Journal of Achievements in Materials and Manufacturing Engineering
AMME
VOLUME 23
ISSUE 1
July
2007

Failure reasons investigations of dumping conveyor breakdown

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Received 13.04.2007; published in revised form 01.07.2007

Analysis and modelling

ABSTRACT

Purpose: The main purposes of the paper are to discuss designing and exploitation problems of machines used in strip mines and investigation of its reasons based on steering frame brake-down of the dumping conveyor.
Design/methodology/approach: Numerical and experimental approach was used to investigate reasons of the break-down of the dumping conveyor. Numerical simulations based on the finite element method were used. Fractographic and microscopic evaluation and chemical analysis, were used to perform material evaluations. The objectives are achieved by analysis of the numerical simulations results of the broken part of machine and data coming from material evaluations. Based on the mentioned results conclusions concerning results of the failure were given. Additionally the new design of the steering frame half-shafts systems was discussed
Findings: The causes of break-down of the steering frame of dumping conveyor were found. Designing and manufacturing problems were the main reasons of the failure. The half shafts systems in undercarriages of the open pit machines are prone to break-downs. They require detailed analysis to be successfully implemented into steering system. Recommendations for the single shaft system are given in the paper.
Research limitations/implications: in the half-shaft undercarriage system, the friction in the supporting areas limits horizontal forces acting on safetying. Investigations of the static and sliding friction coefficients should be performed to estimate correct forces and optimal designing rules.
Practical implications: The study provides practical implication into designing of half-shafts undercarriage systems and their safetyings. Discussed design of the safetying should be redesigned or the half-shaft system should be changed into one shaft design.
Originality/value: The paper provides information backed by evaluation and test results, stating the nexus of causes of the dumping conveyor failure. The experimental and numerical approaches show relationship between designing and manufacturing process of machines. This can be helpful for the designers and researchers looking for reasons, methods of investigations or how to prevent failures of similar machines.
Keywords: Analysis and modelling; CAD/CAM; Constructional design; Materials; Metallography

1. Introduction

The issues relating to damage to and breakdowns and catastrophes of machines and equipment are the subject of numerous researches [13]. Detailed analyses are carried out to determine its causes, course and results. Thanks to the knowledge gained from such investigations similar cases can be prevented in the future. The examination of the causes allows one to identify the source of the problem. Moreover, from the user/owner's point of view this may be important in order to file an insurance claim. A block diagram of interdependencies between the activities involved is shown in fig. 1. To the user the course of a breakdown is not as important as its causes and results while to the researcher it is the course of the breakdown, which is most important since it implies conclusions about both the causes and the results. It is also usually most interesting from researching point of view. This is illustrated below for a breakdown of an overburden dumping conveyor. In opencast mines, dumping conveyors form a link in the EBD (excavator, belt conveyor flight, dumping conveyor) system.

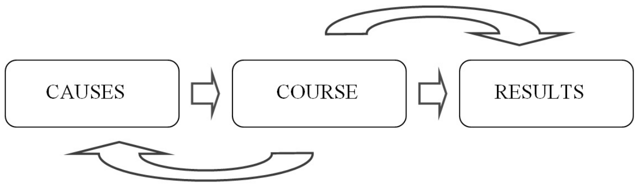

Fig. 1. Interdependencies between causes, course and results of breakdown

The dumping conveyor [6] is shown in fig. 2. The overburden which is to be removed from an open pit is transported to dumping conveyor for distribution [11].

Fig. 2. Breakdown of dumping conveyor's steerable carriage.

The investigated dumping conveyor has a system of two axle shafts (halfshafts) joining the track girders to the carriage (fig. 3). In such machines the halfshafts are usually 3 m long and have a diameter of 500 mm (resulting mass - 3.5 tons). This design solution requires, however, solid support and protection against slipping out.

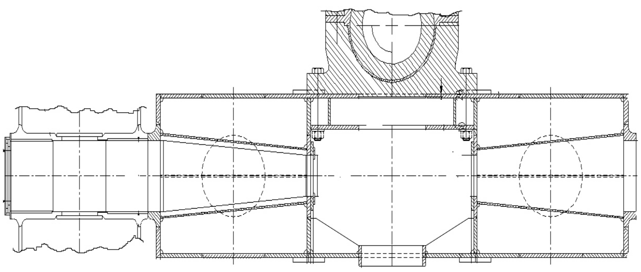

Fig. 3. Halfshaft mounting in carriage structure.

The breakdown occurred because one halfshaft slipped as a result of damage to its slip-out protection (fig. 4).

In order to determine the causes of the breakdown the an analysis of the forces acting in accordance with the current standards [4], FEM strength calculations of the steerable unit with the steering shaft, defectoscopic tests of the carriage and the halfshafts, metallographic examinations of the halfshaft material were carried out.

Fig. 4. Damaged halfshaft of steerable carriage

2. Analysis of forces acting on steerable caterpillar unit

The general principles of calculating caterpillar undercarriages can be found in [6]. By analyzing travelling forces one can determine the loads acting on the particular components of the caterpillar unit. When investigating the causes of the breakdown special attention was paid to the analysis of the forces acting during curvature travel (fig. 5). The forces for the caterpillar unit include: body weight V, transverse force H, the steering force S at the end of the shaft.

$$S = \mu V \frac{i_R}{R_s} \tag{1}$$

where:
μ – the coefficient of track friction against the ground (μ = 0.6),
i_R – the radius of gyration of the caterpillar track contact area,
R_s – the distance between the turn mechanism screw axis and the caterpillar track axis.

In FEM calculations [8, 9] one can substitute supports for the active forces (the latter are defined as reactive forces).

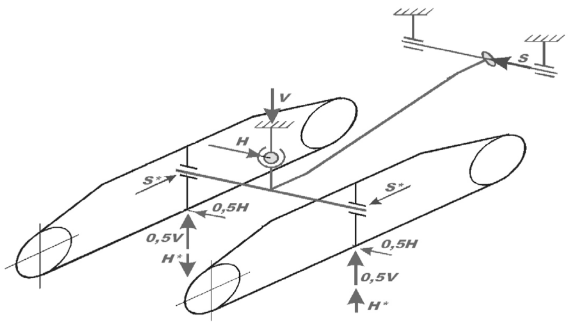

Fig. 5. Forces acting on caterpillar unit during turn

3. FEM strength calculations of caterpillar unit carriage shaft

The caterpillar unit carriage shaft has a box structure. The latter was reproduced using a geometrical model and a discrete shell-beam model. The strength calculations were performed for several cases of loads required by standard [4], using the finite element method [3, 5, 14]. Sample Huber-Misses stresses distribution is shown in fig. 6.

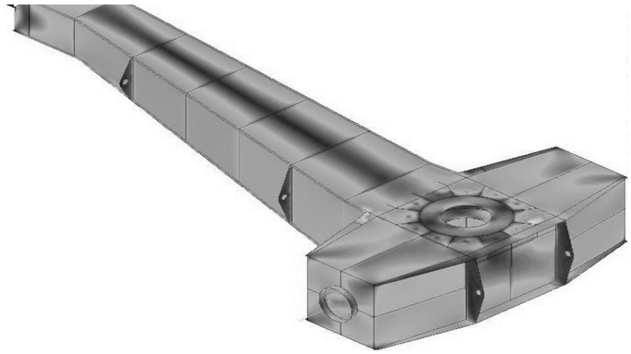

Fig. 6. Distribution of Huber - Misses stress steering carriage

For all the required cases of loading no stress levels specified by the standard [1] were found to be exceeded.

4. FEM strength calculations of halfshaft end

The protection against halfshaft slip out is shown in fig. 7.

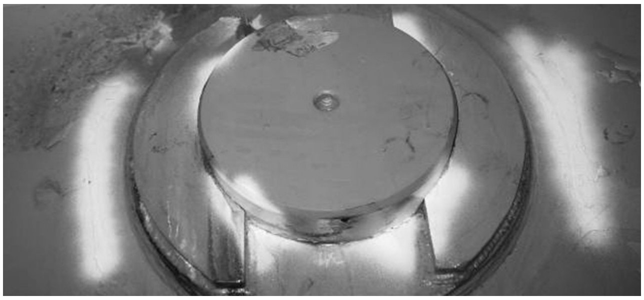

Fig. 7. Halfshaft end - protection against slip out

The plates protecting the halfshaft against slip out were deformed as a result of the longitudinal movement of the halfshaft . Due to the axial clearance of the halfshaft static friction μ = 0.09-0.14 becomes sliding friction μ = 0.01-0.05. The radius at the bottom of the groove for the protective plates equals r = 0.2 mm. The halfshafts were made of toughened constructional alloy steel 42CrMoS4. The FEM strength calculations [7] were performed for two values of the axial force acting on the halfshaft protection: axial force F = 1393 kN (curve travelling with radius R = 60 m, neglected friction on the halfshaft mounting hubs), axial force F = 704 kN (curve travelling with radius R = 60 m, friction μ = 0.09 on the halfshaft mounting hubs). Figure 8 shows the distribution of Huber-Mises stress for radius r = 0.2 mm and r = 4 mm (adopted for comparison purposes) while the maximum Huber-Mises stresses on the groove's bottom are given in table 1. The FEM [15] calculations indicate that at the bottom of the groove with radius r = 0.2 mm stresses considerably exceed the yield point.

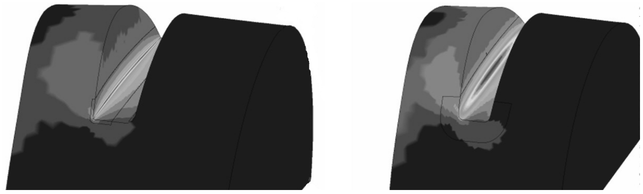

Fig. 8. Distribution of Huber-Mises stress

Table 1.
Maximum Huber-Mises stresses in analyzed halfshaft end

Calculations	Axial force F [kN]	Rounding radius r [mm]	H-M stress σ [MPa]	Safety factor δ
without friction	1393	4	822	1.07
		0.2	1080 (1780*)	0.49
with friction	704	4	405	2.17
		0.2	880	1.00

*) a theoretical value for material elasticity in the full load range.

5. Metallographic examination of halfshaft

Metallographic, macroscopic, microscopic and stereoscopic microscope examinations were carried on elements from the fractured caterpillar halfshaft. The investigations showed that the fractures had fatigue fracture features, including a temporary coarse-grained brittle zone (fig. 9). The fatigue zones are small (2-4 % of the fracture area) and have a lenticular shape [10].

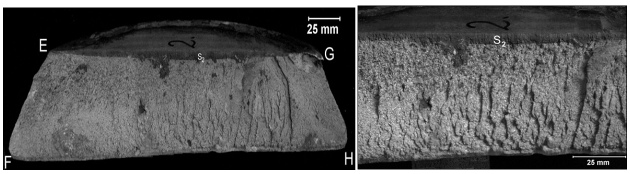

Fig. 9. General view of surface of fracture with marked edges

The small area of the fatigue zone in comparison with that of the temporary zone is evidence of great strain of the halfshaft material. Also improper heat treatment of the element's structural material could be the cause as well as the faulty fabrication of the grooves. On both sides of specimen the radius r=0.2 was measured (fig. 10).

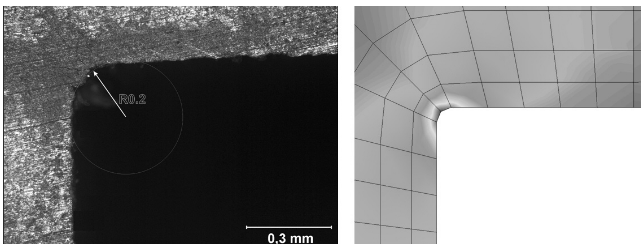

Fig. 10. Groove edge curvature radius r = 0.2 mm on specimen's left side, plastic strain zone on groove's bottom

In the unetched microsections non-metallic inclusions were found. Pearlitic-ferritic structure (locally in the form of the Widmanstätten structure) was observed. Consequently, the material's mechanical properties might have deteriorated [12]. The examinations also showed a crack originating from the groove's edge (fig. 11).

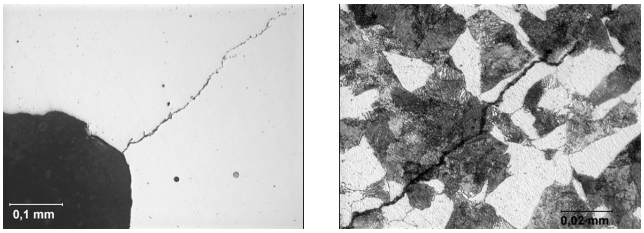

Fig. 11. Fatigue crack propagating along ferrite and pearlite grains

6. Conclusions

The FEM analyses of the caterpillar unit showed that the main cause of the breakdown was the way in which the halfshaft had been protected against slipping out. The fractures in the caterpillar support halfshaft lock are of the fatigue type. This fatigue fracture structure suggests that the largest stresses occurred in the groove's medial part. Because of the low ratio of fatigue zone area to temporary zone area a considerable effort of the structure or the adoption of very low safety factors could have been the case. The investigated element has a pearlitic-ferritic structure with locally occurring Widmanstätten structure. The direct cause of the breakdown was the too small rounding radius of the groove's bottom (r = 0.2), leading to breakdown of the dumping conveyor. A fatigue analysis of the groove's bottom showed that the halfshaft material was in the range of limited fatigue strength (fig. 12).

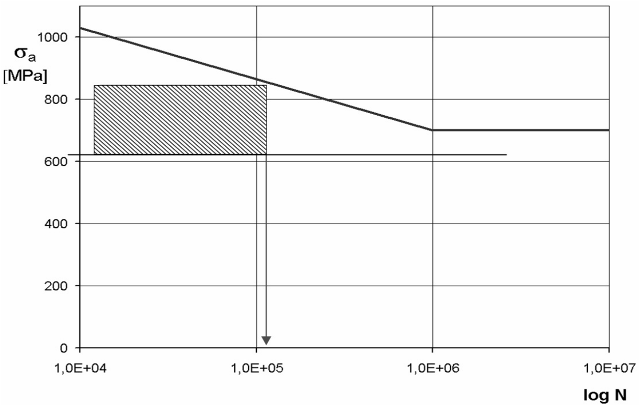

Fig. 12. Wöhler diagram for steel 40HM with marked limited fatigue strength region for radius r = 0.2 mm in halfshaft grooves

It is concluded that the adopted protection is not good for this system of fixing the halfshafts. A much better solution is an annular protection with a properly large rounding radius of the halfshaft groove. The undercarriage system consisting of two halfshafts is advantageous from the fabrication and weight points of view. An alternative could be a single-shaft design in which the slide-out protection is much simpler. However, this system would be much heavier and more expensive. Since many breakdowns of two-shaft undercarriages have been reported it can be concluded that such systems require more care in their design as compared with single-shaft systems. It can be helpful here to run numerical simulations. This approach was adopted in the present research. Also friction forces (static or sliding friction) at the places where the halfshafts are supported can determine the safety of operation of the undercarriage. But the magnitudes of the forces have not been precisely identified yet. Therefore further research is needed to make the design of such systems more accurate.

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