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Parallel IP – Resonant control approach for attenuating the effect of torsion in electric drives with flexible links

Abstract. *Because of their high efficiency, electric drives have become a staple solution in numerous applications, such as robotic manipulators and exoskeletons. However, such constructions need to be designed with lighter materials, which are also more prone to twisting and bending. Because of torsion affecting the mechanical structure, precision of motion is deteriorated. In this study, a classic IP controller is coupled with a resonant filter to improve the performance of the drive affected by the torsional torque. The results include both numerical analysis and experimental verification.*

Streszczenie. *Ze względu na wysoką sprawność, napędy elektryczne stały się klasycznym rozwiązaniem stosowanym w wielu mechanizmach, m.in. manipulatorach oraz egzoszkieletach. Aplikacje te często wymagają wykorzystania lekkich materiałów podatnych na skręcanie i wyginanie. Odkształcenia te mocno utrudniają precyzyjne sterowanie napędem. W niniejszej pracy przedstawione zostało wykorzystanie połączenia klasycznego rozwiązania (regulatora IP) oraz filtra rezonansowego zastosowanego w celu eliminacji składowych odpowiedzialnych za pogłębianie skręcenia wału. Struktura sterowania poddana została zarówno analizie numerycznej, jak i weryfikacji eksperymentalnej.* ***(Równoległy regulator IP – rezonansowy zastosowany do łagodzenia wpływu skręcenia wału w napędach elektrycznych z elastycznym sprzęgłem)***

Keywords: two-mass system, resonant control, vibration suppression, electric drives
Słowa kluczowe: układ dwumasowy, regulacja rezonansowa, tłumienie drgań, napędy elektryczne

Introduction

Electric drives with flexible links (long shafts, belts, gear boxes, etc.) can be represented by a two-mass system [1, 2], where the masses represent the inertia of the electrical motor and the load. In such a model, both elements are connected by an elastic shaft with finite stiffness. Regarding electric drives, two-mass system model can represent various devices, for example: steel rolling mills [3], robotic arms [4], wind turbines [5], humanoid robots [6] and electro-hydrostatic actuators [7].

This model allows to analyze the impact of torsion on the system. It becomes apparent through the difference between the angular speed of the motor and the load machine, and torsional torque value. The phenomenon that occurs in these systems is torsional vibration. One of the sources of said vibrations is a rapid change of the rotational speed of the motor which causes twisting of the flexible link [8]. Vibrations in two-mass systems result in energy loss, performance degradation and dynamic stress [9]. It can also be problematic in precise position control.

Many methods to damp torsional torque oscillations have been introduced. One of them is to use PI control with additional feedbacks. In [10], three different groups of feedback signals were presented. Using two out of three feedback groups allows to modify the damping coefficient and the resonant frequency of the two-mass system. Other approach is to use an IP controller [11]. This solution allows satisfactory vibration damping using controller gains (based on inertia and the resonant frequency values) that ensure stability of system. Vibration damping can be also done using IFT (Iterative Feedback Tuning) [12]. That method utilizes auto-tuning to optimize parameters for the used controllers. State controller with state variables estimation can also mitigate oscillation as shown in [13]. State variable estimation using fuzzy MKF (Multilayer Kalman Filter) provides signals for the state controller allowing proper torsional vibration mitigation.

Regarding torsional oscillations damping not only P, I and D type controllers can be implemented. Resonant controllers are used in order to mitigate vibrations within specified frequency bandwidths. In the context of electric drives these can be introduced for various purposes, e.g. controlling current harmonics in order to minimize torque ripple [14], and mitigating torque vibrations using speed control error [15]. Both of these examples utilize resonant and PI controller in parallel configuration. As in [16] and [17], multiple resonant controllers can be used for damping vibrations in electrical drives at wide scope of frequency bandwidths.

In this paper IPR (integral-proportional-resonant) controller is introduced. The novelty of the proposed solution is the IP controller usage (instead of PI as in other solutions with resonant control) and phase compensator transfer function representation of the resonant element, which allows more intuitive tuning thanks to it's physical interpretation. IPR controller requires only speed feedback as opposed to the other vibration-mitigating methods that require multiple signals to operate properly. IP element of the IPR controller not only ensures proper dynamics of the two-mass system, but also damps vibrations as opposed to the PIR controller, where PI element is only responsible for dynamics. The resonant part of the IPR controller improves vibration mitigation properties of the IP element in the frequency bandwidth corresponding to the two-mass system resonance.

Mathematical description of the controlled plant

The two-mass system is the commonly used model of elastic drives with flexible links. Fig. 1 depicts its schematic diagram. The two-mass system is the mechanical model that

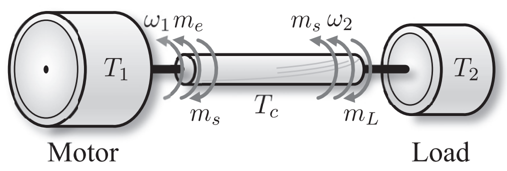


Fig. 1. Schematic diagram of the two-mass system.

represents the vibration phenomenon. Time-dependent variables are generally given: $\omega_1(t)$, $\omega_2(t)$ – The motor and load side speeds. $m_s(t)$, $m_e(t)$, $m_L(t)$ – The torsional, electromagnetic, and load torques, respectively. The electromagnetic torque is the system's input and is regulated by sufficiently fast torque control. This inner torque controller generates the signals applied to the electromagnetic part of the motor through the power converter [10]. Since the load torque is regarded as the disturbance to the load side of the two-mass system, the controller must regulate the load speed against it.

With frictions and torsional damping neglected, the mathematical model of the two-mass system is given as follows [10]:

$$
\frac{d}{dt}\begin{bmatrix}\omega_1(t)\\ \omega_2(t)\\ m_s(t)\end{bmatrix}=\begin{bmatrix}0 & 0 & \frac{-1}{T_1}\\ 0 & 0 & \frac{1}{T_2}\\ \frac{1}{T_c} & \frac{-1}{T_c} & 0\end{bmatrix}\begin{bmatrix}\omega_1(t)\\ \omega_2(t)\\ m_s(t)\end{bmatrix} + \begin{bmatrix}\frac{1}{T_1} & 0\\ 0 & \frac{-1}{T_2}\\ 0 & 0\end{bmatrix}\begin{bmatrix}m_e(t)\\ m_L(t)\end{bmatrix}, \tag{1}
$$

where T_1, T_2, and T_c are the mechanical time constants of the motor, load machine, and flexible links.

IPR controller

Even though the objective of drive control is to make the load speed follow the desired reference $\omega^{\mathrm{ref}}(t)$, due to limitations, such as sensor installation, the motor speed is usually fed back to the controller. An IP controller [11, 18] is a PID controller subclass involving proportional and integral actions for reference and feedback signals. While PI controllers [10] have proportional-integral actions for reference and feedback, in IP control integral action affects both signals, but proportional action only affects the feedback. This contributes to the control system responding with a slightly longer rise time, but significantly reduced overshoot, preventing unnecessary vibrations.

This paper arranges the resonant controller [15] in parallel with the IP controller to suppress vibrations by adjusting the magnitude and phase of the error signal $\omega^{\mathrm{ref}}(t) - \omega_1(t)$. In this paper, the resonant controller is defined in the following transfer function:

$$
G_r(s) = K_r \frac{s^2 + 2\alpha\zeta\omega_r s + \omega_r^2}{s^2 + 2\zeta\omega_r s + \omega_r^2}, \tag{2}
$$

where K_r, α, ζ, and ω_r denote the parameters of the resonant controller. The frequency ω_r and damping coefficient ζ determine the bandwidth. Fig. 2 depicts the bode plots of the resonant controller $G_r(s)$, which vary according to the parameter changes. This figure indicates that the resonant element reshapes error signals by the properties of phase lead (at lower frequencies) and phase lag (at higher ones). The four parameters specifically have the physical interpretation; K_r affects only the entire bandwidth in the gain characteristic. α makes both the gain and phase properties stand out. ζ increases the bandwidth of the controller. ω shifts the band of compensation. The conflicting attributes of these parameters facilitate the design of a resonant controller in the frequency domain tailored to each drive system.

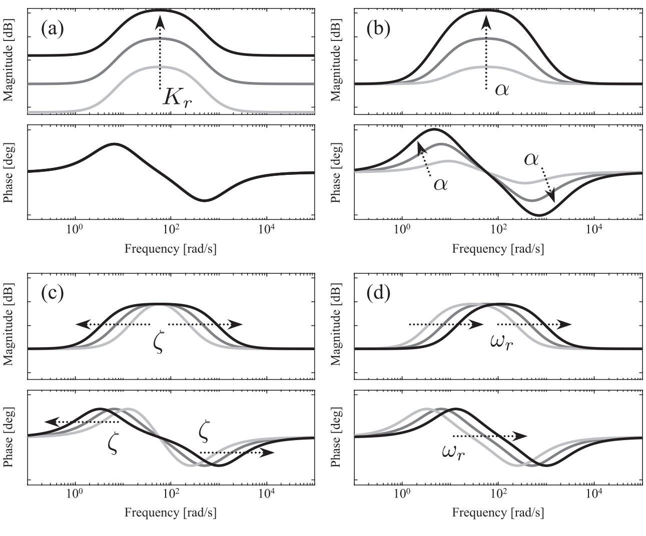


Fig. 2. Bode plots of resonance controller $G_r(s)$ with parameter changes in (a) K_r, (b) α, (c) ζ, and (d) ω_r.

Fig. 3 shows the entire configuration of the IPR controller.

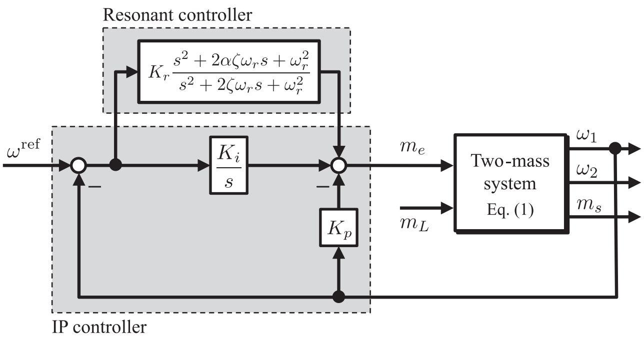


Fig. 3. Block diagram of IPR controller.

Eventually, the electromagnetic torque can be obtained by the IPR controller as follows:

$$
m_e(s) = \frac{K_i}{s}\left[\omega^{\mathrm{ref}}(s) - \omega_1(s)\right] \quad K_p\omega_1(s) + G_r(s)\left[\omega^{\mathrm{ref}}(s) - \omega_1(s)\right], \tag{3}
$$

where K_i and K_p denote the integral and proportional gains, respectively.

Numerical results

The proposed control scheme has been implemented in Matlab/Simulink environment to perform initial verification of the theoretical assumptions. For this purpose, Matlab ver. 2020b has been used. During simulations, step time has been set to 0.0001 s.

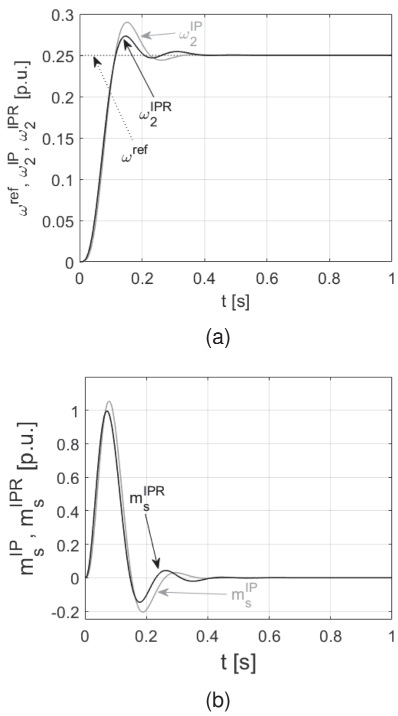


Fig. 4. Initial phase of operation of the two-mass system – comparison of IP and IP-Resonant control: (a) angular speed of the load, (b) torsional torque.

For a better perspective, the verification puts the proposed IP-resonant controller alongside the classic IP approach. Figures showing the obtained results show transients for both control structures.

First, the structure's response to a step function was tested. The comparison is shown in Fig. 4. The resonant term in the control structure provides two visible advantages. The overshoot is limited, and the settling time is shorter for the proposed controller. The advantage is also visible in the torsional torque response.

Robustness against parameter changes was next verified. The ratio H of the resonant frequency of the two-mass system ω_{res} to its anti-resonant frequency ω_{ares} was chosen as the point of analysis:

$$H = \frac{\omega_{res}}{\omega_{ares}} = \frac{T_1 + T_2}{T_2}. \tag{4}$$

Control system capabilities were tested in different operating scenarios. Changes in the time constant of the load machine (its inertia) were applied, thus changing the ratio H. For the nominal values of the time constants of the used machines, the ratio is equal to $H_n = 1,7123$. Additionally, two other scenarios were tested: with the load machine inertia increased, so that the time constant $T_2 = 0.456$ s ($84\%H_n$), and T_2 decreased to $0.171\ s$ ($128\%H_n$). The results are shown in Figure 5. When T_2 is decreased, the overshoot during operation start is mitigated, but at the cost of harsher response to load torque changes. On the other hand, when T_2 value is higher than nominal, reaction to load torque is milder, but the overshoot during the start is bigger. Still, the controller is capable of withstanding wide range of load inertia changes. Selecting the exact value of H_n for the controller's resonant frequency is suggested to achieve a compromise in the system performance during various dynamic states.

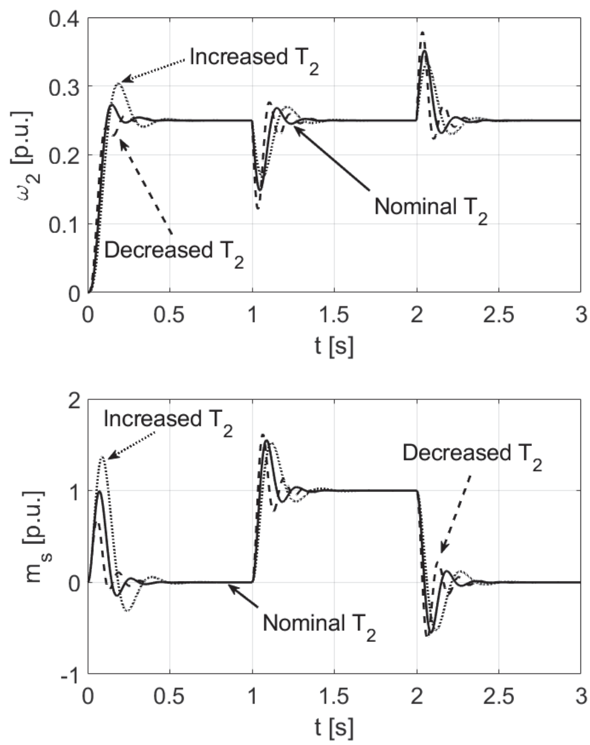


Fig. 5. Two-mass system reaction to change of load machine inertia.

The results are promising in two perspectives. First, better quality of control (faster response with limited overshoot) transfers to more efficient movement of the machine. However, not only the control quality, but also the machine's longevity is improved. With less stress put on the shaft, enhanced operation safety and less mechanical faults can be expected.

Experimental verification

The results from the previous section have been verified on a laboratory workbench. Nominal parameters of the two-mass system drive are given presented in Table 1. The algorithm has been implemented on a dSPACE 1103 fast-prototyping device. More elaborate description of the setup is available in [19].

Table 1. Nominal Parameters of the experimental workbench

Parameter	DC Motor	DC Load machine	Shaft
Nominal power	500 W	400 W	—
Nominal voltage	230 V	230 V	—
Nominal current	3.15 A	1.74 A	—
Nominal angular speed	1450 rpm	1450 rpm	—
Time constant	0.203 s	0.285 s	0.0026 s

The results gathered during the experimental tests prove the prevalence of the IP-resonant control structure. Speed response is similar to what was expected from the numerical studies. In Fig. 6 lowered overshoot and shorter shorter settling time can be observed. Also, torsional torque is lower during the transition. However, there is no significant difference between the compared control structures when load torque is applied and removed (as shown in Fig. 7).

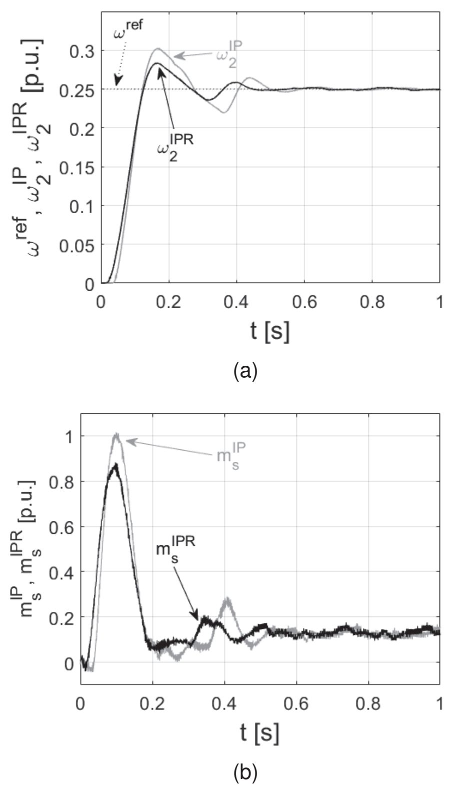
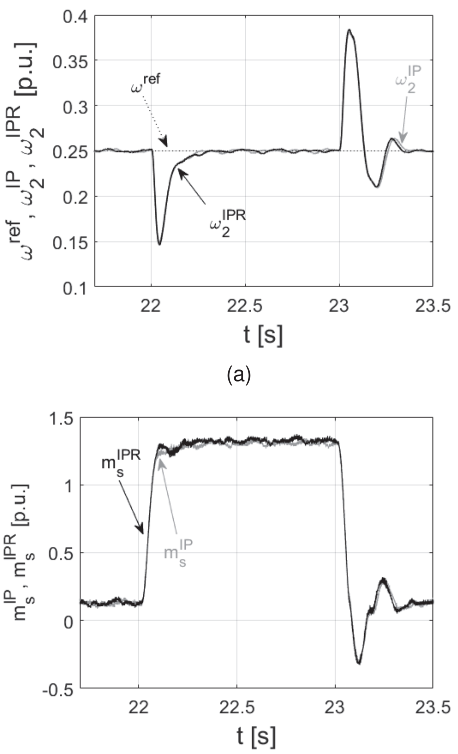


Fig. 6. Experimental results of two-mass system reaction to change of load torque – comparison of IP and IP-Resonant control: (a) angular speed of the load, (b) torsional torque.



(a)

(b)

Fig. 7. Experimental results of the initial phase of operation of the two-mass system – comparison of IP and IP-Resonant control: (a) angular speed of the load, (b) torsional torque.

Discussion

The addition of the resonant component parallel to the IP controller resulted in the improvement of the vibration damping in the transient stage of electrical drive speed control. Both speed and torsional torque have lower oscillations at that stage compared to the structure utilizing only IP component as confirmed by simulation and experimental results.

The mathematical representation of resonant component, shown in the article, enables customization of the IPR controller properties such as gain, bandwidth and phase shift. Such approach allows the controller to be implemented in the wide range of two-mass systems after prior tuning.

It is necessary to determine resonant frequency of the two-mass system ω_{res} for proper IPR controller operation as it should be equal to the frequency ω_r. Further research should focus on expanding the proposed structure with resonant frequency observer.

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