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SUBSTANTIATION OF ADEQUACY OF LOADING CONDITIONS AT BENCH AND FIELD TESTS OF CONSTRUCTION MACHINES

L. Pelevin
PhD, Professor*
E-mail: pelevin_leonid@ukr.net
A. Fomin
PhD, Professor*
E-mail: fomin.av@knuba.edu.ua
Ye. Gorbatyuk
PhD, Associate Professor*
E-mail: gek_gor@i.ua
G. Machishin
PhD, Associate Professor*
E-mail: ma4ichin@ukr.net
*Department of Construction Machinery
Kyiv National University of Construction and Architecture
Povitroflotsky ave., 31, Kyiv, Ukraine, 03037

Описано адекватність режимів навантажень при стендових випробуваннях до навантажень, що діють на будівельні машини при реальних режимах експлуатації. Встановлено необхідність дотримання наступних умов випробування: вузол, що досліджується, не наближається до резонансу; вплив частот повторно-змінного навантаження на процес руйнування від втоми незначний. Дотримання вказаних умов дозволяє використовувати теорію ймовірностей та математичної статистики для: розрахунку на витривалість при різноманітних параметрах навантаження; моделювання різноманітних умов роботи машини. Описана в статті методика дозволяє при створені будівельних машин економити час та гроші. При випробуванні машин та їх вузлів визначати надійність та ресурс безвідмовної роботи. Це дозволяє знизити металоємність та підвищити якість машини. Визначення коефіцієнтів кореляційного зв'язку при випробуваннях з форсуванням за частотою та амплітудою дозволяє визначати зв'язки між характеристиками експлуатаційного і стендового режимів навантаження. Використані показники надійності з використанням фізико-статистичного аналізу робочих процесів будівельної техніки

Ключові слова: прискорені стендові випробування, випадкові навантаження, гіпотеза спектрального підсумовування, ходова частина

1. Introduction

The present-day production of construction machines is impossible without conducting a large test cycle before their delivery to the customer. Tests start from prototype models and end with serial machines. Prototype tests play an important role in improving the construction machine design as they can reveal drawbacks and improve reliability of the machine design.

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In engineering, development of methods and techniques for accelerated bench testing of construction machine drives enables a more qualitative testing of construction machines for reliability and determination of their operability in shorter terms. A separate scientific task consists in development of such test bench designs that make it possible to realize correspondence of reliability characteristics obtained at the test bench with those taking place in operation [1–4].

2. Literature review and problem statement

The calculation methods used for prediction of machine life and reliability are increasingly used in various fields of science and technology. Prediction should be carried out in the early stages of machine designing [5]. However, the authors introduced only an algorithm for implementation of the prediction process.

The problems of strength and life of mechanisms were investigated in [6]. The authors got rid of the empirical study method and applied a procedure of visual inspection. This requires development of special test benches for individual machine mechanisms which is very costly.

In [7], no bench tests are used at all. All data on failures and breakdowns are revealed by long-term observations. This method cannot be used for rapid re-equipment of production and development of new machine designs.

Prediction and ensuring longevity of elements of the chassis of high-speed crawler machines was reduced to the study of solely torque shaft failures [8]. The method for determining life in multiple-cycle fatigue damage is based on the assumption that the material fatigue is determined by action of the force factors similar to the friction forces. The method does not require a preliminary schematization of loading processes. This method is convenient for application as a component of software of on-board control information systems. It is applied to study long-term stress variation processes in modeling machine movements.

The vast majority of earthmoving construction machines have a crawler drive. Studies of movement smoothness of the machines with a crawler-mounted mover are usually limited to definition of vertical and longitudinal oscillations [9, 10]. Earth-moving machines usually move on unlevelled construction sites most of their working time. Thus, the machine chassis and body are strongly influenced by longitudinal angle oscillations that arise because of difference between the left and the right track profiles. The effect of transverse angle oscillations on the chassis and body of crawler machines was modeled in [11, 12] which makes it possible to consider dynamics of spatial movement of a crawler machine to reduce impact of harmful oscillations on machine operators.

The life of crawler construction machines is influenced even by rubber seals [13]. The authors confirmed the results obtained by field tests. Therefore, replacement of these tests with bench tests is relevant.

Empirical dependences connecting the load with deformation of the soil body were also used in the study of loads acting on the operating devices of road construction machinery [14]. However, application of such an approach has a number of significant drawbacks. These dependences suggest a uniaxial loading of the soil body. The use of dependences requires very rough schematization of the construction site surface, often as a set of individual surface sections contacting with the soil and consider the problem as a two-dimensional problem. In this case, deformation of the soil of each section is considered independently. It is obvious that such a description of the bearing surface of a high-speed machine track with a complicated lug form is very approximate. When machine is moving, there is a simultaneous immersion of the track into the soil and moving in a transverse direction accompanied by large deformations, soil raking by the side surface and the like. For an adequate description of these processes, it is necessary to consider the complex stress-strain state of the soil volume taking into account its nonlinear properties and destruction.

Recently, studies have applied a 3D measurement and recording with the help of modern video devices [15]. This method is used by Komatsu Ltd. It is a powerful tool for “visualizing” the current situation that can “stitch” the sequence of processes. The high cost of specialized equipment impedes dissemination of this type of study.

Experimental and theoretical studies of loading of rubber-metal track joints of high-speed machines were conducted during full-scale run tests. Stressed and thermal state of the joints was studied using a specially designed bench. Procedures for calculating durability of joints according to the criteria of fatigue and thermomechanical destruction were developed based on the obtained data [16, 17] but it was not shown how this affects reliability of the machine in general.

Insufficient number of benches for wear testing of the working bodies of construction machines in heavy operating conditions restrains works to improve structures of the machines themselves. It is necessary to create methods and equipment for testing wear, reliability and durability of parts and make necessary theoretical substantiation of choosing forced testing conditions [18].

Thus, the problems of creating scientific foundations of the theory of increasing reliability and service life of construction machines, raising their techno-economic level, reducing time of their design and implementation, the principles of organization and scientific basis of comprehensive tests have not been solved until now.

3. The aim and objectives of the study

This study objective was to improve reliability and service life of construction machines taking into account the random processes that occur during their operation through development of methods and procedures for accelerated bench tests. The methods should make it possible to systematize a set of steps which must be performed using the proposed algorithm of interrelation of statistical loading characteristics based on normal distribution during bench tests of construction machines. The procedures should solve concrete problems with the choice of criteria and mathematical characteristics taking into account spectral density of distribution of amplitudes of disturbing forces with a subsequent reproduction of the equivalent loads on the test bench.

The objective was realized by solving the following main tasks:

– to perform analysis of operating and bench loading conditions;

– to determine characteristics of equivalent bench loading conditions for various forcing methods;

– to determine approximate dependences for estimating the acceleration coefficients and a ratio for determining the number and duration of accelerated bench tests;

– to develop bench designs for accelerated testing of construction machines.

4. Analytical search for, and substantiation of the adequacy of loading conditions during bench and full-scale tests

Compliance of the results of accelerated tests with the operation data is determined in the first place by the type and nature of the failures, the ratio of mean time between failures to the life of the tested structure on the bench and in operation. In order to ensure that the reliability characteristics obtained on the stand and in service meet requirements, it is necessary to establish proper loading conditions. At the same time, various methods of test acceleration and, consequently, loading conditions are possible [19].

The most correct results are obtained in bench tests at operation loading conditions, with the test acceleration achieved by reducing breaks in the work and improvement of shift planning. Such tests are suitable for machines operated rarely during the year, i.e. the machines for seasonal operation. However, this test method gives a small acceleration for some types of earth moving machines operated round the year.

Somewhat higher coefficient of acceleration K_f is achieved by increasing the loading frequency when tested on a bench with preservation of loading amplitudes. However, in full-scale tests, frequency is limited because of a sharp growth of inertial loads. Accordingly, the nature of stressed state of metal structures and the failure nature change. When the loading level is forced, numerous variants of burden stepping up are possible:

– to increase amplitudes at unchanging average loading levels;

– to increase the average level at an unchanging amplitude;

– to increase amplitudes and average loading values.

If the ratio of the loading amplitudes K_a at the stand and under operation conditions is $P_{ac}/P_{ae}=K_a$ and the ratio of the average loading values is $P_{cc}/P_{ce}=K_c$, then the ratio of the loading process dispersions D at the stand and in operation can be represented for the above-mentioned forcing methods in the form:

$$D_e = \sum\left(P_{ai} - P_c\right)^2;$$

$$D_c = \sum\left(K_a P_{ai} - P_c\right)^2 = \sum K_a{}^2 P_{ai}{}^2 - 2K_a \sum P_{ai} P_c + \sum P_c{}^2;$$

$$D_f = \frac{D_c}{D_e} = \frac{\left(K_a^2 - 1\right)\sum P_{ai}^2}{D_e} - \frac{2\left(K_a - 1\right)\sum P_{ai} P_c}{D_e} + 1. \qquad (1)$$

$$D_e = \sum\left(P_{ai} - P_c\right)^2;$$

$$D_c = \sum\left(K_a P_{ai} - P_c\right)^2 = \sum K_a{}^2 P_{ai}{}^2 - 2K_a \sum P_{ai} P_c + \sum P_c{}^2;$$

$$K_f = \frac{D_c}{D_e} = \frac{\left(K_a^2 - 1\right)\sum P_{ai}^2}{D_e} - \frac{2\left(K_a - 1\right)\sum P_{ai} P_c}{D_e} + 1. \qquad (2)$$

$$D_e = \sum\left(P_{ai} - P_c\right)^2;$$

$$D_c = \sum\left(K_a P_{ai} - K_c P_c\right)^2 =$$

$$= K_a^2 \sum P_{ai}{}^2 - 2\left(K_a K_c - 1\right)\sum P_{ai} P_c + \sum P_c{}^2 K_c^2;$$

$$K_f = \frac{D_c}{D_e} = \frac{\left(K_a^2 - 1\right)\sum P_{ai}^2}{D_e} -$$

$$- \frac{2\left(K_a K_c - 1\right)\sum P_{ai} P_c}{D_e} + \frac{\left(K_c^2 - 1\right)\sum P_c^2}{D_e} + 1. \qquad (3)$$

When the load levels are forced by these methods, the cycle asymmetry coefficients r can be determined for appropriate variants as follows:

$$r_e = \frac{\sigma_{e\min}}{\sigma_{e\max}}; \quad r_c = \frac{K_a \sigma_{e\min}}{K_a \sigma_{e\max}} = r_e.$$

$$r_c = \frac{\left(K_c \sigma_c - \sigma_a\right)}{\left(K_c \sigma_c + \sigma_a\right)} \text{ with increase in } K_c \ - \ \eta \to 1. \qquad (4)$$

$$r_c = \frac{\left(K_c \sigma_c - K_a \sigma_a\right)}{\left(K_c \sigma_c + K_a \sigma_a\right)}$$

at

$$K_c = K_a = K, \ r_c = \frac{K\left(\sigma_c - \sigma_a\right)}{K\left(\sigma_c + \sigma_a\right)} = r_e.$$

$\sigma_{e\min}$ is the minimum stress in the structure, $\sigma_{e\max}$ is the maximum stress in the structure.

Thus, in the case of an increase in amplitudes at a constant value of the mean load level as well as at the same forcing of amplitudes and mean values, constancy of the cycle asymmetry coefficient remains. Therefore, such variants of forcing prevail.

The considered methods of accelerated tests are given in Table 1.

In the first case (in the case of unchanged loading conditions), loading conditions are obtained by reproducing the record of the operating conditions or by forming the loading conditions equivalent to the operational ones. If stresses above $0.5\sigma_{-1}$ appear, formation of conditions occurs with the cut-off of lower loads.

In the second case (forcing by amplitude), the loading process on the bench is reproduced with the same frequencies as loading in operation. In this case, for all types of tests with harmonic immersion, amplitude is increased, the action increase in K_a times, that is $A_{ci}=K_a A_{ei}$. When using a random loading process at the bench, the relationship between the spectral density of processes is expressed by dependence

$$f\left(\omega_c\right) = K_f f\left(\omega_e\right),$$

where

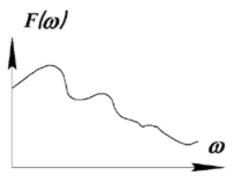
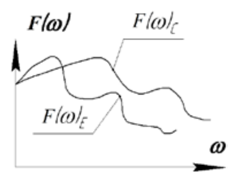
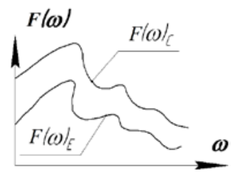
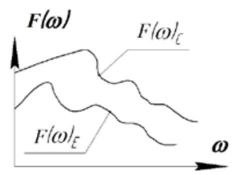
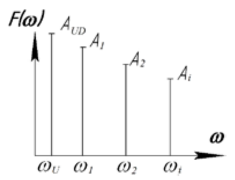
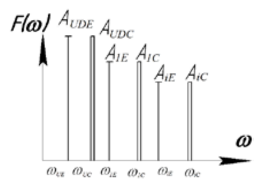
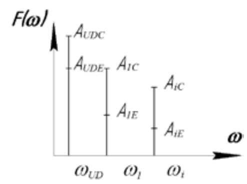
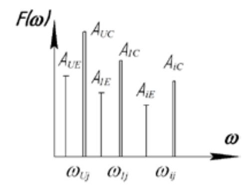
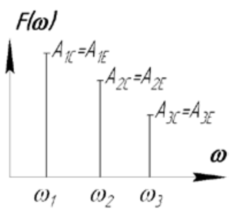
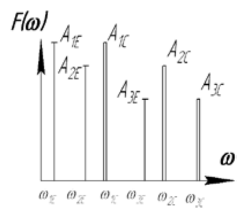
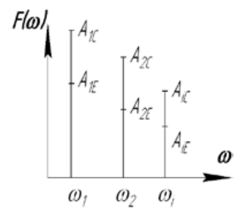
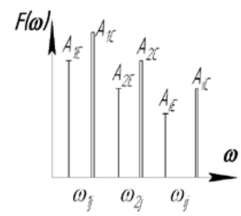
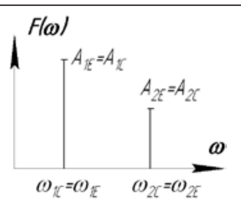
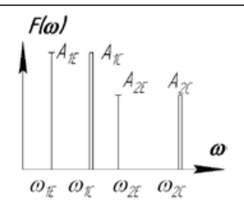
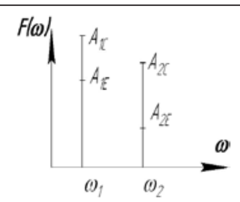
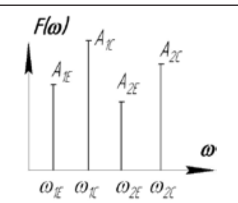
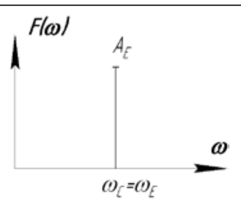
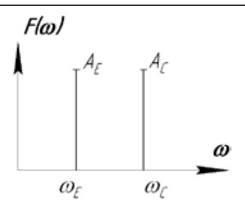
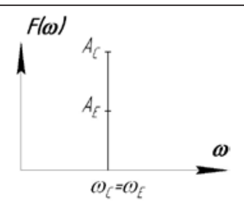
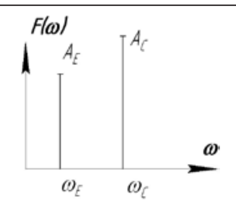
$$K_f = \frac{\sigma_{ac}^2}{\sigma_{ae}^2}. \qquad (5)$$

In the third case (forcing by frequency), loading on the bench is performed at higher frequencies compared with loading in operation. An increase in frequency performed for all characteristic actions is one and the same, that is, the loading conditions shift to the region of higher frequency

$$\omega_{ic} = K_\omega \omega_{ie}.$$

Table 1

Relationship between bench and operating loading conditions. Types of the loading process in operation and at the bench

Item No.	1	2	3	4
random loading process				
multifrequency loading with periodic impacts				
multifrequency (polyharmonic) loading				
double-frequency (biharmonic) loading				
monofrequency (harmonic) loading				
Correlation between amplitudes and frequencies of loading in operation and on the bench	$A_E=A_C$; $\omega_E=\omega_C$ acceleration due to continuous operation (increase in the utilization factor).	$A_E=A_C$; $\omega_E<\omega_C$ forcing in loading frequency	$A_E=A_C$; $\omega_E=\omega_C$ forcing in loading level.	$A_E=A_C$; $\omega_E=\omega_C$ forcing in loading level and loading frequency

For the random process of loading, when describing the correlation function and the spectral density with various expressions, the interrelations between the correlation coefficients determined for the tests in operation conditions and on the bench are given below.

The interrelations of characteristics of the processes in accelerated tests by frequency forcing for the random loading process:

1. Correlation function

$$R(\tau)=R(0)e^{-\alpha(\tau)}\cos\beta\tau,$$

spectral density

$$F(w)=R(0)\frac{2\alpha\left(\alpha^2+\beta^2+w^2\right)}{w^4+2\left(\alpha^2-\beta^2\right)w^2+\left(\alpha^2+\beta^2\right)^2},$$

interrelations between the correlation coefficients

$$B_c=Kw\beta e,$$

$$\alpha_c=\frac{\alpha_e^2+\beta_e^2}{2\alpha_e}-\sqrt{\left(\frac{\alpha_e^2+\beta_e^2}{2\alpha_e}\right)-K^2w\beta_e^2};$$

2. Correlation function

$$R(\tau)=R(0)e^{-\alpha(\tau)}\sin\beta\tau,$$

spectral density

$$F(w)=R(0)\frac{2\beta\left(\alpha^2+\beta^2+w^2\right)}{w^4+2\left(\alpha^2-\beta^2\right)w^2+\left(\alpha^2+\beta^2\right)^2},$$

interrelations between the correlation coefficients

$$B_c=Kw\beta e,$$

$$\alpha_c=\sqrt{Kw\left(\alpha_e^2+\beta_e^2\right)-K_w^2\beta_e^2}.$$

3. Correlation function

$$R(\tau)=R(0)e^{-\alpha(\tau)}(\cos\beta\tau+\sin\beta\tau),$$

spectral density

$$F(w)=R(0)\frac{2(\alpha+\beta)\left(\alpha^2+\beta^2+w^2\right)}{w^4+2\left(\alpha^2-\beta^2\right)w^2+\left(\alpha^2+\beta^2\right)^2},$$

interrelations between the correlation coefficients

$$B_c=Kw\beta e,$$

$$\alpha_c=\frac{\alpha_e^2+\beta_e^2}{2(\alpha_e+\beta_e)}-\sqrt{\left(\frac{\alpha_e^2+\beta_e^2}{2(\alpha_e+\beta_e)}\right)+\frac{Kw\beta_e\left(\alpha_e^2+\beta_e^2\right)}{\alpha_e+\beta_e}-K_w^2\beta_e^2}.$$

4. Correlation function

$$R(\tau)=R(0)\times\left[\sum_{i=1}^{n}a_ie^{-\alpha(\tau)}\cos\beta_i\tau+\sum_{i=1}^{m}a_ie^{-\alpha(\tau)}\sin\beta_i\tau\right],$$

spectral density

$$F(w)=R(0)\times\left[2\sum_{i=1}^{n}\frac{a_i\alpha_i\left(\alpha_i^2+\beta_i^2+w^2\right)}{w^4+2\left(\alpha^2-\beta^2\right)w^2+\left(\alpha^2+\beta^2\right)^2}+2\sum_{i=1}^{m}\frac{a_i\beta_i\left(\alpha_i^2+\beta_i^2+w^2\right)}{w^4+2\left(\alpha^2-\beta^2\right)w^2+\left(\alpha^2+\beta^2\right)^2}\right],$$

interrelations between the correlation coefficients

$$B_{ic}=K_w\beta_{ie},$$

$$\alpha_c=\frac{a_{ci}\left(\alpha_{ei}^2+\beta_{ei}^2\right)}{2a_{ei}\left(\alpha_{ei}+\beta_{ei}\right)}-$$

$$-\left\{\left[\frac{a_{ci}\left(\alpha_{ei}^2+\beta_{ei}^2\right)}{2a_{ei}\left(\alpha_{ei}+\beta_{ei}\right)}\right]^2+\frac{K_wa_{ci}\beta_{ei}\left(\alpha_{ei}^2+\beta_{ei}^2\right)}{a_{ei}\left(\alpha_{ei}+\beta_{ei}\right)}-K_w^2\beta_{ei}^2\right\}^{1/2}.$$

Determination of the correlation coefficients for the accelerated test conditions was carried out under condition $f(0)c=f(0)$, that is, at a constant value of dispersion of the loading processes in operation and at the bench.

In the fourth case (forcing both in amplitude and frequency) the loading process is performed with an increase in amplitude in K_a times ($A_{ci}=K_aA_{ei}$) and frequency in K_ω times ($\omega_{si}=K_\omega\omega_{ei}$). Interrelations between the characteristics of processes at acceleration of tests by forcing in frequency and amplitude:

1. Correlation function

$$R(\tau)_e=R(0)_e\,e^{-\alpha_e(\tau)}\cos\beta_e\tau,$$

$$R(\tau)_e=K_f(0)\;e^{-\alpha_c(\tau)}\cos\beta\;\tau,$$

spectral density

$$F(w)_e=R(0)_e\frac{2\alpha_e\left(\alpha_e^2+\beta_e^2+w^2\right)}{w^4+2\left(\alpha_e^2-\beta_e^2\right)w^2+\left(\alpha_e^2+\beta_e^2\right)^2},$$

$$F(w)_e=K_fK(0)\;\frac{2\alpha_c\left(\alpha_c^2+\beta_c^2+w^2\right)}{w^4+2\left(\alpha_e^2-\beta_e^2\right)w^2+\left(\alpha_e^2+\beta_e^2\right)^2},$$

interrelations between the correlation coefficients

$$\beta_c=K_w\beta_e,$$

$$\alpha_c=\frac{\alpha_e^2+\beta_e^2}{2\alpha_eK_f}-\sqrt{\left(\frac{\alpha_e^2+\beta_e^2}{2\alpha_eK_f}\right)-K_w^2\beta_e^2}.$$

2. Correlation function

$$R(\tau)_e=R(0)_e\,e^{-\alpha_e(\tau)}\sin\beta_e\tau,$$

$$R(\tau)_c=K_f(0)_e\,e^{-\alpha_c(\tau)}\sin\beta_c\tau,$$

spectral density

$$F(w)_e=R(0)_e\frac{2\beta_e\left(\alpha_e^2+\beta_e^2+w^2\right)}{w^4+2\left(\alpha_e^2-\beta_e^2\right)w^2+\left(\alpha_e^2+\beta_e^2\right)^2},$$

$$F(w)_e=K_fR(0)\;\frac{2\beta_c\left(\alpha_c^2+\beta_c^2+w^2\right)}{w^4+2\left(\alpha_c^2-\beta_c^2\right)w^2+\left(\alpha_c^2+\beta_c^2\right)^2},$$

interrelations between the correlation coefficients

$$\beta_c=K_w\beta_e,$$

$$\alpha_c=\sqrt{\frac{K_w}{K_f}\left(\alpha_e^2+\beta_e^2\right)-K_w^2\beta_e^2}.$$

3. Correlation function

$$R(\tau)_e=R(0)_e\,e^{-\alpha_e(\tau)}(\sin\beta_e\tau+\cos\beta_e\tau),$$

$$R(\tau)_c=K_f(0)_e\,e^{-\alpha_c(\tau)}(\sin\beta_c\tau+\cos\beta_c\tau),$$

spectral density

$$F(w)_e=R(0)_e\frac{2(\alpha_e+\beta_e)\left(\alpha_e^2+\beta_e^2+w^2\right)}{w^4+2\left(\alpha_e^2-\beta_e^2\right)w^2+\left(\alpha_e^2+\beta_e^2\right)^2},$$

$$F(w)_c=K_fR(0)_e\frac{2(\alpha_c+\beta_c)\left(\alpha_c^2+\beta_c^2+w^2\right)}{w^4+2\left(\alpha_c^2-\beta_c^2\right)w^2+\left(\alpha_c^2+\beta_c^2\right)^2},$$

interrelations between the correlation coefficients

$$\beta_c=K_w\beta_e,$$

$$\alpha_c=\frac{\alpha_e^2+\beta_e^2}{2K_f(\alpha_e+\beta_e)}-$$

$$-\left\{\left[\frac{\alpha_e^2+\beta_e^2}{2K_f(\alpha_e+\beta_e)}\right]^2+\frac{K_w}{K_f}\beta_e\frac{\alpha_e^2+\beta_e^2}{\alpha_e+\beta_e}-K_w^2\beta_e^2\right\}^{1/2}.$$

4. Correlation function

$$R(\tau)_e = R(0)_e \times \left[\sum_{i-1}^{n} a_{ie} e^{-\alpha_e(\tau)} \cos\beta_{ie}\tau + \sum_{i-1}^{m} a_{ie} e^{-\alpha_{ie}(\tau)} \sin\beta_{ie}(\tau) \right],$$

$$R(\tau)_c = R(0)_e K_f \times$$
$$\times \left[\sum_{i-1}^{n} a_{ic} e^{-\alpha_{ic}(\tau)} \cos\beta_{ic}\tau + \sum_{i-1}^{m} a_{ic} e^{-\alpha_{ic}(\tau)} \sin\beta_{ic}(\tau) \right],$$

spectral density

$$F(w)_e = 2R(0)_e =$$
$$= \left[\sum_{i-1}^{n} \frac{a_{ie}\alpha_{ie}\left(\alpha_{ie}+\beta_{ie}^2+w^2\right)}{w^4+2\left(\alpha_{ie}^2-\beta_{ie}^2\right)w^2+\left(\alpha_{ie}^2+\beta_{ie}^2\right)^2} + \sum_{i-1}^{m} \frac{a_{ie}\beta_{ie}\left(\alpha_{ie}^2+\beta_{ie}^2+w^2\right)}{w^4+2\left(\alpha_{ie}^2-\beta_{ie}^2\right)w^2+\left(\alpha_{ie}^2+\beta_{ie}^2\right)^2} \right],$$

$$F_c(w) = 2K_f R(0)_e =$$
$$= \left[\sum_{i-1}^{n} \frac{a_{ic}\alpha_{ic}\left(\alpha_{ic}+\beta_{ic}^2+w^2\right)}{w^4+2\left(\alpha_{ic}^2-\beta_{ic}^2\right)w^2+\left(\alpha_{ic}^2+\beta_{ic}^2\right)^2} + \sum_{i-1}^{m} \frac{a_{ic}\beta_{ic}\left(\alpha_{ic}^2+\beta_{ic}^2+w^2\right)}{w^4+2\left(\alpha_{ic}^2-\beta_{ic}^2\right)w^2+\left(\alpha_{ic}^2+\beta_{ic}^2\right)^2} \right].$$

Interrelations between the correlation coefficients

$$\beta_{ic} = K_w \beta_{ie},$$

$$\alpha_{ic} = \frac{a_{ic}\left(\alpha_{ie}^2+\beta_{ie}^2\right)}{2a_{ie}K_f\left(\alpha_{ie}+\beta_{ie}\right)} -$$
$$- \left\{ \left[\frac{a_{ic}\left(\alpha_{ie}^2+\beta_{ie}^2\right)}{2a_{ie}K_f\left(\alpha_{ie}+\beta_{ie}\right)} \right]^2 + \frac{K_w a_{ic}}{K_f a_{ie}} \beta_{ie} \frac{\alpha_{ie}^2+\beta_{ie}^2}{\alpha_{ie}+\beta_{ie}} - K_w^2\beta_{ie}^2 \right\}^{1/2}.$$

The bench tests of construction machines at constant loads are carried out in order to check the static strength of parts and units of machines in operation conditions. It is advisable to take into account possible overloads of static nature and the transformation of loads by the oscillational system of the machine.

The tests at constant loads are performed both for individual units and the machine in general. If the machine in general or only an operating device is tested, constant loads should be taken equal to the design values of the cutting forces P_{riz}, N_{riz} or digging forces $P_{roz.kop}$, $N_{roz.kop}$. The values of P_{riz}, N_{riz} are determined in accordance with [20, 21] by the following equality:

a) for sharp cutting elements:

$$P_{riz} = \overline{P_{0kn}}; \quad (6)$$

$$N_{riz} = \overline{N_{0kn}}; \quad (7)$$

b) for worn or dented cutting elements:

$$P_{riz} = \overline{P_{0kn}} + P_{pl.zn}; \quad (8)$$

$$N_{riz} = \overline{N_{kn}} - N_{pl.zn}, \quad (9)$$

where $\overline{P_0}$, $\overline{N_0}$ are mean values of the tangential and normal cutting forces for sharp cutting elements; $P_{pl.zn}$, $N_{pl.zn}$ are additional tangential and normal cutting forces associated with wear of cutting elements; k_n is the coefficient of overload taking into account the random nature of cutting forces.

Calculated values of digging forces can be found from the calculated values of cutting forces by means of relations:

$$P_{roz.cop} = P_{riz.} + P\partial;$$

$$N_{roz.cop} = N_{riz.} + N\partial,$$

where $P\partial$, $N\partial$ are additional resistance forces caused by friction, displacement of the dragging prism, filling of buckets with soil, etc. The forces $\overline{P_0}$, $\overline{N_0}$, $P_{pl.zn}$, $N_{pl.zn}$, $P\partial$ and $N\partial$ can be determined by the procedure given in [22] at the depth of cutting corresponding to a full use of the machine power. The overload coefficient k_n can be determined from the nomogram given in [23] depending on the coefficient of loading variation W_q=W_p and the conditional number of loading cycles, N_{um}. The values W_p and N_{um} are determined by the formulas:

$$W_p = W_{p0}\left(\frac{500}{F}\right)^r; \quad (10)$$

$$N_{um} = 1{,}44\overline{n_0}k_n t, \quad (11)$$

where W_{p0} is the value of variation coefficient for the cutoff section area 400…600 cm; F is the actual area of the cut section in cm; r is dimensionless exponent characterizing the degree of influence of the area F on the coefficient of variation W_p and is determined from Table 2; $\overline{n_0}$ is the average frequency of cutting force oscillations determined by the formula:

$$\overline{n_0} = k_4 \frac{v}{h},$$

where k_4 is the coefficient of frequency depending on the soil category; t is actual time corresponding to the specified time of failure-free operation of the machine or working body; K_p is the coefficient of operation conditions taken equal to one in the absence of periodic changes of the cutting forces, and if such changes take place,

$$K_p = 0{,}483\sqrt{W_p}. \quad (12)$$

Table 2

Value of the indicator r characterizing influence of the cutoff section area on the magnitude of the coefficient of the cutting force variation

Soil category	I	II	III	IV	V	VI	VII	VIII
r	0.38	0.36	0.34	0.31	0.28	0.25	0.22	0.20

When testing multi-bucket excavators, the calculation forces obtained by this method should be applied to all buckets or the working body cutters. When testing the dump machines, the total calculated force should be distributed equally between several sections of the cutting blade edge.

If individual assemblies or elements of metal structures of the machine are tested, then instead of calculated values of the cutting forces, it is necessary to determine the calculated value of some load q induced by them and proportional to the actual or reduced stress in the element. There can be the following two main cases:

1) elastic oscillations of the machine in the process of its operation are practically absent;

2) elastic oscillations of the machine are significant and should be taken into account in the calculation.

In the first case, the load q from the cutting forces is a linear function of the tangent components of these forces

$$q=\sum_{i=1}^{z}C_{1i}P_{ai}+\sum_{i=1}^{z}C_{2i}P_{pl.zn.}, \quad (13)$$

where C_{1i}, C_{2i} are some constant coefficients determined by static calculation; Z is the total number of cutting elements or sections into which the cutting edge is divided.

Assuming the individual cutting forces are statistically independent, we have the following equality of dispersion G_q^2, coefficient of variation W_p and the mean oscillation frequency $\overline{n_{0q}}$ of loading

$$G_q^{\;2}=\sum_{i=1}^{z}C_{1i}^{\;2}G_{pi}^{\;\;2}; \quad (14)$$

$$W_q=\sqrt{\sum_{i=1}^{z}E_i^{\;2}W_{pi}^{\;2}}; \quad (15)$$

$$\overline{n_{0q}}=\frac{1}{G_q}\sqrt{\sum_{i=1}^{z}C_{1i}^{\;2}G_{pi}^{\;2}\overline{n_{0i}^{\;2}}}=\sqrt{\sum_{i=1}^{z}E_i^{\;2}W_{pi}^{\;2}\overline{n_{0i}^{\;2}}}\,/\,W_q, \quad (16)$$

where

$$E_i=C_{1i}\overline{P_{0i}}\,/\,\overline{q_0}, \quad (17)$$

$$\overline{q_0}=\sum_{i=1}^{z}C_{1i}\overline{P_{0i}}. \quad (18)$$

If individual cutting forces are correlated with each other, the formulas for G_q^2, W_p and $\overline{n_{0q}}$ take the following form:

$$G_q^{\;2}=\sum_{i=1}^{z}C_{1i}^{\;2}G_{pi}^{\;2}+2\sum_{i\lg}rij(0)C_{i1}C_{1j}G_{pi}G_{pj}; \quad (19)$$

$$W_q=\frac{G_q}{\overline{q_0}}; \quad (20)$$

$$n_0=\frac{1}{G_q}\sqrt{\sum_{i=1}^{z}C_{1i}^{\;2}G_{pi}^{\;2}\overline{n_{0i}^{\;2}}-\frac{1}{2\pi^2}\sum_{i\langle j}r_{ij}''(0)C_{1i}C_{1j}G_{pi}G_{pj}}, \quad (21)$$

where $r_{ij}(0)$ is the mutual normalized correlation function $r_{ij}(\tau)$ between the i-th and j-th forces at $\tau=0$; $r_{ij}''(0)$ is the second derivative of the function $r_{ij}(\tau)$ in τ at $\tau=0$.

In dump-type machines, the cutting force P_i is a system of correlated random functions with the same statistical characteristics $\overline{P_{0i}}=\overline{P_{01}}$, $\overline{G_{pi}}=\overline{G_{p1}}$, $n_{0i}=n_{01}$, $S_{pi}(\tau)=S_1(\tau)$ and mutual normalized correlation functions

$$r_{ij}(\tau)=e^{-\beta(i=j)}S_{p1}(\tau). \quad (22)$$

Also, taking into account that

$$S_{p1}''(0)=-4\pi^2\overline{n_{01}}^{\,2} \quad [23],$$

it can be found by substitution of the indicated equality in formulas (19) to (21) that for dump-type machines

$$G_q^{\;2}=G_{p1}^{\;\;2}\left(\sum_{i=1}^{z}C_{1i}^{\;2}+2\sum_{i\langle j}e^{-\beta}C_{1i}C_{ij}\right); \quad (23)$$

$$W_q=\frac{G_q}{\overline{q_0}}; \quad (24)$$

$$\overline{n_0}=\overline{n_{01}}. \quad (25)$$

The final stage of determining the calculated value of the load q does not differ practically from the definition of the calculated values of the digging forces. Estimated load from cutting forces is determined by the formulas:

a) for sharp cutting elements

$$q_{roz.}=\overline{q_0}k_n; \quad (26)$$

b) for cutting elements with a wearing area

$$q_{roz.}=\overline{q_0}k_n+q_{pl.zn.}, \quad (27)$$

where $\overline{q_0}$ is the mean value of the load from the cutting forces for sharp cutting elements;

$$q_{pl.zn.}=\sum_{i=1}^{z}C_{2i}P_{pl.zn.i}$$

is the load from additional forces associated with wear of the cutting elements (it can be positive or negative).

The overload coefficient K_{Per} is determined by the nomogram from [24] depending on the coefficient of variation W_q and the conditional number of load cycles N_{um} which is determined by formulas (10), (11) with substitution of $\overline{n_{0q}}$ for $\overline{n_0}$ and W_q for W_p.

Total calculated load from the digging forces

$$q_{roz.cop}=q_{riz.}+q_{\partial}, \quad (28)$$

where q_{∂} is the additional load caused by friction, moving of the dragging prism, filling buckets with soil, etc.

In the second case, when operation of the machine is characterized by significant elastic oscillations, the calculated load $q_{roz.cop}$ can be found based on the periodic component and the spectral density $G_q(\omega)$ of the dynamic load by the following scheme:

1) the dispersion D_q, the variation coefficient and the average frequency of the loading oscillations $q(t)$ are calculated by means of equality

$$D_q=G_q^{\;2}=\int_0^{\infty}G_q(\omega)\mathrm{d}\omega, \quad (29)$$

$$W_q=\frac{G_q}{\overline{q}_{\max}}; \quad (30)$$

$$\overline{n_{0q}^{\;2}}=\frac{1}{4\pi^2D_q}\int_0^{\infty}\omega^2G_q(\omega)\mathrm{d}\omega; \quad (31)$$

2) in the assumption of a normal distribution of dynamic load, the coefficient of overload is calculated

$$K_M=1+W_q\sqrt{2e_n(1{,}44K_p\overline{n_{0q}}t)}; \quad (32)$$

where K_p is the coefficient of operation conditions in the presence of periodic component of the load determined by the expression

$$K_p = 0{,}483\sqrt{W_q}; \tag{33}$$

3) the calculated load from the digging forces is determined by formula:

$$q_{roz.cop.} = \overline{q_{\max}} K_M. \tag{34}$$

Beside the tests for constant loads, construction machines and working bodies should be subjected to special tests for impact loads. In the absence of such tests, the maximum horizontal load $P\partial_{\max}$ on the working body should be calculated. It arises from the impact of the machine on an insuperable obstacle. The obstacle has a given stiffness n_f. Forces $R_{riz.cop}$. and $N_{riz.cop}$ are taken as constant loads if $P_{\text{riz.cop.}} \rangle\rangle P\partial_{\max}$, or as a force $P\partial_{\max}$, if $P_{\text{riz.cop.}} \langle P\partial_{\max}$.

At present, the predominant form of mechanical failure in building machines is the fatigue breakdown of their elements [5, 6].

Current means of calculating fatigue life from random loads [25–27] do not take into account the cycles of variable loading. The existing means of cyclization are as follows: calculation of distribution of realization extrema; distribution of excesses of the specified level; amplitude distribution; construction of a common distribution of amplitudes and mean values of the “cycle”, etc. Depending on the cyclization means, the estimated life values for some loading types can vary tenfold.

An unambiguous solution to the problem of determining equivalent loads will provide use of the hypothesis of spectral summation [28] applied to life calculation of the working elements of construction machines. According to this hypothesis:

– damage from fatigue caused by each of the harmonious components of the total load does not depend on the presence of other components;

– combined action of several harmonics brings about the fatigue damage equal to the sum of damages from fatigue caused by each of the harmonics separately.

Consequently, the actual random load in calculations can be replaced by an equivalent harmonic or polyharmonic load. In this case, the following four typical cases may occur.

1. The load $q(t)$ represents a random narrow-band process. The spectral density $Gq(t)$ of this load close to one of the frequencies ω_1 has a sharp surge. It corresponds to the overwhelming part of the area between the graph of the function $Gq(\omega)$ and the abscissa (Fig. 1).

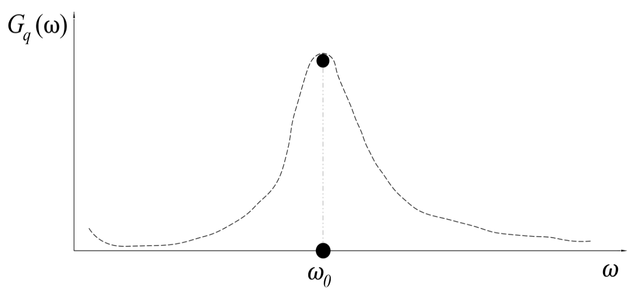


Fig. 1. Example of the spectral density of a narrow-band random process

The above random load $q(t)$ can be replaced by a harmonic load with the same average value $\overline{q}$, with a frequency ω_1 and an amplitude [30]

$$q_a = \sigma_q\sqrt{2}\cdot\sqrt[m]{\Gamma\left(\frac{m}{2}+1\right)}, \tag{35}$$

where G_q is rms deviation of the load; Γ is the Euler gamma function; m is an exponent of the fatigue curve.

2. The load $q(t)$ is the sum n of random narrow-band processes with frequencies ω_1, ω_2,.., ω_n. In this case, its spectral density $G_q(\omega)$ has n sharp surges at frequencies ω_1, ω_2,.., ω_n while the zones of these surges cover the prevailing part of the area between the graph of the function $G_q(\omega)$ and the abscissa.

By analogy with the previous case, the input random load in the calculations can be represented as a polyharmonic load with its constant component equal to $\overline{q}$, and the individual harmonic components have frequencies ω_1, ω_2,.., ω_n and amplitudes

$$q_{ai} = \sigma_{qi}\sqrt{2}\cdot\sqrt[m]{\Gamma\left(\frac{m}{2}+1\right)}, \tag{36}$$

where $\sigma_{qi} = \sqrt{D_{qi}}$; D_{qi} is the part of the load dispersion $q(t)$ which falls on the frequency ω_i.

However, the load under consideration can be reduced to a single frequency ω_0 if the hypothesis of spectral summation is used [28].

In this case, according to this hypothesis, the equivalent harmonic load at a mean value $\overline{q}$ and frequency ω_0 will have amplitude

$$q_{ekv} = \sqrt{\sum_{i=1}^{n} q_{ai}^2\left(\frac{\omega_i}{\omega_0}\right)^{2/m}} \tag{37}$$

or

$$q_{ekv} = \sqrt[m]{\Gamma\left(\frac{m}{2}+1\right)}\sqrt{2\sum_{i=1}^{n}\sigma_{qi}^2\left(\frac{\omega_i}{\omega_0}\right)^{2/m}}. \tag{38}$$

The frequency ω_0 can be chosen arbitrarily. In particular, it can coincide with one of the frequencies ω_1, ω_2,.., ω_n.

3. The load $q(t)$ has a wide spectrum of frequencies and does not contain frequencies to which a significant part of the overall dispersion of the process would correspond (Fig. 2).

In this case, according to the hypothesis of spectral summation, the random load $q(t)$, can be replaced by a harmonic load having a mean value $\overline{q}$, arbitrarily taken cyclic frequency ω and amplitude

$$q_{ekv} = \sigma_a\sqrt{2}\cdot\sqrt[m]{\frac{\omega_{ekv}}{\omega}\Gamma\left(\frac{m}{2}+1\right)}, \tag{39}$$

where

$$\omega_{ekv} = \left[\int_0^{\infty} g_q(\omega)\omega^{2/m}\mathrm{d}\omega\right]^{m/2}, \tag{40}$$

where

$$g_q(\omega) = \frac{G_q(\omega)}{D_q}$$

is the normalized spectral loading density.

The cyclic frequency ω is recommended to be taken proceeding from the condition

$$\omega \approx \omega_{0q} = 2\pi\overline{n}_{0q},$$

where $\overline{n}_{0q}$ is the average frequency of the load fluctuations.

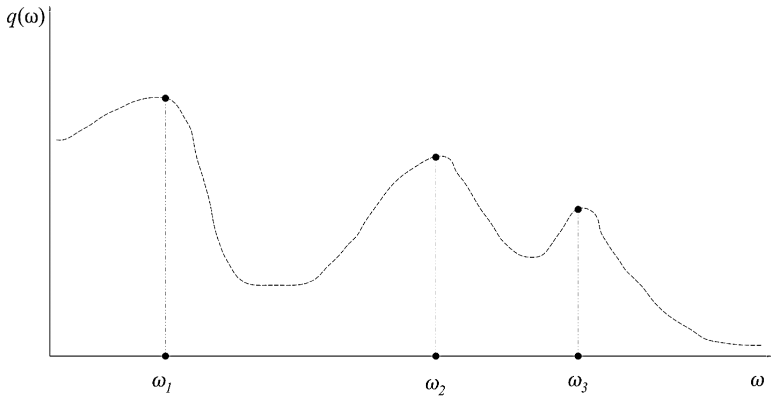


Fig. 2. Example of a broadband random process

Taking into account that parameters of the fatigue curves can vary slightly at a high frequency difference, it is advisable to replace the random actual load with not a harmonic but a polyharmonic load consisting of 2 to 4 harmonics. As cyclic frequencies for these harmonics, one can choose frequencies ω_i that correspond to separate maxima of the spectral density $G_q(\omega)$ or one or another frequency in the region of large ordinates of the $G_q(\omega)$ function. Then the amplitudes of the individual harmonics will be determined by equality

$$q_{ekv} = \sigma_a \sqrt{2} \cdot \sqrt[m]{\frac{\omega_{ekv}}{\omega} \Gamma\left(\frac{m}{2}+1\right)}, \tag{41}$$

$$\omega_{ekv} = \left[\frac{1}{D_{gi}} \int_{\omega_{\min_i}}^{\omega_{\max_i}} G_q(\omega) \omega^{2/m} \mathrm{d}\omega \right]^{m/2}, \tag{42}$$

where

$$\sigma_{qi} = \sqrt{D_{qi}}; \tag{43}$$

$$D_{qi} = \int_{\omega_{\min_i}}^{\omega_{\max_i}} G_q(\omega) \mathrm{d}\omega, \tag{44}$$

where $\omega_{\min i}$ for i=1 is equal to 0 and for other values, i $\omega_{\min i}=0.5(\omega_{i-1}+\omega_i)$; $\omega_{\max i}$ for all values of i except the latter is equal to 0.5 $(\omega_{i-1}+\omega_i)$; for the last value,

$$i \to \omega_{\max_i} = \infty.$$

The essence of this means of equivalent conversion of the input load lies in a preliminary distribution of the random load $q(t)$ into several random components $q_i(t)$. The spectral density of each of them is equal to a certain section of the total spectral density $G_q(\omega)$ with a subsequent substitution of an equivalent harmonic component in accordance with the hypothesis of spectral summation for each of these random components.

4. Combination of polyharmonic and stationary random loads.

The polyharmonic load consists of n harmonic components with amplitudes $q_1, q_2, \ldots, q_n$ and frequencies $\omega_1, \omega_2, \ldots, \omega_n$ with significant differences. Assume that the amplitude q_1 is the largest. Then it is expedient to take frequency ω_1 as the base frequency.

The stationary component has a dispersion $D_q = \sigma_q^2$ and a normalized spectral density $G_q(\omega)$.

The polyharmonic load is replaced by a harmonic component whose frequency is equal to ω_1 and the amplitude

$$q_{ekv_1} = \left[\sum_{ii=1}^{n} \left(\frac{\omega_i}{\omega_1}\right)^{2/m} q_i^2 \right]^{1/2} = \\ = q_1 \sqrt{\sum_{i=1}^{n} \left(\frac{q_i}{q_1}\right)^2 \left(\frac{\omega_i}{\omega_1}\right)^{2/m}}. \tag{45}$$

The random stationary load is replaced by a harmonic component with frequency ω_1 and amplitude

$$q_{ekv} = \sigma_a \sqrt{2} \cdot \sqrt[m]{\frac{\omega_{ekv}}{\omega_1} \Gamma\left(\frac{m}{2}+1\right)}. \tag{46}$$

Next, replace the two obtained harmonic components with the same frequency ω_1 by one component with the same frequency ω_1 and the amplitude equal to

$$q_{ekv} = \sqrt{q_{ekv_1}^2 + q_{ekv_2}^2} = \\ = q_1 \sqrt{\sum_{i=1}^{n} \left(\frac{q_i}{q_1}\right)^2 \left(\frac{\omega_i}{\omega_1}\right)^{2/m} + 2\left(\frac{\sigma_q}{q_1}\right)^2 \left(\frac{\omega_{ekv}}{\omega_1}\right)^{2/m} k_{ekv}^2}, \tag{47}$$

where

$$k_{ekv} = \sqrt[m]{\Gamma\left(\frac{m}{2}+1\right)}.$$

Instead of using the constant component of the load, it is possible to change in calculations of $\overline{q}$ the amplitudes of the harmonic components q_{ekvi} according to formula

$$q'_{ekv_i} = q_{ekv_i} \left(1 + \frac{\varphi \cdot \overline{q}}{\sum_{i=1}^{n} q_{ekv_i}} \right), \tag{48}$$

where ϕ is the coefficient of metal sensitivity to the cycle asymmetry.

Thus, in life calculations, a real random load may be modeled to be equivalent to a harmonic or polyharmonic load. The loading frequency and amplitude can be chosen in three ways: by increasing the loading frequency, by increasing the loading amplitude or by increasing both of these factors simultaneously.

The frequency increase involves the following conditions:

– the assembly under study does not approach resonance;

– the effect of repeatedly changing loading frequencies on the fatigue fracture process is negligible.

When increasing the loading amplitude, it is necessary that the stresses in the construction do not exceed the permissible values.

Compliance with these conditions allows one to carry out fatigue calculations at various loading parameters by modeling various machine operation conditions.

In the process of digging with earth-moving machines or trenchers, resistance to rolling of wheels or movement of caterpillars takes place proportional to normal loading [20, 21] which corresponds to the expression

$$P_t = t(Q\cos\alpha + N_k), \tag{49}$$

where t is the dimensionless coefficient of the machine rolling resistance; Q is the weight of the machine; α is the angle of inclination of the rolling surface to the horizon; N_k is the additional normal reaction to the chassis arising from interaction of the working bodies of the machine with the soil.

Besides, there is a perturbing effect from irregularities of the support surface on the machine described by the random functions $q_1(t)$ and $q_2(t)$.

Though force P_t varies in the process of machine operation, it can be conventionally considered constant at a certain operation stage under a quasi-static loading of the working body. Consequently, when modeling loading of the machine chassis, it is necessary to simultaneously reproduce the constant loading caused by wheel rolling resistance and a random loading from the microscopic profile irregularities in order to approximate the loading conditions to the operation conditions. When designing test benches, such a complex loading is provided either by supporting racing drums or an endless band with irregularities formed on them. The value of the constant load is regulated by the braking devices installed on the drive shaft (a retarder brake, a hydraulic pump, etc.).

Reproduction of a random component from the microirregularities of the transport path profile is associated with great design constraints. Thus, it is necessary to provide the following measures:

– prior to the tests, analyze the section of the path to be reproduced on the test bench;

– determine dispersion of random microirregularities;

– calculate spectral density of distribution of the amplitudes of the disturbing force;

– reproduce on the bench a harmonic loading with an equivalent frequency determined from formula (42);

$$\omega_{ekv} = \left[\frac{2\alpha\sigma\partial}{\pi} \int_0^{\infty} \frac{\omega^{2/}}{(\alpha\sigma\partial)^2 + \omega^2} d\omega \right]^{m/2}. \tag{50}$$

Equivalent distances between irregularities on the band are determined from the relation

$$l = \frac{\sigma\partial}{\omega_{ekv}}. \tag{51}$$

Fig. 3 shows schematic view of the bench for bulldozer run tests. The band 5 has pads 14 attached with a space determined by formula (51). Mounting of the pads results in a growth of the dynamic load on the tractor 1 (Fig. 4) which makes it possible to accelerate the bench test.

In the process of machine manufacture, its internal parameters determining reliability indicators change according to the obtained information about unreliable design elements obtained from test and operation facilities. Improvement of design of both elements and the machine in general increases reliability and stabilizes the manufacturing process. As the manufacture process stabilizes, both the mean values of the failure flow and the dispersion of its distribution are reduced. Such variation of reliability indicators in time is typical for a rhythmic production. If the production of machines is not rhythmic (uneven), periodic deviations from the average long-term nature of the change in the indicators of all machine elements occur which in turn reduces reliability.

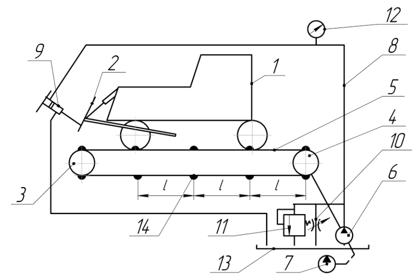


Fig. 3. Schematic view of the bench for bulldozer run tests

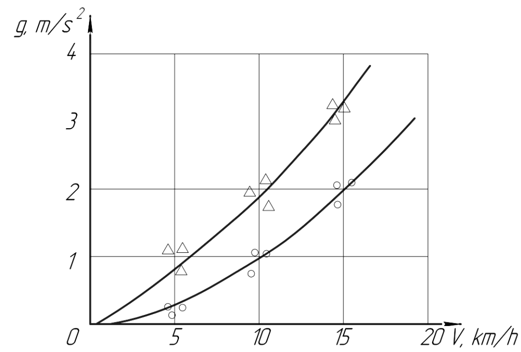


Fig. 4. Dynamic loading of the crawler machine at the bench: without pads (O); with pads (Δ)

5. Discussion of the results obtained in the study of accelerated bench tests of construction machinery drives

To determine equivalent load levels when testing machines and their units, it is necessary to determine the maximum limit of permissible forcing taking into account the type of load distribution, applying the law of normal distribution of loads and determining its frequency spectrum. It is desirable to replace polyharmonic loading and stationary random loading with an equivalent harmonic loading. Simulation of an equivalent loading can be done in three ways: by increasing the loading frequency, by increasing the loading amplitude or by increasing both of these factors simultaneously. In doing so, the following conditions must be observed:

1) do not approach resonance and fatigue breakdown with an increase in the loading frequency;

2) do not exceed the permissible stresses in the structures with an increase in the loading amplitude.

Improvement of reliability of machines and their units is achieved in a short time using accelerated tests at all stages of design. The maximum reduction of the machines design terms is ensured by studies and testing combined with an algorithm of test sequence including the following:

– analytical studies and calculations of work processes;

– determination of loading conditions;

– accelerated testing of machines, their assemblies, parts, samples, models, etc.;

– service tests;

– summarizing data and estimation of reliability.

Application of the test system reduces the total test work volumes, accelerates test conduction, ensures test validity, significantly reduces the final tests after the start of serial production, reduces consumer losses by creation of reliable equipment.

It has been established that reliability and service life of construction machines under interaction with loads should be determined not by analysis of reliability and durability of individual elements of the system but by the joint analysis of the system and calculating the probability of failure-free operation by multidimensional models. This approach opens up new possibilities in clarifying the new laws of the system functioning process and development of dynamic drives for the new generation machines with improved reliability and service life.

6. Conclusions

1. Indicators of the specified reliability and service life of the elements of the actuators of dynamic working bodies of construction machinery with an account of extreme values of the average life of the reserve system with equally reliable elements were substantiated and worked out. It was established that loading by various forcing methods is represented by double-frequency, multifrequency or random processes. In the whole range of high-cycle and low-cycle fatigue, the process was described by correlation functions and expressions for spectral density with various interrelations between the correlation coefficients for operating and bench test conditions. Correlation coefficients for accelerated tests were determined at a constant value of dispersion of the loading processes in operation and on the test bench. Moreover, the operation loads on the machine serve as the input variable parameters and equivalent loads serve as the output parameters.

2. Determination of equivalent harmonic loading parameters makes it possible to conduct tests with a specified acceleration coefficient by changing the loading parameters (the test frequency or amplitude). This makes it possible to force the loading amplitude, e. g., during bulldozer bench run tests, by more than 60 % at a nominal frequency. For the systems of long-term functioning, an estimation of the bench test volumes was made.

3. A procedure for accelerated service life tests of the machine drives with a definition of acceleration coefficients and methods for accelerated bench tests ensuring determination of approximate dependencies for estimating the machine service life in shorter terms were developed.

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