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VIBRATION SPECTRA CHARACTERISTIC FREQUENCIES FOR CONDITION MONITORING OF MINING MACHINERY COMPOUND AND COMPLEX GEARBOXES

The paper gives general procedures for development characteristics frequencies in a simple and elaborate gearbox systems. The gearboxes systems may consists of compound and complex gearboxes. The gearbox classification for compound and complex gearboxes is presented. The complex systems consists of planetary gearboxes. Three different planetary gearboxes are considered. These three types of planetary gearboxes are used in driving systems for mining machinery like bucket wheel excavators, shearers. The introduction to frequency characteristic development is presented. The characteristic frequencies are: sequence of recurrent excitations for short recurrent frequencies, meshing frequencies, shaft frequencies, local fault frequencies.

1. INTRODUCTION

There is a need to classify the gearbox, which are used for mining machinery. They may be classified as compound and complex gearboxes (Banach 1956). In figure 1 there is given a scheme for a compound gearbox, which consists of a bevel and cylindrical gears. If one defines the ratio for bevel gearbox as u_1 and cylindrical gearbox as u_2 the total ratio is given by multiplication:

$$u_T = u_1 u_2 \,. \tag{1}$$

The basic characteristic of the compound gearbox is that the total ratio of the compound gearbox can be given as multiplication of ratios of its stages as is given in the statement (1).

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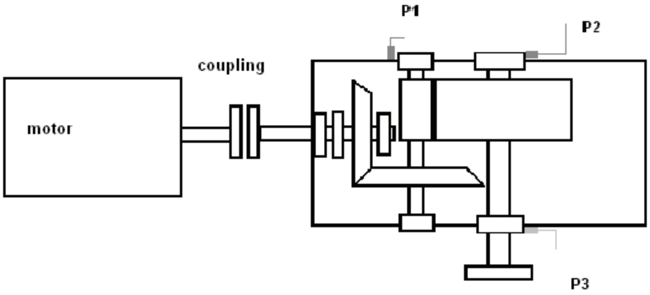

Fig. 1. Scheme of compound gearbox

The characteristic frequencies will be defined for shaft rotation frequencies, gear meshing frequencies, gear local fault frequencies.

The gear shaft rotation/revolution frequencies (Bartelmus 2006) for one stage cylindrical or bevel stage is calculated from the formula:

$$f_{01} = \frac{n_1}{60}, \text{ Hz} \tag{2}$$

$$f_{02} = \frac{n_2}{60} = \frac{n_1}{60u}, \text{ Hz} \tag{3}$$

where: f_{01}, f_{02} – respectively the driving and driven shaft frequency, Hz;
n_1, n_2 – respectively the driving and driven shaft rotational speed, rev/min,
$u = n_1/n_2$ – a gear ratio.

The meshing frequency is calculated from the formula:

$$f_z = \frac{nz}{60} = \frac{n_1 z_1}{60} = \frac{n_2 z_2}{60} = f_{o1} z_1 = f_{o2} z_2 \text{, Hz} \tag{4}$$

where: f_z – the meshing frequency, Hz,
z – the number of teeth,
z_1, z_2 – respectively the driving and driven wheel tooth number.

Components f_{01}, f_{02} occur also as modulation components $f_z + f_0$, $f_z - f_0$, $f_z + 2 f_0$, $f_z - 2 f_0$, etc. in the signal spectrum. A narrowband spectral analysis (zoom) reveals also other side components (in the gearing spectrum), (Bartelmus 2006); their origin can be traced as follows: each entrance into tooth contact results in an excitation of the system. 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 the number of excitations after which the excitation cycle will be repeated is defined. 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

$$\tau_1 = \frac{1}{f_z} = \frac{60}{n_1 z_1}, \text{ s.} \tag{5}$$

Meshing generates a sequence of recurrent excitations with period

$$T_p = \tau_1 N z_1 = \frac{60N}{n_1}, \text{ s.} \tag{6}$$

A sequence of excitations forms a function whose repetition rate is

$$f_p = \frac{1}{T_p} = \frac{n_1}{60N}, \text{ Hz.} \tag{7}$$

For short this repetition rate is called a recurrent frequency.

If a signal spectrum is found, then components if_z (where i – an integer larger or equal to 1) together with a complex structure of modulation components dependent on the distribution of errors/imperfections on the particular teeth will be obtained. An increase in the no uniformity of error distribution in meshing is a measure of unequal wear of the gear teeth. One may also count the frequencies associated with damage to the race, to a rolling element of a bearing. To determine the condition of gear and rolling element bearings (kinematic pairs) it is necessary to know the component frequencies and their intensity. On the base of this above consideration one can estimate the characteristic frequencies for any compound gearbox, which consists of bevel and cylindrical gear stages.

2. PLANETARY GEARBOX CHARACTERISTIC FREQUENCIES

If the system consists, beside of the bevel and cylindrical stages, also of planetary gearboxes the planetary gearboxes should be treated as complex gearboxes. The way of treating planetary gearboxes, three kinds of planetary gearboxes are given in figures 2 to 4 and now considered.

Figure 2a gives a scheme of planetary gears and distribution of peripheral and angular velocities. It is a specific planetary gearbox for which the gear wheel with a radius r_3 is standstill. In figure 2 there are marked radiuses of the gear wheels marked r_1 to r_3. The figure 2b gives the scheme of absolute (ω_1, ω_2, ω_a) and relative (ω_{1a}, ω_{2a}) angular rotation vectors. On the base of figure 2b one may write statements that

$$\omega_{1a} = \omega_1 - \omega_a, \text{ rad/s} \tag{8}$$

where:

ω_1 – a absolute angular velocity of a planetary gearbox sun,

ω_a – absolute angular velocity of a planetary gearbox arm/carrier,

ω_{1a} – relative angular velocity of a planetary gearbox sun.

$$\omega_{2a} = \omega_2 + \omega_a \ , \tag{9}$$

where:

ω_2 – absolute angular velocity of planetary gearbox satellite,

ω_{2a} – relative angular velocity of planetary gearbox satellite.

The ratio of a planetary gearbox given in figure 2a is defined as

$$u_p = \frac{\omega_1}{\omega_a} . \tag{10}$$

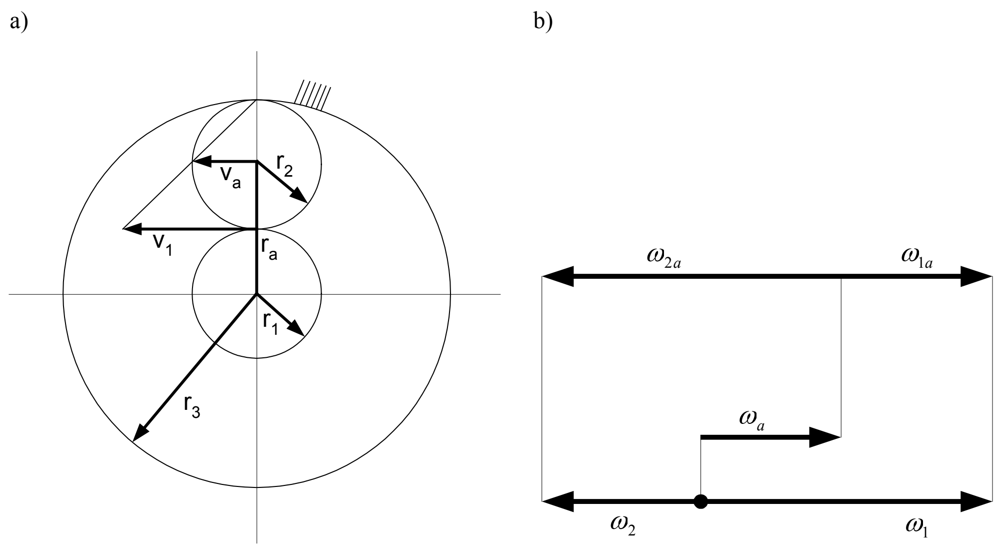

Fig. 2. Scheme of planetary gears with standstill rim a)
and distribution of peripheral and angular velocities b)

On the base of figure 2a and after some developments one can get the statement of the planetary gearbox ratio as fallow

$$u_p = 1 + \frac{r_3}{r_1} = 1 + \frac{z_3}{z_1} , \tag{11}$$

where z_1 and z_3 are numbers of teeth in gears, sun and rim.

A meshing characteristic frequency is

$$f_{12} = f_{23} = \frac{n_1 z_1 z_3}{60(z_1 + z_3)}. \tag{12}$$

The planet angular frequency is

$$\omega_2 = \frac{v_1}{2r_2} = \frac{\omega_1 r_1}{2r_2} = \frac{\omega_1 z_1}{2z_2}, \text{ rad/s.} \tag{13}$$

The arm/carrier rotation speed RPM

$$n_a = \frac{n_1}{u_p} = \frac{n_1 z_1}{z_1 + z_3}, \text{ RPM} \tag{14}$$

and its frequency

$$f_a = \frac{n_1 z_1}{60(z_1 + z_3)}, \text{ Hz} \tag{15}$$

The local fault frequency for the sun gear

$$f_{1l} = \frac{n_{1a} s}{60} = \frac{n_1 - n_a}{60} s, \text{ Hz} \tag{16}$$

where s – number of satellites or

$$f_{1l} = \frac{n_1 z_3 s}{60(z_1 + z_3)} \tag{17}$$

for the planet/satellite gear

$$f_{2l} = 2\frac{n_{2a}}{60} = 2\frac{n_2 + n_a}{60} = \frac{4 n_1 z_1 z_3}{60(z_3^2 - z_1^2)}. \tag{18}$$

The local fault frequency for the rim

$$f_{3l} = \frac{n_{3j}}{60} s = \frac{n_a s}{60} = \frac{n_1 z_1 s}{60(z_1 + z_3)}. \tag{19}$$

Beside the mentioned characteristic frequencies also occur modulation frequency, which is connected with the satellite passing through the constant place of the vibration signal receiving point $f = f_a s$.

In the planetary gearbox figure 2 like in cylindrical gearboxes there are also frequencies, which are connected with relation given in statement (7). The question is how to use this statement in the case of the relative gear motion. One can eliminate the arm/carrier motion in condition of motion observation from the carrier. The issue is treated at the end of the paper (62) to (79).

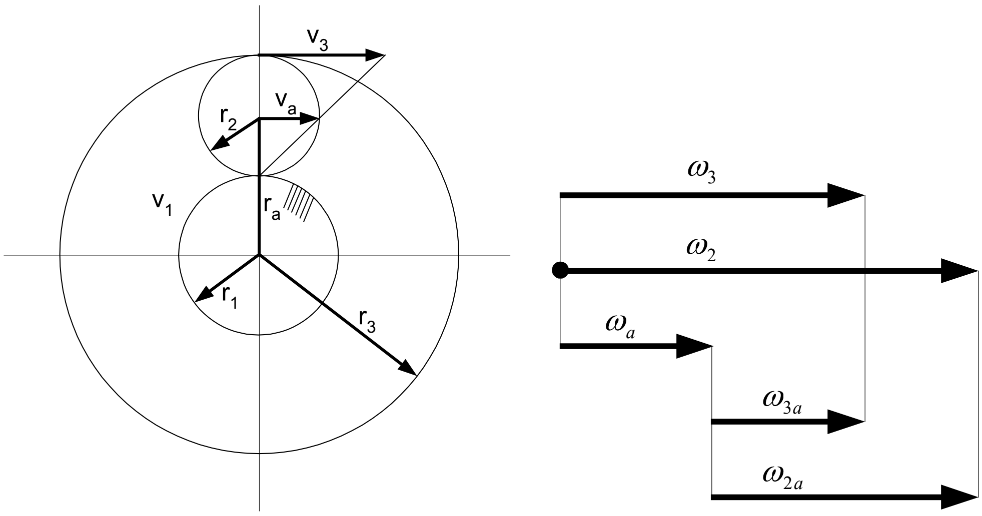

Fig. 3. Scheme of planetary gears with standstill sun
and distribution of peripheral and angular velocities

Planetary gearbox ratio for the system with the standstill sun given in figure 3 is

$$u_p = \frac{n_3}{n_a} = \frac{\omega_3}{\omega_a}, \tag{20}$$

where: ω_3 – rim angular velocity, rad/s,

ω_a – arm angular velocity, rad/s.

The rim rotating frequency is equal to input frequency.

$$f_3 = \frac{n_3}{60}[Hz]. \tag{21}$$

The arm rotating frequency is

$$f_a = \frac{n_3 z_3}{60(z_1 + z_3)}. \tag{22}$$

The planet gear rotating frequency is

$$f_2 = \frac{n_3 z_3}{120 z_2}. \tag{23}$$

The meshing frequency equals to

$$f_{23} = \frac{n_3 - n_a}{60} z_3 = \frac{n_2 - n_a}{60} z_2 = \frac{n_3 z_1 z_3}{60(z_1 + z_3)} . \tag{24}$$

The local fault frequency for the sun gear

$$f_{1l} = \frac{n_a}{60} s \text{ , Hz} \tag{25}$$

where s – number of satellites or

$$f_{1l} = \frac{n_3 z_3 s}{60(z_1 + z_3)} , \tag{26}$$

for the planet/satellite gear

$$f_{2l} = \frac{2n_{2a}}{60} = \frac{2(n_2 - n_a)}{60} = \frac{4n_3 z_1 z_3}{60(z_3^2 - z_1^2)} . \tag{27}$$

The local fault frequency for the rim

$$f_{3l} = \frac{n_{3a} s}{60} = \frac{(n_3 - n_a) s}{60} = \frac{n_3 z_1 s}{60(z_1 + z_3)} . \tag{28}$$

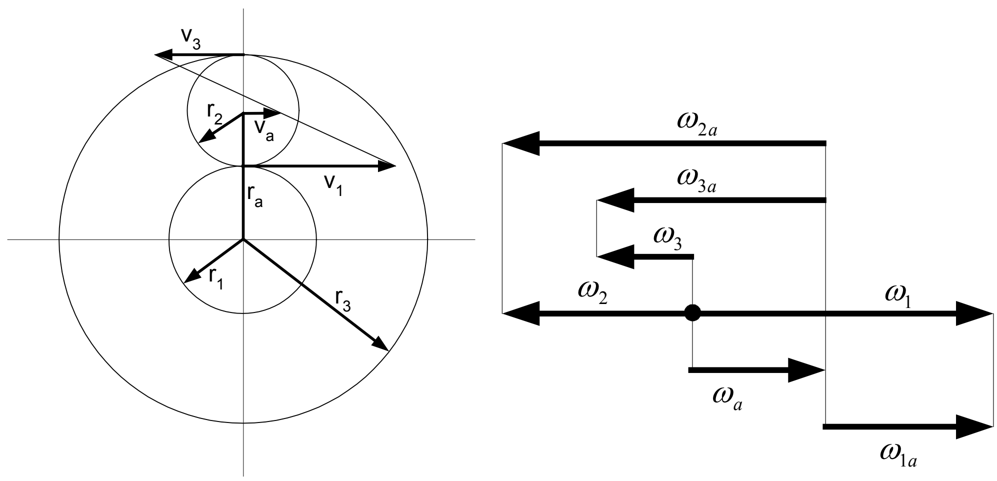

Fig. 4. Scheme of planetary gears with rotating sun and rotating rim, distribution of peripheral and angular velocities

In the planetary gearbox (fig. 3) like in cylindrical gearboxes there are also frequencies, which are connected with relation given in statement (7). The question is how to use this statement in the case of the relative gear motion. One can eliminate the arm/carrier motion in condition of motion observation from the carrier. The issue is treated at the end of the paper (70) to (77).

The ratio of the planetary gearbox of the type given in figure 4 can be given by the statement

$$u_p = \frac{1}{\left(1 - \frac{v_3}{v_1}\right)}\left(1 + \frac{r_3}{r_1}\right). \tag{29}$$

Further consideration on the planetary gearbox given in figure 4 is presented later.

3. CHARACTERISTIC FREQUENCIES FOR GEARBOX SYSTEMS

The system given in figure 5 can be treated as compound gearbox and consists of a planetary gearbox and three stages cylindrical gearbox. As one can see from upper consideration the planetary gearbox is a complex gearbox but after consideration given above can be treated as unit and the total ratio for the system is

$$u_t = u_p u_1 u_2 u_3 , \tag{30}$$

where:

u_t – compound total gearbox ratio,

u_p – planetary gearbox ratio,

u_1, u_2, u_3 – cylindrical gear ratios.

The system given in figure 5 has included the planetary gearbox with the standstill rim. The planetary gearbox given in figure 5 is equivalent to the planetary gearbox presented in figure 2 so the meshing frequency can be evaluated from the statement

$$f_{12} = f_{23} = \frac{n_1 z_1 z_3}{60(z_1 + z_3)} = \frac{950 \cdot 39 \cdot 93}{60(39 + 93)} = 435.067\text{Hz} \tag{31}$$

where n_1 – input rotation velocity RPM

The arm frequency is

$$f_a = \frac{n_1 z_1}{60(z_1 + z_3)} = \frac{950 \cdot 39}{60(39 + 93)} = 4.67 \text{ , Hz.} \tag{34}$$

The rotation frequency of second gear z_2 is

$$f_2 = \frac{n_1 z_1}{120 z_2} = \frac{950 \cdot 39}{120 \cdot 27} = 11.43 \text{, Hz.} \tag{35}$$

Meshing frequencies for three stage cylindrical gearbox are as fallow

$$f_{45} = f_a z_4 = 4.67 \cdot 34 = 158.78 \text{, Hz} \tag{36}$$

$$f_{67} = f_a \frac{z_4}{z_5} z_6 = 4.67 \cdot \frac{34}{117} \cdot 42 = 57 \text{, Hz} \tag{37}$$

$$f_{89} = f_a \frac{z_4 z_6}{z_5 z_7} z_8 = 4.67 \frac{34 \cdot 42}{117 \cdot 145} \cdot 35 = 13.75 \text{, Hz} \tag{38}$$

The condition monitoring and diagnostics for the system given in figure 5 is given in papers (Bartelmus 2007) and (Bartelmus, Zimroz 2009a).

The total ratio of the complex gearbox like scheme of a complex gearbox given in figure 6 is not a given by direct multiplication of the ratio of the given stages. The presented planetary gearboxes given in figure 6b is an other type of a planetary gearboxes, different from ones given before in figure 5.

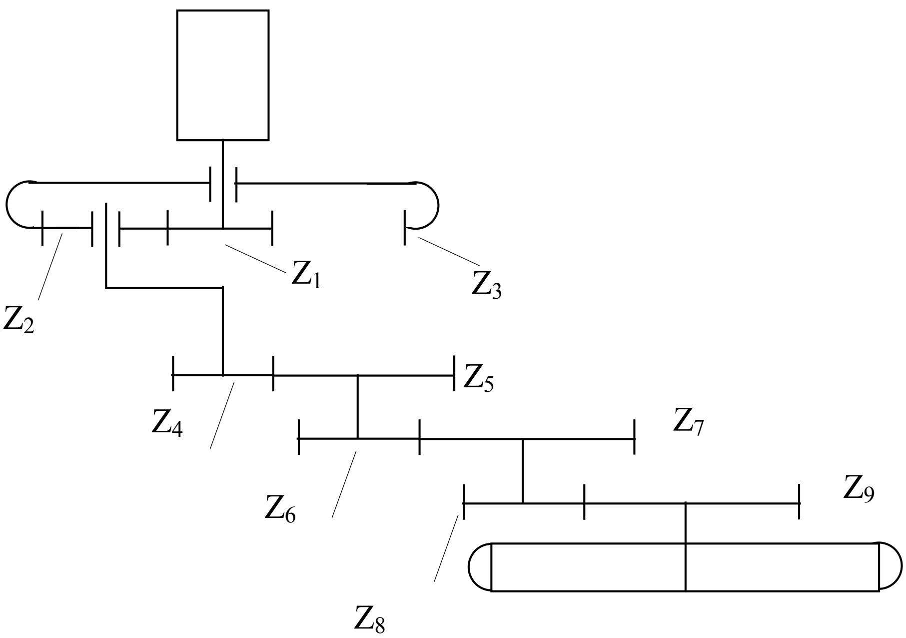

Fig. 5. Scheme of complex gear system

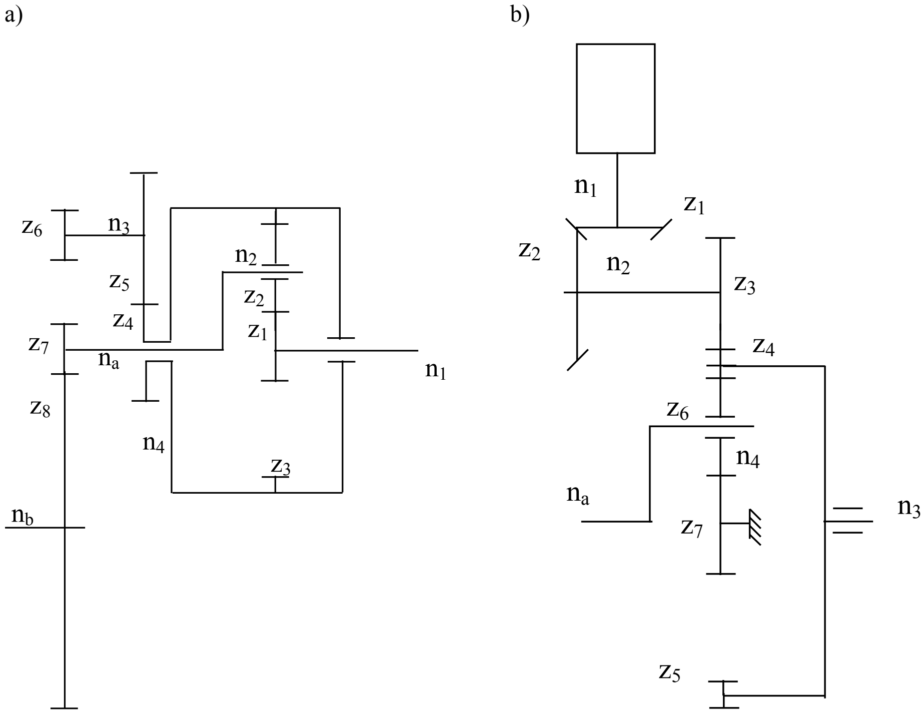

Fig. 6. Gearbox Driving system for bucket wheel with two subsystems, first subsystem input rotation velocity n_1 output rotation velocity n_a, second subsystem input rotation velocity $n_1 = n_a$ output rotation velocity n_b and $n_3 = n_a$

The planetary gearbox presented in figure 6b is of the type given in figure 3 but the planetary gearbox presented in figure 6a is of the type given in figure 4. For finding characteristic frequencies the whole system figure 6 is divided into two subsystems a) and b).

The subsystem b) may be treated as a compound gearbox system which consists of the bevel gears (first stage) z_1 and z_2 , the cylindrical gears (second stage) z_3 and z_4 and the planetary gear of the type figure 3 (third stage) z_5, z_6, z_7.

Taking into consideration the whole subsystem b) the ratio of the system is

$$u_{s1} = \frac{z_2 z_4}{z_1 z_3}\left(1 + \frac{z_7}{z_5}\right) = \frac{27 \cdot 76}{17 \cdot 23}\left(1 + \frac{26}{94}\right) = 6{,}7 \tag{39}$$

As it is given figure 6 an output rotation velocity n_a of the subsystem b) is the input rotation velocity n_1 for the subsystem a).

The subsystem a) is a complex system which consists of planetary gearbox given in figure 4.

The ratio of this planetary gearbox where a sun and rim are rotating is given in the statement (29).

From figure 7 one can write

$$\frac{v_3}{v_4} = \frac{r_3}{r_4}. \tag{40}$$

Further can be written

$$v_3 = v_4 \frac{r_3}{r_4} = \omega_a r_5 \frac{r_3}{r_4}. \tag{41}$$

So

$$\frac{v_3}{v_1} = \frac{\omega_a r_5 r_3}{\omega_1 r_1 r_4} = \frac{1}{u_{s2}} \cdot \frac{r_3 r_5}{r_1 r_4}. \tag{42}$$

After substituting (36) into (29) and some developments the ratio of the sub-system is

$$u_{s2} = 1 + \frac{z_3}{z_1} + \frac{z_3 z_5}{z_1 z_4}. \tag{43}$$

For the considered case

$$u_{s2} = 1 + \frac{82}{23} + \frac{82 \cdot 41}{23 \cdot 32} \cong 9.13. \tag{45}$$

The whole gearbox ratio for the bucket wheel drive given in figure 6 is

$$u = u_{s1} u_{s2} \frac{z_8}{z_7} = 6.7 \cdot 9.13 \cdot \frac{108}{20} = 321.1. \tag{48}$$

From the ratio the bucket wheel rotation velocity n_b can be counted

$$n_b = \frac{n_i}{u} = \frac{958}{321.1} = 2.98 \text{, RPM.} \tag{49}$$

For the meshing frequency evaluation for the planetary gearbox in the first sub-system one should take into consideration, according scheme figure 6b) reduction of a rotation velocity from n_1 to n_3,

$$n_3 = \frac{n_1 z_2 z_4}{z_1 z_3} = \frac{958 \cdot 17 \cdot 23}{27 \cdot 76} = 182.54 \text{, RPM} \tag{50}$$

and a meshing frequency is

$$f_{56} = f_{67} = \frac{n_3 z_5 z_7}{60(z_5 + z_7)} = \frac{182.54 \cdot 94 \cdot 26}{60 \cdot (94 + 26)} = 61.93\text{, Hz.} \tag{51}$$

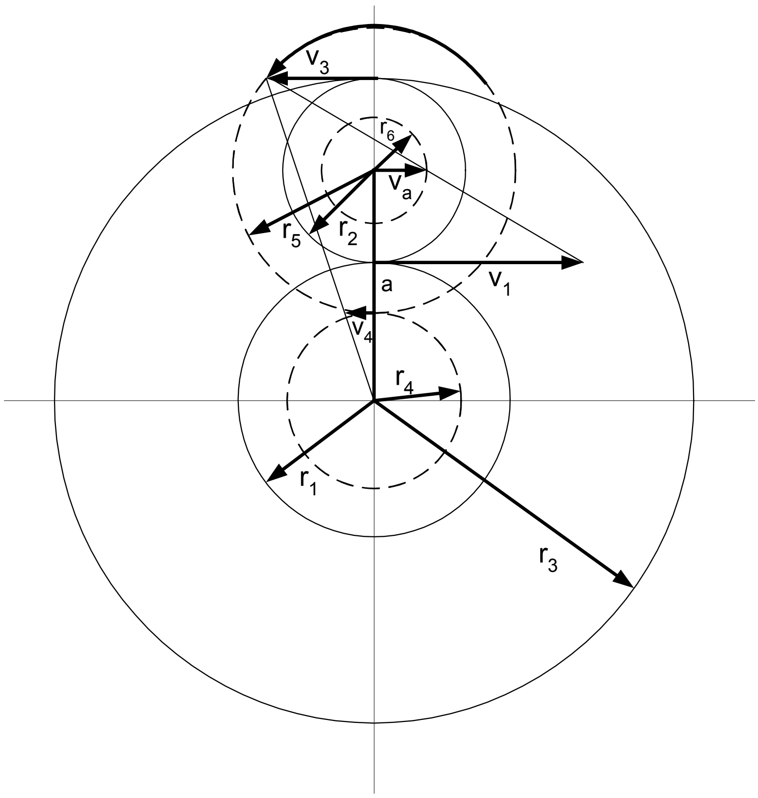

Fig. 7. Linear velocity schema for planetary subsystem given in figure 6b) gears z_5, z_6, z_7

Taking into consideration b scheme given in figure 6 the arm frequency of the first subsystem is

$$f_a = \frac{n_3 z_5}{60(z_5 + z_7)} = \frac{182.54 \cdot 94}{60 \cdot (94 + 26)} = 2.38\text{, Hz.} \tag{52}$$

Considering the sub-system a) given in figure 6. for which input RPM is $n_1 = n_i/u_{s1}$ = 958/6.7 = 60 f_a = 142.98 RPM mashing frequency for the planetary gearbox is if one take into consideration the position/direction of vectors given in figure 7

$$f_{12} = \frac{n_{1a} z_1}{60} = \frac{n_1 - n_a}{60} z_1 = \frac{n_{2a} z_2}{60} = \frac{n_2 + n_a}{60} z_2 = \frac{142.98 - 15.66}{60} 23 = 48.8\text{, Hz,} \tag{53}$$

where

$$n_2 = \frac{n_1 z_1 + n_3 z_3}{2 z_2}; \quad n_a = \frac{n_1}{u_{c2}}. \tag{54}$$

It obvious that $f_{12} = f_{23}$

$$f_{23} = \frac{n_{3a} z_3}{60} = \frac{n_3 + n_a}{60} z_3 = \frac{n_{2a} z_2}{60} = \frac{n_2 + n_a}{2} z_2 = \frac{(20.06 + 15.66)}{60} 82 = 48.8\text{, Hz.} \tag{55}$$

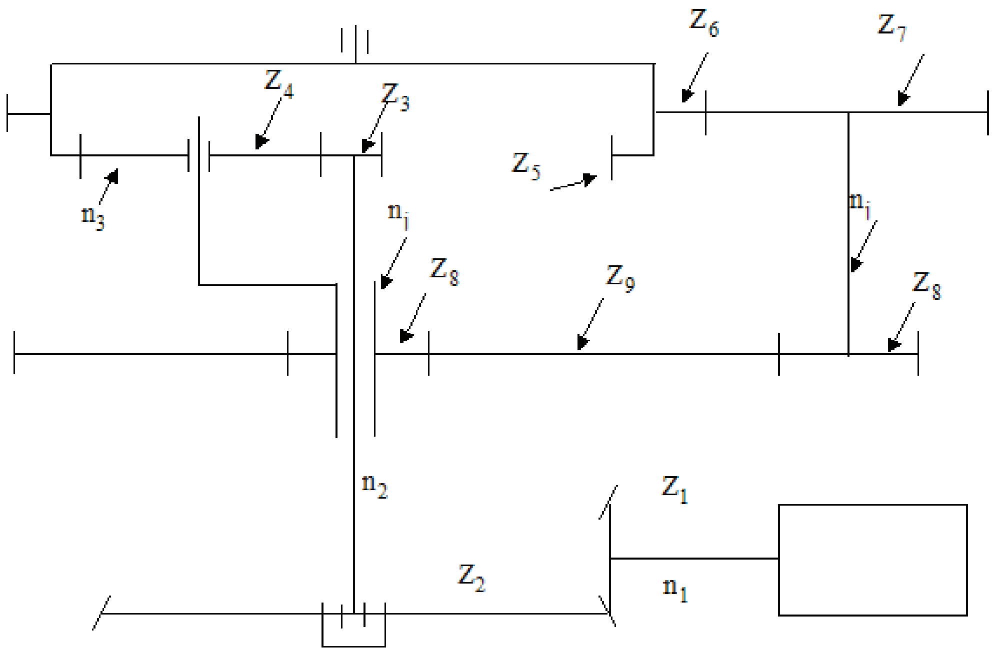

Fig. 8. Scheme for bucked wheel driving system consisting of a bevel stage and a complex gearbox system

The third example of a diagnosed heavy machinery object is a multistage gearbox used in the drive unit of a bucket wheel excavator is given in figure 8. The condition monitoring for the system is given in papers (Bartelmus, Zimroz 2009b; Bartelmus et al. 2010; Bartelmus, Zimroz 2008; Bartelmus 2009; Bartelmus, Zimroz 2010a; Bartelmus, Zimroz 2010b). The system consists of a bevel stage, gear z_1 and z_2 and the system as given in figure 6a. The planetary stage consists of gear marked as z_3, z_4, z_5. The condition of this set of gears is based on a mashing frequency given by a statement

$$f_{34} = f_{45} = \frac{n_{2j} z_3}{60} = \frac{(n_2 - n_j) z_3}{60}, \tag{56}$$

where:

n_{2j} – related speed rotation of a shaft which rotates with the speed rotation n_2 RPM,

n_2 – absolute speed of the second shaft RPM,

n_j – arm/carrier speed rotation RPM,

z_3 – number of teeth in gear 3.

To use the above statement one ought have more statements which are connected with the gearbox system given in figure 8. The complete ratio of the system is

$$u_c = u_s u_p u_w. \tag{57}$$

The bevel stage ratio equals to

$$u_s = \frac{z_2}{z_1}. \tag{58}$$

The planetary gearbox ratio equals to

$$u_p = 1 + \frac{z_5}{z_3} + \frac{z_5 z_7}{z_3 z_6}. \tag{59}$$

The cylindrical stage gear ratio equals to

$$u_w = \frac{z_8}{z_9}. \tag{60}$$

The arm/carrier speed rotation

$$n_j = \frac{n_2}{u_p}. \tag{61}$$

The example of a scheme for driving system used in shearers is given in figure 9. The driving system given in figure 9 (starting from the electric motor) consists of planetary gearbox stage with gears z_1, z_2, z_3; cylindrical gears stage with gears z_4, z_5; cylindrical gears stage with gears z_6, z_7, z_8, z_9, z_{10} where gears z_7, z_8, z_9 are idle gears; and an other planetary gearbox which consists of gears z_{11}, z_{12}, z_{13}. Using the above given principles one can easily calculate the characteristic frequencies. In some cases one even do not need the scheme to calculate the characteristic frequencies on condition that the system is reduced to compound system and for planetary stages are used statements as are given from (12) to (28).

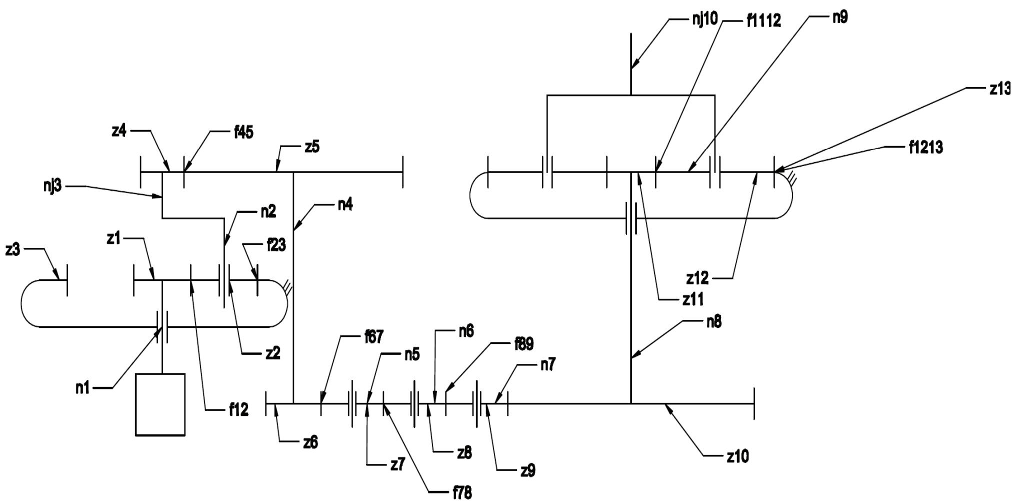

Fig. 9. Example of a scheme for driving system used in shearers

Some consideration should be given for recurrent frequencies for planetary gearboxes according to (5) to (7) for planetary stage figure 2 for gear z_1 and z_2

$$\tau_1 = \frac{1}{f_{12}} = \frac{60(z_1 + z_3)}{n_1 z_1 z_3}, \tag{62}$$

$$T_{p1} = \tau_1 N_{p11} z_1 = \frac{60(z_1 + z_3)}{n_1 z_3} N_{p1}, \tag{63}$$

$$f_{p1} = \frac{1}{T_{p1}} = \frac{n_1 z_3}{60(z_1 + z_3) N_{p1}}, \tag{64}$$

where

$$\frac{z_1}{z_2} = \frac{mM_{p1}}{mN_{p1}} = \frac{M_{p1}}{N_{p1}}, \tag{65}$$

where m is a common divisor.

There is also a need to develop recurrent frequency for planetary stage (fig. 2) for gear z_2 and z_3

$$\tau_1 = \frac{1}{f_{12}} = \frac{60(z_1 + z_3)}{n_1 z_1 z_3}, \tag{66}$$

$$T_{p2} = \tau_1 N_{p2} z_2 = \frac{60(z_1 + z_3)}{n_1 z_3} N_{p2}\,, \tag{67}$$

$$f_{p2} = \frac{1}{T_{p2}} = \frac{n_1 z_1 z_3}{60(z_1 + z_3) z_2 N_{p2}}\,, \tag{68}$$

where

$$\frac{z_2}{z_3} = \frac{mM_{p2}}{mN_{p2}} = \frac{M_{p2}}{N_{p2}}\,, \tag{69}$$

where m is a common divisor.

There is also a need to give some consideration for recurrent frequencies for planetary gearboxes according to (5) and to (7) for planetary stage (fig. 3) for gear z_1 and z_2

$$\tau_1 = \frac{1}{f_{23}} = \frac{60(z_1 + z_3)}{n_3 z_1 z_3} \tag{70}$$

$$T_{p1} = \tau_1 N_{p11} z_3 = \frac{60(z_1 + z_3)}{n_1 z_1} N_{p1} \tag{71}$$

$$f_{p1} = \frac{1}{T_{p1}} = \frac{n_3 z_1}{60(z_1 + z_3) N_{p1}} \tag{72}$$

where

$$\frac{z_3}{z_2} = \frac{mM_{p1}}{mN_{p1}} = \frac{M_{p1}}{N_{p1}} \tag{73}$$

where m is a common divisor.

There is also a need to develop recurrent frequency for planetary stage (fig. 3) for gear z_2 and z_3

$$\tau_1 = \frac{1}{f_{12}} = \frac{60(z_1 + z_3)}{n_1 z_1 z_3} \tag{74}$$

$$T_{p2} = \tau_1 N_{p2} z_2 = \frac{60(z_1 + z_3) z_2}{n_3 z_1 z_3} N_{p2} \tag{75}$$

$$f_{p2} = \frac{1}{T_{p2}} = \frac{n_1 z_1 z_3}{60(z_1 + z_3) z_2 N_{p2}} \tag{76}$$

where

$$\frac{z_2}{z_1} = \frac{mM_{p2}}{mN_{p2}} = \frac{M_{p2}}{N_{p2}} \tag{77}$$

where m is a common divisor.

Special consideration for a driving system with an overload mechanism for a bucket wheel excavator is given in paper (Bartelmus 2011). This consideration is based on consideration given in this paper.

CONCLUSIONS

The presented analysis shows the very complicated structure of characteristic frequencies for the complex and compound gearboxes, which are used in mining machinery systems. In the paper are considered the characteristic frequencies as sequence of recurrent excitations for short recurrent frequencies, meshing frequencies, shaft frequencies, local fault frequencies. Such considerations are not given in literature on subject so the paper should have influence for understanding the problems which are connected with condition monitoring and diagnostics of systems which are used in mining industry.

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REFERENCES

BANACH S., 1956, *Mechanika*, PWN, Warszawa, 338.

BARTELMUS W., 2006, *Condition monitoring of open cast mining machinery*, Oficyna Wydawnicza Politechniki Wrocławskiej, Wrocław.

BARTELMUS W., ZIMROZ R., 2007, *Metoda diagnostyki przekładni planetarnej*, Górnictwo i geologia IX (Red. nauk. Walter Bartelmus), Oficyna Wydawnicza PWr., Wrocław, 3–15.

BARTELMUS W. ZIMROZ R., 2009a, *Vibration condition monitoring of planetary gearbox under varying external load*, Mechanical Systems and Signal Processing, 23, 246–257.

BARTELMUS W., ZIMROZ R., 2009b, *A new feature for monitoring the condition of gearboxes in non-stationary operation conditions*, Mechanical Systems and Signal Processing, 23, 1528.

BARTELMUS W. CHAARI F. ZIMROZ R. HADDAR M., 2010, *Modelling of gearbox dynamics under time varying non-stationary operation for distributed fault detection and diagnosis*, European Journal of Mechanics – A/Solids, 29**,** 637–646.

BARTELMUS W. ZIMROZ R., 2008, *Problems and solutions in condition monitoring and diagnostics of open cast monster machinery driving systems*, Diagnostyka, 3/47, 55–60.

BARTELMUS W., 2009, *Statistical feature estimation of the process describing object condition change for maintenance decision*, Prace Naukowe Instytutu Górnictwa Politechniki Wrocławskiej, Górnictwo i geologia XII, Wrocław.

BARTELMUS W. ZIMROZ R., 2010a, *Way of non-stationary signal analysis generated by machinery for feature extraction and their processing*, International Congress on Noise Control Engineering, June 13–16, Lisbon, Portugal.

BARTELMUS W. ZIMROZ R., 2010b, *Heavy machinery diagnostics and condition monitoring*, Proceedings of International Conference on Condition Monitoring and Machine Failure Prevention Technology, June 22–24, Stratford upon Avon, England.

BARTELMUS W., 2011, *Fundamentals for condition monitoring and diagnostics for driving bucket wheel system of bucket wheel excavator*, Górnictwo i Geologia, Scientific Papers of the Institute of Mining.

CHARAKTERYSTYCZNE CZĘSTOTLIWOŚCI DRGAŃ DO MONITOROWANIA I DIAGNOSTYKI ZŁOŻONYCH I ZESPOLONYCH PRZEKŁADNI ZĘBATYCH MASZYN GÓRNICZYCH.

W pracy przedstawiono procedurę znajdowania charakterystycznych częstotliwości w prostych i złożonych układach przekładni zębatych. Przekładnie mogą tworzyć złożone i zespolone przekładnie. Przedstawiono sposób klasyfikacji przekładni złożonych i zespolonych. Zespolone przekładnie tworzą przekładnie planetarne. Trzy różne przekładnie planetarne są rozpatrywane. Te trzy rodzaje przekładni planetarnych są stosowane w koparkach kołowych i w kombajnach węglowych. Przedstawiono wprowadzenie do wyznaczania częstotliwości charakterystycznych. Przedstawiono charakterystyczne częstotliwości, takie jak: częstotliwość powtarzania się sekwencji spotykania się tych samych zębów, częstotliwości zazębienia, częstotliwości obrotów wałów, częstotliwości uszkodzeń lokalnych.