Group 2 LiniX Linear Actuators

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I. Introduction

The purpose of this lab is to have an introduction on transmission technologies used in industry. For this lab we used a brushless DC motor and a linear belt driven axis.

All values presented regarding the K_p, K_d, and K_i gains are the combination of respective gains and multipliers.

II. Theoretical Questions

What type of control does this experiment deal with?
This experiment deals with the position control and torque control.

What type of control does the system use?
Closed loop.

If more static friction is added in the system stronger tightening, a larger tracking error is to be expected. It could be however compensated by increasing Ki gain, while keeping a quick reaction. Is there any issue of doing so?
Yes, when static friction coefficient is suddenly exceeded, integral term has charged and violent reaction occurs.

III. Industrial Linear Guide Solutions

From the introduction given by the teaching assistant, what are the ideal keys of such a Linear Guide Solution?
High reliability and high static load capacity

What are the benefits of a plain bearing guide ?
The pros and cons of a plain bearing guide are represented in the table I:

Pros	**Cons**
- Simplest	- Higher friction
- No need of lubrication	- Lower accuracy
- Silent	
- Tolerant to dirt	

TABLE I: Plain bearing pros and cons

What are the benefits of a linear ball bearing guide ?
The pros and cons of a linear ball bearing guide are represented in the table II:

Pros	**Cons**
- High accuracy	- Does not like contamination
- Very low friction	- Need of lubrication
- Long life longevity	- Must run on hardened shafts
- High working speed	- Some noise

TABLE II: Linear ball bearing pros and cons

What are the benefits of a track roller guide ?
The pros and cons of a track roller guide are represented in the table III:

Pros	**Cons**
- High load capacity	- Lower working speeds
- Tolerant to some contamination	- Some specialized rails can be expensive
- Can be lubricated for life	- Higher start up static friction
- Can support lateral and moment forces	

TABLE III: Track roller guide pros and cons

IV. Assembly

The three main aspects to take care on the LiniX Linear Actuator assembly are:

1) Rollers tightening on the aluminium profile (guide). The rollers' tightening has an impact of the friction forces, both dynamic and viscous. If play is present between the rollers and the guide, the carriage will not be stable and could degrade the rollers and guide in a unplanned manner.
2) Belt's tension. Loosen belt will induce hysteresis and possible slipping behaviour on the driving pulley. A belt that has too much tension will induce higher friction forces and can degrade itself and rolling elements in bearings faster than what is initially planned.
3) Bearing assembly. When mounting bearings, all the elements that constitute the must be well aligned. One must be careful to not transmit any force or load on the inside bolt, as it is constrained.

Those aspects are also important as they are influencing on the forces present in the system. For a given controller with fixed gains, the effect will be that the closed-loop dynamics will changes depending on those influences.

V. GUI PID Explorations

A. P Gain

In this part, we have explored the effect of different values of K_p on the system. Figures 1, 2 and 3 show the behaviour of the electric motor for proportional gains that lead to its stabilization, sustained oscillation, and diverging oscillations respectively when a disturbance is introduced (forced deviation of the position from the set point followed by a release of the motor's shaft). The table IV lists each parameter that have been used. The low frequency beat of the amplitude from figures 2 and 3 comes from the fact that the acquisition frequency is lower than the oscillating frequency of the motor.

The diverging behaviour has been obtained by increasing the maximum speed. It is visible on the position graph that from an initial perturbation around the set point, the oscillations' amplitude grows until it reaches a stable oscillation. The phenomenon is not truly divergent as the viscous friction and the energy losses will, at some point, equal the power that the motor inputs into the system.

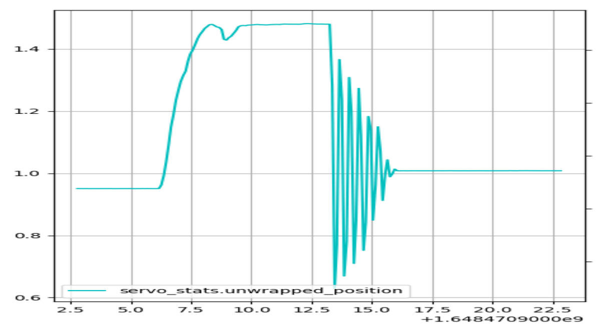

Fig. 1: Position of motor's shaft versus time. Stable behaviour, $K_p = 0.1$

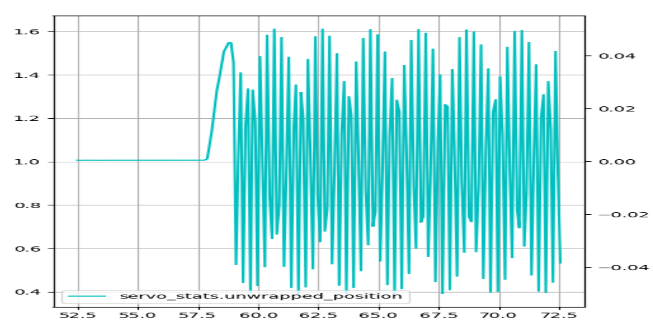

Fig. 2: Position of motor's shaft versus time. Stable oscillations, $K_p = 0.3$

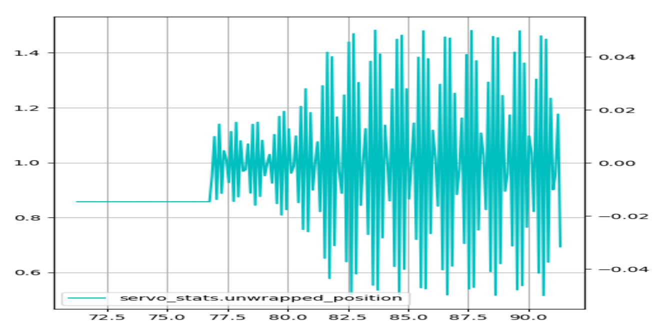

Fig. 3: Position of motor's shaft versus time. Diverging position followed by stable oscillations, $K_p = 0.18$

Stable	0.5 ms	1 Nm	$K_p = 0.1$
Oscillating	0.5 ms	1 Nm	$K_p = 0.3$
Diverging	5.0 ms	0.3 Nm	$K_p = 0.18$

TABLE IV: Values of K_p

We have also experienced the behavior without the belt fixed on the motor. We have observed a huge behaviour difference between the two setups, because the total moment of inertia seen from the electric motor is very different, as well as the friction forces that are added by all the additional moving components.

B. D Gain

In this part, the derivative gain has been set to $K_d = 1.3$. Figure 4 shows how the speed and torque of the motor are linked when the carriage is moved by hand. As expected, the torque is inversely proportional to the displacement speed.

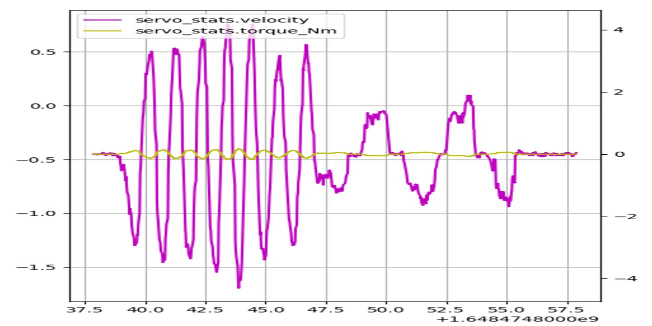

Fig. 4: Velocity and torque of motor's shaft versus time. Visualisation of damping effect when $K_p = 0$ and moving the carriage by hand

C. PID Tuning

In this part, we experience with PID tuning. The first step of PID tuning is to find, having K_d and K_i set to zero, the critical value of K_p which is defined as when the system begins to oscillate. The critical value found is $K_p = 1.5$. The system's behaviour is shown on figure 5. The chosen value of K_p for the controller is half the critical value, thus $K_p = 0.75$, and results into the closed-loop behaviour shown on figure 6.

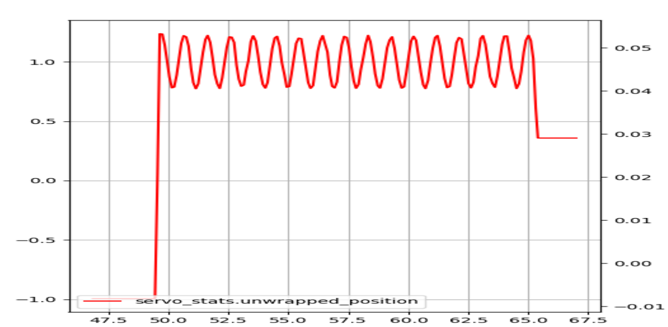

Fig. 5: Position of motor's shaft versus time. The system oscillates at critical value $K_p = 1.5$

The next step is to find the value of K_i that will cancel the steady state error while staying in the stable range of the system. A value of $K_i = 0.8$ presents a good behaviour: figure 7 shows that perturbations applied after the system has reached

Fig. 6: Position of motor's shaft versus time. Closed-loop behaviour at half the critical value $K_p = 0.75$

the set point are cancelled by the controller. A saturation of the integration term of the controller has been set to 1 in order to suppress wind-up effects of the integration term.

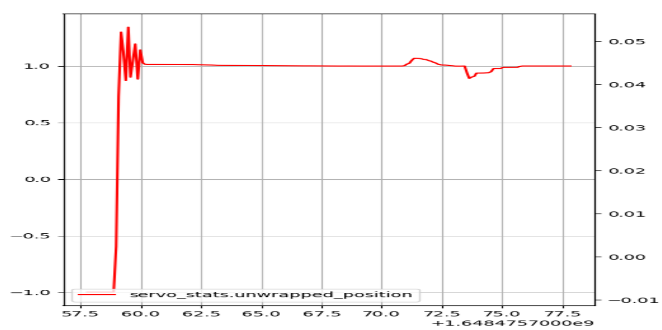

Fig. 7: Position of motor's shaft versus time. Visualisation of cancellation of perturbation made by hand on the carriage when $K_i = 0.8$

The final step is the set the K_d gain in order to suppress the overshoot, but not too high as it would increase the time to reach the set point. The best value found is $K_d = 0.08$. Figure 8 shows the system's behaviour in closed-loop using all gains (K_p, K_d and K_i).

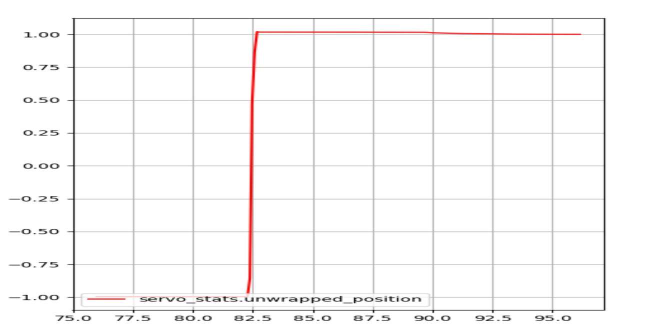

Fig. 8: Position of motor's shaft versus time. Visualisation of the system behaviour with $K_p = 0.75$, $K_i = 0.8$ and $K_d = 0.5$

All the different values of K_d, K_i and K_p used for each step of this experiment can be found in table V.

5.5.2	PID vel 15	1 Nm	kd = 0	ki=0	kp= 1.5
5.5.3	PID vel 15	1 Nm	kd = 0	ki=0	kp= 0.75
5.5.4	PID vel 15	0.3 Nm	kd=0	ki=0.8	kp= 0.75
5.5.5	PID vel 15	max control 1,7	kd=0.5	ki=0.8	kp= 0.75

TABLE V: Values of K_d, K_i and K_p for each step of the PID tuning

VI. Performance Benchmark

On these two last experiments, we had a look on the effect of friction and load over the mechanism's performances. We used rectangular and sine command signals.

A. Friction

In this part, we played with the eccentric nuts to increase/decrease the friction on the linear axis. Figures 9 and 10 represent the plots for the rectangle and the sine command signals respectively, of the position, velocity and torque of the motor's shaft over time when there is no friction, i.e. the bearings are in the most widely spaced configuration. Figures 11 and 12 represent the same plots when there is friction, i.e. the bearings are in the tightest configuration to each other. Figures 13 and 14 represent the same plot for when the bearings are in the middle of the two previous configurations.

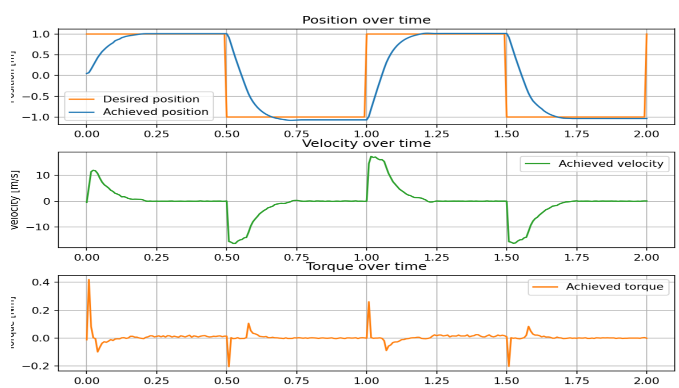

Fig. 9: Frictionless rectangular wave

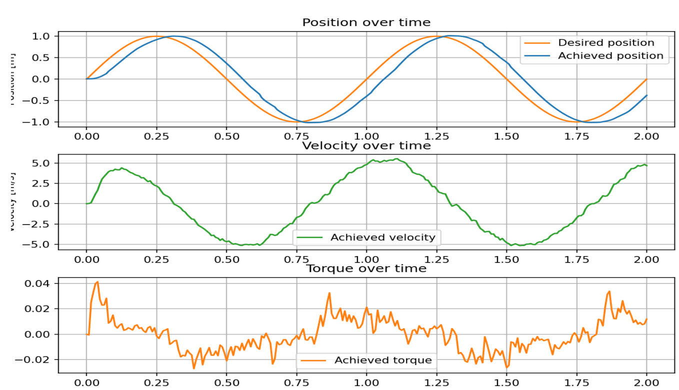

Fig. 10: Frictionless sine wave

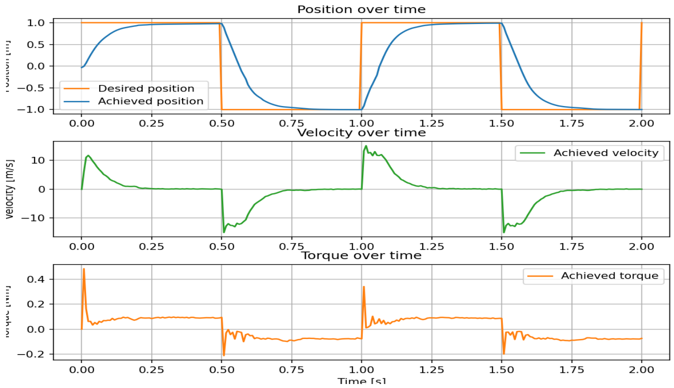

Fig. 11: Friction rectangular wave

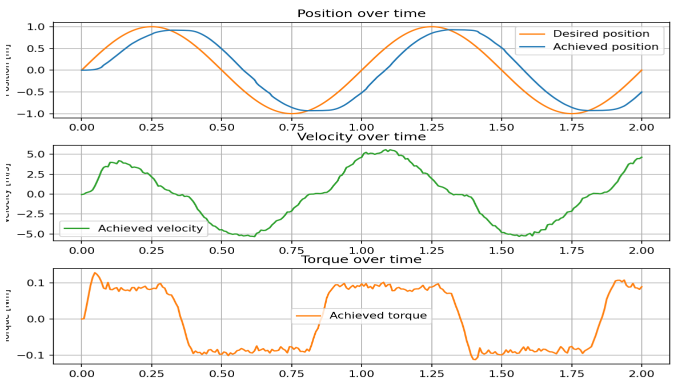

Fig. 12: Friction sine wave

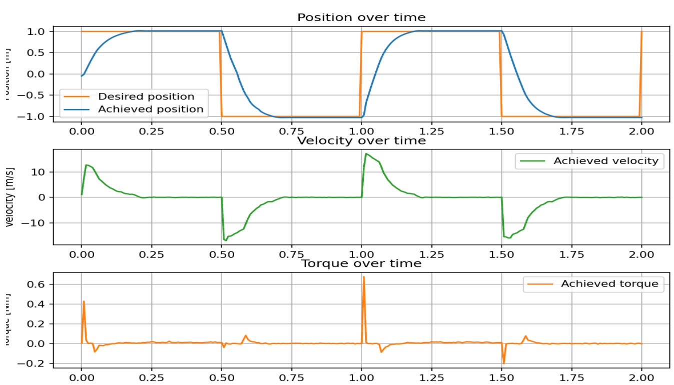

Fig. 13: Nominal rectangular wave

For the rectangular command signal plots, we observe that the more friction there is, the slower the motor's shaft can reach the desired position; there is more noise on the velocity and the torque peaks; the torque values between peaks are never truly null as for the nominal plot. However, the less friction there is, the fastest the motor's shaft can reach the desired position; the velocity peaks are smoother; there is a bit of noise on the signal between the torque peaks.

For the sine command signal plots, we observe that the more friction there is, the slower the motor's shaft can reach the desired position, as for the rectangular command signal; the velocity signal is a very deformed sine wave, it reaches some

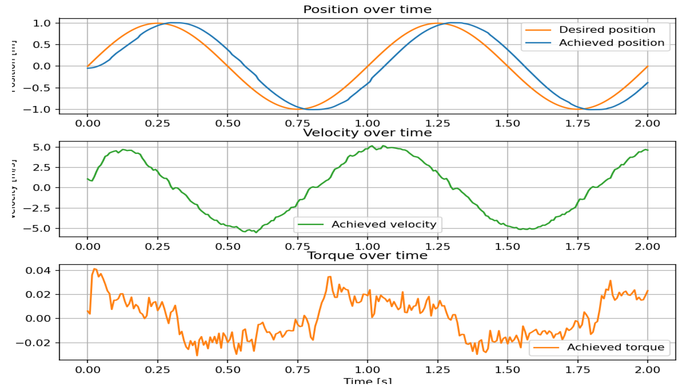

Fig. 14: Nominal sine wave

plateau at the null value; however, the torque plot is better represented than that of the nominal configuration. Moreover, the less friction there is, the fastest the motor's shaft can reach the desired position; the velocity plot is smoother and looks very much like the one for the nominal configuration; however, the torque signal is very noisy.

B. *Load*

In this part, we played with the two masses (201g each) to increase/decrease the load on the linear axis. Figures 15 and 16 represent the plots for the rectangle and the sine command signals respectively, of the position, velocity and torque of the motor's shaft over time when there is no mass. Figures 17 and 18 represent the same plots when there is one mass. Figures 19 and 20 represent the same plot when the are two masses. During all the experiments, the bearings were left at the nominal configuration described earlier.

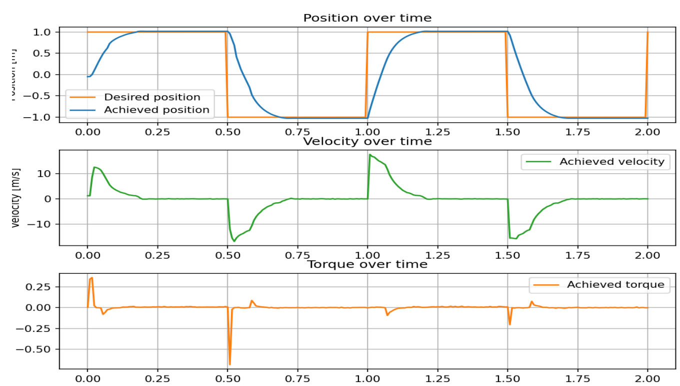

Fig. 15: Rectangular wave with no mass

For the rectangular command signal plots, we observe that the slope of the achieved position is less and less straight as the weight increases; the velocity plot gets noisier when the weight increases; the torque positive peaks get higher and the torque negative peaks get lower as the weight increases.

For the sine command signal plots, we observe that the achieved position and the velocity don't change much depending on the weight; the torque plots are also very noisy for each of the mass configurations.

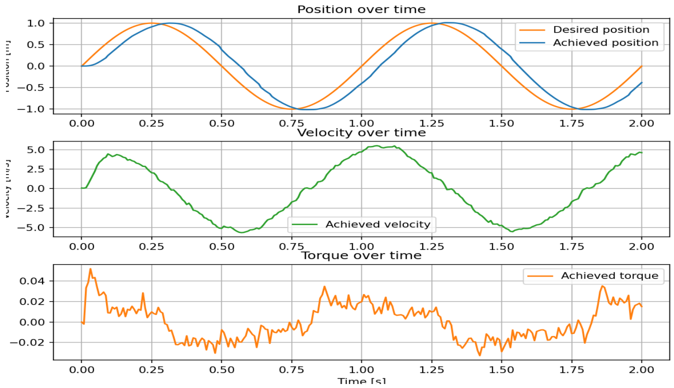

Fig. 16: Sine wave with no mass

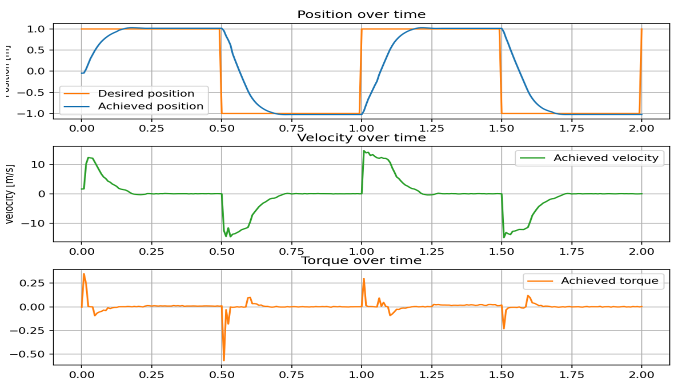

Fig. 17: Rectangular wave with 121g

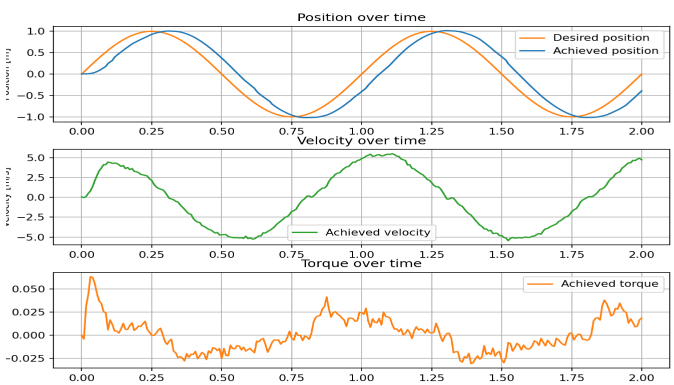

Fig. 18: Sine wave with 121g

VII. Conclusion

During this lab, we experienced the effect of different values of K_p, K_d, and K_i and their respective behaviour on the electric motor. We then could choose the optimal values to tune our PID system.

After that, we observed the effect of friction and load over the mechanism's performances, by changing the bearings spaced configurations thanks to the eccentric nuts, and by adding weights of the motor's shaft. The more friction there is, the more noise is added on the velocity and the torque signals. The heavier the motor's shaft, more noisy is the velocity signal, and the torque increases as well.

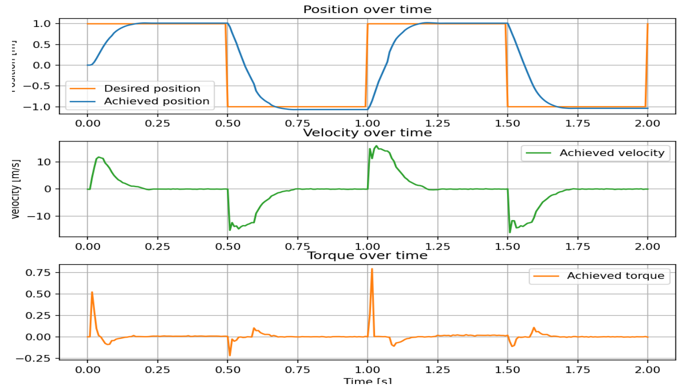

Fig. 19: Rectangular Wave with 242g

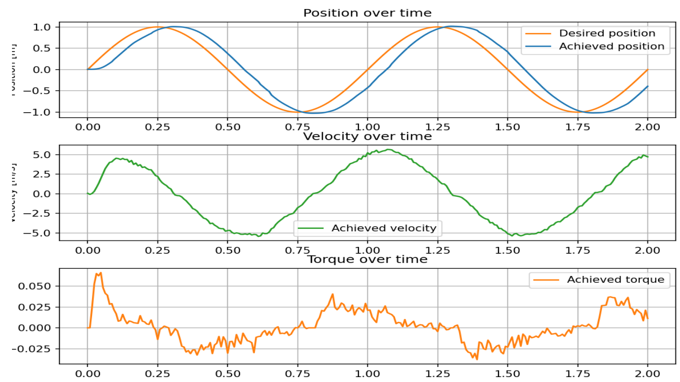

Fig. 20: Sine wave with 242g

We have also learned about the benefits of different bearing guide, and some main aspects on actuator assembly.