UDC 699.844.1
IRSTI 67.01.05
RESEARCH ARTICLE

SINGLE-DEGREE-OF-FREEDOM VIBRATION ISOLATION SYSTEM WITH ONE ADDITIONAL SUPPORT

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Abstract. *Vibration isolation is one of the most effective methods for reducing vibration levels of supporting structures when installing vibroactive equipment (active vibration isolation) or vibration levels of vibro-sensitive objects relative to foundation vibration levels (passive vibration isolation). Damping devices utilizing high-speed fluid flow through apertures have found wide applications in shock vibration isolation and vibration isolation systems in aerospace and defense sectors. Recent research has led to the development of viscous fluid dampers (VFDs) for use in civil engineering, particularly in earthquake-prone areas. Scientists conducted experiments aimed at determining the ability of viscous fluid dampers to reduce damages and displacements of structures without increasing stresses. Mathematical models have been developed and are applied in vibration isolation systems. When vibroactive equipment is installed on building and structure support systems, vibrations with sufficiently high vibration parameters may occur, potentially leading to loss of load-bearing capacity. In such cases, vibration isolation is considered one of the most effective methods for reducing these vibration levels. In this study, a calculation method has been developed, and calculation dependencies and algorithms for calculating vibration protection systems with nonlinear characteristics (additional support connections, viscous fluid damper) have been derived for both single-degree-of-freedom and two-degree-of-freedom systems. The research involved an analysis of normative and scientific-technical literature on the subject: types, structural solutions, calculation, and analysis of vibration protection systems. The main method selected was based on the use of transfer functions of linear systems (the non-traditional "normal form" method). Calculations were performed using computer mathematics systems- Matlab.*

Keywords: *vibration isolation, damper, viscous friction, vibration level, vibro-sensitive objects, damping devices, linear system.*

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https://doi.org/10.51488/1680-080X/2024.2-12
Received 09 April 2023; Revised 17 June 2023; Accepted 24 September 2023

ӘОЖ 699.844.1
ҒТАМР 67.01.05
ҒЫЛЫМИ МАҚАЛА

БІР ЕРКІНДІК ДӘРЕЖЕСІ ЖӘНЕ БІР ҚОСЫМША БАЙЛАНЫСЫ БАР ДІРІЛ ОҚШАУЛАУ ЖҮЙЕСІ

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Аңдатпа. *Дірілді оқшаулау іргетастың діріл деңгейіне (пассивті дірілді оқшаулау) қатысты виброактивті жабдықты (белсенді дірілді оқшаулау) немесе дірілге сезімтал объектілердің діріл деңгейлерін орнату кезінде тірек құрылымдарының діріл деңгейін төмендетудің ең тиімді әдістерінің бірі болып табылады. Апертуралар арқылы жоғары жылдамдықты сұйықтық ағынын пайдаланатын демпферлік құрылғылар аэроғарыштық және қорғаныс секторларында соққы дірілін оқшаулау және дірілді оқшаулау жүйелерінде кең қолданыс тапты. Соңғы зерттеулер азаматтық құрылыста, әсіресе жер сілкіністері қаупі бар аймақтарда қолдануға арналған тұтқыр сұйықтық амортизаторларының (VFD) дамуына әкелді. Ғалымдар тұтқыр сұйықтық амортизаторларының кернеулерді арттырмай құрылымдардың зақымдануы мен жылжуын азайту қабілетін анықтауға бағытталған эксперименттер жүргізді. Математикалық модельдер әзірленді және дірілді оқшаулау жүйелерінде қолданылады. Ғимарат пен құрылысты қолдау жүйелерінде виброактивті жабдық орнатылған кезде, діріл параметрлері жеткілікті жоғары діріл пайда болуы мүмкін, бұл жүк көтеру қабілетінің жоғалуына әкелуі мүмкін. Мұндай жағдайларда дірілді оқшаулау осы діріл деңгейлерін төмендетудің ең тиімді әдістерінің бірі болып саналады. Бұл зерттеуде есептеу әдісі әзірленді және сызықтық емес сипаттамалары бар дірілден қорғау жүйелерін есептеу алгоритмдері (қосымша тірек қосылыстары, тұтқыр сұйықтықтың демпфері) бір еркіндік дәрежесі үшін де, екі дәрежелі үшін де есептелді. еркіндік жүйелері. Зерттеу осы тақырып бойынша нормативтік және ғылыми-техникалық әдебиеттерді талдауды қамтыды: дірілден қорғау жүйелерінің түрлері, құрылымдық шешімдері, есептеулері және талдаулары. Таңдалған негізгі әдіс сызықтық жүйелердің тасымалдау функцияларын қолдануға негізделген (дәстүрлі емес «қалыпты форма» әдісі). Есептер компьютерлік математикалық жүйелер – Matlab көмегімен орындалды*.

Түйін сөздер: *дірілді оқшаулау, демпфер, тұтқыр үйкеліс, діріл деңгейі, вибросезімтал объектілер, демпферлік құрылғылар, сызықтық жүйе.*

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https://doi.org/10.51488/1680-080X/2024.2-12
Алынды 09 сәуір 2023; Қайта қаралды 17 маусым 2023; Қабылданды 24 қыркүйек 2023.

УДК 699.844.1
МРНТИ 67.01.05
НАУЧНАЯ СТАТЬЯ

СИСТЕМА ВИБРОЗАЩИТЫ С ОДНОЙ СТЕПЕНЬЮ СВОБОДЫ И С ОДНОЙ ДОПОЛНИТЕЛЬНОЙ СВЯЗЬЮ

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Аннотация. *Виброизоляция является одним из наиболее эффективных методов снижения уровня вибрации несущих конструкций при монтаже виброактивного оборудования (активная виброизоляция) или уровня вибрации виброчувствительных объектов относительно уровней вибрации фундамента (пассивная виброизоляция). Демпфирующие устройства, использующие высокоскоростной поток жидкости через отверстия, нашли широкое применение в системах виброизоляции и виброизоляции в аэрокосмической и оборонной отраслях. Недавние исследования привели к разработке демпферов на основе вязкой жидкости (VFD) для использования в гражданском строительстве, особенно в сейсмоопасных районах. Ученые провели эксперименты, направленные на определение способности вязкостных демпферов уменьшать повреждения и смещения конструкций без увеличения напряжений. Разработаны математические модели, которые применяются в системах виброизоляции. При установке виброактивного оборудования на опорные системы зданий и сооружений могут возникать вибрации с достаточно высокими параметрами вибрации, потенциально приводящие к потере несущей способности. В таких случаях виброизоляция считается одним из наиболее эффективных методов снижения уровня вибрации. В работе разработана методика расчета, получены расчетные зависимости и алгоритмы расчета систем виброзащиты с нелинейными характеристиками (дополнительные опорные соединения, вязкостной демпфер) как для одностепенных, так и для двухстепенных степеней свободы. системы свободы. В ходе исследования был проведен анализ нормативной и научно-технической литературы по теме: типы, конструктивные решения, расчет и анализ систем виброзащиты. В качестве основного метода выбран метод, основанный на использовании передаточных функций линейных систем (нетрадиционный метод «нормальной формы»). Расчеты проводились с использованием системы компьютерной математики Matlab.*

Ключевые слова:. *виброизоляция, демпфер, вязкое трение, уровень вибрации, виброчувствительные объекты, демпфирующие устройтсва, линейная система.*

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https://doi.org/10.51488/1680-080X/2024.2-12
Поступила 9 апреля 2023 г.; Пересмотрено 17 июня 2023 г.; Принято 24 сентября 2023 г.

ACKNOWLEDGEMENTS/SOURCE OF FUNDING

The study was conducted using private sources of funding.

CONFLICT OF INTEREST

The authors state that there is no conflict of interest.

АЛҒЫС/ҚАРЖЫЛАНДЫРУ КӨЗІ

Зерттеу жеке қаржыландыру көздерін пайдалана отырып жүргізілді.

МҮДДЕЛЕР ҚАҚТЫҒЫСЫ

Авторлар мүдделер қақтығысы жоқ деп мәлімдейді.

БЛАГОДАРНОСТИ/ИСТОЧНИК ФИНАНСИРОВАНИЯ

Исследование проводилось с использованием частных источников финансирования.

КОНФЛИКТ ИНТЕРЕСОВ

Авторы заявляют, что конфликта интересов нет.

1 INTRODUCTION

Vibrations caused by vibration-active equipment can have serious consequences for the structures of buildings and structures. These consequences include cracking, additional precipitation, disruption of processes, discomfort, and exceeding levels permitted by sanitary standards. To reduce the level of vibrations, the vibration isolation method is widely used, which includes active and passive approaches. Active vibration isolation is aimed at reducing the load on the base, and passive isolation is aimed at reducing the level of vibrations of vibration-sensitive equipment.

The most effective use of vibration isolation is observed in the case of machines with periodic loads, such as pumps, fans, compressors and other equipment.

At certain ratios of natural vibration frequencies of vibration-isolated equipment and shock loads, as well as supporting structures, the load transmitted to the supporting structures can be significantly reduced. When assessing the effectiveness of vibration isolation systems for machines with periodic loads, it is necessary to take into account transient modes (starting and stopping) and possible resonance.

However, with large movements in vibration isolation areas, problems with contact with auxiliary equipment, pipelines, etc. may arise. The vibration isolators themselves are also susceptible to destruction due to low-cycle fatigue.

To reduce the level of vibrations in these modes, additional elements can be used, such as connections, dissipative systems and additional masses. In this case, the characteristics of vibration protection systems become nonlinear, and their calculation is reduced to the analysis of nonlinear systems with a finite number of degrees of freedom (DOF).

In this study, a calculation method has been developed, and design dependencies have been derived and algorithms for calculating vibration protection systems with nonlinear characteristics (additional support link, viscous friction damper) as systems with one degree of freedom and two degrees of freedom have been compiled.

2 LITERATURE REVIEW

To calculate systems with a finite number of degrees of freedom and, in particular, vibration protection systems, the method of proffessor Chernov Y. T. (**Chernov Y. T., 2011**). The general theory of calculating linear systems using methods based on transition functions, impulse transition functions and their connections was given by prof. Solodovnikov V. V. (**1960**). When constructing solutions, in contrast to the traditional method of "normal forms," there is no need to construct one's own forms, write and solve equations in principal coordinates, and go back to generalized coordinates" (**Krylov & Bogolyubov, 2004**).

In the literature, there are much fewer works that provide calculation and assessment of vibration levels in vibration isolation systems in transient modes (start and stop).

High levels of vibrations in such modes can cause a breakdown in communication with additional equipment, such as pipelines, and cause destruction of vibration isolators, in particular metal ones, as a result of low-cycle fatigue.

The work (**Rodriquez et al., 1994; Martinez-Rodrigo & Romero, 2003; Osipova, 2014**) describe methods for calculating and analyzing the nature of oscillations in transient modes, including depending on the start and stop time intervals. Many examples of calculating systems with one degree of freedom and two degrees of freedom under different laws of changes in the frequencies of forced oscillations in transient modes are presented in the works of Ivovich V. A. (**1984**).

This study examined the problems of calculating linear and nonlinear vibration isolation systems as systems with a finite number of degrees of freedom (two degrees) in operational and transient modes. Under operating conditions, the calculation formulas were brought into a closed form, in the form of an expansion into the proper forms of linear systems immediately relative to generalized coordinates.

One of the common options for reducing vibration levels in transient modes is the introduction of additional elements that are activated during large movements in resonant zones in transient modes.

The algorithm for calculating nonlinear systems using the “normal forms” method is considered in Chernov (**Chernov, 2011**). This method is well studied and widely used in the study of linear dynamic systems. For example, it was successfully applied to the calculation of nonlinear systems with a finite number of degrees of freedom. This method is especially effective when calculating “systems with a large number of degrees of freedom.” “One of the main stages of calculation using the “normal forms” method is the determination of the system’s own forms and their normalization”. The method is based on the concept of the movement of system masses at an arbitrary point in time, shown in the form of an expansion in eigenvectors (principal coordinates):

$$\vec{y} = \Phi\vec{a} \tag{1}$$

where Φ-matrixnormalized proper forms;$\vec{a}$ - vector of main coordinates.

In this case, the associated equations of motion have the form:

$$M\ddot{\vec{y}} + D\dot{\vec{y}} + K\vec{y} = \vec{q}(t) \tag{2}$$

where M, D and K are the matrices of mass, dissipation and rigidity of the system, respectively.$\vec{y}$,(t)$\vec{q}$ –vectors of displacements of the system and external load applied to the masses.

These equations are transformed into equations of motion unrelated to the principal coordinates, similar to the equations of motion of a system with one degree of freedom":

$$\ddot{a_r} + d_r\dot{a}_r + p_r^2 a_r = b_r(t), \quad (r = 1,2 \dots n) \tag{3}$$

where a_r- main coordinates,

r- own form number;

d_r, p_r- dissipative coefficients and frequencies of natural oscillations;

br(t)=Φ'$\vec{q}$(t)-representation of the external load in the form of expansion in terms of its own forms (Φ' - transposed matrix of normalized eigenforms).

According to the “frequency-independent friction hypothesis,” one should accept:

$$d_r = p_r\,\gamma_r \tag{4}$$

whereγ_r –coefficients of inelastic resistance of the system, all values of which are usually taken equal.

The solution to equation (3) is usually represented using the Duhamel integral:

$$a_r(t) = \frac{1}{p_r^*}\int_0^1 b_r(\tau)e^{-n_r(t-\tau)}\sin p_r^*(t-\tau)d\tau\,, \quad r = 1,2 \dots n) \tag{5}$$

where

$$n_{r=}\frac{d_r}{2} = \frac{\mathrm{p_r\gamma_r}}{2} \tag{6}$$

$p_r^* = \sqrt{p_r^2 - n_r^2} = p_r^2\sqrt{1 - \frac{\gamma_r^2}{4}}$ - natural oscillation frequencies taking into account damping.

Displacements in the original system in generalized coordinates are determined by formula (1).

Research has shown that nonlinear VF dampers are very reliable (**Osipova, 2013**). The medical center in San Bernardino County, California, is a five-story facility that uses 400 high-pressure rubber bearings and 233 α = 0.5 nonlinear VF dampers. In addition, studies conducted on the seismic retrofitting of the Golden Gate Bridge suspension section in San Francisco showed that the use of VF dampers with a coefficient of α = 0.75 leads to the desired efficiency (**Krylov & Bogolyubov, 2004**).

In some cases, VF dampers are used in conjunction with seismic isolation systems. For example, they have been integrated into the vibration isolation system of five buildings of a new medical center in San Bernardino County, located close to two major fault lines, since its construction in 1995 (**Soong & Dargush, 1998**).

Viscous friction dampers are used in some tall engineering structures exposed to gusty winds to mitigate dynamic loads. In China, they have been used, for example, in Beijing Yintai Center, Yizhenyuan Building in Huaiying City, Beijing Exhibition Center Building and Beijing Yintai Center Tower Central Building (**Hongnan & Linsheng, 2010**).

A study of a 9-story building with viscous friction dampers installed on the 5th and 9th floors showed a 32% reduction in displacements and a 53% reduction in accelerations on the 5th floor. On the 9th floor, a decrease in displacements by 36% and accelerations by 75% was noted (**Khan et al., 2014**).

In (**TsNIISK named after V. A. Kucherenko, 1986**), recommendations are given for vibration protection of buildings using vibration dampers and additional vibration insulation, but protection methods using additional support are not considered.

In (**Dubinin et al., 2023**) discusses information modeling and rules for compiling models of objects at different stages of operation of structures, incl. and when resonance phenomena occur during startup and shutdown of the object. In (**Polyakova et al., 2022**) outlines the solution algorithm (system of equations 7) and obtains the main characteristics for dynamic effects on structural elements, but does not consider the use of dampers or other vibration dampers (system of equations 8). As a special case of confirmation of these conclusions, the characteristics of the operation of objects in static mode are given (**Polyakova et al., 2021**).

$$\frac{du_r}{ds} = -\nu_1 E_1 \frac{\cos\theta}{r} u_r - \nu_1 \frac{E_2}{E_1}\frac{\cos\theta}{r}\frac{\partial v}{\partial\varphi} - \sin\theta * \vartheta + \frac{1-\nu_1\nu_2}{E_1 h}\cos^2\theta * Q_r + \frac{1-\nu_1\nu_2}{E_1 h}\sin\theta\cos\theta * Q_z,$$

$$\frac{du_z}{ds} = -\nu_1 \frac{E_2}{E_1}\frac{\sin\theta}{r} ur - \nu_1 \frac{E_2}{E_1}\frac{\sin\theta}{r}\frac{dv}{d\varphi} - \cos\theta * \vartheta + \frac{1-\nu_1\nu_2}{E_1 h}\sin\theta\cos\theta * Q_r + \frac{1-\nu_1\nu_2}{E_1 h}\sin^2\theta * Q_z,$$

$$\frac{dv}{ds} = -\frac{cos\theta}{r}\frac{\partial u_r}{\partial\varphi} - \frac{sin\theta}{r}\frac{\partial u_z}{\partial\varphi} + \frac{cos\theta}{r} v - \frac{h^2}{3r^2} sin\theta \frac{\partial\vartheta}{\partial\varphi} + \frac{2(1+\nu_1)}{E_1 h} S^*, \tag{7}$$

$$\frac{d\vartheta}{ds} = -\nu_1 \frac{E_2}{E_1}\frac{cos\theta}{r}\vartheta + \frac{12(1-\nu_1\nu_2)}{E_1 h^3} M_1 - \nu_1 \frac{E_2}{E_1}\frac{sin\theta}{r^2}\frac{\partial^2 u_r}{\partial\varphi^2} + \nu_1 \frac{E_2}{E_1}\frac{cos\theta}{r^2}\frac{\partial^2 u_z}{\partial\varphi^2} + \nu_1 \frac{E_2}{E_1}\frac{sin\theta}{r^2}\frac{\partial v}{\partial\varphi},$$

$$\frac{dQ_r}{ds} = \frac{E_2 h}{r^2} u_r - \frac{E_2 h}{r^2}\frac{\partial v}{\partial\varphi} - \frac{E_2 h^3}{12 r^3} sin\theta cos\theta \frac{\partial^2\vartheta}{\partial\varphi^2} - (1-\nu_2)\frac{cos\theta}{r} Q_r + \nu_2 \frac{sin\theta}{r} Q_z - \frac{cos\theta}{r}\frac{\partial S^*}{\partial\varphi} + \nu_2 \frac{sin\theta}{r}\frac{\partial^2 M_1}{\partial\varphi} - q_r,$$

$$\frac{dQ_z}{ds} = -\frac{E_2 h^3}{6(1+\nu_2) r^4}\frac{\partial^2 u_z}{\partial\varphi^2} + \frac{E_2 h^3}{6(1+\nu_2) r^3}\frac{\partial^2\vartheta}{\partial\varphi^2} + \frac{E_2 h^3 cos^3\theta}{12 r^3}\frac{\partial^2\vartheta}{\partial\varphi^2} - \frac{cos\theta}{r} Q_z - \frac{sin\theta}{r}\frac{\partial S^*}{\partial\varphi} + \nu_2 \frac{cos\theta}{r^2}\frac{\partial^2 M_1}{\partial\varphi^2} - q_z,$$

$$\frac{dS^*}{ds} = -\frac{E_2 h}{r^2}\frac{\partial u_r}{\partial\varphi} - \frac{E_2 h}{r^2}\frac{\partial^2 v}{\partial\varphi^2} - \frac{E_2 h^3}{12}\frac{sin\theta cos\theta}{r^3}\frac{\partial\vartheta}{\partial\varphi} + \nu_2 \frac{cos\theta}{r}\frac{\partial Q_r}{\partial\varphi} + \nu_2 \frac{sin\theta}{r}\frac{\partial Q_z}{\partial\varphi} - 2\frac{cos\theta}{r} S^* + \nu_2 \frac{sin\theta}{r}\frac{\partial M_1}{\partial\varphi} - q_v,$$

$$\frac{dM1}{ds} = -\frac{E_2 h^3}{12}\frac{\sin\theta\cos\theta}{r^3}\frac{\partial^2 u_r}{\partial\varphi^2} + \frac{E_2 h^3}{12}\frac{\cos\theta}{r^3}\frac{\partial^2 u_z}{\partial\varphi^2} + \frac{E_2 h^3}{6(1+\nu_2) r^3}\frac{\partial^2 u_z}{\partial\varphi^2} + \frac{E_2 h^3}{12}\frac{\sin\theta\cos\theta}{r^3}\frac{dv}{d\varphi} + \frac{E_2 h^3}{12}\cos\theta * \vartheta - \frac{E_2 h^3}{6(1+\nu_2) r^2}\frac{\partial^2\vartheta}{\partial\varphi^2} + \sin\theta * Q_r - \cos\theta * Q_z - \frac{h^2\sin\theta}{3r^2}\frac{dS^*}{d\varphi} - (1-\nu_2)\frac{\cos\theta}{r} M_1,$$

$$\frac{du_r}{ds} = \frac{1-\nu_1\nu_2}{E_1 h} Q_z \sin\theta\cos\theta + \frac{1-\nu_1\nu_2}{E_1 h} Q_r \cos^2\theta - \nu_1 \frac{E_2}{E_1}\cos\theta \frac{u_r}{r} - \vartheta\sin\theta,$$

$$\frac{du_z}{ds}=\frac{1-\nu_1\nu_2}{E_1h}Q_z\sin^2\theta+\frac{1-\nu_1\nu_2}{E_1h}Q_r\sin\theta\cos^2\theta-\nu_1\frac{E_2}{E_1}\sin\theta\frac{u_r}{r}+\vartheta\cos\theta,$$
$$\frac{d\vartheta}{ds}=\frac{12(1-\nu_1\nu_2)}{E_1h^3}M_1-\frac{E_2}{E_1}\nu_1\frac{\cos\theta}{r}\vartheta,$$
$$\frac{dQ_r}{ds}=\frac{E_2h}{r^2}u_r-\frac{(1-\nu_2)\cos\theta}{r}Q_r+\frac{\nu_2\sin\theta}{r}Q_z-q_r, \qquad (8)$$
$$\frac{dQ_z}{ds}=-\frac{\cos\theta}{r}Q_z-q_z\cdot r,$$
$$\frac{dM_1}{ds}=\frac{E_2h^3\vartheta}{12r^2}\cos^2\theta+Q_r\sin\theta-Q_z\cos\theta-\frac{(1-\nu_2)\cos\theta}{r}M_1.$$

3 MATERIALS AND METHODS

To dampen vibrations in transient modes in many fields of technology, including the operation of vibration-isolating equipment, systems with nonlinear elements - limiters and viscous friction dampers - are widely used. Problems and algorithms for solving such systems in operational and transient modes are considered. The system oscillation equation (**Figure 1a**) has the form:

$$m\ddot{y}+\left(1+2\nu\frac{d}{dt}\right)c(y)y=q(t) \qquad (9)$$

Dependence of “reaction – displacement” for the accepted type of nonlinearity:

$$c(y)y=k_1y \text{ for } y\leq y_0;\; c(y)y=k_1y_0+(k_1+k_2)(y-y_0) \text{ for } y>y_0 \qquad (10)$$

Further, during the transformation, the system under consideration, taking into account nonlinearity, will take the form:

$$\ddot{y}+\left(1+2\nu\frac{d}{dt}\right)p_1^2y=\frac{q(t)}{m}-\left(1+2\nu\frac{d}{dt}\right)\frac{k_2(y-y_0)}{m} \qquad (11)$$

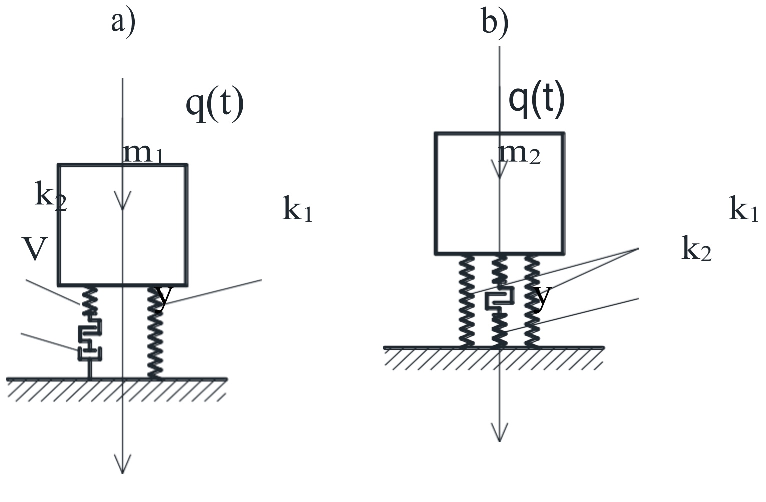

Figure 1 – a) system with limiter; b) system with viscous friction damper (author’s material)

The solution to equation (12) is presented in the form of two solutions; linear system to an external (y_l) and fictitious load, and which will depend on the type of nonlinearity (y_{nl}).

$$y = y_l - y_{nl} \tag{12}$$

The solution to the equation from the external load is written in the form of the Duhamel integral (**Chernov, 2011**):

$$y_л = \frac{1}{p_1^* m}\int_0^1 q(\tau)e^{-n_1(1-\tau)}\sin p_1^*(1-\tau)d\tau = \frac{1}{mp_1}[d_1(t)F_2(t) - \quad d_2(t)F_1(t)] \tag{13}$$

where $2n_1 = 2\nu p_1^2$, $p_1^* = \sqrt{p_1^2 - n_1^2}$,

$$d_1 = e^{-n_1 t}\sin p_1 t \quad d_2 = e^{n_1 t}\cos p_1 t \tag{14}$$

$$F_1 = \int_0^t q(t)\cdot e^{-n_1 t}\cos p_1^* \tau\, d\tau \tag{15}$$

The nonlinear component of the solution is determined from the integral equation

$$y_{nl} = \frac{1}{mp_1^*}\int_0^t \left(1 + 2\nu\frac{d}{dt}\right) k_2(y - y_0)e^{-n_1(1-\tau)}\sin p_1^*(t-\tau)d\tau \tag{16}$$

where t_0 – time to turn on additional support.

Following (13) – (15) we can write down:

$$y_{nl} = \frac{k_2}{mp_1^*}\int_0^t (y - y_0)e^{-n_1(1-\tau)}\,(\sin p_1^* t\, \cos p_1^* \tau - \cos p_1^* t\, \sin p_1^* \tau)d\tau =$$

$$\frac{k_2}{mp_1^*}[d_1(t)F_2(t_0, t) - d_2(t)F_1(t_0, t)] \tag{17}$$

where $F_2(t_0, t) = \int_{t_0}^t (y - y_0)\cdot e^{n_1 \tau}\cos p_1^* \tau d\tau$;

$$F_1(t_0, t) = \int_{t_0}^t (y - y_0)\cdot e^{n_1 \tau}\sin p_t^* \tau d\tau \tag{18}$$

The total displacement is calculated using formula (13), which is solved step by time with iterations at each step.

As an example, a system with the following parameters is considered: system mass – 10 tons; rigidity of the original system – 4000 kN/m; rigidity of additional connection – 2000 kN/m; amplitude of disturbing force – 350 kN; disturbing force frequency – 78 rad/s; z0= 0.015 m;

We use the above approach to calculate a system with a viscous friction damper (**Figure 1b**).

The equation of motion of a system with one degree of freedom with a viscous friction damper takes the form (in particular during start-up and in operating mode):

$$m\ddot{y} + \left(1 + 2\mu\frac{d}{dt}\right)ky + h_k\left|\frac{dy}{dt}\right| = q(t). \tag{19}$$

where h_k – drag coefficient (**TsNIISK named after V. A. Kucherenko, 1986**)

When $y_0<y<y_1$, where $y_0(t_0)$ and $y_1(t_1)$ are the boundaries of the zones for switching on and off the damper. In the rest of the zone h_k is equal to zero.

Taking into account the above conditions, using the Duhamel integral, the solution to equation (19) can also be represented as a sum of two solutions (formulas 12 - 13). Let us write down the solution to the nonlinear part:

$$y_{нл} = \frac{1}{p_1^* m}\int_{t_0}^t h_k \frac{dy}{d\tau}V_1(p_1^*, t-\tau)d\tau - \text{при } t \le t_1 \tag{20}$$

Integrating (20) by parts, we write (in the interval $t_0<t<t_1$)

$$y_{nl} = \frac{1}{p_1^* m}\left\{h_k y V_1(p_1^*, t)\int_{t_0}^t - \int_{t_0}^t h_k y V_2(p_1^*, t-\tau)d\tau\right\}; \tag{21}$$

where $V_1(p_1^*, t) = e^{-n_1 t}\sin p_1 t$- pulse transient function, $V_2 = \frac{d}{d\tau}V_1$,

After such transformations it takes the form:

$$y_{nl}(t) = \frac{h_k}{p_1^* m}\{[n_1 d_1(t) - p_1 d_2(t)]F_2(\tau) + [n_1 d_1(t) - p_1 d_2(t)]F_1(\tau)\}. \quad (22)$$

4 RESULTS AND DISCUSSION

Without doing numerical solutions, we note that the indicated algorithm corresponds to the start-up solution, and the numerical algorithm is similar to the algorithm used in linear and nonlinear vibration isolation problems. In operating mode, the upper limit of the integral should be set to t=t1. Movements of systems in stop mode are determined by the algorithm given above in the interval t3-t, where t3−time when the damper is activated.

This study examines options for nonlinear vibration isolation systems, including systems with a limiter and viscous friction damper. Algorithms for calculating these systems both with one degree of freedom under the action of a harmonic load in operational mode and in transient modes are also presented. The given calculation example shows that the use of a system with a limiter allows one to reduce maximum movements in transient modes by 30-35%.

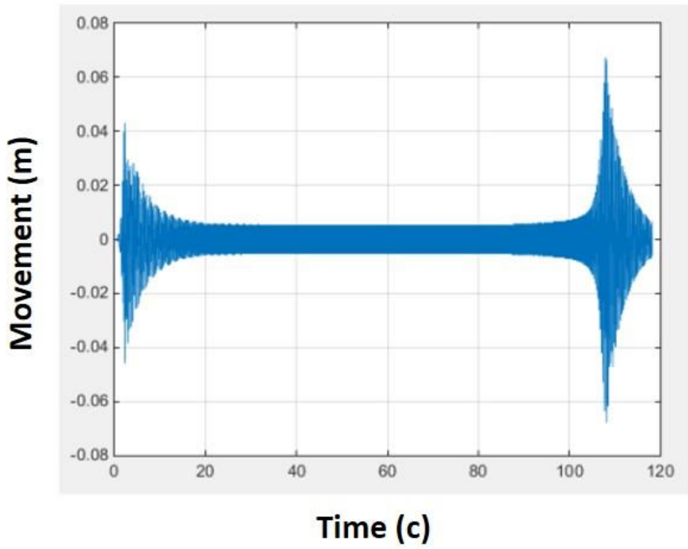

Figure 2 – Traditional system movement graphics (author's material).

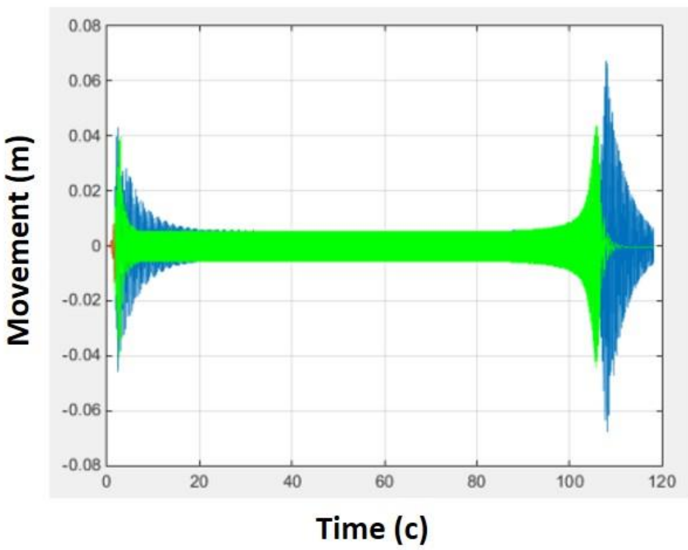

Figure 3 – Movements in the traditional system (blue) and with additional support (green) (author's material).

An assessment was made of two options for vibration protection systems; traditional system and system with additional support.

The maximum displacement values with the traditional vibration isolation system were 0.0598; 0.0893 and 0.0061 m during start-up, stop and operation, respectively. For option 2 - systems with additional support, the greatest effect when turning on support during startup was at k2 = 1500 kN/m. Other stiffness values gave unfavorable results and displacements increased. When stopped, the optimal value of stiffness k_2 was 2500 kN/m. In operational mode, of course, additional connections did not affect the levels of movement.

The maximum reactions transmitted to the supporting structures are also determined. Option 1 was taken as a control for comparison with other response values for other vibration isolation options. For option 2, the optimal stiffness value was k_2 = 2500 kN/m, at which the reaction value increased by 47% when starting and decreased by 48% when stopping.

Table 1
Amplitudes of movements in systems with 4 vibration isolation options [author's material]

Vibration isolation option	Constant parameter	Changed parameter	Amplitude of movement (m)		
			at start-up	when stopping	during operation
Traditional system	m_1 = 10t; k_1 = 4200kN/m	-	0,0598	0,0893	0,0061
System with an additional support	m1 = 10t; k_1 = 4200kN/m	k_2 (kN/m) 500	0,0629	0,0856	0,0061
		1000	0.0644	0.0796	0.0061
		1500	0.0591	0.0696	0.0061
		2000	0.0728	0.0514	0.0061
		2500	0.0704	0.0289	0.61

5 CONCLUSIONS

In the present study, the effect of the duration of transient regimes on displacement amplitudes was studied **(Table 1)**. Increasing the time intervals in transient modes gave different effects. An increase in the time interval by 33% led to an increase in displacement by 9-17% for all vibration isolation options. When stopping, an increase in time by 67% resulted in an increase in the amplitude of movements by 20%-42%. In operating mode, the oscillation amplitudes did not change, which indicates stability of operation during operation and possible malfunctions during startup or shutdown of the structure.

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