Suppression of torsional oscillations based on IMPACT controlling structure

Milan Matijevic [*]   **Milic Stojic** [†]
Slobodan Vukosavic [‡]   **Miladin Stefanovic** [§]

Abstract

In this paper, an algorithm for suppression of the torsional oscillations, in the computer controlled servo systems, is purposed. A few modification of the IMPACT structure (Internal Model Principle and Control Together) in the digitally controlled electrical drives are proposed. The IMPACT structure suitable for suppression of the torsional oscillations in a servo system with the flexible coupling has been presented. The approach, proposed in this paper, gives better solutions for the problem of the mechanical resonance in the modern servo drive than the previous ones. The presented structure is simple with a small number of adjustable parameters that could be easily set to achieve the desired robust, filtering, and dynamic properties of the system.

[*]Faculty of Mechanical Engineering University of Kragujevac, Sestre Janjić 6, 34000 Kragujevac, Serbia, e-mail: milan.matijevic@gmail.com

[†]Elektrotehnički fakultet, Beograd, e-mail: estojic@etf.bg.ac.yu

[‡]Elektrotehnički fakultet, Beograd, e-mail: boban@ieee.org

[§]Faculty of Mechanical Engineering University of Kragujevac, Sestre Janjić 6, 34000 Kragujevac, Serbia, e-mail: miladin@kg.ac.yu

1   Introduction

In many application in the modern tool machines, which must ensure high quality of products and high productivity of production lines, high dynamic performances of speed and positioning servomechanisms with different motors in the role of the actuators are required. Similar requirements are presented in other modern machine systems, such as: devices for production of micro-electronic assemblies, robotic manipulators, automatized machines for quality control, systems for data storage, etc. The main goal is reaching a nominal error of monitoring in the level of resolution of sensors, both in the stationary state and in the transition process. Good robust performances of a designed servosystem with a closed feedback are mandatory, so the implementation of the Tsypkin's IMPACT structure [1-5] in the synthesis of the digitally controlled electric motor drive presents itself as a logical solution [1,2,5].

However, servomotors, in a typical industrial environment, are connected with their actuators with a transfer mechanism with a finite stiffness [6,7]. Elastic coupled masses: motor – load, introduce a finite number of zeros and a pair of conjugate complex poles in a plant transfer function and lead to a possible appearance of the mechanical resonance. This problem is mode marked in the servosystems in which the position sensor is placed on the loaded shaft [6,7]. In that case, the system with the closed feedback will include non-modeled modes of the torsional resonance [6]. The same problem appears in the case when the feedback sensor is placed on an axle of a motor. This phenomenon limits performances of the system and initiate low suppressed oscillations in the feedback system due to the excitation from the referent input of the load [6,7]. Usually ignored in the design of conventional servo systems, the resonant modes could overlap with the bandwidth and provoke forced oscillations of the machine and coupled parts [6]. This process is accompanied with noise and a possible damage of the machine parts joints (due to the material wearing out [6]), wearing and tools breakage, as well as a possible influence on the quality of the working product. A fine review of the previous efforts in this problem solution is presented in [6], while in [6,7] a new approach of design of the anti-resonant compensator is suggested. The efficiency of this model is approved in the case of the synthesis of the angular velocity servomechanism, with the simulation and experimental

results.

Using the genuine IMPACT structure and its modifications [1, 2, 5], we produce a desired character of behavior, preciseness in the steady-state regime, high speed of reaction (wide bandwidth) of servo-drive with the closed feedback, small influence of the external disturbance and measurement noise on the control variable (angular velocity or position) as well as a low sensitivity of the performance quality index on the changes of the parameters inside the controlled plant [5]. The synthesis of the IMPACT structure of the speed servomechanism with an asynchrony motor in the role of the actuator and the polynomials predictive filter, instead of the classical prediction polynomial, is suggested in this paper. The simulation illustrated possibilities of a structure in the suppression of the torsional oscillations produced in the case of appearance of the mechanical resonance [6]. It is shown that the implementation of the suggested solution could improve the performances of the system.

2 Servo system with elastic feedback

The model of a controlled plant (an AC motor with a load), which covers the flexible coupling of the motor and the load is presented in Figure 1 [6]. The moment of the actuator M_{em} is a control variable; the moment on the loaded shaft M_l presents disturbance. The motor inertia J_m and the load inertia J_l are coupled with the transmission mechanism (shaft, reductor and gear belt) with a stiffness coefficient c_s. The elastic coupling doubles the number of the variable stages inside the electromotor drive [6]. Generally, the angular velocity ω_m and the position θ_m of the motor axle are different, retrospectively, and form the angular velocity ω_l and the position θ_l on the loaded shaft e.g. on the loaded side. The coefficient of the viscosity friction b_v generally has very small values and increase existence of the low suppression of the torsional oscillations [6]. The torsional moment M_o is equal to the moment of the load M_l only in the stationary oscillation kinetic state, while during the transition process, the angular velocities of the motor and the load are different, and this is presented with the relation [6,7]

$$M_o = c_s \Delta\theta + b_v \Delta\omega \tag{1}$$
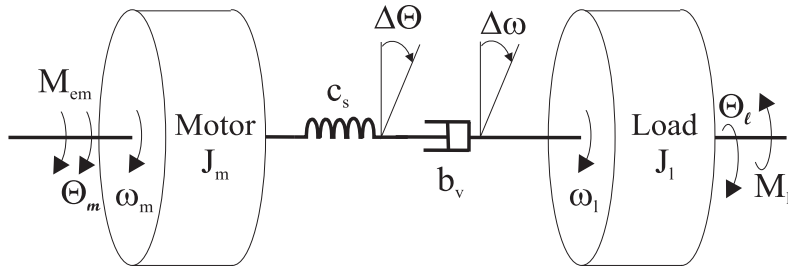
(a) Flexible coupling of motor axle and load
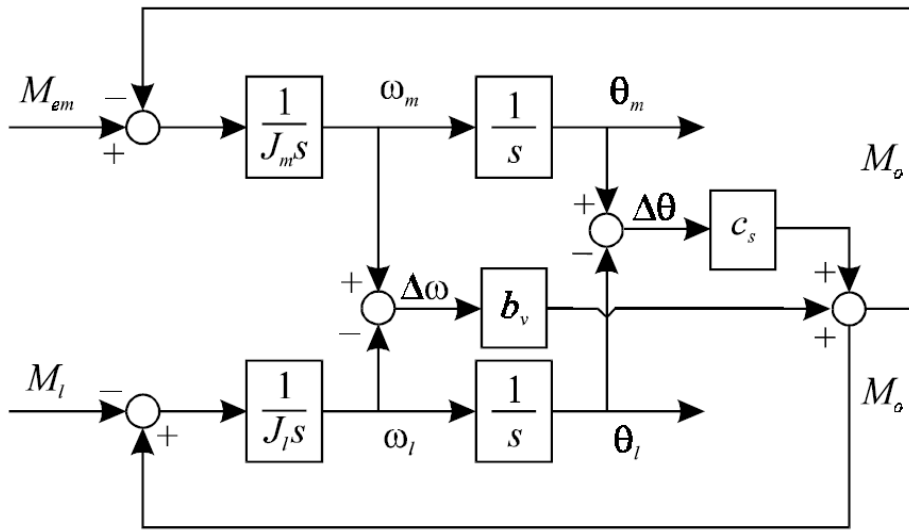
(b) Block diagram of the servo system's plant with the flexible coupling

Figure 1:

So, the traditional approaches in the synthesis of the speed servomechanism starts from a simple plant model, which is presented with the transfer function $1/Js$. However, considering the elastic coupling of the motor and the load, a more precise model is given with the transfer function:

$$W_m(s) = \frac{\omega_m(s)}{M_{em}(s)} = \frac{1}{(J_m + J_l)s} \frac{1 + \frac{b_v}{c_s}s + \frac{J_l}{c_s}s^2}{1 + \frac{b_v}{c_s}s + \frac{J_l J_m}{c_s(J_m+J_l)}s^2}$$

(2)

$$= \frac{1}{(J_m + J_l)s} \frac{1 + \frac{2\varsigma_z}{\omega_z}s + \frac{1}{\omega_z^2}s^2}{1 + \frac{2\varsigma_p}{\omega_p}s + \frac{1}{\omega_p^2}s^2}$$

if the sensor is placed on the axle of motor, or

$$W_l(s) = \frac{\omega_l(s)}{M_{em}(s)} = \frac{1}{(J_m + J_l)s} \frac{1 + \frac{b_v}{c_s}s}{1 + \frac{b_v}{c_s}s + \frac{J_l J_m}{c_s(J_m+J_l)}s^2}$$

(3)

$$= \frac{1}{(J_m + J_l)s} \frac{1 + \frac{2\varsigma_z}{\omega_z}s}{1 + \frac{2\varsigma_p}{\omega_p}s + \frac{1}{\omega_p^2}s^2}$$

if the sensor is placed on the loaded shaft e.g. on the loaded side. In the previous relations (2) and (3), the non-suppressed natural frequencies ω_p and ω_z, and the coefficients of the relative suppression ζ_p and ζ_z, are given by the following equations:

$$\omega_p = \sqrt{\frac{c_s(J_m + J_l)}{J_m J_l}}, \qquad \omega_z = \sqrt{\frac{c_s}{J_l}},$$

(4)

$$\varsigma_p = \sqrt{\frac{b_v^2(J_m + J_l)}{4c_s J_m J_l}}, \qquad \varsigma_z = \sqrt{\frac{b_v^2}{4c_s J_l}}$$

The non-suppresses natural frequencies ω_p and ω_z, are marked as a resonant and an anti-resonant frequency, retrospectively [6], and their quotient presents the resonant ratio:

$$R_r = \frac{\omega_p}{\omega_z} = \sqrt{1 + \frac{J_l}{J_m}}$$

(5)

In the case when the sensor is placed on the axle of the motor, small amounts of the resonant relation reduce influence of the torsional load on the dynamics of the speed servo-mechanism [6]. A large axial mass inertia moment of the rotary mass in the model of the motor prevents a penetration of the torsional oscillations from the load to the motor. However, this approach $(J_m \gg J_l)$, favor performance of the angular velocity

ω_m and the position θ_m of the motor, even though most applications demand good performances on the load side (ω_l and θ_l). These problems, in the case of the placement of the sensors, either on the axle of the motor or on the load side, are analyzed in detail in [6]. It is the emphasized tendency that desired bandwidth in the speed servomechanisms in the modern mechanical processing units almost covers frequencies of the torsional resonance [6]. The resonance is a steady state phenomenon that occurs when the motor's natural resonant frequencies are excited at particular velocities. For example, if we slowly increase the motor's speed, we may notice ,,rough" spots at certain speeds. But, it is not caused by the transient reference inputs. The resonance is affected by the load. Some loads are resonant and can make the motor resonance worse. Other loads can damp the motor resonance. Unlike the resonance, ringing is a transient phenomenon that can be caused both by the acceleration or deceleration to a reference velocity. Namely, when controlled to quickly accelerate to a given velocity, the motor shaft can ,,ring" about that velocity, oscillating back and forth. As the resonance, the ringing causes an error in the motor shaft position. Also, the ringing (or vibration) can cause an audible noise.

So in [6], besides the presented overview of the situation in this field, a special attention was paid to the passive approaches in compensation of the resonant modes in the feedback system. That is why, as it is depicted in Figure 2, a cascade compensator is developed for annulations or averaging of the resonant modes, produced due to the elastic coupling inside the mechanical subsystem. The regular cascade filter (Figure 2) is a notch filter. For the synthesis of the notch filter, it is necessary to have the values ω_p and ζ_p. This assumption presents a great problem in realization of practical applications based on the implementation of the anti-resonant compensator of the type notch filter. In [6,7] a simpler and more robust anti-resonant filter is presented, with only one adjustable parameter:

$$W_{NF}(z^{-1}) = \frac{1 + z^{-n}}{2}, \;\; n = \frac{T_{osc}}{2T} \tag{6}$$

where n is a relation of the sub-period of oscillations of the resonant mod and the sampling period. The period of oscillations,

$$T_{osc} = \frac{2\pi}{\omega_p \sqrt{1 - \varsigma_p}} \tag{7}$$

which is generally adjustable parameter of the FIR filter, (6) could experimentally be defined. The idea of the synthesis of the filter (6) was elaborated in [6].
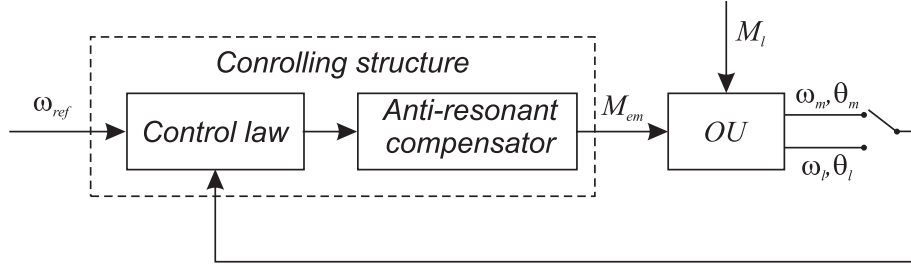

Figure 2: System with anti-resonant compensator

3 Modification of the impact structure and suppression of torsional oscillations

In Figure 3, a special case of the IMPACT structure, when the controlled plant does not contain transport time delay, is presented. This structure could be implemented in a case of the digitally controlled electro-motor drives [1,3]. The signal w_M models an influence of the load moment on the system output y (e.g. a signal of the angular velocity or position, regarding to the speed or position of the controlled servo-mechanism, retrospectively). The control part of the structure in Figure 3 is presented in the function of the polynomial by complex variable z^{-1}. The plant $W_{ou}(s)$ could be presented via its nominal discrete model:

$$W^o(z^{-1}) = \frac{z^{-1-k}P_u^o(z^{-1})}{Q^o(z^{-1})}$$

This is explicitly implemented in the structure as the nominal two-inputs internal plant model. The signal ε estimates influence of the generated disturbance on the controlled variable, e.g. the influence of the external disturbance and uncertainty of the nominal plant model. The model impreciseness could be adequately described by a multiplicative bound of uncertainty $\alpha(\omega)$[4]
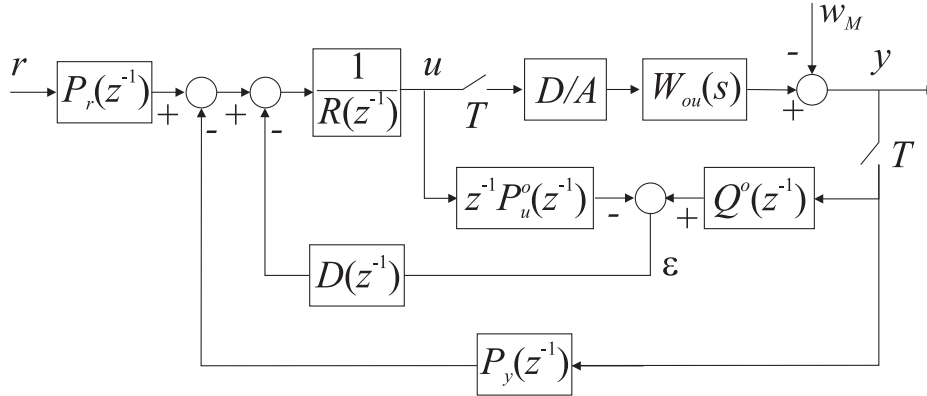
$$W(z^{-1}) = W^o(z^{-1})(1 + \delta W(z^{-1}))$$

Figure 3: IMPACT structure of digital controlled system

$$\left| \delta W(e^{-j\omega T}) \right| \leq \alpha(\omega), \;\; \omega \in [0, \pi/T] \tag{8}$$

Then, the system depicted in Figure 3 fulfills demand of the robust stability, if the nominal system was stable and the relation fulfilled

$$\alpha(\omega) < \left| \frac{Q^o(z^{-1})R^o(z^{-1}) + z^{-1}P_u^o(z^{-1})P_y(z^{-1})}{z^{-1}P_u^o(z^{-1})(P_y(z^{-1}) + Q^o(z^{-1})D(z^{-1}))} \right|_{z^{-1}=e^{-j\omega T}}, \;\; \omega \in [0, \pi/T].$$

This is practically a condition of the system working ability. The robust performance of the system could be achieved by functioning of the internal contour of the system. The task of the inter contour of the system is to suppress the effect of the generated disturbance on the output variable. The internal model of disturbance is designed according to the absorption principle [3,4] with the introduction of signal ε. In the case of the controlled plant, which does not contain a transport lag, the internal model of disturbance is guided to the prediction polynomial $D(z^{-1})$[3,4]

$$(1 - D(z^{-1}))\varepsilon(z^{-1}) = 0, \quad t = nT \geq (\deg(1 - D(z^{-1})))T$$

The purpose of the absorption principle is to include the disturbance model in the control structure in order to suppress influence of the disturbance on the stationary value of the controlled variable. In the sufficient completeness of the information about the disturbance, the prediction polynomial $D(z^{-1})$ is defined simply starting from the disturbance

model, in time domain [3,4]. In Tsypkin's work, commonly used prediction polynomial is

$$D(z^{-1}) = 2 - z^{-1} \qquad (9)$$

and refers to the class of the linear disturbance, and its implementation in the IMPACT structure gives efficient results for different classes of the slowly-changing disturbances. A smaller period of sampling, has justification in the linear approximation of the arbitrary signal in a limited time range [1]. According to the standard procedure of synthesis [3], for minimal phase plants

$$R(z^{-1}) = P_u^o(z^{-1}) \qquad (10)$$

The external contour determinates a dynamic behavior of the nominal system and it could be designed based on the defined function of the coupled transmission of system $G_{de}(z^{=1})$. By selecting a desired function of the discrete transmission of system $G_{de}(z^{=1})$ and solving the equation [2]

$$G_{de}(z^{-1}) = \frac{z^{-1}P_r(z^{-1})}{Q^o(z^{-1}) + z^{-1}P_y(z^{-1})} \qquad (11)$$

we get the polynomial of the control structure $P_y(z^{-1})$ and $P_r(z^{-1})$, which finishes the process of the parametric synthesis. However, as it was noticed in [2], the internal model of the control structure increases sensitivity of the system on the quantization noise, specially in the speed servo-mechanism. The suggested modification of the structure, which includes an observer for the estimation of the signal of feedback, is presented in [2]. A high quality observer was implemented with an ability to evaluate a coordination of the stage during the influence of the disturbance and suppress the effects of the quantization noise in the system. The prediction polynomial continues to be an independent structure part and its synthesis simply goes according to the needs of the absorption of the arbitrary class of the disturbance. The simpler and no less efficient solution is presented in [5]. A clear relation between the prediction polynomial signal and the Newton's predicator is established [8]. It was suggesting usage of the prediction polynomials filters [8] instead the classical prediction polynomials. The efficiency of the polynomial LSN (Linear Smoothed Newton), in the role of the prediction polynomial IMPACT, was researched. It was shown that changing of one adjustable parameter of the LSN predictor could simply influence the dynamical characteristics

of the system, as well as the suppression of the quantification noise and the field of the robust system stability [5].

In order to solve our problem (Figure 1), in this paper, we present a structure consisting of the simplest RLSN (Recursive Linear Smoothed Newton) predictor (Figure 4) instead of the prediction polynomial. The parameter c_p¡1 enables that the RLSN predictor has an amplitude-frequent characteristic of the NF filter. It is shown, in [1], that the changes of this parameter (c_p) could influence the expansion of the robust stability area in the field of the medium frequencies. The synthesis of the IMPACT
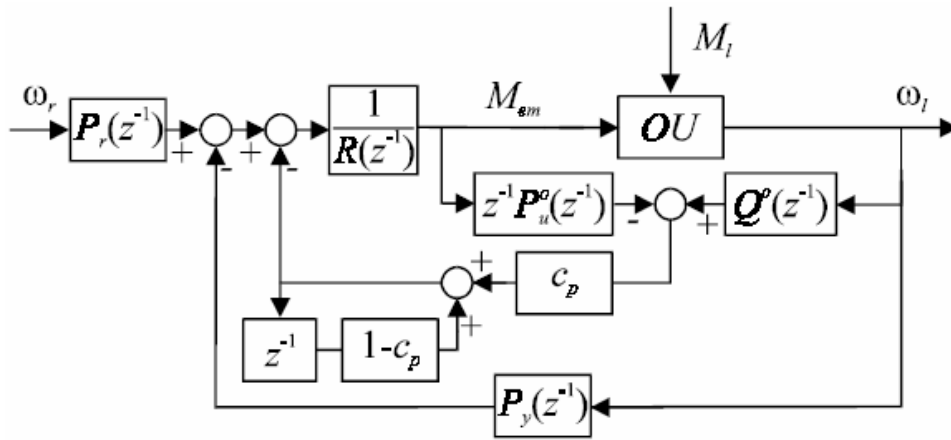

Figure 4: Modification of IMPACT structure of digital speed servo mechanism

structure starts from the following model of the controlled plant

$$W_{ou}(s) = \frac{1}{Js} = \frac{1}{(J_m + J_l)s}, \quad W_{ou}(z^{-1}) = \frac{T}{J_m + J_l} \frac{z^{-1}}{1 - z^{-1}} \qquad (12)$$

although the elastic coupling (Figure 1), in the analysis of the work of the structure, will not be neglected (the effects of the coupling are presenting as a component of the generated disturbance). The selection of the sampling period is coupled with a period of the torsional oscillations, so

$$T = \frac{T_{osc}}{8} = \frac{\pi}{4\omega_p\sqrt{1 - \varsigma_p^2}} \qquad (13)$$

The structure on the internal model of disturbance is adjusted to absorption of the effects of the step disturbance class (Figure 1). Because a very small sampling period is selected, this internal model of disturbance is an adequate solution for a wider class of disturbance. An adjustment of the parameters c_p simply influences the efficiency of absorption of the disturbance effects or the expansion of area of the robust stability and suppression of the torsional oscillations. This will be shown in the following simulations.

According to: 1) the plant nominal model (the AC motor with load) (12), 2) the desired spectrum of the system poles, which could be defined based on the given coefficient of the relative suppression ς and the unsuppressed natural frequency ω_n, and 3) the required zero steady-state error for the step reference signal; the following polynomial are calculated: $P_r(z^{-1})$ and $P_y(z^{-1})$ according to the desired transfer function of the feedback system

$$G_{de}(z^{-1}) = \frac{(1 - (z_1 + z_2) + z_1 z_2)z^{-2}}{1 - (z_1 + z_2)z^{-1} + z_1 z_2 z^{-2}} \quad \left(= \frac{z^{-1}P_r(z^{-1})}{Q^o(z^{-1}) + z^{-1}P_y(z^{-1})} \right)$$

using the relations

$$
\begin{aligned}
P_r(z^{-1}) &= (1 - (z_1 + z_2) + z_1 z_2)z^{-1} \\
P_y(z^{-1}) &= 1 - (z_1 + z_2) + z_1 z_2 z^{-1}
\end{aligned}
\tag{14}
$$

where

$$z_{1/2} = e^{s_{1/2}T}, \quad s_{1/2} = -\varsigma\omega_n \pm j\omega_n\sqrt{1 - \varsigma^2} \tag{15}$$

It should be considered that the selection of the polynomials of the coefficients of the relative suppression ς and the unsuppressed natural frequency ω_n influence the dynamical characteristics of the system, on one hand, and the robust and the filter characteristics of the system on the other hand. Further modification of the structure given in Figure 4 consists of the modification of the pre-filters $P_r(z^{-1})$

$$P_r(z^{-1})\prod_{i=1}^{n}(1 - b_i z^{-1})^i \left/ \prod_{i=1}^{n}(1 - b_i)^i \right.$$

in order to achieve an increased speed of response to the defined value, and remain the achieved quality of stationary oscillation kinetic state
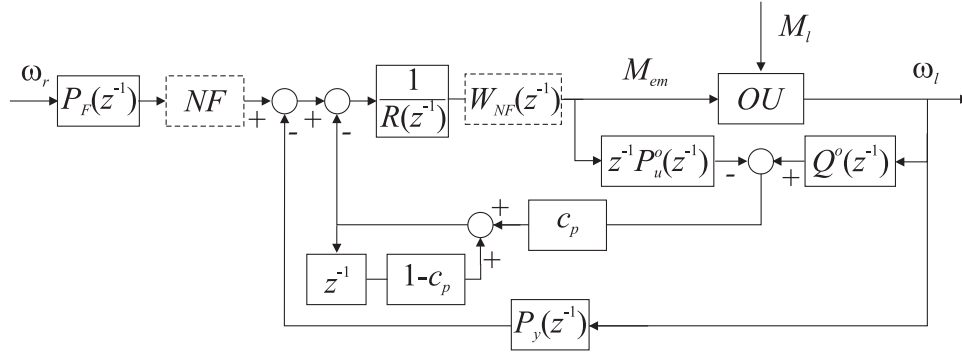
Figure 5: Modification of the IMPACT structure

and the area of the robust stability [1]. In order to extend the area of the robust stability, the prefilter is extended with the factors

$$\prod_{i=1}^{n} (1 - b_i z^{-1})^i, \quad 0 \le b_i \le 0.9$$

accompanied with an effort that parameters n and b_i have the smallest possible value. The value b_i increases to its maximum before the value of the parameter n, is increased. This process lasts until the criteria of the robust stability of the system is reached. The engineering approach, suggested in this paper, means that the prefilter $P_F(z^{-1})$ is previously set to ensure a considerable expanding of the bandwidth of the system with a closed feedback, which will be persevered with re-adjustment of the parameters of the NF filters, and that way, enable a compromise between the speed and the oscillatory of the response from the defined value. The essence is that the expansion of the bandwidth, using a selection of the coefficients of the relative suppression ς and the unsuppressed natural frequency ω_n, of the system with a closed feedback influence the reduction of the area of the robust stability. So, the prefilter $P_F(z^{-1})$ cancels good suppressed poles of the system and introduce the new "faster" poles. That way, the robust stability remains, but the expansion of the bandwidth influences the robust performance of the system (its manifestation is a less suppressed response than the defined value). The oscillatory response from the defined value could be controlled by an insertion of the NF filter as an integral part of the prefilter (Figure 5). By selecting theNF

filter with one adjustable parameter, with a clear physical meaning, we can easily control the oscillatory of the response forming a defined value.

One possible approach to achieve the suppression of the torsional oscillations in the continual response of the system is an implementation of the FIR filter (6), suggested in [6,7], depicted in Figure 5. In this case, the implementation of the prefilter $P_F(z^{-1})$ is mandatory; on the other hand, the NF filter (discussed above) is not mandatory. However, like it was noticed in [1], a wrong selection of the parameter n of filter (6), could have a large influence on the system. For example, the selection of

$$\frac{T_{osc}}{2T} < n \qquad (16)$$

may lead to a destabilization of the system.

4   Illustrative example

The efficiency of the suggested structure, presented in Figure 4, will be illustrated with a simulation, under the same conditions as performed in [6]. In [6], two identical motors are interconnected by an elastic hollow shaft. The motors are independently controlled and used as a motor and a load. The electro magnetic resolver is placed on both of them. We distinguish the following important data J_m=0.000620kgm^2, J_l=0.000220kgm^2, $c_s = 350\,Nm/rad$, $b_v = 0.004\,Nms/rad$.

A synthesis of the control algorithm is required for insurance that a continual step response of the system has no error in the stationary oscillation kinetic state with a coefficient of the relative suppression ζ=0.7 and an unsuppressed natural frequency ω_n=400 rad/s. The following are polynomials of the control structure

$$R(z^{-1}) = P_u^o(z^{-1}) = 0.636881, \qquad Q^o(z^{-1}) = 1 - z^{-1}$$

$$P_r(z^{-1}) = 0.03941938z^{-1},$$

$$P_y(z^{-1}) = -0.701702946 + 0.741122325z^{-1}$$

The sensor is placed on the loaded shaft (Figure 4) and with the selected characteristic c_p=0.2 the result of the simulation is presented in Figure 6. In Figure 7, the influence of the variation of the parameter c_p=0.2 on the

performances of the system is presented. These results are acquired using the software module SIMULINK, an integral part of software package MATLAB. The same amplitudes as in [6] of the input signals are used ($\omega_r(t)$=3·h(t-0.05) [rad/s], $M_l(t)$=1·h(t-0.1) [Nm]). The simulations are performed with a plant, which structure is defined in Figure 1. The results represented in Figure 6 are slightly better then the analog results (the simulation and the experimental results) in [6]. In the case that
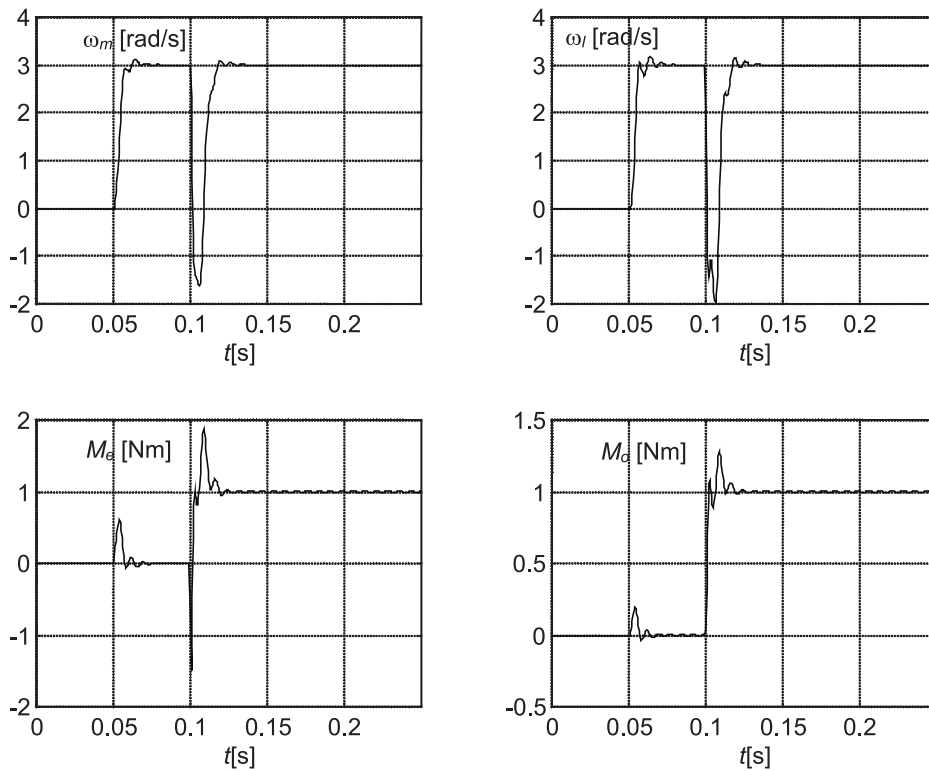

Figure 6: Illustration of the work of structure from Figure 4 ($c_p = 0.2, \omega_n = 400 rad/s, \zeta = 0.7$)

the system possesses a wider bandwidth, for example ζ=0.7 and ω_n=2000 rad/s (ω_n¿ω_p, and that the difference is 37%), we will have the polynomial
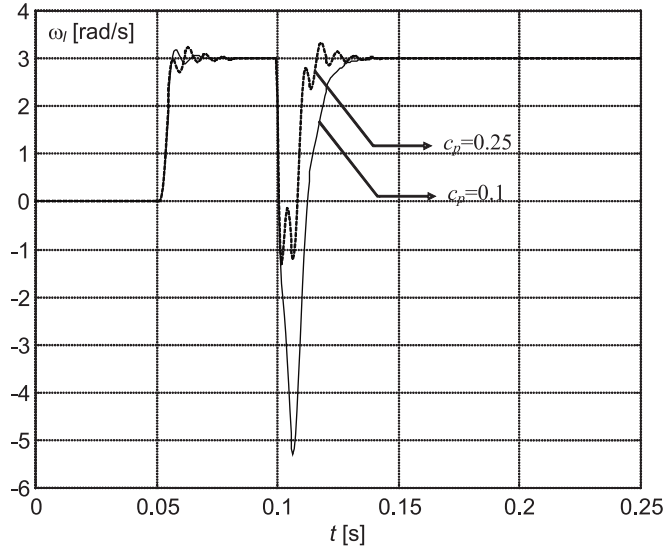
Figure 7: Angular velocity on the loaded shaft – the influence of the variation of the parameter a c_p

control structures

$$P_r(z^{-1}) = 0.54079z^{-1}, \quad P_y(z^{-1}) = -0.3172015325 + 0.22358851z^{-1}$$

but it leads to a destabilization of the system, what is the expected result. The result is a usage of the prefilter

$$P_F(z^{-1}) = \frac{1 - 1.701702946z^{-1} + 0.741122325z^{-2}}{1 - 0.68279846z^{-1} + 0.22358850965z^{-2}} 0.54079z^{-1}$$

like one in the Figure 5, while the result of the simulation is presented in Figure 8.

5   Conclusion

Several concepts of the modification of the IMPACT structure of the speed servomechanisms of high performances are suggested. The goal was the suppression of the torsional oscillations in the system. These

oscillations could appear as a result of the mechanical response produced by the elastic coupling of the motor-load. The simplest RLSN predictor, with one adjustable parameter (c_p), is used in the role of the internal model of disturbance. The results of the simulations clearly present an advantage of the described approach comparing to the previous solution of the same problem. The main advantages are the simplicity of adjustment of the parameters and the realization of the desired performances.
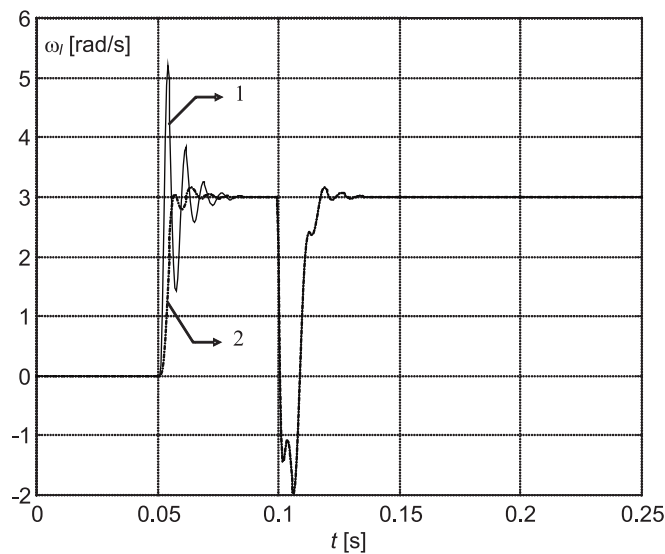

Figure 8: Angular velocity on the loaded shaft - $c_p = 0.2$, 1 – structure with the prefilter $(\omega_n = 2000 rad/s)$; 2 – structure without the filter $(\omega_n = 400 rad/s)$

References

[1] M.S. Matijevic, Development of new structures of digitally controls electromotor drives and industrial processes, PhD thesis, Faculty of Mechanical Engineering, Kragujevac, 2001. (in Serbian)

[2] M.S. Matijevic, M.R. Stojic, Synthesis of digital control system for servomechanism with internal model: suppression of quantization

noise, In Proc. XLII Conf. ETRAN, Vrnjacka Banja, (1998), 291-294,(in Serbian)

[3] Ya.Z Tsypkin, Design of robust optimal control systems having plants with bounded parameter uncertainties, , Avtom.Telemech., N^o9, (1992.), 139-159, (in Russian)

[4] Ya.Z. Tsypkin and U. Holmberg, Robust stochastic control using the internal model principle and internal model control, Int. J. Control, vol. 61, N^o4, (1995), 809-822,

[5] M.S. Matijevic, M.R. Stojic and M.Z. Stefanovic, Application of internal models in the design of digitally controlled electrical drives, In Proc. of 11th Mediterranean Conf. on Control and Automation MED'03, Rhodes, Greece, (2003. June 17-20)

[6] S.N. Vukosavic and M.R. Stojic, Suppression of torsional oscillations in a high-performance speed servo drive, IEEE Trans. on Industrial Electronics, vol. 45, N^o1, (1998.), 108-117

[7] M. Itoh and H. Yoshikawa, Vibration suppression control for an articulated robot: Effects of model-based control applied to a waist axis, International Journal of Control, Automation and Systems, vol. 1, No. 3, (2003)

[8] S.Valiviita, Ovaska and O.Vainio, Polynomial predictive filtering in control instrumentation: A review, IEEE Trans. on Industrial Electronics, vol. 46, N^o5, (1999), 876-888.

[9] M.S. Matijevic, M.R. Stojic: New approach of parameter synthesis of IMPACT structure, In Proc. XLVI Conf. ETRAN, Teslic, (2002.) (in Serbian)

[10] M.S. Matijevic, G. Jakupovic, J. Car, Computer aided measurement and control, Faculty of Mechanical Engineering, Kragujevac, (2005.) (in Serbian)

[11] M.S. Matijevic, M.R. Stojic, IMPACT structure – possible improvement and applications, Tehnika-Elektrotehnika, vol.54, N^o5, Belgrade, (2005.) (in Serbian)

Potiskivanje torzionih oscilacija korišćenjem impact upravljačke strukture

UDK 539.42, 539.421

U radu su razmotrene mogućnosti strukturne sinteze brzinskog servomehanizma sa asinhronim motorom u ulozi izvršnog organa i elastično spregnutim opterećenjem. Elastično spregnute mase: motor – opterećenje dovode do moguće pojave mehaničke rezonanace, što dalje uzrokuje pojavu prinudnih torzionih oscilacija u sistemu. Analizirana su prethodna rešenja data u literaturi i predložena su nova, koja daju bolje rezultate, a zasnovana su na modifikacijama IMPACT (Internal Model Principle and Control Together) strukture. Predložena upravljačka struktura je jednostavna za primenu i ima mali broj podešljivih parametara, kojima se jasno i jednostavno može uticati na robustne, filtarske i dinamičke performanse sistema.