Speed control of a hydrostatic drive using inverse steady state model

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Abstract. In this article a state of the art system inversion based control (SIBC) technique is proposed for regaining the speed of a pedagogical hydrostatic drive to its desired speed under changed load torque on the drive shaft. The input calculation through the SIBC is activated only when steady state value of the output deviates from the desired level. Therefore, the desired output is attained without making the system closed loop. As a result, neither high frequency oscillation (chattering) in the control input nor the risk of closed loop instability comes into the picture as of PID controller where system output is continuously compared with the reference to calculate the input.

1. Introduction

A hydrostatic drive is used for transmitting power using hydraulic oil as a medium [1]. It offers many advantages in comparison to pure mechanical transmissions [2]. With the incorporation of controllers, it is widely being used in the industrial application, where a fast response, high tracking accuracy and the high load variability are important [3-4].

The classical approach of controlling the hydrostatic drive is PID control. On line gain-scheduled PID control is adopted to tackle unknown load torque on the drive shaft, unknown leakage flow, etc. In this direction, various methods such as fuzzy logic, neural network and optimization techniques are used to tune the PID parameters [5-7]. To improve the control performance further, different model-based control methods are established [8-13]. The model-based control is categorized into forward and inverse model based. In inverse model based approach, the desired output is mapped into the input of the controller. The system inversion based control has been of interest to control engineers for many years [14-16]. Some of the important issues associated with system inversion are robust determination of observability, calculation of the desired input in case of unknown initial condition [17], and compensation of dead time, nonlinearity and measured disturbances [18].
This paper deals with a novel steady-state output feedback control of hydrostatic drive using state-of-the-art system inversion technique. The objective is to calculate the control input with unknown initial conditions and to use the calculated input in such a way that improved transient response is achieved. The contribution of this research is that the control input has been calculated at steady state instead of taking derivatives of the states in order to avoid the dependency on initial condition. The effectiveness of the proposed schemes has been demonstrated through the simulation and the experimental responses.

2. The test set-up
A schematic representation of the test set-up is shown in figure. 1. It consists of a variable speed electric motor (item A) driving a fixed displacement vane pump (item B), which delivers pressurized flow to the hydraulic motor (item D) through a directional control valve (item C). The pressure relief valve (item G) protects the system from the pressure exceeding the set limit. For varying the load on the hydro-motor, a loading circuit is provided. The drive shaft of the hydro-motor is connected to the pump (item E) of the loading circuit, where outlet flow of the pump is released to the tank through pressure relief valve (item F). The load torque on the shaft is varied by adjusting the set pressure of this pressure relief valve. The hydraulic motor speed, loading pump pressure are measured by using a proximity sensor (S/S) and a pressure transducer (P/T), respectively.

The load used here is an artificial load which is applied through the pressure relief valve (PRV) (item F in figure 1 of the manuscript). In the loading circuit of the system, the outlet flow of the loading pump is restricted by varying the port opening area of the proportional PRV by changing its cracking pressure. Step load was applied by means of controller to the solenoid of the proportional PRV.

![Figure 1. Schematic test set-up.](image)

3. System inversion
To compensate the effect of the altered load on the drive speed, the flow supplied by the pump needs to be adjusted. In the test setup shown in the figure, 1, the load on the motor can be varied by changing the port opening area of the loading valve \( A_{lv} \) to \( \tilde{A}_{lv} \). Accordingly, the hydraulic system gets transformed into a different state, and all the system responses (e.g. pressure flow rate, torque speed) are changed.

The sequence followed for calculating the desired value of the control input is numbered below:

**Step 1)** The changed steady state system responses corresponding to \( \tilde{A}_{lv} \) are assigned in the state matrix \( \{\tilde{x}\} \). The * marked states represent measurement and remaining states are observed states. Although
the system is observable with only one sensor (\(\omega_m\)), however an additional measurement of loading pump pressure (\(P_{lp}\)) is used to know the torque acting on the hydro-motor shaft.

\[
\{\dot{x}\} = \{\dot{P}_p \hat{P}_m \dot{P}_{lp} \omega^*_m\}^T
\]  

(1)

Steady state equations of the system are:

\[
D_p \omega_{em} - \frac{\dot{P}_p}{R_{\text{kg}_{1}}} - C_d A \sqrt{\frac{2(\hat{P}_p - \dot{P}_m)}{\rho}} = 0
\]  

(2)

\[
C_d A \sqrt{\frac{2(\hat{P}_p - \dot{P}_m)}{\rho}} - \frac{\hat{P}_m}{R_{\text{em}_m}} - D_m \omega^*_m = 0
\]  

(3)

\[
\hat{P}_m D_m - \omega^*_m R_{\text{fric}} - \hat{P}_{lp} D_{lp} = 0
\]  

(4)

\[
D_m \omega^*_m - \frac{\hat{P}_{lp}}{R_{\text{kg}_{lp}}} - C_d \tilde{A}_h \sqrt{\frac{2 \hat{P}_{lp}}{\rho}} = 0
\]  

(5)

**Step 2)** From equation (5), \(\tilde{A}_h\) is calculated as

\[
\tilde{A}_h = \frac{1}{C_d} \sqrt{\frac{\rho}{2 \hat{P}_{lp}}} \left( D_m \omega^*_m - \frac{\hat{P}_{lp}}{R_{\text{kg}_{lp}}} \right)
\]  

(6)

**Step 3)** Assign \(\omega_{em}\) to its changed speed \(\tilde{\omega}_{em}\) so as to regain the speed of hydro-motor \(\omega^*_m\) to its desired value \(\omega^*_m\), replace \(\tilde{A}_h\) by the expression given in equation (6), then update the state matrix to new variable \(\{\tilde{x}\}\).

\[
\{\tilde{x}\} = \{\tilde{P}_p \tilde{P}_m \dot{P}_{lp} \omega^*_m\}^T
\]  

(7)

Steady state equations with new state variables are:

\[
D_p \omega_{em} - \frac{\dot{P}_p}{R_{\text{kg}_{1}}} - C_d A \sqrt{\frac{2(\hat{P}_p - \dot{P}_m)}{\rho}} = 0
\]  

(8)

\[
C_d A \sqrt{\frac{2(\hat{P}_p - \dot{P}_m)}{\rho}} - \frac{\hat{P}_m}{R_{\text{em}_m}} - D_m \omega^*_m = 0
\]  

(9)

\[
\hat{P}_m D_m - \omega^*_m R_{\text{fric}} - \hat{P}_{lp} D_{lp} = 0
\]  

(10)

\[
D_m \omega^*_m = \frac{\hat{P}_{lp}}{R_{\text{kg}_{lp}}} - \left( D_m \omega^*_m - \frac{\hat{P}_{lp}}{R_{\text{kg}_{lp}}} \right) = 0
\]  

(11)

From equation (8) through equation (11), \(\hat{P}_p\), \(\tilde{P}_m\), \(\tilde{P}_{lp}\) and \(\tilde{\omega}_{em}\) are calculated as

\[
P_{lp} = R_{\text{kg}_{lp}} \left( D_m \omega^*_m - \left( D_m \omega^*_m - \frac{\hat{P}_{lp}}{R_{\text{kg}_{lp}}} \right) \right)
\]  

(12)

\[
P_m = \frac{\omega^*_m R_{\text{fric}} + \hat{P}_{lp} D_{lp}}{D_m}
\]  

(13)
\[ P_p = P_m + \frac{\rho}{2(C_d A_v)} \left( \frac{\bar{P}_m}{R_{lm}} + D_m \omega_m^* \right)^2 \]  \hspace{1cm} (14)

\[ \bar{\omega}_{em} = \frac{1}{D_p} \left( \frac{\bar{P}_p}{R_{kg1}} + \frac{\bar{P}_m}{R_{lm}} + D_m \omega_m^* \right) \]  \hspace{1cm} (15)

In order to increase the speed of the hydro-motor to its nominal value, the current pump speed has to be updated into \( \bar{\omega}_{em} \). Validation of this control scheme through simulation and experiment has been given in the next section. The parameter values used in the simulation are listed in Table 1.

### Table 1. Parametric values and set pressure considered for simulation.

| Parameters                          | Values          | Parameters                          | Values          |
|-------------------------------------|-----------------|-------------------------------------|-----------------|
| Port opening area of DCV            | 4.04x10^{-6} m² | discharge coefficient               | 0.90            |
| Bulk stiffness of hydraulic oil     | 5x10^{10} N/m²  | Density                             | 870 kg/m³       |
| Moment of inertia of motor          | 1.5x10^{-2} kg.m² | Pump (B) displacement               | 2.25x10^{-6} m³/rad |
| Pump combination                     |                 | Motor (D) displacement               | 7.77x10^{-7} m³/rad |
| Pump (E) displacement               | 7.77x10^{-7} m³/rad |               |                 |
| Pump (B) Leakage Resistance         | 1x10^{12} N.s/m³ | Frictional resistance               | 1.4x10^{-2} N   |
| Motor (D) Leakage Resistance        | 5x10^{12} N.s/m³ | Atmospheric pressure                | 1 bar           |

### 4. Results and discussions

Initially, up to 13 sec, the system has been operated at constant pump speed. Thereafter, for the same pump speed, the set pressure of the loading valve has been increased. Referring to figure 2 it is clear that for increased set pressure of the loading valve, the speed of the hydro-motor is reduced. In order to increase the speed of the hydro-motor to its previous value, the pump (B) speed has been increased rapidly to the calculated value obtained using Equation. 15. Accordingly, the nominal steady state value of motor speed is achieved.

![Figure 2. Hydro-motor speed recovery through system inversion based control (SIBC).](image-url)
There are many control techniques through which the same objective can be attained without system inversion, such as PID control. PID control is one of the most popular and wide spread control technique applied in industry. The simulation and experimental response obtained from PID control are shown in figure 3. The corresponding input signals are shown in figure 4. In PID control, the control input evolve continuously (see figure 4) to make the output error zero. Therefore, the fluctuation in control input corresponding to PID after applying control action (23 s onward) is higher. Practically, there exists a hysteresis in the control hardware implementation, i.e. instantaneous changing of physical parameter of the control actuator is not possible. Due to this delay, the PID control as compared to SIBC (system inversion based control) yields more steady state error. Additionally, chattering in the control input of PID also yields high heat loss in electrical circuits and high wear of moving mechanical parts. The close loop nature of the PID control may excite the high-frequency dynamics leading to degraded control performance and in some cases may make the system unstable. These issues do not arise in the system inversion method. Therefore, the system inversion method can be a better choice for many similar applications.

5. Conclusion
In this article, the speed recovery of a hydro-motor from an altered speed, due to load variation, to its normal speed has been demonstrated using system inversion technique. Instead of contemporary output feedback, the output is first saved at steady state and then it is used by system inversion module to calculate the desired input of the system. In the inversion technique, the desired output has been attained without making the system closed loop. As a result, neither high frequency oscillation in the control input nor the problem of closed-loop instability comes into the picture as of PID controller. Since step
change in the control input has been used to vary the pump speed to its calculated value, the maximum overshoot and transient fluctuation are high. A gradual change in input may reduce the maximum overshoot and transient fluctuation significantly. Therefore, effect of the different trends of pump speed (gradual input variation) on the transient response of hydro-motor speed would be investigated in the future to find the optimal one.

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