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Linear Analysis to Control an Electro-hydraulic System Using Proportional Directional Control Valve

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Abstract

In the field of hydraulic control system the performance including the stability of the system are important parts of hydraulic system working. In this work electro-hydraulic control circuits including a proportional directional control valve have been designed.

A mathematical model developed to determine the natural frequency and damping ratio using Matlab software.

To find out the stability of the system a proportional plus, integral plus, derivative (PID controller) control was used.

The analysis was conducted with an experimental work to study the pressure drop, loading pressure, and flow rate and extension velocity with and without external load.

A comparison between the theoretical and experimental work and literature showed an agreement results.

الخلاصــــــة

ان اداء المنظومات الهيدروليكية واستقراريتها من العوامل المهمة لتحديد عمل المنظومة لذا تم تصميم دائرة كهروهيدروليكية تحتوي على صمام تناسبي. تم اشتقاق موديل رياضي لايجاد الدبدبة الطبيعية (Natural Frequency) ونسبة التخميد (Damping Ratio) باستخدام برنامج حاسوب (Matlab software).ولايجاد استقرارية المنظومة تم استخدام دائرة السيطرة (PID controller).

ان التحليل النظري تمت مقارنته مع الجانب العملي والذي تظمن قياس الانخفاض في الضغط وقوة الضغط والجريان وحركة المدك باضافة حمل خارجي وبدون حمل.

تمت مقارنة النتائج العملية والنظرية وكذلك تمت مقارنتها مع المصادر وكانت النتائج جيدة.

1. Introduction

Hydraulic drives are often applied in practical system e.g. in concrete pumping manipulators that operate in wide ranges or carry heavy loads.

As the main components for such a drive in this paper a cylinder and a proportional valve are investigated. A linear mathematical modeling of the hydraulic system proves to be very useful for a controller determination.

Rich. Barton etal [1] presented a simple method to estimate a proportional valve parameters such that, spring constant viscous damping coefficient, spring pretension and flow force spring constant. This estimation associated with main stage of a proportional solenoid valve. In general found that a large spool displacement was preferable because information along the complete stroke of the valve could be obtained.

Zheng D.L. [2] presented a method for modeling on electro-hydraulic system. Without directly using the conventional linear model. Accurate models for the electro-hydraulic components have been constructed using the characterization data provided by manufactures and measurements obtained in the laboratory with accustom build electro-hydraulic manipulator. A commonly a vailable general-purpose simulation package, simulink has been used to solve the non-linear differential equations.

Vanghan N.D and Gamble J.B [3] presented models of various configuration of control valves and hysteretic modeling in proportional valves. These models provide some understanding of complex interactions between electrical, mechanical and fluid aspects of hydraulic control valves.

Margolis D.L [4] presented the fundamental smooth nonlinearity in hydraulic models. The square – root term in the orifice flow equations, always presented. This term is essential a nonlinear input gain, which varies as a function of chamber pressure in a cylinder attached to the valve. Non-smooth nonlinearities arise from geometric imperfections of the valve and its spool.

Zhang Q. [5] proposed fuzzy controller for electro-hydraulic steering. He conclude that fuzzy control technology can mimic a humanizes operating strategy in controlling complex system and can handle systems with uncertainty and nonlinearity and it's the most such controllers are designed based on natural language control laws, and presented the speed control of hydraulic cylinder. He used electro-hydraulic system feed forward-plus-PID control. The speed control of a hydraulic cylinder is a third-order system. Its dynamic behaviors are affected by spool valve characteristics, system pressure, and cylinder size.

Norgaard, M.etal [6] presented modeling of hydraulic actuator that is used for controlling the position of acran arm. The crane has four actuator: boom,arm,telescopic extention, and rotation of the whole crane.

It can be concluded from the above that most of the designers of hydraulic system had choosen one particular controlling or modeling type, depending on their machine type and its applications, so, the design of the system were depending on the application and the cost of the system.

2. Theoretical analysis

Fig. (1) Show the schematic diagram for analysis of the hydraulic control system. Although the dynamics of proportional directional control valves are highly nonlinear, simplified expressions are employed in practice to express the flow rates through the valve ports. For a critical center valve connected to a double acting cylinder, they often used load flow equation [7], [8].

$$Q_L = K_q X_v - K_c P_L \tag{1}$$

Provides a linearized approximation of the valve dynamic as a function of spool opening and load pressure. In this expression, the load flow and load pressure are defined respectively as:

$$Q_L = \frac{Q_1 + Q_2}{2} \quad , \quad P_L = P_1 - P_2$$

Where:

K_q = Critical center valve flow gain m^2/s
Kc= valve pressure gain $m^4.s/kg$
PL= Load pressure N/m^2
QL= Load flow m^3/s
Xv= Valve spool position m
Where the subscripts refer to the valve output ports.

However, the load flow in equation (2.1) represents the average of the flows in the lines and does not equal to the instantaneous flow rate at each valve ports. Further more, this flow equation is only valid for a critical center valve.

An improved flow model is developed be figure (1), where the effect of valve lapping on the flow rates is incorporated into the model. This model provides a single set of rate equations for a generic proportional directional control valve, combining the cases of critical center, overlapped, and under lapped proportional valves. Under generally accepted assumptions such as incompressible fluid, the model simplifies considerable, which results approximation of the flow rates at the valve ports.

The accuracy of the model is determined through a non-dimensional analysis, and thus the results hold for any similar system. Given a set of hydraulic system parameters, the analysis allows a designer to determine if this flow model will provide an accurate representation of the valve dynamics for subsequent analysis and control design [9].

2-1 Non – Dimensional Analysis Insights from a Linear Model.

Considering the hydraulic system of fig (3) the application of the continuity equation to the two sides of the cylinder yields:-

$$\begin{aligned} \dot{P}_1 &= \frac{\beta}{V_1}[-\dot{V}_1 - R_{ip}(P_1 - P_2) + Q_1] \\ \dot{P}_2 &= \frac{\beta}{V_2}[-\dot{V}_2 - R_{ip}(P_1 - P_2) + Q_2] \end{aligned} \tag{2}$$

Where Rip is the internal leakage coefficient between the cylinder chambers [10]. Other parameters are defined in the nomenclature. Assumed that the piston is initially centered so that the chamber volumes can be modeled as:

$$\begin{aligned} V_1 &= V_{1,0} + A_1 X_p \\ V_2 &= V_{2,0} + A_2 X_p \end{aligned} \tag{3}$$

Where the initial chamber volumes are $v_{10}=v_{20}=v_{t/2}$ and v_t is the total cylinder volume including the connection fitting, etc.

For a double – acting cylinder with equal piston areas, assumed:

$A_p=A_1=A_2$, so that

$$\begin{aligned} \dot{V}_1 &= V_p . \ddot{X}_p \\ \dot{V}_2 &= -V_p . \dot{X}_p \end{aligned} \tag{4}$$

Assuming that the motion of the piston is limited to the region around the center of the cylinder, as written:

$$V_1 \approx V_2 \approx V_{t/2} \tag{5}$$

Hence the pressure relation equation (2) can be rewritten as:

$$\dot{P}_1 = \frac{2\beta}{V_1}[-A_p \dot{X}_p - R_{ip}(P_1 - P_2) + Q_1]$$
$$\dot{P}_2 = \frac{2\beta}{V_2}[-A_p \dot{X}_p - R_{ip}(P_1 - P_2) + Q_2] \tag{6}$$

The equation of motion of the load mass is [11]:

$$m\ddot{X}_p = F_{hyd} \tag{7}$$

Where m is the total mass of the piston and the load, and

$$F_{hyd} = A_p.P_L \tag{8}$$

Where F_{hyd} is the hydraulic force due to pressure differentia across the piston. Differentiating this and using equation (2-6) obtaining:

$$m\dddot{X}_p = \dot{F}_{hyd} = A_p.\dot{P}_L$$
$$m\dddot{X}_p = \frac{2\beta A_p}{V_t}[-2A_p\dot{X} - 2R_{ip}P_L + 2QL] \tag{9}$$

Where using the relation $P_L = m\ddot{X}_p / A_p$ can simplify equation (2-7) as:

$$m\dddot{X}_p + \frac{4\beta R_{ipm}}{V_t}\ddot{X}_p + \frac{4\beta A_p^2}{V_t}\dot{X}_p = \frac{4\beta A_p}{V_t}Q_L \tag{10}$$

Considering equation (8), could be seen that the dynamics of the hydraulic system is similar so that of a second order linear system in the variable $\dot{x}_p = v_p$ the piston velocity. However, it should be noted the load flow Q_L, is in general a nonlinear function of spool position and chamber pressure, and there fore the above differential equation is in fact nonlinear [12].

It is possible to that a simpler case where the load flow is given by equation (1).

In this case, the response of the system is linear and can be express in terms of the piston velocity using operator (D) methods.

$$\ddot{V}_p + 2\eta_h w_h \dot{V}_p + w_h^2 V_p = w_h^2 \frac{K_q}{A_p} X_v$$
$$\ddot{V}_p + 2\eta_h w_h \dot{V}_p + w_h^2 V_p = w_h^2 \overline{V}_p$$
$$D^2 V_p + 2\eta\ w_h D V_p + w_h^2 V_p = w_h^2 \overline{V}_p \tag{11}$$
$$(D^2 + 2\eta_h w_h D + w_h^2) V_p = w_h^2 \overline{V}_p$$
$$\frac{\overline{V}_p}{V_p} = \frac{D^2 + 2\eta_h w_h D + w_h^2}{w_h^2}$$

Where the hydraulic natural frequency and the hydraulic damping coefficient are defined respectively as: [11]

$$w_h = 2A_p\sqrt{\frac{\beta}{m.V_t}} \tag{12}$$

$$\eta_h = \frac{R_{ip} + K_c}{A_p}\sqrt{\frac{\beta.m}{V_t}} \tag{13}$$

The terms on the right hand side of equation (11) can be combined as:
$\overline{V}_p = (K_q / A_p) X_v$ So that, in terms of Laplace variable (S)

$$\frac{V_p(s)}{\overline{V}_p(s)} = \frac{w_h^2}{S^2 + 2\eta w_h S + w_h^2} \tag{14}$$

$$\frac{V_p(s)}{\overline{V}_p(s)} = \frac{1}{(S/w_h)^2 + 2\eta_h (S/w_h) + 1} \tag{15}$$

Not that variable $\overline{V_p}$ is equivalent to the velocity of the piston as a function of spool opening if the hydraulic fluid is incompressible. It also represents the average behavior of the piston velocity.

To obtain a non-dimensional representation of hydraulic system behavior define a new complex variable S = S/w_h that equation (15)
Becomes

$$\frac{V_p(s)}{\overline{V}_p(s)} = \frac{1}{S^2 + 2\eta_h S + 1} \tag{16}$$

This transfer function suggests that for hydraulic control system.

The relation between the actual and approximate piston velocities is in variant with system properties.

To test the response of transfer function equation (16) to hydraulic control system as:
(a)- Calculate hydraulic natural frequency by using (12).
(b)- Calculate hydraulic damping coefficient by using equation (13).
Table (1) explains the source of parameters for hydraulic control system.

2-2 PID-Controlled for Hydraulic Control System

The selection of the three coefficient PID controllers is basically a search problem in a three-dimensional space. Point in the search space correspond, to different selections of PID controllers three parameters. [9] A PID controller can be determine by many methods which leads to regulate the PID controller for example a trial-and error etc.

The pre-design method consist of three steps:
(1)- Slect the W_h of the close-loop system by specifying the settling time

$$t_s = \frac{4}{t w_h} \qquad or \qquad t_s = \frac{3}{\eta w_h}$$

(2)- Determine the three coefficients using the appropriate optimum equation. [9]
(3)- Determine aprefilter $G_c(s)$ so that the close-loop system transfer function, T(s) does not have any zeros.

Figure (4) shows the block diagram for transfer function of PID controller. To simulation step response for transfer function by using Matlab Software.
The simulation parameters calculated sets shown in table (2).

3. Experimental work

The main objective of the experimental work is to verify the mathematical model and its analysis. The study including the valve pressure drop, loading pressure, flow rate, extension velocity, constant system pressure control of cylinder with and without weight load by using proportional directional control valve type [4WRE 6081X/24 ZAM] and cylinder type [CD70F40/25-300Z11/01 HCDM11T].

3.1 Valve pressure drop

The change in the valve pressure drop is accompanied by a change in flow rate. The term "Valve pressure drop" means the sum of pressure losses at the meter - in (P A) and meter out (b T) throttle cross section. In Figure (3) the relationship between valve pressure drop (ΔP_v), system pressure (P_s), load pressure (P_L) and tank pressure (P_T) is shown.

P_A = pressure at connection B (bar)
P_B = pressure at connection A (bar)
ΔP_v = Valve pressure drop (bar)
P_S = System pressure (bar)
P_T = Tank pressure (bar)
P_L = Load pressure (bar)

$$P_L = P_A - P_B$$

For switching position (b)
$\Delta P_v = P_s - P_L$
For switching position (a)
$\Delta P_v = P_s + P_L$
V = S/t = extension velocity m/s
S = cylinder stroke m
t = extension time sec
A = piston area m^2
Q = flow rate = 60*V*A L/min

3-1-1 Cylinder control without Weight Load

The objective this work are to study the characteristics of the proportional directional control valve with electrical feed back of the control spool position.

In this experiment extending and retracting the cylinder without weight load, to calculate extension velocity (v), flow rate (Q), valve pressure drop (ΔP_v), loading pressure (P_L).

The hydraulic circuit constructed by fitted the hydraulic parts on the test bench as shown in Figure (4).

3-1-2 Cylinder control with Weight Load

In this work a weight load used to study the effect of this load on the valve pressure drop (ΔP_v), loading pressure (P_L), flow rate (Q) and extension velocity (v).
The hydraulic circuit constructed on the test bench as shown in Figure (5).

4. Results and Discussion

To improve the stability of the system, different parameters have been taken in consideration such as piston area A_p, total volumetric flow V_t, Bulk modulus β and the external mass (m) on the electro hydraulic systems. Figure (6, 7) shows the step response of transfer function with overshoot due to instability of the system. To overcome this overshoot and let the system to be stable a PID controller added to system. Different sets of parameter used (table-2) to overcome this short time. Figure (8) shows the step response of transfer function. The system becomes stable with a short time (4.5e-3 sec). This means that the natural frequency, damping coefficient and the bulk modulus of elasticity of the liquid are the main factors to get a good step response and stable system.

A comparison between the experimental and theoretical analysis and to verify the characteristics of the proportional direction control valves shown in figures (9, 11). All figures shows that the pressure drop (ΔP_v) decreases when the input signal increase (Spool stroke X_v). The theoretical value of the valve pressure drop less than the experimental value because of the deceleration of spool movement due to friction and after a certain time the theoretical valve pressure imminence to experimental pressure drop.

A figure (12) shows a comparison with results of ref [13] and the experimental work which shows an agreement between them.

5. Conclusion

- The stability of a hydraulic system needs a proper selection of the controller parameters to get a proper working.
- The natural frequency, damping ratio and Bulk modulus are the main factors to get response i.e. short time to overcome the overshoot and become a stable system.
- The performance of proportional directional control valve used should be checked experimentally and compare its results with documents.

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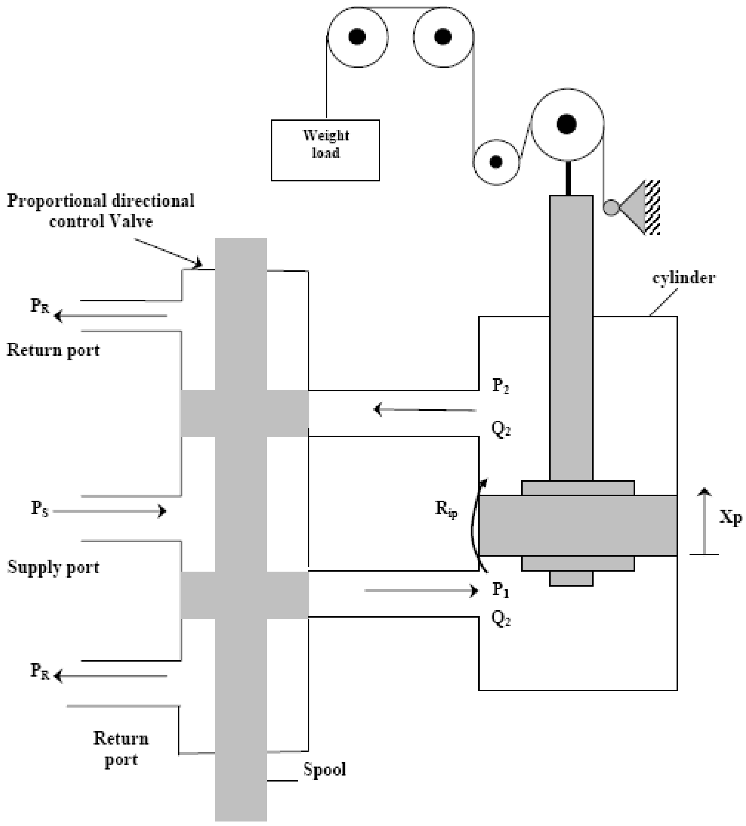

Figure (1) Schematic diagram of electro-hydraulic system

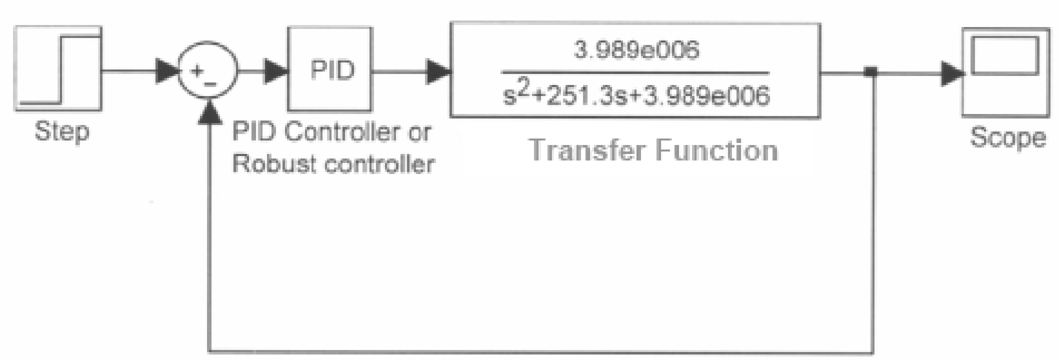

Figure (2) Block diagram of hydraulic control system with PID controller

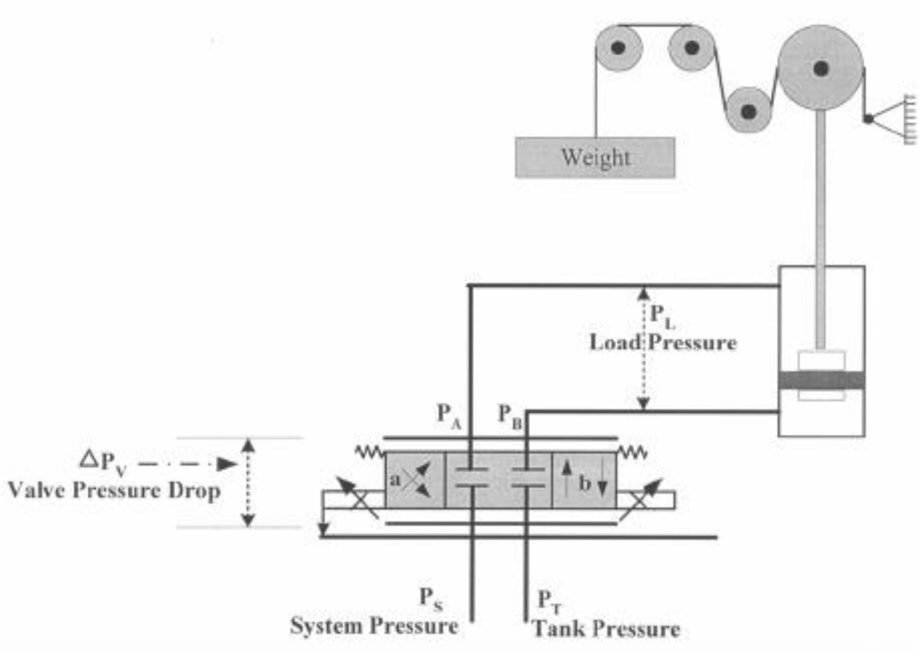

Figure (3) Designation of different pressure in a proportional valve

(1) Variable displacement pump.

(2) Pressure relief valve.

(3) Hydraulic cylinder.

(4) Proportional control valve.

(5) Gauge pressure.

(6) Check valve. (In pipe)

(7) Elect. Motor.

(8) Tank.

(9) Filter with check valve.

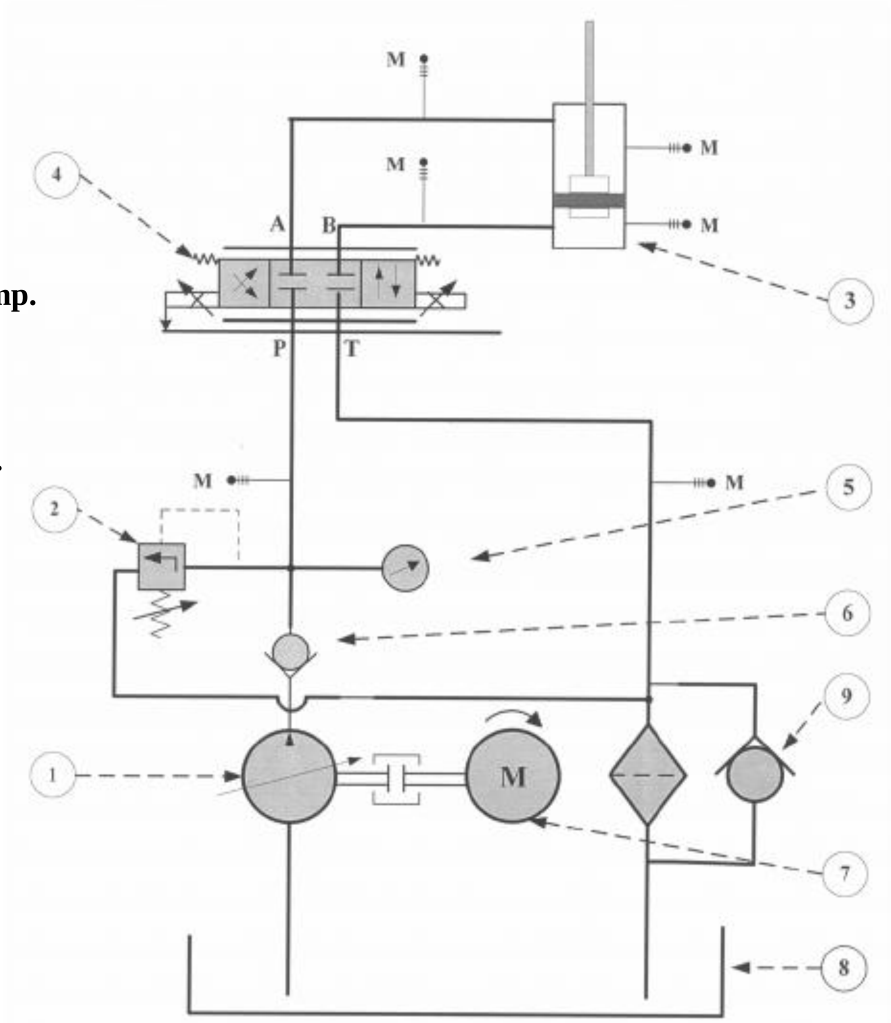

Figure (4) Hydraulic circuit without weight load

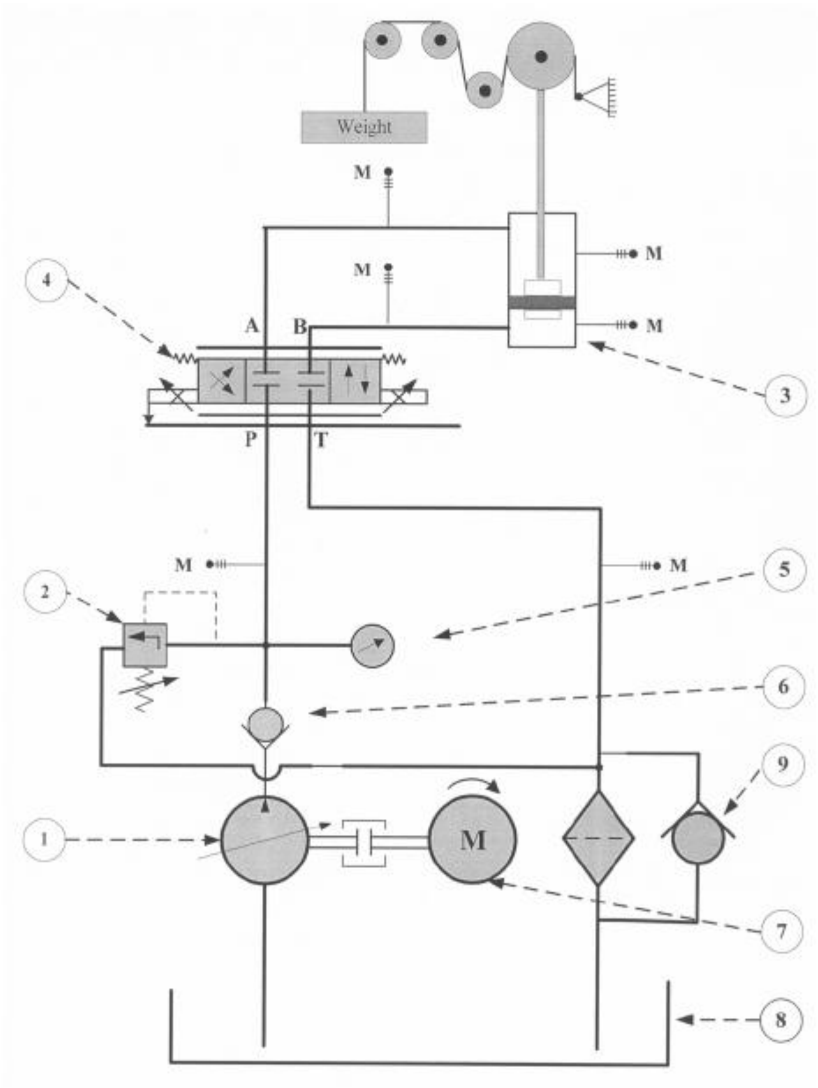

Figure (5) Hydraulic circuit with weight load

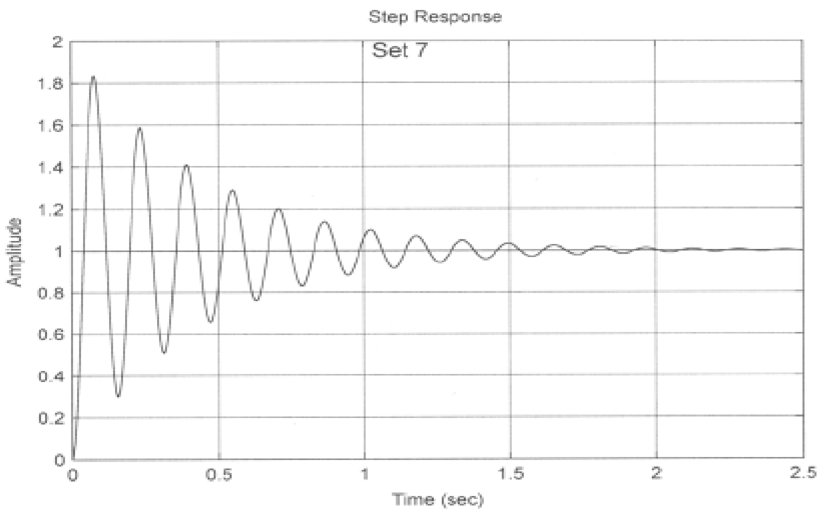

Figure (6) Simulation step response using set (7) of parameters

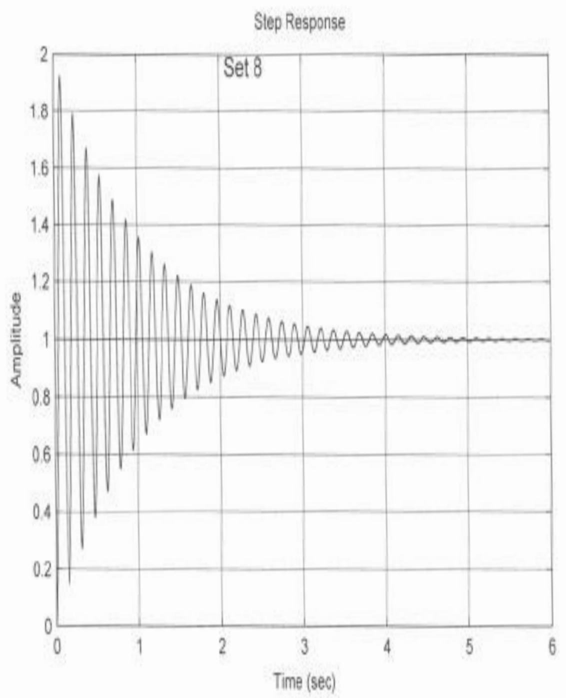

Figure (7) Simulation step respo
using set (8) of parameters

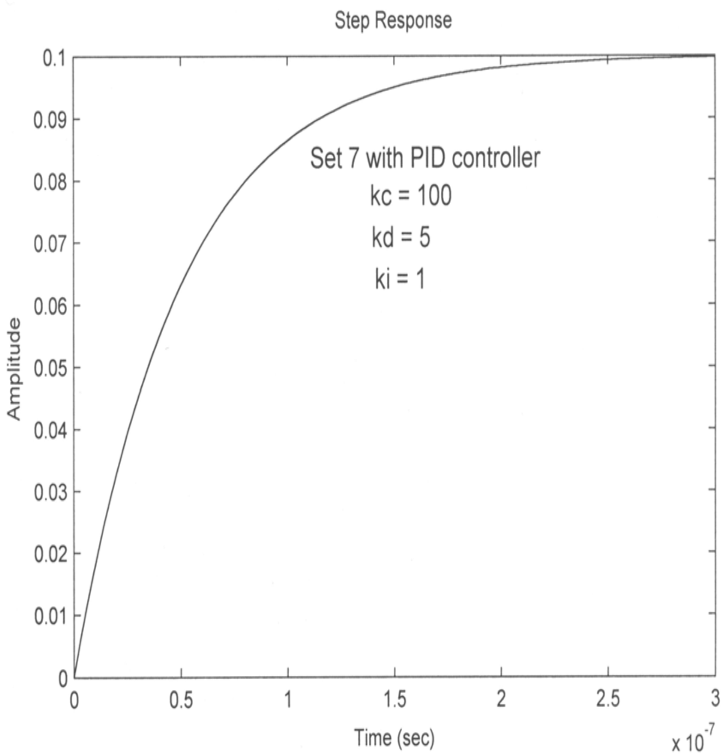

Figure (8) Simulation step response using set of parameters
wirth PID controller

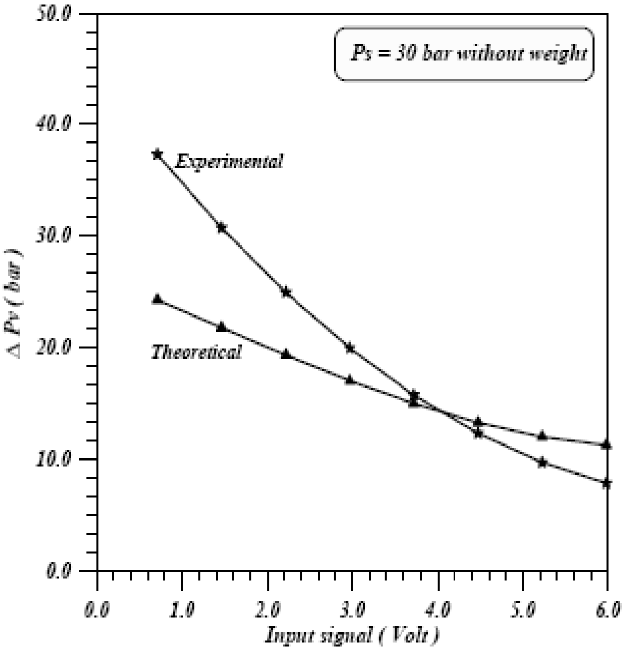

Figure (9) Variation of valve pressure drops (ΔP_v) with input signal for case (1)

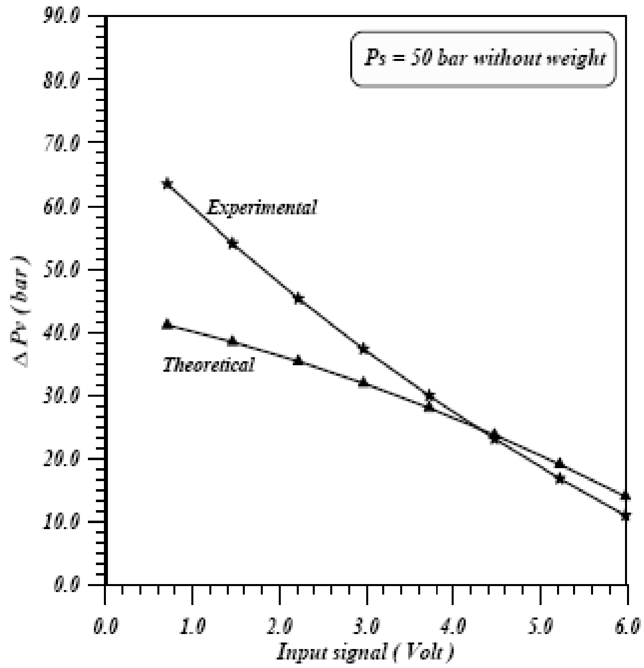

Figure (10) Variation of valve pressure drops (ΔP_v) with input signal for case (2)

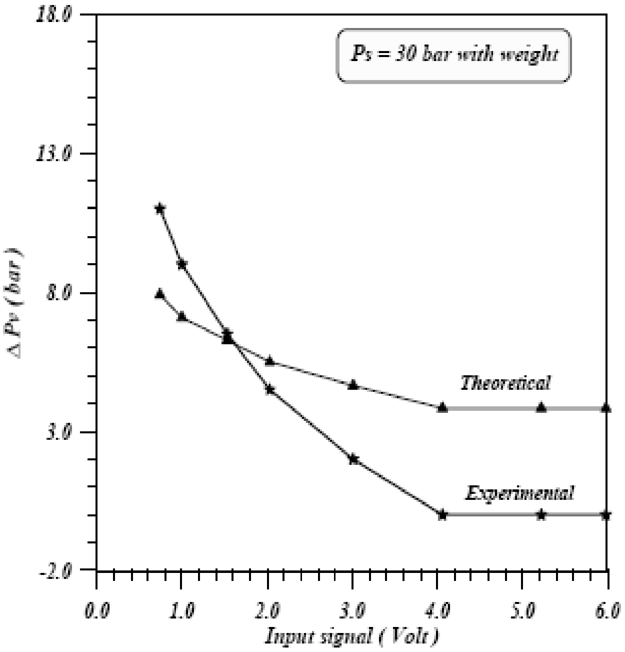

Figure (11) Variation of valve pressure drops (ΔP_v) with input signal for case (3)

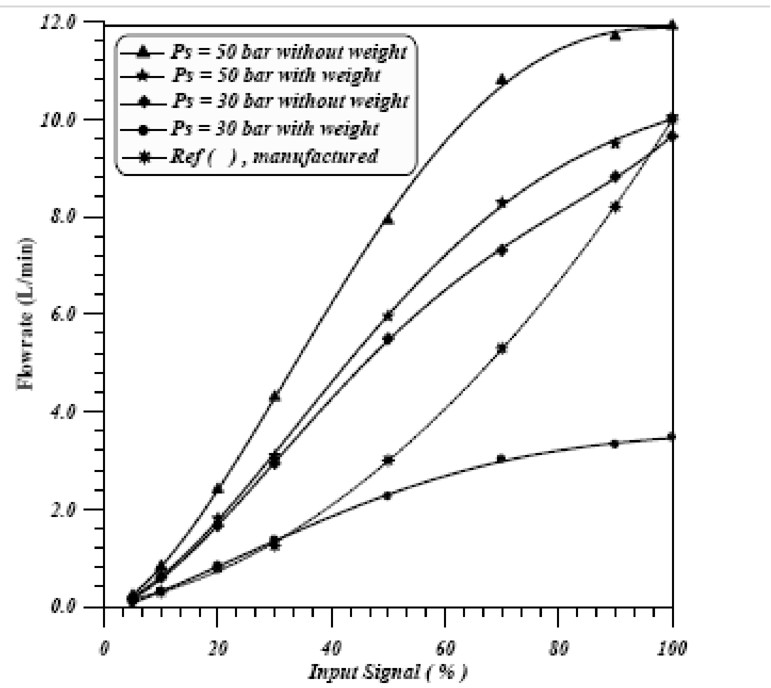

Figure (12) Variation of flow rates with input signals for all cases and manufactured curve

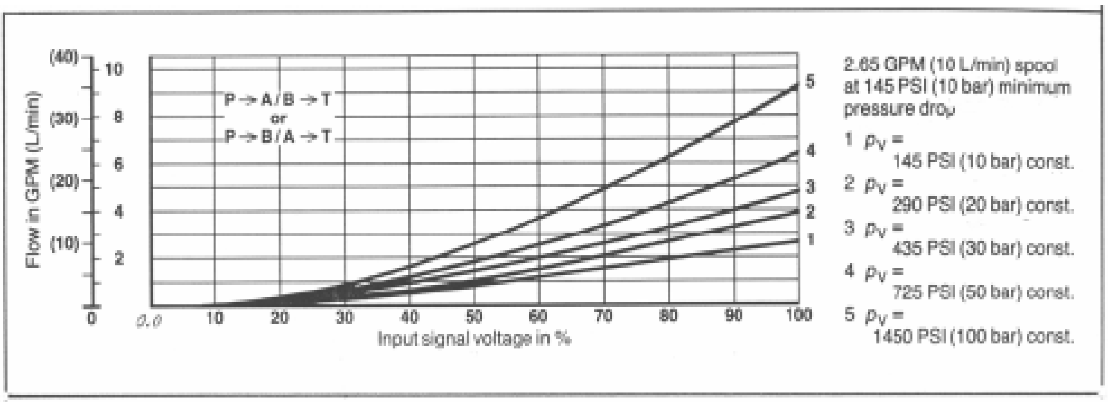

Figure (13) Operating curves for proportional directional valve, spool type; “E”,”EA”,”W”. [13]

Table (1) Source of parameters for hydraulic system control

Symbol	Parameter	Value	Unit	Source of Information
L	Valve port opening	$5*10^{-3}$	m	Manufacturer's data
ε	Spool Lapping	±0.1*L	m	Manufacturer's data
P_S	Pressure supply	$30*10^5$ $50*10^5$	N/m^2	From experimental
A_P	Area of piston	$12.566*10^{-4}$	m^2	Manufacturer's data
m	Mass	5 and 25	kg	From experimental
P_R	Pressure return	Zero	N/m^2	From experimental
V_t	Total cylinder volume	$57.00457*10^{-6}$	m^3	Calculated
S	Piston stroke	0.3	m	Manufacturer's data
α	$=C_d\, w\, (2/\rho)^{1/2}$	0.0004867	$m^{5/2}/kg^{1/2}$	Calculated
β	Fluid bulk modulus	$108*10^8$	N/m^2	Manufacturer's data
R_{iP}	Cylinder leakage coefficient	$1.8*10^{-11}$	$m^4.s/kg$	Manufacturer's data
K_C	Valve pressure gain	$1.9*10^{-12}$	$m^4.s/kg$	Laboratory section
ζ_h	Hydraulic damping coefficient	0.0629278 0.1407043		Calculated
W_h	Hydraulic natural frequency	1997.2088 893.17892	rad.s	Calculated
P	Density for hydraulic HLP	880	kg/m^3	Manufacturer's data [25]
ν	viscosity	48	mm^2/s	Manufacturer's data

Table (2) Simulation parameters sets.

Set	Parameter Changes	ζ_h	W_h (Hz)
1a	Nominal values	0.0629248	1997.2088
1b	Nominal values	0.1407043	893.17892
2	$m \rightarrow m/10$ and $\beta \rightarrow 10\beta$	0.0629248	19972.1
3	$m \rightarrow m/10$ and $A_P \rightarrow A_P/\sqrt{10}$	0.0629248	1996.8481
4	$m \rightarrow m/10$ and $\beta \rightarrow \beta / 10$	0.062924	1997.21
5	$m \rightarrow 10 * m$ and $\beta \rightarrow 10 * \beta$	0.1407042	893.17946
6	$Vt \rightarrow 10 * Vt$ and $m \rightarrow 10 * m$	0.1407042	89.3178
7	Nominal value of damping coefficient	0.1407042	16
8	Nominal value of damping coefficient	0.0629248	16
9a	Nominal value of damping coefficient	0.0629248	1200
9b	Nominal value of damping coefficient	0.1407042	1200
10a	Nominal value of damping coefficient	0.1407042	1600
10b	Nominal value of damping coefficient	0.0629248	1600
11a	Nominal value of damping coefficient	0.0629248	2000
11b	Nominal value of damping coefficient	0.1407042	2000
12a	Nominal value of damping coefficient	0.1407042	2500
12b	Nominal value of damping coefficient	0.0429248	2500
13a	Nominal value of damping coefficient	0.0629248	3000
13b	Nominal value of damping coefficient	0.1407042	3000
14a	Nominal value of natural frequency	0.6	1997.2
14b	Nominal value of natural frequency	0.6	893.17892