STRENGTH ANALYSIS OF TYPICAL CONSTRUCTIONS OF KOEPE PULLEYS APPLIED IN MINING HOISTING INSTALLATIONS

Stanisław WOLNY, Bogusław ŁADECKI

AGH University of Science and Technology
Faculty of Mechanical Engineering and Robotics
Al. Mickiewicza 30, 30-059 Kraków, Poland
e-mail: {stwolny, boglad}@agh.edu.pl

A large number of hoisting installations with machines equipped with Koepe pulleys is presently operating in the Polish mines. In the construction of these Koepe pulleys an occurrence of cracks of a fatigue character was found. Running repairs performed in areas of the cracks occurrences only temporarily limited this effect. In order to eliminate this problem, endeavours were undertaken to identify sources of forming and growing of fatigue cracks of Koepe pulleys. With this aim the numerical models of the Koepe pulleys construction were developed and their analysis performed by the finite element method. The FEM was preceded by the dynamic analysis of the hoisting installation during its normal work cycle. The results of these analyses – dynamic and strength – were verified by experiments on the real object, by measuring forces and stresses in the selected areas of this wheel construction. The obtained results provided the grounds for assessing the fatigue life and for the development of the reconstruction concept of the considered constructions of Koepe pulleys.

Key words : Koepe pulley of hoisting installations, strength, material fatigue, experimental tests, non-destructive tests, FEM

1. INTRODUCTION

Presently the Polish mining utilises a significant number of shaft hoists, which operation period exceeds 30 years. These machines are usually equipped either in Koepe pulleys with diaphragms of a shape of U letter, or with two circumferential rings stiffening the mantel. Numerous cracks of a fatigue character are observed in such wheels [1,2,3], and periodically performed repairs do not eliminate sources of cracks reoccurring in the same structure areas. Fast growing fatigue failures and the necessity of often repairs essentially limit safety and efficiency of hoisting installations. The results of non-destructive dynamic and strength analyses, verified by measuring deformations on real objects are presented in the hereby paper. Investigations were aimed at explaining the reasons of forming Koepe pulleys defects and at proposing such changes in their construction which would eliminate reasons of cracks formations.

2. DYNAMIC LOAD ACTING ON A KOEPE PULLEY UNDER THE TYPICAL OPERATING CONDITIONS

The real loads acting on the pulley drum are obtained from the dynamic analysis of its operation throughout the typical duty cycle [4]. Since tower-type winding installations with pulley blocks are now in widespread use in Polish collieries, this analysis shall be confined to a this type of hoisting installation, shown schematically in Fig 1.

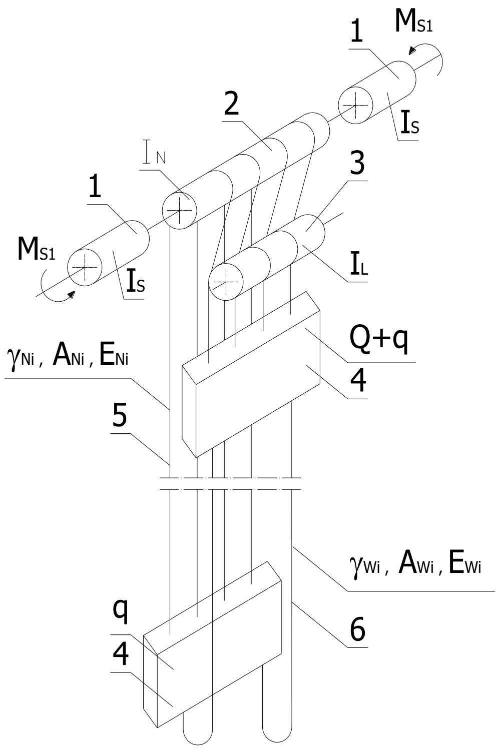


Fig. 1. Schematic diagram of the hoisting installations: 1 – low speed dc motor, 2 – multi-rope pulley block with the diameter D and inertia moment I_N, 3 – baffle wheel assembly, 4 – skips (conveyance) with mass q and loading capacity Q, 5 – branches of parallel – arranged hoisting ropes with the density γ_N and tensile rigidity $A_{Ni}E_{Ni}$, 6 - branches of parallel – arranged balance ropes with the density γ_{Wi} and tensile rigidity $A_{Wi}E_{Wi}$.

Results of dynamic analyses [4] and measurements taken on a real object [5,6] provide the backgrounds for deriving analytical formulas to compute the maximal and minimal loads transmitted from the hoisting ropes onto the drum in the winding gear.

The maximal loads are transmitted from hoisting ropes onto the drum at the instant a full conveyance begins to be hoisted from the bottom level and are given as [4]:

$$S_{\max} = S_{St}^{R} + \Delta S^{R} \tag{1}$$

where:

S_{St}^{R} - maximal static load transmitted from the rope onto the Koepe pulley in the start-up phase,

$$\Delta S^{R} = A_N E_N \frac{a_1}{a_N} \frac{8 l_N}{a_N} \left[1 - e^{-k}\right] \frac{1}{k}, \tag{2}$$

$$k = \frac{A_N E_N}{M_2 a_N};\ M_2 = Q + q,$$

$$\sum_{i=1}^{n} A_{Ni} E_{Ni} = A_N E_N \ ,$$

n - number of hoisting ropes,

a_1 – acceleration in the start-up chase,

a_N – velocity of elastic wave propagation in hoisting ropes,

l_N – hoisting rope length at the instant when the full conveyance begins to move upward from the bottom level.

The minimal loads are transmitted from hoisting ropes onto the drum at the instant when the full conveyance approaching the station begins the braking phase and are given as [5]:

$$S_{\min} = S_{St}^{H} + \Delta S^{H} \quad , \qquad (3)$$

where:

S_{St}^{H} – static load transmitted from the hoisting ropes onto the Koepe pulley at the instant when the full conveyance begins to brake when approaching the top level.

$$\Delta S^{H} \equiv -A_N E_N \cdot \frac{a_2}{a_N} \cdot \frac{20.5 l^{*}}{a_N} \quad , \qquad (4)$$

a_2 – deceleration of braking of the hoisting installation,

l* - hoisting rope length from the Koepe pulley to the conveyance at the instant when the braking phase begins in the normal duty cycle.

Utilising dependencies (1) to (4) for the hoisting installation operating in one of the Polish mines [1] maximal total force values in hoisting ropes on both sides of the Koepe pulley were determined. These values are: S_{max} = 529,5 kN and S_{min} = 441,1 kN, respectively.
Assuming uniform loading of all ropes, for further calculations the loading of an individual rope was taken as:

$S^{(1)}_{max} = S_{max}/n$ = 529,5/4 = 132,4 kN,

$S^{(1)}_{min} = S_{min}/n$ = 441,5/1 = 110,3 kN,

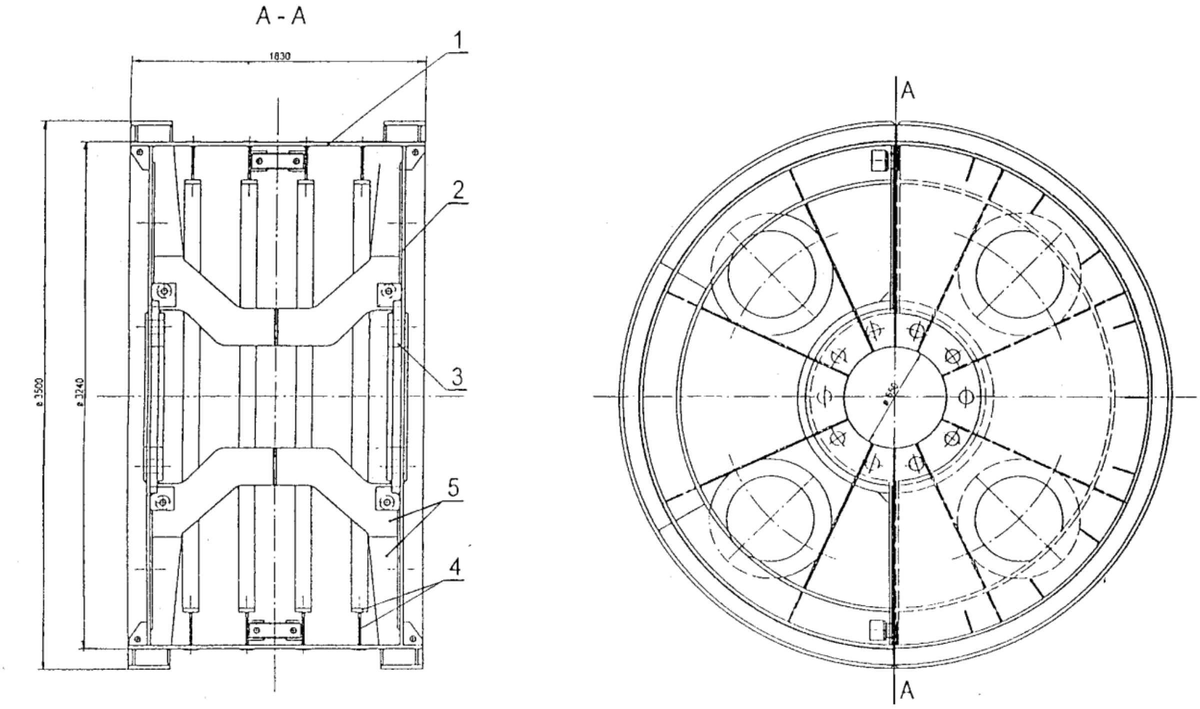
where: n = 4 – number of hoisting ropes.

3. NON-DESTRUCTIVE TESTS OF KOEPE PULLEYS

In order to determine ranges, character and intensity of occurrence of fatigue failures of Koepe pulleys of two considered constructions: with diaphragms of a shape of U letter, or with two circumferential rings stiffening the mantel, used in the selected Polish mines, non-destructive tests were performed by means of the magnetic-powder and ultrasound methods. Performed investigations indicated an occurrence of numerous cracks. Examples of crack types for the pulley with ‘U’ diaphragms are shown in Fig. 2 [1,2], while for the pulley with circumferential stiffening rings in Fig. 3 [3].

Fast rate of forming and growing of fatigue failures, as well as the resulting need of often repairs, essentially limit the safe usage and efficiency of hoisting installations. Having the above in mind, the authors of the hereby paper undertook an attempt to assess the technical state of Koepe pulleys of hoisting installations with taking into account the fatigue problems. After the dynamic analysis of drive wheels structural elements, the strength analysis of the discussed wheels solutions was performed. The results were verified by deformations measurements (stresses) in real objects under normal operational conditions in a few Polish mines.

a)



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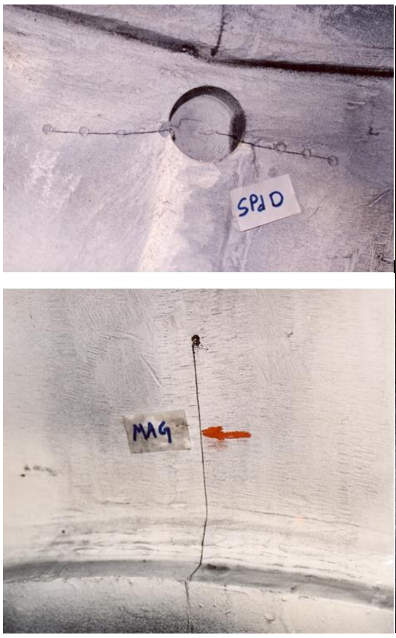

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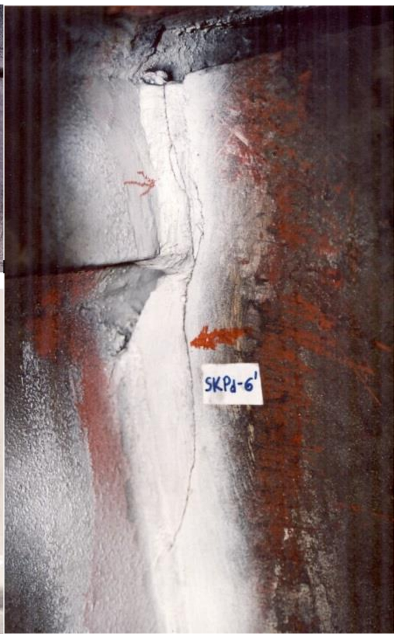
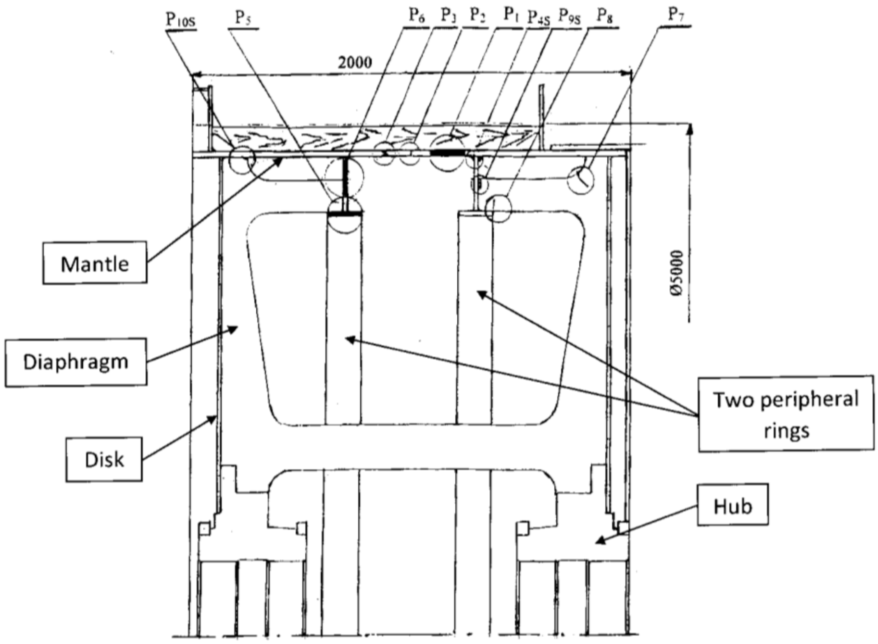


Fig. 2. Schematic presentation of the Koepe pullet with ‘U’ diaphragms: 1 – mantle, 2 – disk, 3 – hub, 4 – circumferential ring, 5 - diaphragm – a; fatigue cracks within a side disks zone: b – circumferential and radial cracks, c – crack in the zone joining the diaphragm ‘U’ with the side disk and hub [1]



Crack types detected on drums:

$P_1(P_{1S})$ – cracking of the mantle material (butt weld) in the direction of the generating line,

$P_2(P_{2S})$ – peripheral cracking of the mantle material (butt weld),

$P_3(P_{3S})$ – transverse cracking of the mantle material (butt weld),

P_{4S} – peripheral cracking of fillet weld connecting the web of the peripheral rib with the mantle,

$P_5(P_{5S})$ – cracking of the shelf (butt weld) of the peripheral rib,

$P_6(P_{6S})$ – radial or skewed cracking of the web (butt weld of the web or the fillet weld of the web bit) of the peripheral ribs,

P_7 – skewed cracking of a transverse reinforcing rib,

$P_8(P_{8S})$ – radial cracking of the transverse rib (butt weld of the transverse rib),

P_{9S} – cracking of a fillet weld connecting the transverse ribs with the web of the peripheral rib,

P_{10S} – cracking of a fillet weld connecting the transverse ribs with the mantle

Fig. 3. Schematic diagram of the Koepe pulley with two peripheral reinforcing rings and the types of detected cracks [3]

4. STRENGTH ANALYSIS

A high intensity of fatigue failures found on the basis of flow detection investigations for both types of pulleys indicates the unsatisfactory fatigue life in the considered period of their usage. In this situation performing detailed strength analysis of the pulley construction, which would provide the grounds for the modernisation allowing their further safe operations, became necessary.

The aim of the undertaken strength analysis of Koepe pulleys of hoisting installations was obtaining a complete information concerning the state of stresses formed in wheel elements under an influence of usage stresses. Knowledge of components of the state of stress in areas of the pulley construction subjected to extreme efforts, constituted the basis for assessing their strength and durability (allowable period of operations). The obtained information concerning values and distributions of stresses should allow to point out such zones of Koepe pulleys structures, which - due to the stress level - can be subjected to fatigue failures and should be periodically inspected by means of non-destructive tests.

On account of the above, the strength analysis of both types of drive wheels was performed with using numerical methods of the structure analysis. The finite element method (FEM) was applied [8] with using computer programs ‘FEMAP’ and ‘NE/Nastran’, on the basis of the developed calculation models of the Koepe pulley structure, consisting of the adequate elements of the plate and chunk type, fully representing the geometry of analysed structures [1,2,3].

4.1. Koepe pulley with diaphragms in a form of 'U' letter

Forces being the result of the hoisting ropes influence on the pulley (determined from equations (1) and (3)) as well as the dead weight of the adequate structure elements constituted loads of the basic numerical calculation model of the Koepe pulley structure ('MP').

The load originated from forces in ropes was adequately distributed along the contact line of ropes and the wheel mantle. The assumed scheme of the load distribution along the rope-drum contact is presented in Fig. 4 [1].

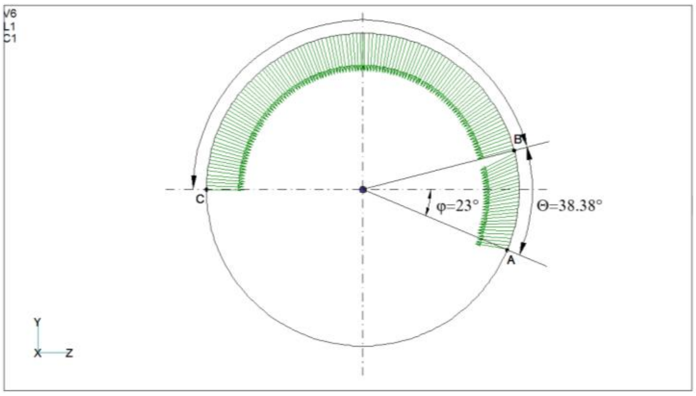


Fig. 4. The assumed scheme of the load distribution along the rope-drum contact [1]

This loading was estimated after taking into account the following parameters of the analysed hoisting installation:

- $\alpha_o = \varphi_{AC} = 203^0$ – angle of contact between ropes and the Koepe pulley drum,
- $\mu = 0{,}25$ - coefficient of friction between the Koepe pulley lining and ropes,
- $\theta = \ln(S_1/S_2)/\mu$ - shear angle of the rope on the Koepe pulley,
- $p_\alpha = 2 \cdot S_\alpha/BD$ – the rope pressure on the Koepe pulley,
- $S_\alpha = S_2 \cdot e^{\mu \cdot \alpha}$ – the rope force as a function of the angle of contact,
- $t_\alpha = p_\alpha \cdot \mu$ - rope friction force on the Koepe pulley,

where: S_1, S_2 – rope tensions on both sides of the Koepe pulley, D – diameter of the Koepe pulley drum, B – width of the rope contact surface with the Koepe pulley.

After introducing the proper equations and parameters, corresponding with the assumed load values, the applied computer program [1] generated automatically values of forces and imposed them on the proper nodes of the finite elements network.

At the A-B arch length (Fig. 4) the load originated from the rope pressure and friction - changing its value in accordance with changing forces in the rope - was applied, while along the B-C arch the load of a constant value originated from the rope pressing the mantel of the Koepe pulley drum.

As the effect of the performed computer simulations the complete information on the deformation and stress states, which were discussed for the skip hoist drum loaded with the maximum total force in ropes [2], were obtained. The selected results of the computer calculations are presented in the form of contour plans, illustrating distributions of the reduced stress values σ_H , in accordance with the Huber-Mises hypothesis, in individual zones of the pulley model. The general view of the distribution of the reduced stress values σ_H for the Koepe pulley (MP model), is shown in Fig. 5.

Performed calculations revealed that extreme stresses occurred in places where diaphragms were connected with the shaft hub (Fig. 5), it means in zones where fatigue cracks were the most often seen.

The analysis of the FEM results indicates that components of the stress state - for diaphragms - are periodically changing their values, together with changing their placement, caused by the pulley angular motion. Extreme values of main stresses (σ_1 and σ_2) are obtained when the diaphragm is situated in parallel to the resultant force in hoisting ropes, i.e. when it is in the plane of the maximum shaft deflection.

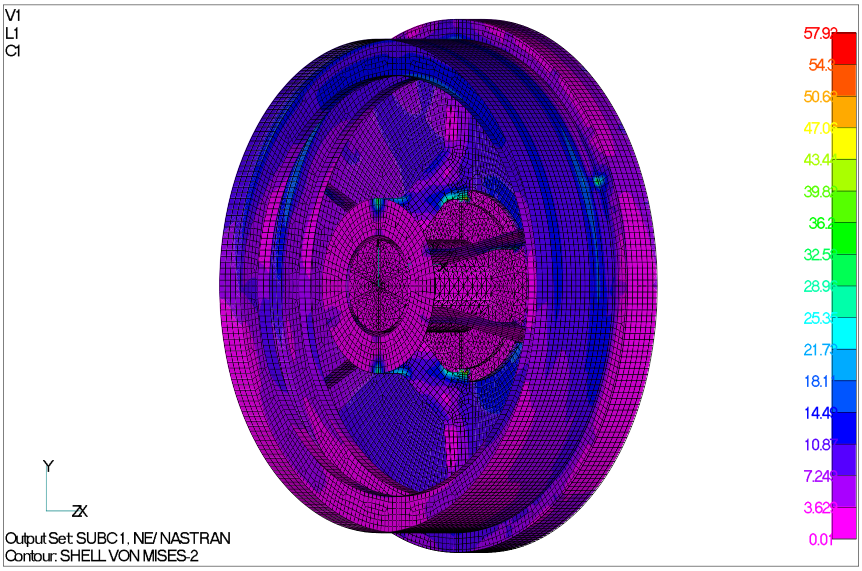


Fig. 5. Distribution of the reduced stress values σ_H [MPa] for the Koepe pulley - MP model [1]

In order to determine the influence of length and geometry of the crack developing within the diaphragm on stresses occurring in the pulley, the expanded FEM analysis based on the cracks development mechanism - identified for this area - was carried out. This analysis was taking into account a successive crack length growth, which is illustrated in Fig. 6. Individual calculation models (M) corresponded to the successive stages of the crack length growth: MA - line A-B, MB - line A-B-C-D, MC - line A-B-C-E. Extreme values of main stresses σ_1, σ_2 and reduced stresses σ_H in diaphragms and side disk, for all analysed models, are listed in Table 1 and 2.

The analysis of the FEM calculations indicates that the crack growth along line A-B (model MA) has no influence on the stress state and disk deformations. However, when the crack will grow to point D or E, it means when it will reach the side disk, its further growth will occur within the disk, causing failures illustrated in Fig. 2b.

a) b)

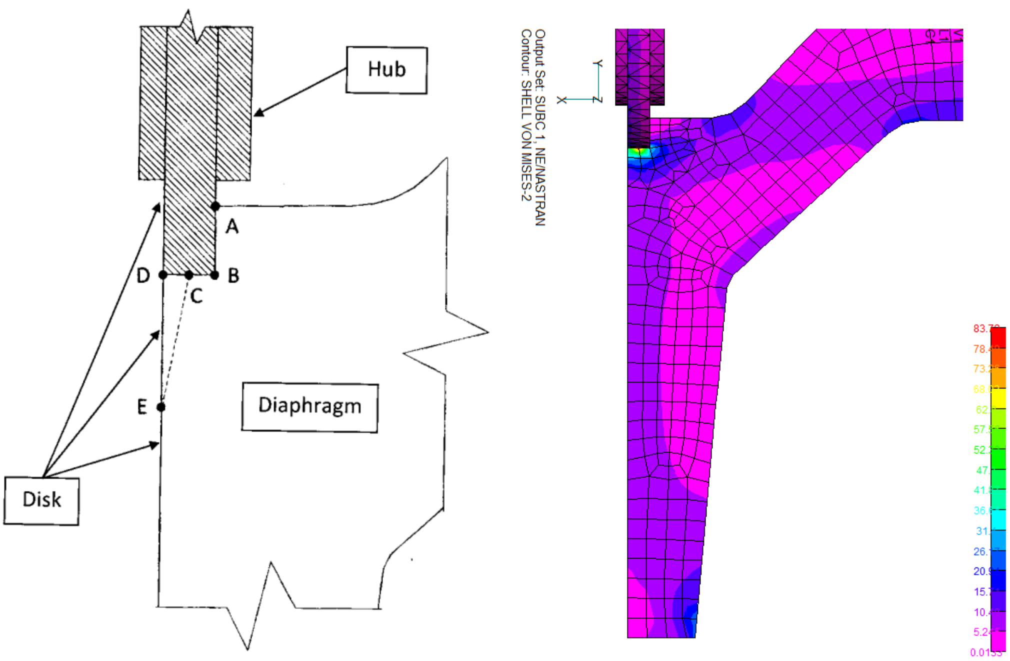


Fig. 6. Schematic presentation of the crack growth in the zone of connecting the diaphragm with side disk – a, distribution of reduced stresses σ_H [MPa] in the stretched diaphragm - model MB [1]

Presented above calculation results indicate that the direct reason of formation and growth of cracks is a significant over-rigid of the pulley construction by diaphragms. Diaphragms locally limit deformations of side disks caused by a shaft deflection, which is the reason of cyclic, alternate exocentric compressions and stretching of diaphragms (in agreement with the Koepe pulley rotations). In consequence, this causes formation of fatigue cracks in the most efforts zones of diaphragms and side disks of pulleys. This fact is confirmed by the calculation results together with the strain gauges measurements results [2,8], shown in Table 3 for extensometers, situated according to Fig. 7.
Significant differences in stress values obtained on the basis of the FEM and of measurements, visible in Table 3, reaching at measuring points P34 and PW approximately 47%, are probably caused by significant geometrical differences as compared with the designing documentation which was assumed in calculations. These differences were caused - among others - by numerous local repairs and reinforcements performed with using the welding process, which was introducing additional welding stresses.

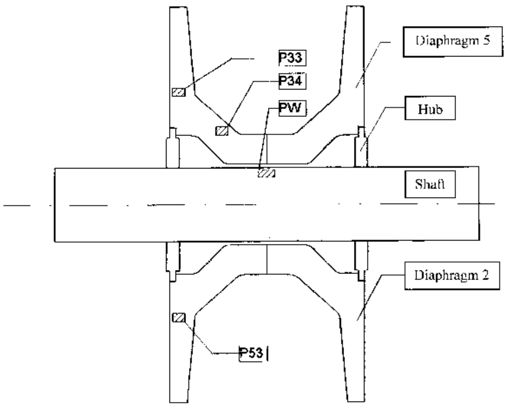


Fig. 7. Scheme of the placements of strain gauges on diaphragms and a shaft

According to the requirements of mining regulations [9,10], concerning a fatigue strength of welded joints (class D) made of steel grade S235 (of which the pulley was made), it can be assumed that this strength for the butt stretched weld equals: R_w = 45MPa, while for the fillet shear weld: R_w = 27MPa. The calculated stress values indicate that the reason of formation and growth of fatigue failures in the Koepe pulley is exceeding the fatigue strength values of welded joints.

The analysis of the FEM results indicates that the removal of the middle part of diaphragms together with the ring joining them, can have a beneficial influence on the state of stresses in the place of their connection with disks. Therefore the additional calculation model 'MK', was prepared. In this model the horizontal parts of diaphragms were removed, while their vertical fragments connected with side disks were remained. The distribution of reduced stresses in diaphragms and in a side disk of the pulley obtained for this model is presented in Fig. 8 and in Tables 1 and 2.

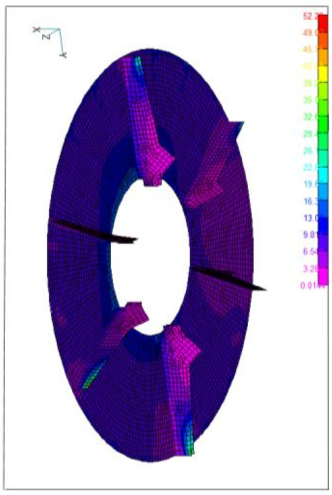


Fig. 8. Distribution of reduced stresses σ_H [MPa] in diaphragms and disk of the MK model [1]

Table 1. List of extreme stress values in the diaphragm [2]

Calculation model	Diaphragm placement:					
	above the shaft			below the shaft		
	Stress values [MPa]					
	σ_H	σ_1	σ_2	σ_H	σ_1	σ_2
MP	53.6	-14.6	-59.4	57.9	64.1	15.4
MA	48.9	-19.9	-54.1	51.3	56.4	18.9
MB	83.7	-0.9	-84.1	74.3	72.5	1.8
MC	43.7	-19.3	-49.8	47.7	53.8	19.3
MK	17.5	-3.7	-19.0	20.3	22.6	5.6

Table 2. List of extreme stress values in the disk [2]

Calculation model	Diaphragm placement:					
	above the shaft			below the shaft		
	Stress values [MPa]					
	σ_H	σ_1	σ_2	σ_H	σ_1	σ_2
MP	8.1	3.6	-6.5	9.6	7.3	-5.1
MA	8.2	3.6	-6.7	9.9	7.5	-5.4
MB	39.8	-8.2	-43.0	38.9	40.8	4.6
MC	60.5	-24.0	-68.8	60.9	69.1	23.0
MK	16.1	3.6	-14.0	18.1	15.2	-4.7

Table 3. List of strain gauges measurements and calculations results [2]

Measuring sensor number (Fig. 7)	Values of stress [MPa] obtained on the basis of:				
	Difference [%]	Measurements	FEM calculations		
		σ_p	σ_H	σ_1	σ_2
P33	-28.9	26.7	20.7	20.1	-1.3
P34	44.9	13.0	23.6	24.3	-
P53	35.2	6.8	10.5	8.5	3.0
PW	46.6	10.1	18.9	18.4	-0.2

The results of calculations - listed in Tables 1 and 2 - indicate that removal of the horizontal parts of diaphragms significantly decreases the stress level within areas of cracks formations, causing simultaneously approximately two-times increase of stresses in side disks, the values of which still remain low. It should be mentioned that the removal of the middle part of diaphragms did not significantly change the drive shaft deflection.

4.2. Koepe pulley with two circumferential rings

Forces being the result of hoisting ropes influences on the Koepe pulley, determined from equations (1) and (3) (at taking dead weight into account), constituted loads of the numerical calculation model of the Koepe pulley with two circumferential rings. On account of finding that fatigue cracks occur in the wheel mantle area, three various stages of rope winding were considered: total rope winding on the pulley, unwinding of one third of coils and unwinding of half of coils.

The distribution of reduced stresses σ_H in the wheel mantle, for circumferential rings and transverse ribs - for the original solution from Fig. 3 - is presented in Fig. 9 [3]. The selected results of the FEM analyses for circumferential rings and transverse ribs for the original solution were compared with the results obtained for the modernised pulley. Three additional circumferential rings were adequately applied in place of the removed one. The results are presented in a form of contour plans, representing distributions of reduced stresses values σ_H , shown in Fig. 10 [3,11].

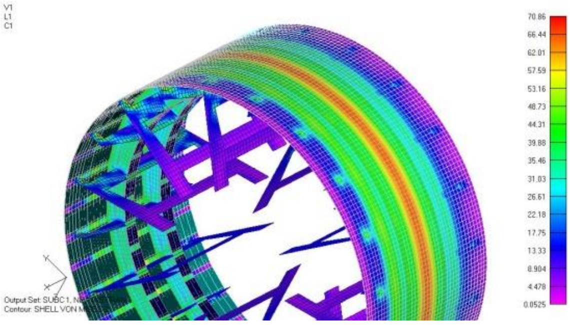


Fig. 9. Distribution of reduced stresses σ_H [MPa] on the inner surface of the mantle - original structure, total winding of the rope [3]

a) b)

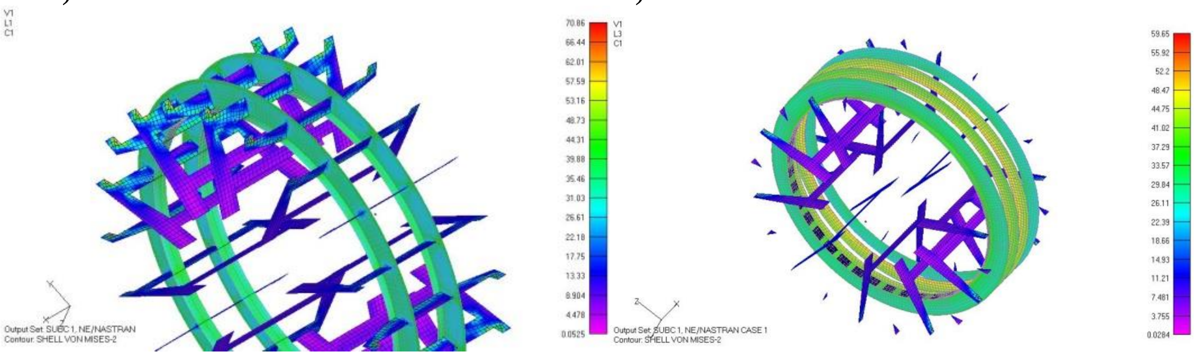


Fig. 10. Distribution of reduced stresses σ_H [MPa] in rings and transverse ribs, for the total winding of the rope: a – original construction, b – modernised construction [3]

Computer simulations indicate that stresses of extreme values occur in zones, in which the occurrence and growth of fatigue cracks were found. Thus, they explicitly indicate that the direct cause of fatigue cracks constitutes unsatisfactory load-carrying capacity and stiffness of the pulley mantle.

In order to adequately increase the pulley mantle strength, several constructional solutions were considered [13]. Finally the solution based on removal of one of the rings strengthening the mantel - (Fig. 10a) and substituting it by three other rings of smaller diameters, properly distributed (Fig. 10b) was selected [3]. The selected results of the performed numerical analysis in a form of diagrams showing stress changes along the mantel generating line are presented in Fig. 11 and 12.

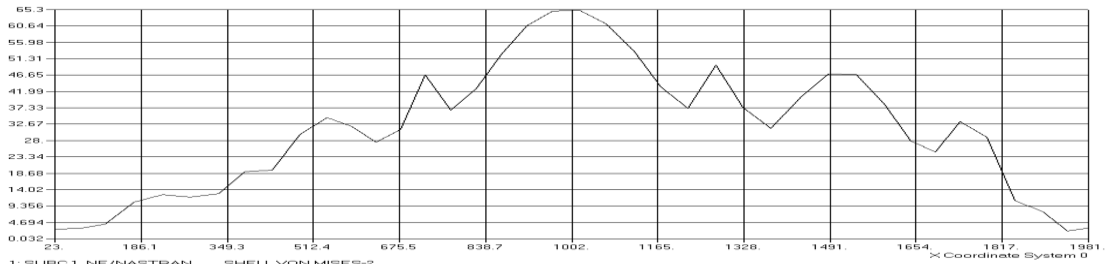


Fig. 11. Changes of the reduced stress values σ_H [MPa] along the mantel generating line. Original solution – total rope winded on the pulley [3]

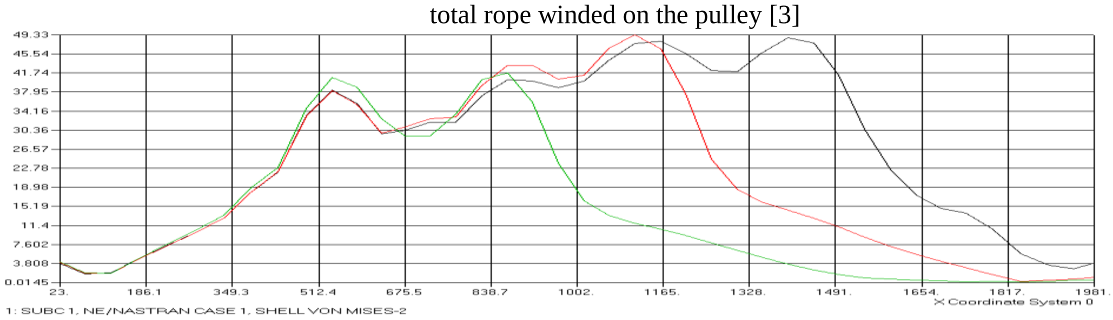


Fig. 12. Changes of the reduced stress values σ_H [MPa] along the mantel generating line for three assumed load cases. Pulley after the reconstruction [3]

The selection of essential information on values and distributions of deformations and stresses in calculation models was carried out on the basis of the numerical calculations results. Radial displacements of the mantel were decreased by 19%, for the modernised pulley construction. Information concerning stress changes are given in Table 4.

Table 4. FEM results

Maximal reduced stress σ_H [MPa]	Model	
	Original design	Re-design
In the model	71	60
On the mantle's external surface	65	45
On the mantle's internal surface	55	49
On rings in contact with the mantle	55	49
On the rings' inside diameter	43	46

Measurements of the operational stresses variability were performed with the application of the strain gauge method for the Koepe pulley, operating in one of the Polish mines [3]. The scheme of strain gauge arrangement is shown in Fig. 13 [3]. By strain gauges: 1, 2 - circumferential stresses in plates of stiffening rings were measured, while by: 3, 4, 5, 6 – circumferential and transverse stresses in two areas of the wheel mantel. Measurements were performed under conditions of license work of the Koepe pulley, consisting of: skip hoist loading with allowable load at the loading station, ascending of the loaded skip (with full kinematic parameters), unloading at the unloading station, and descending of the empty skip (with full kinematic parameters).

The performed measurements indicated that for each work cycle of the pulley circumferential stresses measured by means of extensometers 1,2,3 and 5 are compressible, while transverse stresses measured by extensometers 4 and 6 are stretching. Values of stress ranges measured by individual extensometers are compiled in Table 5 with allowable stress ranges, which were determined on the basis of standard [13] (PN-B-03200:1990) for the investigated wheel made of steel (S235 grade).

The results listed in Table 5 indicate a significant exceeding - in all measuring points - of stress values in relation to admissible values.

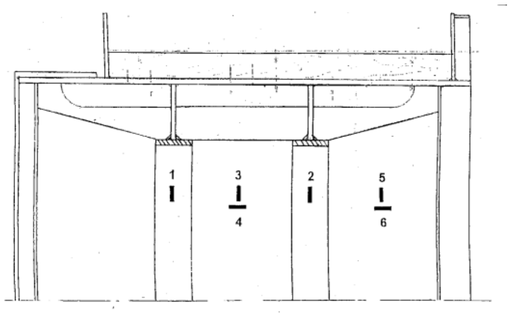


Fig. 13. Strain sensor arrangement on the drum

Table 5. Stress levels obtained by strain gauges measurements

Sensor	Stress range $\Delta\sigma$ [MPa]		Admissible level exceeded by [%]
	Measured	Admissible	
1	49	36	36
2	72	36	100
3	118	57	107
4	90	57	58
5	86	57	51
6	98	57	72

5. FINAL CONCLUSIONS

Numerous fatigue cracks, mainly in the welded joints areas, were found in both discussed constructions of Koepe pulleys, with diaphragms in a letter U form and with two circumferential rings stiffening the mantel, applied widely in the Polish mining for a long-term period significantly exceeding 30 years.

Strength calculations of pulleys performed on the basis of the performed dynamic analysis of hoisting installation, verified by strain gauge measurements of deformations on real objects, indicate that welded joints - applied in the considered constructions of Koepe pulleys - do not satisfy the requirements determined in regulations [9,10,13], concerning a fatigue strength of welded joints of class D of steel S235 grade, of which these wheels were made.

On the bases of the performed computer simulations (FEM) certain modifications of drive wheels were proposed and realised for a few wheels operating in the Polish mines. Experiences collected after a few years of their operations indicate a significant decrease of the number and intensity of fatigue cracks.

These strength calculations provided the basis for pointing out the construction areas of the considered pulleys subjected to the highest efforts, together with determining the time-period of performing their non-destructive tests.

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