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The ways to enhance durability and bearing capacity of the open gear drum mills

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Abstract

Purpose is substantiating ways to enhance durability and bearing capacity of open gears of ore-pulverizing drum mills as well as efficiency of engineering solutions concerning the increase in their unit power at the expense of drive improvement.

Methods. Results of continual experiments and theoretical studies have been generalized as for the abrasion of working surfaces of open gear teeth of drum mills and factors influencing load distribution in terms of a tooth rim width.

Findings. Comparative analysis between domestic mills and the best world-class products has been carried out. Ways of solving problems to design large-capacity mills with a gearbox drive have been demonstrated. Influence of hardness of working teeth surfaces on their durability has been evaluated quantitatively. The factors, governing load distribution in terms of tooth rim width, have been analyzed. Use of self-adjusting gear drives for open gears has been evaluated.

Originality. Functional relation between stress-strain properties of working surface of teeth; the number of running-in modes, determined by operational conditions; and durability of open gear has been identified. The factors, influencing load distribution in terms of tooth rim width, have been considered.

Practical implications. It has been shown that use of such open gears, where hardness of working surface of gear teeth is (500-600) H_1B_1 and that of a tooth rim one is (260-300) H_2B_2, makes it possible to provide almost wear-free operation. Moreover, it is the required condition for the performance of a tooth rim with two drive gears.

Keywords: *drum mill, open gear, load distribution, abrasion, self-adjusting gear*

1. Introduction

Ukraine possesses 20 per cent of global iron ore reserves. Ukraine leads the world in the iron ore raw materials; it ranks #7 in the world as for the output. In the context of global iron ore production, share of Ukraine is almost 5 per cent [1]-[3]. The annual iron ore output in Ukraine is approximately 160 million tons and total explored reserves of iron ores are sufficient for mining enterprises to operate for 180 years, and the reserves of deposits under operation (or those operated until recently) are sufficient for 130 years [4]. A significant part of underground mining (25%) is occupied by PJSC "Zaporizhzhia Iron-Ore Plant", which develops high-grade iron ore (iron content more than 60%) in the Bilozerskyi iron ore region [5].

To ensure market competitiveness, the experts, being engaged in the direct mineral mining as well as in the specific equipment manufacturing, focus on the searching for ways to cut the end product cost at each stage of the procedure [6], [7]. First of all, it concerns mineral mining and processing equipment advance [8]-[10].

Grinding is among capital- and energy-intensive stages of ore processing [11]-[13]. Energy consumption to grind the ore is 7-10 per cent of the whole world output; as for the metal, it is 2 per cent. In this context, energy intensity of fine grinding is 20-60 kWh/t. Currently, drum mills are the most popular facilities for ore grinding. Their global share is more than 80 per cent [14], [15].

In world practice, cut of the expenditures, connected with grinding, follows the way of grinding equipment aggregation. In the 1960s motor output of the largest mills did not exceed 4478 kW; in the year of 1996 a semi-autogenous grinding mill with 20000 kW duty was manufactured [16]. Aggregation of unit power of drum mills has cut substantially capital expenditures at the expense of the decreased number of process facilities and, subsequently, the number of construction areas in terms of one and the same production

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rates. Moreover, the abovementioned results in the decreased amount of supporting machinery; in addition, maintenance intervals for one technological line are similar to those for each section involving smaller facilities. Finally, total downtime is reduced as well [17]-[21].

Currently, ore is grinded worldwide by means of semi-autogenous grinding mills (SAGMs). Especially, it concerns very hard metallic minerals to which iron ore belongs [22], [23]. Physical modeling methods have been developed making it possible to define parameters of semi-autogenous grinding mills being designed relying upon results of laboratory studies as for the material disintegration [24], and precipitation for separation of rare-earth elements [25]-[28].

Use of wet semi-autogenous grinding mills helps intensify ultimately a raw grinding process. The wet semi-autogenous grinding mills result from the progress of wet autogenous grinding mills (WAGMs) involving 15 per cent increase in ball load amount. Ball adding factors into the necessity for further increase in drive output as well as for the increase of bearing capacity of open gear [29], [30].

As far back as 1989, Erdenet enterprise (MPR) put into operation self- and semi-autogenous grinding system involving two sections operating simultaneously. Two wet 9000×3000 A ($N = 4000$ kW and $V = 160$ m^3) semi-autogenous grinding mills perform raw grinding stage; stage two takes place in overflow 5500×6500 ($N = 4000$ kW and $V = 140$ m^3) ball mills. The necessity to remain competitive in the nonferrous metal market resulted in the necessity to introduce highly reliable large-size equipment of the increased unit power into a technological process of the mine-mill Erdenet enterprise. In 2015, self- and semi-autogenous grinding system was put into operation. Raw grinding stage is performed by a wet 9750×4880 ($N = 8400$ kW and $V = 363$ m^3) semi-autogenous grinding mill; stage two takes place in overflow 6700×9750 ($N = 8400$ kW and $V = 330$ m^3) ball mill [31].

The largest mills are equipped with gearless drives [32]. A feature of the design is as follows: the mill drum is actuated by a motor mounted on the drum which rotor is bolted to the mill drum; static stator assembly involves the rotor components. The electric motor is energized by a frequency changer converting input current with 50/60 Hz current to current which frequency is close to 1 Hz.

If high reliability of the mills, equipped with gear drives and involving open gears, is provided then they are quite cheaper than gearless ones [33], [34]. Increase in unit power of gear drives is to design mills where torque effect by motors (or a motor) is transmitted through a tooth rim with the help of several drive gears. Two-motor drives are among them. Paper [35] consider a wet semi-autogenous grinding mill with 10363 mm diameter and 6096 mm length driven by slow-speed AB B motors ($n = 200$ rpm) of 6250 kW output each. Synchronous motors are equipped with frequency changers making it possible to perform smooth motor start, and control a drum speed depending upon grinding behaviour; characteristics of the grinded material; filling degree of the drum; and wear rate of lining.

Mills, which drives are equipped with one motor; lateral gear; and two drive gears, are another promising tendency as for the design of the geared mills with high unit power. In this context, gear design provides uniform load distribution among drive gears [36], [37]. Hence, a possibility emerges to develop in future mills with more than 15000 kW gear drive.

To compare with the best world models, drive capacity of the largest national mills is 4000 kW. Hence, there is a need for domestic mining industry to upgrade facilities grinding mineral raw materials. The abovementioned challenges machine builders to design powerful grinding large-size facilities. Aggregation of mills, equipped with gear drives, is restricted by bearing capacity of open gears. Bearing capacity of heavy-duty machinery is influenced significantly by nonuniformity of load distribution in terms of a tooth wheel width [38]-[40].

Paper [38] represents results of theoretical studies and experiments concerning evaluation of load concentration, and determination of regularities of its distribution in terms of a tooth rim width of open gears of drum mills. It has been shown that taking into consideration face-motion variation of a tooth wheel, the sum angle of gear obliquity is composed of a constant component, and variable component. The variable component of an obliquity angle cannot run in since each following pair of gears interacts with different obliquity angles. It has been determined that depending upon a pair of gears their incomplete contact is possible. Calculation technique for open gears in terms of incomplete gear contact has been developed.

Hence, the problem to improve bearing capacity of drum mills and, consequently, their unit power depends ultimately on the solution of a problem to provide uniform load distribution in terms of a tooth rim width [41]-[43]. For the purpose, self-adjusting drive gears have been designed for slow-speed gear systems to support complete tooth contact. Self-adjusting gears with movable tooth rim and hinging [44] are singled out, and two-piece gears with polymer elastic fastenings of a tooth rim [45]. On the one hand, the self-adjusting gears use lowers load concentration in terms of a tooth rim; on the other hand, it results in the increased dynamic loads within the gearing. In addition, even if tooth contact is complete, absolutely uniform load distribution cannot be provided [38]. That is why the problem of efficient application area of self-adjusting gears within the drives of drum mills is still unsolved.

Currently, science-based background has been developed making it possible to substantiate scientifically the parameters of gear drives for ore-grinding mills and outline ways to improve their durability as well as bearing capacity.

The objective is to demonstrate ways improving durability and bearing capacity of open gears of ore-grinding drum mills as well as efficiency of engineering solutions increasing their unit power.

2. Methods

The methods to calculate open gears as for their wear durability rely upon the experiments carried out using representative line of self-adjusting ball ore-grinding mills, and ore-pebble ones in terms of Inhuletskyi MPP, Novokryvorizkyi MPP, Lebedynskyi MPP as well as in terms of crashing mills of Prydniprovska TPP. Simultaneously, concentration of mechanical impurities in lubricant; their granulometric composition; concentration of metallic and mineral components were determined in addition to the lubricant type effect, and lubrication method applied for the open gear [38].

Gear tooth caliper was used to gauge the tooth thickness. The unworn tooth area was assumed as the basic one. Wear value was gauged in each 5 mm heightwise within three and more points across the tooth width. Then, the wear values were averaged. As a result, dependences of tooth wear values upon the number of mesh intervals were obtained.

Analysis of the experimental data has helped determine that an open gear wear process has three distinctive periods: running-in period, stable period, and catastrophic one. The latter results from 3.5-4.0 mm wear of tooth working surface. In terms of greater wear value, gear operation is prohibited since it factors into sharp durability reduce of the whole gear. The accelerated wear is typical for the running-in and catastrophic periods. A value of running-in gear wear is s_{10} = 0.5-0.8 mm being s_{10} = 0.3-0.6 mm in terms of a rim. Moreover, dependence of tooth and rim wear rates upon rim tooth wear value has been determined.

It has been identified that open gears with relatively low hardness of working surfaces of teeth (i.e. (265-280) H_1B_1 and (260-300) H_2B_2) are subject to intensive abrasion. During the life cycle of a tooth rim, several gears operate with it. Each following gear installation (or its reinstallation to enable operation of teeth of another working surface) initiates running-in mode demonstrating heighten intensity wear of gear and rim teeth. Moreover, wear rate of a drive gear teeth increases progressively along with the increased wear of rim teeth. Hence, durability of each following gear will be shorter than that of a previous one. Experimental data processing has made it possible to derive the dependence of wear rate of gear teeth as well as a rim one upon wear rate of rim teeth:

$$U_{1(i-1)} = U_{10}\left[1 + k\Delta s_2^{\chi}\right], \quad (1)$$

where:

k and χ – constant coefficients determined experimentally.

Methods of gear wear analysis are based upon the studies of abrasive grain mechanics within the teeth contact area, further advanced and deepened by G.Ya. Yampolski, A.P. Natarov and I.V. Kragelski.

Abrasion model of surfaces of friction pairs (particularly, it concerns heavy loaded gears) takes into consideration a characteristic of abrasion influence A; physicomechanical properties of $M_{1(2)}$ materials; and geometrical and kinematic parameters of K gear pair. According to the model, in terms of μм wear per one load will be:

$$U_{1(2)} = 0.69\frac{AK}{M_{1(2)}}, \quad (2)$$

where:

$$A = q_a^{2/3}R^{0.5}\sigma^{2.5};$$

$$M_{1(2)} = \delta_{1(2)}^{z}HB_{1(2)}^{1.5}HB_{2(1)};$$

$$K = \sqrt{\rho^*}\frac{V_1 - V_2}{V_1 + V_2}.$$

In this context, 1 and 2 indices belong to working surfaces of gear and wheel respectively. Moreover, q_a is concentration of abrasive admixtures within the lubricant; R is their average radius, mm; $\delta_{1(2)}$ is strength, MPa; $\delta_{1(2)}$ is plasticity characteristic of surface layers, i.e. failure elongation,%; z is contact friction fatigue coefficient; $HB_{1(2)}$ is hardness of active surfaces of the teeth; R is equivalent radius of curvature of conjugate surfaces; ρ^* are radii of teeth curvature within a contact point, mm; and V_1, and V_2 is sliding velocity of the conjugate surfaces, m/s. K parameter is expressed through involute gear pair parameters.

Relying upon the experimental data and Expression (2), both analysis and evaluation of the average velocity of working teeth surface wear in terms of a stable operation will be performed using the formula:

$$U_{1(2)0} = 60 \cdot \varPhi_{1(2)}Kn_{1(2)}\nu_2 L_{1(2)}, \text{ mm/h}, \quad (3)$$

where:

$\varPhi_{1(2)}$ – experimental parameter determined for the given operational conditions; it involves characteristic of abrasion influence and physicomechanical properties of materials of working surfaces;

K – a design parameter involving geometrical and kinematic parameters of gear pair [37];

$n_{1(2)}$ – is rotation of gear wheel;

ν_2 – the number of gears operating with a tooth rim;

$L_{1(2)}$ – a coefficients involving dissimilarity of abrasion influence of environment, physicomechanical properties of material, and teeth loading conditions for the controlling case from those ones in terms of which $\varPhi_{1(2)}$ parameter was identified.

According to Expression (2), it may look like:

$$L_{1(2)} = \frac{\xi}{k_{qa}^{2/3}k_R^{0.5}k_\sigma^{2.5}\mu_{\delta 1(2)}^{z}\mu_{HB1(2)}^{1.5}\mu_{HB2(1)}}, \quad (4)$$

where:

$$k_{qa} = \frac{q_{ae}}{q_{ap}};$$

$$k_R = \frac{R_e}{R_p};$$

$$k_\sigma = \frac{\sigma_e}{\sigma_p};$$

$$\mu_{\delta 1(2)} = \frac{\delta_P}{\delta_e};$$

$$\mu_{HB1(2)} = \frac{HB_{p1(2)}}{HB_{e1(2)}},$$

where values with e index corresponds to those variables for which $\varPhi_{1(2)}$ parameter was determined experimentally; and p index corresponds to the controlling case.

For the conditions, where open gear is not protected from getting of mechanical admixtures and USsA lubricants, gear wheels should be manufactured from medium-carbon low-alloyed constructional steel.

Hardness of active teeth surfaces should be HB_2 260-300, and HB_2 180-200 with $\varPhi_1 = 3.2 \cdot 10^{-10}$ mm$^{1/2}$, and $\varPhi_2 = 6.4 \cdot 10^{-10}$ mm$^{1/2}$ averaged value [38].

3. Results and discussion

If running-in wear period is ignored then time of i^{th} working gear surface up to $[s_1]$ will be defined on the formula:

$$T_{1i} = \frac{[\Delta S_1] - r_1\Delta S_{10}}{U_{1(i-1)}}, \quad (5)$$

where:

$U_{1(i-1)}$ – average wear velocity of i^{th} working surface of gear teeth:

$$U_{1(i-1)} = \frac{U_{1i} + U_{1(i+1)}}{2}. \quad (6)$$

Apply wear velocity values from (1) to Expression (6). Then, to identify average wear velocity of i^{th} working surface of gear teeth, the Expression (6) will be as follows:

$$U_{1(i-1)} = U_{10}\left\{1 + \frac{1}{2}k\Delta S_2^{\chi}\left[(i-1)^{\chi} + i^{\chi}\right]\right\}. \quad (7)$$

In terms of Expression (7), a value of rim teeth wear may be represented as the total of wear rates they withstand while operating with each working gear surface:

$$\Delta S_{2i} = \sum_{i=1}^{n}\left(\nu_2 r_2 \Delta S_{20} + U_{2(i-1)}T_{1i}\right) i. \quad (8)$$

Apply T_{1i} from (5) to (8), and assume that $\frac{U_{2(i-1)}}{U_{1(i-1)}} = \frac{U_{20}}{U_{10}}$.

Hence, Expression (9) will be written as:

$$\Delta S_{2i} = \left(\nu_2 r_1 \Delta S_{20} + \frac{U_{20}}{U_{10}}\left[\left[\Delta s_1\right] - r_1 \Delta s_{10}\right]\right) i. \quad (9)$$

Gear rim durability is determined using the expression:

$$T_2 = \sum_{i=1}^{n} T_{1i}. \quad (10)$$

Use of more effective lubricants and automatic lubricant systems is one of the ways to improve open gear durability [38], [46], [47]. Consider influence of hardness of working surfaces of gear teeth and rim in terms of open gear of ore-grinding mill MShTs 55×65 (Table 1).

Table 1. Geometry of open gear

Geometry	MShTs 55×65
Motor power N, kW	4000
Rotation, rpm, of:	
– gear n_1	75.0
– rim n_2	13.7
Teeth number of:	
– gear z_1	46
– rim z_2	252
Teeth module m, mm	25
Advance angle β, degrees	6
Tooth rim width b_w, mm	900

The calculation involved following initial data: average value of a coefficient taking into consideration interlocking geometry $K = 77.7$ mm$^{0.5}$; average wear values of working surfaces of gear teeth and a rim from each working surface being $[s_1] = 3.5$ mm and $[s_2] \leq 4$ mm as admissible values, running-in values being $s_{10} = 0.8$ mm and $s_{20} = 0.4$ mm; the number of running-in periods per a wear interval of each working teeth gear surface being $r = 2$; a coefficient taking into consideration dissimilarity of one controlling case from experimental one (if harnesses of active surfaces of teeth are $HB_1 = 270$ and $HB_2 = 190$) being $L_{1(2)} = 1$, and $L_1 = 0.5$, and $L_2 = 0.6$ if $HB_1 = 550$, and $HB_2 = 300$; and experimental parameter $\Phi_{1(2)}$ being $\Phi_1 = 3.2$ mm$^{0.5}$ and $\Phi_2 = 6.4$ mm$^{0.5}$ if one gear with a tooth rim operates and $\Phi_2 = 12.8$ mm$^{0.5}$ if two gears with tooth rim operate.

Analysis of calculation results concerning open gear durability shows that operation of one gear with a rim demonstrates acceptable results even if teeth working surfaces are of relative hardness (Fig. 1a). However, in terms of operation of two gears with tooth rim, the tooth gear durability experiences its more than twofold reduction (Fig. 1b). High durability of open tooth gear results from the increased hardness of working surfaces of the teeth (Fig. 1b). Hardening of working surfaces of gear teeth up to $HB_1 = 550$ value, and hardening of working surfaces of rim teeth up to $HB_2 = 300$ value will make it possible to achieve almost 12 year durability of a tooth rim when two gears operate with it during the whole lifetime.

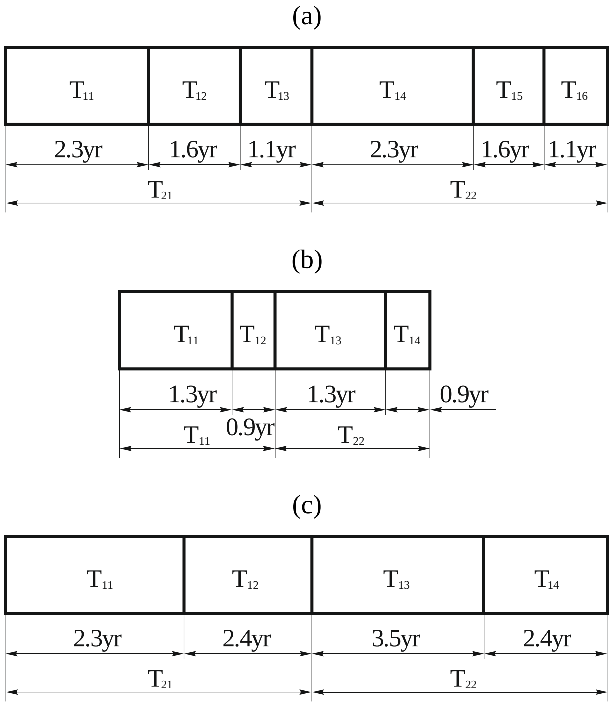

Figure 1. Durability of open tooth gears in terms of their wear: (a) single-engine drive with $HB_1 = 270$ hardness of gear teeth and $HB_2 = 190$ rim hardness; (b) two-engine $HB_1 = 270$ and $HB_2 = 190$ drive; (c) two-engine $HB_1 = 550$ and $HB_2 = 300$ drive; T_{11}, T_{12}, T_{13}, T_{14} – durabilities of 1st, 2nd, 3rd, and 4th working surface of gears; T_{21}, T_{22} – rim durability in terms of 1st and 2nd wear of working surfaces

The power, transferred to a drum from one drive gear to another one through a tooth rim, is limited by the geared drive strength. In the context of other consistent conditions, increase in bearing capacity of the drive gear is possible owing to the increased teeth module, tooth rim width, and working surface hardness. On the other hand, increase in teeth module results in the increased diameter of a drive gear as well as teeth rim. In turn, the abovementioned factors into the increased errors while the gear-tooth system manufacturing and assembling. Among other things, it concerns the increased irregularity of load distribution in terms of tooth rim width. In this connection, teeth module cannot exceed $m = 30$ mm and maximum diameter of the tooth rim should not be more than 12 m.

Pressure line lengthening is determined with the help of tooth rim width and angle of teeth advance. Increase in the angle of teeth advance is limited by axial force value. It is $\beta \approx 0\text{-}6°$ for the current drum mills. The increased bearing capacity of gear, resulting from the extension of tooth rim width, is limited by load distribution uniformity along contact line length. In terms of the specified errors of gear systems, ultimate tooth rim width is available, which exceedence cannot initiate a contact line lengthening and, hence, the gear system strength. Thus, in the majority of cases, tooth rim width of modern mills is not more than b = 1 m.

Increase in hardness of working surface of teeth improves contact strength of the teeth as well as wear properties of the gear system. On the other hand, the abovementioned decelerates running in velocity of a gear system resulting in excessive nonuniformity of load distribution along the contact line length. Consequently, the static load, acting on a gear system F_t (being peripheral force within indexing cylinder) experiences nonuniform distribution along the length of the contact lines as well as between the teeth. Moreover, both external and internal dynamic loads also influence teeth strength. To take into consideration the factors, static peripheral force, acting in the end section, is multiplied by a load coefficient Kn while calculating for contact strength or by K_F while analyzing bending [48]:

$$K_H = K_A \cdot K_{Hv} \cdot K_{H\beta} \cdot K_{H\alpha};$$

$$K_F = K_A \cdot K_{Fv} \cdot K_{F\beta} \cdot K_{F\alpha}, \qquad (11)$$

where:

K_A, K_{Hv}, $K_{H\beta}$, $K_{H\alpha}$, K_{Fv}, $K_{F\beta}$ and $K_{F\alpha}$ – coefficients involve external dynamic load; internal dynamic load; nonuniformity of load distribution along the length of a contact line; and load distribution between teeth respectively.

Errors of tooth direction, deviations of axial pitches, misalignment of axes and their skewness, resulting in mutual teeth skewness, are unavoidable while gear system manufacturing and assembling. According to the data by manufacturers of domestic mills, $\gamma_x = \gamma_y = 0.3 \cdot 10^{-3}$ may be assumed relying upon tolerances to assemble angles of obliquity γ_y and misalignment of axes γ_x. Skewness and misalignment of the toothed wheel axes initiate nonuniform load distribution along the length of contact lines. In terms of the specified angles of obliquity and misalignment of the toothed wheel axes, load distribution along the length of contact lines is influenced by a rim width as well as by hardness of working surfaces of teeth.

Tooth rim mounting right on a drum is the feature of open gears at large drum mills. Significant axial rim runout arises if misalignment of geometrical axes of a drum rotation and a tooth rim takes place. According to the data by NKMZ, axial rim runout for rim gears with d_2 = 5-8 m diameter should not be more than $\Delta_{\delta\max}$ = 1.2-2.0 mm. As for the best world-class product, maximum value of axial runout is not more than $\Delta_{\delta\max}$ = 0.7 mm. Axial runout stipulates mutual teeth skewness which variation follows a harmonic law with $\gamma_\delta = \Delta / d_2$ drum rotation and amplitude.

Taking into consideration axial runout of a rim, total angle of teeth angle of obliquity involves a constant component and variable component. Constant component of angle of obliquity cannot run in since each following teeth pair interacts with different angles of obliquity. For instance, in terms of MShTs 5.5×6.5 mills, similar angles of obliquity for teeth pair will be observed for the period during which gear performs 126 rotations, and rim performs 23 ones [38]. Operation of gear systems of domestic mills with $500H_1B_1$ hardness of working surfaces of gear teeth and $\Delta_{\delta\max}$ = 1.2-2.0 mm axial rim runout resulted in teeth breakdown.

To identify teeth breakdown causes, consider MShTs 5.5×6.5 mills. Table 1 demonstrates some of its parameters.

Calculations show that in terms of $\Delta_{\delta\max}$ = 2 mm and b_w = 900 mm rim width, teeth pair will contact along the b^*_w = 700 mm length with load distribution corresponding to a coefficient of distribution nonuniformity $k_{F\beta}$ = 2 [38]. Hence, in this context bending analysis should involve b^*_w = 700 mm rim width and $k_{F\beta}$ = 2.

If $\Delta_{\delta\max}$ = 0.7 mm then teeth pair will contact full width of a rim b_w = 900 mm with $k_{F\beta}$ = 1.6 coefficient. For illustrative purposes, Figures 2a and 2b demonstrate linearity schemes of load distribution full width of a rim for the two cases.

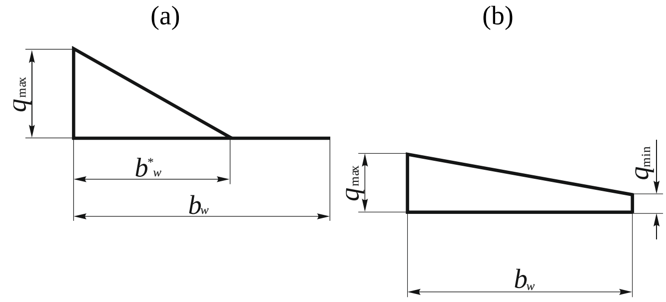

Figure 2. Linearity schemes of load distribution full width of the rim: (a) radial runout of 2 mm rim; (b) radial runout of 0.7 mm rim

Determine bending stresses within a critical tooth section using the formula:

$$\sigma = \frac{M_{Kp}}{W_r} = \frac{6F_t K_F h}{b_w s^2}, \qquad (12)$$

where:

M_{Kp} – gear shaft torque;

W_r – modulus of resistance;

F_t – normal load acting on the tooth;

h – arm of force;

s – tooth thickness within the critical section.

While assuming that dynamic load and load distribution between the teeth are similar for the two considered cases, use Formula (2) to identify how many times bending stresses σ_1 within the critical tooth section are more in terms of $\Delta_{\delta\max}$ = 2 mm than bending stresses σ_2 in terms of $\Delta_{\delta\max}$ = 0.7 mm.

$$\frac{\sigma_1}{\sigma_2} = \frac{k_{F\beta(1)} b_w}{k_{F\beta(2)} b^*_w} = \frac{2 \cdot 900}{1.6 \cdot 700} = 1.6 .$$

Hence, stresses within the critical teeth sections of open gears of the best world-class mills are more than 1.6 times less. The abovementioned as well as lower internal dynamic loads make it possible to provide reliable operation of a gear system differing in higher hardness of working surfaces of teeth. Since open gear systems of domestic mills have not such reserves, their bending strength may result from rapid teeth running-in which is possible if only working surfaces of the teeth are of relatively low hardness.

Nonuniformity of load distribution along the length of contact lines, stipulated by errors made in the process of gear system manufacturing and assembling, restricts a possibility to improve its bearing capacity due to the increased width of a rim. Solving the problem of tooth skew compensation follows the line to design tooth wheels having mobility within *rim-hub* junction. The jointed tooth wheels with elastic fastening of a gear rim and hub are among them. Gear design in the form of a tooth wheel with a rim on a spherical hinge is the most advantageous engineering solution for self-adjusting heavy-loaded gear system. The matter is that in terms of the design, *gear rim-hub* junction has the form of involute profile splines with a dome-shaped modification. Figure 3 demonstrates structural version of self-adjusting tooth wheel. Torque from hub 2 to rim 3 is transferred through a spline joint. In turn, rim 3, being engaged with a double tooth wheel (it is not shown in Figure 3), transfer torque to it. In the context of nonuniform load distribution, the total force, acting on the teeth of the self-adjusting tooth wheel, produces a torque relative to axis passing through a centre of spherical bearing surface *5* and *6*, under the action of which the rim turn takes place until the total force passes through the spherical surface centre.

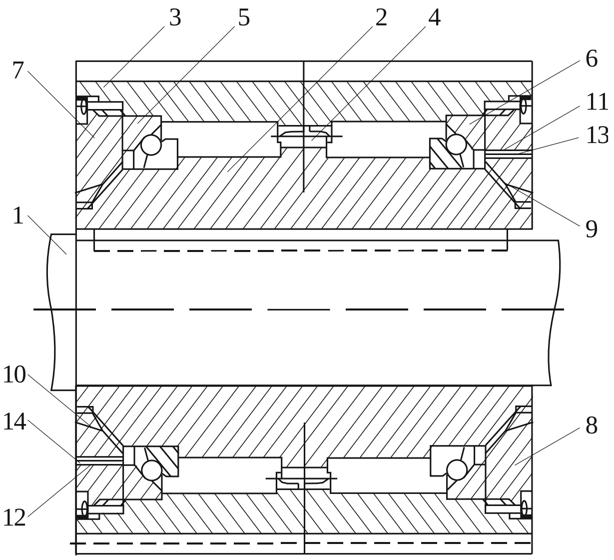

Figure 3. Self-adjusting tooth wheel: 1 – shaft; 2 – hub; 3 – rim; 4 – involute profile splines of a dome-shaped form; 5 and 6 – axial bearings; 7 and 8 – covers; 9 and 10 – consolidators; 11 are 12 – thread holes to supply lubricant; and 3 and 14 – stoppers

As the calculations of gear systems, performed using finite-element method, have shown those self-adjusting gears cannot provide perfect load distribution across the width of a tooth rim [38]. The abovementioned depends on the fact that extension in the rim width experiences increasing influence of elastic deformation of gear system. For instance, in terms of MShTs 5.5×6.5 mill, where tooth rim width is b_w = 0.9 m, full contact of “dangerous” pair of teeth, located above a hub, will help reduce coefficient of load nonuniformity from $K_{F\beta}$ = 2.7 down to $K_{F\beta}$ = 1.2. On the other hand, use of a self-adjusting gear system results in excessive dynamic load within the gear [48]. Theoretical analysis of the self-adjusting gear dynamics has demonstrated that the excessive dynamic loads depend upon the impact teeth speed at the end of a phase of angle of obliquity picking.

Evaluate dynamic load within a gear using dynamic coefficient k_v being equal to ratio between maximum load $F_{\max}$ within the contact and dynamic one F_v:

$$k_v = \frac{F_{\max}}{F_v}.$$

Figure 4 demonstrates analytical values of dynamic coefficient k_v dependence upon the width of a tooth rim b_w of a self-adjusting drive gear of MShTs 5.5×6.5 mill [49].

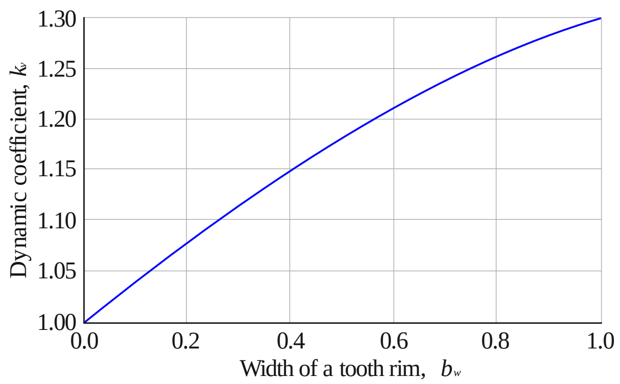

Figure 4. Dependence of dynamic coefficient k_v upon width of a tooth rim b_w of self-adjusting drive gear of MShTs 5.5×6.5 mill if turn angle of a mobile gear rim is γ_0 = 1.2·10^4 rad

As Figure 4 explains, if MShTs 5.5×6.5 mill is equipped with a self-adjusting gear, that may result in the increased dynamic load. For instance, in terms of b_w = 0.9 m width of a tooth rim, dynamic coefficient will be k_v = 1.28. On the other hand, use of the self-adjusting gear will decrease a coefficient of load distribution nonuniformity $K_{F\beta}$ across the tooth rim from 2.7 down to 1.2 making it possible to perform 1.8 times decrease in the load coefficient, i.e. from 2.7 down to $K_{F\beta}$ = 1.2 1.28 = 1.5.

4. Conclusions

Open gears, having relatively low hardness of working surfaces, i.e. (260-300) H_1B_1 and (180-200) H_2B_2, are subject to abrasion advancing in proportion to wear of the open gear teeth. Several drive gears operate with a gear rim during its age. Durability of each following gear reduces progressively.

On the one hand, intensive wear of the open gear teeth decreases its durability. On the other hand, it provides rapid running-in of working surfaces of teeth which decreases load distribution nonuniformity across the tooth rim width. In terms of high nonuniformity of load distribution nonuniformity across the tooth rim width, the case provides teeth bending strength. The increased hardness of working surfaces of teeth decelerates running-in process resulting in their breakdown.

High wear rate of teeth having relatively low hardness of working surfaces prevents from the increase in unit mill capacity due to the use of drives involving operation of two drive gears with one tooth rim.

Application of self-adjusting tooth gears for drum mill drives will help: increase from 4000 up to 6000 kW and more a value of power transferred through a tooth rim by one drive gear; increase hardness of working teeth surfaces up to 600 HB while improving abrasion resistance of a tooth gear; and increase unit capacity of mills by means of power transmission to a drum through a tooth rim using two drive gears.

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Шляхи підвищення довговічності та несучої здатності відкритих зубчастих передач барабанних млинів

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Мета. Обгрунтування шляхів підвищення довговічності та несучої здатності відкритих зубчастих передач рудорозмельних барабанних млинів, а також ефективності технічних рішень зі збільшення їх одиничної потужності за рахунок вдосконалень приводу.

Методика. Узагальнено результати багаторічних експериментальних і теоретичних досліджень щодо абразивного зношування робочих поверхонь зубів відкритих зубчастих передач барабанних млинів, а також факторів, що впливають на розподіл навантаження по ширині зубчастого вінця.

Результати. Проведено порівняльний аналіз вітчизняних млинів з кращими світовими зразками. Показано шляхи вирішення завдань, пов'язаних зі створенням млинів великої одиничної потужності з редукторним приводом. Дана кількісна оцінка впливу твердості робочих поверхонь зубів на їх довговічність. Проведено аналіз факторів, що впливають на розподіл навантаження по ширині зубчастого вінця. Надана оцінка ефективності застосування у відкритих зубчастих передачах самовстановлюваних привідних шестерень.

Наукова новизна. Виявлено функціональний зв'язок між фізико-механічними властивостями робочої поверхні зубів, числом прироботочних режимів, які визначаються умовами експлуатації, і терміном служби відкритої зубчастої передачі, розглянуті фактори, що впливають на розподіл навантаження по ширині зубчастого вінця.

Практична значимість. Показано, що застосування відкритих зубчастих передач з твердістю робочих поверхонь зубів шестерні (500-600) H_1B_1 і вінця (260-300) H_2B_2 дозволяє забезпечити практично без зносний режим роботи і є необхідною умовою при роботі зубчастого вінця з двома приводними шестернями.

Ключові слова: *барабанний млин, відкрита зубчаста передача, розподіл навантаження, абразивний знос, самовстановлювана шестерня*

Пути повышения долговечности и несущей способности открытых зубчатых передач барабанных мельниц

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Цель. Обоснование путей повышения долговечности и несущей способности открытых зубчатых передач рудоразмольных барабанных мельниц, а также эффективности технических решений по увеличению их единичной мощности за счет улучшений в приводе.

Методика. Обобщены результаты многолетних экспериментальных и теоретических исследований по абразивному износу рабочих поверхностей зубьев открытых зубчатых передач барабанных мельниц, а также факторов, влияющих на распределение нагрузки по ширине зубчатого венца.

Результаты. Проведен сравнительный анализ отечественных мельниц с лучшими мировыми образцами. Показаны пути решения задач, связанных с созданием мельниц большой единичной мощности с редукторним приводом. Дана количественная оценка влияния твердости рабочих поверхностей зубьев на их долговечность. Проведен анализ факторов, влияющих на распределение нагрузки по ширине зубчатого венца. Дана оценка эффективности применения в открытых зубчатых передачах самоустанавливающихся приводных шестерен.

Научная новизна. Выявлена функциональная связь между физико-механическими свойствами рабочей поверхности зубьев, числом приработочных режимов, определяемых условиями эксплуатации, и сроком службы открытой зубчатой передачи; рассмотрены факторы, влияющие на распределение нагрузки по ширине зубчатого венца.

Практическая значимость. Показано, что применение открытых зубчатых передач с твердостью рабочих поверхностей зубьев шестерни (500-600) H_1B_1 и венца (260-300) H_2B_2 позволяет обеспечить практически без износный режим роботы и является необходимым условием при работе зубчатого венца с двумя приводными шестернями.

Ключевые слова: *барабанная мельница, открытая зубчатая передача, распределение нагрузки, абразивный износ, самоустанавливающаяся шестерня*