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TESTS AND EXAMINATION OF DRIVE SHEAVE IN MINE SKIP HOIST

BADANIA I OCENA KONSTRUKCJI KOŁA PĘDNEGO MASZYNY WYCIĄGOWEJ GÓRNICZEGO WYCIĄGU SKIPOWEGO

This article describes a certain stage of work aimed at strength analysis of the structure of drive sheaves in mine shaft hoists. The work was initiated in connection with the emergence of cracks in those sheaves as a result of operation, and is an attempt to develop a new improved design of the drive sheave.

The first stage involved several series of strain gauge measurements of stress in the drive sheaves of machines exposed to their rated load. Measurement technology and results are described in detail in (Rokita & Wójcik, 2013). The results of those measurements were the basis for FEM stress analysis and discussion of the changes in the structure of those sheaves.

This article focuses on a description of the developed computational models for the currently used sheaves, and presents the results of strength calculations. As a result of those actions, a new drive sheave design was developed, for which a computational model was also prepared and FEM calculations were performed.

Comparison of the results of calculations for both sheave models is, therefore, also an assessment of the impact of the proposed changes to the design of the drive sheave on its strength and durability. It was decided to adopt the results of strength analysis of the sheave assembly currently in service as the baseline for this assessment. This required not only the performance of a strength analysis on the basis of a specially developed drive sheave calculation model, but also identification and assessment of the allowable stress values for the analysed structure.

Keywords: hoist, drive sheave, hoist tests

Niniejszy artykuł opisuje pewien etap prac mających na celu analizę wytrzymałościową konstrukcji kół pędnych maszyn wyciągowych górniczych wyciągów szybowych. Podjęcie tych prac związane było z pojawiającymi się pęknięciami tych kół w wyniku eksploatacji i próbą opracowania nowej poprawionej konstrukcji koła napędowego.

W ramach podjętych działań w pierwszym etapie przeprowadzono kilka serii pomiarów tensometrycznych naprężeń w kołach pędnych maszyn nominalnie obciążonych. Technologię pomiarów i ich wyniki opisano szerzej w (Rokita & Wójcik, 2013). Wyniki tych pomiarów stanowiły podstawę do analizy wytrzymałościowej metodą MES oraz dyskusji o zmianach w konstrukcji przedmiotowych kół.

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W niniejszym opracowaniu skoncentrowano się na opisie opracowanych modeli obliczeniowych aktualnie eksploatowanych kół oraz zaprezentowano wyniki obliczeń wytrzymałościowych. W wyniku tych działań powstał nowy projekt koła pędnego, dla którego również opracowano model obliczeniowy oraz wykonano obliczenia metodą MES.

Porównanie wyników obliczeń dla obu modeli koła jest więc jednocześnie oceną wpływu zaproponowanych zmian konstrukcji koła pędnego na poprawę jego wytrzymałości i trwałości. Za podstawę do tej oceny postanowiono przyjąć wyniki analizy wytrzymałościowej aktualnie eksploatowanej konstrukcji. Wymagało to nie tylko przeprowadzenia w oparciu o specjalnie opracowany model obliczeniowy koła pędnego analizy wytrzymałościowej ale również ustalenia i oceny dopuszczalnych wartości naprężeń dla analizowanej konstrukcji.

Słowa kluczowe: maszyna wyciągowa, koło pędne, badania maszyn wyciągowych

1. Introduction

Work aimed at increasing the durability and reliability of mine hoist assemblies has been conducted at the AGH University of Science and Technology for many years (Wolny, 2009, 2012; Płachno & Szczygieł, 2013). This article is limited only a description of the tests and upgrades to the design of a mine hoist drive sheave.

After several years of operating hoists in one of the mines, a few cracks were found in their drive sheaves. The photographic documentation shows that the cracks appeared in two areas. One of them is located on the side disc above the hub, where the sleeve for the screws retaining the two halves of the power transmission unit is mounted**.** The second is located on the contact point between the shield and the side disc with a rib closing the halves of the power transmission unit**.** The appearance of those cracks indicates fatigue. Most probably, they initiated in the joints located in those areas. Crack propagation proceed initially through the joint, and then in the material of the joined elements.

The photos below (Fig. 1 and 2) show examples of the areas of drive sheaves where cracks were found.

Fig. 1. Side disc cracks above the hub

Fig. 2. Crack on the contact point between the shield and the side disc

The first stage involved several series of strain gauge measurements of stress in the drive sheaves of machines exposed to their rated load. Measurement technology and results are described in detail in (Rokita & Wójcik, 2013). The results of those measurements were the basis for FEM stress analysis and discussion of the changes in the structure of those sheaves.

2. Tested object

Tests were performed on a drive sheave of a mine skip hoist. The hoist consists of four ropes and machinery located in a tower.

Below are selected technical parameters of the tested hoist.

- Hoisting height $H = 1053$ m,
- Hoisting velocity $v = 20$ m/s,
- Acceleration, delay $a = b = 1.20$ m/s^2,
- Skip capacity $M_p = 33000$ kg,
- Mass of empty skip with suspension $M_C = 37800$ kg,
- Drive sheave diameter $D_1 = 5500$ mm,
- Drive sheave rotation $n_1 = 69.4$ rotations/min,
- Drive sheave mass $G_1 = 33940$ kg,
- Machine shaft mass $G_2 = 19060$ kg,
- Moment of inertia of the sheave with shaft $J_2 = 196047$ kgm^2.

The machine with the above parameters was operated for 6 years and performed about 2 million load cycles during that period.

3. Adopted drive sheave calculation models

The strength analysis was carried out using the finite element method (FEM), based on numerical models developed on the basis of technical documentation of the device. The geometry of the structure in question was mapped using solid elements. In the final strength analysis, the results of calculations obtained on the basis of two main models, described as MOD-AE and MOD-PM, were used. The first of those models, MOD-AE, was developed on the basis of the technical documentation of hoisting machines currently in operation. The second model, MOD-PM, is based on the new documentation of a drive sheave. This documentation contains changes made in the design of the power transmission unit proposed by the author of this article. The proposed changes are the end result of partial calculations and analyses. Their purpose is to enhance the durability and reliability of the abovementioned hoists.

The calculation model MOD-PM included structural changes to such elements of the drive sheave as side discs, sheave shield, and internal ribbing of the sheave. Structural changes were to reduce stress concentrations in areas where the cracks occurred. In the side discs of the sheave, the transition from metal discs on the thicker middle ring, which is used to attach the sheave onto shaft flanges, were changed . In the drive sheave shield, a smooth transition along the thickness of the shied was also applied in locations where lining is present. The internal ribbing of the drive sheave for structure stiffening was adapted to the loads from the lining of the hoisting ropes exerting pressure on the shield.

Due to the symmetry of the hoist, the adopted computational model fully covers only half of the geometry. The impact of the rejected part on the model was replaced with appropriate bonding. The load on the models consisted of the impact of hoisting ropes, the weight of the modelled parts of the power transmission unit and shaft, and the motor rotor. The load on both models was identical as regards the value and the method of its application to the respective nodes and elements. The MOD-AE model and its selected elements are shown in Figures 3 and 4.

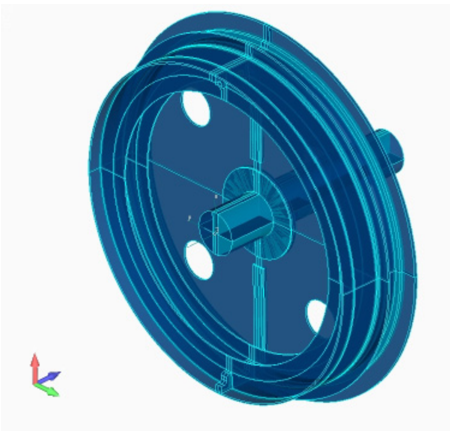

Fig. 3. View of MOD-AE model from the inside of the power transmission unit

Fig. 4. Fragment of MOD-AE model, including shaft, caps, and transverse ribs

4. Results of numerical calculations of stresses in drive sheave models

As a result of the calculations, information was obtained about the distribution and component values of the stress associated with the load applied to the models. Selected results obtained from the numerical calculations, relevant from the point of view of the strength analysis, are illustrated in Figure 5 and 6 for the MOD-AE model, and in Figure 7 to 10 for the MOD-PM model. In the form of contour plans, they show the distribution of reduced stresses σ_H (as per the Huber-von Mises hypothesis) and principal stresses – $\sigma_{\max}$, $\sigma_{\min}$.

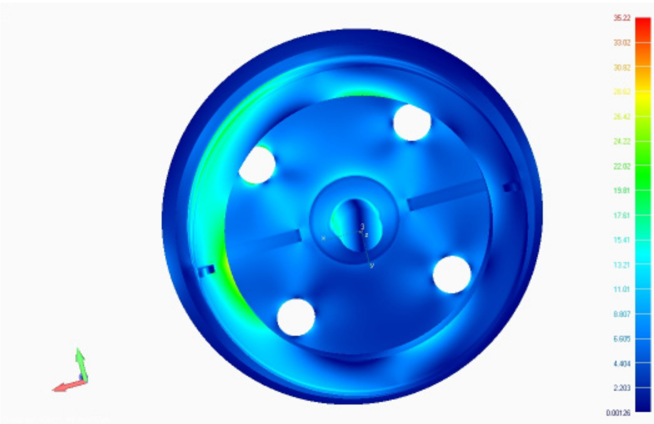

Fig. 5. Distribution of reduced stresses σ_H [MPa].
MOD-AE model, view from motor side

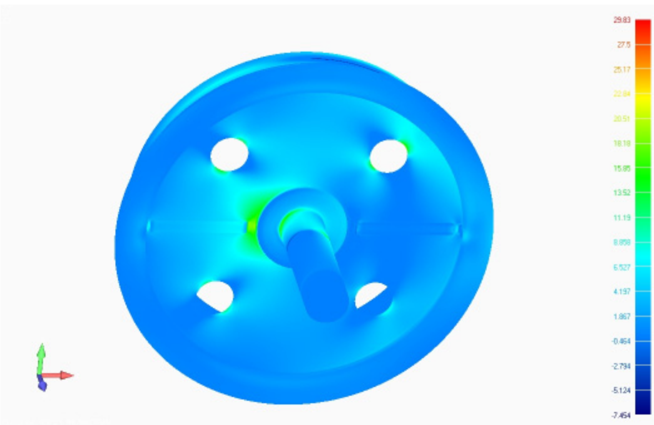

Fig. 6. Distribution of principal stresses $\sigma_{\max}$ [MPa].
MOD-AE model, view from motor side

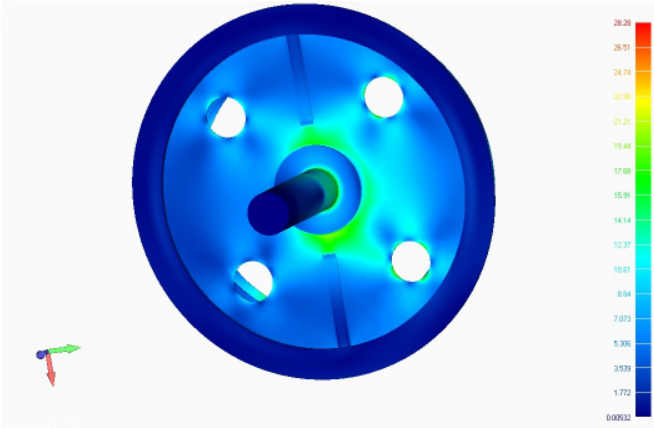

Fig. 7. Distribution of reduced stresses σ_H [MPa].
MOD-PM model, view from motor side

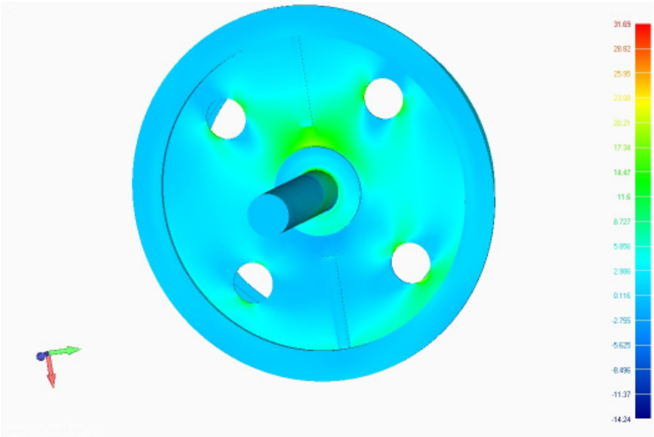

Fig. 8. Distribution of principal stresses $\sigma_{\max}$ [MPa].
MOD-PM model, view from motor side

In order to facilitate the final stress analysis, Tables 1 and 2, for the MOD-AE model and the MOD-PM model respectively, present the limit values of stresses that arise in their characteristic points (nodes) during one revolution of the shaft of the hoisting machine.

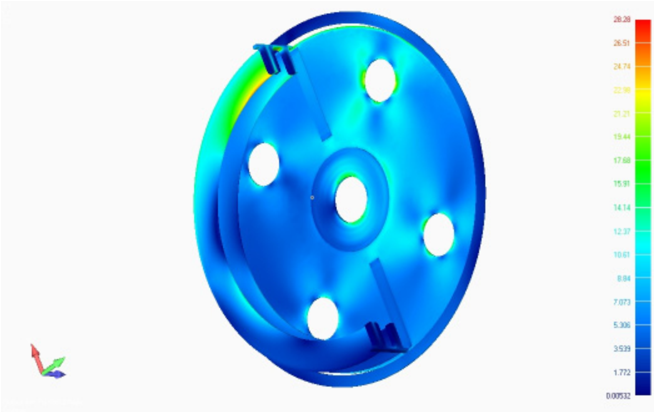

Fig. 9. Distribution of reduced stresses σ_H [MPa].
MOD-PM model, disc, rib, and rings

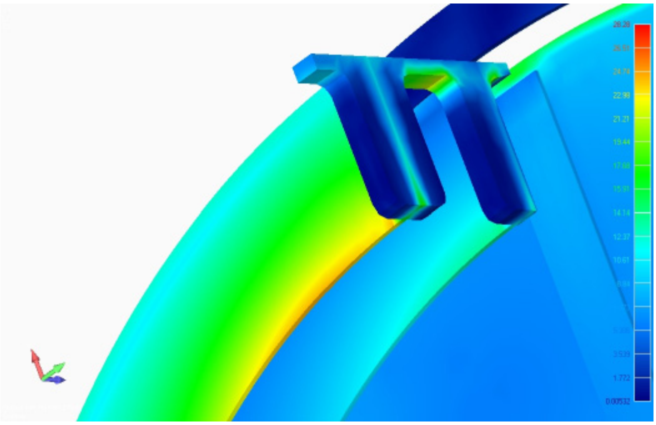

Fig. 10. Distribution of reduced stresses σ_H [MPa].
MOD-PM model, disc, rib, and rings

5. Assessment of the strength and durability of the analysed structures of drive sheaves

Tables 1 to 5 show a summary of the most important information about the state of stresses in the models of drive sheaves. They form the basis for the strength analysis and assessment of the design of power transmission unit. The values of stresses included in Tables 1 and 2 relate to two positions of the same point after revolution of the hoisting machine shaft by 180 degrees. The values given in Table 6 allow a comparison of the fatigue strength of those structures.

TABLE 1

Stress values at characteristic points of MOD-AE model

Element of the structure			Stress [MPa]		
			σ_H	σ_{max}	σ_{min}
Side discs near hub	Outer surface	Above shaft	21.6	–0.1	–22.3
		Below shaft	18.3	18.2	–0.0
	Inner surface	Above shaft	10.1	4.3	–7.3
		Below shaft	9.8	6.8	–4.4
Shield	Outer surface	Above shaft	27.4	29.6	–1.8
		Below shaft	11.0	0.5	–11.7
	Inner surface	Above shaft	25.6	0.6	–25.8
		Below shaft	9.6	8.4	–2.1
Middle ring under rope		Above shaft	27.5	0.0	–27.6
		Below shaft	9.9	9.9	0.0
Transverse rib		Above shaft	35.2	3.6	–34.0
		Below shaft	21.5	20.7	–2.2

TABLE 2

Stress values at characteristic points of MOD-PM model

Element of the structure			Stress [MPa]		
			σ_H	σ_{max}	σ_{min}
Side discs near hub	Outer surface	Above shaft	18.0	–1.7	–19.9
		Below shaft	15.1	15.1	–0.7
	Inner surface	Above shaft	11.3	4.6	–8.0
		Below shaft	10.6	7.1	–4.8
Shield	Outer surface	Above shaft	25.4	30.8	1.7
		Below shaft	12.7	–1.0	–15.4
	Inner surface	Above shaft	27.8	–4.9	–33.8
		Below shaft	14.4	15.1	–0.4
Middle ring under rope		Above shaft	24.2	–0.2	–24.5
		Below shaft	7.2	7.4	0.0
Transverse rib		Above shaft	21.8	31.4	7.6
		Below shaft	7.2	–1.6	–9.7

Table 3 presents extreme stress values that were obtained from calculations for the model of the sheave currently in use (MOD-AE) and for the model of the new design of the drive sheave (MOD-PM).

TABLE 3

Extreme stress values obtained in calculations for the models

Model	Stress [MPa]		
	σ_H	σ_{max}	σ_{min}
MOD-AE	35.2	29.6	–34.0
MOD-PM	27.8	31.4	–33.8

Table 4 and 5 contain extreme stress values at those points of the models the old and the new structure in which the range of variation reaches maximum values.

Based on the stress values listed in the tables, conclusions may be drawn about the fatigue strength (durability) of the analysed structures of drive sheaves.

TABLE 4

Limit values of the stress variability range in characteristic points of the MOD-AE model

Element of the structure		**Stress [MPa]**			**Location of the point**
		σ_{max}	σ_{min}	$\Delta\sigma = \sigma_{max} - \sigma_{min}$	
Side discs near hub	Outer surface	18.2	–22.3	40.5	Weld
	Inner surface	6.8	–4.4	11.2	Native material
Shield	Outer surface	29.6	–11.7	41.3	Native material
	Inner surface	8.4	–25.8	34.2	Native material
Middle ring under rope		9.9	–27.6	37.5	Native material
Transverse rib		20.7	–34.0	51.1	Weld

TABLE 5

Limit values of the stress variability range in characteristic points of the MOD-PM model

Element of the structure		**Stress [MPa]**			**Location of the point**
		σ_{max}	σ_{min}	$\Delta\sigma = \sigma_{max} - \sigma_{min}$	
Side discs near hub	Outer surface	15.1	–19.9	35.0	Weld
	Inner surface	7.1	–8.0	15.1	Native material
Shield	Outer surface	30.8	–15.4	46.2	Native material
	Inner surface	15.1	–33.8	48.9	Native material
Middle ring under rope		7.4	–24.5	31.9	Native material
Transverse rib		31.4	–9.7	41.1	Native material

TABLE 6

Maximum values of the stress variability range in the models

Model	**Stress $\Delta\sigma = \sigma_{max} - \sigma_{min}$ [MPa]**	**Location of the point**
MOD-AE	51.0	Weld
	41.3	Native material
MOD-PM	35.0	Weld
	48.9	Native material

The maximum value of stress variation in Table 6 indicate that the stress in the welds of the newly designed drive sheave has significantly decreased, from 51.0 MPa to 35.0 MPa, while the stress in the native material of the sheaves has increased slightly, from 41.3 MPa to 48.9 MPa.

6. Summary and final conclusions

According to the information in the Polish Standard PN-90/B-03200 Steel structures. Static calculations and design (Table Z3-2), in the case of using "tee and cross connections – other joints sized for full load bearing capacity of the cross-section", the following fatigue strength should be adopted:

- $\Delta\sigma_R$ = 37 MPa for the number of cycles $N = 10^7$,
- $\Delta\sigma_R$ = 23 MPa for the number of cycles $N = 10^8$.

In the case of native material, "non-welded elements with openings for fasteners", the following fatigue strength may be adopted respectively:

- $\Delta\sigma_R$ = 90 MPa for the number of cycles $N = 10^7$,
- $\Delta\sigma_R$ = 57 MPa for the number of cycles $N = 10^8$.

It was assumed that the hoisting machine works 20 hours a day, six days a week, hence the estimated number of load cycles for the machine may be as follows:

$$N = 2 \times 10^7$$

The strength analyses conducted demonstrate that the calculated value of stresses in the joint connecting a transverse rib with the shields exceeds the fatigue limit value $\Delta\sigma_r$ = 37 MPa after the hoisting machines currently in use reach the following number of load cycles: $N = 10^7$.

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