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PROBLEMS OF TRANSMITTING VIBRATIONS TO ELECTRONIC DEVICES AT VIBRATION TESTS CONDUCTED ON VIBRATION STAND

Abstract - The paper represents problems of attachment for tested objects to the table of vibration stand by means of standard appliances. As proved theoretically the vibration stand oscillations do not always coincide with those of the attached object. The possible way for solving this problem has been indicated.

Keywords: vibration, vibration table, resonance, oscillation, attachment appliance.

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ПРОБЛЕМА ПЕРЕДАЧІ ВІБРАЦІЙ КОНСТРУКЦІЯМ РЕА ПІД ЧАС ВИПРОБУВАНЬ НА ВІБРОСТЕНДІ

В доповіді коротко викладена проблема закріплення випробовуваних об'єктів за допомогою стандартних пристосувань на столі вібростенда. Теоретично показано, що коливання столу вібростенда не завжди збігаються з коливаннями об'єкта, закріпленого на ньому. Вказаний ймовірний шлях вирішення проблеми.

Ключові слова: вібрація, вібростіл, резонанс, коливання, кріпильні пристрої, пристосування.

Introduction

The proper design, production and testing of electronic devices requires knowledge and consideration of conditions, in which those will be used.

In case of using devices in conditions of vibration and impacts then the following necessary to identify: diapasons of oscillation spectrum, amplitude, vibration speed, vibration overload and impact impulses characteristics. For electronic devices important conditions to consider are not only conditions in which they are used but also transported and stored.

Such systems are very often installed on mobile objects – helicopters, airplanes, ships, automobile and tracked vehicles, rockets, satellites and other. As the matter of fact they work being exposed the complex of destabilizing factors. The most dangerous of them are: vibration and impact loads, acoustic exposure, road shaking, shots, imbalance of fast rotating parts, shock waves, rapid wind blows, fast turbulent flows etc.

Table 1 demonstrates diapasons and maximal amplitude of oscillations in electronic devices installed on various mobile objects [1].

Table 1

Diapasons and maximal amplitude of oscillations in electronic devices installed on mobile objects

Type of electronic device	Frequency of vibration, Hz	Maximal amplitude, mm
Ground devices installed on the vehicles:		
- in the car case; on tractors, on armoured wheeled vehicles;	0–80	1–2,5
- on tracked tractors;	8–15	1
- on tanks.	400–700	0,25
Ground mobile units		
on airplane:	20–2000	0,25
on units with piston engines;	10–1200	0,15–2,5
- on units with jet engines	5–150	0,15–5
Rockets:	5–500	0,025–0,3
- on acceleration stage;	5–500	-
- on the flight.	30–5000	1–3
Ships:		
- on large tonnage ships	2–35	1
- on small tonnage ships	5–150	1
Used in rail transport.	2–10	≤35

Besides to recognize the sources of oscillation excitation is important in order to control them and in this way to eliminate or reduce vibration and impacts which they generate.

These sources can be internal, for example imbalance of rotor and fan blades, electric engines, transducers and other parts in electronic devices.

External sources of vibrations are considered first of all as vibration of those objects which devices are installed on or transported by. Besides, external sources of excitation include acoustic noise, wind blows, road shakes, sea waves etc.

MAIN PART

The most important link in the system that provides quality transmission of vibration onto the object during vibration and impact tests is appliance for attachment of the tested object to the source of vibration, in particular to electrodynamic vibration stand.

Various methods and appliances are well known and widely used in industry to attach objects as for vibration

tests so for manufacturing. The large part of them includes mechanical attachments designed to use bolts, arcs and clamps. The main requirement to these units is to maximally provide transmitting mechanical energy from the table of vibration stand to tested object, what corresponds to transmission factor equal one in the whole diapason of vibration test.

The most of the known mechanical fastening methods do not comply with such requirements [2]. Their application creates additional difficulties entailed by necessity to produce large amount of equipment for testing multiple products, which have different sizes, shape and do not permit a surface damage.

In the other cases glue, resin and compounds are used for this purpose, what in turn causes another problem. So using compounds requires filling operation and includes labour consuming operation to release an object from irreversible gluing.

Practice of testing objects by vibration and also using them on vibrating supports, within set standards, testifies of cases of damages in individual components and (or) parts of electronic devices. Measuring vibration in places of location of these components in devices has shown on frequencies of excitation that vibration amplitude of the components or vibration speed, or acceleration (vibration overload) dozens times exceeded analogous parameters of vibration detected on the table of vibration stand, or on vibrating support during operation. Developers of components claimed that destruction of components occurred during tests or operation due to unacceptably high levels of vibration. On the contrary consumers of electronic components, those who produce complex electronic devices, did not consider the claimed fact and assumed that testing of devices did not exceed acceptable limits of the standard. The assumption was made by controlling oscillations of the table of vibration stand or of the base, which the object was attached onto, in operational conditions.

In so doing vibration of the whole tested object and its components was considered equal to vibration measured on the table of vibration stand. Nevertheless the real situation is much more complicated.

We will show that vibration of the table (platform) of vibration stand in general does not characterize either vibration of fastening appliances or all the more vibration of the parts and components of structurally complex electronic device. That is why we will review a simplest case.

Figure 1 shows the scheme of the table of vibration stand with the object attached to it through fastening appliance having mass *m*.

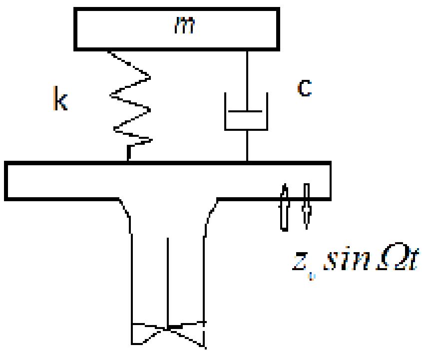

Fig. 1. Oscillating system

The attachment is not totally rigid, so fig. 1 shows the scheme of total mass *m*, which is supported by the table of vibration stand through the springs with rigidity *k* and damper with damping *c*.

Mobile table of vibration stand performs harmonic oscillations with amplitude z_0 from the law $z_0 \sin \Omega t$.

Symbol *z* stands for the downward vertical mass displacement. Then due to displacement of the table of vibration stand downward at the value $z_0 \sin \Omega t$ the spring expansion at any moment of time will be equal $z - z_0 \sin \Omega t$. The elastic force of the spring will be equal $-k(z - z_0 \sin \Omega t)$, and damping force proportional to the speed will be equal $-c(z' - z_0 \Omega \cos \Omega t)$. The next equation of oscillation of our system is obtained on the base of Newton's second law.

$$mz'' = -k(z - z_0 \sin \Omega t) - c(z' - z_0 \Omega \cos \Omega t)$$

$$\text{or} \quad mz'' + k(z - z_0 \sin \Omega t) + c(z' - z_0 \Omega \cos \Omega t) = 0$$

$$\text{or} \quad mz'' + cz' + kz = kz_0 \sin \Omega t + cz_0 \Omega \cos \Omega t \tag{1}$$

To find movement of mass m toward the vibrating table of the stand the derivatives are taken from spring expansion assuming that mass m moved at this moment together with spring at $z_1 = z - z_0 \sin \Omega t$

$$\text{or} \quad z = z_1 + z_0 \sin \Omega t\ .$$

Differentiating results are:

$$z' = z_1' + z_0 \Omega \cos \Omega t$$

$$z'' = z_1'' - z_0 \Omega^2 \sin \Omega t\ .$$

Substituting these expressions into equation (1) we obtain:

$$mz_1'' - mz_0 \Omega^2 \sin \Omega t + cz_1' + cz_0 \Omega \cos \Omega + kz_1 + kz_0 \sin \Omega t = kz_0 \sin \Omega t + cz_0 \Omega \cos \Omega t$$

Having similar expressions reduction at the right and left parts the final expression is:

$$mz_1'' + cz_1' + kz_1 = mz_0 \Omega^2 \sin \Omega t\ . \tag{2}$$

In this equation:

mz_1'' – force of mass inertia m;

cz_1' – damping force;

kz_1 – elastic force of the spring;

$mz_0 \Omega^2$ – inertial disturbing force.

Such disturbance of oscillations is called inertial one [3].

Thus, we have determined that mass displacement m (of electronic device together with fastening appliance) to vibrating base (table of vibration stand) occurs in the same way as absolute movement of the mass m at stationary base (table of vibration stand) when mass is exposed to harmonic disturbing force with amplitude $mz_0 \Omega^2$.

Forced oscillations of such single mass system are well studied so we will not focus on them.

We will define that amplitude and phase of such oscillations depend on closeness of frequency Ω of disturbing force to cyclic frequency ω_c of natural oscillations of the mass

$$\omega_c = \sqrt{\frac{k}{m}}\ .$$

Formula for oscillations of the mass *m* on rigidity *k* when $c = 0$ is

$$z = \frac{z_0 \sin \Omega t}{1 - \left(\frac{\Omega}{\omega_c} \right)^2}\ .$$

The expression demonstrates that only when ω_c is considerably higher than Ω mass *m* will oscillate almost in the same way as the base which is the table of vibration stand.

As Ω is approaching to ω_c oscillation of mass will grow and be out of phase. When $\Omega = \omega_c$ the resonance occurs as mass oscillation tends to maximal (∞), and phase difference – to 90°.

Once resonance is passed and farther more the mass oscillation is reducing and tending to zero, and phase delay is growing so far away from the resonance it reaches 180°, what means that mass (of tested object) oscillation will be directed opposite to oscillation of the table of vibration stand and will try to quench it. Practically since $\Omega = 10\omega$ oscillation of the table of vibration stand will not be transmitted to mass *m.*

The oscillation theory of double mass systems shows possibility to quench oscillations of one mass by means of another one that oscillates in counter-phase and is connected to the first one by the spring. This process is the base of dynamic oscillation quencher.

This problem is not within the frame of the current work and that is why will not considered here. The important fact for us which is theoretically demonstrated is that if mass m is not fastened absolutely firmly on the table of vibration stand then table oscillations may not coincide with oscillations of the mass.

Incidentally, transmitting oscillations from the table to tested object is desirable to conduct undistorted in 1:1 scale at all frequencies of excitation.

The real situation is more complex.

The first thing to consider is that table of vibration stand does not have perfect rigidity and on some frequencies has resonances and correspondent forms of oscillation, at which deformations along the table surface are not uniform, and the table with a rod as a mass is supported by low rigidity supports and has first resonance at frequencies under 5-10 Hz that is to be passed fast.

The transmission of vibration during vibration test is conducted in the next manner. The oscillating movement of the table in general case is first transmitted to fastening appliance. In so doing, as mentioned above, the plane of the table at some frequencies, in particular at resonance, can additionally to moving together with rod have deformation having form of table plane oscillation, which is correspondent to its resonance. And this means that vibrations will not be the same along the table plane. Hence, at different points where fastening appliance contacts with the table plane different values of the forces will be applied.

Noteworthy is that fastening appliance will have its own resonance and will increase or decrease (nonuniformly) vibrations generated by the stand.

Furthermore vibrations are transmitted, through fixtures by which object is attached to fastening appliance, to its case and then from the case to printed boards through their fixtures.

We understand that case components and printed boards have their own resonances and increased deformations and stresses in them. Finally the components on the board can have their own resonance and increased vibration received through places of their fixtures to the board.

In such multi-mass oscillating system every component is represented as mass, rigidity, exciter and inhibiter of oscillations, and growing amplitude can involve another neighbour components thus changing parameters of oscillating system: mass, rigidity, damping.

Therefore the task to reach in practice undistorted transmitting of oscillations from the table to tested object or its components in 1:1 scale is very complicated.

Noteworthy is that the situation becomes more complicated when rod axis of vibration stand and exciting force acting along this axis do not cross the mass centre of the object with fastening appliance. In case when the bend moment appears and so do related to it oscillations of the whole system in planes perpendicular to table plane of vibration stand, which superpose on main oscillations and make the total situation even more complicated.

SUMMARY

In order to avoid problems with resonance of the attachment appliances the no resonance ones are to be created in form of cylinder or cube. They are to be made from light and rigid materials which can be presented by dielectric materials (plastic, textolite etc.) or light materials (titanium, aluminium and duralumin etc.). The tested objects are to be attached to attachment appliances with the help of nanostructure electro rheological suspensions [4].

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