SIMULATION MODAL ANALYSIS OF THE DRIVING SPROCKET OF A SCRAPER-CHAIN CONVEYOR

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ABSTRACT
The issues related to a scraper-chain conveyor have been reviewed and analysed. A computer-aided research has been conducted and the resultant natural frequencies of the driving sprocket obtained. The causes for the occurrence of unfavourable resonance processes have been investigated.
Key words: *scraper-chain conveyor, driving sprocket.*

1. Object of study.

The scraper-chain conveyors are employed in transporting easily conveyable bulk material of medium range pieces. In the mining industry, their main application is in coal transportation. The traction body of the conveyor is formed by an endless chain with scrapers attached onto it. The load is hauled by the scrapers in the direction of the chain movement. Initial tension of the chain is necessary to provide the normal passage through the driving sprocket and the reversing sprocket.

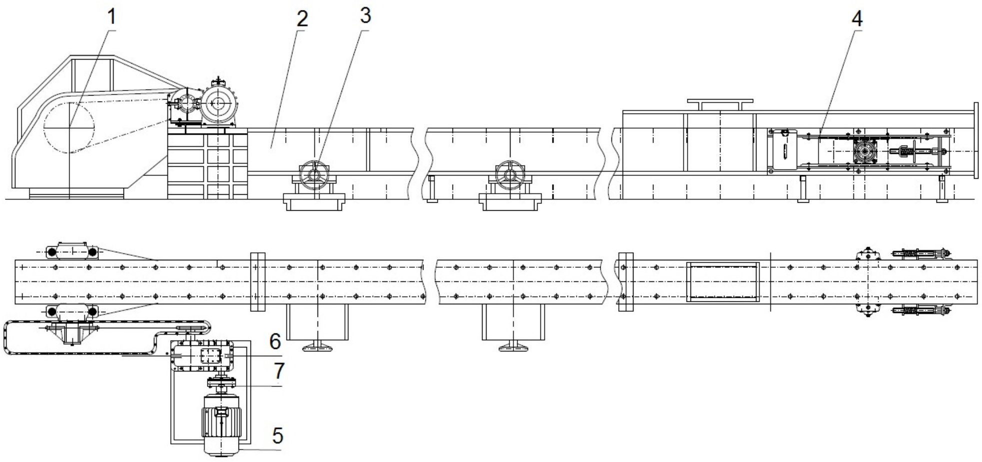

Figure 1. A general view of a scraper-chain conveyor

Figure 1 presents a drawing of the general view of a scraper-chain conveyor with the following elements:

1 - Conveyor drive; 2 - Conveyor section; 3 - Tippling device; 4 - Reversing station; 5 - Electric motor; 6 - Reduction gear, and 7 - Claw coupler.

2. The main calculations in determining the tensile forces in the tractive body and resistance during the operation of a scraper-chain conveyor [1,6].

2.1. Resistance in the load branch

$$W_{load\ branch} = L\left[\left(q_{load}.\omega + \mu.q_{tracktive\ body}\right).cos\beta \pm \left(q_{load} + q_{tracktive\ body}\right).sin\beta\right], [N] \quad (1)$$

where:

- L is the conveyor length, [m]

$-q_{\ load}[N/m]$ – the linear weight of the load;

$-q_{\ tracktive\ body}[N/m]$ – linear weight of the tractive body;

$-\omega = 0{,}6$ – resistance factor when dragging (hauling) the load along the chute;

$-\mu = 0{,}35 -$ resistance factor of the chain motion along the chute;
$-\beta = 10^0 -$ conveyor angle of inclination;

2.2. Resistance in the void branch

$$W_{void\ branch} = L.q_{tracktive\ body}(cos\beta.\mu \pm sin\beta\ ), [N] \tag{2}$$

2.3. Resistance in the reversing sprocket

$$W_{reversing\ sprocket} = \kappa.S_{incoming\ branch}, [N] \tag{3}$$

$-\kappa = 0{,}07 -$ resistance factor in the reversing sprocket;
$-S_{\ incoming\ branch} -$ tensile force in the incoming branch of the chain at the reversing sprocket;

2.4. Resistance in the driving sprocket

$$W_{\ driving\ sprocket} = \lambda.\left(S_{incoming\ branch} + S_{outgoing\ branch}\right), [N] \tag{4}$$

$-S_{\ incoming\ branch}[N]$- tensile force in the incoming branch of the chain at the driving sprocket;
$-S_{\ outgoing\ branch}[N]$- tensile force in the outgoing branch of the chain at the driving sprocket;
$-\lambda = 0{,}07 -$ resistance factor in the driving sprocket;

3. Tensile forces in the tractive body

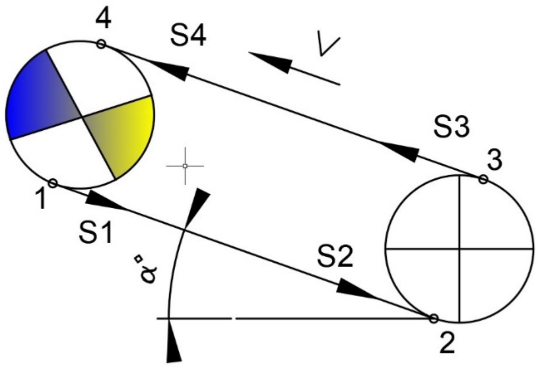

Figure 2. Diagram of the tensile forces in the tractive body

To determine tensile forces in various points along the endless loop of the tractive chain, the so called "point method" is used whose principle can be expressed as follows:

$$S_1 = S_{min}; S_2 = S_1 + W_{void\ branch}; S_3 = S_2 + W_{\ reversing\ sprocket.}; S_4 = S_3 + W_{\ load\ branch.}; \tag{5}$$

where:

S_1, S_2, S_3 и S_4 are the tensile forces in the chain for the specified point along the loop. The tensile force in the chain for any random point along its loop is equal to the tensile force in the preceding point plus the resistance of their intermediate section.

3. Major parameters of the scraper-chain conveyor for determining resistances of motion and the tensile forces [4,5,6].

$-L = 80[m]$ - conveyor length;
$-Q_h = 85[t/h]$ - output by hour;
$-v = 0{,}6\ [m/s]$ - chain speed of motion;
$-n = 27min^{-1}$- revolutions of the driving sprocket or $n = 0{,}45[sec^{-1}]$;
$-q_{\ load} = 386[N/m]$;
$-q_{\ tracktive\ body} = 108[N/m]$;
Upon substituting in equations 1, 2, 3, 4, and 5, the following are obtained:
$-W_{\ load\ branch} = 27792[N]$;

$-W_{void\ branch} = 1494[N]$;
$-S_1 = 3000[N]$ – minimum tensile force;
$-S_2 = 4494[N]; S_3 = 4808[N]; \ S_4 = 32600[N]$;

4. Modal (frequency) analysis of the driving sprocket after the finite element method (FEM) [2,3].

The working process with scraper-chain conveyors is extremely dynamic. One of the structure elements which is loaded the most is the driving sprocket with the shaft. It is particularly unfavourable when they fall into a resonant mode of operation. Resonance occurs when the frequency of a disturbing force coincides with the natural frequency. Therefore, a major technical task is to avoid the possibility of resonance occurrence. For this purpose, the frequencies of the disturbing forces that will act on the machines have to considerably vary from the native frequencies. Resonant modes do not pose a threat provided they pass quickly.

The objective of the current study is to observe if the assembled unit driving sprocket - shaft (fig.3 and fig.4) falls into a resonant mode. The present work examines a type construction of the assembled unit driving sprocket-shaft. Computer modelling is carried out within the *Autodesk Inventor* medium with the highest possible input data discretisation.

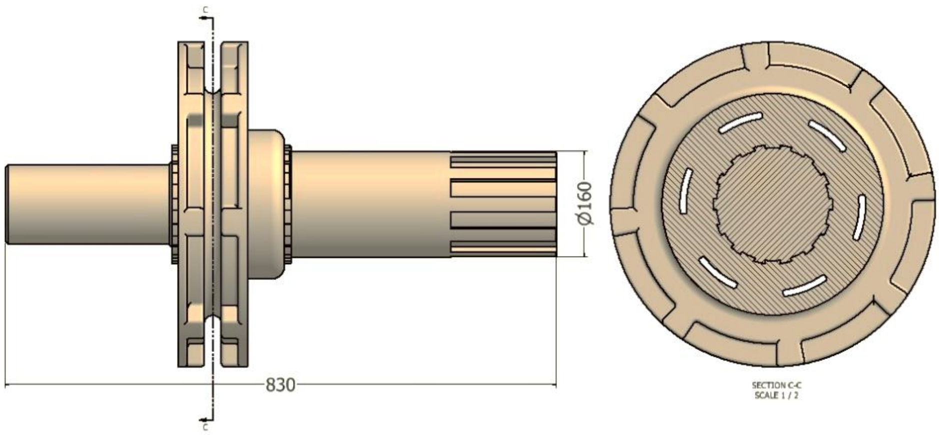

Figure 3. A developed 3D model of the assembled unit driving sprocket - shaft

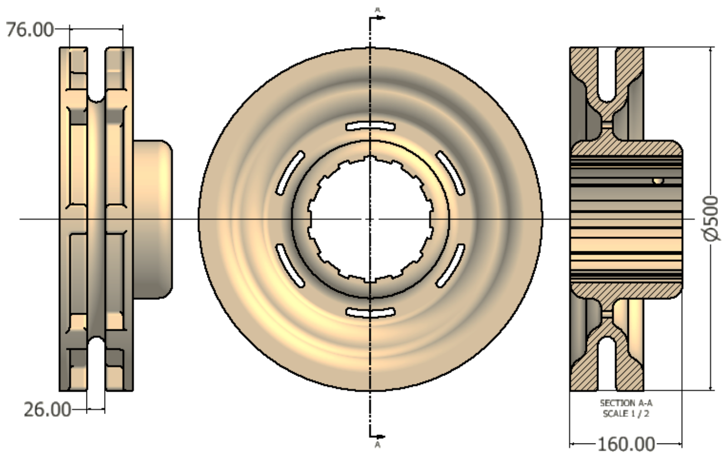

Figure 4. 3D model of the driving sprocket

Model meshing (discretisation) is one of the most significant steps in the study, shown in Fig.5. Through the “Fixed constraint” option, the left end of the shaft, where the grooves are, is set to be immobile (fixed). This is needed for the survey since the computer analysis requires an area of zero displacements. Through the “Force” option, the forces acting on the assembled unit are shown. The accepted base material used for its production is steel grade S235 J203 of the Bulgarian State Standard (BDS) EN 1025, whereby the allowable stress range is $[\sigma] = 160MPa$.

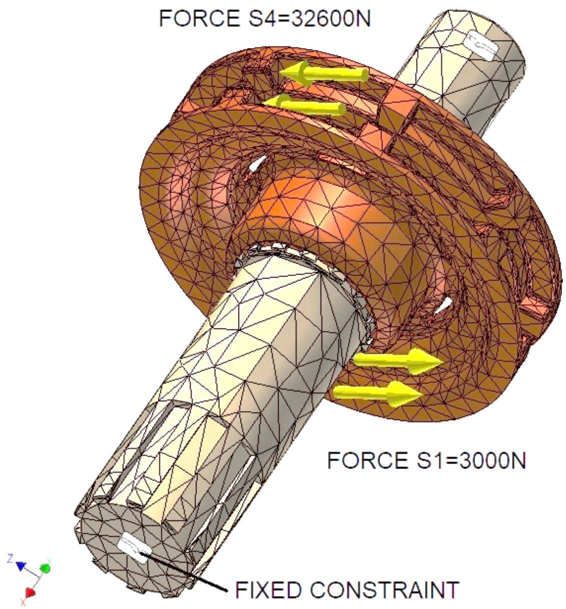

Figure 5. Discretisation of the model and its meshing

5. Results and conclusions.

With the modal analysis of a system with a large number of degrees of freedom, only the several initial natural oscillation frequencies are of interest. From an engineering point of view, only the first mode shape of loss of stability matters.

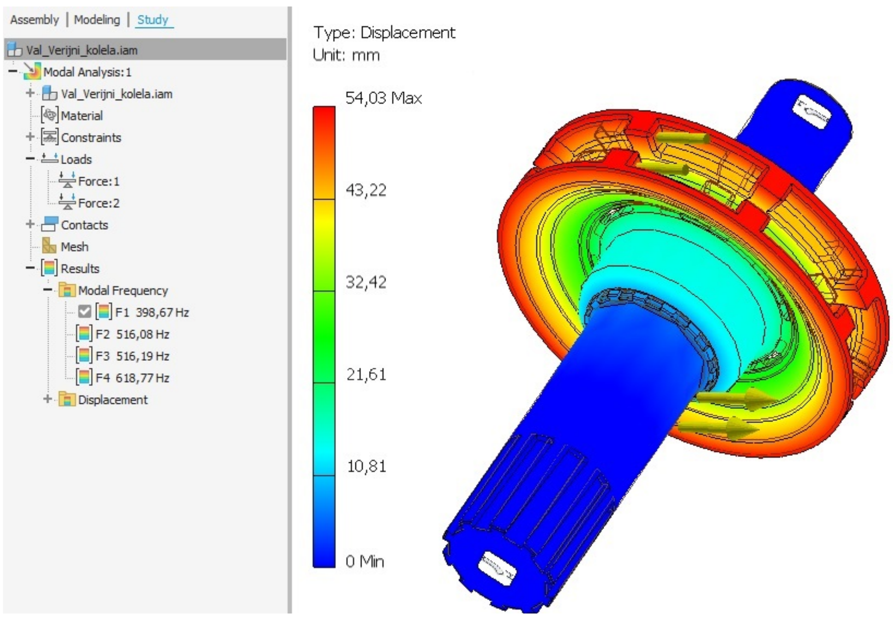

Figure 6. Results obtained for the first mode shape of the driving sprocket - shaft at 398.67[Hz]

Fig. 6 shows the first harmonic frequency of the system "driving sprocket-shaft", which represents the first bending frequency of the driving sprocket $f_1 = 398{,}67[Hz]$.

Figure 7 illustrates the second harmonic frequency of the system studied, which represents the first bending frequency of the driving shaft $f_2 = 516{,}08[Hz]$.

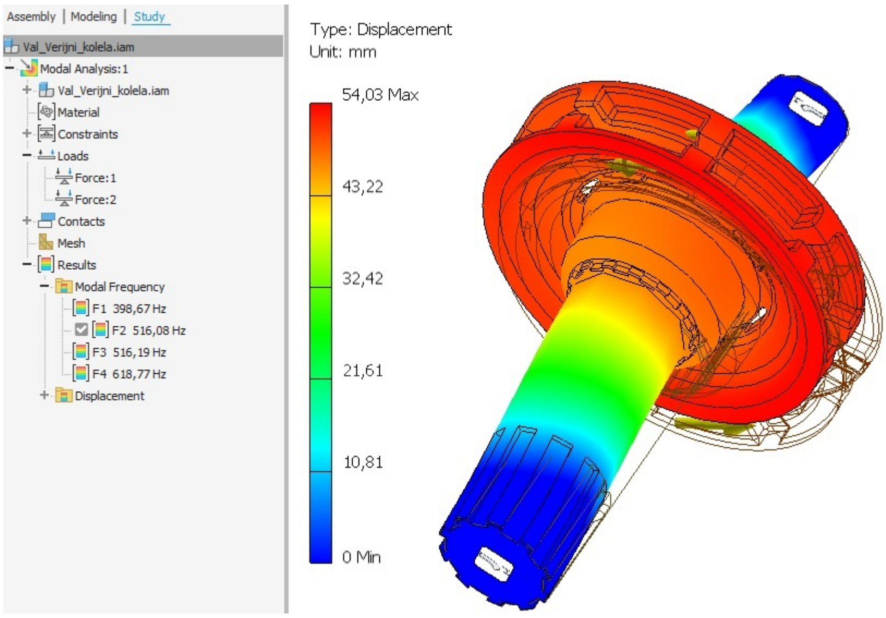

Figure 7. Results obtained for the second mode shape of the driving sprocket - shaft at 516.08[Hz]

Figure 8 presents the third harmonic frequency of the system studied, which is the second bending frequency of the shaft $f_3 = 516{,}19[Hz]$.

Figure 9 shows the results of the fourth harmonic frequency of the assembled unit, which is the second bending frequency of the driving sprocket $f_4 = 618{,}77[Hz]$.

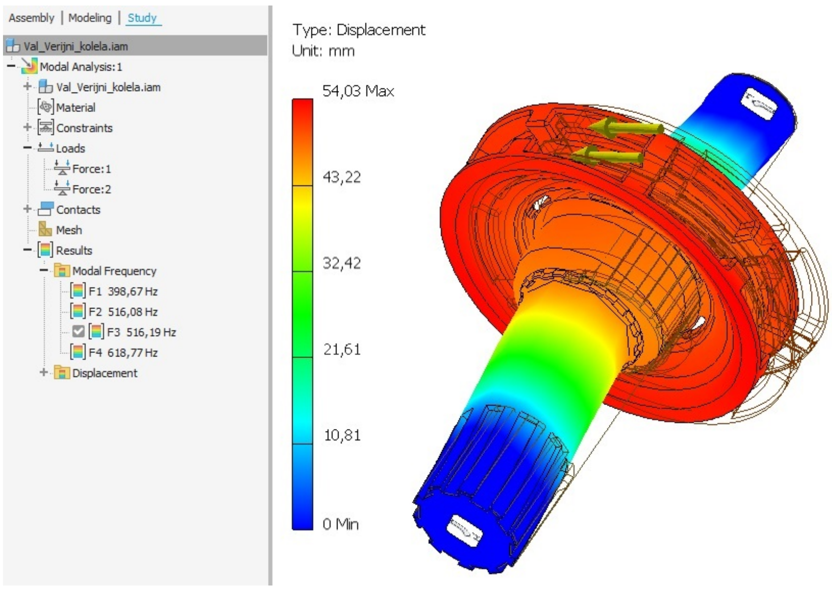

Figure 8. Results obtained for the third natural mode shape of the driving sprocket - shaft at 516.19[Hz]

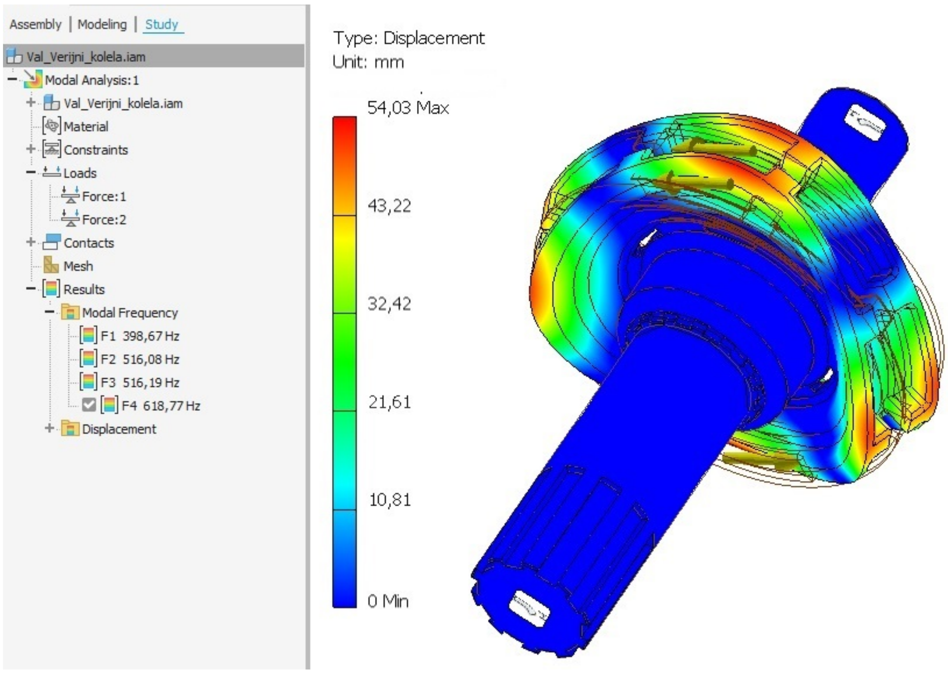

Figure 9. Results obtained for the fourth mode shape of the driving sprocket - shaft at 618.77[Hz]

Considering that the rotational frequency of the “driving sprocket-shaft” system is $n = 0{,}45[Hz]$, we can conclude that it is in no risk of a resonance in view of the vibrations conveyed by the drive. The frequencies thus obtained by the computer modal analysis performed matter with the occurrence of other external mechanical oscillations of a frequency that is close to the natural frequency.

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