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Vibrational Machining with Torsional Spindle Vibrations

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Abstract—The use of a pulsed variable-speed drive as a vibrational drive is studied theoretically and experimentally. The design of a transmission with controlled velocity fluctuation is considered.

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Vibrational machining by the superposition of axial or radial vibrations has been studied by Kumabe in Japan and at Bauman Moscow State Technical University under the leadership of Poduraev [1]. Interest centers, for example, on drilling, which is complicated by pulsed rotation, with the alternation of working and idling passes, while the motion is transmitted by periodic pulses. Devices with such motion are pulsed variable-speed drives for continuous speed regulation, accompanied by torsional vibration of the driven shaft [2]. The variable-speed drive is used both as a gearbox adjusting the speed and as a generator of torsional spindle vibrations [3].

To create a variable-speed drive with specified (optimal) nonuniformity of the path, it is expedient to use a mechanism with a shaped cam. Consider the design of such a cam. The tachogram in Fig. 1a is described by the following equations

$$\left.\begin{aligned} \varphi_1(\alpha) &= \frac{1}{i_c}\left(\frac{3+\delta}{3} - \frac{4\delta}{\pi^2}\alpha^3\right) + R_2; \\ \varphi_2(\alpha) &= \frac{A}{a}\Big[\sin a(\alpha - \pi) \\ &+ \sin a R_1 - B\left(\alpha - \frac{k\pi}{2}\right)\Big] + 2R_2, \end{aligned}\right\} \quad (1)$$

where i_c is the mean gear ratio of the variable-speed drive; δ is the path nonuniformity of the variable-speed drive's shaft; φ and α are the angles of rotation of the driven shaft and drive shaft, respectively; k is the overlap of the parabolic section of the tachogram; a, A, B, R_1, and R_2 are constants.

On the basis of the results, speócial programs may be used to plot the rotational angle φ of the driven shaft and the speed and acceleration of the variable-speed drive as a function of the rotational angle α of the drive shaft (Fig. 1c) [4]. To describe the vibration of this variable-speed drive, we use the amplitude δ of torsional vibrations of the driven shaft.

Since, in speed regulation, the vibrations of the variable-speed drive change nonuniformly, a drive design with a bearing that may move along the slide rod without obstruction is proposed (Fig. 2) [5]. Besides Eq. (1), the operation of the variable-speed drive is determined by the following parameters: the length of the slide rod $BC = L$; the length of the rocker $CD = c$; the distance $ED = p$ determining the position of the mobile bearing E; the roller radius r; the pressure angle γ; and the lengths $EC = l_1$ and $BE = l_2$. We now consider the characteristic positions of the slide rod BC and rocker CD (Fig. 2a).

The length l_1 is determined from the equations

$$l_1 = \sqrt{p^2 + c^2 + 2pc\cos\varphi};$$

$$\beta = \arcsin\frac{c + \sin\varphi}{\sqrt{p^2 + c^2 + 2pc\cos\varphi};}$$

$$\frac{d\beta}{d\alpha} = \frac{c(c + p\cos\varphi)}{p^2 + c^2 + 2pc\cos\varphi}\frac{d\varphi}{d\alpha},$$

where β is the deviation of the slide rod BC.

With variation in β, the length l_2 varies as follows

$$ds = \sqrt{(l_2 d\beta)^2 + dl_2^2};$$

$$\frac{ds}{d\varphi} = \sqrt{\left(l_2 + \frac{d\beta}{d\varphi}\right)^2 + \left(\frac{dl_2}{d\varphi}\right)^2};$$

$$\frac{ds}{d\alpha} = \frac{ds}{d\varphi}\frac{d\varphi}{d\alpha}.$$

The equations for the calculation of the cam's radius vector $R = OB$ and its polar angle θ are obtained by considering the characteristic positions of the slide rod BC. The first position corresponds to the maximum deviation of the slide rod in idling—that is, when $d\varphi_2/d\alpha = 0$ (Fig. 1a; points d_1 and d_3 of the tachogram).

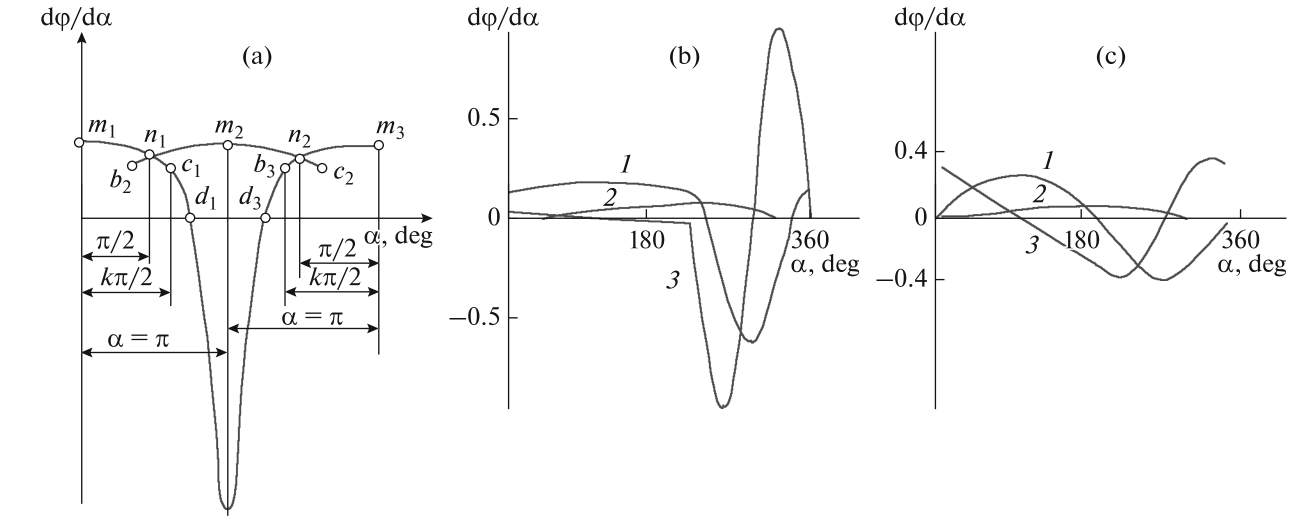

Fig. 1. Tachogram of the variable-speed drive (a) and dependence of dφ/dα (*1*), φ (*2*), and $d^2φ/dα$ (*3*) on the positional angle α for a cam with δ = 0.1 (b) and 1.2 (c).

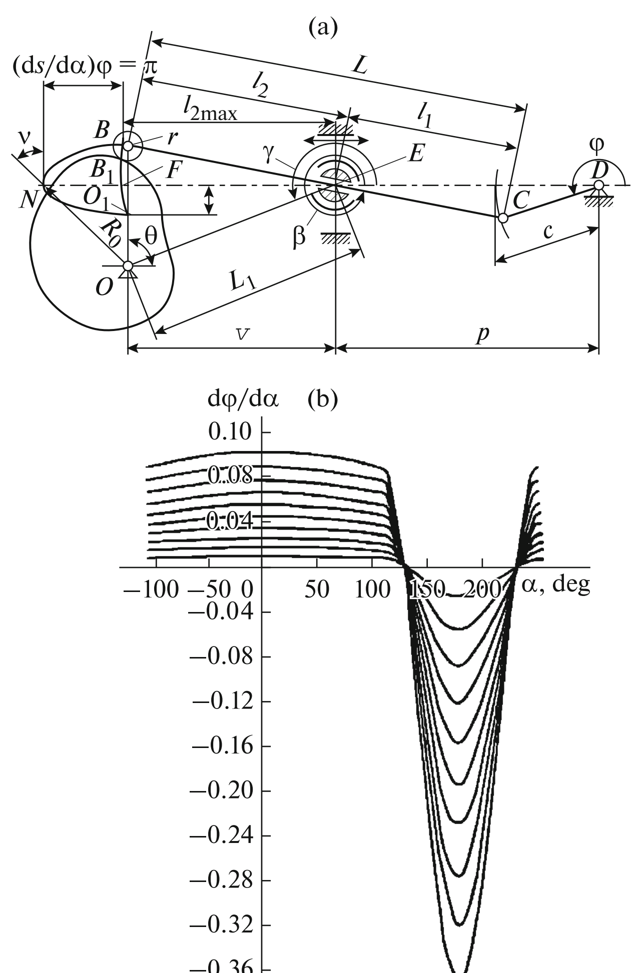

Fig. 2. Variable-speed drive with a slide rod of variable length (a) and its control characteristics (b).

From Eq. (1), we determine the corresponding angle

$$(\alpha_0)_{12} = \pi \pm \frac{1}{a}\arccos\frac{B}{A};$$

$$\varphi' = \varphi_{2\min} = \pm\frac{A}{a}\sin\left[\arccos\left(\frac{B}{A}\right)\right]\mu\frac{B}{A}\arccos\left(\frac{B}{A}\right) + \pi.$$

For the first position, the projection of the point B on the rocking axis DN of slide rod BC is denoted by F, and the distances $BF = q$ and $EF = v$ are determined

$$q = \left(\frac{1}{\sqrt{p^2 + c^2 + 2pc\cos\varphi'}} - 1\right)c\sin\varphi';$$

$$v = \frac{l_2'}{l_1'}(p + c\cos\varphi'),$$

where

$$l_2' = L - l_1' = \sqrt{p^2 + c^2 + 2pc\cos\varphi'}.$$

With variation in β by $d\beta$, the length $BE = l$ of the slide rod changes by dl_2, according to the formula

$$\frac{dl_2}{d\alpha} = \frac{pc\sin\varphi}{\sqrt{p^2 + c^2 + 2pc\cos\varphi'}}\frac{d\varphi}{d\alpha}.$$

If the slide rod BC and rocker CD form a straight line B_1D (the second position of the slide rod; Fig. 2a)—that is, if $\varphi = \pi$ and $B_1E = l_2 = l_{2\max}$—we obtain

$$\left(\frac{dl_2}{d\alpha}\right)_{\varphi=\pi} = 0; \left(\frac{d\varphi_2}{d\alpha}\right)_{\varphi=\pi} = \left(\frac{d\varphi_2}{d\alpha}\right)_{\max}.$$

Taking into account that

$$B_1B = \mathrm{d}s;\ \frac{\mathrm{d}s}{\mathrm{d}\varphi};\ \frac{\mathrm{d}s}{\mathrm{d}\alpha} = \frac{\mathrm{d}s}{\mathrm{d}\varphi}\frac{\mathrm{d}\varphi}{\mathrm{d}\alpha}$$

and that

$$\left(\frac{\mathrm{d}s}{\mathrm{d}\alpha}\right)_{\max} = \left(\frac{\mathrm{d}\beta}{\mathrm{d}\alpha}\right)_{\max} l_{2\max}$$

$$\text{and } \left(\frac{\mathrm{d}\beta}{\mathrm{d}\alpha}\right)_{\max} = \frac{c}{c-p}\left(\frac{\mathrm{d}\varphi}{\mathrm{d}\alpha}\right)_{\max}$$

we find that

$$\left(\frac{\mathrm{d}s}{\mathrm{d}\alpha}\right)_{\max} = c\left(\frac{L}{c-p}+1\right)\left(\frac{\mathrm{d}s}{\mathrm{d}\alpha}\right)_{\max}$$

$$\text{and } l_{2\max} = L + c - p.$$

The minimum permissible radius vector R_0 (ON_1) of the equidistant point on the cam is determined from Fig. 2a

$$R_0 \ge c\left[\begin{array}{l}\left(\frac{L}{c-p}-1\right)\left(\frac{\mathrm{d}s}{\mathrm{d}\alpha}\right)_{\max} + (L+c-p) \\ -\left(\frac{L}{\sqrt{p^2+c^2+2pc\cos\varphi'}}-1\right)(p-c\cos\varphi')\end{array}\right]$$

$$\cot\gamma - \left(\frac{L}{\sqrt{p^2+c^2+2pc\cos\varphi'}}-1\right)c\sin\varphi';$$

Likewise, the pressure angle γ is

$$\tan\gamma = \frac{\left(\frac{\mathrm{d}s}{\mathrm{d}\alpha}\right)_{\max} + [l_{2\max} - v]}{R_0 + q}.$$

To ensure that rotation is possible (with no contact of the cam and bearing E in the course of rotation), we require that

$$R_0 \le \left(\frac{L}{\sqrt{p^2+c^2+2pc\cos\varphi'}}-1\right)c\sin\varphi',$$

$$v = \frac{p + c\cos\varphi}{c\sin\varphi'}.$$

The radius vector $\overline{R}$ of the equidistant point on the cam and its polar angle θ are given by the formulas

$$\overline{R} = \sqrt{L^2 + l^2 + 2l_1l_2\cos(\alpha-\beta)};$$

$$\theta = \arccos\left(-\frac{L_1\cos\gamma + l_2\cos\beta}{R}\right),$$

Here

$$L_1 = \sqrt{(R_0+q)^2 + v^2};\ v = wq;$$

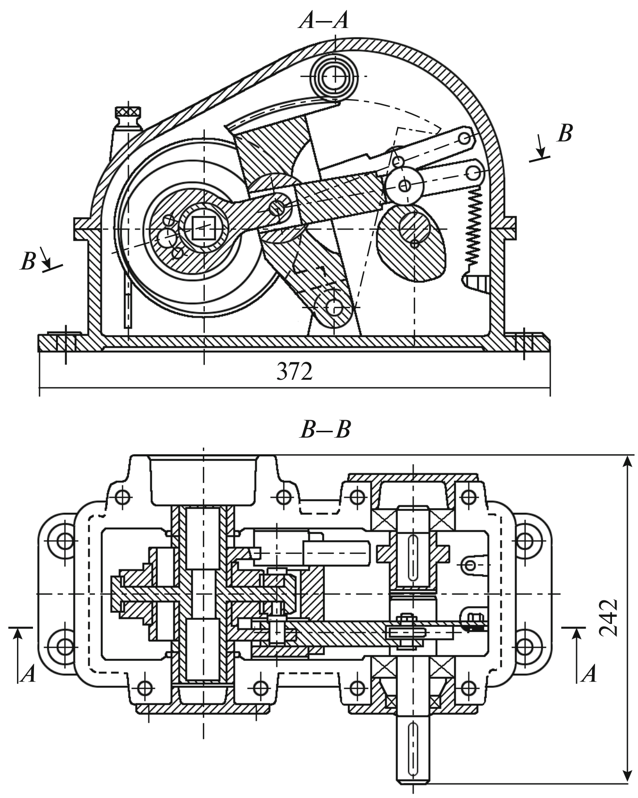

Fig. 3. Pulsed variable-speed drive with a slide rod of variable length.

$$l_2 = L - \left(\frac{1}{\sqrt{p^2+c^2+2pc\cos\varphi'}}\right);$$

$$\gamma - \arctan\left(\frac{R_0+q}{v}\right);$$

$$\beta = \arcsin\frac{c\sin\varphi}{\sqrt{p^2+c^2+2pc\cos\varphi}};$$

and β and γ are the angles of vectors $\overline{L}$ and $\overline{L}_1$, respectively.

To investigate the drive properties of the variable-speed drive, we use a special system. The drilling of steel samples shows that the vibrations generated by the variable-speed drive change the chip formation and increase tool life, thanks to the change in the drill torque with slight path nonuniformity. The experiment does not permit the identification of the optimal path nonuniformity for use in the design of the cam profile. The design of the variable-speed drive is shown in Fig. 3. The constant amplitude of the vibrations over the whole range of speed regulation (Fig. 2b) permits its use in various branches of industry [6]. To simply the system generating torsional vibrations, we may use other transmissions.

(a) (b)

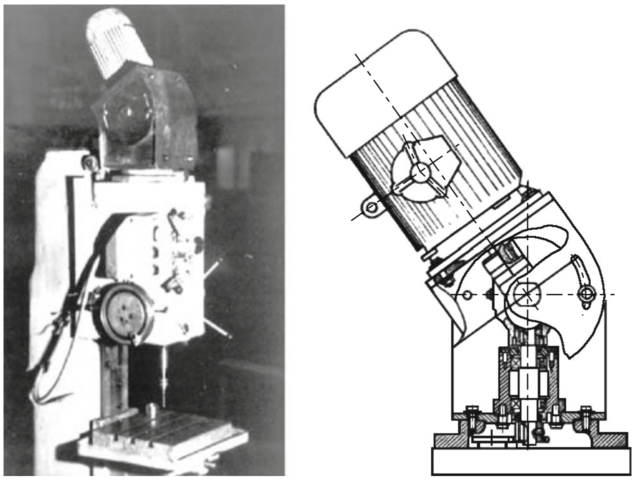

Fig. 4. Vertical drilling machine with a vibrational drive including a universal joint (a) and structure of the joint (b).

When using a pulsed transmission with two cylindrical gears mounted with the same eccentricity e, the possible path nonuniformity is [7]

$$\delta = \frac{4Ae}{\left(A^2 + 4e^2\right) i_c},$$

where i_c is the mean gear ratio; A is the distance between the centers.

The transmission may only be easily installed in the machine-tool drive when $e_{max} = 0.05A$ and $\delta_{max} = 0.4$. That considerably limits its applicability.

To create optimal torsional vibrations, we investigate the change in speed of the mechanism that transmits the rotation, such as a universal joint [8]. For example, the spindle drive in a vertical-drilling machine (Fig. 4) has an electric motor that may rotate relative to the center of the joint. The drill speed depends on the angular position α of the electric motor relative to the vertical axis of the spindle and is determined by the relation

$$\frac{d\varphi_2}{d\varphi_1} = \frac{\cos\alpha}{1 - \sin^2\alpha\cos^2\varphi_1},$$

while the path nonuniformity $\delta = \sin^2\alpha/\cos\alpha$.

If the electric motor is fixed and α is constant, the drill turns at constant speed. If the motor rotates, along with the drive head of the joint, α increases, with corresponding variation in amplitude of the drill vibration, determined by δ. The maximum value $\delta_{max} = 0.70$ corresponds to $\alpha = 45°$. With a vertical motor position—that is, when $\delta = 0$ and $\alpha = 0$—the machine operates without vibration. Rotation of the motor relative to the vertical axis permits assessment of the influence of vibration on machining.

However, joint operation with $\alpha \geq 30°$ is undesirable, and the presence of gear couplings and key couplings in the drilling machine's gearbox reduces the speed vibration. The use of a machine with such a vibrational drive at the Pskovkhimlegmash plant has been accompanied by both change in the chip behavior and increase in tool life.

A vibrational drive with two universal joints is described in [9]. The input heads are established so that the plane of head *3* for the first joint (Fig. 5a) is perpendicular to the plane of head *4* for the second joint. Electric motor *1* and the intermediate transmission (for example, gearbox *2*), connected by clutch *5*, are mounted on a common frame *6*; motion in direction *7* between the extreme positions *A* and *B* of the center of the first joint is possible.

We know, for example, that uniform spindle rotation may be ensured by means of two universal joints in a spindle drive configured so that the vibrations cancel out. In the proposed vibrational drive, the universal joints are configured so that the vibrations of the first joint reinforce those of the second. For such a drive, the gear ratio is

$$i_{21} = \frac{d\varphi_2}{d\varphi_1} = \frac{\cos^2\alpha}{\sin^2\varphi_1\cos^4\alpha + \cos^2\varphi_1}. \quad (2)$$

On the basis of condition of normal drive operation, we consider joint positions such that $\alpha = \alpha_{max} = 45°$ and determine the gear ratio from Eq. (2), with different positional angles of the input shaft in the intermediate transmission

$$i_{21} = \cos^2\alpha = \frac{1}{2} \text{ when } \varphi_1 = 0;$$

$$i_{21} = \frac{2\cos^2\alpha}{\cos^4\alpha + 1} = \frac{4}{5} \text{ when } \varphi_1 = 45°;$$

$$i_{21} = \frac{1}{\cos^2\alpha} = 2 \text{ when } \varphi_1 = 90°.$$

Taking account of the assumed gear ratios and Eq. (2), we determine

$$\delta = \frac{1 - \cos^4\alpha}{\cos^2\alpha}.$$

Thus, when $\alpha = 45°$, the nonuniformity of rotation of the driven component (the tool) is $\delta_{max} = 1.5$, which is twice that for a drive with a single universal joint. By regulation of the path nonuniformity of the intermediate transmission with electric motor *1* in direction *7*, nonvibrational rotation ($\delta = 0$) may be ensured at point *A* when $\alpha = 0$, while $\delta_{max} = 1.5$ at point *B* (Fig. 5a). Note the compactness of the proposed drive: the total motion of the frame with the electric motor $AB = 76.6$ mm when $AO = 100$ mm (Fig. 5b). In addition, the drive is simple, technologically expedient,

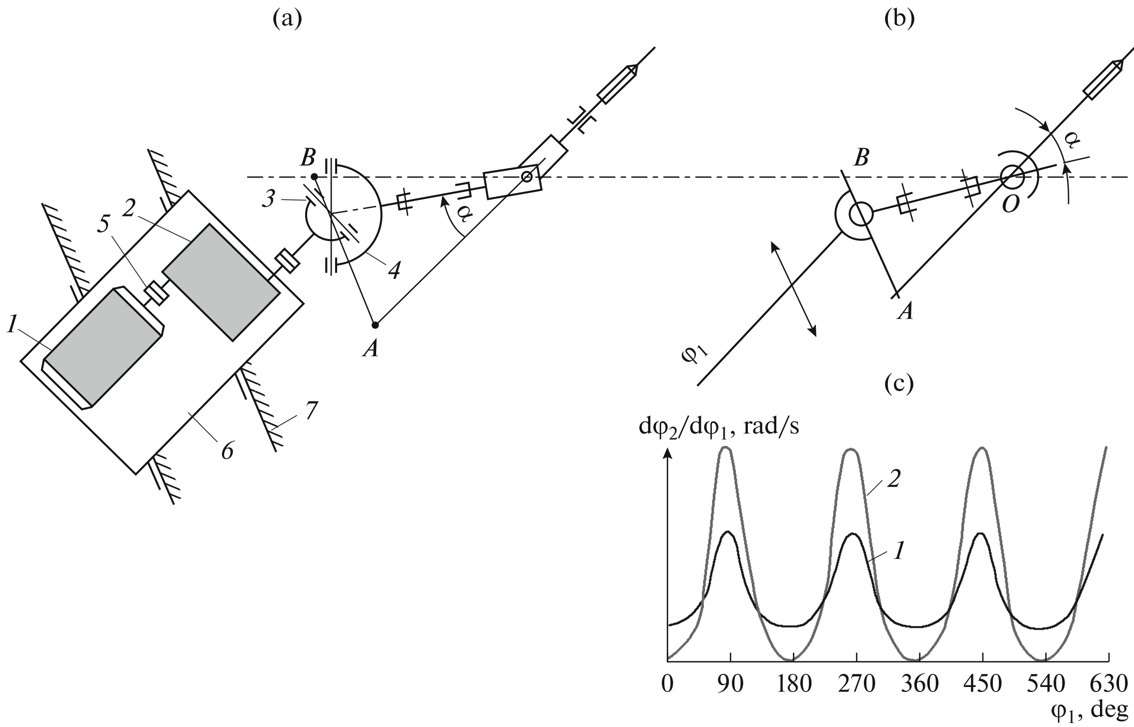

Fig. 5. Vibrational drive with two universal joints (a); configuration of drive (b); and speed $d\varphi_2/d\varphi_1$ of tools with one (*1*) and two (*2*) universal joints as a function of the drive shaft's positional angle φ_1 (c).

reliable, and kinematically compatible with drilling machines.

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