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Load-Bearing Capacity of Spiroid Gears of Mining Machine Drives under Peak Loads

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Abstract—The substantiated dimensionless criterion of jamming characterizes relationship between the temperature of the contact links of spiroid gearing and the load-bearing capacity of an oil film. Based on the review of the existing tribotechnical systems used in scoring resistance testing, a physical model of spiroid gearing is selected. The test data obtained on a rolling-and-disc scheme of a friction assembly are presented. The relations between the friction coefficient in the spiroid gearing and the unit load are determined for the steel–steel material couples in a range of slip velocities and temperatures. A design procedure for spiroid gearing with respect to jamming is proposed.

Keywords: Spiroid gear, gearing, scoring resistance, jamming.

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The review of the modern machines and mechanisms reveals an increase in the application gear transmissions [1]. The current toothing transmissions are being improved while the new, improved types of gears appear. For instance, spiroid gears are widely being used to for the transmission of rotation between crossing shafts.

Spiroid gears are more advantageous as against wormgears and can essentially enhance performance of mining machines, reduce their dimension and weight, extend life, cut down expenses connected with maintenance, repair and idle time, which will increase efficiency of labor of maintenance personnel. .

On the other hand, spiroid gears and wormgears have the same drawback, which is jamming of teeth. Critical tough jamming within a short time results in heavy wear of teeth, the teeth stop moving and the transmission becomes unserviceable. Aimed to avoid jamming, teeth are made of antifriction materials which abate jamming risk and, consequently, reduce capability of the transmission as compared with cylinder gears [1, 2].

Jamming is divided into three stages. The first stage is transition from the contact hydrodynamic to boundary lubrication. Under real conditions, jamming does not take place when the dynamic lubricating film disappears or becomes critically thin as the contact surfaces are yet separated by the boundary lubricant film. The second stage is transition from the boundary to metal contact as a consequence of failure of the boundary film or large deformation of contact surface, or high temperatures in the contact zone, or both. The thirds stage is grasping of metal and failure of jammed galling, etc. The surface temperature grows in the meanwhile, the damages develop and the surface become totally unserviceable [3, 4].

Jamming of spiroid gears was investigated on a physical model, which allowed saving material expenses connected with the manufacture of specimens and experimentation preparation time. The basic requirements imposed on physical model testing are [5, 6]:

—similarity of functions of real and model systems (inlets, outlets and their functional relationships);

—similarity of structures of real and model systems (elements, properties and tribological interactions).

The first requirement is fulfilled using the similarity criterion that describes interaction of key factors which influence jamming process in the real and test systems. The similarity criterion is found with regard to the theory of similarity, using the methods of integral analogs or dimension analysis [6]. The second requirement is connected with the choice of the physical model and design of a lab-scale testing plant such that to ensure setting and control of all parameters which have influence on the model process [5, 7].

The implemented factor analysis and systematization of the current jamming criteria for gears [2] shows that each criterion is applicable either for a certain type of a gear or at a certain stage of jamming. The Delphi procedure reveals the most weighty factors which influence initiation and development of spiroid gear jamming at different stages of the process [6].

At the first stage of jamming, the main influences are the lubricant viscosity and total rolling speed, which affect the oil film thickness. At the second stage, the influences are the temperature and load at the friction surfaces, which promote rupture of the oil film and transition to the direct contact between metals. At the third stage, the contact results in gripping. The key factor is the ability of metals to grip, i.e. properties of materials, hardness and thermal treatment of metal surfaces.

On this ground, it is possible to conclude that the most weighty factor to govern jamming at all stages is temperature. It influences the oil film thickness through viscosity at the first stage, facilitates rupture of the oil film at the second stage and affects mechanical properties of materials, intensifying and promoting gripping at the third stage.

Since the jamming criterion is supposed to be included in the gear design procedure, this criterion must characterize a combination of conditions that promote jamming. This implies exact rationing of gear operation before jamming enters the second stage when the contact conditions of teeth go from boundary friction to direct metal contact. Under poor rationing, the friction coefficient will jump, and so the temperature will do, which is called a temperature raise and favors development of galling. Mostly, the contact of metals under high loads and rubbing velocity leads to irreversible jamming and, finally, to holding and total failure of the gear. For this reason, the main criterion in the analysis of galling resistance of spiroid gear is the thickness of oil film on the contact surfaces, conditioned by the oil viscosity, contact load and rubbing velocity.

By Block's criterion [8], the temperature of a contact is a sum temperature of the friction surface before the contact, t_{s}, and instant temperature at the contact (temperature raise), $t_{\max}$:

$$t_{\Sigma} = t_{\mathrm{s}} + t_{\max} \le t_{\mathrm{cr}} = \mathrm{const}, \tag{1}$$

here, t_{cr}—critical temperature of lost of greasy properties by oil.

The friction surface temperature t_{s} insignificantly differs from the reducing gear oil temperature t_{o}, and these values are assumed equal in calculations. The oil temperature in reducing gear depends on the transmission capacity, efficiency, operating mode, cooling area, material the body of the reducing gear is made of and ambient temperature.

The instant temperature is calculated by the condition of temperature distribution under thermal contact at local friction. According to [9], under stationary thermal contact, the temperature distribution at the contact and in nearby area is proportional to the Newtonian potential of simple layer having density equal to the product of rubbing velocities and contact stresses. Being proportional to the Newtonian potential, the temperature raise is related with the friction power:

$$P_{\mathrm{f}} = qfv_{\mathrm{rub}}, \tag{2}$$

where q—specific load in contact, N/m; f—friction coefficient; v_{rub}—rubbing velocity, m/s.

In a general case, the temperature raise is directly proportional to the friction power and inverse proportional to the heat conductivities λ of materials the gear teeth are made of, which characterize heat transmission from the materials to the oil:

$$P_{\mathrm{f}} = \frac{qfv_{\mathrm{rub}}}{\lambda_1 + \lambda_2}, \tag{3}$$

λ_1, λ_2—heat conductivities of materials in a pair of teeth, W(m·K).

By dividing both values by $t_{\max}$ from (1), we obtain a dimensionless criterion of temperature effect on jamming of contact surfaces [6]:

$$Cr_1 = \frac{qfv_{\mathrm{rub}}}{(t_{\mathrm{cr}} - t_{\mathrm{o}})(\lambda_1 + \lambda_2)}. \tag{4}$$

Beside the temperature, the oil film thickness is governed by kinematic and kinetostatic factors—load in contact and oil feed rate in contact zone.

According to the hydrodynamic lubrication theory, the oil film thickness at interface of two elastic cylinders is directly proportional to dynamic oil viscosity, oil layer flow velocity and inversely proportional to specific load [11]. In a spiroid gear, oil is dragged in the contact zone of of teeth at the maximum rubbing velocity equaling the sum of rotation speed of gear teeth. In this regard, the second criterion is derived for the kinematic parameters of the oil film:

$$Cr_2 = \frac{q}{v_{\Sigma}\eta}, \tag{5}$$

where v_{Σ}—total rolling speed, m/s; η—dynamic viscosity of fluid, Pa·s.

Based on the knowledge on kinematic viscosity and visco-thermal characteristics of oil, we rewrite (5) and use the relation of the kinematic μ_0 and dynamic viscosity of oil: $\eta = \rho\mu_0$, where ρ—fluid density:

$$Cr_2 = \frac{q}{v_{\Sigma}\rho\mu_0}. \tag{6}$$

On the assumption of the first theory of similarity, multiply the criteria (4) and (6). Since the degree of influence of each factor on the initiation of jamming is unknown, in accordance with the second similarity criterion, the criteria are raised to the power of a and b [6]:

$$Cr = \left(\frac{qfv_{\mathrm{rub}}}{(t_{\mathrm{cr}} - t_{\mathrm{o}})(\lambda_1 + \lambda_2)}\right)^a \left(\frac{q}{v_{\Sigma}\rho\mu_0}\right)^b < [Cr], \tag{7}$$

where a, b—coefficients of influence of factors on jamming; [Cr]—permissible value of the criterion.

The expression (7) is taken as a new criterion including two key sets of influences: thermal. i.e. influence of oil temperature on oil film thickness in contact, and hydrodynamic, i.e., influence of feed conditions and physical properties of oil. The principal advantage of the criterion is zero dimension, which allows its application in physical modeling by the method of similarity, and makes it possible to estimate and compare different lubricants and materials the teeth could be made of in terms of their jamming risk.

The criterion efficiency was checked on the physical model of spiroid gear. After analysis of the available friction assemblies used in jamming tests of spiroid toothing [12], a roller–wheel disc friction pair was chosen. This design ensures functional similarity and tribological similarity, i.e. linear contact of the elements. The similarity of tribological interactions also depends on the method of feed, type and brand of oil.

The roller–wheel disc design enables the spiroid gear-specific position of the contact zone relative to axes of the gear elements. The move of the toothing zone increases the angle between the directions of the rubbing velocity of the elements and their total rolling speed, which is favorable for an oil wedge to appear, i.e. oil is dragged into the contact zone. The lateral displacement of the roller axis relative to the wheel disc allows adjusting the angle between the rubbing velocity direction and the contact line, as well as the angle between the linear speeds of the elements in the pair, which ensures similarity of geometrical parameters, including curvature radius and contact line length. The similarity of physical parameters is achieved by means of selecting materials the elements of the friction pair are made of and choosing the method of their treatment.

The similarity of external mechanical actions is ensured by providing their adjustability in the range analogous to the gear being modeled. For the physical model testing of a spiroid gear, the Siberian State Transport University designed a laboratory bench based on the roller–wheel disc friction pair design (Fig. 1) [12, 13].

The elements in the roller–wheel disc friction unit *7* contact along the line equal to the width of the roller. The wheel disc has roller paths of the same width as the roller. These roller paths make it possible to avoid face friction at the contact when a steel roller operates for a long time and shapes a groove on a disc made of softer bronze. As a consequence, additional resistance to the roller rotation is generated by the walls of the groove.

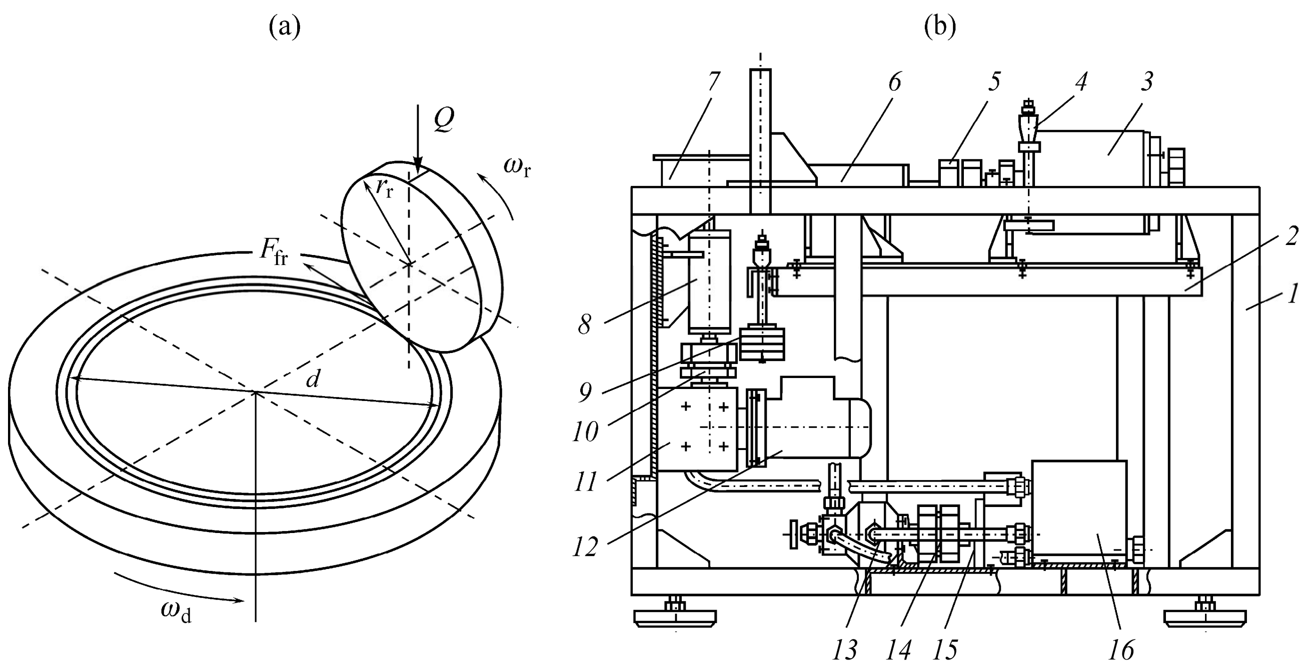

Fig. 1. (a) Friction unit and (b) general view of the roller–wheel disc test bench: *1*—frame; *2*—mobile frame; *3*—electric motor drive of the roller; *4*—measurement device; *5*—elastic pin coupling; *6*—roller drive mounting group; *7*—friction pair; *8*—disc drive mounting group; *9*—loading device; *10*—elastic pin coupling; *11*—reducing gear RS-31.5-49; *12*—electric motor drive of the wheel disc; *13*—gear-type pump; *14*— elastic pin coupling; *15*—electric motor of the pump; *16*—tank; ω_r, ω_d—angular speeds of the roller and wheel disc; F_{fr}—friction force.

The wheel disc and roller have independent drives. The wheel disc drive includes the electric motor *12*, pin coupling *10* and mounting group *8*. The disc velocity is adjusted by steps using spiroid reducing gear transmission. The drive of the roller consists of the cradle DC motor *3* and elastic pin coupling *5* to transfer the moment to the shaft with the single-sided support roller. The rotation speed of the roller is adjusted by the change in the voltage of the armature winding of the motor, which allows simulating operation of a worm with a motor with different number of ports and makes it possible to ensure the wanted velocities of rubbing and rolling in he contact zone.

The wheel disc drive is mounted on the bench frame *1*, and the roller drive is arranged on the mobile frame *2* stabilized by two hinges. The third support point of the frame is the roller that which, at the contact point, transfers the load from the mobile frame and the roller drive to the wheel disc. For the load adjustment at the contact, the loading device *9* is arranged on the mobile frame in the form of two arms with suspended weights to increase or decrease the load at the roller and wheel disc contact by varying downward force of the weights. To change the angle between the contact line and the rubbing velocity direction, the roller drive can be displaced along the perpendicular line to the drive axis.

Oil is fed to the contact zone between the roller and wheel disc by a pump unit including the gear-type pump *13* driven, through the coupling *14*, by the motor *15*, pressure and discharge pipes and the tank *16*. The bulk temperature of oil, which indirectly affects jamming, is adjusted using a fire-bar element installed in the tank and a system of automated oil heating to a preset temperature.

The measurement device *4* serves to determine the friction moment required to overcome the force of friction in the contact zone. Since the roller drive motor is cradle-set on the support, the value of the motor moment can be measured in terms of its deviation from the longitudinal axis. To this end, a special measurement arrow is arranged on the motor to show the motor deviation on a special calibrated dial. After the measurement system calibration, the friction moment in the contact zone of the roller and wheel disc is measured to an accuracy of 3% as the resistance to rotation of the motor shaft in the bearings is lower as compared to the resistance due to friction.

Under jamming, the friction coefficient jumps more than by 30% and the bench starts to vibrate. The friction coefficient is an indication of the jamming.

The tests were carried out with two combinations of the materials the roller and wheel disc were made of: steel–steel and steel–bronze. Regarding the steel–steel tests (wheel disc—steel with, HRC 22…24; roller—steel 40X, HRC 48…53), the results are presented by the curves of the friction coefficient in the contact zone, load and rubbing velocity at different oil temperatures in Fig. 2.

Figure 2 shows the points of the friction coefficients measured at different temperatures, loading and rubbing velocities. The approximating log curves $f = f(q)$ plotted based on these points for each value of the rubbing velocity at a certain accuracy coincide with the real curves exact functions of which are yet unknown.

Thus, in the steel–steel tests, jamming of the active surface was recorded at the temperatures of 60, 80 and 100 °C under loading higher than 250% of the rated load and the rubbing velocities more than 165% of the standard value. It is visible in the curves in Figs. 2b–2d as the jump of the friction coefficient and its deviation from the trend.

Figure 3 shows the pictures of the roller surface before and after the tests. The grooves and polished surface of the roller in Fig. 3b illustrate initiation of jamming.

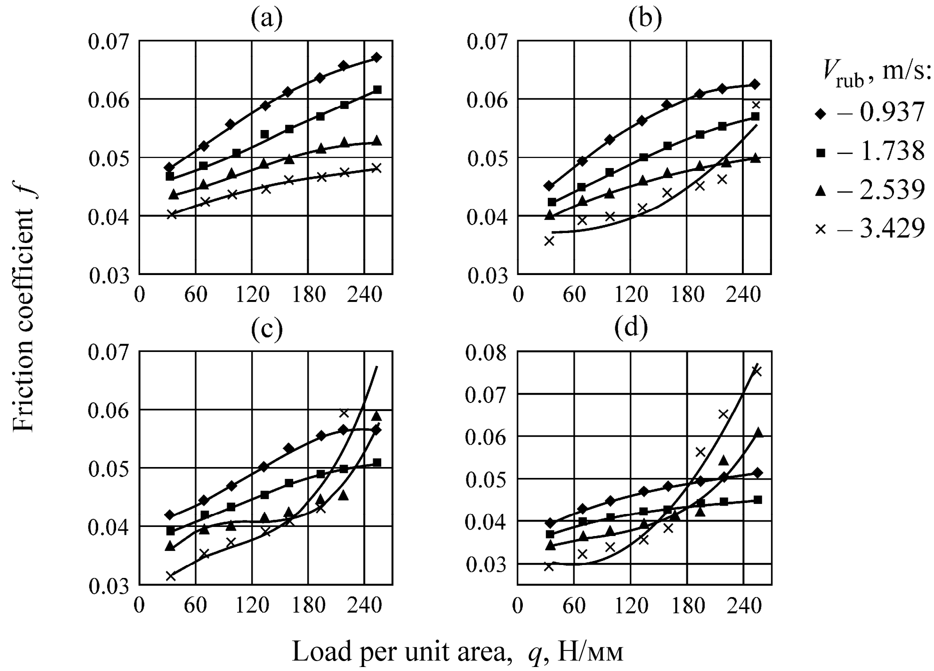

Fig. 2. Friction coefficient versus loading in the rubbing velocity range from 0.937 to 3.429 m/s at different oil temperatures, C: (a) 40; (b) 60; (c) 80; (d) 100.

After the analysis of the obtained curves, the critical value $Cr = 3$ was determined to indicate the onset of jamming. Then, by statistical processing of the test data, the coefficients a and b (7) were calculated and the criterion formula (7) took on the form [2, 3]:

$$Cr = \left(\frac{q f v_{\text{rub}}}{(t_{\text{cr}} - t_{\text{o}})(\lambda_1 + \lambda_2)} \right)^{1.42} \left(\frac{q}{v_{\Sigma} \rho \mu_0} \right)^{0.28} < 3. \quad (8)$$

Based on the proposed criterion, the spiroid gear design procedure with regard to jamming was developed. In case the calculated criterion exceeds a permissible value, the parameters of the gear geometry, kinematics and loading should be adjusted. Therefore, the criterion of jamming should be calculated at the early stage of spiroid gear design to synthesize ideal toothing while calculating force, efficiency and thermal conditions [13]. The flow chart of the jamming procedure is shown in Fig. 4.

The source information are the parameters of geometry and kinematics of spiroid gear: transmission number, center-to-center distance, rotation speed of worm and moment on output shaft. The procedure consists of 7 steps. First, geometrics of the gear elements is calculated. Then, kinematics—rubbing velocity and total rolling velocity—is determined. After that, materials for the elements of the gear are selected, toothing forces are estimated, and the load value is found. Later on, lubricating materials are chosen, and oil temperature is the contact zone is determined. Based on the brand and temperature of oil taken from the catalogs, the kinematic viscosity, density and critical inflammation temperature of the lubricant are set.

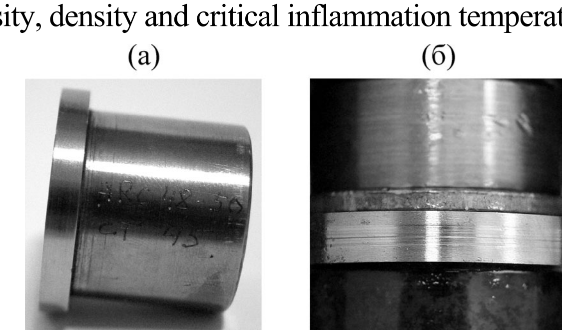

Fig. 3. Pictures of the roller surface (a) before and (b) after the tests.

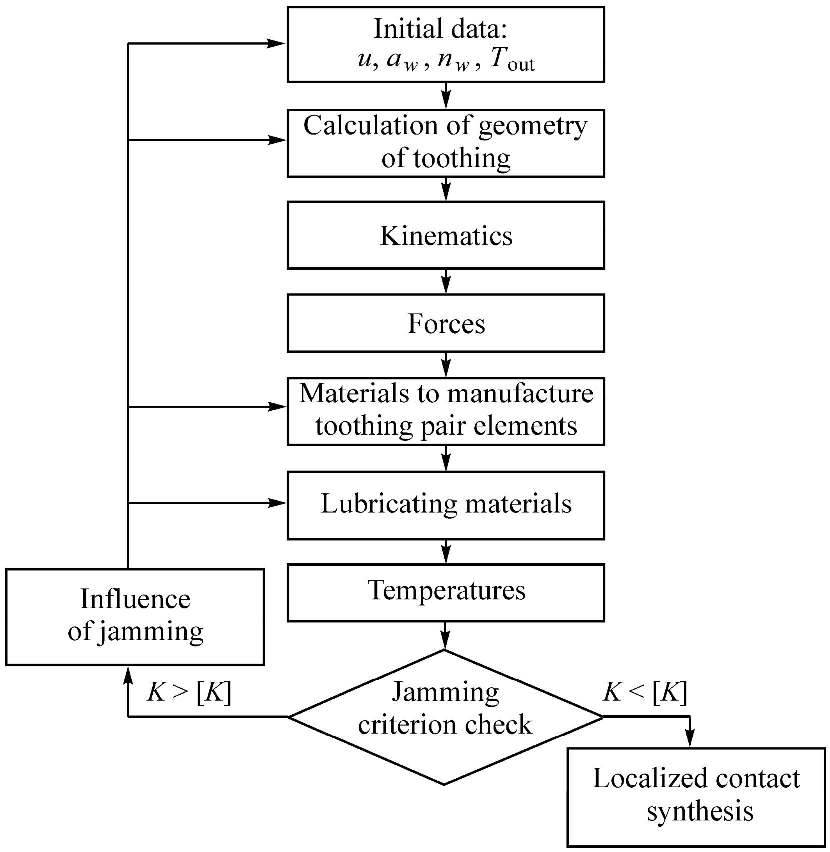

Fig. 4. Flow chart of spiroid gear design with regard to jamming.

The wanted friction coefficient is selected with respect to the calculated loads, rubbing velocities and oil temperatures from the reference data bases on the chosen materials to make the friction pair elements, based on the experimental research findings [2, 12].

Then, the critical jamming criterion is determined to indicate the rupture of oil film and the onset of the third stage of jamming. When the calculated criterion exceeds the permissible critical value $[Cr]$, adjustment of the gear parameters is required, or additional structural facilities are to be provided, e.g., cooling system of reducing gear.

The described procedure has been applied in designing reducing gears for a swing drive group of spreader cams, with the steel–steel combination of materials the elements in the pair were made of, and in improvement of travel mechanism of tower crane model KB-405.

CONCLUSIONS

The paper has proposed the dimensionless criterion of jamming in spiroid gears. The key advantage of the criterion is its applicability in physical modeling by the similarity method. The dimensionless criterion allows estimating and comparing risk of jamming for different lubricants and different combinations of materials the toothing pair elements are made of.

The physical modeling of spiroid gear jamming has been carried out. As a result, the data base on the friction coefficients has been complied for toothing of steel–steel pair. This data base can be used to improve accuracy of design of spiroid gears. The values of the friction coefficients and working conditions of the initiation of jamming in toothing pairs have been determined.

Using the proposed criterion, the procedure has been developed to design spiroid gears with regard to jamming in reducing gear group depending on its geometrics, kinematics and force characteristics, materials the toothing pair elements are made of and lubricants.

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