A Fast Autotuning Method for Velocity Control of Mechatronic Systems

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Abstract: In this paper a fast automatic tuning methodology for velocity controllers of mechatronic systems is proposed. In order to be applicable in general, the method takes into account the position, velocity and torque constraints of the motion control system and it requires a minimum intervention of the operator. Further, it can be implemented also with small computational capabilities which makes it suitable for industrial drives. Simulation results show the effectiveness of the technique.

Keywords: PID controllers, autotuning, mechatronic systems.

1. INTRODUCTION

While automatic tuning methodologies for proportional-integral-derivative (PID) controllers have been widely developed and applied in process control applications (Åström and Hägglund (2006)), the subject needs further developments for industrial drives applied to motion control of mechatronic systems in the context of industrial applications like, for example, hoist and crane systems or injection molding machines (Choi and Chung, 2004).

In fact, when an automatic tuning method is applied to a mechatronic system, it has to fulfill some specific requirements, in addition to those that are typically taken into account also in process control. In particular, depending on the machine on which the autotuning procedure is applied, position, speed and torque/acceleration constraints have to be considered. Indeed, each motor has its own speed and torque limits but they could be further restricted by the specific application, in order to avoid possible damages of the mechanical structure. Also the jerk should be limited as much as possible, because the use of discontinuous accelerations may induce vibrations, which, in general, may reduce the lifespan of the machine. Then, there are peculiar parameters of the dynamic model to estimate (upon which the PID controller can be tuned), such as inertia, static and dynamic friction, elasticities, and so on.

In any case, as in process control, it has to be taken into account that the automatic tuning method should have those characteristics that are very appreciated in an industrial framework. Namely, it has to require a minimum intervention of the operator (who, ideally, should not to take any decision in the procedure) and its duration should be as short as possible in order to avoid to decrease the productivity, to minimize the energy consumption and to avoid possible waste of material (whose production could not be of the required quality when the autotuning procedure is performed).

In this context, different methodologies have been proposed in recent years. In particular, by considering the speed control task, the application of pseudo random binary signal (PRBS) or a slightly more sophisticated stepwise signal to the torque have been proposed in order to estimate the frequency response of the system and, most of all, the presence of resonances (Villwock and Pacas, 2008; Weissbacher et al., 2013; Beineke et al., 1997). This can be done by applying, for example, a Fast Fourier Transform (FFT) or the Welch method (Villwock and Pacas, 2008) to the input and output signals of the system. Note that the identification experiment can be performed either in open loop or in closed loop. In the latter case, of course, an already tuned (PID) controller has to be in place which might be a disadvantage if the commissioning phase is at its very first stage. In any case, as already mentioned, the presence of many steps in the torque signal might not be suitable for the machine as high-frequency vibrations can be induced (even if the excitation of the high-frequency dynamics is actually the purpose of the method). Then, the PRBS signal has to be properly designed.

An alternative method can be the use of a swept sine wave. In (Goubej et al., 2013; Goubej, 2015), a methodology has been proposed in order to effectively deal with measurement noise and nonlinearities of the system. The main disadvantage of this technique is that its duration can be excessively high.

A very interesting two-stage approach has been recently proposed in (Calvini et al., 2015). First, the overall inertia, and the static and viscous friction coefficients are estimated by using a filtered PRBS signal (note the filter bandwidth has been set arbitrarily to 10 Hz). Then, a PRBS signal or simply a torque pulse (depending on the presence of backlash) is applied to estimate the high-frequency parameters. However, in this case position, speed and torque limits are not taken into account explicitly.

Once a model of the system has been obtained, the PID controller can be tuned. In this context, the presence of resonant frequencies (mainly due to the presence of an elastic transmission between the motor and the load) has to be properly handled, for example by considering the limits of PID controllers (Goubej

and Schlegel, 2014; Zhang and Furusho, 2000) and, in case, by using a more complex controller, by exploiting notch or bi-quadratic filters or observers that allow the virtual measurement of the load speed (Ellis and Lorenz, 2000).
Regarding the PID tuning, although optimization methods might yield a significant increment of the tracking performance (Calvini et al., 2015), it has again to be taken into account that the available computational capabilities might be limited, so that the use of simple tuning procedure are desirable in general.
In this paper a new automatic tuning approach that aims at satisfying all the previously mentioned requirements is proposed. The proposed method is able, in just one experiment, to find the appropriate PI controller and filter parameters. The experiment is composed by an initial part where the static friction coefficient is found. Then, a series of torque steps is properly given in order to find the frequency response of the system. In this context, the effect of the static friction is suitably removed from the data. If a resonant frequency is detected, a battery of bi-quadratic filters is designed. Finally, from the obtained frequency response, a first-order transfer function is used to approximate the system and to tune the PI parameters.
It is worth stressing that the operator just needs to set the position, speed and torque limits of the motor for the given machine and then the procedure is fully automatic.
The paper is organized as follows: in Section 2 the proposed autotuning methodology is described in detail; in Section 3 the simulation results obtained by testing the proposed method on both rigid and elastic systems are shown. Section 4 concludes the paper.

2. PROPOSED METHODOLOGY

The automatic tuning approach proposed in this paper is composed by a single procedure (see Figure 1), which can be divided in four parts: (*i*) an initial part where the motor torque is increased by small steps until the system starts moving to identify K_f; (*ii*) a second part where a series of positive and negative torque steps is given to the system in order to estimate the frequency response; (*iii*) a third part where, if necessary, a series of two bi-quadratic filters is tuned to compensate possible resonance and anti-resonance frequencies; (*iv*) a fourth (and last) part where the system is estimated as a first-order transfer function and a PI controller is tuned.
Thus, given the torque limit $\bar{\tau}$, the speed limit $\bar{\dot{\theta}}$, the position limit $\bar{\theta}$, which have to be set by the user depending on the application, the four steps can be outlined as follows.
(*i*) At the beginning, a torque signal is given to the motor in order obtain an estimation of the numerical values of the static friction coefficient $\hat{K}_f$. The torque signal is composed by a series of small steps that continuously increase until the system moves (see Figure 2). Each step has a magnitude computed as

$$\Delta\tau = \frac{\bar{\tau}}{N_{steps}} \quad (1)$$

where $\Delta\tau$ represents the torque step amplitude, $\bar{\tau}$ is the maximum applicable motor torque and N_{steps} represents the numbers of steps on which the maximum torque value is divided. N_{steps} is a tunable parameter, and its choice depends on the desired precision for the friction estimation. To understand if the system moves, a threshold equal to 1.5 times the common velocity measurement noise $\dot{\theta}_{noise}$ is applied. If the velocity exceeds, in magnitude, the velocity threshold, the static friction coefficient $\hat{K}_f$ is estimated, otherwise the torque input is increased by another $\Delta\tau$.

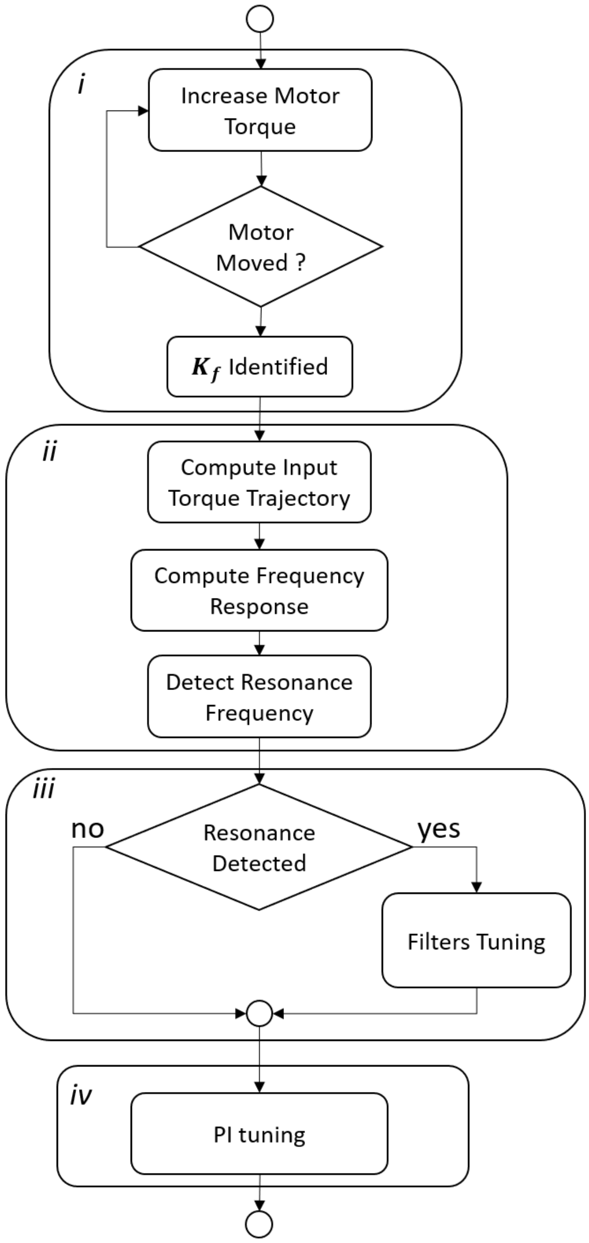

Fig. 1. Schematic flow chart of the autotuning procedure.

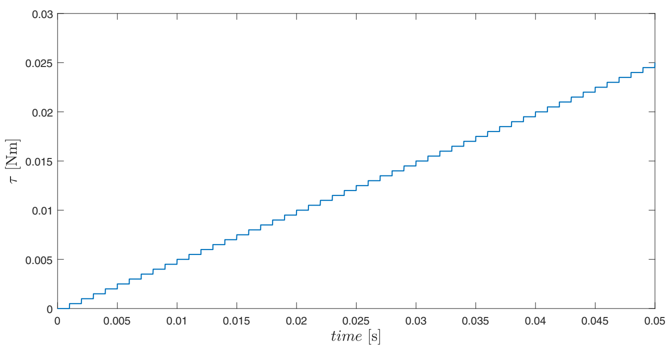

Fig. 2. Proposed torque input for the estimation of K_f.

(*ii*) After having estimated $\hat{K}_f$, a series of torque steps is given to the system in order to estimate its frequency response (see Figure 3). In order to compute the steps amplitudes and the steps times, to better estimate the response of the system over its working conditions, the torque $\bar{\tau}$, speed $\bar{\dot{\theta}}$ and position $\bar{\theta}$ limits of the motor must be taken into account. In this context, a frictionless system is considered with a maximum inertia equal to two times the motor one (denoted as J_m). This choice comes from the fact that in motion control applications a speed reducer is usually employed in such a way that the load inertia seen by the motor (*i.e.*, the load inertia divided by the square of the reduction rate) has a value similar to the motor inertia Giberti et al. (2011). Furthermore, by considering a frictionless system, the presence of friction in the real system ensures that the maximum real velocity and the real maximum reached position are lower than the maximum ones defined by the user. When defining the three traits law of motion, it is important to allow the evolution of the system as much as possible after an excitation signal, therefore a long zero acceleration (that means zero torque) trait must be used between the two positive and negative acceleration traits. Moreover, the torque steps amplitude has to be big enough to overcome the non linearities that affect values of torque that are near the limit of the friction torque K_f. The amplitude of the step torque is thereby set as the torque limit $\bar{\tau}$ for a the first three traits law of motion. The law of motion is then defined by calculating the total time t_{tot} and the acceleration time t_a, that can be expressed in relation to the total time introducing a coefficient α as $t_a = \alpha t_{tot}$. The system that gives the values of α and t_{tot} under the conditions of $\bar{\ddot{\theta}}_1$, $\bar{\theta}$ and $\bar{\dot{\theta}}$ is

$$\begin{cases} \alpha_1 = \dfrac{\bar{\dot{\theta}}^2}{\bar{\dot{\theta}}^2 + \bar{\ddot{\theta}}_1^2} \\ t_{tot_1} = \dfrac{\bar{\dot{\theta}}^2 + \bar{\ddot{\theta}}_1 \bar{\theta}}{\bar{\ddot{\theta}}_1 \bar{\dot{\theta}}} \end{cases} \tag{2}$$

with

$$\bar{\ddot{\theta}}_1 = \frac{\bar{\tau}}{2J_m}.$$

An opposite three traits acceleration law of motion is applied, when the free evolution of the system has ended, in order to bring the motor back to the initial position (see Figure 5). When the free evolution of the system has ended again, in order to better estimate the frequency response of the system, the procedure is repeated decreasing the torque step value to half of the maximum torque $\bar{\tau}$, and the parameters are calculated solving system (2), with

$$\bar{\ddot{\theta}}_2 = \frac{\bar{\tau}}{4J_m}.$$

instead of $\bar{\ddot{\theta}}_1$, resulting in t_{tot_2} and α_2. Given the sampling period T_s of the control system the frequencies used for the estimation of the frequency response of the system are selected as

$$\omega_i = 10^{W_i} \quad \text{for } i = 0 \ldots N \tag{3}$$

where

$$W_i = \log(\underline{\omega}) + \Delta i \tag{4}$$

and

$$\Delta = \frac{\log(\bar{\omega}) - \log(\underline{\omega})}{N}, \tag{5}$$

where $\underline{\omega}$ and $\bar{\omega}$ are the minimum and maximum frequency of the range respectively, N is the number of frequencies to consider and Δ defines the linear relation between the minimum and the maximum frequency. It is worth noting that this kind of selection permits to have the same frequency resolution on all the logarithmic scale.

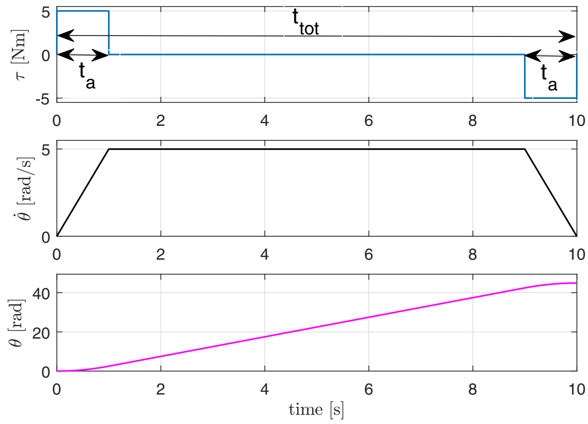

Fig. 3. Proposed three traits law of motion, with $t_{tot} = 10$ [s] and $t_\alpha = 1$ [s].

To obtain the frequency response of the system, the H_v method presented in (Vold et al., 1984) can be used.

(iii) At this point, if a resonance frequency is detected, a series of two bi-quadratic filters is properly designed. The transfer function of a bi-quadratic filter for the resonance compensation is

$$H_r(s) = \frac{s^2 + \frac{\omega_r}{F} s + \omega_r^2}{s^2 + R\omega_r s + \omega_r^2} \tag{6}$$

where ω_r is the resonance frequency, ω_a is the anti-resonance frequency, F is a parameter that allows the compensation of a possible estimation error of ω_r and

$$R = \frac{\omega_a}{\omega_r} + \frac{\omega_r}{\omega_a}. \tag{7}$$

In order to compensate for resonance frequency estimation errors, a suitable value of the parameter F is selected as the difference in decibel between the frequency response at the resonance frequency A_{ω_r} and the frequency response at the anti-resonance frequency A_{ω_a}. This difference becomes

$$F = \left| \frac{10^{A_{\omega_r}/20}}{10^{A_{\omega_a}/20}} \right|. \tag{8}$$

For the compensation of the anti-resonance of the system, the transfer function of bi-quadratic filter is

$$H_a(s) = \frac{s^2 + R\omega_a s + \omega_a^2}{s^2 + \frac{\omega_a}{F} s + \omega_a^2} \tag{9}$$

where R and F are the same as the ones expressed in (7), (8).
(*iv*) At the end, independently from the execution of step *(iii)*, a first-order approximation of the system is performed. The first-order estimation of the transfer function of the system

$$\hat{P}(s) = \frac{k}{t_p s + 1} \tag{10}$$

is obtained as follows:

- the gain parameter k is computed as the mean value of the magnitude of the frequency responses of the three lower frequency values;
- the time constant t_p is estimated as the inverse between the frequency value (denote as ω_p) when the frequency response is $3dB$ less than $20log_{10}(k)dB$, that is ω_p.

Given the PI controller in the form

$$C(s) = K_p \left(\frac{T_i s + 1}{T_i s} \right), \tag{11}$$

by zero-pole cancellation, imposing $T_i = t_p$, the following closed-loop first order transfer function is obtained:

$$\hat{G}(s) = \frac{\hat{P}(s)C(s)}{1+\hat{P}(s)C(s)} = \frac{kK_p}{T_i s + kK_p} \tag{12}$$

Now, by considering the maximum amplitude $\Delta\dot{\theta}_{sp_{max}}$ of a set-point step selected by the user depending on the application, K_p can be tuned, in order to ensure that the actuator will not saturate, as

$$K_p = \frac{\bar{\tau}}{\Delta\dot{\theta}_{sp_{max}}} \tag{13}$$

It is worth stressing that the operator just needs to select the sampling period of the controller, the maximum amplitude $\Delta\dot{\theta}_{sp_{max}}$ of a set-point step and the position, speed and torque limits of the motor. The procedure is then fully automatic.

3. SIMULATION RESULTS

The proposed methodology has been tested in the Matlab/Simulink environment on two different simulated systems: a rigid system and an elastic one. In both systems θ represents the motor position while θ_L represents the load position.
The rigid system that has been used in the simulation has been modelled as

$$\begin{cases} \ddot{\theta} &= \frac{1}{J_{tot}} \left[\tau_r - K_f \,\mathrm{sgn}(\dot{\theta}) - B_m \dot{\theta} \right] \\ \ddot{\theta}_L &= i\ddot{\theta} \\ \dot{\tau}_r &= \frac{1}{t_e} (\tau - \tau_r) \end{cases} \tag{14}$$

where $J_{tot} = J_m + J_L/i^2$ is the total inertia seen by the motor, i is the transmission ratio, K_f is the static friction coefficient and B_m is the viscous friction one, τ_r is the real motor torque, τ is the desired motor torque and t_e is the electric time constant of the electrical drive. The last equation has been added to approximate the effects of the electrical drive. The data of the rigid system parameters that have been used in the simulations are listed in Table 1, along with the torque, velocity and position limits imposed for the automatic tuning procedure.
The results obtained during the part (i) of this simulation are shown in Figure 4. As it is possible to see, the static friction parameter estimated during this part is very close to the real one ($\hat{K}_f = 0.0520$), using $N_{steps} = 20000$.
In part (ii), the frequencies parameters in (5) have been set to the following values:

$$\underline{\omega} = 0.1\ [rad/s]$$
$$\bar{\omega} = \frac{2\pi}{5T_s}\ [rad/s] \tag{15}$$
$$N = 200$$

Parameter	Motor simulator	Unit
J_m	$2.8\,10^{-4}$	kgm^2
J_L	0.0070	kgm^2
K_f	0.05	N
B_m	0.032	$N/(rad/s)$
i	5	− − −
t_e	$2.5\,10^{-4}$	s
$\bar{\tau}$	10	Nm
$\bar{\theta}$	500	rad
$\bar{\dot{\theta}}$	300	rad/s

Table 1. Simulated rigid system data.

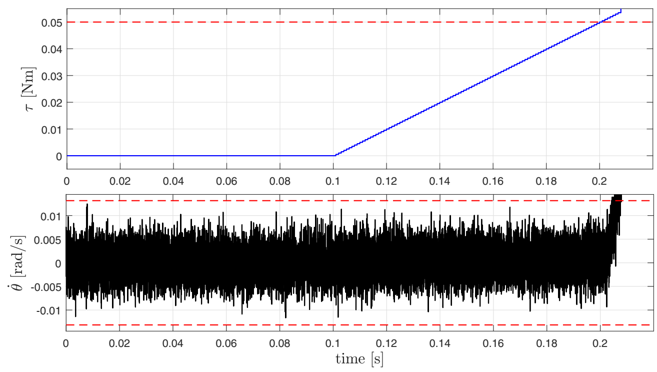

Fig. 4. Estimation of the static friction parameter K_f on the rigid system. The real static friction parameter value is the red dashed line.

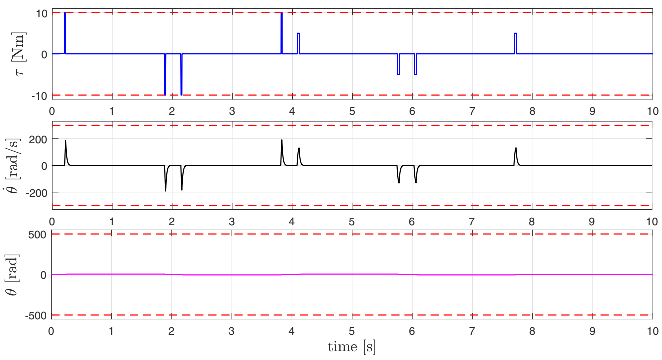

Fig. 5. Proposed input torque set point and the relative measured motor velocity and position of the simulated rigid system during the frequency response identification trial. It can be seen that the second three traits acceleration law of motion is applied only after the rest of the system.

Times $t_{tot_1} = 1.6835$ $[s]$ and $t_{tot_2} = 1.7003$ $[s]$ and coefficients $\alpha_1 = 0.0100$ and $\alpha_2 = 0.0198$ have been determined solving system (2) for the two sequences of the input torque, and the plots of the resulting identification experiment are shown in Figure 5. The obtained system frequency response is shown in Figure 6. As it is possible to see, the estimated frequency response (blue solid line) is very close to the one of the real system without the static friction (black dashed line); also the first-order approximation (green dashed line) is very close to the estimated frequency response of the system. Furthermore, as expected, no resonance and anti-resonance frequencies have been detected. The determined first-order transfer function is

$$\hat{P}(s) = \frac{31.303}{0.0173s + 1} \tag{16}$$

As it is possible to see from Figure 6, no resonance or anti-resonance frequencies have been detected, so, during the part (iii) of this procedure no bi-quadratic filters have been tuned.
In phase (iv), given a maximum set-point step velocity of $\Delta\dot{\theta}_{sp_{max}} = 200$ $[rad/s]$, the resulting PI parameters are

$$T_i = 0.0173 \qquad K_p = 0.050 \tag{17}$$

The step responses obtained from the system are shown in Figure 7, where step response of the closed-loop first-order approximation and the one of the closed-loop real system are compared.

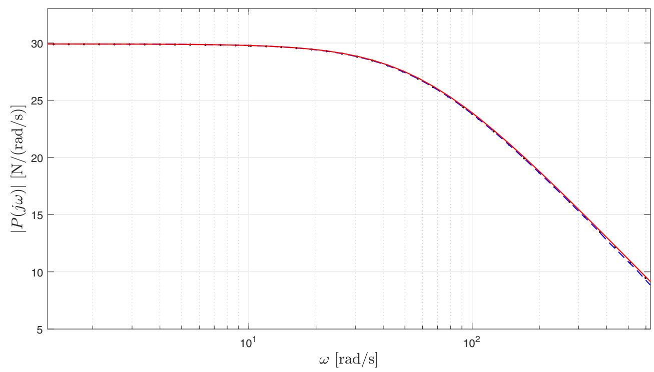

Fig. 6. Frequency response of the simulated rigid system: real linearized response (black dotted line), estimated one with the proposed methodology (blue dashed line) and first order approximation (red solid line).

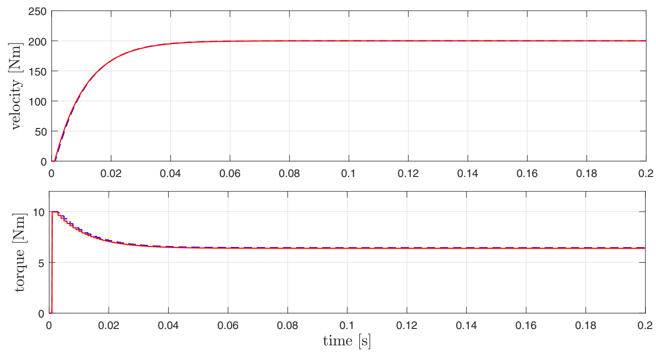

Fig. 7. Step response (top) and control variable (below) of the simulated rigid system with the proposed control strategy with $\dot{\theta}_{sp} = 200[rad/s]$: ideal first-order system response (red solid line) and real system response (blue dashed line).

As a second example, an elastic system described by the following model has been considered

$$\begin{cases} \ddot{\theta} & = \frac{1}{J_m}\left[\tau_r - K_f \operatorname{sgn}(\dot{\theta}) - B_m\dot{\theta} - \frac{K}{i}d\theta - \frac{C}{i}d\dot{\theta}\right] \\ \ddot{\theta}_L & = \frac{K}{i}d\theta + \frac{C}{i}d\dot{\theta} \\ \dot{\tau}_r & = \frac{1}{t_e}(\tau - \tau_r) \end{cases} \tag{18}$$

where $d\theta = \left(\frac{\theta}{i} - \theta_L\right)$, $d\dot{\theta} = \left(\frac{\dot{\theta}}{i} - \dot{\theta}_L\right)$, K is the elastic constant and C is the damping coefficients of the system. The values of the parameters and the user defined limits that have been used in the simulations are equal to the ones of the rigid system, listed in Table 1, exception made for the elastic constant K and the damping coefficient C, whose values are $K = 100$ $[N/rad]$ and $C = 0.30$ $[Ns/rad]$.

The results obtained during the part (i) of this simulation are shown in Figure 8. As it is possible to see, also if there is a resonance in the system, the static friction parameter estimated during this part is very close to the real one ($\hat{K}_f = 0.052$).

In part (ii), times $t_{tot_1} = 1.6835$ $[s]$ and $t_{tot_2} = 1.7003$ $[s]$ and coefficients $\alpha_1 = 0.0100$ and $\alpha_2 = 0.0198$ have been determined, and the plots of the resulting identification experiment are shown in Figure 9. The obtained system frequency response is shown in Figure 10. The obtained first-order transfer function is

$$\hat{P}(s) = \frac{31.3383}{0.0196s + 1} \tag{19}$$

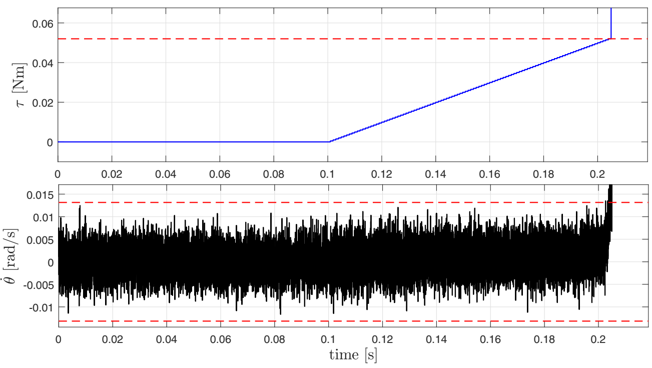

Fig. 8. Estimation of the static friction parameter K_f on the elastic system. The real static friction parameter value is the red dashed line.

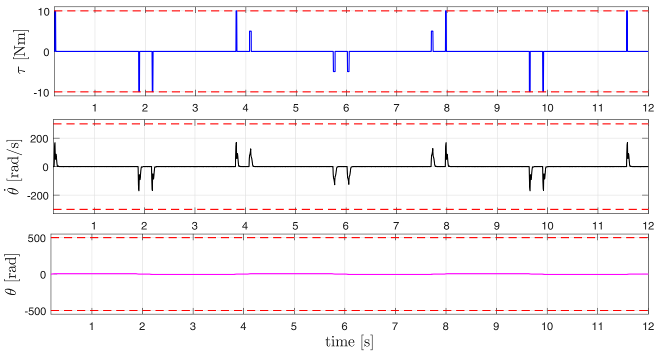

Fig. 9. Proposed input torque set point and the relative measured motor velocity and position of the simulated elastic system during the frequency response identification trial.

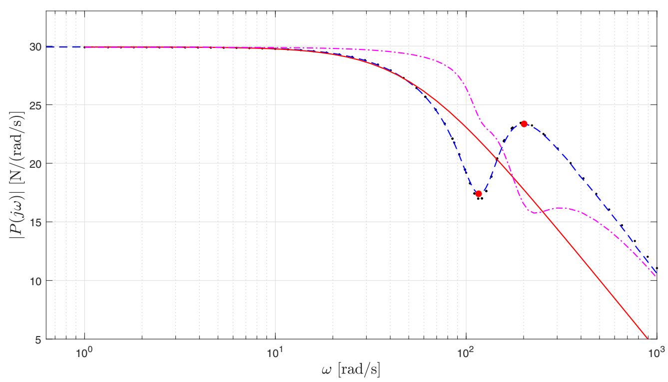

Fig. 10. Frequency response of the simulated elastic system: real linearised response (black dashed line), estimated one obtained with the frequency response analysis (blue dotted line), first-order approximation (red solid line) and system with notch and antinotch (magenta dash-dot line).

As it is possible to see, the estimated frequency response (blue solid line) is very close to the real one (black dashed line). The first order approximation (green dashed line) is able to find properly the low frequency pole of the system. Furthermore, as expected, one resonance and one anti-resonance frequency has been detected.

In (iii) a couple of bi-quadratic filters has been used in order to compensate for the resonance and antiresonance of the system. The two bi-quadratic filters have been tuned by considering

Parameter	Real value	Estimated value	Unit
ω_r	199.5	201.27	rad/s
ω_a	118.6	116.18	rad/s

Table 2. Comparison between real and estimated resonance and anti-resonance frequencies.

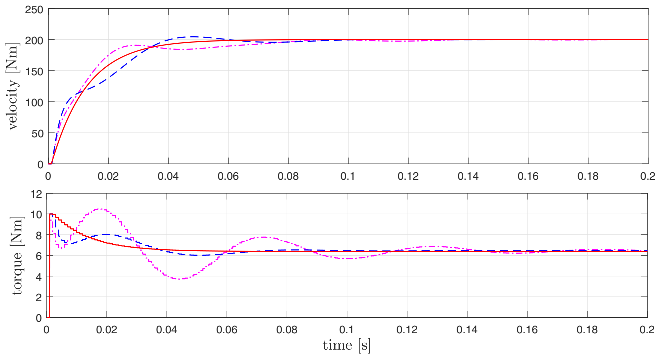

Fig. 11. Step response of the simulated elastic system with the proposed control strategy $\dot{\theta}_{sp} = 200[rad/s]$: ideal first order response (red solid line), response of the system with bi-quadratic filters (magenta dash-dot line), and real system without bi-quadratic filters (dashed blue line).

the estimated resonance frequency $\hat{\omega}_r$ and anti-resonance frequency $\hat{\omega}_a$ shown in Table 2. Finally, in phase (*iv*) of this procedure the tuning of the PI controller has been tested. The resulting PI parameters are

$$T_i = 0.0196 \qquad K_p \quad = 0.050 \tag{20}$$

The step responses obtained from the closed-loop system are shown in Figure 11, where the ideal first-order system and the closed-loop real system with and without the bi-quadratic filters are compared.

4. CONCLUSIONS

In this paper a fast autotuning procedure for the velocity loop of mechatronic systems is presented. The method is based on a single short experiment that permits the identification of the static friction and the frequency response of the system. A battery of bi-quadratic filters is automatically tuned if a resonance frequency is detected. A PI controller is finally tuned based on a first-order approximation of the system, obtaining a first-order closed loop system. During the automatic procedure the limits related to the maximum torque, speed and position of the system are explicitly considered.
To test the procedure, simulations on a rigid and on an elastic system have been made in the Matlab/Simulink environment. The results confirm the effectiveness of the proposed methodology.

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REFERENCES

Åström, K.J. and Hägglund, T. (2006). *Advanced PID Control*. ISA Press, Research Triangle Park, USA.

Beineke, S., Schutte, F., Wertz, H., and Grotstollen, H. (1997). Comparison of parameter identification schemes for self-commissioning drive control of nonlinear two-mass systems. In *Proceedings Annual Meeting IEEE Industry Applications Society*, 493–500. New Orleans, LA.

Calvini, M., Carpita, M., Formentini, A., and Marchesoni, M. (2015). PSO-based self-commissioning of electrical motor drives. *IEEE Transactions on Industrial Electronics*, 62(2), 768–776.

Choi, Y. and Chung, W.K. (2004). *PID Trajectory Tracking Control for Mechanical Systems*. Springer, Berlin, D.

Ellis, G. and Lorenz, R.D. (2000). Resonant load control methods for industrial servo drives. In *Proceedings IEEE Industry Applications Conference*, 1438–1445. Rome, I.

Giberti, H., Cinquemani, S., and Legnani, G. (2011). A practical approach to the selection of the motor-reducer unit in electric drive systems. *Mechanics based design of structures and machines*, 39(3), 303–319.

Goubej, M. (2015). Kalman filter based observer design for real-time frequency identification in motion control systems. In *Proceedings 20th International Conference on Process Control*. Strbske Pleso, SK.

Goubej, M., Krejci, A., and Schlegel, M. (2013). Robust frequency identification of oscillatory electromechanical systems. In *Proceedings 19th International Conference on Process Control*. Strbske Pleso, SK.

Goubej, M. and Schlegel, M. (2014). Robust PID control of electrical drive with compliant load. In *Proceedings 19th IFAC World Congress*, 11781–11786. Cape Town, SA.

Villwock, S. and Pacas, M. (2008). Application of the welch-method for the identification of two- and three-mass-systems. *IEEE Transactions on Industrial Electronics*, 55(1), 457–466.

Vold, H., Crowley, J., and Rocklin, G.T. (1984). New ways of estimating frequency response functions. *Sound & Vibration*, 18(11), 34–38.

Weissbacher, J., Grunbacher, E., and Horn, M. (2013). Automatic tuning of a servo drive speed controller for industrial applications. In *Proceedings IEEE International Conference on Mechatronics*, 700–705. Vicenza, I.

Zhang, G. and Furusho, J. (2000). Speed control of two-inertia system by PI/PID control. *IEEE Transactions on Industrial Electronics*, 41(3), 603–609.