Yildiz Technical University

Department of Mechanical Engineering

Machine Theory, System Dynamics and Control Division

Special Laboratory Report - Position Control of Rotary Servo Base Unit using PIV Controller

Lab Date:

Number:

Name Surname:

Lab Director:

Group/Sub-group: ….. / ….

Lab Location: O Block - Automatic Control Laboratory
Lab Name: Machine Theory - 3
Subject: Position Control of Rotary Servo Base Unit (SRV02) using PV and PIV Controllers

Apparatus and tools:
- Computer with MATLAB-Simulink and QUARC software
- Data acquisition device, power amplifier, and main components of the SRV02 (e.g. actuator, sensors).

Aim of the experiment:
-Deriving the dynamics equation and transfer function for the SRV02 servo plant using the first-principles.
- Design of a proportional-velocity (PV) controller for position control of the servo load shaft to meet certain time-domain requirements.
-Design of a proportional-velocity-integral (PIV) controller to track a ramp reference signal.
-Implementation of the controllers on the Quanser SRV02 device to evaluate their performance.

1 System Modeling

1.1.1.1 Electrical Equations

The DC motor armature circuit schematic and gear train is illustrated in Figure 1.1. As known R_m is the motor resistance, L_m is the inductance, and k_m is the back-emf constant.

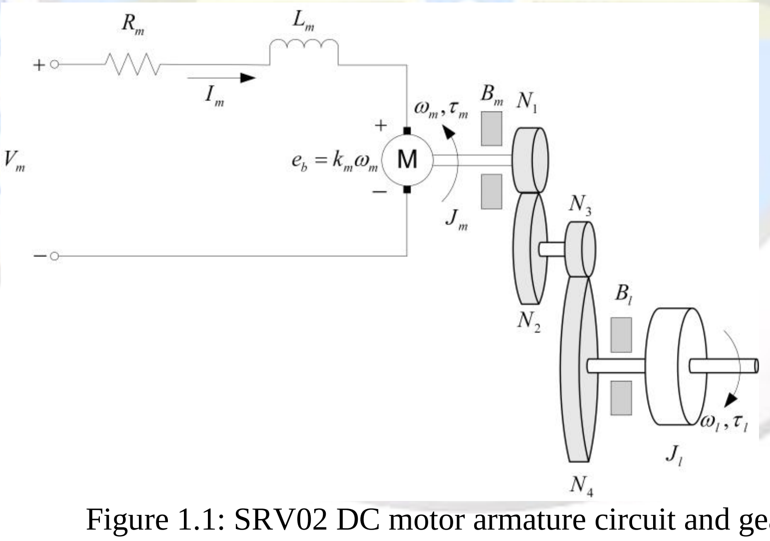

Figure 1.1: SRV02 DC motor armature circuit and gear train

The back-emf (electromotive) voltage $e_b(t)$ depends on the speed of the motor shaft, ω_m, and the back-emf constant of the motor, k_m. It opposes the current flow. The back emf voltage is given by:

$$e_b(t) = k_m\omega_m(t) \tag{1.1.2}$$

Using Kirchoff's Voltage Law, we can write the following equation:

$$V_m(t) - R_m I_m(t) - L_m \frac{dI_m(t)}{dt} - k_m\omega_m(t) = 0 \tag{1.1.3}$$

Since the motor inductance L_m is much less than its resistance, it can be ignored. Then, the equation becomes

$$V_m(t) - R_m I_m(t) - k_m\omega_m(t) = 0 \tag{1.1.4}$$

Solving for $I_m(t)$, the motor current can be found as:

$$I_m(t) = \frac{V_m(t) - k_m\omega_m(t)}{R_m} \quad (1.1.5)$$

1.1.1.2 Mechanical Equations

In this section the equation of motion describing the speed of the load shaft, ω_l, with respect to the applied motor torque, τ_m, is developed. Since the SRV02 is a one degree-of-freedom rotary system, Newton's Second Law of Motion can be written as:

$$J \cdot \alpha = \tau \quad (1.1.6)$$

where J is the moment of inertia of the body (about its center of mass), α is the angular acceleration of the system, and τ is the sum of the torques being applied to the body. As illustrated in Figure 1.1, the SRV02 gear train along with the viscous friction acting on the motor shaft, B_m, and the load shaft B_l are considered. The load equation of motion is

$$J_l\frac{d\omega_l(t)}{dt} + B_l\omega_l(t) = \tau_l(t) \quad (1.1.7)$$

where J_l is the moment of inertia of the load and τ_l is the total torque applied on the load. The load inertia includes the inertia from the gear train and from any external loads attached, e.g. disc or bar. The motor shaft equation is expressed as:

$$J_m\frac{d\omega_m(t)}{dt} + B_m\omega_m(t) + \tau_{ml}(t) = \tau_m(t) \quad (1.1.8)$$

where J_m is the motor shaft moment of inertia and τ_{ml} is the resulting torque acting on the motor shaft from the load torque. The torque at the load shaft from an applied motor torque can be written as:

$$\tau_l(t) = \eta_g K_g \tau_{ml}(t) \quad (1.1.9)$$

where K_g is the gear ratio and η_g is the gearbox efficiency. The planetary gearbox that is directly mounted on the SRV02 motor is represented by the N_1 and N_2 gears in Figure 1.1 and has a gear ratio of

$$K_{gi} = \frac{N_2}{N_1} \quad (1.1.10)$$

This is the *internal* gear box ratio. The motor gear N_3 and the load gear N_4 are directly meshed together and are visible from the outside. These gears comprise the *external* gear box which has an associated gear ratio of

$$K_{ge} = \frac{N_4}{N_3} \quad (1.1.11)$$

The gear ratio of the SRV02 gear train is then given by:

$$K_g = K_{ge}K_{gi} \quad (1.1.12)$$

Thus, the torque seen at the motor shaft through the gears can be expressed as:

$$\tau_{ml}(t) = \frac{\tau_l(t)}{\eta_g K_g} \quad (1.1.13)$$

Intuitively, the motor shaft must rotate K_g times for the output shaft to rotate one revolution.

$$\theta_m(t) = K_g\theta_l(t) \quad (1.1.14)$$

We can find the relationship between the angular speed of the motor shaft, ω_m, and the angular speed of the load shaft, ω_l by taking the time derivative:

$$\omega_m(t) = K_g\omega_l(t) \quad (1.1.15)$$

To find the differential equation that describes the motion of the load shaft with respect to an applied motor torque substitute (1.1.13), (1.1.15) and (1.1.7) into (1.1.8) to get the following:

$$J_m K_g\frac{d\omega_l(t)}{dt} + B_m K_g\omega_l(t) + \frac{J_l(\frac{d\omega_l(t)}{dt}) + B_l\omega_l(t)}{\eta_g K_g} = \tau_m(t) \quad (1.1.16)$$

Collecting the coefficients in terms of the load shaft velocity and acceleration gives

$$(\eta_g K_g^2 J_m + J_l)\frac{d\omega_l(t)}{dt} + (\eta_g K_g^2 B_m + B_l)\omega_l(t) = \eta_g K_g \tau_m(t) \quad (1.1.17)$$

Defining the following terms:

$$J_{eq} = \eta_g K_g^2 J_m + J_l \quad (1.1.18)$$

$$B_{eq} = \eta_g K_g^2 B_m + B_l \quad (1.1.19)$$

simplifies the equation as:

$$J_{eq}\frac{d\omega_l(t)}{dt}+B_{eq}\omega_l(t)=\eta_g K_g \tau_m(t) \tag{1.1.20}$$

1.1.1.3 Combining the Electrical and Mechanical Equations

In this section the electrical equation and the mechanical equation are brought together to get an expression that represents the load shaft speed in terms of the applied motor voltage. The motor torque is proportional to the voltage applied and is described as

$$\tau_m(t)=\eta_m k_t I_m(t) \tag{1.1.21}$$

where k_t is the current-torque constant $(N.m/A)$, η_m is the motor efficiency, and I_m is the armature current. We can express the motor torque with respect to the input voltage $V_m(t)$ and load shaft speed $\omega_l(t)$ by substituting the motor armature current given by equation 1.1.5, into the current-torque relationship given in equation 1.1.21:

$$\tau_m(t)=\frac{\eta_m k_t\left(V_m(t)-k_m\omega_m(t)\right)}{R_m} \tag{1.1.22}$$

To express this in terms of V_m and ω_l, insert the motor-load shaft speed equation 1.1.15, into 1.1.22 to get:

$$\tau_m(t)=\frac{\eta_m k_t\left(V_m(t)-k_m K_g\omega_l(t)\right)}{R_m} \tag{1.1.23}$$

If we substitute (1.1.23) into (1.1.20), we get:

$$J_{eq}\left(\frac{d}{dt}w_l(t)\right)+B_{eq}w_l(t)=\frac{\eta_g K_g\eta_m k_t\left(V_m(t)-k_m K_g\omega_l(t)\right)}{R_m} \tag{1.1.24}$$

After collecting the terms, the equation becomes

$$\left(\frac{d}{dt}w_l(t)\right)J_{eq}+\left(\frac{k_m\eta_g K_g^2\eta_m k_t}{R_m}+B_{eq}\right)\omega_l(t)=\frac{\eta_g K_g\eta_m k_t V_m(t)}{R_m} \tag{1.1.25}$$

This equation can be re-written as:

$$\left(\frac{d}{dt}w_l(t)\right)J_{eq}+B_{eq,v}\omega_l(t)=A_m V_m(t) \tag{1.1.26}$$

where the equivalent damping term is given by:

$$B_{eq,v}=\frac{\eta_g K_g^2\eta_m k_t k_m+B_{eq}R_m}{R_m} \tag{1.1.27}$$

and the actuator gain equals

$$A_m=\frac{\eta_g K_g\eta_m k_t}{R_m} \tag{1.1.28}$$

Questions

1. We obtained an equation (1.1.26) that described the dynamic behavior of the load shaft speed as a function of the motor input voltage. Starting from this equation, find the transfer function $\frac{\Omega_l(s)}{V_m(s)}$.

2. Express the steady-state gain (K) and the time constant (τ) of the process model in terms of the J_{eq}, $B_{eq,v}$, and A_m parameters.

$$\frac{\Omega_l(s)}{V_m(s)}=\frac{K}{(\tau s+1)}$$

3. Calculate the $B_{eq,v}$ and A_m model parameters using the system specifications given in Table 3.1.

4. Calculate the moment of inertia about the motor shaft. Note that $J_m = J_{tach} + J_{m,rotor}$ where J_{tach} and $J_{m,rotor}$ are the moment of inertia of the tachometer and the rotor of the DC motor, respectively. Use the specifications given in Table 3.1.

5. The load attached to the motor shaft includes a 24-tooth gear, two 72-tooth gears, and a single 120-tooth gear along with any other external load that is attached to the load shaft. Thus, for the gear moment of inertia J_g and the external load moment of inertia $J_{l,ext}$, the load inertia is $J_l = J_g + J_{l,ext}$. Using the specifications given in Table 3.2, find the total moment of inertia J_g from the gears . **Hint:** Use the definition of moment of inertia for a disc $J_{disc} = \frac{mr^2}{2}$.

6. Assuming the disc load is attached to the load shaft, calculate the inertia of the disc load, $J_{ext,l}$, and the total load moment of inertia, J_l. (Use m_d and r_d in Table 3.1)

7. Evaluate the equivalent moment of inertia J_{eq}.

8. Calculate the steady-state model gain K and time constant τ.

2 POSITION CONTROL

2.1.1 Steady State Error

Steady-state error is denoted by the variable e_{ss}. It is the difference between the reference input and output signals after the system response has settled. Thus, for a time t when the system is in steady-state, the steady-state error equals

$$e_{ss} = r_{ss}(t) - y_{ss}(t) \tag{2.1.10}$$

where $r_{ss}(t)$ is the value of the steady-state input and $y_{ss}(t)$ is the steady-state value of the output.

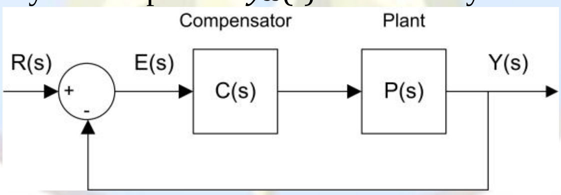

Figure 2.1: Unity feedback system.

We can find the error transfer function $E(s)$ in Figure 2.1 in terms of the reference $R(s)$, the plant $P(s)$, and the compensator $C(s)$. The Laplace transform of the error is

$$E(s) = R(s) - Y(s) \tag{2.1.11}$$

Solving for $Y(s)$ Figure 2.1 yields

$$E(s) = \frac{R(s)}{1 + C(s)\,P(s)} \tag{2.1.12}$$

We can find the the steady-state error of this system using the final-value theorem:

$$e_{ss} = \lim_{s \to 0} sE(s) \tag{2.1.13}$$

In this equation, we need to substitute the transfer function for E(s) from 2.1.12. The E(s) transfer function requires, R(s), C(s) and P(s). For simplicity, let C(s)=1 as a compensator. Then, the error becomes:

$$P(s) = \frac{K}{s\,(\tau s + 1)},\ R(s) = \frac{R_0}{s} \text{ (step input)} \Rightarrow \quad E(s) = \frac{R_0}{s\left(1 + \frac{K}{s\,(\tau s+1)}\right)} \tag{2.1.14}$$

Applying the final-value theorem gives

$$e_{ss} = R_0 \left(\lim_{s \to 0} \frac{(\tau\, s + 1)\; s}{\tau\, s^2 + s + K} \right) \tag{2.1.15}$$

When evaluated, the resulting steady-state error due to a step response is

$$e_{ss} = 0 \tag{2.1.16}$$

Based on this zero steady-state error for a step input, we can conclude that the SRV02 is a *Type 1* system.

2.1.2 SRV02 Position Control Specifications

The desired time-domain specifications for controlling the position of the SRV02 load shaft are:

$$e_{ss} = 0 \tag{2.1.17}$$

$$t_p = 0.20 \text{ s} \tag{2.1.18}$$

$$PO = 5.0 \text{ \%} \tag{2.1.19}$$

Thus, when tracking the load shaft reference, the transient response should have a peak time less than or equal to 0.20 seconds, an overshoot less than or equal to 5 %, and the steady-state response should have no error.

2.1.3 PV Controller Design

2.1.3.1 Closed Loop Transfer Function

The proportional-velocity (PV) compensator to control the position of the SRV02 has the following structure

$$V_m(t) = k_p\, (\theta_d(t) - \theta_l(t)) - k_v \left(\frac{d}{dt}\, \theta_l(t) \right) \tag{2.1.20}$$

where k_p is the proportional control gain, k_v is the velocity control gain, $\theta_d(t)$ is the setpoint or reference load shaft angle, $\theta_l(t)$ is the measured load shaft angle, and $V_m(t)$ is the SRV02 motor input voltage. The block diagram of the PV control is given in Figure 2.4. We need to find the closed-loop transfer function $\Theta_l(s)/\Theta_d(s)$ for the closed-loop position control of the SRV02. Taking the Laplace transform of equation 2.1.20 gives

$$V_m(s) = k_p\, (\Theta_d(s) - \Theta_l(s)) - k_v\, s\Theta_l(s) \tag{2.1.21}$$

To find the voltage-to-position transfer function, we can put an integrator $(1/s)$ in series with the speed transfer function (effectively integrating the speed output to get position). From the Plant block in Figure 2.4 and this transfer function, we can write

$$\frac{\Theta_l(s)}{V_m(s)} = \frac{K}{s\,(\tau\, s + 1)} \tag{2.1.22}$$

Substituting equation 2.1.21 into 2.1.22 and solving for $\Theta_l(s)/\Theta_d(s)$ gives the SRV02 position closed-loop transfer function as:

$$\frac{\Theta_l(s)}{\Theta_d(s)} = \frac{K\, k_p}{\tau\, s^2 + (1 + K\, k_v)\, s + K\, k_p} \tag{2.1.23}$$

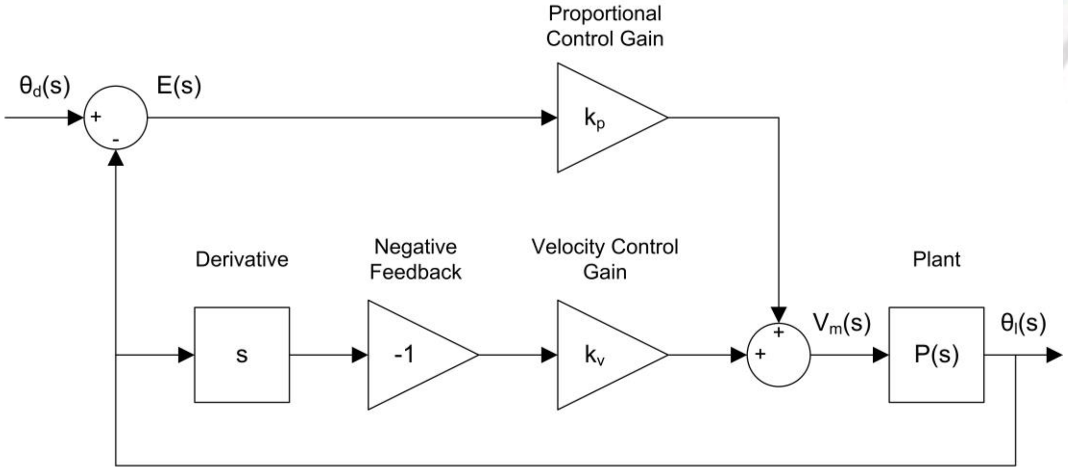

Figure 2.4: Block diagram of SRV02 PV position control

2.1.3.2 Ramp Steady State Error Using PV Control

From our previous steady-state analysis, we found that the closed-loop SRV02 system is a Type 1 system. In this section, we will investigate the steady-state error due to a ramp input when using PV controller. Given the following ramp setpoint (input)

$$R(s) = \frac{R_0}{s^2} \tag{2.1.25}$$

we can find the error transfer function by substituting the SRV02 closed-loop transfer function in equation 2.1.23 into the formula given in 2.1.11. Using the variables of the SRV02, this formula can be rewritten as $E(s) = \Theta_d(s) - \Theta_l(s)$. After rearranging the terms we find:

$$E(s) = \frac{\Theta_d(s)\, s\,(\tau\, s + 1 + K\, k_v)}{\tau\, s^2 + s + K\, k_p + K\, k_v\, s} \tag{2.1.26}$$

Substituting the input ramp transfer function 2.1.25 into the $\Theta_d(s)$ variable gives

$$E(s) = \frac{R_0\ (\tau\, s + 1 + K\, k_v)}{s\ (\tau\, s^2 + s + K\, k_p + K\, k_v\, s)} \tag{2.1.27}$$

2.2 PIV Controller

Adding an integral control can help eliminate any steady-state error. We will add an integral signal (middle branch in Figure 2.6) to have a proportional-integral-velocity (PIV) algorithm to control the position of the SRV02. The motor voltage will be generated by the PIV according to:

$$V_m(t) = k_p\ (\theta_d(d) - \theta_l(t)) + k_i \int (\theta_d(t) - \theta_l(t))\, dt - k_v\ \left(\frac{d}{dt}\,\theta_l(t)\right) \tag{2.1.28}$$

where k_i is the integral gain. We need to find the closed-loop transfer function $\Theta_l(s)/\Theta_d(s)$ for the closed-loop position control of the SRV02. Taking the Laplace transform of equation 2.1.28 gives

$$V_m(s) = \left(k_p + \frac{k_i}{s}\right)\ (\Theta_d(s) - \Theta_l(s)) - k_v\, s\, \Theta_l(s) \tag{2.1.29}$$

From the Plant block in Figure 2.6 and the open-loop voltage-to-position transfer function, we can write

$$\frac{\Theta_l(s)}{V_m(s)} = \frac{K}{(\tau\, s + 1)\ s} \tag{2.1.30}$$

Substituting equation 2.1.29 into 2.1.30 and solving for $\Theta_l(s)/\Theta_d(s)$ gives the SRV02 position closed-loop transfer function as:

$$\frac{\Theta_l(s)}{\Theta_d(s)} = \frac{K\ (k_p\, s + k_i)}{s^3\, \tau + (1 + K\, k_v)\, s^2 + K\, k_p\, s + K\, k_i} \tag{2.1.31}$$

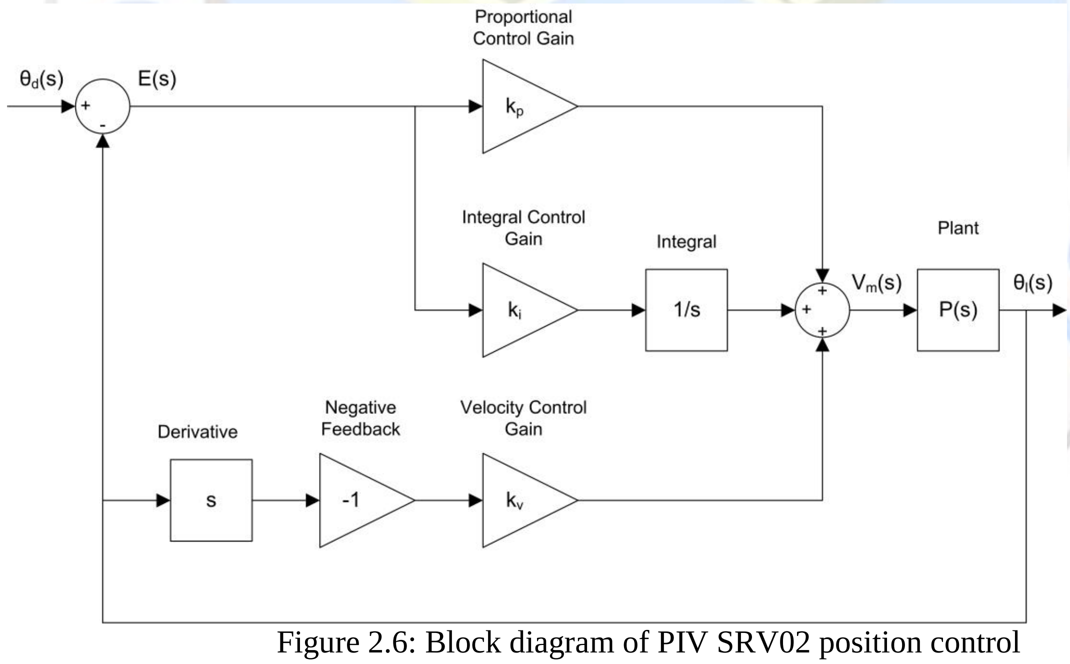

Figure 2.6: Block diagram of PIV SRV02 position control

2.2.1 Ramp Steady-State Error using PIV Controller

To find the steady-state error of the SRV02 for a ramp input under the control of the PIV substitute the closed-loop transfer function from equation 2.1.31 into equation 2.1.11

$$E(s) = \frac{\Theta_d(s)\, s^2\ (\tau\, s + 1 + K\, k_v)}{s^3\, \tau + s^2 + K\, k_p\, s + K\, k_i + K\, k_v\, s^2} \tag{2.1.32}$$

Then, substituting the reference ramp transfer function 2.1.25 into the $\Theta_d(s)$ variable gives

$$E(s) = \frac{R_0\ (\tau\, s + 1 + K\, k_v)}{s^3\, \tau + s^2 + K\, k_p\, s + K\, k_i + K\, k_v\, s^2} \tag{2.1.33}$$

2.2.2 Integral Gain Design

It takes a certain amount of time for the output response to track the ramp reference with zero steady-state error. This is called the *settling time* and it is determined by the value used for the integral gain.

In steady-state, the ramp response error is constant. Therefore, to design an integral gain the velocity compensation (the V signal) can be neglected. Thus, we have a PI controller left as:

$$V_m(t) = k_p\,(\theta_d(t) - \theta_l(t)) + k_i \int (\theta_d(t) - \theta_l(t))\,dt \qquad (2.1.34)$$

When in steady-state, the expression can be simplified to

$$V_m(t) = k_p\,e_{ss} + k_i \int_0^{t_i} e_{ss}\,dt \qquad (2.1.35)$$

where the variable t_i is the integration time.

Questions

1. The SRV02 closed-loop transfer function was derived in equation 2.1.23. Find the control gains k_p and k_v in terms of ω_n and ζ. **Hint:** Remember the standard second order system equation.

2. Calculate the minimum damping ratio and natural frequency required to meet the specifications given in 2.1.17-2.1.19.

$$PO = 100\,e^{\left(-\frac{\pi\,\zeta}{\sqrt{1-\zeta^2}}\right)}$$

$$t_p = \frac{\pi}{\omega_n\,\sqrt{1-\zeta^2}}$$

3. Based on the nominal SRV02 model parameters, K and τ, calculate the control gains needed to satisfy the time-domain response requirements given in 2.1.17-2.1.19.

4. For the PV controlled closed-loop system, find the steady-state error and evaluate it numerically given a ramp with a slope of $R_0 = 3.36$ rad/s. Use the control gains found before.

5. What should be the integral gain k_i so that when the SRV02 is supplied with the maximum voltage of $V_{max} = 10V$ it can eliminate the steady-state error calculated above in 1 second? **Hint:** Start from equation 2.1.35 and use $t_i = 1$, $V_m(t) = 10$, the k_p and e_{ss} you found. Remember that e_{ss} is constant.

2.3.1 Implementing Step Response using PV Controller

In this experiment, we will control the angular position of the SRV02 load shaft, i.e. the disc load, using the PV controller. Measurements will then be taken to ensure that the specifications are satisfied.

Experimental Setup
The *q_srv02_pos* Simulink diagram shown in Figure 2.12 is used to implement the position control experiments. The *SRV02-ET* subsystem contains QUARC blocks that interface with the DC motor and sensors of the SRV02 system. The *PIV Control* subsystem implements the PIV controller, except a high-pass filter is used to obtain the velocity signal (as opposed to taking the direct derivative).

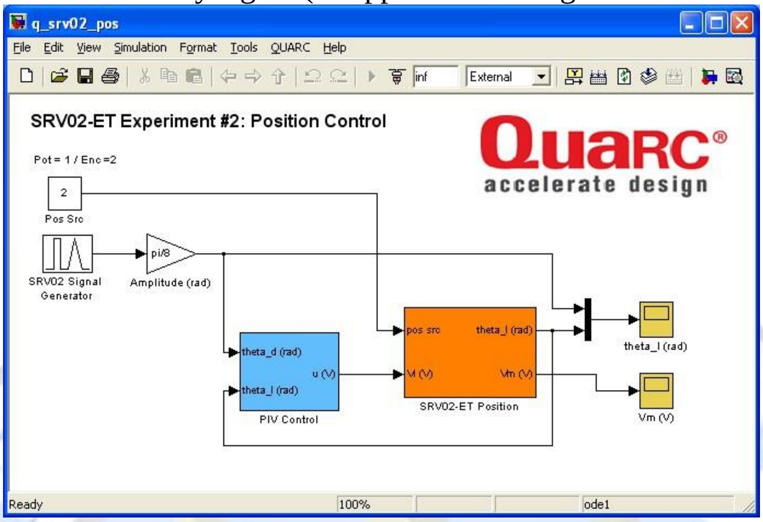

Figure 2.12: Simulink model used with QUARC to run the PV and PIV position controllers on the SRV02.

1. Run the setup_srv02_exp02_pos.m script with CONTROL_TYPE = 'AUTO_PV'.
2. Enter the proportional and velocity control gains.
3. To generate a step reference, ensure the SRV02 Signal Generator is set to the following:
   - Signal type = square
   - Amplitude = 1
   - Frequency = 0.4 Hz
4. Set the *Amplitude (rad)* gain block to $\pi/8$ to generate a step with an amplitude of 45 degrees.
5. Open the load shaft position scope, *theta_l (rad)*, and the motor input voltage scope, *Vm (V)*. Note that in the theta_l (rad) scope, the yellow trace is the setpoint position while the purple trace is the measured position.
6. Click on QUARC | Build to compile the Simulink diagram.
7. Select QUARC | Start to begin running the controller.
8. When a suitable response is obtained, click on the Stop button in the Simulink diagram toolbar (or select QUARC | Stop from the menu) to stop running the code. Generate a Matlab figure showing the PV position response and its input voltage similar to Figure 2.15.

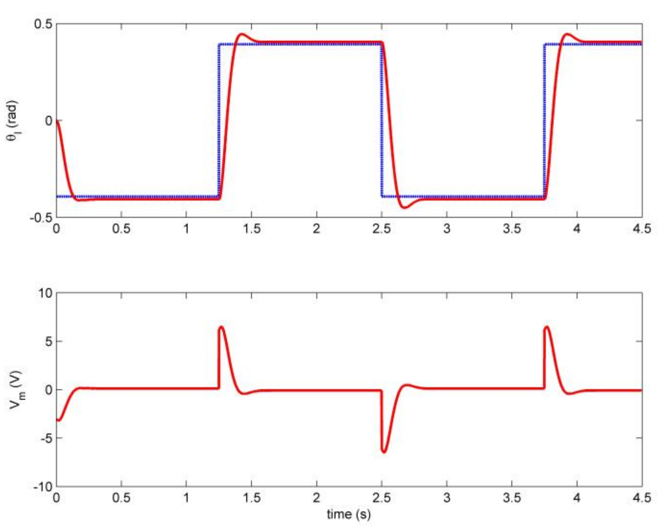

Figure 2.15: Measured SRV02 step response using PV.

9. Measure the steady-state error, the percent overshoot, and the peak time of the SRV02 load gear. Does the response satisfy the specifications given in 2.1.17-2.1.19?

2.3.2 Implementing Ramp Response Using PV

In this experiment, we will control the angular position of the SRV02 load shaft, i.e. the disc load, using a PV controller. The goal is to examine how well the system can track a triangular (ramp) position input. Measurements will then be taken to ensure that the specifications are satisfied.

As in the Step Response experiment, in this experiment you also need to use the *q_srv02_pos* Simulink diagram shown in Figure 2.12 to implement the position control experiments.

1. Run the setup_srv02_exp02_pos.m script with CONTROL_TYPE = 'AUTO_PV'.
2. Enter the proportional and velocity control gains.
3. Set the *SRV02 Signal Generator* parameters to the following to generate a triangular reference (i.e., ramp reference):
   - Signal Type = triangle
   - Amplitude = 1
   - Frequency = 0.8 Hz
4. In the Simulink diagram, set the *Amplitude (rad)* gain block to $\pi/3$.
5. Generate a Matlab figure showing the *Ramp PV* position response and its corresponding input voltage trace similar to Figure 2.21.

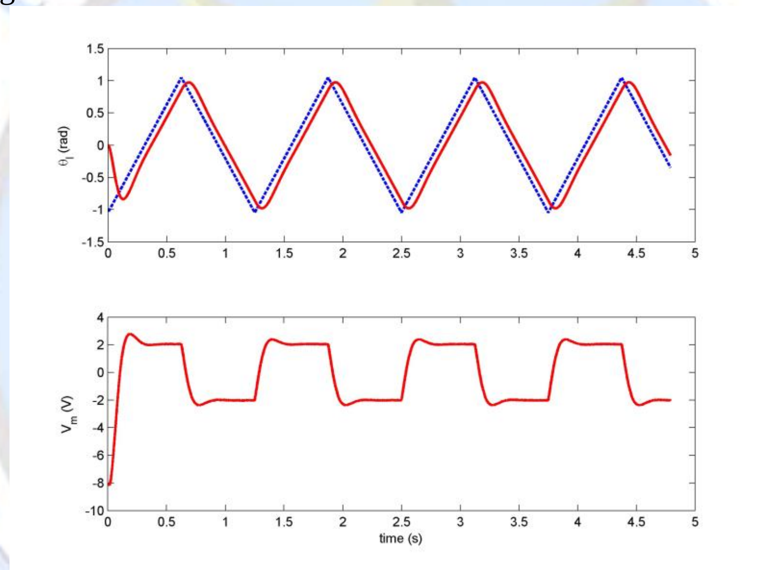

Figure 2.21: Measured SRV02 ramp response using PV.

9. Measure the steady-state error and compare it with the steady-state error calculated before.

2.3.3 Ramp Response with No Steady-State Error

Design an experiment to see if the steady-state error can be eliminated when tracking a ramp input. In this experiment, we will control the angular position of the SRV02 load shaft, i.e. the disc load, using a PIV controller. The goal is to examine how well the system can track a triangular (ramp) position input.

1. Run the setup_srv02_exp02_pos.m script with CONTROL_TYPE = 'AUTO_PIV'.
2. Enter the proportional and velocity control gains.
3. Set the *SRV02 Signal Generator* parameters to the following to generate a triangular reference (i.e., ramp reference):
   Signal Type = triangle, Amplitude = 1, Frequency = 0.8 Hz
4. In the Simulink diagram, set the *Amplitude (rad)* gain block to $\pi/3$.

Questions

1. How can the PV controller be modified to eliminate the steady-state error in the ramp response? State your hypothesis and describe the anticipated cause-and-effect leading to the expected result.

2. List the independent and dependent variables of your proposed controller. Explain their relationship.

3. Your proposed control, like the PV compensator, are model-based controllers. This means that the control gains generated are based on mathematical representation of the system. Given this, list the assumptions you are making in this control design. State the reasons for your assumptions.

4. Generate a Matlab figure showing the position response of the system and its corresponding input voltage similar to Figure 2.27.

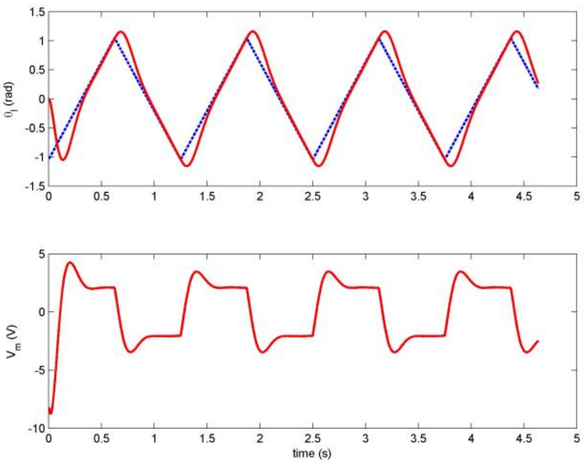

Figure 2.27: Measured SRV02 closed-loop ramp response using PIV.

5. Measure the steady-state error.

2.3.4 Results

Description	Symbol	Value	Unit
Pre-Lab: Ramp Steady-State Error Steady-state error using PV	e_{ss}		rad
Step Response Simulation Peak time Percent overshoot Steady-state error	t_p PO e_{ss}	0.20 5.0 0.00	s % rad
Step Response Implementation Peak time Percent overshoot Steady-state error	t_p PO e_{ss}		s % rad
Ramp Response Simulation with PV Steady-state error	e_{ss}	-0.213	rad
Ramp Response Implementation with PV Steady-state error	e_{ss}		rad
Ramp Response Simulation with no steady-state error Steady-state error	e_{ss}	-0.0125	rad
Ramp Response Implementation with no steady-state error Steady-state error	e_{ss}		rad

Table 2.1: Summary of results for the SRV02 Position Control laboratory.

3 SRV02 SPECIFICATIONS

Table 3.1 lists and characterizes the main parameters associated with the SRV02. Some of these are used in the mathematical model. More detailed information about the gears is given in Table 3.2.

Symbol	Description	Value	Variation
V_{nom}	Motor nominal input voltage	6.0 V	
R_m	Motor armature resistance	2.6 Ω	± 12%
L_m	Motor armature inductance	0.18 mH	
k_t	Motor current-torque constant	7.68×10^{-3} N-m/A	± 12%
k_m	Motor back-emf constant	7.68×10^{-3} V/(rad/s)	± 12%
K_g	High-gear total gear ratio	70	
η_m	Motor efficiency	0.69	± 5%
η_g	Gearbox efficiency	0.90	± 10%
$J_{m,rotor}$	Rotor moment of inertia	3.90×10^{-7} kg-m^2	± 10%
J_{tach}	Tachometer moment of inertia	7.06×10^{-8} kg-m^2	± 10%
J_{eq}	High-gear equivalent moment of inertia without external load	2.087×10^{-3} kg-m^2	
B_{eq}	High-gear Equivalent viscous damping coefficient	0.015 N-m/(rad/s)	
m_b	Mass of bar load	0.038 kg	
L_b	Length of bar load	0.1525 m	
m_d	Mass of disc load	0.04 kg	
r_d	Radius of disc load	0.05 m	
m_{max}	Maximum load mass	5 kg	
f_{max}	Maximum input voltage frequency	50 Hz	
I_{max}	Maximum input current	1 A	
ω_{max}	Maximum motor speed	628.3 rad/s	

Table 3.1: Main SRV02 Specifications

Symbol	Description	Value
K_{gi}	Internal gearbox ratio	14
$K_{ge,low}$	Internal gearbox ratio (low-gear)	1
$K_{ge,high}$	Internal gearbox ratio (high-gear)	5
m_{24}	Mass of 24-tooth gear	0.005 kg
m_{72}	Mass of 72-tooth gear	0.030 kg
m_{120}	Mass of 120-tooth gear	0.083 kg
r_{24}	Radius of 24-tooth gear	6.35×10^{-3} m
r_{72}	Radius of 72-tooth gear	0.019 m
r_{120}	Radius of 120-tooth gear	0.032 m

Table 3.2: SRV02 Gearhead Specifications