Hydraulic drive of vibration stand for testing the robotic systems units by random vibration method

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Abstract. This paper reviews the development of hydraulic drive of vibration stand for testing the robotic systems units (technical products) for vibration resistance by random vibration method. The physical laboratory model was developed for studying the dynamic properties of a test stand which allows to realize smooth sweep excitation frequency f1 within the range of 8...56 Hz and detect resonant modes. The new design allows adequate simulation of vibration during transportation and using gapless narrow-band spectrum of random vibrations from power units. The stand allows to implement a large number of test modes that demonstrate the prospects of its application for testing technical products by random vibration method.

1. Introduction
It is known in mechanical engineering that a significant number of products failures during operating are caused by vibration during transportation and from power units. Dynamic loads caused by operation are significant and can lead not only to functional disorder (loss vibration resistance) of hydro-mechanical, pneumatic, electrohydraulic and similar devices, but can also result mechanical failure of structural components of such equipment (loss of vibration strength). Before launching the production, devices must be tested in more extreme conditions than the actual operational ones [1–8].

One method of vibration tests is carried out as follows: the smooth flat pattern of the excitation frequency in the specified range is used to determine the present frequency of the vibrations of the test elements [9, 10]. After such scanning, the product shall be subject to a vibrational load in the local resonance mode. By the result of the tests it is decided whether changes to design or modifications should be fulfilled [1]. Another efficient method is the method of amplitude-modulated oscillations [1, 3] with assistance of which the harmonic structure can be presented by a wide line spectrum [12, 13] thus adequately simulating the transport vibration and the vibration of operating power units of technological machines. Amplitude-frequency modulation test is the most efficient when amplitude fluctuation is supplemented by changed carrier excitation frequency [14–17]. The maximum vibration testing effect [9, 10] can be achieved when implementing the continuous spectrum with a given width range. Any technical object is characterized by the spectrum of natural frequencies, which can have a wide frequency range depending on its size and structure. At transport vibration, the elements of various technical devices of transport and operational machines are impacted simultaneously by the dynamic within the given range. Therefore, the testing method of random vibration is the most difficult from the technical point of view [1, 2, 18–20].
2. Test stand for excitation of random vibration

Random vibration test method can be carried out in a testing laboratory by using a hydrostatic vibratory drive of linear excitation. Vibration test stand (Figure 1) differs from the construction described above [21], by the presence of:

- test platform 4, hydrocylinder 3, which rod is connected with a spring 5 providing a reverse motion of the platform and its setting to its original position when the initial pressure \( P_0 \) changes in the hydraulic system of oscillating circuit: the piston – a shell – the test platform;
- reverse two-position direction valve 9;
- servo-motor No.2 controlled by a random signal generator through the direction valve 9 which provides parameters change \( a_1 \) and \( a_2 \) of generator signal \( w_{x_1} \) (1).

Hydraulic power drives the eccentric 6 which acts on the generator piston. In this case the alternating fluid flow \( w_{x_1} \) is displaced from the shell 1 into the hydraulic cylinder 3 of platform 4, raising the platform vibration.

![Figure 1. Schematic diagram of the test stand for random vibration excitation:](image)

- \( b\) and \( h\) – shell preload options; \( c\) – spring stiffness \( 5\); \( w_{x_1}\) – fluid volume in the cavities; \( \Delta w_{x_1}\) – alternating fluid flow; \( x_1(\omega_1)\), \( x_2(\omega_2)\) – the laws of generator pistons motion of oscillations and hydrocylinder; \( \varphi_1\), \( \varphi_2\) – rotation angles of the preload regulator 2 and the regulator of displacement volume of the pump 8, accordingly; \( \omega_1\), \( \omega_2\) – angular speed of shaft rotation of hydro-motor 7 and test platform 4 oscillations.
The law of actuating fluid displacement is described by the quadratic function [21]:

\[ w_{x1} = a_1 \cdot x_1(\omega_1) + a_2 \cdot x_1^2(\omega_1) \]  

(1)

where \( x_1(\omega_1) = e \cdot \sin\omega_1 t \) – input signal (generator piston movement); \( e \) – the generator shaft eccentricity; \( a_1 \) and \( a_2 \) – geometric constants of the elastic shell [21].

The dynamic structure of stand in question shown by the diagram is [21].

Accelerometers, the data line, the vibration measurement module and the laptop are the part of the mobile complex for rapid diagnosis of the technological system elements, which principle of operation is described in detail. The pressure measurement module has been added.

The accelerometers are installed on the device for accelerometer 19 and on the test platform 4, oil-pressure sensor is installed on the preloaded elastic shell 1. Data on pressure and movement from sensors 16-18 come in from the data line to measurement modules (analog-to-digital converter) and after digitization arrive on the laptop equipped by special software «Vibroregistrator-M2», created in a graphical programming environment National Instruments LabVIEW [22]. Thus, the values of the oscillations \( x_1 \) (mm) of the generator plunger and \( x_2 \) (mm) of the hydraulic cylinder 3, the pressure \( P \) (MPa) in the shell \( J \), and hence the frequencies rate \( f_1 \) and \( f_2 \) (Hz) are obtained. Registered data are processed in the software environment “Vibroregistrator-M2”. The formulas given in this paper and in [21] made it possible to construct experimental dependences.

3. Experimental Research

To study the dynamic properties of a test stand the physical laboratory model was developed. It allowed to realize smooth sweep excitation frequency \( f_1 \) within the range of 8...56 Hz and detect resonant modes.

The first resonant mode (Figure 2) is characterized by abrupt transition to larger amplitudes in accordance with the relative shell preload:

\[ \Delta = \frac{h}{d_0} \]  

(2)

and by frequency ratio:

\[ \frac{f_1}{f_2} = 1, \]  

(3)

where \( d_0 \) – the inner shell diameter, m; \( h \) – elastic shell preload, m; \( f_1, f_2 \) – input \( x_1(\omega_1) \) and output signals \( x_2(\omega_2) \) frequency, Hz (Figure 1).

The relative amplitude or dynamic factor which is equal to the ratio of amplitude in resonance with respect to the static amplitude when \( \omega_1 = 0 \), reaches in the first resonance values \( k_d=10...11 \). This indicates high quality of the system.

The second resonant mode is carried out in conditions of capture the generator \( f_1 \) drive signal frequency by frequency \( f_2 \) with the ratio of generator drive signal frequencies and test platform equals:

\[ \frac{f_1}{f_2} = 2, \]  

(4)

Equation 4 corresponds to the resonance with the parametric excitation of oscillations, so we can use the Mathieu equation [23–30].

Relating to this system it can be shown that the value range of its parameters, where second (parametric) resonant mode is implemented in the second (parametric) resonant mode can be described by the expressions [23, 24]:

\[ c_{th} = k_1 + \left[ \frac{m(a_k f_e^2)}{2\alpha} \right] \left( 1 \pm \sqrt{1 - \frac{2\alpha^2 k_2 f_2^3 A_{k_2}}{m(a_k f_e^2)}} - c - k_2 f_2^2 \right) \]  

(5)
where $\alpha$ – viscous friction coefficient, N·s/m; $d$ – outer diameters of the shell, mm; $c_{sh}$ – volumetric shell rigidity; $k_1$, $k_2$ – empirical coefficients, correspondingly in N/m and N/m$^2$; $A_{x2}$ – test object amplitude, mm; $c$ – spring stiffness.

$\kappa_2 = \frac{2 \alpha^2 k_2 f^3 A_{x2}}{m(a k_2 e f^2)} = 0$  \hspace{1cm} (6)

If draw the horizontal axis through the points $c$ and expand areas $ac$ around them 180 °, we obtain the limiting areas, approximately corresponding to the experiments results.

Left fields are limited minimally by acceptable preliminary shell preload, taking into account the equation (2):

$$h_{\text{min}} = d - 0.5d_0 + e.$$  \hspace{1cm} (7)

Qualitatively, amplitude frequency characteristics of the main and second resonances (Figure 2) do not differ, but the amplification of the oscillations in the second resonance mode is much higher, so the system dynamic response factor reaches $k_d = 15 \ldots 16$. The quality factor of the system increases and this is of theoretical interest.
Figure 3. Parameters value range $c_{sh}$ and $\Delta$ ($d_0=20$ mm, $b=45$ mm), where the second resonant mode is implemented $1 - e=1.22$ mm; $2 - e=1.67$ mm.

The most informative content about the energy exchange in oscillatory systems has the phase conditions [11, 23]. Figure 4 shows the experimental oscillograms of the second resonance.

Figure 4. The vibration oscillograms in the second resonance: $x_1$ и $x_2$ – the plunger oscillations of generator and hydraulic cylinder, $P$ – pressure in its cavity and shell.
The pressure oscillogram shows that the dynamic interaction in the system is in the section 1-2 – the pressure reaches the maximum value. There is a pause in the 2-3 section – the pressure value approaches to zero and this is explained by the fact that the hydraulic cylinder plunger, while moving upwards, «pulls» out of the shell almost all of the liquid which was accumulated in it at \( P_0 \). There is no interaction in the pause.

The signal section \( x_1 \), at the moment of the interaction pulse (points 1 and 2), corresponds to the rotation angle of the generator eccentric \( \sim 210^\circ ... 235^\circ \). This is much larger than at the first resonance \( 135^\circ ... 150^\circ \) (Figure 4 does not demonstrate). Obviously the interaction angles and the phase shift between \( x_1 \) and \( x_2 \) are linked to energy exchange. However, unlike linear systems, the measurement of the phase shift on the displacement of the oscillation peaks is impossible here, since the phase conditions varies also within the period of oscillation («fast phase»).

The original method was used to processing the oscillograms. The oscillation in a nonlinear system can be represented by a vector rotating with variable frequency. By means of sequential constructions it is possible to establish a change in the «fast phase» depending on rotation angle and the vector of the input signal is \( x_1 \).

Figure 5 shows its variation and the average integral value of \( \psi_{ai} \) at the second resonance at \( \Delta=1.2 \) mm and \( P_0=0.6 \) MPa.

![Figure 5](image_url)

**Figure 5.** The diagram of the definition for the average integral value of the phase shift.

There is energy exchange in the system at the dynamic interaction, so the phase shift is determined on the section between the dashed lines (Figure 5) corresponding to p. 1 and 2 on the oscillogram \( P(t) \) (Figure 4). The average integral value of the phase was determined by the equality of the dashed areas \( s_1 \) and \( s_2 \).

Thus, several oscillograms were processed at different frequencies \( \omega_1 \). At the points of the maximum amplitude \( A_2 \) of the second resonance, the phase error between \( x_1 \) and \( x_2 \) is 39 \(^\circ\), whereas in the first resonance, \( \psi_{ai} \) was 74 \(^\circ\).

The vector diagrams (Figure 5) correspond to the points at the beginning of the interaction (p. a, Figure) at average integral value of the phase (p. b) and at the end of the interaction (p. c).

For linear system, the energy balance on one period is defined as

\[
K + P + E_a = E_0 + E_{in}
\]

where \( E_0 \) – energy level of the system in resonance, \( E_0=K+P=\text{const.} \)
The balance of reactive energies (kinetic $K$ and potential $P$) and the balance of dissipative energy $E_d$ and the input energy $E_{in}$ are saved. The elastic and inertial force vectors as well as driving force and resistance forces are directed opposite to one another and the phase shift is equal to 90°. Here the energy exchange is significantly different. The interaction between the generator and hydraulic cylinder starts at $\psi=64^\circ$, and the output from it corresponds to 8° (Figure 6).

All values in the variation interval $\psi$ are less than 90°, therefore it can be seen on the vector diagrams that the driving force $F$ not only compensates the resistance force, but adds its own part – the component $F \cos \psi$ to the reactive forces $\Phi$ and $F_y$. This component is directed along the $x_2$ axis and performs positive work. This means that during the interaction period (p. 1 and 2 in Figure 4), the generator introduces into the system additional energy $\Delta E_0$.

As a comparison, in the first resonance, the interaction in the system begins at $\psi=105^\circ$, and the output from it at 45°, therefore at the beginning of the interaction the reactive component $F \cos \psi$ is directed opposite to the $x_2$ axis and performs negative work, braking the oscillations of the hydraulic cylinder (Figure 6). The average integral value of the «fast phase» is less than the phase shift in the linear system (74° against 90°), so the amplitude $A_2$ in our case is larger than for a linear system with equivalent dissipative energy scattered during a period, but less than at the same system in the resonance regime at $\psi=35^\circ$. At point 1 (Figure 4), the hydraulic cylinder plug enters with the energy:

$$E_{01}=K_1+P_1,$$

and from point 2 comes out with increased energy:

$$E_{02}=K_2+P_2>E_{01},$$

however:

$$\Delta E_0=E_{02}-E_{01}.$$

![Figure 6. Vector diagrams at parametric resonance: $F$ – vector of the driving force is equivalent to the kinematic excitation, $\Phi$, $\vec{R}$ и $F_y$ – vectors of inertial forces, viscous drag and elastic force.]

As follows from the energy exchange, the generator inputs energy that compensates the dissipative energy of frictional forces while another quotient of the input energy is accumulated in the form of reactive energy of the actuator. The non-return of this energy is due to the «detachment» of the hydraulic cylinder plunger in the pause, when the reactive energy $\Delta E_0$ is converted to the active $E_a$ in the section of 2-3 (Figure 4).

The second resonance effect is of great practical value. This resonance effect allows the significant expansion of the frequency range of the implemented oscillations spectrum with the amplitude and frequency modulation at a variable parameter $h$. Figure 2 shows the amplitude-frequency characteristics of the oscillation circuit, obtained by continuous sweep of the excitation frequency $f_1$ in the range of 8...56 Hz. At first the system passes through the first resonance with slow amplitude $A_{x2}$ increasing at $a$-$b$ range and its following leap to the value at the point $c$. However, transition to point $c$ is a transition process shown in timed sweep oscillations (Figure 7).
Further, the amplitude is reduced in accordance with the gain slope, passing through the minimum at the point \( d \); but at the point \( b \) there is the second leap to the amplitude value \( A_{x2} \) at the point \( f \). During \( e-f \) transition the oscillation frequency of the test platform instantly reduces in half as the generator \( f_1 \) basic frequency is captured by frequency \( f_2 \) at their ratio \( f_1/f_2=\omega_1/\omega_2=2 \). Therefore, at the range of \( e-f \) (Figure 6) the transition process is much shorter. At the point \( f \) there are two possible options for further displacement of the excitation frequency \( f_1 \) in the following direction:

- frequency increases with decrease of amplitude \( A_{x2} \) with simultaneous increase of \( f_2 \);
- strengthening of the amplitude to the maximum value \( A_{x2} \) of dynamic coefficient \( k_d \) with decrease of oscillations frequency of test platform \( f_2 \).

Both designs can be implemented by control system of pump 8 working volume with servo motor No.1 (Figure 1). These designs will change the shape of the modulation and, accordingly, the spectral structure of oscillations. Then the backward displacement of the excitation frequency \( f_1 \) from the point \( f \) to \( g \), the frequency \( f_2 \) will be reduced in accordance with condition [21]:

\[
C+Ca\omega^2.
\]  

The amplitude will grow until it reaches the stability limit of the oscillation circuit in the frequency capture mode. System transition to the point \( h \) is accompanied by instantaneous frequency change \( f_2 \) to \( f_1 \) and a sudden fall of the amplitude and the dynamic factor. Further, displacement of the excitation frequency \( f_1 \) downwards leads to the resonance increase to stability limit of the oscillation circuit at the point \( i \). Behind it, the system returns to its original state – point \( a \) and then there is a repeat cycle.

Graphic illustration of such oscillation process with amplitude and frequency modulation gives a picture of "mobile" line spectrum (Figure 8).

Thus timed sweep oscillations of test stand platform within the frequency range ~8...56 Hz, covering both the resonant modes, demonstrate the feasibility of implementing the amplitude-frequency modulated oscillations, with the possibility to perform control in three ways:

- changing the frequency sweep range of the excitation frequency \( f_1 \), which changes the shape of the amplitude-modulation function, and therefore, the harmonic structure of the oscillations;
- changing the position of the reversal points of the sweep frequency excitation on the AFC system;
- changing the frequency sweep range of the excitation frequency \( f_1 \), which changes the shape of the frequency-modulation function, and therefore, the spectral structure of the oscillations.
random changing of the parameters $a_1$ and $a_2$ (1) of the generator oscillator signal by changing the shell preload $h$ due to the servo motor No. 2, managed by the generator of occasional signals through the valve 9 (Figure 1).

Figure 8. «Mobile» line spectrum as a model for the system passage through the resonant modes with the variables amplitude $A_{x2}$ and frequencies $f_1$ to $f_2$:
point $i$ – the main resonant mode $(f_1/f_2=1)$; point $g$ – the second resonant mode $(f_1/f_2=2)$.

One of the modulation function designs corresponds to the above considered mode of timed sweep excitation frequency and is shown below (Figure 9a).

Figure 9. Modulating functions: $a$ – original; $b$ – fluctuating;
$T$ – original function period; $T'$ – the approximate period of the original function.

The elastic shell regulator of oscillations generator (Figure 1) is connected through the transmission with rotary hydro-cylinder shaft, the latter is connected to the hydraulic system via the electrically operated reversing on-off valve 9. If we use the signals produced by the occasional signal generator [1]
as controllers, then due to the randomly changing supplied volumes $-\Delta w$ and $+\Delta w$, the oscillations amplitude of the test platform $A_x$ will receive unpredictable disturbances that lead to fluctuations in the original modulating function (Figure 9, b).

Obviously, if rotation angles $\varphi_1$ and controller of preload $h$ change randomly (Figure 1), the oscillations spectrum of the test platform will be continuous and unpredictable and fluctuate randomly within frequency range from 8…10 to 60 Hz or any other specified range.

4. Summary
The performed research demonstrates that the changing narrowband random vibration mode is one of the most effective methods for vibration testing, including the broad-band vibration tests, further, a stand design allows to implement a large number of test modes. In addition to practical significance of the second resonance, it is also interesting from the theoretical point of view, since despite the external similarity with the parametric one, the energy processes in the dynamical system considered are essentially different.

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