Vibrodiagnostics of hoisting crane mechanisms using simulation modeling

A Yu Ganshkevich¹ and Ph O Fadeev²

¹ Academy of Water Transport, Russian University of Transport, 22, building 3, Novosuschevskaya street, Moscow, 127055, Russian Federation
² Original Soft LLC, Original Group JSC, 20, building 27, Ogorodny passage, Moscow, 127322, Russian Federation

E-mail: ganshkevich.a.yu@edu.rut-miit.ru

Abstract. The subject of the research is an opportunity to increase the efficiency of vibration diagnostics of hoisting cranes through the use of simulation of the vibration signal. In the article the basic approaches developed by the authors to construction of the type diagram of a mechanism and modeling of a vibrating signal of the crane mechanism with operational defects are resulted. The basic design dependences for the description of a vibrating signal of the mechanism with defects at presence of amplitude and frequency modulation are obtained, approaches to modelling of stages of acceleration and braking of the mechanism are formulated, the technique of modelling of a noise component of a signal is identified. The possible areas of application of simulation modeling during vibration diagnostics are shown and a brief description of the results obtained is given. Pre-imitation modeling will improve the detection of defects during the vibration diagnostics of cranes and reduce the time spent on the analysis of the obtained results.

1. Introduction

Nowadays, one of the most pressing problems in the operation of hoisting machines serving large transport terminals is sudden failures, which lead to unplanned downtime of equipment and serious losses. Statistical analysis of accidents and incidents in Russian ports showed that about 50% of failures arise from technical reasons: wear and tear, fatigue, etc. Decrease in the number of sudden failures can be achieved by introducing early fault diagnosis methods.

Vibrodiagnostics is one of the most effective methods of diagnosing mechanical equipment that allows one to detect defects at the stages of its origin and early development without disassembling and dismantling the units [1, 2]. The interpretation and analysis of crane mechanism vibration signals cause specific problems [3, 4], some of which can be successfully solved by using frequency-time analysis and adaptive filtering [4]. However, when analyzing mechanisms with a large number of vibration sources, the authors have faced a number of problems:

- for effective application of bandpass filters, it is necessary to know a priori the frequencies at which certain defects are most likely to be detected;
- to identify defects by frequency peaks when signals from several defects are superimposed, you need to know the values of combination frequencies
when checking hypotheses about the relationship of individual frequency peaks with specific defects, it is necessary to be able to investigate in detail the contribution of individual defects to the overall vibration spectrum.

To solve these problems, the authors developed a technology of simulation of the mechanism with operational damage, which allowed one to get a complete and more detailed picture of the composition of the vibration signal.

2. Task Setting

In any crane mechanism there are many components that are sources of vibration: clutches, shafts, bearings, gears, etc., each of which is characterized by its forced frequency, depending on the frequency of the excitation force at the input link of the mechanism. This frequency is commonly called the fundamental frequency of the unit. When defects occur, the vibration signals of the units are modulated by frequency, phase or amplitude, which leads to the appearance in the spectrum of harmonics and subharmonics of fundamental frequencies. In addition, some units demonstrate a vibro-response at its natural frequency, which does not depend on the fundamental frequency of the unit [1, 3].

The interaction of different sources of excitation leads to the appearance of a large number of combination frequencies. In case of non-stationary operation of mechanisms, which is caused by changes in the load and repeated short-term operation of the cranes, these frequencies change significantly during the cycle.

The simulation model of the mechanism should consider these factors for correct work.

3. Imitation model of mechanism

3.1. Model schematization

At the first stage of modeling, the structural scheme of the mechanism was constructed, the components of which are units and defects. A unit is a structural element that generates a vibration signal during operation, such as gearing, clutch, bearing ring, etc. Each unit is characterized by its fundamental frequency. A defect is a structural element that modulates the fundamental frequency of a unit, such as static or dynamic coupling unbalance, pitting in a gearing pair, etc. When constructing the structural diagram, the elements were in the same sequence as in the mechanism. In fig. 1, as an example, a fragment of the structural scheme of the winch of the crane hoisting mechanism "Albatros" 10/20-32/16 is given.

![Figure 1. Fragment of the structural diagram of the hoisting mechanism winch.](image)

M – Primary excitation source: electric motor; U₁…U₅ – units: clutch, cage, inner and outer bearing rings, high speed shaft and gearing, respectively; d₁…d₆₃ – possible defects of the respective units: unbalance, displacement, deformation, wear, pitting, etc.

3.2. Excitation effect simulation

Since most of the crane mechanisms mainly operate at variable speed, the vibration signal at the input was simulated to consider the change in the frequency of the excitation force. The variation of the frequency occurs during start and brake of the mechanism, as well as due to the instability of the drive load. The law of excitation frequency change was simulated by function (1):

\[
f_0(t) = f_{on} \cdot k_{ab}(t) \cdot k_f(t). \quad (1)
\]
\( f_{on} \) – rated frequency of the excitation force during steady motion; \( k_{ab}(t) \) – coefficient that takes into account the acceleration or braking of the mechanism and is determined by the formula (2); \( k_f(t) \) – coefficient that takes into account changes in motor speed due to variable load.

The law of motor speed change at start and braking modes was set based on experimental studies. The experiment was conducted on portal slewing cranes "Albatros" 10/20-32/16, "Albrecht" 10-32, "Condor", belonging to "North Port" JSC, Moscow. The mentioned cranes have asynchronous motors with a phase rotor and relay and a contactor control circuit as a part of the electric drive. Estimation of motor rotation speed was carried out based on measurement of the motor supports vibration frequency measured by accelerometers. Accelerometers VS 201 (no. 49619-12 in the Register of Measuring Instruments of the Russian Federation) and analog-to-digital converter ZET 220 produced by CJSC "Electronic Technologies and Metrology Systems" ("ETMS" CJSC) were used for measurements. Measurements were made by the "blinded" method, i.e., the crane operator was not aware of the measurement and worked in the normal mode.

As shown by the analysis of the dynamics of acceleration and braking modes of portal slewing cranes, the dependence of the drive speed on time during the cycle is well described by the piecemeal function of the type (2):

\[
k_{ab}(t) = \begin{cases} 
\sin \left( \frac{\pi t}{t_a} \right), & t < t_a \\
1, & t_a \leq t \leq (t_c - t_b), \\
\sin \left[ \pi (t - t_c - 2t_b) \right], & t > (t_c - t_b)
\end{cases}
\]

(2)

\( t_a \) – acceleration time of the mechanism to the nominal speed; \( t_c \) – operation cycle time of the mechanism; \( t_b \) – braking time of the mechanism.

The law of coefficient change \( k_f(t) \) depends on the mechanism that is modeled. For the hoisting mechanism, its value depends only on the load on the ropes and the direction of movement of the load, and is within the range of \( k_f \in [0.97; 1.05] \), and for the derrick mechanism changes smoothly when the load moment varies, which is well described by the dependence (3):

\[
k_f(t) = \frac{1}{p+1} \cdot \left[ \sin (\varepsilon \cdot t + \vartheta) + p \right].
\]

(3)

\( p, \varepsilon, \vartheta \) – parameters determined empirically for each crane model.

Simulation of vibration signals was performed with the sampling frequency \( f_s \) equal to the sampling frequency of the measuring complex, which was used for measurements. The excitation signal was described by a vector \( \bar{a}_0 \) whose elements were calculated using formula (4):

\[
a_{0i} = \sin \left( 2 \cdot \pi \cdot f_{on} \cdot \int_{t-0.5 \Delta t}^{t+0.5 \Delta t} k_{ab}(t_i) \cdot k_f(t_i) dt \right) \cdot k_{ab}(t_i) \cdot k_f(t_i),
\]

(4)

\( \Delta t = f_s^{-1} \) – sampling interval.

3.3. Defect modeling

The dependence between the excitatory frequency \( f_0 \) and the stimulated frequencies of individual units \( f_{ui} \), is in most cases described by the linear dependence of species (5):

\[
f_{ui} = u_i \cdot f_0
\]

(5)

\( u_i \) – gear ratio for unit \( i \).

In the presence of defects, the forced frequency of the unit is subjected to frequency or amplitude modulation, which manifests itself on the spectrum in the form of side harmonics with frequencies \( (f_{ui} \pm k \cdot f_{di}) \), where \( f_{di} \) – the frequency of defect manifestation, \( k = 1, 2, \ldots \). In addition, shock pulses occurring when some defects get into the contact zone, cause a response of the defective elements in the form of fast damping oscillations. Such response appears on the spectrum in the form of combination frequencies \( (f_{di} \pm k \cdot f_{di}) \), where \( f_{di} \) – shock pulse repetition frequencies, \( k = 1, 2, \ldots \)
The most common defects in crane mechanisms are: coupling and shaft misalignment, radial and angular misalignment of the coupling halves, bearing defects - wear or pitting of the rolling elements and raceways, the destruction of cages, cracks or chipped rings, gear defects - pitting, abrasive wear, cracks, chipped teeth, sticking in the pair, etc. Vibration patterns of such defects are well studied and described in detail in the literature [1, 5-9] and others. Thus, for example, surface tooth decoration manifests itself in the spectrum in the form of amplitude modulation of the tooth-fixing signal, while jam - in the form of frequency modulation of the same signal [9]. In the presence of amplitude and frequency modulation of the mathematical model of the signal defective unit was described by the dependence (6):

$$a_u(t) = \sum_d [1 + M_d \cdot \cos(\Omega_{d1} \cdot t + \Phi_d)] \cdot \cos([2 \cdot \pi \cdot f_u \cdot t + \varphi_u] + m_d \cdot \sin(\Omega_{d2} \cdot t))]$$

(6)

$M_d$ and $m_d$ – indices, and $\Omega_{d1}$ and $\Omega_{d2}$ – angular frequencies of amplitude and frequency modulation generated by the defect $d$; $\Phi_d$ and $\varphi_u$ – initial phases of amplitude modulation signals and fundamental unit vibrations, respectively.

When modeling the mechanism as a whole, the signals of individual units were summed up (7):

$$a(t) = \sum_j [\xi_j \cdot a_{u(j)}]$$

(7)

$j$ – serial number of the unit; $\xi_j$ – coefficient that takes into account the signal attenuation on the way from the source of excitation to the sensor installation point.

Based on the analytical dependence (7), $a_d$ binary signal with a given sampling rate $f_s$ by formula (8) was generated:

$$a_{di} = a \left( \frac{j}{f_s} \right)$$

(8)

An additional factor complicating the interpretation of the results of vibration diagnostics is the presence of the noise component of the signal, which was modeled by adding to the members of the series $\tilde{a}_d$, random value, distributed evenly in the interval [-1,1] and multiplied by the noise factor.

4. Modeling results

The developed simulation model was used for diagnostics of the hoisting mechanism with defects of the first and second stage gears.

The modulating frequency values $\Omega_{u1}$ and $\Omega_{u2}$ for dependence (6) were taken on the basis of known reference data [1, 7, 8] depending on the type and nature of possible defects. The initial phases $\Phi_d$ and $\varphi_u$ were assumed to be zero.

When planning a diagnosis, a simulation model was used to determine the required sampling rate. To avoid loss of diagnostic information, condition (9) must be fulfilled for the sampling rate:

$$f_s \geq 2 \cdot f_{\text{max}}$$

(9)

$f_{\text{max}}$ – maximum frequency in the signal spectrum that can carry information about defects. Manipulation of $M_d$ and $m_d$, modulation indexes made it possible to determine $f_{\text{max}}$ taking into account combinational frequencies that occur when multiple vibration processes occur simultaneously.

When analyzing the diagnosis results, the simulation model was used to specify the bandpass filter cut rates when searching for certain defects. For this purpose, all foreign sources of excitation with corresponding defects were “switched off” in the model signal, which could mask the frequencies of the required defect.

The comparative analysis of the model and experimental signals performed by the same methods allowed to reveal some uncertainties observed in the experimental signal. For example, a mathematical experiment with a simulation model of the hoisting mechanism of the «Albatros» portal slewing crane allowed one to determine that unidentified peaks in the Fourier spectrum, based on the results of a field experiment, correspond to the frequencies of gear defects shifted to the low frequencies region as a result of aliasing.
5. Conclusion

1. The pre-simulation modeling allows one to choose the right characteristics of the measuring system.
2. Pre-simulation with "deactivation" of individual defects allows selecting the frequency bands to search for specific defects.
3. Comparative analysis of the model behavior and the diagnosed unit allows one to reveal uncertainties arising from the analysis of the vibration diagnosis results.

So, the simulation modeling allows increasing the efficiency of application of vibration diagnostic methods.

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