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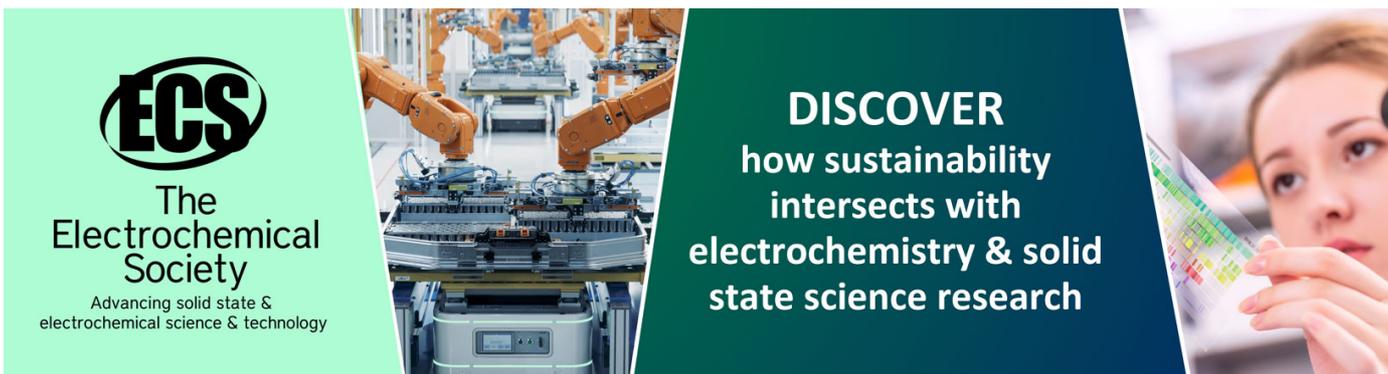
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With frequency simulation to design the monitoring of frame structures

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Abstract: Based on experience from the testing of the mining stacker in the copper mine in Bor, a practical procedure of frequency simulation of the enforced vibration of the structure was demonstrated. Modelling of the excitation was performed by the harmonic function of the eccentric mass of the belt conveyor drive. The paper shows how the drive frequency excitation affects the amplitude of vibration of the supporting structure. From this, the idea is presented - an algorithm for software that would allow for keeping the dynamic properties of the members of the structure within controlled range. An experiment for accelerating the structure and its FFT analysis in the frequency domain was performed. Dynamic modelling of the structure allows obtaining the objectivised concept of the measurement system for monitoring based on the given level of dynamic sensitivity of the parts of the structure.

1. Introduction

External excitation (forced) forces often occur as a product of constant circular motion (frequency) of machine drive devices. If they have a wave (harmonic) character, then they are usually represented by the sinus or cosine function $\sin(\omega t)$, $\cos(\omega t)$, with the period $T = 2\pi / \omega$. Observation of such stationary dynamic processes with frequency excitation has the advantage of being reduced to the observation of only one period of the wave in the continuous duration of excitation period and not for a long period as in the transient processes analysis. Such analyses forced by frequency-excitation are known as linear frequency response analysis or Steady-state response analysis. These analyses have engineering significance because they are studying the frequency domain of machine operations, the appearance of the resonance and the dynamic amplification factors [1, 2]. They are particularly of interest the large transport machines which we can see in mining, energetics and basic industries. Analysis gives an answer on the duration and reliability out of the production plant.

In Figure 1, a miner stacker was shown, where the research was performed numerically and experimentally. This transportation machine disposed to 4800 t/h of the waste-rock and is characterized by continuous operation [3]. Stacker geometry is 55.90x16.88x7.87 (m) and mass 210680 (kg). At the same time, the large structure of the boom relies on a small rotating support of the stand (diameter 1.5 m). Natural question is the dynamic stability of the long boom, oscillation amplitude and dynamic amplification factor in the wider frequency domain. Linear frequency response analysis is performed by the FEM method. The model consists of 2134 finite elements (masses) and



1016 nodes. Such great number of masses makes modelling more realistic and implies advantages of discrete models.

Frequency excitation in these machines makes more process: 1. Fast-eccentric masses on the rotating parts of the conveyor, 2. Fall of load (waste-rock) on the conveyor belt, 3. Weight on the bracket of the belt drive gearbox mounted, 4. Vertical harmonic movement of the material on the belt through equally spaced rollers of the conveyor, 5. The vibrating device excitation for filling (cleaning) the bunkers of the conveyor belt system, 6. Frequency of eccentric mass of propulsion conveyor motor.

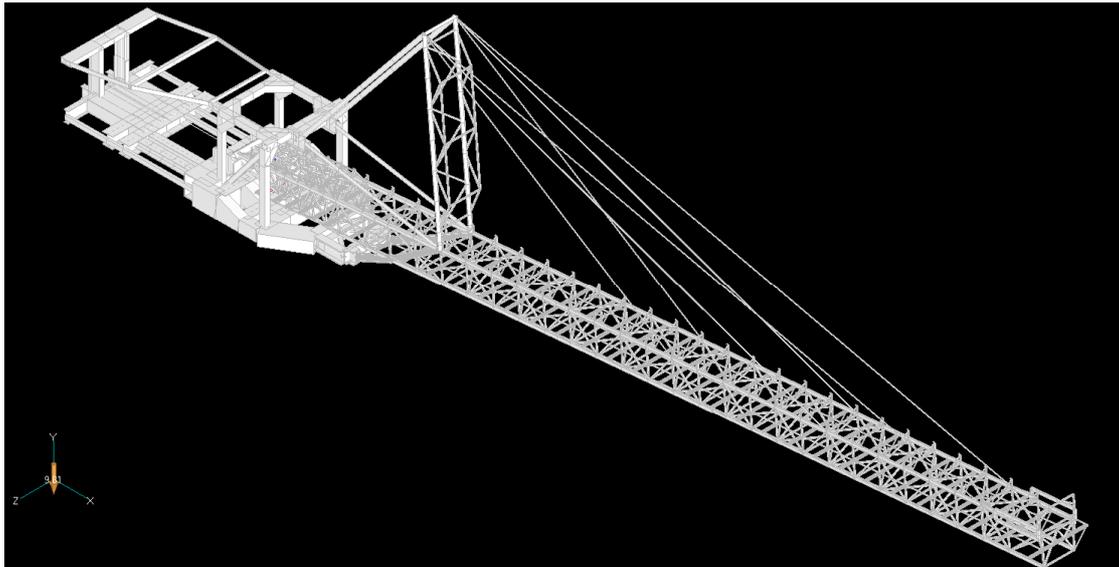


Figure 1. Model of stacker's FEM structure.

2. Theoretical basis

Frequency response analysis is a method to compute structural response to steady-state oscillatory excitation. In frequency response analysis, the excitation is explicitly defined in the frequency domain (forcing frequency). Excitation can be in the form of forces or enforced motions (displacements, velocities, accelerations). The results obtained from a frequency response analysis typically include grid point (node) displacements, grid point accelerations, element forces and stresses. The compute responses are complex form, with magnitude and phase (real and imaginary components). Two different numerical methods can be used in frequency response analysis. First - Direct frequency response analysis solves the coupled equations in terms of forcing frequency. Second method – Modal frequency response analysis uses the normal modes of the structure to uncouple the equations of motion, with the solution for a particular forcing frequency, obtained through the summation of the individual modal responses. Equation of motion of damped forced vibration with harmonic excitation has coordinates $\{x(t)\}$ and form (1), with notation [2]. In this equation, member $[M]$ is the mass matrix, $[B]$ is the damping coefficient matrix, $[K]$ is the global stiffness matrix of the structure and $\{P(\omega)\}$ is an external vector of excitation with frequency ω . Modal frequency response analysis uses the mode shapes of the structure to reduce the size, uncouple the equations of motion (when modal damping is used), and make the numerical solution more efficient. First step in the formulation is transformation of the variables from physical coordinates $\{u(\omega)\}$ to **modal coordinates** $\{\xi(\omega)\}$ using a relationship (2). $\{u(\omega)\}$ is one complex displacement vector with magnitude (u) and phase angle (θ) as real and imaginary components of oscillation.

$$[M] \cdot \{\ddot{x}(t)\} + [B] \cdot \{\dot{x}(t)\} + [K] \cdot \{x(t)\} = \{P(\omega)\} \cdot e^{i\omega t} \quad (1)$$

$$\{x\} = \{u(\omega)\} \cdot e^{i\omega t} = [\Phi] \cdot \{\xi(\omega)\} \cdot e^{i\omega t} \quad (2)$$

Equation (2) represents an equality if all modes are used. The mode shapes $[\Phi]$ are used to transform the problem in terms of the behavior of the modes as opposed to the behavior of the grid points. Number of used modes represents level of model approximation. In case that structural damping is used, the orthogonality property does not diagonalize the generalized stiffness matrix $[\Phi]^T[K][\Phi] \neq \text{diagonal}$ because structural damping matrix has complex form (3). Where $[K]$ is the global stiffness matrix, G is the overall structural damping coefficient, $[K_E]$ is the elements stiffness matrices and G_E is element structural damping coefficient (in the material).

$$[B] = (1 + iG)[K] + i \sum G_E [K_E] \quad (3)$$

In this situation, the modal frequency approach [2, page 131] solves the coupled problem in terms of modal coordinates using the direct frequency approach. When the expression (2) is substituted into equation (1), the following is obtained:

$$[-\omega^2[\Phi]^T[M][\Phi] + i\omega[\Phi]^T[B][\Phi] + [\Phi]^T[K][\Phi]]\{\xi_{(\omega)}\} = [\Phi]^T\{P_{(\omega)}\} \quad (4)$$

If damping is applied to each mode separately, the uncoupled equations of motion can be maintained. When modal damping is used, each mode has damping b_i , where $b_i = 2m_i\omega_i\zeta_i$. The equations of motion remain uncoupled and have the form:

$$-\omega^2 \cdot m_i \cdot \xi_i(\omega) + i \cdot \omega \cdot b_i \cdot \xi_i(\omega) + k_i \cdot \xi_i(\omega) = p_i(\omega) \quad (5)$$

Then, each of the modal responses is computed with relation (6):

$$\xi_i(\omega) = \frac{p_i(\omega)}{-m_i\omega^2 + i \cdot b_i \cdot \omega + k_i} \quad (6)$$

At resonance the three types of damping are possible and they are defined by damping ratio ζ_i and structural damping coefficient G_i , equations (7):

$$\zeta_i = \frac{b_i}{b_{cr}} = \frac{G_i}{2}, \quad b_{cr} = 2m_i\omega_i \quad (7)$$

The uncoupled equation of motion, for final solutions, are:

$$-\omega^2 \cdot m_i \cdot \xi_i(\omega) + (1 + i \cdot G(\omega)) \cdot k_i \cdot \xi_i(\omega) = p_i(\omega) \quad (8)$$

3. Numerical simulation

Dynamic modelling of Stacker is performed by forming discrete system of mass elements of the caring structure and installed machine equipment. Masses are mutually connected by elastic connections of the structure. For more practical research, modelling can be performed by FEM model. For modelling, this paper uses real structure of the Stacker RBB [3]. Analysis is performed by FEM method of finite elements [2] by using FEMAP, PLM Siemens Software [4].

Rotatory platform and boom are in the form of a frame, so they are modelled by finite elements in the form of a beam. Machine equipment, belts, drums and transported material on conveyor belt are modelled by dotted-mass finite elements. The base construction of Stacker (without caterpillars) is rigidly laid on the ground.

Dynamic analysis requires knowledge of the system damping. For this reason, a modal analysis is carried out to determining whether the individual parts of the structure oscillate, by what frequencies, and with what kind (type) of damping. The damping can be viscous (proportional to the speed of oscillation), and structurally, which is proportional to the position (the size of displacement). Therefore, modal analysis are performed to determine the speeds and processes of damping.

Figure 2 shows a modal shape of the free oscillation of the rotary part of the stacker. That is 87 mod (15.9539 Hz), which is characterized by the shape - vertical undulation of the main carrier [5-7]. After the modal analysis, two frequency-based excitations were selected. They were described and introduced in the **modal-frequency analysis**. The analysis sought significant oscillation of structure in order to prevent extreme dynamic phenomena and improve vibro-comfort.

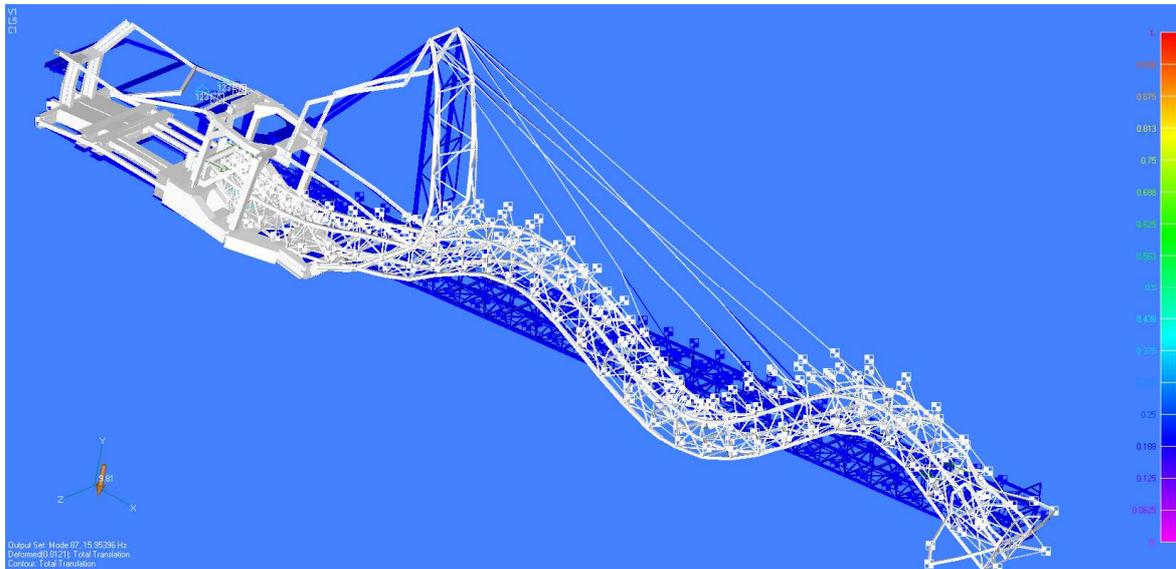


Figure 2. The first step is a modal analysis that determines the own values of the structure; on the image is Mod 87, the frequency is 15.9539 Hz (double vertical sine wave of the boom).

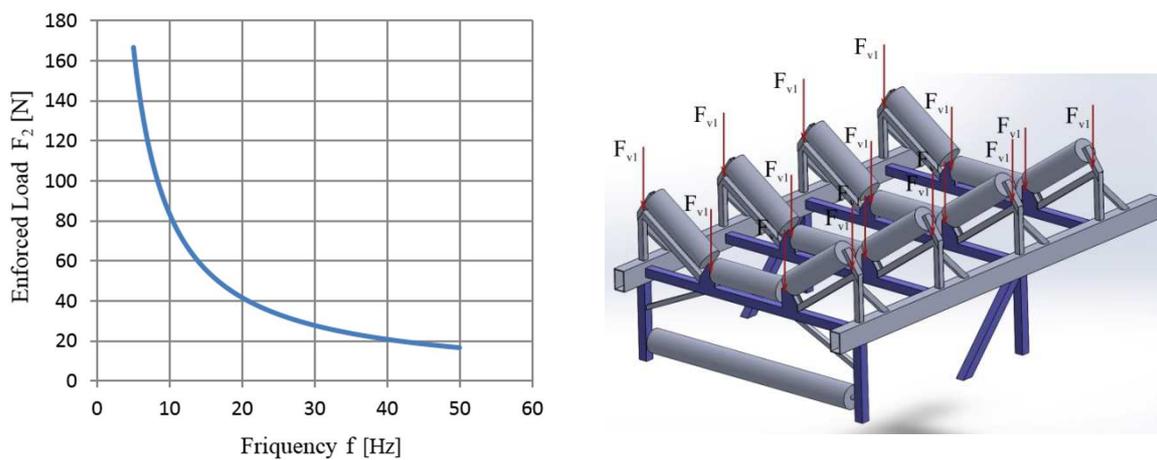


Figure 3. Dynamic forces $F_2=F_{v1}$ due to the entry of material on the conveyor (flow/arrival of material with 5-50 compact pieces in every second).

Dynamic forced action is determined from the conveying flow $Q=4800$ [t/h]=1333 [kg/s]. Frequency of appearance of compact pieces is determined by average material granulation $f=5-50$ [Hz]. The impulse force of the material pieces weight according to Figure 3 acting on $n = 16$ receiving rollers of the belt. The force on a roller of an entry of the conveyor is done by equation (9):

$$F_{2(f)} = \frac{Q \cdot g}{n} \cdot \frac{1}{f} = \frac{1333 \cdot 9.81}{16} \cdot \frac{1}{f} = \frac{817.5}{f} \text{ [N]} \tag{9}$$

The centrifugal force of 1.0 kg of material is stacked eccentrically on the drum with radius $R = 0.5$ m at the frequency of rotation f [Hz], eq. (10), Figure 4. The stationary frequency of the drum conveyor is 1.666 [Hz] (100 rpm).

$$F_{1(f)} = m \cdot R \cdot \omega_D^2 = m \cdot R \cdot (2\pi f)^2 = 1.0 \cdot 0.5 \cdot (2\pi \cdot f)^2 = 19.72 \cdot f^2 \text{ [N]} \tag{10}$$

By introducing these two forces $F_1(f)$ and $F_2(f)$ defined in the frequency domain in the dynamic equation of the structure (1) and solving it with a front exposed modal-frequency procedure, we obtain

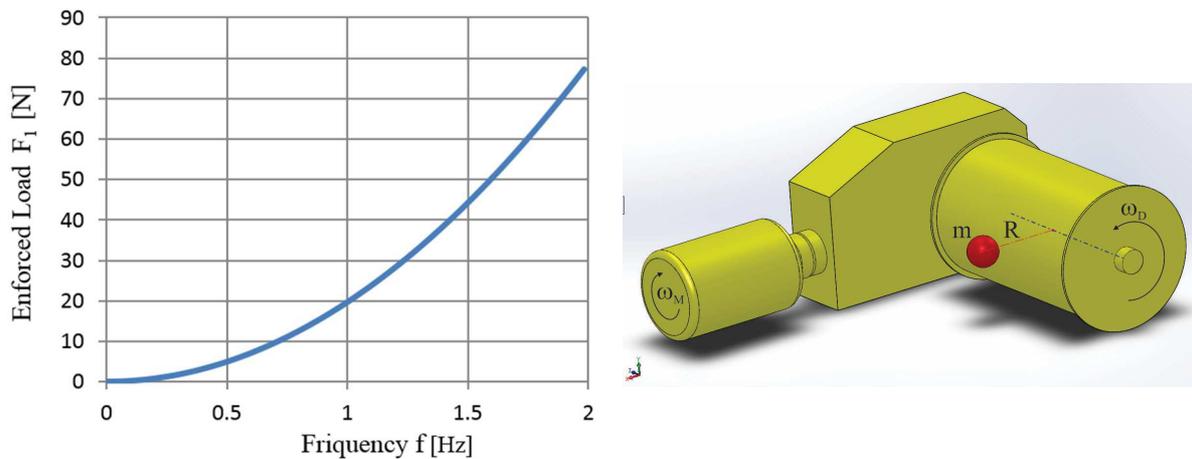


Figure 4. Frequency excitation from rotation of eccentric mass on drum (1.666 Hz); diagram of frequency excitation force and mechanical model of conveyor drive mechanism.

a dynamic response of any point of structure grid. Figure 5 shows the oscillation of the tip of the stacker boom at the point where it is the longest tie rod connected with the horizontal boom structure (Figure 5). The picture shows three curves of acceleration components in the frequency domain of 50 Hz. The smallest oscillation of this point is seen at a frequency of 15 Hz. Oscillations below 10 Hz are due to the flow of material through the conveyor loading bunker.

Oscillations over 20 Hz are conditioned by an eccentric weight of rotor but with a moderate dynamic amplification factor. In the analysis it is possible to introduce the excitation of the electric motor drives. This effect can be seen in Figure 6-b as a single frequency of 25 Hz (corresponding to the speed of the motor).

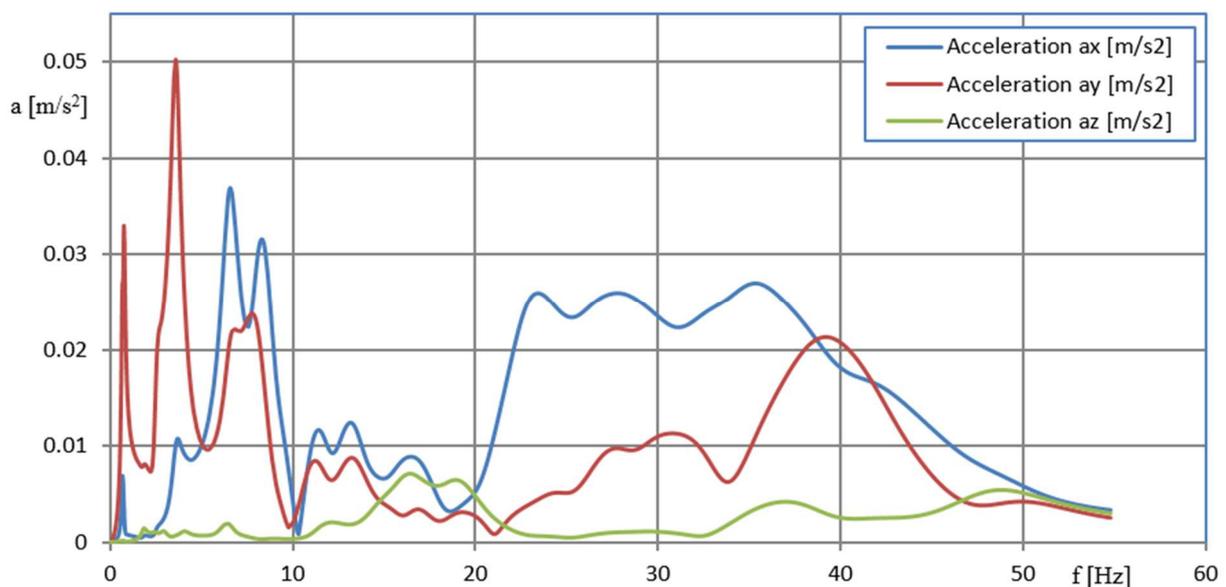


Figure 5. Numerical solution of frequency response analysis (collecting fact); calculated acceleration of the structure a_x, a_y, a_z (m/s^2). Structure boom position: On the end of longest rod (number 3); node 692.

4. Experiment on real structure

Reality of investigated eigenvalue was checked experimentally by measuring the acceleration of the characteristic points of structure [5]. The three component acceleration with excitation caused by more

positions on conveyor were measured in positions of the pylon, ROD-1, ROD-2, ROD-3 and the driving conveyor drum, Figure 6. A three-component piezo sensor ADXL-312 brand MEMS with integrated amplifier Analog Device AD-320 and SD memory of 4 GB were used as the sensor. Record of measurements were analyzed by VIBRAREC software [8]. The acceleration and speed in the frequency-domain were identified using FFT transformations on the basis of measured accelerations in time-domain. They are clearly defined all the present dominant frequencies - caused by them themselves. Figure 6-a shows an experimental record of acceleration in time-domain. These are the three acceleration component directions of the boom, in position near the ROD-3 (the longest supporting lamella-rod). Red curve (\mathbf{a}_y) is the acceleration in the vertical direction, the blue curve (\mathbf{a}_x) is the acceleration in the lateral direction, black curve (\mathbf{a}_z) are boom accelerations in the longitudinal direction. Component accelerations are up to -0.23 m/s^2 . Figure 6-b shows the same acceleration in the frequency domain up to 50 Hz. In the domain of 0-20 Hz dominated the accumulation of natural frequencies. The drive motor excitation appears at 25 Hz (Figure 6-b, red curve).

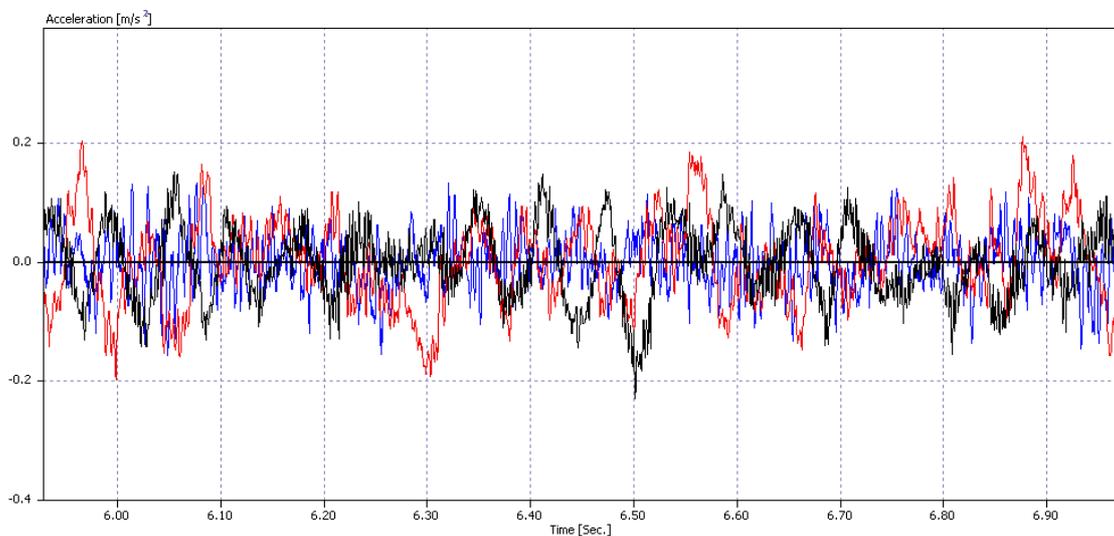


Figure 6a. Experiment [5]: Acceleration in the time domain at the point of boom and connection of the third tie rod. Measuring location according to the mark in Figure 7.

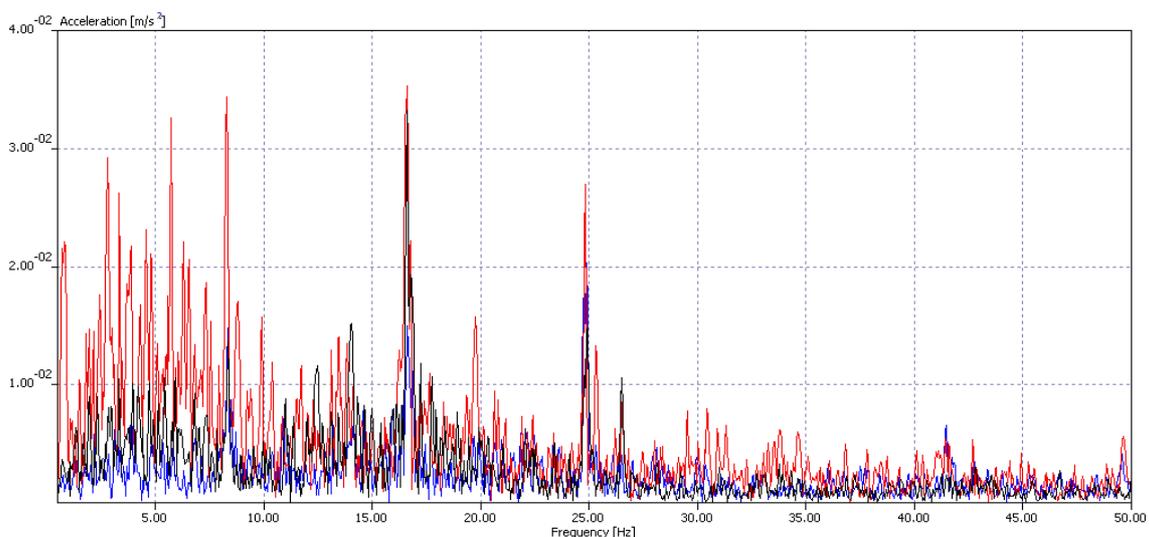


Figure 6b. Experiment [5]: Acceleration in the frequency domain at the point of boom and the third tie rod. Measurement location according to the mark in Figure 7.

5. Algorithm characteristics monitoring

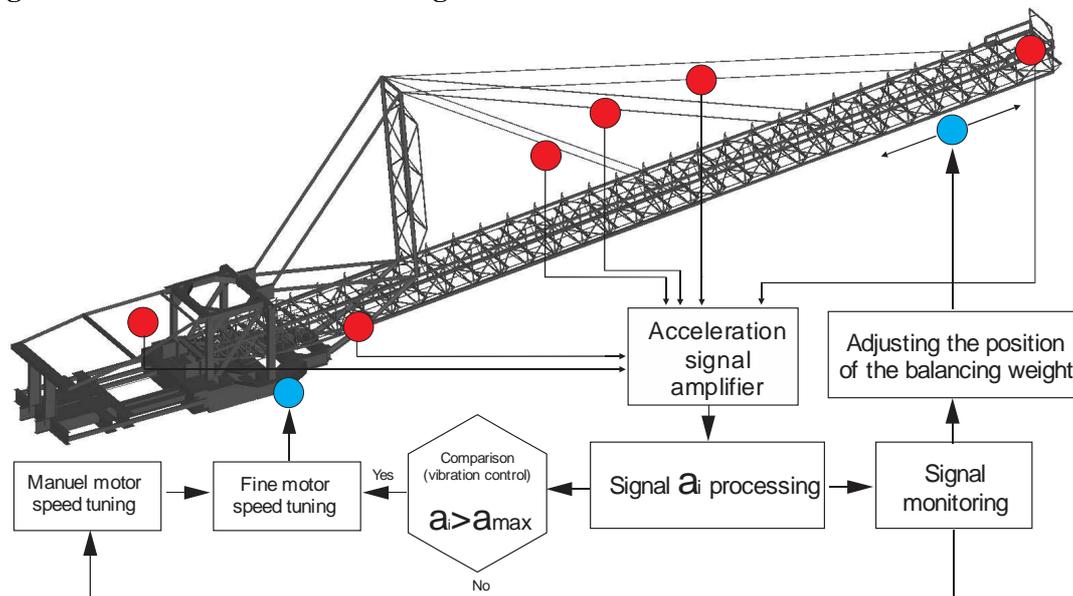


Figure 7. Monitoring algorithm and active vibration control structure.

The monitoring system starts from a number of previously made modal-frequency analyses which determine the level of amplitude of a large number of points of the structure and the resulting stresses by these displacements. The group of obtained points of a structure with significant results is firstly evaluated by the level of responsibility and then the reduction of number of points is performed. Figure 7 shows a scheme of location of the accelerometer sensor and a stress sensor. At the same time the scheme shows the necessary equipment and functions that must be performed by encoders and controllers. Since the stacker is working in different working regimes, it is necessary to introduce in the control of the structure - the active oscillation setting in the domain of small amplitudes. This can be done by installing the movable mass one on the lower side of boom structure (Figure 7, blue mass). By changing the position of weight the frequency-amplitude characteristics of the structure are changing. The outer natural effects have accidental nature (wind), the present control system can fine-tune the weight distribution that minimizes the amplitude. The adjustment should be manual and then autonomous. The development of an autonomous balancing system requires the introduction of a critical acceleration value when the setting starts (a_{max}) and the maximum acceleration when the operation stops. The measuring system requires amplifiers and equipment for monitoring and control, which is a typical control technique. Introducing a measuring system into a structure significantly increases the level of plant safety which ensures the planned and long exploitation of large machines. The first step towards the regulation system is to set up monitoring and introduce the stacker into the phase of observing and registering phenomena. Modal-frequency analyses have the greatest significance for heavy machines that operate in shock regimes. These are bucket-wheel excavator and crushers.

6. Conclusions

1. The stacker is a large structure with model of thousands elements and several load sources inside the machine. For these conditions the modal method is recommended by the software manufacturer.
2. By comparing numerical and experimentally determined acceleration in the frequency domain (Figure 5 and 6-b), frequent closeness of dynamic behavior can be seen in terms of individual dynamic gain factors. This conclusion can be reached by examining (calculating) certain frequencies influence (sources) separately.

3. The defined model enables the simultaneous introduction of a number of frequency influence (frequency environments) in analysis, which contributes to the approximation of amplitude-frequency characteristics.
4. Modal-frequency FEM analysis can also treat large supporting structures from the numerical aspect. The reason for efficiency is in non-coupled equations of oscillation in different modes, which has a simple algorithm for solving and a shorter period for obtaining a solution.
5. Modal-frequency FEM analysis allows to obtaining basic dynamics data of the supporting structures. These are the data in the amplitude-frequency domain of the structure. These data are discrete and show the positions of construction points that have dynamic extremes. These points are crucial for the deployment of control and monitoring equipment.
6. The observed stacker has a low vibration level since it works with a loose, crushed material. However, these constructions also require amplitude checking due to the sensitivity of the bearing rotary part and swinging boom.

Acknowledgment

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