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INTERACTION CONTROL OF A NON-BACKDRIVEABLE MR-COMPATIBLE ACTUATOR THROUGH SERIES ELASTICITY

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ABSTRACT

This research aims at developing a magnetic resonance (MR)-compatible equivalent of an exoskeleton used for wrist movement rehabilitation therapy of neurological patients. As a crucial step towards the accomplishment of this goal, this paper investigates the development of a novel actuation architecture suitable for interaction control in MR environments, the MR-SEA (SEA stands for Series Elastic Actuator). MR-SEA consists of the combination of a non-backdriveable MR-compatible actuator and of a compliant force-sensing element. The preliminary design of a 1 DOF actuator is presented, in addition to non-linear dynamical model of the system featuring the most relevant actuator non linearities. A switching controller is proposed, and the asymptotic stability of the resulting discontinuous dynamical system is demonstrated for force control in blocked output conditions. Simulation results show that the proposed system is adequate for the implementation of force control for wrist movement protocols in fMRI, demonstrating a bandwidth higher than 8 Hz for force control. For stiffness control, simulation results demonstrate that the system is passive for all values of desired virtual stiffness lower than the stiffness of the physical spring, and isolated stability is obtained for the entire range of stiffness values considered.

1 INTRODUCTION

It is widely agreed that a deeper understanding of the neural effects of movement therapy after neurological injury is necessary to develop more effective rehabilitation training programs [1]. In this context, robot-aided rehabilitation [2] has been suc-

cessful in introducing standardization and repeatability to movement rehabilitation techniques, paving the way for the implementation of novel and neuroscience-based rehabilitation protocols. Although it is more or less clear which interaction modalities do not contribute to recovery [3–5], a full identification of the exact therapy modalities that allow a given subject or subject group to recover more effectively is far from having been accomplished. A major barrier towards the development of more effective therapies is our lack of understanding of the exact processes that underlie plasticity and recovery promoted by rehabilitation after neurological injury. Neuroimaging techniques such as fMRI (functional Magnetic Resonance Imaging) offer promise to shed light on such aspects, representing an appealing opportunity to study treatment-effect relationship of robot-aided neurorecovery. However, the same standardization and reproducibility of motor performance recently obtained for the therapy (treatment delivery) phase is far from having been achieved in motor protocols in MRI environments (effect measurement). This problem is mainly due to the difficulty in introducing standard robotic technologies that allow the accurate and systematic measurement and/or assistance of human movements.

In fact, MR compatibility introduces significant technological difficulties. Materials most often used in conventional robotic and mechatronic systems have ferromagnetic characteristics due to the need for mechanical properties such as strength, rigidity, and machinability [6]. Most importantly, MR-compatible robots have to address the lack of suitable off-the-shelf actuation and measurement technologies. The commonly used electromagnetic actuators are intrinsically not MR-compatible due to their principle of operation. Fluidic actuation is instead intrinsically MRI compatible: hydraulic power can be transferred

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through long hoses, thus allowing the power source (pump) to be placed outside the scanning room [7]. Pneumatic systems are mainly suitable for relatively low-force applications and they have limited stiffness and force regulation bandwidth [8]. Non-conventional actuation systems such as electrorheological fluids (ERFs) have provided an alternative way for generating resistive forces inside a MR scanner [9, 10] and recently active ERF devices have also been proposed [11]. Among the systems developed so far, a promising actuation approach is represented by UltraSonic Motors (USMs), featuring intrinsic magnetic immunity, bidirectionality, high torque-to-weight ratio, small size, and compact shape [12, 13]. However, USMs cannot be directly employed for interaction control, due to their high intrinsic output impedance, which makes them non-backdrivable.

This research aims at overcoming the mentioned limitations in interaction control of MR-compatible actuators, by developing a novel MR actuation architecture for MRI environments. The proposed architecture is the paraphrases of the SEA concept [14, 15] to the specific needs of interaction control in movement protocols for fMRI, and consists of the interposition of a compliant force-sensing element in series between a MRI compatible and non-backdrivable actuator and the output. The proposed solution enables the implementation of force-feedback controllers with non-backdrivable actuators through the measurement of macroscopic deflections of the compliant elements, that can be performed via sensors commercially available in MR-compatible versions, such as optical encoders.

2 SYSTEM DESCRIPTION AND MODELING

2.1 MR-compatible linear SEA

The design of a linear MR-SEA prototype is shown in Fig. 1. Its design is based on our recent work [16] in the design of a linear SEA for the compliant version of a wrist exoskeleton, the RiceWrist, clinically used for movement rehabilitation therapy after stroke. The parallel portion of the RiceWrist [17], a serial-in-parallel, 3 RPS (Revolute Prismatic Serial), four degree-of-freedom (DOF) robotic manipulator is designed to provide support to wrist movements during rehabilitation. When worn by a subject, its DOFs decrease to three (translation along the radial axis is ideally prevented by the anatomy of the wrist joint) and supports wrist movements in a range of motion compatible with that of the wrist joint during basic activities. The kinematic structure of the RiceWrist is well suited for the introduction of compliance, i.e. by introducing three linear bidirectional springs in series to the actuators, allowing to reproduce a configuration-dependent intrinsic stiffness on the output. In our previous study [16], we derived the basic specifications for a linear SEA to be included in a 3RPS parallel exoskeleton for wrist rehabilitation, that we report in the following since they have been used also for the present design:

- 20 N of continuous force
- 6 kN/m intrinsic stiffness
- 0.1 N of force measurement resolution / linearity error

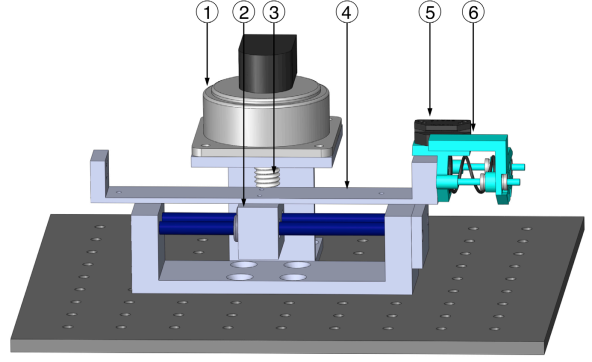


Figure 1. CAD design of the MR-SEA prototype under development. Motor (1) rotation is converted to a translational motion by a cable transmission (cable not shown). The cable is wrapped around the spool (3), and connected at both extremities of a slider (4), supported by two linear bearings (2). A custom compliant element (6) composed of the parallel connection of a compression and an extension spring (not shown) is interposed in series; its deflection is measured through a linear encoder (5).

- 60 mm of range of linear motion
- 800 mm/s maximum velocity

The desired performance is achieved with the design shown in Fig. 1, which includes an ultrasonic motor from SHINSEI, model USR60-E3N, with an integrated 1000 cpt optical encoder, which has been shown to be MR-compatible with proper use of RF filters and shielded cables in a 7 T environment [18]. The motor is non back-driveable owing to its principle of operation; however the manufacturer gives the option of using the motor in conjunction with a velocity controller, which uses the motor position feedback to generate a switching voltage suitable to implement velocity control in a range of velocities as shown in the diagram in Fig. 2.

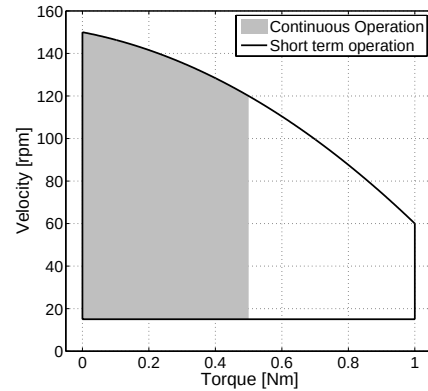


Figure 2. Torque vs. velocity curve for the ultrasonic motor considered. From the manufacturer's catalogue.

A brass cable is wrapped around a threaded aluminum spool (pitch diameter 12 mm) fixed on motor shaft, driving a slider supported with two DryLin R Adjustable Pillow Blocks (Igus GmbH, Köln, Germany), thus converting rotary motion to a translational DOF. A custom compliant element, similar to the one presented in [16], is mounted in series to the slider. The compliant element includes the parallel connection of two linear titanium springs, an extension spring and a compression spring, constituting a bidirectional compliant element, with stiffness $k_s=6$ kN/m, supported by linear bearings to constrain motion to a purely translational DOF. An incremental encoder is mounted in parallel to the spring (combination of the US Digital EM1-0-500-I reading head and LIN-500 strip, already tested for MR-compatibility [19]), allowing for the measurement of spring deflection with a resolution of 0.01 mm (corresponding to 0.06 N of force resolution).

2.2 System modeling and control architecture

The basic model for a Series Elastic actuator in the mechanical domain is presented in Fig. 3. It includes an electrical motor, that can be current controlled to be an ideal force source, driving the output mass through a spring/mass/damper system. The basic differential equation for this system is

$$m_M \ddot{x}_M + b_M \dot{x}_M = F_M - F_L = F_M - k_s(x_M - x_L), \quad (1)$$

with x_M and x_L representing motor and load displacements, respectively, m_M the reflected inertia of the motor through the transmission (the hypothesis of rigid transmission is considered), b_M is the linear coefficient of viscous friction of the geared motor seen at the spring port, and k_s the linear spring stiffness.

Several control approaches have been proposed for torque control of SEAs, including direct feedback force controllers with linear feedforward compensation terms [20], nonlinear compensators to reduce the effects of friction and variability of interaction dynamics [21] and the application of cascaded linear force/torque and position [22] or velocity control [23–25]. Among the mentioned controllers, the last two are of particular

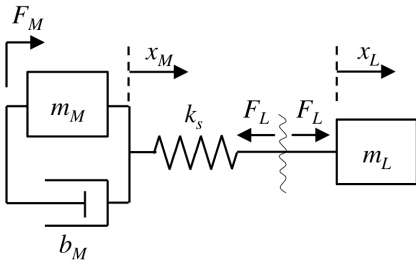


Figure 3. Mechanical schematic of a Series Elastic Actuator. A motor is connected to the load through a spring, whose deflection is measured, thus allowing the measurement of the interaction force F_L .

interest, since they effectively allow the conversion of a force control problem into simpler position or velocity control problems and also allow the application of interaction controllers using non-backdriveable actuators, without requiring a detailed knowledge of system parameters. Moreover, the cascaded force and velocity control scheme was demonstrated to be passive [25], for a wide range of controller gains that do not significantly compromise the torque regulation performance. This holds true even in the absence of viscous friction b_M , which is not generally the case of direct force/torque feedback control [26]. Due to its inherent robustness to nonlinearities introduced by transmissions or actuation in SEAs, the cascaded force and velocity control scheme was selected for this paper. Fig. 4 reports a linear block diagram of this controller applied to the system described in Fig. 3 in the Laplace domain, with C_V and C_F being the transfer function of linear controllers (usually two PI controllers) acting on the velocity and force error, respectively.

This control scheme can be simplified using linear systems theory. To this aim, the inner velocity loop can be modeled by the superposition of two contributions, one describing velocity control performance, and the other one describing the degradation of velocity control due to the interaction with the environment:

$$V_M(s) = H_v(s)V_{des}(s) + D_v(s)F_L(s), \quad (2)$$

where H_v is the velocity control closed loop transfer function, in the absence of torque disturbance ($H_v = C_v G / (1 + C_v G)$), D_v describes the effect of torque disturbance on velocity control output ($D_v = -G / (1 + G_v G)$), and G is the plant subject to velocity control, a mass-damper system ($G = 1 / (m_M s + b_M)$). The simplified block diagram in the Laplace domain is shown in Fig. 5. In the specific case of the selected ultrasonic motor, the relation between torque applied from the output and resulting velocity is highly nonlinear, due to its high intrinsic mechanical impedance introduced by its principle of operation, based on friction. Moreover, the motor is actually shipped with a control driver which implements a feedback velocity controller, based on the velocity estimated through differentiation of the measured position measured through incremental encoders.

An approximated model of the velocity controlled motor will now be introduced. Considering the high intrinsic mechanical impedance of the ultrasonic motor, and assuming a perfect

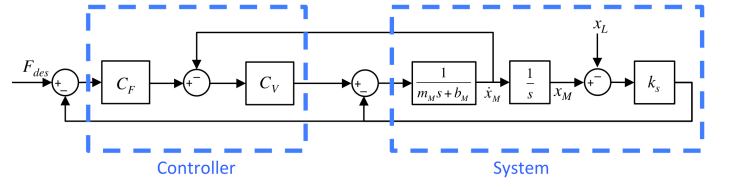


Figure 4. Linear block diagram of a SEA subject to a cascaded force-velocity control.

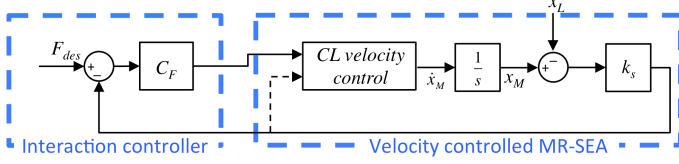


Figure 5. Block diagram of the proposed controller for the MR-SEA. A non-backdriveable motor is controlled to be a velocity source, and an outer loop is closed on the measured force of interaction with the environment. A velocity control disturbance transfer function can be defined to describe the effect of interaction force on error in velocity control.

disturbance rejection from the velocity controller, we hypothesize that the term $D_v(s)F_L(s)$ is negligible compared to the first term in (2), implying no effect deriving from load torque on the motor velocity control performance.

Under the mentioned assumptions, the velocity-controlled ultrasonic motor has been modelled as the series of a non-linear block, which takes into account the range of controllable velocities, and a low-pass filter, which takes into account the time required to change velocity from a current value to a new specified value. The manufacturer declared that velocity regulation is achieved within a rise time of around 6 ms in the admissible velocity range (see Fig. 2). Therefore, the time constant of the low-pass filter used for simulation was set to 10 ms, which corresponds to a 15.9 Hz cut-off frequency, conservative to the parameter declared by the manufacturer. The non-linear block implements the following discontinuous function between the input variable v and the output variable $f(v)$:

$$f(v) = \begin{cases} 0 & \text{if } |v| < v_{min} \\ v & \text{if } v_{min} < |v| \leq v_{max} \\ v_{max}(F_L) & \text{if } |v| > v_{max}(F_L) \end{cases} \quad (3)$$

Fig. 6 shows the effect of the application of a proportional force control to the system modeled as in Fig. 5. By setting proportional control gains in the range that guarantees passivity, the low force regulation performance is degraded, due to the small-velocity dead band present in the plant. In order to avoid such effects, a nonlinear compensation scheme (Fig. 7) will be developed and tested in the followings sections.

3 CONTROL

In order to deal with the non-linearities introduced, a discontinuous force feedback control law $v(\cdot)$ is defined. Force-feedback control will be described for the unperturbed system ($x_L = 0$), in the case of a constant desired force $\dot{F}_{des} = 0$ and its stability proved for this condition. In order to simplify the stability analysis, the function $f(u)$ will be approximated neglecting the dependence between motor velocity saturation and applied force, using a conservative value for $v_{max} = v'_{max}$. The

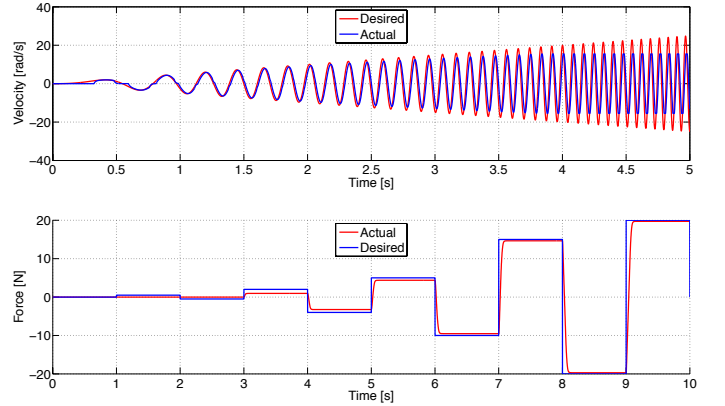


Figure 6. (top) USM velocity control performance, subject to a desired velocity command $v_{des}(t) = 5t \sin(2\pi(0.1 + 3t))$. The effects of dead band at low velocities, attenuation at higher frequencies and saturation at higher velocities are visible. (bottom) Linear cascaded force and velocity control of the system in Fig. 7, with $g(e, v_{min}) = Kp$. The effect of steady state error and low force inaccuracies can be observed.

developed control will be also analyzed during interaction using the numerical simulations described in the later sections. The state equations for the unperturbed system subject to the nonlinear feedback control law $v(x_1)$ can be obtained by defining the state vector $\mathbf{x} = [x_1, x_2]^T$, with state variables $x_1 = (F_L - F_{des})/k_s$ and $x_2 = \dot{x}_M$ as:

$$\dot{\mathbf{x}} = \mathbf{f}(\mathbf{x}) = \begin{bmatrix} x_2 \\ -\frac{x_2}{T} - \frac{v(x_1)}{T} \end{bmatrix} \quad (4)$$

The control input $v(x_1)$ is chosen as

$$v(x_1) = g(x_1)\text{sign}(x_1) \quad (5)$$

using the absolutely continuous function $g(x_1)$, defined as:

$$g(x_1) = \begin{cases} v_{min}, & \text{if } |K_p x_1| \leq v_{min} \\ K_p |x_1|, & \text{if } v_{min} < |K_p x_1| \leq v_{max} \\ v_{max}, & \text{if } |K_p x_1| > v_{max} \end{cases} \quad (6)$$

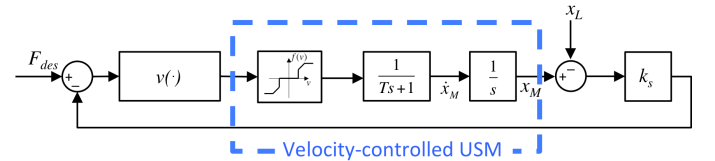


Figure 7. MR-SEA simulation block diagram. The velocity-controlled motor is modeled as the series of a saturation block and a low-pass filter. A non-linear compensator is included to account for the non-linear plant.

3.1 Stability analysis

We are interested in demonstrating that the application of the controller in (5) to the system (4) results in a globally asymptotically stable dynamical system, with equilibrium point $\mathbf{x} = \mathbf{0}$, demonstrating the torque regulation capabilities for any constant F_{des} .

Since the right hand side of (4) is discontinuous, existence and uniqueness of its solutions cannot be commented on in the conventional sense. We consider the solutions $\mathbf{x}$ in the sense of Fillipov [27], which are absolutely continuous and

$$\dot{\mathbf{x}} \in \mathcal{F}(\mathbf{x}). \quad (7)$$

Thus by considering Fillipov solutions, we replace the differential equation (4) by the differential inclusion (7). A differential inclusion specifies that the state derivative belongs to a *set of directions* instead of a single direction. We will use Lyapunov stability theorem based on Fillipov's differential inclusion and Lie derivatives as given by Cortés in [28] to prove convergence of the solution trajectories. We will first reproduce the definitions of the Fillipov set-valued map and set-valued Lie derivative from [28].

Definition 3.1 (Fillipov set-valued map): For $\mathbf{f}(\mathbf{x}) : \mathbb{R}^d \rightarrow \mathbb{R}^d$, the Fillipov set-valued map $\mathcal{F}(\mathbf{x}) : \mathbb{R}^d \rightarrow \mathfrak{B}(\mathbb{R}^d)$ is given by

$$\mathcal{F}(\mathbf{x}) \triangleq \bigcap_{\delta > 0} \bigcap_{\mu(S)=0} \overline{\text{co}}\{\mathbf{f}(\mathbf{B}(\mathbf{x}, \delta) \setminus S)\}, \quad \mathbf{x} \in \mathbb{R}^d. \quad (8)$$

Here $\overline{\text{co}}$ denotes convex closure, μ denotes Lebesgue measure, and $\setminus$ denotes the relative complement operator. S is the set of points where the vector field $\mathbf{f}(\mathbf{x})$ is discontinuous. $\mathfrak{B}(\mathbb{R}^d)$ denotes the set whose elements are all of the possible subsets of $\mathbb{R}^d$, and $\mathbf{B}(\mathbf{x}, \delta)$ is a ball with radius δ and centered at $\mathbf{x}$.

Definition 3.2 (Set-valued Lie derivative): Given a locally Lipschitz function $V(\mathbf{x}) : \mathbb{R}^d \rightarrow \mathbb{R}$ and a set-valued map $\mathcal{F}(\mathbf{x}) : \mathbb{R}^d \rightarrow \mathfrak{B}(\mathbb{R}^d)$, the set-valued Lie derivative $\tilde{L}_{\mathcal{F}}V : \mathbb{R}^d \rightarrow \mathfrak{B}(\mathbb{R})$ of V with respect to $\mathcal{F}$ at $\mathbf{x}$ is defined as

$$\tilde{L}_{\mathcal{F}}V(\mathbf{x}) = \{a \in \mathbb{R} : \text{there exists } \mathbf{v} \in \mathcal{F}(\mathbf{x}) \text{ such that } \zeta^T \mathbf{v} = a \text{ for all } \zeta \in \partial V(\mathbf{x})\} \quad (9)$$

At each $\mathbf{x} \in \mathbb{R}^d$, the set-valued Lie derivative $\tilde{L}_{\mathcal{F}}V(\mathbf{x})$ is a set contained in $\mathbb{R}$. For the empty set, we adopt the convention $\max \emptyset = -\infty$. It can be observed that the set-valued Lie derivative is a generalized notion of derivative for capturing the evolution of a function $V(\mathbf{x})$ along the trajectories of a discontinuous vector field, and it breaks down to the derivative in conventional sense when the vector field is continuous and $V(\mathbf{x})$ is continuously differentiable.

We will apply the Theorem 1 in [28] to show that the function

$$V(X) = x_2^2 + \frac{2}{T} \int_0^{x_1} g(\sigma_1) \text{sign}(\sigma_1) d\sigma_1 \quad (10)$$

is a *strong* Lyapunov function and prove global asymptotic convergence of the solution trajectories of (7) to origin.

Theorem 3.1: For $K_p > 0$ and $T > 0$, all trajectories of the system (7) converge asymptotically to origin $\mathbf{x} = \mathbf{0}$. Moreover, $V(\mathbf{x})$ as defined in (10) is a strong Lyapunov function assuring this property.

Proof: $V(\mathbf{x})$ as defined in (10) can be rewritten as

$$V(\mathbf{x}) = x_2^2 + \frac{2}{T} h(x_1) \quad (11)$$

where

$$h(x_1) = \int_0^{x_1} g(\sigma_1) \text{sign}(\sigma_1) d\sigma_1 = \begin{cases} v_{min}|x_1|, & \text{if } |K_p x_1| \leq v_{min} \\ \frac{1}{2} \frac{v_{min}^2}{K_p} + \frac{1}{2} K_p x_1^2, & \text{if } v_{min} < |K_p x_1| \leq v_{max} \\ v_{max}|x_1| + \frac{v_{min}^2 - v_{max}^2}{2K_p}, & \text{if } |K_p x_1| > v_{max} \end{cases} \quad (12)$$

By inspection, $V(\mathbf{x})$ is locally Lipschitz and regular on $\mathbb{R}$. Also, $V(0) = 0$, and $V(\mathbf{x}) > 0$ for $\mathbf{x} \in \mathbb{R} \setminus \{0\}$.

We then go through the steps needed to compute the Lie derivative as outlined in [28], using the definition given in (9). We first compute the Fillipov set-valued map $\mathcal{F}(\mathbf{x})$ for the system (4) using the calculus given in [29] as

$$\mathcal{F}(\mathbf{x}) = \begin{cases} [x_2 \quad (-\frac{x_2}{T} - \frac{g}{T})]^T, & \text{if } x_1 > 0 \\ [x_2 \quad (-\frac{x_2}{T} - d\frac{g}{T})]^T : d \in [-1, 1], & \text{if } x_1 = 0 \\ [x_2 \quad (-\frac{x_2}{T} + \frac{g}{T})]^T, & \text{if } x_1 < 0 \end{cases} \quad (13)$$

and then compute the generalized gradient ∂V ([30]):

$$\partial V = \begin{cases} [\frac{2}{T}g \quad 2x_2]^T, & x_1 > 0 \\ [\frac{2}{T}dg \quad 2x_2]^T : d \in [-1, 1], & x_1 = 0 \\ [-\frac{2}{T}g \quad 2x_2]^T, & x_1 < 0 \end{cases} \quad (14)$$

With (13) and (14), we can write the set-valued Lie derivative $\tilde{L}_{\mathcal{F}}V(X) : \mathbb{R}^2 \rightarrow \mathfrak{B}(\mathbb{R})$ as

$$\tilde{L}_{\mathcal{F}}V(\mathbf{x}) = \begin{cases} -\frac{2x_2^2}{T}, & x_1 \neq 0 \text{ and } x_2 \neq 0 \\ \emptyset, & x_1 = 0 \\ 0, & x_1 = 0 \text{ and } x_2 = 0 \end{cases} \quad (15)$$

It can be observed that $\max \tilde{L}_{\mathcal{F}} V(\mathbf{x}) < 0$ for each $\mathbf{x} \in \mathbb{R}^2 \setminus \{\mathbf{0}\}$. Thus $V(\mathbf{x})$ satisfies all the hypotheses of Theorem 1 in [28], and is a strong Lyapunov function. Hence we conclude that $\mathbf{x} = \mathbf{0}$ is a strong globally asymptotically stable equilibrium of (7). ■

4 SIMULATION RESULTS

A numerical dynamical model of the system has been implemented in Simulink (The Mathworks Inc.) to simulate the evolution of the system shown in Fig. 7 and described by equations (3), (5-6). Model parameters have been set in order to match the properties of the system described in Sec. 2.1, with the parameters reported in Table 1, using a fixed-step Runge-Kutta solver, with a sampling time $\Delta T = 1$ ms. In order to avoid chattering introduced by the switching control signal action (5), the discontinuity introduced by the signum operator was approximated using the arctan function, and tuning the scaling factor to regulate steady-state error values to less than 1% of desired value.

4.1 Force control

Force control performance of the developed SEA was characterized in blocked output conditions (i.e., $x_L = 0$). In this configuration, different metrics were considered to measure the performance of the torque controller, as described below.

4.1.1 Step response Force controller gains were regulated based on the system response to a step commanded torque, in order to have an overshoot lower than 0.5 N for all F_{des} in the admissible force control range. Fig. 8 shows the response of the system to commanded steps with different amplitudes. The system responds with a 2% overshoot and with rise time of 75 ms for the largest commanded input (transition from -20 N to 20 N). In the low-force range, for $F_{des} = 0.5$ N, a rise time of 9 ms is achieved, with an overshoot of 0.12 N, which represents 24% of the desired value. Despite being significant in percentage terms, this value is considered acceptable in absolute terms for the required application, since it represents only 0.6% of maximum force output.

Table 1. MODEL PARAMETERS

Symbol	Value	Description
T	0.01 s	Velocity inner loop time constant
K_p	2 Ns/m	Force control gain
k_s	6 kN/m	Spring constant
v_{min}	8.2 mm/s	Minimum motor velocity
v_{max}	$a - bT_L - cT_L^2$ [mm/s] $a = 85, b = 49, c = 30$	Maximum motor velocity

4.1.2 Force control bandwidth Force control performance is evaluated in the frequency domain by estimating the transfer function between the desired force F_{des} and the actual force F_L applied by the series elastic actuator, when the output is blocked (again, $x_L = 0$). In the frequency domain, it can be written as $G_F(f) = F_L(f)/F_{des}(f)$, $F_L(f)$ and $F_{des}(f)$ being the Fourier transforms of F_L and F_{des} respectively. The transfer function $G_F(f)$ has been estimated using a non-parametric system identification method via the Welch's method, using the following general relation:

$$\hat{G}(f) = \frac{P_{yu}(f)}{P_{uu}(f)}, \quad (16)$$

with $P_{uy}(f)$ the cross-spectral density and $P_{uu}(f)$ and $P_{yy}(f)$ the auto-spectral density of the input (u) and output (y) signals respectively. To estimate force control bandwidth, the input signal was the desired force F_{des} , while the output signal was the resulting force measured by the spring F_L . In transfer function estimation, a coherence function $Coh(f)$ can be defined as:

$$Coh(f) = \frac{|P_{uy}(f)|^2}{P_{uu}(f)P_{yy}(f)}. \quad (17)$$

Values of $Coh(f)$ close to one indicate coherence between the input and output signals, thus demonstrating a good agreement of the linear model underlying the estimation method. In order to determine the transfer function $G_F(f)$, desired force signals $F_{des}(t)$ corresponding to Schroeder multisines, with a flat power spectral density between 0.05 and 40 Hz, negligible power content above 40 Hz, and different amplitudes, ranging from 0.5 N (5% of maximum force) to 20 N (peak force value) were applied to the system. Fig. 9 shows the performance of the simulated force controller, for different values of peak desired force,

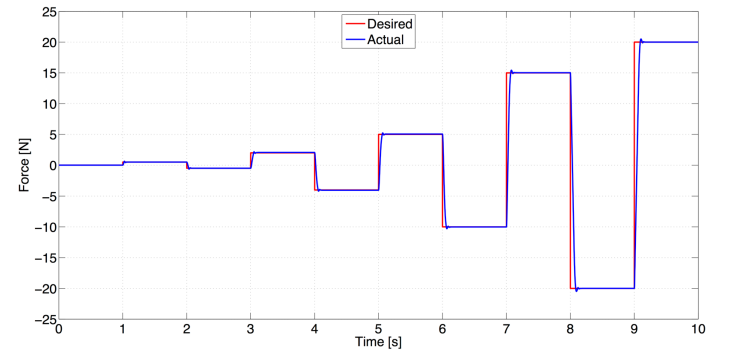


Figure 8. Step response of the system subject to the nonlinear control action. The steady-state error is less than 1% of desired value in all conditions; the maximum overshoot of 0.5 N (highest absolute error in the 20 N case).

demonstrating a *small-force* bandwidth that depends on the desired force values (ranging from 21 Hz for the 0.5 N signal to 10 Hz for the 15 N signal), and a *large-force* bandwidth of 8 Hz, determined by motor velocity saturation. In the hypothesis of sinusoidal force tracking in blocked output conditions, with a desired force $F_{des}(t) = F_d \sin(2\pi f_0 t)$, the resulting actuator motion is $x_{M,des}(t) = F_{des}(t)/k_s$, and the corresponding required velocity $\dot{x}_{M,des}$ is again a sinusoidal function, with amplitude $V_0 = 2\pi f_0 F_{des}/k_s$, and frequency f_0 . Considering the maximum rotational speed of the motor, and the chosen transmission ratio for the spool-cable transmission, the maximum linear velocity of the drive equals $8.5 \cdot 10^{-2}$ m/s, which is 55% of the maximum required velocity for a tracking of a sinusoidal force with amplitude 20 N and frequency 8 Hz, thus demonstrating the attenuation at 3 dB observed at that frequency, fully attributable to motor velocity saturation.

4.1.3 Zero-force control The behavior during interaction has been evaluated simulating the application of a position disturbance from the load side x_L and commanding the actuator to regulate a zero interaction force. This control mode is intended to guarantee transparency of the device during evaluation mode, where the user backdrives the device and therefore the apparent mechanical impedance should be minimized. To simulate interaction, a set of Schroeder multisine signals has been applied from the load side to the model. The amplitude of the signals has been scaled to provide perturbation forces on the spring F_p (when $x_M=0$, i.e. blocked motor conditions), comparable with the forces deliverable by the actuator (in the range [0 - 20] N).

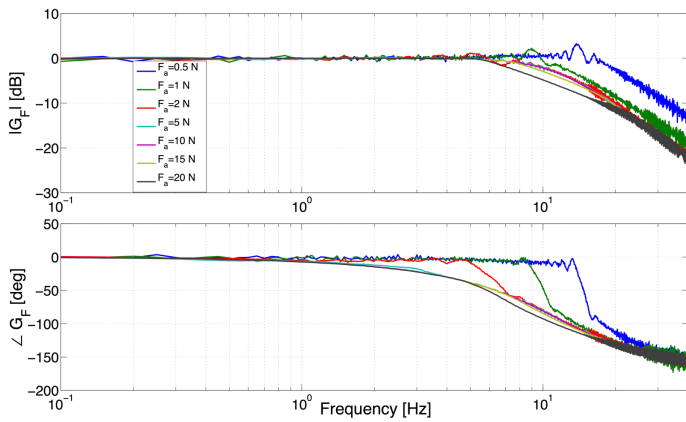


Figure 9. Bode diagram of force control transfer function $G_F(f)$, for different values of $F_{des}(t)$. The desired force signal is a Schroeder multisine, with reported peak amplitude (RMS value: 0.545 peak torque), and flat frequency spectrum in the range [0.05 50] Hz. Force control bandwidth is a function of desired force amplitude, due to velocity saturation effects. Results for cases reported in the plot range from 8 Hz (for a 20 N peak force) to 20 Hz (for the minimum considered force of 0.5 N). The maximum overshoot obtained in the low force range is of only 0.4 N.

The frequency range of the applied perturbation was [0.05 20] Hz, well above those achievable during human interaction. The mechanical impedance transfer function in zero force control mode $Z_0(f) = F_0(f)/\Omega(f)$ has been estimated using (17), considering the applied velocity $\dot{x}_L$ as input signal (Fourier transform $\Omega(f)$) and the measured force as output signal (Fourier transform $F_0(f)$).

The resulting transfer function shows that in these conditions the system mainly behaves as a linear viscous damper in the controlled bandwidth, and reduces to a spring above the controllable bandwidth (i.e. for frequencies higher than 8 Hz for all amplitudes, and higher for smaller perturbations). The apparent damping coefficient equals 100 Ns/m for the high force perturbation conditions (perturbation force higher than 2 N), while is reduced to 10 Ns/m for the small force perturbation conditions. Results are consistent also with other simulations performed applying constant speed perturbation with different amplitude, showing that the damping coefficient at high speeds is higher than that for low speeds, mainly due to the higher relative overshoot of the torque controller at small amplitudes (compare with Fig. 8). The used estimation method provides low (<0.9) coherence values for low frequency (<2 Hz), low force (<2 N) perturbations, implying inaccuracies in the transfer function estimation, mainly due to the non-linearities of the force resulting from imposed motion in the low-force ($|F| < 2$ N) range.

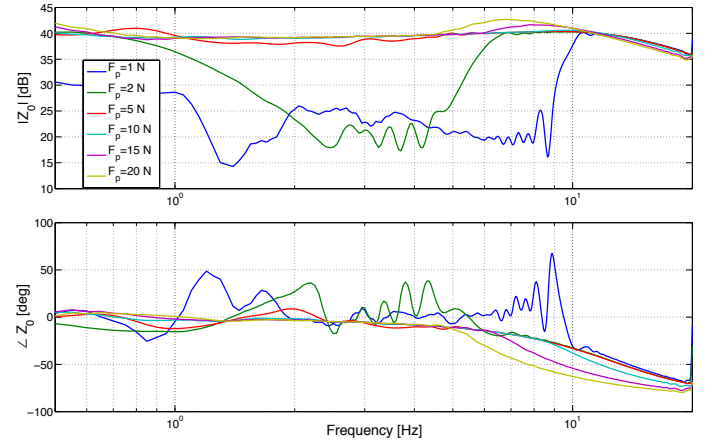


Figure 10. Bode plot of the mechanical impedance transfer function Z_0 , during the zero-force interaction condition. In the simulated interaction conditions, a displacement x_L was applied from the output. The applied displacement was a Schroeder multiline signal, with amplitude scaled to provide a perturbation force on the spring F_p (in $x_M=0$ conditions, blocked motor), as reported in the legend. The apparent damping equals 100 Ns/m for the high velocity perturbations, while it is reduced to a minimum of 10 Ns/m for small velocity perturbations.

4.2 Stiffness control

An outer control loop has been implemented on the controller shown in Fig. 5, to regulate interaction with a virtual environment. The interaction controller is a force-feedback stiffness controller, which implements a relation between applied motion and resulting force by specifying a new value of desired force as:

$$F_{des}(t) = -k_v[x_L(t) - x_{L,d}(t)], \quad (18)$$

where k_v is the desired virtual stiffness coefficient, $x_{L,d}$ is a desired position and x_L is the measured position. The performance of the force-feedback stiffness controller was evaluated for different values of desired virtual stiffness k_v , and for position perturbations of variable amplitude, to assess stability, accuracy and effect of nonlinearities. In all conditions, the mechanical impedance transfer function $Z_k(f)$ was evaluated, defined as the ratio between $F_L(f)$ and applied velocity. The applied perturbation is a position perturbation constructed as a Schroeder multiline signal, with amplitude scaled to provide a perturbation force on the spring F_p within the admissible range and flat power spectral density in the range [0.05 - 50] Hz. The Bode plots of the estimated apparent impedance for the different conditions are reported in Fig. 11. For virtual stiffness values lower or equal to that of the physical stiffness, the system is passive in all considered conditions (i.e. the system is stable and its impedance transfer function has a positive real part [31]). At high frequencies, system behavior reduces to a spring, with stiffness equal to that of the physical spring, while in the controlled range, accurate stiffness control is achieved. In the case of $k_v = 2k_s$, the system is non passive and can be potentially unstable coupled with some passive environments [31], in agreement with previous results on the cascaded force-velocity control of SEAs [32]. However, in the reported idealized case, isolated stability of the system subject to position perturbation is achieved.

Due to plant nonlinearities, the bandwidth of stiffness control depends on the amplitude of the perturbation and on the specific value of desired virtual stiffness. We can define as in [33] the bandwidth of stiffness control f_k as:

$$f_k : \left| \frac{K(f) - k_v}{k_v} \right| = \sqrt{2}/2, \quad (19)$$

where $K(f)$ is a stiffness transfer function, estimated from the input position x_L to the output force (the corresponding Bode plots are reported in Fig. 12). Stiffness control bandwidth is maximized when $k_v = k_s$ (unsurprisingly, considering the modeled actuator non backdriveability). Moreover, stiffness control performance deteriorates when k_v decreases, obtaining a worst-case bandwidth of 7 Hz for maximum perturbation, and $k_v = 0.2k_s$. Velocity saturation is instead the limiting factor for the case $k_v = 2k_s$, thus limiting stiffness control bandwidth to a lower value of 6 Hz, for the highest perturbation amplitude considered.

5 CONCLUSIONS AND FUTURE WORK

This paper describes a novel actuation architecture suitable to implement interaction control in MRI environments, thus allowing performing continuous movement protocols during fMRI in both assisted mode and transparent mode. The developed architecture consists of the combination of a non-backdriveable MR-compatible actuator with a compliant force-sensing element.

The mechatronic design of a 1 DOF actuator is presented, including a description of the main performance limitations introduced by MR-compatibility constraints, followed by a non-linear dynamical model of the specific actuator, which includes the most relevant actuator non linearities, namely velocity dead-

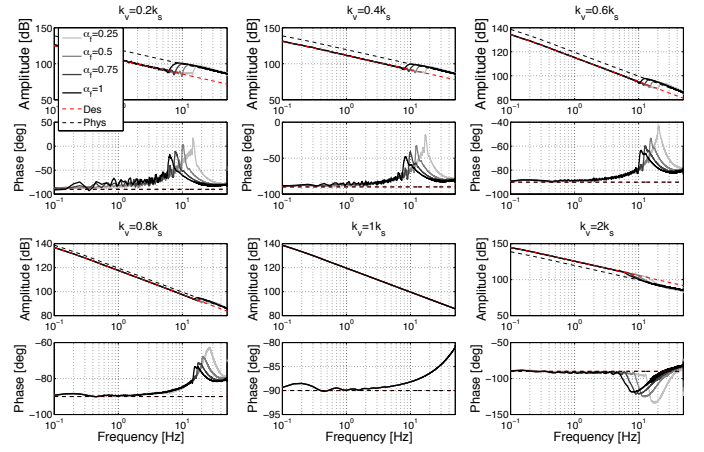


Figure 11. Apparent impedance of the system subject to stiffness control, for different values of virtual stiffness k_v (different Bode plots are reported for the considered values of k_v) and of position perturbation input $x_{L,p}$, scaled such as $k_s x_{L,p} = \alpha_f F_{max}$ (plots are overlaid in every window). The applied position $x_{L,p}$ is a Schroeder multiline signal, with flat spectral power density in the range [0.05 - 50] Hz.

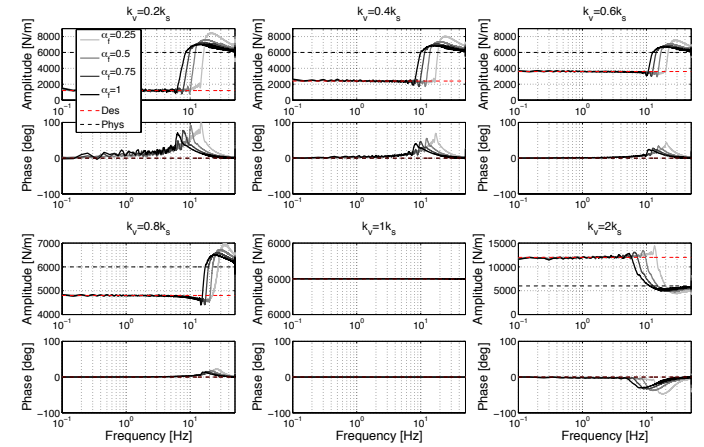


Figure 12. Apparent stiffness of the system subject to stiffness control, for different values of virtual stiffness k_v and of perturbation amplitude.

band and saturation. A non linear controller is developed to exploit the possibility of controlling the MR-compatible motor as a velocity source, and suitable to compensate the nonlinearity of the motor. A detailed stability analysis for force control in blocked output conditions is included, demonstrating that global asymptotic stability is achieved for every desired force in the admissible range, determined by motor mechanical power limitations.

Simulation results quantify the performance of force control, demonstrating a bandwidth of 8 to 21 Hz, largely compatible with slow and limited-extent motor tasks implementable in MRI environments. Furthermore, the possibility of rendering virtual environments with variable stiffness properties is investigated. Simulation results demonstrate that the proposed controller provides isolated stability for all considered values of required stiffness K_v in the range $[0, 2k_s]$, with k_s being the stiffness of the physical spring, and passive behavior for $K_v \leq k_s$, in agreement with similar linear controllers applicable to non MR-compatible and back-driveable motors.

Future work will involve fabrication of the described prototype, and the detailed evaluation of MR-compatibility of the whole mechatronic system. A more accurate model of the velocity-sourced actuator, including the disturbance transfer function $D_v(s)$, will be developed using system identification methods with different values of desired velocity, to account for possible non-linearities in the velocity-control plant. This system identification process will validate the modeling approach used in this paper, or motivate the development of refined controllers, e.g. including integral action and/or compensators to account for the loading effect on the actuator introduced by interaction with the environment. Finally, the interaction controller presented will be implemented in the developed hardware and its performance compared to the simulation results presented in this paper.

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