Increasing the quality of excavators’ planetary reduction gearboxes on the basis of dimensional analysis and geometrical characteristics of tooth wheels

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Abstract. The article describes the problem of extending operational electrically driven excavators’ reducing gear boxes which break down nearly every six months. The main reason of the function loss is the breakdown of the bearing assembly of one of the pinions. The authors performed kinematic, dynamic and geometric calculation of gearing to detect the breakdown reasons. The main reason is that alignment condition is not provided in the differential part and in the power return gear. All toothed gear wheels must be manufactured with the positive bias of the generating rack profile, but in fact all pinions and idle gears are manufactured with negative bias. This lead to an intolerable radial clearance in the gearing and skew of the floating master gears.

1. Introduction
Fuel and energy park has the outmost importance in the world economy. Despite of its turn-down coal is one of the most important among the energy resources of Russia and it stabilizes the situation in the energy industry [1, 2]. Open-cut coal mining has obvious advantages compared with underground mining: higher productivity, low cost-price and safety of mining works. This is the reason why about 70 per cent of coal in Russia are open-cut, and this figure is constantly increasing. Thus in 2015, compared with 2014, it increased by 2 %, and by 41 %, in comparison with 2000 [3]. Working efficiency of opencast coal mines is determined by productive capacity and performance reliability of the in-line equipment and, first of all, excavators. Analysis of Russian mining enterprises’ excavator fleet shows that its main part (63%) includes excavators EKG-10 with bucket capacity of 8...15 m³ produced by OMZ IZ KARTEKS [3, 4].

Mechanical section of an excavator EKG-10 comprises four basic mechanisms: lifting mechanisms (540 kW) and swing mechanisms (200 kW), as well as crowding (125 kW) and traveling (54 kW) gears. Electric motor power is given in brackets. Every mechanism includes electric motors and a reducing gear box. As you can see from the given data bowl lift mechanism has the highest power supply capacity. Operating practice of excavators EKG-10 in opencast coal mines of Kuzbass shows that the reducing gear box of the lifting mechanism breaks down nearly every six months. Given that, the repair, if it is properly organized and necessary spare parts are available, takes not more than three days, that is why it does not influence a lot the breakdown time, but it is always very expensive and excesses the budget for the yearly repair of the whole excavator. The breakdown of the lifting mechanism reducing gear box can hardly be predicted and it is always an emergency.
2. Problem description
The check of broken down reducing gear boxes of lifting mechanisms of excavators EKG-10 at three Kuzbass opencast coal mines showed absolutely the same results. The bearing cage of one of the central part’s pinions was fractured, fragments of ball race collars were broken down, and bearing rollers were lost (Figure 1), then they engaged the toothed wheel rim of the drum, and, as a result the teeth were fractured. The character of wearing of the teeth of the master gear that engages the pinions (Figure 2) shows that it worked with an obvious tilt (Figure 2). This tilt caused sideways bending on the axis of the carrier and the planetary bearing. The result of this bending is shown on the Figure 1.

![Figure 1. Fracture of the planetary bearing of the lifting mechanism reducing gear box differential part.](image1)

![Figure 2. Wearing of the master gear of the lifting mechanism reducing gear box differential part.](image2)

In view of the above the authors set a task to study kinematics and dynamics of the reducing gear box, to analyze the breakdown process and to give recommendations for extending operational life of the gear box.

3. Theory
Bowl lifting mechanism gear box of the excavator EKG-10 comprises two identical closed rack-type differentials with one common output element – it is a drum where two strings of the wire rope cable lifting the excavator bucket are spooled simultaneously (Figure 3). Every closed rack-type differential includes a differential part (central wheels $z_1$ and $z_3$, three pinions $z_2$ and a carrier) and a power return gear train (wheels $z_5$, four idler gears $z_4$ and a tooth wheel rim $z_3$). All toothed gear wheels have module $m = 10$ mm.

![Figure 3. Kinematic diagram of the excavator EKG-10 lifting mechanism reducing gear box.](image3)
The central toothed gear wheel \( z_1 \) is manufactured as floating without a bearing assembly to provide equal distribution of load on the pinions \( z_2 \) of the differential part. Torque from the engine is transmitted to it by the torsion bar via the clutch gear. To prevent axial deflection of the central wheels \( z_1 \) on their axis between them a 30 kg distance rod is installed (it is not shown on Figure 3).

Gravity force of the distance rod can produce bending moment causing the work of the master gear \( z_1 \) with a tilt. To find the reason of the tilt we calculated the forces of pinions influencing it and compared them with the gravity force of the distance rod. According to the certificate maximum heave is \( F_{\max} = 1000 \text{ kN} \). Calculated heave is assumed \( F_n = F_{\max}/2 = 500 \text{ kN} \). One string of the wire rope cable circumferential force is \( F_{\text{тр}} = 500 / 2 = 250 \text{ kN} \). Hoist drums torque:

\[
M_6 = F_{\text{тр}} \cdot \frac{d_6}{2} = 250 \cdot \frac{1.32}{2} = 165 \text{ kNm}.
\]

Gear ratio of the reducing gear box is calculated using the formula for a closed rack-type differential [5]:

\[
u_{13} = \frac{-z_2}{z_5} + \left(1 + \frac{z_3}{z_5}\right) \cdot \left(\frac{-z_2}{z_1}\right).
\]

After substitution of the teeth numbers we will get \( \nu_{13} = -25.71 \).

Then the master gear \( z_1 \) resistive torque, taking into account the reducing gear box efficiency rate, is:

\[
M_b = \frac{M_6}{\nu_{13} \eta} = \frac{165}{25.71 \cdot 0.8} = 8.02 \text{ kNm}.
\]

Master gear circumferential force:

\[
F_t = \frac{2M_b}{d_1} = \frac{2 \cdot 8.02}{0.24} = 66.83 \text{ kN},
\]

where \( d_1 \) is the reference diameter \( (d_1 = mz_1 = 10 \cdot 24 = 240 \text{ mm} = 0.24 \text{ m}) \).

Circumferential force on the master gear from one pinion:

\[
F'_t = \frac{F_t}{3} = \frac{66.83}{3} = 22.28 \text{ kN}.
\]

Then master gear radial stress from one pinion is:

\[
F_r = F'_t \cdot tga = 22.28 \cdot 0.364 = 8.11 \text{ kN},
\]

where \( \alpha \) is the standard pressure angle of a typical generating rack profile \((\alpha = 20^\circ)\). Thus the master gear is centered by three radial forces 8.11 kN each, and therefore distance rod gravity force equal to 0.3 kN cannot influence its tilt.

The second reason of the master gear work with significant tilt can be excessive spaces in toothed gearing of the reducing gear box differential part. Alignment condition preliminary check showed that toothed gear wheels \( z_1, z_2, z_4 \) and \( z_5 \) must have been manufactured with generating rack profile positive bias. Indeed alignment conditions with shifting coefficients \( x_1 = x_2 = x_4 = x_5 = 0 \) for the differential part and power return gear trains are written respectively as:

\[
z_3 = z_1 + 2z_2, \quad z_3 = z_5 + 2z_4.
\]

(1)

Substitution of teeth numbers to the formulae (1) shows that toothed wheel rims must have the number of teeth equal to 106, but not 108 as they have in fact.

To verify this version under the conditions of a servicing station of the open cast coal mine Prokopyevskiy we disassembled a broken down reducing gear box of an excavator EKG-10 and measured the main geometrical parameters of the toothed gear wheels of the differential part.

4. Experimental results

The measurements were performed with reliable measuring tools: gear-tooth micrometer BV-5046, caliper gauges of the types ShTs, ShTsK and ShTsTsK, gear-tooth micrometer of the type MZ with special attachments. The measured parameters are: outside diameter \( d_n \), mm; root diameter \( d_f \), mm; base tangent length \( w_n \) for \( n \) teeth; base tangent length \( w_{n+1} \) for \((n+1)\) teeth. The measurements were
performed five times for each wheel on different sections, and then the arithmetic mean value of the parameter was calculated.

The results of master gear geometrical parameters measurements on an unworn section are presented in the Table 1 as an example.

**Table 1.** The results of calculation the master gear parameters.

| Measured Parameter | 1 meas. | 2 meas. | 3 meas. | 4 meas. | 5 meas. | Mean |
|--------------------|---------|---------|---------|---------|---------|------|
| Outside diameter $d_a$, mm | 268.1 | 268.0 | 268.2 | 268.0 | 268.1 | 268.08 |
| Root diameter $d_f$, mm | 225.75 | 226.0 | 225.5 | 225.25 | 225.5 | 225.6 |
| Base tangent length for $n = 4$ teeth, mm | 109.9 | 109.9 | 109.4 | 109.8 | 109.7 | 109.74 |
| Base tangent length for $n + 1 = 5$ teeth, mm | 139.4 | 139.2 | 138.8 | 139.4 | 139.45 | 139.25 |

Calculation of the shifting coefficient of the master gear was performed according to the following algorithm [6].

1. Normal pitch to addendum angles:
   
   $$ p_h = w_{n+1} - w_n = 139.25 - 109.74 = 29.51 \text{ mm}, $$

   where $w_{n+1}$ is the base tangent length for $(n + 1 = 5)$ teeth, mm;
   $w_n$ – base tangent length for $(n = 4)$ teeth.

2. Tooth thickness at the base circle:

   $$ s_b = w_{n+1} - n \cdot p_h = 139.25 - 4 \cdot 29.51 = 21.21 \text{ mm}. $$

3. Base radius:

   $$ r_b = (mz/2)\cos \alpha = (10 \cdot 24/2)\cos 20^\circ = 112.763 \text{ mm}. $$

4. Shifting coefficient of generating rack profile:

   $$ x = \frac{2[5(b_2/r_b) - \text{inve}) - 0.5\pi}{2 \times \theta_a} = \frac{24[(21.21/2 \times 112.763) - 0.014904] - 0.5 \times 3.14159}{2 \times 0.364} = 0.4514. $$

5. Root diameter:

   $$ d_f = mz - 2(h_a^* + c^* - x)m = 10 \cdot 24 - 2(1 + 0.25 - 0.4514) \cdot 10 = 224.029 \text{ mm}, $$

   where $h_a^*$ is the addendum factor ($h_a^* = 1$);
   $c^*$ – is the clearance coefficient ($c^* = 0.25$).

The obtained result is coherent with the measured ($d_f = 225.6$ mm). Therefore the shifting coefficient calculated according to base tangent lengths of an unworn gear is valid.

Computational error for $d_f$ calculation was:

$$ \Delta = \frac{225.6 - 224.029}{225.6} \cdot 100\% = 0.7\%. $$

On this basis we may conclude that generating rack profile shifting coefficient can be calculated using the root diameter according to the formula obtained from (2):

$$ x = \frac{d_f - mz + 2(h_a^* + c^*)m}{2m}. $$

With this conclusion and the fact that base tangent lengths on the worn toothed gear wheels are less than design values, shifting coefficients for the rest of the toothed gear wheels were calculated according to the formula (3). The results of the measurements and calculations are presented in the Table 2.

**Table 2.** The results of measurement and calculation of gears’ parameters in the reducing gear box.

| Parameters | Pinion 1 | Pinion 2 | Pinion 3 | Central Wheel |
|------------|---------|---------|---------|--------------|
| Outside diameter $d_a$, mm | 425.14 | 424.32 | 424.4 | 1075.478 |
| Root diameter $d_f$, mm | 380.58 | 380.08 | 380.4 | 1121.378 |
| Shifting coefficient $x$ | -0.221 | -0.246 | -0.23 | – |
5. Discussion
Using the results of measurements and calculation of toothed gear wheels' parameters presented in the Tables 1 and 2, we analyzed the real alignment condition of the reducing gear box differential part. On the Figure 4 the calculation model of the reducing gear box differential part shows that:

\[ d_{a3} = d_{a1} + 2d_{f2} + 4c^*m. \]  
(4)

And also:

\[ d_{f3} = d_{f1} + 2d_{a2} + 4c^*m. \]  
(5)

After substitution of the altered parameters to (4) and (5) we got respectively:

\[ d_{a3} = 268.08 + 2 \cdot 380.35 + 4 \cdot 0.25 \cdot 10 = 1038.78 \text{ mm}. \]

\[ d_{f3} = 225.6 + 2 \cdot 424.62 + 4 \cdot 0.25 \cdot 10 = 1084.84 \text{ mm}. \]

The measured outside and root diameters of the central wheel with internal teeth: \( d_{a3} = 1075.478 \text{ mm} \) and \( d_{f3} = 1121.378 \text{ mm} \). The discrepancy is: \( \Delta_{a3} = 1075.478 - 1038.78 = 36.698 \text{ mm} \) and \( \Delta_{f3} = 1121.378 - 1084.84 = 36.578 \text{ mm} \).

6. Summary and conclusions
Thus the results of the analysis of broken down bowl lifting mechanism reducing gear boxes’ real state, measurement and calculation of geometrical parameters of the toothed gear wheels in their differential part as well as dynamic calculation of the gearing allowed to conclude the following:
1. The main reason of bowl lifting mechanism reducing gear box emergency breakdowns is the fracture of the bearing block of one of the pinions in the differential part.
2. Gravity force of the distance rod between floating master gears of the reducing gear box does not influence significantly their behavior.
3. According to alignment condition in the differential part and the power return gear train all wheels must be manufactured with positive bias of the generating rack profile. However in fact pinions and idler gears are manufactured with negative bias. This lead to intolerable radial space in the gearing about 37 mm.
4. Intolerable radial space in the gearing lead to significant tilt of the floating master gears, their oscillating movement and edge contact with the pinions. This contact caused pinions’ bending moment fracturing the bearing block.
5. The radical outcome of the existing situation is eliminating the radial space by manufacturing all toothed gear wheels with the positive bias.

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