777. Active synthesis of machine drive systems using a comparative method

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Abstract. In this work the method of active synthesis of mechanical systems in accordance with the desired frequency spectrum has been formulated and formalised. Active synthesis of a proportional regulation system has been performed in accordance with the method formulated and a verification of the correctness of the results has been carried out.

Keywords: mechanical impedance, passive synthesis, distribution of the characteristic.

Introduction

The reduction of vibrations in the system can be achieved at both the design level and through adaptation of existing machinery with regard to the requirements of production processes. There are many methods and technologies allowing a reduction of unwanted vibrations of the machine [1, 2, 4, 8-11]. The methods can be divided into the following groups: passive vibration reduction, active vibration reduction and semi-active vibration reduction. The usual method of vibration reduction is through the introduction of dampers (passive vibration reduction) to the object, their task being to increase energy dispersion in the system. Previous work by the authors focused on passive synthesis of systems understood in such a way, i.e. a calculation method, by which a mechanical system with parameters is designed to meet the desired characteristics in the form of chosen resonant and anti-resonant frequencies [1, 3-7]. This resulted in the possibility of synthesis of discrete damped systems using models for viscous damping: proportional to inertial parameters and proportional to the stiffness, and combinations of the above (Rayleigh model). Modern computer technology enables more effective methods of vibration reduction, which is active adjustment. This consists of determining the active force exerted on the system, which neutralises dynamic loads causing vibrations. This work will present the active synthesis of machine drive systems as models of torsional vibrations. Such vibrations are more difficult to detect than flexural ones, which are accompanied by noise and vibrations of the adjacent elements (for example, shaft frames). Due to the absence of symptoms, torsional vibrations are particularly dangerous, as they may be unnoticeable until the destruction of subsystems occurs. The objective of designing an active system is to ensure that the system meets the main conditions for correct operation. The primary criterion here is the stability of the system near the resonant condition of the system.

Synthesis of Cascade Drive Systems

The first step in the synthesis of mechanical systems is the creation of mathematical functions, which on the one hand meet the conditions required for systems, and on the other can be accurately performed in a real system. The method presented in this work for analytical determining of the dynamic characteristics is based on the adoption of the resonant and anti-resonant frequencies (poles and zeros of the desired dynamic characteristic). Therefore, if we take a frequency string in the form of: poles ($\omega_0 = 10$ rad/s, $\omega_2 = 30$ rad/s), zeros ($\omega_1 = 0$ rad/s, $\omega_3 = 20$ rad/s), then the function describing dynamic properties of the torsionally vibrating discrete system can be represented in the form of dynamic flexibility:

$$Y(s) = \frac{(s^2 + 20^2)}{(s^2 + 10^2)(s^2 + 30^2)}, \tag{1}$$

or immobility (mechanical impedance):

$$U(s) = H\frac{(s^2 + 10^2)(s^2 + 30^2)}{s(s^2 + 20^2)}, \tag{2}$$

where: $s = j\omega$, H - any positive real number. The characteristic function, in the form of impedance (2) was used to determine the structure and parameters of the model of the machine drive system. The cascade structure of the proposed system, together with the values of the inertial and elastic elements, was obtained using the method of distribution of the dynamic characteristics into a continued fraction in the following form:

$$U(s) = \frac{s^4 + 1000s^2 + 90000}{s^3 + 400s} = J_1 s + \cfrac{1}{\cfrac{s}{c_1} + \cfrac{1}{J_2 s + \cfrac{c_2}{s}}}, \tag{3}$$

where: $J_1 = 1\left[\text{kgm}^2\right], J_2 = 2.4\left[\text{kgm}^2\right]$ - value of the inertial elements of the sought system, $c_1 = 600\left[\text{Nm/rad}\right], c_2 = 360\left[\text{Nm/rad}\right]$ - values of elastic elements of the sought system. Figure 1 shows a discrete mechanical system as the physical realisation of the synthesised characteristic.

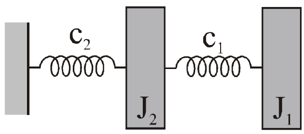

Fig. 1. A model of the drive system corresponding to the characteristic distribution (2)

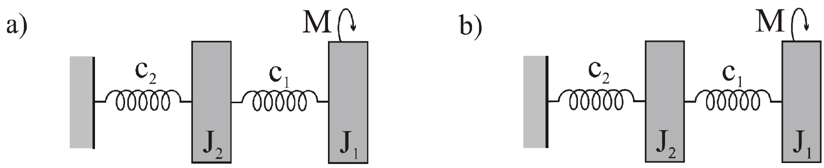

Fig. 2. A model of a drive system with active force:
a) acting on the first inertial element, b) acting on the second inertial element

After determining the parameters and structure of the passive system, it is possible to proceed to determine the force that will allow stabilising and reduction in vibrations of the system in the vicinity of the resonant state of the system. The implementation of the excitation force, as the setting value, can be made in the system, both for the first and the second inertial element – Fig. 2. In considering the desired characteristics (1), the sought force is determined for the first inertial element – Fig. 2a.

For this purpose, the dynamic characteristics (1) are modified, introducing the parameter h for the decrease in frequency of the chosen resonant frequency, in the form:

$$Y1(s) = \frac{(s^2 + 20^2)}{(s + h + j10)(s + h - j10)(s^2 + 30^2)}, \tag{4}$$

and

$$Y2(s)=\frac{\left(s^2+20^2\right)}{\left(s^2+10^2\right)\left(s+h+j30\right)\left(s+h-j30\right)}. \tag{5}$$

As a result of such modifications to the characteristics (1), a reduction in vibrations of the system in the area of the first resonant frequency is achieved - characteristic (4) or the second resonant frequency - characteristic (5). An example of vibration reduction defined in such a way is shown in Fig. 3.

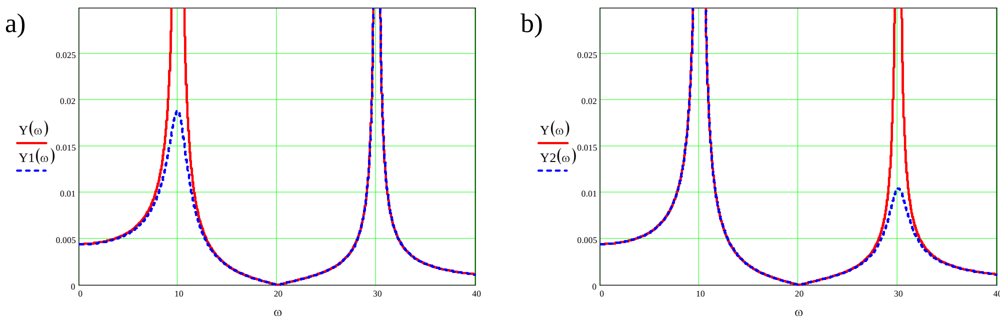

Fig. 3. Dynamic characteristics of a system subjected to active synthesis: a) reduction of the first resonant frequency of the system, b) reduction of the second resonant frequency of the system

There exists also a possibility of reducing vibrations in the area of both resonant frequencies and then the characteristic will take the following form:

$$Y3(s)=\frac{\left(s^2+20^2\right)}{\left(s+h_1+j10\right)\left(s+h_1-j10\right)\left(s+h_2+j30\right)\left(s+h_2-j30\right)}. \tag{6}$$

In order to decrease the vibrations near the resonant frequency of the analysed system, a control law is adopted, allowing the calculation of the excitation force as a function of the force feedback in the following form:

$$M=-\left(k_{p1}\varphi_1+k_{v1}\dot{\varphi}_1+k_{p2}\varphi_2+k_{v2}\dot{\varphi}_2\right), \tag{7}$$

where: k_{p1}, k_{p2}, k_{v1}, k_{v2} - coefficients of the gain of the control system dependant of the position and velocity of inertial elements of the analysed system. In the following part, the method for calculating these coefficients has been shown, which in turn will permit the determination of the control force. For this purpose a block diagram is built of a closed system including the controllers for the force inductors in the system, as shown in Fig. 4.

Dynamic flexibility between the first input and output of the system with the vibration eliminator in the form of a dynamic force is as follows:

$$Y1r(s)=\frac{Y_{11}(s)}{1+Y_{11}(s)\cdot Y_{11r}(s)+Y_{21}(s)\cdot Y_{21r}(s)}, \tag{8}$$

where: $Y_{11}(s)=\dfrac{2.4s^2+960}{2.4s^4+2400s^2+216000}$ - the transfer function between the first input and output of the analysed system in Fig. 2, $Y_{21}(s)=\dfrac{600}{2.4s^4+2400s^2+216000}$ - transfer function between the second output and the first input of the analysed system; $Y_{11r}(s)=k_{p1}+k_{v1}s$ - transfer function of the controller in the force feedback loop from the first movement $Y_{21r}(s)=k_{p2}+k_{v2}s$ - transfer function of the controller in the force feedback loop from the second movement. The value of amplification factors of the control system is determined by comparing the characteristics (4), (5) and (6) with the characteristic of dynamic flexibility of the control system in the force feedback loop (8). As a result of the comparison of characteristics, including expressions with equal powers and taking into account the gain in the characteristic of the analysed system (8), three sets of equations have been determined, which will be used to determine the gain coefficients of the control force. In the case of reduction of the first and second resonant frequency (comparing the characteristics (6) and (8)), it has the form:

$$\begin{cases} k_{v1}=2h_1+2h_2, \\ k_{p1}=h_1^2+4h_1h_2+h_2^2, \\ \dfrac{960k_{v1}+600k_{v2}}{2.4}=2h_1h_2^2+2h_2h_1^2+200h_1+1800h_2, \\ \dfrac{960k_{p1}+600k_{p2}}{2.4}=h_1^2h_2^2+100h_1^2+900h_2^2. \end{cases} \tag{9}$$

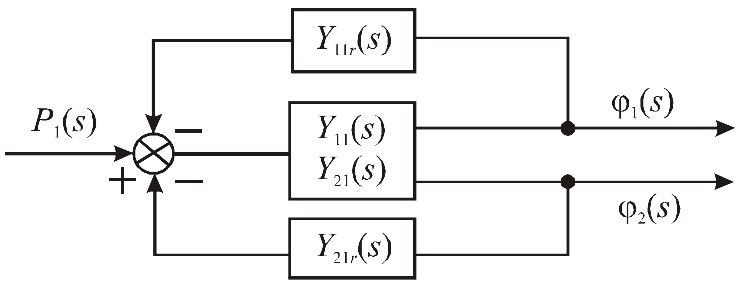

Fig. 4. Block diagram of the analysed control system

Assuming the decline in value of natural vibration frequency h = 1 [rad/s] (considering the last case, the two coefficients were deemed to equal one), the values of gain for the control force will adopt the values indicated in Table 1. After determining all cases of application of the control force on the first inertial element, one can proceed to determine the force acting on the second inertial element (Fig. 2b). For this purpose, the starting point for designating the active force is a characteristic in the form of:

$$Y(s)=\frac{(s^2+24.49^2)}{(s^2+10^2)(s^2+30^2)}. \tag{10}$$

In order to reduce vibrations near the resonant frequency, the parameter for decline in frequency of vibration is introduced:

- for the first resonant frequency:

$$Y4(s)=\frac{\left(s^2+24.49^2\right)}{(s+\mathrm{h}+j10)(s+\mathrm{h}-j10)\left(s^2+30^2\right)}, \tag{11}$$

- for the second resonant frequency:

$$Y5(s)=\frac{\left(s^2+24.49^2\right)}{\left(s^2+10^2\right)(s+\mathrm{h}+j30)(s+\mathrm{h}-j30)}, \tag{12}$$

- for both resonant frequencies:

$$Y6(s)=\frac{\left(s^2+24.49^2\right)}{(s+\mathrm{h}_1+j10)(s+\mathrm{h}_1-j10)(s+\mathrm{h}_2+j30)(s+\mathrm{h}_2-j30)}. \tag{13}$$

Through assuming the control law (7) and setting the dynamic flexibility between the second input and output of the system, the values of the control force have been set - the same way as in the case of forces acting on the first inertial element. Table 1 summarises the gain values of the control force acting on the first and second inertial element.

Table 1. Values of gain of the control system

	$Y1(s)$	$Y2(s)$	$Y3(s)$	$Y4(s)$	$Y5(s)$	$Y6(s)$
k_{p1}	1	1	6	1.2	-2	-10.396
k_{v1}	2	2	4	2.4	-4	-1.584
k_{p2}	2	-1.2	-5.596	2.4	2.4	14.4
k_{v2}	4	-2.4	1.616	4.8	4.8	9.6

Numerical verifications of the results were carried out for the case when the system will be operating with the control force exerted determined for the characteristic (6) – $Y3(s)$. In the simulation, the excitation signal was determined as forces with unit amplitude and circular frequencies corresponding to the resonant frequency of natural vibrations of the system. In addition, it was assumed that the control force is activated after 20 seconds. Movements from the dynamic model are shown in Fig. 5.

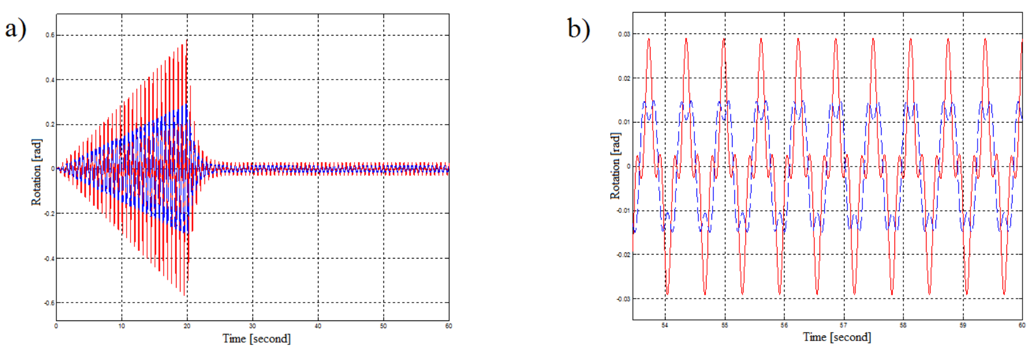

Fig. 5. Movements of the system with excitation linked to the first and second resonant frequency of the system: a) general view, b) zoomed in when in a steady state

Conclusions

The exertion of an active control force is tantamount to the provision of additional energy from the outside. When designing an active vibration reduction system, the values of gain of the control force should be determined so as to achieve the desired effect of reducing vibration at the smallest cost. The presented active synthesis makes it possible to meet criteria determined in such a way. This is due to the large number of gain parameters of the control force obtained through synthesis, which may substantially affect the optimal choice of control parameters.

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