Substantiation of connection of diagnostic indicators with parameters of technical condition of hydraulic manipulators

B Martynov, N Sidorenkov, M Taraban

1 St. Petersburg State Forest Technical University, 5 Institutskiy Lane, St. Petersburg 194021, Russian Federation
2 JSC «Podyemnye Mashiny», 182112, 8Korniyenko street, Velikiye Luki, Pskov oblast, Russian Federation

*Corresponding email: lesbisnes@mail.ru

Abstract. At the organization of proactive system of maintenance of mechanisms the technique based on determination of relative probability is applied. But in contrast to the technique, when the machine is for maintenance and adjusted the type and frequency of the hydraulic manipulators (HM) it is necessary to carry out constant monitoring of the technical condition of the main elements of the mechanism. The main elements of the HM, which primarily determine its technical condition, are the distribution system, including the pump, oil pressure distributors in the executive bodies of the HM, as well as the column, the extension-grip joint and the rotator. Nominal and permissible pressure values in the distribution system are determined by the technical documentation on the HM. The paper considers the dependence of the vibration signal index upon the technical condition of the coupling shaft-gear column HM.

1. Introduction
By organizing proactive system maintenance of mechanisms, the technique is applied based on determination of relative probability [1]. However, unlike the technique when the machine is currently on maintenance where the type and frequency are adjusted, for hydraulic grapple it is necessary to control constantly the technical condition of the main elements of the mechanism. The main elements of the grapple, which define primarily its technical condition, are the distribution system, including pumps, valves oil pressure in the Executive apparatus, the hydraulic grapple and the column joint extension tube-grab and rotator. Minor and allowable pressure in the distribution system are defined by technical documentation of the hydraulic grapple. In the work the dependence of the vibration signal index upon technical condition of the finger joint of the grapple column’s shaft gear.

The purpose of the studies was to define the correlation of vibro-acoustic signal and the technical condition’s parameters of the grapple column’s pairing.

2. Methods and Materials
The technical condition of grapple’s column will be determined specifically directly by the finger joint of the shaft gear and by the technical condition of the bearing of this pairing.

In the work [2] the matter are reviewed concerning appearance of the errors in finger joint and determination of the force causing oscillation in transmission.

The impact force of finger joint defines by:
where, \( J_1, J_2 \) – moments of inertia of the drive and driven wheels; \( \omega_1, \omega_2 \)–the angular velocities of the wheels; \( r_i \) – radius of pitch circle; \( \frac{\Delta \iota}{\iota} \) - relative deviation of the transmission ratio.

For coupling a rack pinion of the grapple’s mechanism, expression for impact force is presented as following:

\[
g_0 = \frac{J_{r1}J_{r2}V}{J_{r1}V + J_{r2}\omega_2} + \frac{\Delta \iota}{\iota}
\]

where, \( J_{r1}, J_{r2} \) — the moments of inertia of the rack and pinion respectively; \( V \) – linear velocity of the rack; \( \omega_2 \) – angular velocity of the pinion.

Considering that angular velocity is connected to linear relationship \( V = R \omega \), and solving the moments of inertia, we will get:

\[
g_0 = \frac{m_{r1}m_{r2}R_{r1}R_{r2}V}{2(m_{r1}R_{r1}V + 0.5m_{r2}R_{r2}V)} + \frac{\Delta \iota}{\iota}
\]

where, \( m_{r1}, m_{r2} \)– masses of the rack and pinion respectively; \( R_{r1}, R_{r2} \)–radiuses of the rack and pinion respectively; \( V, V_2 \) – linear velocities of the rack and pinion respectively.

A relative deviation of the transmission ratio of the coupling rack pinion will depend on error of the teeth profiles and side clearance of gearing teeth [3]. On one hand, the error profiles of the teeth appears by their cutting, therefore, it will be systematic error of the gear. On the other hand, the error of increase of side clearance appears at the operating conditions. Therefore, the final expression for impact force value will be presented as following:

\[
g_0 = \frac{m_{r1}m_{r2}R_{r1}R_{r2}V}{2(m_{r1}R_{r1}V + 0.5m_{r2}R_{r2}V)} \cdot (\delta + c h) = \frac{K_2V}{2K_2V + K_3V_2}, (\delta + c h)
\]

where, \( K_1, K_2, K_3 \) – constant values, depending on geometric size and masses of coupling; \( \delta \) – error teeth profile; \( c \)–average teeth stiffness; \( h \) – value of the side clearance.

For defining the amplitude of vibration signal perceived by the sensor, we use the second Newton’s law presented in differential form [4].

\[
m \frac{d^2y}{dt^2} + \varepsilon \frac{dy}{dt} + cy = F(t)
\]

where, \( m = \frac{m_{r1}m_{r2}}{m_{r1} + m_{r2}} \) provided mass of the sensor-object system for diagnosing; \( \varepsilon \) – coefficient of the tenacious deformation; \( c \)–stiffness of sensor binding to object of diagnosing; \( F(t) \) – force oscillating transmissions in object of diagnosing.

Assuming that sensor-object system is stiff system where coefficient of the tenacious deformation is almost equal zero and considering that amplitude of the impact force determines by (4), the formula can be modified as following:

\[
A = \frac{K_3V}{2K_2V + K_3V_2}, (\delta + c h) + K_4
\]

where, \( K_4 \) – coefficient of the stiffness for the sensor-object system; \( A \) – amplitude of the vibration signal.

Considering that error teeth profiles is at least an order of magnitude smaller than stiffness value and knowing value of the velocities and stiffness of the binding sensor, the amplitude of the vibro-acceleration can be made a simple formula:

\[
A = f(h)
\]

Under operating conditions in the process of wear of a rolling bearing, various types of malfunction occur, leading to an increase in the vibration signal [5].
An arising vibration signal is momentary, but it has a wide range of frequencies from 2 Hz to 25 kHz.

We will consider the case of a rolling bearing used on a grapple. A spherical roller bearing is set on the rotary grapple column. In many cases, the technical condition of the bearing affects the reliability and safety of such low-speed mechanisms.

The main malfunctions of spherical bearings are wear of the rolling path of the rings due to combined loads during operation, chipped ring edges, a shell and cracks on the rolling surfaces, installation defects.

All types of defects occur at the low-frequency area, with the exception of installation defects, which occur at medium frequencies. Moreover, the frequency of occurrence of defects is equal to [5]:

\[ f_o = \frac{1}{2} f_y (1 - \frac{r_c}{r_y} \cos \alpha) \cdot z \]  

(8)

where \( f_o \) – frequency of occurrence of outer ring; \( f_y \) - shaft speed; \( \alpha \) - contact angle of the rollers with rolling path; \( z \) – number of the rolling elements; \( r_c, r_y \) – the radii of the rollers and shaft, respectively.

In [6], it was proved that a change in the frequency of rotation of a defective bearing does not affect the change in the frequency of free oscillations of the node in which it is installed. Therefore, the noise in the form of vibration can be neglected from the frequency of oscillation of the structure.

As the result, the dependence of the vibration acceleration pulse from shock loads in the bearing can be written as follows:

\[ m \frac{d^2 y}{dt^2} + \varepsilon \frac{dy}{dt} + cy = F(t) + Q(t) \]  

(9)

where, \( F(t) \) – restoring force due to contact elasticity [7]; \( Q(t) \)– radial load.

Since the grapple’s operation occurs in the start-stop mode, i.e. the main impact energy is concentrated in the impulse of force arising at the beginning of work, therefore, the restoring force does not have a noticeable effect on the amplitude of vibration due to the duration of the process.

Moreover, the radial load will be characterized by the impulse of force, which is determined from the expression [8]:

\[ g_1 = S(1 + l) \left( \frac{F_m}{r} \sin \frac{\pi}{N} \right)^{0.5} \]  

(10)

where, \( g_1 \) – force impulse; \( l \)–coefficient of restoring; \( F \) – radial force acting on the bearing; \( m \) – bearing mass; \( S \) – gap; \( r \) – rolling elements radius; \( N \) –number of the rolling elements.

Radial load is associated with the clearance correlation [2, 7]:

\[ F = N l 1.5 \cdot 2 r S^{1.5} \]  

(11)

Substituting (11) into (10) in conclusion for the force impulse we will get:

\[ g_1 = KS^{1.25} [2N(\sin \frac{\pi}{N})]^{0.5} \]  

(12)

where, \( K \)– constant depending on the type of the bearing.

Because of changing the parameters of the technical state, the value of the disturbing force changes. This is due to the presence of amplitude modulation of the signal:

\[ g_1 = KS^{1.25} [1 + \xi \sin(\psi t + \alpha)](2N \sin \frac{\pi}{N})^{0.5} \]  

(13)

where, \( \xi \) – modulation coefficient, which generally considers the errors of the rings, rolling elements, etc.; \( \psi \)–modulation frequency; \( \alpha \) – modulation phase.

Substituting expression (13) into (9), we obtain the dependence of the amplitude of vibration acceleration on bearing errors.

\[ m \frac{d^2 y}{dt^2} + \varepsilon \frac{dy}{dt} + cy = KS^{1.25} [1 + \xi \sin(\psi t + \alpha)](2N \sin \frac{\pi}{N})^{0.5} \]  

(14)
Since the stiffness condition of the sensor-object system is preserved, equation (14) is converted to:

$$\frac{d^2y}{dt^2} = \frac{1}{m} KS^{1.25} \left[ 1 + \xi \sin(\psi t + \alpha) \right] (2N \sin \frac{\pi}{n})^{0.5}$$

(15)

As has been said, all types of bearing defects mainly occur in the low frequency area. Therefore, in this case, it is necessary to measure not the amplitude of vibration acceleration, but the amplitude of vibration velocity, since it more accurately describes the impact energy [5].

3. Results and Discussion

For determining the relationship of the amplitude of the velocity of the vibration signal from the parameters of the technical condition of the bearing, firstly a particular solution of the equation (15) must be found in the form [3]

$$y = b \sin(\omega t + \varphi)$$

(16)

where, \( b \) – the amplitude of the excitation force; \( \omega \) – excitation force frequency; \( \varphi \) – excitation force phase.

After that, we must find the first derivative of the obtained value of the vibration movement. Because of these transformations, we obtain:

$$\frac{dy}{dt} = \frac{2\omega_0 \omega_1}{(\omega_0 - \omega_1^2)} [1 + \xi \sin(\psi t + \alpha)] \sin(\omega t + \varphi)$$

(17)

where, \( \omega_0 \) – natural system frequency.

Since the natural frequency of the system is at least two orders of magnitude lower than the frequency of the exciting force, and the frequency and the modulation phase are almost the same as the carrier frequency, equation (17) reduces to the following:

$$B = \frac{KL^{1.25}}{\omega m} (1 + \xi)$$

(18)

where, \( B \) – amplitude of the vibration velocity.

Thus, analyzing the formula (18), we can conclude that the amplitude of the vibration velocity increases with growing radial clearance in the bearing and modulation coefficient.

References

[1] Martynov B G 2005 Substantiation of individual strategy of effective operation of forest machines by results of diagnostics (in Russian – Obosnovanie individualnoi strategii effektivnoi ekspluanacii lesnyh machin porezultatam diagnostirovanija) Izvestia Sankt-Peterburgskoj lesotehnicaskoj akademii (Saint-Petersburg: SPbFTA) issue 172, pp 62-65

[2] Dynamic processes in mechanisms with gears 1976 (in Russian – Dinamicheskie processy v mehanizmah s zubchatymi peredachami) ed M D Genkin (Moscow: Nauka) 154

[3] Martynov B G 2005 Justification of effective operation of individual machines according to the results of their technical condition (in Russian – Obosnovanie effektivnoi ekspluanacii individualnyh machin po rezultatam ih tehniceskogo sostoiannia) Doctoral thesis. St. Petersburg, 361

[4] Shkalikov V S 1980 Measurement of vibration and shock parameters (in Russian – Izmerenie parametrov vibracii i udara) (Moscow: Publishing house of standards), 280

[5] Barkov AV, Barkova N A and Azovtsev A Yu 2000 Monitoring and diagnostics of rotary machines by vibration (in Russian – Monitoring i diagnostika rotornyh machin po vibracii) (Saint-Petersburg: Sbpgmtu), 158
[6] Kostyukov V N 1985 Rank method of vibroacoustic diagnostics and quality assessment of machines (in Russian – Rangovyi metod vibroakusticheskoi diagnostiki I oценki kachestva machin) *Hydraulic Drive and control systems of construction, traction and road machines*, (Omsk: OMPI), pp 113-124

[7] Sheftel B T and Shanitsyn A A 1973 Vibration of ball bearing with radial clearance (in Russian – Vibraciia sharikopodshipnika s radialnym zazorom), *Machine science* (Mashinovedenie), Moscow, 4, pp 29-35

[8] Pavlov B V 1971 Acoustical diagnosis of mechanisms (in Russian – Akusticheskaia diagnostika mehanizmov) (Moscow: Mashinostroenie), p 221