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Numerical Tests of Roadheader's Boom Vibrations

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Abstract

The work presents a dynamic model of a telescopic boom of a roadheader. The boom represents a load–carrying structure of cutting heads and of their drive system. Together with the cutting heads' drive, it represents a cutting system of a roadheader performing the roadheader's basic function, that is cutting the heading face. A physical model with a discrete structure was created for the purpose of analysing the vibrations accompanying the operation of a roadheader. Due to the design of the telescopic boom, three vibrating masses are distinguished in this model concentrated in the centre of gravity of rigid bodies representing: the fixed part of the boom, the extendable part (telescope) and a reduction gear (with transverse cutting heads mounted in the output shaft journals) fitted to the extendable part of the boom. It is a spatial model with 18 degrees of freedom. The mathematical model established was used in simulation tests the aim of which was to identify the value and sources of vibrations in the selected structural nodes of the boom during the performance of a working process. The excitation of vibrations is an effect of a computer simulation of the rock cutting process with transverse heads with the set stereometry. The article presents selected results of numerical tests using the established dynamic model.

Keywords: roadheader, boom, dynamic model, dynamic loads, vibrations

1. Introduction

Roadheaders are the fundamental cutting machines used in mechanised technologies for tunnelling in underground mines and civil engineering. The key process carried out by such type of machine is the cutting of rock deposited in the cross section of the drilled headings. In case of boom–type roadheaders commonly used in hard coal mines, cutting is accomplished with picks mounted to cutting heads. The heads are mounted at the end of a boom which is inclined in the line parallel and perpendicular to the floor, hence they can be advanced along the face surface of the drilled heading along any trajectory. A roadheader boom represents a load–carrying structure for cutting heads and their drive system. It ensures the required range of cutting determining the maximum size and shape of the drilled tunnel.

The drive of cutting heads, their carrying structure (boom) and other roadheader subassemblies functionally related to the cutting system (including boom swinging mechanisms) are subject to the strong excitations of vibrations accompanying the working process carried out. The process of rock cutting is indeed a source of high dynamic loads, especially when cutting rocks with poor workability. The excitation of vibrations is of a stochastic nature and similarly, as in the case of, e.g. machines for earthworks (excavators, bulldozers, etc.), it results to a large degree from the properties of the worked medium [1]. For the purpose of a dynamic state analysis of cutting machines, they are considered, however, as determined (polyharmonic), provided the cutting process conditions remain unchanged [2]. An excessive dynamic load and overloads resulting from cutting process performance may lead to a boom's failure conditions caused by immediate or fatigue damages to its parts. An analysis of dynamic states of a roadheader cutting system (cutting heads' drive and their load–carrying structure – boom) is indispensable for assessing the design correctness of the roadheaders currently produced and sets a starting point for developing new design solutions.

2. Dynamic model of roadheader boom

A dynamic model of a roadheader boom consists of a physical model and a mathematical model. Three rigid bodies connected with each other with weightless viscoelastic elements are distinguished in the structure of the physical model (Fig. 1) [3]. The bodies represent the key parts of a roadheader boom, i.e.: fixed part (1), extendable part (2) and a reduction gear (3) in the cutting head (6) drive attached to extendable part. The fixed part of the boom is mounted to the movable part of the turntable by means of two slide bearings and is supported with two hydraulic lifting cylinders (4) – a right one (SPR) and a left one (SPL). The extendable part of the boom (2) is seated as sliding in the fixed part of the boom (1), and its extended by means of a telescopic mechanism's cylinders (5) – a right one (STR) and a left one (STL). The length of the boom may change within the range of L_1 to $L_1+\Delta L_1$.

The fixed part of the boom has the form of a rigid body with the mass m_{WS}, concentrated in its centre of gravity O_{WS}, and with moments of inertia I_{WSX}, I_{WSY} and I_{WSZ}, determined in relation to the parallel axes to the axis of the system of coordinates $X_W Y_W Z_W$, passing through the point O_{WS}. The mounting of the fixed part of the boom to the turntable is modelled as three viscoelastic constraints with specific rigidity marked, respectively, as: k_{WSX}, k_{WS1} and k_{WS2} and with the damping coefficients: c_{WSX}, c_{WS1} and c_{WS2}. The boom lifting actuators are modelled as viscoelastic elements with substitute specific rigidity k_{SPR} (an actuator on the right side of the boom) and k_{SPL} (an actuator on the left side of the boom) and the damping coefficient: c_{SPR} and c_{SPL}. The elastic properties of the working fluid in the cylinder were considered in the model of the hydraulic cylinders, along with piston rod flexibility and deformability of the cylinder walls [4]. The substitute values of the parameters mentioned are determined as for a serial connection of flexible parts.

The coupling of the fixed part of the boom with the extendable part was modelled as 16 viscoelastic constraints with the specific rigidity k_{Ti} and the damping coefficient c_{Ti} (for i=1, 2, ..., 16). The constraints are arranged in the same point of application of resultant reactive forces in the fulcrums of the extendable part relative to the fixed part of the boom. Eight of the reactions are parallel to the plane $Y_W Z_W$ (points 1 – 8), while others are parallel to the plane $X_W Y_W$. The elements which are modelling a connection of the extendable part with the fixed part of the boom carry compressive loads only.

The possible occurrence of clearances in this connection is predicted. The temporary location of a body modelling the fixed part of the boom is described by means of six coordinates: three translation coordinates – x_{WS}, y_{WS} and z_{WS} and three rotation coordinates – φ_{WSX}, φ_{WSY}, and φ_{WSZ} (six degrees of freedom).

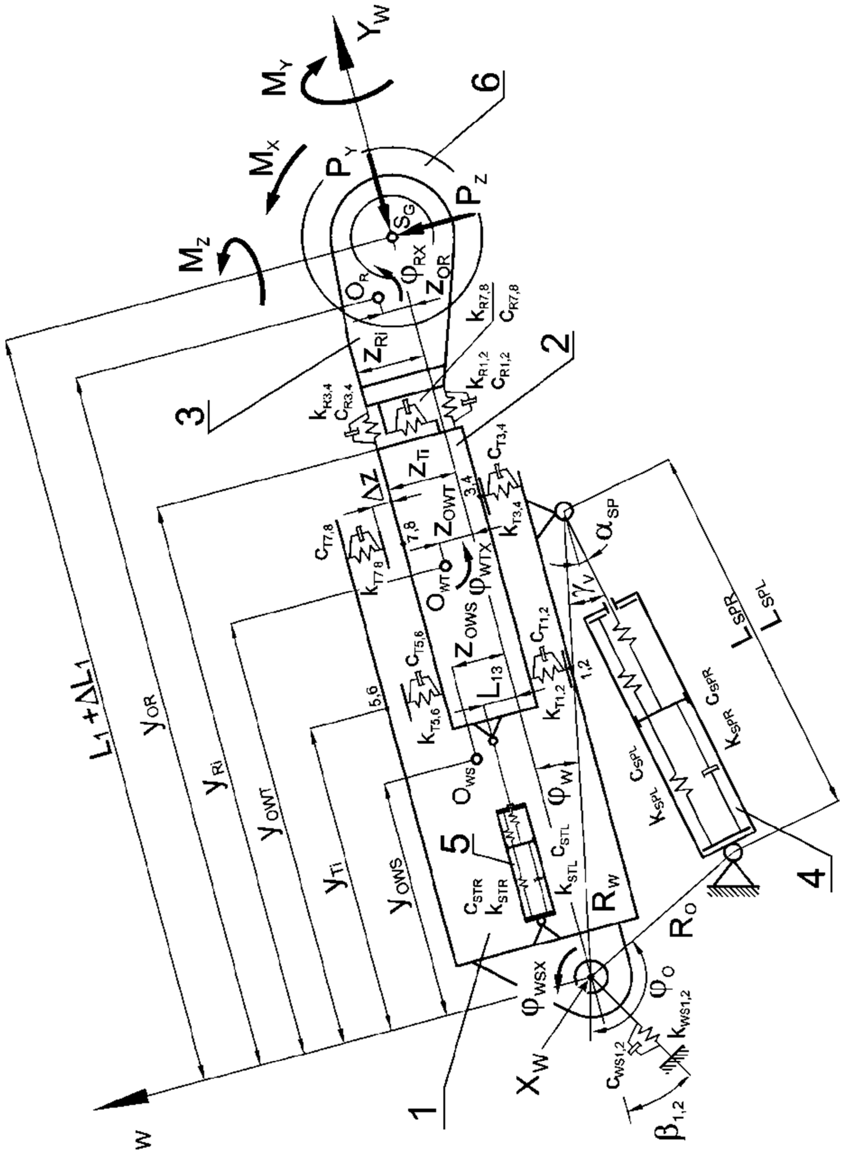

Figure 1. Physical model of a roadheader telescopic boom [3]
1 – fixed part, 2 – extendable part (telescope), 3 – reduction gear in the drive of transverse cutting heads, 4 – hydraulic lifting cylinders, 5 – hydraulic cylinders of telescopic mechanism, 6 – cutting heads

The extendable part of the boom is modelled as a rigid body with the mass m_{WT} concentrated in its centre of gravity O_{WT} and with the moments of inertia: I_{WTX}, I_{WTY} and I_{WTZ}. The location of this part relative to the fixed part of the boom is determined by the length of telescopic mechanism cylinders: L_{STP} and L_{STL}. Similar, as in the case of boom lifting cylinders, the dynamic properties of boom extension are characterised with substitute rigidities k_{STR} and k_{STL} and damping coefficients c_{STR} and c_{STL}. The extendable part of the boom also has six degrees of freedom. The temporary location of this mass is hence described with the six coordinates: x_{WT}, y_{WT}, z_{WT}, φ_{WTX}, φ_{WTY}, φ_{WTZ}.

The third element of a physical model of the telescopic boom is a reduction gear together with transverse cutting heads mounted in the output shaft journals. It is modelled as a rigid body with the mass m_R concentrated in its centre of gravity O_R and the moments of inertia I_{RX}, I_{RY} and I_{RZ}. The reduction gear is coupled with the extendable part of the boom by means of eight weightless viscoelastic elements with the specific rigidity k_{Ri} and the damping coefficient c_{Ri} (i=1, 2, ..., 8). The temporary location of the considered body is defined here with the following coordinates: x_R, y_R and z_R as well as φ_{RX}, φ_{RY}, and φ_{RZ}.

A spatial discreet model of the telescopic boom is subject to the activity of vibration excitations from the external load which are the result of working process performance (cutting the heading face). This load was reduced to the intersection point of the boom longitudinal axis with an axis of rotation of cutting heads and was described with six components – the concentrated forces applied in the point S_G (P_X, P_Y and P_Z) and the moments of forces (M_X, M_Y and M_Z).

The equations of motion in the physical model were written using the Lagrange second order equation. A system of 18 ordinary nonlinear second–order differential equations were obtained this way. For example, the equations of motion for the extendable part of the boom (telescope) assume the following form:

$$\begin{aligned} m_{WT}\cdot\ddot{x}_{WT}-\sum_{i=9}^{16}\{H[A_{Ti}]\cdot(k_{Ti}\cdot A_{Ti}+c_{Ti}\cdot B_{Ti})\}+\sum_{i=5}^{6}(k_{Ri}\cdot A_{Ri}+c_{Ri}\cdot B_{Ri})+ \\ +Q_{WTXW}+W_X\cdot Sign(\dot{x}_{WT}-\dot{x}_{WS})=0 \end{aligned} \tag{1}$$

$$\begin{aligned} m_{WT}\cdot\ddot{y}_{WT}-k_{STP}\cdot A_{ST}-c_{STP}\cdot B_{ST}-k_{STL}\cdot A_{ST}-c_{STL}\cdot B_{ST}+\sum_{i=1}^{4}(k_{Ri}\cdot A_{Ri}+c_{Ri}\cdot B_{Ri})+ \\ +Q_{WTYW}+W_Y\cdot Sign(\dot{y}_{WT}-\dot{y}_{WS})=0 \end{aligned} \tag{2}$$

$$\begin{aligned} m_{WT}\cdot\ddot{z}_{WT}-\sum_{i=1}^{8}\{H[A_{Ti}]\cdot(k_{Ti}\cdot A_{Ti}+c_{Ti}\cdot B_{Ti})\}+\sum_{i=7}^{8}(k_{Ri}\cdot A_{Ri}+c_{Ri}\cdot B_{Ri})+ \\ +Q_{WTZW}+W_Z\cdot Sign(\dot{z}_{WT}-\dot{z}_{WS})=0 \end{aligned} \tag{3}$$

$$\begin{aligned} I_{WTX}\cdot\ddot{\varphi}_{WTX}-k_{STP}\cdot A_{ST}\cdot(z_{OWT}-L_{13})-c_{STP}\cdot B_{ST}(z_{OWT}-L_{13})\ + \\ -k_{STL}\cdot A_{ST}\cdot(z_{OWT}-L_{13})-c_{STL}\cdot B_{ST}(z_{OWT}-L_{13})+ \\ +\sum_{i=1}^{8}\{H[A_{Ti}]\cdot(k_{Ti}\cdot A_{Ti}+c_{Ti}\cdot B_{Ti})\cdot(y_{OWT}-y_{Ti})\}+\sum_{i=1}^{2}[(k_{Ri}\cdot A_{Ri}+c_{Ri}\cdot B_{ri})\cdot(z_{OWT}+z_{Ri})]+ \\ +\sum_{i=3}^{4}[(k_{Ri}\cdot A_{Ri}+c_{Ri}\cdot B_{ri})\cdot(z_{OWT}-z_{Ri})]-\sum_{i=7}^{8}[(k_{Ri}\cdot A_{Ri}+c_{Ri}\cdot B_{ri})\cdot(y_{OWT}-y_{Ri})]+W_{TX}=0 \end{aligned} \tag{4}$$

$$\begin{aligned} & I_{WTY} \cdot \ddot{\varphi}_{WTY} + \sum_{i=1}^{4} \left\{ H[A_{Ti}] \cdot \left(k_{T(2i-1)} \cdot A_{T(2i-1)} + c_{T(2i-1)} \cdot B_{T(2i-1)} \right) \cdot x_{T(2i-1)} \right\} + \\ & \quad - \sum_{i=1}^{4} \left\{ H[A_{Ti}] \cdot \left(k_{T(2i)} \cdot A_{T(2i)} + c_{T(2i)} \cdot B_{T(2i)} \right) \cdot x_{T(2i)} \right\} + \\ & \quad - \sum_{i=5}^{8} \left\{ H[A_{Ti}] \cdot \left(k_{T(2i-1)} \cdot A_{T(2i-1)} + c_{T(2i-1)} \cdot B_{T(2i-1)} \right) \cdot \left(z_{T(2i-1)} - z_{OWT} \right) \right\} + \\ & \quad + \sum_{i=5}^{8} \left\{ H[A_{Ti}] \cdot \left(k_{T(2i)} \cdot A_{T(2i)} + c_{T(2i)} \cdot B_{T(2i)} \right) \cdot \left(z_{T(2i)} + z_{OWT} \right) \right\} + \\ & \quad - \sum_{i=5}^{6} \left(k_{Ri} \cdot A_{Ri} + c_{Ri} \cdot B_{Ri} \right) \cdot \left(z_{OWT} - z_{Ri} \right) + \\ & \quad - \left(k_{R7} \cdot A_{R7} + c_{R7} \cdot B_{R7} \right) \cdot x_{R7} + \left(k_{R8} \cdot A_{R8} + c_{R8} \cdot B_{R8} \right) \cdot x_{R8} = 0 \end{aligned} \tag{5}$$

$$\begin{aligned} & I_{WTZ} \cdot \ddot{\varphi}_{WTZ} - \sum_{i=9}^{16} \left\{ H[A_{Ti}] \cdot \left(k_{Ti} \cdot A_{Ti} + c_{Ti} \cdot B_{Ti} \right) \cdot \left(y_{OWT} - y_{Ti} \right) \right\} + \\ & \quad + \sum_{i=1}^{2} \left[\left(k_{R(2i-1)} \cdot A_{R(2i-1)} + c_{R(2i-1)} \cdot B_{R(2i-1)} \right) \cdot x_{R(2i-1)} \right] + \\ & \quad - \sum_{i=1}^{2} \left[\left(k_{R(2i)} \cdot A_{R(2i)} + c_{R(2i)} \cdot B_{R(2i)} \right) \cdot x_{R(2i)} \right] + \\ & \quad + \sum_{i=5}^{6} \left[\left(k_{Ri} \cdot A_{Ri} + c_{Ri} \cdot B_{Ri} \right) \left(y_{OWT} - y_{Ri} \right) \right] + W_{TZ} = 0 \end{aligned} \tag{6}$$

where:

A_{Ti}, B_{Ti}, A_{Ri}, B_{Ri} – auxiliary values connecting the coordinates of individual vibrating masses and their speeds;

A_{ST}, B_{ST} – auxiliary values describing displacement and speed of piston rods of telescopic mechanism cylinders;

$H[A_{Ti}]$ – the Heaviside function modelling a possibility of conveying compressive loads only by the nodes situated in the fulcrums of the extendable part relative to the fixed part;

Q_{WTXW}, Q_{WTYW}, Q_{WTZW} – components of the gravity force of the boom extendable part in the direction of the axis of the reference system $X_W Y_W Z_W$;

W_X, W_Y, W_Z, W_{TX}, W_{TY}, W_{TZ} – components of friction forces in the fulcrums of the extendable part relative to the boom fixed part and moments of friction forces relative to axes parallel to the axis of the system of coordinates $X_W Y_W Z_W$.

The excitation curve of boom vibrations (components of an external load) is determined during a simulation of the process of cutting the heading face surface with cutting heads. The values of load components of the picks taking part in the cutting process are determined starting with the projections of cuts made by individual cutting head picks based on the values of cut parameters determined on the basis thereof. Next, by simulating the rotary motion of a cutting head, the curves of the following forces are determined numerically: P_X, P_Y, P_Z, M_X, M_Y and M_Z being an external load of the roadheader boom in the dynamic model created.

3. Analysis of dynamic state of telescopic boom

The cutting of rock was simulated for the purpose of determining dynamic loads and vibrations in constructional nodes of the telescopic boom with the rock compressive strength of R_c=120 MPa with transverse heads fitted with 80 picks. The cutting of a rock layer with the height of h=0.45 m and with the web of z=0.2 m in the working motion was accompanied by boom deflection in a plane parallel to the floor with the angular velocity of ω_{ow}=0.022 rad/s. A rotary motion of the cutting head was simulated here with the angular velocity of ω_G=4.4 rad/s. It was also assumed that the telescope is completely extended, so that boom length is $L_1+\Delta L_1$=4 m.

Figure 2 shows fragments of vibration curves in the intersection points of the boom's longitudinal axis with the axis of rotation of cutting heads (point S_G). The vibrations are the response of the examined system to an excitation from cutting. The load generated with the cutting process is exciting the boom to vibrations in all the considered directions, with the largest displacement of the point S_G along the axis Z_W (Fig. 2a – dotted thick line). As far as the maximum displacement of this point in the direction of the axis X_W and Y_W is, respectively:10^{-2} and $-6\cdot10^{-3}$ m, then in the direction of the axis Z_W it reaches $4\cdot10^{-2}$ m (the symbol "–" means displacement in the direction opposite to the assumed turn of the axis of the system $X_WY_WZ_W$). A much higher level of vibrations in the direction of the axis Z_W results from the fact that the investigated system is highly flexible in the lifting plane (Y_WZ_W), which is a result of supporting the boom with hydraulic cylinders. Due to relatively low rigidity of the cylinders as compared to other elastic constraints, the boom is performing high–amplitude rotational vibrations around the axis X_W. A manner of loading the external boom is also important. The forces exciting the vibrations, coming from cutting resistance, reduced to the point S_G, are acting on a large arm (equivalent to the boom length). The boom is, therefore, subject to the activity of a large torque in the plane Y_WZ_W.

The velocity of vibrations in the point S_G of the boom, in the direction of the axis X_W, ranges between ±0.13 m/s (Fig. 2b – thin continuous line), while acceleration varies within the range of –17 to +8 m/s^2 (Fig. 2c – thin continuous line). The effective values of velocity and acceleration of vibrations in this case are: 0.05 m/s and ~3 m/s^2. The maximum values, according to a relative value, of velocity and acceleration of vibrations in the direction of the axis Y_W are, respectively: 0.3 m/s and 70 m/s^2 (thick continuous line), with the effective values of such values of: 0.11 m/s and 17 m/s^2. On the other hand, the maximum values of the analysed boom vibration parameters in the direction of the axis Z_W equal to: 0.6 m/s and 75 m/s^2 (thick dotted line), while the effective vibration speed and acceleration values in this direction are, respectively: 0.2 m/s and 18 m/s^2. Boom vibrations in the direction of the axis X_W are weakest. It is because the effective values of the parameters characterising the vibration motion in this direction are 4 times (velocity) and 6 times (acceleration) smaller as compared to the effective values determined for vibrations in the direction of the axis Z_W. At the same time, the effective values of vibration velocity and acceleration in the direction of the axis Y_W are only by 45% and 6% smaller compared to the effective values of such parameters for vibrations in the direction of the axis Z_W. The highest intensity have, therefore, the vibrations in the boom lifting plane perpendicular to the displacement direction of cutting heads in the working motion.

A spectral analysis of the studied curves has revealed that a number of characteristic vibration components exists (Fig. 3). The share of such components varies for relevant vibration directions. For example, vibrations with the frequency of 35 rad/s (Fig. 3a) dominate in the vibration acceleration spectrum in the direction of the axis X_W. Components are also evident with the frequency of: 53; 87; 139 and 244 rad/s. Meanwhile, three components with the frequency of: 35; 209 and 297 rad/s dominate in the vibration acceleration spectrum in the direction of the axis Z_W (Fig. 3b). The following vibrations: 35; 87 and 139 rad/s are the first three own frequencies of the studied system. Vibrations with the frequency of 53 rad/s result from the fact that picks are positioned along the helixes with a small twisting angle (helix frequency), and vibrations with the frequency of 209 rad/s are the result of the next picks advancing to the cutting zone (pick frequency). The other identified vibration components are higher harmonics of vibration excitations.

4. Conclusions

The created dynamic model allowed to perform comprehensive simulation tests in order to determine dynamic loads in the constructional nodes of a roadheader telescopic boom and analyse its vibrations. The numerical tests carried out allowed to identify the basic sources of boom vibrations and – for the set cutting process performance conditions – to determine the values of the parameters characterising telescopic boom vibrations with the defined constructional form. Conclusions from simulation tests were used for formulating requirements for dynamic properties of a telescopic boom for a newly designed roadheader in terms of reduction in dynamic loads and vibrations.

The presented dynamic model of a roadheader's telescopic boom has been created for the design of roadheader's telescopic boom. The usefulness of this model and the reliability of the results will be soon verified based on the results of experimental studies of roadheader after manufacture developed telescopic boom.

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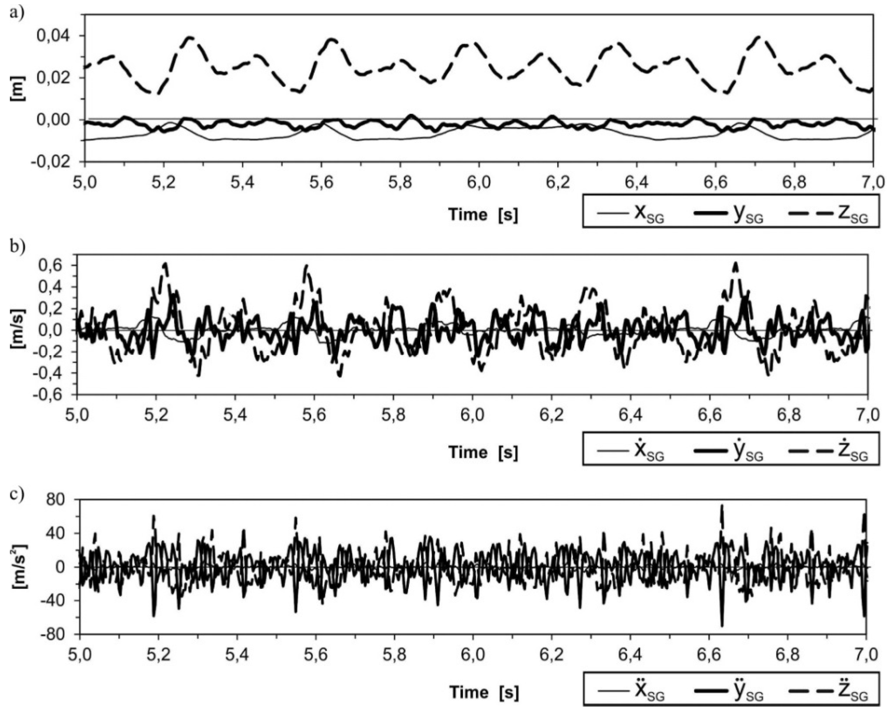

Figure 2. Boom vibration components in the intersection point of the boom longitudinal axis with the axis of rotation of cutting heads: a) displacement, b) speed, c) acceleration

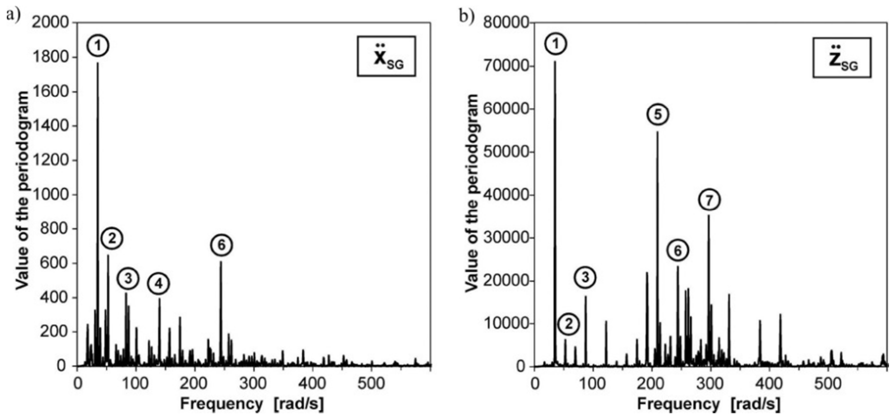

Figure 3. Spectrum of vibration acceleration in the direction of the axis: a) X_W, b) Z_W
1 – 35 rad/s 2 – 53 rad/s 3 – 87 rad/s 4 – 139 rad/s 5 – 209 rad/s 6 – 244 rad/s 7 – 297 rad/s