Problems of Tribology, V. 28, No 4/110-2023, 13-21

Problems of Tribology
Website: http://tribology.khnu.km.ua/index.php/ProbTrib
E-mail: tribosenator@gmail.com

DOI: https://doi.org/10.31891/2079-1372-2023-110-4-13-21

Analysis of the influence of shaker table rigidity on the accuracy of vibration test results for electronic equipment

A. Goroshko*, M. Zembytska

Khmelnytskyi National University, Ukraine
**E-mail: iftomm@ukr.net*

Received: 14 September 2023: Revised: 14 October 2023: Accept: 1 4 November 2023

Abstract

During vibration tests of structurally complex equipment on electrodynamic shakers, the measured vibrations on the shaker table, test fixtures and various places of the tested product may differ. One of the reasons is the deformable shaker table, the stiffness of which can significantly affect the results of vibration tests. The electronic equipment being tested, as a rule, does not have an axisymmetric shape and therefore, during testing, it is very difficult to align the center of gravity of the product and the vibrating table or equipment with the axis passing through the center of the shaker table. The paper theoretically shows that the shaker table can be considered a rigid body only if there is no displacement of the weight of the tested product from the center of the table, as well as at excitation frequencies significantly lower than the critical frequency of the table. For theoretical analysis, the disc-shaped shaker table is presented in the form of a thin plate. Vibrography of the vibrating platform experimentally confirmed the effect of increasing vibrations when moving away from the center of the shaker table. If during vibration tests it is not possible to place the electronic unit coaxially with the center of the vibrating table, it is necessary to monitor the value of vibrations at different points of the tested product, for example, at the points farthest from the center.

Keywords: vibration test, shaker, test fixtures, electronic equipment

Introduction

Products of electronic equipment used in space and aviation equipment are subject to significant overloads (10…20)g [1]. One of the main parameters of such equipment is their vibration resistance. The specified equipment is subject to increased requirements for vibration protection, since the failure of a relatively inexpensive interconnection, electronic unit or component can lead to the destruction of the entire aircraft [2]. Therefore, during production and operation, on-board electronic equipment is subject to mandatory testing on various types of shakers. Electrodynamic shakers are used for testing electronic devices, assemblies and aircraft units [3, 4]. The accuracy of vibration test results significantly depends on the accuracy of the measured vibration parameters. Idealizing a shaker table or test fixture as an absolutely rigid body increases vibration measurement errors. To improve the quality of vibration tests, it is necessary to take into account the deformability of the vibrating table and equipment.

Literature review

The most important link in the system for ensuring the quality of vibration impacts transmitted to the object during vibration and impact testing of products is the fixture for attaching the test product to the table of an electrodynamic vibration stand [5, 6]. Such fixturte can introduce distortions into the process of vibration transmission from the vibrating table to the product being tested [7]. Vibration monitoring based on readings from sensors installed on the shaker does not reflect real vibrations. Analysis of the results obtained from many vibration tests of electronic control units of aerospace equipment showed a significant spread between the level of vibrations measured on the shaker table and the vibrations of the test fixture or products being tested [8].

Copyright © 2023. A. Goroshko*, M. Zembytska. This is an open access article distributed under the Creative Commons Attribution License, which permits unrestricted use, distribution, and reproduction in any medium, provided the original work is properly cited.

When testing electronic equipment, which, as a rule, does not have an axisymmetric shape, it is very difficult to align the center of mass of the product and test fixture with the axis passing through the center of the shaker table. If the axis of the shaker rod and the exciting force acting along this axis do not pass through the center of mass of the product with the table, a bending moment occurs. The associated vibrations of the entire system in planes perpendicular to the plane of the shaker table are superimposed on the main vibrations and further complicate the vibration response. This has prompted many researchers to further investigate vibration transmission models during testing. [9-13].

The authors in [9, 13] created a mathematical model of an electrodynamic shaker, which takes into account the transmit of mechanical load from the launch vehicle to the spacecraft. Possible distortions of the center of gravity of the tested product are taken into account. For this purpose, modal analysis and the design of experiments (DOE) method were used. As a result, three shapes of the shaker head expander were proposed to select the geometry with the best vibration characteristics.

The work [10] shows the principle mechanical and electrical parts of a lumped-parameter, one-dimensional, three DoFs, electromechanical shaker model and describes a basic model of a Modal shaker. It was shown in [11,12] that due to the unbalanced radial electromagnetic force and the displacement of the center of gravity of the tooling or test sample, in practice, an electrodynamic shaker can generate significant vibration in the transverse direction. As a result, the accuracy of vibration control of the equipment under test is greatly reduced. To take into account the lateral vibration of the moving system, a dynamic model of an electrodynamic shaker with 7 DOF was developed in [11].

Purpose

The purpose of this work was a theoretical analysis and experimental study of the influence of the deformability of the VEDS-200 shaker table on the magnitude of the measured vibrations.

Research methodology

Dynamic model of vibrations in the assumption of an absolutely rigid shaker table

Let us consider the process of vibration testing of an electronic product mounted on a shaker, containing a housing and a printed circuit board. In this case, we assume that the shaker table is a rigid body with a mass that much exceeds the mass of the test product. The electronic unit under test, as an oscillatory system, consists of two passive masses, representing its body and board, which are interconnected by elastic elements.

In the general case, the oscillatory system is elastically connected to the movable platform of the shaker. The movement of the moving platform is determined by the kinematic excitation of oscillations of the system under study. Assuming a significant excess of the mass of the moving platform of the vibration stand compared to the masses of the body and board, a discrete model with 2 DOF, which is presented in Fig. 1.

The motion of such a two-mass oscillating system is described by a system of differential equations

$$\begin{cases} m_1\left(\ddot{z}_1-\ddot{z}_0\right)+k_1 z_1+k_2\left(z_1-z_2\right)+c_1\dot{z}_1+c_2\left(\dot{z}_1-\dot{z}_2\right)=0, \\ m_2\left(\ddot{z}_2-\ddot{z}_0\right)+k_2\left(z_2-z_1\right)+c_2\left(\dot{z}_2-\dot{z}_1\right)=0, \end{cases} \tag{1}$$

where m_1 is the mass of the body of the block, represented by a solid body installed on the elastic-dissipative support of the fastening node with a stiffness coefficient k_1 and a vibration resistance coefficient c_1;

m_2 is the mass of the electronic board, whose elastic-dissipative support models the elastic-dissipative properties of the board itself with a stiffness coefficient k_2 and a vibration resistance coefficient c_2;

z_i (i = 1, 2) is displacement of masses m_i;

$z_0=Z_0\sin\omega t$ – is the harmonic oscillations generated by the vibrating platform. The function $z_0(t)$ describes the movement of the moving platform of the shaker, causing the kinematic excitation of the vibrations of the system under study.

The solution of the system of equations (1) has the form:

$$\begin{cases} z_1=U_1\cos\omega t+V_1\sin\omega t, \\ z_2=U_2\cos\omega t+V_2\sin\omega t. \end{cases} \tag{2}$$

After substituting (2) into (1), we obtain a system of four linear algebraic equations for unknowns U_i, V_i (i = 1, 2):

$$\begin{cases} U_1\left(k_1+k_2-m_1\omega^2\right)+V_1\left(\left(c_1+c_2\right)\omega\right)+U_2\left(-k_2\right)+V_2\left(-c_2\omega\right)=-m_1 Z_0\omega^2, \\ U_1\left(\left(c_1+c_2\right)\left(-\omega\right)\right)+V_1\left(k_1+k_2-m_1\omega^2\right)+U_2\left(c_2\omega\right)+V_2\left(-k_2\right)=0, \\ U_1\left(-k_2\right)+V_1\left(c_2\omega\right)+U_2\left(k_2-m_2\omega^2\right)+V_2\left(c_2\omega\right)=-m_2 Z_0\omega^2, \\ U_1\left(c_2\omega\right)+V_1\left(-k_2\right)+U_2\left(c_2\omega\right)+V_2\left(k_2-m_2\omega^2\right)=0. \end{cases} \quad (3)$$

The amplitudes of mass oscillations m_1 and m_2 are determined by the formulas:

$$Z_1=\sqrt{U_1^2+V_1^2},\ Z_2=\sqrt{U_2^2+V_2^2}\ . \quad (4)$$

Given the statement of the problem, from system (3), using formulas (4) we obtain expressions for determining the relative amplitude of oscillations $\bar{Z}_n=\dfrac{Z_n}{Z_0}$ in particular, for mass m_2, which simulates the concentrated mass of the tested electronic device:

$$\bar{Z}_2=\frac{D_1 m_2\omega^2}{D_2}, \quad (5)$$

where

$$\begin{cases} D_1=k_1+k_2-m_1\omega^2+c_1\omega^2+c_2\omega^2, \\ D_2=(k_1+k_2-m_1\omega^2)(k_2-m_2\omega^2)-k_2^2-c_1c_2\omega^2+\omega\left[(c_1+c_2)(k_2-m_2\omega^2)+c_2k_1-c_2(k_2-m_1\omega^2)\right]. \end{cases} \quad (6)$$

In the considered model, the shaker table is presented as a rigid body. However, monitoring vibrations at various places on the shaker table shows that this is not the case, and in fact the table can become deformed [13, 14]. We theoretically study the influence of finite stiffness on the deflections of a shaker table made in the form of a disk.

Mathematical model of a shaker table represented by a flexible thin plate

A typical design of the moving part of an electrodynamic vibration stand is shown in Fig. 2. A vibration table with a thickness *h* and a radius *b* is rigidly bolted to a cylindrical rod with a radius *a*, which performs vertical reciprocating movements at the frequency of the generator of the shaker control system. As a rule, for most vibrating tables the following condition is met $0,025 \le h/a \le 0,2$, therefore, to estimate the rigidity and deflections of the vibrating table, you can use the theory of calculations symmetrical bending of thin circular plates with small deflections [15].

Let us consider an approximate model of a shaker table in the form of an elastic round plate, rigidly clamped along the contour with a radius *r*=*a* (Fig. 2). The load on the shaker table be distributed uniformly with intensity *q*.

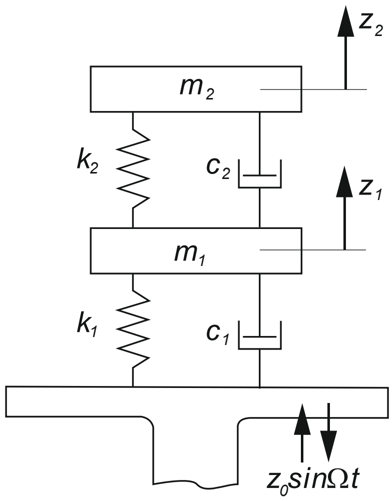


Fig. 1. Dynamic model of oscillations of an electronic product fixed on a rigid shaker table

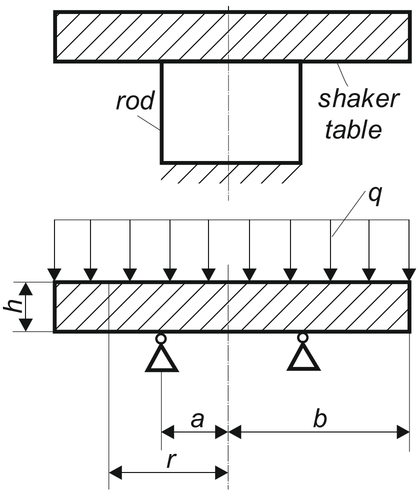


Fig. 2. Load scheme of the deformable shaker table by its own weight

In the polar coordinate system, plate deflection and load are functions of r and θ, respectively: $w(r,\theta)$, $q(r,\theta)$. The differential equation of a curved plate has the form [15]

$$D\left(\frac{d^2}{dr^2}+\frac{1}{r}\cdot\frac{d}{dr}+\frac{1}{r^2}\cdot\frac{d^2}{d\theta^2}\right)\left(\frac{d^2w}{dr^2}+\frac{1}{r}\cdot\frac{dw}{dr}+\frac{1}{r^2}\cdot\frac{d^2w}{d\theta^2}\right)=q\ , \tag{7}$$

where $w=w(r)$ is the deflection as a function of the coordinate r;
q is the uniformly distributed load on the surface;
$D=\dfrac{Eh^3}{12\left(1+\mu^2\right)}$ is the bending stiffness of the plate;
μ is the Poisson's ratio.

The problem is axisymmetric if the load on the plate, as well as the conditions for securing its edges, do not depend on the polar angle θ. Then the deflections do not depend on the polar angle θ, and they are only a function of the coordinate $w=w(r)$. The general differential equation for the equilibrium of the curved middle surface of a circular plate has the form

$$\frac{d^4w}{dr^4}+\frac{2}{r}\cdot\frac{d^3w}{dr^3}+-\frac{1}{r^2}\cdot\frac{d^2w}{dr^2}+\frac{1}{r^3}\cdot\frac{dw}{dr}=\frac{q}{D}\,. \tag{8}$$

Integrating (8), we obtain the equation of the rotation angles

$$w=\frac{qr^4}{64D}+C_1+C_2\ln r+C_3r^2+C_4r^2\ln r\,. \tag{9}$$

The linear radial bending moment is equal to

$$M_r=-D\left(\frac{d^2w}{dr^2}+\frac{\mu}{r}+\frac{dw}{dr}\right)=D\left(\frac{d\varphi}{dr}+\mu\frac{\varphi}{r}\right) \tag{10}$$

where $\varphi=-\dfrac{dw}{dr}$ is the angle between the tangent to the curved middle surface and the axis r.

To determine the integration constants in solution (7), we have the following boundary conditions:

1) $r=a$, $w=0$;
2) $r=a$, $\varphi=0$;
3) $r=b$, $M_r=0$;
4) $r=b$, $Q=\dfrac{dM_r}{dr}=-D\dfrac{d}{dr}\left(\dfrac{d^2w}{dr^2}+\dfrac{\mu}{r}+\dfrac{dw}{dr}\right)=0$;

Substituting the deflection functions into these boundary conditions $w(r)$ (7), moment $M_r(r)$ (8), angle $\varphi(r)$ and reduced shear force $Q(r)$, we obtain the following matrix system of equations:

$$\begin{pmatrix} 1 & \ln a & a^2 & a^2\ln a \\ 0 & a^{-1} & 2a & a\left(1+2\ln a\right) \\ 0 & \dfrac{\mu-1}{b^2} & 2\left(1+\mu\right) & 2\ln b\left(1+\mu\right)+3+\mu \\ 0 & \dfrac{2\left(1-\mu\right)}{b^3} & 0 & 2b^{-13}\left(1+\mu\right) \end{pmatrix}\cdot\begin{pmatrix} C_1 \\ C_2 \\ C_3 \\ C_4 \end{pmatrix}=\begin{pmatrix} -\dfrac{qa^4}{64D} \\ -\dfrac{qa^3}{16D} \\ -\dfrac{qb^2}{16D}\left(3+\mu\right) \\ -\dfrac{qb}{8D}\left(3+\mu\right) \end{pmatrix}. \tag{11}$$

Solving system (11) we find the integration constants C_1, C_2, C_3, C_4.

To illustrate the nature of the behavior of dependence (7), an analysis of the obtained results was carried out on the example of the deflection of the shaker table of the laboratory vibration stand VEDS-200, the appearance of which is presented in fig. 3. The technical characteristics of the shaker are given in the table. 1. The characteristics of the shaker table of this vibrating stand are given in the table. 2 To calculate the maximum deflections, we will use the maximum permissible load conditions from the technical specifications. At the same time, the transverse force is uniformly distributed over the surface of the vibrating table with intensity q=11493 Н/м^2.

Table 1.

VEDS-200 Shakers specifications

Maximum of force shaker	Maximum payload	Displacement	Frequency range	Acceleration without load	Acceleration with maximum load
2000 N	45 kg	12,5 mm	5-5000 Hz	40g (392 m/s^2)	4g (39 m/s^2)

Table 2.

Shaker table specifications

Material	Table's Radius	Rod's Radius	Height	Young's modulus	Poisson's ratio
Aluminum	b=230 mm	a=90 mm	h=25 mm	E=70 GPa	μ=0,34

Having solved system (9), we obtain: C_1=-0.0538·10^{-3}, C_2=-0.0172·10^{-3}, C_3=-0.8828·10^{-3}, C_4=-0.9994^{-3}. Analytically calculated according to dependence (7), the shape of the middle surface bend in XYZ coordinates is presented in Fig. 4, where $r = \sqrt{x^2 + y^2}$.

Fig. 3. Shaker VEDS-200 with a test electronic unit

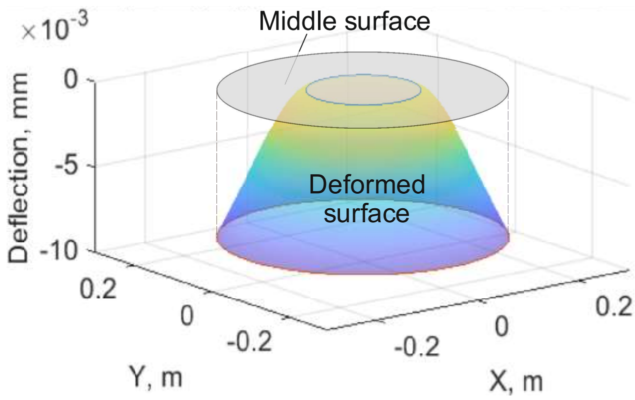


Fig. 4. Static deflection of the shaker table at absence of displacement

The analysis of dependence (7) shows that the rigidity of the shaker table decreases sharply when moving away from point O to the edge of the table. Therefore, when testing large-sized blocks, the attachment of which to the vibrating table is located at a distance $r>a$, the idealization of the shaker table as a rigid solid body in the dynamic model (1) is unjustified. In the model of the testing process, it is necessary to introduce an additional spring with a stiffness of k_3, which takes into account the elastic properties of the shaker table. Such a spring is connected in series with a spring with stiffness k_1, simulating the stiffness of the body of the electronic product. The resulting rigidity of the vibrating table-body system k_{13} is less than k_1:

$$k_{13} = \frac{k_1 k_3}{k_1 + k_3},$$

which explains the phenomenon of a decrease in the real critical speeds of the shaker-product system and an increase in vibrations compared to those expected in the range of operating frequencies of the shaker.

Results of experimental studies and discussion

To experimentally confirm the deformability of the shaker table and assess the influence of elasticity on the level of vibrations at different points on its surface, experimental studies were carried out. The subject of study was the VP-90M platform of the VEDS-200 shaker expansion table in the form of an aluminum plate. Before the study, the platform was divided into 12 sectors.

In each sector at a distance of 100; 330; and 560 mm from the center, the working vibration sensor was alternately fixed (Fig. 5). A control vibration sensor was fixed in the center of the shaker table. Both vibration sensors, complete with the amplification equipment of the shaker, were pre-calibrated for an acceleration of 1g. A vibration sensor of the ABC 027-01 type (Fig. 6) complete with a matching amplifier and a VZ-38 millivoltmeter was used as a reference. Sinusoidal oscillations with an acceleration of 1g were applied to the shaker table. The acceleration values were monitored by a working vibration sensor fixed alternately around a circle with a radius of 100, 330, and 560 mm in each of the 12 sectors of the surface of the vibrating table platform. The study was conducted for 28 frequency values in the frequency range of 5-1000 Hz. Based on the obtained values of vibration overloads, diagrams of the distribution of vibration overload on the surface of the shaker table were constructed. Examples of diagrams for sampling frequencies {100, 300, 450, 650, 800, 1000}Hz are presented in fig. 7.

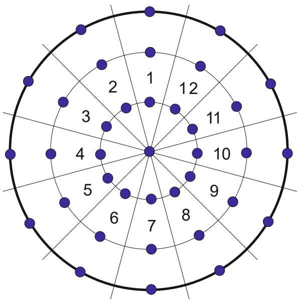
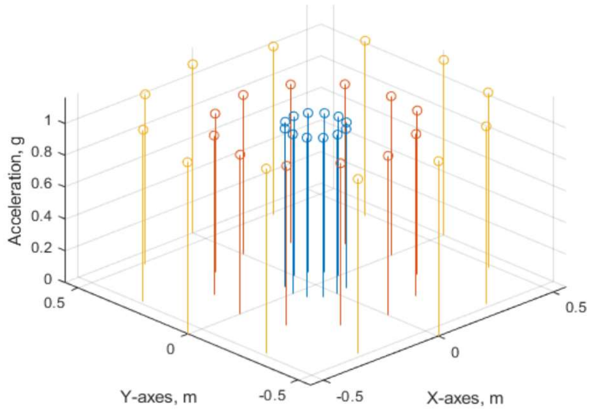
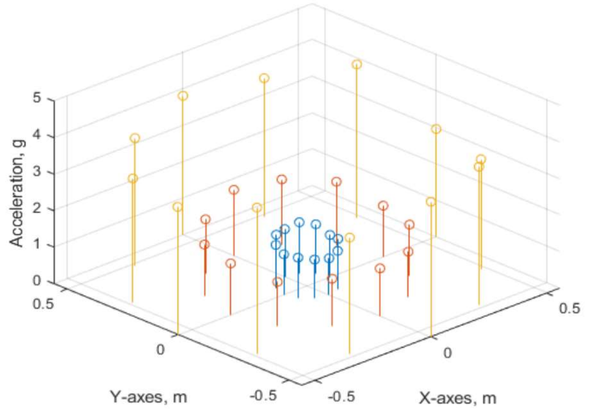


Fig. 5. Scheme of the location of the sensors

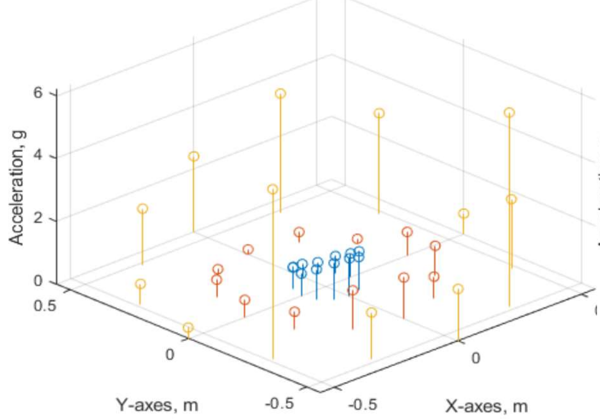
Fig. 6. Sensor ABC 027-01



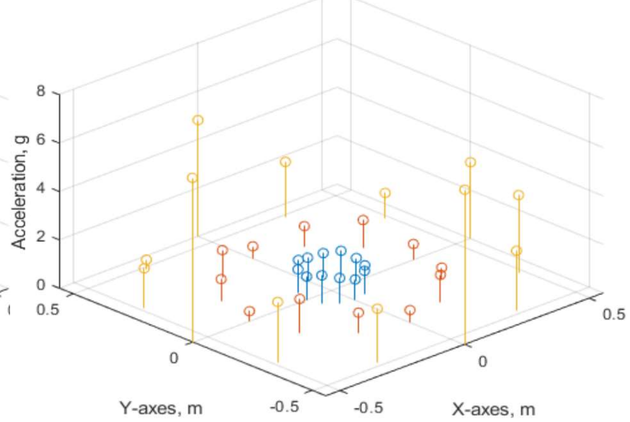
f = 100 Hz



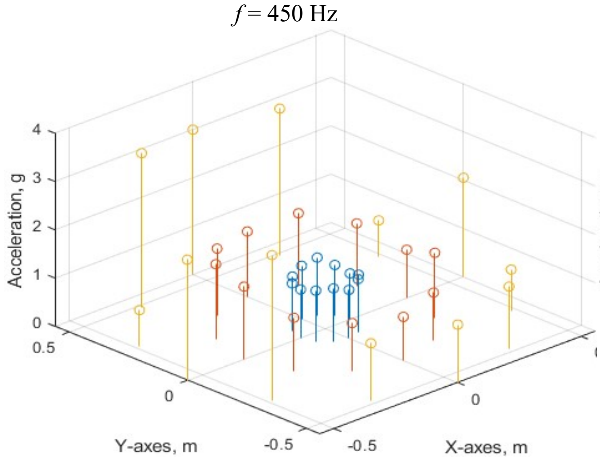
f = 300 Hz



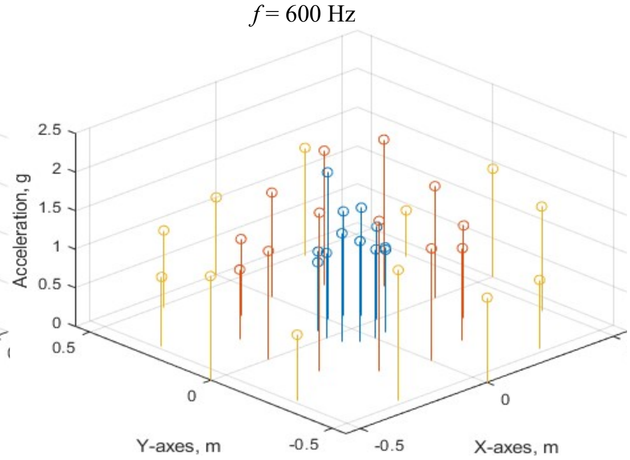
f = 450 Hz



f = 600 Hz



f = 800 Hz



f = 1000 Hz

Fig. 7. Diagrams of distribution of vibration accelerations over the surface of the expansion shaker table at an acceleration value of 1g in the center

Research results show that the values of vibration overloads at different points of the platform of the vibrating table differ significantly from the values measured in the center. They can be from 3 to 6 times less than specified (for example, at f = 450 Hz), or from 7 to 17 times greater (for example, at f = 300 Hz).

Electronic equipment that undergoes vibration tests on electrodynamic vibration stands can have a weight from units to thousands of Newtons. The design of structurally complex electronic devices can be considered as a system consisting of a large number of masses with elastic-damper connections (case, printed circuit boards, components, fasteners, connectors, etc.). Such an electronic unit, having many natural critical frequencies, installed on the vibrating table platform, will further increase the difference in vibration values at the center and other points of the shaker table.

Conclusions

A real shaker table can be considered a rigid body only if the axis of the shaker table rod and the line of action of the exciting force passes through the center of gravity of the test product and test fixture, and also at excitation frequencies significantly lower than the critical frequency of the table. When modeling and simulating electrodynamic shakers, as well as during vibration testing of electronic device structures, it is necessary to take into account the stiffness of the shaker table. Varying the flexibility and deformability of the shaker table makes it possible to more accurately determine the real resonant frequencies of the table testing system. These resonant frequencies are lower than in the case when the shaker table is considered a rigid body. In addition, deformation of the shaker table leads to the fact that the vibrations at different points on its surface are not the same. In practice, this means that when testing an electronic unit, control of the vibration level in the center of the vibration table is inaccurate. Vibrations exceeding the control value can be transmitted from the table to the test fixture and the electronic product being tested, and there they can intensify many times over.

In order to increase the accuracy of test results, it is necessary to comply with the condition that the line of gravity of the device being tested coincides with the axis of the shaker table rod or lies as close as possible to it. In this case, the rigidity of the table is much higher than the rigidity of the product or fastening, and the elastic properties of the table can be neglected.

If during vibration tests it is not possible to place the electronic unit coaxially with the center of the vibrating table, monitoring the vibration value must be carried out at different points of the tested product, for example, at the most distant points from the center.

Acknowledgements

This study was funded by the Ministry of Science and Education of Ukraine (Project No. 0120U102067).

References

1. Veeramuthuvel, P., Shankar, K., Sairajan, K. K. (2016). Experimental investigation of particle damper-based vibration suppression in printed circuit board for spacecraft applications. Proceedings of the Institution of Mechanical Engineers, Part g: Journal of Aerospace Engineering 230.7, 1299-1311 https://doi.org/10.1177/0954410015607552
2. Galler D., Slenski, G. (1991). Causes of aircraft electrical failures. IEEE Aerospace and Electronic Systems Magazine, 6(8), 3-8, https://doi.org/10.1109/62.90949
3. ISO 5344:2004, Electrodynamic test equipment for generating vibration — methods of describing equipment characteristics, International Organization for Standardization, Switzerland, (2004)
4. Lang, G. F., Snyder, D. (2001). Understanding the physics of electrodynamic shaker performance. Sound and vibration, 35(10), 24-33.
5. Shuster, M., Briano, B., Kitchin, Ch. (2002). Mounting considerations for ADXL series accelerometers. AN-379, Analog Devices Application Note
6. Dal, H., Baklaci, M. (2019). Design, fabrication and vibration analysis of a lightweight head expander for a high frequency electrodynamic shaker. Materials Testing, 61(10), pp. 965-972. https://doi.org/10.3139/120.111407
7. Avitabile, P. (1999). Why you can't ignore those vibration fixture resonances. Sound and Vibration, 33, 20-27.
8. Kovtun, I., Boiko, J., Petrashchuk, S. (2019). Assessing Enclosure Case Design on Excitation and Transmission of Vibration in Electronic Packages. In 2019 IEEE 2nd Ukraine Conference on Electrical and Computer Engineering (UKRCON), IEEE, 265-270 https://doi.org/10.1109/UKRCON.2019.8879782
9. Waimer, S., Manzato, S., Peeters, B., Wagner, M., & Guillaume, P. (2016). Numerical modelling and simulation of a closed-loop electrodynamic shaker and test structure model for spacecraft vibration testing. In Proceedings of the European Conference of Spacecraft Structures, Materials and Environmental Testing, Toulouse, France.

10. Waimer, S., et al. (2016). A multiphysical modelling approach for virtual shaker testing correlated with experimental test results. Special Topics in Structural Dynamics, Volume 6. Springer, Cham, 87-99.
11. Zuo, S., Feng, Z., Pan, J., & Wu, X. (2020). Electromechanical coupling dynamic modeling and analysis of vertical electrodynamic shaker considering low frequency lateral vibration. Advances in Mechanical Engineering, 12(10), 1687814020963851
12. Hoffait, S., Marin, F., Simon, D., Peeters, B., Golinval, J. C. (2016). Measured-based shaker model to virtually simulate vibration sine test. Case Studies in Mechanical Systems and Signal Processing, 4, 1-7
13. Waimer, S., Manzato, S., Peeters, B., Wagner, M., Guillaume, P. (2018). Modelling and simulation of a closed-loop electrodynamic shaker and test structure model for spacecraft vibration testing. Advances in Aircraft and Spacecraft Science. 5(2), 205-223
14. Siswanto, W. A., Ibrahim, M. N., Madlan, M. A., & Mohamad, S. M. (2011). Shaker Table Design for Electronic Device Vibration Test System. International Journal of Engineering and Technology, 3(6), 663 http://dx.doi.org/10.7763/IJET.2011.V3.302
15. Timoshenko, S., Woinowsky-Krieger, S. (1959). Theory of plates and shells. Vol. 2. New York: McGraw-hill

Горошко А.В., Зембицька М.В. Аналіз впливу жорсткості вібростола на точність результатів вібраційних тестувань електронних виробів

При вібраційних випробуваннях структурно-складної електронної апаратури на електродинамічних вібростендах виміряні вібрації на столі, оснастці і різних місцях тестованого виробу можуть відрізнятись. Однією з причин є деформівний стіл вібростенда, жорсткість якого може суттєво впливати на результати вібраційних випробувань. Тестоване електронне обладнання як правило не має вісесиметричної форми і тому при випробуваннях дуже складно сумістити центр мас виробу і вібростола або оснастки с віссю, яка проходить через центр вібростола. В роботі теоретично показано, що вібростіл можна вважати жорстким тілом лише за умови відсутності зміщення сили ваги тестованого виробу від центру вібростола, а також на частотах збудження, значно менших за критичну частоту стола. Для теоретичного аналізу вібростіл у формі диску представлено у вигляді тонкої пластини. За допомогою вібрографування вібраційної платформи експериментально підтверджено ефект зростання вібрацій при віддаленні від центру вібростола. Якщо під час вібраційних випробувань немає можливості розмістити електронний блок співвісно з центром вібростола, контроль за значенням вібрацій необхідно обов’язково проводити у різних точках тестованого виробу, наприклад у найвіддаленіших точках від центру.

Ключові слова: вібраційні випробування, вібростенд, оснастка, електронний виріб