Hindawi
Shock and Vibration
Volume 2018, Article ID 3732746, 10 pages
https://doi.org/10.1155/2018/3732746

Research Article

Global Sensitivity Analysis of the Vibration Reduction System with Seated Human Body

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Received 2 March 2018; Accepted 5 July 2018; Published 26 July 2018

Academic Editor: Stefano Marchesiello

The paper deals with the global sensitivity analysis for the purpose of shaping the vibroisolation properties of suspension systems under strictly defined operating conditions. The variance-based method is used to evaluate an influence of nonlinear force characteristics on the system dynamics. The proposed sensitivity indices provide the basis for determining the effect of key design parameters on the vibration isolation performance. The vibration transmissibility behaviour of an exemplary seat suspension system is discussed in order to illustrate the developed methodology.

1. Introduction

A selection process of the system dynamic properties is difficult due to a presence of the conflicted vibroisolation criteria [1]. In the case of an ideal system isolation, vibration amplitude q_{1x} of the isolated body is equal to zero (Figure 1(a)) and the human vibrations are perfectly dissipated by a suspension system. Unfortunately, in such a situation, the relative displacements $q_{1x} - q_{sx}$ of the suspension system are considerably large, and they are the same as the input vibration q_{sx}. In an extreme case, the recommended deflection limitation requires the suspension travel equal to zero (Figure 1(b)). Then, the isolated body oscillates with the vibration amplitude q_{1x} equal to the input vibration q_{sx} and endangers driver's health [2].

There are a lot of design parameters having an influence on the system dynamics [3, 4]. Usually, not very effective design process is related to a significant number of the decision variables that affect the vibroisolation properties of the suspension system [5, 6]. Global sensitivity analysis is a tool frequently used to explore how the variations in the model outputs can be attributed to variations of the model inputs [7]. Such an analysis can be applied in order to investigate the relative influence of model parameters on the system performance and also for the purpose of evaluating the dominant controls of a system (model). The process of determining the impact of input factors on system evaluation criteria is proposed further in this paper.

The fundamental method of evaluating the effectiveness of vibration reduction system is to perform an experiment in the laboratory [8, 9]. Unfortunately, the duration of the test and its high cost are the important aspects when the test must be performed repeatedly for different design parameters of the system. Therefore, the authors of the following paper recommend to carry out a simulation experiment based on a mathematical model of the vibration isolation system. In traditional experimental designs, the effect of input factors is customarily estimated over two levels, that is, at the lower and upper bounds [10]. Unlike the traditional design's analysis, the variance-based method takes into account the whole input space using the Monte Carlo method [11]. Thus, the variance-based method can measure the nonlinear effects of individual input factors and allows to estimate the effect of interactions in nonadditive systems [12].

In this paper, an original methodology of shaping the vibroisolation properties is presented for the purpose of improving the suspension dynamics. The proposed iterative

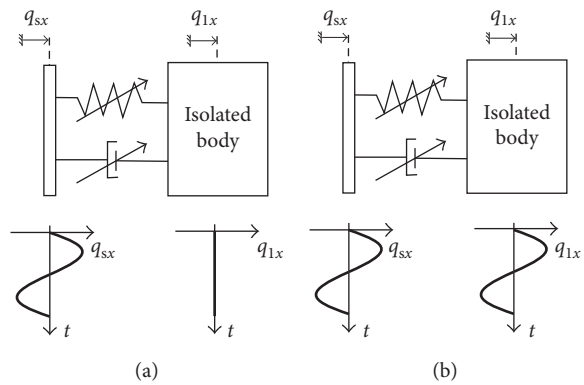


Figure 1: Conflicted requirements for the vibroisolating properties: ideal reduction of the body vibration (a); ideal limitation of the suspension travel (b).

procedure supports analysing the nonlinear system characteristics that yield a desirable vibration response in view of the conflicted requirements for modern vibration reduction systems.

2. Global Sensitivity Analysis

2.1. Sample Matrix. The sample matrix $\mathbf{X}_{\mathrm{AB}}$ has to be generated with respect to the probability distributions of the input factors. Using a uniform random number generator (rand), the initial starting points for sensitivity analysis may be expressed as follows:

$$\mathbf{X}_{\mathrm{AB}} = \mathrm{rand}\,(N, 2\mathrm{p}), \tag{1}$$

where N is the number of rows in the matrix that indicates the number of simulation runs required. The number of columns is the doubled number $2p$ of input factors to be analysed. The resulting sample matrix takes the following form:

$$\mathbf{X}_{\mathrm{AB}} = \left[\begin{array}{cccc|cccc} x_{\mathrm{A}11} & x_{\mathrm{A}12} & \cdots & x_{\mathrm{A}1p} & x_{\mathrm{B}11} & x_{\mathrm{B}12} & \cdots & x_{\mathrm{B}1p} \\ x_{\mathrm{A}21} & x_{\mathrm{A}22} & \cdots & x_{\mathrm{A}2p} & x_{\mathrm{B}1p} & x_{\mathrm{B}22} & \cdots & x_{\mathrm{B}2p} \\ \cdots & \cdots & \ddots & \cdots & \cdots & \cdots & \ddots & \cdots \\ \underbrace{x_{\mathrm{A}N1} \quad x_{\mathrm{A}N2} \quad \cdots \quad x_{\mathrm{A}Np}}_{\mathbf{X}_{\mathrm{A}}} & & & & \underbrace{x_{\mathrm{B}N1} \quad x_{\mathrm{B}N2} \quad \cdots \quad x_{\mathrm{B}Np}}_{\mathbf{X}_{\mathrm{B}}} & & & \end{array}\right],$$
$$i = 1, \ldots, N,\ j = 1, \ldots, p, \tag{2}$$

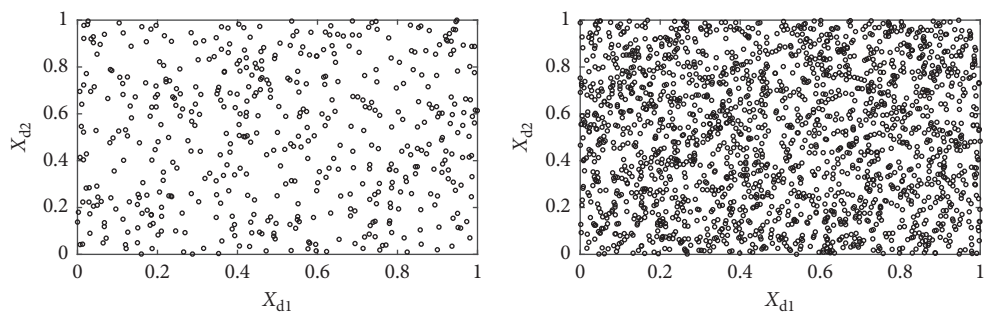
where $\mathbf{X}_{\mathrm{A}}$ and $\mathbf{X}_{\mathrm{B}}$ are the sample matrices which consist of two independent starting points in the p-dimensional unit hypercube.

The higher number of N samples contributes to a better estimation of sensitivity indices [13]. Two-dimensional representation of uniformly distributed random starting points at different number of samples is presented in Figure 2.

2.2. Variance-Based Method. The variance-based method uses randomly generated starting points in order to calculate the output variances decomposed into fractions that can be attributed to the system inputs [14]. The method is based on the combination of two independent sampling matrices $\mathbf{X}_{\mathrm{A}}$ and $\mathbf{X}_{\mathrm{B}}$ (2) in such a way that the recombined matrices $\mathbf{X}_{\mathrm{C}1}, \mathbf{X}_{\mathrm{C}2}, \ldots, \mathbf{X}_{\mathrm{C}j}$ contain the elements of matrix $\mathbf{X}_{\mathrm{A}}$ except the jth column which is taken from matrix $\mathbf{X}_{\mathrm{B}}$. The resulting matrices are determined as follows:

$$\mathbf{X}_{\mathrm{C}1} = \begin{bmatrix} x_{\mathrm{B}11} & x_{\mathrm{A}12} & \cdots & x_{\mathrm{A}1p} \\ x_{\mathrm{B}21} & x_{\mathrm{A}22} & \cdots & x_{\mathrm{A}2p} \\ \cdots & \cdots & \ddots & \cdots \\ x_{\mathrm{B}N1} & x_{\mathrm{A}N2} & \cdots & x_{\mathrm{A}Np} \end{bmatrix}, \quad i = 1, \ldots, N,\ j = 1, \ldots, p,$$
$$\mathbf{X}_{\mathrm{C}2} = \begin{bmatrix} x_{\mathrm{A}11} & x_{\mathrm{B}12} & \cdots & x_{\mathrm{A}1p} \\ x_{\mathrm{A}21} & x_{\mathrm{B}22} & \cdots & x_{\mathrm{A}2p} \\ \cdots & \cdots & \ddots & \cdots \\ x_{\mathrm{A}N1} & x_{\mathrm{B}N2} & \cdots & x_{\mathrm{A}Np} \end{bmatrix}, \quad i = 1, \ldots, N,\ j = 1, \ldots, p,$$
$$\vdots$$
$$\mathbf{X}_{\mathrm{C}j} = \begin{bmatrix} x_{\mathrm{A}11} & x_{\mathrm{A}12} & \cdots & x_{\mathrm{B}1p} \\ x_{\mathrm{A}21} & x_{\mathrm{A}22} & \cdots & x_{\mathrm{B}2p} \\ \cdots & \cdots & \ddots & \cdots \\ x_{\mathrm{A}N1} & x_{\mathrm{A}N2} & \cdots & x_{\mathrm{B}Np} \end{bmatrix}, \quad i = 1, \ldots, N,\ j = 1, \ldots, p. \tag{3}$$

The model outputs should be calculated for each design point in the $\mathbf{X}_{\mathrm{A}}$, $\mathbf{X}_{\mathrm{B}}$, and $\mathbf{X}_{\mathrm{C}1}, \mathbf{X}_{\mathrm{C}2}, \ldots, \mathbf{X}_{\mathrm{C}j}$ matrices; therefore, a total of $N(2+p)$ model evaluations are required. The corresponding model outputs should be saved by using the following vectors:



(a) (b)

Figure 2: Two-dimensional representation of uniformly distributed random starting points at different numbers of simulation runs: $N = 500$ (a) and $N = 2000$ (b).

$$f(\mathbf{X}_A) = \begin{bmatrix} f(\mathbf{X}_A)_1 \\ f(\mathbf{X}_A)_2 \\ \cdots \\ f(\mathbf{X}_A)_N \end{bmatrix},$$
$$f(\mathbf{X}_B) = \begin{bmatrix} f(\mathbf{X}_B)_1 \\ f(\mathbf{X}_B)_2 \\ \cdots \\ f(\mathbf{X}_B)_N \end{bmatrix},$$
$$f(\mathbf{X}_{C1}) = \begin{bmatrix} f(\mathbf{X}_{C1})_1 \\ f(\mathbf{X}_{C1})_2 \\ \cdots \\ f(\mathbf{X}_{C1})_N \end{bmatrix}, \tag{4}$$
$$f(\mathbf{X}_{C2}) = \begin{bmatrix} f(\mathbf{X}_{C2})_1 \\ f(\mathbf{X}_{C2})_2 \\ \cdots \\ f(\mathbf{X}_{C2})_N \end{bmatrix},$$
$$f(\mathbf{X}_{Cj}) = \begin{bmatrix} f(\mathbf{X}_{Cp})_1 \\ f(\mathbf{X}_{Cp})_2 \\ \cdots \\ f(\mathbf{X}_{Cp})_N \end{bmatrix}, \quad i = 1, \ldots, N, j = 1, \ldots, p.$$

The output vectors: $f(\mathbf{X}_A)$; $f(\mathbf{X}_B)$; and $f(\mathbf{X}_{C1}), f(\mathbf{X}_{C2}), \ldots, f(\mathbf{X}_{Cj})$ consists of numerically calculated system evaluation criteria that are obtained for the ith set of initial starting points.

2.3. Sensitivity Indices. Sensitivity analysis is performed in the basis of the first-order sensitivity index S_j and the total effect index S_{Tj}. The first-order sensitivity index (S_j) indicates the main effect of model output variance relating to the jth input factor; however, the total effect sensitivity index (S_{Tj}) includes the higher order effects as the sum of the first, second, third order effects, and so on [13]. According to Saltelli [11], the first-order sensitivity index S_j and the total effect sensitivity index S_{Tj} can be estimated by using the following expressions:

$$S_j = 1 - \frac{(1/2N)\sum_{i=1}^{N}\left(f(\mathbf{X}_B)_i - f\left(\mathbf{X}_{Cj}\right)_i\right)^2}{(1/N)\sum_{i=1}^{N} f^2(\mathbf{X}_A)_i - f_0^2}, \quad i = 1, \ldots, N, j = 1, \ldots, p, \tag{5}$$

$$S_{Tj} = \frac{(1/2N)\sum_{i=1}^{N}\left(f(\mathbf{X}_A)_i - f\left(\mathbf{X}_{Cj}\right)_i\right)^2}{(1/N)\sum_{i=1}^{N} f^2(\mathbf{X}_A)_i - f_0^2}, \quad i = 1, \ldots, N, j = 1, \ldots, p, \tag{6}$$

where $f(\mathbf{X}_B)_i$ and $f(\mathbf{X}_A)_i$ are the model outputs calculated using independent sets of initial starting points and $f(\mathbf{X}_{Cj})_i$ is also the model output but calculated in the basis of recombined sets of initial starting points (Section 2.2). Expected value f_0 of model outputs can be estimated by using the arithmetic mean in the following form [11]:

$$f_0 = \frac{1}{N}\sum_{i=1}^{N} f(\mathbf{X}_A)_i, \quad i = 1, \ldots, N, \tag{7}$$

where N is the number of simulation runs.

2.4. Block Diagram for the Sensitivity Analysis of Vibration Reduction Systems. In order to analyse the vibroisolation properties of vibration reduction systems, the global sensitivity analysis is employed. By using such analysis, the relative importance of input factors with respect to the model output should be quantified to identify the key model parameters that affect the system performance significantly. The variance-based method [13] is used in the presented study because such a method is suitable for nonlinear models that are generally used to describe a specific vibroisolation process realized by the system [15]. Block diagram for the sensitivity analysis of vibration reduction systems is presented in Figure 3.

Based on both the selected number N of simulation runs and the double number $2p$ of input factors to be analysed, the matrix $\mathbf{X}_{AB}$ of initial starting points should be generated by using a uniform random number generator (Section 2.1). Then, the resulting sample matrix has to be divided into two separated matrices $\mathbf{X}_A$ and $\mathbf{X}_B$ of the same size ($N \times p$) that

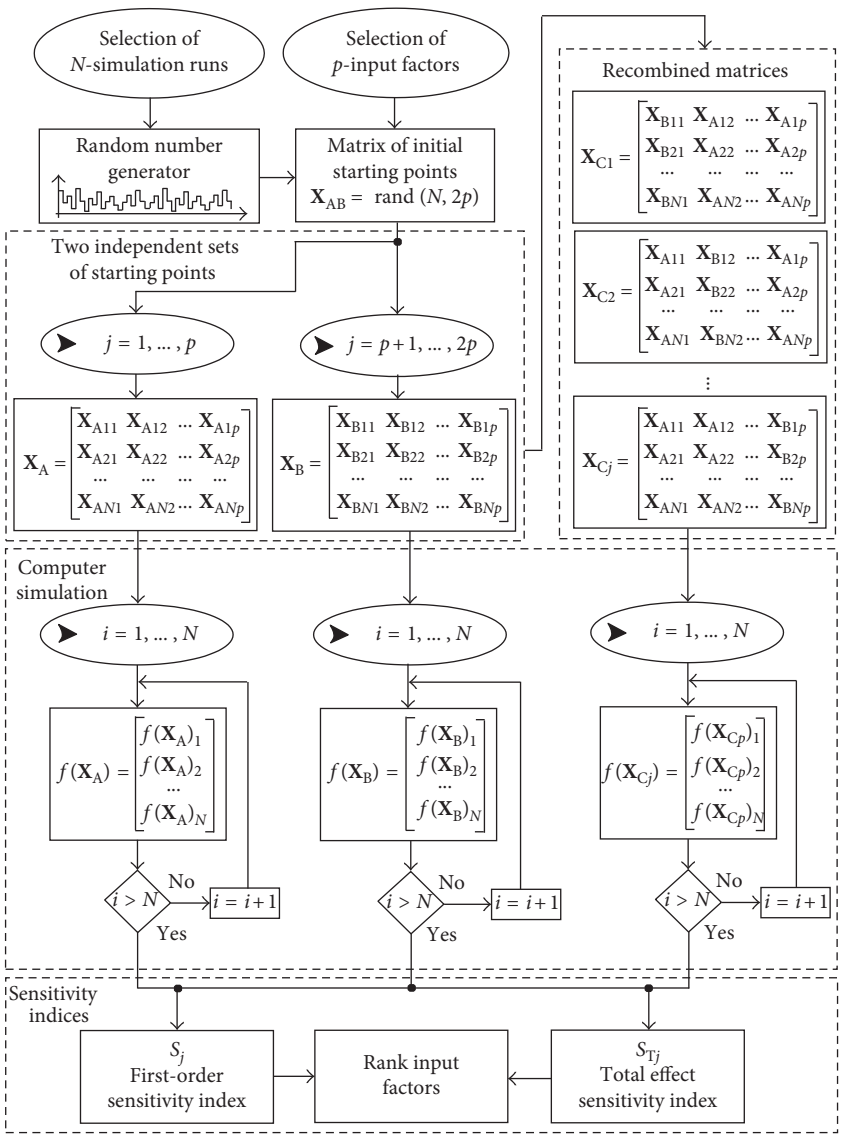


Figure 3: Block diagram for the sensitivity analysis of vibration reduction systems.

include two independent sets of initial starting points. Additionally, the recombined matrices $\mathbf{X}_{C1}, \mathbf{X}_{C2}, \ldots, \mathbf{X}_{Cj}$ must be created in such a way that these matrices contain the elements of matrix $\mathbf{X}_A$ except the jth column which is taken from matrix $\mathbf{X}_B$. Further analysis requires $N(2+p)$ model calculations by using a computer simulation. The corresponding model outputs have to be stored as the following vectors: $f(\mathbf{X}_A)$, $f(\mathbf{X}_B)$, and $f(\mathbf{X}_{Cj})$. When all of the model outputs are iteratively evaluated, then the sensitivity indices shall be estimated. The first-order sensitivity index S_j and the total effect sensitivity index S_{Tj} are calculated using (5)–(7), and the obtained results can be used to provide a reliable ranking of the input factors.

3. Example: Sensitivity Analysis of the Horizontal Seat Suspension

3.1. Model of the System. In Figure 4, the physical model of the horizontal seat suspension with seated human body is shown. The object of vibration isolation is assumed as a lumped mass body which consists of three interconnected masses by means of linear springs and dampers [16]. The first equivalent mass m_1 is employed for modeling the upper part of the frame equipped with a cushion on which the person sits. The second and third masses are used to model the human body parts that are in contact (m_2) and without contact (m_3) with the back support. The equivalent stiffness c_{12x} and damping d_{12x} coefficients are utilized to determine viscoelastic characteristics of the human body part to be in contact with the back support, while the stiffness c_{23x} and damping d_{23x} coefficients define biodynamic response of the human body part that is not in contact with the back support. Finally, the equivalent stiffness c_{2x} and damping d_{2x} coefficients are used to express the reactions transmitted through hands on the steering wheel.

Such a simple 3-DOF model is extensively discussed in the literature [16] and captures many essential characteristics of the system—seated human body and cushioned seat.

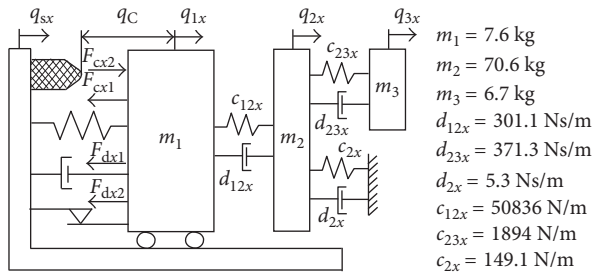


Figure 4: Physical model of the passive horizontal seat suspension with seated human body.

The displacements q_{1x}, q_{2x}, and q_{3x} represent the movement of elements contained in the biomechanical model of a human body. The mechanical vibrations are generated by using kinematic excitation q_{sx} of the system.

A set of three independent equations of motion is formulated in the matrix form:

$$\mathbf{M}_x\ddot{\mathbf{q}}_x + \mathbf{D}_x\dot{\mathbf{q}}_x + \mathbf{C}_x\mathbf{q}_x = \mathbf{F}_{sx}, \tag{8}$$

where $\mathbf{q}_x$ is the displacement vector of the isolated body; $\mathbf{M}_x$, $\mathbf{D}_x$, and $\mathbf{C}_x$ are the inertia, damping, and stiffness matrices, respectively; and $\mathbf{F}_{sx}$ is the vector of applied forces describing the nonlinear vibration isolator.

The diagonal inertia matrix $\mathbf{M}_x$ represents the masses of each element contained in the human body model as follows:

$$\mathbf{M}_x = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{bmatrix}. \tag{9}$$

The damping $\mathbf{D}_x$ and stiffness $\mathbf{C}_x$ matrices are symmetric, and they are defined for a specific direction of the vibration exposure:

$$\begin{aligned} \mathbf{D}_x &= \begin{bmatrix} d_{12x} & -d_{12x} & 0 \\ -d_{12x} & d_{2x} + d_{12x} + d_{23x} & -d_{23x} \\ 0 & -d_{23x} & d_{23x} \end{bmatrix}, \\ \mathbf{C}_x &= \begin{bmatrix} c_{12x} & -c_{12x} & 0 \\ -c_{12x} & c_{2x} + c_{12x} + c_{k23x} & -c_{23x} \\ 0 & -c_{23x} & c_{23x} \end{bmatrix}. \end{aligned} \tag{10}$$

The human body model parameters, that is, body masses: m_1, m_2, and m_3, damping coefficients: d_{12x}, d_{23x}, and d_{2x}, and stiffness coefficients: c_{12x}, c_{23x}, and c_{2x}, have been identified for the seated human body with cushioned seat system and backrest contact in the lumbar region [16].

The vectors describing the human body displacements and applied forces are presented in the following form:

$$\begin{aligned} \mathbf{q}_x &= \begin{bmatrix} q_{1x} \\ q_{2x} \\ q_{3x} \end{bmatrix}, \\ \mathbf{F}_{sx} &= \begin{bmatrix} -F_{cx1} + F_{cx2} - F_{dx1} - F_{dx2} \\ 0 \\ 0 \end{bmatrix}. \end{aligned} \tag{11}$$

The mathematical models of the basic forces in the system, that is, the spring force F_{cx1}, the force from end-stop buffers F_{cx2}, the force of hydraulic shock absorber F_{dx1}, and the overall friction force of suspension system F_{dx2}, have been shown in the author's previous paper [17]. In this paper, similar model has been formulated and experimentally verified for the purpose of optimizing the vibroisolation properties of horizontal seat suspension. The present paper concerns a global sensitivity analysis; therefore, another scientific problem is discussed. However, a novel research methodology is applied for a specific seat suspension system in order to investigate the relative influence of nonlinear force characteristics on the system dynamics.

3.2. Input Factors. An influence of nonlinear force characteristics of the conservative (F_{cx1}, F_{cx2}) and dissipative (F_{dx1}, F_{dx2}) elements is investigated for chosen excitation signals. The following forces as functions of the system relative displacement $q_{1x} - q_{sx}$ or velocity $\dot{q}_{1x} - \dot{q}_{sx}$ are taken into account:

(i) Spring force $F_{cx1} = f(q_{1x} - q_{sx}, d_s)$

(ii) Force from the end-stop buffers $F_{cx2} = f(q_{1x} - q_{sx}, c_{b1}, c_{b3})$

(iii) Force of the hydraulic shock absorber $F_{dx1} = f(\dot{q}_{1x} - \dot{q}_{sx}, d_o, l_o)$

(iv) Overall friction force $F_{dx2} = f(\dot{q}_{1x} - \dot{q}_{sx}, \delta_f)$.

In order to enable adjusting the force characteristics of the horizontal seat suspension, the input factors and their ranges are defined as the set of important design parameters:

(i) Wire diameter of the helical springs $d_s = (1.4\text{–}2.2) \cdot 10^{-3}$ m

(ii) Linear stiffness coefficient of the end-stop buffers $c_{b1} = (3\text{–}30) \cdot 10^3$ N/m

(iii) Cubic stiffness coefficient of the end-stop buffers $c_{b3} = (50\text{–}500) \cdot 10^6$ N/m^3

(iv) Orifice diameter of the shock absorber $d_o = (0.75\text{–}1.15) \cdot 10^{-3}$ m

(v) Orifice length of the shock absorber $l_o = (2\text{–}20) \cdot 10^{-3}$ m

(vi) Reduction ratio of the friction force $\delta_f = (0.2\text{–}1.2)$.

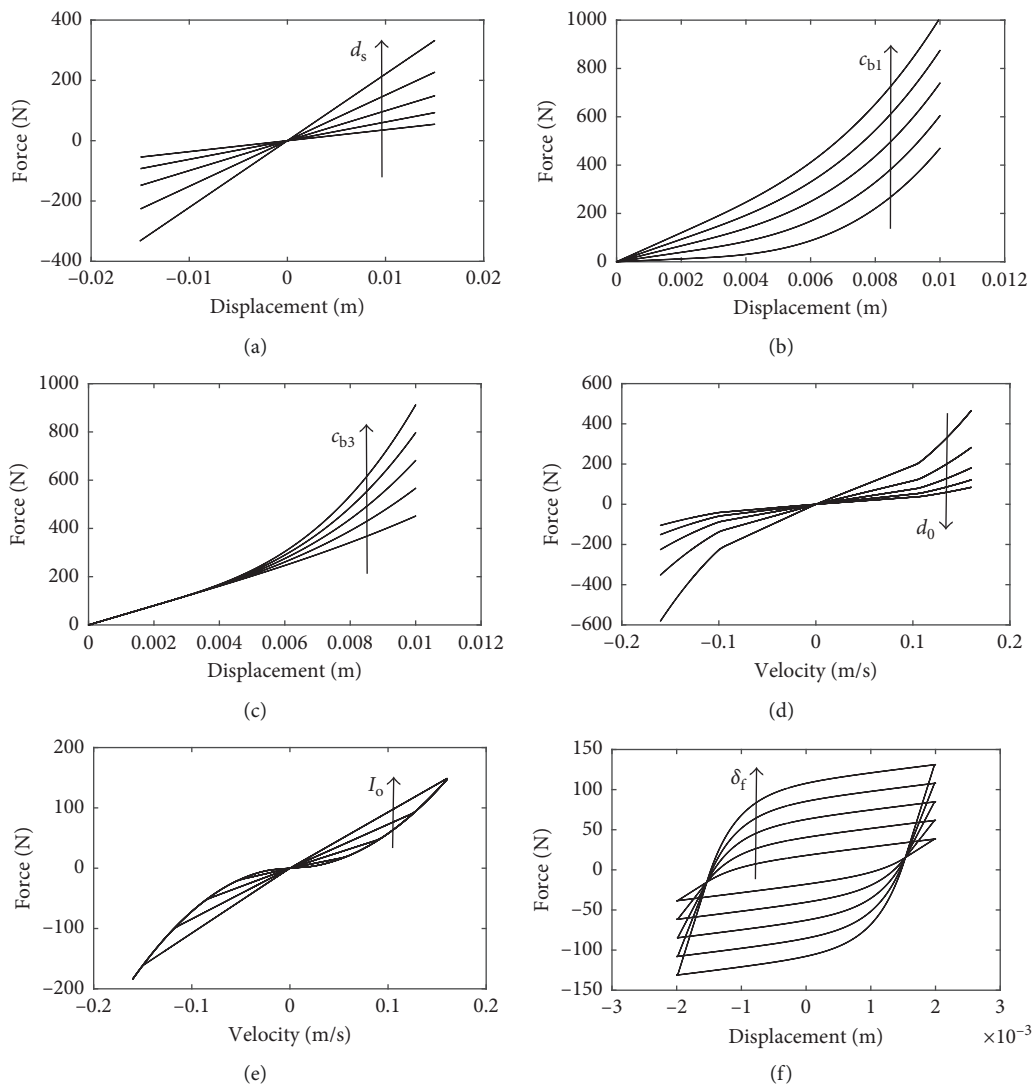


Figure 5: Force characteristics F_{cx1} of the helical spring for different wire diameter d_s (a), force characteristics F_{cx2} of the end-stop buffer for different linear stiffness coefficient c_{b1} (b), force characteristics F_{cx1} of the end-stop buffers for different cubic stiffness coefficient c_{b3} (c), force characteristics F_{dx1} of the shock absorber for different orifice diameter d_o (d), force characteristics F_{dx1} of the shock absorber for different orifice length l_o (e), and friction force F_{dx2} for different reduction ratio δ_f (f).

Individual characteristics of the viscoelastic elements included in horizontal seat suspension are presented in Figure 5. The helical springs can be modified by means of the wire diameter d_s, so its stiffness characteristic can be changed (Figure 5(a)). The stiffness characteristics of end-stop buffers can be also adjusted using the linear c_{b1} (Figure 5(b)) and cubic c_{b3} (Figure 5(c)) stiffness coefficients. A change of the damping characteristics in the velocity domain is also possible by modifying the circular orifice inside the shock absorber. Changing the orifice diameter d_o allows to adjust the value of damping force (Figure 5(d)), and the orifice length l_o influences the nonlinear shape of damping characteristics (Figure 5(e)). Furthermore, the friction force of the suspension system can be modified by using the reduction ratio δ_f (Figure 5(f)).

3.3. Model Outputs. The conflicted vibroisolation criteria [18], which correspond to the model outputs in this paper, provide a simple numerical assessment of the suspension system efficiency. For the purpose of evaluating the vibroisolation properties of vibration reduction systems, the following indicators are chosen:

(i) Frequency-weighted transmissibility factor TFE_x

(ii) Suspension travel s_{tx}.

At first, the frequency-weighted transmissibility factor is defined in order to evaluate the effectiveness of the vibration reduction system:

$$\mathrm{TFE}_x = \frac{(a_{1xw})_{\mathrm{RMS}}}{(a_{sxw})_{\mathrm{RMS}}}, \tag{12}$$

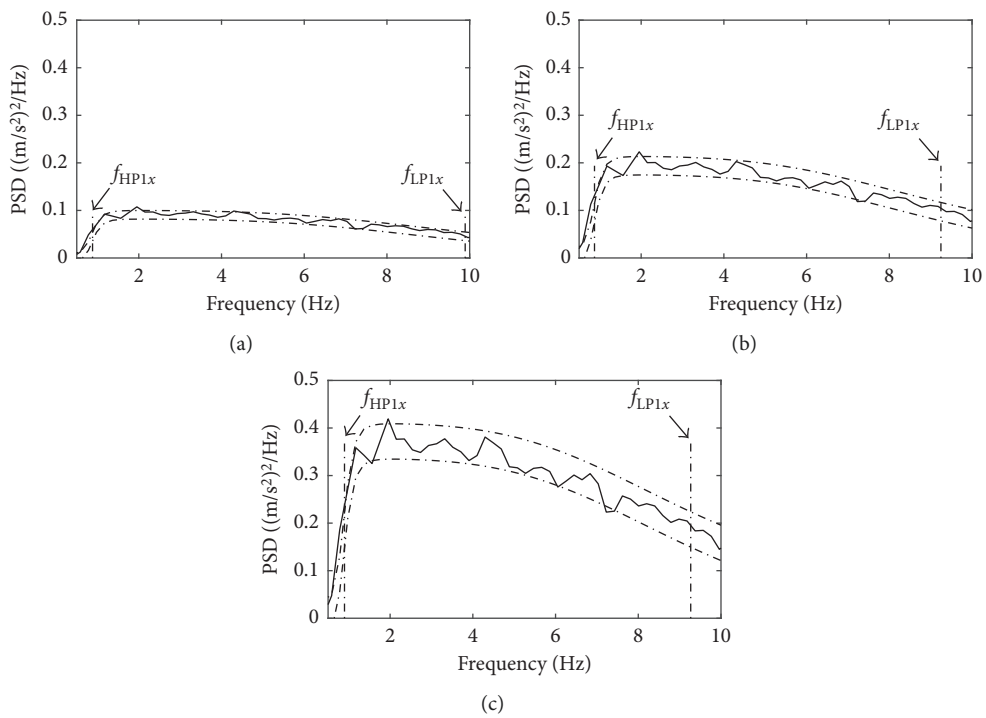


Figure 6: Power spectral densities of the simulated input vibration (solid line) at different intensities: WN1x (a), WN2x (b), and WN3x (c) and their tolerances (dashed line).

Table 1: Numerical values of the cutoff frequencies and filter orders for the input vibration at different excitation intensities: WN1x, WN2x, and WN3x.

Input vibration	High-pass filter		Low-pass filter		Excitation intensity
	Cutoff frequency f_{HP1x} (Hz)	Filter order n_{HP1x}	Cutoff frequency f_{LP1x} (Hz)	Filter order n_{LP1x}	$(\ddot{q}_{sx})_{RMS}$ (m/s^2)
WN1x	0.771	4	9.89	2	1.02
WN2x	0.742	4	9.24	2	1.36
WN3x	0.776	4	9.26	2	1.91

where $(a_{1xw})_{RMS}$ is the frequency-weighted root mean square value of the isolated body acceleration for the longitudinal x direction and $(a_{sxw})_{RMS}$ is the frequency-weighted root mean square value of the input acceleration. The frequency weightings defined in ISO-2631 [19] have to be used for the purpose of calculating the frequency-weighted accelerations.

For further approach, the suspension travel is a simple numerical assessment of the system performance as well [1]. The suspension travel is defined as the measure of distance from the bottom to the top of suspension stroke using the following relation:

$$s_{tx} = \max_{t\in[0,t_k]}\left(q_{1x}(t) - q_{sx}(t)\right) - \min_{t\in[0,t_k]}\left(q_{1x}(t) - q_{sx}(t)\right), \tag{13}$$

where $q_{1x}(t)$ is the displacement of the isolated body for the longitudinal x direction, $q_{sx}(t)$ is the displacement of the input vibration, t is the current time instant, and t_k is the observation time.

3.4. Simulation Results. The sensitivity analysis is performed by using the simulation model of the horizontal suspension system. The simulations are performed for the excitation signals that are similar to the white noise, band-limited noise. These excitation signals are obtained using a signal generator with the random wave form. Spectral characteristics of normally distributed random signals are subsequently formed by the Butterworth filters: high-pass (HP) and low-pass (LP) filters in a specific frequency range. Simulated power spectral densities of the input vibration at different excitation intensities are presented in Figure 6. Numerical values of the cutoff frequencies and filter orders are presented in Table 1.

The first-order and total effect sensitivity indices ((5) and (6)) of transmissibility factor TFE_x and suspension travel s_{tx} are calculated for chosen input vibrations, that is, WN1x, WN2x, and WN3x. According to the results obtained by using the variance-based method (Figure 7), the friction force (input factor δ_f) demonstrates the greatest influence on the system dynamics for most of the excitation signals. Less

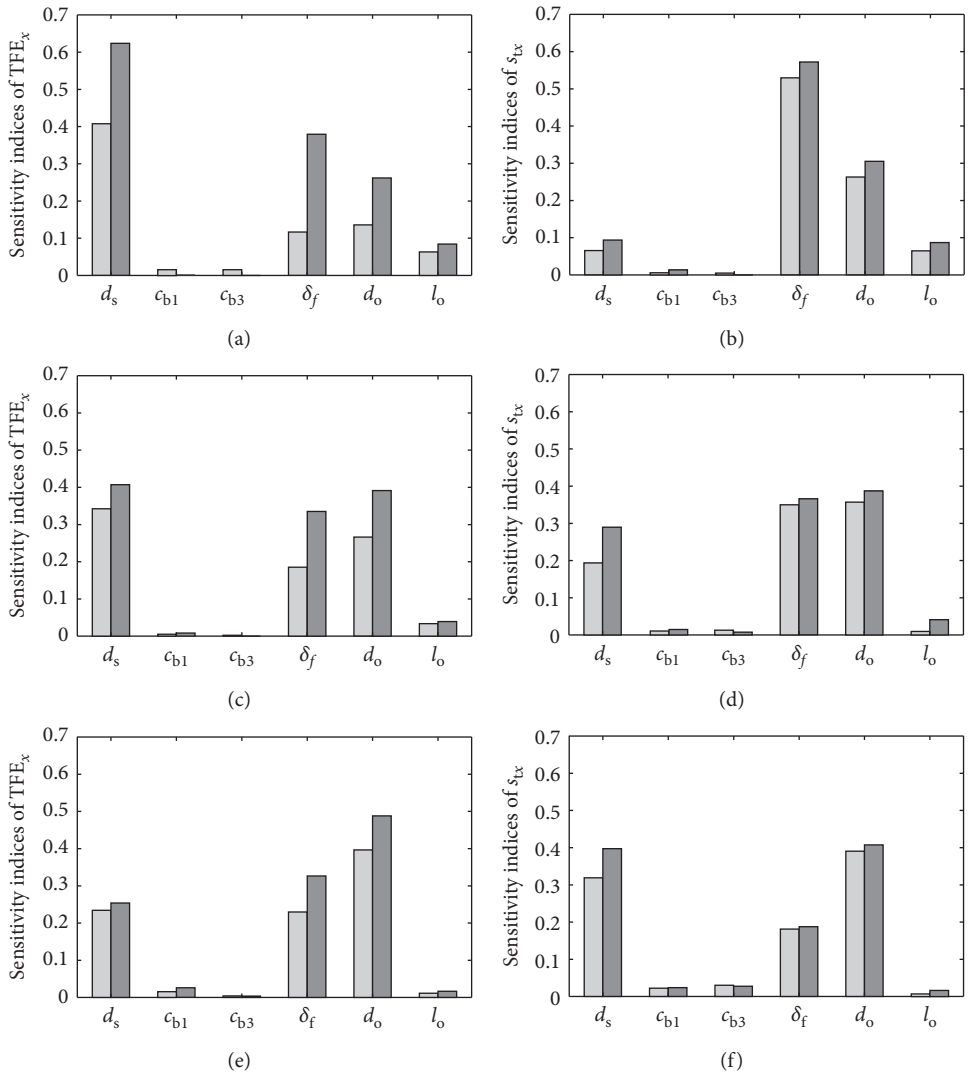


Figure 7: First-order (■) and total effect (■) sensitivity indices of transmissibility factor TFE$_x$ for different input vibrations: WN1x (a), WN2x (c), and WN3x (e); first-order (■) and total effect (■) sensitivity indices of suspension travel s_{tx} for different input vibrations: WN1x (b), WN2x (d), and WN3x (f).

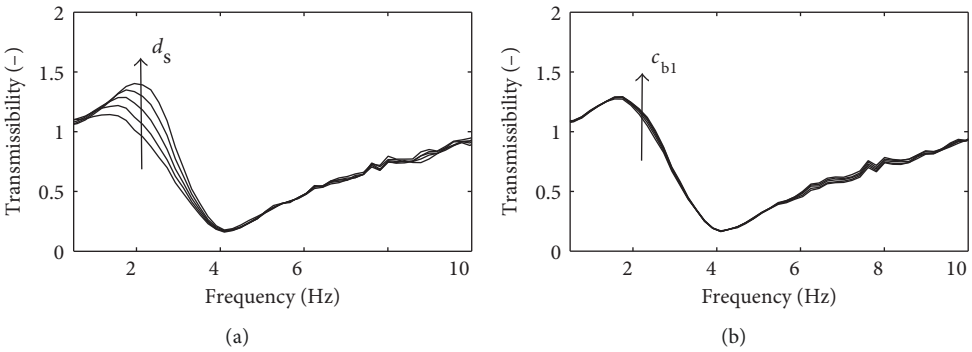


Figure 8: Continued.

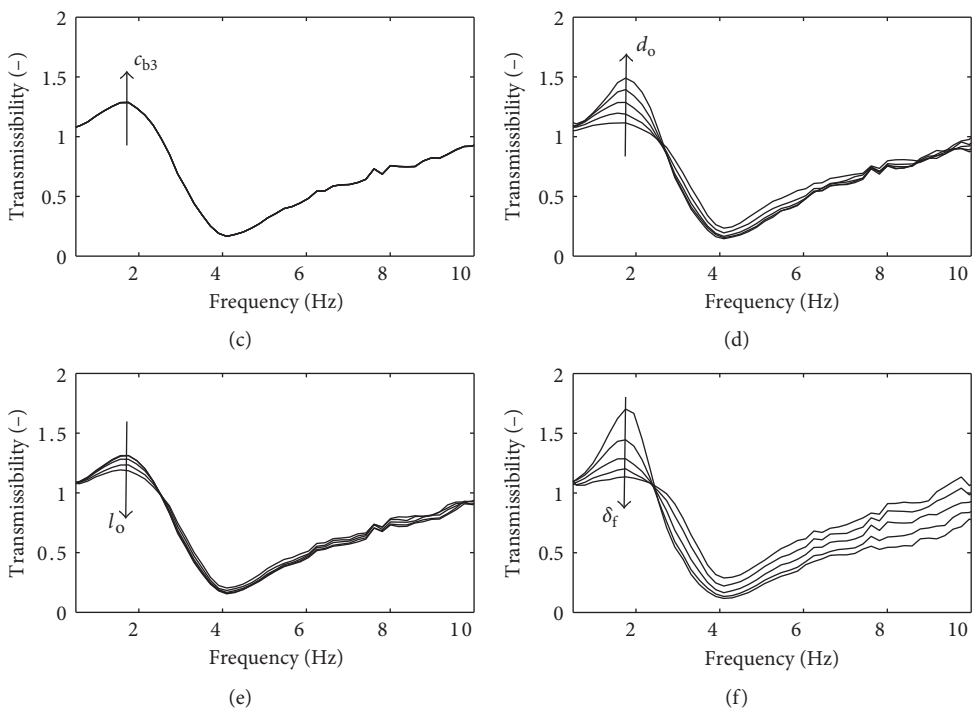


Figure 8: Transmissibility functions of the horizontal seat suspension at different input factors: wire diameter of the helical springs d_s (a), linear stiffness coefficient of the end-stop buffers c_{b1} (b), cubic stiffness coefficient of the end-stop buffers c_{b3} (c), orifice diameter of the shock absorber d_o (d), orifice length of the shock absorber l_o (e), and reduction ratio of the friction force δ_f (f).

influence of the suspension friction is observed for signal WN3x with relatively high amplitudes of the input vibration. In the case of such a signal, the damping force (input factor d_o) is dominant; therefore, the sensitivity indices achieve substantial values. Stiffness characteristics of the end-stop buffers (input factors c_{b1} and c_{b3}) point out the lowest influence on the system vibroisolation properties for each excitation intensities. An undesirable end-stop impact in a suspension seat occurs only occasionally; therefore, the suspension stroke is sufficient for the considered excitation signals.

Transmissibility functions of the horizontal seat suspension at various system configurations are presented in Figure 8. The presented dynamic behaviour of the system under examination corresponded well to the sensitivity indices that are calculated by using the variance-based method. This evidently proves that the proposed sensitivity analysis is correctly formulated and yields the desired overview of nonlinear system dynamics for strictly defined operating conditions.

4. Conclusions

In this paper, an effective procedure of analysing the vibroisolation properties of vibration reduction systems is proposed. The global sensitivity analysis is applied to determine the sensitivity of evaluation criteria to the design parameters by means of which the correct system configuration can be found. The developed variance-based method is suitable to analyse an influence of the nonlinear force characteristics and allows to estimate the sensitivity indices based on the conflicted vibroisolating criteria, that is, the frequency-weighted transmissibility factor and suspension travel. The correctness of the proposed procedure is evaluated by using the exemplary horizontal seat suspension that is exposed to random vibration at different excitation intensities.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

Acknowledgments

This work was part of the research project "Methods and procedures of selecting vibro-isolation properties of vibration reduction systems" funded by the National Science Center of Poland under the Contract no. UMO-2013/11/B/ST8/03881.

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