Quantitative evaluations of the effectiveness of vibrating and vibroimpact machines

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Abstract. Using the example of an ultrasonic production machine, the problem of quantitative evaluation of the effectiveness of a vibrating machine is considered. Along with the classical approach, a more convenient and informative approach for vibrating and vibroimpact machines is considered, which is called a working (practical) approach. This approach is based on quantitative evaluations of the marginal and real utility of a technological process when performing this workload. It is shown that when implementing a resonant tuning of the machine, which provides the maximum working efficiency, the maximum possible efficiency in the classical sense can be equal to 50%.

1. Introduction

Mathematical modeling of vibrating and, in particular, ultrasonic production machines [1-21], as a rule, takes into account the drive (vibration generator), the vibration system (vibration line), as well as the actuating devices equipped with some tools. Actuating devices interact with the environment to be processed (structural materials, parts, blanks, etc.)

At the same time, reasoning qualitatively, it is necessary to recognize that a production machine that, with the minimum consumption of resources, puts the maximum possible energy into the processing zone is efficient. Thus, it is advisable to recognize machines that correctly solve the assigned technological problems, making minimal waste of external energy and other resources as efficient. For samples of vibration technology, here we can talk about implementation of the maximum possible values of amplitudes or other actuating devices with reasonable resource and energy limitations. That is, for example, heavy-duty and therefore bulky drives are not considered.

An important subclass of vibrating, vibroimpact, and, in particular, ultrasonic production machines is the so-called resonant machines [1-4, 11, 15]. At the same time, since building of adequate models of these machines certainly requires taking into account all the most significant acting factors, including dynamic characteristics of a technological process, we can mainly talk about nonlinear resonance phenomena. Frequencies of nonlinear resonances have a rather complicated structure and differ from natural frequencies of linear parts of systems, and resonance effects in the presence of nonlinear forces acquire completely new properties [2, 3, 5, 13-15].

During operation of vibrating machines, when large amplitudes of the actuating device are reached, a large amount of power is invested in the medium to be processed. Therefore, in the case when the actuating (working) device of the machine is in resonance, the machine itself operates with greater efficiency. In resonant states, the engine operates with the greatest efficiency. Consuming relatively
small external energy, due to the dynamic properties of nonlinear systems, it gets a ‘happy opportunity’ to work in the most economical mode. External energy is spent mainly on the essential effective work and on compensation of the work of always present dissipation forces. At the same time, the actuating device finds itself under conditions, in a sense, similar to the ideal ones: its oscillations are determined mainly by the intrinsic properties of the resonant system. This is exactly why resonant type machines differ from ordinary ones. Here, the operation of the machine is organized in such a way that in order to obtain a technological effect, the engine gives out an extremely high possible power. At the same time, there is no need to use super-powerful systems, and scarce resources are saved. All mentioned is justified by theory and experimental data (references are presented above).

2. Two approaches to the concept of ‘efficiency’ in vibrating machines

Hereinafter, we will assume that it is precisely the resonant machines that turn out to be the most efficient. Let us examine the connection between such an understanding of efficiency and an understanding that goes back to the classics of science who believed that efficiency

\[ \kappa = \frac{W_1}{W_1 + W_2} \times 100\% . \]  

Here \( W_1 \) is the total value of power consumed by the machine to solve a technological problem, and \( W_2 \) is the power dissipated due to inevitable dissipative factors. The denominator in formula (1) determines the total power spent by the engine (vibration generator). We will call variable \( \kappa \) a classical efficiency coefficient or, simpler, a classical efficiency.

Variable \( W_1 \) is determined by the features of technological processes. For vibratory hammers, this is the power required to drive piles or pipes into a soil. For vibratory mills, respectively, it is the power required, for example, to obtain the required highly dispersed powders. In an ultrasonic lathe – it is the power that provides turning, that is, spent on overcoming resistance of the processed material to the working cutter.

Let the machine operate in such a way that the value of variable \( W_1 \) is so large that \( W_1 \gg W_2 \). Then the power supplied by the production machine significantly exceeds the lost power and therefore \( W_1 + W_2 \approx W_1 \). In this case, therefore, it can be seen from formula (1) that variable \( \kappa \approx 100\% \).

We will evaluate the working efficiency in this case. Since the value of power \( W_1 \) is very high, the resistance to processing will also be very high. At the same time, the working amplitude of oscillations of an actuating device should naturally be low. Then, all real efficiency indicators will turn out to be low, close to zero: both the processing (cutting) speed, and, say, the amount of liquid pumped by the vibrating pump, and the shock pulse in the vibratory hammer, and, in general, working efficiency.

Let’s assume that losses of energy \( W_2 \) are very great now. Then variable \( \kappa \approx 0 \). The amplitude of tool oscillations will again be very low, since almost all the drive energy is converted into heat (dissipated). It is this situation that usually develops in very many (non-resonant) machines, for example, in standard ultrasonic machines, where the coefficient of classical efficiency cannot exceed several percent.

Thus, when \( \kappa \approx 100\% \) and \( \kappa \approx 0\% \), the work of production machines is practically ineffective. Therefore, it is reasonable to assume that there is an intermediate value of coefficient \( \kappa \) at which practically realized working efficiency reaches its maximum value.

Let us denote as \( Y \) a certain parameter of the actuating device of the machine, which determines the technological process. Usually, such a parameter is chosen so that it is clearly associated with some integral of motion in the corresponding conservative model. This, for example, can be the total mechanical energy of the actuating device. Now, let \( Y^* \) denote in some way an estimated maximum possible value of the taken parameter for the given characteristics of the drive and resistance of the processed material: \( Y \leq Y^* \). We will call variable

\[ \mu = \frac{Y}{Y^*} \times 100\% , \]  

(2)
the coefficient of working (practical) efficiency. It will be shown below that, for ultrasonic machines, the actuating device carries out resonant oscillations, and implementation of maximum capabilities is possible in the case when classical efficiency \( \kappa = 50\% \) and there is a coordination of power costs that go to overcome the dissipation and resistance forces of the processed material \( W_i = W_j \). It turned out that in this particular case it is possible to achieve the maximum realizable speed of the technological process (cutting). Other values of coefficient \( \kappa \) correspond to already non-resonant settings of the machine.

Thus, for resonant vibrating machines, the use of classical efficiency coefficient seems rather inconvenient. It is advisable to express the measure of working (practical) efficiency by coefficient \( \mu \) (2), since resonant tuning corresponds to value \( Y = Y^* \).

3. Resonant tuning

Let \( x(t) \) be the motion law of an actuating device of a vibrating (ultrasonic) production machine. Assuming that the vibration system (vibration line) is linear, we will write the following operator equation of motion using [1–3, 5]:

\[
x(t) = x_0(t) - L(\omega)\Phi[i\omega x(t); P_j].
\]

(3)

Here \( L(\omega) \) is dynamic compliance at the point of processing; \( x_0(t) = A_0(\omega)\cos \omega t \) is the law of motion during idle mode. In the equation of motion (3), \( \Phi \) is a nonlinear characteristic of the process. It can depend on the speed and higher derivatives of the law of motion, as well as on constant forces \( G_j \). For an ultrasonic machine designed for vibration turning [2, 3, 6–10, 19], the feed force must be taken into account.

The actuating device of the machine is often tuned to resonance in the idle mode: \( \omega = \omega_0 \). At the same time, force \( \Phi \) is described using a class of threshold functions [5] and begins to influence the system only after coordinate \( x \) exceeds a certain value, which corresponds to the beginning of the process. Then it turns out that the resonant frequency is usually quite different from the linear resonance frequency, which corresponds to the idle mode.

Studies of nonlinear resonances in ultrasonic machines were originally carried out in [10], and were thoroughly described in monographs [2, 3]. However, these studies need further development.

The standard amplitude-frequency characteristics \( A = A(\omega) \) for a system under consideration (3) are shown in Figure 1.

\[ A_0^* \]

\[ \omega \]

\[ \omega_0 \]

\[ A \]

\[ 1 \]

\[ 2 \]

\[ 3 \]

\[ 4 \]

Figure 1. Amplitude-frequency response: 1 - idle mode; 2 - low feed force; 3 – nominal feed force; 4 - enveloping curve.
With relatively low feed forces $G \leq \lambda_0^* b_0/2$, the amplitude-frequency response in the idle mode (curve 1) and when a load is applied (curve 2) practically coincide, although the maximum (amplitude values at the resonance) turns out to be shifted to the high-frequency zone with increasing of value $G$; coefficient $b_0$ is determined by the actual dissipation and depends on values $\text{Im}L(i\omega)$ and the dissipative component of force $\Phi$. An increase in the feed force ($G > \lambda_0^* b_0/2$) noticeably affects the form of the amplitude-frequency response (curve 3), leading to appearance of the so-called unstable branch, indicated by a dashed line.

Curve 4 is an enveloping curve that defines the set of resonances for all allowable forces $G$. Creating an effective impact requires maintaining movements that have parameters corresponding to this particular curve.

Let us emphasize that to implement this condition, systems of autoresonant settings for ultrasonic production machines and devices were developed and created, and that they make it possible to ensure implementation of resonant modes of motion in wide ranges of parameter changes [1 - 4, 11-15].

Methods of organizing the functioning of vibrating, in particular, ultrasonic machines, based on autoresonance, make it possible to create systems with high working (practical) efficiency.

4. Calculations and comparison of approaches

Let us carry out the calculations and consider a method for determining the working and classical efficiency for, taken as an example, an ultrasonic lathe, which is configured by the so-called load tuning method. In this case, an additional body with mass $m$ is attached to the actuating device. Such an addition gives rise to compensation for detuning resulting from the interaction of the tool and the workpiece.

When this compensation is performed, the resonant tuning occurs at a frequency of $\omega = \omega_0$. The mass of the attached body is presented by the equality $m = k(A)/\omega_0$. In this case, function $k(A)$ is determined and built using the characteristic of technological process $\Phi$, which determines the operator equation (3). It can be proved [2, 3] that the amplitude of vibration in the presence of a load is found as follows

$$A^* = \lambda_0^* \left(1 - \frac{2h}{\pi \lambda_0^* F_0} \sin^2 \frac{\pi G}{h}\right).$$

(4)

Here, variable $h$ is determined by the workpiece material, an area of the tool working surface, the type of abrasive and the state of the abrasive suspension directly near the cutting point. The value of variable $F_0$ gives the level of external excitation. Carrying out calculations using formula (4), we can determine the cutting speed

$$V = \frac{\lambda_0^* \omega}{\pi} \sin^2 \frac{\pi G}{h} \left(1 - \frac{2h}{\pi \lambda_0^* F_0} \sin^2 \frac{\pi G}{h}\right).$$

(5)

Here we will find clamping forces $G^*$ using condition $dV/dG = 0$ (maximum speed condition):

$$G^* = \frac{h}{\pi} \arcsin \left(\frac{\pi \lambda_0^* F_0}{4h}\right).$$

(6)

Usually in practice $\lambda_0^* F_0 \ll h$, and relation (6) is reduced to the following form $G^* = 0.5 \pi^{-1} \sqrt{\pi \lambda_0^* h F_0}$. From expressions (5) and (6), we will determine the maximum cutting speed:

$$V^* = \frac{1}{h} \lambda_0^* \omega_0 F_0/h,$$

(7)

and the amplitude of the tool vibration

$$A^* = \frac{\lambda_0^*}{2}.$$
Let us turn to the characteristic values of the motion parameters for actually operating machines derived from relation (7). When $A_0 = 20.5 \mu m$, we will get $V^* = 49 \text{ mm/min}$. This cutting speed is obtained when force $G^* = 10^4 \text{ H}$. It can be shown that in a mode with a maximum possible productivity, the power that is spent on a production task (destruction of material) $W_p$ turns out to be equal to power $W$ that dissipates in the vibration system:

$$W_p = W_0 = \frac{V^*}{8} A_0^2 \omega_0 F_0. \tag{9}$$

Then it turns out that this setting gives better coordination of the vibration system with both the elastic and dissipative components of the load. Due to its resonance, it also makes it possible to realize ultimate capabilities of the machine. At the same time a classic efficiency coefficient $\kappa = 50\%$

Formulas (7) and (8) are very convenient to use when calculating the vibration system of an ultrasonic machine with a given performance. If the feed force is higher than the value given by formula (6), then the value of the coefficient of classical efficiency will increase, but the cutting speed will decrease. Thus, for machines of this class, such a characteristic cannot be selected in any way as a practically significant indicator of the system tuning efficiency.

In formula (2) it is advisable to suppose that $Y = V$ and $Y^* = V^*$, that is, the working efficiency is the ratio of the processing speed to its maximum possible value: $\mu = V / V^* \times 100\%$ [11]. At the same time, value $V^*$ is given by relation (7), therefore, the operating efficiency in the idle mode is calculated as $\mu = V / V^* \times 100\% = 2.4 A_0^2 F_0 / h \times 100\%$. Value $A_0^2 F_0$ characterizes the level of external excitation and therefore has the order of dissipative forces. For ultrasonic vibration systems with a power of $0.1-10$ kW, the limits of this value can be estimated: $A_0^2 F_0 = 10 \div 10^7 \text{ H}$. The values of $h$ must lie within the limits: $h = 10^3 \div 10^6 \text{ H}$. Thus, we can conclude that the idea of tuning the system in idle mode leads to very inefficient working results: here $\mu = 2.4\%$. This means that in this case only $2.4\%$ of the ultrasonic machine capabilities will be used.

At the same time, a load resonance tuning makes it possible to achieve coefficient of working efficiency $\mu \approx 70\%$. In addition, the introduction of compensation allows to get values close to $\mu \approx 70\%$.

A further study of the above formulas shows that load tuning can be especially useful when machining hard workpieces on low-power machines with vibration systems with high quality factor. (‘Hard’ here means workpieces with a large processing area or made of materials that are difficult to process). Other authors made similar conclusions based on previous experiments [2, 3].

The above results can find application in assessing the working efficiency and tuning of numerous ultrasonic production apparatuses and devices for surface hardening, welding of plastic products and synthetic fabrics, excitation of vibrations of metal-cutting tools in installations for vibration cutting, and in many other cases.

Figure 2, a gives graphs of dependence of the cutting speed on the clamping force at different magnitudes of amplitude $A_0^*$. Here, the experimentally obtained data were plotted. Figure 2, b shows similar dependences found for the case of resonance tuning. It can be seen that a resonance case provides an almost tenfold increase in the cutting speed.

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Figure 2. Cutting speed: idle mode (a); resonance (b): 1–8; 2-11; 3-16.5; 4- 20.5 mkm.

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