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Active reduction of identified machine drive system vibrations in the form of multi-stage gear units

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1. Introduction

The dynamical processes occurring in machine drive systems are caused by vibrations occurring in the system. They have a direct impact on noise emission, fatigue strength, controllability and stability. Therefore, one of the basic criteria used in designing contemporary mechanical structures are the dynamical properties of the structure described in the form of frequency transfer functions. By knowing them, it is possible to avoid operating the system within resonance ranges which can compromise the machine life and reliability [1-17]. The structural and parametric identification of systems considering the dynamical machine properties is one of the methods for extending its life and improving reliability. This task can be performed by using the algorithms of passive and active synthesis [1–8, 12, 16]. The paper presents the method for determining the structural and dynamical parameters of machine drive systems by using the algorithm of active synthesis. This method makes it possible to choose the parameter and the controlling force of the system to maintain the loads at specified working conditions within rated limits.

The active synthesis of mechanical systems in this paper is understood as a calculation method used to identify the structure of a mechanical system, including its parameters and active force, meeting the desired characteristics in the form of selected resonance and anti-resonance frequencies. The characteristic feature of the obtained active mechanical systems is the fact that the determined parameters do not change over time and the fact that obtaining the assumed properties requires delivering an external source of energy.

The research on the passive and active synthesis of torsionally oscillating mechanical drive systems is developed by the authors of this paper. The introduction to the synthesis of machine drive systems, acting as torsionally oscillating systems, is presented in [1]. This section presents the method of synthesizing the distribution of characteristics on a continued fraction making it possible to obtain the parameters and the model of a single-axis system. In addition, paper [2] present the synthesis of drive systems using the method of characteristic distribution on partial fractions (branched systems), the mixed method (cascade-branched systems). Then, in the case of dynamical models of the whole drive system obtained, the authors determine the controlling force applied on the system that counteracts the oscillation-inducing dynamical loads – active synthesis [3, 4, 6, 8]. Afterwards, in [7] the authors presented an attempt to extend the existing methods of passive synthesis, aimed at obtaining the models of drive systems including multi-stage gear units. This paper constitutes an extension of the latter issue, by the problem of vibration reduction in identified machine drive systems. The reduction method was based on using the active synthesis making it possible to obtain the desired mechanical effect by properly selecting the dynamical properties of the system, including the calculation of active force as a function of system coupling.

2. Idea of active synthesis of non-reduced machine drive systems

The first stage of synthesizing the mechanical systems constitutes creating mathematical functions that, on one hand, meet the requirements set for those systems, while on the other, can be precisely implemented in real systems. The problem is about creating rational functions meeting the feasibility requirement and the desired dynamical requirements. The method for determining the analytic form of the dynamical characteristics consists in assuming the sequence of resonance and anti-resonance frequencies (poles and zeroes of the searched dynamical characteristics – Fig. 1). The task formulated in the above-mentioned way, aimed at determining the form of dynamical characteristics, does not require performing the approximation process, but instead it makes it possible to start the very synthesis at once.

If the function describing the dynamical properties of the discrete oscillating system is assumed as dynamic stiffness or the dynamical flexibility, then the following transforms are used for individual characteristics:

$$V(s) = sY(s), \tag{1}$$

where $V(s)$ is mobility (mechanical admittance), $Y(s)$ is dynamical flexibility on the plane of complex variable s, $s = \sqrt{-1}\omega$,

$$U(s) = \frac{1}{s} Z(s), \tag{2}$$

where $U(s)$ is immobility (mechanical impedance), $Z(s)$ is dynamical stiffness on the plane of complex variable s.

By using the Eqs. (1) and (2) it is possible to transform the system motion description from generalized displacements and force generalized into a motion description by using generalized velocity and force values. This is the way to transform the dynamical flexibility function of

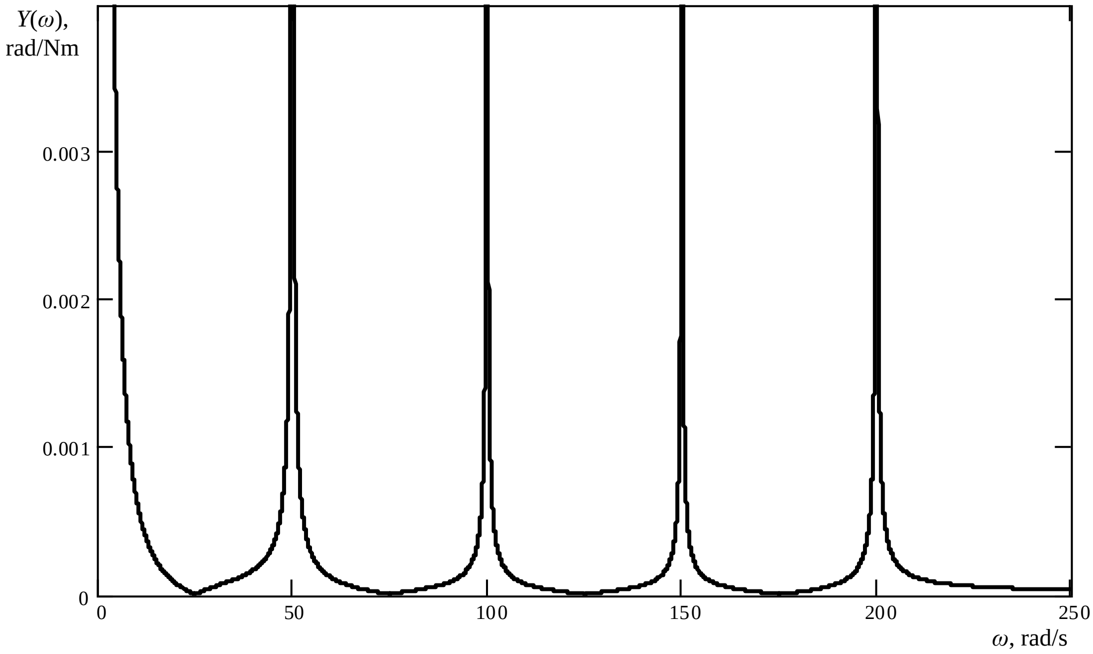

Fig. 1 Example of dynamical characteristics

the system into the mobility function or the dynamical stiffness function of the system into the immobility function, used both to synthesize mechanical systems.

Because of the class of mechanical systems under consideration, the synthesis task will be limited to the case of semi-definite systems, when the number of the obtained system elements is odd. The dynamical function meeting these assumptions is the immobility function described by the following formula:

$$U(s) = H \frac{s\prod_{i=1}^{n}\left(s^2 + \omega_{bn}{}^2\right)}{\prod_{j=1}^{n}\left(s^2 + \omega_{zn}^2\right)}, \tag{3}$$

where: H is any positive number, $\omega_{b1}, \omega_{b2}, \ldots, \omega_{bn}$ are sequence of resonance frequencies, $\omega_{z1}, \omega_{z2}, \ldots, \omega_{zn}$ are sequence of anti-resonance frequencies. At the first step it is necessary to subject the characteristics (3) to synthesis by using the method of distribution on a continued fraction (used for the first time to synthesize electrical circuits – the Cauer method). Finally, it is possible to obtain the immobility value in the form of a continued fraction having the following form:

$$U(s) = J_1 s + \cfrac{1}{\cfrac{s}{c_1} + \cfrac{1}{J_{2z}s + \cfrac{1}{\cfrac{s}{c_{2z}} + \cfrac{1}{J_{3z}s + \cfrac{\vdots}{+\cfrac{1}{J_{k-1z}s + \cfrac{1}{\cfrac{s}{c_{k-1z}} + \cfrac{1}{J_{kz}s}}}}}}}}. \tag{4}$$

As a result of synthesizing the immobility characteristics (3) we can obtain a discrete oscillating system presented in Fig. 2, having the assumed properties in the form of resonance and anti-resonance frequencies.

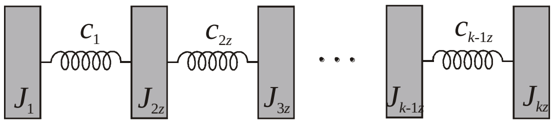

Fig. 2 Cascade model of a drive system

The system obtained constitutes a start point for determining the following: the controlling reducing force and the forces eliminating vibrations and the non-reduced drive system parameters in the form of a multi-stage gear unit.

Once the parameters and structures of the passive system are determined, it is possible to commence determining the value of force making it possible to stabilize, reduce or eliminate system vibration near the resonance system focus. The force excitation, as a setting value, can be performed in the system on any identified inertial element. The paper limits the search for the controlling force to the first inertial element J_1. To do this, the dynamical characteristics of flexibility (1) are modified by introducing:

- vibration frequency drop parameters h for the selected resonance frequencies (an example of the vibration reduction defined in this way is presented in Fig. 3), in the following form:

$$Y1(s) = H\frac{\prod_{j=1}^{n}\left(s^2+\omega_{zn}^2\right)}{s^2\prod_{i=1}^{n}\left(s^2+2h_{bn}s+h_{bn}^2+\omega_{bn}{}^2\right)}, \tag{5}$$

- the parameters of shifting the vibration frequencies $\Delta\omega$ for the selected resonance frequencies (an example of the vibration elimination defined in this way is presented in Fig. 3), in the following form:

$$Y2(s) = H\frac{\prod_{j=1}^{n}\left(s^2+\omega_{zn}^2\right)}{s^2\prod_{i=1}^{n}\left(s^2+\left(\omega_{bn}+\Delta\omega_{bn}\right)^2\right)}. \tag{6}$$

- the parameters of reduction and shifting for the selected resonance frequencies of the system, by introducing parameters h and $\Delta\omega$ (an example of the characteristics is presented in Fig. 3), in the following form:

$$Y3(s) = H\frac{\prod_{j=1}^{n}\left(s^2+\omega_{zn}^2\right)}{s^2\prod_{i=1}^{n}\left(s^2+2h_{bn}s+h_{bn}^2+\left(\omega_{bn}+\Delta\omega_{bn}\right)^2\right)}. \tag{7}$$

Introducing those characteristics modifications Eq. (2) makes it possible to reduce system vibrations near the resonance frequency range – characteristic Eq. (5), resonance frequency shifts – characteristics Eq. (6) and hybrids of the above-mentioned cases – characteristics Eq. (7).

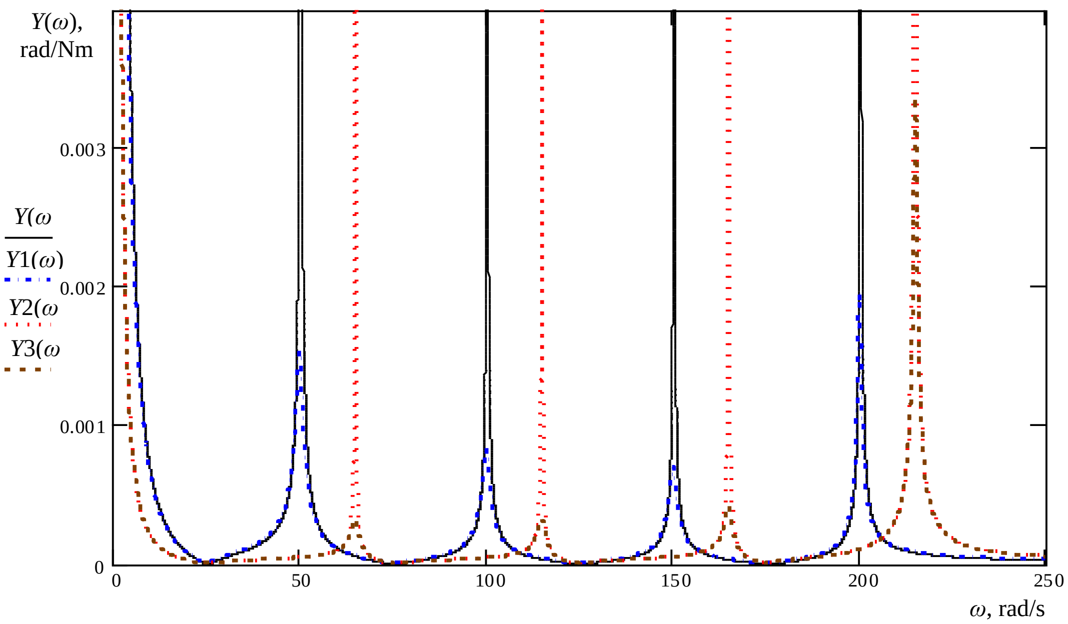

Fig. 3 Example of dynamical characteristics: $Y1(\omega)$ - created by taking into account the free vibration frequency drop parameters, $Y2(\omega)$ - by taking into account the free vibration frequency shift parameters, $Y3(\omega)$ - created by taking into account the free vibration frequency drop and shift parameters

To obtain a vibration drop near the resonance frequency range of the system under analysis (Fig. 2), we adopt the rule of control [9, 13], making it possible to calculate the force excitation as a feedback function in the following form:

$$M(s) = -\left(k_{p1}+k_{v1}s+k_{p2}+k_{v2}s+\ldots+k_{pn}+k_{vn}s\right), \tag{8}$$

where $k_{p1}, k_{p2}, \ldots k_{pn}; k_{v1}, k_{v2}, \ldots, k_{vn}$ are amplification coefficients of the control system depending on the position and velocity of the inertial elements of the system analyzed.

Then, the paper presents the method for determining the above-listed coefficients, which, consequently, will make it possible to define the controlling force.

By using the identified structure parameters (Fig. 2) it is possible to construct a dynamic stiffness matrix in the following form:

$$Z(s) = \left[\begin{array}{ccc|c} J_1s^2+c_1 & -c_1 & 0 & 0 \\ -c_1 & J_{2z}s^2+c_1+c_{2z} & -c_{2z} & 0 \\ 0 & -c_{2z} & J_{3z}s^2+c_{2z}+c_{3z} & 0 \\ \hline 0 & 0 & 0 & J_{kz}s^2+c_{k-1z} \end{array}\right], \tag{9}$$

and the active force matrix, because of the fact that the searched active force acts on the first inertial element only, in the following form:

$$M_1(s)=\left[\begin{array}{cccc} k_{p1}+k_{v1}s & k_{p2}+k_{v2}s & k_{p3}+k_{v3}s & k_{pn}+k_{vn}s \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \end{array}\right]. \quad (10)$$

Using the matrix $M_1(s)$ and matrix $Z(s)$ allows us to determine the following polynomial:

$$det\left(Z(s)+M_1(s)\right)=A_{2n}s^{2n}+A_{2n-1}s^{2n-1}+ \\ +A_{2n-2}s^{2n-2}+\ldots+A_1s^1+A_0s^0. \quad (11)$$

To calculate the value of active force, the obtained polynomial should be divided by the coefficient A_{2n}, and then compared to the polynomial characterizing the adopted dynamical properties of the resonance and free vibration drop coefficients for selected frequencies. The presented equation takes the following form depending on the case under consideration, when an active force is applied:

$$\frac{det(Z(s)+M_1(s))}{A_{2n}}=\prod_{i=1}^{n}\left(s^2+2h_{bi}s+\omega_{bi}^2+h_{bi}^2\right), \quad (12)$$

$$\frac{det(Z(s)+M_1(s))}{A_{2n}}=\prod_{i=1}^{n}\left(s^2+\left(\omega_{bi}+\Delta\omega_{bi}\right)^2\right), \quad (13)$$

$$\frac{det(Z(s)+M_1(s))}{A_{2n}}= \\ =\prod_{i=1}^{n}\left(s^2+2h_{bi}s+\omega_{bi}^2+2\omega_{bi}\Delta\omega_{bi}+\Delta\omega_{bi}^2+h_{bi}^2\right). \quad (14)$$

Once the coefficients found at the same polynomial exponents Eqs. (12)-(14) are compared, we can determine the parameter of the active force reducing the identified system vibrations. The earlier papers of the authors [4-6] presented numerical calculations of the active force applied in the torsionally and longitudinally oscillating identified mechanical systems.

The obtained system presented in Fig. 2 is a model of the reduced system for the first shaft axis and constitutes a starting point for creating a drive system with couplings – toothed gears (primary system).

The use of the reverse reduction rule, saying that the kinetic and potential energy values of the reduced and primary systems are the same, does not provide an unambiguous solution. To do this, we introduce an additional condition saying that all subsystems of the primary system should meet the desired dynamical requirements in the form of selected resonance values.

The synthesis of the primary system begins from building a transmittance matrix:

$$Y(s)=\left[\begin{array}{cccc} Y_{11}(s) & Y_{12}(s) & Y_{13}(s) & Y_{1k}(s) \\ Y_{21}(s) & Y_{22}(s) & Y_{23}(s) & Y_{2k}(s) \\ Y_{31}(s) & Y_{32}(s) & Y_{33}(s) & Y_{3k}(s) \\ Y_{n1}(s) & Y_{n2}(s) & Y_{n3}(s) & Y_{nk}(s) \end{array}\right], \quad (15)$$

where $Y_{ij}(s)=X_i(s)/F_j(s),(i=1,2,\ldots n;\ j=1,2,\ldots k)$ are operator transmittances between i input and j input and are equal to the proportion of the i response transform to the j forcing transform to be used to determine zeroes for the dynamical characteristics describing the dynamical properties of primary systems. Then the system is disengaged to create two subsystems in relation to the first reduced element J_{2z} and the resonance frequencies of the disengaged subsystems are determined in relation to the first reduced mass. By taking the highest values from the sequence of the pre-set resonance frequency values and from the sequence of the anti-resonance values determined, it is possible to construct the dynamical characteristics describing the properties of the searched primary subsystem at the first axis. Such a function is subject to synthesis by using the method of characteristics distribution on a continued fraction to determine the system parameters before performing the reduction at its shaft with stiffness c_1, in the following form:

$$U_1(s)=H'\frac{s\left(s^2+\omega_{bn}{}^2\right)}{\left(s^2+\omega_{2z}^2\right)}=H'J_{2'}s+\cfrac{1}{\cfrac{s}{H'c_{1'}}+\cfrac{1}{H'J_{1'}s}}, \quad (16)$$

where ω_{2z} is max anti-resonance frequency determined for system 1 in relation to J_{2z}, $J_{1'}, J_{2'}$ are the values of inertial elements of the subsystem obtained, $c_{2'}$ is the value of elastic element. For the values obtained it is necessary to select the constant of proportionality H' to meet the following requirement:

$$H'J_{1'}=J_1 \Rightarrow H'=\frac{J_1}{J_{1'}}. \quad (17)$$

The synthesis performed results in obtaining a system structure taking into account the first disengagement presented in Fig. 4.

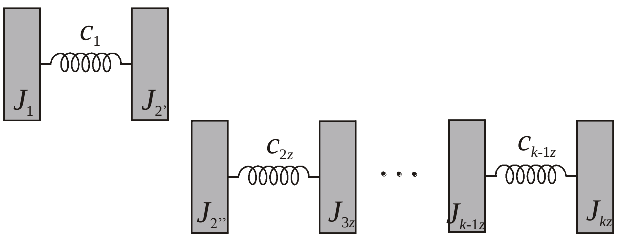

Fig. 4 Model of a drive system taking into account the first disengagement

Further synthesis used the subsystem of the searched drive system reduced at the shaft with stiffness of c_2, obtained as a result of designing the structure and parameters of the separated primary subsystem with stiffness of c_1 (Fig. 4). The parameters of the subsystem under analysis take the following values:

$$J_{2''}=J_2-H'J_{2'}, c_2,\ldots,c_{k-1z}, J_{kz}. \quad (18)$$

At the later stage of the synthesis, the obtained subsystem is disengaged in relation to the subsequent inertial element J_{3z} reduced and the subsystem anti-resonance frequencies are determined. After assuming the dynamical properties of the system in the same way as for the engagement J_{2z} in relation to the inertial element described above, the dynamical characteristics distributed on a continued equation is created, taking the following form:

$$U_2(s) = H'\frac{s\left(s^2+\omega_{bn}{}^2\right)}{\left(s^2+\omega_{3z}^2\right)} = H'J_{3'}s + \frac{1}{\dfrac{s}{H'c_{2'}} + \dfrac{1}{H'J_{2'''}s}}, \quad (19)$$

where $H' = \dfrac{J_{2''}}{J_{2'''}}$ is the constant of proportionality, ω_{3z} is max anti-resonance frequency determined for subsystem 2'' in relation to J_{3z}, $J_{3'}, J_{2'''}$ are the values of inertial elements of the subsystem obtained, $c_{2'}$ is the value of elastic element. This step of the synthesis results in obtaining the system structure by taking into account the first and second disengagement presented in Fig. 5.

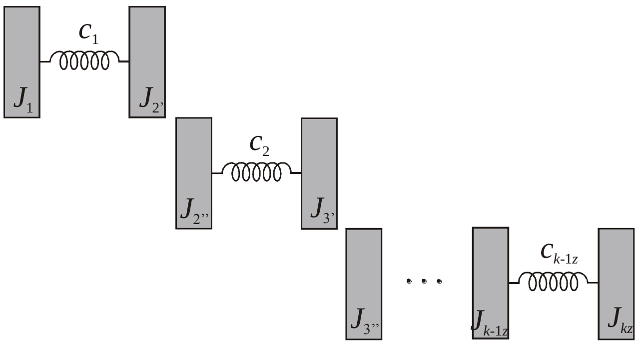

Fig. 5 Model of a drive system taking into account the first and second disengagement

On the other hand, the subsystem of the drive system reduced at the shaft with stiffness of c_3 takes the following value:

$$J_{3''} = J_{3z} - H'J_{3'}, c_3, \ldots, c_{k-1z}, J_{kn}. \quad (20)$$

The above-described process continues until performing the disengagement in relation to the $J_{k\text{-}1z}$ element.

When synthesizing drive systems with couplings, it is necessary to assume the requirements regarding the total system transmission ratio. This ratio is the product of selected single transmission ratios at each gear unit stage expressed by the following formula:

$$i_{1k} = \prod_{i=1}^{k-1} i_{i,i+1}, \quad (21)$$

where i_{1k} is total transmission ratio, $i_{i,\,i+1}$ is ratio for subsequent gear unit stages. To obtain the real parameters of the synthesized toothed gear it is necessary to take into account (23). It can be done by multiplying the drive system parameters obtained by the selected ratio at individual stages (it refers to the case, when the system is reduced at the driveshaft), in the following form:

- inertial elements:

$$J_{1\mathrm{r}} = J_1\ ;\ J_{2'\mathrm{r}} = J_{2'};\ J_{2''\mathrm{r}} = J_{2''}\cdot i_{12}{}^2;\ J_{3'\mathrm{r}} = J_{3'}\cdot i_{12}{}^2;$$
$$J_{3''\mathrm{r}} = J_{3''}\cdot i_{13}{}^2;\ \ldots;\ J_{k-1'\mathrm{r}} = J_{k-1\mathrm{z}}\cdot i_{1k-1}{}^2,\ J_{k-1''\mathrm{r}} = J_{k-1''\mathrm{z}}\cdot i_{1k}{}^2,$$
$$J_{k\mathrm{r}} = J_{k\mathrm{z}}\cdot i_{1k}{}^2.$$

- elastic elements:

$$c_{1\mathrm{r}} = c_1;\ c_{2\mathrm{r}} = c_{2\mathrm{z}}\cdot i_{12}{}^2;\ldots,\ c_{k-1\mathrm{r}} = c_{k-1\mathrm{z}}\cdot i_{1k}{}^2.$$

Finally, we obtain the drive system before reduction presented in Fig. 6, meeting the desired properties in the form of resonance frequencies assumed in advance and ratios for individual drive system stages.

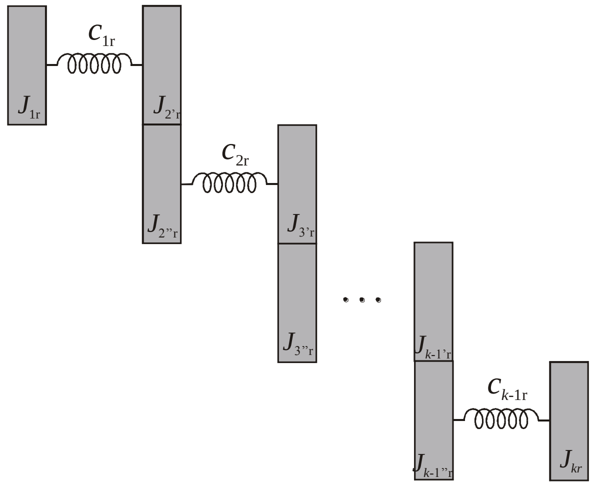

Fig. 6 Model of a drive system taking into account all couplings

The synthesis results in obtaining a machine drive system taking into account couplings in the form of a multi-stage gear unit. The basic property of the system model created and its subsystems is narrowing down the critical (resonance) frequencies to the number assumed at the beginning of the task.

3. Numerical example

Numerical calculations used to determine the parameters and structures of the system model in the form of two-stage gear unit are presented. The input values, in order to obtain the structure and parameters of the system, consisted of:

- the resonant and anti-resonant frequencies (poles and zeros of the desired dynamic characteristic) in the form of: $\omega_{b1} = 0$ rad/s, $\omega_{b2} = 50$ rad/s, $\omega_{b3} = 100$ rad/s, $\omega_{b4} = 150$ rad/s, $\omega_{z1} = 25$ rad/s, $\omega_{z2} = 75$ rad/s, $\omega_{z3} = 125$ rad/s;
- vibration frequency drop parameters *h* for the selected resonance frequencies in the form of: $h_{b2} = 1$ rad/s, $h_{b3} = 2$ rad/s, $h_{b4} = 1$ rad/s;
- the parameters of shifting the vibration frequencies $\Delta\omega$ for the selected resonance frequencies in the form of: $\Delta\omega_{b2} = 15$ rad/s, $\Delta\omega_{b3} = 15$ rad/s, $\Delta\omega_{b4} = 15$ rad/s;

which were used to describe the dynamic properties of the discrete vibrating system in the form of dynamical flexibility:

$$Y(s) = \frac{\left(s^2+25^2\right)\left(s^2+75^2\right)\left(s^2+125^2\right)}{s^2\left(s^2+50^2\right)\left(s^2+100^2\right)\left(s^2+150^2\right)}, \quad (22)$$

$$Y1(s)=\frac{(s^2+25^2)(s^2+75^2)(s^2+125^2)}{s^2(s^2+2s+1+50^2)(s^2+4s+4+100^2)(s^2+2s+1+150^2)},\tag{23}$$

$$Y2(s)=\frac{(s^2+25^2)(s^2+75^2)(s^2+125^2)}{s^2(s^2+(50+15)^2)(s^2+(100+15)^2)(s^2+(150+15)^2)},\tag{24}$$

$$Y3(s)=\frac{(s^2+25^2)(s^2+75^2)(s^2+125^2)}{s^2(s^2+2s+1+(50+15)^2)(s^2+4s+4+(100+15)^2)(s^2+2s+1+(150+15)^2)},\tag{25}$$

and the total gear ratio of the synthesised gar:

$$i_{13}=i_{12}\cdot i_{23}=\frac{1}{4}\cdot\frac{4}{3}=\frac{1}{3}.\tag{26}$$

The characteristic functions Eqs. (22)-(25) were used to determine the structure and set of parameters in accordance with the formal arrangement presented in the previous chapter. The results of the various stages of drive system synthesis are summarised in Table 1 and Table 2.

Table 1

The results of the various stage of drive system synthesis

	Cascade model of a drive system					Model of a drive system taking into account the first disengagement			
	J, kgm^2		c, Nm/rad			*J*, kgm^2		c, Nm/rad	
1	J_1	1	c_1	13125	2	J_1	1	c_1	13125
	J_{2z}	2.1	c_{2z}	11812		$J_{2'}$	1.4	c_{2z}	11812
	J_{3z}	2.52				$J_{2''}$	0.7		
	J_{4z}	4.62	c_{3z}	8662		J_{3z}	2.52	c_{3z}	8662
						J_{4z}	4.62		
	Model of a drive system taking into account the first and second disengagement					Model of a drive system taking into account all couplings			
	J, kgm^2		c, Nm/rad				*J*, kgm^2	c, Nm/rad	
3	J_1	1	c_1	13125	4	J_{1r}	1	c_{1r}	13125
	$J_{2'}$	1.4				$J_{2'r}$	1.4		
	$J_{2''}$	0.7	c_{2z}	11812		$J_{2''r}$	0.04	c_{2r}	1312.44
	$J_{3'}$	2.11				$J_{3'r}$	0.13		
	$J_{3''}$	0.39	c_{3z}	8662		$J_{3''r}$	0.04	c_{3r}	962.44
	J_{4z}	4.62				J_{4r}	0.51		

Table 2

The values of controlling force parameter values

	k_{p1}, Nm/rad	k_{v1}, Nms/rad	k_{p2}, Nm/rad	k_{v2}, Nms/rad	k_{p3}, Nm/rad	k_{v3}, Nms/rad	k_{p4}, Nm/rad	k_{v4}, Nms/rad
$Y1(s)$	26	8	-29.39	2.41	-10.86	-8	14.26	-2.4
$Y2(s)$	9675	0	2089.8	0	-5145.14	0	-6619.66	0
$Y3(s)$	9701	8	2076.92	10.66	-5177.54	-6.26	-6600.38	-12.4

4. Conclusion

This paper presents the problem of identifying the torsionally vibrating discrete systems, considered as machine subassemblies featuring desired mechanical properties. The presented identification method makes it possible to obtain the parameters and structures of non-reduced machine drive system on account of the assumed resonance and anti-resonance values and the transmission ratio. In addition, the paper presents the method of active vibration reduction of the identified machine drive systems. The aim of identifying an active system is to ensure that the system meets the key operation requirements. The basic criterion here is the reduction and elimination of vibrations near the location in the system where the resonance originates.

The identified structure of multi-stage toothed gear unit is not the only model featuring the pre-set dynamical properties. Using the presented method makes it possible to obtain not only the cascade-structured systems, but also the branched and branched-cascade structures. The structures obtained in such a way and parameter values can be used to select an optimum drive system. Further research on synthesizing drive systems will focus on finding the criterion for choosing appropriate drive system obtained as a result of the synthesis.

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MAŠINŲ PAVAROJE NUSTATYTŲ MECHANINIŲ VIRPESIŲ AKTYVUS SLOPINIMAS DAUGIAPAKOPĖSE KRUMPLIARATINĖSE PAVAROSE

R e z i u m ė

Straipsnyje analizuojamas mašinų pavaroje nustatytų mechaninių virpesių aktyvus slopinimas daugiapakopėse krumpliaratinėse pavarose. Slopinimo metodas pagrįstas aktyvia sinteze leidžiančia pasiekti norimą mechaninį efektą teisingai parenkant sistemos dinamines savybes. Norimos dinaminės savybės suprantamos, kaip rezonansų ir antirezonansų dažnių seka dinaminių charakteristikų formoje. Tokiu būdu parinktos savybės yra išeities taškas diskretinių virpančių sistemų parametriniam ir struktūriniam identifikavimui.

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ACTIVE REDUCTION OF IDENTIFIED MACHINE DRIVE SYSTEM VIBRATIONS IN THE FORM OF MULTI-STAGE GEAR UNITS

S u m m a r y

The paper presents the problem of vibration reduction in the identified machine drive systems in the form of single or multi-stage gear units. The reduction method was based on using an active synthesis making it possible to obtain a desired mechanical effect by properly selecting the dynamical system properties. The desired dynamical properties are understood throughout the paper as a sequence of resonance and anti-resonance frequencies in the form of dynamical characteristics. The properties assumed in such a way constitute a starting point for the parametric and structural identification of discrete oscillating systems.

Keywords: mechanical impedance, mechanical admittance, resonant and anti-resonant frequencies.

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