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VERIFICATION BY ANALYTICAL CALCULUS OF THE WINDING PULLEY SHAFT OF LONEA MINING PLANT NEW SHAFT WITH SKIP

IOSIF DUMITRESCU, VILHELM ITU, AND MIHAI CARMELO RIDZI

ABSTRACT. The pulley mounted in the tower at the winding installation with the entailing part of the winding cable on the soil, are winding pulleys. The winding pulleys are auxiliary installation devices with the role of supporting and guiding the winding cable on the vertical symmetry axis of the winding shaft. The pulley unit is made up of the pulley shaft and its bearings by which the unit is fixed to the winding tower structure, the mobile pulley and fixed pulley are mounted to the pulley shaft. The paper presents the verification calculus of the Lonea type winding shaft pulley for Lonea Mining Plant with new skip with analytical method.

1. INTRODUCTION

The extracting installation that operates on the new skip well from Lonea Mining Plant, is destined for the extraction from the underground of minerals. The extraction is done from the horizon +169,140 to the surface (the surface level is +704,5m; and the skip unloading level is +715,5m). The installation (Fig. 1) is ballanced and has an extracting machine type MK 5x2 equipped with two motors type P2S-1000-213-HUHLH/1998, of 1000 kW power and a nominal rpm of 54 rot/min.

FIGURE 1. Lonea Mining Plant New Shaft winding installation with skip

FIGURE 2. Winding pulley's platform

FIGURE 3. The metallic tower

The cables are wrapped around a moving wheel of Φ 5000 mm. The extracting cables with diameters of Φ 46,5 mm and a mass (on a linear meter) of 8,049 kg/m are wrapped around the two extracting pulleys of Φ 5000 mm with a mass (the pulley, the axel of the pulley and the bearing of the axel) of 5430 kg (Fig. 2), laying on the tower at a height of 47 m (pulley axel). The ballanced cables have a section of 135x20 mm and a mass (on

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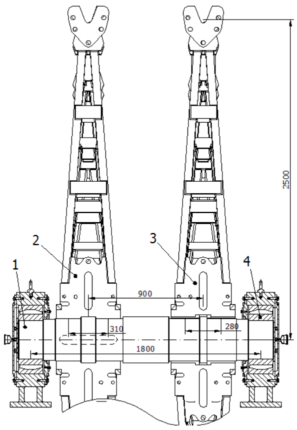

Figure 4. The pulley shaft

a linear meter) of 9,062 kg. The extracting vessels are skips having a mass (own mass, plus D.L.C., plus D.E.C. and suplimentary mass) of 21600 kg. and the effective load is 7000-8000 kg/skip. Another main component of the extracting installation is the metallic tower (Fig. 6) with a height until the pulley axel of 51 m.

The structure of the tower is composed of the extracting pulley platform sustained by the leading component and the one abutment set up as a frustum pyramid The extracting machine lies on the ground (at a height of 6,45 m to the 0 level of the well (well collar), sideways from the tower (well tower), at a distance (of the wheel axel), towards the vertical portion of the extracting cables which enter the well of 44m. The length of the cable chord (the distance between the tangent points of the cable to the deviating pulley from the tower and the wheel of the extracting machine, in the central position of the chord (perpendicular on the wheel axel)), is for the bottom branch L_{ci} = 52,91690824m, and L_{cs} = 59,30405431m for the top branch. The incline angles of the cables chords are $\beta_i = 43°43'31'', 37317428$ for the bottom branch and $\beta_s = 42°21'58'', 20794964$, for the top branch.

2. Shaft with pulleys

For Fig. 4 shows the pulley shaft, cross sectioning the separation plane of the semi pulleys, where: 1 – shaft; 2 – fixed pulley; 3 – mobile pulley; 4 – pulley shaft support bearings. The fixed pulley is positioned on the shaft through a shoulder and a parallel wedge, and the mobile pulley through a shoulder and two sliding bearings. Semi pulleys are assembled through cantering wedges, fastening screws and fastening rings. The mobile pulley has the role of reducing tensions in the cable and wear of the groove for the cable, due to different lengthening of the two cables when winding and unwinding on and from the drive roll of the winding engine.

3. Verification of shaft with pulleys

At the beginning of the year 2007, the shaft broke down at point 6, shown in Fig. 5. It is noticed that the break was initiated in the area of the wedge groove and was produced in time due to fatigue.

The design shape and size of the shaft are given in Fig. 5 and the mathematical model of loading the shaft in Fig. 6.

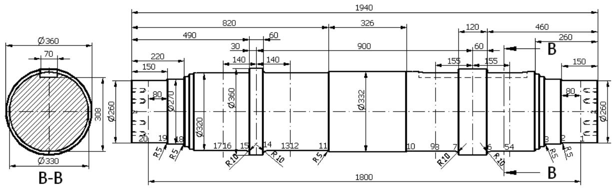

FIGURE 5. Constructive size of the shaft

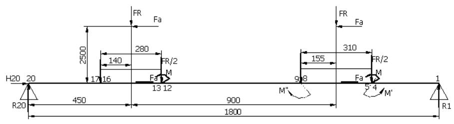

FIGURE 6. Mathematical model of loading of the shaft

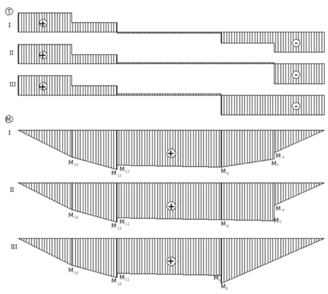

FIGURE 7. Shaft loading for the three cases

Each pulley is considered to rest on two planes of the shaft, with a force concentrated on the middle of the planes.

Fig. 6 shows the loading method of the shaft, and the dashed line shows the case where the fixed pulley does not stay on both the levels of the shaft, case 2 and 3.

Based on the mathematical model in Fig. 6, a shaft size verification programme for three work situation of the fixed pulley was devised in the 20 points on the shaft, for the three situations: I – when the fixed pulley rests on both planes; II – when the fixed pulley rests only on the plane at the end of the shaft; III – when the fixed pulley rests only on the plane at the middle of the shaft. The three work situations were given only for the fixed pulley, since the mobile one rests on the both planes of the shaft due to the wear of the sliding bearing lining.

For case I, when the fixed pulley rests on both planes (fig. 8, 9, 10, 11).

For case II, when the fixed pulley rests only on the plane at the end of the shaft (fig. 12, 13, 14, 15).

For case III, when the fixed pulley rests only on the plane at the middle of the shaft (fig. 16, 17, 18, 19).

4. CONCLUSIONS

Based on the mathematical model in Fig. 6, a shaft size verification programme for three work situation of the fixed pulley was devised.

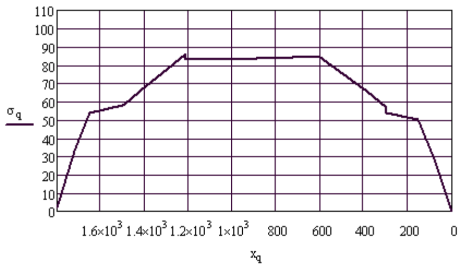

FIGURE 8. Maximum bending stress, N/mm^2

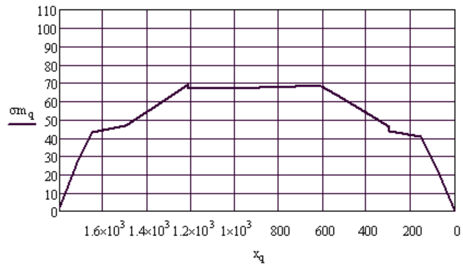

FIGURE 9. Minimum bending stress, N/mm^2

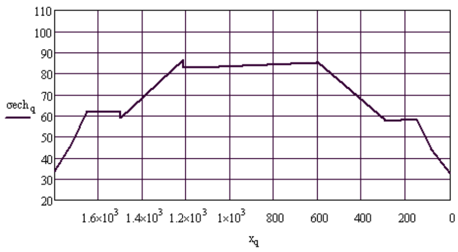

FIGURE 10. Equivalent stress, N/mm^2

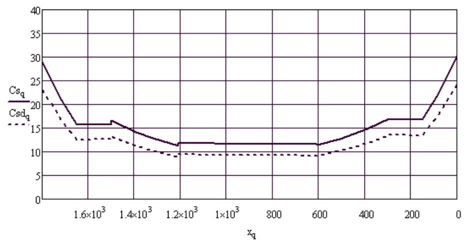

FIGURE 11. Safety coefficient in simple bending; as to C_{sdq} flow limit; as to C_{sq} resistance to breaking

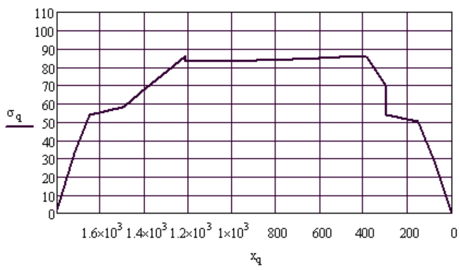

FIGURE 12. Maximum bending stress, N/mm^2

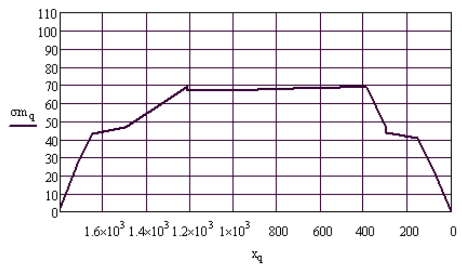

FIGURE 13. Minimum bending stress, N/mm^2

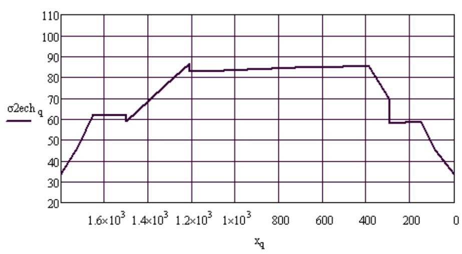

FIGURE 14. Equivalent stress, N/mm^2

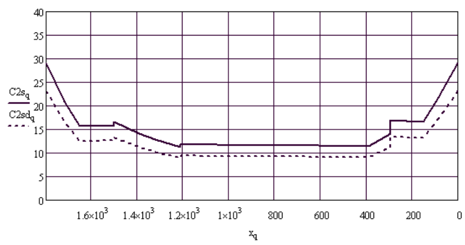

FIGURE 15. Safety coefficient in simple bending; as to C_{sdq} flow limit; as to C_{sq} resistance to breaking

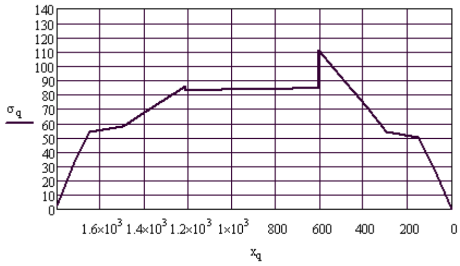

FIGURE 16. Maximum bending stress, N/mm^2

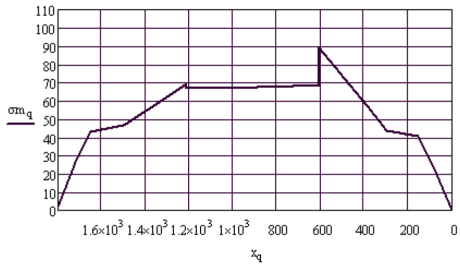

FIGURE 17. Minimum bending stress, N/mm^2

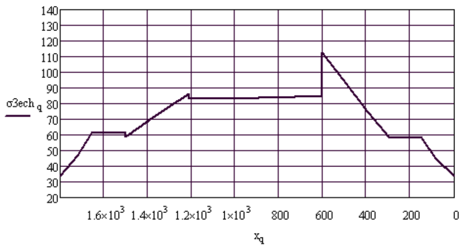

FIGURE 18. Equivalent stress, N/mm^2

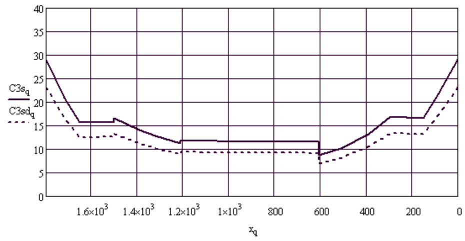

FIGURE 19. Safety coefficient in simple bending; as to C_{sdq} flow limit; as to C_{sq} resistance to breaking

Figures 11, 15 and 19 shows the variations of the (σr=980 MPa) safety coefficient compared to the break resistance of the shaft steel, 40Cr10, and the (σc=780 MPa) flow limit in the 20 points on the shaft, for the three situations: I – when the fixed pulley rests on both planes(Fig. 11); II – when the fixed pulley rests only on the plane at the end of the shaft (Fig. 15); III – when the fixed pulley rests only on the plane at the middle of the shaft (Fig. 19).

It is seen that in all the three situations the safety coefficient is higher than 10, except case III, when between points 6 and 7 a value less than 10 is noticed.

Even with these values of the safety coefficient, the shaft broke down at point 6, shown in Fig. 4.

It is noticed that the break was initiated in the area of the wedge groove and was produced in time due to fatigue.

In the area of the fixed pulley, tension concentrators, due to diameter leap(Φ330 to Φ360) of a 2,5 mm radius and wedge groove, overlap.

For the three possible cases of mounting the pulley on the shaft, with the help of Gh. Buzdugan's formula,

$$C_\sigma = \frac{1}{\sqrt{\left(\frac{\beta k_{\sigma 1}\cdot \beta k_{\sigma 2}}{\varepsilon_\sigma \cdot \gamma}\right)^2 \cdot \left(\frac{\sigma_a}{\sigma_{01}}\right)^2 + \left(\frac{\sigma_m}{\sigma_c}\right)^2}} \qquad (1)$$

safety coefficients to fatigue shown in Table 1 were determined.

In the execution of the two parts of the support plane of the pulley, two level adjustment of the tool should be made, these leading to size deviations of the two parts, even when shape and position deviations didn't exist.

Pulley mounting case	Point on the shaft	
	6	7
I	1,37	1,18
II	1,061	1,066
III	1,22	0,95

TABLE 1. Safety coefficients to fatigue

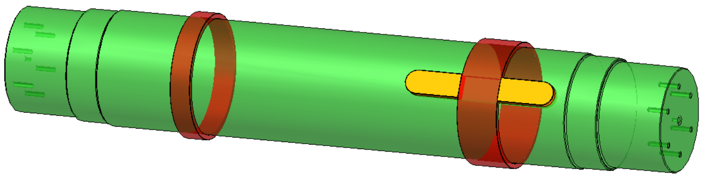

FIGURE 20. Pulleys' shaft proposed for Lonea Mining Plant

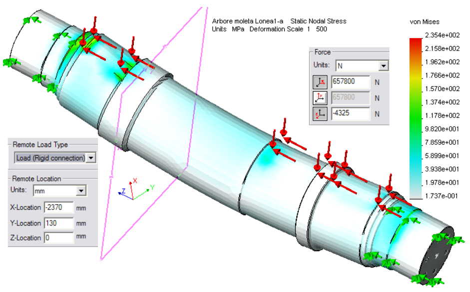

FIGURE 21. The pulley shaft stressed in the wedge canal on its opposite face on the level on the bearing sidet

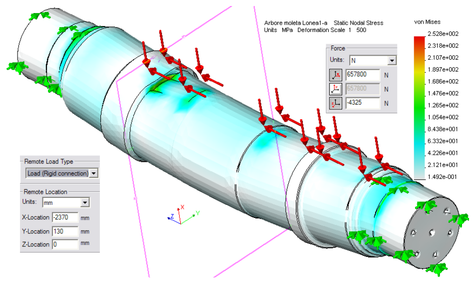

FIGURE 22. The pulley shaft stressed in the wedge canal plane on it opposite side on the inside level

These size deviations lead to fixing the pulley on only one side, its boring being made by only one pass, which brings an additional bending moment to the shaft, reducing the safety coefficient to fatigue even more, (less than 0,4).

The situation shown is confirmed by the use of a copper foil on one of the sides, secured with a bracelet against rejection.

Following the results of the dimensional verifications by analytical calculus constructive shaft solution was suggested, which is shown in Fig. 20.

The proposed shaft has symmetrical head pins and one central plane, allowing its easy and precise processing.

The mobile pulley's ring is tightened to the shaft by the two wedges, is mounted loosely.

Figs. 21 and 22 shows the analysis with finite elements (with the help of Cosmos Design Star), for the two cases. Case II was analyzed, when the fixed pulley is fixed only to the outside plane and case III – when the fixed pulley rests only on the plane at the middle of the shaft. Tthe equivalent stress of the shaft is highe, duet to the geometrical shape and stress concentrators.

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University of Petroșani
Department of Mechanical Engineering
Universității 20, 332006 Petroșani, România

University of Petroșani
Department of Mechanical Engineering
Universității 20, 332006 Petroșani, România

University of Petroșani
Department of Mechanical Engineering
Universității 20, 332006 Petroșani, România