This is a postprint version of the following published document:

Monje, C. A., Balaguer, C., Deutschmann, B. y Ott, C.
Control of a soft robotic link using a fractional-order
controller. In *Handbook of Fractional Calculus with
Applications*, De Gruyter, pp. 321-338, 2019.

DOI: <https://doi.org/10.1515/9783110571745-014>

Concepción A. Monje*, Carlos Balaguer, Bastian Deutschmann,
Christian Ott

Control of a soft robotic link using a fractional-order controller

Abstract: The purpose of this work is to present a novel control approach for a tendon-driven elastic continuum mechanism using a Fractional Order (FO) controller. The mechanism is composed of a silicon continuum, tendons and antagonistic actuation which yields a highly complex mechanical model. This model is heavily simplified to that of a linear time invariant second order system and a robust control approach is applied to cope with the neglected dynamic effects. Apart from the specifications for the speed and the overshoot of the system response, the controller must be extensively robust to model parameter mismatches and uncertainties. The FO controller is used to meet the control specifications, taking advantage of the introduction of its fractional order α . Simulation and experimental data are presented to validate the approach in comparison to results from two standard integer-order controllers that are designed using the same specifications.

Keywords: Fractional Order Control, Model-based Control, Soft Robotics, Robust Control

***Corresponding author: Concepción A. Monje, Carlos Balaguer, Systems Engineering and Automation Department, University Carlos III of Madrid, Av. de la Universidad 30, 28911, Leganés, Madrid, Spain, e-mail: cmonje,balaguer@ing.uc3m.es**

Fig. 1: Left: Test setup including two linear actuators which are connected via tendons (blue) to the tip lever plate of the soft continuum mechanism. top: Close up in straight and deflected configurations. bottom: Left side of the antagonistic setup. Right: Tendon-driven soft continuum mechanism as a neck in a humanoid robot [12].

1 Introduction

In recent years, inherently soft robots got in focus of research. These systems ensure that collisions with humans or the environment can be handled by the hardware of the robot. The soft robotic system examined in this chapter is a tendon-driven elastic continuum mechanism (ECM) and the softness is gained by the use of silicon. Its compliance allows the structure to deform as a whole, no deflection of joints can be measured directly for feedback control.

The majority of continuum mechanisms are long, thin tubes used in medical applications as a steerable needle and the flexibility is mainly used to bend continuously whereas axial or shear deformations can be neglected. The present system, however, is able to deform considerably in all directions due to a large cross section and the use of silicon [3]. It is a planar prototype for a spine and neck for a future humanoid robot [12] (Fig. 1). To control the motion of such a system, model-based control approaches are proposed in the literature. For example, a linearized model is used in [6] to design a vibration damping set point controller and a feedback-linearization based on a nonlinear dynamic model is applied in [2].

Bastian Deutschmann, Christian Ott, Institute of Robotics and Mechatronics, German Aerospace Center (DLR), Oberpfaffenhofen, 82234, Wessling, Germany, e-mail: Bastian.Deutschmann,Christian.Ott@dlr.de

A big challenge for model-based approaches are parameter mismatches or changing loading conditions, as they cannot be handled by these approaches instantaneously. Fractional Order (FO) control can be used for robot motion control with considerable structural flexibility, as successfully demonstrated in [8]. The introduction of the fractional order of the controller provides a performance enhancement of the system even when its structural flexibility incorporates complex dynamics and demands high robustness requirements.

This chapter will provide a more detailed discussion of the control approach established by [11] including the recent passivity investigation from [4]. The approach is a model-based approach in the task space of the robot and the complex system is heavily simplified to that of a second-order system. With that, a FO controller [9] is designed in the frequency domain and modeling errors, changing loading conditions, or external disturbances are covered by the inherent robustness properties of the approach. Two well-established integer-order controllers are designed with the same specifications in the frequency domain and are compared to the FO controller with respect to (w.r.t.) two established performance measures in experiments. These experiments cover the desired nominal behavior and investigate the robustness of each controller by varying model parameters and applying high-frequency noise through external disturbances.

2 Problem Statement & System Model

The problem at hand is to control the tip angle $\theta_L \in \mathbb{R}$ of the planar soft continuum mechanism, see Fig. 1, whereas the control action is the external torque $\tau_L \in \mathbb{R}$ at the tip generated by two, antagonistically acting tendon forces $\mathbf{f}_t \in \mathbb{R}^2$ of each actuator while measuring tendon forces and tendon positions $\mathbf{x}_{LM} \in \mathbb{R}^2$ only. As the full dynamic model of this mechanism includes partial differential equations and a nonlinear dynamic coupling between the tip and the actuator motion, it is not suitable for the design of a linear controller. Instead, a linear time invariant transfer function $G(s) \in \mathbb{R}$ is used as an approximation for the dynamics of the tip position $\Theta_L(s)$ w.r.t. the torque input $\tau_L(s)$,

$$G(s) = \frac{K\omega_n^2}{s^2 + 2\delta\omega_n s + \omega_n^2} = \frac{\Theta_L(s)}{\tau_L(s)}, \quad (1)$$

with the steady-state gain $K = 0.0631$, the Eigenfrequency of the system $\omega_n = 37.8021$, and the linear damping $\delta = 0.4$. The parameters have been estimated experimentally by measuring step responses. A comparison between the model and the real system can be found in Fig. 2 for the dynamic and the static case.

Fig. 2: Comparison of the model (mod) and the real system (meas). Left: Dynamic characteristic with an input step of $\tau_L = 8\text{Nm}$. Right: Static characteristic while the mechanism is slowly moved through the workspace.

Table 1: Parameters of the simplified model

steady-state gain	$K = 0.0631$
eigenfrequency	$\omega_n = 37.8021$
damping	$\delta = 0.4$
poles of the system	$p_{1,2} = -15.1208 \pm 34.6462i$
magnitude at $\omega_{cg} = 3\text{rad/s}$	$ G(j\omega_{cg}) _{\text{dB}} = -22.6\text{dB}$
phase at $\omega_{cg} = 3\text{rad/s}$	$\arg(G(j\omega_{cg})) = -3.6^\circ$

3 Control Strategy

The model of the soft robotic system has been approximated to that of a second order system with the transfer function (1) and the parameters in Table 1. The input to the system is the torque applied by the tendons, $\tau_L[\text{Nm}]$, and the output is the angular position of the tip, $\theta_L[\text{rad}]$. Three controllers will be designed for the system: a FO controller, a PID controller and a lag compensator. A discussion on the required specifications as well as the constraints that need to be fulfilled are presented in the following.

3.1 Control Specifications

The objective of this control approach is to design a controller $C(j\omega)$ in the frequency domain so that the system is robust to system uncertainties and load disturbances. For this reason, specifications related to gain crossover frequency ω_{cg} , phase margin φ_m and a robustness constraint are going to be considered,

due to their important significance regarding performance and stability [10]. The design problem is formulated as follows:

- **Phase margin and gain crossover frequency specifications.** Gain and phase margins are important measures of robustness. It is known that the phase margin is related to the damping of the system and therefore can also serve as a performance measure [5]. The equations that define the gain crossover frequency and the phase margin are:

$$|C(j\omega_{cg})G(j\omega_{cg})|_{dB} = 0dB. \quad (2)$$

$$\arg(C(j\omega_{cg})G(j\omega_{cg})) = -\pi + \varphi_m. \quad (3)$$

- **Robustness to variations in the gain of the plant.** According to [1], this requirement can be addressed by demanding:

$$\left. \frac{d(\arg(C(j\omega)G(j\omega)))}{d\omega} \right|_{\omega=\omega_{cg}} = 0s. \quad (4)$$

The second specification forces the phase of the open loop system $F(j\omega) = C(j\omega)G(j\omega)$ to be flat at $\omega = \omega_{cg}$ and so, to be almost constant within an interval around ω_{cg} . It means that the system is more robust to gain changes and the overshoot of the response is almost constant within a gain range (iso-damping property of the time response). As for this robustness specification, there are two important considerations to make:

1. In our system, the gain of the model is mainly affected by the load of the device (top masses or pending masses in the experimental results section). Besides, a change in the pulley location can also result in a different gain value. Therefore, with this specification the controller ensures the robustness of the system to changes in its physical parameters, guaranteeing a good performance for a range of uncertainty around the nominal configuration.
2. The design method proposed here for the FO controller will be based on a graphical approach, avoiding the analytical resolution of (4) for the obtaining of the controller parameters, see section 3.2.

For the case study presented in this work, the following constraints are used in order to compare the different controllers:

- The workspace of the system is $\theta_L = \pm 0.35\text{rad} (\approx 20^\circ)$.
- The pretension force is $f_{pre} = 10\text{N}$.
- The maximum control torque is $\tau_L = 8\text{Nm}$ as the maximum tendon force is $f_{t,max} = 110\text{N}$.
- It is desirable to have a negligible steady-state error and a step response with a settling time of $\approx 1\text{s}$, which corresponds to about $\omega_{cg} = 3\text{rad/s}$.

- To achieve a better performance, it is desirable to have a slight overshoot. Thus $\varphi_m = 80^\circ$ is demanded.

The different controllers proposed next will be designed ensuring that these specifications are fulfilled. For the sake of comparison, two integer-order controllers including a Lag compensator and a PID controller are designed with the specifications above. The latter ensures no steady-state error as it includes a pure integral term. However, this integral action could brake the system, e.g. a tendon, when the mechanism is in contact with the environment.

Furthermore, a FO PD^α controller is designed as it is able to fulfill the control specifications and ensures good robustness properties. A deeper discussion on the advantages of using this controller over the previous mentioned integer-order ones will be presented in Section 4 of experimental results.

3.2 FO Controller

As a novel control approach for the soft continuum mechanism, we propose the use of the generalized FO PD^α controller, formulated as [9]:

$$PD^\alpha(s) = \left(\frac{\lambda s + 1}{x\lambda s + 1} \right)^\alpha, \quad (5)$$

which corresponds to a FO lead compensator that can be identified as a PD^α controller plus a noise filter. The parameters $\alpha, \lambda, x \in \mathbb{R}$ are now obtained by the subsequent method.

The design method proposed here for the tuning of the controller is based on the work previously conducted by the authors in [10]. As an improvement, considerations on the dual use of the controller as a lead-lag compensator are introduced. For a specified phase margin φ_m and gain crossover frequency ω_{cg} , the following relationship for the open-loop system is given in the complex plane:

$$\begin{aligned} G(j\omega_{cg}) \cdot \left(\frac{j\lambda\omega_{cg} + 1}{jx\lambda\omega_{cg} + 1} \right)^\alpha &= e^{j(-\pi + \varphi_m)}, \\ \left(\frac{j\lambda\omega_{cg} + 1}{jx\lambda\omega_{cg} + 1} \right)^\alpha &= \frac{e^{j(-\pi + \varphi_m)}}{G(j\omega_{cg})} = a_1 + jb_1, \\ \left(\frac{j\lambda\omega_{cg} + 1}{jx\lambda\omega_{cg} + 1} \right) &= (a_1 + jb_1)^{1/\alpha} = a + jb, \end{aligned} \quad (6)$$

where $G(s)$ is the plant to be controlled and (a_1, b_1) is called the “*design point*”, which is directly obtained from the system (1) with Table 1 and the values of

Fig. 3: Left: Lead regions for the controller $C(j\omega_{cg})$ for $0 \leq \alpha \leq 2$. Right: Parameter pairs (x, λ) for $\alpha_{\min} \leq \alpha \leq 2$.

φ_m and ω_{cg} . After solving (6) for x and λ , we have

$$x = \frac{a-1}{a(a-1)+b^2}, \quad (7)$$

$$\lambda = \frac{a(a-1)+b^2}{b\omega_{cg}}. \quad (8)$$

Studying the conditions for a and b to find a solution, it can be concluded that a lead compensator is obtained when $a > 1$ and $b > 0$ [9]. According to this condition and considering (7) and (8), the lead compensation regions in the complex plane for different positive values of α are obtained, as shown in Fig. 3, left. The zone to the right of each curve is the lead region, and any design point in this zone can be fulfilled with a FO compensator having a value of α equal or bigger than the one defining the curve which passes through the design point $\alpha_{\min}$. For instance, for the design point $\alpha_{\min} = 0.48$ in Fig. 3, left, Fig. 3, right shows the pairs (x, λ) obtained for each value of α in the range $\alpha_{\min} = 0.48 \leq \alpha \leq 2$. Therefore, once the design point is computed, the selection of the controller parameters is direct and flexible through the use of the curves in Fig. 3.

For the sake of robustness to gain variations, and according to the discussion in Section 3.1, the phase curve of the open-loop system should present a flat shape (zero slope) around the frequency of interest ω_{cg} . It is then desirable for the FO controller to contribute with a phase curve that both guarantees the amount of phase required for the fulfillment of the phase margin specification

at that frequency and ensures that the flat phase constraint for the open-loop system is achieved.

Taking into account the phase characteristic of the system $G(s)$, see (1) with Table 1, which presents a slope close to zero at $\omega_{cg} = 3\text{rad/s}$, the condition for the phase curve of the FO controller is also to have a flat behavior around that frequency, ensuring this way that the open-loop system phase is as flat as possible at ω_{cg} . It can be observed from Fig. 3 that the maximum value of x is obtained for $\alpha_{\min}$. In other words, by choosing the minimum value $\alpha_{\min}$, the distance between the zero and the pole of the compensator will be the maximum possible, which ensures that the phase curve of the compensator is the flattest possible and variations in a frequency range centered at ω_{cg} will not produce a significant phase change as in other cases. That improves the robustness of the system. This will be the criterion to be followed here and the controller will be tuned according to this $\alpha_{\min}$ condition.

In case a phase lag (φ_{lag}) is required for the compensator in order to fulfill the design specifications, it will be designed as a lead compensator giving a phase $|\varphi_{\text{lead}}| = |\varphi_{\text{lag}}|$, and later the sign of α will be changed so that the phase contribution is negative. Besides, it has to be taken into account that a change in the sign of α for the lag compensation leads to an inverted magnitude of the designed compensator. So, in order to keep the gain unchanged (fulfilling already the specification of crossover frequency), the lag compensator should be multiplied by a gain $k_{\text{lag}} = 1/|G(j\omega_{cg})|^2$, as can be reached from (6). Therefore, the fractional lag compensator will be given by:

$$C_{\text{FOC}}(s) = k_{\text{lag}} \left(\frac{\lambda s + 1}{x\lambda s + 1} \right)^{-\alpha}, \quad (9)$$

with α a positive real number.

Now a FO controller of the form in (9) is designed so that the specifications $\varphi_m = 80^\circ$ at $\omega_{cg} = 3\text{rad/s}$ are achieved, together with the requirement of robustness to gain variations. After the design process the resulting controller is:

$$C_{\text{FOC}}(s) = 905.7688 \left(\frac{6.97s + 1}{0.0069s + 1} \right)^{-1.12}, \quad (10)$$

with $\lambda = 6.97$, $x = 9.9291 \cdot 10^{-04}$, $\alpha = 1.12$. The Bode plots of the open-loop system with this FO controller are shown in Fig. 4, where it can be observed that the phase margin, gain crossover frequency, and robustness constraints (flat phase) are fulfilled.

In order to evaluate the robustness of the closed-loop system to gain variations, Fig. 5, left presents the step responses for varying gains K (1), showing that the overshoot keeps almost constant.

Fig. 4: Bode plots of the open-loop system of the three designed controllers

Fig. 5: Left: Nominal step responses of the closed-loop system. Top: $C_{FOC}(s)$ controller with different values of the gain (Table 1): K , $0.5K$ and $1.5K$. Bottom: $C_{PID}(s)$ and $C_{LAG}(s)$. $\theta_{L,d}$ in black. Right: Nyquist plots of the $P(s)$ functions corresponding to the three controllers. The dashed, red curve corresponds to the finite approximation of the FO controller.

Integer Order Approximation

When FO controllers have to be implemented, fractional transfer functions are usually replaced by integer transfer functions with a behavior close enough to the one desired, but much easier to handle. There are many different ways of finding such approximations but unfortunately it is not possible to say that one of them is the best, because even though some of them are better than others in regard to certain characteristics, the relative merits of each approximation depend on the differentiation order, on whether one is more interested in an accurate frequency behavior or in accurate time responses, or on how large admissible transfer functions may be, among others. A good review of these approximations can be found in [9].

In our particular case, to implement the FO controller in the real setup, the transfer function (10) has been approximated to that of an integer-order controller using a frequency identification method performed by the MATLAB function *invfreqs*. An integer-order transfer function is obtained which fits the frequency response of the FO controller in a frequency of two decades around the gain crossover frequency, with two poles/zeros:

$$\tilde{C}_{\text{FOC}}(s) = \frac{0.3908s^2 + 67.3s + 774.4}{s^2 + 7.465s + 0.8204}. \quad (11)$$

This method has been chosen due to its accuracy in the frequency range of interest, as successfully applied for real implementations in [8], [10], but any other of the techniques in [9] could also be suitable for this purpose.

3.3 Integer-order PID Ccontroller

A PID controller is designed with the graphical user interface of the *Automated PID Tuning* method incorporated in the *Control Systems Toolbox* in MATLAB. The tuning program uses the *Robust Response Time* method which allows the specification of the bandwidth and the phase margin to be $\omega_{cg} = 3\text{rad/s}$ and $\varphi_m = 90^\circ$, respectively. Note that the maximum contribution in phase of this controller is -90° , which makes it impossible to fulfill the phase margin specification of 80° and, in general, a wider range of values for the phase margin. The PID controller designed following these instructions is

$$C_{\text{PID}}(s) = 41.17 \cdot \frac{(1 + 0.024s)^2}{s}, \quad (12)$$

whose open-loop Bode plots are presented in Fig. 4. The nominal step responses of the closed-loop system is shown in Fig. 5, bottom left.

A physical interpretation of this PID controller is that by choosing both zeros of the controller to be at $z_{1,2} \approx 41.6\text{rad/s}$, the controller approximately cancels the poles of the system (Table 1), which extends the inherently flat phase to the system (see Fig. 4). As the chosen design method maximizes bandwidth and optimizes phase margin, it is a compromise between robustness and performance.

3.4 Integer-order Lag Compensator

An integer-order lag compensator is designed for the system. However, the integer order limits the fulfillment of the phase margin specification, since the maximum phase contribution is -90° . Besides, the flat phase robustness constraint cannot be guaranteed due to the lack of an extra parameter.

For the case study presented in this work, an integer-order lag compensator is designed fulfilling the frequency specifications $\omega_{cg} = 3\text{rad/s}$ and $\varphi_m = 90^\circ$. The general transfer function of an integer-order lag compensator is

$$C_{\text{LAG}}(s) = \frac{s + z}{s + p}, \quad \text{with } z > p. \quad (13)$$

The pole p and the zero z are determined using the specifications that at $\omega_{cg} = 3\text{rad/s}$ the magnitude of compensator needs to be 22.6dB with a phase of -90° . By solving (2) and (3), we find the integer-order lag compensator to be

$$C_{\text{LAG}}(s) = \frac{s + 39.74929}{s + 0.3007329}. \quad (14)$$

Fig. 4 and Fig. 5 show the Bode plots and the step response of the system with this lag compensator, respectively. In these simulation results, it can be seen that the steady-state error is higher in comparison with the FO and PID controllers.

3.5 Passivity Analysis

The controller design in the previous section was based on a linear system model. It was shown that the proposed FO controller has advantageous robustness properties, for example w.r.t. gain variations. The experimental results in Section 4 will confirm that the inaccuracy in the linear approximation of the system dynamics can be handled by these robustness properties. In this section we want to discuss to which extent one can make analytical stability statements for the application of the (linear) FO controller to the nonlinear system dynamics. In particular, we look at the passivity properties of the closed-loop system. As a purely mechanical structure, the continuum mechanism clearly is a passive system

with the physical energy as a storage function, considering the input τ_L and the output $\dot{\theta}_L$. This input-output pair represents the physical power between the controller and the mechanism. In the following, we will show that the designed FO controller represents a passive system w.r.t. the input $\dot{\theta}_L$ and the output $-\tau_L$, i.e. in feedback interconnection with the mechanism. Since passivity is preserved by feedback interconnection of passive subsystems [14], we can then conclude passivity of the closed-loop system. This even extends to physical interaction of the mechanism with its environment. As long as the environment is passive, the same holds for the overall system.

For showing the passivity of the controller we have to consider the transfer function $P(s) = -C_{\text{FOC}}(s)\frac{1}{s}$. Passivity requires $\text{Re}(P(j\omega)) \geq 0$ (positive realness) for all frequencies ω , which can be checked with a Nyquist plot [13]. Figure 5, right shows the Nyquist plots of the $P(s)$ functions corresponding to the three controllers discussed in the previous sections. One can easily observe that the FO controller as well as its finite-order approximation are positive real, while this is not the case for the PID controller.

Notice that this analysis assumes that the electrical dynamics of the force-controlled actuators is sufficiently fast so that it can be neglected and thus the desired control torque τ_L can be realized instantaneously. While unmodeled dynamics at the level of the continuum mechanism can be handled in this way, unmodeled dynamics at the actuator level would need a different approach.

4 Experimental Results

4.1 Test Setup

The setup used for the experiments is depicted in Fig. 1. Two linear actuators from Linmot[®] with incorporated position sensors are equipped with an axial force sensor from Omega[®] to measure position and external force at each slider. The designed motion controllers and the force control loops are implemented in MATLAB/Simulink[®] using real time workshop on a QNX-neutrino 6.5 target. An EtherCAT bus sends the generated control signals to the current controller of the linear motors and receives sensor data within a control cycle of 1kHz. By applying EtherCAT as an industrial standardized real time communication protocol, a high degree of determinism is ensured. The sensor data are the actuator positions $\mathbf{x}_{\text{LM}}$ and tendon forces $\mathbf{f}_t$. The error level of the force sensors is $\pm 0.1\text{N}$ with 0.1% of linearity. The mean error of the position encoders is $\pm 0.01\text{mm}$. The soft continuum mechanism is molded out of silicon from Dragonskin[®] and

connected to a 3D-printed bottom plate and a tip lever plate with an anchored area to ensure a firm connection. The polyethylene tendons are looped around a ball bearing, and are mounted to the soft continuum mechanism at a bearing seat on the top lever, see Fig. 1. The tendons are routed with pulleys at each side towards the linear actuators.

4.2 Evaluation

As introduced earlier, the performance of each controller is evaluated using the ISE- index,

$$\text{ISE} = \int_0^{t_{\text{end}}} e(t)^2 dt, \quad (15)$$

with $e(t) = \theta_{L,d}(t) - \theta_L(t)$ being the error between the desired and the actual control output. The ISE measure is associated to the error energy and is a well-known performance index in control design [7]. A low ISE-value indicates a small control error, which is desirable. Furthermore, we want to assess the control action by

$$\text{CA} = \frac{1}{t_{\text{end}}} \int_0^{t_{\text{end}}} (\tau_L(t=0) - \tau_L(t))^2 dt \quad (16)$$

to indicate the amount of energy created by the control law. A low CA-value indicates less control action, which is also desirable.

4.3 Nominal Responses

Fig. 6, left shows the experimental step responses of the system in closed loop with the FO controller, the PID controller and the integer-order lag compensator, together with their corresponding control laws. As can be seen in the figure, the FO controller presents a slightly underdamped response (peak value of 20.44°), as expected, and a soft control law under the saturation limit of 8Nm. The final value of the response is 19.65° , presenting a small but negligible steady-state error due to the absence of an integral component in the controller. The PID controller presents zero steady-state error due to the effect of the integral action. However, the integrator may cause the drifting of the control action when an external force blocks the device, which may eventually break the tendon. The lag compensator presents a higher steady-state error (final value of 17.77°). Though

Table 2: ISE and CA of the three different controllers

Controller	ISE m^2s			CA 10^4 (Nm)^2		
	PID	LAG	FOC	PID	LAG	FOC
Nominal	12.34	17.05	7.63	2.52	2.26	2.94
1 Mot.	9.56	13.23	5.94	1.88	1.71	2.19
097	20.91	27.34	13.09	1.87	1.75	2.96
Mass 194	22.07	27.65	12.54	1.82	1.73	2.93
380	20.51	26.84	12.90	1.82	1.71	2.84
566	20.72	26.46	12.87	1.73	1.63	2.75
Disturb. 0°	0.90	1.19	0.69	0.28	0.23	0.12
20°	0.80	0.90	0.40	0.32	0.30	0.25

the three controlled systems present a transient with similar settling time, the rise time for the FO controller is lower and its steady-state error negligible, providing a better response than the integer-order controllers. The assessment of the three controllers w.r.t. the performance measures is shown in Table 2. For the nominal response (first row), one can observe that the FO controller has the lowest ISE value, meaning the least error over time to the desired motion. However, the FO controller needs the highest control action, indicated in a higher CA value.

In order to test that the device behaves experimentally as a linear system around the nominal working point, the FO control system has been tested for different step inputs with amplitudes in the range $(5^\circ, 25^\circ)$. The results are presented in Fig. 6, right, showing that only slight variations are appreciated w.r.t. the nominal performance.

4.4 Mass Variation

To evaluate the robustness of the closed-loop system, step responses of the three controllers are measured when one of the tendons is disconnected. This means that mass and frictional damping of the real system decrease w.r.t. the model and its parameters. The results are shown in Fig. 7, left for the three controllers. All the controllers result in a stable system.

Another, more critical robustness test of the closed-loop system is the placement of additional masses on the top of the mechanism. This mass implies a change in

Fig. 6: Left: Experimental step responses θ_L of the system in closed-loop and corresponding control inputs τ_L of the three controllers. Right: Experimental step responses of the system with the FO controller (red) for different step inputs with amplitudes in the range ($5^\circ, 25^\circ$).

Fig. 7: Left: Experimental step responses and tendon forces of the system in closed loop with only one connected motor. Right: Experimental step responses and tendon forces for different external masses placed on the top of the mechanism with the FO controller.

Fig. 8: Experimental step responses and control laws of the system in closed loop for different external masses placed on the top of the mechanism. Left: PID controller. Right: LAG compensator

the total impedance of the system. Particularly, the assumption that the dynamics of the soft continuum mechanism are neglected is jeopardized. The results are shown in Fig. 7, right for the FO controller and in Fig. 8 for the PID and LAG controller. The additional masses on the top are $m_{\text{ext}} = \{97, 194, 380, 566, 663, 849\}\text{g}$. The FO controller presents a robust performance to this variation of the overall impedance, while the PID controller and the lag compensator become unstable for $m_{\text{ext}} = 566\text{g}$. For a mass variation, the performance measures were also evaluated and yield the same results as in the nominal response, being a low ISE value and a higher CA value for the FO controller (row 2-6 in Table 2). The more mass loaded on top of the system, the less control action is necessary to deflect the mechanism, which is displayed in the course of the CA values.

4.5 External Disturbances

A requirement for the present mechanism is to handle external disturbances to a certain extent. To investigate the behavior of the control system while it is subjected to external disturbances, the following experiments are carried out. In our approach, θ_L is controlled at static positions and an external mass of 849g disturbs the control system in the following way:

1. A disk is attached to the tip lever plate with a cord; the cord is guided with an additional pulley to ensure a disturbance in parallel to the actuation.

2. The disk is dropped from a fixed distance, causing a high impact on the tip. The distance traveled by the disk is about 15cm.

This experiment has been done for two static positions, 0° and 20° tip deflection. The results are shown in Fig. 9. A robust performance is obtained in all the cases. However, the disturbance rejection property of the FO controller is superior to the integer-order controllers. This is further substantiated by the two performance measures in the last two lines of Table 2. Here, a very interesting result can be reported as the FO controller still impresses with a low ISE value but also with a small control action with lower CA-values than both integer-order controllers. The reported behaviour is substantiated by a video that shows the experiments, see <https://www.youtube.com/watch?v=ivR-3bN0LVA&feature=youtu.be>.

5 Conclusions and Future Works

This chapter reports on a model-based approach to control the position of a tendon-driven system. The complex mechanical system is heavily simplified to that of a linear time invariant second-order system. A fractional-order lag compensator is designed with this model and is proven to be passive. The required specifications for the closed-loop system are met with this controller and the incorporated robustness is able to cope with unmodeled dynamic effects.

In order to compare the FO controller, two standard linear controllers are designed with the same specifications whereas three different experiments are conducted: a PID controller, which is typically not passive, and a lag compensator, which is passive. The nominal motion corresponds to the system step response of the link angle $\theta_L = 20^\circ$. The three controllers prove an experimentally stable response. However, the rise time of the FO controller is superior to the others while having a negligible steady-state error of 0.35° , which is also reflected in a low ISE value compared to the integer-order controllers.

In order to evaluate the robustness w.r.t. additional parameter variations, several different masses are mounted on the tip of the soft continuum mechanism and again step responses are conducted. Again, the FO controller presented a superior performance, as it showed a stable behavior with higher additional masses than both integer-order controllers, which become unstable for masses higher than 566g.

Since external collisions are also of interest, the controlled soft robotic system is investigated under these conditions as well. The controllers show a stable behavior in the testbed. However, the superior disturbance rejection of the FO controller

Fig. 9: Experimental responses with the FO controller, the PID controller and the integer-order lag compensator in the presence of disturbances. Left: 0° tip deflection. Right: 20° tip deflection.

needs to be emphasized. This property is also reflected in the performance indices that were investigated: a lower ISE value for the FO controller combined with a lower CA value, meaning that less control energy is necessary in order to track the desired position while being externally disturbed.

Acknowledgement

The research leading to these results has partially received funding from the HUMASOFT project, with reference DPI2016-75330-P, funded by the Spanish Ministry of Economy and Competitiveness.

References

- [1] Y. Chen, C. Hu, and K. L. Moore, “Relay feedback tuning of robust pid controllers with iso-damping property,” in *Decision and Control, 2003. Proceedings. 42nd IEEE Conference on*, vol. 3. IEEE, 2003, pp. 2180–2185.
- [2] B. Deutschmann, A. Dietrich, and C. Ott, “Position control of an underactuated continuum mechanism using a reduced nonlinear model,” in *Decision and Control (CDC), IEEE Conference on*, 2017, pp. 5223–5230.
- [3] B. Deutschmann, T. Liu, A. Dietrich, C. Ott, and D. Lee, “A method to identify the nonlinear stiffness characteristics of an elastic continuum mechanism,” *IEEE Robotics and Automation Letters*, pp. 1450–1457, 2018.
- [4] B. Deutschmann, C. Ott, C. A. Monje, and C. Balaguer, “Robust motion control of a soft robotic system using fractional order control,” in *Conference on Robotics in Alpe-Adria Danube Region*. Springer, 2017.
- [5] G. F. Franklin, J. D. Powell, and A. Emami-Naeini, “Feedback control of dynamics systems,” *Addison-Wesley, Reading, MA*, 1994.
- [6] I. A. Gravagne, C. D. Rahn, and I. D. Walker, “Large deflection dynamics and control for planar continuum robots,” *IEEE/ASME Transactions on Mechatronics*, vol. 8, no. 2, pp. 299–307, 2003.
- [7] W. S. Levine, *The control handbook*. CRC press, 1996.
- [8] C. Monje, F. Ramos, V. Feliu, and B. Vinagre, “Tip position control of a lightweight flexible manipulator using a fractional order controller,” *Control Theory & Applications, IET*, vol. 1, no. 5, pp. 1451–1460, 2007.
- [9] C. A. Monje, Y. Chen, B. M. Vinagre, D. Xue, and V. Feliu-Batlle, *Fractional-order systems and controls: Fundamentals and applications*. Springer Science & Business Media, 2010.

- [10] C. A. Monje, B. M. Vinagre, V. Feliu, and Y. Chen, “Tuning and auto-tuning of fractional order controllers for industry applications,” *Control Engineering Practice*, vol. 16, no. 7, pp. 798–812, 2008.
- [11] C. A. Monje, C. Balaguer, B. Deutschmann, and C. Ott, “Fractional order control of a soft robotic system,” in *Advances in Cooperative Robotics*. World Scientific, 2017, pp. 771–779.
- [12] J. Reinecke, B. Deutschmann, and D. Fehrenbach, “A structurally flexible humanoid spine based on a tendon-driven elastic continuum,” in *Robotics and Automation (ICRA), IEEE Conference on*, 2016, pp. 4714–4721.
- [13] R. Sepulchre, M. Jankovic, and P. Kokotovic, *Constructive Nonlinear Control*. Springer-Verlag, 1997.
- [14] A. van der Schaft, *L_2 -Gain and Passivity Techniques in Nonlinear Control*, 2nd ed. Springer-Verlag, 2000.