

## Simulation of the frequency response of the ERC 1400 Bucket Wheel Excavator boom, during the excavation process

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Date of submission to the Editor: 05/2019  
Date of acceptance by the Editor: 07/2019

### GENERAL REQUIREMENTS

The Bucket Wheel Excavator (BWE) is a machine with continuous operation which cuts the rock with the help of the buckets installed on the rotor, performing at the same time the transfer of the excavated material to the main conveyor (Rasper, 1975), with the help of the conveyor belts installed on its boom (Fig. 1).

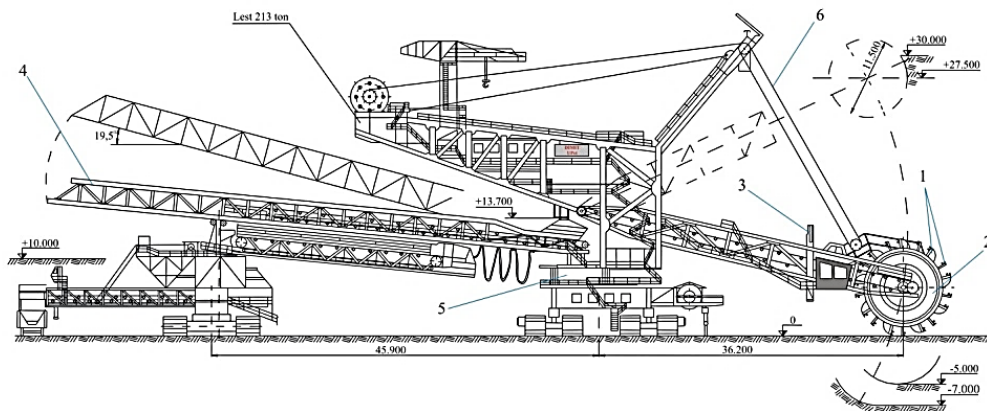


Fig. 1 Bucket Wheel Excavator type ERC 1400-30/7

The operating equipment is the rotor (bucket wheel) which performs a rotation movement in vertical plane and, with the help of the boom, a horizontal pivoting movement and a vertical rising-lowering movement (Nan, 2007, ROMINEX, 2007).

The analysis of the dynamic behavior of BWE is very important, both in design and operation process, mainly in order to prevent the possible occurrence of the resonance in the system's elements and to create a basis for the analysis of the stress conditions in structural subsystems and for the assessment of lifetime. The most vulnerable subsystem of the BWE is represented by the boom, which generally is realized as a spatial truss having different stiffness and damping characteristics compared with the excavator's other subsystems.

Not only the shape and constructive parameters of the boom are responsible for its vulnerability, but also the nature, source and character of the external exciting loads of which it is exposed. The influence of this last one has been for a long time neglected, the current engineering codes and national standards used in design calculations (German Institute for Standardization, 2015) ignoring the dynamic character of external loads, their value being replaced by equivalent static loads amplified with a so called dynamic coefficient.

This approach seems to be far from reality, because it cannot predict the occurrence of failures caused by vibration, which has to effects not covered by it, namely the fatigue due by the load's cyclical variability and the possibility of huge deformation produced by resonant vibration of some constitutive elements. The vibration is attributable to the extremely difficult operating conditions generating loads with a very pronounced dynamic and stochastic character as result of repeated soil-bucket contact, the unbalance of the driving mechanism's elements (the bucket wheel and the rotational components of the on board belt conveyor) and the strokes of the excavated rock fragments in the course of emptying the buckets (Bošnjak et al., 2015).

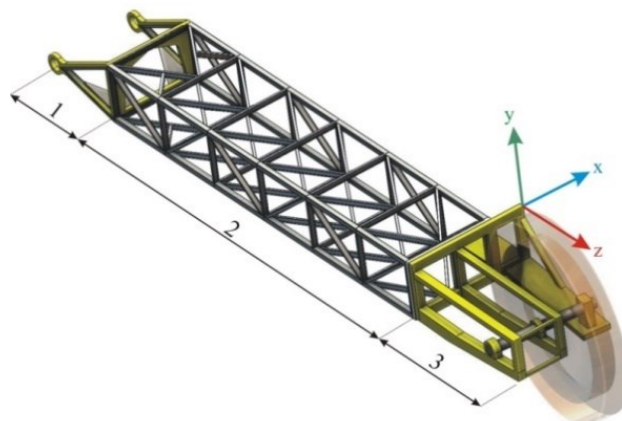
The importance of the investigations related to BWE dynamics is proved by the increasing number of publications dealing with this subject This increasing interest is justified by the recent trend towards an increasing automation of this class of earthmoving machines, focused mainly on the control of vibration which affect their working performance.

The studies (Boskovic et al., 2015) include problems of determining excitation induced by resistance-to-excitation problems of determination and measurement of natural frequencies of BWEs' structures as well as their vibrations during mining process (Rusinski et al., 2012).

Furthermore, it is important to define the response of BWE superstructure to excitation caused by the resistance-to-excitation.

The paper (Chudnovskii, 2007) details the analysis of vibration in horizontal plane at the level of tooth mounting angles.

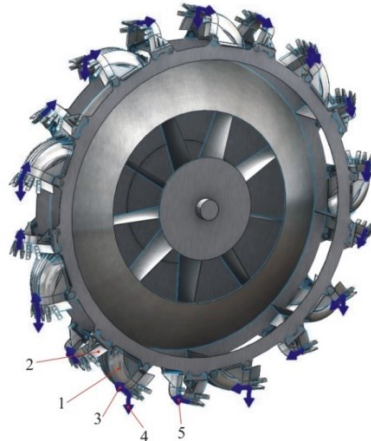
The boom of the ERc 1400-30/7 bucket wheel excavator is a spatial, load-bearing structure that can be divided (Vilceanu, 2018) into three sections (Figure 2).



**Fig. 2 Sections of the boom**

1. The joint section between the boom and structure, which allows both vertical and horizontal movement;
2. The intermediate section on which the conveyor belt is mounted, for the discharge of the excavated material;
3. The bucket wheel support section on which the drive mechanisms are mounted, as well as the boom hoist cables attachment device (Drebenstedt, Paessler, 2006).

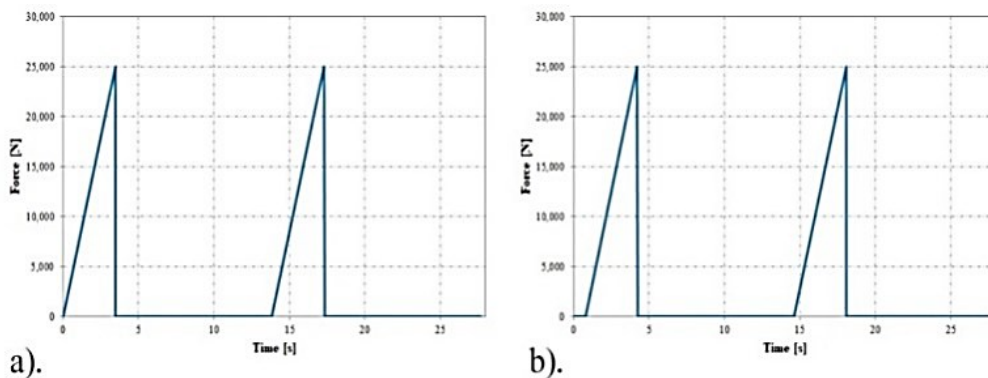
The BWE we analyzed has 9 cutter-loader buckets and 9 cutter buckets fitted on the wheel (Figure 3).



**Fig. 3 ERC 1400-30/7 BWE bucket wheel**

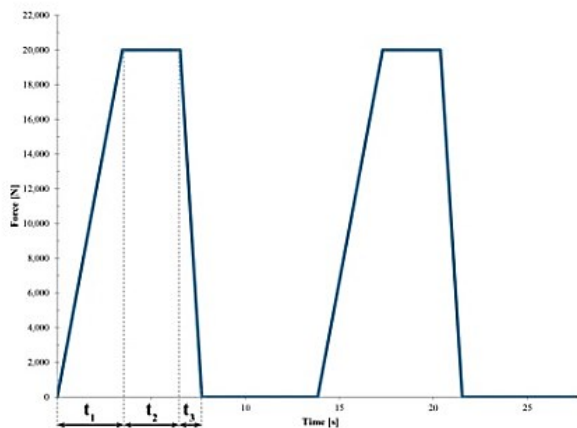
During excavation there are two types of forces:

1. the cutting forces, that are tangent to the circle described by cutting edges and act both on the cutter-loader and the cutter buckets (Figures 4a, 4b);
2. forces corresponding to the weight of the material, the action of which occurs as soon as a cutter-loader bucket enters the excavation process until the material in this bucket is discharged to the conveyor in the boom. (Figure 5). Here we highlighted the time intervals  $t_1$  corresponding to the loading of the cut material,  $t_2$  corresponding to the lift of the loaded bucket up to the discharge level and  $t_3$  corresponding to the material discharge on the conveyor (Kertesz (Brînaș), 2018).



**Fig. 4 Cutting forces diagram:**  
a). for cutter – loader buckets, b). for cutter buckets

The forces' variation diagrams shown in Figures 4.a, 4.b, and 5 correspond to a single cutter loader bucket and cutter bucket respectively and are plotted for two complete rotations of the bucket wheel, during which each bucket performs cutting twice.

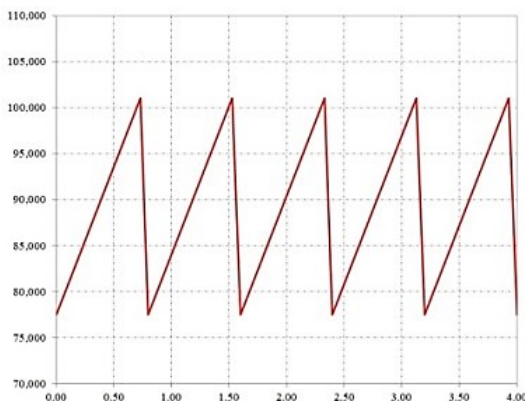


**Fig. 5 Diagram of the forces corresponding to the weight of material with highlighted time intervals**

The offset time that occurs between two successive curves in Figures 4a and 4b respectively, is given by the rotation speed of the bucket wheel. For the studied BWE the speed is 4.33 rpm which leads to an offset time of 13.86 second between two successive cuts of the same bucket. The time gap between the diagrams in Fig. 4.a and 4.b corresponds to an angular distance between two successive buckets (one cutter and one cutter-loader) of  $2\pi/18$  during a 0.77 s time interval. The weight of the loaded material only refers to the cutter-loader buckets, at a discharge rate of 39 buckets/min, which means an identical gap of 13.86 s, only the duration of the process is longer, as to the time of the cut  $t_1$  the lifting and discharge times ( $t_2$  and  $t_3$ ) are added.

By overlapping the three forces (offset as shown) and reducing them to the bucket wheel shaft, the time variation of the resultant force on the shaft is obtained.

The variation in continuous regime of the thus determined resulting force at the bucket wheel shaft is shown in Figure 6 (Radu et al. 2018).



**Fig. 6 Time variation of the force at the bucket wheel shaft**

This excitation force is the main source of vibration of the boom, vibrations that are transmitted through the boom to the entire structure of BWE.

### THE MODEL OF THE BOOM

Based on the constructive characteristics (ROMINEX, 2007) of the ERC 1400-30/7 BWE, a model of the bucket wheel boom was built in SOLIDWORKS®. This model will be used to analyze the frequency response to the forces afore explained, generated during the excavation process. Prior to this, the static stresses acting on the bucket wheel boom are first determined.

The main static loads (mass forces) to which the boom and its elements are subjected are presented in Table 1, where the type of SolidWorks® Simulation specific type of load is also defined.

**Table 1 Static loads (mass forces) and SOLIDWORKS specific loads correspondence**

No.	External Loads	Unit	Value	SOLIDWORKS® type of load
1	Conveyor belt mounted on the boom	kg	25000	Remote Loads/Mass
2	Cinematic chain of bucket wheel drive	kg	29500	Distributed Mass
3	Virtual bucket wheel	kg	39600	Part
4	Hoist cables of the boom	N/m	2x35000000	Spring

### MODAL ANALYSIS OF THE BUCKET WHEEL BOOM STRUCTURE

The modal analysis implies determining the vibration modes of a structure. It is realized under the assumption of free vibrations and lack of damping. If the damping is considered during the determination of own frequencies, then the results obtained refer to the resonance frequencies of the structure.

A vibration mode can be defined as a preferred way of vibration of a structure. It is characterized by:

- the vibration frequency;
- percentage of masses participating in vibration;
- the shape of the structure deformation.

If modal analysis is done using the Finite Element Analysis (FEA), it is essential to evaluate the effective mass participation factor (EMPF) that provides a measure of the energy of each vibration mode (Muhammad et al. 2016), because it represents the part of mass of the system contributing to that mode. Its equation is:

$$p_i = \frac{\vec{\varphi}_i^T \cdot [M] \cdot \vec{r}}{(\vec{\varphi}_i^T \cdot [M] \cdot \vec{\varphi}_i)^2} \quad (1)$$

where:

[M] is the mass matrix of the structure;

$\vec{\varphi}_i$  – normalized form of deformations

$\vec{r}$  – is the coefficient of the exciting movement influence (Priestley et al. 1996).

For a given direction, the sum of all effective mass participation factors is the Cumulative Effective Mass Participation Factor (CEMPF). When this is in the range of 80% to 90% in any direction of response, it is considered that the dominant dynamic response of the structure was determined.

$$80 \leq 100 \cdot \sum_{i=1}^n p_i \leq 90 \tag{2}$$

where:

$n$  – is the number of modes considered.

Practically it is necessary to verify that the CEMPF  $\sum_{i=1}^n p_i$  is between 80% and 90% for those directions (X, Y or Z) where significant dynamic load is expected. In the case of the structure of the bucket wheel excavator boom, we will consider that the modal analysis (Gottvald, 2010) is done when equation (2) is satisfied on X and Y directions.

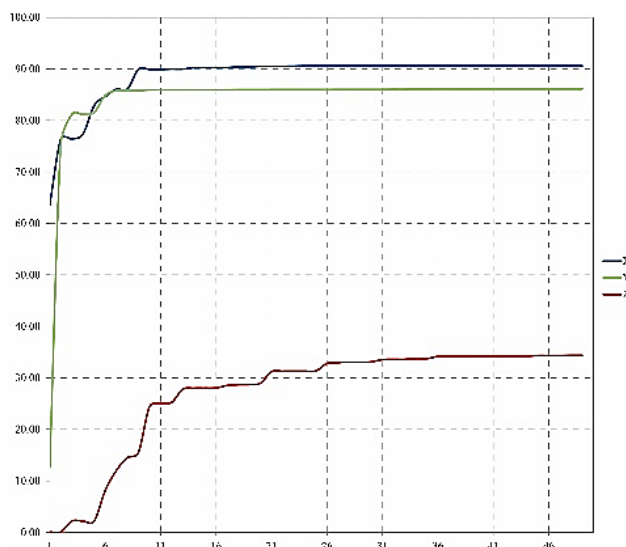
We analyzed the modal frequencies for 49 modes. Figure 7 shows the CEMPF for all three directions of the coordinate system (see Figure 2) and in Table 2 the list of modal frequencies and the corresponding modes are listed.

**Table 2 List of modal frequencies**

Mode number	1	2	3	4	5	6	7	8	9	10	11	12	13
Frequency [Hz]	1.98	2.11	5.32	5.92	8.19	9.30	11.14	13.73	13.96	16.84	16.96	17.13	17.22

Analyzing Figure 7, we can conclude that the 49 modes analyzed are sufficient because the CEMPF for X and Y directions starting with Mode 11 exceed 80% and remain constant up to Mode 49. These directions are the ones that are actually expected to have vibration induced stresses due to the excavation process and also because of the constructive structure of the boom.

As far as the CEMPF for direction Z is concerned, it exceeds 30% after mode 26 and has small variations up to mode 49. Because no vibration induced stresses are expected on this direction, we considered that the 49 modes analyzed are sufficient, particularly because after mode 36 the CEMPF for Z direction remains constant.



**Fig. 7 Cumulative effective mass participation factor [%]**

Figure 8 shows a histogram of the EMPF depending on the vibration modes for X direction. It can be noted that mode 1, with a corresponding frequency of

1.9779 Hz, is the one for which the EMPF for X direction has the highest percentage, (of over 60%). A percentage above 10% is also shown for mode 2 with a corresponding frequency of 2.1092 Hz.

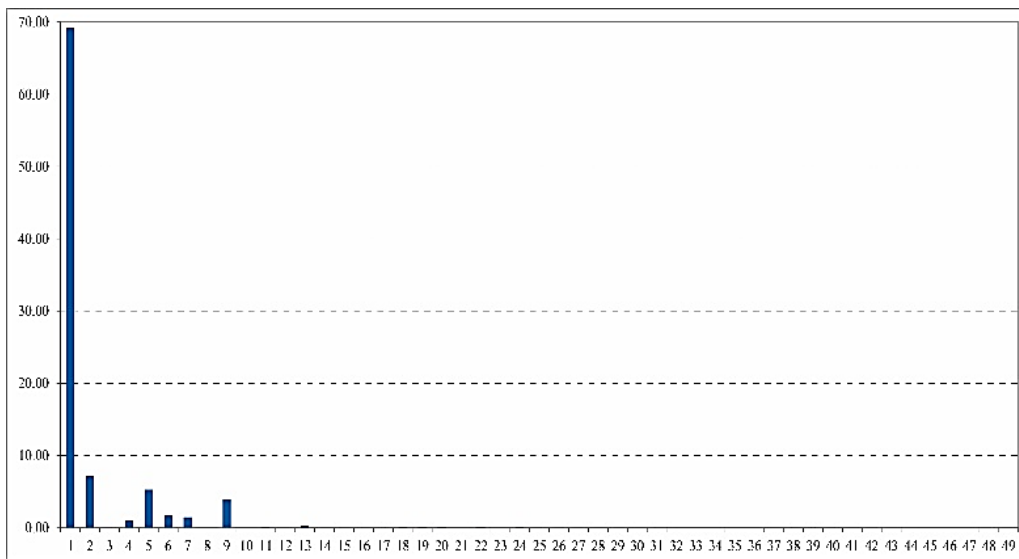


Fig. 8 The EMPF histogram for direction X

For Y direction (Figure 9), mode 1 has a EMPF percentage value of more than 10%, and mode 2 has a EMPF value of over 60%.

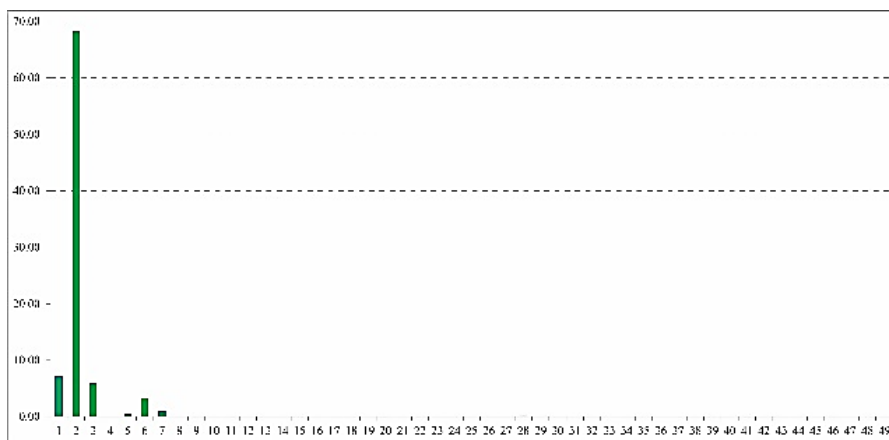


Fig. 9 The EMPF histogram for direction Y

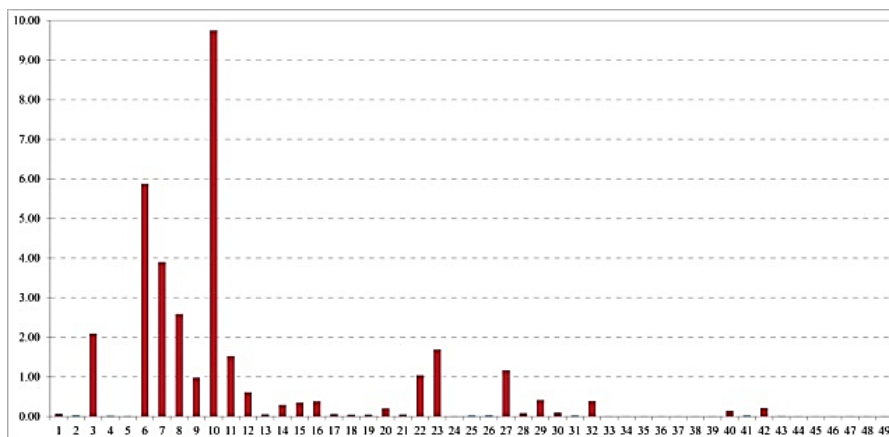


Fig. 10 The EMPF histogram for direction Z



Figure 10 shows the histogram of the EMPF for Z direction. It can be seen that values of the first two modes are negligible. Significant values are present starting with mode 3, with a maximum value of around 9% for mode 10, corresponding to a frequency of 16.841 Hz. The value of 16.841 Hz will be the reference in the study of the dynamic regime of frequency response, being the limit to which the force corresponding to the excavation process will be developed in Fourier time series.

## THE DYNAMIC REGIME OF FREQUENCY RESPONSE DURING EXCAVATION

The characteristic equation for the dynamic regime of frequency response is:

$$[M]\ddot{d} + [C]\dot{d} + [K]d = F(A \cos(\omega t) + B \sin(\omega t)) \quad (3)$$

where:

$M$  – is the mass matrix,

$C$  – is the damping matrix,

$K$  is the stiffness matrix,

$F(t)$  – is the vector of nodal loads, expressed as a function of time,

$d$  – is an unknown vector of the nodal displacements (Kurowski, 2015, 2016).

To avoid resonance issues, it is necessary for the analyzed system to satisfy the condition:

$$f_{rk} \leq 0,8 \cdot f_0 \quad (k = 1 \dots n) \quad (4)$$

where:

$f_0$  – is the exciting frequency (i.e.  $f_0$  is the expected dynamic load frequency),

$f_{rk}$  – is a resonance frequency of the analyzed system (frequency of  $k$  order),

$n$  – is the maximum number of modes analyzed. In the same way, we can write that:

$$f_{rk} \geq 1,2 \cdot f_0 \quad (k = 1 \dots n) \quad (5)$$

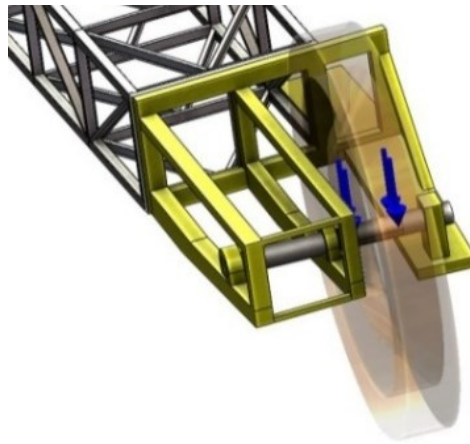
which means that the resonance frequencies considered should be 20% lower or higher than the expected frequency (Muhammad et al. 2016, Priestley et al. 1996).

The dynamic analysis of the frequency response is suitable for excitation frequencies that have a slow variation and require the definition of global damping as a percentage of the critical one (Aggogeri et al., 2017, Rea et al., 1971). In the actual paper the values of 2%, 5% and 10% were considered for the global damping coefficient.

The exciting force is applied to the surface of the bucket wheel shaft, as a uniformly distributed force on its support surface, and it is a function of frequency (Figure 11).

In order to express the force in relation to frequency we approximated the force variation (shown in figure 6) for one period, using Fourier series development.





**Fig. 11 Applying the force to the bucket wheel shaft**

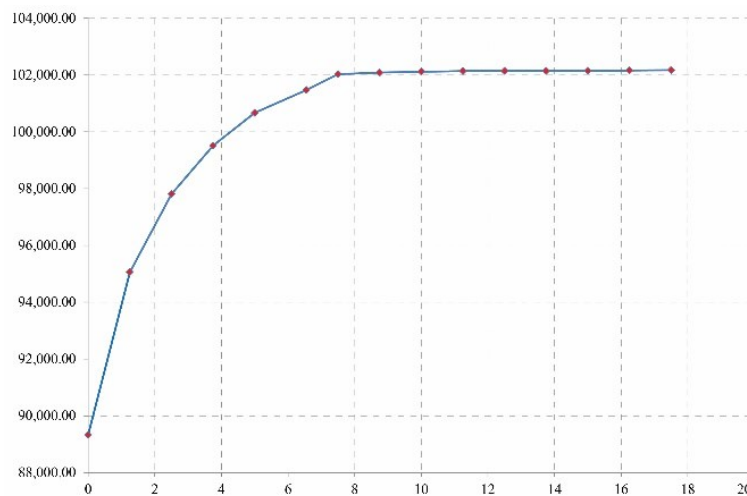
The equation of the approximation function is:

$$f_1(t, N) = \frac{1}{2} \cdot A_0 + \sum_{n=1}^N [A_n \cdot \cos(n \cdot \omega \cdot t) + B_n \cdot \sin(n \cdot \omega \cdot t)] \quad (6)$$

where:

$$B_n = \frac{2}{T} \cdot \int_0^T f(t) \cdot \sin(n \cdot \omega \cdot t) dt \quad A_n = \frac{2}{T} \cdot \int_0^T f(t) \cdot \cos(n \cdot \omega \cdot t) dt \quad (7)$$

The actual force and corresponding frequency values are shown in Table 3, and Figure 12 shows the graph of the force variation in relation to frequency.



**Fig. 12 Variation of force in relation to frequency**

**Table 3 Force values corresponding to frequency values**

Frequency [Hz]	0	1.25	2.5	3.75	5	6.25	7.5	8.75	10	11.25	12.5	13.75	15
Effective force [kN]	89	5.73	2.75	1.70	1.15	0.80	0.55	0.053	0.034	0.019	0.007	0.006	0.006

Using the bucket wheel boom model presented in paragraph 2 of the paper and imposing an exciting force with a frequency variation graph as in Figure 12, we performed the dynamic frequency response analysis.

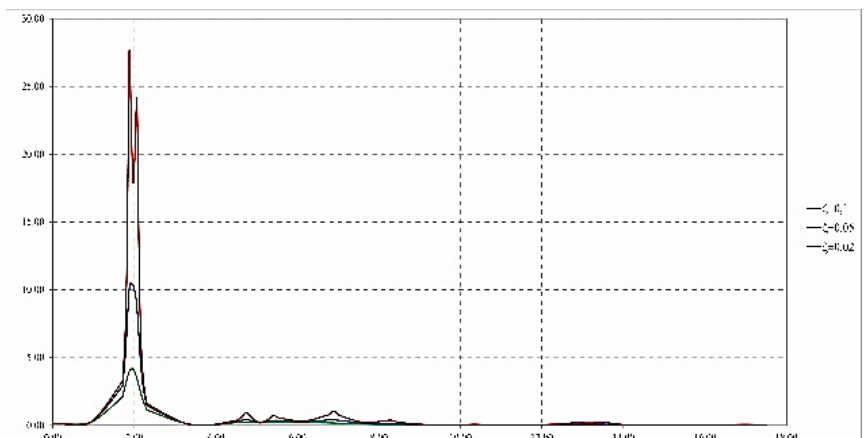
Table 4 presents the frequencies obtained from the modal analysis, the frequencies resulting from the dynamic frequency response analysis and the

percentage changes in the modal frequencies if the analysis was performed in considering damping (resonance frequencies).

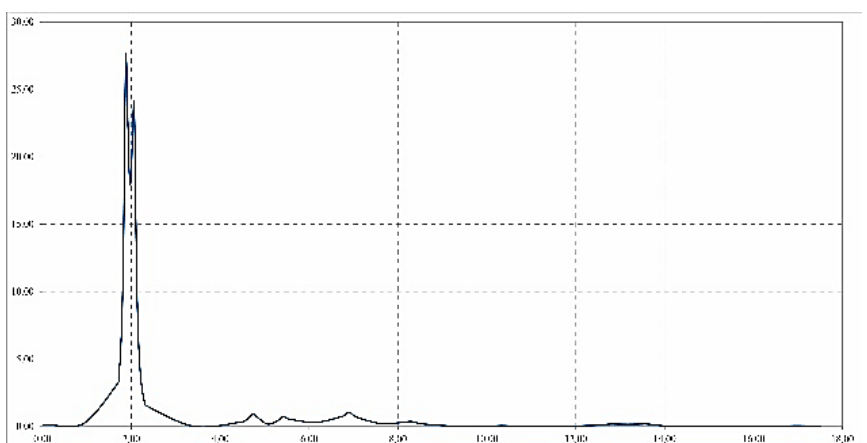
**Table 4 Results of modal analysis, dynamic frequency response analysis and percentage changes**

Mode number	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Resonance freq. [Hz]	1.875	2.07	4.752	5.396	6.906	8.308	10.31	12.73	13.63	13.95	16.77	17.02	17.22	17.37	17.5
Modal freq. [Hz]	1.978	2.109	5.321	5.917	8.187	9.30	11.14	13.73	13.96	16.84	16.96	17.13	17.22	17.35	17.4
Deviation perc. [%]	5.21	1.85	10.71	8.81	15.65	10.66	7.49	7.28	2.36	17.17	1.12	0.64	0.01	-0.13	-0.75

As with modal analysis, the maximum value of EMPF for direction X is obtained for the first frequency at all three global damping values. The frequency response curves for the X direction at global damping of 2%, 5% and 10% are shown in Figure 13. It can be seen that the maximum response (deformation) is obtained for the frequency of 1.875 Hz. A significant value of the response is also determined by the second frequency having the value of 2.07 Hz. The other resonant frequencies do not determine significant values of the response in X-direction. Separately we presented the frequency response on all directions at 2% damping (Figures 14, 16 and 20) because this overall damping is specific to lattice (truss like) beams (Pietrusiak et al. 2017).

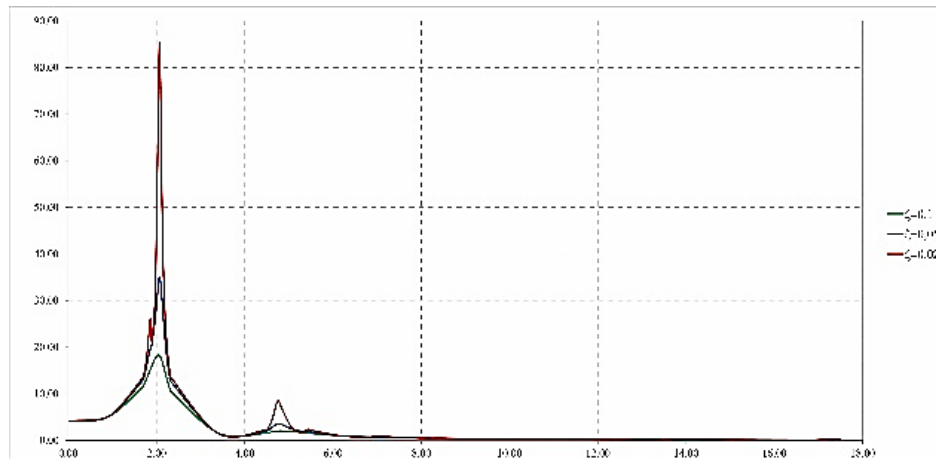


**Fig. 13 Response for X direction to the dynamic frequency response analysis**

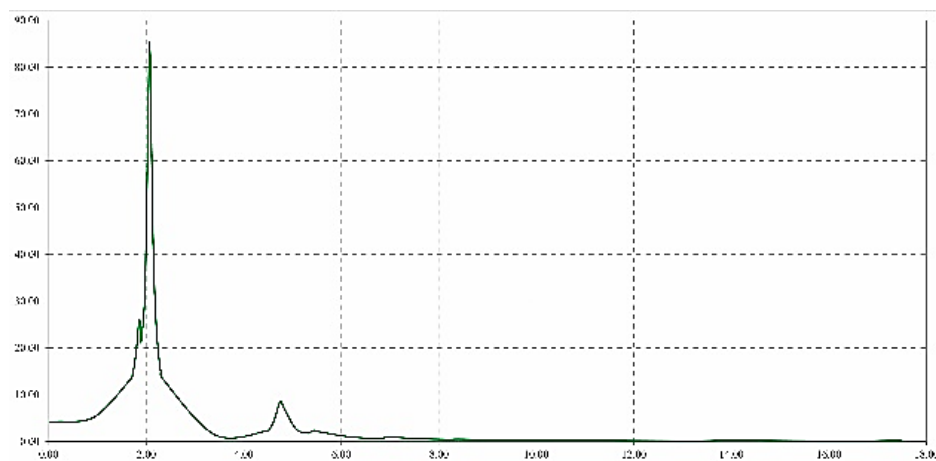


**Fig. 14 Response for X direction to the dynamic frequency response analysis with 0.02 damping**

The maximum value of EMPF for direction Y is obtained for the second frequency (Table 4). The frequency response curves for Y direction at global damping of 2%, 5% and 10% are shown in Figure 15. The maximum response (deformation) is obtained for the frequency of 2.07 Hz. A significant value of the response is also determined by the first frequency of 1.875 Hz. The other resonant frequencies do not determine significant values of deformation.



**Fig. 15** Response for Y direction to the dynamic frequency response analysis



**Fig. 16** Response for Y direction to the dynamic frequency response analysis with 0.02 damping

The highest value of EMPF for Z direction is obtained for the 8<sup>th</sup> frequency of 12.73 Hz (Figure 17). As in previous cases, Figure 18 shows the comparative curves of the dynamic frequency response analysis for the three global damping ratios.

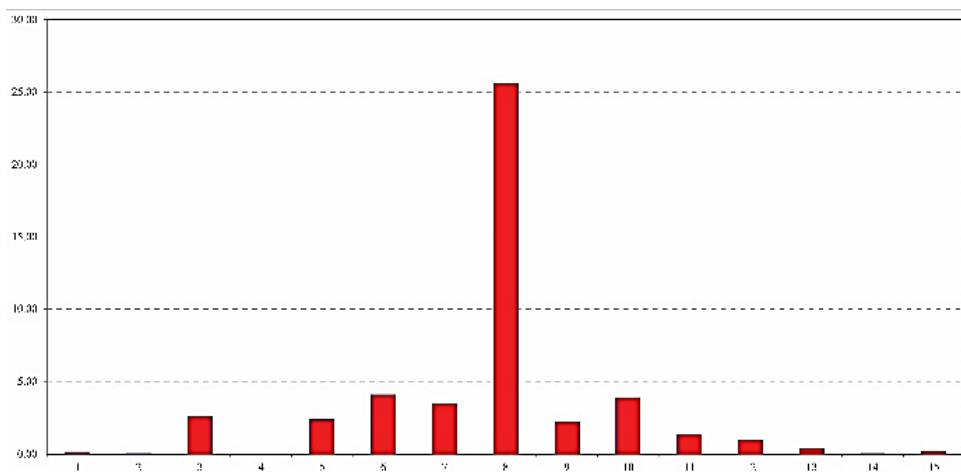


Fig. 17 Histogram (EMPF) for Z direction - Dynamic frequency response analysis

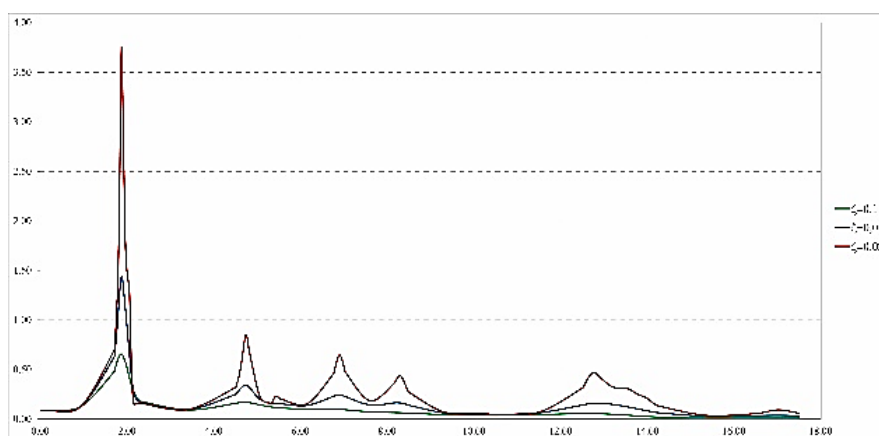


Fig. 18 Response for Z direction to the dynamic frequency response analysis

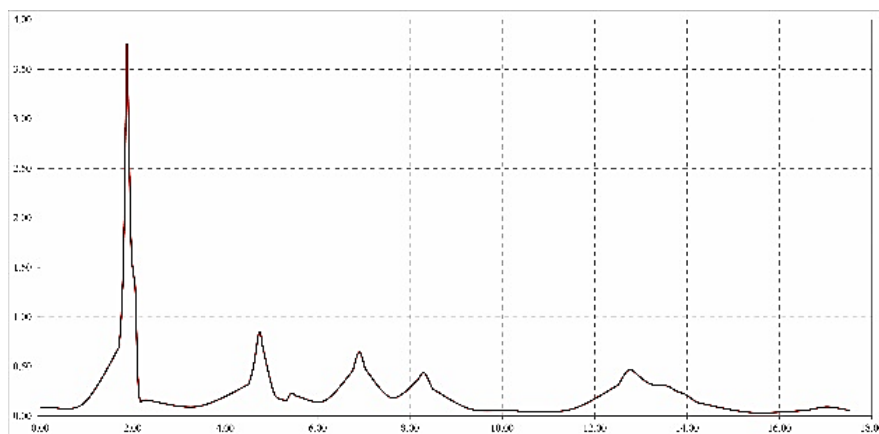


Fig. 19 Response for Z direction to the dynamic frequency response analysis with 0.02 damping

### CONCLUSIONS

Using SolidWorks®, a model of the ERC 1400-30/7 bucket wheel excavator boom was built, that we used to perform the frequency response analysis of the stresses generated by the bucket wheel during excavation. For this purpose a mathematical model was defined of the resultant of the excavation forces, which are the main source of vibration of the boom, and we determined the static stresses acting on the bucket wheel boom.

We performed modal analysis for 49 modes and we determined CEMPF on the three directions of the coordinate system and calculated the modal frequency values.

We carried out the dynamic analysis of the frequency response of the excavator boom using series evolution of time variation of the resultant excavation forces and using the build boom model, considering 2%, 5% and 10% overall damping values.

For direction X (lateral), mode 1 corresponding to a frequency of 1.9779 Hz, the value (EMPF) has the highest percentage (over 60%), mode 2 having a value below 10%.

For Y direction (vertical), mode 2 corresponding to 2.11 Hz corresponds to an (EMPF) of over 60%, and mode 1 has a EMPF of less than 10%.

Frequencies near 2 Hz correspond to maximum amplitudes in all three directions (x lateral, y-vertical and z-longitudinal). They are close to the fundamental frequency and to the first harmonic of the excitation force. At the same time, the cumulative mass participation factor for modes 1 and 2 is over 60%.

Higher amplitudes at these frequencies are recorded in the Y direction, followed by X, and are much lower in the Z direction.

The existence of significantly high fundamental components indicates the inertial damping effect of the concentrated and distributed masses, pointing at a low probability of self-sustaining vibrations.

It is noticed that the maximum response is obtained for the frequency of 2.07 Hz. The limits of the safety interval for this frequency are:

$$\begin{aligned} f_{\min} &\leq 0.8 \cdot f_{\text{resonance}} = 0.8 \cdot 2.07 = 1.656 \text{ [Hz]} \\ f_{\max} &\leq 1.2 \cdot f_{\text{resonance}} = 1.2 \cdot 2.07 = 2.484 \text{ [Hz]} \\ f_{\text{excitation}} &\notin (1.656 \dots \text{ [Hz]}) \end{aligned} \quad (8)$$

The force causing the vibration was considered 1.25 Hz. Since this value is outside the interval (1.656...2.484) Hz, there is no danger of resonance.

With respect to the Z (longitudinal) direction, the largest percentage share of the masses is obtained for the 8th frequency of 12.73 Hz. As can be seen in Figures 18 and 19, the response for this frequency is lower than that obtained for lower order frequencies having a lower percentage of participation.

The explanation of this result resides in the fact that at high frequencies there is a tendency to maximize elastic energy and minimize kinetic energy.

Consequently, it is explicable why although the percentage participation of the masses for the 8th frequency is over 25% the response is inferior to the 3rd frequency of 4.75 Hz with a percentage participation of the masses of 2.5%, of the 5th of 6.9 Hz with a mass participation percentage of 2.5% and approximately equal to that of the 6th frequency of 8.3 Hz with a mass participation percentage of 8%.

If for the same structure, with the same loading regime, the modal analysis and the dynamic analysis of the frequency response is performed concurrently – as presented in this paper – that is beneficial because the two methods provide complementary and concordant information.

The proposed method is approachable by easily available numeric methods, and it is useful both in the design phase of new load-bearing structures of truss type subjected to high-variability forces, and also in refurbishment or improvement phases of the existing structures of this kind.

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**Abstract.**

The paper deals with the modal analysis and frequency response analysis of a bucket wheel excavator (BWE) boom, obtained by simulation, based on a virtual model of an existing BWE boom. The boom, which generally is realized as a spatial truss, is the most vulnerable subsystem of the BWE, being submitted to severe operational loads characterized by very pronounced cyclical, dynamic and stochastic variability. This vulnerability is the consequence of its shape and constructive parameters and the nature, source and character of the external exciting loads to which it is exposed. The classical approach recommended by standards and norms cannot predict the occurrence of failures caused by vibration, which produces fatigue due to the load's cyclical variability and the deformation produced by resonant vibration of some constitutive elements. As exciting load we considered the operational forces acting on the bucket wheel. In this manner we can take into account the constructive features – with modal analysis, and the vibration regime – with frequency response analysis. The proposed method is useful both in the design phase of new load-bearing structures of truss type subjected to high-variability forces, and also in refurbishment or improvement phases of the existing structures of this kind.

**Keywords:** Bucket wheel excavator, damping, frequency response, modal analysis, resonance