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A Single-Actuator Control Design for Hydraulic Variable Displacement Pumps

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Abstract
In this paper, it has been shown that the conventional two-actuator configuration for the discharge pressure control of variable displacement axial piston pumps essentially is an open loop unstable system, which imposes inherent constraints on the stability and performance of the closed loop control design. In addition, a large force is required to balance the load generated by the two parallel actuators, which in turn requires large size bearings to hold the swashplate in place. To overcome both of the drawbacks in the conventional pump configuration, a novel control configuration is proposed in this paper. By balancing the torque induced from the pressure carry-over angle, the presented control design uses only one actuator that is arranged in a manner such that the open loop dynamics are strictly stable. Efficient control algorithms have been developed for the proposed new scheme. Experimental tests show some significant advantages of the proposed design over the conventional counterpart.

1. Introduction

Axial-piston variable displacement pumps are widely used in hydraulic systems to provide pressure and flow for various machines in industry. With a constant shaft speed, the discharge flow-rate can be changed by regulating the swashplate angle, and in this way the pump discharge pressure can also be controlled. Ideally, the pump discharge pressure needs to be capable of tracking a desired pressure profile based on the change of the system load pressure. In this aspect, many control disciplines have been applied to develop electro-hydraulically pressure controlled pump systems, among which mostly linear control methods are employed (Akers and Lin, 1988; Lantto, et al, 1989; Lin and Akers 1990). Using a proportional solenoid valve to provide control flow for regulating the swashplate angle, a nonlinear control strategy has been developed for pump discharge pressure control which is capable of on-line adaptive compensating for the change of the pressure carry over angle (Du and Manring, 2000). To further get insight into the system dynamics, and to investigate the effects of particular parameters on the control design, it would be useful to classify the system states into different categories. By doing this, the dominating "slow" variables can be identified and the less important "fast" variables can be neglected to reduce system complexity for control design and implementation purposes. In addition, a reasonable reduced order model will reveal the most important aspects that affect fundamental control system behaviors.

As shown in Figure 1, an axial piston swashplate hydraulic pump normally consists of several pistons within a common cylindrical block. The pistons are nested in a circular array within the block at equal intervals about the shaft axis. The cylinder block is held tightly against a vale plate using the force of the compressed cylinder-block spring and a less obvious pressure force within the cylinder block itself. A ball-and-socket joint connects the base of each piston to a slipper. The slippers themselves are kept in contact with the swashplate and the swashplate angle, α, is controlled by a servomechanism based upon the requirements of the discharge pressure and/or discharge flow rate. For practical applications, the loads on any hydraulic actuator can change from time to time, which in turn requires the hydraulic pump to provide a different operating pressure accordingly. Specifically, for a pressure control, the objective is to control the pump discharge pressure to follow the desired pressure time history (which is determined by the working loads and the environment) with bounded tracking error, while considering system uncertainties and implementation limitations.

There are two main drawbacks in the conventional configuration of control actuators. First, the discharge pressure feedback to increase the pump displacement results in an open loop unstable system, which imposes inherent constraints on the stability and performance of the closed loop control design. Although this argument can be rigorously proved as presented in the following sections, some intuitive analyses would help to reveal the instability physically. Assuming that the pump discharge flow rate does not change, and feedback control is not taking place (open loop), an external disturbance is introduced and this causes the discharge pressure to deviate from its operating point. An increase in the pump discharge pressure will be followed by an increase of the swashplate angle, which in turn results in a further increase in discharge pressure. This positive feedback process quickly raises the pump discharge pressure to its maximum value corresponding to the mechanically set maximum swashplate angle, which makes a pressure relief valve absolutely necessary for safety. For the same reason, a decrease in discharge pressure will cause the output pressure to drop down to its minimum value. Second, a large force is required to balance the load generated by the two parallel actuators, which in turn requires large size bearings to hold the swashplate in place.

To overcome both of the drawbacks in the conventional pump configuration, a novel control configuration is proposed in this paper. By balancing the torque induced from the pressure carry-over angle, the presented control design uses only one actuator that is arranged in a manner such that the open loop dynamics are stable and the force exerting on the swashplate bearings is significantly reduced. In this paper, to more physically track the pump pressure discharge control system, a reduced open loop model is developed using a singular perturbation technique. An analysis of each system state and the design details for synthesizing system controllers are also provided. Experimental results are used to validate the proposed control system design. The paper is organized as follows. Section 2 describes the servo control configuration and the dynamics of the variable displacement pump. Section 3 develops a reduced order pump control model, the dynamic analysis, and the control design. Section 4 provides the experimental results and the discussion to validate the proposed model reduction and control design. And, finally, Section 5 summarizes the work with conclusions.

2. System Modeling

The schematic of the pressure servo control system, with conventional actuators configuration, for variable displacement pumps is shown in Figure 2. With the cylinder block running at a constant angular velocity, ω, each pistons periodically passes over the discharge and intake ports on the valve plate. The inclination of the swashplate, with a positive angular position α, makes the pistons undergo an oscillatory displacement in and out the cylinder block thus drawing fluid from the low pressure intake-port and pumping it out through the high pressure discharge-port. The swashplate angle control torque is generated using two servos as shown in the figure.

A small servo, along with a spring, provides a bias torque to increase the pump stroke and a larger servo, on the other hand, is used to decrease the pump stroke. The discharge pressure of the pump is directly fedback to the small servo while the pressure acting on the large servo comes from the output port of a controlled servo valve. In this research, a four way single stage servo valve was purchased and used in the control configuration. One of the output ports was connected to the large servo and the other was blocked to prevent oil leakage. For this configuration, only a 3-way valve is needed. The position of the valve spool controls fluid flow in or out of the large servo chamber to generate the control torque for positioning the swashplate angle. The servo valve is assumed to have a much higher bandwidth than that of the pump such that its dynamics may be neglected in the analysis of the system dynamics and in the control design. Thus, the task of pressure servo-control design becomes finding an appropriate law to position the spool of the servo valve so that the pump discharge pressure would track a desired pressure time history.

2.1 Dynamics of Axial Piston Swashplate Hydraulic Pumps

Using previous research (Du and Manring 2000, Manring 1999), it can be shown for the configuration of Figure 2 that the equation of motion for the swashplate is given by

$$I(\alpha)\ddot{\alpha}+G(\alpha,\dot{\alpha})=T_f\cos^2\alpha+a_pP-T_{fpst}-a_cP_c+d \quad (1)$$

where T_f is the nonlinear friction exerting on the swashplate, T_{fpst} is the magnitude of the Coulomb friction between a piston and its bore, and

$$I(\alpha)=\frac{L_1^2m_1+L_c^2m_2+0.5nr^2m_p}{\cos^2\alpha}+J_{sp}\cos^2\alpha \quad (2)$$

$$G(\alpha,\dot{\alpha})=(c_{sp}\cos^2\alpha+\frac{c_1L_1^2+c_2L_c^2}{\cos^2\alpha}+\frac{nr^2c}{2\cos^2\alpha})\dot{\alpha} \quad (3)$$
$$+\frac{(2L_1^2m_1+2L_c^2m_2+nr^2m_p)\dot{\alpha}^2\sin\alpha}{\cos^3\alpha}+(kL_1^2+\frac{nr^2m_p\omega^2}{2})\tan\alpha$$

$$a_p=A_1L_1-\frac{rnA_p\gamma}{2\pi} \quad (4)$$

$$a_c=A_cL_c \quad (5)$$

$$d=\frac{2nF_{fpst}r}{\pi}+plL_1 \quad (6)$$

The physical meaning of every parameter is listed in the Nomenclature. It should be noted that among these parameters, shown in Figure 3, γ represents the so called pressure carry over angle (Manring 1996) which changes with the operating conditions of the pump (Eq. (4)).

The pump discharge flow dynamics can be described by

$$\dot{P}=\frac{\beta}{V_t}(K_G\omega\alpha-C_{th}P-Q_L) \quad (7)$$

where V_t is the volume of the discharge chamber of the pump, P is the pump discharge pressure, Q_L is the load flow rate which is determined by the actuator, and the other parameters are listed in the Nomenclature. The control pressure within the large servo chamber is governed by the pressure-rise rate equation and is given by

$$\dot{P}_c=\frac{\beta}{V_c}(Q_c-Q_{leak}+a_c\dot{\alpha}) \quad (8)$$

where P_c is the control pressure, Q_c is the flow rate controlled by the servo valve and Q_{leak} represents the flow leakage through the clearance between servo piston and its bore. The control flow rate provided by a proportional valve can be approximately obtained using the following equation,

$$Q_c=k_q'x_v\sqrt{\frac{\Delta P}{\Delta P_N}} \quad (9)$$

where k_q' is the flow gain of the valve, ΔP_N is the rated pressure difference across the valve port, and $\Delta P=\begin{cases}P-P_c & \text{for } x_v>0\\ P_c & \text{for } x_v<0\end{cases}$. Rewrite Eq. (9) as follows for convenience,

$$Q_c=k_qx_v\sqrt{\Delta P} \quad (10)$$

where $k_q=\frac{k_q'}{\sqrt{\Delta P_N}}$. Further,

$$\dot{P}_c=\frac{\beta}{V_c}(k_qx_v\sqrt{\Delta P}-C_{lc}P_c+a_c\dot{\alpha}) \quad (11)$$

Eqs. (1), (7), and (11) form the open loop dynamics of the pump control system. These equations represent a highly nonlinear fourth order system with the valve spool position x_v and the load flow rate Q_L as inputs and the discharge pressure, P, as the output.

3. Reduced model and control design

In addition to the assumptions made earlier, it is further assumed that the spool in the servo valve is operating near its null position and that the operating range of the swashplate angle is small such that $\cos\alpha\approx 1$ and $\sin\alpha\approx\alpha$.

3.1 Linearization and order reduction

In this subsection, we neglect friction terms and linearize Eq. (1) around the operating point: $P=P_{op}$, $\dot{\alpha}=0$, and $\alpha=0$. This yields

$$a_2\ddot{\alpha}+a_1\dot{\alpha}+a_0\alpha=a_pP_c-a_cP \quad (12)$$

where

$$a_2=J_{sp}+L_1^2m_1+L_2^2m_2+\frac{nr^2m_p}{2} \quad (13)$$

$$a_1=c_{sp}+c_1L_1^2+c_2L_2^2+\frac{nr^2c}{2} \quad (14)$$

$$a_0=kL_1^2+\frac{nr^2m_p\omega^2}{2} \quad (15)$$

It should be noted that the pre-compression force of the bias spring term at $\alpha=0$, which is a constant, is neglected without loss of generality. In fact, this torque can be compensated by adding an additional constant term to the control pressure. Further, at the operating angle α_0, the steady state control pressure and the pump discharge pressure are related by,

$$P_c \approx \frac{a_p}{a_c}P + \left(kL_c^2 + \frac{nr^2 m_p \omega^2}{2}\right)\alpha_0 \quad (16)$$

Thus it can be assumed that at the operating point

$$\Delta P_{op} \approx \begin{cases} \left(1-\frac{a_p}{a_c}\right)P_{op} - \left(kL_c^2 + \frac{nr^2 m_p \omega^2}{2}\right)\alpha_0, & x_v > 0 \\ \frac{a_p}{a_c}P_{op} + \left(kL_c^2 + \frac{nr^2 m_p \omega^2}{2}\right)\alpha_0, & x_v < 0 \end{cases} \quad (17)$$

Let

$x_1 = \alpha$, $x_2 = \dot{\alpha}$, $x_3 = P$, $z = P_c$, $u_1 = x_v$, $u_2 = Q_L$, and output $y = P$.

Given these definitions, the state space form of the linearized system dynamics can be expressed as

$$\dot{x}_1 = x_2 \quad (18a)$$

$$a_2\dot{x}_2 = -a_1x_2 - a_0x_1 + a_px_3 - a_cz \quad (18b)$$

$$\dot{x}_3 = b_0x_1 - b_px_3 - b_qu_2 \quad (18c)$$

$$\varepsilon\dot{z} = -a_cx_2 - C_{lc}z + c_xu_1 \quad (18d)$$

$$y = x_3 \quad (18e)$$

where $b_0 = \frac{\beta}{V_t}K_G\omega$, $b_p = \frac{\beta}{V_t}C_{hl}$, $b_q = \frac{\beta}{V_t}$, $c_x = k_q\sqrt{\Delta P_{op}}$, and $\varepsilon = \frac{V_c}{\beta}$. With a small control volume V_c and a large bulk modulus β, ε is a very small positive number. By using a singular perturbation technique (ε is set to zero), it can be shown that

$$0 = -a_cx_2 - C_{lc}z + c_xu_1 \quad (19)$$

Thus, the pump dynamics are reduced to

$$\dot{\xi}_1 = \varsigma \quad (20a)$$

$$C_{lc}a_2\dot{\varsigma} = -(a_1C_{lc} + a_c^2)\varsigma - a_0C_{lc}\xi_1 + a_pC_{lc}\xi_2 - c_xa_cu_1 \quad (20b)$$

$$\dot{\xi}_2 = b_0\xi_1 - b_p\xi_2 - b_qu_2 \quad (20c)$$

$$y = \xi_2 \quad (20d)$$

where $\xi_1 = \alpha$, $\varsigma = \dot{\alpha}$, $\xi_2 = P$, and output $y = P$. In addition, the leakage for the control servo is very small compared to the other parameters and therefore the coefficient $\varepsilon_1 = C_{lc}a_2$ is also a small positive number. Again, by using singular perturbation theory, setting $\varepsilon_1 = 0$, we have

$$0 = -(a_1C_{lc} + a_c^2)\varsigma - a_0C_{lc}\xi_1 - a_pC_{lc}\xi_2 + c_xa_cu_1. \quad (21)$$

Therefore, a further simplified system model can be written as

$$\dot{\xi}_1 = -\frac{a_0C_{lc}}{(a_1C_{lc} + a_c^2)}\xi_1 + \frac{a_pC_{lc}}{(a_1C_{lc} + a_c^2)}\xi_2 - \frac{c_xa_c}{(a_1C_{lc} + a_c^2)}u_1 \quad (22a)$$

$$\dot{\xi}_2 = b_0\xi_1 - b_p\xi_2 - b_qu_2 \quad (22b)$$

$$y = \xi_2 \quad . \quad (22c)$$

The physical meaning of the model reduction is to separate the system dynamics into fast and slow subspaces. With $g_{\dot{\alpha}} = -(a_1C_{lc} + a_c^2)\varsigma - a_0C_{lc}\xi_1 - a_pC_{lc}\xi_2 + c_xa_cu_1 \neq 0$ and $g_{P_c} = -a_cx_2 - C_{lc}z + c_xu_1 \neq 0$, the states $\dot{\alpha}$ and P_c change very fast, with change rate $g_{\dot{\alpha}}/\varepsilon_1$ and g_{P_c}/ε respectively, so that they will rapidly converge to the roots of Eqs. (19) and (21) which are the equilibrium Eqs (18d) and (20b). The reduced model (Eqs. (22a – 22c)) includes the slow states, the swashplate angle α and the pump discharge pressure P, which dominate the system dynamics at low frequencies.

3.2 Design of an open loop stable system

The open loop dynamic system represented by Eqs. (23a – 23c) is an open loop unstable system. This can be shown as follows: The characteristic equation for the system is

$$(a_1C_{lc} + a_c^2)s^2 + (b_p(a_1C_{lc} + a_c^2) + a_0C_{lc})s + (a_0C_{lc}b_p - a_pb_0C_{lc}) = 0 \quad (23)$$

Since the leakage coefficient C_{hl} is very small compared with the other parameters, it is usually the case that $a_0b_p << a_pb_0$. Hence for Eq. (23), we have $a_1C_{lc} + a_c^2 > 0$, $b_p(a_1C_{lc} + a_c^2) + a_0C_{lc} > 0$, and $a_0C_{lc}b_p - a_pb_0C_{lc} < 0$, which means that one of its roots is in the right half s-plane. This conclusion proves that the pump pressure control system as shown in Figure 2 is an essentially open loop unstable system, which inherently imposes limitations on any closed loop control design.

To construct an open loop stable system and to reduce the control force exerted on the swashplate bearings, a novel control configuration is proposed in Figure 4. In this design, only a single actuator is used for force reduction. A soft spring is introduced to force the pump into stroke during start-up when there is no fluid pressure available for actuation. Instead of de-stroking the swashplate, the control servo is placed at an up-stroking position to balance the torque induced on the swashplate by the pressure carry-over angle. For this configuration, we follow the same procedure cited earlier. The linearized governing equation for the swashplate can be obtained as

$$a_2'\ddot{\alpha} + a_1'\dot{\alpha} + a_0'\alpha = a_cP_c - a_p'P \quad (24)$$

where

$$a_2' = J_{sp} + L_2^2m_2 + \frac{nr^2m_p}{2} \quad (25)$$

$$a_1' = c_{sp} + c_2L_2^2 + \frac{nr^2c}{2} \quad (26)$$

$$a_0' = kL_c^2 + \frac{nr^2m_p\omega^2}{2} \quad (27)$$

$$a_p' = \frac{rNA_p\gamma}{2\pi} \quad (28)$$

Furthermore, by using the singular perturbation theory, a linear reduced order system model can be expressed as

$$\dot{\xi}_1 = -\frac{a_0'C_{lc}}{(a_1'C_{lc} + a_c^2)}\xi_1 - \frac{a_p'C_{lc}}{(a_1'C_{lc} + a_c^2)}\xi_2 + \frac{c_xa_c}{(a_1'C_{lc} + a_c^2)}u_1 \quad (29a)$$

$$\dot{\xi}_2 = b_0\xi_1 - b_p\xi_2 - b_qu_2 \quad (29b)$$

$$y = \xi_2 \quad . \quad (29c)$$

From these equations, it can be shown that the eigenvalues of the system matrix are all in the left half of the s-plane. Therefore, an open loop stable system is obtained by using the control configuration shown in Figure 4.

3.3 Control design

Control torque

The control torque is generated by the control pressure and the torque induced on the swashplate by the pressure carry-over angle, which can be expressed as

$$T_c = a_cP_c - a_p'P \quad (30)$$

It should be noted that the control torque magnitude is limited by the physical constraints, of the parameters a_p' and a_c. In addition, the pump discharge pressure inherently imposes upper and lower bounds on the control torque as well. Considering $0 \le P_c \le P$, it can be shown that.

$$-a_p P \le T_c \le (a_c - a_p')P \tag{31}$$

Eq. (31) reveals two important points related to the control performance of the pump design. First, a higher operating pressure corresponds to a wider range of available control torque, which implies that smaller control gains could be able to generate sufficient control torque to move the swashplate to a desired position within a short period of time. For low operating pressures, a larger system response time should therefore be expected. Secondly, a larger difference between a_c and a_p' helps in increasing the least upper bound on control torque, while a larger pressure carry-over angle decreases the largest lower bound on control torque. This also shows that increasing both a_p' and a_c could benefit control system performance, since smaller feedback control gains and faster response times are the main trade off between robust stability and robust performance.

Control design and the selection of control parameters

With a guaranteed stability, the performance objective of control design is to regulate the pump discharge pressure to track the desired pressure value without overshoot. To meet this requirement, the control scheme proposed in this paper is to form a closed loop system with an essentially first order error dynamic system. To achieve this, a further analysis would be helpful in controller design and its parameter selection. Let

$$N(s) = c_x a_c b_0 \tag{32}$$

and

$$M(s) = (a_1' C_{lc} + a_c^2)s^2 + \left((a_1' C_{lc} + a_c^2)b_p + a_0' C_{lc}\right)s + \left(a_0' C_{lc} b_p + a_p' C_{lc} b_0\right) \tag{33}$$

Considering the pump discharge flow rate as the system disturbance, shown in Figure 5, the closed loop transfer function can be written as follows:

$$T(s) = \frac{C(s)N(s)}{M(s) + C(s)N(s)} \tag{34}$$

Several points should noted. (1) $M(s)$ is a second order polynomial with all its non-zero coefficients being positive, which implies that the two roots of $M(s)$ will always be in the left half of s-plane. (2) $N(s)$ is a very large positive constant compared with the coefficients in $M(s)$ which implies that, with a properly constructed controller $C(s)$, the properties of the transfer function $T(s)$ will be dominated by the forward loop transfer function, $C(s)N(s)$. (3) If the polynomial $M(s) + C(s)N(s)$ is of second order, with all of the coefficients being strictly positive, the system stability will be guaranteed as long as the unmodeled dynamics can be neglected. This means a proportional control (P control) or proportional plus derivative control (PD control) will be appropriate from a system stability and implementation point of view, provided that the control gains are properly chosen such that the high frequency unmodeled system uncertainties are not excited.

Based upon the above discussion, a PD controller, $C(s) = k_d s + k_p$, is used for the pump discharge pressure to track its desired time history in this research. Hence, the closed loop transfer function, Eq. (34), can be written as

$$T(s) = \frac{c_x a_c b_0 (k_d s + k_p)}{(a_1' C_{lc} + a_c^2)s^2 + \left((a_1' C_{lc} + a_c^2)b_p + a_0' C_{lc} + k_d c_x a_c b_0\right)s + \left(a_0' C_{lc} b_p + a_p' C_{lc} b_0 + k_p c_x a_c b_0\right)} \tag{35}$$

From Eq. (35), it is can be seen that at low frequencies the transfer function is essentially a first order dynamic system, which implies that no over shoot for a step input can be expected. For this essential first order system, the system rising time and steady state tracking error will be basically determined by feedback gains, k_p, and k_d. The closed loop time constant and the DC steady state error can be expressed respectively as follows:

$$\tau \approx \frac{k_d}{k_p} \tag{36}$$

$$\underset{t \to \infty}{e(0)} \approx \frac{a_0' C_{lc} b_p + a_p' C_{lc} b_0}{a_0' C_{lc} b_p + a_p' C_{lc} b_0 + k_p c_x a_c b_0} P_d(0) \quad . \tag{37}$$

4. Experimental Studies

Experimental Setup

A schematic of the pump discharge pressure control test is shown in Figure 6. The experimental setup consists of (1) a speed controlled 335.7 kW (450 HP) electrical motor to provide a constant pump shaft running speed; (2) a variable displacement axial piston pump with 9 pistons and a maximum displacement of 247 cc/rev; (3) a MOOG D633 direct drive servo-proportional control valve with rated flow of 5 l/min at 6.895 MP (1000 psi) operating pressure to provide the control pressure; (4) a variable orifice to restrict the pump discharge flow rate and to provide a discharge pressure and a flow disturbance; (5) a pressure relief valve with an adjustable relief pressure for safety (the pressure relief valve is set much higher than the test pressure range during the experimental tests); (6) A Pentium II based 333-MHz computer with with USB DT9802 Data Acquisition system used for data acquisition and control; (7) two pressure transducers and an angular position sensor to measure the system variables; and (8) custom built signal conditioners. 10W hydraulic oil is used in the experimental tests and the temperature of the oil is controlled and limited to less than $65°C$. Other related parameters are listed in Table 1. The voltage output from the computer is limited to $\pm$ 10V(20mA). A digital Butterworth low pass filter with a 50 Hz cutoff frequency is implemented to remove the effects of high frequency electrical noise. The A/D board has 16-bit resolution on analog inputs and 12-bit resolution on the analog outputs. The control software was developed using VC++ language.

Experimental Results

For the PD control design, generally speaking, the smaller τ and the larger k_p, the faster system response and the better performance, provided unmodeled dynamics are not excited and the actuator saturation is prevented. In the tests, the two parameters are selected as τ= 0.1 sec and the proportional gain kp = 1.2e10^{-3} mm/kPa.

The regulation performance is studied using a square wave trajectory for P_d of amplitude 3 MPa centered at 17 MPa and with a period of 4 or 8 seconds. The sampling frequency is limited by 60 Hz. The pump running speed is set at 2000 rpm, 1900 rpm, 1800 rpm and 1700 rpm respectively. A large number of experimental tests have been conducted and the test results are very consistent. Figures 7 and 8 show the typical closed loop control system performance for hardware implementation of the proposed strategies and algorithms. In the figures, the high frequency component corresponds to the pressure fluctuation caused by the cycling of the pistons which usually is not modeled. It can be seen from the figures that for a PD controller, as predicted in the analysis, no overshoot occurs for the step command inputs and the system is stable with tracking errors being bounded. The relatively slower de-stroking of the swashplate, which can be improved with a gain scheduling scheme, basically causes the offset of the steady state average error. It should also be noted that the higher pump running speed corresponds to a quicker system response when pump swashplate de-stroking is required. During the tests, it has also been found that if the feedback gains are kept as constants at different pressure operating points, the tracking error and the system response time become larger and longer respectively. Gain scheduling based on different operating points improves the performance significantly.

5. Conclusion

In this paper, a discharge pressure control of hydraulic variable displacement pumps using single actuator is proposed. The control designs presented in this paper provide an approach for controlling the discharge pressure of variable displacement pumps to track the desired pressure time history using single actuator. It has been shown that, using the reduced second order system model, the locally linearized open loop system is strictly stable. A proportional control (P control) or proportional plus derivative control (PD control) will be able to form a strictly stable closed loop system, provided that the control gains are properly chosen such that the high frequency unmodeled system uncertainties are not excited. With properly selected PD gains, the closed loop transfer function is essentially a first order dynamic system, which implies that no over shoot for step response can be expected. For this essential first order system, the system rising time and steady state tracking error will be basically determined by feedback gains. For high-pressure applications, better performance can be obtained since the unmodeled dynamics, especially the swashplate kinematics, are much less important and a wider range of control torque can be obtained. Experimental results are provided to validate the proposed control approaches.

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Table 1. Physical parameters for the pump dynamics

J	0.127 kg.m^2	k	3185 N/m	C_d	0.61	c_{sp}	0.1 Nm/rad/sec	r_{sp}	0.005 m
m_2	0.264 kg	ω	209.44 rad/sec	N	9	P_{op}	17.5 MPa	L_1	0.092 m
V_c	0.000061 m^3	c_1	0.05 Nm/rad/sec	γ	19^0	C_{lc}	1.02x10^{-13} m^3/sec.Pa	V_t	0.0012 m^3
r_h	0.0127 m	c	0.05 Nm/rad/sec	A_c	0.00064 m^2	C_{lh}	.23x10^{-11} m^3/sec.Pa	L_c	0.092 m
β	1500 MPa	c_2	0.05 Nm/rad/sec	A_p	0.00078 m^2	r	0.062 m		

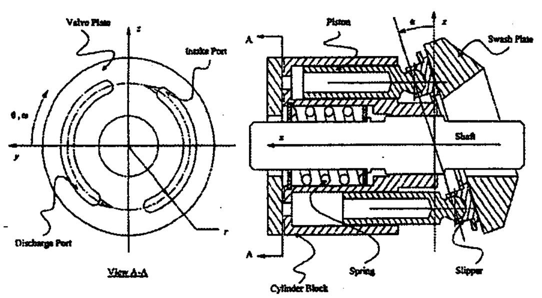

Figure 1. The general configuration of a axial piston swashplate.

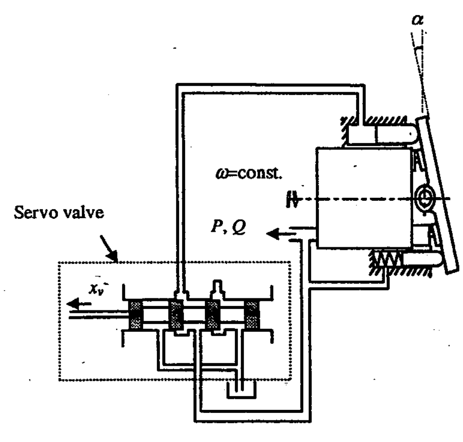

Figure 2. Schematic of pump conventional control configuration.

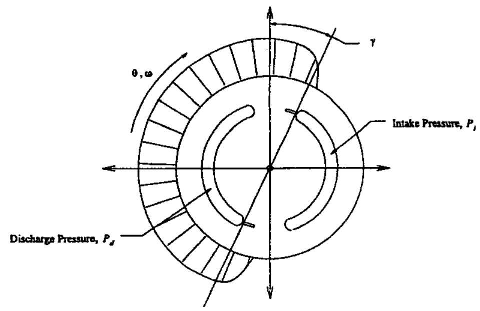

Figure 3. A schematic of the pressure profile.

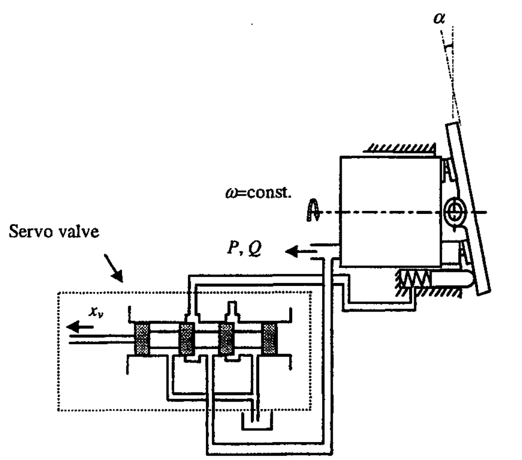

Figure 4. Schematic of pump control configuration using single actuator.

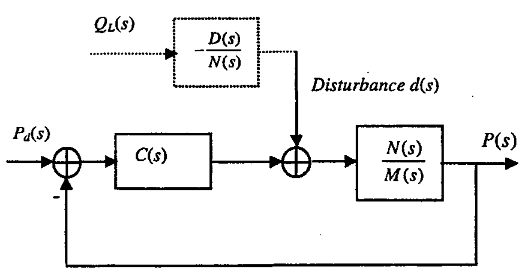

Figure 5. Overall control diagram

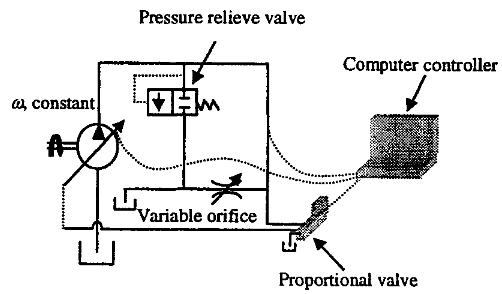

Figure 6. Schematic of the experimental test setup.

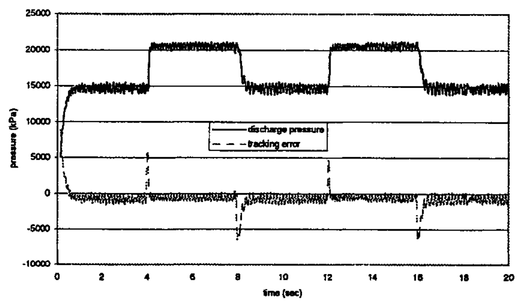

Figure 7. Pump discharge pressure regulation for PD control design. Pump running speed 2000 rpm.

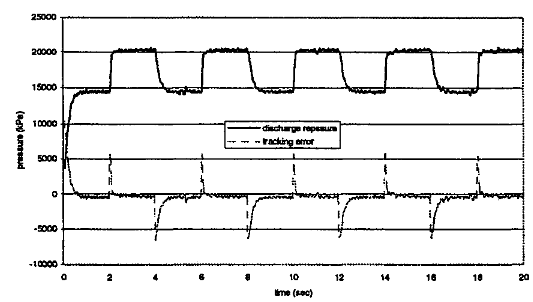

Figure 8. Pump discharge pressure regulation for PD control design. Pump running speed 1700 rpm.