Gantry Crane Project Report

Tarek Al-Bakri
albakritarek@mines.edu
Electrical Engineering Senior

Brian Durland
bdurland@mines.edu
Mechanical Engineering Senior

Eddie Silva
easilva@mines.edu
Electrical Engineering Senior

Abstract—This project aims to create a control system for a simulated crane gantry. Attention is focused on the controller, which ensures that the mass attached to the gantry oscillates less than the simulated model. This was done by creating both non-linear and linear models for the subsystem, and subsequently developing a discrete-time control system for the operation of the gantry. The results of this project were a theoretical controller that closely matched the given simulation, confirming the accuracy of the discrete-time control system. Validated on the discrete-time controller, $\%OS = 0$, $t_{settling} = 120$ **seconds**, $\dot{z}_{max}(t) \leq 1$ **m/s.** Frequency analysis for this system reported extremely high phase and gain margins, indicating robust controller stability.

I. Introduction

When a gantry crane is used by an operator, as depicted in the problem statement[1][2], they can only control the direction and speed of motion, and position control relies on the skill of the operator. The controller developed in this project will instead move the payload on the crane to a desired position by internally adjusting the speed and direction of the gantry motor system. This controller is widely applicable to automation in manufacturing, as it eliminates the need for operators and is highly repeatable. This controller also has more niche applications, such as gravity reduction simulators. Given the large masses that these gantries often have to move, safe handling is of the utmost importance. For these heavyweight applications, time can be sacrificed to prevent overshoot, which could lead to potential crashes. Thus, the proposed controller in this project will focus on these critical safety applications, which define a longer rise and settling time to ensure absolute precision in meeting the reference signal.

One key result obtainable from the developed controller is the analysis of the order required to meet the performance criteria. To increase system efficiency, creating a circuit to mirror the system's behavior is generally preferred over software implementations. However, this may be unwieldy for higher-order systems, and some computational time is sacrificed for system simplicity.

The rest of the paper will follow the following structure: Section II Modeling, Section III Design, Section IV Analysis. The Modeling section discusses the methodology used to find ℓ, the non-linear and linear models, their relevant equations, an introduction to the controller design, and the step response of the controller. The Design section elaborates on the constraints for this controller, the design choices made, and discusses why each decision is vital to consider. Finally, the Analysis section discusses the results of the controller, performance measurements, and potential improvements.

The team aimed to learn how to create a realistic, safe, and stable controller for a crane gantry carrying a load of some mass. This was accomplished by defining and adhering to design constraints and developing a suitable controller. The proposed controller involved some trade-offs between time to move and the speed at which it is being moved. Still, the safety of the potential user is paramount over the theoretical performance of the system.

II. Modeling

To model the payload dynamics with the crane gantry system, the team constructed an open-loop non-linear plant with hoist velocity $u(t) = \dot{z}(t)$ and a gantry output position $y(t)$ using integrated mechanics of $\phi(t)$ to model the payload, at a distance ℓ, from the crane. Once this non-linear system was validated, it was linearized using small-angle approximations.

The methodology used to find ℓ, the distance between the gantry and the payload, was to evaluate the amplitude of the steady state response for the given experimental data. The team iteratively tested a range of lengths, comparing them to the steady state amplitude, which was extracted using the MATLAB `max` function, and refining the range tested. The final length, ℓ chosen, which visually matched the given data, was found to be 3.7 m, which was then validated against the simulation in figure 1[2].

To implement the nonlinear model, the given angular velocity function, equation (1), was created, where the angular position, $\phi(t)$, is defined in equation (2). These equations were then built up block-wise in a Simulink model.

$$\dot{\phi}(t) = \frac{1}{\ell}\cos\left(\phi(\tau)\right)\dot{z}(\tau) + \int_0^t \left(\frac{1}{\ell}\sin\left(\phi(s)\right)\dot{\phi}(s)\dot{z}(s) - \frac{g}{\ell}\sin\left(\phi(s)\right)\right)ds \tag{1}$$

$$\phi(t) = \int_0^t \dot{\phi}(\tau)d\tau \tag{2}$$

This set of equations required the plant model to be closed-loop, which is not inherently problematic, but is not the final solution. During the validation and calibration of the model to the given data, the model pulled in the crane velocity from the MATLAB workspace and sent the position back to be easily graphed. However, in developing the controller, these inputs were made more generic, and any workspace integration went through the control block. In addition to aiding in controlling development, the general inputs into the model subsystem allowed the team to test the step response using built-in Simulink blocks.

To then linearize the model, the team used the small-angle approximations $\sin(x) = x$ and $\cos(x) = 1$, replacing all the corresponding blocks in the model. In addition to simplifying the model by removing the transcendental components, it also reduced the number of closed feedback paths in the plant by eliminating the dependency on $\cos(\phi(\tau))$. The linearized transfer function from Simulink was equation (4). This agrees with the theoretical plant model in equation (3). Equation (4) has an integrator, an implicit zero at $s = \infty$, and two repeated poles at $s = \pm j\sqrt{2.6771}$. In discrete time, with a zero-order hold and sampling time of 0.02 seconds, or 50 Hz[3], the linearized transfer function, equation (4), becomes equation (5).

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[2]Exact value used was $\ell = 3.6644$ m where $g = 9.81$ m/s
[3]as described in section III

$$G(s) = \frac{g/\ell}{s(s^2 + g/\ell)} \quad (3)$$

$$G(s) = \frac{2.6771}{s^3 + 2.6771s} \quad (4)$$

$$G(z) = \frac{3.569 \times 10^{-6} z^2 + 1.428 \times 10^{-5} z + 3.569 \times 10^{-6}}{z^3 - 2.999 z^2 + 2.999 z - 1} \quad (5)$$

Equation (5), the plant equation used in the model, has zeros at $z_{1,2} = \{-3.732, -0.268\}$, a real pole at $z = 1$, and a pair of complex poles nearly on top of the real pole, barely outside the unit circle. This makes sense, as the given data shows only marginal stability in the steady state response.

As discussed previously, the step response, pictured in figure 2, was tested by converting the linear and nonlinear plants into Simulink subsystems and then passing the built-in Simulink step function. There was no concern regarding sampling, as all of the blocks operated in continuous time. Recall, in the Laplace domain, $u(t) = 1/s$, which removes the integrator from the linearized transfer function, and the nonlinear model will behave similarly. Although it lacks an algebraic transfer function, the accuracy of the small-angle approximation validates this.
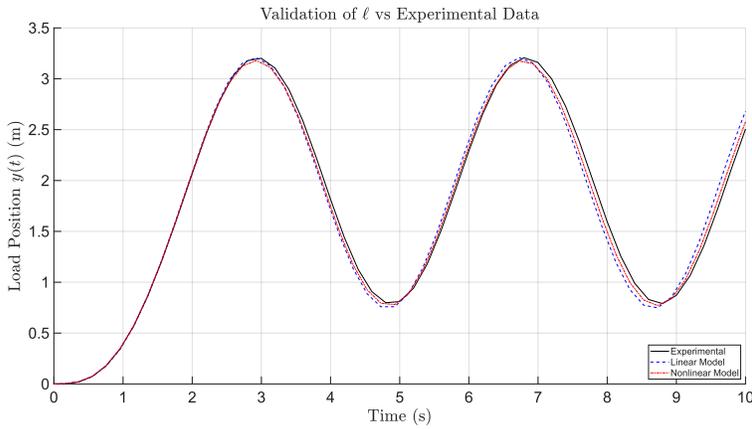

Fig. 1: Validation of ℓ value against experimental data, using both the non-linear approximation, and the linear calculation
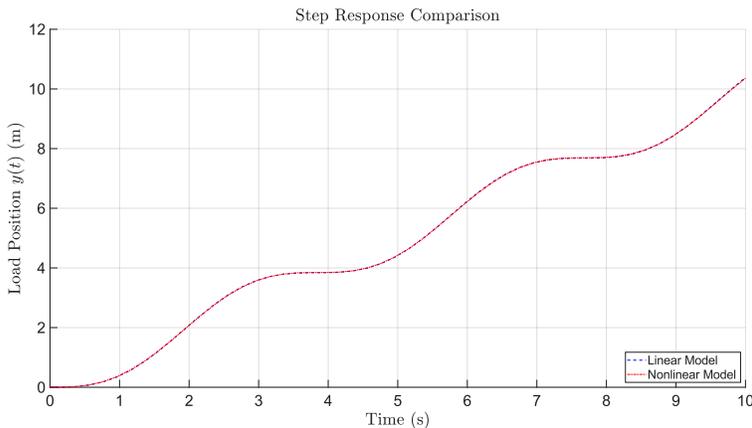

Fig. 2: Step Response for linear and non-linear models

III. DESIGN

For the design of the team's controller, option B was chosen from the project description: "First, discretize your linearized plant, and then use discrete-time techniques to design your discrete-time controller directly in the z-domain."[2]

The design of the controller was a multi-step process. First, no controller should be developed without a plant, so the subsystem plant model was implemented into the top-level model. SISOtool was used to create a transfer function, equation (7), that would be in series with the plant controller. The purpose of the transfer function is to draw the system to the desired response in accordance with the chosen performance criteria, which are described below. The design philosophy was based on the assumption that the gantry would hold a significant mass and have an equally large inertia. Thus, the gantry should ideally move slowly and with high levels of control to ensure the mass does not move unpredictably. With this design approach, it is better to move slowly and steadily to ensure no damage occurs.

The following characteristics of the controller were implemented:

- $\%OS = 0$
  Closed loop step response overshoot of 0%. The purpose of this is to prevent the gantry from overshooting, which would cause the mass to move forward and hit other objects. In order to prevent the system from ever crossing the vertical bound set by the reference signal, $\%OS = 0$.

- $t_{settling} = 120$ seconds
  Closed loop settling time, t_s, of ≤ 2 minutes. The reasoning behind this is that a worker would be moving the item over any given arbitrary distance and should move slowly but efficiently. In conclusion, taking 2 minutes to move safely is preferable to moving much quicker, as it can lead to overshooting, long settling times relative to rise time, or possible danger to operators.

- $\dot{z}_{max}(t) \leq 1$ m/s
  Hoist velocity, $\dot{z}$, which would be < 1 m/s given a reference distance of 1 m. Ensuring the safety of potential workers using this crane gantry remains the main priority. Not only was this a given parameter [2], but it also ensures that the crane gantry operates smoothly.

- Phase margin and Gain margin as large as possible.
  These parameters are desired as the margins measure how close the loop is to oscillation (instability) if parameters, dynamics, or delays change.

$$PM = 48°, \text{ from MATLAB} \quad (6)$$

A sampling rate of 0.02 seconds was chosen to ensure that the crane gantry, regardless of its speed, remains slow-moving in comparison. The objective of $t_s < 120$ seconds benefits from thousands of control updates across the transient, discerning a negligible discretization error. This also ensured the avoidance of excessive sampling due to hardware limitations in real life. The goal was to choose a realistic rate at which the controller still operated functionally well.

Following these parameters and the goal for the transfer function outlined previously, equation (7) was found. This function has a pair of zeros at $\{0.989488, 1.00951\}$, and then three poles at $\{0.8225, 0.6689, 0.9273\}$, which can be observed in the equation.

$$C(z) = 0.51077 \frac{z^2 - 1.999z + .9989}{(z - 0.8225)(z - 0.6689)(z - 0.9273)} \quad (7)$$

Then, placing a near-symmetric zero pair around $z = 1$ to shape the loop where low-frequency tracking and phase are most critical, especially for a heavy payload on the gantry. This enabled a large measured phase margin of $48°$, equation (6), whilst also avoiding

overshoot. Controller poles, equation (7), were chosen to align well inside the unit circle, ensuring the closed-loop dynamics stayed predictable. This also helped avoid high-frequency amplification, which aligns with the goal of $\dot{z}(t)$'s maximum speed.

The poles and zeros are set to prioritize the phase and smoothness of movement over speed, allowing the 50 Hz sampling rate to easily replicate the slow physics of the gantry.

Given these parameters, the resultant controller focused on stability, safety, and predictability. Using the transfer function within the feedback loop outlined in figure 8, a controller was developed that adhered to the above goals.

IV. ANALYSIS

Upon review of the data, the proposed controller satisfies the main objectives and is within the safety guidelines. The standard hoist velocity remains within the 1 m/s bound, given a reference $r = 1$ m. The load swing angle is not greater than $1°$, and for the crane hoist length of ≈ 3.7 m, it is roughly 3 inches of swing, and the loop shows large gain and phase margins.

Figure 3 shows that the system takes just under 2 minutes to slowly settle and reach the 1 meter reference, a consequence of the controller's design. This is a deliberate and justified tradeoff for a heavy suspended load. Slower motion not only reduces the chance of overshoot, tipping, and collision, but also minimizes peak accelerations that can make the system very dangerous if used in the real world.

Figure 4 illustrates the closed-loop behavior of the crane when commanded to move a fixed distance. This response is smooth and behaves as expected, without oscillatory instability. This confirms the controller successfully maintains system stability while preventing the load from swinging beyond the desired range. This ultimately shows the safety of the proposed system and verifies that the controller gains were tuned precisely, showing a stable system.

Figure 5 represents the commanded hoist velocity over time. As shown, it remains constant within the 1 m/s bound throughout the simulation. This confirms that the system respects the safety limits and that the controller maintains a consistent rate of motion without excessive acceleration or abrupt changes.

Figure 6a and figure 6b display the margins of stability in the system through a Bode plot. They reflect a mechanical system with integral control action, showing an initial phase lag of around -450 degrees, and its massive resonance peak spikes to about $+200$ dB. The decrease in magnitude at higher frequencies is characteristic of the system's inertia. The mass of the gantry, along with the electrical time constants of the motor and control filtering, acts as a low-pass filter, causing the system's gain to diminish rapidly as frequency increases. A positive Phase Margin is desired, with 45 to 60 degrees ideal for performance. This margin generally guarantees closed-loop stability, as calculated from the PM provided in the plot, equation (6), which falls within the desired range for performance criteria.

Analysis of the phase margin, given in equation (6), helps understand the delay at which the closed-loop system becomes unstable. MATLAB's margin tool was used to determine the open-loop phase margin of $48°$, or 0.83 rad at 1.62 rad/s. To find the crossover time that creates instability, the phase margin was divided by the crossover frequency, which gives 0.5 s. For the team's system frequency of 50 Hz, instability will occur after a 25-sample delay. To validate this result, a concentration delay of 25 samples was added to the controller before the transfer function. The instability is marginal, but with some simple scripting to determine the settling time, one finds that it does not exist for this delayed system.

The results of the system show that the identified nonlinear plant and its small-angle approximation are a safe, stable, and valid controller design. The discrete-time controller, implemented on the nonlinear model, tracks the 1 m reference, r, without overshoot, and settles within a reasonable window of time, whilst also respecting the 1 m/s hoist velocity limitation. These outcomes reflect the intended trade-off between speed and performance and the predictability, safety, and stable motion of the payload.
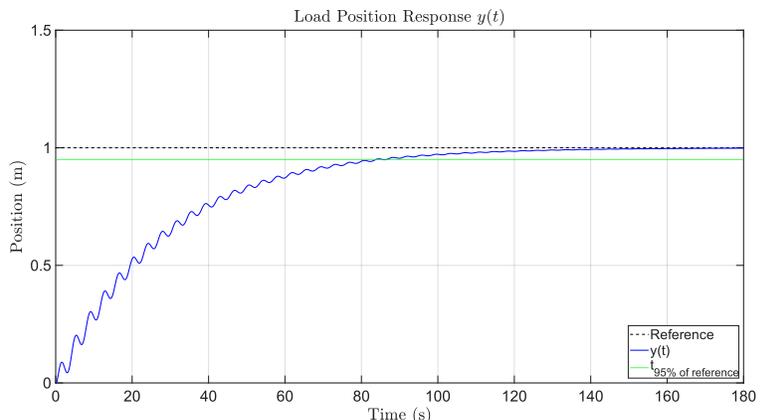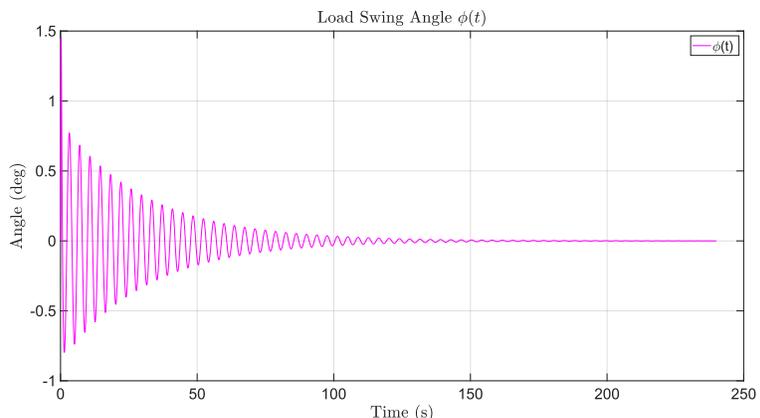

Fig. 3: Load position over Time

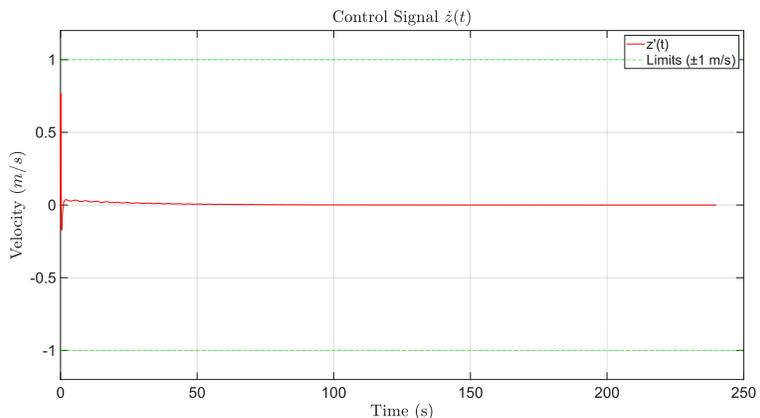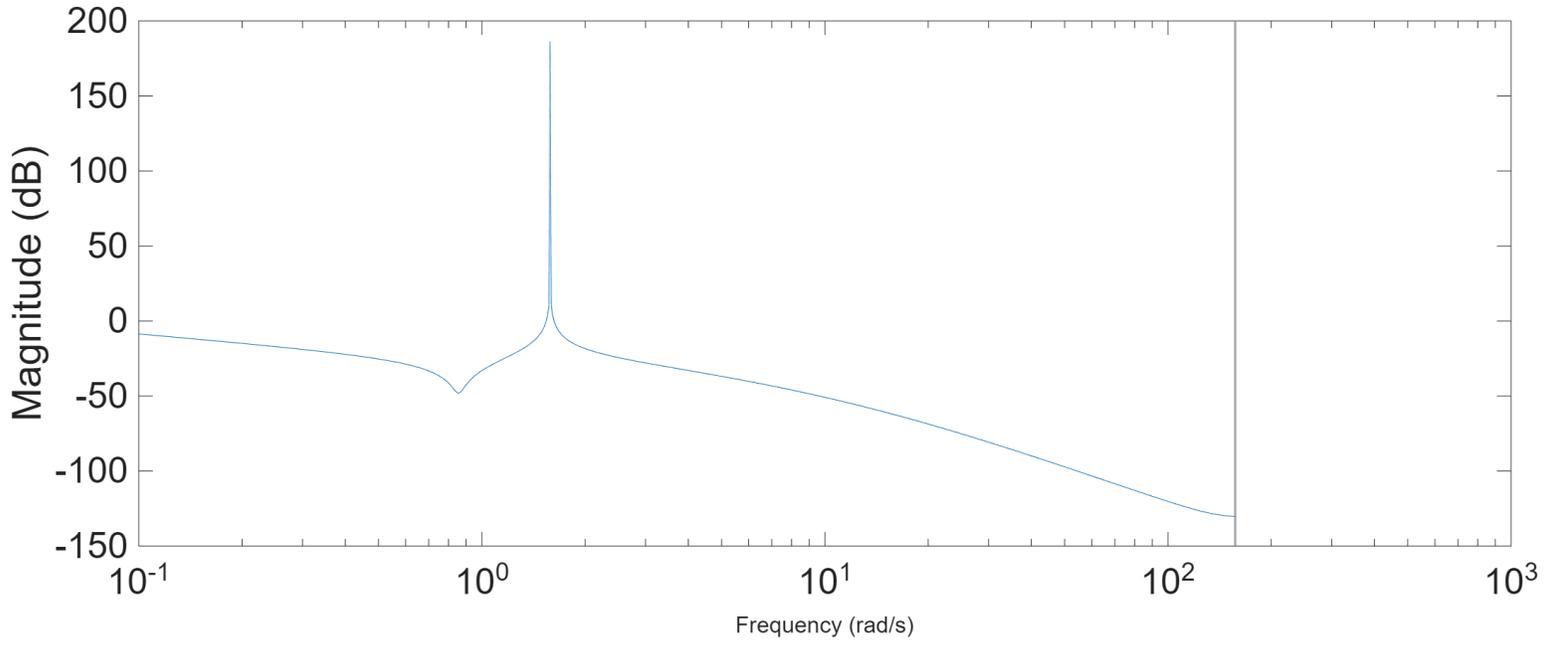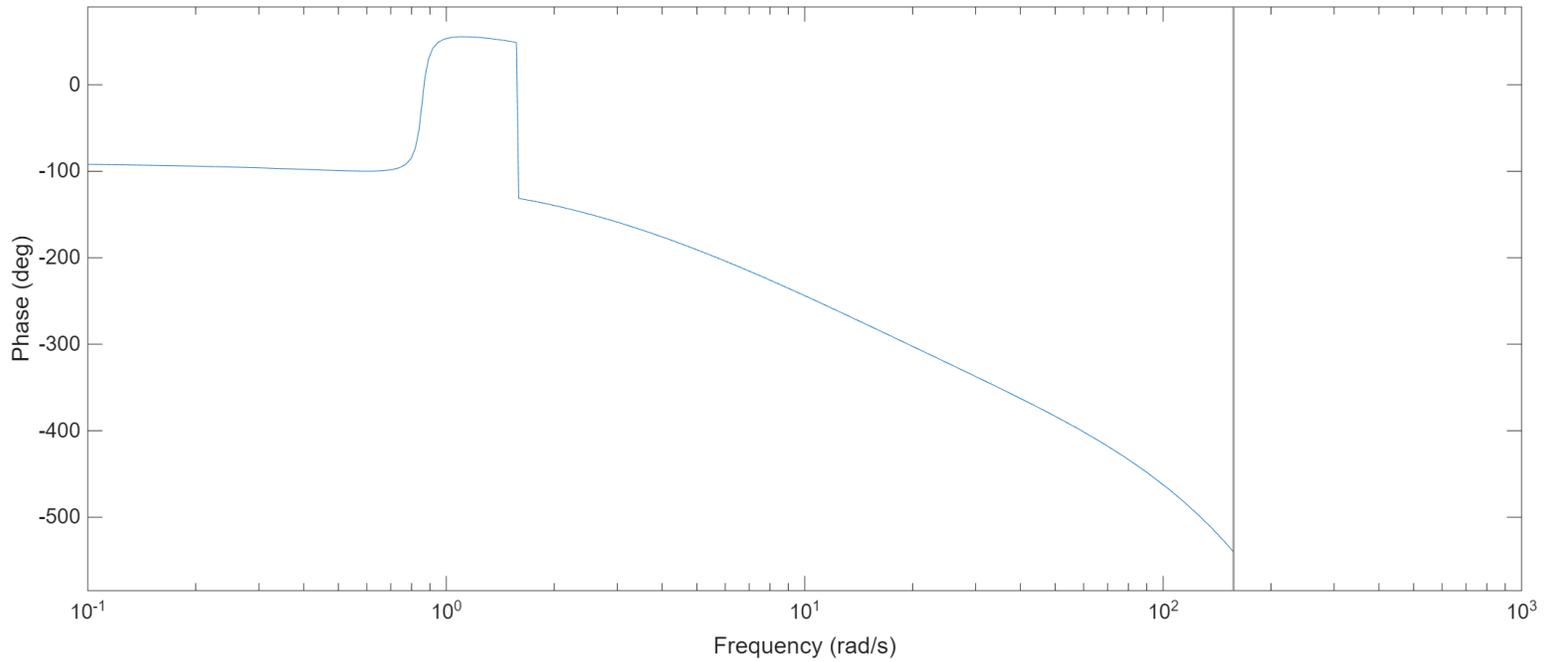Fig. 4: Load swing angle (in degrees) over time

Fig. 5: Load Velocity over Time

Bode Diagram

(a) Open-loop magnitude

Bode Diagram

(b) Open-loop phase

Fig. 6: Open-loop Bode plots for the designed controller.

Bode Diagram

(a) Closed-loop magnitude

Bode Diagram
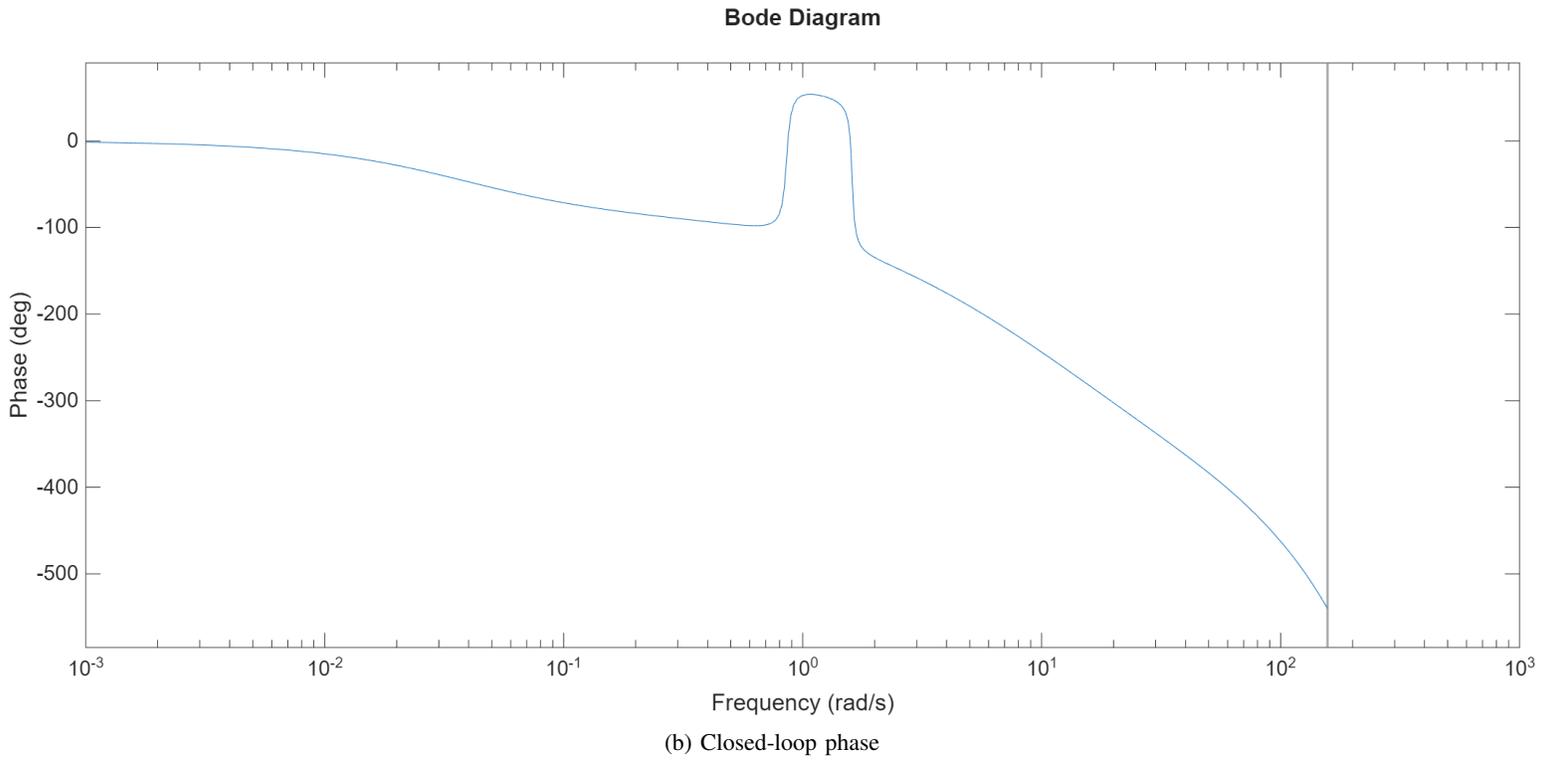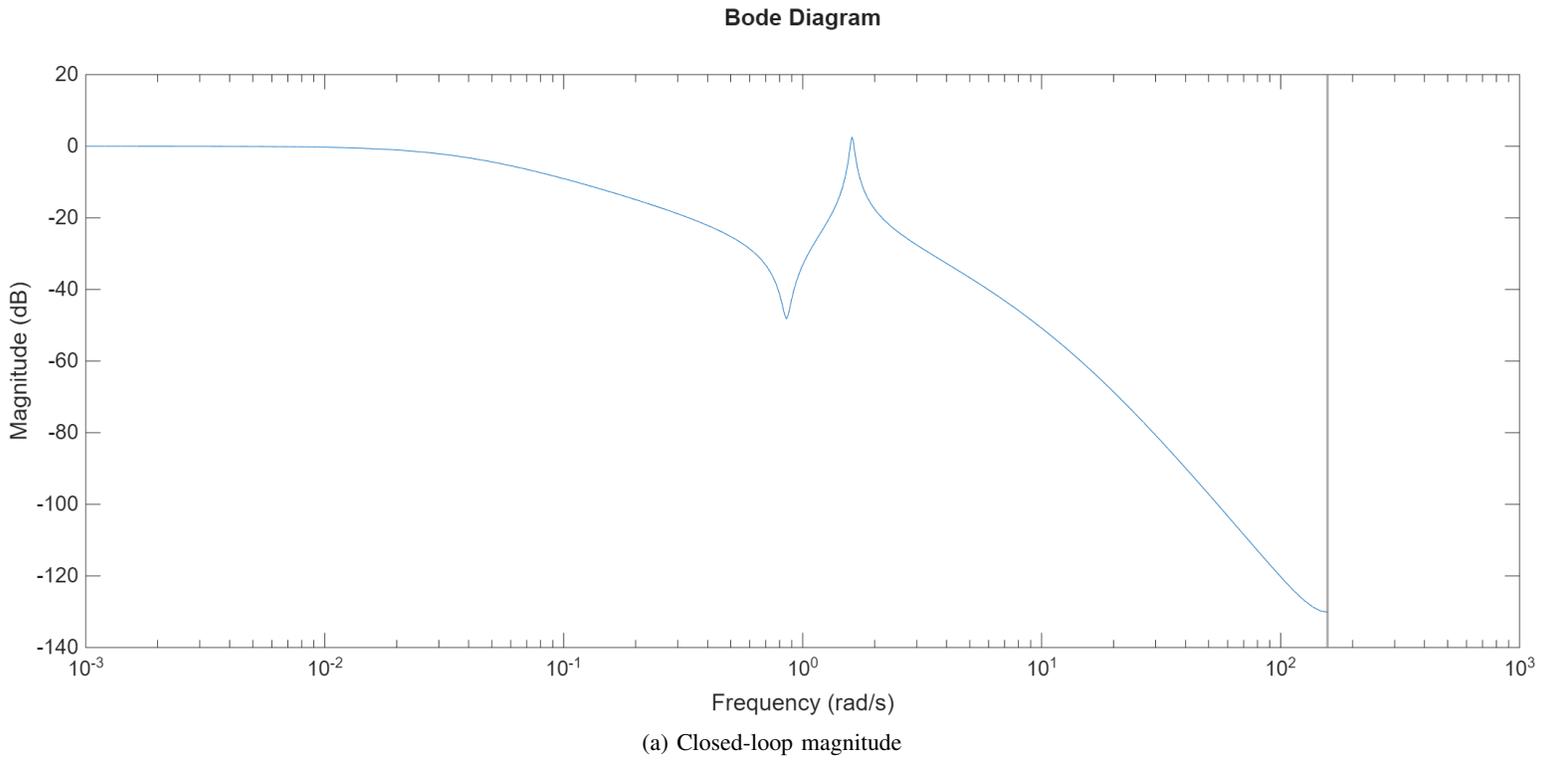

(b) Closed-loop phase

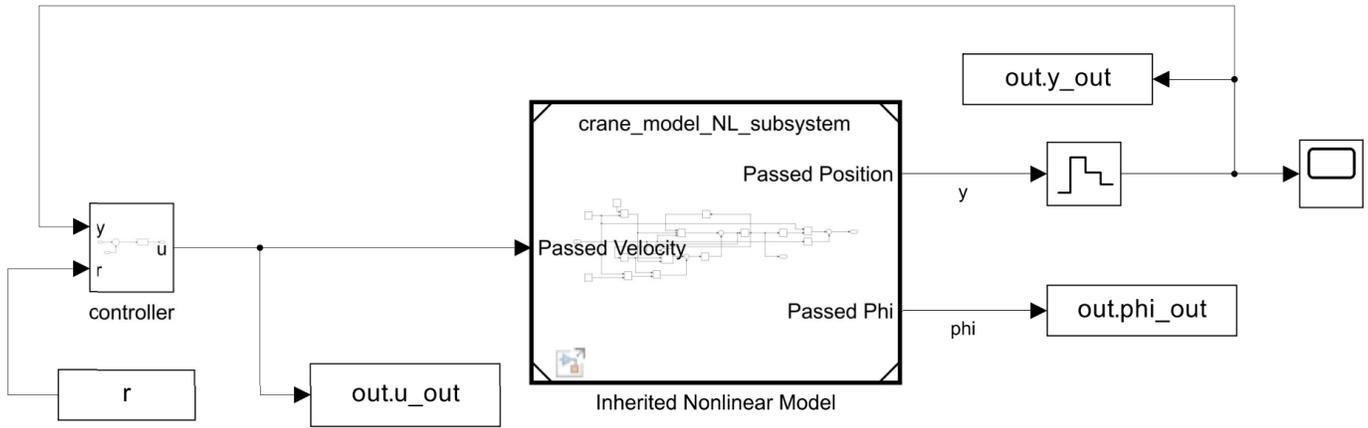Fig. 7: Closed-loop Bode plots for the designed controller.

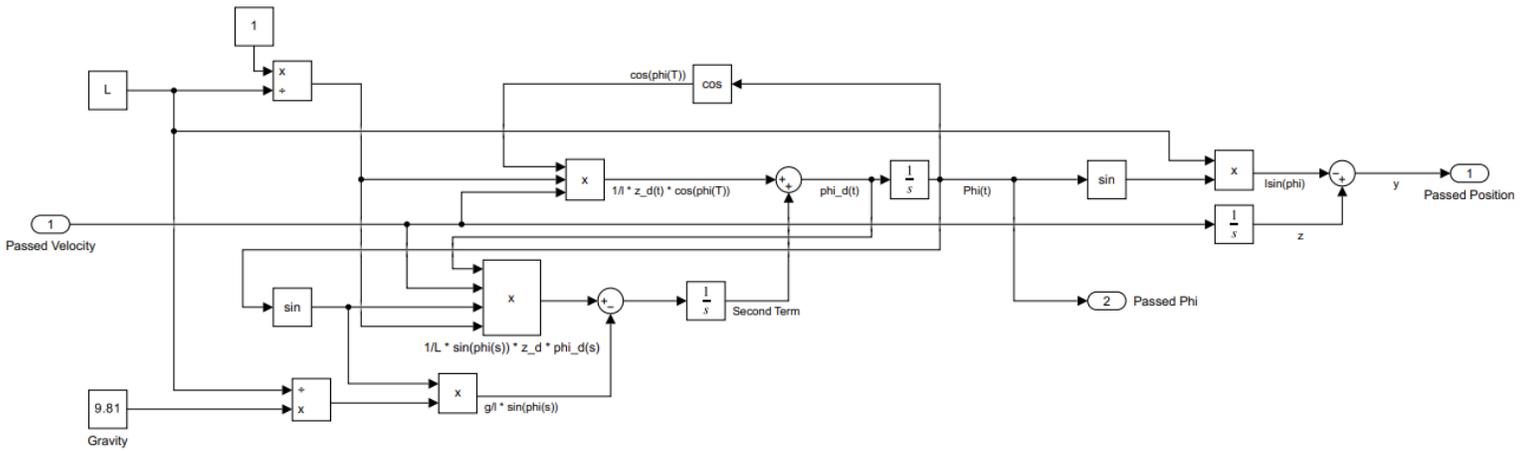Fig. 8: The proposed closed-loop controller used for the gantry crane system

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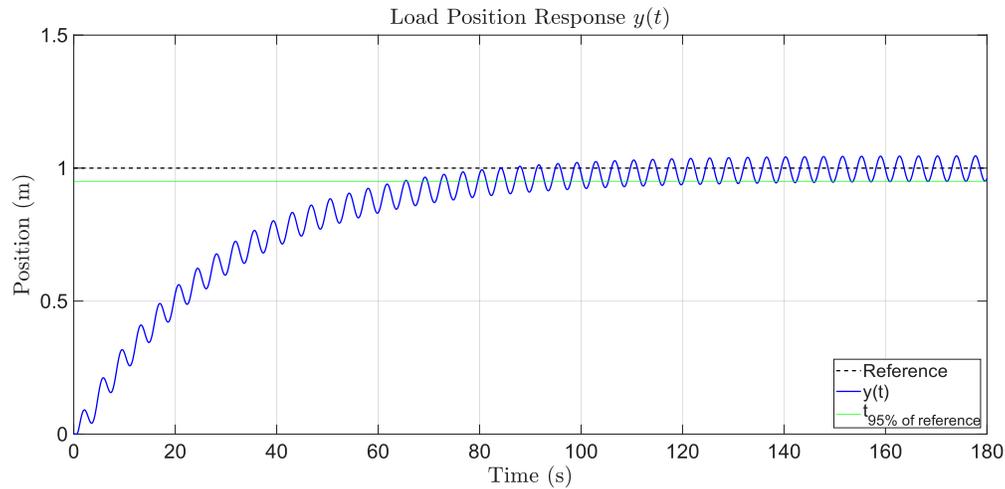

Fig. 9: Simulink model for the crane gantry subsystem

Fig. 10: Unstable controller after 25-sample delay

REFERENCES

[1] IEEE, *Ieee conference template*, https://www.overleaf.com/latex/templates/ieee-conference-template/grfzhhncsfqn, Last updated 6 years ago, 2025.
[2] T. Vincent, *Eeng417: Modern control design, project 1*, 2025.
[3] T. Vincent, *Example project report for eeng417: Modern control design*, 2025.

[4] For all the Bode plots, the title is not an item that can be edited, and therefore has a smaller font than the rest of the figures.