HYDRAULIC POSITION CONTROL SYSTEM WITH VARIABLE SPEED PUMP

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ABSTRACT
In this study, a valveless energy saving hydraulic position control servo system controlled by two pumps is investigated. In this system, two variable speed pumps driven by servomotors regulate the flow rate through a differential cylinder according to the needs of the system, thus eliminating the valve losses. The mathematical model of the system is developed in MATLAB Simulink environment. A Kalman filter is applied to reduce the noise in the position feedback signal. In the test set up developed, open loop and closed loop frequency response and step response tests are conducted by using MATLAB Real Time Windows Target (RTWT) module, and test results are compared with the model outputs.

NOMENCLATURE

<table>
<thead>
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<th>Description</th>
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<tr>
<td>A</td>
<td>System matrix</td>
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<tr>
<td>A_A</td>
<td>Cap end piston area</td>
</tr>
<tr>
<td>A_B</td>
<td>Rod end piston area</td>
</tr>
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<td>B</td>
<td>Input matrix</td>
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<td>C</td>
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<tr>
<td>C_l</td>
<td>Internal leakage coefficient of the pump</td>
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<tr>
<td>C_e</td>
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<td>C_cy</td>
<td>Internal leakage coefficient of the cylinder</td>
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<td>D_p</td>
<td>Pump displacement</td>
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<td>E</td>
<td>Bulk modulus of hydraulic oil</td>
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<td>H</td>
<td>Measurement Matrix</td>
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<td>P</td>
<td>Non-dimensional power</td>
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<td>P</td>
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<td>Measurement noise covariance matrix</td>
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<td>Q</td>
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<td>Q_pump</td>
<td>Pump flow rate</td>
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<td>Cap end cylinder volume</td>
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<tr>
<td>V_B</td>
<td>Rod end cylinder volume</td>
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<td>b</td>
<td>Viscous friction coefficient of the load</td>
</tr>
<tr>
<td>m</td>
<td>Mass of the load</td>
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<td>n</td>
<td>Pump speed</td>
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<td>p_a</td>
<td>Cap end chamber pressure</td>
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<tr>
<td>p_b</td>
<td>Rod end chamber pressure</td>
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<td>p_sum</td>
<td>Hydraulic cylinder chambers pressure sum</td>
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<td>p_l</td>
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<td>q</td>
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<td>Non-dimensional valve opening</td>
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<td>β</td>
<td>Offset speed ratio</td>
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<td>γ</td>
<td>Area ratio</td>
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<td>Ψ</td>
<td>Conversion factor between the desired sum pressure and pump speed</td>
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INTRODUCTION
Most conventional hydraulic control systems are based on valve controlled cylinders, in which valves located next to the actuator regulate the flow rate by changing their orifice areas. In spite of their high precision and fast dynamic behavior, considerable amount of hydraulic energy is wasted as heat loss to the environment due to throttling in control valves, increasing the oil temperature. The basic approach to improve the energy efficiency in hydraulic systems is to eliminate valve losses by regulating the flow according to the load requirements. Hence, in energy efficient hydraulic systems pump control techniques become the center of the focus. There are mainly two methods to control the flow rate of a pump. In the first method, the flow rate is regulated by changing the pump displacement whereas in the
second one, the flow rate is regulated by changing the drive speed of a constant displacement pump. Furthermore, the combination of these two methods that is changing the flow rate by both changing the displacement and drive speed of the pump can also be used.

In recent years, engineering systems are forced to be energy efficient due to limited and high-priced energy resources and the increasing environmental sense. For this reason, factors like the total energy usage, noise level, amount of oil used, and oil replacement cost are becoming important performance criteria as well as the dynamic response. In this study, a variable speed pump control principle is investigated and applied on a test setup. The position of a differential cylinder is controlled with two variable speed pump units. The flow rate is fully adapted to the load requirements by regulating the drive speed of the pumps. By this way, the throttle losses are eliminated completely and only leakage and friction losses of motor and pump unit will remain as factors to reduce the efficiency. In this study, it is aimed to eliminate the valve losses without conceding from the dynamic performance [1].

INVESTIGATION OF VALVE LOSSES IN CONVENTIONAL HYDRAULIC SYSTEMS

In constant pressure hydraulic control systems, the pump output pressure is always constant and equal to the set pressure of the relief valve. By this way, the pump output pressure is not affected from the changes in valve opening.

For simplicity, the pressure-flow relationship and the characteristic curves of a zero lapped 4-way valve are considered in this study. In a zero lapped valve, only two of the arms are open at any one time, and the configuration becomes a simple series circuit. The non-dimensional valve characteristic equation of this valve can be written as [2]

\[ p_L = 1 - \frac{q_L^2}{\alpha^2} \] (1)

In Fig. 1, a family of parabolas for different valve openings shows the characteristics defined by Eq. (1). The load locus shown in Fig. 1 is drawn for an ideal fictitious load whose maximum power requirement intersects with the maximum power output of the valve. Thus, the intersection point A represents the maximum efficiency of a valve controlled system. The coordinates of this point can be found by writing the power equation transmitted to the load, which is the area formed by drawing perpendicular lines to the axes and defined by Eq. (2).

\[ P = p_L q_L = q_L \left(1 - \frac{q_L^2}{\alpha^2}\right) \] (2)

The derivative of the Eq. (2) with respect to \( q_L \) gives the non-dimensional pressure and flow rate which are found to be \( p_L = 2/3 \) and \( q_L = \sqrt{1/3} \). At that point, the maximum non-dimensional power that can be transmitted to the load is written as

\[ P_{\text{max}} = p_L q_L = \frac{2}{3} \sqrt{\frac{1}{3}} = 0.385 \] (3)

In Fig. 1, the maximum power that can be transmitted to the load is shown by Area I which is 38.5% of the total supplied power. Area II shows the power losses on the relief valve and Area III shows the power losses on the flow control valve. Therefore, a great portion of the hydraulic power supplied by the pump is lost on the pressure relief valve and the flow control valve. On the top of these losses, some additional energy is consumed to cool the heated oil returning to the oil tank. All these form the basic rationale behind eliminating control valves for an energy efficient hydraulic system.

VARIABLE SPEED PUMP DRIVE SYSTEM

In variable speed pump control systems, the servo valve, as the control element in conventional hydraulic systems, is replaced by a servo pump. Therefore, all throttling losses within the hydraulic power circuit are eliminated [4]. Hence, the hydraulic system used in this study is composed of three parts; a hydraulic actuator, a constant displacement pump, and a servo motor to drive the pump. A differential cylinder is used as the hydraulic actuator because, due to its compact design, low cost and ease of manufacture in most of the industrial applications, single rod hydraulic actuators are used. A servo motor is used to drive the pump since the dynamic response is an important criterion for the system performance.
The variable speed pump controlled hydraulic system is shown in Fig. 2. The CCW rotation of the pumps and upwards extension of the hydraulic cylinder are assumed to be positive. Pump 2 controls the direction and the velocity of the actuator, while pump 1 compensates the asymmetric flow rate due to the difference in areas on two sides of the piston. During the extension, pump 1 provides the lacking flow for the cap end and during the retraction it absorbs the excess flow of pump 2 [5].

Assuming that the leakages are compensated, the speed of pump 1 to compensate for lacking flow due to the area difference should be

$$Q_{\text{pump} 1} = (\gamma - 1)Q_{\text{pump} 2}$$

(4)

The pumps used in this study are identical and their displacements are the same, then Eq. (4) can be written in the form;

$$n_1 = (\gamma - 1)n_2$$

(5)

In a stationary position without any load, pump 2 rotates in negative direction and pressurizes the rod end of the cylinder. Pump 1 rotates in positive direction supplies the flow needed by pump 2 as well as internal and external leakages while pressurizing the cap end of the cylinder.

The speed ratio $\beta$ of the pumps, which is negative and depends on their internal and external leakage coefficients as well as the internal leakage of the cylinder, can be derived by a simple static analysis. It is used to relate the offset pump speeds as

$$n_{10} = \beta n_{20}$$

(6)

If the pumps are driven at speeds corresponding to different speed ratio than $\beta$, the cylinder chambers will still be pressurized but the piston will not be stationary and move forward or backward.

The offset speed signals $n_{10}$, $n_{20}$ are constant and determined according to the desired sum pressures of the chambers. The relation between the offset pump speed of the second pump and the desired sum pressure of the chambers is given as

$$n_{20} = \Psi p_{\text{Sum}}$$

(7)

Here, constant $\Psi$ depends on the leakage coefficients, area ratio and the pump displacement. It is derived by a simple static analysis.

After pressurizing the cylinder chambers, the dynamic pressure changes due to external loading will change around these static chamber pressures. Note that the desired sum pressure of the chambers $p_{\text{Sum}}$ should be selected such that the cap end dynamic pressure change due to an external loading should not be greater than the cap end static pressure. Otherwise, when the load is applied or the piston is accelerated, a negative pressure will be needed to balance the applied load.

To sum up, there are two control signals in this variable speed pump drive system. The first signals $n_{10}$ and $n_{20}$ are the pressure control signals. They are constant and applied to preload the cylinder chambers while compensating the leakages. The second signals $n_1$ and $n_2$ are the position control signal of the closed loop control system. The relation between these synchronous pump speeds are defined by Eq. (5) and Eq. (6). The speed setting of each pump becomes the sum of the open loop pressure control signal and the closed loop position control signal as follows.

$$n_1 = n_1 + n_{10} = (\gamma - 1)n_2 + \beta n_{20}$$

(8)

$$n_2 = n_2 + n_{20}$$

(9)
MATHEMATICAL MODELING OF THE SYSTEM

Servo motors are assumed to be ideal angular velocity sources as each has a controller inside and both have faster dynamics than the hydraulic system. Also, the hydraulic transmission lines are assumed to be lossless and are not modeled. The hydraulic capacitances constituted by the transmission line volumes are lumped into the associated hydraulic cylinder chamber volumes. The models of the pumps and the actuator as the remaining parts of the system are given below.

Hydraulic pump model

Two identical internal gear pumps are used in this application. Due to their symmetric geometry, these pumps have equal resistances to the flow in both directions and can be driven in 4 quadrants. This is an important property as the load locus of the system is in 4 quadrants, so these pumps should be able to operate in 4 quadrants.

Operating in 4-quadrant means that the pump unit can both work as a hydraulic pump or a hydraulic motor; that is, both the high pressure port and the flow direction can change.

Assuming the flow rate passing through the leakage paths are very low so that the flow is laminar, a simple linear model relating the leakage flow rate to the pressure difference is used. According to the assumed positive directions, the A side ports of pump 1 and pump 2 are the outlet ports, and the B side port of pump 2 is the inlet port. The flow rate equations for the inlet and the outlet ports of these pumps are given below [6]. These equations are valid both for pump and motor model.

\[ Q_{p_{1A}} = D_p n_1 - C_{i_1} (p_A - p_i) - C_{i_1} p_d - D_p n_1 - C_{i_1} p_d \] (10)

\[ Q_{p_{2A}} = D_p n_2 - C_{i_2} (p_A - p_b) - C_{i_2} p_d \] (11)

\[ Q_{p_{2B}} = D_p n_2 - C_{i_2} (p_A - p_b) + C_{i_2} p_b \] (12)

Hydraulic actuator model

Because the extension of the hydraulic cylinder is assumed to be the positive direction, the positive \( Q_a \) represents the incoming flow to the cap end of the actuator and the positive \( Q_b \) represents the outgoing flow from the rod end of the actuator. Then continuity equations of the hydraulic cylinder are given below.

\[ Q_a = A_a \Delta x + \frac{V_A}{E} \frac{dp_A}{dt} + C_{cy} (p_A - p_b) \] (13)

\[ Q_b = A_b \Delta x - \frac{V_B}{E} \frac{dp_B}{dt} + C_{cy} (p_A - p_b) \] (14)

Static behavior of the system

For the stationary case, the continuity equation for the rod end of the cylinder can be written as follows by using Eq. (12) and Eq. (14),

\[ D_p n_{20} - C_{i_2} (p_A - p_b) + C_{2e} p_b = C_{cy} (p_A - p_b) \] (15)

For zero static loading, the cap end and the rod end chamber pressures can be written as follows by using the static equilibrium.

\[ p_A = \frac{P_{sum}}{\gamma + 1} \] (17)

\[ p_b = \frac{\gamma P_{sum}}{\gamma + 1} \] (18)

where \( P_{sum} \) is the desired sum pressure of the hydraulic cylinder chambers.

If Eq. (17) and the Eq. (18) are inserted in Eq. (15) and Eq. (16), the offset speed of the pump 2 and pump 1 will be found as,

\[ n_{20} = \frac{-(\gamma - 1)(C_{2e} + C_{cy}) + \gamma C_{2e}}{(\gamma + 1)D_p} P_{sum} = \Psi P_{sum} \] (19)

\[ n_{10} = \frac{(\gamma + 1)C_{2e} + C_{i_1}}{(\gamma + 1)D_p} P_{sum} \] (20)

Then the ratio between the offset speeds are given as

\[ \beta = \frac{n_{10}}{n_{20}} = -\frac{(\gamma + 1)C_{2e} + C_{i_1}}{(\gamma - 1)(C_{2e} + C_{cy}) + \gamma C_{2e}} \] (21)

Dynamic behavior of the system

In the dynamic analysis, the flows due to the rod movement and the compressibility are added to the continuity equations defined by Eq. (15) and Eq. (16). The offset speeds do not affect the dynamic behavior of the system, so the continuity equations are written in terms of the position control signal speeds \( n_1 \) and \( n_2 \).
By using Eq. (12) and Eq. (14), the continuity equation for the rod end can be written in s-domain as
\[
D_pN_2(s)-A_gx(s)=\left(C_{2i}+C_{oy}\right)P_d(s)-\left(C_{2i}+C_{oy}+C_{ze}+\frac{V_a}{E}s\right)P_b(s)
\]  

(22)

By using Eq. (5), Eq. (10), Eq. (11), and Eq. (13), the continuity equation for the cap end can be written as follows,
\[
\gamma\left[D_pN_2(s)-A_gx(s)\right]=\left(C_i+C_{oy}+C_{ze}+\frac{V_a}{E}s\right)P_i(s)-\left(C_{2i}+C_{oy}\right)P_b(s)
\]

(23)

The force balance on the load gives
\[
(\gamma P_i(s)-P_b(s))A_B=(ms^2+bs)X(s)
\]

(24)

Arranging Eq. (22), Eq. (23), and Eq. (24), the transfer function between the speed of pump 2 and the rod position can be written as follows,
\[
\frac{X(s)}{N_2(s)}=rac{\left(V_a+\gamma\frac{V_a}{E}s\right)+F}{\left[m\left(\frac{V_a}{E}\right)^2+(mG+b+H+A_{g}\gamma)\right]s^2+(b(2(C_{2i}+C_{oy})+A_{g}\gamma))s}
\]

(25)

If the numerical values of the system parameters are used in this transfer function, it will be seen that the system has a zero and a pole next to each other. This is due to the chamber pressure relations. It is seen that the left hand sides of Eq. (22) and Eq. (23) are proportional with the area ratio \(\gamma\). From these two equations the relation between the dynamic chamber pressure changes can be written as follows,
\[
P_i(s)=-\frac{(\gamma-1)(C_{2i}+C_{oy})+\gamma C_{ze}+\frac{V_a}{E}s}{C_{i}+C_{ze}+\left(C_{2i}+C_{oy}\right)(1-\gamma)+\frac{V_a}{E}s}P_b(s)
\]

(26)

Equation (26) implies that for the following specific volume ratio and leakage coefficients pressure changes of the chambers will become identical but in opposite directions.

\[
V_a=\gamma V_b
\]

\[
C_i=(\gamma-1)\left(2\left(C_{2i}+C_{oy}\right)+C_{ze}\right)
\]

(27)

That is, \(P_3(s)=-P_6(s)\). When this condition holds, the order of the system is reduced to 3, and a simpler transfer function is obtained as
\[
\frac{X(s)}{N_2(s)}=rac{A_pN_2(\gamma+1)}{s^2+(b(2(C_{2i}+C_{oy})+A_{g}\gamma))s}
\]

(28)

TEST SET-UP

The block diagram representation of the test set-up used in this study is shown in the Fig. 3.

![FIGURE 3. TEST SET-UP BLOCK DIAGRAM](image)

The hardware and the software used in the test set-up are given below.

- **Hydraulic pump:** Bucher hydraulics QXM32-16 series internal gear pump. The pump displacement is 15.6 cm³/rev. The maximum speed in pump mode is 3,900 rpm and in the motor mode is 5,500 rpm. The maximum pressure is 210 bar and the maximum torque is 80Nm.

- **Servomotor:** Teco JSDA 30 AC series servo motor. The nominal power of the motor is 2kW, the maximum speed is 2,000 rpm and the maximum torque is 14.5Nm. The servo motor is driven single phase with 220VAC source.

- **Data Acquisition Card:** NI 6024E (PCMCIA) 16 bit card.

- **Position transducer:** Balluff BTL series contactless linear position transducer. The stroke of the transducer is 0-100 mm and has a 0-10 V analog output. The resolution is 10 microns.
• **Load:** A steel plate of a mass of 11.6 kg

• **Software:** The mathematical model and the real-time control algorithm are prepared in MATLAB Simulink environment. The system is controlled in real time by using the MATLAB RTWT algorithm.

### KALMAN FILTER DESIGN

A Kalman filter is designed to filter the noisy position feedback signal in this study. The Kalman filter estimates a process by using a form of feedback control. The filter estimates the process state at some time and then obtains feedback in the form of (noisy) measurements. The equations for the Kalman filter fall into two groups: time update equations and measurement update equations. The following time update equations

\[ \hat{q}_k = A\hat{q}_{k-1} + Bu_{k-1} \]  

\[ P_k = AP_{k-1}A^T + Q \]  

are responsible for projecting the current state and error covariance estimates at time step \( k-1 \) to obtain the priori estimates for the time step \( k \).

The measurement update equations are responsible for incorporating a new measurement into priori estimate and obtain an improved a posteriori estimate [7-8]. The first task in measurement update equation is to compute the Kalman gain \( K_k \).

\[ K_k = P_k H^T (HP_k H^T + R)^{-1} \]  

The next step is to actually measure the process to obtain \( \hat{q}_k \) and then to generate an a posteriori estimate of states by incorporating the measurement as

\[ \hat{q}_k = \hat{q}_k + K_k (z_k - H\hat{q}_k) \]  

The final step is to obtain a posteriori error covariance as

\[ P_k = (I - K_k H)P_k \]  

In Eq. (30), \( Q \) is the process noise. It is the covariance matrix of errors in the state variables that have been caused by not truly representative of the system. \( R \) is the measurement noise. It is the covariance matrix of the likely errors in the measured values.

After each time and measurement update pair, the process defined through Eq. (30) to Eq. (33) is repeated with the previous a posteriori estimates used to project or predict the new a priori estimates. The MATLAB Simulink model representing this process is shown in Fig. 4.

Since the Kalman filter is a discrete time algorithm and also to be compatible with real time digital computing algorithm, the state space equations of the system are written and discretized by using the MATLAB \( c2dm \) command with zero order hold method and with a sampling time of 2 ms.

The states, \( q \), of the system are chosen as the actuator position, velocity, and the chamber pressures. The output, \( y \), of the system is cylinder position, while the control input is the speed of pump 2.

\[ \begin{bmatrix} q \n x \n p_A \n p_B \end{bmatrix}^T, \ y = [x], \ u = [n_2] \]  

The system matrix \( A \), and the input matrix \( B \) can be written by using Eq. (22), Eq. (23), and Eq. (24) as

\[ A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & -\frac{b}{m} & \gamma A_B & -\frac{A_B}{m} \\ 0 & \frac{E_A d}{V_A} & \frac{E(C_2 + C_{2i} + C_{1o})}{V_A} & \frac{E(C_2 + C_{1o})}{V_A} \\ 0 & \frac{E_A d}{V_B} & \frac{E(C_2 + C_{1o})}{V_B} & \frac{E(C_{2i} + C_{1i})}{V_B} \end{bmatrix} \]  

\[ B = \begin{bmatrix} 0 & 0 & \frac{E_A d}{V_A} & \frac{ED_P}{V_B} \end{bmatrix}^T \]  

The actuator position is the only filter output, then the output matrix becomes...
The measured position and the filtered position are shown in the Fig. 5. Unlike from the valve controlled hydraulic systems, the pump controlled systems are linear. Therefore, as there is no linearization and loss of information due to higher order terms, the linear Kalman filter performs well while estimating the future state Eq. (29) and the error covariance Eq. (30).

\[
C = [1 \ 0 \ 0 \ 0] \quad (37)
\]

In the open loop frequency response test, a sinusoidal signal with an exponentially decaying magnitude is applied to the system in order to prevent the piston rod to reach the end of the stroke at low frequencies. The amplitude of the test signal starts from 5 V at high frequencies around 30 Hz and decreases exponentially while its frequency decreases continuously down to 0.1 Hz. The test signal is shown in Fig. 6.

To find the frequency response of the system FFT of the input signal and the system output are taken to determine the amplitudes of the constituting harmonics and their frequencies. FFT’s are taken with MATLAB `fft` command. To increase the accuracy eight consecutive tests are conducted and their mean is used to draw the Bode diagram.

Figure 6 shows the experimental and the theoretical open loop frequency response of the system. Since the type number of the system is one due to the free s term in the denominator of the transfer function defined by Eq. (28), the slope of the Bode diagram at the low frequency region is –20 dB/dec, as expected. It is seen from the Bode diagram that the resonance frequency of the system is around 280 Hz. At this frequency region, damping dominates the dynamic behavior and some time should pass for the system to reach steady state. However, at low frequency region the system rapidly responds to the input signal and there is no need to wait for the system to reach steady state. Thus continuously changing the test signal frequency is not a problem for this frequency response tests.

After adding a position feedback, the bandwidth of the closed loop position control system can be adjusted by a proportional controller gain. The bandwidth of the system will increase by increasing the proportional gain. Theoretically the proportional gain can be increased as much as possible without making the system unstable. However, in practice, the higher order fast dynamics of the servo motor will prevent to use high gains. In this study the servo motor dynamics are neglected, which is true for low frequencies and small inertial loads. However, for higher frequencies or higher inertias, the motor dynamics should be considered.

A constant amplitude test signal is used for the closed loop tests. In Fig. 7, the closed loop frequency responses of the test set-up and mathematical model are compared.

![FIGURE 5. POSITION SIGNAL ESTIMATION VIA KALMAN FILTER](image)

![FIGURE 6. OPEN LOOP FREQUENCY RESPONSE AND TEST SIGNAL](image)

![FIGURE 6. OPEN LOOP FREQUENCY RESPONSE AND TEST SIGNAL](image)

![FIGURE 7. CLOSED LOOP FREQUENCY RESPONSE](image)
The test signal used has amplitude of 0.25 V and its frequency is varied from 0 to 20 Hz in 120 seconds. The proportional gain is taken as 24. The maximum pump speed at this test is 1,030 rpm. As seen from the Bode diagram the bandwidth of the closed loop system is around 20 Hz.

In Fig. 8 the step response of system is compared with the mathematical model results by employing square wave inputs. The proportional gain is chosen as 16. For this gain the maximum pump speed is 1,800 rpm.

**CONCLUSIONS**

In this study, an energy efficient variable speed pump control system investigated. First, the valve losses in a conventional hydraulic system are investigated. It is seen that in a conventional valve controlled system the maximum power that can be transmitted to the system is 38.5% of the total energy supplied by the pump. As an alternative configuration, a more energy efficient valveless hydraulic circuit is also investigated and its mathematical model is obtained. In this system, pump speeds are regulated according to the systems needs, thus eliminating the throttle losses and only leakage and friction losses of motor and pump units will remain as factors to reduce the efficiency. To demonstrate the dynamic performance of the system, open-loop and closed-loop frequency response tests are conducted and their results are compared with the mathematical model. From the test results it is seen that the dynamics of the pump controlled system can be comparable to valve controlled systems. Using a simple proportional controller with gain 24 the bandwidth of the closed loop system is found to be 20 Hz, which is satisfactory for most of the hydraulic applications. Besides, Kalman filtering is applied to reduce the noise in the position feedback signal. The valveless pump speed control system shows a good performance when a linear Kalman filter is employed to filter the noisy position measurements due to the linear nature of such a system. As a future work, the estimation of chamber pressures with a linear Kalman filter is aimed.

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