Methods for Vibration Reduction in Enclosed Electronic Packages

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Abstract. Excitation and transmission of vibration inside electronic packages are strongly connected with mechanical characteristics of their enclosure, which are in the objective for the current research aimed at developing methods and means to reduce vibration inside the enclosure case. The main idea of the introduced methods is in creating elastic and dissipative mechanical joints between structural elements, which appear to be the links for vibration transmission, inside and outside of the electronic package enclosure. The effectiveness of developed methods is proved by experimental verification and theoretical foundation.

1 Introduction

Vibration analysis of enclosed electronic packages represented in [1-3] disclosed that printed circuit boards (PCB) installed inside electronic packages may receive resonant oscillations, which amplitudes increase manifold and cause breakages of parts and components or deviations of electronic parameters. Excitation and transmission of vibration inside electronic packages are strongly connected with mechanical characteristics of their enclosure, which are in the objective for the current research aimed at developing methods and means to reduce vibration inside the enclosure case.

Despite variety of vibration isolators [4-7] the reliable and simple wideband isolator applicable to electronic packages has yet to be created. Now the close attention is paid to damping methods having inherent drawbacks though. For an instance, the widely used damping by encapsulation of electronic packages with vibration absorbing materials makes the package not maintainable and causes internal stress and hence damages to sealed components and sealant when subjected to thermal impacts as it was demonstrated in [8,9].

The previous research [1-3] verified that developing methods for protection against vibration and avoiding resonances is achievable by complex using of several methods unlike improving the single one. However the leading role in the task belongs to damping.

2 Mechanical characteristics of enclosure case

Enclosure case of electronic package represents one of the links in vibration transmission to PCBs installed inside. Hence varying its mechanical characteristics, which are elastic and dissipative parameters, such as stiffness, are expected to effect vibration transmission in purpose to reduce it. In this purpose the enclosure case was varied by changing stiffness of the bearing walls and introducing three modifications: 1) enclosure case with walls of 1 mm

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thick aluminum; 2) enclosure case with walls from damping material (1 mm thick fiberglass); 3) enclosure case with walls removed.

The tested objects were represented by two types of printed circuit boards PCB_1 and PCB_2, which differed in elastic and inertial characteristics: stiffness $C_1 < C_2$ and mass $m_1 < m_2$. Vibration analysis for PCBs was conducted by two parameters measured on their surface: vibration amplitude and strain. Vibration sensor (V) and strain gauges (S) were attached to PCBs as shown in fig. 1.
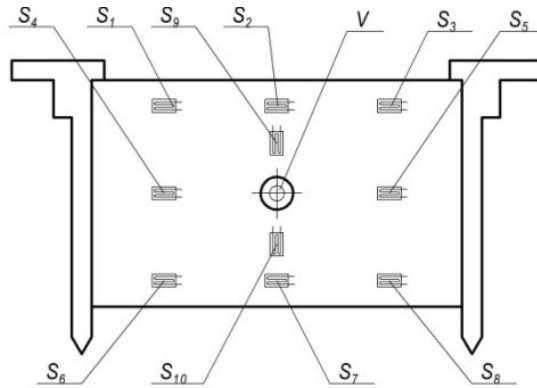
Fig. 1. Layout of vibration sensor and strain gauges on printed circuit boards.

The generated frequency and amplitude of testing vibration was controlled by the shaker VEDS-200A. In the experimental setup the tested electronic package was installed on the shaker so that PCBs inside were in horizontal position and direction of generated vibration coincided with their minimal stiffness; in such position oscillation amplitude of PCBs is maximal. Vibration tests were performed by method of fixed resonance frequencies, which had been identified by trial tests conducted before by using varying frequency method in diapason from 0 to 500 Hz; important that generated vibration acceleration was maintained stable. Noteworthy is that tests were conducted in frequencies under 100 Hz, since the higher diapason (100-5000 Hz) had no excitation of oscillations in the tested objects observed. Tests were performed for all modifications of cases. Tightening torque of bolts mounting tested case onto shaker table was maintained equal to 2.8 N·m in order to eliminate its effect to vibration transmission.

Results of experiments were represented by amplitude frequency responses obtained for two types of PCBs in three modifications of the cases (fig. 2). To unify measured values they were converted into relative values introduced as vibration transmission:
$$w = a_b / a_0 \tag{1}$$
where: a_b – vibration acceleration measured on printed circuit boards, m/s²; a_0 – vibration acceleration generated by the shaker, m/s².
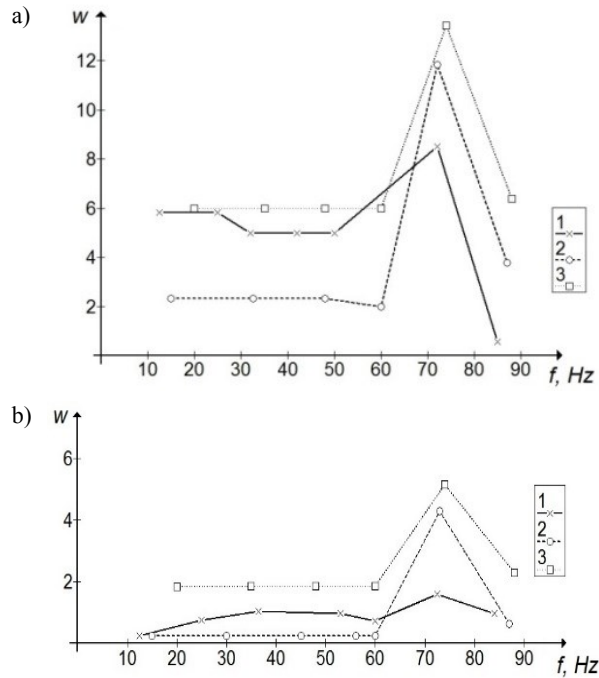
Obtained amplitude frequency responses of vibration transmission revealed that different enclosure case modifications demonstrate different vibration transmission to PCBs under the same level of applied dynamic force. Such difference is explained by their different damping level and confirms the statement that structural damping surpasses the damping produced by internal friction in material of structure parts.

The issue of the research was that decreasing case stiffness, in particular bearing walls, increases vibration transmission from the case supports to printed circuit boards. *w* in case with walls removed is 2-4 times higher, and in case with walls from fiberglass – 1.5-2 times higher in comparison with regular case with aluminum walls.

At the same time fig. 2 demonstrated negligible changes in resonant frequencies for both types of PCBs in different modification of enclosure case: type 2 increased resonance

frequency by 1-1.5 Hz and type 3 – by 1.5-2.5 with respect to type 1. Thus, offered changes in design of enclosure case were not effective for detuning from resonance.

Analysis of vibration acceleration measured for two types of PCBs revealed that decreasing elastic and inertial characteristics, such as mass and stiffness, increases amplitude response at resonance frequencies. This fact attracts closer attention to PCB_1.

Fig. 2. Amplitude frequency response of vibration transmission to PCB_1 (a) and PCB_2 (b) in types of cases: 1, 2, 3.

In order to estimate level of stress caused by vibration tests in PCB_1 strain values measured at resonant frequencies were recalculated [10,11] to stress values by Hooke's law for linear deformation. Mean values of stress were calculated for PCB substrate material (fiberglass) whose elasticity module is $E = 0.2 \cdot 10^5$ MPa.

The maximal vibration acceleration and stress were observed at frequency 72 Hz, which is the natural frequency for the researched PCB, whilst at other resonances, which were produced by other parts of enclosure case, these values are much lower. Those additional resonances depended on the stiffness of the case walls, which also effects to vibration transmission from the case to PCBs. Vibration acceleration and stress observed at additional resonances were negligibly low. The highest stress $\sigma = 30$ MPa was measured in PCBs installed in enclosure case without side walls. This stress was 3 times higher than that of the case with aluminum walls. Vibration transmission reached almost 9 times inside the case with aluminum walls and almost 13 times for the case without side walls.

Another experiment revealed influence of tightening torque of bolts mounting case to the shaker's platform on vibration transmission from the shaker to PCBs – growing tightening torque reduces vibration transmission to PCBs. This issue confirms the statement made above, that reduction of case stiffness increases vibration transmission to PCBs. In the test the case of type 1 with PCB_1 installed inside was subjected to vibration at the resonant frequency 72 Hz and vibration acceleration 50 m/s^2 along with gradual changing tightening torque of bolts.

Thus, high stiffness is required for building enclosure case even from damping materials in order to avoid additional resonances capable to increase vibration transmission inside the enclosure. Among the cases whose design differed in stiffness and other elastic, inertial and dissipation parameters the case with aluminum walls (type 1) was considered as the most effective for resonant oscillation reduction. Nevertheless, vibration transmission to PCBs in this case remains yet unacceptably high. Therefore searching effective methods reducing vibration transmission and excitation was considered actual for the next research.
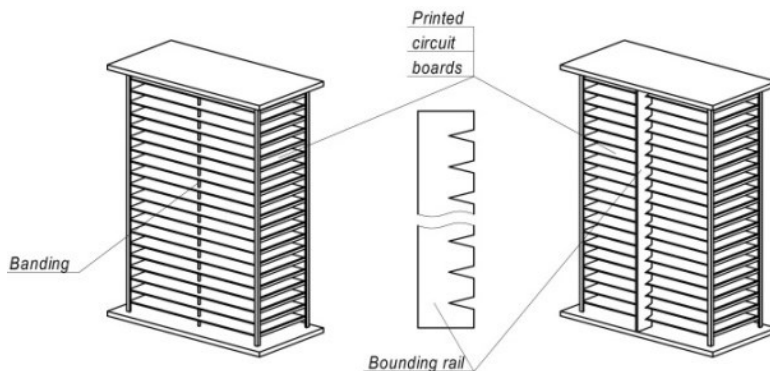
3 Elastic and dissipative joints of enclosure case

PCB_1 installed in case of type 3 was in objective in the next research. Case without side walls was selected in purpose to eliminate effect that walls produce to amplitude frequency response. The main goal of the research was developing methods for reduction of vibration transmission and excitation in PCBs, and the main idea consisted in building and studying elastic and dissipative joints in two options: 1) outside of enclosure case – between case and source of vibration; 2) inside of enclosure case – between its bearing parts.

The first option was introduced by two variants for installation of enclosure case on the shaker: 1) rigid installation, when enclosure case is firmly fixed on the shaker; 2) flexible installation, when enclosure case is set on the shaker through rubber spacers.

The second option was introduced by building three variants of mechanical joints into PCB installation: 1) banding; 2) inserting bounding rail; 3) inserting bounding rail through rubber spacers. All variants were tested in flexible installation of the package on the shaker.

Banding of PCBs was executed by using 4 mm thick cable passed through technological holes in all PCBs inside the case (fig. 3). Banding was applied with reasonable tension to provide necessary stiffness of mechanical joint.

Rails made from composite material were profiled with recesses cut at one side with spaces equal to installation steps for PCBs in the case (fig. 3). Another variant of using rails was inserting them through the rubber spacers.


Fig. 3. Parts of elastic and dissipation joints inside enclosure case.

Reading vibration was provided by vibration sensor IS567A-III attached to PCBs in their center. The second vibration sensor was attached to the shaker's platform. Procedure of vibration tests was identical to the previous chapter. Electronic package was subjected to vibrations at resonance frequencies, which had been detected before. The shaker generated vibration acceleration of 10 m/s^2 in frequency diapason 20-40 Hz and 50 m/s^2 in frequency diapason 50-2000 Hz. Every test was repeated three times to reduce measuring errors. Values of vibration transmission measured on PCBs in all variants of elastic and dissipative joints are represented in fig. 4. Tightening torque of fixing bolts was maintained equal to 1.4 N·m.

Figure 4 indicates natural frequency of PCB$_1$ around 60 Hz, what was then verified by striking method. All variants tested in the research demonstrated their effectiveness to reduce vibration transmission to PCB$_1$ although its natural frequency was not noticeably changed.

The highest effectiveness was achieved by inserting bounding rail, which more than 5 times reduced vibration transmission. Application of bounding rail has its obvious advantage as its usage is complimentary and does not change production technology. Close result was reached by inserting bounding rail with rubber spacers.
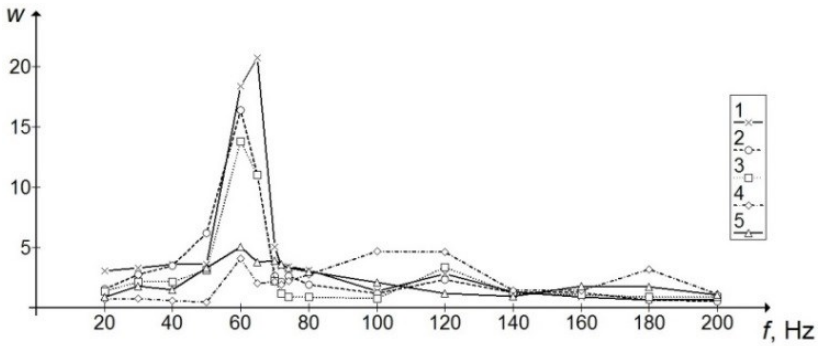
Fig. 4. Amplitude and frequency response of vibration transmission for enclosure case installations: 1 – rigid; 2 – flexible; and PCB installations: 3 – banding; 4 – inserting bounding rail; 5 – inserting bounding rail through rubber spacers.

4 Installation of printed circuit boards

Since vibration of PCB in resonance exceeds manifold vibration of its enclosure the next research was aimed at creation of elastic and dissipative joints in fixtures of PCBs inside the enclosure case. To perform variety of the fixtures in tests the vise appliance was built. PCBs fixed in vice were set on the shaker.

The following modifications represented PCBs fixtures as:
1. regular plastic fixture (fig. 5, a);
2. vibration isolating fabric tape (fig. 5, b);
3. vibration isolating rubber tape (fig. 5, b);
4. regular plastic fixture with foam rubber inserts;
5. special fixture pasted over by rubber tape;
6. vibration isolating fabric tape with dry friction damper (fig. 5, c).

The test procedure was identical to the previous research. Vibration accelerations were read from two piezoelectric sensors: one attached to the center of PCB and another – to the shaker's platform. Amplitude and frequency response of vibration transmission shown in fig. 5, d indicated the fixture of PCB on the vibration isolating fabric tape with dry friction damper as the most effective protection method against vibration.
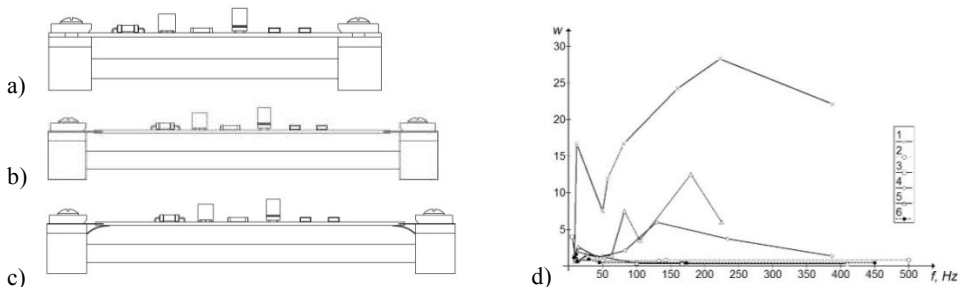

Fig. 5. Modifications of PCBs fixtures and their amplitude and frequency responses.

The dry friction damper was made from 0.3 mm thick bronze plate of 40 mm width and 15 mm length. The clamping force applied by the plate to PCB made 1.48 N. The value of this force was found with consideration of the plate deflection measured by the clock indicator and elasticity modulus.

Analysis of the complex application of the most effective methods, represented before, given as amplitude and frequency responses (fig. 6, a) and diagram of resonance values (fig. 6, b) for vibration isolating fabric tape with dry friction damper as a PCB fixture method and mentioned types of elastic and dissipative joints inside and outside of the electronic package enclosure, brought to the following statements:

1. using flexible installation of the case on the shaker governs increasing of vibration excitation of PCBs irrespective to types of mechanical joints inside the case. Herewith the highest degree of such excitation is observed for banding;

2. using bounding rail demonstrates the highest efficiency to reduce vibration when the case is firmly set on shaker as rigid installation. This is explained by the maximal level of structural damping provided by mutual movement of contacting surfaces introduced by dry friction damper;

3. increasing level of vibration excitation for flexible installation of the case on the shaker is explained by reduction of structural damping of vibration caused by insignificant mutual movement of contacting surfaces.
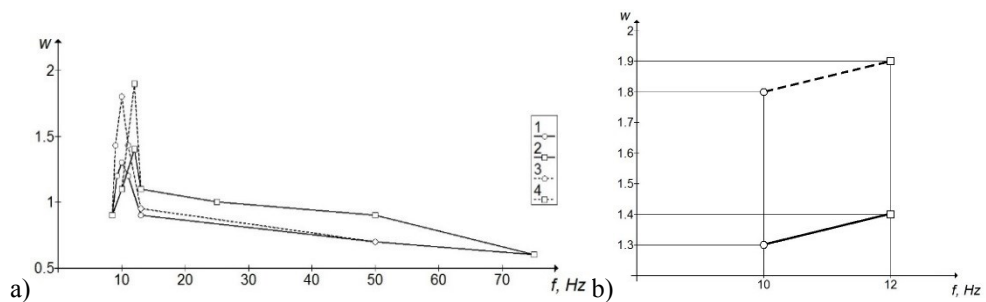
Fig. 6. Amplitude and frequency responses (a) and diagram of resonance values (b) of vibration transmission from the case supports to PCBs, fixed by using vibration isolating fabric tape with dry friction damper, in combination with using elastic and dissipative joints inside the case (ʃ – banding; ○ – bounding rails) and outside the case (solid line - rigid installation; dash line - flexible installation).

## 5 Vibration isolation of the whole enclosure case

In order to secure the whole electronic package, including all components, subjected to external vibration another attempt was made by offering seismic enclosure case suspension. The suspension is introduced by insertion of rubber pneumatic chambers in between external and internal cases of enclosure, so their elastic joint is maintained by pressure in the chambers. In such kind of joint adjusting chamber pressure controls resonance frequency of the whole system and shifts it away from the resonance.

The experimental tests of the offered idea were conducted on the shaker for the following variants:

1. rigid installation of the case on the shaker;
2. foam rubber is inserted between PCBs;
3. the case is installed inside vibration isolation case, pneumatic chambers are under high pressure, foam rubber is inserted between PCBs;
4. the case is installed inside vibration isolation case, pneumatic chambers are under high pressure;
5. the case is installed inside vibration isolation case, lower pneumatic chambers are under half-pressure.

For all the variants the case was fixed on the shaker by rigid installation.

The test procedure was identical to the previous research. Vibration accelerations were read from two piezoelectric sensors: one attached to the center of PCB and another – to the shaker's platform. Amplitude and frequency response of vibration transmission shown in fig. 7 indicated the variant 3 as the most effective protection method against vibration.
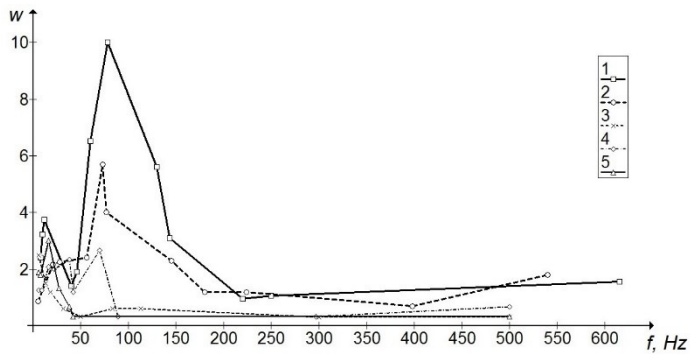
Fig. 7. Amplitude and frequency responses of vibration transmission from the case supports to PCB_1, for variants of vibration protection.

The advanced variant of vibration isolation was introduced by using dry friction damper (DFD) with suspension of the internal case by the soft springs inside the external enclosure (fig. 8, a) so that the natural frequency of the suspended case would make 3-8 Hz. Vibration at these frequencies is reduced by non-linear DFD, whose design is introduced by the following components: two plate springs 2, set in the bottom of external case 1, create wedge-shaped guides, which accommodate the rod 3 fixed to the internal case 4. The rod has two disks 5, made from hard rubber or fiberglass, attached to its surface.
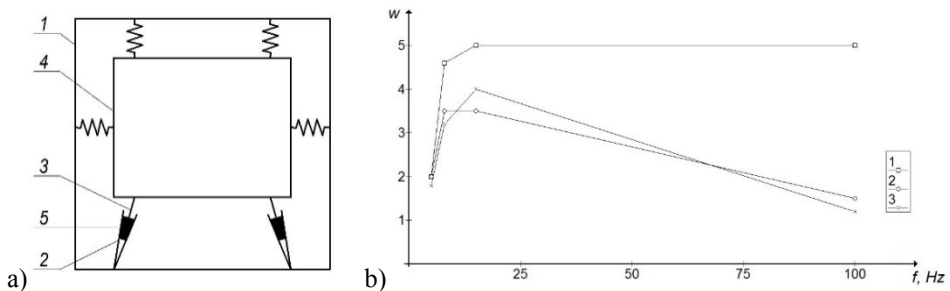
Fig. 8. Design of vibration isolation setup (a) and amplitude and frequency responses of vibration acceleration (b)

On the low frequency and high amplitude resonance of the external case, the wedge-shaped guides 2 moving along the rod 3 overcome friction force and dissipate vibration energy. The design of DFD illustrates that higher vibration amplitude increases movement of one case against another and hence increases the friction force because of growing normal pressure produced by the plate springs onto the rod. Quite to the contrary, on the high-frequency vibration, when amplitudes are small, DFD is not engaged at all.

The effectiveness of represented vibration isolation is demonstrated (fig. 8, b) by values of vibration acceleration measured on the external case (curve 1), internal case (curve 2) and printed circuit board (curve 3) correspondingly during vibration tests.

6 Mathematical model of suspension on the fabric tape with dry friction damper

Analysis of experimental data has verified that the most effective and simple methods for protection against vibration and hereby providing strength and reliability of enclosed

electronic packages subjected to external dynamic forces, like vibration, are: the installation or suspension on the fabric tape with dry friction damper for PCBs; and vibration isolation of the whole electronic package by using dry friction damper.

Suspension of PCBs on the fabric tape with dry friction damper gives the range of advantages: 1) PCB undergoes vibration in narrow frequency diapason – a few Hertz, depending on its weight. At higher frequencies outside of this diapason PCB remains still due to inertia and flexibility of the fabric tape – such structures are called as mechanical low-frequency filters; 2) low frequency vibration in the system proceeds without deformations of PCB, what secures through-hole and surface mount components; 3) easy access to components makes the assembly of such package maintainable unlike encapsulated packages sealed by compounds.

Since the damping in the fixture (fig. 9, a) is performed by friction of the friction plate 2 against the bearing pad 1, the magnitude of energy dissipation depends on friction force, vibration amplitude and other characteristics, which can be found from solution of differential equation that describes vibration of this system.

In mathematical model the fixture design is introduced by scheme that represents plate or beam oscillation on two elastic and damping supports (fig. 9, b, c). When stiffness of supports is considerably lower than that of PCB, PCB undergoes two resonances: 1st – by symmetrical form (fig. 9, b); 2nd – by skew-symmetric form of oscillation (fig. 9, c); further resonances are produced by deformations of PCB on the rigid supports.
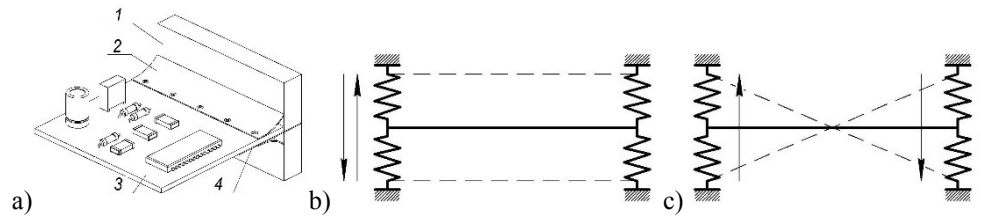
Fig. 9. Fixture design (a): 1 – bearing pad; 2 – friction plate; 3 – PCB; 4 – fabric tape, and PCB oscillation forms: b) symmetrical; c) skew-symmetric

Fig. 10 represents the 1st form of oscillation (fig 9, b), which is the most dangerous in condition of the offered fixture at low-frequency vibration. Mass m represents PCB 3 suspended by the spring 4 with stiffness k. The joint of the spring (case of enclosure) undergoes oscillation by the law $z = z_{max} \sin \omega t$, which engages mass movement. Friction plates 2 are attached to mass. They are pressed to bearing pads by the force that produces specified deflection.

Symbol x stands for PCB's movement, and z – for the supports' movement, which is the forced oscillation. Stiffness of the spring k is generalized parameter, which depends on tension (stiffness) of fabric tape, pressure force for friction plates, friction and other factors.

The mathematical model is brought to model of forced oscillations in single-mass system (fig. 10, a) [1,2] at presence of inelastic resistance forces produced by friction plates friction against surface of bearing pads:

$$R = \pm \frac{\alpha A^n}{\pi} \sqrt{1 - \left(\frac{x}{A}\right)^2} \qquad (2)$$
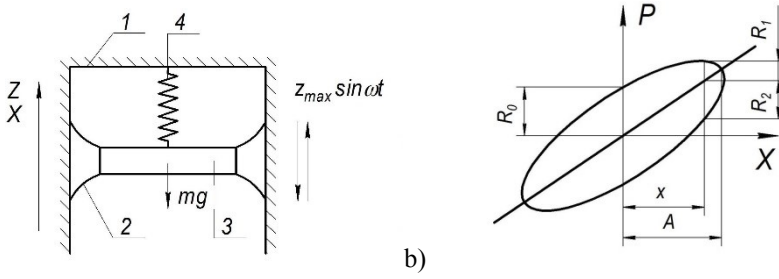
where: A – PCB's oscillation amplitude; α, n – constants, which depend on material.

Inelastic resistance force (2), indicated in the curve (fig. 10, b), represents deviation of the curve "force–movement" from linear Hook's law by hysteresis loop.

In accordance to [1,2] and considering (2) the forced oscillation of the single-mass system is described by equation:

$$-A\omega^2 \sin(\omega t - \gamma) + A\omega_c^2 \sin(\omega t - \gamma) + \frac{\alpha A^n}{\pi \cdot m} \cos(\omega t - \gamma) = \ddot{z} \sin \omega t \qquad (3)$$

Fig. 10. Oscillation model with dry friction damper (a) and force – movement diagram (b).

Assuming complexity of precise accounting effects for inelastic resistance force, which has nonlinear relationship with velocity $\dot{x}$, $R = b\dot{x}|\dot{x}|^{n-1}$, where b, n – constants of the structure, it was substituted by energetically equivalent linear force $b_0\dot{x}$ using method of energy balance. Coefficient of equivalent inelastic resistance b_0 was found from condition of equal works for these forces in half-period of oscillation.

By the acceptable substitution of inelastic resistance force by the friction force $R = \pm F_{\text{fr}}$, when $n = 0$ (dry friction), amplitude is represented as:

$$A = \pm \frac{\ddot{z}}{\omega_0^2} \sqrt{\frac{1 - \left(\frac{4F_{\text{fr}}}{\pi \cdot m \cdot \ddot{z}}\right)^2}{1 - \left(\frac{\omega}{\omega_0}\right)^2}} \qquad (4)$$

The first multiplier in the right part of equation represents static deviation, and the second one – dynamic factor, which has real values only in condition of $\frac{F_{\text{тр}}}{m \cdot \ddot{z}} < \frac{\pi}{4}$. Friction force is defined as:

$$F_{\text{fr}} = f_{\text{fr}} \cdot \delta \cdot E \frac{bh^3}{4l^3} \qquad (5)$$

where: f_{fr} – friction between surface of friction plate and bearing pads; δ – spring deflection.

Dimensions of the plate (spring) $l \times b \times h$ are specified by design considerations that comply with condition $F_{\text{fr}} < 0{,}785m\ddot{z}$. Given width b, length l, material E and stiffness $k = (2 \cdot \pi \cdot f)^2 m$ of the plate its thickness is calculated as:

$$h = \sqrt[3]{\frac{4Cl^3}{Eb}} \qquad (6)$$

By known friction the deflection of plate is found as:

$$\delta = \frac{F_{\text{тр}}}{2f_{\text{тр}}N} \qquad (7)$$

Obtained mathematical model provides optimization of oscillation system dynamic factor (4) with respect to geometric, elastic and dissipative characteristics of elastic and damping supports.

7 Conclusions

Vibration tests of different enclosure case modifications indicated that decreasing case stiffness increases vibration transmission to parts and components inside. Vibration transmission in case with walls removed is 2-4 times higher, and in case with walls from fiberglass – 1.5-2 times higher in comparison with regular case with aluminum walls.

High stiffness is required for building enclosure case even from damping materials in order to avoid additional resonances increasing vibration transmission inside the enclosure.

Building elastic and dissipative joints outside and inside of enclosure case demonstrated high effectiveness to reduce vibration transmission to printed circuit boards although its natural frequency does not change noticeably. The highest effectiveness was achieved by inserting bounding rail, which more than 5 times reduced vibration transmission.

Analysis of experimental data has verified that the most effective and simple methods for protection against vibration and hereby providing strength and reliability of enclosed electronic packages subjected to vibration, are: the installation or suspension on the fabric tape with dry friction damper for PCBs; and vibration isolation of the whole electronic package by using dry friction damper.

Obtained mathematical model provides optimization of oscillation system dynamic factor with respect to geometric, elastic and dissipative characteristics of elastic and damping supports.

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