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Arch. Min. Sci. **62** (2017), 4, 775-793

Electronic version (in color) of this paper is available: http://mining.archives.pl

DOI 10.1515/amsc-2017-0054

PIOTR CHELUSZKA*#

THE ANTI-RESONANCE CRITERION IN SELECTING PICK SYSTEMS FOR FULLY OPERATIONAL CUTTING MACHINERY USED IN MINING

KRYTERIUM ANTYREZONANSOWE W DOBORZE UKŁADU NOŻY NA ORGANACH ROBOCZYCH GÓRNICZYCH MASZYN URABIAJĄCYCH

This article discusses the issue of selecting a pick system for cutting mining machinery, concerning the reduction of vibrations in the cutting system, particularly in a load-carrying structure at work. Numerical analysis was performed on a telescopic roadheader boom equipped with transverse heads. A frequency range of the boom's free vibrations with a set structure and dynamic properties were determined based on a dynamic model. The main components excited by boom vibrations, generated through the process of cutting rock, were identified. This was closely associated with the stereometry of the cutting heads. The impact on the pick system (the number of picks and their arrangement along the side of the cutting head) was determined by the intensity of the external boom load elements, especially in resonance zones. In terms of the anti-resonance criterion, an advantageous system of cutting head picks was determined as a result of the analysis undertaken. The correct selection of the pick system was ascertained based on a computer simulation of the dynamic loads and vibrations of a roadheader telescopic boom.

Keywords: dynamics, roadheader, telescopic boom, pick system, excitation of vibrations, resonance vibrations

W artykule zajęto się zagadnieniem doboru układu noży organów roboczych maszyn urabiających w aspekcie redukcji drgań występujące w układzie urabiania, a w szczególności w jego ustroju nośnym podczas realizacji procesu roboczego. Badania numeryczne zrealizowane zostały na przykładzie wysięgnika teleskopowego kombajnu chodnikowego wyposażonego w głowice poprzeczne. W oparciu o model dynamiczny określone zostały zakresy częstości drgań własnych wysięgnika o zadanej strukturze i własnościach dynamicznych. Dokonana została identyfikacja głównych składowych wymuszenia drgań wysięgnika generowanych procesem urabiania skały, które powiązane zostały ze stereometrią głowic urabiających. Określono wpływ układu noży (liczby noży oraz ich sposobu rozmieszczenia na pobocznicy głowicy urabiającej) na intensywność składowych obciążenia zewnętrznego wysięgnika, szczególnie w jego obszarach rezonansowych. W efekcie zrealizowanych badań określony został korzystny, ze wzglę-

* SILESIAN UNIVERSITY OF TECHNOLOGY, FALCULTY OF MINING AND GEOLOGY, DEPARTMENT OF MINING MECHANISATION AND ROBOTISATION, AKADEMICKA 2, 44-100 GLIWICE, POLAND

Corresponding author: Piotr.Cheluszka@polsl.pl

du na kryterium antyrezonansowe, układ noży na głowicy urabiającej. Poprawność doboru układu noży potwierdzona została w oparciu o symulację komputerową obciążeń dynamicznych i drgań wysięgnika teleskopowego kombajnu chodnikowego.

Słowa kluczowe: rezonans, drgania, kombajn chodnikowy, wysięgnik teleskopowy, dynamika, układ noży

1. Introduction

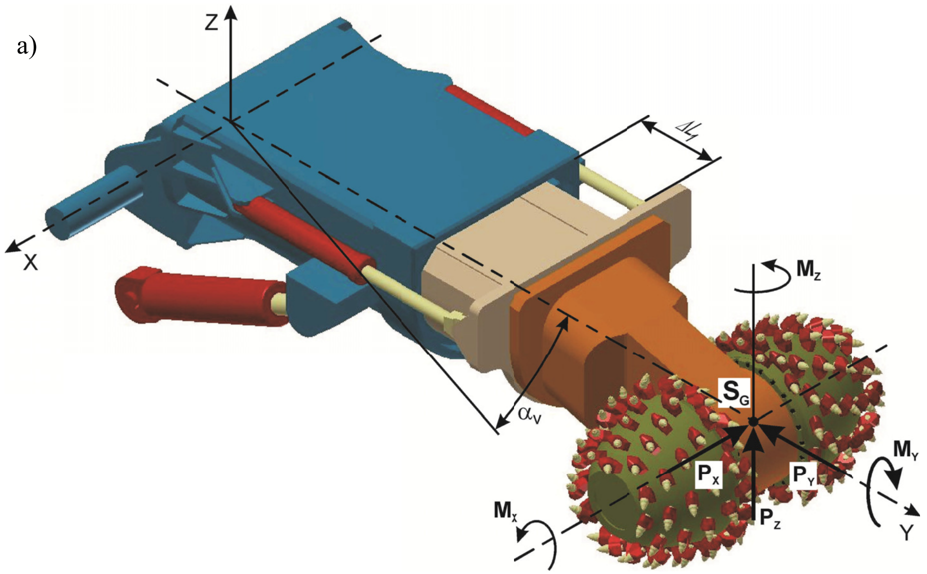
The process of cutting rock with roadheaders is highly dynamic. The value and character of dynamic loads and the vibrations experienced in a drive system, as well as in the load-carrying structure of a roadheader cutting system, depend on both the conditions of the cutting process and the dynamic properties of the subassembly (mass, spring and damping parameters). The progress of the cutting process depends upon the mechanical properties of the rock being excavated, the operational parameters of the roadheader (angular speed and advancing speed of cutting heads, size of the web and height of cut) and the roadheader's power variable (output power, active shaft torque and advancing force). The construction features of active machinery is important (cutting heads for roadheaders, cutting drums for longwall shearers). This especially relates to their stereometry, i.e. the number, arrangement and setting of picks, as well as, their geometry.

The designed system of picks for the cutting heads/drums of roadheaders/shearers for particular operating conditions should take into consideration a number of criteria to ensure the ability to display the expected functional and operating features, while satisfying the conditions of a given solution's technical feasibility (Dolipski et al., 2007). Under consideration here is the optimisation of multi-dimensional functionality in which the ideal solution accounts for a number of objective functions.

When allowing for the dynamics of mining machinery, the anti-resonance criterion is fundamental to the design of pick systems. The phenomenon of resonance strongly amplifies vibrations, due to the fact that the excitation frequency of vibrations originating from the work performed is in tune with the frequency of free vibrations from the system subjected to the activity of the respective excitation. This may increase dynamics loads on the machine's constructed joints, resulting in overloads and leading to immediate or long-term damage of various elements. This relates to drivetrains, as well as the load-carrying structure of the machinery.

In the case of boom-type roadheaders and longwall shearers in which cutting heads/drums are placed on the end of a deflected boom/arm, states of resonance are created by higher lateral and/or longitudinal vibrations which affect the values of operating parameters performed during the mining process. This, in turn, may gradually lead to stronger vibrations.

This article discusses the terms of selection for a system of picks running along the side of roadheader transverse cutting heads in order to eliminate boom resonance vibrations on the cutting system. The object studied was a telescopic boom designed for a roadheader with a cutting system capacity of 200÷250 kW (Fig. 1a). Numerical analysis was performed on transverse cutting heads with a maximum pick casing diameter of 1100 mm and length of 750 mm. Hard rocks were considered for the cutting process. Computer simulations included cutting a heading face layered parallel to the floor (Fig. 1b) with set z ranging between 0.1 and 0.2 m, for cutting heads with an angular speed of $\omega_G = 4.4$ rad/s and their advancing speed on a plane parallel to the floor v_{ow} over a wide range, i.e. from 0.05 to 0.25 m/s. A dynamic model of the roadheader telescopic boom described in the work (Dolipski et al., 2014) was used to evaluate the dynamic state of the boom.



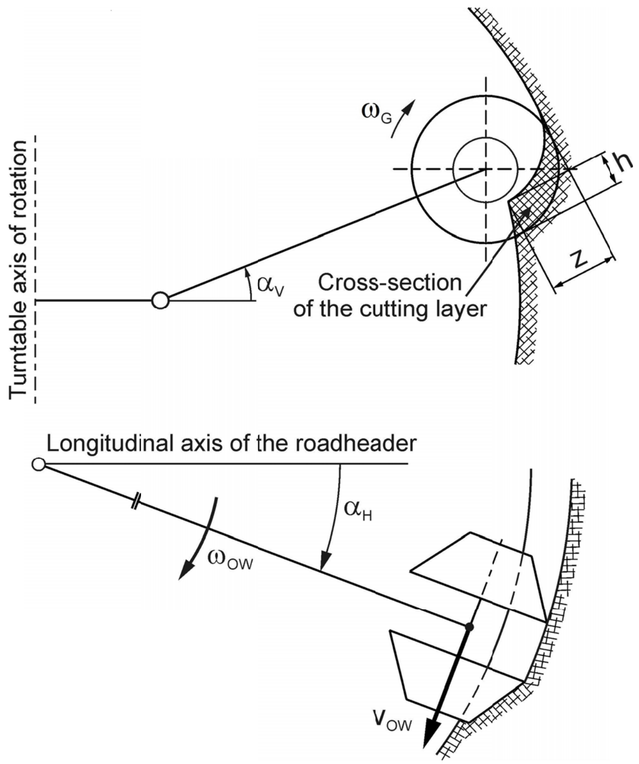


Fig. 1. The study object – roadheader telescopic boom (a) and cutting process parameters (b): z – web of cut, h – cut height, ω_G, v_{OW} – angular speed and advancing speed of cutting heads, α_H, α_V – deflection angles of the boom, ω_{OW} – angular speed of deflection of the boom in the plane parallel to the floor

2. Modal analysis of a roadheader boom

The starting point for evaluating states of resonance from a roadheader subject to the vibration excitations experienced in excavation, and the possibility of eliminating them, will determine the frequency of its free vibrations. Analysis was performed using a spatial discrete physical model of a telescopic boom. Three vibrating masses were distinguished in this model with rigid solids modelling the fixed part of the boom, the extendable part (telescope) and the reduction gear casing in the drive of the cutting heads, together with the heads themselves. The coupling of individual vibrating masses were modelled as weightless viscoelastic constraints representing the boom supports, the guiding of the extendable part relative to the fixed part, the mounting of the reduction gear and cylinders of the boom deflection and telescopic mechanisms. The model studied was a system containing internal elastic and dissipation couplings, with 18 degrees of freedom. For the purpose of implementing computer simulations, it was assumed that energy dissipation mainly took place in the fluid of the hydraulic cylinders (the damping in the supports of particular boom components and the reduction gear mounting were omitted). The fluid damping coefficient values in the hydraulic cylinders of the boom lifting mechanism and the telescopic mechanism were determined based on data provided by the following work (Bartnicki et al., 2010).

The physical model considered was subject to movement from vibration excitations from an external load which was the result of the work performed (cutting the heading face of a excavated tunnel). This load was reduced to a point of intersection between the boom's longitudinal axis (axis Y – Fig.1a) with the axis of rotation of the cutting heads (point S_G) and was described with six components – concentrated forces (P_X, P_Y and P_Z) and moments of forces (M_X, M_Y and M_Z) (Dolipski et al, 2014).

The spatial movement of masses in the physical model considered is described with the arrangement of the following ordinary nonlinear second-order differential equations:

$$\mathbf{M} \cdot \ddot{q}_i + \mathbf{C} \cdot \dot{q}_i + \mathbf{K} \cdot q_i = \mathbf{F} \quad \text{for} \quad i = 1,2,\ldots,18 \tag{1}$$

where:

M, **C**, **K** — the matrices of: inertia, damping and stiffness,
F — vector of components of vibration excitation forces,
$q, \dot{q}, \ddot{q}$ — vectors of generalised: displacement, speed and acceleration.

When additional assumptions are included while clearances and frictional forces in guiding the telescope are omitted, assumptions formulated in the work are then satisfied for a modal analysis of the structural model of the object. The frequencies of free vibrations from the model were determined in a theoretical manner. This consisted of solving an eigenvalue problem for a linear model (Findeisen, 2000). For this purpose, the equations of motion in the physical model were initially transformed into algebraic form, as a result of applying the Laplace transform. The elements of the transfer function matrix $\Phi(s)$ of the system considered, were determined in the next stage as quotient of the minor of a principal determinant in regard to the system of equations of motion and this principal determinant (Marchelek, 1991):

$$\Phi_{ik}(s) = \frac{\Delta_{ik}(s)}{\Delta(s)} \quad \text{for} \quad i,k = 1,2,\ldots,18 \tag{2}$$

After the expansion of the numerator and denominator of the expression (2) to polynomials, after including the Fourier transform ($s = j \times \omega$) and after appropriately grouping the terms of such polynomials, the particular elements of a transfer function matrix take the following form:

$$\Phi_{ik}(j\omega) = \frac{\mathrm{Re}_{M(ik)} \cdot \mathrm{Re}_{WG} + \mathrm{Im}_{M(ik)} \cdot \mathrm{Im}_{WG}}{\mathrm{Re}_{WG}^2 + \mathrm{Im}_{WG}^2} - j \cdot \frac{\mathrm{Re}_{M(ik)} \cdot \mathrm{Im}_{WG} - \mathrm{Re}_{WG} \cdot \mathrm{Im}_{M(ik)}}{\mathrm{Re}_{WG}^2 + \mathrm{Im}_{WG}^2} \tag{3}$$

while:

$$\begin{aligned} \mathrm{Re}_{WG} &= 1 - A_2 \cdot \omega^2 + A_4 \cdot \omega^4 - A_6 \cdot \omega^6 + \ldots \\ \mathrm{Im}_{WG} &= A_1 \cdot \omega - A_3 \cdot \omega^3 + A_5 \cdot \omega^5 + \ldots \\ \mathrm{Re}_{M(ik)} &= B_0 - B_2 \cdot \omega^2 + B_4 \cdot \omega^4 - B_6 \cdot \omega^6 + \ldots \\ \mathrm{Im}_{M(ik)} &= B_1 \cdot \omega - B_3 \cdot \omega^3 + B_5 \cdot \omega^5 + \ldots \end{aligned} \tag{4}$$

where:

Re_{WG}, $\mathrm{Re}_{M(ik)}$ — real parts of, respectively: a principal determinant of the system of equations of motion and its minor obtained by deleting i-th column and k-th row of the principal determinant,

Im_{WG}, $\mathrm{Im}_{M(ik)}$ — imaginary parts of, respectively: a principal determinant of the system of equations of motion and its minor,

A, B — coefficients of polynomials being the expansion of the principal determinant of the system of equations of motion and its minor,

ω — vibration frequency of the studied system.

The zeroing of the real part in the expression (3), when an extremum of the imaginary element of the transfer function occurs, corresponds to the eigenfrequencies ω_{0i} (for $i = 1,2,\ldots,18$) of the studied system (Findeisen, 2000).

Eighteen frequencies of free vibrations from the studied system were determined by solving the eigenvalue problem for the considered model in this way (Tab. 1). The distinguished forms of free vibrations manifest themselves in the curves of the transfer functions $\Phi = f(\omega)$ determined for particular components of the excitation of vibrations (P_X, P_Y, P_Z, M_X, M_Y and M_Z) – Fig. 2. Table 1 shows that the eigenfrequencies of the studied system span a wide range, reaching up to 30000 rad/s. The first five eigenfrequencies are within a range of zero to 800 rad/s. A spectra analyses of boom vibration excitation (described further in this article) revealed that a frequency range not exceeding 800 rad/s is of practical relevance.

The frequencies of the boom's free vibrations depend on its deflection on a plane perpendicular to the floor (described with the boom deflection angle α_V) and on the telescope extension (described by the value of advancement of the extendable part of the boom relative to its fixed part ΔL_1). As a result, the values of free vibrations are within a certain range – between the minimum and maximum value (Fig. 3). It is visible, most of all, in the case of the first seven forms of free vibrations. Within the variability range of parameters studied ($\alpha_V = -18 \div +55°$; $\Delta L_1 = 0.1 \div 0.6$ m), the maximum value of the first frequency of free vibrations is more than 4 times higher when compared with the minimum value (green line in Fig. 3). A maximum value more than twice as high as the minimum value was also noted for the second frequency of free vibrations in the system studied. The variability ranges of free vibrations frequencies correspond-

TABLE 1

Forms of vibrations in transfer functions determined for external boom load components – the presented frequencies of free vibrations concern a horizontal boom ($\alpha_V = 0$) and a fully extended telescope ($\Delta L_1 = 0.6$ m)

No. of the form of vibrations	Frequency of free vibrations ω_{0i} [rad/s]	Transfer function					
		Φ_{PX}	Φ_{PY}	Φ_{PZ}	Φ_{MX}	Φ_{MY}	Φ_{MZ}
1	31,8		+	+	+		
2	75,2		+	+	+		
3	190,8	+				+	+
4	503,0		+	+	+		
5	738,2	+				+	+
6	1335,8	+				+	+
7	2104,8		+	+	+		
8	5581,9	+				+	+
9	6287,4	+				+	+
10	8650,4		+	+	+		
11	8741,9		+	+	+		
12	12151,4	+				+	+
13	13624,8		+	+	+		
14	15485,3	+				+	+
15	20130,9	+				+	+
16	20624,0		+	+	+		
17	26705,8	+				+	+
18	29317,0		+	+	+		

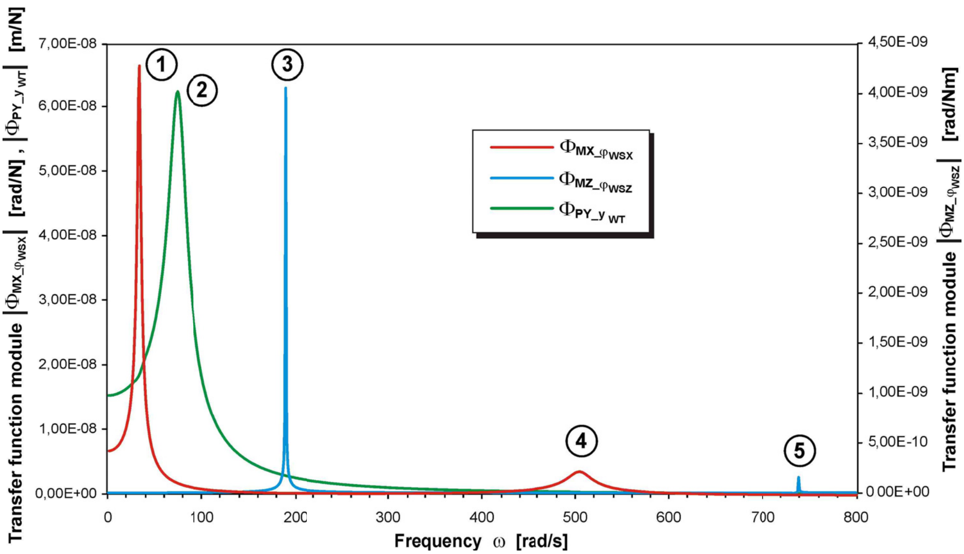


Fig. 2. Elements of amplitude – frequency characteristic for the three selected transfer functions – horizontal boom ($\alpha_V = 0$) and fully extended telescope ($\Delta L_1 = 0.6$ m): Point 1 – $\omega_{01} = 31.8$ rad/s; Point 2 – $\omega_{02} = 75.2$ rad/s; Point 3 – $\omega_{03} = 190.8$ rad/s; Point 4 – $\omega_{04} = 503$ rad/s; Point 5 – $\omega_{05} = 738.2$ rad/s

ing to the 4th and 7th form of free vibrations were the widest. Ranges for such frequencies of free vibrations were, respectively: 273 and 244 rad/s (for the sake of comparison, the range of the variability interval for the first eigenfrequency was 45.2 rad/s).

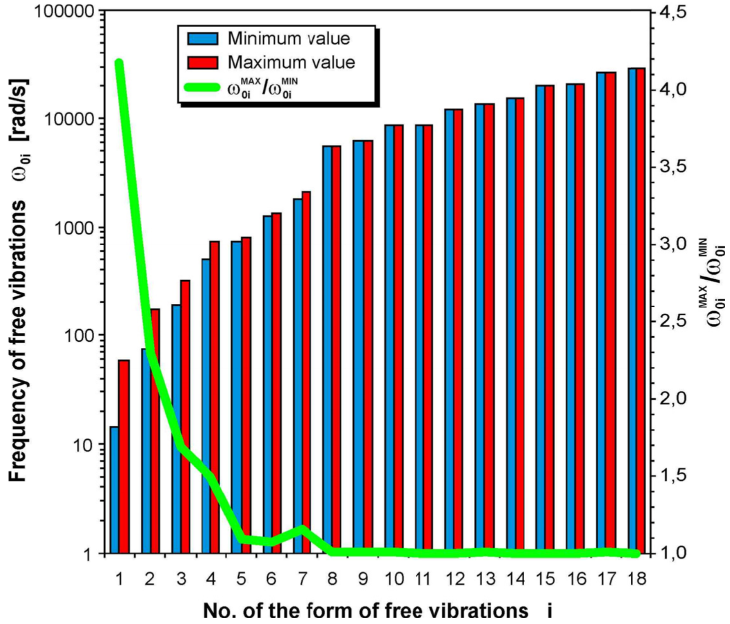


Fig. 3. Variability ranges of eigenfrequencies of roadheader boom free vibrations in the variability range of the boom deflection angle on a plane perpendicular to the floor $\alpha_V = -18 \div +55°$ and a telescopic extension of $\Delta L_1 = 0.1 \div 0.6$ m

The effect of boom deflection and telescope extension on the boom eigenfrequencies, mainly results from the variable stiffness of the boom the deflection mechanism's hydraulic cylinders during angle deflection, and the variable stiffness of the telescopic mechanism's hydraulic cylinders during it's extension. Such stiffnesses are inversely proportional to the level of fluid in the cylinders of the hydraulic actuators, which depends on the degree of extension between the piston rods and the cylinders (Podsiadła, 2000). This is a hyperbolic dependency (Fig. 4). As the boom is deflected upwards (towards the positive values of the angle α_V), the specific stiffness of the hydraulic cylinders for lifting the boom k_{SP} is decreasing. The value of this stiffness corresponding with $\alpha_V = 0$ (k_{SP0}) is decreasing within the range 3.6 to 0.3, i.e. 12 times (black line).

A clear effect of the boom deflection angle on a plane perpendicular to the floor is only visible for the first form of vibrations (Fig. 4). This effect is substantiated as the first form of free vibrations represents the vibrating motion of the boom on a plane perpendicular to the floor. The first eigenfrequency changes as the boom deflection angle changes according to the hyperbolic function. In the case of a contracted telescope (red line), in the considered variability range of the boom deflection angle, the frequency of free vibrations ω_{01} is changing within the limit of 60 rad/s – for $\alpha_V = -18°$ to 16.5 rad/s – for $\alpha_V = +55°$, consequently, 3.5 times. The maximum

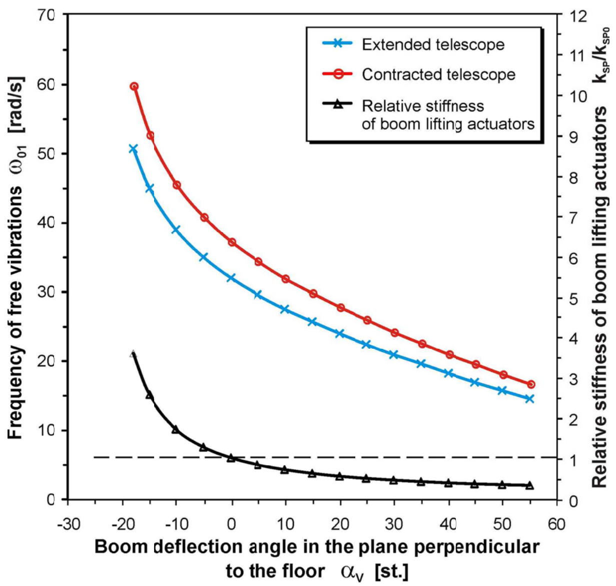


Fig. 4. Effect of boom deflection on a plane perpendicular to the floor on the value of the first eigenfrequency of telescopic boom free vibrations

extension of the telescope (blue line) causes the value of the first eigenfrequency of the boom to drop. Within the examined variability range of the angle a_V, the value of this frequency changes in a range of 50.5 to 14.3 rad/s.

3. Identification of vibration excitation elements of the cutting system generated through the process of cutting a heading face

The components of boom vibration excitation were identified by analysing the time curve spectrum of a load on the roadheader boom. The curves were obtained by way of a computer simulation of the cutting process of a heading face with cutting heads equipped with N_Z = 56 conical picks arranged on the base of the heads. The actual number of picks utilized in cutting the heading face, depends on the size of the web z which is cut, and on the height of the layer cut h (layer cutting when the heads cutting parallel to the floor surface are advancing is considered here).

Fig. 5 shows a spectrum of one of six external boom load components – the moment of load forces M_X (cf. Fig. 1a) – obtained for one rotation of the cutting heads with a set stereometry. The computer simulation pertained to cutting a rock layer with a web of cut of z = 0.2 m and a cut height of h = 0.45 m with the cutting head advancing at a speed of v_{ow} = 0.2 m/s.

Six vibration components are clearly visible for a frequency up to 1500 rad/s. The frequency ω_D = 17.6 rad/s (point 1) is a helix frequency, resulting from the fact that the picks are positioned along four helixes with a large rotational angle (the frequency is 4 times higher in relation to the angular speed of the cutting heads ω_G). Vibrations with a frequency of ω_K = 52.8 rad/s (point 2)

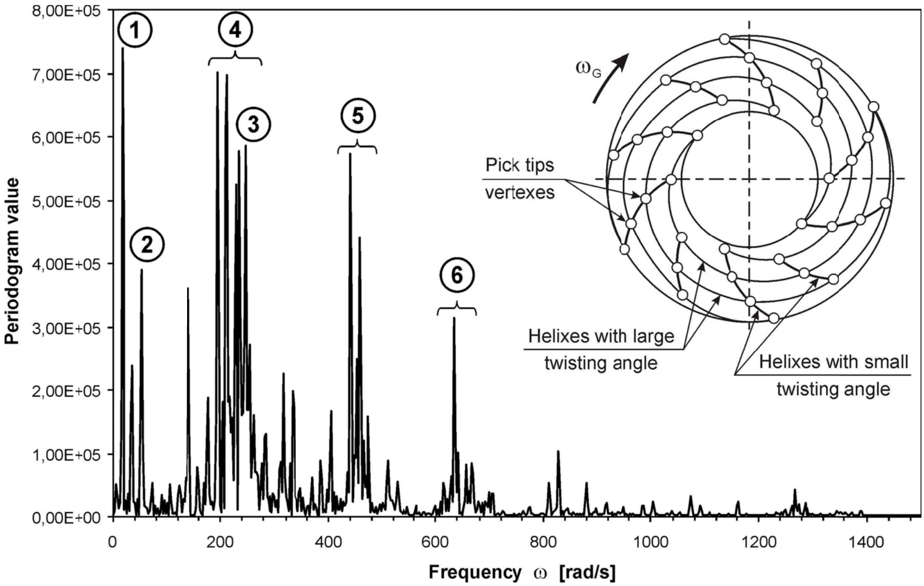


Fig. 5. Spectrum of the external boom load generated by the rock cutting process

are 12 times larger than the angular speed of the cutting heads and result from the fact that the picks are positioned along 12 helixes with a small rotational angle. The frequency $\omega_N = 246.4$ rad/s (point 3) is the pick frequency. These vibrations are due to the fact that the other picks being utilized are entering the cutting zone.

The roadheader vibration excitation components mentioned above, not including vibrations with the frequency ω_K, were already identified under earlier studies analysing the dynamic state of boom-type roadheader cutting head drives (Dolipski & Cheluszka, 1998). No helix frequency ω_K of vibrations were distinguishable, however, as the cutting head picks studied were not arranged as orderly, then. Vibrations with a frequency corresponding to the angular speed of the cutting heads (ω_G) are not visible on the periodogram presented (Fig. 5). This shows that fragments of the external load curve of the boom which corresponded to a singular rotation of the cutting heads were subjected to spectral analysis. A vibration component with the radial frequency of the cuttings heads is shown in the analysis of extended fragments of the cutting system load. This was a result of the assumption that the progression of the cutting process was repeatable in subsequent rotations of the cutting heads. Just as in the case of the cutting head drive, the circular frequency of the cutting heads is a basic frequency of the roadheader boom vibration excitation.

Apart from the vibration components which have been distinguished above, vibrations with frequencies spanning certain range are clearly visible in the boom load spectrum (Fig. 5 – intervals: 4, 5 and 6). The presence of such vibrations in the spectrum of the load analysed is related to the progression of the rock cutting process with particular cutting head picks. This derives from the fact that the load on the picks is characterised by significant variations. Variations in the value of the forces acting on the picks as a function of the cutting path (or as a function of time) can be presented as a triangular curve with a gap (Dolipski & Cheluszka, 1998). The load vibrations of picks have, therefore, a shape similar to a sawtooth curve (Fig. 6). The character of the load curve of the conical picks corresponds with findings from experimental and numerical studies pursued by various research institutions – e.g. (Gehring, 1973; Keleş, 2005; Rojek et al., 2009; Vašek & Pinka, 2006; Van Wyk, 2012; Yu & Khair, 2007). When cutting heads rotate, periodic vibrations with; differing amplitudes, frequencies and initial phases; arising from the subsequent picks contacting the rock, come together forming the total sum. The amplitude and frequency of these vibrations (referred to as the detachment frequency ω_S) depend on the cutting depth g of a given pick. The function $\omega_S = f(g)$ is nearly hyperbolic (Fig. 7a – continuous line). On the other hand, the initial phase and duration of vibrations t_U (the number of detachment cycles during each cutting head rotation) depend on the position of the chosen pick on the cutting head, on the web and on the height of the rock layer excavated, which determine the value of the angle the pick enters ϑ_e and exits ϑ_a the cutting zone (Fig. 6).

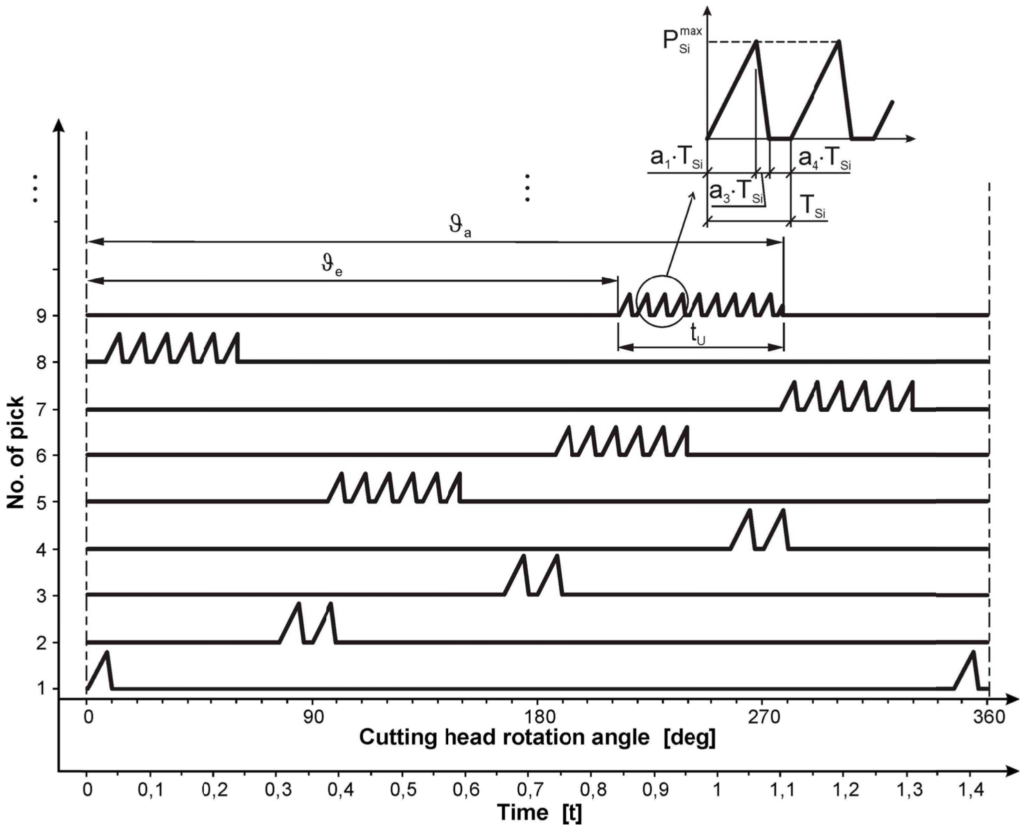


Fig. 6. Elements of sequence of rock cutting with cutting head picks with the applied load curves of the picks

The average power values of vibration components on the load of the selected picks of the cutting head studied were determined, in order to identify the effect pick load vibrations from the detachment curve of rock grains have on the external boom load. The vibrations are treated as a periodical signal with the period $T_G = 2\pi/\omega_G$. For example, for the considered com-

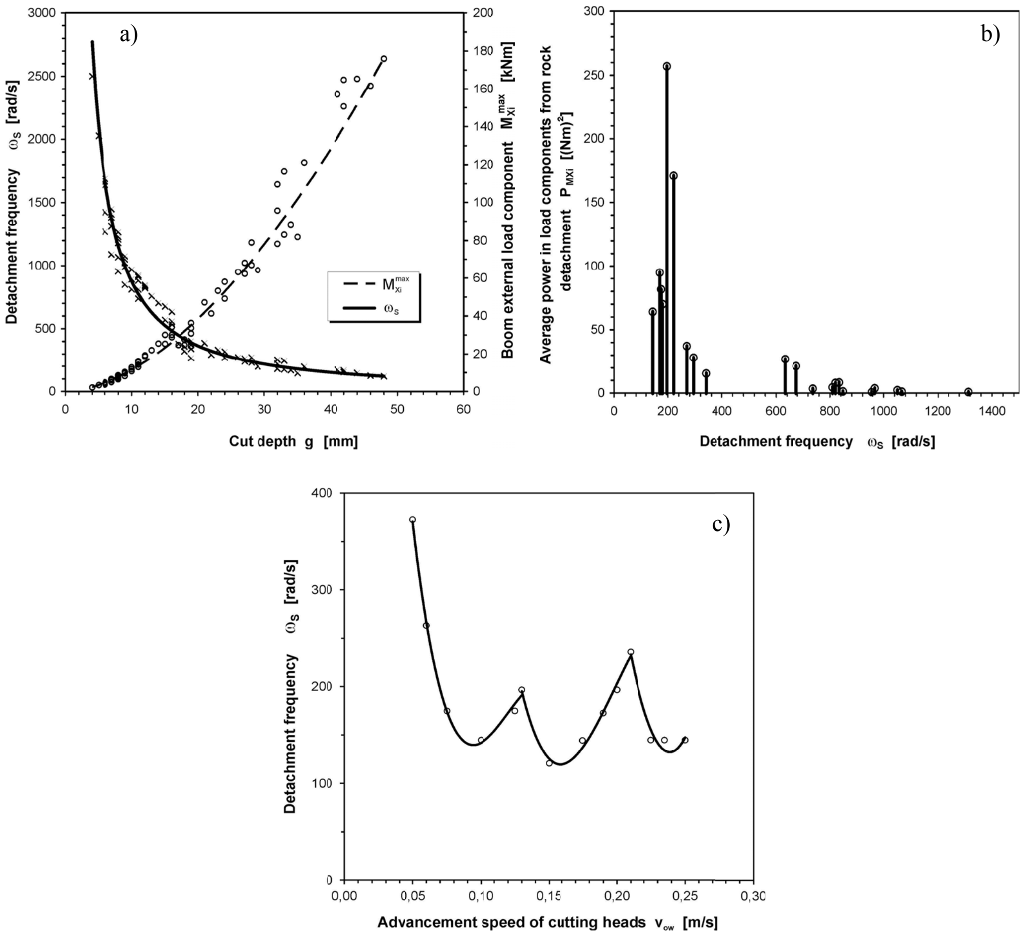


Fig. 7. Detachment frequency: a) the influence of the depth of cuts, b) distribution of power density of components of vibrations generated with the detached rock fragment phenomenon for v_{ow} = 0.2 m/s, c) the influence of the cutting head's speed of advancement

ponent of boom vibration excitation – the moment of load force M_X – the average power P_{MXi} of the vibration component resulting from the acting i-th pick, is described as follows (marked as in Fig. 6):

$$P_{MXi} = \frac{1}{T_G}\int_0^{T_G} M_{Xi}^2(t)\,dt = \frac{T_{Si}}{3\cdot T_G}\cdot(a_1 + a_3)\cdot\left(M_{Xi}^{\max}\right)^2\cdot \mathsf{E}\left[\frac{\vartheta_{ai}-\vartheta_{ei}}{T_{Si}\cdot\omega_G}\right] \quad \text{for } i = 1,2,3,\ldots N_U \qquad (5)$$

where:

$M_{Xi}^{\max}$ — the maximum external load value M_X caused by the activity of the i-th pick in particular detachment cycles,

T_G — rotational period of cutting heads,

T_{Si} — detachment period:

$$T_{Si} = \frac{K_g}{\omega_G} \cdot \frac{g_i}{r_i} \tag{6}$$

K_g — detachment factor (cf. (Dolipski & Cheluszka, 1998)),

a_1, a_3 — coefficients of the ratio of the cutting force variability interval in the detachment cycle (cf. (Dolipski & Cheluszka, 1998)),

g_i — the depth of the cut performed by the *i*-th pick,

r_i — the distance of the pick's *i*-th vertex from the axis of rotation of the cutting head,

N_U — the number of picks active in the process of cutting a layer with the defined cut height *h* with the defined web of cut *z*,

ϑ_{ei}, ϑ_{ai} — respectively: the angle under which the *i*-th pick enters the cutting zone and the angle under which the *i*-th pick exits the cutting zone,

w_G — angular speed of the cutting heads,

E[ ] — the total part of the argument in brackets.

In the case of the advancing cutting speed analysed, the detachment frequencies corresponding to the depth of the cuts made by the cutting head picks studied, span a wide range, i.e. 145 to 1314 rad/s. The highest average power values from load vibrations caused by fragments detaching from the rock, however, correspond to the detachment frequencies of: 196 rad/s and 221 rad/s (Fig. 7b). When cutting rock with a compressive strength of R_c = 120 MPa, the average power of the vibrations for the frequency values ω_S then reaches: 257 and 171 (Nm)^2, respectively. For a detachment frequency higher than 250 rad/s, the average power of the vibrations for the external boom load components being considered is decisively lower, which signifies that such vibrations have little effect on the curve of the boom load generated by the rock cutting process. Vibrations from an external boom load with a frequency of 193 to 246 rad/s (interval 4 in Fig. 5), therefore, correspond to elements of the progression of detaching rock fragments with average power. The frequency intervals: 5 and 6 in Fig. 5 correspond to the increased harmonics of the elements of detachment. The load on the picks from cutting a depth *g* of 36 and 32 mm had a decisive effect on the excitation of roadheader boom vibrations in the study. Such depths correspond to the maximum values of the average power of vibrations from the pick load. The formula (6), however, shows that not only is the depth of the picks' cuts important, but also the relationship between the depth and the distance of the considered tip of the pick vertex, from the rotational axis of the cutting head (*r*).

Detachment frequency depends on the advancing speed of the cutting heads (Fig. 7c). In the study, the frequency ω_S corresponds to the highest average power value of load vibrations caused by rock fragments detaching, spans a relatively wide range, i.e. 120.6 – for v_{ow} = 0.15 m/s to 372,5 rad/s – for v_{ow} = 0.05 m/s. As a result, the position of the vibration elements generated by detached rock fragments on the periodogram, changes as the advancing speed of the cutting heads changes. The intensity of their interaction also varies.

Analysis of external boom load periodograms indicates that vibration elements with a frequency up to 800 rad/s are important in terms of boom dynamics. Components with higher frequencies have a much lower amplitude.

Figure 8 shows a spectrum of boom vibration excitation obtained during a computer simulation of rock excavating with transverse cutting heads equipped with N_Z = 56. The simulations included different associations of operational parameters for a roadheader. The web z was changed within the range 0.1 to 0.2 m (of the maximum value for the cutting heads studied when cutting hard rock). Two heights of the layer cut h were considered for each value of the web – a maximum one (for the given web) and another equal to half of the maximum value. Computer analysis was performed for the advancing speed of cutting heads v_{ow} at a broad range, i.e. 0.05 m/s to 0.2 m/s. The values of the relative amplitude of vibration components were analysed to identify the intervals of vibration excitation frequencies valid to the context of intensity of interaction. A relative amplitude is understood as the relationship of a vibration amplitude (A) with a given frequency to a maximum amplitude ($A_{\max}$) in a set of values obtained for all the investigated associations of the parameter values in the cutting process.

An envelope of the analysed spectrum (red line in Fig. 8) has several clear gains, of which the most important ones correspond to vibration frequencies smaller than 18 rad/s and within a range of 140 to about 220 rad/s. Dominant vibration components of the external boom load occur in these intervals. Vibration components corresponding to the radial frequency (ω_G) and helix frequency (ω_D) of the cutting heads are in the first frequency range (ω < 18 rad/s). The other frequency intervals mentioned include vibrations with the pick frequency (ω_N) and a detaching rock fragment frequency of (ω_S). Vibrations in this range are the most intense (posses the highest amplitudes).

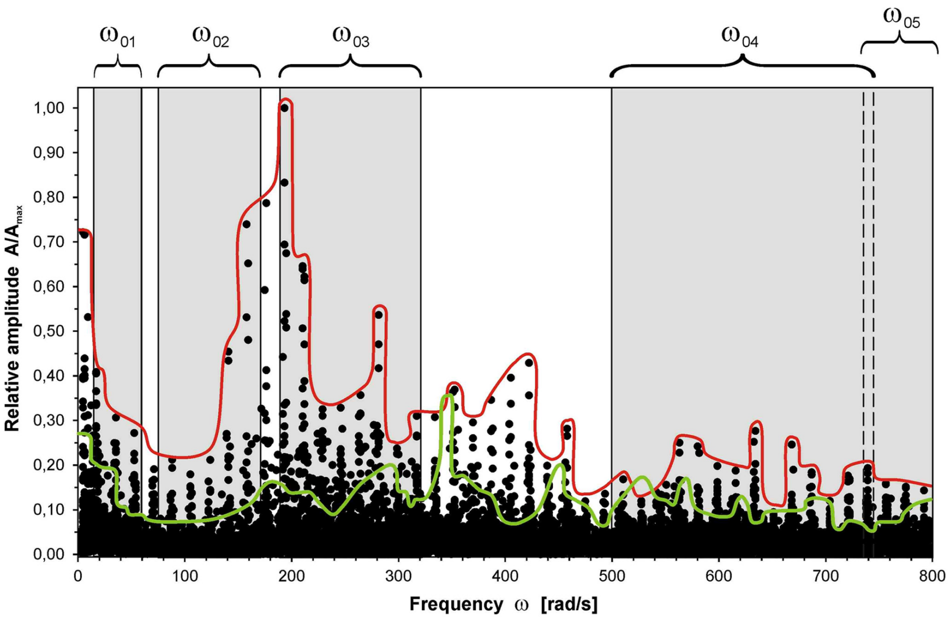


Fig. 8. Spectrum of vibrations excitation obtained from a simulation of the process of cutting with $N_Z = 56$ conical picks placed on the heads of the principal component (envelope in red), for differentiating associations of values of roadheader operating parameters against the first five eigenfrequencies of telescopic boom

The dominant external boom load spectrum generated by the cutting process is located in the band which corresponds to the third eigenfrequency of the boom (ω_{03}). This, therefore, indicates that resonance vibrations are likely to be present during the cutting process for the following grouping of roadheader operational parameter values: $z = 0.2$ m; $h = 0.85$ m and $v_{ow} = 0.15$ m/s.

Another element with an elevated amplitude is seen in the lower frequency range. This corresponds to the circular frequency of the cutting heads. This component is, however, outside the area of the studied system's frequencies of free vibrations.

The external load components possess a low amplitude in a range corresponding to the 4^{th} and 5^{th} eigenfrequency of the boom. This does not exceed one-third of the maximum value of this range.

4. Impact of the pick system on the excitation of vibrations in the cutting system

Investigations into the impact the number of picks and their arrangement on the cutting head made on the boom vibration excitation frequency, were carried out on the four cutting heads with the following number of picks arranged along the principal component (N_Z), i.e. 44; 51; 56 and 69. The heads had the same shape and the same dimension of pick envelope. The systems of picks considered differed in the number of helixes with a large twisting angle (N_D) and a small twisting angle (N_K), along which the picks were arranged (Tab. 2). Computer simulations of the cutting process were performed for cutting rock layers with a web z within the range 0.1 to 0.2 m and with different heights h. The cutting movement was simulated (parallel to the floor) for various advancing cutting heads speeds v_{ow} of 0.05 to 0.25 m/s.

TABLE 2

The list of values of parameters of pick systems used for simulation studies and their corresponding characteristic values of boom vibration excitation

N_Z	N_D	N_K	ω_G	ω_D	ω_K	ω_N
			[rad/s]			
44	2	10	4,4	8,8	44,0	149,6 ÷ 193,6
51	3	12		13,2	52,8	171,6 ÷ 224,4
56	4	12		17,6	52,8	176,0 ÷ 246,4
69	3	15		13,2	66,0	224,4 ÷ 303,6

As the number of helixes along which the picks are arranged varies, this has an effect on helix frequencies of boom vibration excitation (ω_D and ω_K) – Tab. 2. In the instance where picks are arranged along two or three helixes with a large twisting angle, the helix frequency ω_D is smaller (for the defined circular frequency value of heads $\omega_G = 4.4$ rad/s) than the lower limit of the first eigenfrequency of the system studied ($\omega_{01}^{\min} = 14.3$ rad/s). The vibration excitation component with the frequency ω_D is, therefore, outside the resonance area in such cases. In instances where the picks are arranged along four helixes with a large twisting angle, the helix frequency ω_D is situated in the range determined for the first eigenfrequency of vibrations, near its lower limit. The helix frequency ω_K reaches a value outside the ranges corresponding to the considered eigenfrequencies of the boom studied, for only 15 helixes with small twisting angles. In other cases, the values of this element of vibration excitation are situated in the range of the first eigenfrequency ω_{01}.

The pick frequency ω_N depends not only on the number of picks arranged on the main part of the cutting head, but also on the parameter values of the layer being excavated. Hence, this frequency is within a range resulting from the variability range of the considered web z. In the first instance of pick systems considered ($N_Z = 44$), the pick frequency will be outside the resonance range for the web lying within a narrow range of only, i.e. 0.14 to ~0.2 m. For $N_Z = 51$ and 56 – this will be possible for a web value ranging: 0.1 to 0.14 m and ~0.1 to 0.12 m, respectively. On the other hand, for the highest considered number of picks located on the principal component of the cutting head ($N_Z = 69$), in the case of sets smaller than 0.2 m, the pick frequency is within the resonance zone of the boom studied.

The boom vibration excitation curves generated by the cutting process possess highly diversified parameter values. This results from the broad variability range of operational parameter values observed in trials, and from the interconnectedness inherent in the process. If variations in the eigenfrequency of vibrations of a boom for its various working positions are considered, it turns out that it is difficult, or even impossible, to differentiate excitation frequencies from eigenfrequencies.

The intensity of excited vibrations not only depends on the relationship between excitation frequency and eigenfrequency, but also on the value of the excitation forces' amplitude. The listed vibration excitation parameters should be jointly considered in the context of the resonance phenomenon. For this reason, the basis for comparing the systems was both the average and maximum amplitude values of the external boom load vibration components, generated during the cutting process with frequencies in a range relative to the first three free frequencies of the system studied (Fig. 9). The following values were used as a reference point: the average and maximum amplitude of vibration components obtained for a system of picks in which 44 picks are positioned on the cutting head's principal component.

Amplitudes of vibration components do not exceed 10% of the maximum value ($A/A_{max} \leq 0.1$) in over 85% of the analysed set of amplitude values (A) and frequencies of vibration components (ω) forming an amplitude – frequency spectrum of the external boom load, obtained for a variety of configurations of roadheader operational parameter values. In order to identify, in resonance areas, the components with particularly high amplitudes likely to cause significant excitation of boom vibrations, the average values of the amplitude were determined for the values of the function: $A = f(\omega)$ higher than the threshold value: $0.1 \times A_{max}$ (where A_{max} is the highest value from the set of amplitudes of vibration excitation elements obtained for all the pick systems analysed).

Figure 9 shows that the number of picks and their arrangement on the side surface of the cutting head largely influences the vibration intensity of boom excitation components. Variations in both the average and maximum amplitude values of the vibration excitation elements are visible in all three frequency ranges corresponding to the first three eigenfrequencies of a given object's free vibrations. For the four pick systems considered, the lowest average values of amplitudes in all the frequency ranges analysed were recorded for a cutting head equipped with $N_Z = 56$ picks (Fig. 9a). This is especially clear in reference to the 2nd and 3rd frequency of free vibrations. The average amplitude value of external boom load vibration elements, in relation to the highest value, is smaller by: 36% and 26%, respectively. The highest average amplitudes of external boom load vibration elements were recorded for $N_Z = 44$ (in the 1st and 2nd range of boom free vibration frequency) and for $N_Z = 51$ (the 3rd frequency of free vibrations).

A similar effect is also visible when analysing the maximum amplitudes of the external boom load vibration elements (Fig. 9b). Here, as well, for the 1st and 2nd frequency of boom

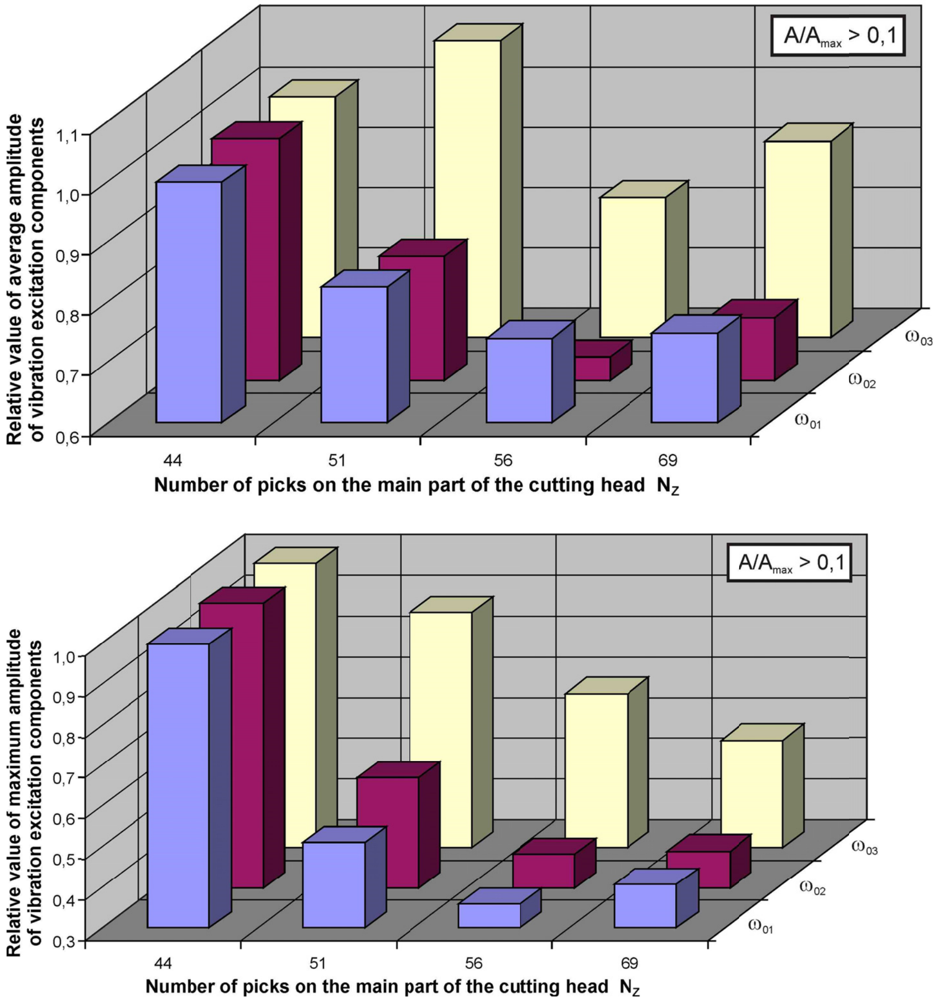


Fig. 9. Impact of the system of picks on the amplitude of boom vibration excitation elements in the range of its first three eigenfrequencies

free vibrations, the maximum amplitude of vibration elements were smallest for the pick system in which the number of picks on the principal component of the cutting head N_Z is 56. Compared to the system with the smallest number of picks ($N_Z = 44$), the maximum amplitude of vibration elements were smaller by about two-thirds. In the frequency range corresponding to the 3rd eigenfrequency of the boom, the smallest value of the amplitude of maximum vibration elements of its external load, was noted for the system with the largest number of picks ($N_Z = 69$). It accounted for nearly half of the maximum amplitude value achieved in the frequency range for the system with the smallest number of picks. In the case of the pick system in which 56 picks were arranged on the principal component of the cutting head, the maximum amplitude of the boom vibration excitation elements was 10% higher than the value noted for $N_Z = 69$.

5. Conclusions resulting from numerical analysis

Two of the four pick systems investigated proved to be most advantageous, in relation to minimising the amplitude of vibration excitation elements, i.e. those with 56 and 69 picks on the principal component of the cutting head. In the case of the first of the previously mentioned pick systems, where the picks were arranged along four helixes with a large twisting angle concurrently with 12 helixes having a small twisting angle, the helix frequencies of ω_D and ω_K are within a range corresponding to the first frequency of free boom vibrations. This effect does not exist in the case where $N_Z = 69$ (picks are arranged along 3 helixes with a large twisting angle and 15 helixes with a small twisting angle). Additionally, in the amplitude – frequency spectrum obtained for the pick system in which $N_Z = 56$ picks are provided along the principal cutting head component, the external load elements on the boom with the highest amplitudes were exhibited at a periodicity lying in the third frequency of free vibrations of the boom studied (Fig. 8).

Considering the aforementioned, as well as the technical criteria (space available for mounting pickboxes along the side of cutting heads with a complex shape and dimension) that fulfills the anti-resonance criterion and is technically possible, the solution is the pick system in which about 56 picks align along 3 helixes with a large twisting angle concurrently with 15 helixes with a small twisting angle, arranged along the principal cutting head component. As a result of the modification made, the system of picks creates 57 picks arranged on the principal head component. Computer simulations of the cutting process have revealed that the amplitudes of external boom vibration excitation elements for this system of picks clearly have smaller values when compared to the cutting head, where $N_Z = 56$. The envelope of the amplitude and frequency spectrum attained for the modified pick system (green line in Fig. 8) does not exceed 35% of the maximum value of amplitudes of vibration elements for $N_Z = 56$. The components with the highest amplitudes here, correspond to frequencies from outside the resonance zones.

The results of a computer simulation using a dynamic model of a roadheader boom confirm that the pick system selection criterion considered in this work has been satisfied. Table 3 confirms the obtained values of telescopic boom vibration amplitude values and the dynamic

TABLE 3

Relationship of amplitudes of dynamic load of boom lifting cylinders (P_{SP}), telescopic mechanism cylinders (P_{ST}) and boom vibrations in the point S_G generated by a cutting process with cutting heads with the following number of picks on the principal component: $N_Z = 57$ ($A^{[57]}$) and $N_Z = 56$ ($A^{[56]}$)

v_{ow} [m/s]	$\Delta L_1 = 0$				$\Delta L_1 = 0,6$ m			
	0,05	**0,10**	**0,15**	**0,20**	**0,05**	**0,10**	**0,15**	**0,20**
	$A^{[57]}/A^{[56]}$							
P_{SP}	0,94	0,55	0,47	0,50	0,76	0,34	0,33	0,40
P_{ST}	0,44	0,43	0,23	0,62	0,52	0,45	0,20	0,56
y_{SG}	0,51	0,56	0,26	0,54	0,50	0,43	0,17	0,61
z_{SG}	0,92	0,82	0,46	0,68	0,75	0,52	0,33	0,57
$\dot{y}_{SG}$	0,44	0,43	0,31	0,61	0,38	0,47	0,22	0,61
$\dot{z}_{SG}$	0,81	0,77	0,43	0,75	0,62	0,47	0,31	0,54
$\ddot{y}_{SG}$	0,39	0,54	0,41	0,64	0,46	0,43	0,26	0,57
$\ddot{z}_{SG}$	0,80	0,62	0,33	0,68	0,96	0,44	0,37	0,72

load of the corresponding hydraulic cylinders while cutting rock with cutting heads equipped with 56 and 57 picks, arranged as described above. The dynamic load amplitude of the boom deflection cylinders on a plane perpendicular to the floor (P_{SP}) for $N_Z = 57$ is smaller by 5%, up to even 67%, versus the amplitude of the load obtained for $N_Z = 56$. Even a greater reduction of vibrations was achieved for the dynamic load on the telescopic mechanism cylinders (P_{ST}). In this case, the amplitude of this load was decreased by a range of 38%÷80%.

The relationship of the boom's defined amplitude of vibrations in the point S_G (Fig. 1a) in the direction of the Y and Z axis for an XYZ system for the two pick systems compared ($N_Z = 56$ and 57) was within 0.17 to 0.82. In the case of longitudinal vibrations for the boom (coordinate y_{SG}) for the pick system in which 57 picks were arranged along the principal cutting head component, a reduction of vibration amplitudes from 39% to even 83%, was achieved. The amplitude of lateral vibrations on a plane perpendicular to the floor (coordinate z_{SG}) was decreased in a range of 18%÷67%. Vibrations in the direction of the axis X were not analysed here. The dynamic model used for the studies did not consider the boom deflection mechanism on a plane parallel to the floor which, as expected, would significantly influence boom vibrations in this direction.

A reduced speed and acceleration of vibrations was also noted for the pick system in which 57 picks were arranged on the principal component. The boom vibration speeds of $\dot{y}_{SG}$ and $\dot{z}_{SG}$ were reduced from 19% to nearly 80%. The acceleration values of boom vibrations: $\ddot{y}_{SG}$ and $\ddot{z}_{SG}$ were smaller by 4÷74%, compared to the values of such acceleration obtained from a cutting head equipped with $N_Z = 56$ picks.

6. Conclusion

The computer simulations performed have pointed out that, by selecting a pick system with a cutting head having an appropriate number of picks with the appropriate arrangement, the dynamics of a boom, being an important element of a roadheader cutting system, are manageable. Vibrations can also be reduced, and dynamic loads minimised, by adapting the system of picks, which is responsible, to a high extent, for the form of vibration excitation during the cutting process, to the dynamic properties of the system.

The issue of eliminating roadheader boom resonance vibrations in the fashion considered in this work is very difficult to implement. This is due to the fact that, it is practically infeasible to differentiate the vibration excitation frequencies in this system from its free vibrations. On one hand, this is due to the wide spectrum of elements of external boom load vibrations from rock cutting, implemented for the various relations between the parameters that characterise it. On the other hand – the reasons for this are the variations in the dynamic properties of the boom in its functioning positions (deflection, telescopic extension). As a result, the frequencies of free vibrations of the boom create certain ranges which may partially coincide.

One way for solving the dilemma presented in this work is to minimise the external boom load vibration amplitude, especially in relation to the vibration elements in the resonance zones. As the amplification of vibrations in resonance is a result of the value of the amplitude multiplication factor (depending on the damping value in the system), resonance vibrations can therefore be reduced by reducing the excitation amplitude of vibrations. The objective was achieved as a result of the comprehensive numerical analysis performed. This stems from the fact that, in the context of achieving the anti-resonance criterion, an advantageous system of picks on roadheader cutting heads was determined.

Hence, the anti-resonance criterion should be an important element in the process of selecting and optimising pick systems on active units of mine excavation machinery. If this criterion is taken into account during the cutting heads/drums design stage, it will contribute to improved cutting process efficiency, enhanced durability and reliability in the excavation machinery used. Improved ergonomics and working conditions for roadheader operators will be experienced as well, even under the difficult environmental conditions involved with mine headings. Reduced boom vibrations will lead to limited variations in the parameter values experienced during the cutting process, and will also contribute to a reduced level of vibrations transferred onto roadheader subassemblies. A precondition for the practical use of the anti-resonance criterion in the design process of active mine excavation machinery, is to have dedicated software with implemented mathematical models representing the dynamics of particular mining excavation roadheader subassembly systems implemented.

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