TECHNICAL UNIVERSITY OF CLUJ-NAPOCA

ACTA TECHNICA NAPOCENSIS

Series: Applied Mathematics, Mechanics, and Engineering
Vol. 63, Issue IV, November, 2020

THE MODAL ANALYSIS, USING SIMULATION AND MODELLING, OF THE BOOM OF THE ERC-1400 BUCKET-WHEEL EXCAVATOR DURING OPERATION

Florin Dumitru POPESCU, Sorin Mihai RADU, Andrei ANDRAȘ, Ildiko BRÎNAȘ, Mihaela Olga MARIȚA, Narcis Ionel REBEDEA

Abstract: Bucket-wheel excavators (BWEs) are subjected to vibrations produced by the excavation process. The boom of the BWE supports and is used to position the bucket-wheel at cutting heights up to 30 meters. The frequency of the vibrations induced by the excavation process is dependent on the number of installed buckets, and the rotation speed of the bucket-wheel. In this paper we propose a model for the boom for which modal analysis is performed. Based on the modal analysis, for the excavation forces we have studied the frequency response of the excavator boom.
Keywords: *Bucket-wheel excavator, modal analysis, resonance frequency, vibration, ERC-1400, simulation, excavation.*

1. INTRODUCTION

The ERc 1400-30/7 model BWE (figure 1) is a continuously digging equipment used in open-pit coal mines. It digs overburden or coal using an array of buckets mounted on the circumference of a rotating wheel, while simultaneously moving the material to its discharge point with the help of conveyors mounted on its structure.

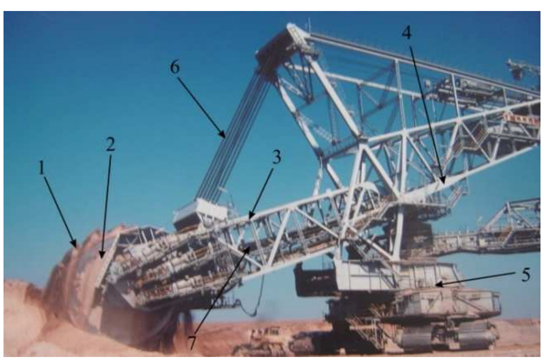


Fig.1. The studied model BWE
(1 – bucket, 2 – bucket-wheel, 3 – boom of the bucket-wheel, 4 – boom of the conveyor belt, 5 – pivoting mechanism, 6 – hoisting cables, 7 – conveyor belt)

The actual tool of the BWE is the bucket-wheel. This rotates, and at the same time, it performs a pivoting horizontal movement using its boom and a vertical positioning at the desired height using the hoisting cables.

2. MODELLING OF THE ERC 1400-30/7 BWE BOOM

The boom of the BWE is a load-supporting spatial structure. It is sectioned into three parts (Figure 2): 1 – the connection part of the boom to the rest of the excavator, allowing both the slewing and the horizontal positioning of the boom, 2 – the middle part which has a support role for the discharge conveyor belt, and 3 – the support part of the bucket wheel which also houses the drive mechanism and has the hoisting cables attached.

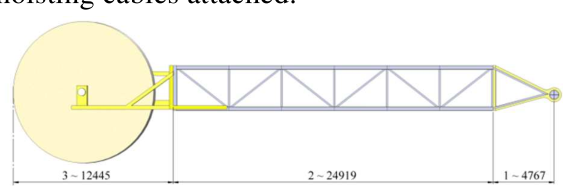


Fig.2 The parts of the model of the BWE boom

The middle part of the boom (figure 3) has the following parts: an upper and a lower chord, each made of two longitudinal load bearing beams (coloured in red), struts (coloured in green) and wind bracings (coloured in yellow) as well a left and a right girder, each made of vertical struts (coloured in grey) and wind bracings (coloured in blue).

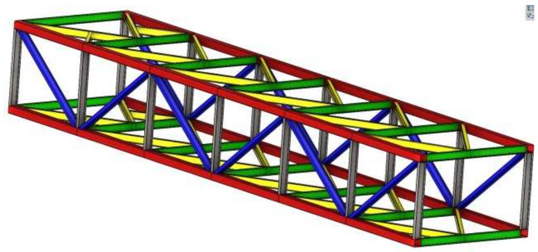


Fig.3. The middle section of the BWE boom

From a structural point of view, this part is a truss made up of vertical, horizontal and inclined beams which are subjected mainly to bending (flexural) stresses. In addition to the bending stress there are shear forces, axial forces and torsional forces acting on the structure. Discretization used in the finite element analysis involves the following options:

- choosing solid type finite elements in case the analysed body has comparable dimensions on the three coordinate axes;
- choosing surface type finite elements (shells) if the analysed body is assimilated with a metal sheet;
- choosing beam type finite elements if we have to evaluate a structural model.

2.1. The bucket-wheel modelling

The rotor (bucket-wheel) is the main functional part of the excavation process which causes stresses to the boom [2], [3], [4].

We created a simplified model, similar in size to the real one, for the bucket-wheel, (figure 4). The static stress exerted on the boom by the bucket-wheel is determined by its weight. We imposed a density of $\delta = 373 \frac{kg}{m^3}$ on the material of the model, so that the model has a similar mass to that of the real bucket-wheel.

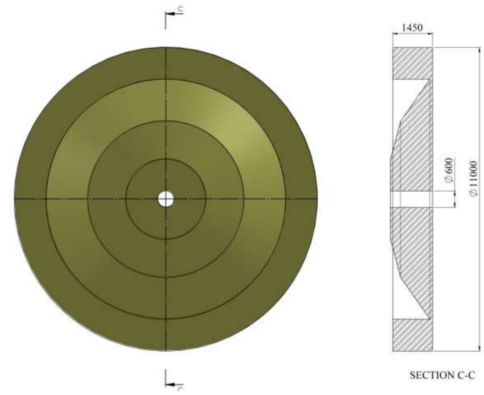


Fig.4. The bucket-wheel model

2.2. Modelling of the bucket-wheel drive mechanism kinematic chain

The drive mechanism of the bucket wheel is a kinematic chain consisting of the following parts (figure 5): the electric motor – 1, the flexible coupling – 2, the gear-box – 3, the drive shaft – 4, the bucket-wheel reducer gear – 5, the bucket-wheel – 6 (cutting buckets – 6.1, cutting-loading buckets – 6.2), and the bucket wheel axle – 7 [4], [7].

All these elements are modelled by a uniform distributed mass (figure 6). The placement of the bucket-wheel model is also shown in this image.

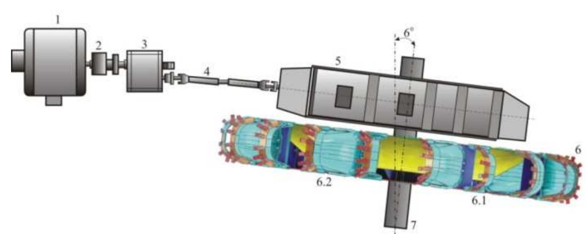


Fig.5 The parts of the drive mechanism of the bucket wheel

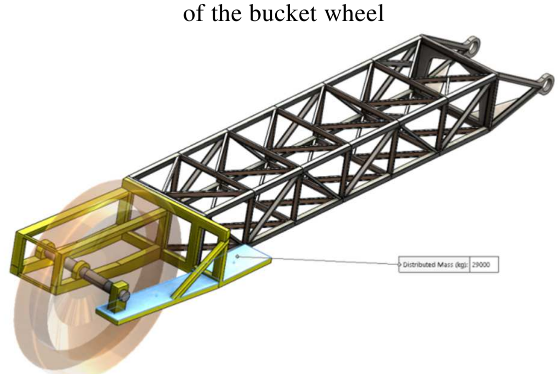


Fig.6 Model of the drive mechanism using a uniform distributed mass

2.3. Modelling of the conveyor belt mounted on the BWE boom

The conveyor installed on the boom was modelled as a remote load/mass (fig. 7). The technical documentation of the BWE gives a value of 25,000 kg for this [6].

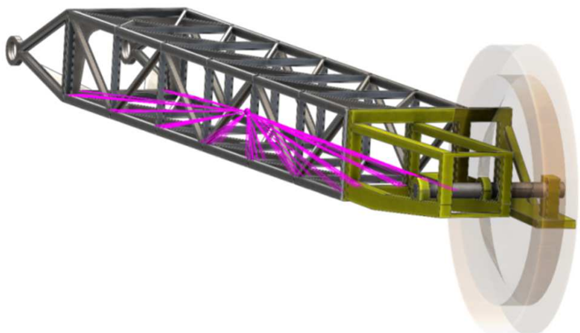


Fig.7 Model of the conveyor installed on the boom as a remote mass

2.4. Modelling the boom hoisting cables

The 10 galvanised hoisting cables (type WS40-6x36) used for lifting the boom to the working position and support its weight are modelled by two springs which are subject to extension (figure 8). Their equivalent constant of elasticity is equal to $35\text{x}10^6 \frac{N}{m}$ per cable. [6].

The major static strains exerted on the bucket-wheel and the part that generates these, are shown in Table 1. The description of the SolidWorks Simulation specific load type is also detailed.

Figure 9 shows the nodal network of the beam structure of the BWE boom, and the coordinate system used as reference for the interpretation of results.

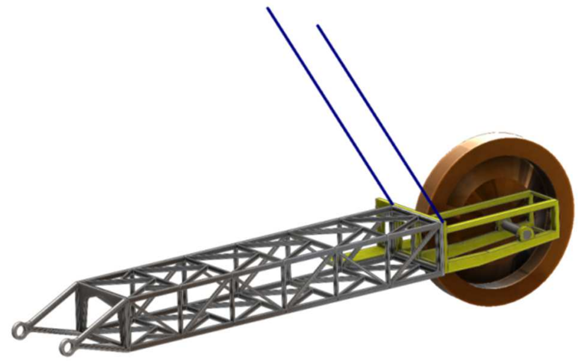


Fig.8 Model of the boom hoisting cables using two springs

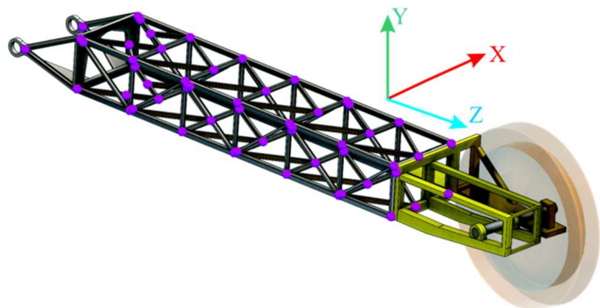


Fig.9 The nodal network of the beam structure and the coordinate system

3. THE MODAL ANALYSIS OF THE BOOM MODEL OF THE BWE

3.1 Theoretical considerations regarding modal analysis

A rigid body which does not possess any movement restrictions displays 6 movement components within a system of coordinates: 3 translations noted $(\bar{u}_1, \bar{u}_2, \bar{u}_3)$ and 3 rotations noted $(\bar{r}_1, \bar{r}_2, \bar{r}_3)$. These determine how many degrees-of-freedom (DoF) the rigid body has. (figure 10) [1].

Table 1

No	External static stress	Unit	Value	SolidWorks® type
1	The conveyor system of the boom	Kg	$25\ \text{x}10^3$	Remote loads / mass
2	The drive mechanism kinematic chain	Kg	$29.5\ \text{x}10^3$	Distributed mass
3	A simplified bucket-wheel model	Kg	$39.6\ \text{x}10^3$	Part
4	The boom hoisting cables	N/m	$2\text{x}35\text{x}10^6$	Spring

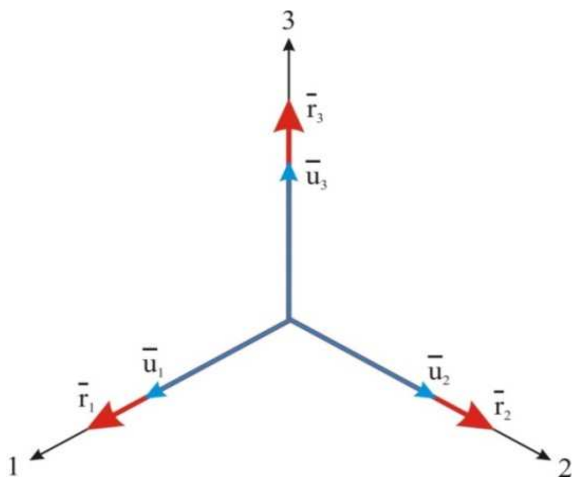


Fig.10 A rigid body's degrees-of-freedom (DoF) in the local coordinates system

The discrete systems subject to vibrations are characterized by the fact that mass and elasticity can be separated conceptually so that some of their parts, due to their mass, can determine the inertial behaviour (mass), and other parts determine the elastic behaviour of the system (the elasticity constant).

In the case of distributed systems subject to vibrations, the conceptual mass and elasticity components can no longer be separated. All the parts of a body determine to the same extent both the inertial and the elastic behaviour of the system. In theory, these systems have unlimited degrees-of-freedom (DoF). If the study is performed using finite elements (FEM), then the degrees-of-freedom becomes limited, and is determined by the number of nodes of the discretized structure. Each node of a structure discretized through the finite element method displays 3 degrees of freedom. Thus for a discretized distributed system we will have $3 \cdot N_{nodes}$ degrees of freedom.

Modal analysis entails the calculation of the specific modes of vibrations of a structure. It is performed within the free vibrations hypothesis and in the lack of amortization.

If in the calculation of the specific frequencies, the amortization is also taken into consideration, the obtained results refer to the structure's frequencies of resonance [6]. A vibration mode is defined as one favourite modality of vibration for a given structure.

The characteristics of a vibration mode are:

- the frequency of the vibration;
- the Effective-Mass Participation Factor (EMPF) for that vibration;
- the structure's deformation shape.

The number of the vibration modes is equal to the number of the degrees-of-freedom (DoF) of the structure that vibrates.

The fundamental vibration mode of a structure or mode 1, corresponds with the lowest specific frequency. The other modes are called superior order specific modes and, if *k* is a random specific mode of the structure (*k*=1, …, *N*) then it is possible to write:

$$\begin{aligned} & f_1 < f_2 < \ldots < f_k < \ldots < f_N \text{ or} \\ & T_1 > T_2 > \ldots > T_k > \ldots > T_N \end{aligned} \tag{1}$$

Where

f_k – is the specific frequency of *k* order;

T_k – is the specific time of *k* order.

The inferior modes (with low frequencies) determine the maximization of the kinetic energy and the minimization of the elastic energy of the vibrating structure, while the superior modes (with high frequencies) determine the maximization of the elastic energy and the minimization of the kinetic energy.

The solution to the problem of vibrations is the modal superposition analysis method. This analysis method can be associated with the deformation direction according to the elasticity. Consequently, we can consider an object in a certain vibration mode as a system with a single DoF, which has the characteristics (elasticity, mass, deformation direction) of the properties of the mode in question. Determining the number of modes subject to analysis is extremely important. If one considers that 3 modes are significant then we can emulate the system using 3 oscillators with one degree of freedom. In this respect, the number of the DoF of the system is reduced to three, regardless of its complexity. Taking into consideration the linear system, the problem consists in determining the response to vibrations through the superposition of three systems with one degree of freedom. Figure 11 shows the principle of the method of modal superposition analysis [7], [9].

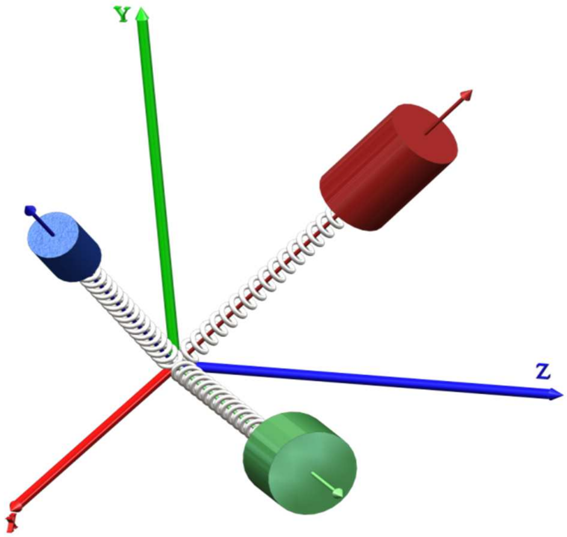


Fig.11 The principle of the method of modal superposition analysis

3.2 The resonance of the vibrating structures

A structure exposed to dynamic loads, with a certain frequency similar to any of its natural frequencies, will present major effort and deformations, a phenomenon known as resonance. Thus we address the problem of establishing the possibility that a specific dynamic load would determine the resonance of the structure in which it operates [7], [11].

To avoid resonance problems, the analysed system must accomplish the relation:

$$f_{rk} \le 0{,}8 \cdot f_0 \quad (k = 1 \ldots n) \tag{2}$$

where:

f_0 – is the frequency of excitation (in other words, f_0 is the frequency of the dynamic load expected);

f_{rk} – is the resonant frequency of k order of the system analysed;

n –the maximum number of modes analysed (usually $n \le 6$).

In a similar way it can be stated that:

$$f_{rk} \ge 1{,}2 \cdot f_0 \quad (k = 1 \ldots n) \tag{3}$$

meaning that the resonant frequencies taken into consideration should be 20 % more or less as compared to the expected frequency or, in order to avoid the resonance phenomenon, the dynamic load frequency should be outside the closed interval in figure 12.

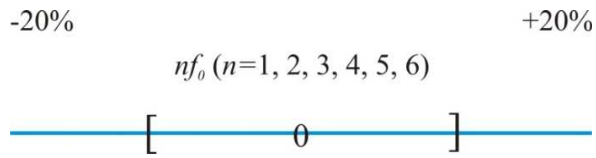


Fig.12 The safety limits of the resonance frequencies

This criterion applies to every natural frequency taken into consideration to assess the possibility of the occurrence of the resonance phenomenon.

A significant problem regarding modal analysis consists in establishing the minimum number of modes which determine satisfactory results. A correct evaluation of the number of modes to be analysed is based on the concept of EMPF (Effective Mass Participation Factor). This offers a value for the energy that each mode of resonance contains.

It represents the mass of the system participating in a given mode. Its expression is [7]:

$$p_i = \frac{\vec{\varphi}_i^T \cdot [M] \cdot \vec{r}}{\left(\vec{\varphi}_i^T \cdot [M] \cdot \vec{\varphi}_i\right)^2} \tag{4}$$

where:

$[M]$ – mass matrix of structure;

$\vec{\varphi}_i$ – normalized deformation shapes;

r – coefficient of the movement influence.

A mode having a high effective-mass, contributes significantly to the system's response. The EMPF can be calculated for distinct deformation directions X, Y and Z, on condition that the sum of the effective-masses for all the modes in one specific direction is equal to the total mass structure.

For a certain direction, the sum of all the EMPF (Effective Mass Participation Factors) represents the CEMPF (Cumulative Effective Mass Participation Factor).

When the CEMPF is between 80% and 90% interval in any response direction, it means that the main dynamic response of the structure was obtained [7]:

$$80 \le 100 \cdot \sum_{i=1}^{n} p_i \le 90 \tag{5}$$

where:

n – the considered number of modes.

Consequently, if it's expected to have a vibration in direction X, the calculus modes must be evaluated till the sum of all EMPFs in that direction becomes approximately 80 to 90%.

Practically, one must verify the condition that CEMPF $\sum_{i=1}^{n} p_i$ is 80% to 90% for those indications (X, Y or Z) where it is expected to have a dynamic load.

3.3 Determining the modal frequencies of the ERc 1400 BWE boom

Using the model of the boom presented above, we have performed its modal analysis under two different circumstances: with boom hoisting cables and without hoisting cables which lift the BWE boom.

As we have conducted the modal analysis in the presence of gravity, thus under the circumstances of preloading the boom's structure, we have used the solving method "Direct sparse solver" of the SOLIDWORKS® application [7].

It is necessary to underline that one must not mistake the preloading of a structure for the stress which causes the vibration. The modal analysis is performed in preloading conditions or when the analysed structure is considered free (without fastening and preloading restrictions).

With the analysed structure, gravity determines a preloading of the excavator boom through its specific mass, the drive mechanism kinematic chain mass, the boom mounted conveyor mass, and the bucket-wheel mass [5], [8].

We have performed the modal frequency analysis for the first 49 modes. Figure 13 presents the CEMPF for the 3 directions of the coordinates system, while figure 14 displays a screenshot of the modal frequencies and their corresponding modes.

Based on figure 13 one can reach the conclusion the number of modes analysed are enough, since the CEMPF in the X and Y directions, beginning with mode 11 (approximately 17 Hz) will exceed 80%. They will also remain constant until mode 49.

In the directions X and Y, vibrations that induce stress will appear. This is expected because of the construction characteristics of the bucket-wheel boom and the actual digging process [4], [10].

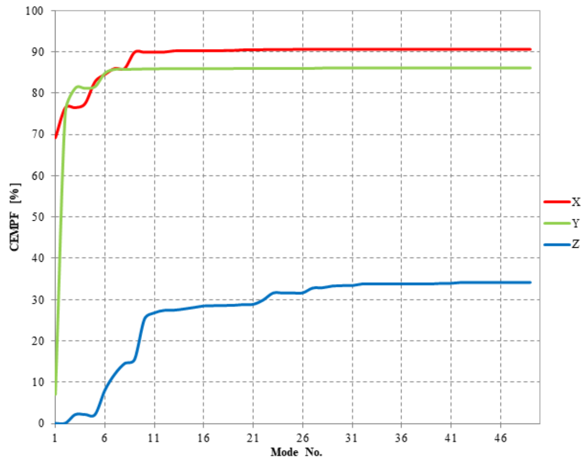
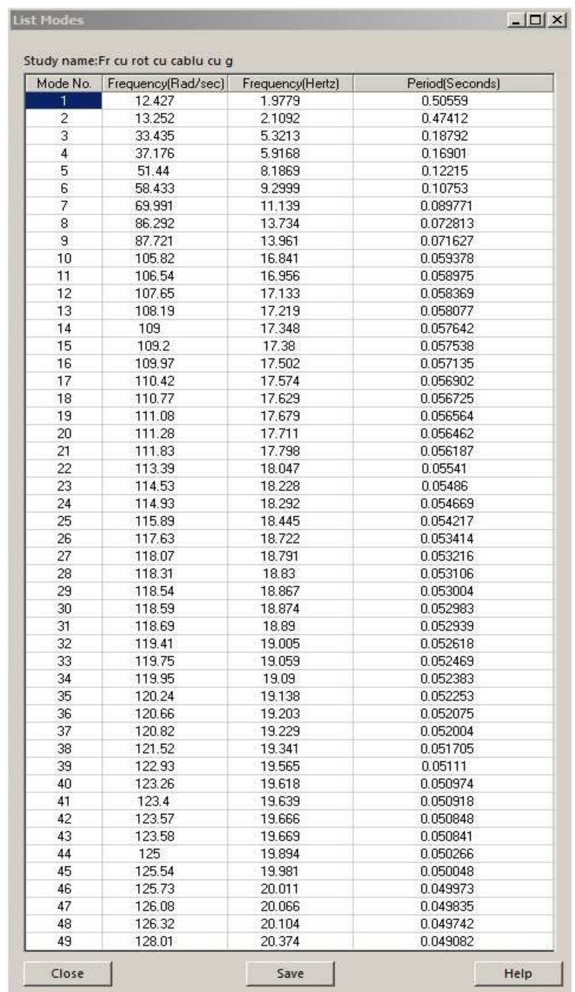


Fig.13 CEMPF for directions X, Y and Z



List Modes

Study name:Fr cu rot cu cablu cu g

Mode No.	Frequency(Rad/sec)	Frequency(Hertz)	Period(Seconds)
1	12.427	1.9779	0.50559
2	13.252	2.1092	0.47412
3	33.435	5.3213	0.18792
4	37.176	5.9168	0.16901
5	51.44	8.1869	0.12215
6	58.433	9.2999	0.10753
7	69.991	11.139	0.089771
8	86.292	13.734	0.072813
9	87.721	13.961	0.071627
10	105.82	16.841	0.059378
11	106.54	16.956	0.058975
12	107.65	17.133	0.058369
13	108.19	17.219	0.058077
14	109	17.348	0.057642
15	109.2	17.38	0.057538
16	109.97	17.502	0.057135
17	110.42	17.574	0.056902
18	110.77	17.629	0.056725
19	111.08	17.679	0.056564
20	111.28	17.711	0.056462
21	111.83	17.798	0.056187
22	113.39	18.047	0.05541
23	114.53	18.228	0.05486
24	114.93	18.292	0.054669
25	115.89	18.445	0.054217
26	117.63	18.722	0.053414
27	118.07	18.791	0.053216
28	118.31	18.83	0.053106
29	118.54	18.867	0.053004
30	118.59	18.874	0.052983
31	118.69	18.89	0.052939
32	119.41	19.005	0.052618
33	119.75	19.059	0.052469
34	119.95	19.09	0.052383
35	120.24	19.138	0.052253
36	120.66	19.203	0.052075
37	120.82	19.229	0.052004
38	121.52	19.341	0.051705
39	122.93	19.565	0.05111
40	123.26	19.618	0.050974
41	123.4	19.639	0.050918
42	123.57	19.666	0.050848
43	123.58	19.669	0.050841
44	125	19.894	0.050266
45	125.54	19.981	0.050048
46	125.73	20.011	0.049973
47	126.08	20.066	0.049835
48	126.32	20.104	0.049742
49	128.01	20.374	0.049082

Close Save Help

Fig.14 Modal frequencies (in the presence of the boom's hoisting cables and considering gravity)

The cumulative-effective mass participation factor in direction Z, will exceed 30% after mode 26 (approximately 19 Hz) and will present slight variations until mode 49. As in this direction we do not expect any vibration stresses, it's assumed that for direction Z, all the 49 analysed modes are enough, since from mode 36 onward the CEMPF in Z direction stays constant.

Figure 15 presents the EMPF for direction X. It can be stated that the first mode, correspondent to a 1.9779 Hz frequency, is the mode for which the top percentage (over 60%) of the EMPF is cumulated for direction X. Mode 2 also reveals the registered values, it can be observed that for the first two modes in the Y direction the values are reversed compared to those obtained in direction X. For direction Y, mode 1 presents an EMPF exceeding 10%, while mode 2 presents an EMPF exceeding 60%.

Figure 17 illustrates the EMPF for direction Z. It can be noted that the values are almost negligible for the first two modes. They become significant starting with mode 3, and the

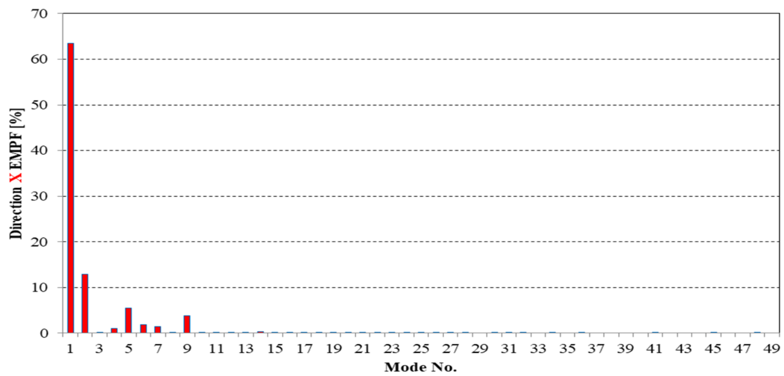


Fig.15 The EMPF for direction X (in the presence of the boom's hoisting cables and considering gravity)

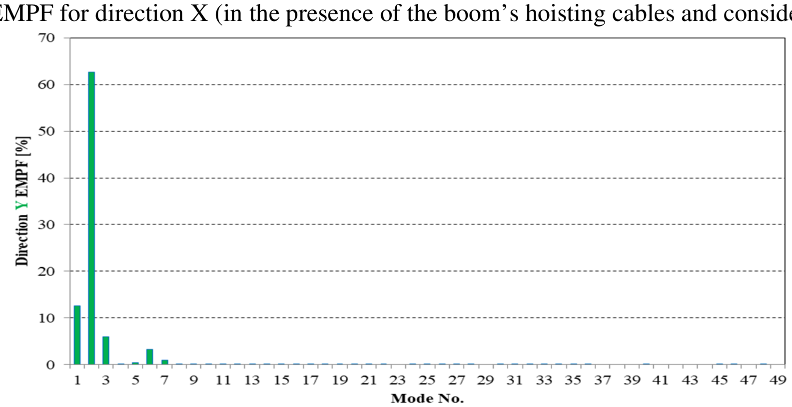


Fig.16 The EMPF for direction Y (in the presence of the boom's hoisting cables and considering gravity)

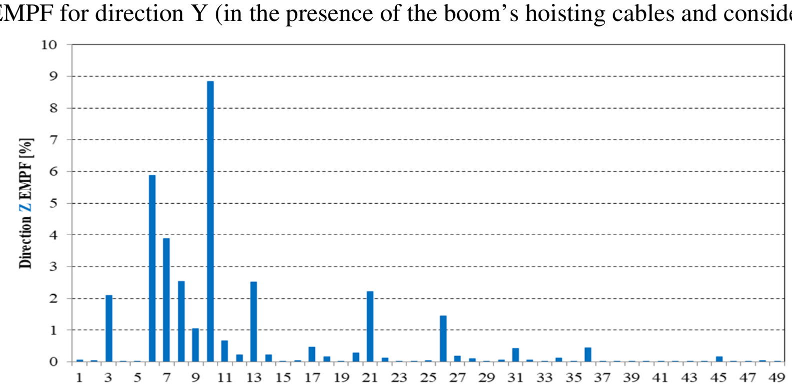


Fig.17 The EMPF for direction Z (in the presence of the boom's hoisting cables and considering gravity)

a percentage of over 10%, a 2.1092 Hz frequency being displayed.

EMPF for direction Y is represented in figure 16. From the point of view of the proportions of maximum value is registered for mode 10, with a value close to 9%. This corresponds to a 16.841 Hz frequency.

In figure 18 we illustrated the deformation trend of the excavator boom in direction X for vibration mode 1 (1.9779 Hz).

Figure 19 highlights the deformation tendency of the excavator's boom in the Y direction for the vibration mode 2 (2.1092 Hz).

Figures 20 and 21 illustrate the deformation trend of the excavator boom in the Z direction for modes 6 and 10, with the corresponding frequencies of 9.2999 Hz and 16.841 Hz respectively.

Analysing figure 20 we can conclude that mode 6 shows a bending of the load-bearing beams in the upper and lower chord, and also the flexure of wind bracings in the left and right girder.

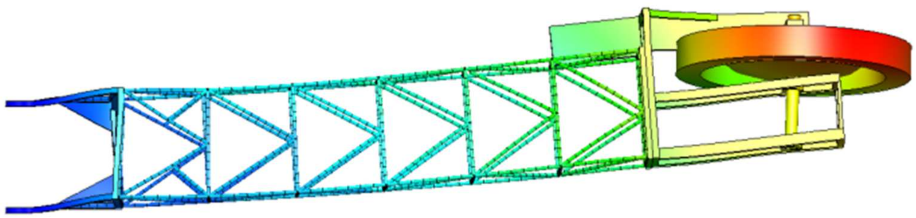


Fig.18 Deformation of the boom on direction X for mode 1
(1.9779 Hz - in the presence of the boom's hoisting cables and considering gravity)

Fig. 19 Deformation of the boom on direction Y for mode 2
(2.1092 Hz - in the presence of the boom's hoisting cables and considering gravity)

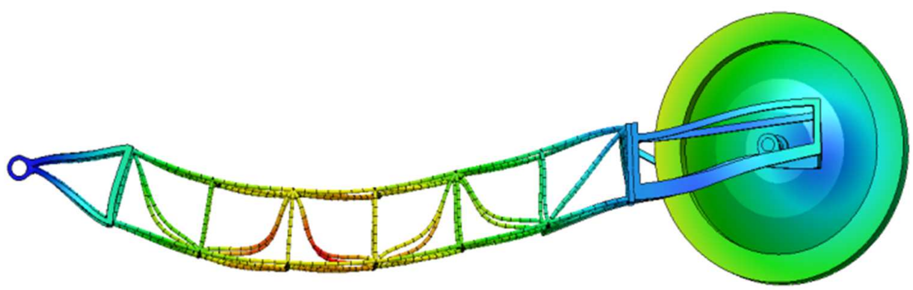


Fig. 20 Deformation of the boom on direction Z for mode 6
(9.2999 Hz - in the presence of the boom's hoisting cables and considering gravity)

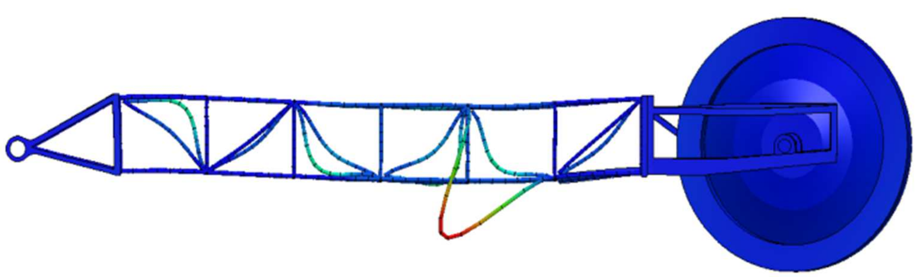


Fig. 21 Deformation of the boom on direction Z for mode 10
(16.84 Hz - in the presence of the boom's hoisting cables and considering gravity)

Figure 21 shows for mode 10 flexure of wind bracings in the left and right girder, which are more accentuated for the left part of the beam structure.

In an analogue mode we performed the modal analysis of the excavator boom structure in the absence of its hoisting cables and without taking gravity into consideration.

5. CONCLUSIONS

The results obtained shows after analysis that the presence of the hoisting cables of the boom, has a significant role in the modal analysis.

They reverse the proportion of the EMPF in the X and Y directions for which an important vibration contribution is expected.

Thus, taking into consideration the BWE boom hoisting cables, the maximum value of the EMPF is determined for mode 1 in direction X, and for direction Y, its maximum value is obtained for mode 2.

The absence from the analysis of the boom hoisting cables, determines a maximum value of the EMPF in the X direction for mode 2 and in the Y direction for mode 1.

The presence of the cables in the modal analysis of the boom structure determines a doubling of the frequency of the first mode.

The obtained results illustrate a diminished influence of gravity in the modal analysis of the structure of the BWE boom.

6. REFERENCES

[1.] Craifaleanu, I.,G. (2008). *Introducerea în calculul structural cu programul SAP2000*, Editura MATRIX ROM, Bucureşti.

[2.] J. Ed Akin, (2009). *Finite Element Analysis Concepts via SolidWorks*, World Scientific.

[3.] Kertesz, I., Popescu, F.D., Radu, S.M. (2017). *The analysis of the deformations and stresses of the cutting-teeth of bucket wheel excavators (BWEs) during operation, using finite elements method (FEM)*, Annals of the University of Petroşani, Mechanical Engineering, pp. 81-90.

[4.] Kertesz (Brînaş) Ildiko (2019). *Contribuţii la studiul echipamentelor miniere de excavare prin modelare şi simulare în vederea îmbunătăţirii performanţelor funcţionale ale acestora*, (Doctoral thesis), University of Petroşani.

[5.] Kovacs, I., Nan, M.S., Andraş, I., Jula, D. (2008). *Study of the working regime of the bucket wheel excavators in the conditions of Romanian open pit lignite mines*, 8th WSEAS International Conference on SIMULATION, MODELLING and OPTIMIZATION (SMO '08), Santander, Cantabria, Spain, September 23-25.

[6.] Kurowski, P.,M. (2016) *Vibration Analysis with SOLIDWORKS® Simulation 2016,* SDC Publications, Mission, KS.

[7.] Ahmad, M.S., Jamil, M., Iqbal, J., Khan, M.N., Malik, M.H., Butt, S.I. (2016). *Modal Analysis of Ship's Mast Structure using Effective Mass Participation Factor*, Indian Journal of Science and Technology, Vol 9(21), DOI: 10.17485/ijst/2016/v9i21/94830

[8.] Nan, M.S. (2007). *Parametrii procesului de excavare la excavatoarele cu rotor*, Editura Universitas, Petroşani, 2007

[9.] Popescu, F.D., Radu, S.M., Kotwica, K., Andraș, A., Kertesz (Brînaș), I. (2019). *Simulation of the Time Response of the ERc 1400-30/7 Bucket Wheel Excavator's Boom during the Excavation Process*, Sustainability, 11(16)

[10.] Popescu, F.D., Radu, S., Andraș, A., & Kertesz Brînaș, I. (2019). *Simulation of the frequency response of the ERC 1400 Bucket Wheel Excavator boom, during the excavation process*, New Trends in Production Engineering, 2(1), 153-167.

[11.] Priestley, M.J.N., Seible, F., Calvi, G.M. (1996). *Seismic design and retrofit of bridges*, A Wiley – Interscience Publication, JOHN WILWY & SONS, INC. New York, pp. 242–243

Analiza modală prin simulare și modelare a brațului excavatorului cu rotor ERc-1400 în timpul procesului de excavare

Rezumat: Excavatoarele cu rotor sunt supuse vibrațiilor generate de tăierea materialului. Cel mai expus subansamblu este brațul excavatorului, care are atât rol de susținere al roții port-cupe cât și de poziționare verticală a acesteia la poziția de tăiere. Frecvenţa vibraţiilor datorate tăierii rocilor depinde de numărul cupelor de pe rotor şi de viteza de rotaţie a acestuia. În lucrare propunem un model al braţului excavatorului pentru care am realizat analiză modală utilizând SolidWorks Simulation. Analiza modală este fundamentală în studiul răspunsului dinamic în frecvență al unei structuri, studiu care va face obiectul unei lucrări viitoare.

Florin Dumitru POPESCU, PhD Eng. Habil., Professor, University of Petroșani, Department of Mechanical, Industrial and Transport Engineering, fpopescu@gmail.com, 0723719303, 20 Universității street, 332006, Petroșani, HD

Sorin Mihai RADU, PhD Eng., Professor, University of Petroșani, Department of Mechanical, Industrial and Transport Engineering, sorin_mihai_radu@yahoo.com, 0722642340, 20 Universității street, 332006, Petroșani, HD

Andrei ANDRAȘ, PhD Eng., Assoc. Prof., University of Petroșani, Department of Mechanical, Industrial and Transport Engineering, andrei.andras@gmail.com, 0722366653, 20 Universității street, 332006, Petroșani, HD

Ildiko BRÎNAȘ, PhD Eng., Assistant, University of Petroșani, Department of Mechanical, Industrial and Transport Engineering, kerteszildiko@ymail.com, 0726703067, 20 Universității street, 332006, Petroșani, HD

Mihaela Olga MARIȚA, PhD student, University of Petroșani, mihaela_marita98@yahoo.com, 20 Universității street, 332006, Petroșani, HD

Narcis Ionel REBEDEA, PhD student, University of Petroșani, 20 Universității street, 332006, Petroșani, HD