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TESTING OF THE OPENCAST MINE SNAIL CONVEYORS

ZKOUŠENÍ ŠNEKOVÝCH DOPRAVNÍKŮ POVRCHOVÝCH DOLŮ

Abstract

Descriptions of the problems of snail conveyors transport. 3D FEM model of a snail. Phases of calculation of the snail conveyor. Verification of calculations by the comparison with results of the FEM model analyses. Conclusion and proposed steps to obtain more accurate input data for more detailed analyses. Description of designed snail conveyors testing device.

Abstrakt

Popis problematiky dopravy pomocí šnekových dopravníků. Model šneku v MKP. Etapy výpočtu šnekového dopravníku. Verifikace výsledků výpočtu metodou MKP. Závěr a navržená opatření pro získání přesnějších vstupních údajů. Zkušební zařízení šnekových dopravníků

1 INTRODUCTION

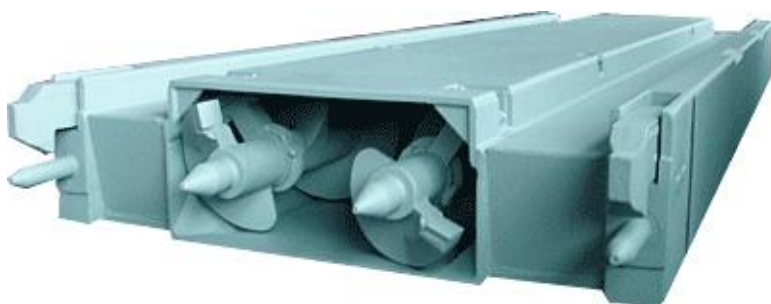Fresh coal in opencast mines in USA is transported by snail conveyors, which are composed of single sections. These sections (fig. 1) are placed successively. If a specific number of sections are assembled, it often results in a destruction of the weld which is situated between the tube and the end part of the snail. This happens most often in one of the last sections of the conveyor.

Fig.1

2 SNAIL CONVEYOR CALCULATION

The goal of the calculation was to prove whether the destruction of the weld occurs because of a manufacturing technique or overloading of the conveyor sections. It was proved that in this case the failure is caused by a breach of the manufacturing technique – the encircling weld of the tube was interrupted by the spiral of the snail. According to the technical documentation the weld should be also under the spiral of the snail. Customer provided a meter reading of the maximal torque load of the snail.

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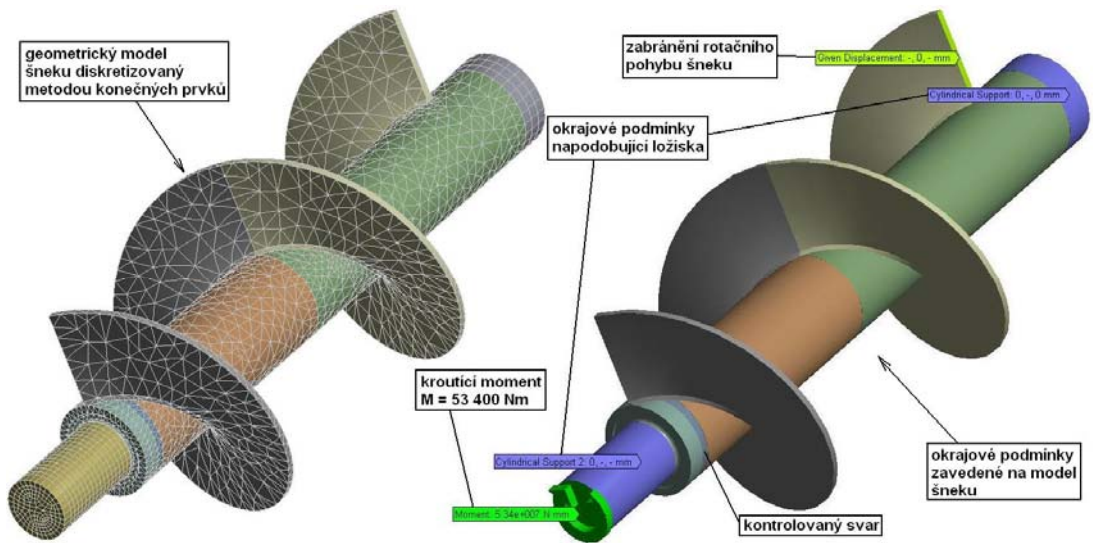
Fig. 2

The manufacturer gave us a drawing documentation of the snail and detailed informations about load type and maximal limits of the external loads. Geometry of the snail was created in AN-SYS Workbench 9.0 software. Then a mesh of elements for finite element method was created. Edge conditions were defined on this mesh to simulate the given real external and internal loads acting the snail. The simulation focused on determination of state of stress in the critical weld as it is shown on fig. 2.

The critical weld between the hub and the snail tube was simulated in two ways. In the first of them is the encircling weld simulated without the discontinuation under the spiral of the snail. In the second way the weld is simulated with the discontinuation which is caused by the manufacturing process. Both of them are shown on fig. 3.
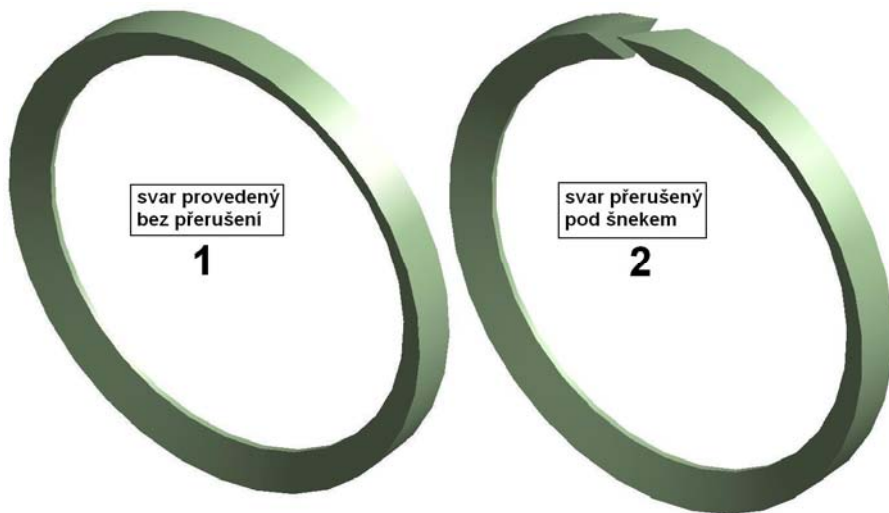

Fig. 3

Material properties used in the stress simulation are E = 210 000 MPa for Young's modulus and μ = 0,3 for Poisson's ratio. Shear and reduced stress (HMH) are graphically presented as a result of the simulation.

Firstly, the joint between the snail tube and the snail hub was analyzed without the spiral of the snail. A pure torque acts in the section where the bearing is situated. It can be expressed as:

$$\tau = \frac{M}{W_k} = \frac{16 \cdot M}{\pi \cdot d^3} = \frac{16 \cdot 53400000}{\pi \cdot 100^3} = 272 \, MPa \,,$$

where d is a diameter of the hub under the bearing and M is a torque. Results of the analyses are shown on fig. 4. Left one shows the shear stress and the right one shows the reduced stress (HMH).
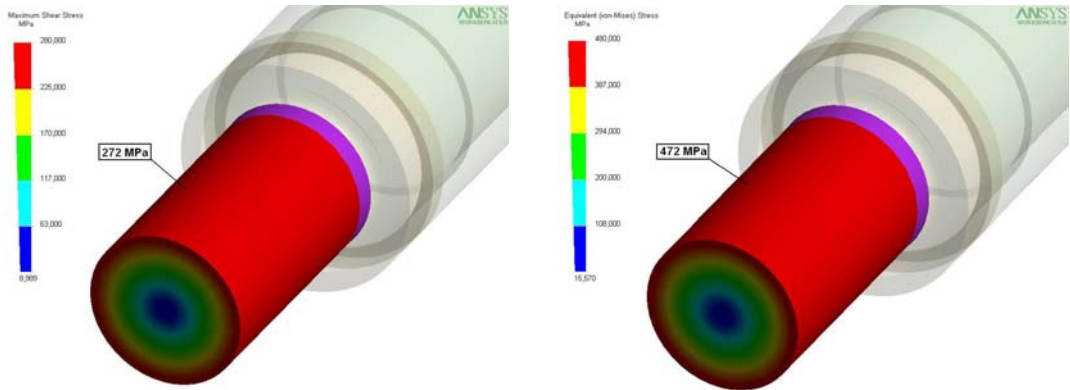

Fig. 4

Result of the simulation is a shear stress $\tau = 272$ MPa. Material of the snail hub is a cast steel 42 2712.5 whose yield point in tension is R_e=300 MPa and ultimate tensile strength R_m=530 MPa.

Yield point in shear can be calculated:

$$R_{e\tau} \approx 0,577 \cdot Re = 0,577 \cdot 300 = 173 MPa \,.$$

This calculation shows that the load exceeds the material yield point in shear. The required maximal torque 53400 Nm will probably produce permanent deformation of the material. Thus, the snail can withstand few load cycles with the maximum torque but after some time it will lead to a material failure of the snail hub.

Shear load of the used material shouldn't be over its yield point in shear. Also the transition of geometry in the area of the bearing fit causes a big concentration of stress. This stress concentration should be represented by a notched coefficient $k_\tau = 1,5$. With a safety coefficient of $k = 1,2$, the maximal shear stress shouldn't overcome value given by:

$$\tau_{dov} = \frac{R_{e\tau}}{k \cdot k_\tau} = \frac{173}{1,2 \cdot 1,5} = 96 \, MPa \,.$$

And the maximal operating torque can be expressed as:

$$M = \frac{\pi}{16} \cdot d^3 \cdot \tau_{dov} = \frac{\pi}{16} \cdot 100^3 \cdot 96 = 18849556 \, Nmm = 18849,9 \, Nm \,.$$

Fig. 5 shows the result of the stress analysis. The left one shows the shear stress and the right one shows the reduced stress (HMH). For this analysis the weld was simulated without the discontinuation and a spiral was modelled on the tube.
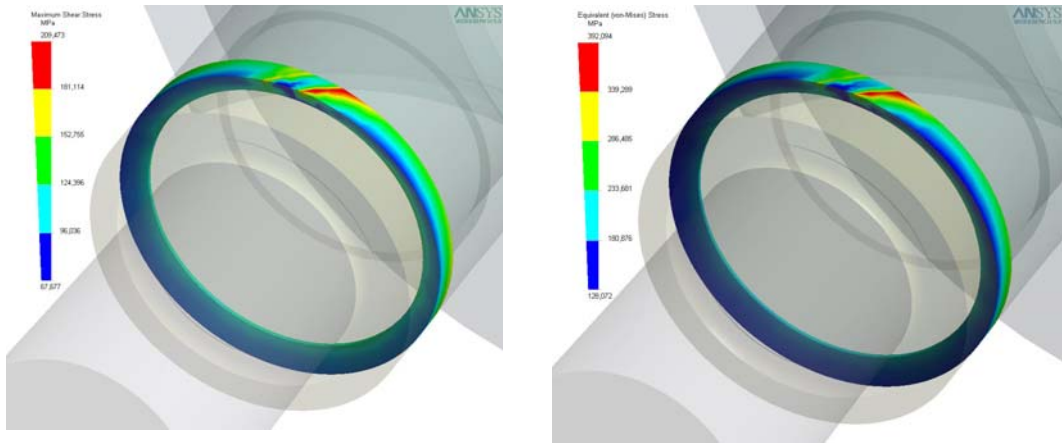
Fig. 5

The shear stress is $\tau = 210$ MPa ~ 100% and the reduced stress is $\sigma_{HMH} = 392$ MPa ~ 100%. These values of stress are a reference for the comparison with values obtained by the analysis where the weld is simulated with the discontinuation (fig. 6).
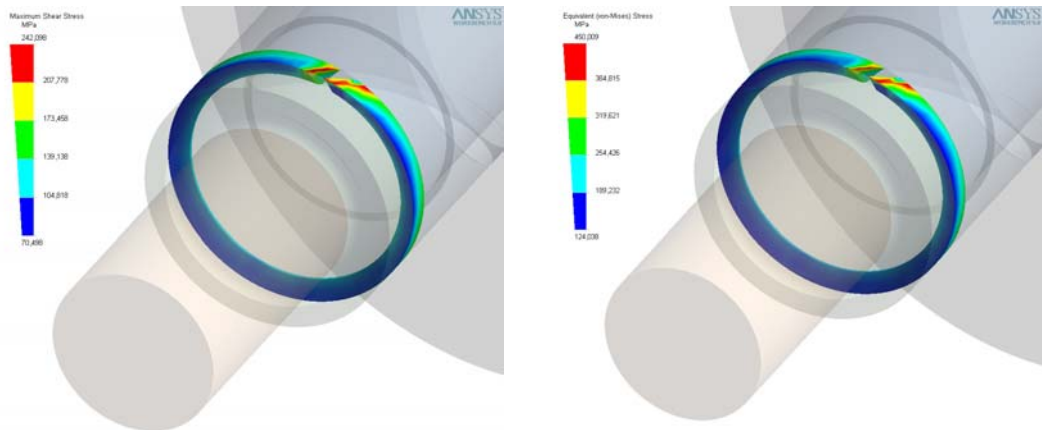

Fig. 6

The shear stress is $\tau = 242$ MPa ~ 115% and the reduced stress is $\sigma_{HMH} = 450$ MPa ~ 115%. The discontinuation of the weld causes a 15% increase of the stress.

The material of the snail hub is 42 2712.5 and the material of the tube is 11 523.1 whose yield point in tension is $R_e = 330$ MPa and the ultimate tensile strength is $R_m = 510$ MPa. For the calculations it is necessary to use the material properties of the snail hub 42 2712.5.

It is obvious that the torque load 53400 Nm significantly exceeds the yield point in shear of the used basic material. Moreover, there is a v-shaped butt weld located in the critical area. Computational coefficient for this kind of weld is $k_2 = 0,7$. The allowed shear stress for the weld and a safety coefficient $k = 1,2$ is:

$$\tau_{dov} = \frac{R_{e\tau} \cdot k_2}{k} = \frac{173 \cdot 0,7}{1,2} = 101 MPa .$$

The maximal safety torque for the weld without the discontinuation (reduced stress is the result of the FEM analysis) is:

$$M = \frac{\tau_{dov}}{\tau_{MKP}} \cdot M_{MKP} = \frac{101}{210} \cdot 53400 = 25683\, Nm \; .$$

The maximal safety torque for the interrupted weld (reduced stress is the result of the FEM analysis) is:

$$M = \frac{\tau_{dov}}{\tau_{MKP}} \cdot M_{MKP} = \frac{101}{242} \cdot 53400 = 22287\, Nm \; .$$

3 device for testing of the snail conveyors

The device for testing of snails was designed and manufactured. It allows a verification of the analyses results and allows an output manufacturing control of the snails. The designed device is shown on fig. 7.
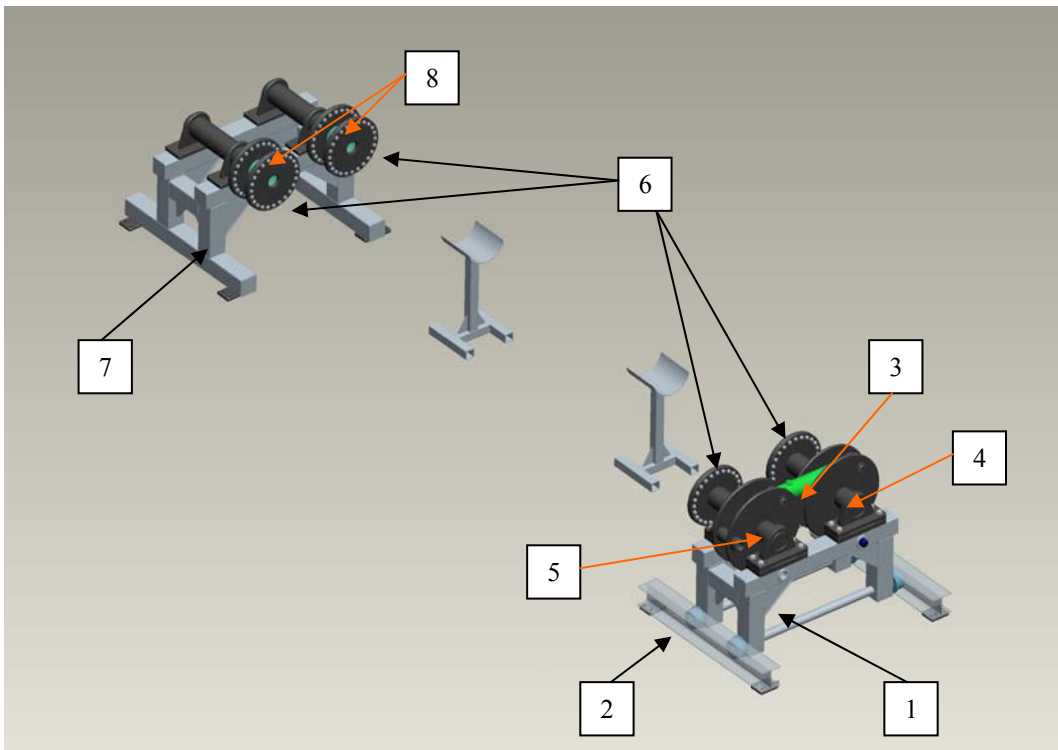

Fig. 7

The testing device consists of a Pressing stand (1) which is on a Travel (2). The pressing stand contains a Linear hydraulic motor (3). The motor affects Shaft 1 (4) or Shaft 2 (5) (depending on the mode of arrestment). The shaft transmits the torque load to the snail. One shaft is for left handed spirals of the snails and the second one is for right handed spirals. End sections of the snails will be welded on Circular flanges (6) as shown on drawing num. SHM-28. Opposite end sections of the snail will be welded on Circular flanges (6) which are bolted to a Fixed stand (7). This allows that the tested snail is fastened the same way as in the working process. On the fixed stand there are situated Measuring units (8). The unit consists of resistance strain gauges and electronics for data transport to control unit.

On fig. 8 is shown a photo of the manufactured testing device.

Fig. 8

4 Conclusions

Based on results of the analyses it is obvious that:

1) Stress increases about 15% due to the discontinuation of the weld between the tube and the hub of the snail under the spiral of the snail.

2) The torque load of 53400 Nm significantly exceeds the yield point in shear of the used basic material.

3) For the interrupted weld the maximal allowed torque with safety coefficient 1,2 is about 22290 Nm.

4) It is recommended to get load spectrums of the conveyors sections in working process for more exact determination of the real acting load and lifetime of the sections.

5) The testing device was designed for maximal torque of 70 000 Nm. During the testing it was verified that the snails of conveyors are able to withstand the required torque of 53 400 Nm as a static load. Failures of the snails are probably caused by the dynamic effects of the coal transportation. The device will be used for testing of lifetime of snails under a repeated load.

REFERENCES

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