

ГЕОТЕХНІЧНА І ГІРНИЧНА МЕХАНІКА, МАШИНОБУДУВАННЯ

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THE STRESS STATE OF HEAVY LOADED OPEN GEARING WITH INCOMPLETE TOOTH CONTACT

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НАПРУЖЕНИЙ СТАН ВАЖКОНАВАНТАЖЕНИХ ВІДКРИТИХ ЗУБЧАСТИХ ПЕРЕДАЧ ПРИ НЕПОВНОМУ КОНТАКТІ ЗУБІВ

Purpose. Development of a design procedure of tooth bending durability upon incomplete tooth contact by a finite element method and its application for research of a stress state of tumbling mill open gearing.

Methodology. Calculation of the stress state is carried out with the finite element method. At the first stage the design model is formed. Further the construction and discretization of gearing geometrical model is done taking into account an axis misalignment angle. Displacements and stress in a tooth gearing upon incomplete tooth contact are calculated. Received distribution diagrams of displacements and stress are used for calculation of tooth bending stress.

Findings. The article is devoted to the development of a design procedure of tooth bending durability taking into account gearing errors. The current state of the issue of tooth bending durability calculation upon incomplete tooth contact is investigated. The factors having a significant impact on a gearing error are defined. It is shown that the greatest impact on tooth gearing durability has an inclination angle of teeth in the plane of teeth contact. The estimation of the value of total inclination angle of teeth of tumbling mill МШРГУ 4500 × 6000 is given. The design model for calculating the tooth bending durability upon incomplete tooth contact with the finite element method is proved. Comparison of the results of calculation by the developed technique and by the engineering of ISO 6336-1:2006, 6336-3:2006 is made. It is shown that the existing engineering technique does not consider the features of large-sized tooth gearings stress state technique enough. The correction factors which allowing adapting an engineering technique for calculating tooth bending durability upon incomplete tooth contact are calculated.

Originality. Significant influence of the central plate on tooth bending durability is established. It is established that upon incomplete tooth contact the load is distributed linearly along the theoretical line of the contact.

Practical value. The algorithm of a design procedure of tooth bending durability upon incomplete tooth contact with the finite element method is developed. The quantitative estimation of tooth bending durability of tumbling mill МШРГУ 4500x6000 open gearing is received.

Keywords: *gearing, stress state, incomplete tooth contact, finite element method*

Introduction. Tooth gearings are the most common elements of machine drives. The existing tendency of increase in individual unit capacity of tumbling mills, tube mills and ball mills, as well as tube furnaces has led to creation of the high-loaded tooth gearings with a

torque on a shaft of about hundreds and thousands kilonewton per metre. Reliability of such machines essentially depends on reliability of tooth gearings.

Practice shows that the common cause of large-sized tooth gearings fatigue involves errors of gearing design due to inaccurate determination of load capacity of gearings [1]. Inaccuracy of gear manufacturing,

elastic deformations of gears, shafts, supports and teeth, wear of bearing brass, foundation subsidence and face runout of a rim lead to nonuniform distribution of loading along the length of contact lines. To calculate loading capacity of spur and helical gears according to the AGMA [2, 3] and ISO [4, 5] standards the Navier equation are used; the standards were obtained in the assumption of uniform distribution of load and correcting factors which consider the inconsistency between theoretical and real load distribution. These correcting factors are obtained under operating conditions of gears of general mechanical engineering and they do not consider the loading conditions of the heavy-loaded gearings precisely enough. The application of modern computer aided engineering systems (CAE-systems) along with the methods of theory of elasticity allows estimating the actual behaviour of the computer model under operation conditions and ensuring operability of a product without significant time consumption and finance. However, the application of design procedures on the basis of CAE-systems is a knowledge-intensive process requiring careful analysis of data and verification of results. Thus, the development of a reliable gearing design procedure is quite topical.

Analysis of the recent research. Due to the intensive wear process and plastic deformations of teeth working surfaces the running-in occurs during the operation period. The running-in induces redistribution of loadings across the gear width, conducive to their alignment. However, occurrence of the variable components of a gearing error leads to the failure of complete load distribution. As a result, a shift of the line of contact occurs (Fig. 1).

There are a lot of scientific works which are devoted to the theoretical and experimental study of the effect of gearing error on the stress state of gearing. Among them works [1, 6–10] can be pointed out.

Applying FEM, Hani A. [6], and later Lias M., Khan M. [6] showed the essential effect of a misalignment on bending stiffness of the rigidly fixed tooth. Thus incomplete contact was simulated by the application of non-uniformly distributed loading according to the parabolic law. Shuting Li [8, 9, 10] experimentally investigated the non-uniformity of load distribution on tooth width of small module ring gears (the torque is $294 \text{ N} \cdot \text{m}$) in a transmission and has also shown that the load distribution along the line of contact is close to the linear one (Fig. 2). With the use of FEM the design procedure of a solid gears stress state upon incomplete tooth contact is developed. It is shown that a skew corner in the tooth contact plane ε_3 influences the stress in the tooth root the greatest (Fig. 1). The dependence of the maximum bending stress on a distortion corner ε_3 has linear character.

Sanchez, Pedrero and Pleguezuelos used the analytical model which considers nonuniform distribution of loading along the length of the contact line. The model is based on the potential energy of the elastic strain and linear strain of teeth. Considering the load distribution law, the model allows defining the

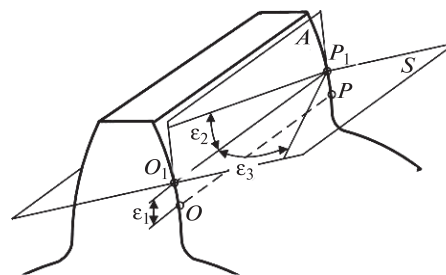


Fig. 1. Specification of components of a gearing error:

A – the plane tangent to a working surface of tooth; *S* – the plane of teeth contact; *PO* – the contact line in the absence of gearing errors; *P₁O₁* – the contact line at parallel shift of axes; ε_1 – parallel shift of axes; $\varepsilon_2, \varepsilon_3$ – skew corners of axes in the planes *A* and *S* respectively

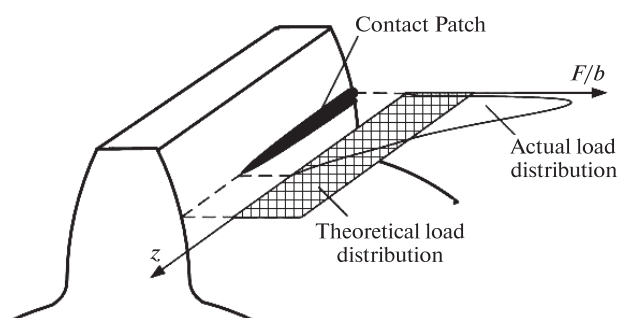


Fig. 2. Load distribution along the line of tooth contact upon complete and incomplete contact:

F – normal force, *b* – width of tooth; *z* – longitudinal coordinate

position and value of the maximum bending stress in the tooth root.

The works listed above are devoted to the research of small module ring gears of the general mechanical engineering. In work [1] it is noted that the gears of complex design are used in large-sized tooth gearings; their stiffness is considerably non-uniform across the width of the contact line. Elastic deformations of a gear contribute significantly to the size of a skew corner of teeth. Experimental studies were conducted regarding the non-uniformity of loading distribution along the length of contact lines in a spur gear open tooth gearing of mill МШРГУ 4500×6000 at Mikhaylovsky mineral processing plant with use of strain gage equipment. The results have shown, that even after the teeth running-in, the considerable non-uniformity of loading distribution is observed. With the width of a gear of 800 mm before and after running-in, the length of the loaded part of tooth has made 395 and 702 mm respectively. Linear nature of loading distribution is observed simultaneously. At the present time there is no reliable design procedure of estimating stress state of large-sized tooth gearings which allows considering the above-stated features of loading.

Objective of the study is the development of a reliable design procedure of calculating the stress state of large-sized tooth gearings with the use of FEM which

allows considering the non-uniformity of loading distribution along the length of contact lines due to gear coupling misalignment and elastic deformation of gears.

Statement of the problems. To reach the stated objectives of the study, the contact interaction problem of ring gear and pinion is solved. The present task features the discrepancy of theoretical contact lines of ring gear and pinion teeth which results in considerable stress concentration in local area of working surface and tooth root. The complex design of the gear rim causes significant 3D deflected mode and possibility of stress concentration in a rim body. To succeed in achieving the purpose, it is necessary:

- to construct a three-dimensional geometrical model of ring gear and pinion;
- to set an exact relative position of ring gear and pinion taking into account the calculated value of a skew corner of teeth;
- to make a rational FE discretization of the geometrical model considering the expected places of stress concentration;
- to apply the constraints and calculated loading;
- to carry out the calculation and to compare the results of the calculation to those of experimental studies, than to confirm adequacy of the developed design procedure.

Substantiation of the analytical model. In the present study the open tooth gearing of tumbling mill МШРГУ 4500 × 6000 is investigated. The geometrical model of a tooth gearing is constructed according to drawings of the manufacturer and the data are presented in Table 1. At the same time such features of the design as chamfers, mounting holes and the neighbouring teeth are excluded, as being insignificant. The form of ring gear 2 (Fig. 3) is limited by 120° rigidly fastened segment, the pinion 1 is the solid cylinder with one degree of freedom. The diameter of a pinion landing surface 3 is no more than 2/3 of the pinion diameter. The torque is applied to landing surface 3 and its value is presented in Table 2. The longitudinal plane of symmetry of the ring gear tooth 4 coincides with the longitudinal plane of symmetry of the gusset 5.

To bring the geometrical model in accordance with the analytical model, the pinion is rotated according to the misalignment angle γ calculated beforehand in the contact plane S . Contact plane S passes through the theoretical lines of contact the working surfaces of the gear rim and pinion teeth. At the same time teeth touch each other at point P – the extreme point of the contact line of on the working surface of tooth.

Discretization of the model is carried out by splitting the model into tetrahedron finite elements. Since the big gradient of stress is expected in the area of the contact line and point P, the grid compaction is carried out in the specified area to ensure the calculation accuracy. The contact type is contact with friction (the friction coefficient is 0.05). Recommended SAPR – FreeCAD, the grid generator – GMSH, the processor and the post-processor – Calculix being the free software.

Results of calculations and their discussion. The maximum misalignment angle is calculated by

Table 1

Key geometrical parameters of gear pair

Parameter	Symbol, unit	МШРГУ 4500 × 6000	
		pinion	ring gear
Number of teeth	z	28	252
Tooth width	b , mm	800	800
Pitch diameter	d , mm	700	6300
Pressure angle	a , °	20	
Normal module	m , mm	25	
Axle base	a_w , mm	3511	

Table 2

Initial data on the load of tumbling mill МШРГУ 4500 × 6000

Parameter	Symbol, unit	Value
Rotational frequency of pinion	n_1 , rpm	150
Torque	T_1 , kN · m	159,155
Gear-ratio	u	9

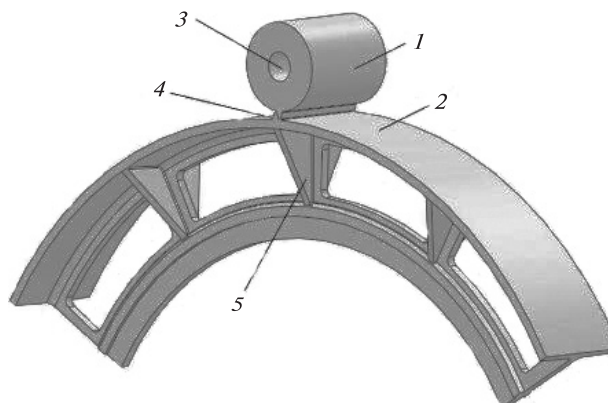


Fig. 3 Geometrical model of gearing:

1 – pinion; 2 – ring gear; 3 – landing surface of pinion; 4 – gear tooth; 5 – gusset

Petrusevich formula

$$\gamma_{\max} = \gamma_{\delta} + \gamma_{\Delta}, \quad (1)$$

where γ_{δ} is the misalignment angle, caused by face runout of ring gear; γ_{Δ} is the misalignment angle, caused by manufacturing and assembling errors.

Misalignment angle γ_{δ} , caused by face runout of ring gear is calculated on the basis of the technical specifications of the manufacturer, NKMZ, according to which the face runout $\Delta_{\delta_{\max}}$ for ring gears with diameters $d_2 = 5-8$ m should not exceed 1.2–2.0 mm. Then, assuming $\Delta_{\delta_{\max}} = 1.2$ mm and $d_2 = 6300$ mm (Table 1)

$$\gamma_{\delta} = 1.2/2 \cdot 3150 = 0.190 \cdot 10^{-3}. \quad (2)$$

Misalignment angle γ_{Δ} , caused by manufacturing and assembling errors, is calculated as

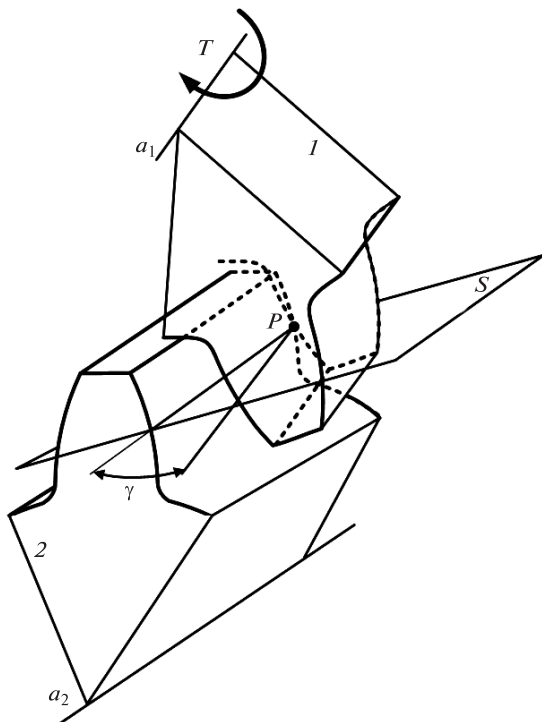


Fig. 4. Analytical model:
1 – pinion, 2 – ring gear; a_1, a_2 – axis of rotation of ring gear and pinion; S – contact plane; P – pitch point; T – torque; γ – misalignment angle

$$\gamma_{\Delta} = \sqrt{\gamma_{\beta 1}^2 + \gamma_{\beta 2}^2 + \gamma_y^2 \cos^2(\alpha_w) + \gamma_x^2 \sin^2(\alpha_w)}, \quad (3)$$

where $\gamma_{\beta 1(2)}$ is ratio of total alignment error of tooth to working tooth width of pinion and ring gear respectively, making $0.052 \cdot 10^{-3}$; γ_x, γ_y are skew angles and axis misalignment respectively, which can be assumed as $0.3 \cdot 10^{-3}$ according to the assembling tolerance; a_w is axle base (Table 1).

Substituting the selected values in the formula (3), we get

$$\begin{aligned} \gamma_{\delta} &= \sqrt{\gamma_{\beta 2}^2 + \gamma_y^2 \cos^2(\alpha_w) + \gamma_x^2 \sin^2(\alpha_w)} = \\ &= \sqrt{2 \cdot (0.052 \cdot 10^{-3})^2 + (0.3 \cdot 10^{-3})^2 \cdot 0.940^2 + (0.3 \cdot 10^{-3})^2 \cdot 0.342^2} = \\ &= 0.309 \cdot 10^{-3}, \end{aligned}$$

then, substituting the received value in (1), we get

$$\gamma_{\max} = \gamma_{\delta} + \gamma_{\Delta} = 0.309 \cdot 10^{-3} + 0.190 \cdot 10^{-3} = 0.499 \cdot 10^{-3}.$$

The results of the solution of the contact problem of gearing are presented in Fig. 5–6. In Fig. 5 a three-dimensional diagram of equivalent (Von-Mises) stress is presented, it occurs in a body of ring gear at value of misalignment angle $\gamma = \gamma_{\max}$.

The form of the contact patch presented in Fig.5 gives evidence of incomplete tooth contact due to skew angle. Stresses are distributed non-uniformly in relation to the central web of the ring gear. In Fig. 6 the curves of equivalent (Von-Mises) stress in tooth root are presented. Line 1 corresponds to complete contact; line 2 corresponds to incomplete contact. It is

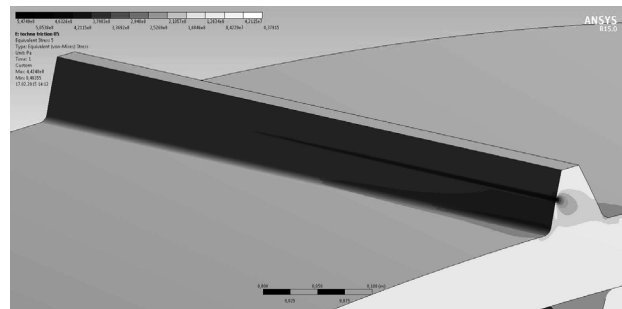


Fig. 5. Equivalent (Von-Mises) stress in a ring gear of open tooth gearing of mill МШПГУ 4500 × 6000 under $\gamma_{\max} = 0.499 \cdot 10^{-3}$

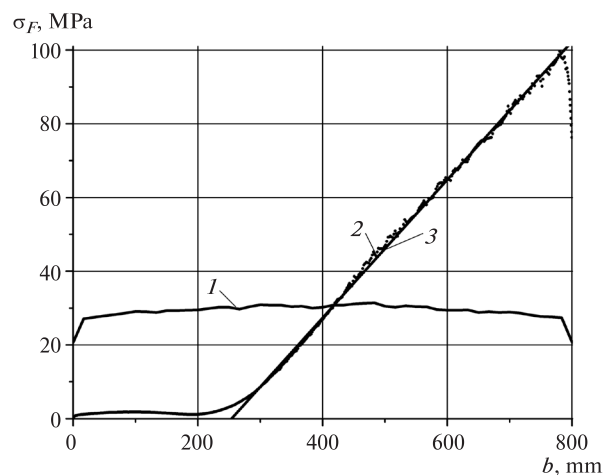


Fig. 6. Equivalent (Von-Mises) stress σ_F in the root of dangerous tooth of ring gear with length of b of open tooth gearing of mill МШПГУ 4500 × 6000:
1 – $\gamma_{\max} = 0$; 2 – $\gamma_{\max} = 0.499 \cdot 10^{-3}$; 3 – linear interpolation of line 2

shown that upon incomplete contact of tooth the stresses in tooth root are distributed according to the law which is close to the linear one. The calculated facewidth b_{cal} is of 550 mm.

Table 3 presents the results of gearing stress calculation. Data achieved by using ISO 6336-1, 6336-3 and FEM calculation technique are presented below.

Face load factor (root stress), $K_{F\beta}$, is calculated on the basis of tooth root equivalent stress diagrams obtained using FEM by formula

$$K_{F\beta} = \frac{\sigma_{F\max}}{\sigma_{Fm}},$$

where $\sigma_{F\max}$ and σ_{Fm} are the maximum and average values of equivalent normal stress in tooth, MPa. Then face load factor (contact stress), $K_{H\beta}$, is defined according to [5].

Calculation of maximum tooth stiffness per unit facewidth (single stiffness) of a tooth pair is done by formula

$$c_{\gamma} = F_n / (\delta \times b),$$

where F_n is normal force, kN; δ is displacement along the contact line in the direction of the line of action, μm .

Table 3

Results of stress calculation of open tooth gearing of mill МШРГУ 4500 × 6000

Parameter	Calculation technique			
	Complete contact		Incomplete contact	
	FEM	ISO	FEM	ISO
Face load factor (root stress), $K_{F\beta}$	1.075	1	2.69	3.79 (2.76)
Face load factor (contact stress), $K_{H\beta}$	1.081	1	2.90	4.19 (2.98)
Maximal tooth root stress, σ_F , МПа	38.42	76.49	99.60	289.70 (105.4)
Mesh stiffness, c_γ , кN/mm × μm	6.30	12.50	6.30	12,50 (6,30)

Table 3 shows that the data obtained by the method of ISO 6336-3:2006 have overstated values of stress in case of complete and incomplete contact. The reason lies in the overstated value of mesh stiffness c_γ . The method of stiffness calculation does not consider the flexibility of gear body properly. This, in turn, results in the overstated value of face load factor $K_{H\beta}$. Factor C_R , which considers the effect of tooth form on mesh stiffness [5], does not consider the presence of cutouts in web and gussets (Fig. 3). The results of FEM calculations show that factor C_R is to be multiplied by the correction factor of 0.5.

The data obtained confirm the experimental data obtained by B. Vinogradov [1]. He showed the mismatch of calculated stiffness to stress state of ring gear. Indeed, substitution of $b_{cat} = 550$ mm and $K_{H\beta} = 2.98$ in the formula for calculating $K_{H\beta}$ under condition of incomplete contact ($K_{H\beta} > 2$) [5]

$$K_{H\beta} = \sqrt{\frac{2F_{\beta\gamma} C_\gamma}{\frac{F_m}{b}}}$$

gives $c_\gamma = 7.5$ кN/mm · μm. This value corresponds to the value obtained by the developed technique within the accuracy of 15 %.

Furthermore, within the ISO method of calculating load capacity only the rim thickness factor, Y_B , considers the design of ring gear. As the investigations show, factor Y_B is correct in case of solid web and symmetrical position of the web relating to the rim. Otherwise, the value of factor Y_B must be corrected by analysing carefully the three-dimensional stress state of the ring gear. The results of FEM calculation show that factor Y_B is to be multiplied by the correction factor of 0.5.

In Table 3, the data of bending strength calculation along with the corrected values of factors C_R and Y_B are presented in parentheses. The data lead to the conclusion that application of corrected values of factors C_R and Y_B allow conforming the methods of ISO 6336-1 (Method C), 6336-3 (Method B) to high-loaded tooth gearings.

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Мета. Розробка методики розрахунку максимальної жорсткості пари зубів при однопарному зачепленні зубчастих коліс методом кінцевих елементів та її використання для оцінки жорсткості зачеплення відкритих зубчастих передач барабаних млинів.

Методика. Розрахунок напружено-деформованого стану здійснюється методом кінцевих елементів. На першому етапі формується розрахункова схема задачі. На базі геометричних параметрів та креслень здійснюється побудування тривимірної

геометричної моделі зубчастого зачеплення. Здаються фізичні характеристики конструкційних матеріалів. Геометрична модель розбивається на кінцеві елементи. Здаються граничні умови, що забезпечують кінематичну незмінність моделі, та умови контакту робочих поверхонь зубів. Здається зовнішнє навантаження у вигляді крутного моменту, що прикладений до шестерні. Здійснюється чисельне розв'язання рівнянь рівноваги. Визначається мінімальне переміщення вздовж контактної лінії зубів. Обчислюється жорсткість пари зубів.

Результати. Стаття присвячена розробці методики розрахунку жорсткості відкритих зубчастих передач барабаних млинів. Досліджено сучасний стан питання про розрахунок жорсткості зубчастих передач. Особливу увагу приділено недолікам існуючих стандартів стосовно великогабаритних зубчастих передач. Показані переваги методу кінцевих елементів при розрахунку деформованого стану елементів зубчастої передачі. Представлено алгоритм і методику розрахунку жорсткості зубчастих передач за допомогою методу кінцевих елементів, здійснено обґрунтування геометричної моделі та розрахункової схеми. Алгоритм розрахунку включає побудову геометричної моделі передачі, завдання механічних властивостей конструкційних матеріалів, розбиття геометричної моделі на кінцеві елементи, розрахунок переміщень і жорсткості зачеплення. Зроблене порівняння результатів розрахунку жорсткості зубчастої передачі барабанної млини МШРГУ 4500 × 6000 за розробленою методикою, стандартною методикою та результатів експериментального дослідження. Встановлено значне розходження результатів стандартної методики та результатів експериментів. Показано, що розроблена методика дає більш точну оцінку жорсткості завдяки повнішому врахуванню конструктивних особливостей вінця.

Наукова новизна. Встановлено істотний вплив конструкції центрального диска вінця на міцність зуба при згинанні. Встановлено, що при неповному контакті зубів навантаження розподіляється лінійно уздовж теоретичної лінії контакту зубів.

Практична значимість. Розроблено алгоритм методики розрахунку жорсткості відкритих зубчастих передач барабаних млинів методом кінцевих елементів. Отримана кількісна оцінка згинальної міцності зуба відкритої зубчастої передачі барабанного млина МШРГУ 4500 × 6000.

Ключові слова: барабанний млин, зубчаста передача, жорсткість, метод кінцевих елементів

Цель. Разработка методики расчета жесткости пары зубьев при однопарном зацеплении зубчатых колес методом конечных элементов и применение ее для оценки жесткости зацепления открытых зубчатых передач барабанных мельниц.

Методика. Расчет напряженно-деформированного состояния осуществляется методом конечных элементов. На первом этапе формируется расчетная схема задачи. На основании основных

геометрических параметров и чертежей производится построение трехмерной геометрической модели зубчатой передачи. Здаються фізическіє свойства конструкционных материалов. Геометрическая модель разбивается на конечные элементы. Здаються граничные условия, обеспечивающие кинематическую неизменяемость модели, и условия контакта рабочих поверхностей зубьев. Здається зовнішня нагрукка в виде крутящего момента, приложенного к посадочной поверхности шестерни. Производится численное решение уравнений равновесия. Определяется минимальное перемещение вдоль контактной линии зубьев. Рассчитывается жесткость пары зубьев.

Результаты. Стаття посвящена разработке методики расчета жесткости открытых зубчатых передач барабанных мельниц. Исследовано современное состояние вопроса о расчете жесткости зубчатых передач. Особое внимание уделено недостаткам существующих стандартов применительно к крупногабаритным зубчатым передачам. Показаны преимущества метода конечных элементов при расчете деформированного состояния элементов зубчатой передачи. Представлен алгоритм и методика расчета жесткости зубчатых передач с помощью метода конечных элементов, осуществлено обоснование геометрической модели и расчетной схемы. Алгоритм расчета включает построение геометрической модели передачи, задание механических свойств конструкционных материалов, разбиение геометрической модели на конечные элементы, расчет перемещений и жесткости зацепления. Произведено сравнение результатов расчета жесткости зубчатой передачи барабанной мельницы МШРГУ 4500 × 6000 по разработанной методике, стандартной методике и результатов экспериментального исследования. Установлено значительное расхождение результатов стандартной методики и результатов экспериментов. Показано, что разработанная методика дает более точную оценку жесткости благодаря более полному учету конструктивных особенностей венца.

Научная новизна. Установлено существенное влияние конструкции центрального диска венца на прочность зуба при изгибе. Установлено, что при неполном контакте зубьев нагрузка распределяется линейно вдоль теоретической линии контакта зубьев.

Практическая значимость. Разработан алгоритм методики расчета прочности зуба на изгиб открытых зубчатых передач барабанных мельниц методом конечных элементов. Получена количественная оценка изгибной прочности зуба открытой зубчатой передачи барабанной мельницы МШРГУ 4500×6000.

Ключевые слова: барабанная мельница, зубчатая передача, жесткость, метод конечных элементов

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