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CONCEPT FOR EVALUATING THE SAFETY OF A DYNAMICALLY LOADED GEARBOX REDUCER

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Abstract. The safety of dynamically loaded gearbox reducers is of interest to companies and the scientific community, and any research and conclusion in this direction is a significant contribution, as this problem has not been widely discussed in scientific articles. This article presents a concept for evaluating the safety of a dynamically loaded reducer, specifically built into a rotating excavator used in surface coal mines for its working wheel. The presented concept offers three ways to evaluate safety: according to factors of safety, according to operating parameters, and according to load capacity and loads. A methodology was presented for all three ways, which was applied to a specific reducer under specific working conditions, and output results were obtained. The output results were analysed and conclusions were drawn using the specific reducer. However, it was established, based on the author's experience, that these results can be generalised for similar reducers operating under the same or similar working conditions. The set concept can be applied to the safety assessment of other dynamically loaded gear reducers.

Keywords: Gearbox reducer, dynamically loading, reducer safety, reducer evaluation.

1. Introduction

The author has been involved with the problem of gearbox reducers for three decades, starting with the research for her Master's thesis [1]. Of particular interest are the dynamically loaded reducers incorporated in mechanised machines operating in surface mines. Most of the theoretical and experimental researches concern the reducers built in rotating excavators working in surface coal mines, for whose activity more articles have been published. Research has been carried out in several directions, ranging from experimental measurements of the loads on reducers, with an interest in the measurement itself [2–4] and the distribution of the load [5], to the optimisation of the reducers as a whole and their constituent components [6].

The starting point for this article is a previous study on the evaluation of the factor of safety of a reducer of a working wheel in specific excavators operating under the same working conditions [7] and the established methodology for this purpose [8]. A set methodology was also used to determine the driving factor of reducers from rotating excavators [9], as an important characteristic of every dynamically loaded machine and the most difficult to determine in these excavators considering the difficult working conditions and the expensive working hours.

The conceptualisation of this article does not require the use of additional literature, given that this problem is not much exposed in scientific articles and that the approach depends on the machine and working conditions, so it relies mainly on the author's experience in dealing with this problem. Based on the author's previous experiences and previously conducted research, for article purposes, an in-depth research and analysis have been carried out to specify some author's research interests about the reducer of the working part (working wheel) of the SRs-630 rotating excavator which is used in the mine "Suvodol" – Bitola, R. N. Macedonia.

In the reducer that is the subject of the research in this paper, a weakness was observed in the first, second, and last gear pair (marked with 1b, 2, and 5, respectively, shown in Figure 1). The scheme of the reducer with marked gear pairs drawn according to [10] is shown in Figure 1. For the third and fourth gear pairs (3 and 4 in Figure 1) no problems have occurred during the working life of the reducer so far, so they are virtually as supplied by the manufacturer. Due to the above, only the first, second, and last gear pairs will be the subject of a further research.

The first pair of gears is double and consists of a driving gear, which is made of two cylindrical helical gears set on a common carrier, and two following gears appropriately placed on the same shaft. Depending on which part of the pair is engaged, two output shaft speeds can be achieved for one input speed and two gear ratios of the reducer. In practice, the excavator works at the higher speed, so only the right part of the gear pair (gear pair 1b) is of interest.

The second pair of gears are conical gears with spiral teeth – miter gears with spiral teeth, and the last pair of gears are cylindrical gears with double helical teeth – double helical gears.

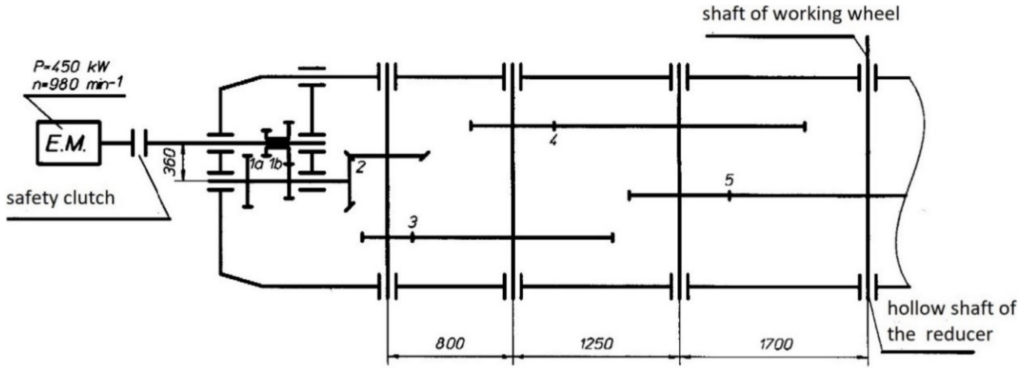

Figure 1. Scheme of the reducer.

Conditional factor of safety	Gear pair 1b		Gear pair 5	
	$z_1 = 56$	$z_2 = 45$	$z_1 = 23$	$z_2 = 98$
$S_H K_A$	7.9	6.4	2.9	11.9
S_H	4.9	3.99	1.8	7.4
$S_F K_A$	9.5	9.99	7.2	6.4
S_F	5.9	6.25	4.5	3.99

Table 1. Conditional factors of safety for gear pair 1b and gear pair 5.

2. Materials and methods

2.1. Evaluation of the reducer according to factors of safety

2.1.1. For gear pairs 1b and 5

Conditional factor of safety from surface pressure:

$$\begin{aligned} S_H K_A =& 0.87171 \cdot \frac{m_n^2 Z^2 bn}{10^{10} P \cos^2 \beta} \cdot \frac{u}{u+1} \\ & \cdot \frac{(Z_L Z_R Z_V Z_N Z_W)^2}{K_V K_{H\alpha} K_{H\beta} Z_{H\beta}^2 Z_\varepsilon^2} \cdot \sigma_{H\mathrm{lim}}^2. \end{aligned} \tag{1}$$

Conditional factor of safety against teeth root bending:

$$\begin{aligned} S_F K_A =& \frac{2\pi}{10^6} \cdot \frac{m_n^2 Z bn}{P} \\ & \cdot \frac{Y_X Y_N Y_\delta Y_R Y_{Eht}}{K_V K_{F\alpha} K_{F\beta} Y_{FS} Y_\varepsilon Y_\beta \cos \beta} \cdot \sigma_{F\mathrm{lim}}. \end{aligned} \tag{2}$$

Conditional factors of safety [11] are calculated for the nominal load of the reducer, which is determined based on data from experimental measurements for all characteristic working regimens of the excavator [12].

At the input shaft of the reducer, the calculated nominal power is 329.43 kW. The coefficient of utilisation of the first gear pair is $\eta = 0.98$, so the second gear pair transmits a power of 322.85 kW. At the output shaft of the reducer, this power is 303.08 kW.

The values of the conditional factors of safety, calculated with application software, according to the specified Equations (1) and (2), for the gears of gear pairs 1b and 5, are given in Table 1.

Conditional factor of safety	Gear pair 2	
	$z_1 = 29$	$z_2 = 38$
$S_H K_A$	9.5	11.9
S_H	5.9	7.4
$S_F K_A$	6.8	6.9
S_F	4.2	4.3

Table 2. Conditional factors of safety for gear pair 2.

2.1.2. For gear pair 2

Conditional factor of safety from surface pressure:

$$\begin{aligned} S_H K_A =& 1.654 \cdot \frac{m_{mn}^2 Z^2 bn}{10^{10} P \cos^2 \beta} \cdot \frac{u}{\sqrt{u^2+1}} \\ & \cdot \frac{(Z_L Z_V Z_R Z_N Z_W Z_{Eht})^2}{K_V K_{H\alpha} K_{H\beta} Z_{H\beta}^2} \cdot \sigma_{H\mathrm{lim}}^2. \end{aligned} \tag{3}$$

Conditional factor of safety against teeth root bending:

$$\begin{aligned} S_F K_A =& 5.34 \cdot \frac{m_{mn}^2 Z bn}{10^{10} P \cos^2 \beta} \\ & \cdot \frac{Y_X Y_R Y_N Y_\delta}{K_V K_{F\alpha} K_{F\beta} Y_{FS} Y_\varepsilon Y_\beta Y_{LS}} \cdot \sigma_{F\mathrm{lim}}^2. \end{aligned} \tag{4}$$

The values of the conditional factors of safety, calculated with application software, according to the specified Equations (3) and (4), for the gears of gear pair 2, are given in Table 2.

2.2. Evaluation of the reducer according to the operating parameters

This evaluation is made according to data from references units [13, 14], and refers to the following constituent components of the reducer:

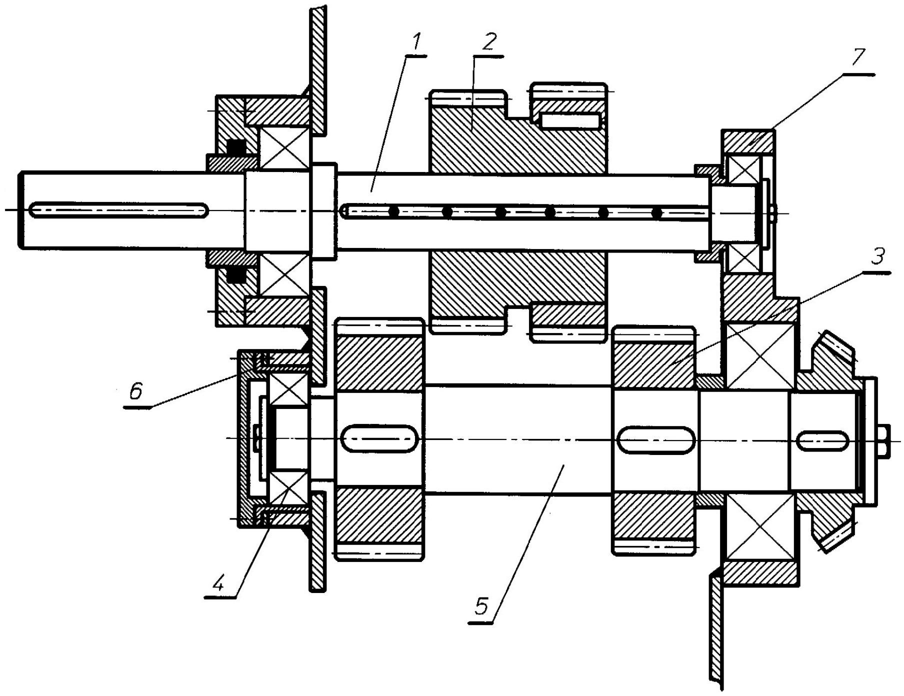

Figure 2. Input gear pair in the reducer.

- Input gear pair (1b, as shown in Figure 1) – this gear pair is replaced once a year, during a general overhaul of the excavator, or twice a year on rare occasions when it behaves unpredictably. The change is made due to the wear of the bearings (see Figure 2), causing deformation and damage to the gears.

  The gears of this gear pair are also damaged by the displacement of the driving gear (gear 2 as shown in Figure 2) along the groove for the wedge of the first shaft of the reducer (shaft 1 as shown in Figure 2), under stochastic loads, which displacement is allowed by the design.

  The input gear in the reducer (marked as 2 in Figure 2) is designed to be able to move along the groove to conjugate with one of the two driving gears placed on the second shaft (marked as 5 in Figure 2) and allows the realisation of two gear ratios of the reducer [10], and accordingly, two digging speeds of the working organ of the excavator (the rotating wheel of the excavator).

- Miter gear pair (marked as 2 as shown in Figure 1) – this gear pair ischanged at least once a year during a general overhaul of the excavator, with arepaired spare gear pair. The reason for damage to the gears is not the quality of manufacture and load capacity, but a practical one – poor mounting and centring. Centring is almost impossible to perform ideally due to the working conditions on the plateau itself, where the excavator is being repaired.

  Damage occurs in the form of broken tips of the teeth, resulting in a poor loading image, uneven gear wear, causing heating of the gears, noise, and vibration during the operation of the reducer.

- Output gear pair (5, as shown in Figure 1) – this gear pair is checked 2 to 3 times a year by opening the reducer's cover, and is replaced as needed. In service to date, the second gear of this pair has been replaced only a few times with a spare gear during the working life of the reducer. Damage to this gear occurs in the form of tiny holes on some of the gear flanks, which are characterised as initial pitting. The established reason for this behaviour is not the quality of the flanks, but the occasionally used inappropriate lubricating oil as well as dust and other impurities that settle on the gear. No significant damage was observed in the driving gear of this gear pair that would require its replacement.

- Bearings – the bearings of this reducer are usually replaced due to their damage in the form of wear. In 90 % of cases, the reducer stops working due to bearing damage.

- Output shaft of the reducer (shown in Figure 3) – this shaft is designed as a hollow shaft and supported on sliding bearings. Bearings (marked as 1 and 2 in Figure 3) are frequently changed due to the possibility of damage being transmitted to both the shaft and the bushings that are set down onto the shaft and over which the bearings are placed. Most often these bearings are damaged from clogging of the lubrication channel.

2.3. Evaluation of the reducer according to the load bearing capacity and the loads

2.3.1. Load capacity of the reducer

Based on an appropriate methodology [1], values for the carrying capacity of the gears of the reducer for

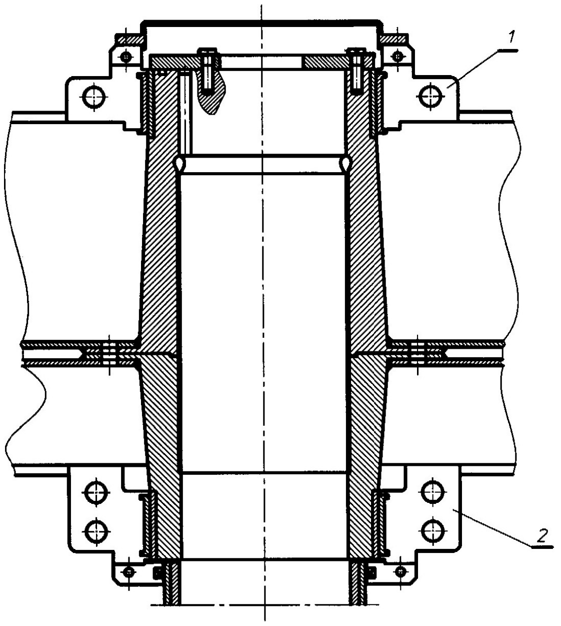

FIGURE 3. The hollow shaft of the reducer with its bearings.

surface pressure and bending are calculated theoretically, for three values ($N = 10^0$, $N = 10^4$, and $N = 10^7$, where N is the number of load changes on the reducer).

For gear pairs 1b and 5

- Load capacity of surface pressure
  - ▷ Permanent load capacity:
    Maximum peripheral force on the tooth flank:

$$F_{tH\infty} = \frac{bd}{S_{H\lim}} \cdot \frac{u}{u+1} \cdot \frac{(Z_X Z_L Z_R Z_\nu Z_N Z_w Z_{Eht})^2}{K_\nu K_{H\alpha} K_{H\beta} Z_\varepsilon^2 Z_E^2 Z_{H\beta}^2} \cdot \sigma_{H\lim}^2. \tag{5}$$

Maximum torque from peripheral force:

$$T_{H\infty} = \frac{F_{tH\infty} d}{2}. \tag{6}$$

  - ▷ Static load capacity:
    Maximum peripheral force on the tooth flank:

$$F_{tH0} = \frac{bd}{S_{H\lim}} \cdot \frac{u}{u+1} \cdot \frac{(Z_X Z_L Z_R Z_\nu Z_N Z_w Z_{Eht})^2}{K_\nu K_{H\alpha} K_{H\beta} Z_\varepsilon^2 Z_E^2 Z_{H\beta}^2} \cdot \sigma_{H0}^2. \tag{7}$$

Maximum torque from peripheral force:

$$T_{H0} = \frac{F_{tH0} d}{2}. \tag{8}$$

  - ▷ Load capacity at $N = 10^4$ load changes:

$$T_{H(10^4)} = \sqrt{T_{H\infty} T_{H0}}. \tag{9}$$

- Load capacity of tooth root bending
  - ▷ Permanent load capacity:
    Maximum peripheral force at the tooth root:

$$F_{tH\infty} = \frac{bd\cos\beta}{z} \cdot \frac{\sigma_{F\lim}}{S_{F\lim}} \cdot \frac{Y_x Y_{ST} Y_N Y_\delta Y_R Y_{Eht}}{K_\nu K_{F\alpha} K_{F\beta} Y_{FS} Y_\varepsilon Y_\beta}. \tag{10}$$

Maximum torque from peripheral force:

$$T_{F\infty} = \frac{F_{tF\infty} d}{2}. \tag{11}$$

  - ▷ Static load capacity:
    Maximum peripheral force at the tooth root:

$$F_{tF0} = \frac{bd\cos\beta}{z} \cdot \frac{\sigma_{F0}}{S_{F\lim}} \cdot \frac{Y_x Y_{ST} Y_N Y_\delta Y_R Y_{Eht}}{K_\nu K_{F\alpha} K_{F\beta} Y_{FS} Y_\varepsilon Y_\beta}. \tag{12}$$

Maximum torque from peripheral force:

$$T_{F0} = \frac{F_{tF0} d}{2}. \tag{13}$$

  - ▷ Load capacity at $N = 10^4$ load changes:

$$T_{F(10^4)} = \sqrt{T_{F\infty} T_{F0}}. \tag{14}$$

The influencing factors in the previously mentioned formulas are determined according to the appropriate methodology for this purpose.

The factor of safety is still used to cover the remaining influences on the operation of the gears which have not yet been covered by their own factor. Since the maximum load capacity should be determined here, its minimum values are taken for the factor of safety: $S_{H\lim} = 1$ and $S_{F\lim} = 1.2$.

According to the known data about the gears and the Equations (5)–(14), the load capacity of the gear pairs 1b and 5 was calculated with application software and are shown in Table 3.

The specified load capacity for gear pair 5 in the table refers to a half gear. For a whole gear with double helical teeth, the given values are multiplied by 2 and the load capacity of the surface pressure and tooth root bending is obtained.

For gear pair 2

- Load capacity of surface pressure
  - ▷ Permanent load capacity:
    Maximum peripheral force on the tooth flank:

$$F_{tH\infty} = \frac{bd_m}{S_{H\lim}} \cdot \frac{u}{\sqrt{u^2+1}} \cdot \frac{(Z_X Z_L Z_R Z_V Z_W Z_N Z_{Eht})^2}{K_V K_{H\alpha} K_{H\beta} Z_{H\beta}^2} \cdot \sigma_{H\lim}^2. \tag{15}$$

Maximum torque from peripheral force:

$$T_{H\infty} = \frac{F_{tH\infty} d_m}{2}. \tag{16}$$

Carrying capacity	Gear pair 1b		Gear pair 5	
	$z_1 = 56$	$z_2 = 45$	x 2 $z_1 = 23$	x 2 $z_2 = 98$
$F_{tH\sim}$ [N]	127 164.2	103 166.1	447 077.43	2 198 172.23
$T_{H\sim}$ [N mm]	25 390 875.81	16 553 000.74	143 210 077.3	3 000 186 363
F_{tH0} [N]	5 208 646.8	3 337 790.79	16 094 788.15	54 954 306.2
T_{H0} [N mm]	1 040 010 506	535 548 533.5	5 155 563 010	75 004 659 000
$T_{H(10^4)}$ [N mm]	162 501 623.3	94 153 785.21	859 260 482.6	15 000 931 000
$F_{tF\sim}$ [N]	127 918.8	135 094.1	892 999.97	819 305.64
$T_{F\sim}$ [N mm]	25 542 546.8	21 675 848.3	286 050 216.6	1 118 233 401
F_{tF0} [N]	447 715.7	472 829.2	3 162 708.6	3 098 378.6
T_{H0} [N mm]	89 395 392.8	75 865 443.1	1 013 094 644	4 228 837 513
$T_{F(10^4)}$ [N mm]	47 783 853	40 551 791.9	5 383 226 984.5	21 745 867 708

Table 3. Load capacity for gear pairs 1b and 5.

▷ Static load capacity:
Maximum peripheral force on the tooth flank:

$$F_{tH0} = \frac{bd_m}{S_{H\lim}} \cdot \frac{u}{\sqrt{u^2+1}} \cdot \frac{(Z_X Z_L Z_R Z_V Z_W Z_N Z_{Eht})^2}{K_V K_{H\alpha} K_{H\beta} Z_{H\beta}^2} \cdot \sigma_{H0}^2. \tag{17}$$

Maximum torque from peripheral force:

$$T_{H0} = \frac{F_{tH0} d_m}{2}. \tag{18}$$

▷ Load capacity at $N = 10^4$ load changes:

$$T_{H(10^4)} = \sqrt{T_{H\infty} T_{H0}}. \tag{19}$$

- Load capacity of tooth root bending

▷ Permanent load capacity:
Maximum peripheral force at the tooth root:

$$F_{tF\infty} = \frac{bd_m \cos\beta_m}{z} \cdot \frac{\sigma_{F\lim}}{S_{F\lim}} \cdot \frac{Y_X Y_N Y_R Y_\delta Y_{Eht}}{K_V K_{F\alpha} K_{F\beta} Y_{FS} Y_\varepsilon Y_K Y_{LS}}. \tag{20}$$

Maximum torque from peripheral force:

$$T_{F\infty} = \frac{F_{tF\infty} d_m}{2}. \tag{21}$$

▷ Static load capacity:
Maximum peripheral force at the tooth root:

$$F_{tF0} = \frac{bd_m \cos\beta_m}{z} \cdot \frac{\sigma_{F0}}{S_{F\lim}} \cdot \frac{Y_X Y_N Y_R Y_\delta Y_{Eht}}{K_V K_{F\alpha} K_{F\beta} Y_{FS} Y_\varepsilon Y_K Y_{LS}}, \tag{22}$$

where $d_m = d - b\sin\phi$ represents the diameter of the middle dividing circle.
Maximum torque from peripheral force:

$$T_{F0} = \frac{F_{tF0} d_m}{2}. \tag{23}$$

Carrying capacity	Gear pair 2	
	$z_1 = 29$	$z_2 = 38$
$F_{tH\sim}$ [N]	2 667 868 701	3 354 330 573
$T_{H\sim}$ [N mm]	375 996 070 000	619 460 990 000
F_{tH0} [N]	58 100 247 000	64 939 839 000
T_{H0} [N mm]	9 422 117 200 000	13 799 715 000 000
$T_{H(10^4)}$ [N mm]	1 882 200 500 000	2 923 762 000 000
$F_{tF\sim}$ [N]	41 943.4	42 162.1
$T_{F\sim}$ [N mm]	5 911 293.1	7 786 285.8
F_{tF0} [N]	114 924.9	115 524.1
T_{H0} [N mm]	18 637 382.9	24 548 855.1
$T_{F(10^4)}$ [N mm]	10 496 239	13 825 498.3

Table 4. Carrying capacity for gear pair 2.

▷ Load capacity at $N = 10^4$ load changes:

$$T_{F(10^4)} = \sqrt{T_{F\infty} T_{F0}}. \tag{24}$$

The influencing factors listed in the previous formulas are determined for the gears of gear pair 2.

For the factor of safety, the following is taken: $S_{H\lim} = 1.2$ and $S_{F\lim} = 1.4$.

According to the determined data for the gears and the Equations (15)–(24), the carrying capacity of the gear pair 2 was calculated with application software and is shown in Table 4.

2.3.2. Loadings of the reducer

The loads on the reducer are determined by measuring the loads [4] on the output shaft of the reducer, as the most loaded part of the reducer, since the shaft of the excavator's working wheel is mounted on this shaft, which receives the resistances from digging. For realising the experimental measurement of the loads on the output shaft of the reducer, a methodology has been set up for that purpose, presented in [12]. This methodology requires a long analysis and preparation considering the complex design of the connection of the output shaft of the reducer with the excavator's working wheel shaft and the nature of the excavator's work. From the output shaft, the load is transferred

Regimes	ε $\times 10^{-6}$ [m]	T_M [N mm]	τ [N mm^{-2}]	P_i [kW]	P_ν [kW]
Calculated mechanical quantities based on the measured magnitude ε					
First	891.48	1 286 377 308	144.01	960.61	1 044.14
Second	629.28	908 031 041	101.65	678.08	737.04
Third	506.92	731 469 450	81.89	546.23	593.73
Fourth	122.36	176 561 591	19.66	131.84	143.30
Fifth	804.08	1 160 261 886	129.89	866.43	941.77
Sixth	856.52	1 230 159 266	137.71	916.63	996.34

TABLE 5. Presentation of systematic measurement results.

to the gear that is placed on that shaft, that is, to the following gear z_2 of gear pair 5.

The load on the driving shaft of the working wheel is measured when the excavator is working with the horizontal carving and at the higher rotation speed of the working wheel ($7.2\,\mathrm{min}^{-1}$) in each of the following defined working regimes:

- first regime – turning left by going down, maximum working load,
- second regime – turning left by going down, normal working load,
- third regime – turning left by going down, maximum working load at which the safety clutches disengage the working wheel,
- fourth regime – the excavator working and advancing (frontal biting),
- fifth regime – turning right by going down, normal working load,
- sixth regime – turning right by going down, maximum working load.

These different working regimes cover all the working conditions [15] in the total working life of the excavator. For each of these working regimes, Table 5 shows the processed output results of the measurement, i.e. maximum deformations ε on the measuring points where the measuring strain gages are placed, as well as the mechanical quantities calculated with application software (torque T_M, tangential stress of torsion τ, the power of output shaft of the reducer – driving shaft of the working wheel P_i, and the power of the input shaft of the reducer P_ν).

The calculated quantities presented in Table 5 are obtained using the basic settings of the theory of strength of materials and known characteristics of the reducer, as shown below:

$\boldsymbol{\varepsilon}$ relative elongation (dilatation) at the measurement point (from strain gages measurements) in µm,

$\boldsymbol{T_M}$ torque at the measuring point, calculated on the basis of the measured deformation ε:

$$T_M = \frac{E}{1+\nu} \cdot W_o \cdot \varepsilon, \tag{25}$$

where

E modulus of elasticity of the material of the shaft of the gearbox, $E = 0.21 \times 10^6\,\mathrm{N\,mm^{-2}}$,

ν Poisson's ratio, $\nu = 0.3$,

W_o polar moment of inertia of the shaft of the reducer, for the diameter d where the measuring strain gages are placed, $d = 354.8\,\mathrm{mm}$, $W_o = 0.2 \cdot d^3 = 0.2 \cdot 354.8^3 = 8\,932\,660.4\,\mathrm{mm}^3$,

$\boldsymbol{\tau}$ tangential stress at the measurement point of the shaft:

$$\tau = \frac{T}{W_o}, \tag{26}$$

$\boldsymbol{P_i}$ power of the output shaft of the reducer (the driving shaft of the working wheel), calculated on the basis of T_M:

$$P_i = \frac{T_M \cdot n}{159\,155}, \tag{27}$$

where n is number of rotations of the shaft per second, $n = 7.2\,\mathrm{min}^{-1} = 0.11885\,\mathrm{s}^{-1}$,

$\boldsymbol{P_\nu}$ power of the input shaft of the reducer:

$$P_v = \frac{P_i}{\eta_R}, \tag{28}$$

where η_R is coefficient of utilisation of the reducer.

3. RESULTS AND DISCUSSION

3.1. ANALYSIS OF SAFETY FACTORS OF THE REDUCER

The following can be concluded from the specified values for the conditional factors of safety shown in Tables 1 and 2:

- The factors of safety for surface pressure and tooth root bending for the two gears of gear pair 1b are higher than the recommended minimum values previously stated: 4.9–3.9 times for surface pressure and 4.9–5.2 times for tooth root bending (the first number refers to the driving gear and the second number to the following gear of the pair). According to this, it is concluded that the gears are oversized for the nominal loading and can reliably withstand significantly higher loads.
- The factors of safety for gear pair 2 are greater than the minimum recommended values, 3–3.1 times for tooth root bending and 4.9–6.2 times for surface pressure. The reasoning leads to the conclusion that the factor of safety for surface pressure is greater than the factor of safety for tooth root bending, for

both gears. In the nominal mode of operation of the excavator, theoretically, no damage to the gears would be expected for this gear pair.

- The factors of safety for tooth root bending for gear pair 5 are 3.8–3.3 times higher than the minimum recommended values, so theoretically, damage of this type should not be expected for the gears during the nominal mode of operation of the excavator. The factor of safety for the surface pressure of the large gear (gear z_2 of gear pair 5) is 7.4 times higher than the minimum values and leads to the conclusion that there is no risk of pitting at the nominal operation and increased loads on the excavator.
- The lowest factor of safety for surface pressure has been found for the driving gear of gear pair 5, $S_H = 1.8$ which is greater than $S_{H\mathrm{lim}} = 1$. Theoretically, there is no danger of pitting damage on the flanks of a tooth at the nominal loading mode. During the operation of the reducer with the maximum power of 400 kW, for which it is designed, this factor of safety decreases to $S_H = 1.4$, and when the maximum loading is reached in the first operating mode of the excavator, it reaches a value close to $S_{H\mathrm{lim}}$. This theoretical reasoning leads to the conclusion that extreme loads on the reducer should be avoided in order to ensure the safety of this gear and the safety of the reducer as a whole.

3.2. Analysis of operational parameters of the reducer

Practical experience shows that the reducer as a whole is well chosen for the operating conditions in which it works and is quite reliable with proper maintenance.

Work stoppages were minimal, according to the assessment of the personnel authorised to maintain the reducer. The most common interventions are due to the replacement of damaged bearings of the reducer.

During the previous work on the reducer, deviations from the expected behaviour of the gears were observed, which is why it was examined during the annual general overhaul of the excavator.

3.3. Analysis of the load capacity and the loads on the reducer

The load capacity of the reducer is equal to the permissible time-constant torque T_2 acting on the following shaft of the reducer during N load changes, that is, the ability of the material moulded into the gear to receive certain types of maximum loads with a determined number of changes, under certain working conditions.

The load capacity curves for gear z_2 of gear pair 5 for surface pressure and tooth root bending are shown in the diagram in Figure 4. The following can be derived from these gear curves (Figure 4):

- The value of the maximum and minimum loads that cause the material to break,
- The value of the load that causes the material to break at a certain number of changes in load.

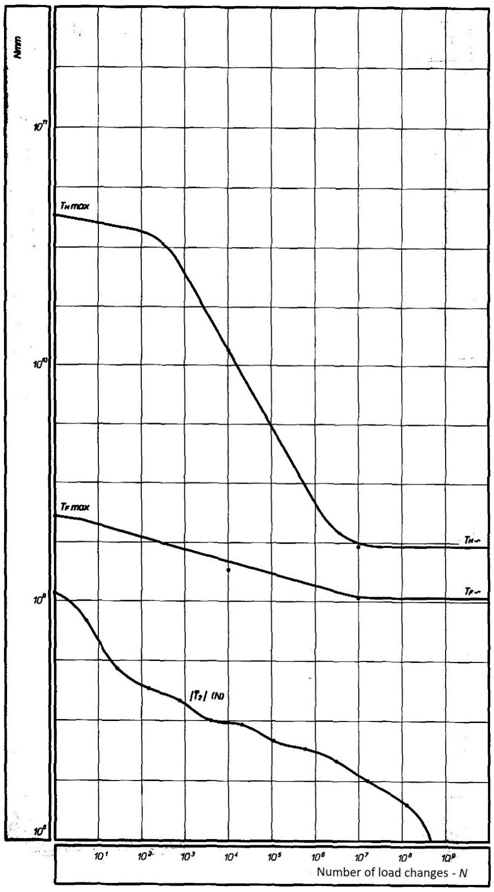

Figure 4. Comparison diagram of the load capacity and the loads for the reducer (for gear z_2 of gear pair 5).

The loading on the reducer represents a change-flow of torque T_2 depending on time t, on the following shaft of the last gear pair, in the total working life of the reducer.

The loading function $T_2(N)$ is the result of processing data of the deformations on the output shaft of the reducer for the specified characteristic operating modes of the excavator, whereby the deformations are obtained by experimental measurements, as shown in the diagram in Figure 4.

Comparing the loading function of gear z_2 of gear pair 5 with the load capacity functions, according to Figure 4, we can conclude that:

- The loading function with maximum value $T_2 = 1.2 \times 10^9$ N mm for $N = 10^0$ and decreasing values for T_2 with increasing number of load changes N, is set significantly lower than the surface pressure capacity function and the load capacity function for tooth root bending. Since these two curves (the loading function and the surface pressure capacity function, as well as the loading function and the load capacity function for tooth root bending) do not intersect and do not touch, based on theoretical settings, it is concluded that the reducer is oversized and has a high reliability in the nominal working mode of the excavator.

4. Conclusion

In this article, a concept for the safety evaluation of a dynamically loaded reducer (gear reducer) is presented.

According to this concept, three criteria are proposed for evaluating the safety of reducers:

- factors of safety,
- exploitation parameters (operating parameters),
- the load capacity and the loads.

For the working conditions in the Suvodol-Bitola coal mine, a number of safety factors for the most heavily loaded gears in the reducer of the SRs-630 excavator have been calculated according to the methodology established for this purpose, and an analysis of the permissible factors of safety has been carried out. Knowing this data gives a clear indication of the factor of safety of the reducer as a whole. The data for the factor of safety of the reducer are evaluated as real and objective parameters, based on a comparison of these data with the data for the same excavators working in similar operating conditions.

The evaluation of the safety of the reducer depending on the operating conditions is carried out according to the analysis methodology of the behaviour of the reducer components (gears, shafts, and bearings) and the systematisation of the most critical parts, their types of defects and the methods of intervention. The input data for the analysis comes from the technical records of the implemented planned and unplanned maintenance interventions carried out on the reducer. The general assessment according to this criterion is that the reducer of the working wheel of the SRs-630 rotating excavator in the Suvodol-Bitola coal mine is well-dimensioned and reliable with proper maintenance for its operating conditions.

The third set criterion for the evaluation of the reducer in this article is the most reliable, but also the most difficult to establish, because it is related to the determination of two mechanical quantities – the load capacity of the reducer and the loads. Taking into account the several components that the reducer is composed of and the operating conditions in which the reducer works, this procedure becomes significantly more complicated. For this criterion, a model is established in this article to determine the load capacity of the reducer based on the determination of the load capacity of the most loaded gear pairs. Also, in this article, the function of the loading of the reducer for its total projected working life is presented, which was obtained based on conducted experimental measurements and processing of the measurement results according to an appropriate methodology. A comparison diagram is given for the load capacity and the loads of the reducer, which displays a visual picture of their ratio for different numbers of load changes.

The presented concept for evaluating the safety of a dynamically loaded gear transmission reducer for a specific excavator and specific operating conditions can be used to evaluate the safety of reducers of other rotating excavators, but also for reducers installed in other mechanisedmachines, as concluded by the author based on the author's experience according to [16].

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