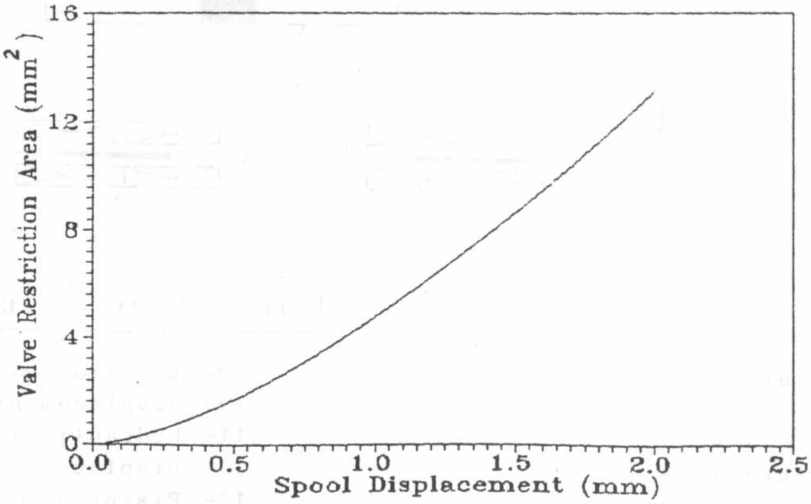
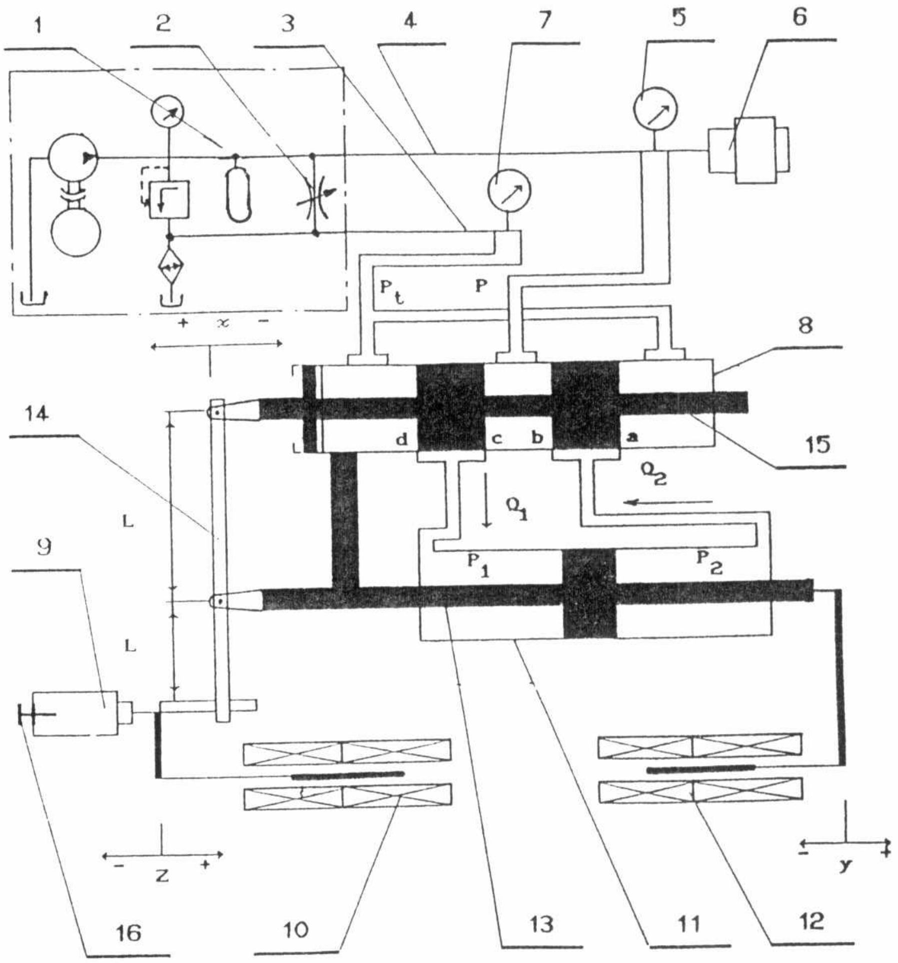


Fig.2. Variation of the directional control valve restriction area with the spool displacement

SYSTEM SIMULATION

The studied HSM is described mathematically by equations 2 to 16. The simulation of the HSM is carried out by the exploitation of these equations using the TUTSIM simulation program [13]. This program establishes the analogy between the real system and computer language. It has the convenience of the analog computers and the speed and accuracy of the digital computers. The simulation by this program is based on the pictorial representation of the mathematical model in terms of functional blocks of the simulation program . The available blocks enable to take into consideration, practically, all of the system nonlinearities as well as the generation of the recommended inputs. The constructional parameters of the studied system have been found out by applying direct measurements on the elements of the studied HSM.



1- Hydraulic accumulator	9- Electric solenoid
2- By-pass valve	10- displacement LVDT
3- Return line	11- hydraulic cylinder
4- Pressure line	12- Displacement LVDT
5- Pressure gauge	13- Piston rod
6- Pressure LVDT	14- Feedback lever
7- Pressure gauge	15- Spool of DCV
8- Directional control valve	16- Stroke adjusting screw

Fig.3 Test setup

EXPERIMENTAL WORK

The transient response of the studied HSM is evaluated experimentally. Figure 2 shows the arrangement of the built test rig. The HSM is supplied by the pressurized oil from a constant pressure hydraulic generator. The control lever is displaced rapidly by using a high power electric solenoid (9) of adjustable stroke. The input displacement, z, and piston rod displacement, y, are picked up by means of two LVDT transducers (10 & 12).

Keeping the supply pressure constant (8.2 MPs), the transient response of the HSM is recorded for different magnitudes of the input displacement, with the HSM unloaded. A part of the experimental results are plotted in nondimensional form in Fig.4.

The recorded input displacement z(t), is introduced in the simulation program. The transient response of the HSM is calculated, taking into consideration the operating conditions of the experimental work. The calculations were carried out for different magnitudes of input displacement. The theoretical results are plotted in Fig. 4. The study of these results shows good agreement between the theoretical and experimental results, specially on the level of the rise time, settling time and maximum percentage overshoot. This good agreement validates the developed model and the simulation program. Therefore the simulation program can be used to predict the static and dynamic behavior of the studied HSM.

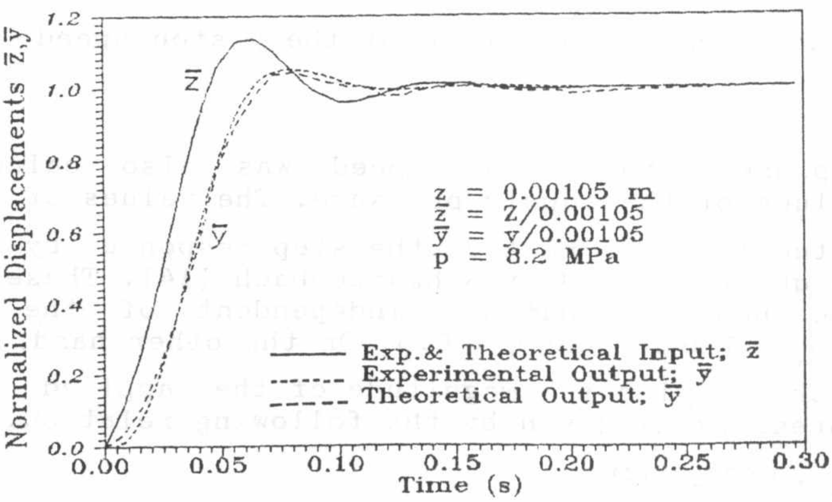


Fig.4 Experimental and simulation results of the transient response of the HSM with unity feedback

DEVELOPMENT OF THE PDF CONTROLLER

The development of the PDF controller necessitates the representation of the controlled plant by means of a transfer function. The parameters of this controller are calculated in terms of the coefficients of the characteristic equation. Therefore, the simulation program is used to develop a representative model; transfer function, for the studied HSM.

The transient response of the piston speed, v(t), to step spool displacement x(t) of different magnitudes was calculated. The calculation results, plotted in Fig.5, show that this response is similar to that of underdamped second order element. Therefore the transfer function relating the piston speed to the spool displacement is assumed to have the following form.

$$G_r = \frac{V}{X}(s) = \frac{K}{s^2/\omega_n^2 + 2\xi s/\omega_n + 1}$$

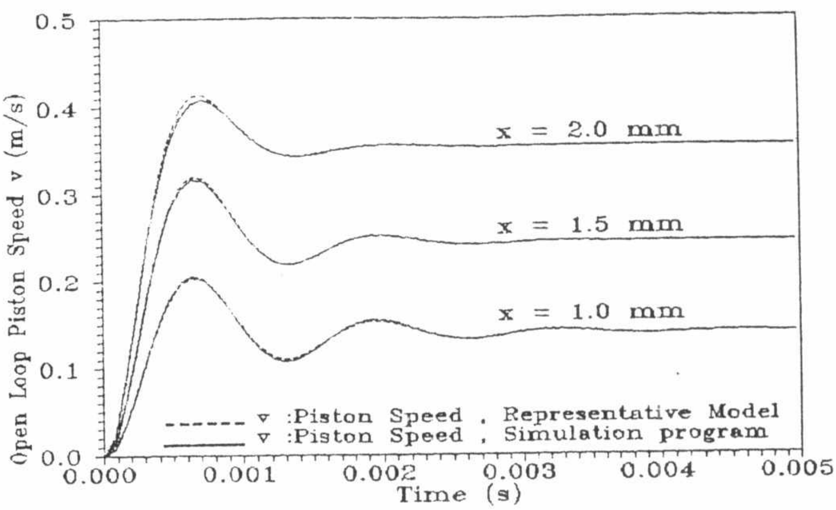


Fig.5 Transient response of the piston speed

The step response of the piston speed was also calculated for different values of the supply pressure. The values of ω_n and ξ were calculated from the plot of the step response by using the simple technique presented by Schwarzenbach [14]. These values are found to be nearly constant, independent of the operating conditions; ω_n = 750 Hz and ξ = 0.5. On the other hand, the gain is found to change with the magnitude of the applied step input and supply pressure as given by the following relation.

$$K = 245\ P^{0.532} x^{0.421}$$

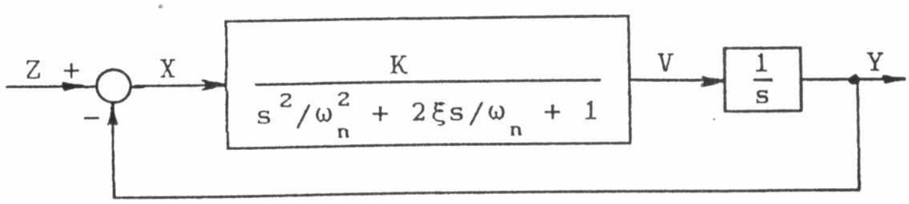


Fig.6 Block diagram of the representative model of the HSM with unity feedback

The transient response of the piston speed, calculated by this representative model is also plotted in Fig.5. This figure shows very good agreement between the response of the simulation program and that of the representative model.

Considering the HSM with unity feedback, its overall representative model is given in Fig.6.

The step response of the piston rod displacement of the HSM with unity feedback is calculated by using the representative model and the simulation program. The calculation results are plotted in Fig. 7, which shows excellent agreement between the response of the two models. Therefore the transfer function of the representative model is used for the calculation of the parameters of the PDF controller.

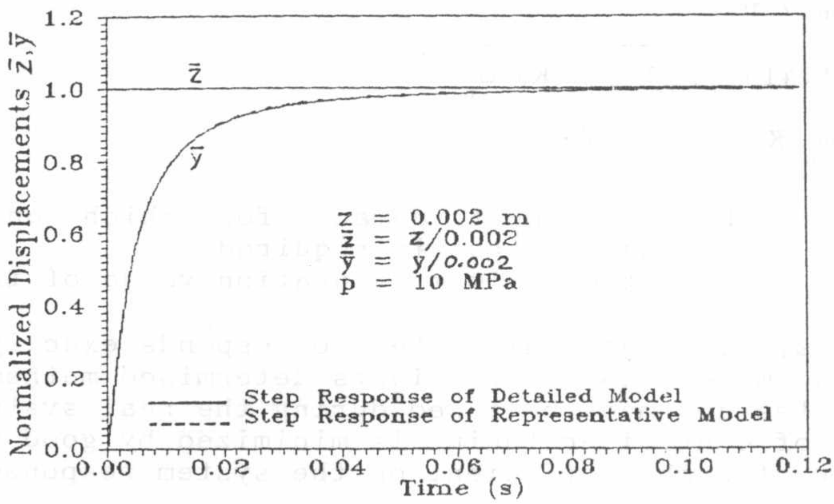


Fig.7 Step response of the HSM with unity feedback, calculated by the simulation program and representative model

A block diagram of the feedback controller, developed by Phelan [10], is given in Fig. 8. In this study, the derivative term; K_{ds}, is found to have no significant effect on the system behavior, therefore it is neglected. The final control element, FCE, is chosen as a simple proportional element of unity gain. it acts also as a position limiter, limiting the maximum spool displacement. The block diagram of the HSM, described by its transfer function, equipped with the PDF controller is given in Fig. 9.

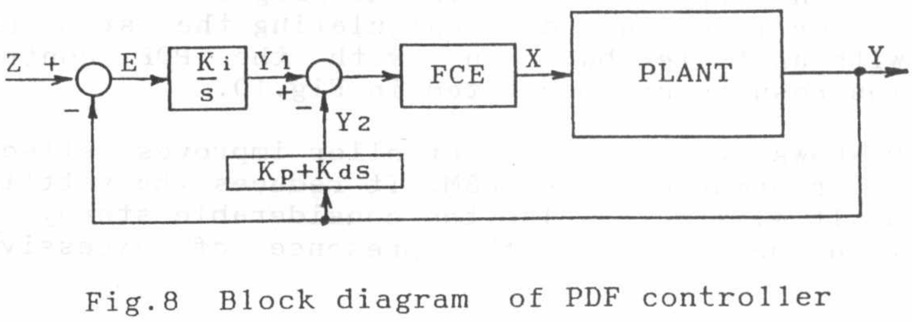


Fig.8 Block diagram of PDF controller

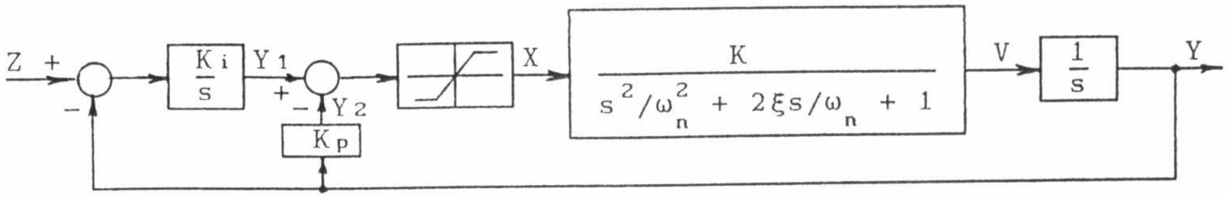


Fig.9 Block diagram of the HSM with PDF controller

A number of formulae for estimating a first guess of the controller coefficients are developed by Flower [15]. These formulae are deduced on theoretical and experimental bases. In the case of the studied HSM, these formulae are as follows.

$$K_p = m / R$$

$$K_d = 1.414 \sqrt{2 \xi K K_p/\omega_n}$$

$$K_i = \omega_n (K K_d + 1)/10\xi$$

Where R is the input demand for which an essentially linear output is required.
m is the maximum saturation value of the FCE.

In all cases, since no real system corresponds exactly to the used mathematical model, the coefficients determined mathematically are only estimates and must be tuned during the real system operation. The effort of controller tuning is minimized by good understanding of the role of each coefficient on the system response.

The coefficient K_i acts to increase x, therefore increasing K_i will increase the system speed of response. This in turn, increases the possibility of overshooting and decreases the system stability. However, a large enough value of K_i reduces the steady state error.

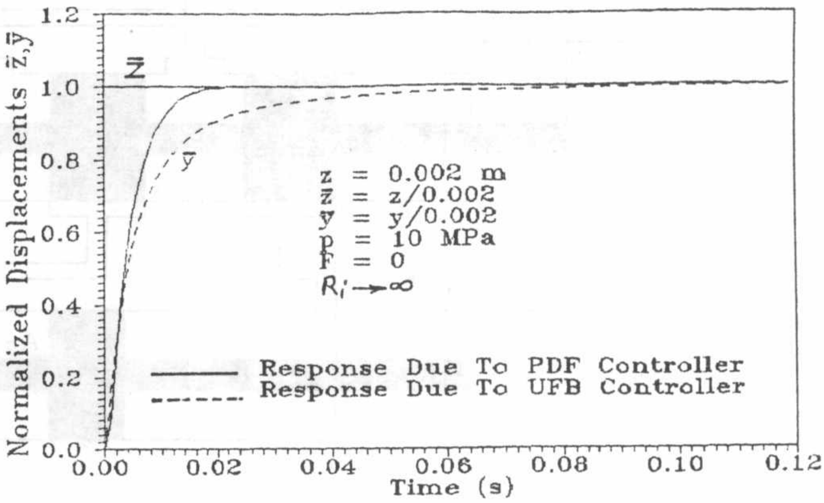
The coefficient K_p is more effective as the response approaches the steady state, i.e. when y is relatively large and dy/dt approaches zero. Therefore, increasing K_p will eliminate overshoot, but it will also slow down the response near the steady state value.

In this study the tuning of the parameters is carried out by using the validated simulation program of the HSM, replacing the unity feedback by the PDF controller in the program. The effect of the PDF controller is evaluated by calculating the step response of the HSM with unity feedback and with the PDF controller. The calculation results are presented in Fig.10.

Figure 10 shows that the PDF controller improves effectively the transient performance of the HSM. It reduces the settling time and rise time. It eliminates also the considerable steady state error appearing in the case of the presence of excessive leakage

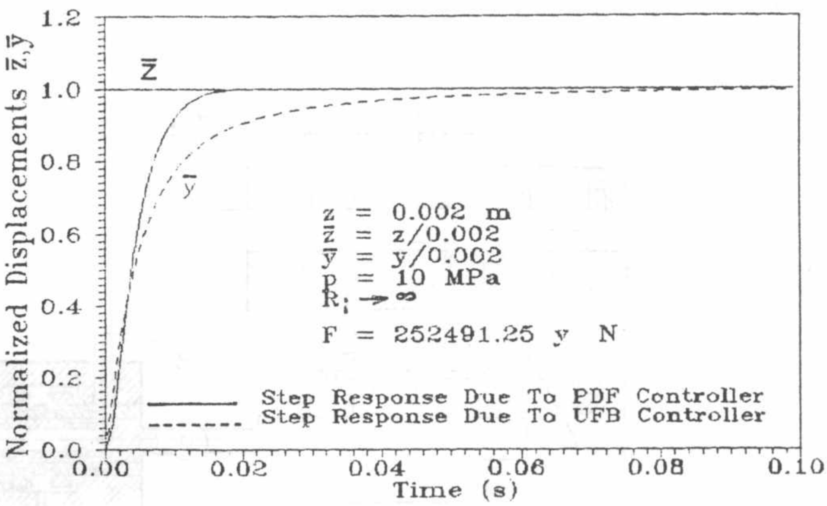
(A)

	PDF	UFB
Tr (ms)	7.2	18.3
Ts (ms)	12	30
SSE %	0	0



(b)

	PDF	UFB
Tr (ms)	7.7	18.7
Ts (ms)	11.5	30.3
SSE %	0	0



(c)

	PDF	UFB
Tr (ms)	9.5	20.4
Ts (ms)	12	50
SSE %	0	4.5

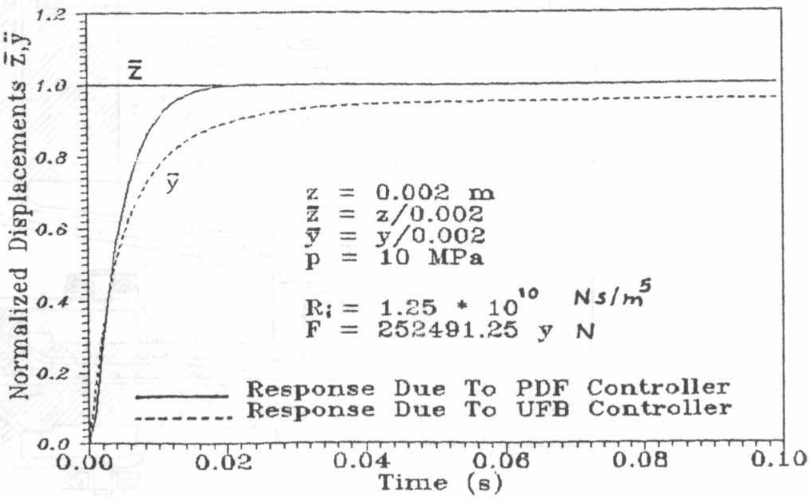


Fig.10. Step response of the HSM equipped with the PDF controller and with unity feedback.

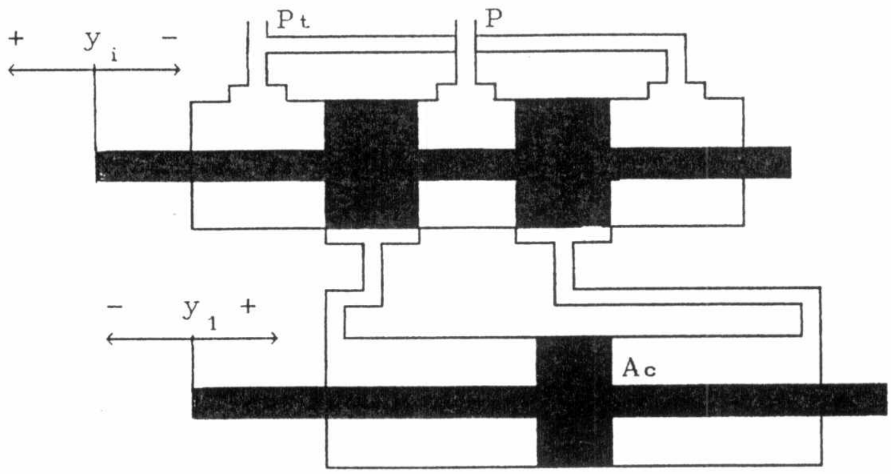


Fig.11 Valve controlled actuator used as an integrating element

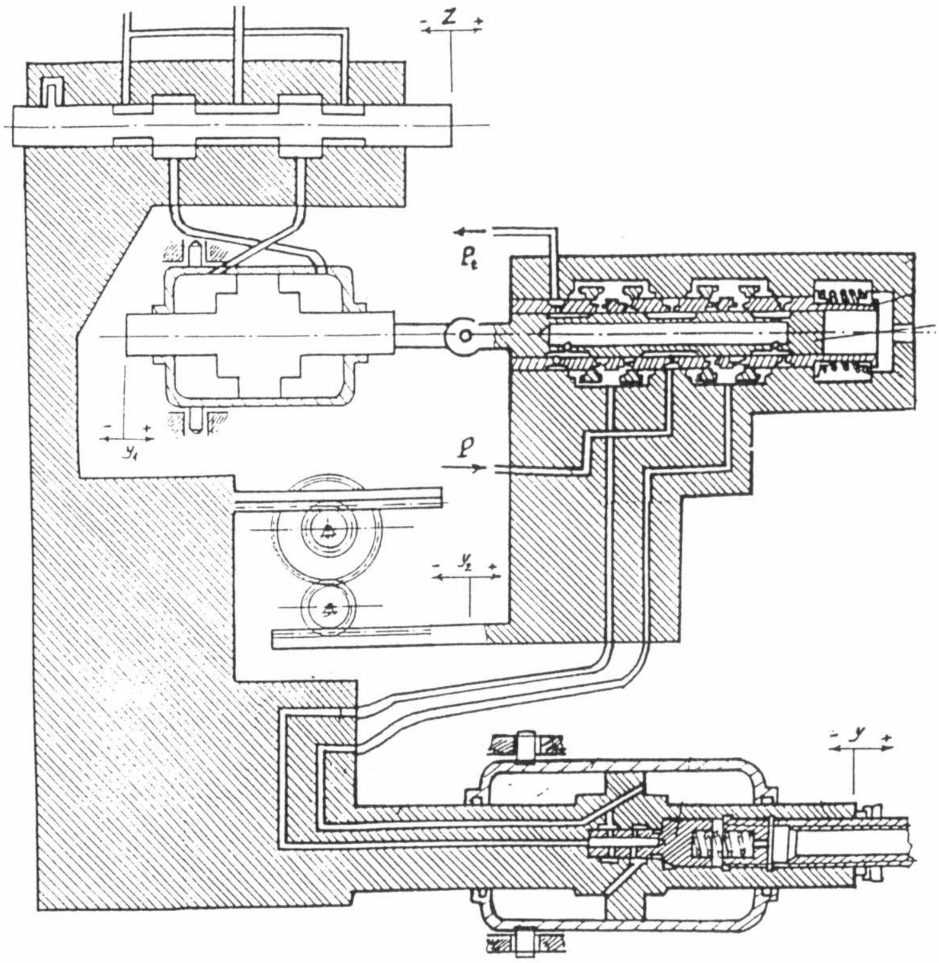


Fig.12 Scheme of the HSM equipped with PDF controller

DESIGN OF THE PDF CONTROLLER

In the case of the electrohydraulic servosystems, the PDF controller can be easily implemented in the electronic card of the servovalve or in the computer program of the controller. But in the case of the HSM, a special design of the controller should be carried out by introducing additional mechanical and hydraulic elements.

The proposed PDF controller, Fig. 9, includes two basic elements; an integrating element in the forward path and a proportional element in the feedback. The integrating element can be realized by using a valve controlled actuator, Fig.11. The constructional parameters of this element should satisfy the following requirements:

* The spool valve is of ideal, matched symmetrical type.
* The viscous friction is negligible.
* Negligible inertia of moving parts and transmission lines.
* Perfect internal and external tightness.

If these conditions are satisfied, the valve controlled actuator acts as a simple integrating element. Its transfer function is given by the following relation:

$$Y_1/Y_i = K_i/s$$

The proportional element can be designed as two racks with two spur gears as shown in Fig.12. The proportionality coefficient is given by $K_p = y_2/y = r_2/r_1$. A proposal for the layout of the HSM equipped with the PDF controller is given in Fig. 12.

CONCLUSION

This paper deals with the design of a pseudo derivative feedback, PDF controller for a hydraulic servo mechanism, HSM. A simulation program is developed for the HSM equipped with its unity feedback mechanism. The simulation program is validated on the basis of experimental results and used to deduce the transfer function of the HSM.

The PDF controller is developed and its parameters were calculated by using the coefficients of the transfer function of the HSM. The effect of the PDF controller is evaluated theoretically by comparing the transient response of the HSM equiped by the PDF the controller with that of the HSM equiped by the original unity feedback. The study showed that the PDF introduced the following improvement on the response of the HSM.

* Considerable reduction of the settling and rise times.
* Elimination of the steady state error appearing in the case of the increase of the internal leakage in the cylinder.

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