Article citation info:
JUREVICIUS M, TURLA V, BUREIKA G, KILIKEVICIUS A. Effect of external excitation on dynamic characteristics of vibration isolating table. Eksploatacja i Niezawodnosc – Maintenance and Reliability 2015; 17 (2): 260–265, http://dx.doi.org/10.17531/ein.2015.2.13.

Mindaugas JUREVICIUS
Vytautas TURLA
Gintautas BUREIKA
Arturas KILIKEVICIUS

EFFECT OF EXTERNAL EXCITATION ON DYNAMIC CHARACTERISTICS OF VIBRATION ISOLATING TABLE

WPŁYW WZBUDZANIA ZEWNĘTRZNEGO NA WŁAŚCIWOŚCI DYNAMICZNE STOŁU WIBROIZOLACYJNEGO

Scientific publications available on sandwich panels in evaluating fundamental frequency with a non-dimensional parameter have been discussed in this article. Effectiveness of optical table with pneumatic vibration insulation supports have been analysed in low (1-50) Hz and higher (500-1200) Hz frequency range. Experiments of vibration transmissibility performed using vibration excitation apparatus and other special test equipment. The dynamics characteristics and application ranges of a table as low frequency vibration damper have been defined. Theoretical and experimental modal analysis of the main part of the system – top surface of table – has been performed. This analysis enabled to determine four resonant eigen-frequencies at higher frequency range. Research results show the reliability of vibration table usage and the dangerous zones of its exploitation.

Keywords: *vibro-isolator, low and higher frequency, optical table, honeycomb core.*

W artykule omówiono dostępne publikacje naukowe dotyczące oceny częstotliwości podstawowej paneli przekładkowych z wykorzystaniem parametru bezwymiarowego. Analizowano wydajność stołu optycznego wyposażonego w podpory wibroizolacyjne w niskim (1-50 Hz) i wyższym (500-1200 Hz) zakresie częstotliwości. Doświadczenia dotyczące charakterystyk przenoszenia drgań prowadzono z zastosowaniem aparatury do wzbudzania drgań i innych specjalnych urządzeń badawczych. Określono właściwości dynamiczne i zakres wykorzystania stołu jako tłumika drgań o niskiej częstotliwości. Dokonano teoretycznej i eksperymentalnej analizy modalnej głównej części systemu – górnej powierzchni stołu. Analiza ta pozwoliła wyodrębnić cztery częstości drgań własnych z wyższego zakresu częstotliwości. Wyniki badań potwierdzają niezawodność stołu wibracyjnego oraz określają strefy jego eksploatacji wykazujące niekorzystne właściwości.

Słowa kluczowe: *wibroizolator, niskie i wysokie częstotliwości, stół optyczny, rdzeń o strukturze plastra miodu.*

1. Introduction

Most of metrological instruments are sensitive to low-frequency mechanical vibrations and higher frequency – acoustic noise. These oscillations can occur from both outside and inside of the building, i.e. from passing automobiles, wind, heating, ventilation and air-conditioning equipment, and these building vibrations effect on laboratory equipment operation as well. The possibility and method of time varying vibration decomposition are discussed in the scientific workof professor Radkowski [11].

Honeycomb sandwich structures are currently being used in various engineering solutions, both within and outside of aerospace engineering. Lightweight honeycomb materials can be used in the construction of composite panels, shells, and tubes with high structural efficiency. The dynamic characteristics of Multilayer Polyurethane foam glass/fiber composite sandwich panels have been determined through Experimental Investigations by Havaldar and others [4].

Sandwich shell structures including metal alloys and composites are being increasingly applied to many aircraft, automobile and construction industries due to their lightweight and highly load carrying capacities. Furthermore, the use of cellular cores for panels and shells not only can further reduce the structural weight, but also potentially possess superior heat dissipation, vibration control and energy dissipation characteristics. Filament winding and twice co-curing processes were proposed to make a carbon fiber reinforced composite (CFRC) sandwich cylinder with Kagome cores. Axial compression was carried out to reveal the stiffness and load capacity of the fabricated sandwich cylinder. Compared with the stiffened cylinder with similar dimensions and mass, the sandwich cylinder is shown to be stiffer and stronger by several times [3]. Linear and nonlinear analysis using the ABAQUS finite element is described in paper [15]. Some mechanical properties such as compression, bending, impacting and vibration have been reported by many scientists worldwide. Analytical calculation methods have a clear advantage when designing multi-layered composite cylindrical shells. Compared to Finite Element Methods (FEM) they enable more precise physical interpretations and weighting of the different design variables to be made and, hence, significantly facilitate problem-specified parameter studies and sensitivity analysis [5]. Sandwich cylindrical shells are the major components of aerospace apparatus structures [17]. Nevertheless, an analytical investigation was carried out to examine the response of CFRC sandwich cylinders with lattice cores.

Researchers Xiong et al [20] manufactured sandwich-walled cylindrical shells with aluminium pyramidal truss core of constant curvature employing an interlocking fabrication technique for the metallic core. Thereafter is carried out axial compression tests on some representative samples to investigate the failure modes of these structures and compared with an analytical failure map developed to account for Euler buckling, shell buckling, local buckling between reinforcements and face-crushing. A combined experimental and numerical study is conducted to assess the effects of impact energy, impact site and core density on the compression-after-impact (CAI) strength of pyramidal truss core sandwich structures [24]. It is found that the severity of impact damage highly depends on the impact site. The CAI tests show that the local buckling occurs for both the un-impact specimens and the specimens impacted under lower energy, while debonding is observed for the specimens impacted under higher energy. In addition to experimental tests, the numerical simulation performs well in capturing the failure modes for impact-damaged specimens under compressive load. Composite pyramidal lattice structures with hollow trusses afford a convenient means to enable functionality by inserting elements into free volumes within or between trusses are presented by researchers Yin, Wu and others [23]. In this study, vibration and low-velocity impact tests were carried out to investigate the dynamic behaviour of hollow composite pyramidal lattice structures filled with silicone rubber. Frequencies and the corresponding damping ratios were obtained, which revealed that the damping ratios of space-filled composite pyramidal lattices increased by two times but those of hybrid composite pyramidal lattices decreased by 2% for the first three orders compared with hollow composite pyramidal lattices.

FEM has been used to predict the modal properties of free-free FRP plates, and the predictions were verified experimentally by scientists Maheri, Adams and Hugon [8, 9]. Results are presented for materials for which little dynamic data have yet been available, including the thermoplastic matrix material PEEK. Also, it is shown how using improved experimental techniques can lead to closer theoretical and experimental damping results. In the paper [10] laminated composite shells are frequently used in various engineering applications in the aerospace, mechanical, marine, and automotive industries. This article follows a previous book and review articles published by the leading author. It reviews most of the research done in recent years (2000–2009) on the dynamic behaviour (including vibration) of composite shells. This review is conducted with emphasis on the type of testing or analysis performed (free vibration, impact, transient, shock, etc.), complicating effects in material (damping, piezoelectric, etc.) and structure (stiffened shells, etc.), and the various shell geometries that are subjected to dynamic research (cylindrical, conical, spherical and others). A general discussion of the various theories (classical, shear deformation, 3D, non-linear etc.) is also given. The main aim of this review article is to collate the research performed in the area of dynamic analysis of composite shells during the last 10 years, thereby giving a broad perspective of the state of art in this field. The paper [12] deals with the free vibration analysis of composite sandwich cylindrical shell with a flexible core using a higher order sandwich panel theory. The formulation uses the classical shell theory for the face sheets and an elasticity theory for the core and includes derivation of the governing equations along with the appropriate boundary conditions. The model consists of a systematic approach for the analysis of sandwich shells with a flexible core, having high-order effects caused by the nonlinearity of the in-plane and the vertical displacements of the core. The behaviour is presented in terms of internal resultants and displacements in the faces, peeling and shear stresses in the face–core interface and stress and displacement field in the core.

The paper [13] explores the partial coverage of cylindrical shells with a constrained viscoelastic damping layer, with emphasis on examining the minimum area of coverage that will yield optimal damping. The distribution of damping patches on the structure is based on strain energy intensity distribution maps derived for the purpose. The analysis uses the FEM, and a suitable curved shell element is formulated for the add-on damping treatment. Numerical studies show that a partial coating procedure can be a viable approach in optimal damping designs. A semi-analytical finite element for doubly curved, multi-layered shells of revolution, based on an extension of the displacement field proposed by Wilkins et al, is proposed in paper [14]. Numerical analysis is done to study the vibration and damping characteristics of multi-layered fluid filled shells with alternating elastic and viscoelastic layers. The effect of varying the number of viscoelastic layers on the vibration and damping characteristics is also studied. The effect of the fluid is incorporated by the added mass concept. The effect of shear parameter on natural frequency and modal loss factor is studied for various circumferential and axial modes. The vibration and damping characteristics of free–free composite sandwich cylindrical shell with pyramidal truss-like cores have been conducted using the Rayleigh-Ritz model and FEM is presented in paper [22]. The predictions for the modal properties of composite sandwich cylindrical shell with pyramidal truss-like cores showed good agreement with the experimental tests. The influences of fiber ply angles on the natural frequency and damping loss factor were investigated. Three types of such composite sandwich cylindrical shells were manufactured using a hot press moulding method and the relevant modal characteristics of various sandwich cylindrical shells could be obtained by modal tests. The natural frequencies of composite sandwich cylindrical shell increased with the increasing of the ply angle of the inner and outer curve face sheets, whereas the damping loss factors of present shells did not increase monotonically. The natural frequencies of composite sandwich beams with lattice truss core are investigated by combining the Bernoulli–Euler beam theory and Timoshenko beam theory were analysed by Xu and Qiu [21].

Latterly, the scientists concerning honeycomb sandwich structures have been focused on effective numerical modelling methods, vibration properties, crash- worthiness, damage, and failure and impact response. Researchers Adams and Maheri[1] investigated the damping of composite honeycomb sandwich beams in steady-state flexural vibration using the method extended from that for monolithic beams.

The material properties such as elastic modules and strengths are various in different directions, and even the compressive and tensile properties are different in the thickness-direction, primarily due to the initial deflection of cell walls. Vibration frequencies and mode shapes of honeycomb sandwich panels with various structural parameters were studied by Qunli Liu and Yi Zhao [7] using computational and experimental methods. Two computational models were used to predict the mode shapes and frequencies of honeycomb sandwich panels. Plate elements were used for honeycomb cell walls to reflect the geometric nature of the hexagonal cells.

Optical table vibrations typically are between 2 Hz and 7 Hz, because it is eigen-frequency at which the optical table resonates. However, especially for low (from 0.7 Hz) frequencies a better insulator, working with scanning probe microscopy and interferometry is required. Existing quasi-zero (negative) stiffness isolators resonates from 0.5 Hz [6].This frequency has almost no power, because it would be very unusual to find large oscillations at 0.5 Hz[18]. Optical tables and active systems do not work very well when placed in a vacuum, especially at high or low temperature and radiation. Such an environment occurs during specific investigations of semiconductors. Quasi-zero stiffness system can work in vacuum, high and low temperatures and under radiation [6].

Optical tables with pneumatic vibration isolators are suitable for laser centres. Theoretical analysis of vibration parameters and analysis of experimental results allows to assess of honeycomb systems reliability. Comprehensive analysis of vibration theoretical methods are described by scientists Cveticanin, Mester and Biro [2], Siljak, Subasi [16] and Wicher, Więckowski [19].

A lot of studies presented in bibliographic sources are related to one of the attributes (high strength/weight or increased energy absorption) mentioned above. With regard to the development of a honeycomb panels, one issue that has been overlooked is the scaling of honeycomb properties with respect to cell size. The variation in cell size may have a large influence on the dynamic properties of honeycomb panels. The goal of this study was to reveal the effect of cell size on the fundamental frequency of honeycomb panels. The results of the experimental investigation are presented and discussed. Nevertheless, authors described the determination of mechanical passive isolation systems ability to isolate low (from 0.7 to 50 Hz) and higher (from 500 to 1200 Hz) frequency oscillations.

2. Research objects, instruments and equipment

The experimental research combination of optical table with pneumatic vibration insulators is shown in Fig. 1.

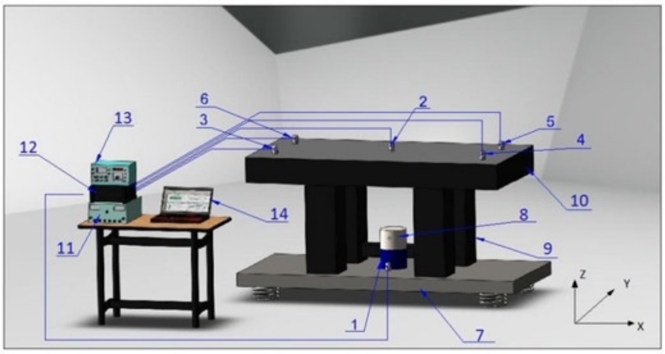

Fig. 1. Research scheme of optical table dynamic characteristic: 1, 2, 3, 4, 5, 6 – vibration transducers; 7 – platform (base); 8 – vibrator; 9 – supports of vibration isolation; 10 – optical table with experimental of vibration isolation; 11 – impulse generator; 12 – amplifier; 13 – generator; 14 – computer with analyser

Top and bottom countertop surfaces of analysed optical table are made of cold-rolled ferro-magnetic steel sheets, which combine lightweight structure made of corrosion resistant cellular steel, giving the table exceptional toughness. The optical table is usually mounted on special vibration isolating supports. The optical table structures resist to static and dynamic forces not only vertically, but also horizontally considered being highly important defining quality factors.

Idealized "seismic" mounting system of optical tables is a rigid table, mounted on a massive foundation or on the supports that inhibit vibrations. Various types of vibration isolation bearings with compressed air dampers are used in world practice. These supports must ensure the stability of the table in vertical and horizontal directions. Horizontal environmental vibration effect is particularly striking when the laboratories are installed on the upper floors of the buildings.

The lightweight honeycomb structures for manufacture objects that are resistant to the dynamic and static forces are widely used. Honeycomb cells are characterized by the size, wall thickness, the material from which they are made, etc. Typical features of honeycomb structures are lightness and resistance to compression and bending. These qualities are especially important when it is required for optimal characteristics ratio of mass and stiffness. Therefore cellular structures are used in aircraft, helicopters, and other plant production.

This type of structures is widely used for optical laboratory tables also. Cellular tables have good vibration damping properties, they are much lighter than the massive tables made of granite. In most cases not resistant for mechanical loads heavy granite table, but lightweight cellular structure have been chosen.

The most important quality criteria of insulating pillars is characteristics of mechanical vibration transmission from a base supports the table top. These characteristics determine the applicability of various experimental techniques. The specific method is selected according to a number of factors, among which the more important one is dominant frequency range.

Study includes analysis and measurement of vibration and other dynamic properties by using "Brüel&Kjær" firm equipment. The portable measurement results processing device connected with computer, and vibration sensors (type 8341 and 8306) with vibration meter 2511 were used as well. An experimental part of modal analysis was carried out by using this equipment as well.

The vibration excitation platform with a vibrator and other special test equipment were used in and tested for research of dynamic parameters of pneumatic isolator of vibrations. Easily tuned vibration excitation platform has been built, allowing the test subject to excite vibrations of (1–50) Hz frequency range in any of three directions: vertical transverse, horizontal transverse and horizontal longitudinal directions.

Using aforementioned equipment modal analysis of top plate of table was performed. This analysis was done with purpose to obtain eigen-frequencies of top plate which are unwelcome in precise measuring process. During experiment top plate of vibro-isolating table was treated as single deformable body instead of construction with supports.

3. Results of theoretical and experimental modal analysis

The results of typical damping (Fig. 1) of vibration isolation supports excited by harmonic vibrations, impulse and white noise, shown in Fig. 2. (a, b, c, d, e).Oscillations are not isolated when there is a harmonic excitation at 2 Hz; isolation starts with 4 Hz. Thus, optical tables are not effective for frequencies up to 4 Hz.

Following are presented the most important results of vibration isolating supports at 1–50 Hz frequency range at different sizes of load (transmission dependence on frequency curves T_v, corresponding resonant frequency of the f_{vr} and transmissibility coefficient values at resonance 5 Hz to 10 Hz frequencies). Maximum value of transmissibility coefficient at different loads varies from 3 to 4 Hz: without load – 2.9 Hz; 100 kg – 2.7 Hz; 250 kg – 2.4 Hz; 500 kg – 2.1 Hz. This shows that the vibration isolating supports with the load forms an elementary single mass vibrating system. Resonance frequency decreases by increasing the load of vibration isolating supports. With increasing frequency above the resonance transmissibility steadily decreases, and at frequencies of 50 Hz is less than 0.01.

Further results of modal analysis are provided.The experimental results were compared with the analytical model of the vibro-isolating table; approximate simplified model of vibro-isolating table built. This analytical model is thoroughly described in paper [6]. In current paper mathematical model of vibro-isolating table was built in ANSYS environment and modal analysis was performed. The table top was modelled using SHELL63 finite element; mesh of 25x25 elements, which gave converged results of eigen-frequencies (see Fig. 3).

Table was measured in 16 points using same equipment as in previous experiments. As top plate of the vibro-isolating table is permitted to freely bend, there are many different shapes in which the top plate can bend. An eigen-mode describes the shape of bended top plate; an eigen-frequency describes how fast bending occurs. Eigen-mode vibrates at its eigen-frequency and the total bending and frequency of the top plate is their sum. Eigen-modes depend on the support configuration of the vibro-isolating table, and the natural frequencies depend on the stiffness and mass components of the top plate and its shape. Natural modes with the highest frequencies are usually not very important because their amplitude is relatively small. Only four lowest eigen-frequencies of the vibro-isolating table are considered to be significant.

Eigen-mode shapes of experiment match ones of mathematical model. Results of mathematical model showed good corresponding

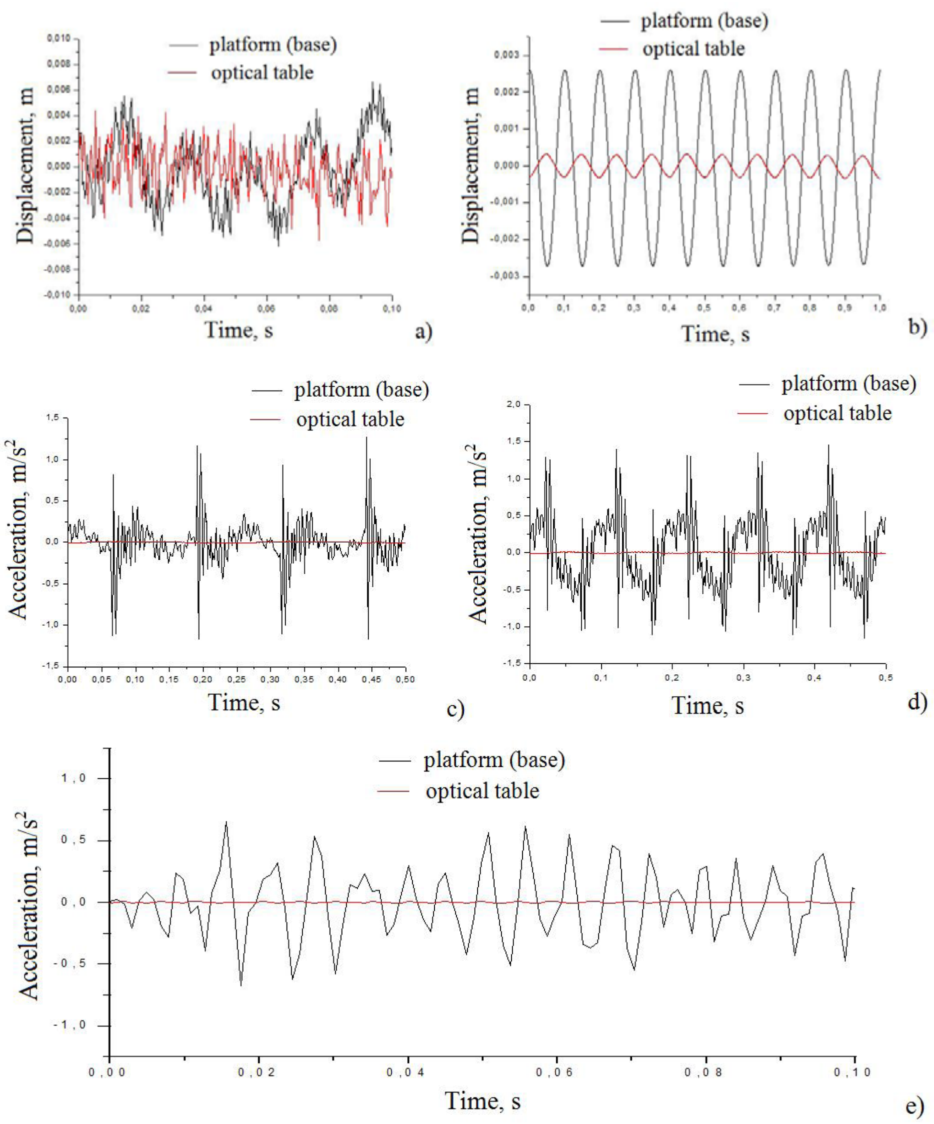

Fig. 2. Damping characteristics of vibration isolation supports (black signal – of platform, red – of optical table): a – 2Hz harmonic excitation frequency; b – 10 Hz harmonic excitation frequency; c – 4 Hz impulse excitation; d – 10 Hz impulse excitation; e – excitation by white noise

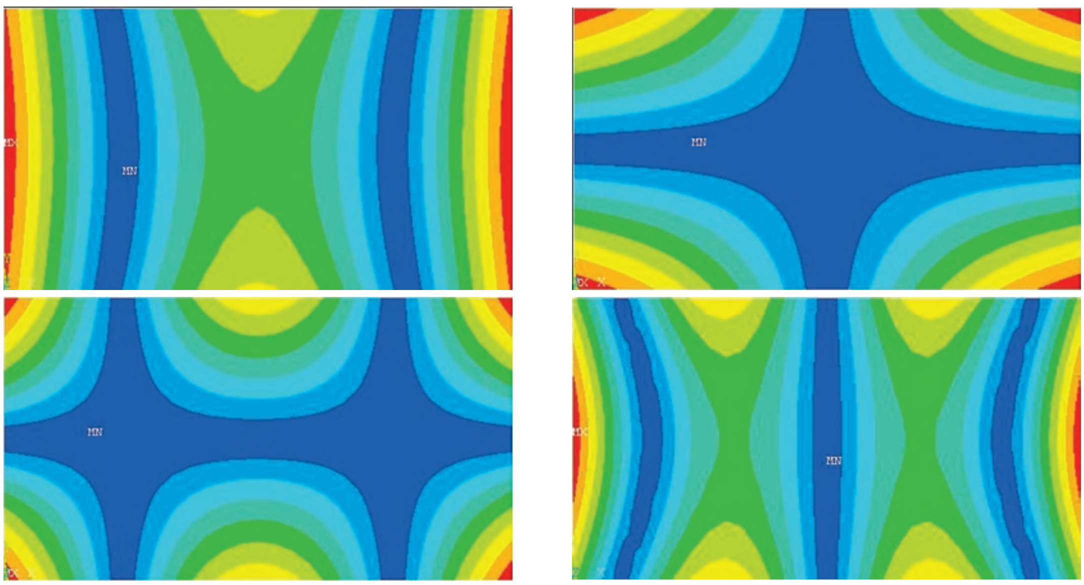

Fig. 3. Lowest four natural modes of the top plate

Table 1. Comparison of modelling and experiment results

Mode	Experimental, f_E, Hz	Theoretical, (using ANSYS) f_T, Hz	Discrepancy, Δ, %
1	504.4	495.8	1.7
2	535.8	551.2	2.8
3	1012.8	989.7	2.3
4	1172.0	1205.6	2.9

with experiment;i.e. discrepancy does not exceed 2.9% between results of experimental and mathematical tests (Table 1).

As seen in Table 1, the discrepancy of theoretical and experimental results varies from 1.7 % up to 2.9 %. This shows enough high reliability of above mentioned modelling method.

4. Conclusions

1. Designated vibration excitation test equipment intended for identification of dynamic characteristics of the investigated objects was designed by authors and tested. The following pneumatic vibration isolator dynamic parameters identified: coefficient of transmissibility in vertical direction – oscillation characteristics at 50 Hz frequency range is less than 0.01; resonant frequency in vertical direction – 3–4 Hz depending on load; and damping efficiency at 5 Hz to 10 Hz depending on optical table load was derived. It was proved, that optical table has less than unitary transmissibility coefficient (vibration isolating properties) in analysed frequency range.
2. Performed modal theoretical and experimental analysis of upper share of the table enable to define four resonant eigen-frequencies at higher frequency diapason: 504.4 Hz, 535.8 Hz, 1012.8 Hz and 1172.0 Hz.
3. The results of this study show thatduring forthcoming experiments with this equipment fixed on the table it is required to avoid these four dangerous resonant eigen-frequecies of the table and table top surface.

References

1. Adams RD, Maheri MR. Dynamic Shear Properties of Structural Honeycomb Materials. Composites Science and Technology. 1993. 47(1): 15–23, http://dx.doi.org/10.1016/0266-3538(93)90091-T.
2. Cveticanin L., Mester G., Biro I. Parameter Influence on the Harmonically Excited Duffing Oscillator. Acta polytechnica Hungarica. 2014.11(5): 145–160.
3. Fan HL, Fang DN, Chen LM, Dai Z and Yang W. Manufacturing and testing of a CFRC sandwich cylinder with Kagome cores. Compos Sci Technol. 2009. 69: 2695–700, http://dx.doi.org/10.1016/j.compscitech.2009.08.012.
4. Havaldar SS, Sharma RS, Antony AP and Bangaru M. Effect of cell size on the fundamental natural frequency of FRP honeycomb sandwich panels. Journal of Minerals and Materials Characterization and Engineering. 2012. 11: 653–660.
5. Hufenbach W, Holste C, Kroll L. Vibration and damping behaviour of multilayered composite cylindrical shells. Compos Struct. 2002;58:165–74, http://dx.doi.org/10.1016/S0263-8223(02)00025-9.
6. Jurevicius M, Kilikevicius A, Berba M. Impact of external excitations on the dynamic properties of negative-stiffness vibration isolation table, Journal of Vibroengineering. 2011. 13(2): 352–357.
7. Liu QL, Zhao Y. Role of Anisotropic Core in Vibration Properties of Honeycomb Sandwich Panels, Journal of Thermoplastic Composite Materials. 2002. 15 (1): 944–952, http://dx.doi.org/10.1106/089270502022860.
8. Maheri MR, Adams RD. Finite-element prediction of modal response of damped layered composite panels. Compos Sci Technol. 1995. 55:13–23, http://dx.doi.org/10.1016/0266-3538(95)00074-7.
9. Maheri MR, Adams RD, Hugon J. Vibration damping in sandwich panels. J Mater Sci. 2008. 43: 6604–6618, http://dx.doi.org/10.1007/s10853-008-2694-y.
10. Qatu MS, Sullivan RW, Wang W. Recent research advances on the dynamic analysis of composite shells: 2000–2009. Compos Struct. 2010. 93: 14–31, http://dx.doi.org/10.1016/j.compstruct.2010.05.014.
11. Radkowski S. Use of vibroacoustical signal in detecting early stages of failures. Eksploatacja i Niezawodnosc – Maintenance and Reliability. 2007. 35(3): 11–18.
12. Rahmani O, Khalili SMR, Malekzadeh K. Free vibration response of composite sandwich cylindrical shell with flexible core. Compos Struct. 2010. 92:1269–81, http://dx.doi.org/10.1016/j.compstruct.2009.10.021.
13. Sainsbury MG, Masti RS. Vibration damping of cylindrical shells using strain energy-based distribution of an add-on viscoelastic treatment. Finite Elem Anal Des. 2007. 43:175–92, http://dx.doi.org/10.1016/j.finel.2006.09.003.
14. Saravanan C, Ganesan N, Ramamurti V. Vibration and damping analysis of multilayered fluid filled cylindrical shells with constrained viscoelastic damping using modal strain energy method. Comput Struct. 2000. 75: 395–417, http://dx.doi.org/10.1016/S0045-7949(99)00099-1.
15. Shariati MSM, Saemi J, Eipakchi HR and Allahbakhsh HR. Numerical and experimental investigation on ultimate strength of cracked cylindrical shells subjected to combined loading. Mechanika. 2010. 4:12–9.
16. Siljak H, Subasi A. Fourier spectrum related properties of vibration signals in accelerated motor aging applicable for age determination. Eksploatacja i Niezawodnosc – Maintenance and Reliability. 2014. 16 (4): 616–621.
17. Sun FF, Fan HL, Zhou CW, et al. Equivalent analysis and failure prediction of quasi-isotropic composite sandwich cylinder with lattice core under uniaxial compression. Compos Struct. 2013. 101:180–90, http://dx.doi.org/10.1016/j.compstruct.2013.02.005.
18. Vijayan V, Karthikeyan T. Design and Analysis of Compliant Mechanism for Active Vibration Isolation Using FEA Technique. Int. Journal of Recent Trends in Engineering. 2009. 1(5): 77-81.
19. Wicher J, Więckowski D. Influence of vibrations of the child seat on the comfort of child's ride in a car. Eksploatacja i Niezawodnosc – Maintenance and Reliability. 2010. 4(48): 102-110.
20. Xiong J, Ghosh R, Ma L, Vaziri A, Wang YL, Wu LZ. Sandwich-walled cylindrical shells with lightweight metallic lattice truss cores and

carbon fiber-reinforced composite face sheets. Compos: Part A. 2014. 6: 226–238, http://dx.doi.org/10.1016/j.compositesa.2013.10.008.

21. Xu MH, Qiu ZP. Free vibration analysis and optimization of composite lattice truss core sandwich beams with interval parameters. Compos Struct. 2013. 106: 85–95, http://dx.doi.org/10.1016/j.compstruct.2013.05.048.
22. Yang J, Xiong J, Ma L, Zhang G, Wang X and Wu L. Study on vibration damping of composite sandwich cylindrical shell with pyramidal truss-like cores. Composite Structures 2014. 117: 362–372, http://dx.doi.org/10.1016/j.compstruct.2014.06.042.
23. Yin S, Wu LZ, Yang JS, Ma L, Nutt S. Damping and low-velocity impact behavior of filled composite pyramidal lattice structures. J Compos Mater. 2013. 1–12.
24. Zhang GQ, Wang B, Ma L, Xiong J, Yang JS and Wu LZ. The residual compressive strength of impact-damaged sandwich structures with pyramidal truss cores. Compos Struct. 2013. 105: 188–98, http://dx.doi.org/10.1016/j.compstruct.2013.05.016.

Mindaugas JUREVICIUS
Arturas KILIKEVICIUS
Department of Mechanical Engineering, Vilnius Gediminas Technical University
J. Basanavičiaus str. 28, LT-03224 Vilnius, Lithuania

Vytautas TURLA
Department of Printing Machines, Vilnius Gediminas Technical University
J. Basanavičiaus str. 28, LT-03224 Vilnius, Lithuania

Gintautas BUREIKA
Department of Railway Transport,Vilnius Gediminas Technical University
J. Basanavičiaus str. 28, LT-03224 Vilnius, Lithuania

E-mails: mindaugas.jurevicius@vgtu.lt, arturas.kilikevicius@vgtu.lt;
vytautas.turla@vgtu.lt; gintautas.bureika@vgtu.lt