318. Vibration Dampers for Transmission of Mechatronic Systems

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Abstract. The paper reviews original torsional vibration dampers retaining their efficiency in a wide disturbing frequency band. Some potential design alternatives are considered. The basic structural element of the damper is a rotary flexible ring. The mathematical model of the system estimates inertia, flexible and gyroscopic terms. The equations to calculate the system motion are derived and stability conditions for the system dynamic balance are formulated, proper consideration is given to other possible stability loss forms and regions of the flexible ring. The analysis of the system natural frequencies is made. Efficiency estimation of the damper versus various parameters is effected following the expression of the equivalent inertia moment and its limit values.

Keywords: transmission of mechatronic system, torsional vibration damper, rotary flexible ring, vibration, stability, equivalent inertia moment.

1. Introduction

New devices for mechatronic systems should be ones of high efficiency. High efficiency may be ensured by power- and speed-related properties.

Increase of transmittable powers and speeds of motion is accompanied by intensification of vibrations in the systems and such vibrations frequently exceed their dynamic loads. Level of vibrations becomes one of the key criteria of quality and reliability of machines. Because of this, a limitation of dynamic overloads of machine assemblies is an urgent problem, directly related to increase of efficiency, reliability, accuracy and longevity of machines, mechanisms, assemblies and devices.

There is a variety of designs of dampers of torsional vibrations that may be naturally inserted into the structure of relevant unit. Seeking for natural arrangement is one of causes of the above-mentioned variety of designs; however, successful designs of vibration dampers are few.

An advantage of frictional dampers of torsional vibrations is their capability to preserve their efficiency in a certain frequency range. However, seeking for essential efficiency in such case leads for non-proportional increase of sizes, weight and so on, the more that frictional damping of vibrations is bound with elimination of heating energy, wear and use of special materials.

Dynamic dampers of torsional vibrations, in spite of their small sizes, are capable to reduce efficiently vibrations of single fixed frequency, however, they become almost unfit, if they operate by variable revolution or floating frequencies of disturbance.

Many works – both theoretical and experimental – were devoted to investigations on vibration processes in mechanical rotor's systems [1…3].

The Authors set the task to develop such dynamic damper of torsional vibrations that would be tunable for wide range of disturbing harmonics (frequencies), remaining a natural element of the rotating system.

2. The schemes of constructions and principle of operations of vibration dampers

The scheme of construction of the simplest damper of torsional vibrations in shape of rotating elastic ring [4] is presented in Fig. 1,a. The key element of such vibration damper is the ring 2 connected to the principal system 1 with two opposite frames 3. The ring may be equipped with supplemental masses 5 (Fig. 1,b,c), perpendicular to masses 4.

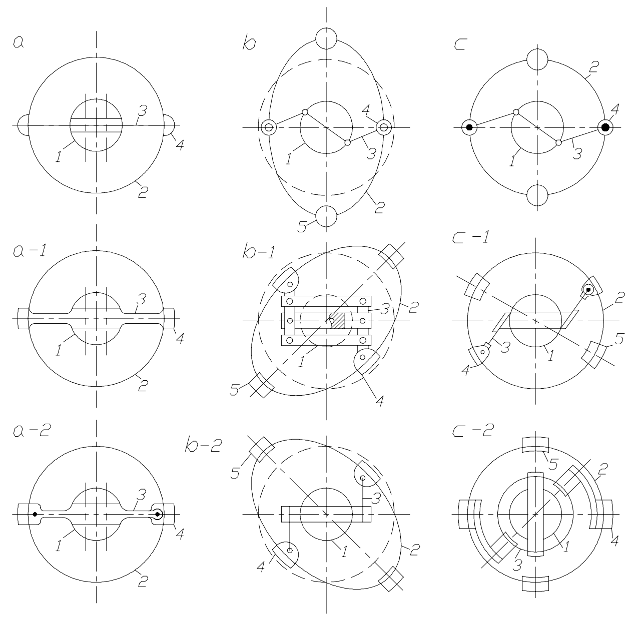

Fig. 1. Schemes of dampers of torsional vibrations based on an elastic ring

In case of absence of rotation or ideally uniform rotation, the axial line of the ring is an ideal circle in the limits of stability. Torsional vibrations of the principal system cause bending of the elastic frames 3 and periodical compression of the ring 2 in the transversal direction. The ring 2, because of its elasticity and „centrifugal mass“, may efficiently damp torsional vibrations of the shaft 1 upon certain parameters in wide range of harmonics.

The technical realization of the vibration damper according to this scheme is presented in Fig. 1, a-1, a-2. In Fig. 1, a-2, the elastic frames are connected with the ring by swivel clamps.

If two supplemental masses of certain size are fixed to the ring, symmetrically to the axis of rotation in the plane perpendicular to the plane of the elastic frames (see Fig. 1,b), the ring is extended into an ellipse-shaped body on the rotation. In many cases, such extended ring distinguishes for improved vibration damping properties. The bent centrifugal pendulums 3 may be stabilized by swivel parallelogram (Fig. 1, b-1) or replaced with symmetric elastic frames 3, tilted for a certain angle with respect to the radius (Fig. 1, b-2).

In many cases, efficiency of vibration damper may be increased by fixing the elastic ring on two symmetrically tilted pendulums (see Fig. 1, c). The tilted pendulums may be elastic frames (Fig. 1, c-1) or elastically fixed tilted pendulums (Fig. 1, c-2). The vibration damper presented in Fig. 1, c-2 includes one more elastic ring of small diameter 3 in the middle, clamped in two opposite points. The ring is an elastic swivel.

3. Investigation on operation of the vibration damper

The investigation is based on Lagrange equations of the second degree [5].

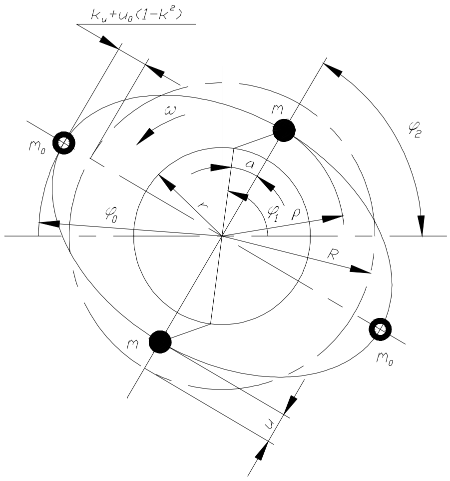

Fig. 2. The estimate scheme of the vibration damper

Fig. 2 presents the estimated scheme of vibration damper, where φ_1 and φ_2 are independent angular coordinates, R – the initial radius of the elastic ring, m – the reduced mass of the part of the ring with the relevant concentrated mass, m_0 – the reduced mass of the part of the ring with the supplemental mass, ρ - the distance of the mass m from the axis of rotation, ρ_0 – the distance of the mass m_0 from the axis of rotation, ρ and ρ_0 in general case are the complex function of rotational deformation of the vibration damper. If the radial shift of the mass $m(u_1)$ is a function of (φ_1 - φ_2), the radial shift of the mass m_0 includes also the independent component u_0 that is the third generalized coordinate.

After the differentiation, we find the equations that describe motion of the system, neglecting the friction forces and finally formulate the conditions of stability of the system.

On experimental investigation on various schemes of dynamic vibration dampers on base of elastic ring, some other forms of loss of stability were obtained: 1) because of an excessive increase of the ring's radius for its extension, 2) because of symmetrical "deflection" of the ring from the axis of rotation, 3) because of non-symmetrical "deflections" sideward and so on. On the base of the experimental data, an analytic investigation on stability of dynamic balances of the elastic ring had been carried out at the preset static deformation and some peculiarities had been cleared up.

The criterion of stability of dynamic balance shall be considered an existence of the maximum of kinetic potential in the preset position (point). In such case, if kinetic energy itself is equal to the maximum, the stability will be considered "natural" and the position of balance will not depend on the mode of speed.

If in the point under investigation, kinetic energy is equal to the minimum or at least does not depend on the disturbance under discussion, the forced stability will be possible only in this point, i.e. we'll consider that rigid forced stabilization is possible due to elastic elements. In other cases, ensuring of forced stability is possible on a certain shift of the point of dynamic balance. The size of such shift depends, among other factors, on the mode of speed. However, such shift usually is bound with an appearance of a certain instability that is impermissible in vibration damping system.

Let's discuss upon various cases:

1. Stability of ideally symmetric concentric ring in case of its uniform rotation.

In such case is supposed that concentrity of the ring is ensured in any case and no static bend of it exists and the limit value of the angular speed will be:

$$\omega^2 \langle \frac{EF}{\gamma_n R^2}. \tag{1}$$

The condition (1) identifies the limit of the zone of stable extension of the ring.

2. Symmetrical longitudinal extension of the ring.

As the carried out investigations showed, the maximum efficiency of vibration damping is achieved, when $z > 1$ (in Fig. 2, $z = m_0/m$) and the ring operates being extended. If we consider that the ring is deformed according to the scheme provided in Fig. 2, the element and more precise investigation provides that the angle α=0, if $z < 1$, and $\alpha > 0$, if $z > 1$, i.e. deformation, identified with the torsion angle α, when $z > 1$, has stable fixed value, dependent on rigidity of the elastic ring and other elements.

It is notable that bending rigidity of the ring itself does not affect the limits of zones of stability (z = 1), it only defines (together with other parameters) the value of static deformation.

3. Symmetrical "deflection" of the ring sideward.

Element investigation of stability of positions of dynamic balance upon the condition of extremity of kinetic potential allowed making the following conclusions:

a) zero deformation of the ring is possible only on z < 1/3 (if the ring is fixed on swivels of pendulum frames);

b) if z = 1/3, a certain statistical deformation of the ring is set dependently on the mode of speeds and rigidity of the rings;

c) if z > 1/3, the ring does not achieve a natural balance, so a symmetric deflection may transform into non-symmetrical one.

Some other possible disturbances of the ring were discussed upon and the conditions of balance were explored as well.

4. Investigation on efficiency of operation of the vibration damper

As specific quantitative calculations showed, it is sufficient to describe the deformed ring in many constructions of vibration dampers with two generalized coordinates only, i.e. it may be considered that a transversal compression of the ring is proportional (in some cases equal) to its longitudinal extension [5].

In such case, the expression of kinetic energy is reduced to the well-known quadratic trinomial.

According to the methods [5], we find linearized equations of small torsional vibrations that may be used for assessment of stability of dynamic balance and efficiency of vibration damper. To a certain extent, such cases are described in works [4, 5] as well.

In this case, stable positions of dynamic balance of the ring are described by one of the following conditions:

a) if $z < 1$, $\alpha = 0$,

b) if $z > 1$, $$\alpha = \arccos \frac{C_{n1} - 4zm\omega^2}{C_{n1} - 2(1+z)m^2}, \quad (2)$$

where $C_n = \frac{1}{4}\left(\frac{\pi}{4} - \frac{2}{\pi}\right)\frac{R^3}{EI}$, E – module of longitudinal elasticity, I – moment of inertia of cross-section of the ring.

For b) case, the minimum cross-section of the ring (its axial moment of inertia) will be found from the following inequality:

$$I \geq \frac{\pi^2 - 8}{16\pi} \frac{(3z+1)m\omega^2}{E}. \quad (3)$$

The condition (3) can be used for definition of critical frequency (ω_c) of rotation and the zone of stability (Fig. 3).

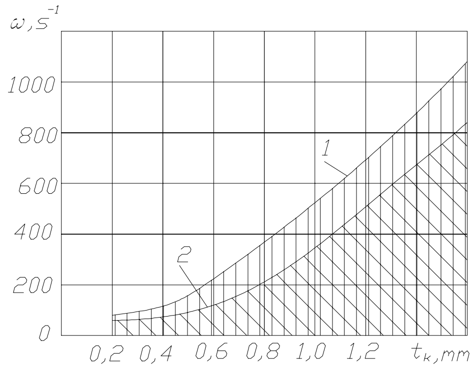

Fig. 3. Dependence of the critical frequency of rotation (ω_c) of the damper on the thickness of the elastic ring (t_k), when b = 20 mm, z = 2,0, m = 0,03 kg, where the curve 1 corresponds to R = 80 mm and the curve 2 – R = 100 mm (the zones of stability are shaded)

The efficiency of the vibration damper depends on the natural frequencies of damping system. Mentioned frequencies were established by solving the equations of small free vibrations around the position of stacionary motion. The frequency of resonance tuning of the vibration damper p^c can be established according to simplified equality:

$$p^c = \omega * \\ * \sqrt{\frac{\frac{C_{n1}}{2m\omega^2}\left(\cos\alpha - \cos 2\alpha - \left[zk(1+k)\cos\alpha - (1+zk^2)\cos 2\alpha\right]\right)}{1 + z + 2zk(1-\cos\alpha)(1+k)}}. \quad (4)$$

Fig. 4 illustrates some curves of natural frequencies of vibration damper.

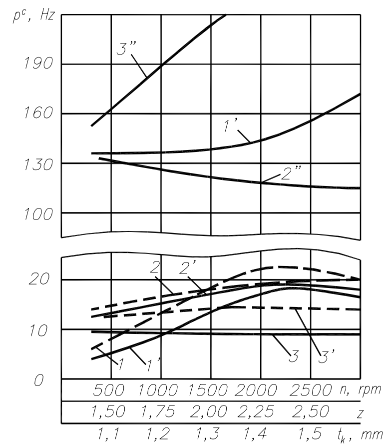

Fig. 4. Dependence of the natural frequencies of vibration damper (p^c) on its structural parameters and the number of revolutions, when the radius of the elastic ring R = 80 mm, m = 0.03 kg, n = 1500 rev/min., z = 1.25, t_k = 1.1 mm, b = 20 mm (the solid lines are obtained using the equations of small free vibration and the dotted ones – using the simplified calculation (4)), where the curves 1, 1', 1'' – p^c = f (n), 2, 2', 2'' – p^c = f(z), 3, 3', 3'' – p^c = f (t_k), the index ' corresponds to the first frequency and the index '' – to the second frequency

Conclusions

The general results of the investigation are provided in the presented schemes of vibration dampers:

- the schemes a, a-1, a-2 (Fig. 1) distinguish themselves for natural stability, if $z < 1/3$. In such case, frames 3 may be designed even in shape of strings that resist to extension only,
- if $z > 1/3$, a rigid forced stabilization may be achieved because of elasticity of the frames 3,
- the vibration dampers, showed in the schemes b and c may be stabilized by introduction of a relevant elastic element, resisting to deflection of the frames 3,
- the remained schemes distinguish themselves for rigid stability and preserve their strict symmetry on relevant rigidity of elastic elements. Side stabilization of extended elastic ring may be ensured by connection of the deflected centrifugal pendulums 3 with swivel parallelogram (see Fig. 1, b-1) or replacement of centrifugal pendulums with symmetric elastic frames 3, situated at a certain angle with respect to the radius (see Fig. 1, b-2),
- the analytical investigation on efficiency of vibration damper, applying special sets of programmes, enables to state that a vibration damper may be sufficiently precisely presented as a vibrating system with two degrees of freedom for most practically important ranges,
- in many cases, methods of simplified calculation ensure a sufficient accuracy,
- efficiency of a vibration damper increases on increasing its natural frequencies.

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