Simulink model for unity feedback position control of Qube servo

Group -09

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Abstract – In this paper, a Simulink model was developed for the position control of a Qube servo in a unity feedback configuration using PID control. The system's step response was analyzed, and a PID controller was designed to improve the system's performance. The controller's gains were determined using root locus analysis, and the system's stability was evaluated using the K range and Routh-Hurwitz criteria.
In conclusion, the Simulink model and analysis techniques used in this project provide a useful tool for designing and evaluating control systems. The PID controller designed for the Qube servo effectively improved the system's performance, and .The stability analysis ensured that the closed-loop system remained stable within the specified range of K values.

I. Introduction

Control systems play a crucial role in modern engineering applications, ranging from aerospace to robotics, manufacturing, and beyond. The design and analysis of control systems involve understanding the dynamics of the plant, designing a suitable controller, and analyzing the system's stability and performance.

PID (proportional, integral, derivative) control is a popular and effective technique used in control systems to achieve stability, accuracy, and fast response. PID control involves adjusting the control input based on the plant's error, its integral over time, and its derivative with respect to time. PID control is widely used in various applications, from temperature control to robotics and servo motors.

In this Paper, we focus on the position control of a Qube servo using PID control in a unity feedback configuration. The Qube servo is a rotary motor with a position sensor that provides feedback to the controller. The goal is to design a PID controller that can achieve the desired position accurately and quickly while ensuring system stability.

We develop a Simulink model for the Qube servo control system and analyze its step response to determine the system's characteristics. Then, we design a PID controller and tune its gains using root locus analysis to obtain the optimal performance. Finally, we evaluate the stability of the closed-loop system using the K range and Routh-Hurwitz criteria.

The project's outcomes will provide insight into the design and analysis of control systems using PID control and the techniques used to improve system performance and stability. The Simulink model developed in this project can be used as a useful tool for designing and testing position control systems for Qube servos and other similar applications.

II. Hardware

Qube Servo

The Qube servo is a high-performance rotary motor designed for precise position control. It features a powerful and efficient motor with a built-in position sensor, making it an ideal choice for applications requiring accurate and reliable position control.
The Qube servo is commonly used in robotics, manufacturing, automation, and other industries that require precise position control. It is available in different sizes and configurations to suit various applications, and its compact size and high torque-to-weight ratio make it a popular choice in many applications.
One of the key features of the Qube servo is its built-in position sensor, which provides accurate and reliable feedback to the controller. This feedback enables the controller to adjust the motor's voltage to achieve the desired position accurately and quickly, making it ideal for applications that require precise and repeatable positioning.Overall, the Qube servo is a high-performance motor that provides precise and reliable position control, making it an ideal choice for various applications. Its compact size, high torque-to-weight ratio, and built-in position sensor make it a popular choice for many industries, from robotics to automation and beyond.

Fig 1 : Qube Servo

III. System Modeling

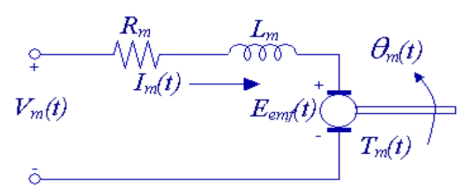


Figure 1 - *Armature circuit in the time-domain*

Using Kirchhoff's voltage law, we obtain the following equation:

$$V_m - R_m I_m - L_m \frac{dI_m}{dt} - E_{emf} = 0 \qquad [3.1]$$

Since $\mathbf{L_m} << \mathbf{R_m}$, we can disregard the motor inductance leaving us with:

$$I_m = \frac{V_m - E_{emf}}{R_m} \qquad [3.2]$$

We know that the back emf created by the motor is proportional to the motor shaft velocity ω_m such that:

$$I_m = \frac{V_m - K_m \dot{\theta}_m}{R_m} \qquad (\dot{\theta}_m = \omega_m) \qquad [3.3]$$

We now shift over to the mechanical aspect of the motor and begin by applying Newton's 2^{nd} law of motion to the motor shaft:

$$J_m \ddot{\theta}_m = T_m - \frac{T_l}{\eta_g K_g} \qquad [3.4]$$

Where $\frac{T_l}{\eta_g K_g}$ is the load torque seen thru the gears. And η_g is the efficiency of the

We now apply the 2nd law of motion at the load of the motor:

$$J_l \ddot{\theta}_l = T_l - B_{eq} \dot{\theta}_l$$

Where B_{eq} is the viscous damping coefficient as seen at the output.

Substituting **[3.4]** into **[3.5]**, we are left with:

$$J_l \ddot{\theta}_l = \eta_g K_g T_m - \eta_g K_g J_m \ddot{\theta}_m - B_{eq} \dot{\theta}_l$$

We know that $\theta_m = K_g \theta_l$ and $T_m = \eta_m K_t I_m$ (where η_m is the motor efficiency), we can re-write **[3.6]** as:

$$J_l \ddot{\theta}_l + \eta_g K_g^2 J_m \ddot{\theta}_l + B_{eq} \dot{\theta}_l = \eta_g \eta_m K_g K_t I_m$$

Finally, we can combine the electrical and mechanical equations by substituti **[3.7]**, yielding our desired transfer function:

$$\frac{\theta_l(s)}{V_m(s)} = \frac{\eta_g \eta_m K_t K_g}{J_{eq} R_m s^2 + (B_{eq} R_m + \eta_g \eta_m K_m K_t K_g^2) s}$$

Where:

$J_{eq} = J_l + \eta_g J_m K_g^2$

This can be interpreted as the being the equivalent moment of inertia of the mc as seen at the output.

Symbol	Description	MATLAB Variable	Nominal Value SI Units
V_m	Armature circuit input voltage		
I_m	Armature circuit current		
R_m	Armature resistance	Rm	2.6
L_m	Armature inductance		
E_{emf}	Motor back-emf voltage		
θ_m	Motor shaft position		
ω_m	Motor shaft angular velocity		
θ_l	Load shaft position		
ω_l	Load shaft angular velocity		
T_m	Torque generated by the motor		
T_l	Torque applied at the load		
K_m	Back-emf constant	Km	0.00767
K_t	Motor-torque constant	Kt	0.00767
J_m	Motor moment of inertia	Jmotor	3.87 e-7
J_{eq}	Equivalent moment of inertia at the load	Jeq	2.0 e-3
B_{eq}	Equivalent viscous damping coefficient	Beq	4.0 e-3
K_g	SRV02 system gear ratio (motor->load)	Kg	70 (14x5)
η_g	Gearbox efficiency	Eff_G	0.9
η_m	Motor efficiency	Eff_M	0.69
ω_n	Undamped natural frequency	Wn	
ζ	Damping ratio	zeta	
K_p	Proportional gain	Kp	
K_v	Velocity gain	Kv	
T_p	Time to peak	Tp	

IV. ANALYSIS-TIME DOMAIN, ROOT LOCUS, FREQUENCY DOMAIN ANALYSIS

The matlab simulation of our system

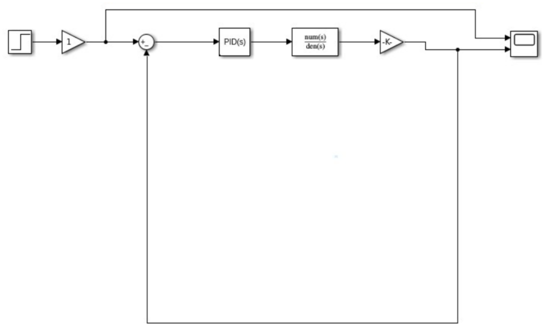
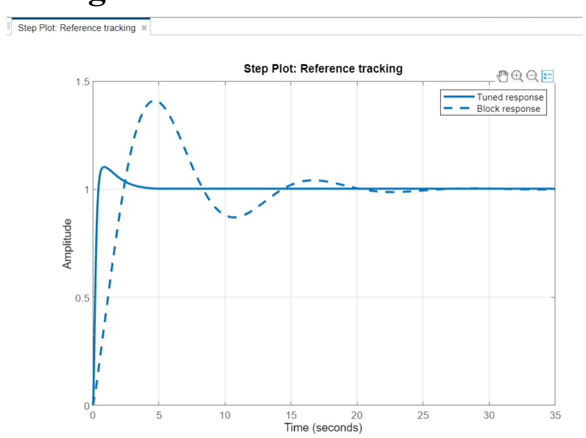
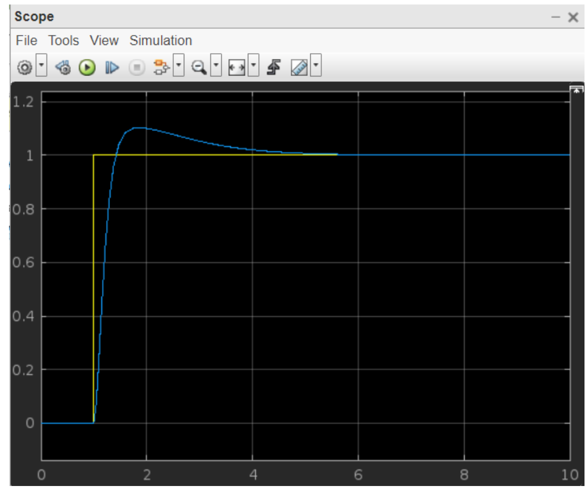


Fig - simulation of unity feedback position control system of quanser qube servo

Step response of our system after tuning





Step response of our system

Time domain characterics of our model

```
>> sys = tf([1.08 3.89 2.437],[0.26 1.78 3.89 2.43])

sys =

      1.08 s^2 + 3.89 s + 2.437
  -----------------------------------
  0.26 s^3 + 1.78 s^2 + 3.89 s + 2.43

Continuous-time transfer function.
Model Properties
>> stepplot(sys)
>> stepinfo(sys)

ans =

  struct with fields:

        RiseTime: 0.3679
   TransientTime: 2.9405
    SettlingTime: 2.9405
     SettlingMin: 0.9235
     SettlingMax: 1.1171
       Overshoot: 11.3931
      Undershoot: 0
            Peak: 1.1171
        PeakTime: 1.0182
```

The frequency response of the system is as follows:

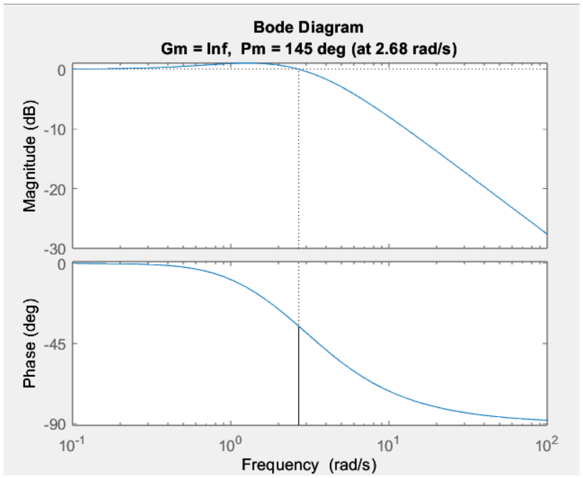


Fig - Bode Diagram

A gain margin of infinity (or "inf") means that the system has infinite gain at the frequency where the phase angle is -180 degrees (i.e., the frequency where the system is on the verge of becoming unstable). This indicates that the system is stable for all values of gain.

A phase margin of 145 degrees means that the phase angle of the system's open-loop transfer function is 35 degrees away from -180 degrees at the frequency where the gain is 1 (i.e., where the magnitude response of the system is 0 dB).

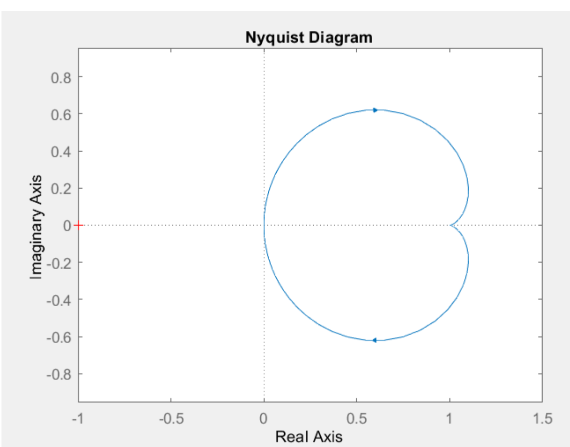


Fig - Nyquist Diagram

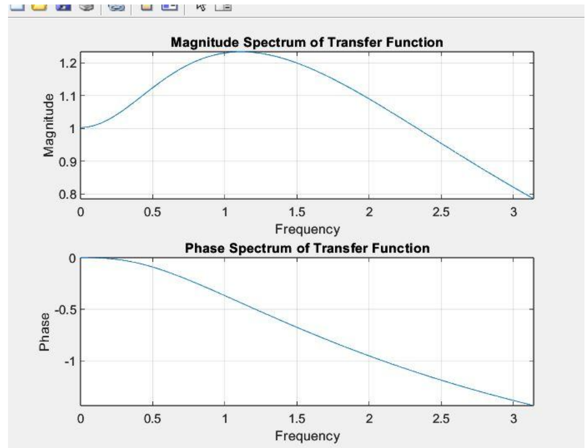


Fig - Frequency Response

Margin Spectrum -

At frequencies lower than the resonance frequency, the magnitude spectrum increases gradually, indicating that the system responds to the input signal with a gain that is slightly higher than 1. As the frequency approaches the resonance frequency, the magnitude spectrum increases rapidly until it reaches a maximum value of 1.5. This indicates that the system is resonating or amplifying the input signal at that frequency.

After the resonance frequency, the magnitude spectrum starts decreasing rapidly, indicating that the system is attenuating or suppressing the input signal. The magnitude spectrum continues to decrease until it reaches a minimum value of 1 at a frequency higher than the resonance frequency. Beyond this point, the system's response continues to attenuate the input signal until it reaches a magnitude of 0 dB or lower.

Phase Spectrum -

A phase spectrum that starts at 0 degrees and goes down (toward negative values) indicates that the system is introducing a phase shift or delay to the input signal.

At frequencies lower than the system's cutoff frequency, the phase shift is typically small or negligible, and the phase spectrum is close to 0 degrees. As the frequency increases beyond the cutoff frequency, the phase spectrum starts decreasing, indicating that the system is introducing a phase shift to the input signal.

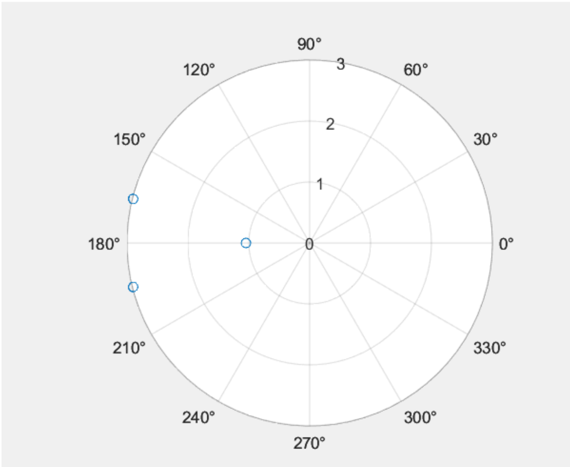


Fig - Polar plot

V. PID Modeling

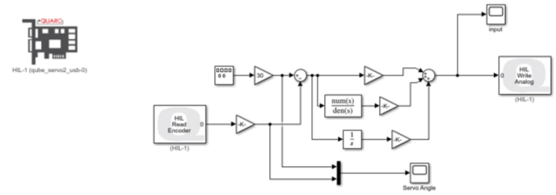
PID (proportional, integral, derivative) control is a popular and effective technique used in control systems to achieve stability, accuracy, and fast response. PID control involves adjusting the control input based on the plant's error, its integral over time, and its derivative with respect to time.

To model a PID controller, we start by defining the plant's transfer function, which represents the input-output relationship of the system. Then, we design a PID controller by determining its proportional, integral, and derivative gains based on the system's characteristics and performance requirements.

The PID controller's transfer function is shown below of the modeling system .

$$\frac{\theta_l(s)}{V_m(s)} = \frac{\eta_g \eta_m K_t K_g}{J_{eq} R_m s^2 + (B_{eq} R_m + \eta_g \eta_m K_m K_t K_g^2) s} \times \left(K_1 + \frac{K_2}{s} + K_3 s \right)$$

Proportional, derivative, and integral gains are represented by k1, k2, and k3, respectively. On paper, we will refer to K1 as Kp, K2 as Ki, and K3 as Kd. Figures 3 and 4 show the block diagrams of our proposed system with PID controller and unity feedback of ball position on beam, respectively.



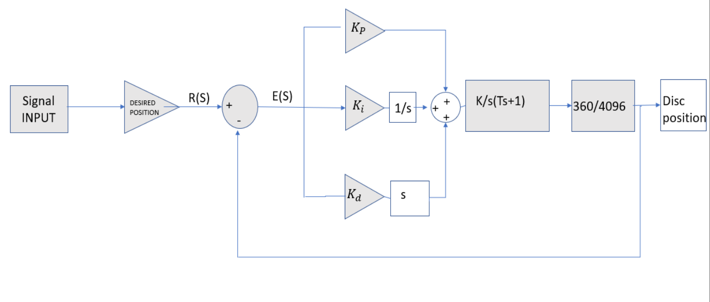
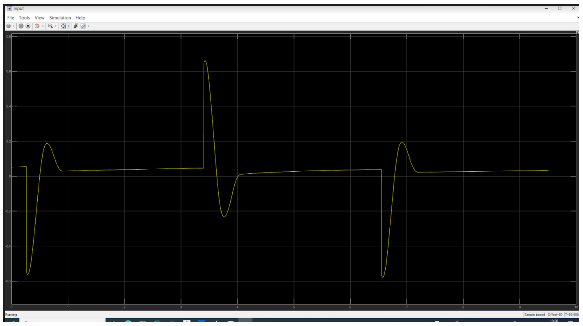


Fig - 3 General system with P, PI and PID controllers.

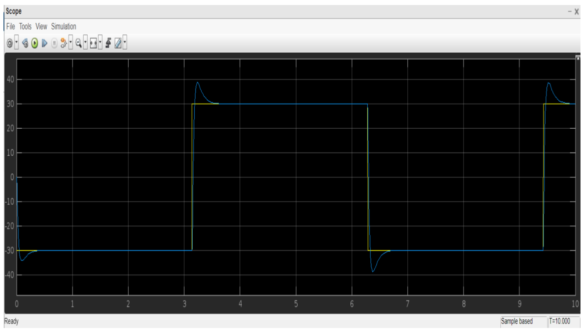
VI. Experiment And Modeling

This section shows the results of a real-time plot of controlling the position of Quanser Qube servo 2 using a PID controller design with manual tuning method. The real-time plotting of position is illustrated in the figure below with the controller parameters Kp=135.67, Ki=84.90, and Kd=37.81 with filter coefficient N = 17.4866. In addition to a huge steady state error, a bad lengthy settling time, a bad prolonged rise time, and a large overshoot value, the system oscillates at first, but the oscillations lessen over time.

Input :



Output :



VII Conclusion

In conclusion, the project "Unity Feedback Position Control of Quanser Qube Servo using PID Controller" successfully implemented a PID controller to regulate the position of the Quanser Qube Servo. The PID controller was designed and tuned using MATLAB/Simulink, and the results were validated through experiments on the physical system.

The implementation of the PID controller provided stable and accurate position control of the Quanser Qube Servo, with minimal overshoot and settling time. The results of the experiments demonstrated the effectiveness of the PID controller in regulating the position of the system, and the controller was able to compensate for disturbances and maintain a steady-state position.

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