586. Relation between numerical model and vibration: behavior diagnosis for bucket wheel drive assembly at the bucket wheel excavator

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Abstract: A drive assembly of bucket wheel at the bucket-wheel excavator (BWE), as a rule, is a unit characterized by large force having huge torque and mass with structures in different arrangement including the support structures as well. Behavior prediction is of crucial importance in order to apply the right approach of operation and maintenance on gearbox, electrical engine as well for structural elements supporting this assembly. It is obvious that vibrations measured at distinctive points of the entire drive assembly and support structure represent the main prediction parameter. The measurement results are used to confirm mathematical model developed by means of finite element method, which is based on theory on flexibility, static and dynamical review for obtained results.

Keywords: prediction, drive assembly, numerical model, vibrations

1. Introduction

Vibrodiagnostics provides picture of current state of a technical system and real tendencies of anomalies in time as well as in frequency domain. So far a number of research papers have been published in the field of vibrations. The reason is very simple – vibrodiagnostics has solved numerous problems in industry. Moreover, the basis for predicative and proactive maintenance [1] for the large part consists of vibrodiagnostics. This paper presents mutual relation between numerical modeling and measured vibrations. In other words, measured vibrations are used to validate model correctness for the case of drive assembly of bucket wheel at the bucket wheel excavator.

Many researchers deal with modeling and vibrodiagnostics of technical systems and their elements.

Mass unbalance of any system has extremely negative effect. Incident of mass unbalance due to eccentricity of elements in rotor-bearing system generates contact between them [2]. This kind of identification contributes to understanding of nonlinear dynamic contact at a rotating system resulting from the same or uneven mass unbalance. In this way all drive assemblies with large dimensions can be surveyed, as well as in drive assembly of bucket wheel at the bucket-wheel excavator (BWE).

Large number of devices is set up on a foundation that moves independently. Making of shaft model is defined on the basis of two shaft operating regimes: slow rotation and periodical motion (swinging) in the lower part of bearing reliance [3]. Both problems can be solved by applying the ways of motion: division on slow and fast motion and using the perturbation method. Such device is also the drive unit of the BWE bucket wheel.

Modeling can be applied with goal of simulating the dynamic performance of rotating machines. One of the approaches is to approximately input into the model the measured vibrations of rotating element, despite all the difficulties and conditions [4]. This modeling method leads to problem of identifying natural frequencies and damping ratio. This method is largely acceptable for analyzing drive unit of BWE bucket wheel, largely because vibrations are used to prove the accuracy of numerical model.

Simulation model of bucket wheel drive unit must meet the following requirements: model must reflect real system proportions as accurately as possible; correlation between the actual system and the idealization achieved in the model should be visible in all places; basic parameters of model system should be determined according to technical data or by using the real system with sufficient accuracy [5]. Another important aspect in model building is the choice of system limit. Outer system parameters are marked as dimensions defined in advance, without the influence of the system. Firstly, analysis of natural frequencies is performed using the linear model. Discretization of drive assembly into stiff mass and stiffness without mass allows for the exact arrangement of frequencies according to the appropriate drive assembly components. This enables feedback conclusions about impulses of unwanted frequencies. This model reveals the complexity of bucket wheel drive unit of the bucket wheel excavator.

Dynamic performance of rotating elements of the rolling bearings in the presence of localized defect surfaces can be presented as a system with three degrees of freedom [6]. Generating vibrations of defects at bearings is a function of bearing rotation, load distribution in bearing, elasticity of main structure, characteristics of lubricant zone and route between bearings and measuring sound. Based on this, new software has been designed [6], in order to simulate vibrations from localized damages.

Defining resonance as a phenomenon at bearings of gyroscope rotor determines the dynamics of gyroscope system [7]. Due to rotation, radial cavity in this system can induce resonance. In this case, the resonance is experimentally identified. This kind of approach can also be applied on mechanical drives with certain characteristics.

All aforementioned citations can achieve implementation of drive unit of bucket wheel, but each in its own manner. What distinguishes this paper is the approach to a certain problem and the peculiarity of the system itself. This is particularly emphasized from the aspect of vibrations source. Dynamic system of drive unit of bucket wheel is a high-quality example of the relations between vibrations, elasticity and stiffness level.

The most loaded parts of the bucket wheel excavator with respect to dynamic tensioning are bucket wheel boom and drive assembly of the bucket wheel with bucket wheel. Each rotating, that is slewing part, as well as each part, which moves at the bucket wheel, is a source of low frequency vibrations. When the excavator is in operation (i.e. when it is digging by his natural low frequency vibrations, which are of extremely complex character due to lot of different sources), vibrations caused by outside dynamic load have to be added. The latter vibrations have a large number of sources as well. It is mentioned due to very simple reason. Because it is well known that the most favorable condition for one structure is resonant, that is when there occurs overlapping of natural low frequency vibrations and vibrations caused by external dynamic loads. Having in mind that bucket wheel drive is at the end of the boom (console), such dynamic system has even bigger complexity, since drive in addition to natural vibrations receives boom vibrations, too, that is other boom elements depending on it rigidity and support, geometry, respectively.

Prediction of behavior of drive assembly at the mining machines, and particularly at bucket wheel excavators, has gained greater importance during the nineties of the last century. Two reasons may be identified. The first one is related to small number of manufactured bucket wheel excavators at that time, while other reason is that bucket wheel excavators that are in operation were in operation for several decades. Both manufacturers and users of these excavators approached repair procedures due to extension of lifetime, but also to maintain or improve existing performance levels. As regards, the second reason, it can be mentioned that bucket wheel drives at the majority of excavators are of old structure arrangement with prolonged service-life. Numerical models and experiments based on vibration measurements were used for both reasons. However, there is very small number of scientific publications where numerical FE-based models were verified against measured vibrations at the drive assemblies of mining machines, particularly at BWEs.

Prediction of drive assembly behavior at the bucket wheel is of crucial importance to adopt right approach on operation, maintenance or overhaul and modernization, for the gearbox as well as for other elements of torque transmission. This also has to be complemented with analysis of assembly supporting structure. Methodology, i.e. conceptual approach to resolution of this issue has been strictly specified: analytical segment that provides introduction to the problem, modeling serving as a guideline for engineering approach and experimental part, which has to confirm methodology correctness and accuracy with respect to capability to determine reason of anomaly and its prevention. At the example of bucket wheel excavator SRs470.17/1.5 (400 kW), that is it bucket wheel drive – gearbox, has to be provided exact review of diagnostic behavior and condition with the aim for making a final attitude on further operation of this gearbox. Manufacturer of this bucket wheel excavator is company TAKRAF, and excavator is in operation at the open pit lignite mine Kostolac. Fig. 1 illustrates bucket wheel drive assembly at this excavator.

Fig. 1. Bucket wheel drive at the excavator SRs470.17/1.5 (400 kW)

Fig. 2 provides principal model (scheme) for this drive assembly type, which indicates long distance between the drive assembly supports, which can be represented due to difference of rigidity level, as gerber with joint in the middle [9].

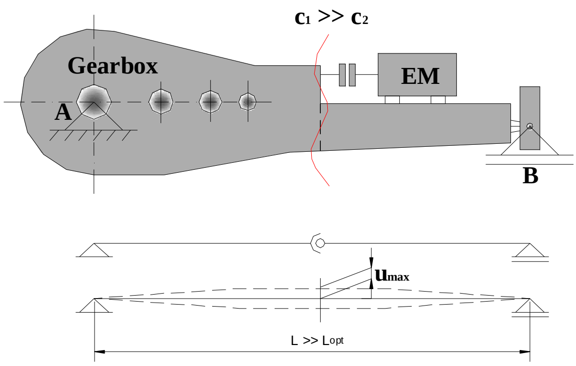


Fig. 2. Principal model of the bucket wheel drive assembly (level of stiffness ratio, $c_1 >> c_2$)

Oscillation of such drive assembly solution way is mainly similar if it is not the same as it is frequency excitation, but it next harmonic, having negative influence to operation of the drive assembly. Such approach to this issue has lead to analytical approach in details, later on defined as numerical-experimental calculation.

2. Experimental details

2.1. Numerical calculation for establishing bucket wheel drive model at the excavator SRs470.17/1.5

Due to large number of vibration sources, natural frequencies and characteristic frequency excitation setting aside is of great importance at elements, subassemblies and assemblies for any drive group at the bucket wheel excavator and particularly to bucket wheel drive. The main oscillation form is associated with natural oscillation frequencies and the corresponding mode shapes. It has to be particularly emphasized excitation and natural oscillating frequencies appeared as frequency calculations in analytical part.

Calculation of support elements structures for respective construction is part of a package connected to a processor. Used program package is KOMIPS (Computer Modeling and Structure Calculations) developed by prof. Maneski at the Faculty of Mechanical Engineering in Belgrade [8]. Developed calculation programs provide static, dynamic and thermal calculation of line, surface and volume issues. Communication between programs has been provided via database sets (files).

Dynamic calculation comprises defining of free oscillations, forced oscillations, and distribution of kinetic and potential energy. Within dynamic calculation, all values are still within the time function. Since, the static calculation is a special case of dynamic (time t = 0), global rigidness matrix remains the same, is formed at the same way, respectively. On the final element during dynamic analysis beside static influence have dynamic forces (inertial and dissipative forces), too.

Moving f and speed $\dot{f}$ for any element point in coordinate system *x, y* and *z*, in time *t*, is:

$$\{f(x,y,z,t)\} = \begin{Bmatrix} u(x,y,z,t) \\ v(x,y,z,t) \\ w(x,y,z,t) \end{Bmatrix} = [N(x,y,z)]\{\delta(t)\}_e$$
$$\{\dot{f}\} = [N]\{\dot{\delta}\}_e \tag{1}$$

Where, *N(x, y, and z)* is a form of final element function, $\{\delta(t)\}_e$ is a global motion vector within a global system as a function of time, and $\{\dot{\delta}\}_e$ is a global speed vector within a global system.

Kinetic energy E_k, dispersion function R and potential energy E_p for the entire model are:

$$E_k = \sum_{e=1}^{m} e_k = \frac{1}{2}\{\dot{\delta}(t)\}^T [M]\{\dot{\delta}(t)\}$$
$$R = \sum_{e=1}^{m} r_e = \frac{1}{2}\{\dot{\delta}(t)\}^T [B]\{\dot{\delta}(t)\} \tag{2}$$
$$E_p = \sum_{e=1}^{m} e_p = \frac{1}{2}\{\delta(t)\}^T [K]\{\delta(t)\} - \{\delta(t)\}^T \{F(t)\}$$

Designations e_k, r_e and e_p refer to kinetic energy, dispersion function and potential elements energy comprising model. Other marks are T matrix for local to global system transformation, *[M]* global masses matrix, *[B]* global absorption, *[K]* global rigidity matrix, $\{F(t)\}$ global forced forces vector as a function of time.

Dynamic equation for structure movement can be performed by application of Lagrange equation or by using Hamilton principle. Lagrange dynamic equation is:

$$\frac{d}{dt}\left\{\frac{\partial L}{\partial \dot{\delta}}\right\} - \left\{\frac{\partial L}{\partial \delta}\right\} + \left\{\frac{\partial R}{\partial \dot{\delta}}\right\} = \{0\} \tag{3}$$

Where derivation between kinetic and potential energy is,

$$L = E_k - E_p \tag{4}$$

Differentiation of derived values leads to dynamic equation for compulsively absorbed oscillations in matrix form and global system:

$$[M]\{\ddot{\delta}(t)\} + [B]\{\dot{\delta}(t)\} + [K]\{\delta(t)\} = \{F(t)\} \tag{5}$$

Where: $\{\ddot{\delta}(t)\}$ is global acceleration vector within a global system in time domain.

Dynamic model for support structure is reduced to final free level number. Free, undamped oscillations within matrix form are:

$$[M]\{\ddot{\delta}(t)\} + [K]\{\delta(t)\} = \{0\} \tag{6}$$

Natural values of the system dynamic matrix represent natural system (model) frequencies. Defining of free frequencies for all system free levels have no technical sense (the first ten are largely important). Computer programs have possibility to determine a small number of natural frequencies. As natural frequencies follow setting of main oscillation shapes that have to be described by the main normal *q* (standardized per masses) coordinates.

Main modes of oscillation have deformation features under ’’imagined’’ loading. The first oscillation mode, then in order, testifies the worst structural behavior. Structure has good dynamic oscillation if:

- The first frequency is high, and
- The difference between frequencies is large.

This is possible to achieve if the structure is arranged with a maximal rigidity and minimal mass. Natural frequency is proportional to $\sqrt{k/m}$, where *k* is element rigidity and *m* is element mass.

Solution for compulsive absorbed oscillation within a frequency range is represented by support structure frequency characteristic. Firstly, they require specification of natural frequencies expressed at the main normal coordinates and with the main oscillation modes. Thereby coupled differential equations are reduced to uncoupled ones. A differential equation written at the main normal coordinates (q) is:

$$[I]\{\ddot{q}\} + [B]\{\dot{q}\} + \lfloor \Omega^2 \rfloor\{q\} = [\mu]^T\{F(t)\} \tag{7}$$

For oscillation form r differential equation within modal form and with excitation at the one-knot and direction (one free level) is:

$$\ddot{q}_r + 2\xi_r\omega_r\dot{q}_r + \omega_r^2 q_r = \mu_{pr}F_p(t) \tag{8}$$

Frequency equation (dispersion function, dynamic indulgency) by application of Laplace transformation is:

$$W(i\omega)_p^r = \frac{q_r(i\omega)}{F_p(i\omega)} = \mu_{pr}\frac{1}{\omega_r^2 - \omega^2 + 2i\xi_r\omega_r\omega} \tag{9}$$

Since the modal matrix is standard, then the relation $\{\delta(i\omega)\} = \{\mu\}\{q(i\omega)\}$ is valid. Frequency characteristic can be now written as relation between Laplace transformation of s rectangle coordinate $\delta_s(i\omega)$ and Laplace p transformation of excitation $F_p(i\omega)$ as:

$$[W(i\omega)]_p^s = \frac{q_s(i\omega)}{F_p(i\omega)} = \sum_{r=1}^{n} \frac{\mu_{sr}\mu_{pr}}{\omega_r^2 - \omega^2 + 2i\xi_r\omega_r\omega} \tag{10}$$

From the previous equation real and imaginary parts can be obtained:

$$[U(\omega)]_p^s = R_e[W(i\omega)]_p^s = \sum_{r=1}^{n} \frac{\mu_{sr}\mu_{pr}\left(\omega_r^2 - \omega^2\right)}{\left(\omega_r^2 - \omega^2\right)^2 + \left(2\xi_r\omega_r\omega\right)^2} \tag{11}$$

$$[V(\omega)]_p^s = I_m[W(i\omega)]_p^s = \sum_{r=1}^{n} \frac{\mu_{sr}\mu_{pr}\left(-2\xi_r\omega_r\omega\right)}{\left(\omega_r^2 - \omega^2\right)^2 + \left(2\xi_r\omega_r\omega\right)^2} \tag{12}$$

Amplitude-frequency and phase-frequency characteristic is as follows:

$$[A(\omega)]_p^s = \sqrt{\left([U(\omega)]_p^s\right)^2 + \left([V(\omega)]_p^s\right)^2} \tag{13}$$

$$[\varphi(\omega)]_p^s = arctg\frac{[V(\omega)]_p^s}{[U(\omega)]_p^s} \tag{14}$$

2.2. Bucket wheel gearbox behavior and condition diagnosis

Program for prediction of gearbox behavior and conditions is outlined in such a way so as to provide relevant and precise approach to actual problem regarding bucket wheel gearbox drive. As concept, this program is possible to be applied at the other power gears and other torque transmitting assemblies. A sequence of preconditions has to be fulfilled for application of his program. In this case, by using repair activities at the excavator, disassembly of the gearbox has been performed and it has been dispatched to special maintenance workshop (testing desk).

According to accurate sequence of activities the causes of malfunctions were identified and verified [9].

2.3. Finite element analysis if main oscillation shapes of support structure for gearboxes, shafts and gears

Application of the program package KOMIPS enabled determination of natural vibration frequencies and their corresponding modes shapes, i.e. main vibration modes were established for the gearbox, shaft and gears support structure.

Figs. 3-7 indicates gearbox model with all main four vibration modes shapes.

The obtained mode shapes enabled identification of support structure to be of highly unfavorable for gearbox dynamic behavior. The reason for this effect is attributed to small gearbox rigidity and unfavorable geometry. For the classically arranged gearbox drive at the bucket wheel (concept solution is from the sixties of the last century), as it is here a case, after a lifelong mining in a very difficult operation conditions, deterioration of proper functionality is unavoidable and has manifested in the form of deficient dynamic behavior.

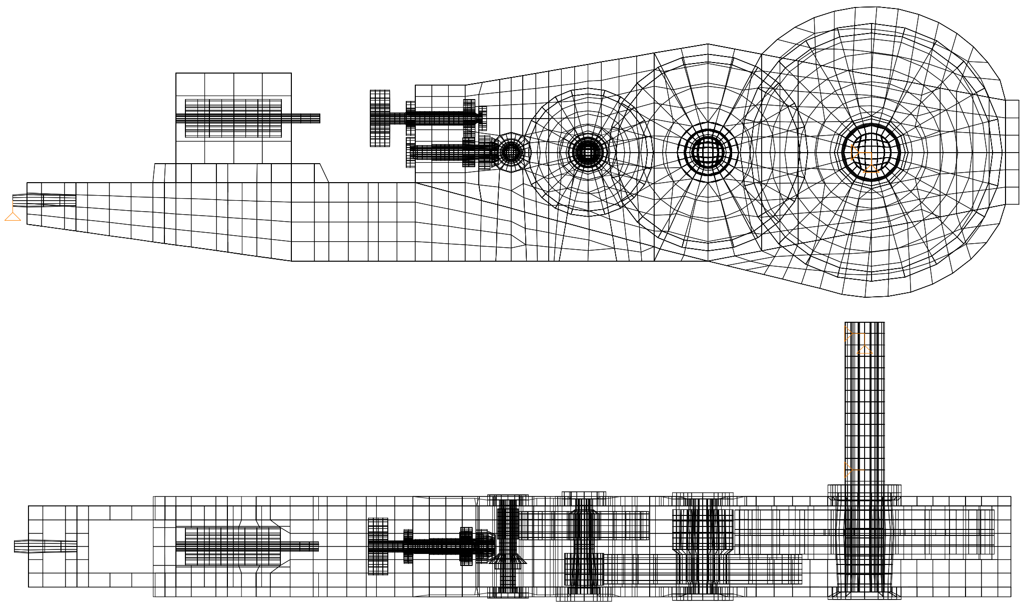


Fig. 3. Gearbox model and bucket wheel drive support structure at the excavator SRs470.17/1.5(400kW)

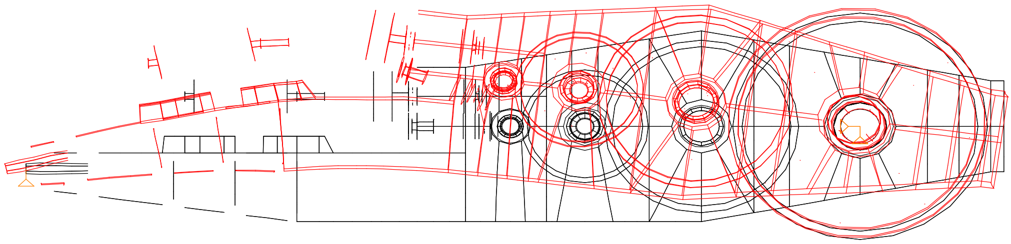


Fig. 4. The first vibration mode shape of support structure - f_{01}= 15,3 Hz

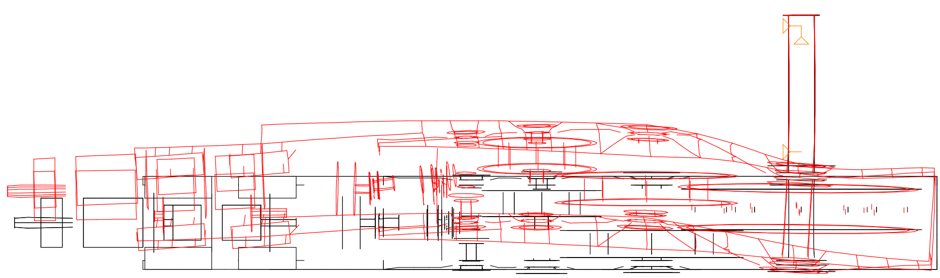


Fig. 5. The second vibration mode shape of support structure - f_{02}= 18,7 Hz

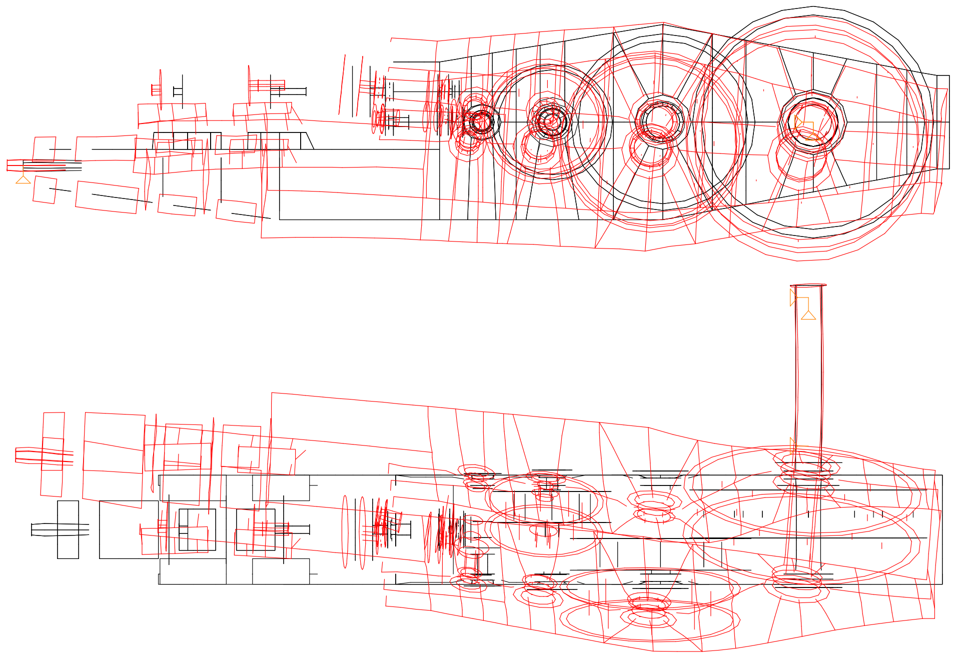


Fig. 6. The third vibration mode shape of support structure - f_{03}= 28,6 Hz

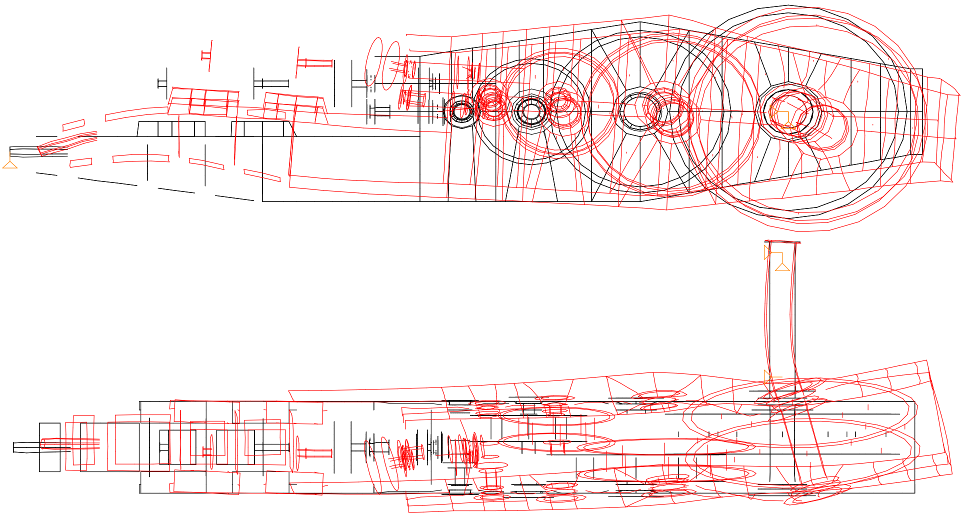


Fig. 7. The fourth vibration mode shape of support structure - f_{04}= 34 Hz

2.4. Excitation and natural frequencies of bucket wheel gearbox

According to generally known transformations and constructive element parameters, characteristic frequencies were determined:

- Gearbox shaft excitation frequencies,
- Tooting excitation frequencies,
- Natural bearings frequencies,
- Electric engine excitation frequencies, and
- Natural frequencies of gearbox and shaft with gears.

Based on performed dynamic computations, the following comments may be provided with regard to obtained forced excitation and natural frequencies:

- The first gearbox natural frequency is under and near the first three shafts' frequencies. During each drive strating entries are through this frequency. This frequency is very unfavorabl due to bending of drive system around it transversal axle as well as fracture of engine shaft and gearbox at the vertical plane.
- The fourth natural gearbox frequency (bending with twisting) is two times greater than the frequency of the first three shafts and is very unfavorable for exploitation of the drive system.
- Driven gears at the fourth, fifth and particularly sixth shaft has very unfavorable oscillation with frequency that are a whole number products with frequency of the first three shafts.

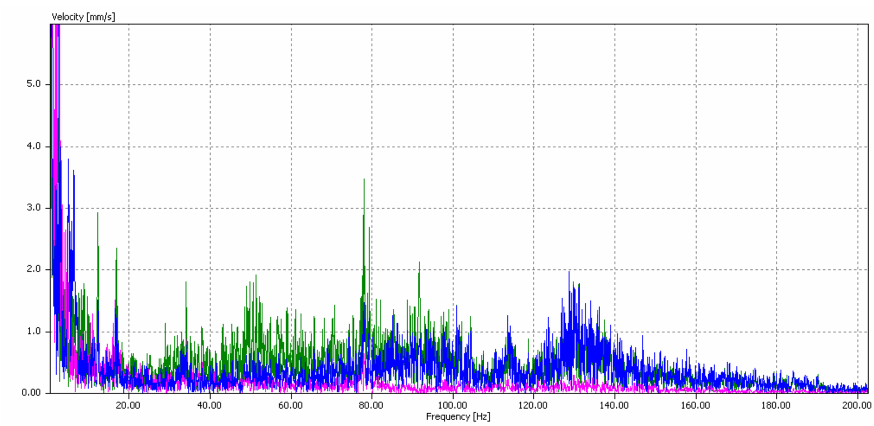
2.5. Vibration measurement of the gearbox in operation (during digging process) before repair

The equipment that includes modern transducer (sensor) of the three-component acceleration transducer, analog-digital (AD) signal converter and USB communication to a computer, was used to perform vibration measurement of the drive gearbox at the bucket wheel. Software package supports analyses of time and frequency acceleration signal. Frequency signal has been obtained by application of FFT analysis.

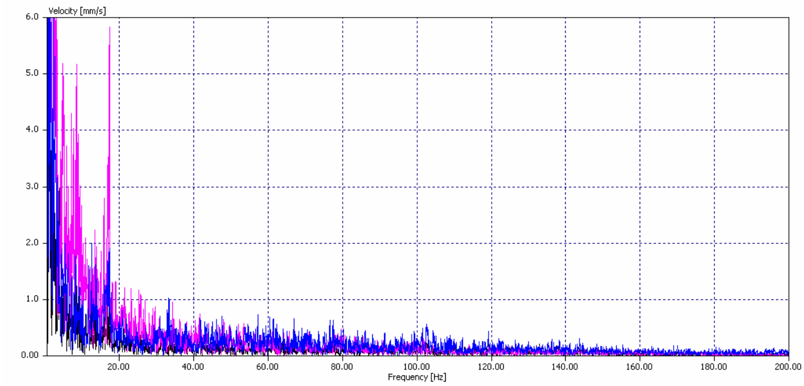
Measured vibrations confirm the findings of the numerical model. Gearbox-measured vibrations at the excavator in operation have revealed dominant frequencies. Its origins are:

1. Frequency 12.4 Hz is present via entire gearbox (later it is shown that it originates in the last gear pair).
2. Frequency 16.5 Hz originates in the first and second shaft.
3. Frequency 17.3 Hz is mostly expressed at the second, third and fourth shaft at the right shaft side (toward the boom). Directly after this measurement has occurred smaller damage at that point (the third shaft at the right side, right bearing has been damaged).
4. Frequency 79 Hz present at the first shaft originates from the electrical engine bearing.
5. Frequency 103 Hz originates from the bearing in the gearbox.

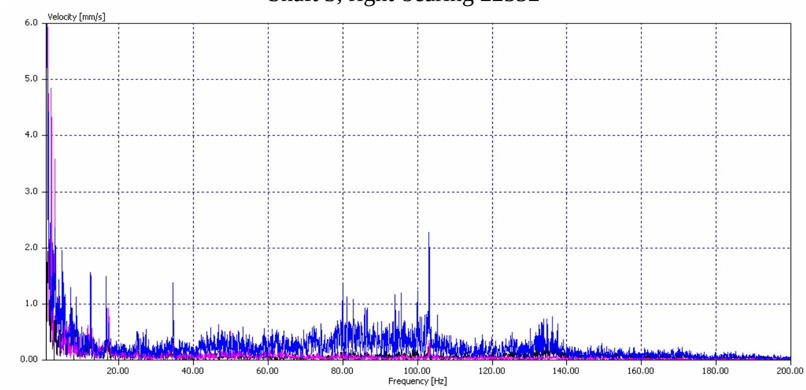
Here we present only four characteristic diagrams (Fig. 8).



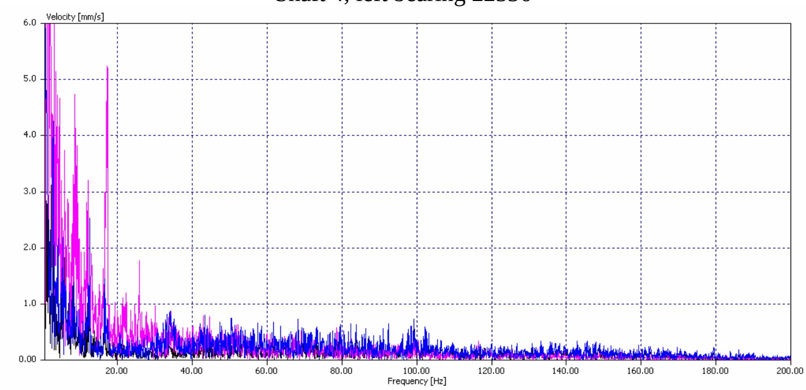
Shaft 1, bearings 22326 and 22318A



Shaft 3, right-bearing 22332



Shaft 4, left bearing 22336



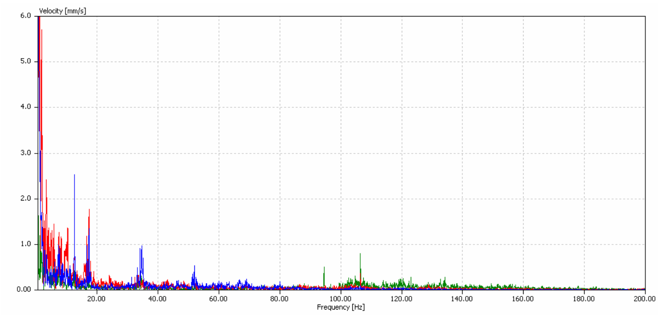
Shaft 4, right-bearing 22252

Fig. 8. Gearbox vibration measurement in [mm/s], during excavator operation; Z – green or black color, axial vibrations; P – blue color, horizontal vibrations; C – red color, vertical vibrations

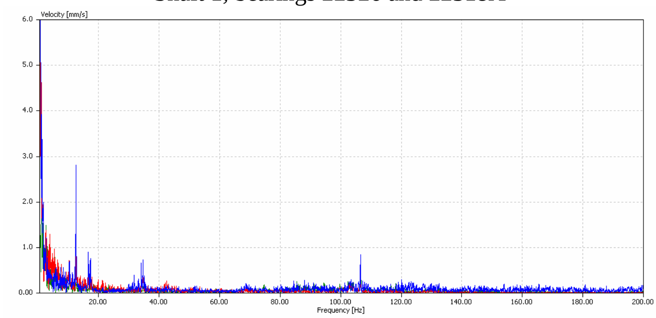
2.6. Vibration measurement of the gearbox after repair at the testing desk

Bucket wheel gearbox at the excavator SRs470.17/1.5 after its disassembly from the bucket wheel shaft, has been brought to large factory workshop, where it undergone repair and overhaul at the gearbox subassemblies, that is replacement of some bearing, shafts and gears.

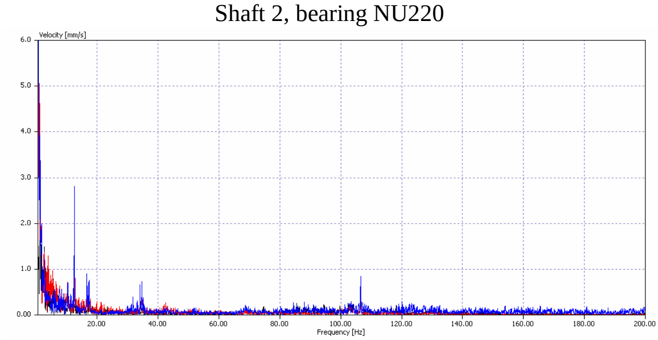
Gearbox has been connected from the testing desk to electrical engine drive with power of 90 kW and number of revolutions of ca. 955 min^{-1}. Measuring has been performed after 40 min of undisturbed operation. The following characteristic diagrams were obtained (Fig. 9):



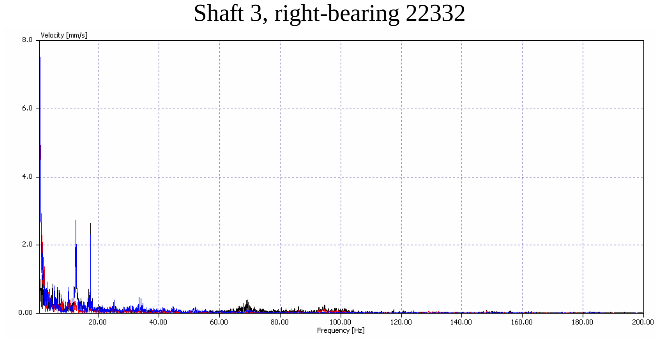
Shaft 1, bearings 22326 and 22318A



Shaft 2, bearing NU220



Shaft 3, right-bearing 22332



Shaft 6, right bearing, plain

Fig. 9. Gearbox vibration measurement in [mm/s] at the testing desk

Conclusions regarding this measurement are as follows:

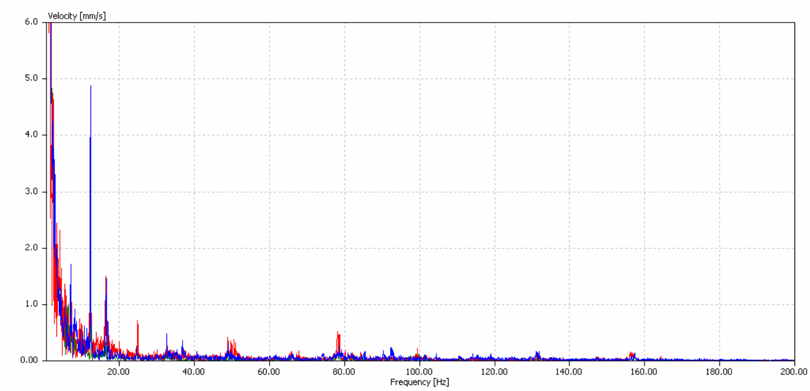
1. The first and the second gear pair has somehow worst behavior after repair because they were not replaced during repair.
2. Frequency 12, 4 Hz has been amplified, thereby indicating that it originates in the last gear pair.
3. Frequency 106 Hz originates at the outside bearing ring 22332.

2.7. Vibration measurement of the gearbox after repair in operation (during digging process)

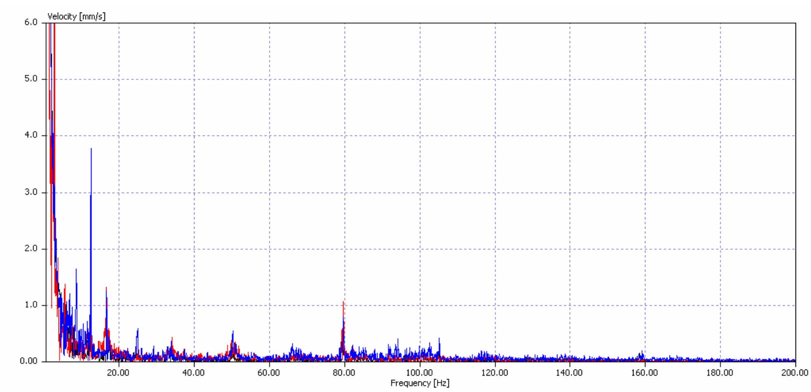
Dominant frequencies are 12.4, 16.5, 17.3 and 79 Hz. Between them the most outstanding frequency is 12.4 Hz. Dominant frequencies and its origin are as follows:

1. Frequency 12.4 Hz is present through entire gearbox. It originates from the last gear pair.
2. Frequency 16.5 Hz originates from the first and the third shafts.
3. Frequency 17.3 Hz is mostly expressed at the second, third and fourth shafts at the right gearbox side (toward the boom).
4. Frequency 79 Hz is present at the first shaft and originates from the electrical engine bearing.

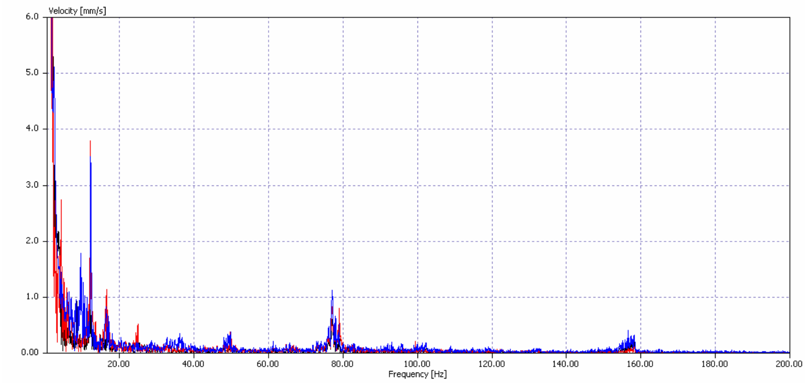
The following characteristic diagrams were obtained (Fig. 10):



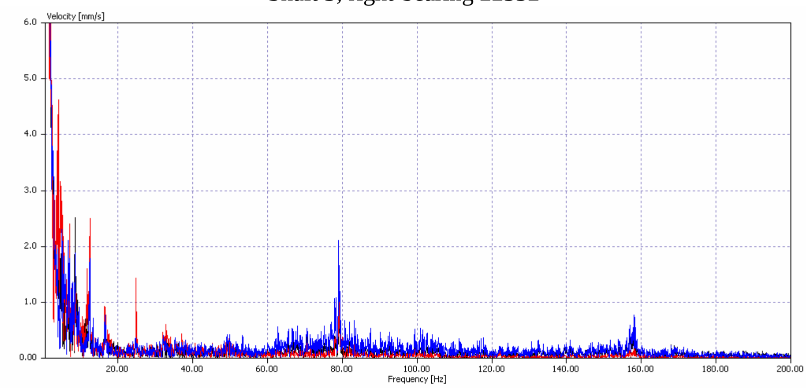
Shaft 1, bearings 22326 and 22318A



Shaft 3, left bearing 22332



Shaft 3, right-bearing 22332



Shaft 4D, right-bearing 22252

Fig. 10. Gearbox vibration measurement in [mm/s] operation after repair

3. Results and Discussion

Comparison of vibrations during excavator operation and vibrations of the gearbox at the testing desk is presented in Table 1.

Table 1. Gearbox dominant vibrations in operation and at the testing desk (amplitude values)

f[Hz]	Shaft 1		Shaft 2, left bearing		Shaft 3, right bearing		Shaft 6, right bearing	
	in operation	repair	in operation	repair	in operation	repair	in operation	repair
12,4	**hor 5** ver 2	hor 2.5	**hor 4** **ver 4**	hor 3	**hor 4** **ver 4**	ver 1.5	**hor 7.8**	ax 3 hor 3
16,5	hor 2 ver 2	hor 1.2 ver 1.2	hor 1.5 ver 1.5	hor 1	ax 1 hor 1 ver 1	-		-
17,3	hor 2 ver 2	hor 2 ver 2	hor 1.5 ver 1.5	hor 1	ax 1 hor 1 ver 1	-		hor 3 ver 3
79	hor 1 ver 1	-	hor 1 ver 1	-	ax 1 hor 1 ver 1	-	ax 1.5	-

4. Conclusions

Dynamic behavior of the bucket wheel drive assembly mainly depends on gearbox type, arrangement of structure, torque arm that is engine support, drive group support, age and time of drive assembly operation (clearance occurrence and wear of larger elements, which are used for torque transmission), structural arrangements of different elements (shafts, gears, bearings), input main parameter (input number of revolutions, revolving torque, force at the operating body, current and voltage). Vibrodiagnostics influences adoption of final decision on assembly conditions because its results are applied to all elements, subassemblies and assemblies. In addition, vibration measurement provides data required for verification of numerical results so that it is confirmed that the model yields accurate results during development of dynamic model.

Comparison of vibration measurements during operation of gearbox at the excavator and operating desk allows to conclude that the highest increase of vibrations has occurred only at the frequency of 12.4 Hz. It is understandable due to larger and unfavorable loading of the gearbox in operation at the excavator with respect to the operating desk. Testing revealed that the entire gearbox has deteriorated dynamic behavior. Detailed vibrodiagnostics of behavior and condition of bucket wheel gearbox at the excavator SRs470.170/1.5 leads to unambiguous conclusion that the gearbox needs replacement and proper adjustment to difficult operating conditions. New gearbox and support system have to eliminate existing weaknesses detected by the applied vibrodiagnostics approach. It is certain that the new gearbox must possess correctly arranged smaller mass, different gearbox interior concept and better relations between transmission efficiency, i.e. better torque transmission. It is recommended to perform vibration monitoring more frequently. It can point out causes of anomalous dynamic behavior and result in timely intervention at the specific drive assembly point thereby preventing possible damage.

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