ELIMINATING INSTABILITIES IN COMPUTER CONTROLLED MOTION CONTROL SYSTEMS CAUSED BY TORSIONAL RESONANCE

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Abstract: Torsional resonances are encountered in high performance motion control systems because the mechanical coupling between the motor and the load has finite stiffness. The presence of torsional resonance in motion control systems limits the maximum achievable performance and causes undesirable oscillations in the control system response. The correct application of the IMPACT (Internal Model Principle and Control Together) can eliminate the instability introduced by the torsional resonance and therefore permit higher performance levels to be achieved. This paper outlines the control strategies for compensating torsional resonance in high performance servo drives, and illustrates advantages of the proposed IMPACT structure. The special case of the IMPACT structure is proposed in order to establish the damping effect on the mechanical part.

Keywods: Torsional resonance, Oscillation suppression, IMPACT strucutre, Controlled electrical drives, Robust control, Distrubance rejection.

1. INTRODUCTION

Most of the machine centers, industrial robots, servomechanisms and other rotating machinery have the geared reduction mechanisms between output shafts of motors and driven machine parts. The insufficiency of the torsional stiffness of the geared reduction mechanism often induces transient vibrations mainly related to eigenvalues of the mechanical parts in the lower-frequency range when the motor starts or stops. Elastic couplings and joints within the machine system are major impediments to the performance enhancement, since high loop gains often destabilize torsional resonance modes associated with the transmission flexibility. Vibration suppression of rotating machinery is an important engineering problem [1-7].

The problem on torsional oscillation suppression and disturbance rejection in flexible system originates in steel rolling mill systems, where the load is coupled to the driving motor by long shaft. The small elasticity of the shaft is magnified and has a vibrational effect on the load speed. Vibrations caused by the load impact and the step input endanger the integrity of the mechanical structure and detriorate the product quality. This vibration is not only undesirable but also the origin of th instability of the system in some cases. As the newly required speed response is very close to the first resonant frequency, the conventional controllers are not longer effective [3,5,6].

To overcome the problem, various control strategies have been proposed, that may be divided into the following three groups [3]: 1) control strategies based on the direct measurement of motor- and load- side variables, 2) strategies involving only one feedback device attached to the motor and the observer that estimating remaining states [5,6], and 3) vibration supression strategies based upon the notch filtering and phase-lead compensation applied in conventional control structures [3]. In this paper, a brief review of them will be given. Next, a new control strategy based on IMPACT structure will be proposed as a simple and practical control algorithm. The proposed control technique should be to satisfy new requirements in the quality of control, i.e., 1) faster speed control response, 2) disturbance rejection on the load speed, and 3) robustness to parameter variations including gear backlash. This paper will show that the IMPACT structure [7-10] is suitable for suppressing of torsional oscillations in servo drives with flexible coupling. The special case of the IMPACT structure will be proposed.

2. TORSIONAL RESONANCE AND SERVO SYSTEM WITH FLEXIBLE COUPLING

Resonance is a steady state phenomenon that occurs when motor's natural resonant frequencies are excited at particular velocities. For example, if we slowly increase motor's speed, we may notice „rough“ spots at certain speeds. The „roughness“ is resonance (Fig.1). But, it is not caused by transient reference inputs. Resonance is affected by the load. Some loads are resonant, and can make motor resonance worse. Other loads can damp motor resonance. Unlike resonance, ringing is a transient phenomenon, that can be caused both by accelerating or decelerating 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 resonance, ringing causes error in motor shaft position. Also, ringing (or vibration) can cause audible noise.

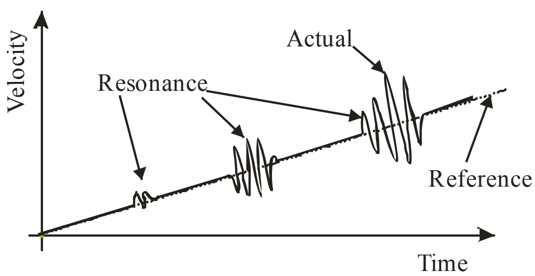

Fig.1 Illustration of a resonance phenomenon in servo drives

In order to slove these problems, system designers will sometimes attach a damping load, such as an inertial damper, to the back of the motor. However, such a load has the undesired effects of decreasing overall performance, and increasing system cost. Again, designers of the control part of a servo system, usually use the simplest motor/load models that haven't information about resonance modes and fast dynamics.

The more realistic model of an AC motor with load is illustrated on Fig.1. and Fig.2, as a two-mass motor/load system with flexible coupling.

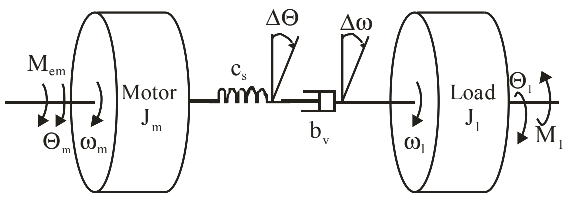

Fig.2. Flexible coupling of the motor shaft and load

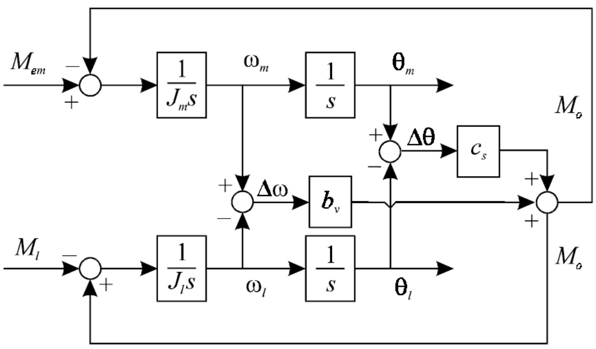

Fig.3. Block diagram of the servo system's plant with flexible coupling

The electromagnetic torque M_{em} is control variable, and the tork on loaded shaft M_l presents disturbance. The motor inertia J_m and load inertia J_l are coupled by the shaft or the transimssion system having a finite stiffness coeficient c_s. The friction coefficient b_v generally assumes very low values, giving rise to weakly damped mechanical oscillations [3]. The torsional torque M_o equals the load torque M_l only in the steady state. During transients, the speeds of motor and load differ, and torsional torque M_o is given by

$$M_o = c_s \Delta\theta + b_v \Delta\omega \qquad (1)$$

Contrary to the traditional model $W_m(s)=1/(J_l+J_m)s$, if the shaft sensor is mounted on the motor, the transfer function of the mechanical subsystem is defined by

$$W_m(s) = \frac{\omega_m(s)}{M_{em}(s)} = \frac{1}{(J_m+J_l)s}\frac{1+\frac{b_v}{c_s}s+\frac{J_l}{c_s}s^2}{1+\frac{b_v}{c_s}s+\frac{J_lJ_m}{c_s(J_m+J_l)}s^2} = \frac{1}{(J_m+J_l)s}\frac{1+\frac{2\varsigma_z}{\omega_z}s+\frac{1}{\omega_z^2}s^2}{1+\frac{2\varsigma_p}{\omega_p}s+\frac{1}{\omega_p^2}s^2} \qquad (2)$$

where undamped natural frequencies (ω_p, ω_z) and relative damping coefficients (ζ_p, ζ_z) are given by

$$\omega_p = \sqrt{\frac{c_s(J_m+J_l)}{J_mJ_l}}, \quad \omega_z = \sqrt{\frac{c_s}{J_l}}, \quad \varsigma_p = \sqrt{\frac{b_v^2(J_m+J_l)}{4c_sJ_mJ_l}}, \quad \varsigma_z = \sqrt{\frac{b_v^2}{4c_sJ_l}} \qquad (3)$$

Undamped natural frequency ω_p and ω_z of the pole- and zero-pairs in (2) are refered as the resonance and antiresonance frequencies [2,3], and their quotient is known as the resonance ratio

$$R_r = \frac{\omega_p}{\omega_z} = \sqrt{1+\frac{J_l}{J_m}} \qquad (4)$$

In the case under consideration, a low value of resonance ratio reduces the influence of torsional load on dynamics of the speed control loop. With $J_m >> J_l$, oscillations of torsional torque are filtered by a large motor inertia J_m and their influence on the control of the motor speed becomes smaller. A damped control of Θ_m and ω_m is favorable, but most applications require fast and precise control of the load variables Θ_l and ω_l [3]. Also, in that case, the estimation of resonance modes from detected signals (Θ_m and ω_m) is not possible, and the load speed and position might exhibit weakly damped oscillations that cannot be disclosed and compensated from the feedback signals [3].

In the case that sensor is mounted on the load shaft, the mechanical subsystem of the drive has the transfer function $W_l(s)$ given by

$$W_l(s) = \frac{\omega_l(s)}{M_{em}(s)} = \frac{1}{(J_m+J_l)s}\frac{1+\frac{b_v}{c_s}s}{1+\frac{b_v}{c_s}s+\frac{J_lJ_m}{c_s(J_m+J_l)}s^2} = \frac{1}{(J_m+J_l)s}\frac{1+\frac{2\varsigma_z}{\omega_z}s}{1+\frac{2\varsigma_p}{\omega_p}s+\frac{1}{\omega_p^2}s^2} \qquad (5)$$

where undamped natural frequencies (ω_p, ω_z) and relative damping coefficients (ζ_p, ζ_z) are given by (3), too.

3. STRATEGIES FOR COMPENSATING TORSIONAL RESONANCE

Many controllers already exist in the field of motion control, but all most of them are designed by assuming an ideal, rigid transmission train [3]. However, the desired speed-loop bandwidth in modern machining centres approaches the frequency of torsional resonance and coincides, at the same time, with most disturbing statistical and deterministic noises. Under these condition, P&I control laws are not suitable. Standard improvement of conventional motion control laws and structures is based on antiresonant compensator inclusion as it is shown on Fig.4. Also, in the literature are proposed model based control approaches, control tehniques based on the disturbance observers, as well as approaches based on two freedom structure based on H_2 control, etc. [2-6,11].

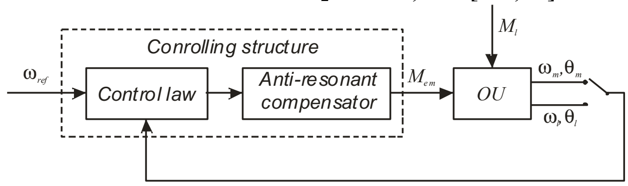

Fig.4. System with antiresonant compensator

H_2 two freedom controller (Fig.5) minimizes functional

(6)

The software for calculation of H_2 controller (G_{fb} and G_{ff}) is available on the cite [11]. In this manner is possible to get more appropriate PI control law ($G_{fb} \equiv G_{ff}$, $G_d \equiv 1$, see Fig.5). But, as one drawback, we can note that H_2 controller design uses trial and error method in tuning parameters to meet the performance specifications (by choosing of different λ and α_h values in the simulation trials).

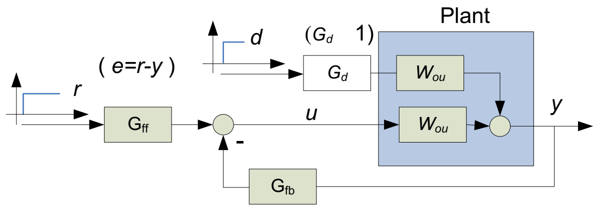

Fig.5. H_2 controlling structure

The notch filter compensator

$$W_{notch}(s) = \frac{s^2 + 2\varsigma_{zz}\omega_{nf}s + \omega_{nf}^2}{s^2 + 2\varsigma_{pp}\omega_{nf}s + \omega_{nf}^2}, \ \varsigma_{pp} \quad \varsigma_{zz} \tag{7}$$

as antiresonance comensator (Fig.4) is most frequently used in practice. The notch filter zeros cancels critical poles (of the torsional load), while the poles of the filter become a new pair of conjugate complex poles with increased relative damping ($\varsigma_{pp} \quad \varsigma_{zz}$). Digital implementation of notch filter ($\varsigma_{pp} = \varsigma_p, \varsigma_{zz} = \varsigma_z, \omega_{nf} = \omega_p$) is given by discrete transfer function

$$W_{notch}(z^{-1}) = \frac{e^{-(\varsigma_p - \varsigma_z)\omega_p T} - 2e^{-\varsigma_p\omega_p T}\cos(\omega_p T\sqrt{1-\varsigma_z^2})z^{-1} + e^{-(\varsigma_p + \varsigma_z)\omega_p T}z^{-2}}{1 - 2e^{-\varsigma_p\omega_p T}\cos(\omega_p T\sqrt{1-\varsigma_p^2})z^{-1} + e^{-2\varsigma_p\omega_p T}z^{-2}} \tag{8}$$

For an exact cancellation of resonance modes, both the resonance frequency and damping factor must be known while tuning all parameters of the notch filter [3]. But, the exact location of critical poles is unknown and, thus, the cancellation is generally imprecise. The notch filter (7) suppresses the resonant mode by the ratio $\varsigma_{pp}/\varsigma_{zz}$. Since a low damping coefficient of zeros increases greatly the snesitivity to parameter variations, the ratio $\varsigma_{pp}/\varsigma_{zz}$ is limited. Hence, the excitation of resonance modes can be only reduced, but not eliminated completely, by the notch serial compensator. The notch compensator si very sensitive to parameter variation, and it presents a serious problem in tuning and implementation the notch filter [3].

As a more robust and more practical solution of vibration supression strategy based upon the structure on Fig.4; in [3] is proposed antiresonance compensator

$$W_{NF}(z^{-1}) = \frac{1 + z^{-n}}{2}, \ n = \frac{T_{osc}}{2T} \tag{9}$$

where the n stands for the ratio between the resonance mode half-period ($T_{osc}/2$) and the sampling time (T) of the discrete time controller. The oscillation period of the resonance mode T_{osc} is given by

$$T_{osc} = \frac{2\pi}{\omega_p\sqrt{1-\varsigma_p^2}} \tag{10}$$

and it is adjustable parameter of the FIR filter (8), that could be experimentally defined. The idea of the synthesis of filter (8) was elaborated in [3]. The conceived cascade antiresonance compensator is simpler, less sensitive to parameter changes, and requiring a setting of only one parameter, but parameter n have to be identified precisely. The theoretical value of the suppression at $\omega_{osc} = 1/T_{osc}$ frequency is infinite, rather then a finite ζ/ζ_p notch filter suppression value.

The resonance ratio control is proposed as an improvement of model-based control techniques (i.e. model following control, application of disturbance observer, time derivate feedback, state feedback control).

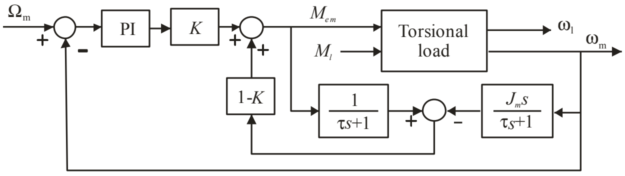

Fig.6. Resonance ratio control

The resonance ratio is defined by relation (4), and should be about $\sqrt{5}$ because of effective vibration suppression. In [6], as a simple and practical strategy, it is proposed the resonance ratio control based on the fast disturbance observer (see Fig.6 and Fig.2), with optimal resonance ratio $0.8\sqrt{5}$. In conventional disturbance observer applications 100% of the estimated disturbances is feed back. In the case on Fig.6, 1-K of the estimated disturbances is used. Parameter K ($0<K\leq1$) and time costant τ (which defines observer's cutoff frequency) are adjustable parameter for vibration suppression. But, as it is previous commented, this control strategy cannot efficiently provide vibration suppression on the load side.

A good review of control strategies of vibration supression is given in [3], where FIR antiresonant filter (9) is proposed as improvement of previous approaches. This solution is investigated and compared with IMPACT structure possibilities in [7].

4. IMPACT STRUCTURE

Fig.7 depicts the special case of IMPACT controlling structure that corresponds to control plants without the transport lag (dead time). Thus the structure may be conveniently applied for digitally controlled electrical drives [7]. In that case, signal w_M modeled the influence of load torque disturbance on system output y which may be shaft speed or angular position depending on the type of servomechanism.

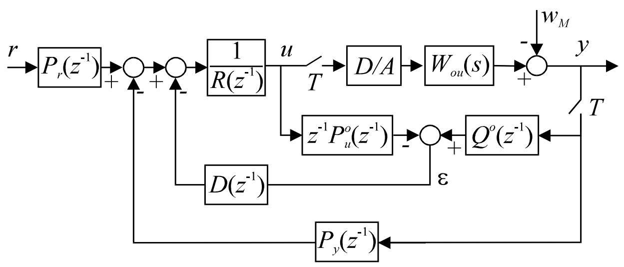

Fig. 7. IMPACT structure of digital control system

The control portion of the system of Fig.7 is given by polynomials in complex variable z^{-1}. In the IMPACT structure, the control plant $W_{ou}(s)$ is given by its simplified nominal discrete model

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

developed at the low-frequency band. This model is included into the control part of the IMPACT structure as a two-input internal plant model. Signal ε estimates effects of generalized external disturbance and uncertainness of nominal plant model on the system output. Uncertainness of nominal plant model can be adequately described by the multiplicative boundary of uncertainness $\alpha(\omega)$

$$\begin{aligned} W(z^{-1}) &= W^o(z^{-1})(1+\delta W(z^{-1})) \\ \left|\delta W(e^{-j\omega T})\right| &\le \alpha(\omega),\ \omega \in [0,\pi/T] \end{aligned} \tag{11}$$

Then the system in Fig. 7 satisfies the condition of robust stability if the nominal system is stable and if the following inequality is fulfilled

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

The robust performance is achieved by the local minor loop of the system in Fig. 7. Namely, the main role of this loop is suppression of effects of the generalized disturbance on the system output. This loop comprises internal model of disturbance implicitly and two-input nominal plant model determined by polynomials $z^{-1}P_u^o(z^{-1})$ and $Q^o(z^{-1})$, explicitly. In the case of a control plant without the dead time, the internal model of disturbance is reduced to the prediction polynomial $D(z^{-1})$.

$$\Phi(z^{-1})F(z) = 0, \qquad t = kT \ge (\deg \Phi)T$$

The choice of this polynomial affects the robust performance of the system and effectiveness in absorption of the given class of disturbance. For example, for constant and ramp disturbances, the proper choice of prediction polynomials are $D(z^{-1}) = 1$ and $D(z^{-1}) = 2 - z^{-1}$, respectively. Smaller sample period, has justification in the linear approximation of arbitrary signal on a limited time range. Thus, polynomial $D(z^{-1}) = 2 - z^{-1}$ refers on calsses of slowly-changing disturbances, too. In that manner, principle of absorption in IMPACT structure is implemented in the minor loop, that enables - estimation of influence of generalized disturbance, its prediction and feedforward compensation [7-10].

According to the standard procedure of IMPACT structure synthesis, for a minimum phase control plant, polynomial $R(z^{-1})$ should be taken on as $R(z^{-1}) = P_u^o(z^{-1})$.

The polynomials $P_r(z^{-1})$ and $P_y(z^{-1})$ in the main external loop of the controlling structure in Fig. 7 determine the dynamic behavior of closed-loop system and these polynomials are determined independently from the design of local inner control loop of the structure. The desired pole spectrum of the closed-loop control system may be specified by the relative damping coefficient ς and undamped natural frequency ω_n of the system dominant poles. In doing so and taking into account the required zero steady-state error for step reference signal, the desired second order discrete closed-loop system transfer function becomes

$$G_{de}(z^{-1}) = \frac{(1-(z_1+z_2)+z_1z_2)z^{-2}}{1-(z_1+z_2)z^{-1}+z_1z_2z^{-2}}$$

where

$$z_{1/2} = e^{s_{1/2}T}, \quad s_{1/2} = -\varsigma\omega_n \pm j\omega_n\sqrt{1-\varsigma^2} \tag{12}$$

Then polynomials $P_r(z^{-1})$ and $P_y(z^{-1})$ are calculated in a straightforward manner from

$$\begin{aligned} P_r(z^{-1}) &= (1-(z_1+z_2)+z_1z_2)z^{-1} \\ P_y(z^{-1}) &= 1-(z_1+z_2)+z_1z_2z^{-1} \end{aligned} \tag{13}$$

However, like it was noticed in [7,10], internal model of control structure (Fig.7) increases sensitivity of the system on quantization noise, specially in speed servosystems. As one solution of this problem, it can be suggested usage of prediction polynomials filters [8] instead the classical prediction polynomials. Generally, the predictive filter is defined as an algorithm that estimates future values of the input signal and suppresses the noise contamination [12]. The relatively simple forms of digital predictive filters corresponding to polynomial disturbances are treated. In this paper we present structure consisting of the simplest RLSN (Recursive Linear Smoothed Newton) predictor (Fig. 8) instead prediction polynomial. Parameter c_p<1 enables that the RLSN predictor has amplitude-frequent characteristic of NF filter. In [7] is shown that changes of this parameter (c_p) could influence on expansion of robust stability area in the field of medium frequencies.

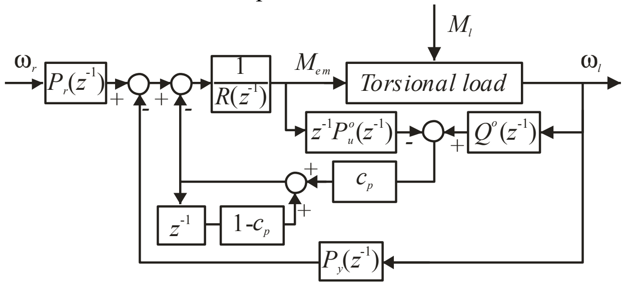

Fig.8. Modified IMPACT structure of digitally controlled speed servosystem

Synthesis of IMPACT structure starts from following plant model (see Fig.2 – flexible coupling is neglected)

$$W_{ou}(s) = \frac{1}{Js} = \frac{1}{(J_m+J_l)s}, \Rightarrow W_{ou}(z^{-1}) = \frac{T}{J_m+J_l}\frac{z^{-1}}{1-z^{-1}} \tag{14}$$

Selection of sampling period is coupled with period of torsional oscillation, so

$$T = \frac{T_{osc}}{8} = \frac{\pi}{4\omega_p\sqrt{1-\varsigma_p^2}} \tag{15}$$

Inner countour of the structure on Fig.8 contains a model of step disturbances. Because very small sampling period is selected, this internal model of disturbance is adequate solution for wider class of disturbance. Adjustment of parameters c_p simply influences on efficiency of absorption of disturbance effects or expansion of area of robust stability and suppression of torsional oscillations. The presented structure is simple with small number of adjustable parameters that could be easily set to achieve the desired robust, filtering, and dynamic properties of the system.

However, possibilities of the structure on Fig.8 to meet robust stability condition by changing parameter c_p are limited. From this reason, on Fig. 9 is proposed modified IMPACT structure.

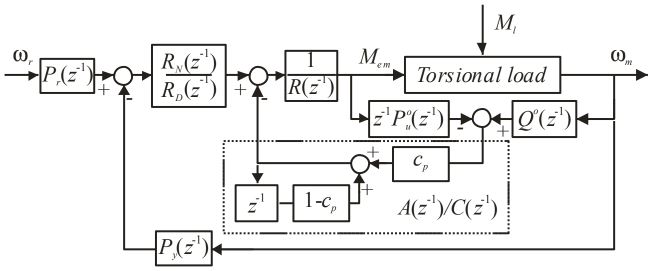

Fig.9. Modified IMPACT structure

Practically, because of elastic drive train, our goal is wide robust stability area about resonant frequency ω_p. In order to lift up frequency curve of the multiplicative bound of uncertainties of IMPACT structure about resonant frequency ω_p, we can choose following polinomial $R_N(z^{-1})$

$$z^{-1}P_u^o(z^{-1}) \qquad (16)$$

where, smaller paremeter ς means wider robust stability area about resonant frequency ω_p. In the special case, in oreder to expand robust stability area about resonant frequency ω_p, polinomial (16) can be factor of implicit disturbance internal model $A(z^{-1})$, too.

The polynomials $P_r(z^{-1})$, $P_y(z^{-1})$ and $R_D(z^{-1})$ in the main external loop of the controlling structure in Fig.9 determine the dynamic behavior of closed-loop system and these polynomials are determined independently from the design of local inner control loop of the structure

$$G_{de}(z^{-1}) = \frac{\left(1-(z_1+z_2)+z_1z_2\right)z^{-2}}{1-(z_1+z_2)z^{-1}+z_1z_2z^{-2}} \qquad (17)$$

Selection of $T_p(z^{-1})$ is free, but poles of polinomial $T_p(z^{-1})$ should be stable and good damped.

As in previous case, according to the standard procedure of IMPACT structure synthesis, for a minimum phase control plant, polynomial $R(z^{-1})$ should be taken on as $R(z^{-1}) = P_u^o(z^{-1})$.

5. ILLUSTRATIVE EXAMPLE

The proposed control algorithms based on IMPACT structure are tested by simulation trials, under the same conditions as performed in [3]. In [2], two identical motors are interconnected by elastic hollow shaft. Motors are independently controlled and used as a motor and a load. The electromagnetic resolver is placed on both of them. We distinguish following important data (see Fig.2) J_m=0.000620kgm^2, J_m=0.000220kgm^2, $c_s = 350Nm/rad$,

$J_m = 0.000620\text{kgm}^2$, $J_l = 0.000220\text{kgm}^2$, $c_s = 350Nm/rad$, $b_v = 0.004Nms/rad$.

Fig.10 shows simulation results of IMPACT structure presented on Fig.8. Results presented on Fig.10 are slightly better then analog simulation and experimental results concerning with antiresonance compensator (9), presented in [3]. The system simulation is performed when the sensor is placed on loaded shaft (Fig.8), with selected characteristic c_p=0.2, and following polynomials of control structure: $R(z^{-1}) = P_u^o(z^{-1}) = 0.636881$, $Q^o(z^{-1}) = 1 - z^{-1}$, $P_r(z^{-1}) = 0.03941938z^{-1}$ and $P_y(z^{-1}) = -0.702 + 0.741z^{-1}$. As in [3], the same input signals ($\omega_r(t)$=3·h(t-0.05) [rad/s], $M_l(t)$=1·h(t-0.1) [Nm]) are used (see Fig.3 and Fig.8). The desired close-loop system transfer function is specified by undamped natural frequency ω_n=400 rad/s and relative damping coefficient ζ=0.7.

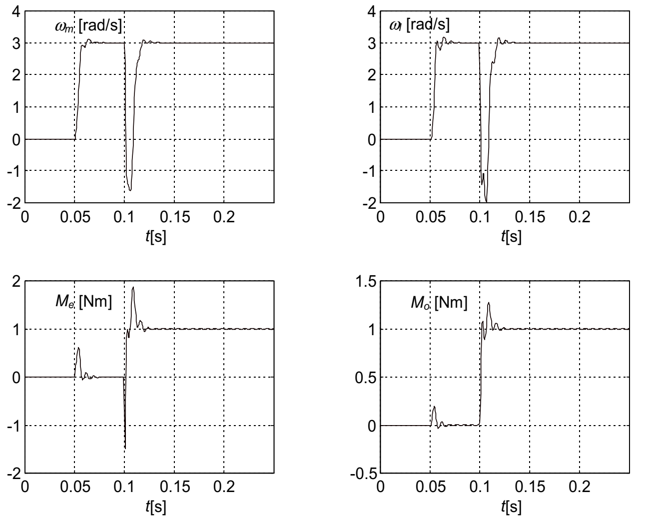

Fig.10. Operation of IMPACT structure on Fig.8 (c_p=0.2, ω_n=400 rad/s, ζ=0.7, $\omega_r(t)$=3·h(t-0.05) [rad/s], $M_l(t)$=1·h(t-0.1) [Nm])

But, our aim is to control the load shaft speed in the presence of torsional vibration, system parameter variation, disturbance torque, and in the absence of a dedicated loadside speed sensor. Robustness and efficiency of IMPACT structure proposed on Fig.9 is illustrated on Fig.11. and Fig.12. As on Fig.9 illustrated, the sensor is mounted on motor shaft, and in that case the system simulation is performed.

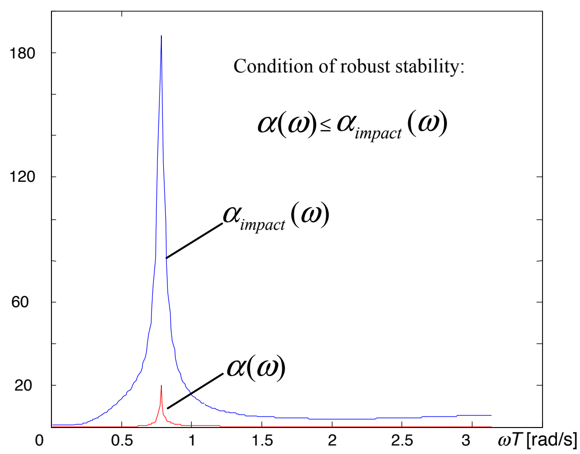

Fig.11. The multiplicative bound of uncertainties of IMPACT structure

Namely, the same experimental conditions as in [3] and the previous case are simulated (plant characteristics, torsional load, input signals). According to described procedure synthesis, polinomials of controling structure on Fig. 9 are defined. As inner contour filter $A(z^{-1})/C(z^{-1})$, again adopted simple RLSN predictor with c_p=0.2. Polinomials $R(z^{-1}) = P_u^o(z^{-1}) = 0.636881$ and $Q^o(z^{-1}) = 1 - z^{-1}$ are same as in the previous case. As the tuning parameter ς of (16) is choosen ς = 0.025 , and polinomial $R_D(z^{-1})$ is $R_D(z^{-1}) = 1 - 1.55951z^{-1} + 0.06594z^{-2}$. In order to can solve Diophant's equation from (17), polinomial $T_p(z^{-1}) = (1 - 0.9z^{-1})(1 - 0.1z^{-1})$ is adopted. From (17) following polynomials of control structure

$R(z^{-1}) = P_u^o(z^{-1}) = 0.63688095$, $Q^o(z^{-1}) = 1 - z^{-1}$

$P_r(z^{-1}) = 0.0394193797z^{-1}$, $P_y(z^{-1}) = -0.701702946 + 0.741122325z^{-1}$

and $P_y(z^{-1}) = -0.055199 + 0.061z^{-1}$ are calculated.

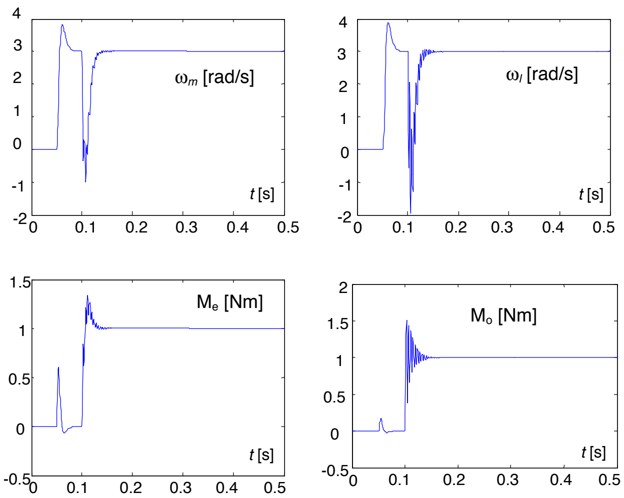

Fig.12. Operation of IMPACT structure on Fig.9 (c_p=0.2, ω_n=400 rad/s, ζ=0.7, $\omega_r(t)$=3·h(t-0.05) [rad/s], $M_l(t)$=1·h(t-0.1) [Nm])

6. CONCLUSION

Mechanical resonance is a current problem in servo systems, and falls into two categories: low-frequency and high-frequency. High-frequency resonance usualy causes instability at the natural frequency of the mechanical system, typically between 500Hz and 1200Hz. Low-frequency resonance occurs more often in general industrial machines, at the first phase crossover, typically between 200Hz and 400Hz. Standard servo control laws are structured for rigidly coupled loads. However, instability results when a such high-gain control law is applied to a flexible coupled servo system. Often, the resultin rigidity of the transmission is so low that instability results when servo gains raised to levels necessary to achive desired performance.

This paper presents several methods for dealing with torsional resonance in servo drives, and proposes a special case of the IMPACT strucutre in order to improve the existing control structures and motion control algorithms to make them compatible with mechanical subsystem. The antiresonant feature of the structure is not based on the exact cancellation of resonance poles. Due to the simplicity and robustness of the proposed structure, it can be easily applied to various flexible systems with different regulator combinations.

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