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FUNDAMENTALS FOR CONDITION MONITORING AND DIAGNOSTICS FOR DRIVING BUCKET WHEEL SYSTEM WITH OVERLOAD MECHANISM OF BUCKET WHEEL EXCAVATOR

The paper gives fundamentals for condition monitoring and diagnostics for the driving bucket wheel system of a bucket wheel excavator, which is designed for operation in increased digging resistance of strata. The fundamentals are given by development the characteristic frequencies as: meshing frequences, local fault freqencies, carrier frequencies for planetary gearboxes and so on.

1. INTRODUCTION

Condition monitoring and diagnostics of driving systems for bucket wheels of bucket wheel excavators given in figure 1 and 2 are described in papers (Bartelmus, Zimroz, 2007; Bartelmus, Zimroz, 2009a; Bartelmus, Zimroz, 2009b; Bartelmus et al., 2010; Bartelmus, Zimroz, 2008; Bartelmus, Zimroz, 2010a; Bartelmus, Zimroz, 2010b; Bartelmus, Zimroz, 2011). In a presented paper one can find fundamentals for the condition monitoring of the new design of a gearbox system which drives a bucked wheel of a bucked wheel excavator. The scheme of a new system is given in figure 3. The characteristic difference between design of systems given in figure 1 or 2 and 3 is that in system (fig. 3) is given an overload mechanism.

The term of a complex and compound gearbox is explained in (Bartelmus, Zimroz, 2010b). The system given in figure 1 consists of planetary stage with gears z_1, z_2, z_3 and three cylindrical stages, gears from z_4 to z_9.

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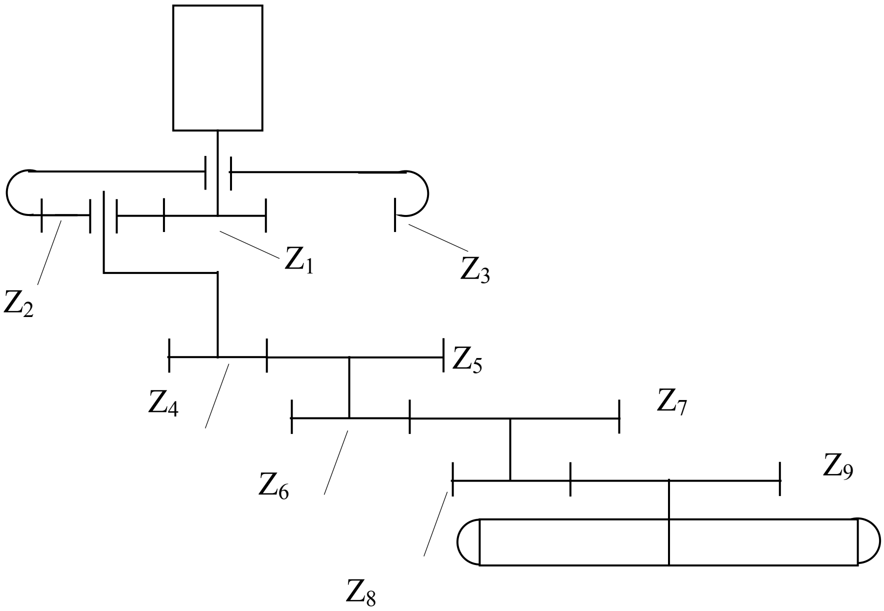

Fig. 1. Scheme of complex gear system

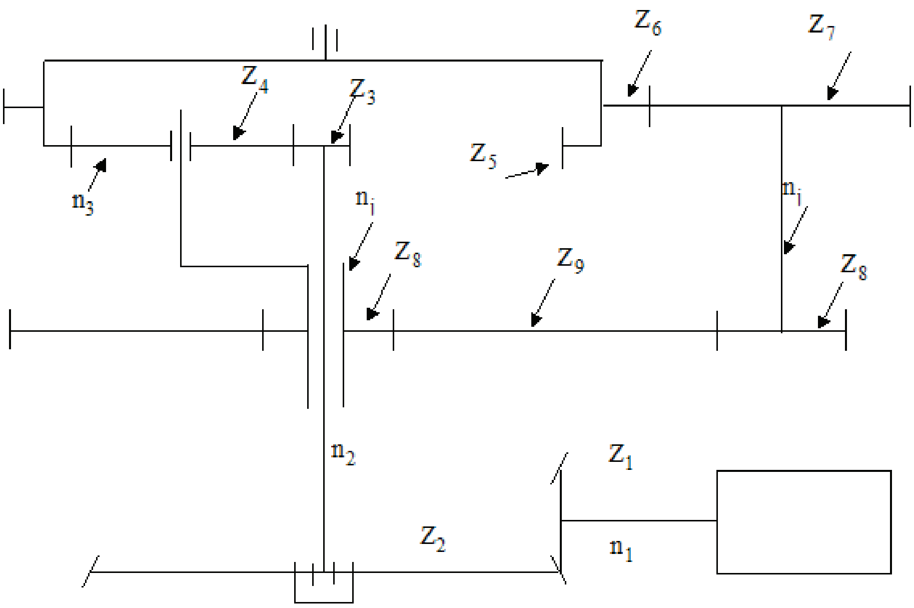

Fig. 2. Scheme complex gear system used for bucked wheel driving

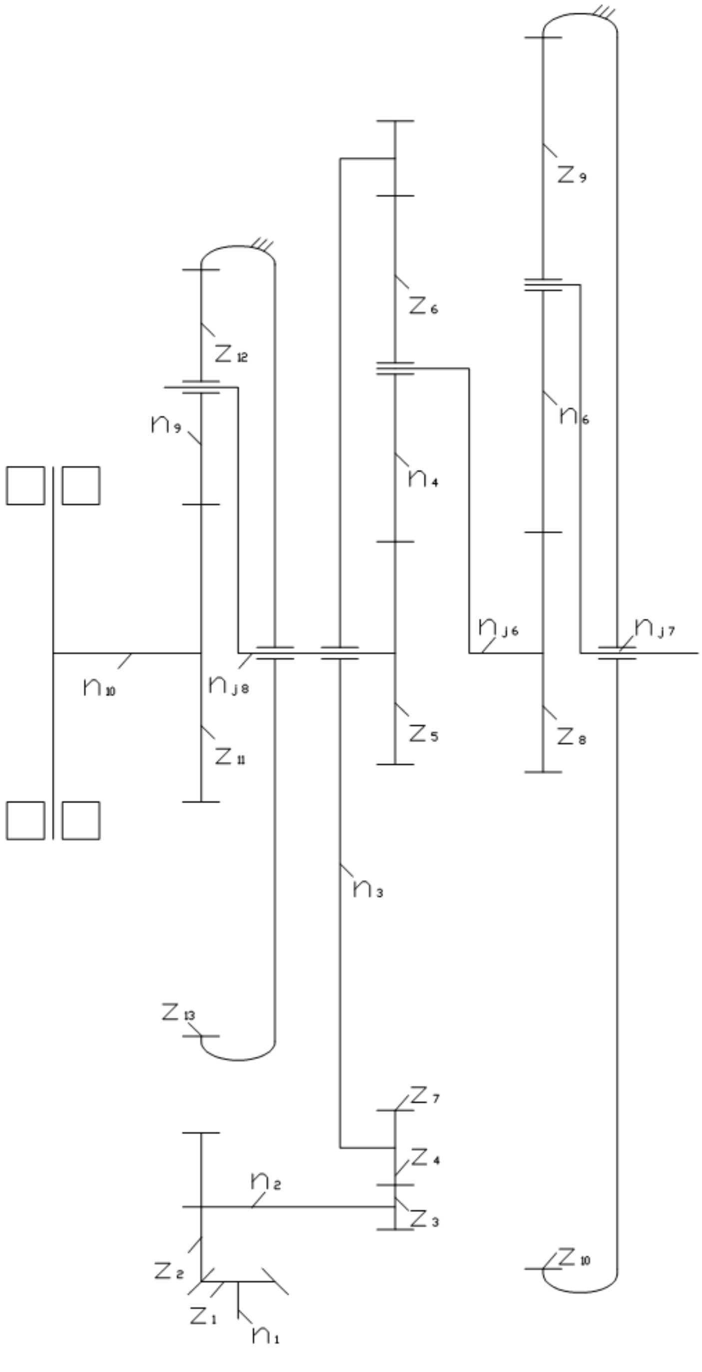

Fig. 3. Scheme of driving system for bucked wheel of bucked wheel excavator with overload mechanism

The system given in figure 2 transmits the power through the bevel stage and next by two different ways. The first way is through the planetary stage z_3, z_4, z_5 and a cylindrical stage z_8 and z_9. It should be noticed that the planetary stage is characterised by two degrees of rotation freedom, which gives the possibility to rotate a sun z_3 and ring z_5 simultaneously so the second way also goes though a planetary gearbox and two stages of cylindrical stages z_6 to z_9. It should be noticed that the pinion z_9 have the same rotation velocities n_j [RPM]. On this condition there is developed the ratio of the planetary stage in (Bartelmus, Zimroz 2010b). The possibility of condition evaluation for the planetary stage is given in (Bartelmus, Zimroz 2009b).

2. OBJECT DESCRIPTION

The considered object description is based on the paper (Wocka, 2003) and the scheme of the drive is give in figure 3.

In the described system there is incorporated an overload mechanism. It is presumed that in normal operation the power is transmitted through a bevel stage (gears z_1, z_2) a cylindrical stage (gears z_3, z_4) a planetary stage (gears z_5, z_6, z_7). The gear z_5 is fixed during normal operation of the drive. It is fixed by the overload mechanism. The last stage of the system is a planetary (gears z_8, z_9, z_{10}) stage with a fixed gear z_{10}. During the overloading the power is transmitted to the overload mechanism. The power is transmitted through a bevel stage (gears z_1, z_2), a cylindrical stage (gears z_3, z_4), a second cylinder stage with a third wheel z_6. It is presumed that the arm/carrier of a planetary gearbox (gears z_5, z_6, z_7) is now fixed because the bucked wheel is stopped by overloading,

3. DEVELOPMENT OF CHARACTERISTIC FREQUENCIES

The system presented in the figure 3 during normal operation works using a bevel stage gearbox with a ratio

$$u_b = \frac{z_2}{z_1}, \tag{1}$$

a cylindrical stage with ratio

$$u_{cy} = \frac{z_4}{z_3}, \tag{2}$$

a planetary gearbox with ratio

$$u_{p1} = 1 + \frac{z_5}{z_7}, \tag{3}$$

in this case the gear z_5 is stopped by the brake which is adjusted to the constant moment M_b [Nm]. The system also includes a planetary gearbox with a ratio

$$u_{p2} = 1 + \frac{z_{10}}{z_8}. \tag{4}$$

The planetary gearboxes are complex gearboxes for, which principles for finding their ratios are given in a paper (Bartelmus, Zimroz 2010b).

Knowing the above relation one may treat the system as a compound gearbox with a total ratio

$$u_t = u_b u_{cy} u_{p1} u_{p2} = \frac{z_2}{z_1}\frac{z_4}{z_3}\left(1 + \frac{z_5}{z_7}\right)\left(1 + \frac{z_{10}}{z_8}\right). \tag{5}$$

Under condition of overloading the bucked wheel is stopped and the power is transmitted to the brake through a bevel stage gearbox with a ratio as given by (1) and a cylindrical stage as given by (2).

In the case of overloading the and the bucket wheel is stoped and $n_{j5} = 0$ so the planetary gearbox is redused to the cylindrical stage with idle gear or third wheel z_6 and the ratio of the fomer planetary gear (gears z_5, z_6, z_7) is now

$$u_{cy2} = \frac{z_5}{z_7} \tag{6}$$

and transmited moment from the electric motor equels to M_b.

Further the power is transmited by a planetary gearbox with (gears z_{11}, z_{12}, z_{13}) which works as multiplicater with a ratio

$$u_{p3} = \frac{z_{11}}{z_{11} + z_{13}}. \tag{7}$$

And now one may treat the system as a compound gearbox with a total ratio of the gearbox from the motor to the brake is

$$u_t = u_b u_{cy} u_{cy2} u_{p3}. \tag{8}$$

The braking toque/moment is

$$M_b = u_t k_o M_m, \tag{9}$$

where M_m [Nm] – the nominal/rated moment of the electric motor,

k_o – coeficient of overloading $k_o \geq 1$.

For the condition monitoring of the system there is a need find meshing frequencies.

The meshing frequences are given in notation as is given in article (Bartelmus, Zimroz 2011) for example f_{12} means the frequency for gears z_1 and z_2 and is evaluated from the statment

$$f_{12} = \frac{n_1 z_1}{60}. \tag{10}$$

Where n_1 rotation valocity of an electric motor in RPM. Meshing frequency for gears z_3 and z_4 is

$$f_{34} = \frac{n_2 z_3}{60}, \tag{11}$$

where

$$n_2 = \frac{n_1 z_1}{z_2}. \tag{11a}$$

A meshing frequency for the planerary stage (gears z_5, z_6, z_7) according to the developments given in (Bartelmus, Zimroz 2010b) is in (12). The meshing frequency is according to an above notation, and should be also noticed that in that planetary gearbox the gear z_5 is fixed during normal operation.

$$f_{56} = f_{67} = \frac{n_3 z_5 z_7}{60(z_5 + z_7)}, \tag{12}$$

where

$$n_3 = \frac{n_1 z_1 z_3}{z_2 z_4}. \tag{13}$$

Meshing frequency for the planerary stage (gears z_8, z_9, z_{10}) according to the developments given in (Bartelmus, Zimroz, 2010b)

$$f_{89} = f_{910} = \frac{n_{j5} z_8 z_{10}}{60(z_8 + z_{10})}, \tag{14}$$

where

$$n_{j5} = \frac{n_1 z_1 z_3}{z_2 z_4} \frac{z_7}{(z_5 + z_7)}. \tag{15}$$

The output rotation velocity in RPM is given by the statment

$$n_{j7} = \frac{n_1 z_1 z_3}{z_2 z_4} \frac{z_7}{(z_5 + z_7)} \frac{z_8}{(z_8 + z_{10})}. \tag{16}$$

The output rotation can be also given by (17)

$$n_{j7} = \frac{n_1}{u_t}. \tag{17}$$

Rotation valocities for satalites z_6 and z_9 according to the depiction in figure 3 accoding to article (Bartelmus Zimroz, 2011) are

$$n_4 = \frac{n_3 z_7}{2 z_6}, \tag{18}$$

$$n_6 = \frac{n_{j5} z_8}{2 z_9}. \tag{19}$$

Next step for a gearbox fault identyfication is connected with local faults. Local fault freqencies are given equivelently for gears z_1, z_2, z_3 and z_4

$$f_{1l} = \frac{n_1}{60};\ f_{2l} = \frac{n_2}{60} = f_{3l};\ f_{4l} = \frac{n_3}{60}, \tag{20}$$

where f_{1l} means a frequency of a local fault for gear z_1 and so on for the other gears.

For the planetary stage (gear z_5 fixed) with gears z_5, z_6, z_7, according to (Bartelmus, Zimroz 2011)

$$f_{5l} = \frac{n_3 z_7 s}{60(z_5 + z_7)};\ f_{6l} = \frac{4 n_3 z_5 z_7}{60(z_7^2 - z_5^2)};\ f_{7l} = \frac{n_3 z_5 s}{60(z_5 + z_5)}, \tag{21}$$

where s is a number of satalites in a planetary stage.

For the planetary stage (gear z_{10} fixed) with gears z_8, z_9, z_{10}, according to (Bartelmus, Zimroz 2011)

$$f_{8l} = \frac{n_{j5} z_{10} s}{60(z_8 + z_{10})};\ f_{9l} = \frac{4 n_{j5} z_8 z_{10}}{60(z_{10}^2 - z_8^2)};\ f_{10l} = \frac{n_{j5} z_8 s}{60(z_8 + z_{10})}, \tag{22}$$

where s is a number of satalites in a planetary stage.

Carrier frequencies equivelently for the planetary stage with gears z_5, z_6, z_7, and z_8, z_9, z_{10}

$$f_j = \frac{n_3 z_7}{60(z_5 + z_7)};\ f_j = \frac{n_{j5} z_8}{60(z_8 + z_{10})}. \tag{23}$$

Passing frequencies

$$f_{p1} = \frac{n_{j5}}{60} s_1;\ f_{p2} = \frac{n_{j7}}{60} s_2. \tag{24}$$

Frequency of meeting the same teeth.

Folowing the discusion given in (Bartelmus, Zimroz 2011) and (Bartelmus 2006), here it is repeated. One may ask a question: after how many rotations do the same teeth meet producing a similar excitation? If this number is denoted by N and multiplied by number of teeth z_1, then it is the number of excitations after which the excitation cycle will be repeated. Number N corresponds to the number of revolutions of the pinion after which the same teeth will meet again. It is calculated from ratio z_1/z_2, e.g. 38/50, and after the elimination of common divisors a ratio is 19/25, where $N = 25$, is obtained. Thus the same teeth meet after 25 revolutions of the pinion. The duration of one excitation is

$$\tau_1 = \frac{1}{f_z} = \frac{60}{n_1 z_1}, \text{ [s]} \tag{25}$$

a meshing generates a sequence of recurrent excitations with period

$$T_p = \tau_1 N z_1 = \frac{60N}{n_1}, \text{ [s]} \tag{26}$$

a sequence of excitations forms a function whose repetition rate is

$$f_p = \frac{1}{T_p} = \frac{n_1}{60N}, \text{ [Hz]} \tag{27}$$

Following the statements (25) to (27) one can evaluate the recurrent frequencies for the bevel stage (gears z_1, z_2) and cylindrical stage (gears z_3, z_4).

More attention should be given for planetary gear stages. Let start with a planetary stage with gears z_5, z_6, z_7, the gear z_6 is fixed during the normal operation.

Following the statement (25) the duration of one excitation should be evaluated which in the case of the considered planetary stage with fixed z_5 is given by the statement

$$\tau_1 = \frac{1}{f_{56}} = \frac{60(z_5 + z_7)}{n_3 z_5 z_7}. \tag{28}$$

From (26) and (27)

$$f_{p1} = \frac{n_3 z_7}{60(z_5 + z_7) N_{p1}}. \tag{29}$$

In the statement (29) is given the recurrent frequency for the gears z_7, z_6.

Where N_{p1} is evaluated from ratio z_7/z_6 after elimination of common divisors the ratio goes to the form

$$\frac{z_7}{z_6} = \frac{mM_{p1}}{mN_{p1}} = \frac{M_{p1}}{N_{p1}}, \tag{30}$$

where m is a common divisor.

Following the presented above procedures the recurrent frequency for gears z_6, z_5 is

$$f_{p2} = \frac{n_3 z_5 z_7}{60(z_5 + z_7) z_6 N_{p2}}, \tag{31}$$

where

$$\frac{z_6}{z_5} = \frac{mM_{p2}}{mN_{p2}} = \frac{M_{p2}}{N_{p2}}. \tag{32}$$

The recurrent frequency for gears z_8, z_9 for the planetary stage with gears z_8, z_9, z_{10} is

$$f_{p1} = \frac{n_{j5} z_{10}}{60(z_8 + z_{10}) N_{p1}}, \tag{32}$$

where

$$\frac{z_8}{z_9} = \frac{mM_{p1}}{mN_{p1}} = \frac{M_{p1}}{N_{p1}}. \tag{33}$$

The recurrent frequency for gears z_8, z_9 for the planetary stage with gears z_8, z_9, z_{10} is

$$f_{p2} = \frac{n_{j5} z_8 z_{10}}{60(z_8 + z_{10}) z_9 N_{p2}} \tag{34}$$

where

$$\frac{z_9}{z_{10}} = \frac{mM_{p2}}{mN_{p2}} = \frac{M_{p2}}{N_{p2}}. \tag{35}$$

As it was mansioned in the case of overloading the bucket wheel is stoped and $n_{j5} = 0$ so the planetary gearbox is redused to the cylindrical stage with idle gear or third wheel z_6 and the ratio of the fomer planetary gear (gears z_5, z_6, z_7) is now is given by (6) and transmited moment from the electric motor equels to M_b.

Further the power is transmited by a planetary gearbox with (gears z_{11}, z_{12}, z_{13}) which works as multiplicater with a ratio (7).

The meshing frequencies for bevel and cylindrical gear as is given in (10) and (11). But further consideration ought to based on that $n_{j5} = 0$. The bucked wheel is overloaded so the meshing frequency for the cylindrical stage with the third wheel z_6 is

$$f_{76} = f_{65} = \frac{n_3 z_7}{60}. \tag{36}$$

Meshing frequency for a planetary stage with (gears z_{11}, z_{12}, z_{13}) is

$$f_{1112} = f_{1213} = \frac{n_{10} z_{11} z_{13}}{60(z_{11} + z_{13})} . \tag{37}$$

Frequency of local faults can be evaluated from the following statements

$$f_{7l} = \frac{n_3}{60};\; f_{6l} = \frac{2n_4}{60};\; f_{5l} = \frac{n_{j8}}{60}, \tag{38}$$

where

$$n_4 = \frac{n_3 z_7}{z_6} . \tag{39}$$

The local faults for a planetary stage with (gears z_{11}, z_{12}, z_{13}) and gear z_{13} fixed) according to (Bartelmus, Zimroz 2010b)

$$f_{11l} = \frac{n_{10} z_{13} s}{60(z_{10} + z_{13})};\; f_{12l} = \frac{4 n_{10} z_{11} z_{13}}{60(z_{13}^2 - z_{11}^2)};\; f_{10l} = \frac{n_{10} z_{11} s}{60(z_{11} + z_{13})}, \tag{40}$$

where s is a number of satalites in a planetary stage.

The carrier frequency is

$$f_j = \frac{n_{10} z_{11}}{60(z_{11} + z_{13})} = \frac{n_{j6}}{60} . \tag{41}$$

The passing frequency

$$f_{p1} = \frac{n_{j6}}{60} s , \tag{42}$$

where s – is a number of satalites in a planetary stage.

The recurrent frequency for gears z_{11}, z_{12} for the planetary stage with gears z_{11}, z_{12}, z_{13} is

$$f_{p1} = \frac{n_{10} z_{13}}{60(z_{11} + z_{13}) N_{p1}} , \tag{43}$$

where

$$\frac{z_{11}}{z_{12}} = \frac{m M_{p1}}{m N_{p1}} = \frac{M_{p1}}{N_{p1}} . \tag{44}$$

The recurrent frequency for gears z_{12}, z_{13} for the planetary stage with gears z_{11}, z_{12}, z_{13} is

$$f_{p2} = \frac{n_{10} z_{11} z_{13}}{60(z_{11} + z_{13}) z_{12} N_{p2}} , \tag{45}$$

where

$$\frac{z_{12}}{z_{13}} = \frac{mM_{p2}}{mN_{p2}} = \frac{M_{p2}}{N_{p2}}. \tag{46}$$

4. FINAL CONSIDERATION

Gearbox is a system which may appear as a one stage gearbox or as a compound or complex gearbox. Most research on the gearbox damage process is done on the one stage gearbox. One stage gearbox system consists of gears, bearings and a case. Considering the damage process of the gearbox all the elements are treated separately by researchers. For example one may investigate the possibility of monitoring bearing faults making artificial fault on one element of the bearing, an inner, outer ring or a ball. One may investigate on condition monitoring of a making artificial breakage of a gear tooth. It as a wrong way of investigating on degradation process and possible condition monitoring or investigation and assessment of a gear fault. It is basic methodology error, because this investigation do not have too much in common with real degradation process. First of all one ought to treat the gearbox stage as a unity. For this unity it is a need to develop the measure of its condition. The measure should be made under deferent values of external loads. The measure of one stage gearbox condition appeared to be a linear function of applied external load (Bartelmus, Zimroz 2009b). This measure should evaluate after gearbox run in. In the case of the compound or complex gearbox the measure of condition should evaluate for each stage of the gearbox treating the stage as a unity. That means separately for a cylindrical stage, bevel stage or worm stage if one is considering the compound gearbox. In the case of complex gearboxes which incorporates planetary gearboxes one should give more attention to division the system into subsystems and reduce the complex system into the compound system. An example of a complex gearbox system is given in figures 1 to 3. For each stage or subsystem there is a need to evaluate the measure of its condition. In the process of a gearbox condition change.

5. CONCLUSIONS

In the paper is given consideration on developing the characteristic frequencies which can be used for condition monitoring of the new design of the drive of a bucked wheel for a bucked wheel excavator with overloading mechanism.

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PODSTAWY MONITOROWANIA I DIAGNOSTYKI UKŁADU NAPĘDOWEGO Z MECHANIZMEM PRZECIĄŻENIOWYM KOŁA CZERPAKOWEGO KOPARKI KOŁOWEJ

Praca przedstawia podstawy monitorowania I diagnostyki stanu układu napędowego koła czerpakowego, który został skonstruowany do eksploatacji złóż o zwiększonych oporach urabiania. Podstawy przedstawiają częstotliwości charakterystyczne zazębienia, częstotliwości uszkodzeń lokalnych, częstotliwości obrotów jarzma itp.