Determination of rigidity of constructive elements of the drive of the spherical mill

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Abstract. The stiffener constructive elements are members of the theoretical diagram electromechanical drive system the ball mill is selected calculated dynamic, the model performed in the form of a two mass electrochemical system, which contains two focused, inertia mass, the united one elastic-viscous loop rotor engine and working body machine. In a result of a calculation found analytical parameters which determine the hardness camshaft mesh, stiffness drive shafts and stiffness elastic couplers, connecting ends of shafts drive basket ball mills. The proposed methodology allows you to determine the stiffness of elements of kinematic chain any electromechanical drive systems and select, in further, their constructive options.

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Introduction

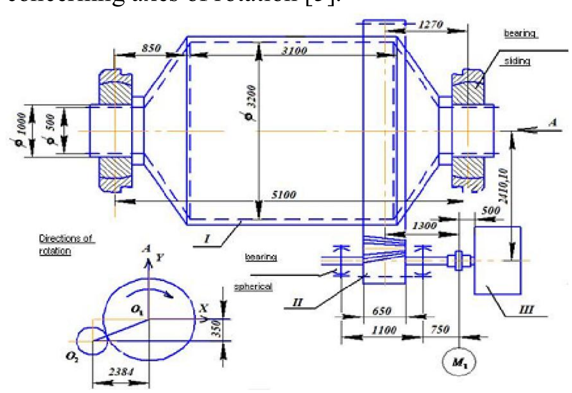
The problem of drawing up calculating dynamic models of various production mechanisms and cars with the adjustable electric drive, including with the drive of working body of the technical car with the big moment of inertia, is extremely actual in the conditions of growing number of new constructive decisions [1]. As a rule, calculating model represents two-mass electromechanical system which contains two concentrated inertial masses connected by one elastic viscose connection. As one concentrated inertial weight acts the engine rotor, and as the other - working body of the car [2].

The main function which characterizes dynamic behavior of the system is the period of the induced excitement in gearing with variable rigidity. This phenomenon is a consequence of change of quantity of the teeth couple connected at the same time that is function of angular situation in gearing [3].

Theoretical prerequisites

Usually by drawing up physical models of machine units for research of dynamic processes proceeding in them, they are represented in the form of system with the concentrated inertial parameters in which the engine rotor, separate parts of the drive, for example, wheels, connecting couplings, etc., and also working part of the machine unit are represented in the form of material bodies and the points possessing a certain masses, the sizes and the inertia moments. Idealized connections between them don't possess weight, but have elastic and dissipative characteristics [4].

According to calculating schemes of mills (fig. 1) all constructive elements of their drive, from the electric motor to the drum, are presented by cylindrical tooth gearings, shafts in the support of swing connected by couplings, with axes of rotation parallel to an axis of rotation of the drum. By drawing up dynamic models of spherical mills we will consider elastic properties (rigidity) of tooth gearings of the drive of the drum, shafts and couplings connecting them; and as resistance forces the moments of friction of support of sliding of the drum, and also the moments of inertia of forces concerning axes of rotation [5].



I – drum with the gear wreath $z_K = 278$; *II* – shaft driving with a gear wheel $z_{III} = 22$; *III* – the electric motor synchronous $n_\partial = 250$ cap/min; M_1 – the coupling connecting

Figure 1. Constructive and kinematic scheme of the spherical mill with the gear wreath on the drum and the without reducer electric drive

In order to consider elasticity (rigidity) of constructive elements of the drive of the drum, we will define rigidity of tooth gearings, shafts and the couplings connected consistently in one kinematic chain.

In operating time of mills in gear gearings of wreath 1 on the drum I and a driving gear wheel 2 (II), (also as well as in gear gearing of wheels of reducer, in the scheme on the figure 2) act forces given by them deforming teeth .

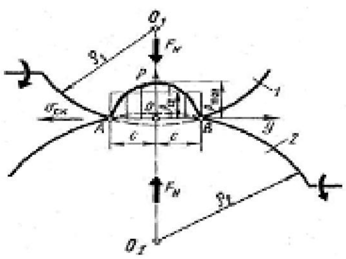


Figure 2. Calculating model to determination of pressure in the contact zone of two elastic cylinders

Statement of the main material

We will consider one of components of this force P_t directed on the tangent to initial circles of cogwheels of transfer (fig. 2), and also component of small elastic deformation of teeth in the same direction S_t.

Force P_t and elastic deformation S_t are connected by the ratio:

$$P_t = c \cdot S_t, \quad (1)$$

where c – linear rigidity of gear gearing, which is proportional to length of teeth *B* (wreath width) is determined by the formula:

$$c = a \cdot B. \quad (2)$$

with =a·B. (2)

Here a –coefficient which for steel wheels is accepted equal = 15·103, MPa [6].

In order to pass from small linear movement S_t to angular $\Delta\varphi$, we will fix unmovable gear wheel 2, and we will put to the wreath drum *1* the moment $M_1 = (M_1, M_C)$. Under its influence teeth of wheels are deformed and the wreath *1* will turn to the small corner:

$$\Delta\varphi_1 = \frac{S_t}{r_1},$$

,

from where $M_2 = c_{21}\Delta\varphi_2$,

$$S_t = \Delta\varphi_1 \cdot r_1, \quad (3)$$

Here r_1 – is the radius of the initial circle of the wheel 1.

Thus $P_t = \frac{M_1}{r_1}$

Substituting the received expressions in equality (1), we will have:

$$M_1 = c \cdot r_1^2 \Delta\varphi_1$$

or

$$M_1 = c_{12}\Delta\varphi_1$$

where the angular rigidity of gear gearing given to the wreath *1*, at not movably fixed gear wheel, is determined by a formula.

$$c_{12} = c \cdot r_1^2. \quad (4)$$

Similarly, if to fix the wreath *1*,

$$S_t = \Delta\varphi_2 \cdot r_2, \quad (5)$$

where r_2 - the radius of the initial circle of the gear wheel 2. If to put to the gear wheel 2 the moment:

$$M_2 = M_1 u_{21},$$

that it will turn on the small corner $\Delta\varphi_2$.

Thus $M_2 = c_{21}\Delta\varphi_2$,

where

$$c_{21} = c \cdot r_2^2 \quad (6)$$

the angular rigidity of gear gearing given to the gear wheel 2, at the motionless wreath 1. Then

$$c_{21} \neq c_{12},$$

as from equality of expressions (3) and (5) follows:

$$\Delta\varphi_1 \cdot r_1 = \Delta\varphi_2 \cdot r_2,$$

$$c_{21} = c \cdot r_1^2 \left(\frac{r_2}{r_1}\right)^2 = c_{12}u_{12}^2.$$

Thus

$$c_{21} = c_{12}u_{21}^2, \quad (7)$$

where $u_{21} = \frac{r_2}{r_1} = \frac{z_2}{z_1}$ - the transfer relation of wheels 2 and 1.

Following the stated technique, we find rigidity of gear gearing of wheels of a reducer in the scheme of the drive drum of mill according to fig. 3 for which, according to formulas (1)-(7), we will have:

$$P_t^{/} = c \cdot S_t^{/}.$$

Here $c = aB^{/}$, $B^{/}$ - width of the wreath of wheels 3 and 4; $S_t^{/} = \Delta\varphi_3 r_3$; $M_3 = P_t^{/} r_3 = M_2 u_{21}$ - the moment

wheel 3; $\Delta\varphi_3$ - small angle of rotation of the wheel 3 at the fixed gear wheel 4.

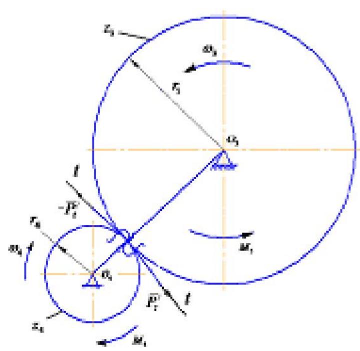


Figure 3. For determination rigidity of gearing of cogwheels of the reducer III in the scheme of the drive drum of the mill on the figure 1

Thus

$$P_t^{/} = \frac{M_3}{r_3} \text{ and } M_3 = (cr_3^2)\Delta\varphi_3 = c_{34}\Delta\varphi_3, \qquad (8)$$

where $c_{34} = cr_3^2$ - angular rigidity of gearing of the wheels 3 and 4, given to the wheel 3 at the motionless wheel 4.

Similarly, if to fix the wheel 3,then $S_t^{/} = \Delta\varphi_4 r_4$.

If to put the moment 4 to the gear wheel: $M_4 = P_t^{/} r_4 = M_3 u_{43}$,

that it will turn to the small corner $\Delta\varphi_4$.

Thus

$$M_4 = (cr_4^2)\Delta\varphi_4 = c_{43}\Delta\varphi_4, \qquad (9)$$

where $c_{43} = cr_4^2$ – angular rigidity of gearing of the wheels 3 and 4, given to the gear wheel 4, at the fixed wheel 3.

The rigidity of shaft of the drive drums of mills are determined by the formula:

$$c_d = \frac{\sigma J_p}{l}, \qquad (10)$$

where где $\sigma = 8 \cdot 10^4$ MPa - the elasticity module on shift of the shaft material o (steel); $J_p \cong 0{,}1d^4$ - polar moment of inertia of shaft section ; d - diameter of the shaft.

Rigidity of the step shaft can doubly be defined:

- or addition of rigidity of separate sites with different diameters and length;
- or on equivalent diameter on the calculating length.

If the site of the shaft has veneer and lisovy connection, then determination of rigidity is made with their account.

Rigidity of the elastic couplings connecting the ends of shaft of the drive drums of mills generally is defined as the relation:

$$c_M = \frac{\Delta M}{\Delta\varphi}, \qquad (11)$$

where $\Delta M = M_2 - M_1$; $\Delta\varphi = \varphi_2 - \varphi_1$ - changes of the given moment and corners of the twisting of the shaft ends corresponding to them.

For operation of drives damping abilities of couplings which are estimated by the size of energy absorbed by them at deformation of their elastic elements due to external or internal friction have special value.

Let, for example, overall dimensions of the coupling with plates are given and its rigidity C [7] is known. It is required to receive the most power-intensive coupling of V and number of plates of z (figures 4, 5).

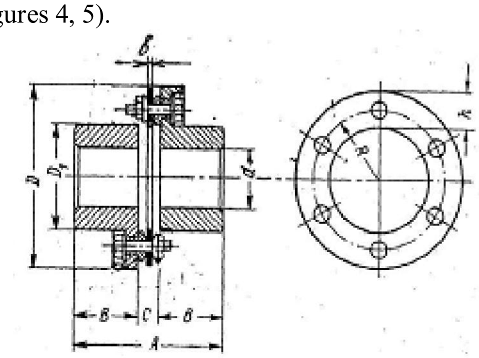
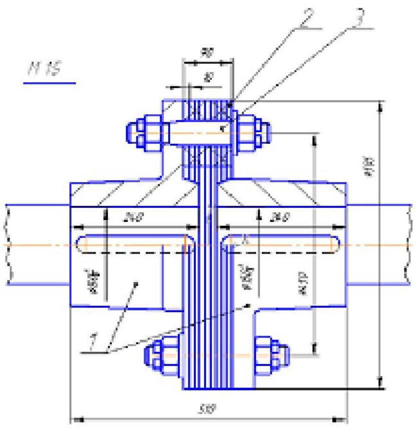


Figure 4. The semi rigid coupling with steel disks



1 – semi-couplings; 2 – disks; 3 - fingers
Figure 5 - Design and sizes of the coupling *M2* of the drive of the mill

The thinner plates are chosen, the more they need to be taken on the volume for obtaining the set rigidity. Therefore, the coupling with such minimum thickness of plates will be the most power-intensive at which in the coupling their greatest number z is located under condition that tension σ and τ and doesn't surpass the admissible $\sigma \le [\sigma]$; $\tau \le [\tau]$. .

Let the torque at which tension in plates reaches allowed is equal to $M_{\max}$. Then the maximum energy accumulated by the linear coupling is equal to

$$A_\sigma = \frac{1}{2} M_{\max} \varphi .$$

But as

$$\varphi = \frac{M_{\max}}{C},$$

Then

$$A_\sigma = \frac{1}{2} \frac{M_{\max}^2}{C}.$$

On the other hand,

$$A_\sigma = k_\sigma \frac{[\sigma]^2}{2E} zV ,$$

and, so

$$\frac{1}{2} \frac{M_{\max}^2}{C} = k_\sigma \frac{[\sigma]^2}{2E} zV .$$

.

Here we obtain

$$M_{\max} = [\sigma] \sqrt{\frac{k_\sigma C}{E} zV} . \quad (12)$$

That is at same it is possible $[\sigma]$ a bigger torque can be transferred in cases when rather bigger volume is occupied by elastic elements.

The given reasons explain aspiration to carry out couplings with the packages gathered from separate plates. Thus it is necessary to consider that thanks to friction between steel plates the coupling gains ability to damping of fluctuations [8].

By comparison the couplings which elastic elements are executed from the same material, it is enough to compare only values of sizes of coefficients k_σ, k_τ . For steel it is possible and when elements of one couplings work for bend, others - on torsion as for steel ratios [9] are carried out:

$$\frac{E}{G} \approx 2,6 \text{ and } \frac{[\sigma]}{[\tau]} \approx 1,6 .$$

Consequently

$$\left(\frac{[\sigma]}{[\tau]} \right)^2 \approx 2,6$$

and it means that

$$\frac{[\sigma]^2}{2E} \approx \frac{[\tau]^2}{2G} .$$

Values of coefficients k_σ и k_τ can be calculated, using formulas of work of deformation of an elastic body:

$$A_\sigma = \int_0^{P_{дин}} P dy \text{ or } A_\tau = \int_0^{M_{кр}} M_{кр} d\varphi . \quad (13)$$

The sizes and parameters of the couplings used in the drive of spherical mills are given in table 1.

Table 1. The sizes (in mm) and parameters of semi rigid disk couplings

$d_{\max}$	n , об/мин	N, кВт	D	B	C	E	A	D_1
101,6	4 200	42,9	279,4	117,4	36,5	36,5	271,4	168,2
114,3	3 600	55,9	320,9	133,3	38,8	46,0	305,5	187,3
127,0	3 200	85,7	361,9	146,0	44,4	52,3	336,5	211,1
139,7	2 900	119,3	400,0	168,2	49,2	60,3	385,7	231,7
152,4	2 600	167,7	444,5	184,1	53,9	66,6	422,2	254,0
177,8	2 300	258,7	504,8	206,3	63,5	73,0	476,2	292,1
190,5	2 200	331,0	552,4	219,0	68,2	79,3	506,4	317,5

Rigidity of this coupling can be determined by the following approximate formula offered by N.Z. Suponitsky:

$$C \approx \frac{Ezm^2 bh}{6,5\left(\frac{1}{R} + 0,01 \frac{1}{h} \right)}, \quad (14)$$

where b - thickness of disks;
h, R - on figure 4;
z - number of disks;

m - number of deformable sections;

E - the module of elasticity of the material of the disk in kg/cm2.

As the material of an elastic disk by transferring of the small moments can also be applied skin, rubber, rubberized fabric, etc. [10]. In this case the design of this coupling, to a certain extent, will be similar to a design of the coupling shown in figure 5.

Conclusions

1. The calculating scheme is developed for determination of rigidity of teeth gearings, shafts, couplings connected in one kinematic chain of the drive of the spherical mill, executed in the form of two-mass electromechanical system which unlike the known contains two concentrated inertial masses connected by one elastic viscous connection of the rotor of the engine and working body of the car.

2. On the basis of the calculating scheme of two-mass electromechanical system is defined the procedure of payments of its key parameters: the moments of inertia, rigidity of transfers and others, allowing effectively carrying out design of electromechanical systems of spherical mills.

3. The analytical parameters defining rigidity of gear gearings, rigidity of shaft of the drive and rigidity of the elastic couplings connecting the ends of shaft of the drive of drums of spherical mills are found.

4. The offered technique allows to define rigidity of elements of any electromechanical systems of drives entering the kinematic chain and to choose their design data in the future.

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