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Use of the Belleville spring package in the vibration protection mechanism of the operator’s seat

Use of the Belleville spring package in the vibration protection mechanism of the operator's seat

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Abstract. Reducing the vibration effects on the operator of a construction, road and other machines is a relevant objective, since its accomplishing improves the efficiency and accuracy of the work performed by machines and reduces the risk of operators' occupational diseases. Vibration protection systems with the effect of quasi-zero stiffness are promising. To produce this effect, Belleville springs packages can be used. They are a way to create a simple, reliable and inexpensive passive mechanism of the vibration protection system of the operator's seat. The influence of the static characteristic center slope of two Belleville springs packages on the maximum acceleration of the load under the sinusoidal kinematic excitation of the seat base movements is investigated. The static characteristic center slope of the spring packages was characterized by the coefficient that affects the type of the static characteristic according to the known analytical expressions for the separate spring force. To obtain the chair acceleration values in a fixed coordinate system, we used a simulation mathematical model developed in the Simscape Multibody package for modeling mechanical systems of the MATLAB mathematical system. The model used blocks of a fixed coordinate system, a sliding joint and a rigid body forming a chain. The maximum acceleration of the chair in a fixed coordinate system was defined during a predetermined period of time at the end of the transition process under the steady state oscillations. When conducting the computational experiment, the specified oscillations amplitude of the chair base and the angular frequency of these oscillations varied. For a proper comparison of the experimental results in the form of graphical dependencies of the maximum acceleration on the amplitude and frequency, the ranges of acceptable and unacceptable arguments combinations were selected for the two spring packages. The combinations of amplitude and frequency were considered unacceptable when the maximum deformation of the spring package went beyond the static characteristic maximum. For the two packages of springs, the acceptable regions were different, therefore their intersection used for the comparative analysis, was obtained. The springs package with a relatively high stiffness in the center of the static characteristic is found out to generally provide better vibration protection of the chair in the entire acceptable range of arguments. Simultaneously, a package with lower stiffness in the center of the characteristic dampens vibrations better in the small amplitudes area, i.e. when operating in the quasi-zero stiffness area.

Key-words: vibrations, vibration protection, acceleration, Belleville spring, Belleville spring package

1. Introduction

The characteristics of spring elements play an important role in determining the properties of the vibration isolation systems used in many industries, in particular, in construction, road and handling machines [1]. The most urgent problem of these machines is the protection of the operator from the effects of vibrations that occur during the machine operation [2]. The analysis of academic publications shows that operators of all major types of soil-moving machines, such as bulldozers [3], graders [4], excavators [5], road cleaning machines [6], front loaders [7], scrapers [8], are exposed to significant vibrations. Hand tools for breaking soil, rocks, or asphalt-concrete pavement also transmit significant vibrations to the operator [9].

Vibration protection systems of the cab [11] and operator's chairs [12] being the vibration supports of various designs [13] are used to reduce vibrations that cause occupational diseases of machine operators [10]. Active [14] and passive [15] vibration supports are distinguished based on utilizing the external energy. The widespread use of active vibration isolation systems for operator's seat is limited by the high cost and design complexity. Passive systems being inferior to the active ones in versatility and efficiency [16] have the advantages, such as the simplicity of design, lower cost of production and maintenance, improved reliability and durability [17].

Vibration protection systems capable of implementing the quasi-zero stiffness mode [18] provide a wider frequency range of vibration protection compared to the linear vibration isolators and more effective vibration protection against low-frequency vibrations [19].

Passive vibration protection systems based on the Belleville springs [20], which are also able to provide a quasi-zero stiffness mode [21] in the specified movements range [22] are reliable, simple and durable. To expand the range of quasi-zero stiffness, the number of springs is increased and spacing washers or rings are placed between the springs [22].

A disadvantage of most such systems is a certain value of the force corresponding to the quasi-zero stiffness, and consequently if the human operator's weight differs from the average values, the static equilibrium position of the system is shifted from the center of the quasi-zero stiffness range. Thus, a useful feature of the vibration protection system is its ability to adapt to changing operational conditions [23].

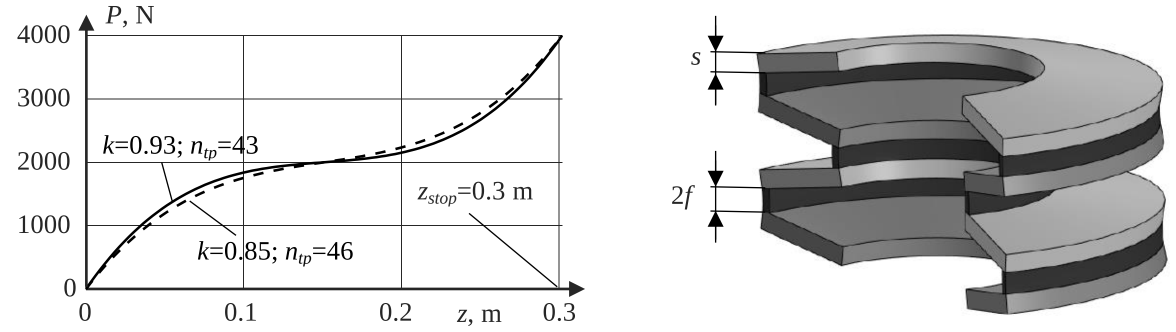
If we consider the static power characteristic of passive vibration protection systems with a quasi-zero stiffness section, then most of these systems use a section close to the horizontal one [24].

However, a minor change in the geometric dimensions of the Belleville springs can provide the static power characteristic center slope of a separate spring and entire Belleville springs package [22]. A characteristic with a slope is more universal than a characteristic with a horizontal section in the center, since it changes the weight of the human operator within the narrow limits. Furthermore, the study investigating the effect of the static power characteristic type of the springs package on the dynamic indicators of the human operator's chair oscillations is of practical interest.

In this regard, the paper considers the influence of the coefficient determining the general form of the static power characteristic of the spring package on the maximum acceleration of the human operator's chair at the specified sinusoidal vibrations of the seat base.

2. Problem statement

The dynamic system is represented in the problem by a single concentrated weight of the chair with a human operator m, which is connected to the movable base by means of the antivibration suspension having a specified static power characteristic $P(z)$ (figure 1a). The characteristic is provided by a package of Belleville springs separated from each other by spacing washers for a sufficient value of the axial movement. The entire antivibration suspension is generally characterized by a constant damping coefficient b of the vertical movement z. The weight of the operator's seat payload is balanced in the midstroke of the spring package, when moving $z=z_{stop}/2$, where z_{stop} is the full stroke of the package. The mechanism is assumed to stop uncushioned by the stoppers at $z<0$ and $z>z_{stop}$.



(a) (b)

Figure 1. Examples of static force characteristics with a lower (—) and higher (- - -) slope in the center of the quasi-zero stiffness range (a) which is created by two different Belleville spring packages with spacing washers (b).

It is necessary to compare the vibration protection mechanisms with a different slope of the static force characteristic of the Belleville springs package in the center of the quasi-zero stiffness range according to the dynamic criterion of the maximum acceleration of the chair in the fixed coordinate system $a_{0\max}$ which is achieved during the transition process.

The separate springs and package as a whole are characterized by the following parameters that have the constant values: D is the outer diameter of a separate Belleville spring; d is the inner diameter of the spring; E is the elastic modulus of the spring material; $P_{nom} = m \cdot g$ is the static force nominal value to be produced by the vibration protection mechanism in the center of the quasi-zero stiffness section; n_{tp} is the number of Belleville springs stacked in series in the package; s is the wall thickness of a separate Belleville spring; f is the height of a separate unloaded spring; k is the coefficient influencing the type of the static characteristic. Zero slope of the characteristic center (quasi-zero stiffness) corresponds to the value of k=1. The stiffness increase in the center of the characteristic corresponds to decreasing the value of k in the range$k$<1.

The current values of the separate loaded spring force and height are indicated by P and λ, respectively.

The studies of the vibration protection mechanisms on a simulation model involve setting the parameters of the model and transition process. These parameters include the following: the simulation final time T_{kon}, the measurement time of the load acceleration at the end of the simulation process T_{izm}, the time step of the simulation results output dt, the weight of the human operator's chair m and the damping coefficient b.

Under the sinusoidal specified base displacements, these displacements are characterized by the following parameters: A_{mp} is the amplitude of the specified base vibrations; F_{req} is the angular frequency of the base vibrations.

To compare two spring packages, it is necessary to very A_{mp} and F_{req} within the certain limits and define $a_{0\max}$ for each combination of the variable parameters.

3. Theory

It was assumed that the spacing washers height of the Belleville springs package of the vibration protection mechanism (figure 1b) is selected in such a way that the full stroke of the individual Belleville spring up to the stop would amount to $2f$. The spacing washers height should also be $2f$ [22].

The static force characteristic of a separate Belleville spring is known to be described by the following nonlinear dependence [21, 22]:

$$P=\frac{8\cdot\lambda\cdot\pi\cdot s\cdot E\cdot\left(\frac{s^2\cdot\ln\left(\frac{D}{d}\right)}{12}-\left(\frac{d+D}{2\cdot d-2\cdot D}+\frac{1}{\ln\left(\frac{D}{d}\right)}\right)\cdot(f-\lambda)\cdot\left(f-\frac{\lambda}{2}\right)\right)}{(D-d)^2}. \tag{1}$$

If antivibration mechanism of the chair contains Belleville springs of the same size $z=n_{tp}\cdot\lambda$, then the equation of the entire mechanism static power characteristic according to (1), will have the following form:

$$P=\frac{8\cdot z\cdot\pi\cdot s\cdot E\cdot\left(\frac{s^2\cdot\ln\left(\frac{D}{d}\right)}{12}-\left(\frac{d+D}{2\cdot d-2\cdot D}+\frac{1}{\ln\left(\frac{D}{d}\right)}\right)\cdot\left(f-\frac{z}{n_{tp}}\right)\cdot\left(f-\frac{z}{2\cdot n_{tp}}\right)\right)}{n_{tp}\cdot(D-d)^2}. \tag{2}$$

Examples of the two static characteristics obtained from (2) are shown in figure 1a. It is known that The separate Belleville spring dimensions *s* and *f* are known to be related both to each other and to the coefficient *k*, on which the stiffness and the slope angle in the center of the spring static characteristic depend, by the ratio [21, 22]:

$$f=\sqrt{2}\cdot k\cdot s. \tag{3}$$

The range of values $k<1$ corresponds to the positive slope angle of the curve (figure 1a) [22]. According to (1), a single force value *P* will correspond to the given constant values of the spring diameters *D* and *d*, the wall thickness *s* and the deformation λ. Provided that P_{nom} is reached in the middle of the characteristic having a symmetric shape, the analytical expression *s* obtained from (1) and (3), will have the following form:

$$s=\left(\sqrt{D-d}\right)\cdot\sqrt[4]{\frac{\frac{3\cdot\sqrt{2}}{4\cdot k}\cdot P_{nom}}{E\cdot\pi\cdot\ln\left(\frac{D}{d}\right)}}. \tag{4}$$

To construct the static force characteristic *P*(*z*) of the Belleville springs package from the given values of *D*, *d*, *E*, *k*, P_{nom}, n_{tp}, the equations (4), (3), (2) were applied sequentially in the order given.
When applying the expression (2), the vertical coordinate *z* of the chair lift varied with a specified step *dz* within the range of $z\in\left[0;2\cdot f\cdot n_{tp}\right]$.

Following the static power characteristic construction, the antivibration suspended seat oscillations were simulated using a dynamic simulation model. The simulation model was developed in the mechanical systems modeling package Simscape Multibody of the mathematical system MATLAB [25].

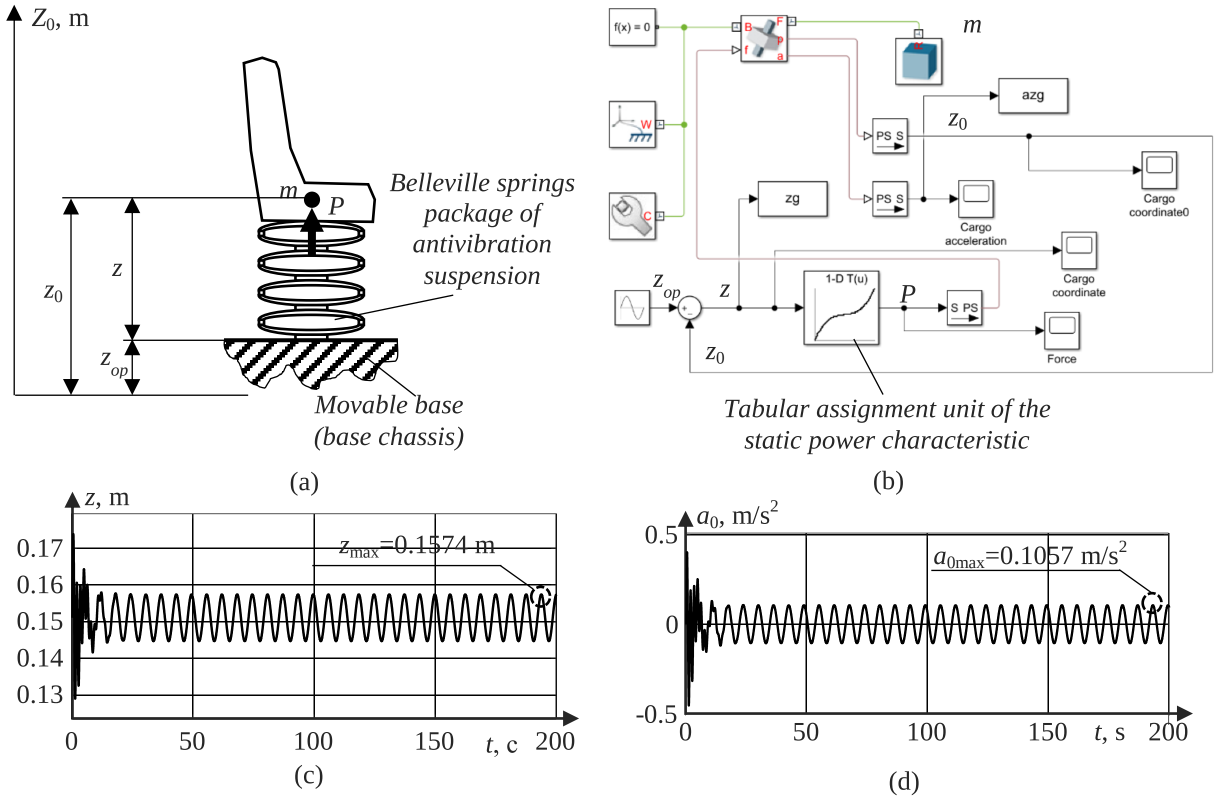


Figure 2. Calculation model (a), simulation model (b) and examples of time dependences of movements (c) and accelerations (d) of the antivibration suspended chair with a movable base.

According to the calculation scheme of the antivibration suspended chair with a movable base (figure 2a), a simulation mathematical model (figure 2b), which used the main blocks of the Simscape Multibody package, such as *World Frame* to set the fixed coordinate system, *Prismatic Joint* to describe the translational degree of freedom of the vibration protection mechanism, *Solid* to set the human operators' chair weight, was developed. The model is based on three blocks chain *World Frame-Prismatic Joint-Solid* connected by a physical bidirectional signal Simscape Multibody. The joint unit *Prismatic Joint* has an additional input supplied with the current force value $P(z(t))$ depending on the current internal deformation of the spring package $z(t)$ and two additional outputs with the current values of the chair vertical coordinate in the fixed coordinate system $z_0(t)$ and acceleration in the fixed coordinate system $a_0(t)$. In the settings of this unit, the constant damping coefficient of the vibration protection mechanism b is set.

Moreover, *Simulink-PS Converter* blocks for converting the "digital" unidirectional Simulink signal into a "physical" bidirectional Simscape Multibody signal and *PS-Simulink Converter* for reverse conversion were also used as auxiliary units in the model. To specify the base movement in the fixed coordinate system $z_{op}(t)$, the model used the Simulink *Sine Wave* signal source unit, which generates a sinusoidal effect. To store the simulation results in the memory of the MATLAB workspace for their subsequent processing, the Simulink signal recipients blocks *To Workspace* were used.

The accepted criterion value of the chair maximum acceleration in the fixed coordinate system $a_{0\max}$ was calculated after the simulation of a separate transient process of the forced oscillations was completed for the time sequence of the chair accelerations $a_0\ (t)$ in the finite time interval $t \in \left[\left(T_{kon} - T_{izm}\right); T_{kon}\right]$. An assumption was made that in the specified time interval at (T_{kon}>100 s; $T_{izm} < \dfrac{T_{kon}}{2}$), the forced oscillations constantly have a steady character, which was confirmed by the results of the computational experiments.

4. Experimental results

When comparing the vibration protection mechanisms with a different static force characteristic slope of the Belleville springs package in the center of the quasi-zero stiffness range, selecting the number of strings n_{tp} in the two compared packages is difficult. It is necessary that there should be the same maximum deformation of the two packages being compared, which according to the accepted assumption is limited by the value $2 \cdot f \cdot n_{tp}$ above which the contact with the bump stops occurs. The size value f of the individual spring, in turn, depends on the value of the coefficient k. Given that n_{tp} accepts only integer values, it is necessary to simultaneously select the values of two combinations of [k; n_{tp}] for two spring packages, so that the displacement value $2 \cdot f \cdot n_{tp}$ for the two specified combinations to be the same.

For example, two packages of Belleville springs with different parameter values (but with the same maximum displacement) were compared: $\left[k = 0.93; n_{tp} = 43\right]$ - package No. 1; $\left[k = 0.85; n_{tp} = 46\right]$ - package No. 2. The given combinations were chosen so that to provide almost the same maximum deformation values of two packages: $2 \cdot f \cdot n_{tp} \approx 0.3\, m$. At the maximum deformation, the static power was also the same: 4000 N. The stiffness at the midpoint of the characteristic accepted the values of 1912 and 3777 N/m for packages No. 1 and No. 2, respectively, i.e. it increased almost twice.The other parameters took the same values for the two packages: D=0.2 m; d=0.1 m $E = 2.06 \cdot 10^{11}$; MPa (spring material is steel); P_{nom}=2000 N (the operator's chair weight is $m = \frac{P_{nom}}{g} = \frac{2000}{9.81} = 203.874$ kg).

The amplitude of the specified base vibrations during the computational experiment varied withing the range of $A_{mp} \in \left[0{,}005; 0{,}15\right]$ with a step of 0.0025 m. The angular frequency of the base oscillations varied within the range of $F_{req} \in \left[0{,}5; 10\right]$ with a step of 0.125 rad/s.

The other parameters constant values of the dynamic process are given below. Simulation time is T_{kon}=200 s. Accelerations measurement time at the end of the process is T_{izm}=50 s. Time step of the results output is dt=0.01 s. Damping coefficient is b=100 N/(m/s).

Figure 2 c, d shows the time dependences of the internal displacement and absolute acceleration of the antivibration suspended chair with a movable base for package No. 2, at the amplitude and frequency of A_{mp}=0.1 m; F_{req}=1 rad/s.

The experimental results in the form of the graphs of the chair maximum acceleration $a_{0\max}$ dependence on the amplitude A_{mp}, and frequency F_{req} of the specified base oscillations, are presented in figure 3 a, b.

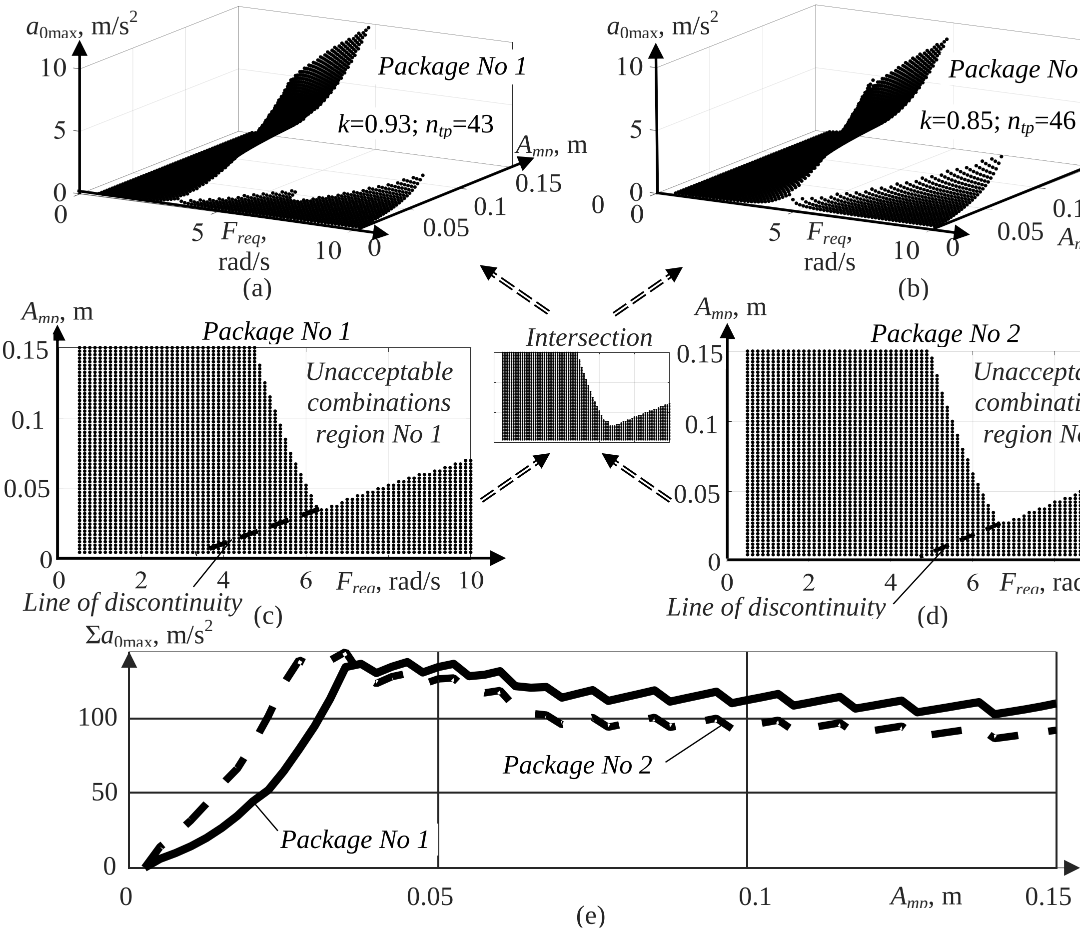


Figure 3. The results of a computational experiment on modeling the antivibration suspended chair oscillations under sinusoidal excitation: (a) is the maximum accelerations, package No. 1; (b) is the maximum accelerations, package No. 2; (c) is the acceptable combinations of arguments, package No. 1; (d) is the acceptable combinations of arguments, package No. 2; (e) the sum of the maximum accelerations for all the frequencies considered depending on the amplitude

For certain combinations of the variable parameters (arguments) values A_{mp} and F_{req}, the spring package movements relative to the base exceeded the limits of the considered static characteristic (z<0; z>$z_{\max}$), which according to the accepted assumptions corresponded to the uncushioned stop. Such combinations have not been considered. Figure 3 c, d shows the arguments values combinations (marked with dots, the values changed simultaneously with the steps indicated above), at which mechanism does not move uncushioned for spring packages No 1 and No 2, respectively. Thus, the accepted combinations of arguments values are given when the vertical movements of the suspension z are within the static power characteristic (figure 1a). Figure 3d shows the sum of the maximum accelerations over all the frequencies considered depending on the amplitude, taking into account the acceptable combinations of arguments (maximum accelerations of both packages in the unacceptable combinations for either of the two packages, were assumed to be zero).

5. Results discussion

The ranges of argument values acceptable combinations (figure 3c, d) for the two spring packages under consideration are somewhat different from each other. The set representing the intersection of two sets (i.e., the acceptable regions No 1 and No 2) was used to compare the obtained graphs for two spring packages No 1 and No 2 (figure 3a, b). Therefore, for proper comparison of two spring packages, the maximum acceleration function values shown in figure 3a, b were simultaneously calculated only for the acceptable arguments combinations of both packages.

When comparing two packages, the criterion of a higher hierarchical level compared to the maximum acceleration was used. The mentioned criterion includes the maximum acceleration as a lower-level criterion. This is the sum S of the chair maximum accelerations on the set of argument values visualized as the intersection described above and presented by the following formula:

$$S = \sum_{i=1}^{i_{\max}} \sum_{j=1}^{j_{\max}} \left(\tilde{a}_{0max}\right)_{ij} , \tag{5}$$

where

$$\left(\tilde{a}_{0max}\right)_{ij} = \begin{cases} \left(a_{0max}\right)_{ij} & at \left(\underset{(Package\ No1)}{\left(z_{max}\right)_{ij} \le z_{stop}} \right) \wedge \left(\underset{(Package\ No2)}{\left(z_{max}\right)_{ij} \le z_{stop}} \right); \\ 0 & \text{in other cases.} \end{cases} \tag{6}$$

The indexes i and j in equations (5) and (6) corresponded to the arguments A_{mp} and F_{req}. These equations assume all combinations of argument values to be in equilibrium with each other, i.e. they are equally probable during use of the vibration protection system.

The values of the criterion S were obtained as follows: S=6057 for package No. 1; S=5752 for package No 2. This means that an increase in the stiffness and slope of the middle section of the package static power characteristic (by 9% in terms of the coefficient k, by 2 times in terms of stiffness) causes an insignificant (by 5 %) decrease in the sum of accelerations in the total acceptable range of arguments.

The values of $\Sigma a_{0\max}$ shown in figure 3d were calculated according to the following dependence:

$$\left(\Sigma a_{0max}\right)_i = \sum_{j=1}^{j_{\max}} \left(\tilde{a}_{0max}\right)_{ij} .$$

Furthermore, the values of the number of points in the proper acceptable ranges of displacements (at the values of the argument steps given above): N=3061 for the package No. 1; N=3025 for the package No 2 have been calculated for two spring packages.

Discontinuity lines of the function surfaces presented in the graphs are shown in the top views (figure 3c, d). They are producing the lines that separate the acceptable and unacceptable ranges of the displacement arguments.

6. Conclusions

The study of the oscillations of the antivibration suspended chair with a movable base with a single degree of freedom was carried out on a simulation mathematical model developed in the Simscape Multibody package of the MATLAB system. The mechanical system had one translational degree of freedom. Static power characteristics of the vibration protection suspension with quasi-zero stiffness effect have been obtained from the known and supplemented dependences for Belleville springs. The parameters and number of springs from the two compared packages have been selected to provide a different slope of the middle section of the characteristic, at the same maximum displacements and force values at the center of the characteristic and at the point of the maximum displacements.

Kinematic excitation of the seat base vibrations was set using a sinusoidal function. The amplitude and frequency influence of the seat base vibrations on the maximum acceleration in a fixed coordinate system was studied for two springs packages taking into account the restrictions on the maximum deformation of the suspension.

It is found out that according to the criterion of the maximum load accelerations sum, the springs package with a high slope of the center of the static characteristic (by the coefficient k, a change is 9 %) provides a better (by 5% in the considered range of amplitudes and frequencies) vibration protection of the chair compared to the springs package with a lower slope of the characteristic.

However, the surface area of the maximum acceleration function in the high frequency region (from 6 to 10 rad/s) is characterized by high acceleration values, when using the springs package with a high slope of the center of the static characteristic.

Therefore, providing better vibration protection over the entire range of amplitude and frequency changes, the antivibration suspension with a relatively high slope of the static characteristic is inferior to the suspension with a lower slope when operating in the increased frequency range.

When analyzing the graphs of the maximum accelerations sums for all the considered frequencies depending on the amplitude (figure 3d), it is evident that the springs package which middle section is close to the horizontal, i.e., which is close to having quasi-zero stiffness, more effectively dampens vibrations in the region of small amplitudes (up to 0.03 m). Nevertheless, in working on the end sections of the characteristic, i.e., at the high amplitudes, the package with a high value of the coefficient k is more effective.

Besides, the antivibration suspension with greater stiffness in the center of the static characteristic has a slightly lower (by 1.2 %) range of amplitudes and frequencies which is permissible for maximum displacements than the antivibration suspension with a lower slope.

The results obtained may be of interest to the researchers and developers of the vibration protection systems of the construction, road and other machines operator's seats.

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