THE VERIFICATION BY CALCULUS OF THE TRANSMISSION OF THE EXTRACTION MACHINE'S REDUCER FROM THE OLD PIT WITH SKIP – LONEA MINIG PLANT

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Abstract: *The extraction machine includes as components a cylindrical reducer with two steps, between the two electrical engines of 500 kW and the drive wheel. At the level of the revolution gear of the extraction machine, in time, damage to the gears' teeth has appeared, a phenomenon which manifests continually, without having a specific cyclicality, but permanently evolving. The breaking of the teeth of the gear's gears in the extreme areas of the dentition – areas that before breaking had a wear accented by contact usury (pitting) - is a deterioration phenomenon that depends on many factors, such as: the material of the gears, the size and the geometrical and kinematic elements of the dentition, the characteristics of the flanks' surface, the mechanical solicitations, the oiling and the quality of the lubricant. These problems that appeared regarding the exploitation of the reducer of the extraction machine lead to the decrease of functioning safety, and even to its halting. In this paper, we present the calculus for verifying the transmission of the extraction machine's gearbox the conclusions that can be drawn from it.*

Keywords: Verification, Gearbox, Extraction machine.

Introduction

The extraction machine has, among its components, between the two electric engines and the driving wheel, a cilindric reducer (figures 1 and 2). The starting and the regulation of speed is done by coupling and uncoupling the benches of rotor resistance and at the positioning manoeuvres or at movements with reduced speeds (for the revision of the pit), we also use the manoeuvre mechanical brake. In order to reduce shocks, the main reducer (fig. 2) with 2 benches, 2 inputs and 1 output was fitted on sustaining arches (fig. 1). Between the reducer and the main shaft there is a fast-coupling with bolts. In order to oil the reducer, we use a pump with gears (the oiling of the last bench is done supplimentary by barbotage as well).

Fig. 1. The extraction machine

Fig. 2. The reducer of the extraction machine

The reducerof the extraction machine

The type of the reducer is 2TD-14 with a transmission report of 6, the reducer's mass (without oil) is of 16060 kg, the quantity of oil in the reducer is of 600 litres, and the maximum temperature of the oil is 60 degrees Celsius. The maximum number of rotations is of 750 rotations/min, the maximum moment on the main shaft is of 20 Nm, the rotation direction is reversible, and the other technical characteristics are shown in table 1. The shaft of the first bench coupled with the operating engine is tubular and, at the opposite end of the end coupled with the engine, has a dentured coupling, which allows the reducer to rotate around the main shaft that constitutes the link with the driving wheel.

The main dimensions of the reducer (figure 3) are shown in table 2.

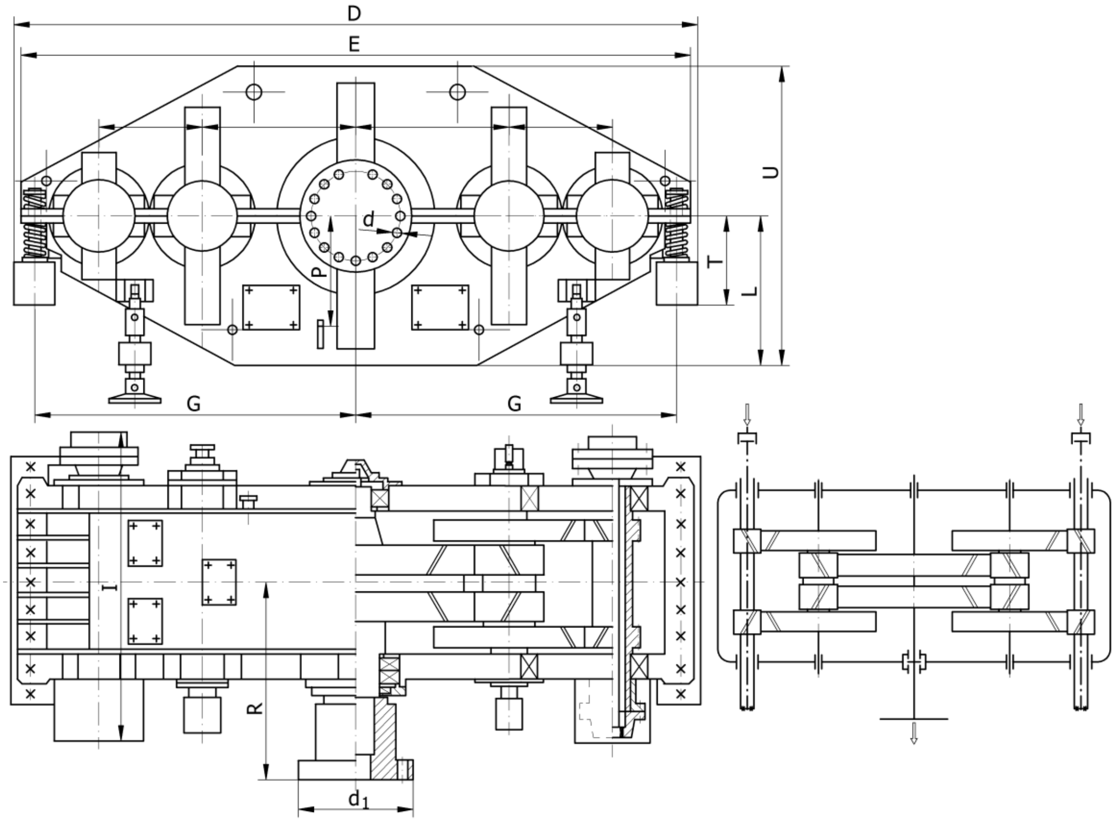

Fig. 3. The construction and the dimensions of the extraction machine's reducer.

Table 1. The reducer's technical characteristics.

Denomination	Value	Denomination	Value
Bench I		Bench II	
- The module	6	- The module	8
- Number of teeth of leading wheel	61	- Number of teeth of leading wheel	41
- Number of teeth of led wheel	113	- Number of teeth of led wheel	134
- Gradient	$29^0 32'30''$	- Gradient	$28^0 57'17''$
- Number of gears per bench	2	- Number of gears per bench	2
Number of electric engines of 500 kW	2	Nominal revolution of the electric engine	490rot/min
Electric engine's mass	6080 kg	Distance between the axis of the entrance shafts	2x1400
Impulse moment at the main shaft	$12\ t\ m^2$	Oiling gearings and bearings	spraying
The debit of the oiling pump	50 l/min	Number of sustaining spiral arches	2x4
Number of hydraulic shock absorbers	2x1	Arches' camber at fitting	20 mm

Table 2. The main dimensions of the reducer

Dimension	D	E	K	M	L	U	T	G	I	R	H	P	d	d1
Size [mm]	4000	3820	600	800	900	1790	755	1850	1060	1000	2190	560	85	760

Verifying by calculus the transmission of the reducer of the extraction machine

From the technical characteristics of the reducer that were shown, starting from the number of teeth, the normal module and the gradient of the denture, we have drawn the calculus breviary of the reducer's gearing. By calculating the geometrical elements of the denture, we verified the distance between the axis of the gears and the exterior diameter of the wheels, which coincide with the information contained in the technical documentation of the reducer. Choosing the gradient of the denture at the level of seconds (β_I = 29°32'30"; β_{II} = 28°57'17") is explained only by using it as a variable element for obtaining the whole values of the distances between the axis of the reducer which are 600 and 800 mm, respectively. Using such values leads to the difficult conditioning of the denture, from the point of view of the adjustment of the grinding machine and of the control of the denture, and as a result the inter-cheangeability of the gears is rendered impossible. Obtaining the contact spot which corresponds to the precission class 8 ((40x50%) between the flanks of the teeth that are in gearing, especially of the central wheel with its two pinions, is done by the wear-in of the gearing or of the reducer after assembling it on special stands. Because of this technical fabrication issue, we undertook a verification of the resistance of the denture at fatigue by contact pressure (S_H) and at fatigue by flexure at the base of the tooth (S_F) depending on the width of the gear (b) and the mode of variation of safety coefficients is shown in figure 4 for the first gearing 61/113, m = 6 mm; and in figure 5 for the second gearing 41/134, m = 8 mm.In the cyclogram of the loading the gearings, we have considered that in 5% of the functioning time of 10.000 hours, the gearing is overstrained up to 2 times (1000 kW), as in the case of starting under load, and the reducer functions with medium shocks on a mining machine with the capacity factor of 2.

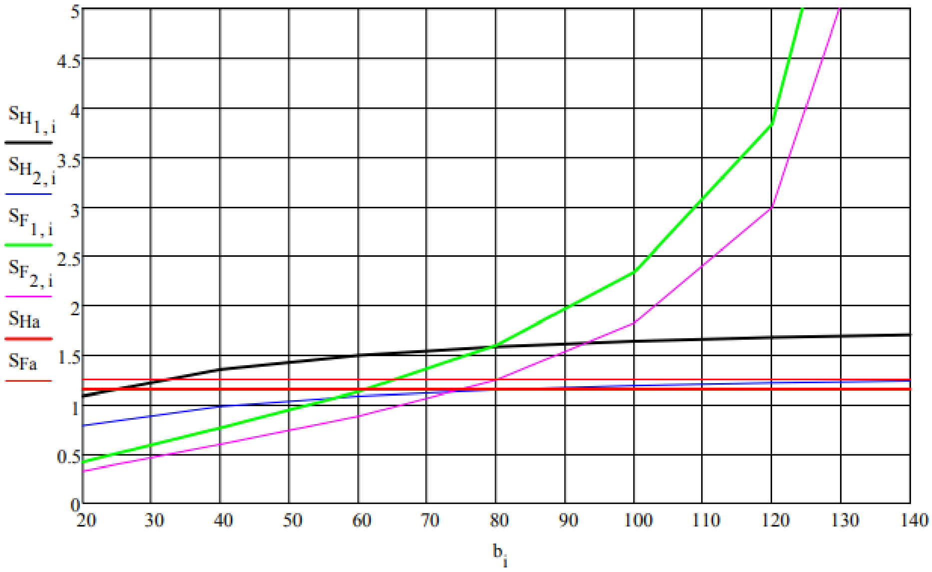

Fig 4. The variation mode of safety coefficients at fatigue by contact pressure and flexure at the base of the tooth, depending on the width of the wheel on which the contact at the first gearing is produced

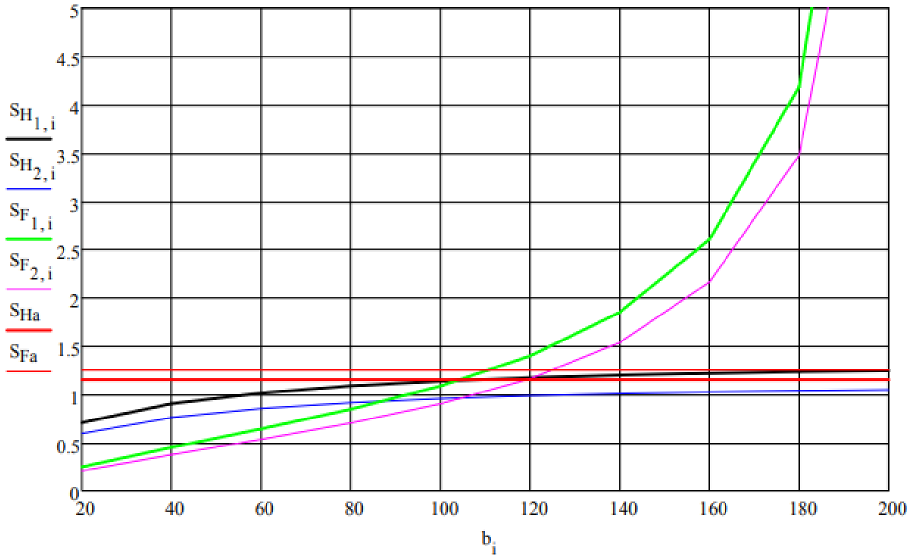

Fig. 5. The variation mode of safety coefficients at fatigue by contact pressure and flexure at the basys of the tooth, depending on the width of the wheel on which the contact at the second gearing is produced.

Verifyng by calculus the shafts of the reducer's transmission

Based on the general scheme of each shaft of the reducer, using the documentation provided by Lonea Mining Plant, we drew the calculus models for the three shafts in figures 6, 7 and 8.

The verification of the reducer's shafts was realised by the determination of reactions in the ball bearings, the flexure moments and the equivalent tensions in the critical sections of the shaft. We also verified the assambling by keying-on of the gear on the shaft, the durability of the bearings and the deformation of the shafts.

For a OLC 45 steel of producing the shaft, for an admissible tension of 150 N/mm^2 we obtain values of the safety coefficients greater than 9 (C_{saI} = 17,2; C_{saII} = 15,07; C_{saIII} = 9,26). In the case of the central shaft, the case of exiting the reducer, we considered the case of operating the reducer with a single engine, so that in the case of operating with both engines, the shaft is not solicited due to the balancing of forces through the central wheel.

The keying-on of gears on the shaft was verified at contact pressure, and the result was values of the safety coefficient greater that 3, and at shearing, where we obtained values of the safety coefficients that are greater than 11,8.

The minimal durability of the bearings on shaft II is 737.700 greater than the time length of 360.000 hours obtained for 50 years, 300 days per year and 24 hours per day of functioning of the reducer.

The camber of the shafts in different points is lower than the minimal admissible values f_{adm} = 0,06 mm (f_I = 0,007; f_{II} = 0,034; f_{III} = 0,014 mm).

The tolerance of the deviation of the tooth's direction pn the contact length of 160 mm (first bearing) and 230 mm (second bearing) at accuracy class 8 is of de F_β=0,040 mm and at class 7 of F_β=0,020 mm, STAS 6273-81, table11,and so the results are $tg\alpha_I$ = 2,5 10-4 and $tg\alpha_{II}$ = 1,7 10^{-4} for F_β=0,040 mm and $tg\alpha_I$ = 1,25 10^{-4} and $tg\alpha_{II}$ = 8,5 10^{-5} for F_β=0,020 mm. The permissible deviationof the angle of deflection of the shafts which is given in the speciality literature is $tg\alpha \leq 10^{-4}$ or $\alpha \leq 10^{-4}$ rad (Pavelescu,D.-M.O. page 269).The angles of deflection of shafts II and II in the area of the gears, point 2, is in a vertical plane of φ_{II} = 7,33 10^{-5} şi φ_{III} = 1,96 10^{-5}times closer to the admissible value of 10 10^{-5}m especially at shaft II, first bearing.

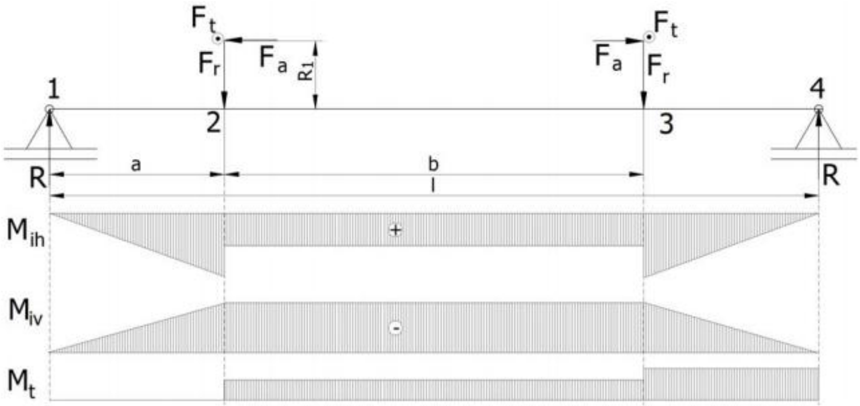

Fig. 6. Calculus model for the first shaft in the reducer.

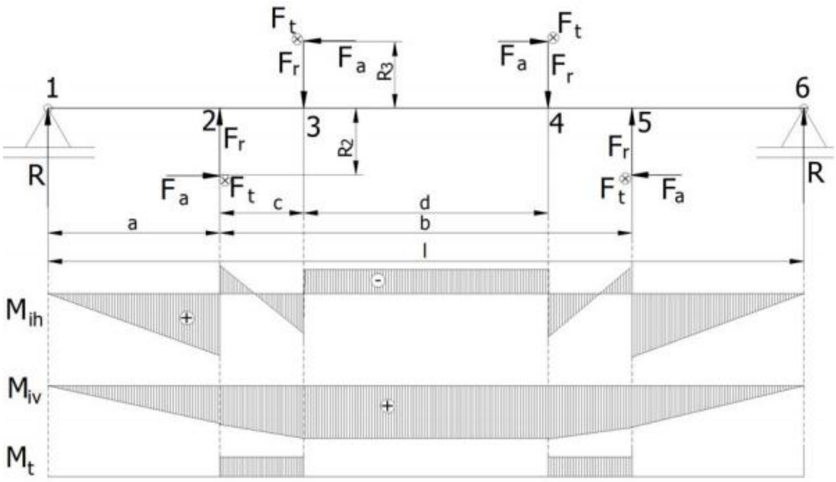

Fig. 7. Calculus model for the second shaft in the reducer.

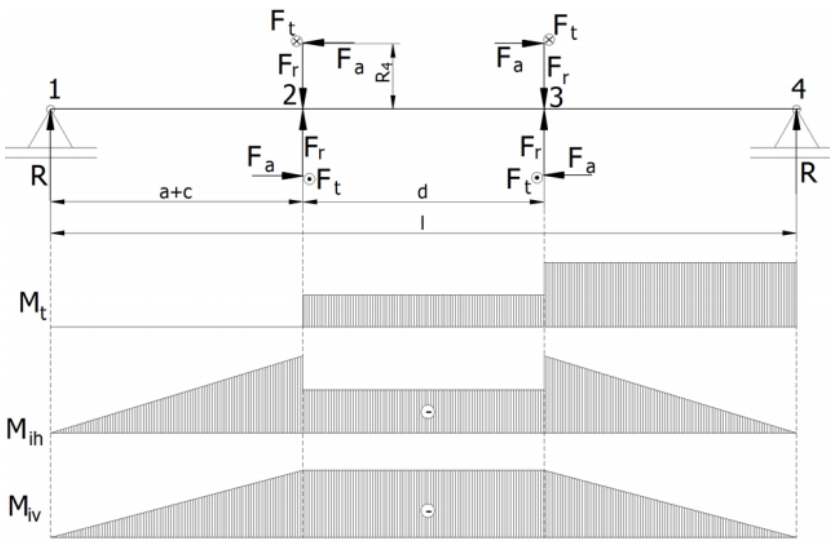

Fig. 8. Calculus model for the third shaft in the reducer.

Conclusions

From the graphics above and from the data in the calculus breviary we have the following results:

The variation of the safety coefficient regarding fatigue by contact pressure for blistered steel pinion $S_{H1}$and the gear made from enhancement alloy steel S_{H2} vary depending on the width of the denture in a reduced range;

The variation of the safety coefficient regarding fatigue by flexure at the base of the tooth for blistered steel pinion $S_{F1}$and the gear made from enhancement alloy steelS_{F2}vary depending on the width of the denture in a reduced range;

For denture widths of under 60%, 80 mm respectively 120 mm, the safety coefficient drops under the admissible value (S_{Ha} = 1,15; S_{Fa} = 1,25);

In order to confirm the calculus breviary of the gearings we must measure the hardness of the denture of the gears inside the reducer, which is hard to do without taking apart the reducer, and from the hardness measurements done on the denture of the gears that were changed, we obtained the results that the pinion on shaft I has the hardness of blistered steel, the pinion and the gear on pinion II have a denture hardness which is much lower, under 300 HB;

In the case of changing a gear in the reducer we must undertake a fast lapping of the gearing with an abrasive suspension by jointing the bolts, suspending the cables and using the brakes in order to obtain the braking moment until we obtain a contact spot which is appropiate for both rotation directions;

If we don't undertake a lapping of the reducer and the deformations of the shaft cumulate with the manufacture misalignments, the contact spot is reduced and it moves towards the extremity of the gear, causing the usury of the contact zone and even the fracture of teeth.

From what we have shown, we can deduce that theese reducers are unique and they must be changed completely when a defect appears in the gears transmission.This is to be done because the interchangeability of the gears is almost impossible to perform in the conditions in which the extraction machine is exploited.

References

[1] Pavelescu, D. and others. *Machine organs, vol. I,* Didactic and Pedagogic Press, Bucharest, 1985.

[2] Radu, D., *Strengh of materials, course and applications*, UNIVERSITAS Press, Petroşani, 2017.

[3] Radu, D., Tigae, I., Ridzi, M.C., *Collecting of strengh of materials, issues, vol.2,* UNIVERSITAS Press, Petroşani,

[4] *** *Acquisition order no.182/ 18.10.2013* issued by Mining Plant Lonea.

[5] *** *Technical documentation*, extraction installation from Mining Plant Lonea.