Investigation of a Frequency-Controlled Electromagnetic Vibration Motor with Nonlinear Characteristics of Elastic Elements

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Vibratory electric drives are widely used in various industrial applications, including bulk material transportation, vibratory feeders, presses, separators, crushers, and compaction equipment. Their popularity is explained by their simple design, reliability, and ability to efficiently transfer oscillatory energy to working elements. Modern technological processes require improved energy efficiency, an extended range of operating modes, and enhanced controllability of drives. Traditional vibratory drive designs provide limited capabilities for regulating amplitude–frequency characteristics and do not always ensure stable performance under variable loads.

INTRODUCTION

The limiting load of a frequency-controlled single-stroke electromagnetic vibratory motor (EMVM) with biasing is not determined by its electromagnetic parameters, since they are designed with a significant power margin, but rather by the onset of an impact operating mode associated with a considerable increase in the constant component of the armature oscillations. The inclusion of buffers or other elements with asymmetric characteristics generates a constant component of the armature oscillations that is directed opposite to the component A0, which arises under the influence of F0. Proper selection of the elastic element parameters of the EMVM makes it possible to mutually eliminate, or at least minimize, the constant component of the armature oscillations, thereby reducing the likelihood of impact operation and increasing the permissible load of the EMVM.

EXPERIMENTAL RESEARCH

The introduction of buffers and elastic elements into the EMVM leads to the differential equation of the armature motion becoming essentially nonlinear. These nonlinearities arise primarily from the stiffness characteristics of elastic elements and the asymmetry of buffer forces, which deviate from the ideal linear Hookean model. As a result, the vibratory system may exhibit amplitude-dependent natural frequencies, resonance shifts, bifurcation phenomena, and even chaotic oscillatory regimes. Such effects significantly complicate the analysis and control of the motor, but at the same time provide opportunities for optimizing its performance through careful adjustment of elastic parameters and control strategies.

$$m\ddot{\chi} + p(\chi)\dot{\chi} + K(\chi)\chi = F(\chi, t) \tag{1}$$

Although several exact methods, such as the integral equation method and the method of point transformations [1], can be applied to equation (1), their use leads to overly complex results that hinder practical analysis of the system’s behavior. Therefore, these methods are more suitable for verifying the accuracy of approximate solutions to equation.

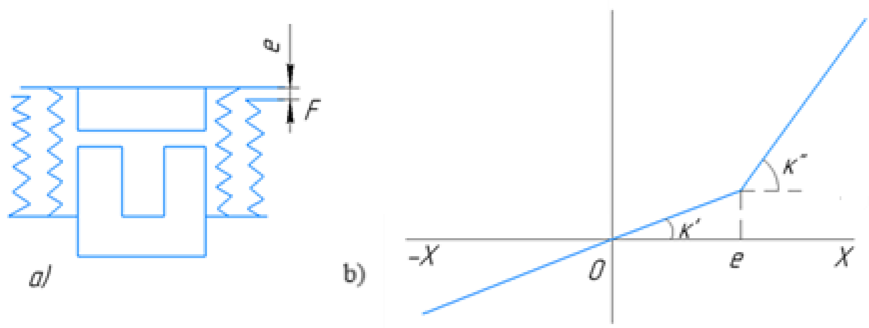

Fig.1. The basic diagram of a single-stroke EMVM with buffers is shown in Fig. a), while Fig. b) illustrates the nonlinear characteristic of the elastic elements.

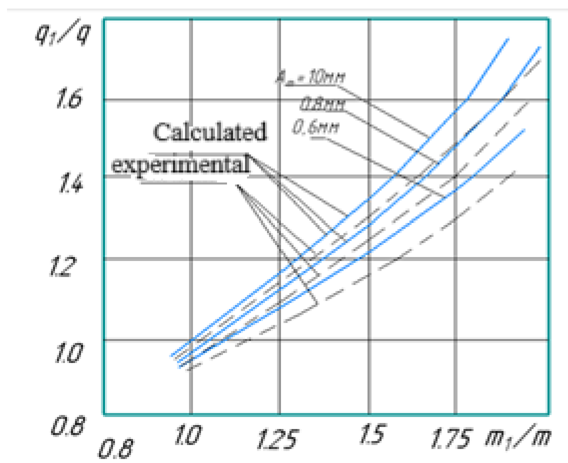

Fig.2. Curves of the dependence q1/q=f(m1/m) for a single-stroke EMVM with biasing and nonlinear elastic elements.

Various authors employ a number of approximate methods to solve equation (1), such as the harmonic balance method (Galerkin–Ritz method) [2, p.126], the small parameter method [3, p.66], and the direct linearization method [4, p.56]. Each author argues that alternative methods are unsuitable for different reasons. However, it was shown in [5, p.185] that these approximate methods are essentially equivalent when applied to equations of type (1) and yield practically identical results. Unlike [6, p.58], where an oscillatory system under harmonic excitation is considered, here we assume the action of the constant component of the nonsinusoidal traction force in a single-stroke EMVM

$$.F = F_0 + F_n \sin \omega t \qquad (2)$$

He armature is displaced by a distance A0 from its equilibrium position (in the absence of an external disturbing force) and oscillates around this point according to the law where a is the oscillation amplitude, ω is the angular frequency, and φ is the initial phase.

$$\chi = A_0 - B_0 + A_1 \, sin(\,\omega t - \gamma_1) + A_2 \, sin(\,2\omega t - \gamma_1) + \cdots. \qquad (3)$$

When buffers are introduced into the elastic elements of the EMVM, as shown in Fig. 1a, the constant component of the armature oscillations B0, caused by the presence of elastic elements with nonlinear asymmetric restoring force characteristics, will be directed away from the EMVM core and opposite to A0. If B0 is determined by the equation

$$B_0 = \frac{k''+k'}{k''-k'} \cdot A_1 - e - \sqrt{\left[\frac{k''+k'}{k''-k'} \cdot A_1 - e\right]^2 - (A_1 - e)^2} \quad (4)$$

the buffer gap e and its stiffness k must be selected so as to satisfy the condition B0=−A0. Then e is determined as;

$$e = (A_1 - A_0) \pm \sqrt{(A_1 - A_0)^2 - \left[A_1 - \left[\frac{k''+k'}{k''-k'} - 1\right] \cdot A_0\right]^2} \quad (5)$$

Here, A0 is determined as if the system were linear and subjected to the traction force (2), and thus the armature oscillations occur according to the law

$x = A_0 + A_1 \sin(\omega t - \gamma)$. Then, in equations (3), (4), and (5), the oscillation amplitude is determined as

$$A_1 = \frac{F_m}{m\sqrt{\left[\omega_0^2(A) - \omega\right]^2 + 2n^2(A)\omega^2}} \quad (6)$$

$$\omega_0^2(A) = \frac{K(\chi)}{m} = \frac{1}{m}\left[K' + \frac{2}{\pi}(K'' - K')\left[\arcsin\frac{e}{A} - \frac{e}{A}\sqrt{1 - \left[\frac{e}{A}\right]^2}\right]\right] \quad (7)$$

$$\rho(\chi) = 2\pi n(A) = \frac{\mu}{\omega} = K' + \frac{2\mu_2}{\pi\omega}(K'' - K')\left[\arcsin\frac{e}{A} - \frac{e}{A}\sqrt{1 - \left[\frac{e}{A}\right]^2}\right] \quad (8)$$

Just as in the single-stroke EMVD with bias magnetization and linear characteristics of elastic elements, in the EMVD with nonlinear characteristics of elastic elements it is also possible to choose one of two control options for A_1: either by changing the supply frequency with a simultaneous change in the U/f ratio at fixed values of Φq, or by changing f and Φq while keeping the applied voltage constant.

The constant component of the armature vibrations B_0, caused by the presence of nonlinear elastic elements, depends-apart from other mechanical quantities-also on the amplitude of the fundamental harmonic of the armature vibrations A_1. Substituting its value from (6) into equation (3), one can determine the dependence of B_0 on the electromagnetic parameters of the EMVD. However, for vibration machines where it is necessary to maintain A_1 unchanged when the mass of the vibrating load varies, B_0 will also remain constant (Fig. 3).

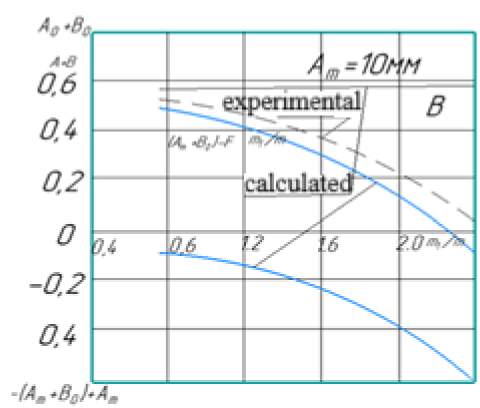

Fig. 3. Dependence of A_0, B_0, and $A_0 + B_0 = f(m_1/m)$ of a single-stroke EMVD with bias magnetization

In Fig. 4, the amplitude-frequency characteristic of a single-stroke EMVD with bias magnetization (by the amplitude of the fundamental harmonic of vibrations) and piecewise-linear characteristics of the elastic elements is presented, constructed from the calculation results using the analytical dependencies (6) and (7). Unlike the linear system, the resonance curve in this case is distorted and inclined to the right, where each frequency in the range from ω_e to ω_s corresponds to three amplitude values.

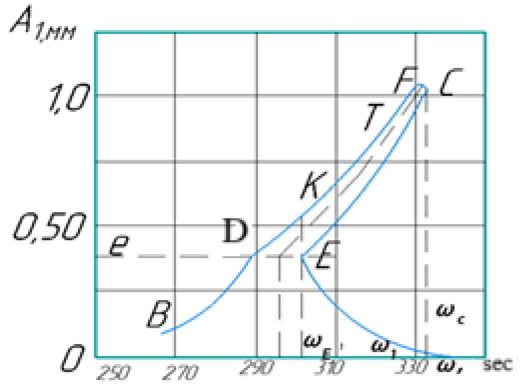

Fig. 4. Amplitude-frequency characteristic of a single-stroke EMVD with bias magnetization

However, not all of these amplitudes are stable. With an increase in frequency, the segments of the characteristic **BDKC** and **EM** correspond to the stable amplitude of the armature vibrations, whereas the segment **EC** belongs to the unstable zone and can practically not be obtained experimentally. Point **C** is the point of breakdown, and point **E** is the point of amplitude jump.

RESEARCH RESULTS

The segment **KFC** of the resonance curve is flatter than the corresponding branch of the amplitude–frequency characteristic of an EMVD with a linear elastic characteristic. This feature of nonlinear systems of the considered type is of particular importance, as it enhances the stability of vibration machine (VM) amplitudes in the resonance region and substantially improves their operational reliability under load variations (in the case of non-regulated motors).

However, when the technological load varies over a relatively wide range-for instance, in vibration platforms-even this extension of the stable operating region of the VM by incorporating a nonlinear structure proves to be insufficient. In such cases, it becomes necessary to regulate **A** by simultaneously adjusting both the frequency and the supply voltage magnitude (q).

It should be emphasized that maintaining **A** at a prescribed level within the **KF** zone of the amplitude–frequency characteristic (Fig. 4) requires frequency adjustment in only one direction-towards higher frequencies. To transition along the characteristic from point **M** to point **T**, the oscillation frequency must first be reduced to a value $\omega < \omega_e$, and only then increased to $\omega = \omega_0$.

CONCLUSIONS

The study demonstrates that effective amplitude control of the EMVD working body is achieved only after tuning the system to resonance. The proposed frequency control strategies and recommended measures ensure reliable regulation of output parameters while preventing armature–core impact. Furthermore, the analysis of nonlinear circuit elements and the corresponding analytical expressions provide a basis for determining the harmonic composition of the winding current, thereby offering deeper insight into the electromagnetic behavior of EMVD systems.

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