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STATIC AND DYNAMIC ANALYSIS OF TELESCOPIC BOOM OF SELF-PROPELLED TUNNELLING MACHINE

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The paper presents selected aspects of the strength analysis of the self-propelled tunnelling machine boom. The principles of creating calculation models for numerical simulations with use of the finite element method are given. The study also presents two ways of conducting numerical calculations in both the static and dynamic range. A detailed example of numerical FEM tests of the telescopic boom is provided.

Key words: finite element method, strength calculations, static and dynamic analysis, self-propelled mining machines

1. Introduction

Difficult geological conditions and more intense tunnelling and mining processes taking place today in many building sites lead to a high mechanisation level of building works. Because of this, specialised self-propelled tunnelling machines are constructed, which enable a sufficient progress in the tunnelling works. Among the machines most frequently used in the tunnelling and building sites, there are those directly used in the preliminary works. These are the vehicles used for slabbing, drill rigs and bolt setters. The common feature of those machines is the fact that the working tools are placed on a boom. The boom mounted on self-propelled tunnelling machines should have a sufficient number of degrees of freedom to minimise the time related to changing the location of the machines. Most frequently, this is a straight-line structure ended (in the case of drill rigs and bolt setters) with a rotating head (turnover fixture). An example of this kind of machine is presented in Fig. 1.

In the case of drill rigs, this part allows one to position the drill rig mast so as to ensure stable perpendicularity of the drill rig axis to the surface of

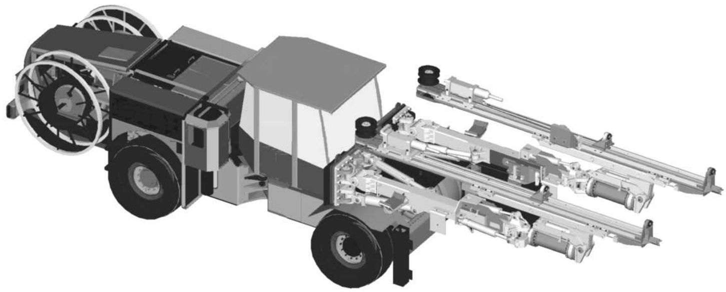

Fig. 1. Drilling machine Face Master with two booms (www.minemaster.eu [13])

the walls or ceiling. The size of the machine and its working range are, in this case, determined by the size of the excavation tunnel and the size of the drilling mast mounted on the boom. The mast length ranges between 4 and 7 m depending on the needs, determining the size of the boom mounted on the self-propelled tunnelling machines. In order to reduce the costs of manufacturing the entire machine, it is important to standardise particular parts and units and to design a universal boom which could be mounted on various types of machines. Because the boom with the mast is a significant load for the structure, its mass should be as smallest as possible. This brings about serious engineering problems stemming from the operation, manufacturing technology, material limitations, etc. Today, this problem is solved by means of a straight-line feeding mechanism which guarantees a sufficient working range and, after moving, reduced distance between the centre of gravity of the working part (boom with the drilling mast) in relation to the front axis of the machine. Another important feature, besides the limitations related to the preliminary works, are loads related to motion of self-propelled tunnelling machines. Many years of experience gained while operating the booms in mines suggest that most failures take place in the transport position (with the boom protruding). There are several reasons for that. The operational loads in the course of preliminary works (e.g. drilling) may be closely determined both with respect to the direction and value. When a self-propelled tunnelling machine moves from one place to another, the loads affecting the structure and boom are of dynamic character which is hard to specify. In such circumstances, it is necessary to carry out experimental research or computer simulation based e.g. on the finite-element method. The computer simulation, in this case, should be given priority over the research since without preparing an expensive prototype we are able to determine both the temporary values of forces and the directions of their action, and the effort value for a given spot of the structure. It should be kept in mind that any change of the boom structure geometry will influence both its strength and load (change of mass).

This paper is aimed at strength analysis of the boom structure using the finite-element method (Rusiński, 1994; Rusiński *et al.*, 2000) in the dynamic and static scope, taking into consideration the material and geometrical nonlinearity (Kleiber and Woźniak, 1991; Woźniak and Kleiber, 1982). The analysis comprised the simulation of the strength test connected with the vehicle front wheels going over a barrier 150 mm high with the maximum speed of 12 km/h, and the analysis of the influence exerted by the drill mast position with respect to the boom while drilling on the effort of the boom structure.

2. Computational model

Based on the technical documentation of the boom structure (prepared in CAD/FEM system, Fig. 2) the geometrical (shell) model has been created (Fig. 3). Then based on the geometrical model the discrete model of the boom were created. The boom structure is supported on two motor operators (rising and rotating ones) and connected by means of joints with the platform. Its frontal part is loaded with a feeding mechanism or a mast which can rotate on two planes (horizontal and vertical) in relation to the boom. All connections and additional masses are taken into consideration in the discrete model, and are presented in Fig. 4.

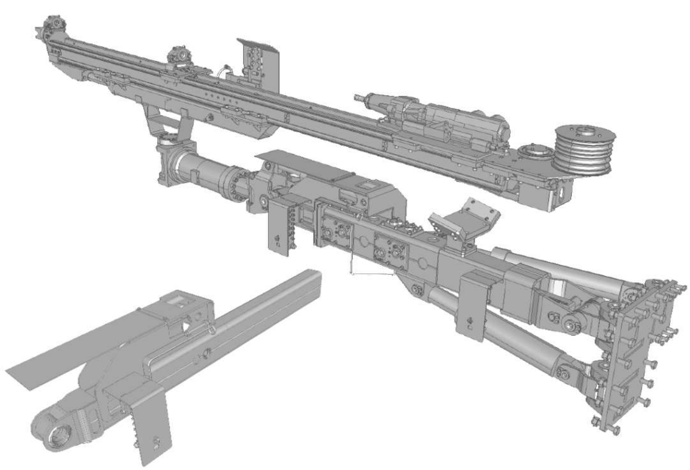

Fig. 2. The working unit with a separate boom

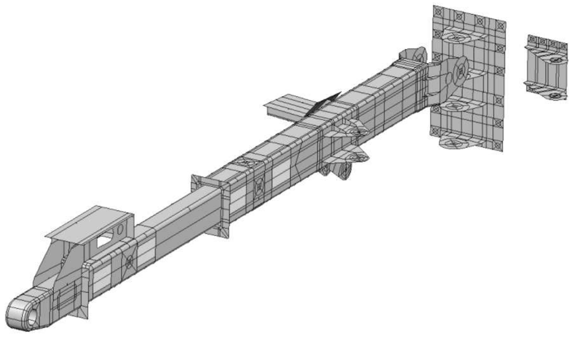

Fig. 3. Shell model of the mining machine boom

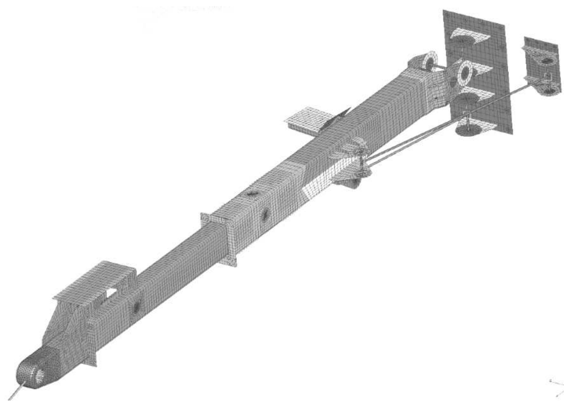

Fig. 4. Discrete model prepared for static analysis

The strength calculations were prepared using the finite-element method. They were divided into the following stages:

- static linear strength calculations carried out using FEM (Karliński and Wach, 2006; Kleiber and Woźniak, 1991; Rusiński, 1994) of the boom structure; determination of shifts and strains in particular points of the structure;
- analogous dynamic calculations.

In the case of dynamic analysis, they were carried out taking into consideration the geometrical and material nonlinearity, using the explicite type algorithm (Jones and Wierzbicki, 1993; Karliński and Wach, 2008; Pam-Crash User Manual, 1989; Wierzbicki and Abramowicz, 1983) for solving the equations of motion. The static analysis was carried out within the linear range using the implicite type algorithm included in the module of one of computer-aided design packages.

The boom structure was made of steel S355 and WELDOX 700. The static material properties of WELDOX 700 are as follows:

plasticity limit	–	$(R_e)_{min} = 700\,\mathrm{MPa}$
tensile strength	–	$(R_m)_{min} = 780\text{-}930\,\mathrm{MPa}$
extension	–	$A_5 = 14\%$

for E350 steel, the material properties are as follows:

plasticity limit	–	$(R_e)_{min} = 355\,\mathrm{MPa}$
tensile strength	–	$(R_m)_{min} = 490\text{-}630\,\mathrm{MPa}$
extension	–	$A_5 = 20\text{-}22\%$

The dynamic linear analysis of the unit reaction to external excitation was carried out using CAD/FEM system (Rusiński *et al.*, 2000; ,18]. To achieve that, the TRANSIENT RESPONSE analysis was used. It can serve to determine a dynamic response of the unit (shifts, speeds, acceleration, deformations and strains) to the task changing in time, such as shift, speed, acceleration or force (Karliński and Iluk, 2000; Karliński *et al.*, 2006).

The dynamic linear analysis using the finite-element method is carried out in compliance with the following equation of motion

$$\mathbf{K}\boldsymbol{U}(t) + \mathbf{C}\dot{\boldsymbol{U}} + \mathbf{M}\ddot{\boldsymbol{U}}(t) = \boldsymbol{F}(t) \tag{2.1}$$

where $\mathbf{K}$, $\mathbf{M}$ and $\mathbf{C}$ are matrices of rigidity, mass and damping, $\boldsymbol{U}(t)$, $\dot{\boldsymbol{U}}(t)$ and $\ddot{\boldsymbol{U}}(t)$ are displacements, velocities and accelerations, respectively and $\boldsymbol{F}(t)$ is a time function of force.

It is assumed that the matrices of rigidity, mass and damping do not change in time.

Equation (2.1) was solved using the method of direct integration of the equations of motion using the β Newmark algorithm (Rusiński *et al.*, 2000).

The β Newmark method is a more precise linear acceleration method and uses the following equations to calculate the speed and acceleration

$$\dot{\boldsymbol{U}}_{t+\Delta t} = \ddot{\boldsymbol{U}} + (1-\gamma)\Delta t\ddot{\boldsymbol{U}}_t + \gamma\Delta t\ddot{\boldsymbol{U}}_{t+\Delta t} \tag{2.2}$$

and

$$\boldsymbol{U}_{t+\Delta t} = \boldsymbol{U}_t + \dot{\boldsymbol{U}}_t\Delta t + \Big(\frac{1}{2}-\beta\Big)\Delta t^2\ddot{\boldsymbol{U}}_t + \beta\Delta t^2\ddot{\boldsymbol{U}}_{t+\Delta t} \tag{2.3}$$

When we extend Eq. (2.3) so as to express $\ddot{\boldsymbol{U}}_{t+\Delta t}$ using $\boldsymbol{U}_t$, $\dot{\boldsymbol{U}}_t$, $\ddot{\boldsymbol{U}}_t$ and $\boldsymbol{U}_{t+\Delta t}$ components

$$\ddot{\boldsymbol{U}}_{t+\Delta t} = \frac{1}{\beta\Delta t^2}(\boldsymbol{U}_{t+\Delta t} - \boldsymbol{U}_t) - \frac{1}{\beta\Delta t}\dot{\boldsymbol{U}}_t - \Big(\frac{1}{2\beta}-1\Big)\ddot{\boldsymbol{U}}_t \tag{2.4}$$

Using Eq. (2.4), equation (2.1) may take the following form

$$\dot{\boldsymbol{U}}_{t+\Delta t} = \frac{\gamma}{\beta\Delta t}(U_{t+\Delta t} - U_{t+\Delta t}) + \Big(1-\frac{\gamma}{\beta}\Big)\dot{\boldsymbol{U}}_t + \Big(1-\frac{\gamma}{2\beta}\Big)\Delta t\ddot{\boldsymbol{U}}_t \tag{2.5}$$

and henceforth

$$\begin{aligned}\mathbf{K}\boldsymbol{U}_{t+\Delta t} &+ \mathbf{C}\Big[\frac{\gamma}{\beta\Delta t}(\boldsymbol{U}_{t+\Delta t} - \boldsymbol{U}_{t+\Delta t}) + \Big(1-\frac{\gamma}{\beta}\Big)\boldsymbol{U}_t + \Big(1-\frac{\gamma}{2\beta}\Big)\Delta t\ddot{\boldsymbol{U}}_t\Big]\\ &+ \mathbf{M}\Big[\frac{1}{\beta\Delta t^2}(\boldsymbol{U}_{t+\Delta t} - \boldsymbol{U}_t) - \frac{1}{\beta\Delta t}\dot{\boldsymbol{U}}_t - \Big(\frac{1}{2\beta}-1\Big)\ddot{\boldsymbol{U}}_t\Big] = \boldsymbol{R}_{t+\Delta t}\end{aligned} \tag{2.6}$$

When we take all the coefficients to the left-hand side and then abridge, we get

$$\begin{aligned}\Big(\mathbf{K} + \frac{1}{\beta\Delta t^2}\mathbf{M} + \frac{\gamma}{\beta\Delta t}\mathbf{C}\Big)\boldsymbol{U}_{t+\Delta t} &= \boldsymbol{R}_{t+\Delta t}\\ &+ \mathbf{M}\Big[\frac{1}{\beta\Delta t^2}\boldsymbol{U}_t + \frac{1}{\beta\Delta t}\dot{\boldsymbol{U}}_t + \Big(\frac{1}{2\beta}-1\Big)\ddot{\boldsymbol{U}}_t\Big]\\ &+ \mathbf{C}\Big[\frac{1}{\beta\Delta t^2}\boldsymbol{U}_t + \Big(\frac{\gamma}{\beta}-1\Big)\dot{\boldsymbol{U}}_t + \frac{\Delta t}{2}\Big(\frac{\gamma}{\beta}-2\Big)\ddot{\boldsymbol{U}}_t\Big]\end{aligned} \tag{2.7}$$

With known displacements, velocities and accelerations for time t we can determine the displacements for time $t+\Delta t$, and then find the velocities and accelerations for $t+\Delta t$, using Eqs. (2.2) and (2.4).

For the modelled dynamic system, the algorithm takes the following form in this method:

1. Assumption of the initial conditions

   $\boldsymbol{U}_0$, $\dot{\boldsymbol{U}}_0$, $\ddot{\boldsymbol{U}}_0$

2. Calculation of the integration constants for the chosen time Δt

   $a_0 = \dfrac{1}{\beta\Delta t}$, $a_1 = \dfrac{\gamma}{\beta\Delta t}$, $a_2 = \dfrac{\gamma}{\beta\Delta t}$, $a_3 = \dfrac{1}{2\beta} - 1$, $a_4 = \dfrac{1}{\beta} - 1$,

   $a_5 = \dfrac{\Delta t}{2}\Big(\dfrac{\gamma}{\beta} - 2\Big)$

3. Assumption of the effective rigidity matrix $\mathbf{K}^*$ in the form

   $\mathbf{K}^* = \mathbf{K} + a_0\mathbf{M} + a_1\mathbf{C}$

4. Abridging of the rigidity matrix $\mathbf{K}^*$

5. Calculation of the effective force vector for time $t+\Delta t$

   $\boldsymbol{R}^*_{t+\Delta t} = \boldsymbol{R}_{t+\Delta t} + \mathbf{M}(a_0\boldsymbol{U}_t + a_2\dot{\boldsymbol{U}}_t + a_3\ddot{\boldsymbol{U}}_t) + \mathbf{C}(a_1\boldsymbol{U}_t + a_4\dot{\boldsymbol{U}}_t + a_5\ddot{\boldsymbol{U}}_t)$

6. Calculation of displacements for $t+\Delta t$

   $\mathsf{K}^{*}\boldsymbol{U}_{t+\Delta t}=\boldsymbol{R}^{*}_{t+\Delta t}$

7. Calculation of acccelerations for $t+\Delta t$

   $\ddot{\boldsymbol{U}}_{t+\Delta t}=a_0(\boldsymbol{U}_{t+\Delta t}-\boldsymbol{U}_t)-a_2\dot{\boldsymbol{U}}_t-a_3\ddot{\boldsymbol{U}}_t$

8. Calculation of velocities for $t+\Delta t$

   $\dot{\boldsymbol{U}}_{t+\Delta t}=\dot{\boldsymbol{U}}_t+(1-\gamma)\Delta t\dot{\boldsymbol{U}}_t+\gamma\Delta t\ddot{\boldsymbol{U}}_{t+\Delta t}$

9. Steps 5-8 should be repeated for each time period.

The analysis was carried out integrating the equation of motion with changing the time period equal to:

- for the time from 0 to 0.03 s – 480 integration steps with the duration value of 0.0000625 s
- from 0.03 to 0.21 s – 140 integration steps of 0.001285 s
- from 0.21 to 0.27 s – 40 integration steps of 0.0015 s
- from 0.27 to 1.00 s – 100 integration steps of 0.0073 s.

The analysis takes also into consideration damping amounting to 3% of the critical damping for the first form of natural vibrations of the entire structure. The value is based on research carried out on similar objects (the first form of free vibrations of the telescopic booms used in underground mining in mobile anchor machines) (Karliński *et al.*, 2007)

3. FEM strength calculation

The boom discrete model was developed with three- and four-joint shell finite elements, based on the thick shell theory. The average size of the finite element was ca. 30 mm. Because during the strength test the material may become partially plastic (material nonlinearity) and the configuration may significantly change as a result of large deflections (geometrical nonlinearity), all the finite elements used were adopted for calculations with both types of nonlinearity.

The discrete boom models for static and dynamic analysis (Pam-Crash User Manual, 1989) are presented in Figs. 4 and 5, where all necessary connections to the machine platform are included and shown. Different thickness of the metal plates are indicated by colours of finite elements.

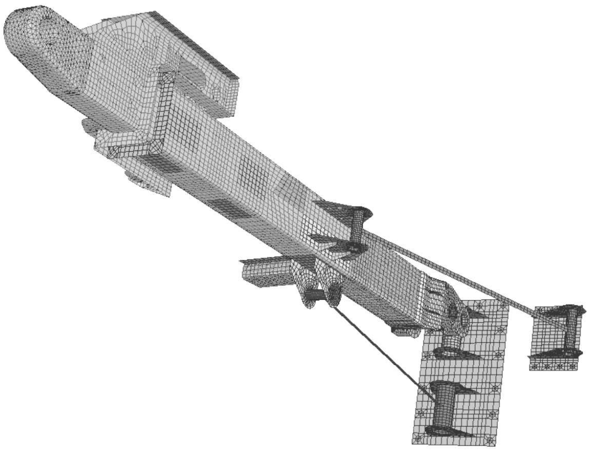

Fig. 5. Discrete model prepared for dynamic analysis

The strength calculations were conducted for the assumed boundary conditions for both static and dynamic cases. The previously mentioned boundary conditions assumed the following (Karliński *et al.*, 2004):

- non-linear strength calculations of the boom structure in the dynamic range: simulation of the machine passing through the rectangular obstacle of height of 150 mm with speed 12 km/h
- linear strength calculations of the boom structure in the static range: simulation of the work of the boom the most adverse configuration – maximum turning moment working on the boom) loaded by the feeding force 15000 N.

The static analysis takes into consideration the load of the structure mass and forces connected with the drill advance for the most adverse drill mast position in relation to the boom (drilling with the boom perpendicular to the wall, the boom protruding to the maximum length) (Derlukiewicz and Karliński, 2004).

The dynamic load affecting the boom, in the case of the dynamic analysis, stems from the machine front wheels going over an obstacle 150 mm high with the maximum speed available for the machine, which is a kinematic coercion, Fig. 6. To achieve the adequate analysis conditions, the entire kinematic system of the boom was modelled (Karliński, 2001). The model of dynamic load taken into consideration in the dynamic calculation is based on the experience related to this type of machines operating in KGHM Polish Cooper mine, where all machines designed by leading producer Mine Master Ltd. are analysed during the designing process according to such a extreme condition.

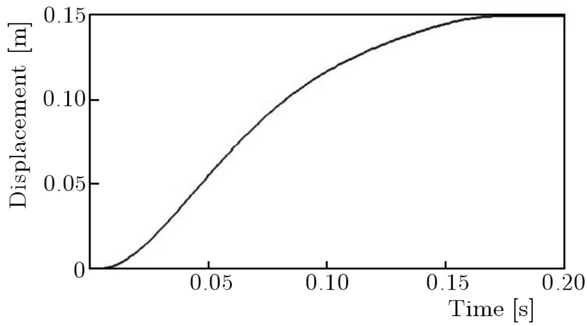

Fig. 6. Characteristics of the kinematic coercion

Exemplary results in the form of contour lines of maximum stresses according to Huber-Misses hypothesis were presented in Figs. 7 and 8 for static analysis. Results for the dynamic analysis in a selected step of time (where Huber-Misses stresses were the highest in the area of the contact of external and internal pipes) are presented in Fig. 9.

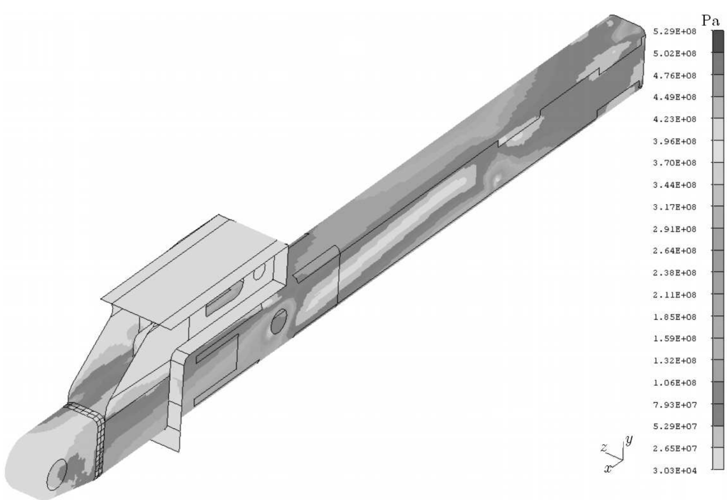

Fig. 7. Stress contour lines according to Huber-Misses hypothesis in the internal tube of the boom – static analysis

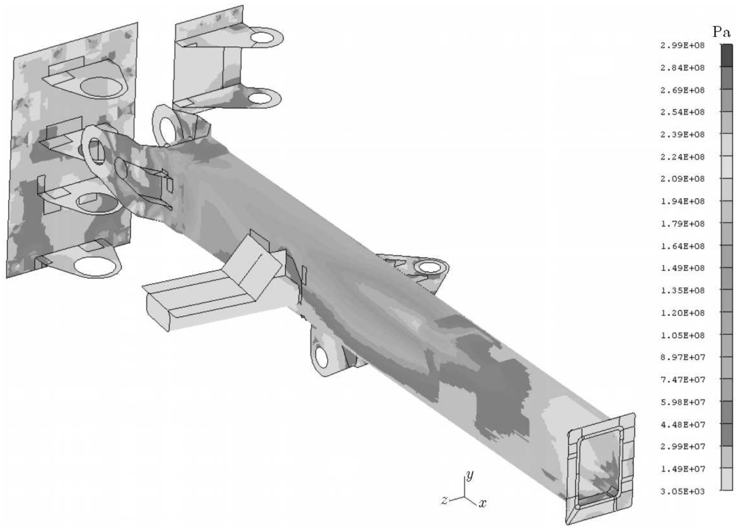

Fig. 8. Stress contour lines according to Huber-Misses hypothesis in the external tube of the boom – static analysis

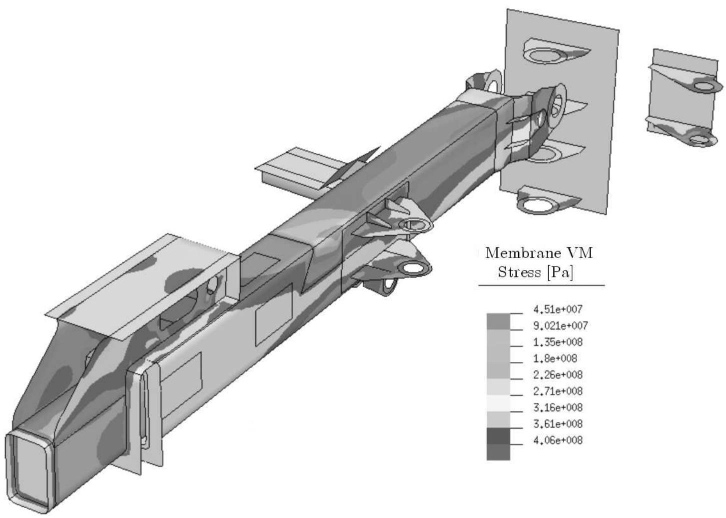

Fig. 9. Maximum stress contour lines according to Huber-Misses hypothesis for a dynamic test in the selected time step $t = 0.153\,\mathrm{s}$

4. Conclusions

The advanced CAD/FEM techniques enable to conduct not only the basic but also sophisticated strength analyses of any structure. In most cases, experimental tests are used to verify the fulfillment of standard requirements for the approval of implementation the machine into practise. With the use of FEM and computer simulations, these structures which do not meet the requirements are eliminated already at the design stage. Sometimes, numerical simulations are the only tests which are done to approve the machine.

Results of FEM calculations are always on the safe side, providing a sufficiently accurate answer to the set loading states and boundary conditions. In virtual models of protective structures, the simplification is limited to models of the material and its behaviour under impact loads and to the quality of manufacturing technology.

The study of the boom stress resulted in the development of a boom with a sufficient number of degrees of freedom (Fig. 10). The boom strength requirements were met thanks to use of a computer-aided design combined with advanced strength analysis carried out using the finite-element method.

Fig. 10. Mining machine Face Master 2.5 with two booms installed (www.minemaster.eu [13])

This problem was solved thanks to the cooperation between the designers and the entity preparing calculations. Simultaneous designing and strength analysis enabled obtaining of the optimum shape of the structure taking into consideration the costs and technologies of manufacturing in a short period of time. Thanks to that it was possible to avoid the costs related to constructing a failed product, which in this case, is the boom (Karliński *et al.*, 2008; Karliński and Wach, 2006; Koziolek *et al.*, 2010). The calculations made confirm the usefulness of numerical methods, including the finite-element method for both the dynamic and static analysis of complex issues of the structure mechanics (physical, geometrical nonlinearities and contact).

In such circumstances, the most important part is to establish the design guidelines (kinematic requirements, choice of materials and the power transmission method, etc.) and to define appropriately and adequately boundary conditions for the structure model. It is also necessary to take into consideration the geometrical and physical nonlinearity, and to choose an appropriate model describing material properties specifying the type of material characteristics and the phenomena taking place such as isotropic hardening, hardening proportional to the speed of deformation etc.

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Statyczna i dynamiczna analiza wysięgnika teleskopowego samojezdnej maszyny górniczej

Streszczenie

W artykule zaprezentowano wybrane zagadnienia dotyczące metodyki wykonywania analiz wytrzymałościowych wysięgników samojezdnych maszyn górniczych przy pomocy metody elementów skończonych. Przedstawiono zasady budowania modeli obliczeniowych. W pracy zaprezentowano przykłady liczbowe obejmujące analizy wytrzymałościowe wykonane dla obciążeń statycznych i dynamicznych.

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