

ACTIVE CONTROL OF GEARS MODULATED VIBRATIONS IN MECHATRONICS SYSTEMS

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Studiul controlului activ al vibrațiilor modulate în amplitudine sau frecvență a fost realizat în scopul determinării eficienței acestei tehnologii moderne pentru atenuarea vibrațiilor și zgomotului sistemelor cu roți dințate. Testele pentru controlul activ al vibrațiilor modulate au fost realizate în bandă îngustă, cu ajutorul unui dispozitiv de control cu o intrare și o ieșire, folosind un filtru adaptiv FIR (Finite-Impulse-Response). În lucrare sunt prezentate cercetările realizate de autori pentru stabilirea posibilităților de realizare a controlului adaptiv feedforward, cu o singură intrare și o ieșire (SISO), în scopul reducerii vibrațiilor transversale ale unei bare simplu rezemate, excitată de forțe perturbatoare periodice, modulate în amplitudine și frecvență.

The gears vibrations occurring during the operation in the mechatronics systems can be studied in terms of amplitude and/or frequency modulation. The paper presents some theoretical and experimental results in order to determine the feasibility of the active control of the gears vibrations in thin bands with/without resonance. In this purpose, it has been realized in/out control system based on an adaptive filter FIR (Finite-Impulse-Response), using the LabVIEW soft. The theoretical studies evaluate the positioning actuators and transducers optimization in the case of a support shaft gear subjected to amplitude and/or frequency modulated disturbances. The experimental results show the efficiency of the active control of these modulated vibrations.

Keywords: Vibration, Active control, Gears, Mechatronics systems, Actuators

1. Introduction

In the gears from the mechatronics systems, the theoretical and experimental studies [1] highlighted amplitude and/or frequency modulation phenomena of the meshing forces. Also, it has been noticed the combinations of these modulations due to the teeth loads fluctuation and gear angular speed variation. These modulations generate frequency sidebands around the gear fundamentals and harmonics frequencies. Furthermore, the modulation process produces high amplitude vibrations in sidebands of the meshing frequencies,

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determining a noise level increase. The corresponding noise of the frequency sidebands vibration has a significant influence on the global noise generated in the gear mesh process. The noise and vibrations frequency spectrum generated within a mechatronics system have a distributed energy on a certain harmonics components.

The active vibration control difficulties are related to noise in measurement data from sensors, which can cause instability of the process control. The adaptive systems require real time processing of measured data and the establishment of a minimal dynamic model, incorporating a systemic effect of all outstanding disturbances, internal or external to vibrating process.

In this purpose, the research is needed to find solutions to ensure system robustness, which also implies the complication of a control strategy. The active vibration control problem can be interpreted as a problem to eliminate the disturbances achieved by: the feedback control or the feedforward control.

The bandwidth where it can be achieved a significant reduction of error with a feedback control system is limited to an amount less than the inverse of the delay realized in the open loop. The feedforward control is preferred when a reference signal correlated with the disturbance is available and uses an adaptive filter for filtering disturbances.

The aim of the active vibration control study, using the amplitude or frequency modulation concept, is to evaluate the efficiency of this modern technology related to the noise and vibration attenuation of the gear mechatronics systems.

The paper presents authors' research in order to establish the possibilities to create a tool (feed-forward in/out adapting control - SISO). This tool is useful to reduce the transverse vibrations of a simple supported axis, excited by periodic perturbation forces, amplitude and frequency modulated. The tests were realized in thin sidebands using in/out control device with an adapting filter FIR (Finite-Impulse-Response).

2. Theoretical approach

Theoretical research was conducted on a simple system (support shaft gears) in order to establish the active control feasibility of the perturbations produced by the modulation phenomenon.

This is to characterize the vibration behaviour of a simple resonant structure, as a mechatronics component. The theoretical studies estimate the actuators and transducers optimum placement in the case of a simple support shaft gears subjected to the amplitude and frequency modulated perturbation [2].

Fig.1 schematically presents: the position x_p of the perturbation source F_p ; the x_c position of the control source F_c ; the x_{ae} position of the error accelerometer

a_e and the x_{ac} position of the control accelerometer a_c , along the support shaft with l length.

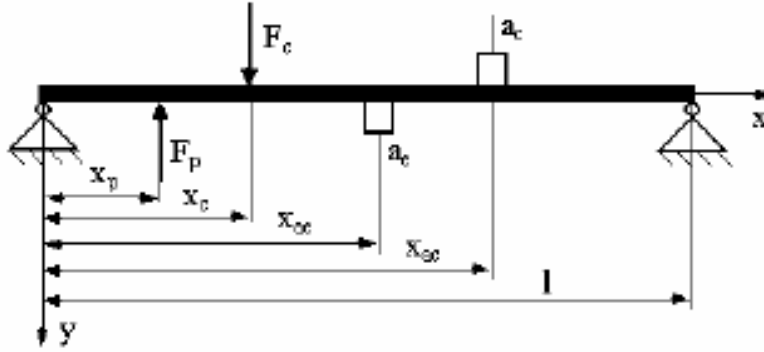


Fig. 1. Perturbation and control sources position on the support shaft gears

The perturbation source was positioned near the shaft support to excite a large number of natural vibration modes. The positions of the control source and error transducer were based on an analytical study, as below described. They can be able to attenuate the harmonic vibrations on a large beam length, not only near the error transducer position.

Considering the shaft composed by n concentrated masses, its transverse vibrations equations for a viscous damping can be expressed in a vectorial form, as follows:

$$[m] \cdot \{\ddot{y}\} + [c] \cdot \{\dot{y}\} + [k] \cdot \{y\} = \{f\} \quad (1)$$

where: $\{y\}$ is the displacements vector; $[m]$ is the inertia matrix; $[c]$ represents the damping coefficients matrix; $[\delta]$ is the influence coefficients matrix; $[k] = [\delta]^{-1}$ is the elastic constants matrix; $\{f\}$ represents the perturbation forces vector.

For a single harmonic perturbation, $F_p(t) = \bar{F}_p \sin \omega t$, with angular frequency ω at $x = x_p$, the shaft response in section x can be determined by the summation of the natural vibration modes:

$$y_p(t) = \sum_{n=1}^{\infty} \frac{\mu_n(x) \cdot \mu_n(x_p)}{M_n \sqrt{(\omega_n^2 - \omega^2)^2 + (2\zeta_n \omega)^2}} \cdot \bar{F}_p \sin(\omega t - \theta_n) \quad (2)$$

where: $\mu_n(x)$ is the natural vibration mode related to the natural frequency ω_n ; ζ_n is the critical damping fraction corresponding to the n natural vibration mode; M_n is the modal mass; θ_n is the phase difference.

Similarly, it can be determined the shaft displacement thanks to the control force $F_c(t) = \bar{F}_c \sin \omega t$ that acts at $x = x_c$.

The shaft response to the simultaneously action of perturbation and control forces can be obtained by summation:

$$y(x, t) = y_p(x, t) + y_c(x, t) = \sum_{n=1}^{\infty} \frac{\mu_n(x) \cdot \sin(\omega t - \theta_n)}{M_n \sqrt{(\omega_n^2 - \omega^2)^2 + (2\zeta_n \omega)^2}} \cdot [\bar{F}_p \mu_n(x_p) + \bar{F}_c \mu_n(x_c)] =$$

$$= \sum_{n=1}^{\infty} a_n(x) \cdot \mu_n(x) \cdot \sin(\omega t - \theta_n) \quad (3)$$

where the coefficients a_n are given by:

$$a_n = \sum_{n=1}^{\infty} \frac{\bar{F}_p \mu_n(x_p) + \bar{F}_c \mu_n(x_c)}{M_n \sqrt{(\omega_n^2 - \omega^2)^2 + (2\zeta_n \omega)^2}} \quad (4)$$

The control estimation can be realized using as cost function the medium value of the amplitude acceleration variation of the transversal vibration $\ddot{y}(x, t)$, on the l shaft length, due to a primarily harmonic perturbation $F_p(t)$ and due to the secondary control force $F_c(t)$, defined by the relation:

$$E = \frac{1}{l} \int_0^l |\ddot{y}(x, t)|^2 dx \quad (5)$$

The cost function expression results from the equations (3) and (5) and, consequently, results the control force that minimize the cost function:

$$E(\omega) = \omega^4 \sum_{n=1}^{\infty} |a_n|^2 \quad (6)$$

The actuator position along the shaft, giving maximum vibration attenuation can be determined by assessing the cost function for different frequencies. If the control force position is the same with the perturbation force,

A digital filter FIR for the reference signal correlated to the vibration signal was used to generate the control force. The digital filter coefficients were adjusted with the least-mean-square LMS algorithm using a piezoelectric accelerometer as error detector. Fig.2 presents the block scheme of the control system. Fig. 3 presents the equivalent diagram of the control system.

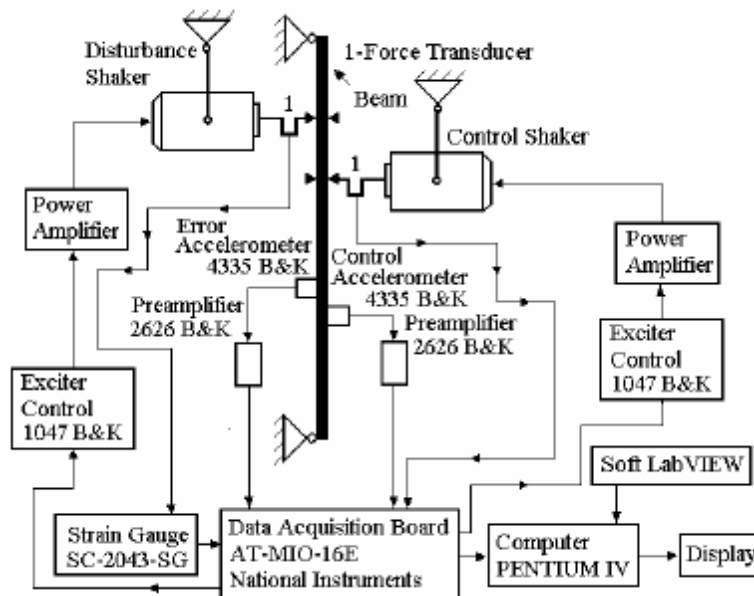


Fig. 2. Block scheme of the control system

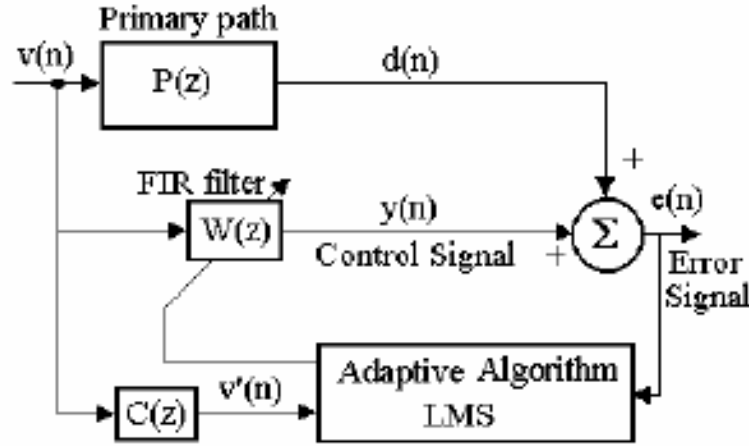


Fig. 3. Equivalent diagram of the control system

The perturbations and the control forces were generated with electrodynamic shakers controlled by virtual signal generators using the acquisition data board AT-MIO-16E (National Instruments) and power amplifiers.

3. Experimental approach

Experimental studies followed the possibilities of active control for vibration frequency modulated (FM) in two cases (one with resonance and the second non-resonant). It was used modulation, a sinusoidal signal with frequency of 10 Hz, which created a narrow band excitation. The shaft response was measured with the control accelerometer in 8 equidistant points. The acceleration power spectrum was recorded in each point. The filter order value was initially determined for each case to determine maximum vibration attenuation. In general, too low order filters cannot simultaneously attenuate the spectral components. Instead, the higher order filters reduce the convergence and the active control performances.

On the other side, the time-varying terms of the adaptive process cannot be eliminated through the reference signal synchronization to an appropriate frequency. This is due to the modulated signal and, as result, more coefficients are required to minimize the time-varying terms. Increasing the size of the filters produced the best signal attenuation up to a point, after which there was a reduction of attenuation. The size filter which produces the most important attenuation may be considered optimal to filter disturbances.

The perturbation signal was generated using the LabVIEW soft with the functions for amplitude and frequency modulation, respectively:

$$y_{AM}(t) = A \sin \omega t + B \sin \omega t \cdot \sin \omega_m t \quad (7)$$

$$y_{FM}(t) = A \sin[\omega t + B \sin \omega_m t] \quad (8)$$

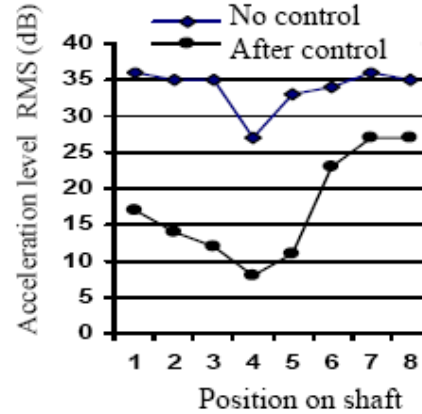
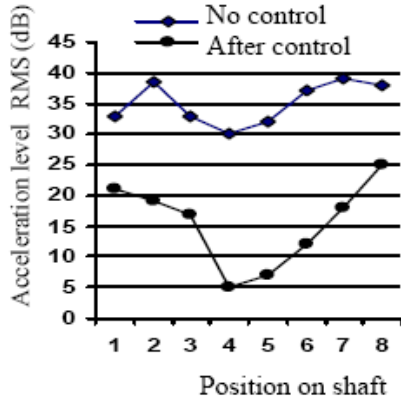


Fig. 4. Acceleration for amplitude modulated signal Fig. 5. Acceleration for frequency modulated signal

The procedures to obtain the perturbation signal are:

- 220 Hz frequency signal was modulated with 10 Hz in amplitude and frequency for the non-resonant forces;
- 325 Hz frequency signal was modulated with 10 Hz in amplitude and frequency for resonance tests.

Fig.4 gives the acceleration size along the shaft for the signal modulated in amplitude. Fig. 5 presents the acceleration for the signal modulated in frequency.

From analysis of the above diagrams it can be noticed that the maximum attenuation of 25 dB is obtain near the error accelerometer.

4. Conclusions

The experimental results analysis highlighted the feasibility of the feed-forward active control of the modulated vibrations.

Higher attenuations of the overall vibrations level were obtained for the resonance perturbations versus perturbations no coincident with the natural frequencies.

The most efficient active control was obtained when the perturbation source position was the same with the control source position.

In general, more the control source is closer to the perturbation source, the controlled frequency band is greater.

R E F E R E N C E S

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