VILNIUS GEDIMINAS TECHNICAL UNIVERSITY

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RESEARCH OF PASSIVE LOW-FREQUENCY VIBRATION ISOLATION SYSTEMS

SUMMARY OF DOCTORAL DISSERTATION

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VILNIAUS GEDIMINO TECHNIKOS UNIVERSITETAS

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Introduction

The research problem

Most metrological tools are sensitive to low-frequency mechanical vibrations and acoustic noise. These may originate both from the outside and from the inside of a building (e.g. from passing vehicles and wind and from the operation of heating, ventilation, and air conditioning equipment and other laboratory installations).

The need for isolating low-frequency vibrations is growing because the proper operation of precision scales, optical microscopes, and other sensitive equipment installed in satellites, space stations, and space telescopes requires isolation from vibrations. Vibrating nanoscales create a number of problems when work must be conducted at high speeds. At present, several methods for isolating vibrations exist: active and passive methods. Passive vibration isolation systems produce less heat than active systems, which is important if it is necessary to carry out work with tools that are sensitive to heat. Furthermore, passive systems are considerably cheaper and more reliable than active ones and do not require any special measures for adjusting damping or stiffness. Currently, passive vibration isolation systems, such as optical tables with pneumatic vibration isolators in laser centres, are widely used to isolate low-frequency vibrations. Typical vibrations of an optical table are found in the range from 2 to 7 Hz because this is its own frequency at which it resonates. However, in the case of very low frequencies (from 0.7 Hz), there is a need for better isolators in order to achieve better results when working with probe microscopes and interferometers. The developed quasi-zero (negative) stiffness isolators resonate at frequencies from 0.5 Hz. This frequency is nearly energy-free since it would be fairly unusual to find any major vibrations at a frequency of 0.5 Hz. The operation of optical tables and active systems is not very good if these are placed in vacuum, very high or low temperatures, or are exposed to radiation. Environments such as described above are found when performing certain studies with semiconductors. Quasi-zero (negative) stiffness systems can operate in a vacuum, at very high and low temperatures, and when exposed to radiation. Quasi-zero (negative) stiffness systems are compact and easy to move from one location to another.

The present work focuses on the determination of the ability of mechanical passive isolation systems to isolate low-frequency (0.7–50 Hz) vibrations.

Relevance of the thesis

Optical tables with pneumatic vibration isolators are suitable for laser centres. However, with the purpose to achieve the better results at working with probe microscopes and interferometers, there is a need of new-generation vibration isolation systems that will enable the sensitive to vibrations equipment (e.g. probe microscopes, micro sweep testers, profilers, and scanning electronic microscopes) to operate in harsh conditions and in the presence of major vibrations (all this cannot be achieved at using even premium-quality optical tables or other active vibration isolation systems). That is why the new isolation methods and systems which are based on the mechanical concept, offer suitable ergonomic properties, are considerably more compact if to compare with optical tables; they also need to be more simple than active vibration isolation systems, and easier to move from one location to the other. All this will ensure the proper isolation of vibrations and allow to use the systems at workstations – at placing them either on desks or on the floor, in vacuum chambers, high or low temperatures, and exposed to radiation.

The research object

The research object are mechanical passive low-frequency vibration isolation systems.

Aim

The aim of the theses is to study mechanical low-frequency passive vibration isolation systems and to formulate the methods of study of this type systems.

Tasks

In order to achieve the aim of the work, the following tasks must be completed:

1. Overview the scientific literature about types of vibration isolation, vibration isolation systems, and studies and analysis of these systems.
2. Analysis and substantiation of efficiency of the mechanical low-frequency vibration isolation systems.
3. To propose the methods of improvement of research of optical tables and pneumatic vibration isolators.
4. To perform the experimental research of dynamic properties of vibration isolation systems of the quasi-zero (negative) stiffness.
5. To perform the studies of dynamics of the complex vibration isolation systems and evaluate the uncertainty of obtained results.

Research methodology

The research involved theoretical and experimental studies. The following study methods were used: principles of digital analysis, comparative measurements of vibration isolation systems, and mathematical statistical analysis to evaluate the reliability of the obtained results.

Scientific novelty

The following results, new for the mechanical engineering, were obtained in the course of preparing the thesis:

1. The analysis of low-frequency vibration isolation systems was made by using comparative methods for evaluation of efficiency of the systems. The obtained theoretical dynamic properties (transmissibility) of the mechanical low-frequency passive vibration isolation systems allowed to evaluate the efficiency of vibration isolation. It was determined that systems of this type can isolate vibrations in the frequency range of 0.7–50 Hz.
2. The proposal methods for the study of optical tables and pneumatic vibration isolators allowed to design and enhance systems of this type in the cases where their dynamic properties are unknown and where any other methods are not applicable. The presented results of the experimental study show that optical slabs and platforms at frequency up to 80 Hz are absolutely stiff bodies.
3. The developed methods of quasi-zero stiffness vibration isolation enabled to deal with low-frequency (0.7–50 Hz) vibration isolation at action of harmonious, impulse and white noise excitation.

Practical importance of the results

At use of the developed mechanical low-frequency passive vibration isolation system, the vibration isolation process becomes considerably easier, cheaper, and is less time and work-consuming if to compare with the other currently available vibration isolation methods.

Defended propositions

1. The new method of quasi-zero stiffness vibration isolation method based on the mechanical concept and weight balancing.
2. Methods allowing evaluation of the dynamic properties of vibration isolation systems and the numerical values of their properties.
3. Substantiation of the opportunities of the practical use of the new quasi-zero stiffness vibration isolation method based on the analysis of the results of completed vibration measurement experiments.

Scope of the thesis

The thesis consists of the introduction, four chapters, conclusions, three appendices, a list of literature, and a list of publications. The total scope of the dissertation are 126 pages, 67 figures, 9 tables, and 113 numbered formulas. A total of 139 scientific literature sources were studied.

1. Development evolution and research analysis of vibration isolation systems

This chapter provides a review of the scientific literature. Major focus in research publications is given to active, semi active, and passive vibration isolation methods and systems, their modelling, analysis, experimental research, and also searching for ways and methods to isolate low-frequency vibrations. The analysis of the scientific literature shows that next generation metrology and research equipment requires higher level systems for isolating vibrations, particularly in the low-frequency range. The standard passive systems for isolating vibrations (optical tables) are frequently used for isolation of high and low frequencies (4−5 Hz) and for vibration resonance control where a higher level of vibration isolation is required. Some authors maintain that active systems in certain cases may provide a better isolation of low-frequency vibrations. However, research publications also show that active systems are more expensive, complicated, and less reliable than passive ones. The main restriction in respect to the use of active systems is that vibration isolation requires an external power supply. Despite the rapid development of active vibration isolation systems and good results obtained in the isolation of low- frequency vibrations, an increase in research publications and interest in new fully mechanical passive low-frequency vibration isolation systems can be observed; these systems are compact, easy to move from one location to another, and can operate under specific conditions, i.e. in vacuum, at high and low temperatures, and when exposed to radiation, while optical tables and active vibration isolation systems cannot do any of this. Such research, therefore, is relevant since no data has been found in the scientific literature that the research in the low-frequency range using quasi-zero stiffness vibration isolation methods has been conducted.

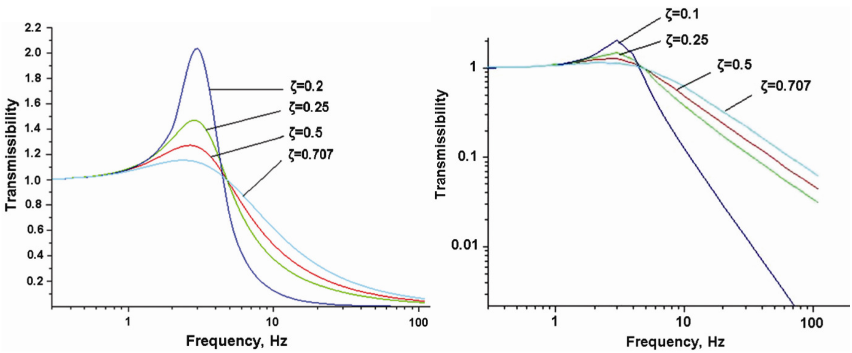
At the end of the chapter conclusions and the thesis tasks are provided.

2. Dynamic research of low-frequency vibration isolation systems

This chapter focuses on dynamic research of low-frequency vibration isolation systems constructed by the author – dynamic substantiation of the platform, optical tables where the tabletop and the floor are subjected to excitation, also quasi-zero stiffness systems and combined systems consisting of an optical table and the quasi-zero (negative) stiffness vibration isolation system under harmonic, impulse and accidental base excitation. On the basis of the research, a mechanical concept of the low-frequency vibration isolation system was developed and patented. A mathematical model of vibrating platform was created according to Lagrange's equation of the second kind and the suitability of the platform for low-frequency harmonic vibration excitation was analysed and identified.

The analytical research showed that the resonance frequency of the tables under consideration ranges from 3 to 4 Hz and that the tabletop dynamic properties are best estimated by the dynamic deflection. The research also showed that in addition to the dynamic deflection, dynamic tabletop characteristics can be well characterised by the dynamic deflection coefficient and the maximum relative tabletop motion.

The study of the behaviour of the optical table on a vibrating platform and the absolute and relative transmissibility coefficients obtained show that vibration isolation starts from 5 Hz under a resonance frequency of the table of 3 Hz (Fig. 2.1).

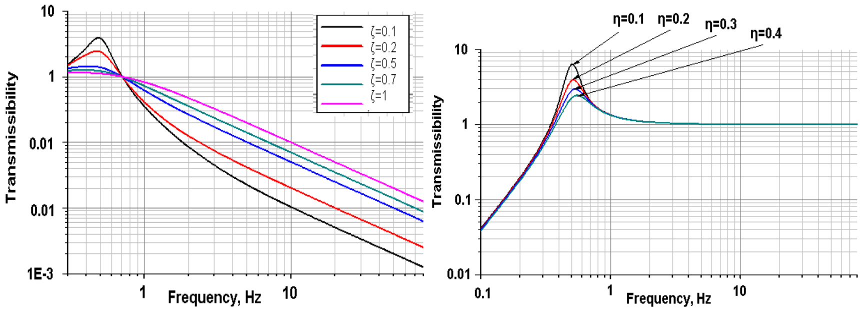


a b

Fig 2.1. Transmissibility: a – absolute; b – relative

Following the study of the opportunities of isolating quasi-zero stiffness vibrations when the system has elastic and viscous or elastic damping and when the internal friction depends on vibration frequency and when it does not depend on frequency, the values of transmissibility established show that vibration isolation starts from 0.7 Hz (Fig. 2.1) and that vibrations are efficiently damped throughout the frequency range.

When the platform undergoes accidental vibrations, the accidental transmission function can be reduced not only by altering the absorption coefficient but also by reducing the system's resonance frequency.



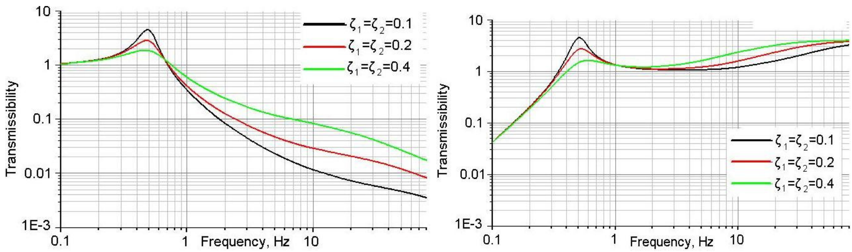
a b

Fig 2.2. Transmissibility: a – viscoelastic damping; b – hysteretic damping

The complex system that was studied theoretically comprises a vibrating platform, an optical table (the weight of the optical slab of the table is not considered) with pneumatic vibration isolators, and a quasi-zero stiffness vibration isolation system with weight *m*; the values of transmissibility of systems of this type were also established.

Under the kinematic harmonious excitation of the platform , the graphs for the absolute acceleration amplitude transmissibility are shown in Figure 2.2a,b.

We can see from Figure 2.3a that if $\omega/\omega_o \geq \sqrt{2}$, the efficiency of vibration isolation is ensured using any damping. In Figure 2.2b, vibration isolation efficiency is ensured throughout the frequency range, if $\xi > 1/\sqrt{2}$, and where $\xi < 1/\sqrt{2}$, then it is ensured in the range $0 < \omega/\omega_o < 1/\sqrt{2}(1 - 2\xi^2)$.



a b

Fig. 2.3. Transmissibility of a complex system: a – absolute; b – relative

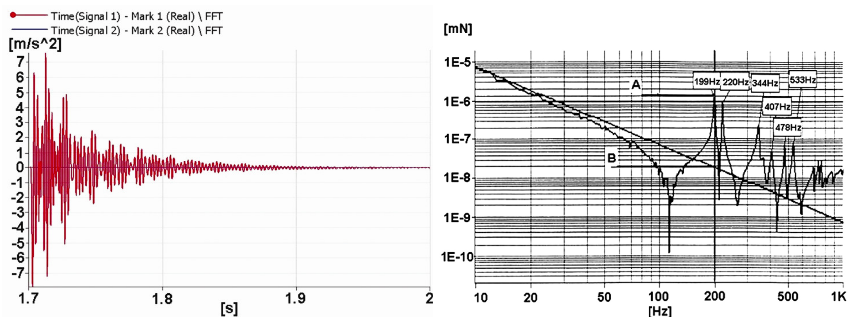
In the fixed frequency ω/ω_o, the degree of the efficiency increases as damping increases; when $\xi = 0$, the efficiency range corresponds to $0 < \omega/\omega_o < 1/\sqrt{2}$.

When the quasi-zero stiffness system is with only viscous damping, the complex system ensures harmonic vibration isolation throughout the low-frequency range; white noise isolation has proved to be most efficient.

3. Experimental research of dynamic charakteristics of optical table and platform

This chapter examines experimental research of the dynamic properties of optical tables and vibrating platforms. Optical tabletops and platforms as well as pneumatic isolators under harmonic, impulse and accidental base excitation were studied. Methods and tools for research of such systems were created. Characteristics of the acceleration time of the honeycomb structure optical slab and platform vibrations were obtained (Fig 3.1a) where all components were quickly damped and two vibration components were damped slowly and their run-out was clearly visible which is confirmed by two high resonance peaks of 199 Hz and 230 Hz (Fig 3.1b) in the dynamic deflection curve. We can see (Fig. 3.1b) that the deflection curve up to 80 Hz fluctuates down without disturbances. This means that the optical table slab and platform are absolutely stiff bodies.

Methods for such studies and tools for studies of systems of this type were developed. Figure 3.1 shows a scheme of study of the dynamic properties of an optical table.



a b

Fig. 3.1. Dynamic properties of an optical slab: a – change of acceleration amplitude response to time; b – dynamic deflection characteristics

Figure 3.2 shows study schemes of the dynamic properties of an optical table.

The dynamic properties of isolation supports of optical tables: resonance frequencies, transmissibility in all three directions (Fig. 3.3), and isolation efficiency (Fig. 3.4) were researched.

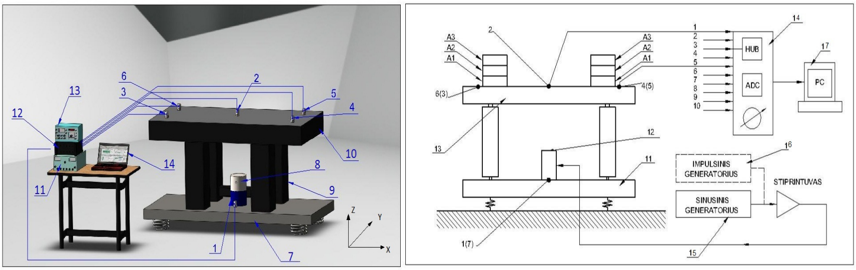
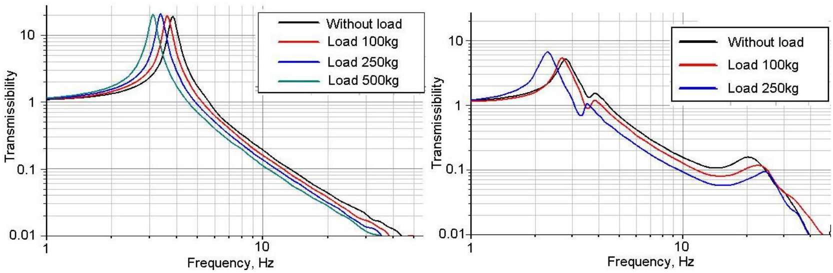


Fig. 3.2. Scheme of study of the dynamic properties of an optical table: 1, 2, 3, 4, 5, 6 – vibration sensors; 7 – platform (base); 8 – vibrator; 9 – vibration isolation supports; 10 – optical table with test vibration isolation supports; 11 – impulse generator; 12 – amplifier; 13 – generator; 14 – computer with an analyser. Directions of vibration excitation: Z – vertical, Y – horizontal transverse, X – horizontal longitudinal



a b

Fig. 3.3. Transmissibility curves: a – vertical direction; b – horizontal direction

In the graphs showing transmissibility in vertical direction one relatively slightly damped resonance can be seen. It shows that vibration isolation supports together with the load create an elementary single mass vibrating system. Research in horizontal direction established that in this direction vibration isolation supports together with the load constitute a complex system with several resonances. This can be explained by the fact that excitation works not through the system mass, but lower. Because of this, torque appears and not only longitudinal vibrations of the load, but also rotating vibrations are excited.

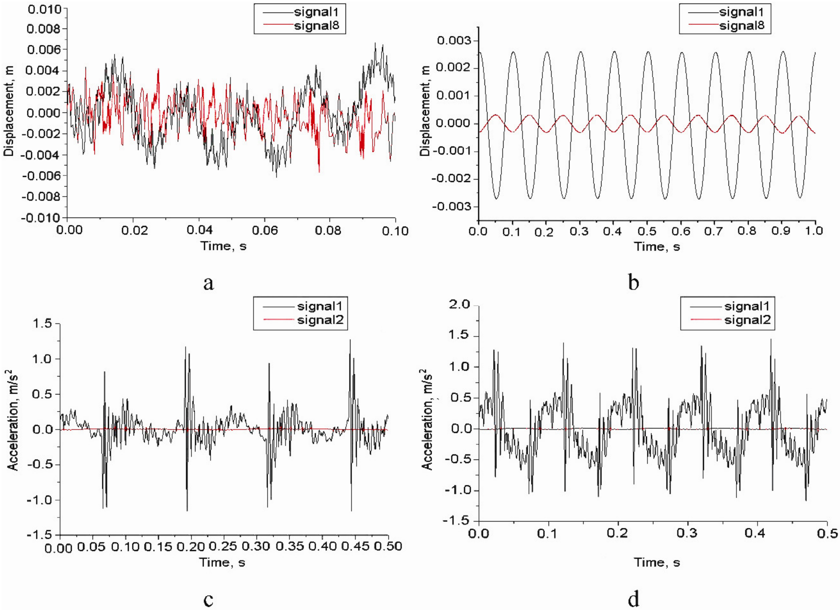


Fig. 3.4. Damping properties of vibration isolation supports: a – 2 Hz harmonious excitation frequency; b – 10 Hz harmonious excitation frequency; c – 4 Hz impulse excitation; d – 10 Hz impulse excitation

The obtained results show that the resonance frequency error when comparing theoretical and experimental research is within 1% limits while damping starts from the frequency of 4 Hz in the case of impulse excitation. During the research of the impact of vibrations of laboratory equipment and the environment spreading through the floor and the research of the optical table isolation capacities it was identified (Fig. 3.5) that vibrations of laboratory equipment and the environment are accidental and their deflection amplitude in longitudinal, transverse, and vertical directions varies from 2.4 μm to 3.2 μm and the acceleration amplitude from 50 $\mu m/s^2$ to 100 $\mu m/s^2$. Deflection amplitude of spectral density under frequency of 0.5 Hz reaches 0.42 μm.

Spectral density of acceleration shows that throughout the frequency range amplitudes reach minus 20 dB and only in the range of 5−50 Hz acceleration amplitudes in vertical direction vary from 4 to 25 dB. So at such low-frequency vibrations optical tables filter only high-frequency acceleration amplitudes and for low-frequency vibration isolation next generation systems are required (Fig. 3.6).

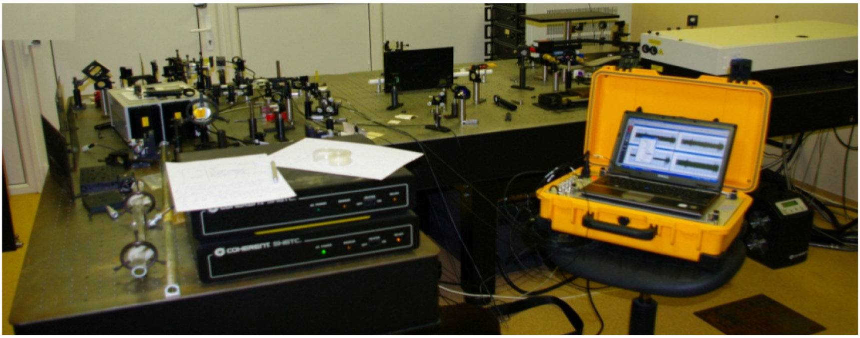
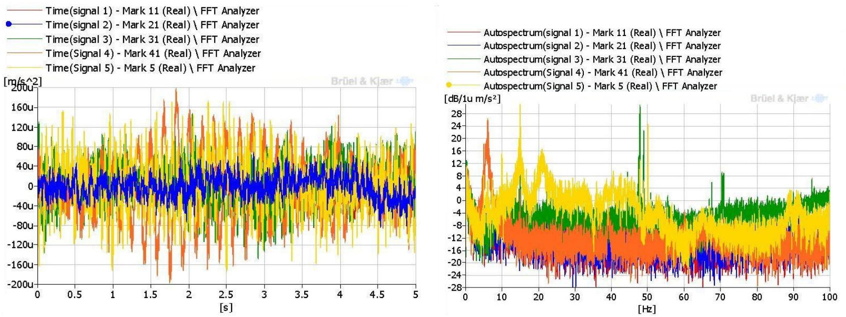


Fig. 3.5. Picture of optical tables studied



a b

Fig. 3.6. Experimental results: a – acceleration amplitudes; b – spectral density

4. Experimental research of mechanical vibration isolation system

The quasi-zero stiffness vibration isolation system and the complex system comprising a vibrating platform, an optical table and a quasi-zero stiffness system were studied and the uncertainty of measurement of vibrations was calculated. Figure 4.1 shows the scheme of study of the quasi-zero stiffness system.

When the platform is under 2 Hz frequency of excitation, vibration amplitude on the quasi-zero (negative) stiffness system decreases 18 times; when the frequency of excitation is 4 Hz, vibration amplitude on the isolation system decreases 90; and when the frequency of excitation is 10 Hz, vibration amplitude decreases 100 times. These results show that the quasi-zero stiffness system efficiently isolates low-frequency vibrations from 0.7 to 100 Hz. This is shown on the transmissibility curve (Fig. 4.2a) and the degree of damping in the case of impulse excitation (Fig. 4.2b). We can see that compared to theoretical transmissibility, when the damping coefficient is between 0.1 and 0.25, the overlay error at multiple frequencies varies from 0.5 % to 1%. Compared to negative stiffness system transmissibility, quasi-zero system isolation capacities are better between 0.8 and 3 Hz; at other frequencies they are similar.

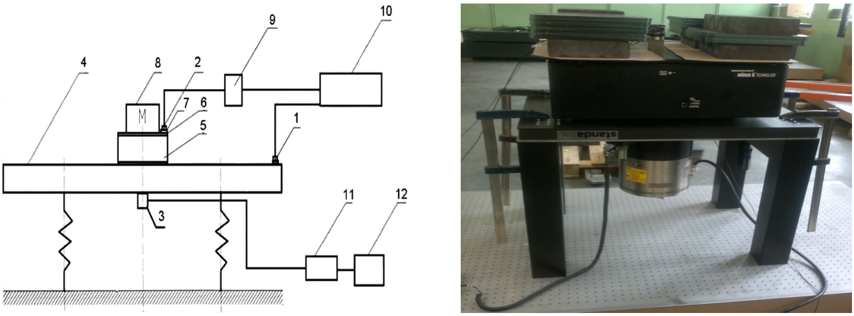
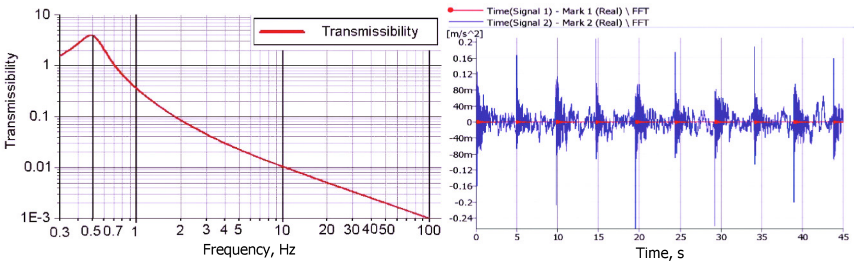


Fig. 4.1. Excitation of vibrations using a vibrator: 1 – seismic accelerometer 8318; 2 – seismic accelerometer 8306; 3 – vibrator; 4 – vibrating platform; 5 – quasi-zero (negative) stiffness vibration isolation system; 6 – attachment slab; 7 – orientation log; 8 – weight: 200 kg; 9 – vibrometer 2511; 10 – portable measurement results processing equipment Machine Diagnostics Toolbox Type 9727 with a DELL computer; 11 – power amplifier 2706; 12 – vibration generator 1027



a b

Fig. 4.2. Quasi-zero (negative) stiffness vibration isolation: a – transmissibility curve; b – impulse excitation

After completion of the research of the capacity to isolate vibrations in a complex system (Fig. 4.3), it was established that this capacity is limited and not suitable for isolating low-frequency vibrations (Fig. 4.4a, b). Certain structural changes must be introduced into the system with a view to reducing the weight of the optical slab to a level where the weight is considerably less than the weight of the isolated load and with a view to solving the system's stiffness and stability problems.

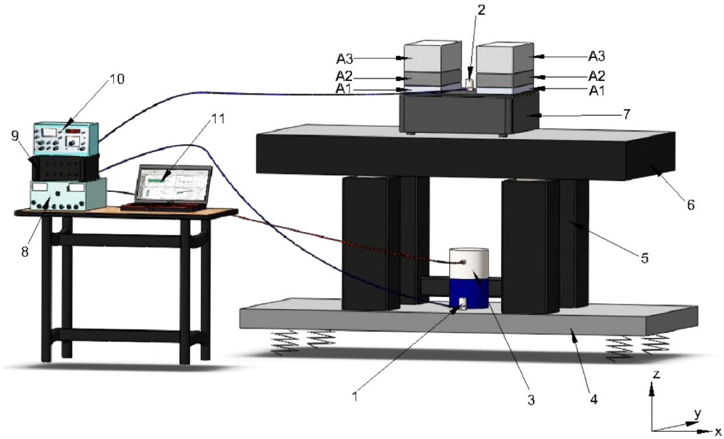


Fig. 4.3. Main view of the complex research system: 1–2 – accelerometers; 3 – vibrator; 4 – vibrating platform; 5 – pneumatic vibration isolators; 6 – optical slab; 7 – quasi-zero stiffness vibration isolation system; A1-A3 – load 330 kg; 8 – sinus vibration generator; 9 – amplifier with vibration analyser; 10 – impulse vibration generator; 11 – measurement results processing station Machine Diagnostics Toolbox Type 9727 with DELL computer

Temporal parameters of low-frequency vibrations are shown in Figure 4.4.

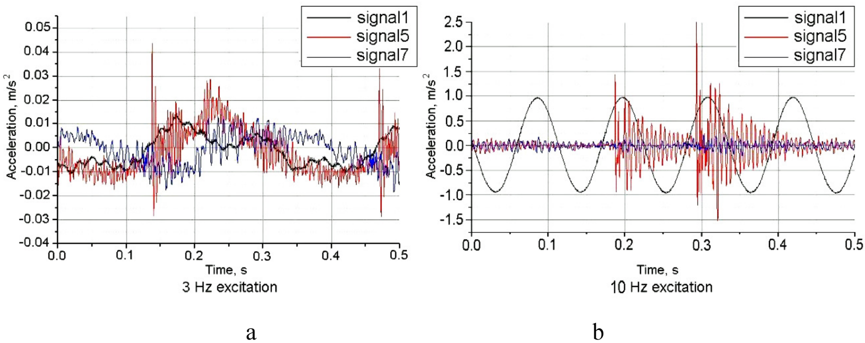


Fig. 4.4. Results of harmonic vibration isolation in a complex system: a – 3 Hz excitation; b – 10 Hz excitation

The uncertainty of the absolute vibrations measurements obtained is 0.45%, which means that the results obtained are reliable.

General conclusions

1. Analysis of sources of information shows that the use of passive and active methods for isolating of low-frequency vibrations is insufficient and/or very expensive, when objects that need to be isolated, precision scales, optical microscopes and other sensitive equipment located in satellites, space stations, and space telescopes are exposed to a vacuum, high or low temperatures, smart vibrations, or radiation.
2. An analytical research of the mechanical quasi-zero stiffness low-frequency vibration isolation systems showed that the isolation of vibrations starts at a frequency of 0.7 Hz and that the efficiency of isolation increases when frequency increases. A patent on the quasi-zero stiffness isolation system was obtained based on the study conducted.
3. The dynamic properties of the complex vibration isolation system were researched and analysed and it was established that systems of this type are effective at damping low-frequency vibrations from 0.8 Hz when the weight of the optical table slab is not considered – that is, when a single-weight system is considered.
4. Methods for conducting experimental studies of honeycomb-structured slabs and pneumatic vibration isolators and for determining their dynamic parameters were proposed and tested. The results of the study show that the dynamic compliance curves of honeycomb-structured optical tables and platforms up to 80 Hz are straight lines, which means that they are absolutely stiff; the first resonance starts at 199 Hz and dynamic deflection coefficients and maximum relative movements of the slabs satisfy the requirements of the best manufacturers. The values of vibration transmissibility of the pneumatic isolators of the tables in the vertical, horizontal transverse, and horizontal longitudinal directions describe the isolation opportunities that start at 5 Hz, while resonance frequencies, depending on the load, vary from 3.1 to 3.9 Hz, which complies with the results of the theoretical study with an error of 1%.
5. The developed methods for determining the dynamic parameters of vibration isolation systems can be used to design the optical tables and other mechanical low-frequency vibration isolation systems. Effective isolator structures can be ensured during operation; the values of parameters and parameter alteration rules can be adjusted. Parameters of systems of this type can be

optimised when the dynamic properties of accidental vibrations changing in time and impacting the systems are unknown.

6. It was experimentally established that a quasi-zero stiffness vibration isolation system efficiently isolates low-frequency vibrations in the 0.7–50 Hz frequency range. The non-compliance error of theoretical and experimental vibration transmissibility for individual frequencies is from 0.5 to 1 %.
7. The dynamic properties of complex systems were studied experimentally and it was established that a complex vibration isolation system is suitable for isolating vibrations only in the 0.8–2 Hz frequency range and in the 10–50 Hz frequency range, while in the 3–10 Hz frequency range it acts as an amplifier and does not isolate vibrations. Consequently, the development of systems of this type calls for additional research with a view to obtaining a single-weight system.

List of published works on the topic of the dissertation

In reviewed scientific periodical publications

Kilikevičius, A.; Jurevičius, M; Berba, M. 2010. Research of dynamics of a vibration isolation platform, *Journal of Vibroengineering* 12(3): 361–367. ISSN 1392-8716 (ISI Web of Science).

Jurevičius, M.; Kilikevičius A.; Berba, M. 2011. Impact of external excitations on the dynamic properties of a negative stiffness vibration isolation table, *Journal of Vibroengineering* 13(2): 352–357. ISSN 1392-8716 (ISI Web of Science).

In other editions

Meškelytė, V.; Jurevičius, M.; Berba, M. 2011. Neigiamo standumo vibroizoliacinio staliuko „Minus K 500BM-1“ tyrimas, *Mokslas – Lietuvos ateitis* 3(6): 56–60. Vilnius: Technika. ISSN 2029-2341.

Berba, M.; Jurevičius, M. 2012. Research of dynamic properties of a complex vibration isolation system, in *The 7th International Symposium "Machine and Industrial Design in Mechanical Engineering": selected papers*, Vol. 3, May 24–26, 2012 Balatonfured, Hungary, 381–384. ISBN 978-86-7892-399-9.

Patent

Berba, M.; Baranov, A.; Jurevičius, M. 2012. *A mechanical quasi-zero stiffness vibration isolation system*. Lithuanian patent LT 5883.

About the author

Michail Berba was born on 20 February 1958 in Kostanay (Kazakhstan). In 1984, he was awarded the diploma of mechanical engineer at Moscow Aviation Institute. In 1984–1987, he worked as a design engineer at the Institute of Physics of the Lithuanian Academy of Sciences. In 1987 M. Berba received an award of the Lithuanian Council of Ministers. Since 1987 he has been the director, chief engineer and technical ideologist of UAB Standa. In 2009–2012 M. Berba was a doctoral student at the Faculty of Mechanical Engineering of Vilnius Gediminas Technical University.

PASYVIŲJŲ ŽEMOJO DAŽNIO VIRPESIŲ IZOLIAVIMO SISTEMŲ TYRIMAS

Problemos formulavimas

Daugelis metrologinių priemonių yra jautrios mechaniniams žemojo dažnio virpesiams ir akustiniam triukšmui. Šie virpesiai gali kilti tiek iš pastato išorės, tiek iš vidaus, t. y. nuo pravažiuojančių mašinų, vėjo, šildymo, vėdinimo ir oro kondicionavimo įrangos bei kitų laboratorinių įrenginių veikimo.

Žemojo dažnio virpesių izoliavimo poreikis auga, nes precizinės svarstyklės, optiniai mikroskopai ir kita jautri įranga, esanti palydovuose, orbitinėse stotyse ir orbitiniuose teleskopuose, turi būti izoliuota nuo virpesių, kad galėtų tinkamai veikti. Virpančios nanoskalės sukelia daug problemų, kai dirbama didesniais greičiais. Šiuo metu taikomi keli žemojo dažnio virpesių izoliavimo metodai – aktyvieji ir pasyvieji. Pasyviosios virpesių izoliavimo sistemos sukuria mažiau šilumos nei aktyviosios sistemos, o tai svarbu, kai dirbama su šilumai jautriomis priemonėmis. Be to, pasyviosios sistemos yra daug pigesnės ir patikimesnės nei aktyviosios ir nereikalauja specialių priemonių slopinimui arba standžiui reguliuoti. Dabartiniu metu pasyviosios virpesių izoliavimo sistemos, tokios kaip optiniai stalai su pneumatiniais virpesių izoliatoriais, lazerių centruose yra plačiai naudojamos žemojo dažnio virpesiams izoliuoti. Optinio stalo tipiniai virpesiai yra nuo 2 iki 7 Hz, nes tai nuosavasis dažnis, kuriame optinis stalas rezonuoja. Tačiau ypač žemiems (nuo 0,7 Hz) dažniams reikia geresnių izoliatorių, dirbant su zondiniais mikroskopais ir interferometrais. Sukurti kvazinulinio (neigiamo) standžio izoliatoriai rezonuoja nuo 0,5 Hz. Šis dažnis beveik neturi energijos, nes dideli virpesiai būtų labai neįprasti esant 0,5 Hz dažniui. Optiniai stalai ir aktyviosios sistemos ne itin gerai veikia, kai patenka į vakuumą, ypač aukštoje arba žemoje temperatūroje ir radiacinėje aplinkoje. Tokia aplinka pasitaiko atliekant specifinius tyrimus su puslaidininkiais. Kvazinulinio (neigiamo) standžio sistemos gali dirbti vakuume, aukštose ir žemose temperatūrose bei veikiant radiacijai. Kvazinulinio (neigiamo) standžio sistemos yra kompaktiškos ir lengvai perkeliamos iš vienos vietos į kitą.

Darbe pagrindinis dėmesys skirtas nustatyti mechaninų pasyvių izoliavimo sistemų gebėjimą izoliuoti žemųjų (0,7–50 Hz) dažnių virpesius.

Darbo aktualumas

Optiniai stalai su pneumatiniais virpesių izoliatoriais tinka lazerių centrams, tačiau dirbant su zondiniais mikroskopais ir interferometrais reikia naujos kartos virpesių izoliavimo sistemų, kurios leistų virpesiams jautriai įrangai, tokiai kaip zondinis mikroskopas, mikroskverbiklio testeris, profilio matuoklis ir skenuojantis elektroninis mikroskopas, veikti šiurkščiomis sąlygomis ir smarkių virpesių aplinkoje, ko negalima būtų pasiekti naudojantis geriausių charakteristikų optiniais stalais ar kitomis aktyviomis virpesių izoliavimo sistemomis. Todėl reikalingi nauji izoliavimo būdai ir sistemos, pagrįstos mechanikos principais, aukšto ergonomiškumo lygio, gerokai kompaktiškesnės nei optiniai stalai ir paprastesnės negu aktyvios virpesių izoliavimo sistemos, kurias būtų galima daug lengviau perkelti iš vienos vietos į kitą. Tai užtikrintų tinkamą virpesių izoliavimo kokybę, leistų naudotis ten, kur to reikia darbo vietose, ant stalų arba grindų, vakuuminėse kamerose, aukštose arba žemose temperatūrose ir radiacinėje aplinkoje.

Tyrimo objektas

Tyrimų objektas – mechaninės pasyviosios žemojo dažnio virpesių izoliavimo sistemos.

Darbo tikslas

Ištirti mechaninę žemojo dažnio pasyviąją virpesių izoliavimo sistemą ir suformuluoti tokių sistemų tyrimo metodiką.

Darbo uždaviniai

Darbo tikslui pasiekti reikia spręsti šiuos uždavinius:

1. Atlikti mokslinės literatūros apžvalgą apie virpesių izoliavimo tipus, sistemas, jų tyrimus ir analizę.
2. Išanalizuoti ir pagrįsti mechaninių žemojo dažnio virpesių izoliavimo sistemų dinaminį efektyvumą.
3. Pasiūlyti optinių stalų ir pneumatinių virpesių izoliatorių eksperimentinių tyrimų metodiką.

4. Atlikti kvazinulinio (neigiamo) standžio virpesių izoliavimo sistemų dinaminių charakteristikų eksperimentinius tyrimus.
5. Atlikti sudėtinių virpesių izoliavimo sistemų dinamikos eksperimentinius tyrimus ir įvertinti gautų rezultatų neapibrėžtį.

Tyrimų metodika

Rengiant darbą atlikti teoriniai ir eksperimentiniai tyrimai, taikant skaitmeninės analizės principus, atliekant lyginamuosius virpesių izoliavimo sistemų matavimus ir taikant matematinę statistinę analizę gautų rezultatų patikimumui įvertinti.

Darbo mokslinis naujumas

Rengiant disertaciją buvo gauti šie mechanikos inžinerijos mokslui nauji rezultatai:

1. Atlikta žemojo dažnio virpesių izoliavimo sistemų analizė, taikant lyginamuosius jų efektyvumo vertinimo metodus. Gautos mechaninių pasyviųjų žemojo dažnio virpesių izoliavimo sistemų dinaminės charakteristikos (perduodamumas) leidžia įvertinti virpesių izoliavimo efektyvumą. Nustatyta, kad tokios sistemos gali izoliuoti virpesius 0,7–50 Hz dažnių ruože.
2. Pasiūlyti optinių stalų ir pneumatinių virpesių izoliatorių eksperimentiniai tyrimo metodai, leidžiantys projektuoti ir tobulinti tokias sistemas tais atvejais, kai jų dinaminės charakteristikos yra nežinomos, o kitus metodus taikyti sunku. Pateikti eksperimentinio tyrimo rezultatai rodo, kad optinės plokštės ir platforma iki 80 Hz yra absoliučiai standūs kūnai.
3. Sukurti kvazinulinio standžio virpesių izoliavimo metodai leidžia spręsti žemojo dažnio (0,7–50 Hz) virpesių izoliavimą, veikiant harmoniniam, impulsiniam ir baltajam triukšmui.

Darbo rezultatų praktinė reikšmė

Taikant parengtą mechaninės koncepcijos žemojo dažnio pasyviąją virpesių izoliavimo sistemą, yra labai supaprastinamas virpesių izoliavimo procesas, jis tampa pigesnis, užima mažiau laiko ir darbo sąnaudų, palyginti su kitais šiuo metu taikomais virpesių izoliavimo būdais.

Ginamieji teiginiai

1. Naujas kvazinulinio standžio virpesių izoliavimo metodas pagrįstas mechanikos principais ir masės išlyginimu.

2. Sukurta eksperimentinio tyrimo metodika leidžia įvertinti virpesių izoliavimo sistemų dinamines charakteristikas ir jų skaitines vertes.

Darbo apimtis

Disertaciją sudaro įvadas, keturi skyriai, bendrosios išvados, naudotos literatūros ir autoriaus publikacijų disertacijos tema sąrašai, trys priedai.

Įvadiniame skyriuje aptariama tiriamoji problema, darbo aktualumas, aprašomas tyrimų objektas, formuluojamas darbo tikslas ir uždaviniai, aprašoma tyrimų metodika, darbo mokslinis naujumas, darbo rezultatų praktinė reikšmė, ginamieji teiginiai. Įvado pabaigoje pristatomos autoriaus paskelbtos publikacijos ir pranešimai konferencijose disertacijos tema, disertacijos struktūra.

Pirmajame skyriuje analizuojami virpesių izoliavimo sistemų tipai, pasyviųjų, pusiau aktyvių ir aktyviųjų virpesių izoliavimo sistemų pranašumai ir trūkumai.

Antrajame skyriuje pateikti naujo tipo virpesių izoliavimo sistemų analizė ir dinaminiai tyrimai.

Trečiajame skyriuje pateikta optinių stalų ir platformos dinaminių charakteristikų eksperimentiniai tyrimai.

Ketvirtajame skyriuje pateikti kvazinulinio standžio ir sudėtinių sistemų eksperimentiniai tyrimai ir virpesių matavimo rezultatų neapibrėžčių tyrimas.

Bendrosios išvados

1. Literatūros šaltinių analizė rodo, kad pasyviųjų ir aktyviųjų metodų taikymas žemojo dažnio virpesiams izoliuoti yra nepakankamas arba labai brangus, kai izoliavimo objektai, precizinės svarstyklės, optiniai mikroskopai ir kita jautri įranga, esanti palydovuose, orbitinėse stotyse, orbitiniuose teleskopuose, yra veikiami vakuumo, aukštos arba žemos temperatūrų, didelės amplidutės virpesių ir radiacijos.
2. Mechaninės kvazinulinio standžio žemojo dažnio virpesių izoliavimo sistemos analitinis tyrimas parodė, kad virpesių izoliavimas prasideda nuo 0,7 Hz dažnio, o dažniui didėjant izoliavimo efektyvumas didėja.
3. Analitiškai ištirti sudėtinės virpesių izoliavimo sistemos dinaminiai parametrai ir nustatyta, kad tokios sistemos gerai slopina žemojo dažnio virpesius nuo 0,8 Hz, kai optinio stalo plokštės masė nevertinama, t. y. kai nagrinėjama vienos masės sistema.
4. Pasiūlyta ir aprobuota korinės konstrukcijos plokščiųjų ir pneumatinių virpesių izoliatorių eksperimentinio tyrimo bei dinaminių parametrų nustatymo metodika. Pateikti tyrimo rezultatai rodo, kad korinės konstrukci-

jos optinių plokščių ir platformos dinaminio slankio charakteristikos iki 80 Hz gali būti vertinamos kaip tiesinės. Tai reiškia, kad jos yra absoliučiai standžios, pirmasis rezonansas prasideda nuo 199 Hz, o dinaminio įlinkio koeficientai ir maksimalūs reliatyvieji plokščių judėjimai atitinka geriausių gamintojų reikalavimus, stalų pneumatinių izoliatorių virpesių perduodamumas vertikaliąja, horizontaliąja skersine ir horizontaliąja išilgine kryptimi nusako izoliavimo galimybes, kurios prasideda nuo 5 Hz, o rezonansiniai dažniai atsižvelgiant į apkrovą svyruoja nuo 3,1 iki 3,9 Hz. Tai atitinka teorinių tyrimų rezultatus su 1 % paklaida.

5. Sukurta virpesių izoliavimo sistemų dinaminių parametrų eksperimentinio nustatymo metodika gali būti taikoma optinių stalų ir kitų mechaninių žemojo dažnio virpesių izoliavimo sistemoms projektuoti, eksploatavimo metu randant efektyvias izoliatorių struktūras, jų parametrų vertes, kitimo dėsnius, optimizuojant tokių sistemų parametrus, kai nežinomos laike kintančios atsitiktinių sistemą veikiančių virpesių dinaminės charakteristikos.
6. Eksperimentiškai nustatyta, kad kvazinulinio standžio virpesių izoliavimo sistema gerai izoliuoja žemojo dažnio virpesius 0,7–50 Hz dažnių juostoje. Teorinės ir eksperimentinės virpesių perdavimo nesutapimo paklaida, esant šiems dažniams, svyruoja nuo 0,5 % iki 1 %.
7. Eksperimentiškai ištirtos sudėtinių sistemų dinaminės charakteristikos ir nustatyta, kad sudėtinė virpesių izoliavimo sistema tinka izoliuoti virpesius tik nuo 0,8 iki 2 Hz dažnių juostoje ir nuo 10 iki 50 Hz dažnių juostoje, o nuo 3 iki 10 Hz dažnio juostoje veikia kaip stiprintuvas ir virpesių neizoliuoja. Taigi tokių sistemų kūrimui reikalingi papildomi tyrimai, siekiant gauti vienos masės sistemą.

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RESEARCH OF PASSIVE LOW-FREQUENCY
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