Simulation Study on Pressure Control using Nonlinear Input/Output Linearization Method and Classical PID Approach

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ABSTRACT

This paper deals with the comparison of linear PID and a nonlinear control strategy to control pressure in a constant volume using a 4/3 way control valve. For the PID control at least three parameters have to be tuned. Rule sets for PID pressure control are very powerful if they derive directly from parameters which can be found in data sheets of the hydraulic components. Nonlinear control techniques are more complex. Their advantage, however, is that they take the nonlinearities of the plant into account and compensate them. The applied nonlinear approach in this paper is the input/output linearization in combination with full state feedback and state observer.

The pole placement problem is approached by analyzing a set of PID control rules which is widely accepted in industry and choosing the pole placement for the nonlinear controller accordingly. In addition, this paper presents another set of PID tuning rules which are derived using ITAE performance criterion. The closed loop poles of these two PID designs have been used to calculate the feedback controller gains of the nonlinear controller. The performance of these two linear PID and nonlinear controller designs is evaluated and compared in simulation based on overshoot, settling time, robustness and complexity.

1. INTRODUCTION

Hydraulic pressure control systems can be found in various industrial and vehicle applications such as in clutch actuation of automatic transmissions, anti-lock brake systems, presses and injection molding machines, paper machines and many more. These systems mainly consist of a proportional control valve, a hydraulic actuation cylinder and a pump system which supplies the pressurized hydraulic fluid for the system. The common point in the above mentioned applications is the dominance of pressure dynamics, which yields to neglect of motion dynamics of hydraulic actuation cylinder.
Different classical linear pressure control schemes have been proposed in the literature (1, 2). Especially the linear PID control scheme proposed by Boes in (1) offers an easy tuning of PID control parameters using known plant parameters. This approach is examined extensively in this work. This study mainly focuses on two different control strategies, the classical PID and input/output linearization based pressure control. In this context, the mathematical model of a pressure control application is derived in section 2. In section 3 PID pressure control and tuning methods are examined, namely the PID tuning rules proposed by Boes in (1) and an alternative way of PID parameter tuning method using an ITAE performance criterion. At the end of this section, pole/zero location analysis for both tuning rules is done. In section 4, input/output linearization based pressure control is introduced and derivation of input/output linearized system, state feedback controller and state observer designs are explained. Additionally, the problem of setting the state controller and state observer gains is discussed and a solution is proposed based on previously designed PID control poles. Finally, simulation results using hydraulic simulation environment DSH plus are presented in section 5 and controller performances are compared based on overshoot, settling time as well as complexity.

2. MATHEMATICAL MODEL OF PRESSURE CONTROL APPLICATION

The plant model for pressure control in this study shall be realized by a 4/3 way proportional control valve, a constant pressure supply and a fixed but adjustable hydraulic capacity such as a cylinder with a piston locked at a specific position in which pressure has to be controlled. This system is shown in figure 1.

![Figure 1: Plant model for pressure control](image)

The dynamics of the 4/3 way proportional control valve can be modeled as a second order lag system (PT2 element) with gain $K_V$, natural angular frequency $\omega_N$ and damping ratio $D_V$ (3, 4). The input is the control voltage $u$ and the output is $x_V$ valve displacement. The differential equation of the valve dynamics reads:
The capacity connected to the valve is modeled by the pressure equation which relates the balance of input/output flow to the pressure gradient. Two flows are considered, the valve flow \( Q_A \) and leakage \( Q_{\text{leakage}} \). The pressure equation then is:

\[
\frac{d}{dt} p_A(t) = \frac{E(p_A)}{V_A} [Q_A(t) + Q_{\text{leakage}}(t)]
\]  

(2.2)

with \( E(p_A) \) being the pressure dependent bulk modulus and \( V_A \) the controlled fluid volume. According to (1), the bulk modulus which varies at different pressures can very well be assumed constant for the purpose of pressure control synthesis. Introducing hydraulic capacity \( C_H \) as a constant

\[
C_H = \frac{V_A}{E}
\]  

(2.3)

and combining equations 2.2 and 2.3 gives

\[
\frac{d}{dt} p_A(t) = \frac{1}{C_H} [Q_A(t) + Q_{\text{leakage}}(t)]
\]  

(2.4)

To complete the modeling the orifice flow equation should be described and added to the model. According to the stationary Bernoulli equation the flow through a sharp edged orifice is proportional to the square root of the pressure difference at its ports (4). For positive valve displacement, \( x_v > 0 \), flow is entering into the capacity:

\[
Q_A(t) = B_V x_v(t) \sqrt{p_{\text{sup}} - p_A(t)}
\]  

(2.5)

and for negative valve displacement, \( x_v < 0 \), flow is going out of the capacity:

\[
Q_A(t) = B_V x_v(t) \sqrt{p_A(t) - p_{\text{tank}}}
\]  

(2.6)

In these equations \( B_V \) is the valve discharge coefficient which is calculated using valve data-sheet parameters of maximum flow \( Q_{\text{Nominal}} \) at rated pressure \( P_{\text{Nominal}} \):

\[
B_V = \frac{Q_{\text{Nominal}}}{\sqrt{\Delta(P_{\text{Nominal}}) x_{\text{max}}}}
\]  

(2.7)
Combining equation 2.4 and 2.5 and also equation 2.4 and 2.6, the following equations can be reached. For positive valve displacement, \( x_V > 0 \):

\[
\frac{d}{dt} p_A(t) = \frac{(B_V x_V(t) \sqrt{p_{\text{sup}} - p_A(t)} + Q_{\text{leakage}}(t))}{C_H} \tag{2.8}
\]

and for negative valve displacement, \( x_V < 0 \):

\[
\frac{d}{dt} p_A(t) = \frac{(B_V x_V(t) \sqrt{p_A(t) - p_{\text{tank}}} + Q_{\text{leakage}}(t))}{C_H} \tag{2.9}
\]

Equations 2.1, 2.8 and 2.9 constitute the mathematical model of a plant for pressure control.

For the controller development, the nonlinearities in equations 2.8 and 2.9 should be eliminated. This can be done by linearizing the system around an operation point \( (p_{\text{tank}} < p_{A0} < p_{\text{sup}}) \) using taylor series expansion. After linearizing equations 2.8 and 2.9 around \( p_{A0} \) and \( x_{V0} \) as well as neglecting leakage flows which act as disturbances, transfer equations of the system for both positive and negative valve openings using equations 2.1, 2.8 and 2.9 can be written as:

\[
\frac{\Delta p_A(s)}{\Delta u(s)} = \frac{V_{\text{QU}} \omega_V^2}{C_H s^3 + 2C_H D_V \omega_V s^2 + C_H \omega_V^2 s} \tag{2.10}
\]

where the flow gain parameter \( V_{\text{QU}} \) depends on the operating point, that means at which pressure level we observe the system and whether the valve is opened positively or negatively.

\[
V_{\text{QU}} = B_V \sqrt{p_{\text{sup}} - p_{A0}} K_V \quad \text{For positive valve opening} \tag{2.11}
\]

\[
V_{\text{QU}} = B_V \sqrt{p_{A0} - p_{\text{tank}}} K_V \quad \text{For negative valve opening} \tag{2.12}
\]

For the following analysis and synthesis, the choice of the flow gain value depends on which opening direction of the valve is more sensitive. Sensitivity depends on the square root of pressure difference in equations 2.11 and 2.12. If the pressure difference between supply pressure \( p_{\text{sup}} \) and operating pressure \( p_{A0} \) is greater than that of between tank pressure \( p_{\text{tank}} \) and \( p_{A0} \), then equation 2.11 should be used because positive opening direction is more sensitive. If opposite is true, then equation 2.12 should be used, because negative opening direction is more sensitive now.
3. PID CONTROL AND TUNING METHODS

The PID control method which consists of proportional, integral and derivative feedback is a very powerful control method and is used in industrial processes today. The challenging part in this control method is the tuning of the control parameters for P, I and D parts which is done by trial-error method in practice. Therefore, it is valuable to have a good first guess for these parameters. In this section, a PID tuning method for pressure control application from the literature is examined, and then an alternative approach for a PID parameter tuning rule based on ITAE performance criterion is derived. The power of both methods lies on the easy tuning of parameters using simple formulas which consist of only known plant parameters. The block diagram of a PI-D pressure control system is shown in figure 2.

![Figure 2: Block diagram of PID pressure control system](image)

The PI-D controller has the derivative part only acting on the feedback rather than also on the input. The reason is to avoid peaks produced by the derivative part due to the step changes of the input signal. The transfer function of the closed loop system is:

\[
\frac{\Delta P_A(s)}{\Delta P_{A_{ref}}(s)} = \frac{V_{QU} \omega_v^2 (K_p s + K_1)}{C_H s^4 + 2 C_H D_v \omega_v s^3 + (C_H + V_{QU} K_D) \omega_v^2 s^2 + V_{QU} \omega_v^2 (K_p s + K_1)}
\]

(3.1)

3.1 Method 1: PID tuning according to Boes (1)
The PID tuning method proposed by Boes described in (1) has been established in industry and therefore shall be used as a benchmark for the following control schemes. This method offers a powerful and easy tuning of PID parameters for pressure control applications. Depending only on the plant parameters, three simple formulas are given which are; (1)
The derivation of these rules is not explained in (1) but it is stated that they are based on experiments and linear pole placement. If the closed loop transfer function in equation 3.1 is analyzed, it can be clearly seen that the system has 4 poles and one zero. The characteristic equation of the closed loop transfer function given in equation 3.1 divided by $C_H$ is:

$$s^4 + 2D_V \omega_V s^3 + \left(1 + \frac{\omega_V^2 V_{QU} K_D}{C_H}\right)s^2 + \frac{\omega_V^2 V_{QU} K_P}{C_H}s + \frac{\omega_V^2 V_{QU} K_I}{C_H} = 0$$  (3.5)

The analysis of equation 3.5 shows that the terms with proportional, integral and derivative gains include the flow gain and hydraulic capacity which are canceled when P, I and D gain formulas (equations 3.2 to 3.4) are employed. Thus the location of poles of the closed loop system becomes independent of the flow gain and hydraulic capacity. Another observation is that the damping ratio (cosine of the angle between secant from origin to pole and negative real axis) is always equal for both pole pairs of the closed loop transfer function. This can be easily seen on the zero-pole diagram in figure 3 using the following typical numerical values.

$$\omega_V = 690.8 \ \text{rad/s}, \ D_V = 0.525, \ V_{QU} = 6.758 \cdot 10^{-5} \ \frac{m^3}{V \cdot s}$$

$$C_H = \frac{V_A}{E} = 1.442 \cdot 10^{-12} \ \frac{m^3}{Pa}$$

where $V_A = 2.148 \cdot 10^{-3} \ m^3$ and $E = 1.486 \cdot 10^9 \ \frac{N}{m^2}$

In figure 3, among the poles marked with asteriks, the pole pair which is closer to the imaginary axis clearly dominates the behaviour of the system. A third observation of our analysis is that as the angular frequency and damping of the valve increase, the pole pairs and the zero move to the left while keeping the trapezoidal shape.
3.2 Method 2: PID tuning according to ITAE performance criterion

The PID tuning method for pressure control introduced in the preceding subsection is simple and powerful. However, the derivation of the controller parameter formulas is not clear. Also, as will be seen from simulation results, this approach can be well improved in performance. With the aim of finding a clearly derived and better set of parameters with good transient behaviour, an alternative approach using ITAE performance criterion has been derived, which, similarly to the previous method, only uses known plant parameters.

ITAE is an integral performance criterion which can be used to optimize control system performance. According to ITAE performance criterion, the best system is defined as the system that minimizes the ITAE index which is (5):

$$I_{ITAE} = \int_0^\tau t|e(t)|\,dt$$  \hspace{1cm} (3.6)
Using this index, the characteristic polynomial coefficients of transfer functions of different orders which minimize the ITAE index can be found. According to (6), the transfer equation which minimizes the ITAE criterion for a 4th order all-pole system reads:

\[
T(s) = \frac{\omega_n^4}{s^4 + a_1 \omega_n^3 s^3 + a_2 \omega_n^2 s^2 + a_3 \omega_n s + \omega_n^4}
\]  

(3.7)

where \(a_1 = 2.1\), \(a_2 = 3.4\), \(a_3 = 2.7\). It should be noted that these parameters minimize the ITAE index for a step input. This means that making parameters of the denominator of the closed loop transfer function of our PI-D pressure control problem in equation 3.1 equal to the parameters in equation 3.7, the tracking behaviour to a step input should be optimal.

Comparing the coefficients of \(s^3\) gives:

\[2.1\omega_n = 2D_V\omega_V\]

which leads to the conclusion that \(\omega_n\) must be set to

\[\omega_n = 0.95 D_V\omega_V\]

(3.8)

The interpretation of this relationship is that the pressure control dynamics is limited by the valve dynamics and not by the pressure dynamics. Now using \(\omega_n\) calculated in equation 3.8, the other coefficients are made equal and comparison of coefficients gives the following controller parameters:

\[K_p = 2.33 \frac{D_V^3 \omega_V C_H}{V_{QU}}\]

(3.9)

\[K_I = 0.82 \frac{D_V^4 \omega_V^2 C_H}{V_{QU}}\]

(3.10)

\[K_D = \frac{(3.08 D_V^2 - 1) C_H}{V_{QU}}\]

(3.11)

Since the ITAE minimized functions in (6) are based on systems that have only poles, the resulting closed loop transfer function of the pressure control system (Eq. 3.1) has to be modified slightly to be also an all pole system, that is, no zeros should appear in the system. The zero in the system described by equation 3.1 can be canceled using a prefilter which satisfies the following equation:
where \( G_C \) is the controller, \( G_P \) is the plant to be controlled and \( G_{\text{pref iter}} \) is the prefilter, that should be placed behind the reference signal generator. So the transfer function of the prefilter reads:

\[
G_{\text{pref iter}} = \frac{C_H \omega_n^4}{\omega_V^2 V_{\text{QU}} (K_p s + K_1)} = \frac{D_V \omega_V}{2.84s + D_V \omega_V} \tag{3.12}
\]

Considering equation 3.5 and equations 3.9 to 3.11, it can be stated that poles of the closed loop system with ITAE PID tuning do not depend on flow gain and hydraulic capacity, similar as in the case for the system with Boes PID tuning. For a comparison with the pole-zero locations of the controller proposed by Boes, the same parameters in subsection 3.1 are used and results are presented in figure 3.

Unlike the poles from the system proposed by Boes, the pole pairs are so close to each other that it cannot be said which one is dominant. In addition, there is no zero in this system. A comparison of step response in time domain will be given in section 5.

4. INPUT/OUTPUT LINEARIZATION BASED PRESSURE CONTROL

The general idea of input/output linearization is to transform the nonlinear system dynamics algebraically into a (fully or partly) linear one, so that linear control techniques can be applied (3). This is a powerful control technique and applications on hydraulic drives can be found in the literature (8, 9). In order to achieve this, a nonlinear transformation of the control signal should be derived (linearizing control law) and the complete model should be expressed in new coordinates \( z \) as a linear model after applying a coordinate transformation. A general nonlinear system is defined as:

\[
\frac{dx}{dt} = f(x) + g(x)u(t) \quad y(t) = h(x) \tag{4.1}
\]

To find such a nonlinear transformation for the control signal, first the output signal must be differentiated so many times until an explicit relationship between input \( u \) and output \( y \) is obtained. The number of successive differentiations until obtaining an explicit relationship between input and output signals is called the relative degree. And when the relative degree is equal to the system order, then input/output linearization takes a special form and is called exact input/output linearization. When the pressure plant model in 2.8 and 2.9 (neglecting leakage flows) are expressed in state space form together with equation 2.1 for positive valve displacement, \( x_V > 0 \), we have:
and for negative valve displacement, $x_\nu < 0$:

$$f(x) = \left( \frac{B_\nu x_2 \sqrt{p_{\text{sup}} - x_1}}{C_H} x_3, g(x) = \begin{pmatrix} 0 \\ 0 \end{pmatrix}, y = x_1 \right)$$

where $x_1 = p_A$, $x_2 = x_\nu$, $x_3 = \dot{x}_\nu$ and the output $y = h(x)$ is defined as the chamber pressure $p_A$.

The first derivative of the output function yields:

$$y^{(1)} = \frac{\partial h(x)}{\partial x} \frac{dx}{dt} = \frac{\partial h(x)}{\partial x} (f(x) + g(x)u) = L_f h(x) + L_g h(x)u$$

where $L_g h(x) = 0$ for all operation points in the control range for both positive and negative valve openings. So the first derivative of the chamber pressure is not influenced by the valve input voltage. If differentiation is continued, at third step we can get a nonzero input in the expression as following:

$$y^{(3)} = \frac{\partial L_f^2 h(x)}{\partial x} \frac{dx}{dt} = \frac{\partial L_f^2 h(x)}{\partial x} (f(x) + g(x)u) = L_f^3 h(x) + L_f^2 L_g h(x)u$$

where $L_f^2 L_g h(x) \neq 0$ for all operating points in the control range for both negative and positive openings. This means that the valve input voltage has a direct influence on the third derivative of the chamber pressure. It also means that the pressure plant model has a relative degree of 3 and we can perform an exact input/output linearization. For positive valve displacement, $x_\nu > 0$, the input function is:
\[ L_f^2 L_g h(x) = \frac{B_y \sqrt{p_{\text{sup}} - x_1 \omega^2} K_V}{C_H} \] (4.6)

and for negative valve displacement, \( x_V < 0 \), the input function is:

\[ L_f^2 L_g h(x) = \frac{B_y \sqrt{x_1 - p_{\text{tank}} \omega^2} \omega^2 K_V}{C_H} \] (4.7)

In order to define an input/output linearized system, new \( z \) coordinates can be defined for positive valve displacement, \( x_V > 0 \), as:

\[
\begin{align*}
\dot{z}_1 &= \dot{h}(x) - \frac{p_A}{B_y x_V \sqrt{p_{\text{sup}} - p_A}} \frac{1}{C_H} - \frac{B_y^2 x_V^2}{2 C_H} \frac{1}{C_H} + \frac{B_y x_V \sqrt{p_{\text{sup}} - p_A}}{C_H} \\
\dot{z}_2 &= \dot{y} \frac{B_y x_V \sqrt{p_{\text{sup}} - p_A}}{C_H} \\
\dot{z}_3 &= \dot{h}(x) - \frac{B_y x_V \sqrt{p_{\text{sup}} - p_A}}{C_H} \\
\end{align*}
\] (4.8)

and for negative valve displacement, \( x_V < 0 \):

\[
\begin{align*}
\dot{z}_1 &= \dot{h}(x) - \frac{p_A}{B_y x_V \sqrt{p_A - p_{\text{tank}}}} \frac{1}{C_H} - \frac{B_y^2 x_V^2}{2 C_H} \frac{1}{C_H} + \frac{B_y x_V \sqrt{p_A - p_{\text{tank}}}}{C_H} \\
\dot{z}_2 &= \dot{y} \frac{B_y x_V \sqrt{p_A - p_{\text{tank}}}}{C_H} \\
\dot{z}_3 &= \dot{h}(x) - \frac{B_y x_V \sqrt{p_A - p_{\text{tank}}}}{C_H} \\
\end{align*}
\] (4.9)

With these new state definitions, we write the derivatives of \( z \) in state space form as:

\[
\begin{pmatrix}
\dot{z}_1 \\
\dot{z}_2 \\
\dot{z}_3
\end{pmatrix} =
\begin{pmatrix}
0 & 1 & 0 \\
0 & 0 & 1 \\
0 & 0 & 0
\end{pmatrix}
\begin{pmatrix}
z_1 \\
z_2 \\
z_3
\end{pmatrix} +
\begin{pmatrix}
0 \\
0 \\
L_f^3 h(x) + L_f^2 L_g h(x)
\end{pmatrix}
\] (4.10)

\[ u, y = z_1 = p_A \]
Now, we have reached a state space system which has a linear dynamics with a nonlinear input function. Defining an appropriate nonlinear feedback and a new input variable \( u \):

\[
  u = \frac{v - L_f^3 h(x)}{L_f^2 L_g h(x)},
\]

we can linearize the system as:

\[
  \dot{z} = \begin{pmatrix}
    \dot{z}_1 \\
    \dot{z}_2 \\
    \dot{z}_3
  \end{pmatrix} = \begin{pmatrix}
    0 & 1 & 0 \\
    0 & 0 & 1 \\
    0 & 0 & 0
  \end{pmatrix} \begin{pmatrix}
    z_1 \\
    z_2 \\
    z_3
  \end{pmatrix} + \begin{pmatrix}
    0 \\
    0 \\
    1
  \end{pmatrix} v
\]

(4.12)

The linearization is also illustrated in figure 4 below:

Equation 4.11 is called the linearizing control law and by introducing this expression to the nonlinear system, the nonlinear system turns into a linear one as defined in 4.12. The derived linearizing control law for the pressure plant model is a long expression which is not printed here due to the space limitation.

### 4.1 Pole placement problem

In the previous subsection, the derivation of input/output linearization for the pressure plant model has been completed. To control such a system, full state feedback controller can be used and the motivation of the full state feedback control is to provide the new virtual input \( v \) as a function of all three states and the reference value. If the states are fed back to the system, multiplied with a state gain matrix and supplied to the system as input, the input function becomes:
Using the state space representation in 4.12 and equation 4.13, a closed loop system forms and reads as:

\[
\frac{dz}{dt} = (A - bk_{\text{state}})z
\]  

where:

\[
A - bk_{\text{state}} = \begin{pmatrix}
0 & 1 & 0 \\
0 & 0 & 1 \\
-k_{\text{state1}} & -k_{\text{state2}} & -k_{\text{state3}}
\end{pmatrix}
\]  

The problem of choosing the state controller gains \(k_{\text{state1}}, k_{\text{state2}}\) and \(k_{\text{state3}}\) is called pole placement problem. Depending on the values of these parameters, the roots of the characteristic polynomial change, so the places of the poles in the Re-Im plane. To guarantee the stability, these poles have to be in the left side of the Re-Im plane, however being stable is not sufficient since the design also has to meet desired transient response characteristics of the closed loop system, such as settling time and overshoot. There are overall 3 poles to placed in case of input/output linearized pressure control system, so the pole patterns of previously designed PID pressure control systems can be used, because of the proved performance of these poles. However, there are 4 poles in the PID control system described in (1). For the state controller, only dominant pole pair and the real part of the other pole pair from the PID system in (1) will be used in simulations, which results in the following state controller gains:

\[
k_{\text{state1-Boes}} = 3.93 \cdot 10^6 \frac{V}{Pa}, \ k_{\text{state2-Boes}} = 3.79 \cdot 10^4 \frac{V}{s \cdot Pa}, \ k_{\text{state3-Boes}} = 400 \frac{V \cdot s}{Pa}
\]

In case of the ITAE PID system, instead of taking directly the pole pattern from the PID system, the 3\textsuperscript{rd} order ITAE function is used to calculate state controller gains, which are according to (5):

\[
k_{\text{state1-ITAE}} = 4.12 \cdot 10^7 \frac{V}{Pa}, \ k_{\text{state2-ITAE}} = 2.75 \cdot 10^5 \frac{V}{s \cdot Pa}, \ k_{\text{state3-ITAE}} = 604 \frac{V \cdot s}{Pa}
\]

4.2 Observer design

To be able to use a full state feedback controller, all the states of the system have to be known or measured. Since measurement of valve spool displacement and valve spool acceleration is impractical, estimation of these states via a Luenberger’s state observer is a reasonable solution. Together with the state observer, closed loop system can be defined as:
where $k_{\text{Obs}}$ is the observer gain vector, $e$ being the error vector, $z_{\text{est}}$ being the estimated state vector and $c^T = \begin{pmatrix} 1 & 0 & 0 \end{pmatrix}$. With the addition of the state observer, another pole placement problem arises. The observer gain vector has to be determined so that matrix $(A - k_{\text{Obs}} c^T)$ has all its eigenvalues (or poles) in the left side of Re-Im plane. In case of input/output linearized pressure control system, there are 3 observer gains to be set. However, when the observer is combined with a state feedback controller, having a stable observer will not be a sufficient criterion. In this case the observer should operate 2 to 5 times faster than the controller (7). This means that the poles of the observer should be 2 to 5 times of the left of the dominant poles of the system with state controller. But in terms of experimentation, the farther the poles of the observer go to the left, the more sensitive the observer becomes to measurement noises (7). In simulations with nonlinear controller, 3 observer poles are placed at -800 on real axis when ITAE pole pattern used. In case of pole pattern from (1), the 3 observer poles are placed at -90 on the real axis.

5. SIMULATION RESULTS AND COMPARISON OF CONTROLLER PERFORMANCES

In order to see the performances of the controllers, the system has been modeled and simulated using hydraulic simulation environment DSH® plus. The reference pressure...
signal used in simulations is a rectangular signal with upper value of 85 bar and lower value of 55 bar. The plant parameters are already given in subsection 3.1 and PID controller parameters can be calculated using corresponding formulas. For the PID controllers, a time constant of 0.5 ms has been used for the derivative parts. In figure 5 pressure signals after a step signal from 55 bar to 85 bar can be seen. In terms of transient behaviour, the PID controller with Boes settings shows the most overshoot among others. Least overshoot exhibits the nonlinear controller with ITAE pole pattern. If settling time is defined for the 2% percentage band, it can be seen from figure 5 that the PID controller with Boes settings as well as the nonlinear controller with pole pattern from Boes have longer settling times compared to the others. In terms of steady state behaviour, the nonlinear controller with pole pattern from Boes shows undamped oscillations, which indicates that the choice of the pole pattern which were derived from PID controller poles with Boes settings is not the best choice. Among the PID controllers, PID controller with ITAE settings shows better damping and settling characteristics. When the complexity is taken into consideration, it can be seen that nonlinear controllers are more complex compared to the PID controllers.

6. CONCLUSIONS

In this paper, two different control schemes for pressure control, namely PID and nonlinear control have been analyzed. A well-known PID scheme for pressure control has been examined and an alternative PID scheme has been derived using ITAE performance criterion. In addition, a nonlinear controller which is based on input/output linearization has been designed. To analyze the performances of the controllers, a pressure control system has been modeled and simulated using DSH plus. Results show that nonlinear controller with pole pattern from ITAE has the best steady state and transient behaviour. Also, the PID controller with ITAE settings has a smaller overshoot and better settling behaviour compared to the PID control scheme proposed in (1). Robustness of the controllers has not been analyzed here, and could be a subject for further research. In terms of complexity, nonlinear controllers need more effort for design and implementation compared to PID controllers and don’t show significant better performance.

LIST OF SYMBOLS

- $B_V$: Valve discharge coefficient
- $C_H$: Hydraulic capacity
- $D_V$: Valve damping coefficient
- $E$: Bulk modulus of hydraulic fluid
- $K_D$: Derivative gain
- $K_I$: Integral gain
- $K_P$: Proportional gain
- $k_{obs}$: Observer gain vector
- $K_V$: Valve input gain
- $k_{state}$: State feedback controller gain
- $P_A$: Pressure in the controlled volume
- $P_{AO}$: Operation pressure in the controlled volume
\( \Delta P_{\text{Nominal}} \) Nominal pressure difference of valve 
\( p_{\text{sup}} \) Supply pressure from the pump 
\( p_{\text{tank}} \) Reservoir pressure 
\( Q_A \) The flow rate in- and out from controlled volume 
\( Q_{\text{Nominal}} \) Nominal flow rate of the valve 
\( u \) Valve input signal 
\( V_A \) Volume of the hydraulic capacity 
\( V_\text{QU} \) Valve flow gain 
\( x_V \) Valve stroke 
\( \dot{x}_V \) Valve spool velocity 
\( x_{V0} \) Operation stroke of the valve spool 
\( x_{\text{max}} \) Maximum valve spool stroke 
\( \omega_V \) Valve eigenfrequency

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