CONCEPT FOR DETERMINING DRIVING FACTOR OF GEARBOX SPEED REDUCER

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Abstract: *The driving factor is an important technical feature of dynamical loaded working machines. It is especially important to know its value for the working wheel speed reducer of the rotating excavators. In this paper is presented a concept for determining the driving factor of a concrete gearbox speed reducer. To calculate the value of the driving factor according to the specified concept, knowledge of the load regime and the carrying capacity of the gearbox is necessary. The load regime is known from the load function, which is given in the paper, and is determined on the basis of the experimental measurements carried out for this purpose. The carrying capacity of the speed reducer is calculated by theoretical analysis according to the established methodology in this paper, for the characteristic pairs of gears determined in this analysis. As the most important gear pair, from the aspect of determining the driving factor, the most loaded gear pair is taken, in which during the exploitation of the speed reducer the most interventions are made, compared to the other gear pairs of the gearbox.*

Keywords: DRIVING FACTOR, GEARBOX REDUCER, ROTATING EXCAVATOR, COALMINE

1. Introduction

The load regime of the gearbox speed reducer of the working wheel of rotating excavator for normal and specific exploitation conditions is defined by the torque distribution function of the output shaft, depending on the number of load changes in the expected working life of the speed reducer.

The load regime of the speed reducer of the working wheel of the rotating excavator SRs-630 which is used in the coalmine Suvodol-Bitola in Macedonia, is determined on the basis of conducted experimental measurements and extensive theoretical analysis. The load function of this speed reducer is shown in this paper.

The carrying capacity of the gearbox speed reducer of the working wheel of rotating excavator is determined for its most loaded gear pair, i.e. for the gear pair that has certain operational weaknesses.

The carrying capacity for all gear pairs from the working wheel speed reducer of the rotating excavator SRs-630 has been obtained theoretically, with a set methodology for this purpose, presented in the continuation of the paper. The practice of maintaining the speed reducer showed that certain weaknesses during the exploitation were observed in the first, second and last (the fifth) gear pair, but most was intervened at the last gear pair which is the most loaded in the speed reducer, that is, the following gear of this gear pair which is mounted on the output shaft of the speed reducer.

The driving factor is a technical feature of dynamical loaded working machines. The driving factor is an important indicator of the dynamic load of the speed reducers in the rotating excavators, in particular for the working wheel speed reducer of the excavator.

In this paper, the concept for determining the driving factor of the working wheel speed reducer of the rotating excavator SRs-630 is presented and a numerical value for the driving factor based on such a methodology has been obtained.

2. Load regime of the speed reducer

The loading of the speed reducer is a change in the torque $\boldsymbol{T_2}$ of the following shaft depending on the time $\boldsymbol{t}$ in the total exploitation life of the speed reducer.

The load function of the speed reducer of the excavator SRs-630 (shown in Figure 1) was obtained by processing the deformations of the output shaft of the speed reducer under the characteristic working regimes for the excavator, which deformations were obtained by experimental measurements.

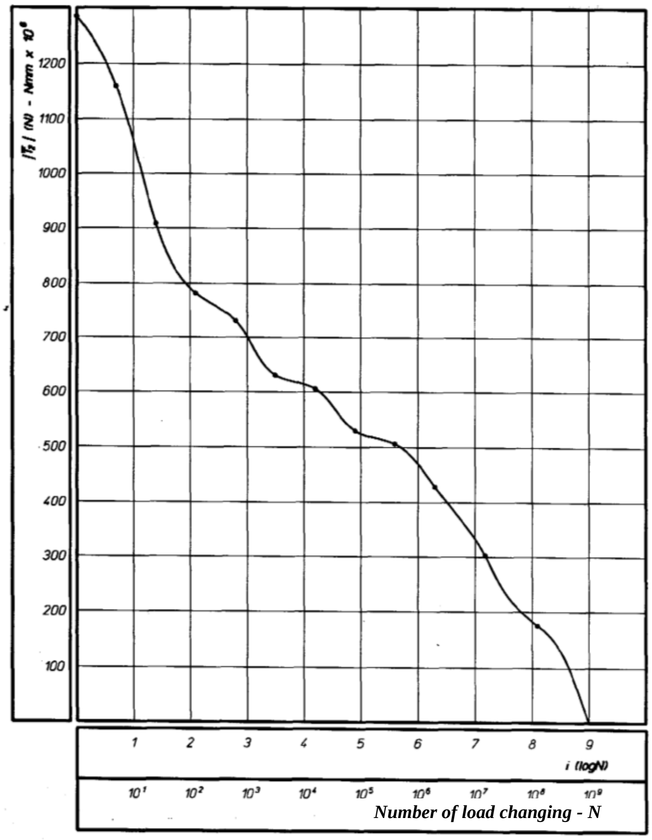

Fig. 1 *Load function of the working wheel speed reducer of the rotating excavator SRs-630 under working conditions at the coalmine Suvodol-Bitola*

3. Carrying capacity of the speed reducer

3.1. Generally for the carrying capacity and characteristics of the speed reducer

The carrying capacity is the ability of the speed reducer to receive certain types of maximum loads with a certain number of changes, under certain working conditions. The carrying capacity of the speed reducer can be associated with the Veler's curves of fatigue of the material for the gears in the speed reducer, obtained by examining the gears (in particular on the surface pressure of the tooth side, and in particular the flexion in the root of the tooth). This is necessary for theoretically obtaining the curve of the load function, while for practical use it is sufficient to find only two points of the curve and to determine the value of the driving factor for those two points.

The initial basis for this calculation is the knowledge of the materials characteristics from which the gears are made:
- Fatigue strength for Hertz surface pressure $\sigma_{H\lim}$
- Static strength for Hertz surface pressure σ_{H0}
- Fatigue strength for strain at the root of the tooth $\sigma_{F\lim}$
- Static strength for strain at the root of the tooth σ_{F0}

Since this is a speed reducer that is in exploitation, whose scheme is shown in Figure 2, in addition to these data are also known the type of stresses, working conditions, shape and dimensions of the gears, etc., which can determine the value of the static and fatigue capacity.

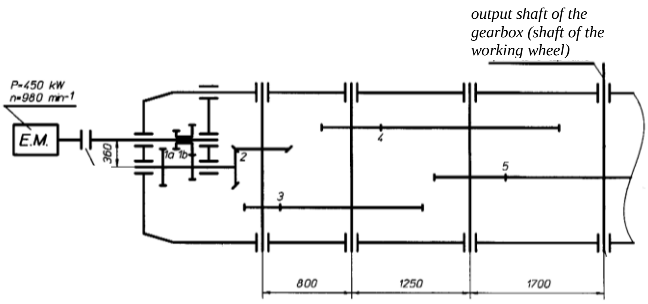

Fig. 2 *Sheme of the gearbox speed reducer with the working wheel of the excavator SRs-630*

The practice showed that in the work of the speed reducer, the weakness of the first, second and last gear pair was observed (the gear pair 1, the gear pair 2 and the gear pair 5, marked in Figure 2). For the third and fourth gear pair, no problems were encountered in the speed reducer's current exploitation life, so they are practically working as delivered by the manufacturer. Because of the above, only the first, second and last gear pair will be treated.

The first gear pair is double, its driving gear is made of two cylindrical gears with slanted teeth made on a common carrier that fit together with two following gears respectively, whereof the following gears are mounted on the same shaft. Depending on which part of the pair is coupled, two speeds can be achieved by rotating the output shaft for one speed at the input shaft and thus two gear ratio of the speed reducer. In practice, the excavator operates with the higher speed, so for this reason only the right part of the gear pair is analyzed in the paper (gear pair 1b).

The second gear pair is conical with curved teeth (conical helical gear pair), and the last gear pair is a cylindrical with a arrow teeth (herring-bone gear pair).

Data on the characteristics of the materials from which the gear pairs are made are used by the theoretical data from the excavator manufacturer, TAKRAF-Germany.

3.2. Nominal loading of the speed reducer

The nominal loading of the speed reducer defines the average load during a lasting drive.

The nominal load is calculated as the average load for all working regimes of the excavator (for which measurements were made), with the exception of the extreme values achieved during the experimental measurements. The indicator of the load is taken showing of the ammeter during the measurement (the measurement methodology is not subject to analysis in this paper). The average value of the electric current during the experimental measurements was:

$$I_{sr} = \frac{40+48+30+40+24+32+30+60}{8} = 38 \ \text{A}$$

The power corresponding to this load is rated (nominal) power, so on the output shaft of the speed reducer has a value

$$P_R = (\sqrt{3}\cdot 6\cdot 38\cdot 0{,}86)\cdot 0{,}97\cdot 0{,}92 = 303{,}08 \quad \text{kW}$$

and the nominal torque of the output shaft is:

$$T_{20} = 159155\cdot\frac{P_R}{n} = 159155\cdot\frac{303{,}08}{0{,}11885} = 405858445{,}3 \ \text{Nmm}$$

3.3. Calculation of the carrying capacity of the gear pairs 1b and 5

3.3.1. Carrying capacity of surface pressure

a) Fatigue carrying capacity

- Maximum peripheral force on the tooth side

$$F_{tH\infty} = \frac{b\cdot d}{S_{H\lim}}\cdot\frac{u}{u+1}\cdot\frac{(Z_x\cdot Z_L\cdot Z_R\cdot Z_v\cdot Z_N\cdot Z_w\cdot Z_{Eht})^2}{K_v\cdot K_{H\alpha}\cdot K_{H\beta}\cdot Z_\varepsilon{}^2\cdot Z_E{}^2\cdot Z_{H\beta}{}^2}\cdot\sigma_{H\lim}{}^2$$

- Maximum peripheral torque from the peripheral force

$$T_{H\infty} = \frac{F_{tH\infty}\cdot d}{2}$$

b) Static carrying capacity

- Maximum peripheral force on the tooth side

$$F_{tH0} = \frac{b\cdot d}{S_{H\lim}}\cdot\frac{u}{u+1}\cdot\frac{(Z_x\cdot Z_L\cdot Z_R\cdot Z_v\cdot Z_N\cdot Z_w\cdot Z_{Eht})^2}{K_v\cdot K_{H\alpha}\cdot K_{H\beta}\cdot Z_\varepsilon{}^2\cdot Z_E{}^2\cdot Z_{H\beta}{}^2}\cdot\sigma_{H0}{}^2$$

- Maximum peripheral torque from the peripheral force

$$T_{H0} = \frac{F_{tH0}\cdot d}{2}$$

c) Carrying capacity at N = 10^4 load changes

$$T_{H(10^4)} = \sqrt{T_{H\infty}\cdot T_{H0}}$$

3.3.2. Carrying capacity of flexion in the root of the tooth

a) Fatigue carrying capacity

- Maximum peripheral force at the root of the tooth

$$F_{tH\infty} = \frac{b\cdot d\cdot\cos\beta}{z}\cdot\frac{\sigma_{F\lim}}{S_{F\lim}}\cdot\frac{Y_x\cdot Y_{ST}\cdot Y_N\cdot Y_\delta\cdot Y_R\cdot Y_{Eht}}{K_v\cdot K_{F\alpha}\cdot K_{F\beta}\cdot Y_{FS}\cdot Y_\varepsilon\cdot Y_\beta}$$

- Maximum peripheral torque from the peripheral force

$$T_{F\infty} = \frac{F_{tF\infty}\cdot d}{2}$$

b) Static carrying capacity

- Maximum peripheral force at the root of the tooth

$$F_{tF0} = \frac{b \cdot d \cdot \cos\beta}{z} \cdot \frac{\sigma_{F0}}{S_{F\lim}} \cdot \frac{Y_x \cdot Y_{ST} \cdot Y_N \cdot Y_\delta \cdot Y_R \cdot Y_{Eht}}{K_v \cdot K_{F\alpha} \cdot K_{F\beta} \cdot Y_{FS} \cdot Y_\varepsilon \cdot Y_\beta}$$

- Maximum peripheral torque from the peripheral force

$$T_{F0} = \frac{F_{tF0} \cdot d}{2}$$

c) Carrying capacity at N = 10^4 load changes

$$T_{F(10^4)} = \sqrt{T_{F\infty} \cdot T_{F0}}$$

The influential factors in the preceding formulas are determined by an appropriate methodology for this purpose, for gears of the gear pairs 1b and 5.

In order to cover the remaining impacts on the work of gears, which are not yet covered by their own factor, the factor of safety is still being used. Because the maximum potential for loadings on the gears (maximum carrying capacity) should be determined here, its minimum values are taken for the factors of safety: $S_{H\lim} = 1$ and $S_{F\lim} = 1.2$.

According to the known data for the gears and the formulas of items 3.3.1. and 3.3.2. the carrying capacity of the gear pairs 1b and 5 are calculated and given in Table 1 for the gear pair 1b and table 2 for the gear pair 5.

Table1

carrying capacity		gear pair 1 b	
		$z_1 = 56$	$z_2 = 45$
$F_{tH\sim}$	[N]	127164.2	103166.1
$T_{H\sim}$	[Nmm]	25390875.81	16553000.74
F_{tH0}	[N]	5208646.8	3337790.79
T_{H0}	[Nmm]	1040010506	535548533.5
$T_{H(10^4)}$	[Nmm]	162501623.3	94153785.21
$F_{tF\sim}$	[N]	127918.8	135094.1
$T_{F\sim}$	[Nmm]	25542546.8	21675848.3
F_{tF0}	[N]	447715.7	472829.2
T_{F0}	[Nmm]	89395392.8	75865443.1
$T_{F(10^4)}$	[Nmm]	47783853	40551791.9

3.4. Calculation of the carrying capacity of the gear pair 2

3.4.1. Carrying capacity of surface pressure

a) Fatigue carrying capacity

- Maximum peripheral force on the tooth side

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

where $d_m = d - b\sin\varphi$ and it is a diameter of the pitch circle

- Maximum peripheral torque from the peripheral force

$$T_{H\infty} = \frac{F_{tH\infty} \cdot d_m}{2}$$

Table 2

carrying capacity		gear pair 5	
		x 2 $z_1 = 23$	x 2 $z_2 = 98$
$F_{tH\sim}$	[N]	447077.43	2198172.23
$T_{H\sim}$	[Nmm]	143210077.3	3000186363
F_{tH0}	[N]	16094788.15	54954306.2
T_{H0}	[Nmm]	5155563010	75004659000
$T_{H(10^4)}$	[Nmm]	859260482.6	15000931000
$F_{tF\sim}$	[N]	892999.97	819305.64
$T_{F\sim}$	[Nmm]	286050216.6	1118233401
F_{tF0}	[N]	3162708.6	3098378.6
T_{F0}	[Nmm]	1013094644	4228837513
$T_{F(10^4)}$	[Nmm]	5383226984.5	21745867708

Note for Table 2: The listed carrying capacity for gear pair 5 in the table refers for half gear. For whole herring-bone gear the given values are multiplied by 2 and the carrying capacity is obtained with respect to surface pressure and flexion.

b) Static carrying capacity

- Maximum peripheral force on the tooth side

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

- Maximum peripheral torque from the peripheral force

$$T_{H0} = \frac{F_{tH0} \cdot d_m}{2}$$

c) Carrying capacity at N = 10^4 load changes

$$T_{H(10^4)} = \sqrt{T_{H\infty} \cdot T_{H0}}$$

3.4.2. Carrying capacity of flexion in the root of the tooth

a) Fatigue carrying capacity

- Maximum peripheral force at the root of the tooth

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

- Maximum peripheral torque from the peripheral force

$$T_{F\infty} = \frac{F_{tF\infty} \cdot d_m}{2}$$

b) Static carrying capacity

- Maximum peripheral force at the root of the tooth

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

- Maximum peripheral torque from the peripheral force

$$T_{F0} = \frac{F_{tF0} \cdot d_m}{2}$$

c) Carrying capacity at N = 10^4 changes

$$T_{F(10^4)} = \sqrt{T_{F\infty} \cdot T_{F0}}$$

The influential factors listed in the preceding formulas are determined for the gears of the gear pair 2. For the factors of safety, it is taken: $S_{H\lim} = 1.2$ and $S_{F\lim} = 1.4$.

According to the specified data for the gears and the formulas of items 3.4.1. and 3.4.2. the carrying capacity of the gear pair 2 is calculated and given in Table 3.

Table 3

carrying capacity	gear pair 2	
	$z_1 = 29$	$z_2 = 38$
$F_{tH\sim}$ [N]	2667868701	3354330573
$T_{H\sim}$ [Nmm]	375996070000	619460990000
F_{tH0} [N]	58100247000	64939839000
T_{H0} [Nmm]	9422117200000	13799715000000
$T_{H(10^4)}$ [Nmm]	1882200500000	2923762000000
$F_{tF\sim}$ [N]	41943.4	42162.1
$T_{F\sim}$ [Nmm]	5911293.1	7786285.8
F_{tF0} [N]	114924.9	115524.1
T_{F0} [Nmm]	18637382.9	24548855.1
$T_{F(10^4)}$ [Nmm]	10496239	13825498.3

4. Driving factor

4.1. Standardized load numbers

$$a_{(10^0)} = \frac{T_{2(10^0)}}{T_{20}} = \frac{1286377308}{405858445{,}3} = 3{,}17$$

$$a_{(10^4)} = \frac{T_{2(10^4)}}{T_{20}} = \frac{624750000}{405858445{,}3} = 1{,}54$$

$$a_{(10^7)} = \frac{T_{2(10^7)}}{T_{20}} = \frac{333200000}{405858445{,}3} = 0{,}82$$

$A_1 = 4$ $A_2 = 1.6$ $A_3 = 1$

The magnitudes $T_{2(10^0)}$, $T_{2(10^4)}$, $T_{2(10^7}$ are read from the diagram in Figure 1 and the magnitudes T_{20} is from item 3.2.

4.2. Standardized carrying capacity numbers

$$b_{(10^0)} = \frac{T_{(10^0)}}{T_{F\infty}} = \frac{4228837513}{1118233401} = 3{,}78$$

$$b_{(10^4)} = \sqrt{b_{(10^0)}} = 1{,}94$$

$$b_{(10^7)} = 1$$

$B_1 = 3.15$ $B_2 = 1.8$ $B_3 = 1$

$T_{(10^0)} = T_{F0}$, the magnitudes T_{F0} и $T_{F\infty}$ are from Table 2 for the following gear of the gear pair 5.

Because the carrying capacity of flexion in the root of the tooth is less favorable (smaller), it is considered the most relevant when defining the standardized carrying capacity numbers.

4.3. Value of the driving factor

K_A = max [A_1/B_1, A_2/B_2, A_3/B_3] = max [4/3.15; 1.6/1.8; 1/1] = max [1.27; 0.89; 1] = 1.27

It is adopted $K_A = 1.25$ for the value of the driving factor of the speed reducer, which is the number from the standard array R20.

5. Conclusion

In this paper is shown the load function of the working wheel speed reducer for a real excavator and known working regimes, which is a function derived from the experimental measurements carried out for that purpose. The load function gives a description of the speed reducer's load regime in its total lifetime.

The paper presents a methodology for theoretical calculation of the carrying capacity of the speed reducers, that is, for its gears, both fatigue and static carrying capacity, for surface pressure on the tooth side and flexion in the root of the tooth. The presented methodology is confirmed by analysis of the specific reducer. Also this paper presents a concept for determining the value of the driving factor of the speed reducer as an important characteristic of its dynamic load, and defines the value of that factor on the basis of this concept for the mentioned speed reducer.

The gain value for the driving factor can serve as a benchmark for the driving factors of the rotating excavators with similar technical characteristics operating in approximate working conditions.

6. References

Hristovska E.: Own papers with this topic and data from experimental measurement

Manual handling for the excavator SRs-630 and other technical documentation from the manufacturer TAKRAF-Germany

Numerous documentation for modifications and repairs of gearboxes from the coalmine Suvodol-Bitola.