Дослідження базується на методі віброcтабілізаційної обробки, який використовується для зниження залишкових напружень в литих та зварних деталях та є альтернативою термодеформаційним методам, оскільки є невибагливим до маси, форми і габаритів деталі.

Зазвичай вібростабілізаційна обробка здійснюється за допомогою дебалансних електромеханічних систем, які є простими за конструкцією силової частини та системи керування. В таких системах обробка одночасно відбувається тільки на одній резонансній частоті. Оброблювана деталь, як правило, характеризується декількома резонансними частотами, що мають тенденцію до зміщення у низькочастотну область при здійсненні вібраційного впливу. Технологічний процес послідовної обробки на кожній змінюваній резонансній частоті є доволі тривалим у часі та не є ефективним з позиції витрат електричної енергії. З метою скорочення витрат часу та енергії, в роботі запропоновано використання найбільш прогресивних способів обробки відразу на декількох резонансних частотах. Виходячи з алгоритмів послідовної вібростабілізаційної обробки на декількох резонансних частотах деталі та їх зміни у бік низьких, запропоновано здійснення обробки полігармонічною збурюючою силою на обмеженій смузі частот. Цей вплив має таку ширину смуги частот, що містить усі можливі частоти деталі, на яких відбувається вібростабілізаційна обробка. Подібний вплив може бути реалізований за допомогою електродинамічного лінійного двигуна у якості виконавчого органу. Перевагою електродинамічного лінійного двигуна є пропорційність створюваного зусилля струму, що підводиться до рухомого провідника, і повторення його за формою.

Шляхом математичного моделювання для обраного прикладу встановлено, що звуження частотного діапазону в області низьких частот у 5 разів зменшує енергетичні витрати більш, ніж в 4000 разів у порівнянні з широкосмуговим законом механічної дії на деталь. Визначений теоретично енергетично ефективний закон може бути програмно реалізований у системах керування електродинамічними лінійними двигунами для здійснення вібростабілізаціної обробки

Ключові слова: вібростабілізаційна обробка, резонансна частота, енергетично ефективний закон, амплітудний спектр, електродинамічний лінійний двигун

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DEFINITION OF ENERGY EFFICIENT LAW OF MECHANICAL IMPACT IN VIBRATORY STRESS RELIEF OF METAL PARTS

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1. Introduction

Warping of basic parts, as well as distortion of rotational parts (shafts, spindles, gears, etc.), is one of the main reasons for the premature loss of required mutual movement accuracy of various machine components in space. In some cases, this may lead to the complete loss of performance for machines and instruments [1–3]. Changes in the shape and size of the cast or welded parts are caused by residual internal mechanical stresses resulting in the process of their uneven cooling, as well as during mechanical processing.

To reduce residual stresses and stabilize the geometric dimensions of the cast or welded parts, thermal deformation and mechanical deformation methods are used.

Common methods for reducing residual stresses are based on the thermal and force factors affecting the metal. Depending on the prevailing factor, there are two main types of methods [4] (Fig. 1):

– when the part is not subjected to significant force effects (natural aging, annealing);

– when a special type of force is created that performs mechanical stabilization treatment (vibratory stress relief, static loading).

The method of dynamic loading or the method of vibratory stress relief is the impact on the part by means of mechanical pulses generated according to a certain law [4, 5]. It differs from the static load in that the loading and unloading of the part take place cyclically, and the loaded state time is relatively short. Vibratory stress relief causes a superposition of stresses, during which local plastic deformations of the metal occur for a certain period of time. During this, the residual stresses caused by the deformation of the crystal lattice, are reduced due to the metal imparting energy pulse in the process of vibration [5].

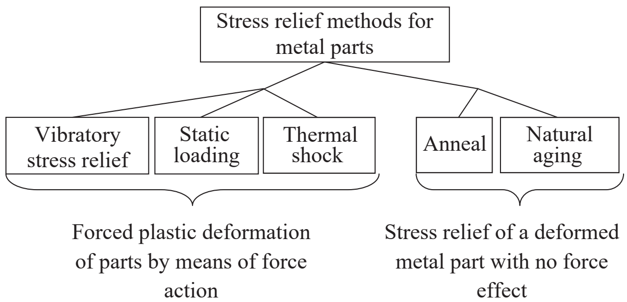

Fig. 1. Stress relief methods classification

Unlike most of residual stress relief methods, vibratory stress relief is undemanding to mass, shape and dimensions of the part. Therefore, it is one of the most versatile ways to reduce residual stresses in cast and welded parts. Energy consumption during vibratory stress relief is an order of magnitude lower than during heat treatment, therefore, improving methods of its implementation is an actual scientific and technical problem.

2. Literature review and problem statement

Traditionally, vibration processing to remove residual stresses is carried out at the resonant frequencies of the part or at frequencies close to the resonant ones, if processing at resonant frequencies is impossible for any technical reasons [6].

The main advantage of vibratory stress relief at the resonant frequencies of a part against processing at frequencies other than resonant is the low energy consumption per unit mass. This approach makes it possible to remove internal stresses in large and massive parts [7].

From the standpoint of a systems approach, the installation for vibratory stress relief can be represented as an electromechanical system, which implements an algorithm of the required processing method. In this case, the electromechanical system can be represented as three subsystems: control, executive and interface. The executive subsystem has a structure, allowing it to perform the reaction, formed by the control subsystem and to process the part applying mechanical pulses of a given shape and duration, performing on the resonant frequency of the part [8].

Electromechanical systems with unbalanced mass (Fig. 2), consisting of a platform or vibration-insulating supports 2 used to install a part 1; vibration exciter 3, attached to a part with clamps or bolts are widely used. Such a system includes a control panel with recording devices 4; a vibration sensor 5 attached to the part for the frequency and amplitude of oscillations feedback. For example, in [9] such a system is used for vibratory stress relief of welds.

A vibration block consisting of a DC motor and several unbalanced masses, mounted on its shaft, acts as the executive body [10].

Significant disadvantages of vibratory stress relief method, implemented by an electromechanical system with unbalanced mass are:

1. The laborious and time-consuming process of searching for resonant frequencies, due to the necessity of DC motors rotational frequency change with a small step.

2. The need to use expensive high-speed DC motors.

3. The operating point shift on the non-rigid motors mechanical characteristic when entering or leaving the resonant mode, which requires the feedback in the control system.

4. The change in motor current used to control the vibratory stress relief process depends on a large number of permanent and variable losses in the vibration system. The part installation method, the vibration block attachment quality and other factors unrelated to the process can also affect currents value and change. In addition, the percentage change in power consumption during the process is small and comparable to the measurement error.

5. The method is limited by the vibration system capabilities that implements it: processing is carried out simultaneously at only one resonant frequency, which cannot exceed 200 Hz.

6. Processing is carried out sequentially only at two or three frequencies, whereas complex-shaped parts have much more significant resonant frequencies.

7. Increased time and energy costs due to processing sequentially at each resonant frequency.

8. The shape of mechanical pulses acting on the part is determined by the unbalanced mass design and can't be changed dynamically during processing. To obtain pulses of the required shape, it is necessary to produce a special unbalanced mass, which requires additional material costs.

9. The method does not allow realizing a polyharmonic disturbing force for simultaneous processing at several resonant frequencies.

10. Low level of automation: often the operator regulates the motor speed controller manually, records the resonant points and then sets the desired processing frequency.

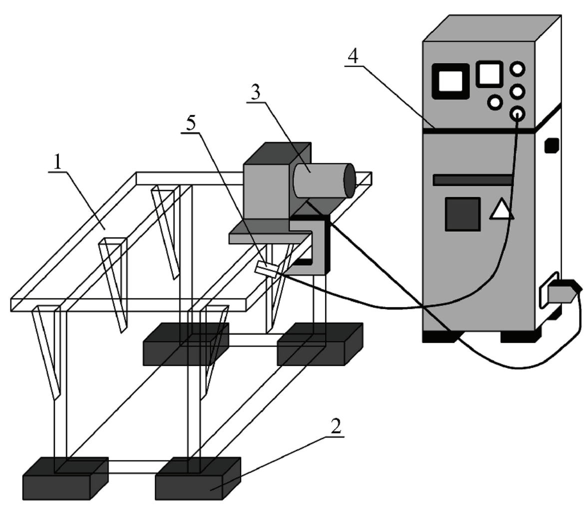

Fig. 2. Typical electromechanical vibration system with unbalanced mass: 1 – part; 2 – platform or vibration-insulating supports; 3 – vibration exciter; 4 – control panel; 5 – vibration sensor

Metal parts of low rigidity may be subjected to vibratory stress relief at nonresonant frequencies. However, this method of vibratory stress relief is technologically inefficient and is used only in limited cases [11].

The most progressive from the energy consumption point of view are vibratory stress relief methods with simultaneous multiple resonant frequencies vibration exciting. Traditionally, as in the method implemented in [8], processing is carried out simultaneously at only one resonant frequency, and in [9] – at two.

In general, studies [5–7] and works [8–10] did not carry out investigation related to the assessment of unproductive energy costs, which inevitably arise with a constant shape and pulse duty cycle during processing.

Thus, it is advisable to conduct a study on the energy efficiency of mechanical action during vibratory stress relief of metal parts. This, ultimately, will allow solving the problem of improving the methods of metal parts vibratory stress relief.

3. The aim and objectives of the study

The aim of this study is to define an effective law of mechanical action during the vibratory stress relief of metal parts, ensuring a reduction in energy consumption, a reduction in processing time, and an increase in the quality of part dimensions stabilization. To achieve this goal, it is necessary to solve the following tasks:

– to investigate ways of mechanical action, providing simultaneous vibration treatment in the frequency band containing resonant frequencies of the metal part;

– to carry out an energy losses estimation during the vibratory stress relief implementation using polyharmonic mechanical action in a variable frequency band;

– to select the executive body of the electromechanical system performing vibratory stress relief.

4. Investigation of methods of mechanical action, providing simultaneous vibration treatment in the frequency band

Often the processed parts have several resonant frequencies. In addition, in the process of vibratory stress relief, the stiffness coefficient decreases, and therefore the natural frequencies of the part move to the low range. Therefore, during vibratory stress relief it is constantly necessary to set the values of natural frequencies. Traditionally, vibration processing is carried out at all frequencies one by one and this process is quite time-consuming and has a low energy efficiency.

Let us consider vibratory stress relief processing of a metal part with a harmonic mechanical signal of the type:

$$P(t)=P_0 e^{i\omega t}, \tag{1}$$

where P_0 and ω – amplitude and angular frequency of processing; i – imaginary unit.

Consider the mechanical system of the vibratory stress relief installation as linear.

For the linear mechanical system with one degree of freedom, the equation of motion with harmonic action (1) can be represented as:

$$m\frac{d^2z}{dt^2}+\beta\frac{dz}{dt}+kz=P_0 e^{i\omega t}, \tag{2}$$

where m – system's mass; β – damping factor; k – stiffness factor; z – part displacement.

If $t>t_{free}$ the solution of equation (2) can be represented as:

$$z(t)=P_B e^{i(\omega t+\alpha)}, \tag{3}$$

where t_{free} – time of the system's free oscillations damping; P_B – vibration exposure amplitude on the part; α – phase shift between force and displacement.

The exposure amplitude depends on the frequency detuning factor:

$$P_B=\left|z(t)\right|=\frac{z_{cm}}{\sqrt{\left(1-\nu^2\right)^2+4\delta^2\nu^2}}, \tag{4}$$

where $\nu=\omega/\omega_0$ – frequency detuning factor; $\omega_0=\sqrt{k/m}$ – system's natural angular frequency; $z_{cm}=P_0/k$ – system's static displacement due to the force P_0; $\delta=\beta/2m$ – viscous damping factor.

Using (4) the dynamic force factor is determined as:

$$k_D=\frac{P_B}{z_{cm}}=\frac{1}{\sqrt{\left(1-\nu^2\right)^2+4\delta^2\nu^2}}. \tag{5}$$

If $\nu=1$, i. e. at resonance, the dynamic force factor can take values from 0.5 to 5 and more, depending on the damping factor. Therefore, the vibratory stress relief process is carried out at frequencies equal to the resonant ones under harmonic exposure.

Often, the processed parts have several resonant frequencies, in addition, in the process of vibratory stress relief, the stiffness factor decreases, and therefore the part's natural frequencies ω_{0i} move to low frequencies. Therefore, in the process of a vibratory stress relief it is constantly necessary to determine the natural frequencies' values. Traditionally, vibratory stress relief is carried out at all frequencies sequentially and this process is quite time-consuming and technically difficult to implement. Therefore, there is a problem of vibratory stress relief simultaneously at all frequencies in the frequency range:

$$\Delta\omega=\omega_{0\max}-\omega_{0\min}, \tag{6}$$

where $\omega_{0\max}$ – the maximum possible part's natural frequency before vibratory stress relief; $w_{0\min}$ – the minimum part's natural frequency after vibratory stress relief.

Thus, the mechanical pulses exposure is a polyharmonic disturbing force. If the oscillatory system is linear, then the total effect of the action of the polyharmonic perturbing force represented by the Fourier series (7) will be expressed as the sum of the partial effects from the action of each of the harmonics:

$$P(t)=a_0+\sum_{n=1}^{\infty}\left(a_n\cos\left(\omega_n t\right)+b_n\sin\left(\omega_n t\right)\right), \tag{7}$$

where $\omega_n=n\omega$ – harmonics frequency; $\omega=2\pi/T$ – the first harmonics angular frequency.

In the general case, the energy-optimal amplitude spectrum with simultaneous vibratory stress relief processing at all «floating» natural frequencies of a part can be represented as:

$$S(\omega)=\begin{cases}S_0,\omega_{0\min}\le\omega\le\omega_{0\max},\\ 0,\omega_{0\max}<\omega<\omega_{0\min},\end{cases} \tag{8}$$

where S_0 – amplitude of all harmonics in the frequency domain $\Delta\omega$.

The graph of this continuous in the interval from $\omega_{0\min}$ to $\omega_{0\max}$ spectrum, is shown in Fig. 3.

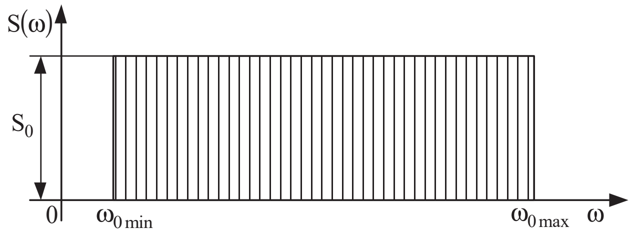

Fig. 3. Energy-efficiency optimal amplitude spectrum $S(\omega)$

To obtain an exposure function $P(t)$ having a spectrum (8), it is necessary to apply the inverse Fourier transform (9) and relations (10) between trigonometric functions and the exponential function:

$$P(t)=\frac{1}{\pi}\int_{0}^{+\infty}d\omega\int_{-\infty}^{+\infty}f(\tau)\cos\omega(t-\tau)d\tau. \tag{9}$$

$$\left.\begin{aligned} &e^{iz}=\cos z+i\sin z;\\ &\cos z=\frac{e^{iz}+e^{-iz}}{2};\\ &\sin z=\frac{e^{iz}-e^{-iz}}{2i}. \end{aligned}\right\} \tag{10}$$

The inverse Fourier transform can be quite simply represented with the use of computing. To implement this function, (8) illustrating the energy-optimal amplitude spectrum (Fig. 3) needs to be converted. The conversion is performed using the Heaviside single function $\Phi(t)$, which has the Fourier transform $\pi\cdot\delta(\omega)-i/\omega$, where $\delta(\omega)$ – is the Dirac function. $\Phi(t)$ takes the value 1 at $t\geq 0$ and the value 0 at $t<0$. The energetically optimal amplitude spectrum can be realized using the Heaviside function and the shift theorem.

The resulting dependence takes the form (11), convenient for computing:

$$\begin{aligned} S(\omega)=&\frac{S_0}{2}\left(\Phi(-\omega+\omega_{0\max})-\Phi(\omega-\omega_{0\max})\right)-\\ &-\frac{S_0}{2}\left(\Phi(-\omega-\omega_{0\max})-\Phi(\omega+\omega_{0\max})\right)-\\ &-\frac{S_0}{2}\left(\Phi(-\omega+\omega_{0\min})-\Phi(\omega-\omega_{0\min})\right)+\\ &+\frac{S_0}{2}\left(\Phi(-\omega-\omega_{0\min})-\Phi(\omega+\omega_{0\min})\right). \end{aligned} \tag{11}$$

After applying the inverse Fourier transform (9) in symbolic form, the following expression is obtained:

$$P(t)=\frac{S_0}{2}\cdot\frac{i}{\pi}\cdot\frac{-e^{i\omega_{0\max}t}+e^{-i\omega_{0\max}t}+e^{i\omega_{0\min}t}-e^{-i\omega_{0\min}t}}{t}. \tag{12}$$

Using (10), dependence (12) can be transformed to:

$$P(t)=\frac{S_0}{\pi}\cdot\frac{\sin(\omega_{0\max}t)-\sin(\omega_{0\min}t)}{t}. \tag{13}$$

At point $t=0$ function (13) has a discontinuity and must be extended as follows:

$$\lim_{t\to 0}\frac{\sin(\omega_{0\max}t)-\sin(\omega_{0\min}t)}{t}=\omega_{0\max}-\omega_{0\min},$$

$$P(t)=\frac{S_0}{\pi}(\omega_{0\max}-\omega_{0\min}).$$

The final expression for the force providing the optimal amplitude spectrum $S(\omega)$ will have the form:

$$P(t)=\begin{cases}\frac{S_0}{\pi}(\omega_{0\max}-\omega_{0\min}), \text{ if } t=0,\\ \frac{S_0}{\pi}\cdot\frac{\sin(\omega_{0\max}t)-\sin(\omega_{0\min}t)}{t}, \text{ if } t>0.\end{cases} \tag{14}$$

The mechanical action pulse of type (14) can be implemented using an electrodynamical linear motor of reciprocating motion.

It should be noted that the closer the values $\omega_{0\max}$ and $\omega_{0\min}$ are to each other the more difficult the polyharmonic mechanical action that the motor needs to perform, therefore, a more complex and functional control system is needed.

Without reducing the quality of vibratory stress relief, the law $P(t)$ can be simplified by accepting $\omega_{0\min}=0$. In this case, the processing will continue to be carried out at all the resonant frequencies and the process will increase slightly in time, but the control algorithms will be significantly simplified. To illustrate this idea, below are the dependencies $P(t)$ for $\omega_{0\min}=80$ s^{-1}, $\omega_{0\max}=100$ s^{-1} and the value $S(\omega)=4{,}000$ N (Fig. 4) and the same dependence, but for $\omega_{0\min}=0$ s^{-1}, $\omega_{0\max}=100$ s^{-1} and the value $S(\omega)=4{,}000$ N (Fig. 5).

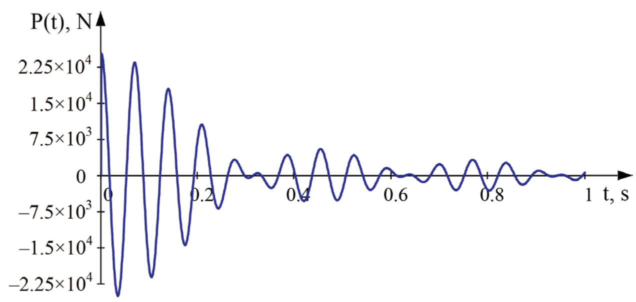

Fig. 4. $P(t)$ plot for $\omega_{0\min}=80$ s^{-1}, $\omega_{0\max}=100$ s^{-1}

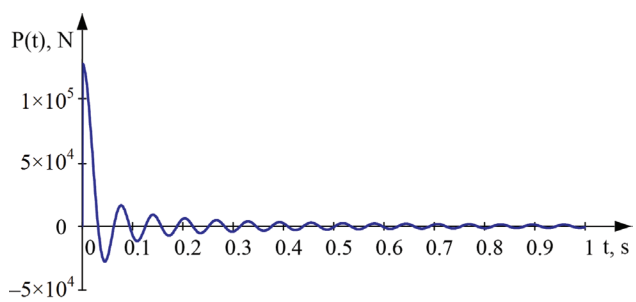

Fig. 5. $P(t)$ plot for $\omega_{0\min}=0$ s^{-1}, $\omega_{0\max}=100$ s^{-1}

In general, the graph of the action function $P(t)$, having a spectrum (11) for $\omega_{0\min}=0$, is shown in Fig. 6.

Based on Fig. 6, we can conclude that the proposed energy-efficient law of mechanical action on metal parts during vibratory stress relief involves the use of a narrowed band of resonant frequencies. As a result, the control system and the corresponding executive body (vibration motor) should be able to form a polyharmonic force action on the part. Let us perform an assessment of energy losses in the implementation of such processing.

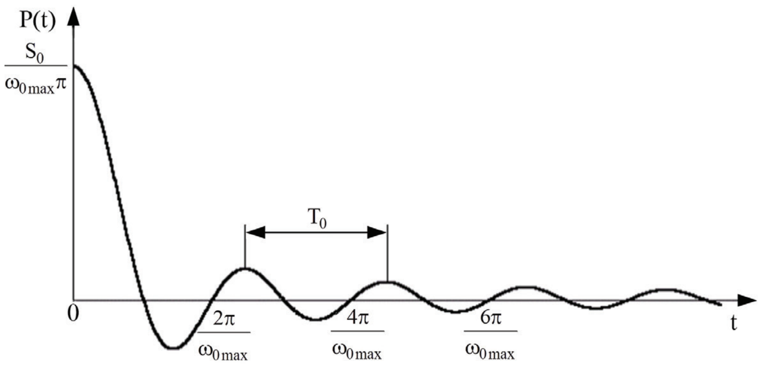

Fig. 6. $P(t)$ plot, having an optimal spectrum for $\omega_{0\min}=0$, represented in the general form

5. Energy losses evaluation for the implementation of vibratory stress relief process by polyharmonic mechanical action

Let us estimate the energy losses when applying the method of $\omega_{0\min}$ bias to zero. To simplify the calculation, let us assume that the efficiency of the motor is constant, that is $\Delta A=\Delta W$ – the change in the work of the force is equal to the change in the energy expended on this change. Elementary work dA in rectangular Cartesian coordinates is determined by the formula:

$$\delta A = F_x dx + F_y dy + F_z dz = \left(F_x \vartheta_x + F_y \vartheta_y + F_z \vartheta_z\right) dt, \qquad (15)$$

where x, y, z – coordinates of the force application point; F_x, F_y, F_z – projections of force vector on coordinate axes; ϑ_x, ϑ_y, ϑ_z – projections of velocity vector on coordinate axes.

The obtained theoretical dependence $P(t)$ is universal for vibratory stress relief processing of any part with any resonant frequencies.

In this case, the force acts only along one of the axes, and the velocity is also directed along the same axis. Therefore, (15) takes the form

$$\delta A = F_z dz. \qquad (16)$$

In the simplest case, in the absence of damping and elastic elements, (2) takes the form:

$$m\frac{d^2 z}{dt^2} = P(t). \qquad (17)$$

Double integrated (17) gives the dependence:

$$z(t) = \frac{S_0}{\pi m}\cdot\left(t\cdot Si(\omega_{0\max}t) + \frac{1}{\omega_{0\max}}\cdot\cos\left(\omega_{0\max}t\right)\right) - \\ - \frac{S_0}{\pi m}\cdot\left(t\cdot Si(\omega_{0\min}t) + \frac{1}{\omega_{0\min}}\cdot\cos\left(\omega_{0\min}t\right)\right). \qquad (18)$$

Let us define the differential of the obtained function:

$$dz = \frac{S_0}{\pi m}\cdot\left(Si\left(\omega_{0\max}t\right) - Si\left(\omega_{0\min}t\right)\right)dt. \qquad (19)$$

Next, using the formula (16), let us find the relationship (20), which is necessary to estimate the energy consumption reduction for vibratory stress relief when the frequency range is narrowed, which simplifies the form of vibration pulses:

$$A(t) = \int\frac{S_0^2}{\pi^2 m}\cdot\frac{\sin\left(\omega_{0\max}t\right)-\sin\left(\omega_{0\min}t\right)}{t}\times \\ \times\left(Si\left(\omega_{0\max}t\right)-Si\left(\omega_{0\min}t\right)\right)\mathrm{d}t = \\ = \frac{1}{m}\int\left(P(t)\int P(t)\mathrm{d}t\right)\mathrm{d}t. \qquad (20)$$

Integral (20) has no analytical solution. It can be calculated using numerical methods for specific data. Take for example two cases. In the first case $\omega_{0\min}=80$ s^{-1}, $\omega_{0\max}=100$ s^{-1}, $S(\omega)=4{,}000$ N, the processed part mass is $m=50$ kg, the pulse duration – 1 second (Fig. 4). Value of $A=19.072$ J.

In the second case $\omega_{0\min}=0$, $\omega_{0\max}=100$ s^{-1}, $S(\omega)=4.000$ N, the processed part mass is $m=50$ kg, the pulse duration – 1 second (Fig. 5). Value of $A=7.904\cdot10^4$ J. Thus, the frequency range narrowing by 5 times leads to a decrease in energy costs by more than 4,000 times.

This leads to the conclusion that when determining the frequency spectrum of vibration, it is necessary to take into account the energy costs in the frequency range from 0 to $\omega_{0\min}$, since the wrong choice of frequency range can lead to significant energy losses.

A graph of $A(\omega_{0\min})$ for $\omega_{0\max}=100$ s^{-1}, $S(\omega)=4{,}000$ N, the processed part mass is $m=10$ kg, the pulse duration is 1 s and when processed by a pulse with a frequency spectrum from 0 to $\omega_{0\max}=100$ s^{-1} is shown in Fig. 7.

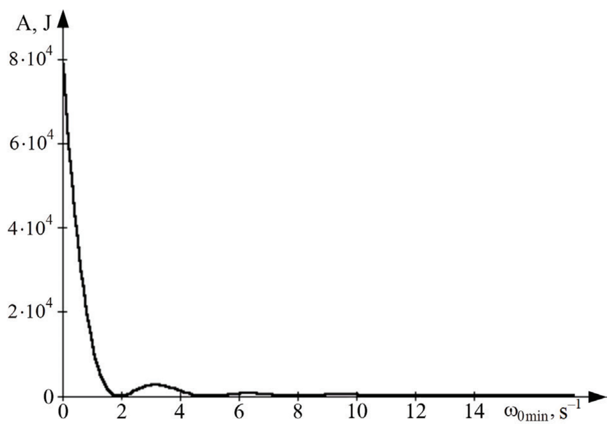

Fig. 7. Work dependency on $\omega_{0\min}$ for $\omega_{0\max}=100$ s^{-1}, $S(\omega)=4{,}000$ N, $m=10$ kg, pulse duration — 1 s ($\omega_{0\min}$ is changed from 0 to 15 s^{-1})

The main energy spectrum (Fig. 7) is in the frequency range $\Delta\omega=0$–10 s^{-1}, therefore the limiting frequency must be at least 10 s^{-1} from zero, otherwise the energy losses will increase significantly.

The work expended on vibratory stress relief processing (Fig. 7, 8) at initial frequencies close to zero is two orders of magnitude higher than the work that needs to be done when processing with initial frequencies ranging from 0 by 5–10 Hz.

This statement is well illustrated by the plot shown in Fig. 9, which is built in relative units. Here, on the abscissa axis is the ratio of the minimum circular frequency (variable value) to the maximum (constant value – this ratio is a kind of processing frequency range). The ordinate axis is the absolute value – work in relation to its maximum value, taken at zero value of the lower limit of the frequency range.

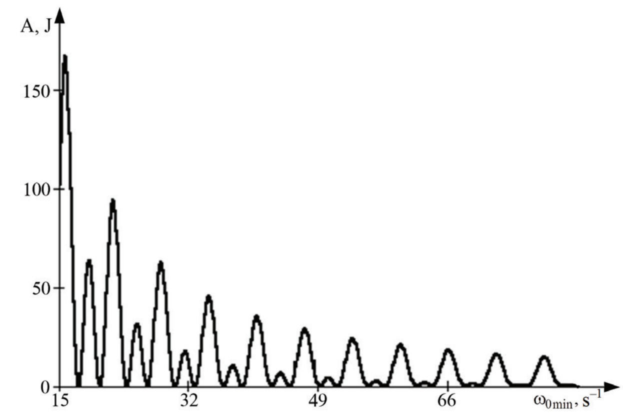

Fig. 8. Work dependency on $\omega_{0\min}$ for $\omega_{0\max}=100$ s^{-1}, $S(\omega)=4{,}000$ N, $m=10$ kg, pulse duration – 1 s ($\omega_{0\min}$ is changed from 15 to 100 s^{-1})

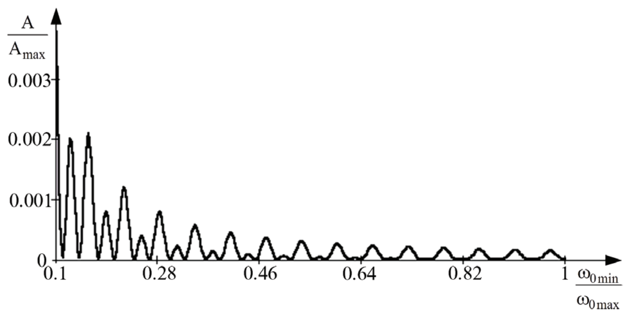

Fig. 9. Work dependency on $\omega_{0\min}$, in relative units, pulse duration – 1 s

The plot in Fig. 9 is generalizing and valid for any numerical values of the frequency range over a time interval with a pulse duration of 1 s.

Thus, when choosing the optimal mechanical action of vibratory stress relief of metal parts, there is a contradiction between the width of the frequency range and the energy consumption for processing.

The use of a wide frequency range has the following advantages:

– reduces the time of vibratory stress relief processing and increases its performance due to the simultaneous exposure on all the resonant frequencies of the part;

– there is no need to monitor the displacement of each of the resonant frequencies in the low range, since the original frequency band covers the entire possible range of resonant frequencies;

– allows simplifying the control system through the use of a simple algorithm.

However, all the advantages of wide frequency range processing fade in assessing the amount of unproductive energy costs that occur when processing at frequencies other than resonant ones.

6. Selection of the executive body for vibratory stress relief

Asynchronous motors and DC motors widely used in industry do not allow polyharmonic force action. Therefore, when creating an electromechanical system, linear reciprocating electric motors should be used for the technical implementation of the proposed law of mechanical action.

The main advantage of such motors is their wide functionality, since their principle of operation is based on the interaction of the current flowing through a moving conductor with a constant magnetic field. In this case, the force generated by the exciter is proportional to the current supplied to the moving conductor, and completely repeats it in shape.

There are the following types of reciprocating motion electrodynamic linear motors:

– with a single-rod magnetic core;

– with a double magnetic core;

– with a double gap in the magnetic core;

– with a double-rod magnetic core.

The disadvantage of vibrators with a single-rod magnetic circuit is a strong magnetic field of scattering, so there is a need to use special screens that weaken its action. It is very difficult to create a powerful motor design using a single-rod scheme due to the excessively large diameter of the moving coil. In addition, increasing the size of the coil reduces the rigidity of the structure, which is an undesirable factor from a technological point of view. The disadvantages of motors with a dual magnetic core and with a double gap in the magnetic core include the presence of a magnetic field in the plane of the table. The main advantage of a linear motor with a double-core magnetic circuit is the absence of the need for shielding the processed parts from magnetic flux fields.

The diagram of reciprocating motion electrodynamic linear motor, proposed for the implementation of vibratory stress relief, is shown in Fig. 10. This motor is made according to a double-rod scheme, its design is technological, convenient in operation and has good maintainability.

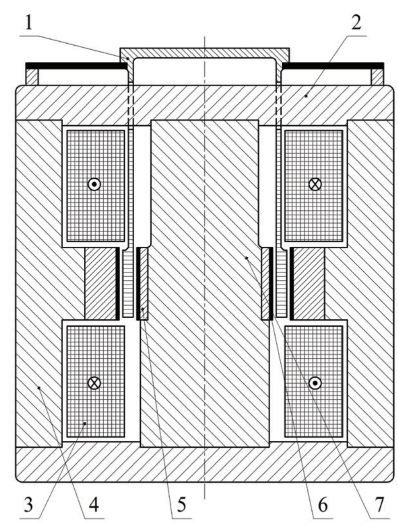

Fig. 10. Diagram of reciprocating motion electrodynamic linear motor: 1 – moving coil; 2 – magnetic flange; 3 – bias coil; 4 – magnetic housing; 5 – control coil; 6 – screen; 7 – rod

The electromechanical system with the proposed design motor will reduce the energy consumption for the implementation of vibratory stress relief. However, the reciprocating motion electrodynamic linear motors are not commercially produced by the industry and are manufactured only by an individual order. This can create certain difficulties in the implementation of the proposed solutions in production.

7. Discussion of the research results and recommendations for their implementation

When a part is exposed to mechanical pulses with a frequency spectrum containing both resonant and nonresonant part's frequencies, nonresonant harmonics are not productive for the process of relieving internal stresses. Vibratory stress relief is only effective at resonant frequencies.

The energy efficiency of the mechanical action by the law determined in this study during vibratory stress relief of metal parts is explained by the narrowing of the low-frequency band to the minimum necessary.

The proposed solution, in contrast to the traditional vibratory stress relief techniques carried out by unbalance electromechanical systems, has several advantages. Among them: elimination of time-consuming and labor-consuming process of finding resonant frequencies, reducing processing time and improving its performance.

The proposed law of formation of force action on a part allows providing a polyharmonic disturbing force for simultaneous processing at several resonant frequencies. Due to this, all the advantages of signal processing with a continuous band of frequencies remain.

The typical limitation of the proposed law of mechanical action during vibratory stress relief is the fact that it requires a special design for its executive body technical implementation (vibration exciter). A linear reciprocating electric motor, the principle of operation of which is based on the interaction of a current flowing through a moving conductor with a constant magnetic field, can act as such an executive body. The force created by the exciter is proportional to the current supplied to the moving conductor, and completely repeats it in shape. Thus, having formed the necessary polyharmonic current time dependence, one can obtain the desired law of the vibratory stress relief process.

As a disadvantage of the proposed solution, the need to periodically control the magnitude of the range $\Delta\omega$ of the resonant frequencies of the part can be mentioned. This leads to the need to use a microcontroller or microprocessor control system and the corresponding sensors. However, the determination of $\Delta\omega$ can be carried out without stopping the processing and does not require high accuracy. In the future perspective, for parts of certain classes, mathematical models can be created to reduce the range of resonant frequencies, which means no need to use sensors.

With the technical implementation of the proposed solution, difficulties may arise with the vibration exciter and the electric drive control system coupling. This is due to the fact that most modern control systems for electric drives do not have standard software for polyharmonic control actions.

The developed electromechanical system can be applied in mechanical instrument engineering and production for vibratory stress relief of both large and massive or small metal parts in order to stabilize their geometric dimensions and reduce residual stresses. The application of the proposed method of vibratory stress relief allows, while reducing the number of frequencies in the spectrum of a vibration signal, to reduce significantly energy consumption.

Further research prospects are related to the design and manufacture of a linear reciprocating motor. The use of such motor as an executive body in the electromechanical system of a laboratory test bench will make it possible to conduct experimental studies and confirm the energy efficiency of the proposed solutions.

8. Conclusions

1. The energy-efficient law of mechanical action during vibratory stress relief of metal parts was determined as a result of the performed research. Its feature is the narrowing of the low-frequency band in the process of processing to the minimum necessary.

2. As a result of the simulation, it was found that a narrowing of the frequency range by 5 times leads to a decrease in energy costs by more than 4,000 times. However, in practice, reducing energy costs largely depends on the shape and dimensions of the processed part.

3. A reciprocating motion electrodynamic linear motor was chosen as the executive body of the electromechanical system that performs vibratory stress relief. The double-rod design of such motor is constructively technological, convenient in operation and has good maintainability.

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CONTOUR MILLING PROGRAMMING TECHNOLOGY FOR VIRTUAL BASING ON A CNC MACHINE

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Представлена нова технологія і прикладна програма для автоматизації програмування операції фрезерування при віртуальному базування заготовки на столі верстата з ЧПУ, коли точна установка заготовки за координатним осями верстата ускладнена. Вирішення такої науково-технічної проблеми дозволяє виконувати контурне фрезерування деталей при їх довільному розташуванні на столі верстата з ЧПУ з гарантованим вирівнюванням припуску за траєкторією формоутворення.

Методика передбачає послідовне виконання трьох етапів з паралельним використанням створеної прикладної програми. На першому етапі готується електронна копія креслення деталі, яка містить виділені різними кольорами контур деталі і заготовки. Таким чином, при скануванні забезпечується автоматичне створення цифрових двомірних масивів геометричних образів, необхідних для вирішення завдання. На другому етапі в створену програму вводяться виміряні щупом на верстаті координати трьох точок заготовки. На підставі введених даних в створеній програмі вирішується завдання вирівнювання припуску методом Гаусса-Зейделя при використанні розмірності Хаусдорфа. Такий підхід дозволяє отримати кількісну оцінку подібності полігональних об'єктів, що необхідно для вирішення задачі мінімаксу розташування припуску. Задача полягає у визначенні корекції управляючої програми за двома лінійними координатами і однією кутовою навколо центра мас заготовки. На третьому етапі визначені в програмі величини корекції вводяться у стійку ЧПУ верстата і починається оброблення контуру.

Запропонована методика і створена прикладна програма були апробовані при обробленні контуру деталі на фрезерному верстаті VF-3 HAAS. Практична апробація показала ефективність методики, яка полягає в забезпеченні фрезерування гарантовано без перевантаження інструменту і скорочення часу обробки при віртуальному базування заготовки

Ключові слова: віртуальне базування, фрезерування контуру на верстаті з ЧПУ, підготовка управляючої програми

1. Introduction

It is known that the program for processing any surface of a part on a CNC machine must be tied to a specific position of the workpiece in the machining zone [1]. Typically, this problem is solved in two ways: either the workpiece is installed in a device that is pre-oriented relative to the coordinate axes of the machine, or the workpiece is installed directly on the machine table with the subsequent alignment of its actual position relative to the axis of the machine.

To implement the first method, it is necessary to ensure the presence of base surfaces of the workpiece, by which it is installed in the device, and when such conditions are met, binding the control program to the actual position of the workpiece on the machine does not cause difficulties [2]. When it is necessary to perform similar operations for large