EFFICIENCY OF VIBRATION MACHINES

Vladimir Astashev[1],Vitaly Krupenin[1,2]
[1]IMASH RAN, Russia; Moscow Polytechnic University, Russia
v_astashev@mail.ru, krupeninster@gmail.com

Abstract. The problems of enhancing efficiency of vibrating machines, which are one of the main tools of modern technology, are considered. They are now used in almost all sectors of industrial production and agriculture, as well as in medicine and everyday life. Technological processes implemented by such machines and devices consist in systematic, often periodic, oscillatory movement of a working body, which affects the workpieces and treated media. Organization of modern vibration machines and devices, and in particular ultrasonic ones, should be based on the principle of resonance. It is this resonant vibration process which makes it possible for the working body to influence on the treated medium most efficiently while external energy consumption is minimal. The submitted paper discusses and compares two concepts: "classic efficiency" and "practical efficiency". Possibility of resonance tuning devices is discussed; a quantitative characteristic of the practical efficiency is introduced. Using the ultrasonic technological machine as an example, it can be shown that when the regime with maximum practical efficiency is implemented, the classic efficiency turns out to be 50%. This at first glance paradoxical result is being discussed.

Keywords: ultrasonic machine, efficiency, resonance tuning, idling.

Introduction

Universal methods of analysis and calculation of modern vibration machines are based on the models representing the machine as a set of drive and executing bodies, interacting with the work media (parts, blanks, etc.).

The question arises as which machines should be considered as effective and which ones – as ineffective. The purpose of this article is to introduce a characteristic that best determines the actual efficiency of a vibratory (ultrasonic) machine.

Let us consider the question qualitatively. For a large number of vibration and vibro impact machines the largest efficiency is obviously connected with the highest amplitude of the oscillations of the working body and (or) the impact force developed. There can be several ways to achieve this. For example, provided the machines are equipped with heavy-duty drives, the values mentioned can be arbitrarily large. However, devices implementing this method are cumbersome and overly metal-consuming and energy-intensive.

It follows that the machines which can solve technological problems with minimum energy consumption should be considered as the most effective. Such requirements are met by resonant machines [1-4].

The amplitude of the vibrations of a resonant machine working body in operating mode reaches its maximum at an excitation frequency different from the values of the natural frequency of the oscillating system in idling due to the influence of the working process and its non-linearity. At the maximum amplitude, the maximum possible energy is applied to the treated medium and, therefore, the machine, the working body of which is in the resonant state, happens to be the most efficient. When the vibration machine works in the resonant mode, the drive operates with the greatest utility, and consuming relatively little energy from external sources, it spends its energy most effectively. The energy is spent mostly to fill the inevitable friction losses and useful work.

This is what counts in favor of resonant machine in comparison with "non-resonant" ones. If before to increase productivity one had to increase the capacity of the drive, then now, to achieve the desired technological effect one can organize the work so that the machine gives its maximum possible power.

Heavy-duty engines and large external energy are unnecessary, and, of course, consumption of materials and dimensions of a machine are reduced. All abovementioned is substantiated by thoroughly developed theory and well-confirmed on practice.

Note that in the case the resonant machine is designed with regard to operating process, the vibration system proves to be inevitably non-linear. Therefore, the resonant frequencies tend to differ significantly from the frequencies of linear resonance, while resonance phenomena acquire new properties [2; 3; 5].

Classic and real efficiency of machines

Having accepted the thesis that resonant machines prove to be most effective, let us consider the manner in which this efficiency is connected with the concept of efficiency in a classic sense.

By definition, classic efficiency is a ratio of the power the machine spends to perform a technological problem (denoted by N_1) to the total power spent by the drive, expressed as a percentage. This total capacity is the sum of two terms.

The first one is power N_1, the second one is N_2 that is power dissipation in the system due to inevitable energy losses on friction and other dissipative factors. Classic performance is evaluated by the formula

$$\eta = \frac{N_1}{N_1 + N_2} \times 100\,\%\,. \tag{1}$$

The value N_1 is determined depending on the specifics of the technological process. For example, for vibration pumps this value is the power required for fluid pumping. For vibro-hammers, it is the power spent on the plastic deformation of the workpiece. For ultrasonic machines, N_1 is the power spent on cutting, i.e. the power which is necessary to overcome the resistance of the processed medium to the tool action. Let us assume the machine is running in such a way that the value N_1 is very high. Then at conventional energy losses in the technological machine itself, the useful power is a lot larger than the power lost $N_1 >> N_2$, and therefore, $N_1 + N_2 \approx N_1$. Per the formula (1) the classic efficiency $\eta \approx 100$ % in this case.

Let us estimate the actual efficiency of such a machine. If the valueN_1 is large, the resistance of the treated medium should also be large, while, naturally, the tool oscillation amplitude will be low: under such conditions the amplitude should decrease. This means that the cutting speed will also be very small, almost zero, and the same will be the amount of the liquid pumped and the impact pulse, and hence, the real efficiency of the machine.

In the case that the useless losses of energy N_2 are very large, the oscillation amplitude of the tool is very small and classic efficiency ≈ 0 %, because almost all energy is wasted.

Such situation takes place in many modern non-resonance machines, for example, ultrasonic machines, classic efficiency of which does not exceed a few percent.

Thus, in the two limiting cases, at $\eta \approx 100\,\%$ and $\eta \approx 0\,\%$, the machine works practically ineffectively. It can be expected, therefore, that there is a certain optimum, and at some intermediate value of the classic efficiency η the practical efficiency will reach its maximum.

Suppose J is a parameter of movement of the working body of the machine, which characterizes the operating process. Typically, the parameter J is one-to-one associated with some integral of motion of the system, for example, with the energy of the working body in the technological process.

Suppose further J^* is the maximum possible value of the chosen parameter at the preset characteristics of the oscillating system and the resistance of the treated medium: $J \leq J^*$.The value

$$\vartheta = \frac{J}{J^*} \times 100\,\% \tag{2}$$

will be referred to as practical efficiency coefficient. It is shown that extreme capabilities of, for example, tuned to resonance ultrasonic machines can be realized at $\eta \approx 50$ % (see the books [2; 3] and later in this article). When $N_1 = N_2$, it is possible to achieve the maximum possible cutting speed.

Larger or smaller values of η correspond to settings with smaller practical efficiency. Thus, for resonance machines a classic efficiency coefficient appears to be an inconvenient characteristic. It is better to express the measure of their efficiency through the value ϑ (2), because quite general case, if it is precisely resonance tuning that allows obtaining magnitude $J = J^*$.

Resonance tuning

If the oscillation system of the machine is linear, the law of motion of the vibration machine working body $u(t)$ can be written in an operator form through dynamic compliance at the point of the working process $L(i\omega)$ [1-3; 5]

$$u(t) = u_0(t) - L(i\omega)\Phi\lfloor i\omega; u(t); P_j \rfloor, \tag{3}$$

where $u_0(t) = a_0(\omega)\cos\omega t$ is the law of motion of the working body at idling;
Φ is a non-linear function of the working process, which depends on the derivatives of the law of motion and possibly some permanent forces;
P_j – determined by the specifics of the process.

For example, for ultrasonic technology machines to treat fragile materials the feed force should be taken into account [2; 3; 6].

Quite often, the working body of the machine is tuned to the resonant frequency of idling $\omega = \omega_0$. However, usually, the non-linear force Φ starts to act only after a coordinate u overcomes a certain threshold value corresponding to the beginning of the interaction between the tool and the workpiece. Therefore, during operation, the resonance frequency is typically different from the frequency corresponding to the idling mode.

Analysis of resonant states of ultrasonic technological machines by approximate analytical methods was initiated in the article [6], and sufficiently detailed in works [2; 7]. However, a full description of all dynamic phenomena accompanying ultrasonic processes has not been given yet.

Typical amplitude-frequency characteristics of this system are shown in Fig. 1.

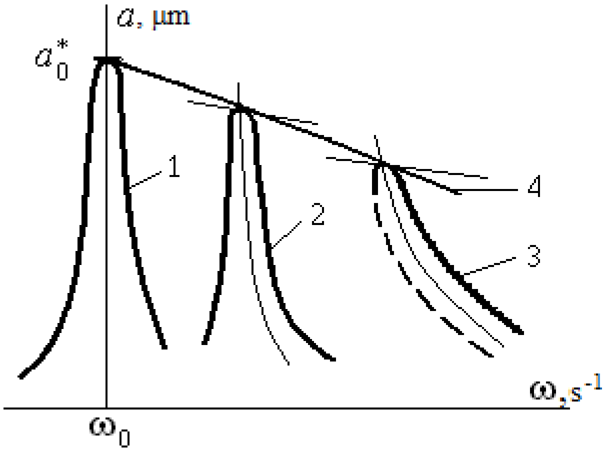

Fig. 1. **Resonance curves:** 1 – idling; 2 – low feed force ($P \le a_0^* B_0/2$); 3 – considerable feed force ($P > a_0^* B_0/2$); 4 – envelope of resonance curves

With relatively small pressing forces $P \le a_0^* B_0/2$ the forms of resonance curves under load (curve 2) and at idling (curve 1) are identical, but their maximum shifts to higher frequencies area as P increases. Value B_0 is determined by dissipative losses in the oscillatory system.

A further increase in the pressing force ($P \le a_0^* B_0/2$) significantly changes the nature of the resonance curve (curve 3), giving rise to the unstable branch, as shown by the dotted line.

Line 4 demonstrates the envelope of the resonance curves. It defines a set of resonant mode amplitudes for all possible values of the feed force P.

To maintain effective influence, it is necessary to ensure implementation of regimes with parameters corresponding to this curve.

Note that with this particular aim in view a system of auto resonant excitation of ultrasound machines has been developed and implemented. The system allows ensuring of resonance tuning in wide range of changes in the system parameters [1-4]. It is autoresonant schemes of excitement of vibration and, in particular, ultrasonic processing machines that provide realization of the systems with high practical performance, being described below.

Comparison of two approaches to the "efficiency" concept

As an example, we consider one of the possible methods for tuning of the ultrasound machine under load, as proposed in [6; 2; 3]. The method consists in attaching additional mass M to the tool, which results in compensation of the mistuning caused by the tool-workpiece interaction. With such compensation, the resonance tuning under the load is achieved at a frequency $\omega = \omega_0$. The value of the compensating additional mass is determined by the ratio $M = k(a)/\omega_0^2$. Here, the function $k(a)$ is defined and constructed based on the type of characteristics Φ of the working process, part of the equation of motion (3). It can be proved that the amplitude of the oscillations under load is determined by expression [2; 3]

$$a^* = a_0^* \left(1 - \frac{2H}{\pi a_0^* V_0} \sin^2 \frac{\pi P}{H} \right), \tag{4}$$

where the value H depends on the workpiece material, the square of the tool, the type of abrasive slurry and condition of the abrasive in the cutting area, while the value a_0^* indicates the level of the system excitation. Using (4), after the calculation, we obtain the cutting speed

$$v = \frac{a_0^* \omega}{\pi} \sin^2 \frac{\pi P}{H} \left(1 - \frac{2H}{\pi a_0^*} \sin^2 \frac{\pi P}{H} \right) \tag{5}$$

From the condition $dv/dP = 0$ we find the pressing force P^* when the cutting speed is maximum

$$P^* = \frac{H}{\pi} \arcsin \sqrt{\frac{\pi a_0^* V_0}{4H}}. \tag{6}$$

In real conditions $a_0^* V_0 << H$, as a rule, and the formula (6) takes the form

$$P^* = \sqrt{\frac{\pi a_0^* H V_0}{4\pi}}.$$

According to (5) and (6), the maximum cutting speed is

$$v^* = \frac{1}{8} a_0^{*2} \omega_0 V_0 / H , \tag{7}$$

while the amplitude of tool oscillations in this mode is

$$a^* = a_0^* / 2 . \tag{8}$$

Here are typical values for real ultrasonic machines, calculated using the formula (7). When a_0^* = 20.5 µm we obtain v^* = 49 mm·min^{-1}. This cutting speed is achieved when the pressure force is $P^* = 10^4$ N. It can be shown that at the realization of a mode with a maximum output, the power N_p spent on the destruction of the material equals to the power dissipated in the oscillatory system, i.e.

$$N_p = N_k = \frac{1}{8} a_0^{*2} \omega_0 V_0. \tag{9}$$

Thus, the tuning under consideration provides the best agreement between the oscillating system with both elastic and dissipative components of the load and allows realization of limiting capabilities of the ultrasonic machine. The value of the classic efficiency in this case is η = 50 %.

It is convenient to use relations (7) and (8) when calculating the oscillating system of the machine with the specified performance. When the feed force increases, and overcomes the value (6), the classic efficiency increases, however, the cutting speed decreases. Therefore, in this case for the machines of the class given, such a characteristic cannot serve as an indicator of the practical efficiency of tuning.

It is reasonable to take v and v^* as values J and J^* in the formula (2), that means that practical efficiency in this case is the ratio of processing speed obtained to its limiting value:

$$\vartheta = \frac{v}{v^*} \times 100\% \text{ [1].}$$

At the same time, the limiting value v * is defined by the equation (7), so the practical efficiency of the tuning at idling $\vartheta = v_0 / v^* = 2{,}4 a_0^* V_0 / D \times 100\%$. The value $a_0^* V_0$ characterizes the level of excitation of the system and is of the order of relatively small dissipative forces. For ultrasonic systems of 0.1-10 kW capacity, this value lies in the range $a_0^* V_0 = 10 \div 10^4$ N. The magnitudes of values H fall in the range $H = 10^3 \div 10^6$ N. This implies that tuning at idling is completely inefficient: for the above considered case of such tuning $\vartheta = 2.4$ %. That means that only 2.4 % of the potential capacity of the machine is used in this case.

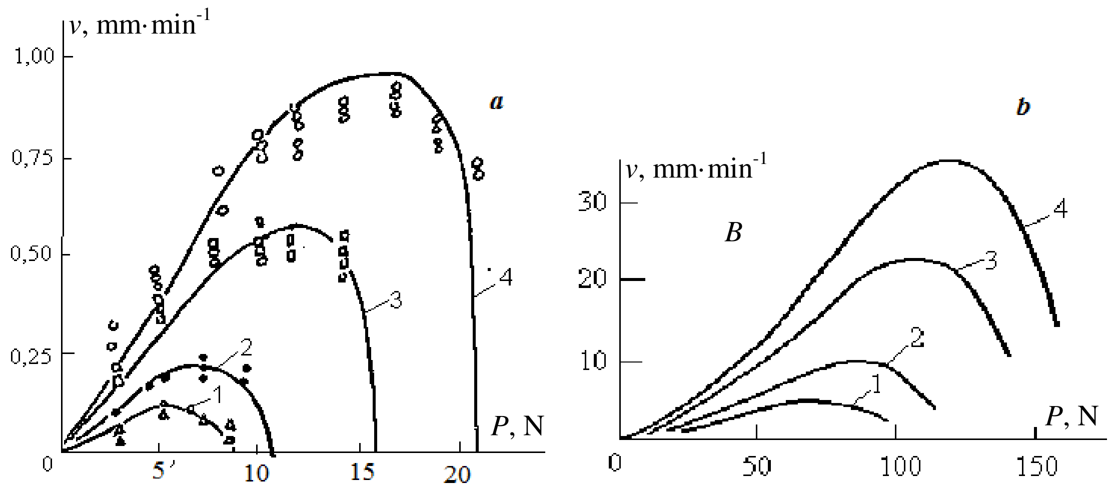

Fig.2. **Cutting speed at idling (a); at resonance tuning (b):**
$a_0^* = 1 - 8$ µm; 2 – 11 µm; 3 – 16.5 µm; 4 – 20.5 µm

At the same time resonance tuning enables an effective coefficient $\vartheta \approx 70$ %, while including compensation one can obtain the efficiency coefficient being close to $\vartheta = 100$ %.

Analysis of the relations above shows that tuning under load can be especially effective for treatment of rigid products by low power machine tools with oscillating systems with a high Q factor. (Rigid refers to products with a large square of processing or manufactured of poorly treatable materials.) Similar conclusions were reached earlier based on the experiments performed [8].

Note that these results can be used to assess efficiency and tuning of various ultrasonic technological ultrasound installations for surface hardening of workpieces, welding of plastics and synthetic fabrics, excitation of high-frequency oscillations of tools in apparatus for vibratory drilling, turning, smoothing, and many others. As mentioned, similar considerations can also be made in the analysis of other types of machines and devices of vibration and vibroimpact nature of action [9-11].

Fig.2 reveals cutting speed dependencies on the pressing force for different values of the amplitude a_0^*. Shown here are also the experimental data. Fig. 2, b represents analogous dependences calculated for resonance tuning. It is easy to see that the cutting speed is more than an orderof magnitude higher in the resonant case.

Conclusions

Consideration of high technologies may require refusal from even such classic concepts as efficiency, which appeared in the era of heatengines. For the analysis of vibration and ultrasound machines the notion of practical efficiencyis significantly easier and clearer than classic efficiency, as it characterizes the limiting capabilities of the process at minimum expenditure of energy.

The use of the theory of practical efficiency makes it possible to choose the optimal tuning of a vibratory (ultrasonic) machine among a set of fundamentally possible ones.

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