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Research Paper

Synthesis of Leaf-spring Suspension Parameters for High-comfort Vehicles

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Abstract

This paper develops the new leaf-spring suspension parameters synthesis methods based on the use of modern mathematical and computer modeling tools to provide vehicle high in-motion comfort. The research aims to develop the new loading characteristics and design parameters synthesis methodology for the vehicle suspension system components, exactly for the elastic element (leaf-spring), the damping element (hydraulic shock absorber) and the transverse stabilizer bar. The authors demonstrate approaches to the use of vehicle dynamics simulation modeling, topological and parametric optimization, and finite element method and explain their role in the new developed vehicle leaf-spring suspension system parameters synthesis methodology. As main results, the authors propose a formation method for a primary suspension system elastic element non-linear loading characteristic to provide for the in-motion vehicle comfort. They develop a synthesis technique for the variable profile longitudinal section geometry of a leaf-spring and its other design parameters to generate the loading characteristic close to the one required for high motion comfort. They also proposed a loading characteristics synthesis for the damping element of the primary suspension with a leaf-spring-type elastic element and selection the most suitable one for a specific vehicle method. The authors also propose a stiffness calculation procedure for a randomly shaped transverse stabilizer bar when selecting its design parameters. As a contribution, based on the comparative virtual tests results of the motion comfort indicators of a vehicle with the loading characteristics generated using the developed methodology for the primary leaf-spring suspension and the initial basic configuration of the classic leaf-spring suspension, it was recorded that the vehicle motion comfort improved by 10-25% when the developed synthesis methodology was used.

Keywords: Vehicle; Comfort; Leaf-spring suspension; Damping element; Stabilizer bar

1. Introduction

Increasing the speed and efficiency of transportation requires solving multiple questions. Considering automobiles, increasing the vehicle power output is the most popular way to improve the cargo and passenger transportation efficiency as it helps increase cruise speed. However, it simultaneously affects the in-motion comfort and vibration safety.

The most well-known ways of improving the vehicle in-motion comfort include the reduction of the elastic element stiffness and damping levels in suspension systems [1], [2]. However, the reduction of the elastic element stiffness with linear loading characteristics results in the increased suspension strokes [3], while reduced damping increases the risk of vehicle rocking, which has a negative effect on the driver's and passenger's comfort and health, as well as traffic safety [4], [5].

Modern suspension system design frequently uses the finite element method [6], [7], [8], [9], [10]. in durability calculations to test the operability and safety of the vehicle components design solutions [11], [12], [13], [14], [15]. The suspension system elements stiffness is determined by the properties of the material they are made of [16], [17]. Besides, the finite element method is used to refine the vehicle movement simulation models by introducing elastic bodies to the solid body dynamic system [18], [19], [20], [21], [22] and optimizing the structure to improve its durability [23], [24].

Leaf-springs are the basic components of vehicle suspension systems that operate in various climate conditions [25]. To study how leaf-spring suspension properties impact comfort, the scientists use the dynamic models that use a precise description of the leaf-spring model to produce reliable results [26]. Dynamic calculations are used for the precise calculation of deformations occurring in leaf-spring leaves, strain distribution, and leaf-spring service life forecasting [27], [28].

There are various ways of setting elastic bodies in vehicle dynamic models. The most popular ones include the introduction of a reduced finite-element model by a modal Craig-Bampton method [29], [30], [31] and the division of a single solid body into several parts with their subsequent joining with a hinge pivot with a set stiffness [32], [33]. The transverse stabilizer bar is one of the elastic bodies in vehicle suspension. Modern methods suggest viewing the stabilizer as an elastic bar that works in torsion [3]. If the stabilizer bar has a complex geometry, it can be difficult to use such methods due to the significant discrepancies between the analytical calculations and the experimental data.

Therefore, it is necessary to develop new synthesis methods for the vehicle suspension systems that can satisfy the current power output, average motion speed, and comfort requirements as shown in **Figure 1**.

Because of this, it is necessary to generate the suspension systems non-linear elastic element characteristics to maintain the suspension strokes [34], and the required damping characteristics must be generated together with the elastic characteristics for a specific vehicle [1]. When developing a suspension system, it is necessary to correctly determine the transverse stabilizer bar stiffness and its design parameters considering its geometry. Due to this research gap, the developed methodology shall define clearly the required values of forces formed by the elastic and damping elements depending on their deformations and their rates.

Therefore, based on the identified problems and available opportunities, the research aims to develop the new loading characteristics and design parameters synthesis methodology for the vehicle suspension system components, exactly

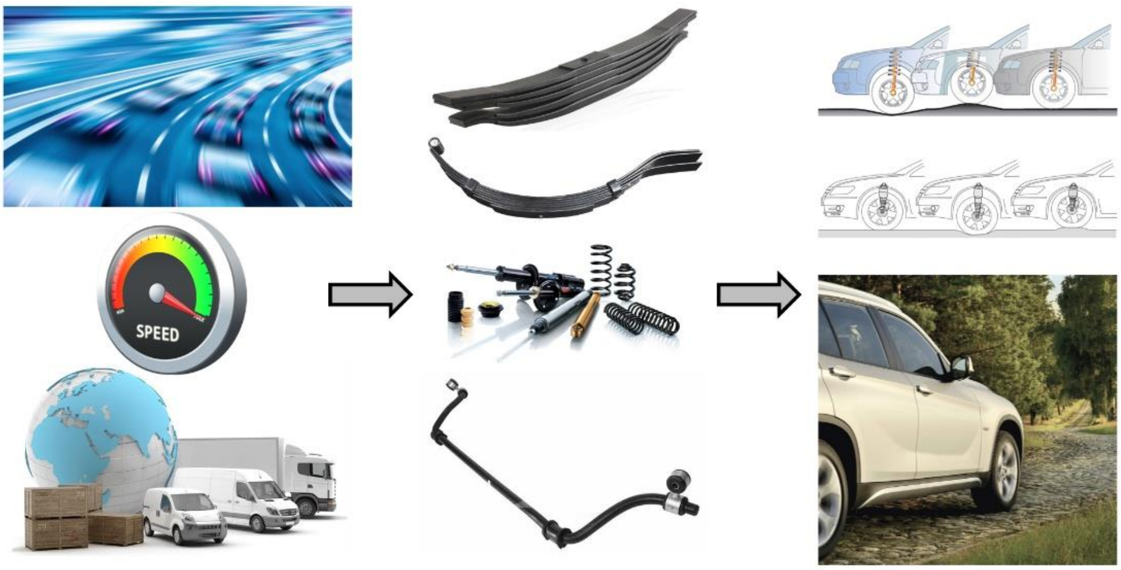

Figure 1. The problem of increasing automobile speed and efficiency

for the elastic element (leaf-spring), the damping element (hydraulic shock absorber) and the transverse stabilizer bar.

2. Methods

2.1. *Vehicle Suspension Elastic Element Non-Linear Loading Characteristic Formation Method*

This method is based on assigning suspension stiffness based on the natural oscillation frequency of the suspended vehicle mass installed on the chassis. Since a vehicle is operated in the total weight and curb weight modes, it is necessary to cover the minimum possible equal natural oscillation frequencies for these modes. Their value has to be determined with account to the natural oscillation frequencies of other vehicle systems (frame, unsuspended masses, primary and secondary suspension systems) so that these natural frequencies were not aligned and the natural frequency of the suspension was implementable and remained at the minimum possible permitted value range [3] adjacent to the natural frequency ranges of other systems. The set suspension natural frequency is used to generate the elastic characteristic.

For the generated elastic element characteristic, it is necessary to select the linear sections corresponding to automobile oscillations with small amplitudes for both the total and the curb weight. The amplitudes and the widths of the corresponding sections must be determined based on the vehicle operating conditions of the vehicle. To determine the small elastic element oscillation amplitudes relative to the suspension static position, it is necessary to obtain the root mean square (RMS) deviation of road microprofile roughness RMS_q for the roughness categories that the studied vehicle will be used on [5].

When RMS_q is determined, it is necessary to calculate the forces applied to the suspension at the reference points: the static travel length of the elastic element loaded by the curb-weight automobile, the static stroke of the elastic element loaded by the total-weight automobile, the dynamic elastic element stroke, and the stiffness of the characteristic linear sections in the these forces application area. The elastic element stiffness is determined using the known natural frequency ω by Eq. (1) where mass m is the curb weight m_{cw} and the total weight m_{tw}. Thus, we get two linear section stiffness values: c_{cw} and c_{tw}. To determine static load P applied to the elastic element of the curb weight automobile $P_{st.c}$ and the total weight automobile $P_{st.t}$, use Eq. (2).

$$\omega = \sqrt{c/m} \tag{1}$$

$$P = \frac{m \cdot g}{2} \tag{2}$$

where, g = 9.81 m/s² is the acceleration of gravity.

When the independent suspension is used, one should account for its transmission ratio. To determine the highest reference point of the loading characteristic (dynamic force on the compression pass under total weight P_d), it is necessary to select the product of the static vehicle suspension force under total weight $P_{st.t}$ and the dynamic factor k_d from the recommended ranges [3]. It is necessary to determine the elastic element deformation value Δh between the static position of the curb weight and total weight automobile. To achieve this, it is necessary to set the maximum longitudinal vehicle tilt φ that occurs when only one of its axles is loaded. If we assume that with the curb weight the longitudinal tilt $\varphi = 0^{\circ}$ and, when only the axle in question is loaded, the deformation of the second axle suspension does not change, the value of Δh can be determined using Eq. (3).

$$\Delta h = l \cdot tg(\varphi) \tag{3}$$

where l is the wheelbase of the vehicle.

The acceptable vehicle tilt values are justified as limits beyond which the driver or passenger may feel uncomfortable [5]. Then we need to determine the reference points of the linear section of the loading characteristic: $(0, 0)$, $(h_{st.c}, P_{st.c})$, $(h_{st.t}, P_{st.t})$ and (h_d, P_d). To determine the deformation values of the elastic element, we need to divide the respective forces by the linear section stiffness factor obtained previously and the lacking forces should be calculated as a product of the respective deformations and stiffnesses. Further we need to move half of the RMS_q to both sides from the elastic element static position in the curb weight automobile $h_{st.c}$ and the elastic element static position of the total weight automobile $h_{st.t}$. These sections are required to ensure the suspension smoothness near static equilibrium points. After that, we need to move curb weight automobile suspension deformation values along the horizontal axis to meet the required distance of

Δh. Thus, we can obtain the elastic element stroke range with the minimum stiffness ($h_{st.min}$, $h_{st.max}$) under the main movement modes. The obtained reference points can be used to generate the elastic element loading characteristic.

To obtain the elastic characteristic with a smooth increase, it is necessary to approximate it in three sections using the limiting conditions: the values in two extreme points and the tilt angle at the limits of the linear sections. Depending on the number of the limiting conditions, the polynomial will have a value smaller than this number by one. Thus, in sections from (0, 0) to ($h_{c.min}$, $P_{c.min}$) and from ($h_{t.max}$, $P_{t.max}$) to (h_d, P_d), the characteristic polynomial will have the value of 2 because these cases feature three limiting conditions: passing through points and being tangent to one linear section of the characteristic. In the case of the section from ($h_{st.c}$, $P_{st.c}$) to ($h_{st.t}$, $P_{st.t}$), there will be 4 limiting conditions: two points and two tangencies. Based on the number of the limiting conditions, the characteristic polynomial will have a value of 3 and be the same as Eq. (4).

$$y(x) = a_0 + a_1 \cdot x + a_2 \cdot x^2 + a_3 \cdot x^3 \tag{4}$$

This expression has 4 unknown coefficients that can be calculated by drawing up and solving a system of 4 Eq. (5). The equations can be composed based on the limiting conditions: 2 points and tangencies to linear sections. In the case of the tangencies, we need to take the derivative from the polynomial, while linear section stiffnesses calculated previously can be used as a function here because they determine the tilt angle of the tangencies.

$$\begin{cases} y_1 = a_0 + a_1 \cdot x_1 + a_2 \cdot x_1^2 + a_3 \cdot x_1^3 \\ c_1 = a_1 + 2 \cdot a_2 \cdot x_1 + 3 \cdot a_3 \cdot x_1^2 \\ y_2 = a_0 + a_1 \cdot x_2 + a_2 \cdot x_2^2 + a_3 \cdot x_2^3 \\ c_2 = a_1 + 2 \cdot a_2 \cdot x_2 + 3 \cdot a_3 \cdot x_2^2 \end{cases} \tag{5}$$

Using the obtained Eq. (5), we can compose matrices $\boldsymbol{A}$ and $\boldsymbol{B}$, and the polynomial can be represented as Eq. (6) consisting of the obtained matrices and used to determine matrix $\boldsymbol{K}$. The obtained matrix $\boldsymbol{K}$ is a set of the required factors (a_0, …, a_3) that can be used to compose the polynomial. The other 2 sections of the elastic element loading characteristic can be approximated using the same procedure.

$$\boldsymbol{A} = \begin{bmatrix} 1 & x_1 & x_1^2 & x_1^3 \\ 0 & 1 & 2 \cdot x_1 & 3 \cdot x_1^2 \\ 1 & x_2 & x_2^2 & x_2^3 \\ 0 & 1 & 2 \cdot x_2 & 3 \cdot x_2^2 \end{bmatrix} \quad \boldsymbol{B} = \begin{bmatrix} y_1 \\ c_1 \\ y_2 \\ c_2 \end{bmatrix} \quad \boldsymbol{K} = \boldsymbol{A}^{-1} \times \boldsymbol{B} \tag{6}$$

2.2. *Leaf-spring Variable Longitudinal Section Profile Geometry Generation Method*

The usage of the topological optimization method in this methodology helps obtain the optimal distribution of material over the leaf-spring leave, provided that the leaf-spring leaves maintain the same durability along their entire lengths. When generating the longitudinal profile geometry for the leaf-spring leaves, we need to observe the following:

- Provide the lowest possible leaf-spring stiffness to improve vehicle comfort;
- The leaf-spring must withstand dynamic loading;
- The force required to compress the leaf-spring until stop against the suspension stroke limited shall be greater than the leaf-spring static loading;
- The leaf-spring attachment point shall correspond to the actual ones on a real vehicle;
- The yield point of the leaf-spring material shall not be exceeded under dynamic loading;
- Strains occurring in the leaf-spring leaves shall be distributed evenly along the leaf lengths.

To generate the leaf-spring longitudinal profile geometry, it is necessary to perform several topological optimizations of the simplified workpiece model to facilitate the subsequent approximate shape of the leaf-spring. For the initial model, we used a rectangle because the permitted design area with the attachment points in the same spots as the attachment fixtures of the suspension system of an actual vehicle. The width of the rectangle is set equal to the permitted width of the leaf-spring leave pack, and its height can be random. The initial model for the leaf-spring shape optimization is shown in **Figure 2**.

The 1st topological optimization iteration shown in **Figure 3a** features the areas where material has to be preserved to achieve lower weight and provide the required stiffness. Most of the material to be preserved is located in the lower

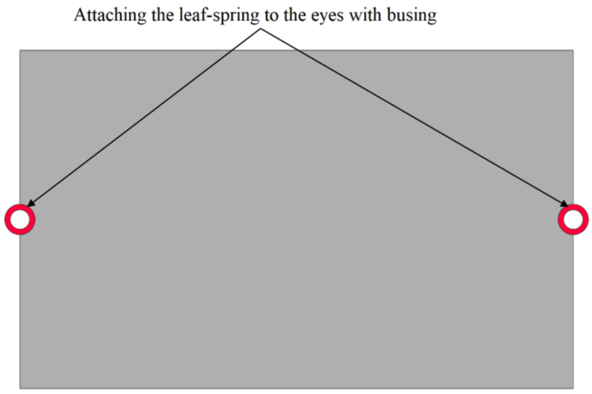

Figure 2. Leaf spring design area for topological optimization.

part. The material appears on the top because there is a deflection and tensile stress at the top of the initial model. Therefore, the position of the top of the model can be modified and taken down. At the bottom, we should remove the material on the sides and reduce the distance between the lower edge and the attachment point so that tension is distributed evenly across the lower part. Then it is necessary to change the geometry of the initial model iteration by iteration considering the obtained results as shown in **Figure 3**.

To determine the correlations between the number of leaf-spring leaves and their stiffness and durability, we used the longitudinal profile shape obtained during the topological optimization to compose the simplified few-leaf springs models and their finite-element models to perform durability calculations. At leaf-spring eyelets, we need to set limitations applied to the components through the centers of holes. The left limitation only allows for the rotation around the Z-axis, and the right limitation allows for the movement along the X-axis and the rotation around the Z-axis. Also, the contact interaction is between the leaf-spring leaves. The friction is so low it can be neglected because the calculations are performed using a simplified model that is only used to obtain the dependencies between the spring stiffness and durability and its geometry. Modern structures use several design solutions that help reduce interleaf frictions. Finite element method (FEM) calculations use leaf-spring deformation that is equal in all cases and corresponds to the largest stroke distance (until the full compression of the buffer) within the vehicle suspension system.

2.3. Leaf-Spring Suspension Damping Element Loading Characteristics Synthesis Method

The damping element loading characteristic must include 4 areas: 2 for the closed relief valves and 2 corresponding to the valve modes reflecting shock absorber compression and rebound whose beginning is determined as $\upsilon_{rod} = \upsilon_{rod\_valve} = (0.25…0.35)$ *m/sec* where υ_{rod} is the shock absorber rod speed, υ_{rod_valve} is the rod and piston speed when relief valves are open [3].

To produce the loading characteristics of the shock absorber (the dependencies of rod forces and rod speed) that would allow for the high vehicle moving comfort, we need to determine the permitted range of the performance characteristics. To achieve this, we need to determine the reference points of each characteristic based on the compression start and rebound valve modes and the zero point (*0*; *0*). To determine the forces applied in these points, it is necessary to set several parameters [3]:

- The natural oscillation frequency of the suspension system;
- The relative damping factor of suspension oscillations $\psi_z = 0.2…0.3$;
- The ratio $n_{sh\text{-}a}$ of damping factors during rebound and compression [3];
- The power value with open relief valves (in the general case, $i = 1$);
- The suspension system ratio by stroke is i_h.

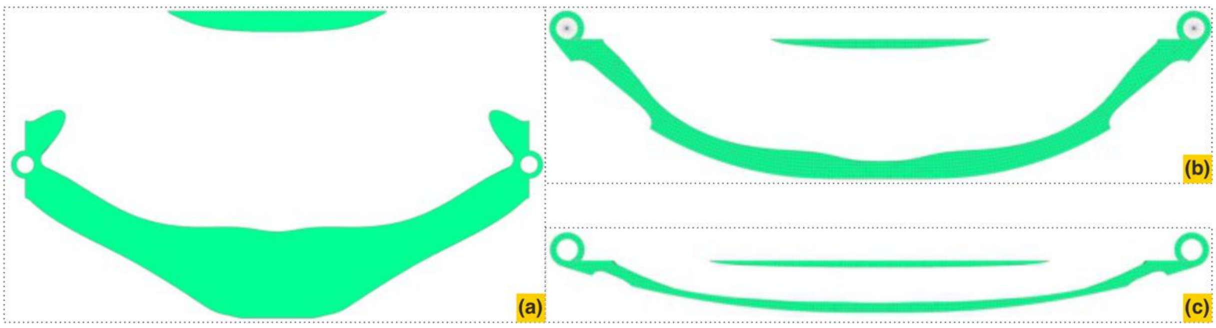

Figure 3. The leaf-spring topological optimization iterations: (a) 1st; (b) 2nd; (c) 3rd

To form the damping element loading parameters selection in the permitted operating parameters range, we need to know 2 the maximum and the minimum factors ψ_z and $n_{sh\text{-}a}$. With these parameter values, we need to calculate the shock absorber resistance factor k_p applied to the vehicle wheel using Eq. (7), and then determine the shock absorber resistance factor $k_{sh\text{-}a}$ using Eq. (8), the $k_{sh\text{-}a}$ factor can be determined on the other side using the Eq. (9), where, $k_{sh\text{-}a\_c}$ is the shock absorber resistance factor during compression, $k_{sh\text{-}a\_r}$ is the shock absorber resistance factor during rebound, also $k_{sh\text{-}a\_r} = n_{sh\text{-}a} \cdot k_{sh\text{-}a\_c}$.

$$k_p = 2 \cdot \pi \cdot \psi_z \cdot 0.5 \cdot m \tag{7}$$

$$k_{sh-a} = k_p \cdot {i_h}^2 \tag{8}$$

$$k_{sh-a} = 0.5 \cdot (k_{sh-a_c} + k_{sh-a_r}) \tag{9}$$

Then it is possible to determine the forces required to open the relief values during compression using Eq. (10) and rebound using Eq. (11) respectively.

$$P_{sh-a_c} = k_{sh-a_c} \cdot {\vartheta_{rod}}^i \tag{10}$$

$$P_{sh-a_r} = k_{sh-a_r} \cdot {\vartheta_{rod}}^i \tag{11}$$

Thus, we can select various force values. After grouping the values of $k_{sh\text{-}a\_c}$ and $k_{sh\text{-}a\_r}$, we can identify the minimum and maximum values (extreme reference points) and set the range of the permitted values and loading characteristics variants for the vehicle suspension damping element as shown in **Figure 4**.

Based on the reference points obtained during the synthesis, we can compose shock absorber non-linear loading characteristics following the procedure [5] that considers the calculation models for single-tube, double-tube, and other damping element types [35]. The required values of the geometrical parameters in these calculations have to be taken based on recommendations from [3]. The obtained equations can be used to determine the dependencies between the forces and the shock absorber rod speeds as shown in **Figure 4**. We need to select the optimal damping characteristic from the synthesized range that satisfies the vehicle comfort requirements. This can be done by selecting the range variants from the closest to the X-axis to the farthest until the vehicle in-motion rocking prevention requirement is fulfilled.

To select the required damping characteristic and confirm its operability and efficiency, we need to perform the simulation modeling of vehicle model dynamics [35] in various weight modes with various shock absorber parameters shown in **Figure 4**. The characteristic closest to the X-axis (with the lowest rod forces) should be considered the minimum and the farthest one should be deemed the maximum.

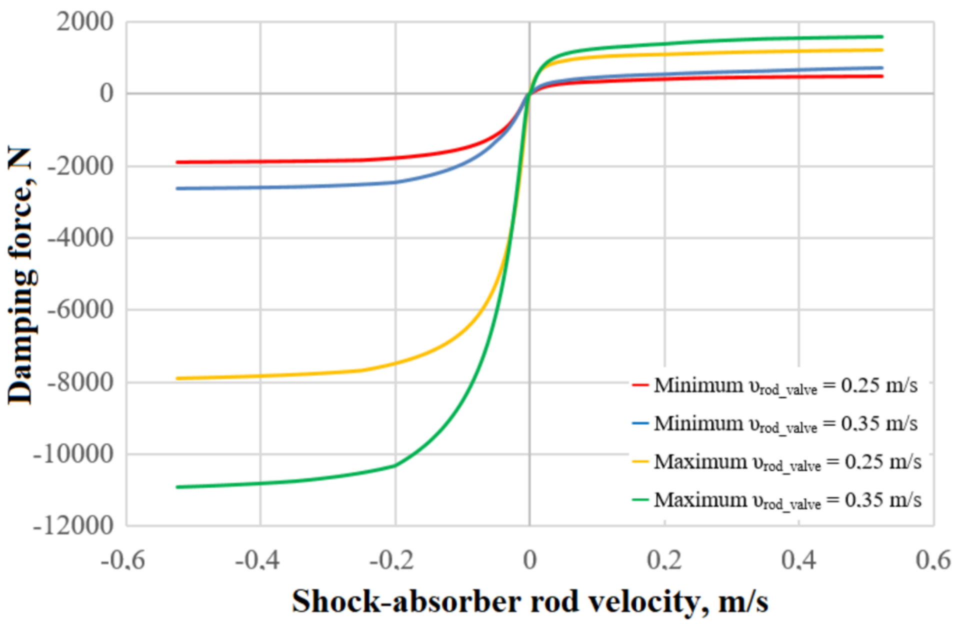

Figure 4. The dependencies between the damping forces and shock absorber rod speed

2.4. Randomly Shaped Transverse Stabilizer Bar Stiffness Calculation Method when Selecting its Design Parameters

The transverse stabilizer bar is a round profile with a randomly shaped guide, certaing parts of which work in torsion, while others work in bending [36] as shown in **Figure 5**. The stabilizer bar stiffness calculation method includes the following stages:

- The stabilizer is split into several simple sections;
- Bending (s_1, s_3, s_5, s_7) and torsion (l_2, l_4, l_6) sections are determined;
- Stabilizer section stiffness values are calculated taking into account their loading types.

The stiffness values for the bending stabilizer sections can be calculated using the Eq. (12).

$$c_i = \frac{E \cdot I_x}{s_i} \tag{12}$$

where, $i = 1, 3, 5, 7$ are the numbers of the bending stabilizer sections, E is the Young's module for the stabilizer bar material, I_x is the central inertia moment for the section during bending, s_i are bending stabilizer bar section lengths.

The stiffness values for the torsion stabilizer sections can be calculated using the Eq. (13).

$$c_j = \frac{G \cdot I_p}{l_j} \tag{13}$$

where, j = 2, 4, 6 are the numbers of the torsion stabilizer sections, G is the module of shear elasticity for the stabilizer bar material, I_p is the polar moment of the section inertia, l_j are the torsion stabilizer bar section lengths.

Ultimately, we need to determine the total stiffness of the transverse stabilizer bar.

Since all the stabilizer sections are connected in series, the total stiffness can be calculated using Eq. (14).

$$\frac{1}{c_{sb}} = \frac{1}{c_1} + \frac{1}{c_2} + \frac{1}{c_3} + \frac{1}{c_4} + \frac{1}{c_5} + \frac{1}{c_6} + \frac{1}{c_7} \tag{14}$$

where, c_1, c_3, c_5, c_7 are the stiffnesses of the bending stabilizer sections and c_2, c_4, c_6 are the stiffnesses of the torsion stabilizer sections.

3. Results and Discussion

3.1. Synthesized Suspension Elastic Element Loading Characteristic

Performing the described in subsections 2.1 action sequence results in the loading characteristic generation. An idealized characteristic for a suspension system elastic element is shown in **Figure 6**.

3.2. Leaf-spring Variable Longitudinal Section Profile Geometry Generation Results

For the synthesized according to subsections 2.2 leaf-spring models, we performed FEM calculations, the results of which are shown in **Figure 7**, to determine the geometrical parameters impact that vary during parametric optimization on their stiffness and durability. The width and height of the leaf-spring central part cross-section were set and varied.

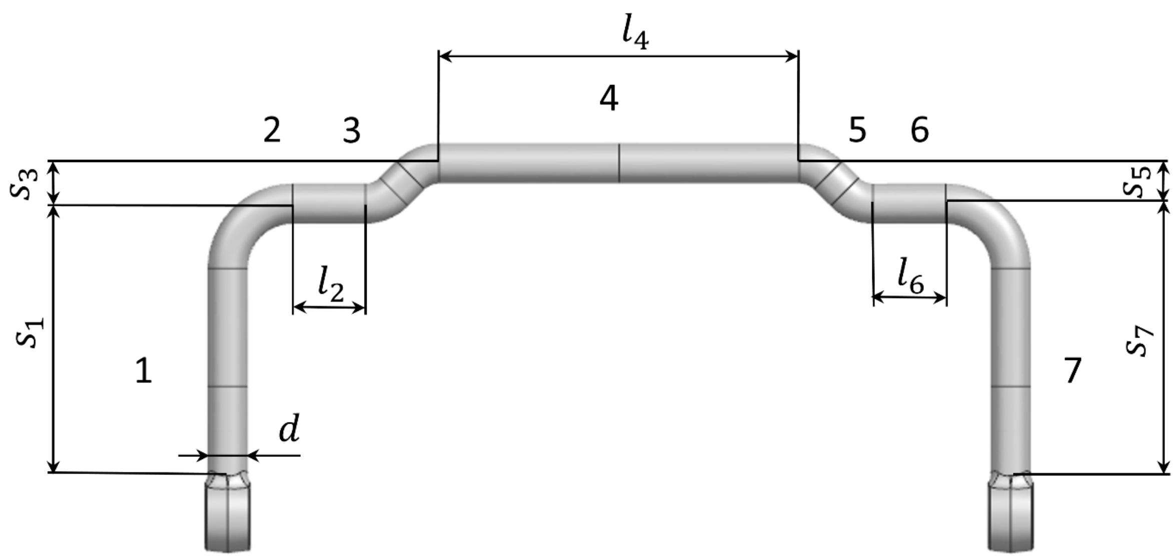

Figure 5. The geometry of the randomly shaped transverse stabilizer bar

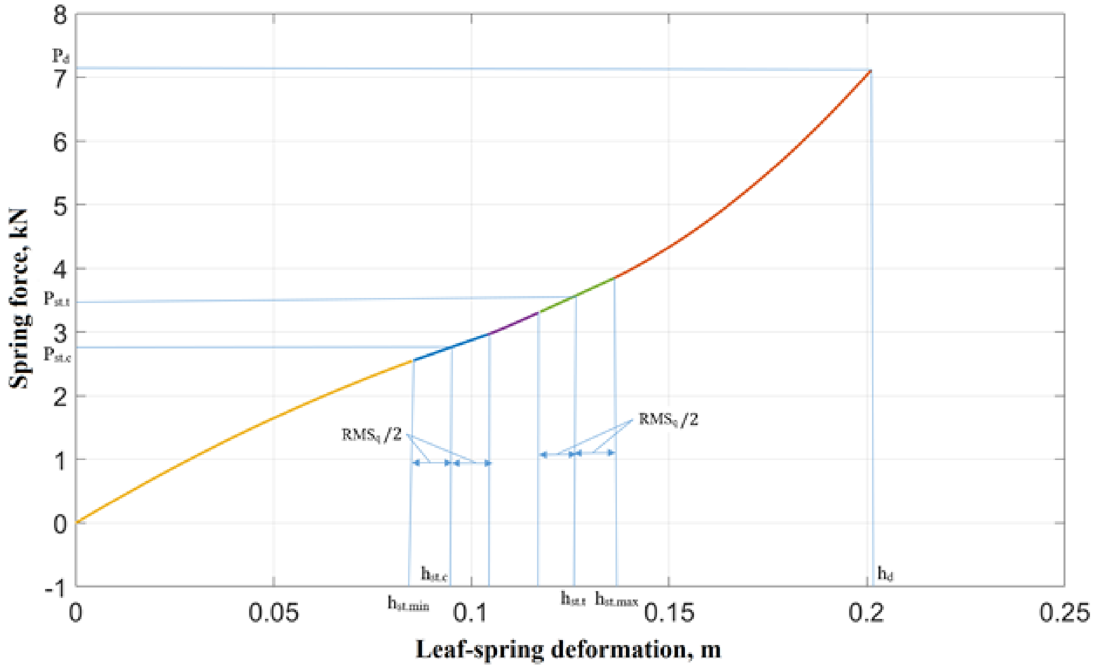

Figure 6. The synthesized suspension elastic element loading characteristic

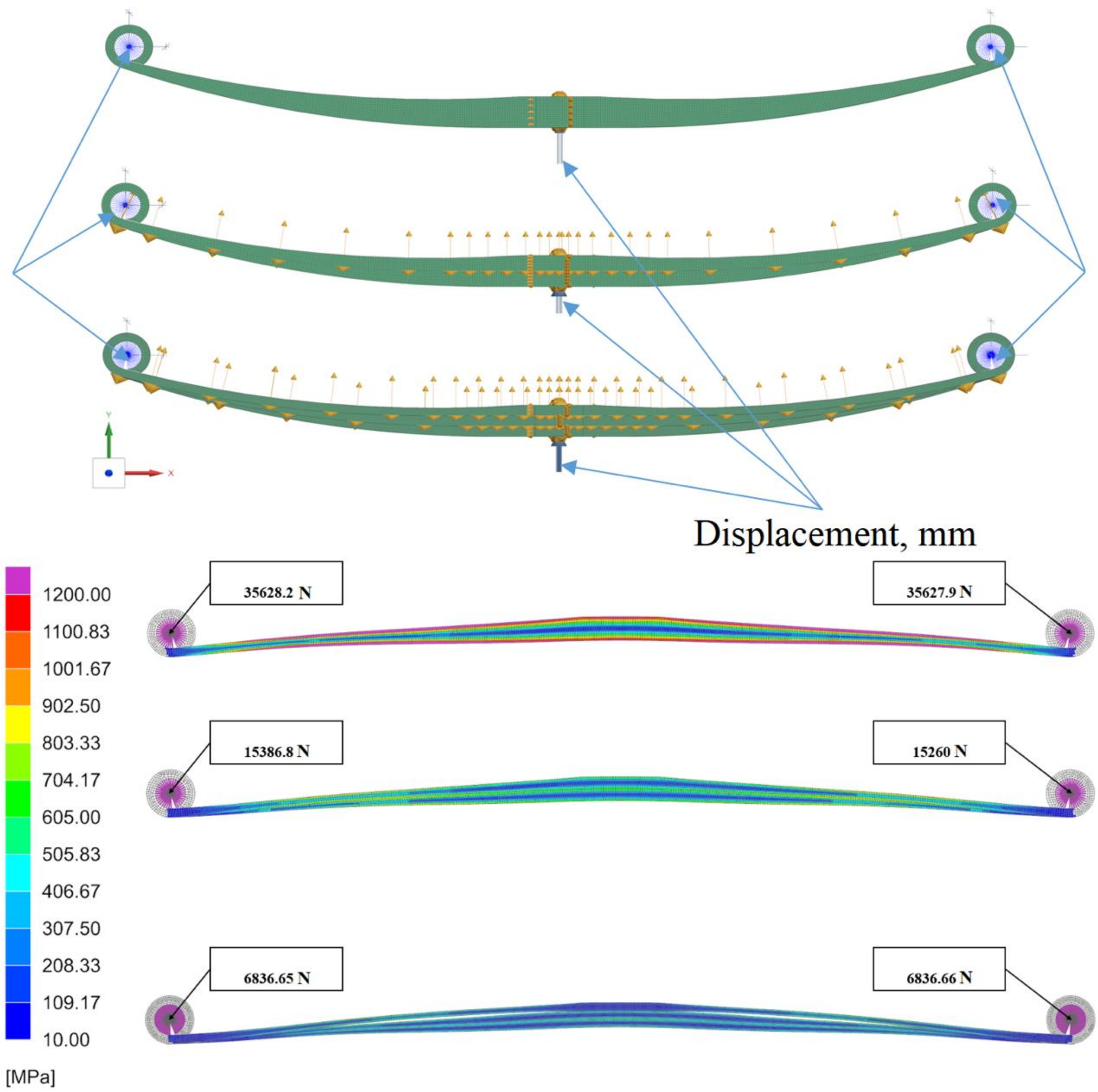

Figure 7. FEM models of leaf-springs and the results of their FEM durability calculations

The dependencies between stiffness and durability and geometry for the calculated leaf-spring types can be obtained as shown in **Figure 8**. With these dependencies composed using the developed leaf-spring profile synthesis method, we can select the most suitable configuration for the vehicle in question based on the weight and dynamic factor requirements, as well as other operating conditions. When a single-leaf-spring is used in the suspension, we can avoid interleaf friction. However, this reduces the dynamic factor and increases stress. The stiffness of a single-leaf-spring can be reduced to 30% compared to other types. When several leaves are used in the leaf-spring pack, there is interleaf friction, but the developed according to subsections 2.2 method helps reduce the leaf-spring stiffness.

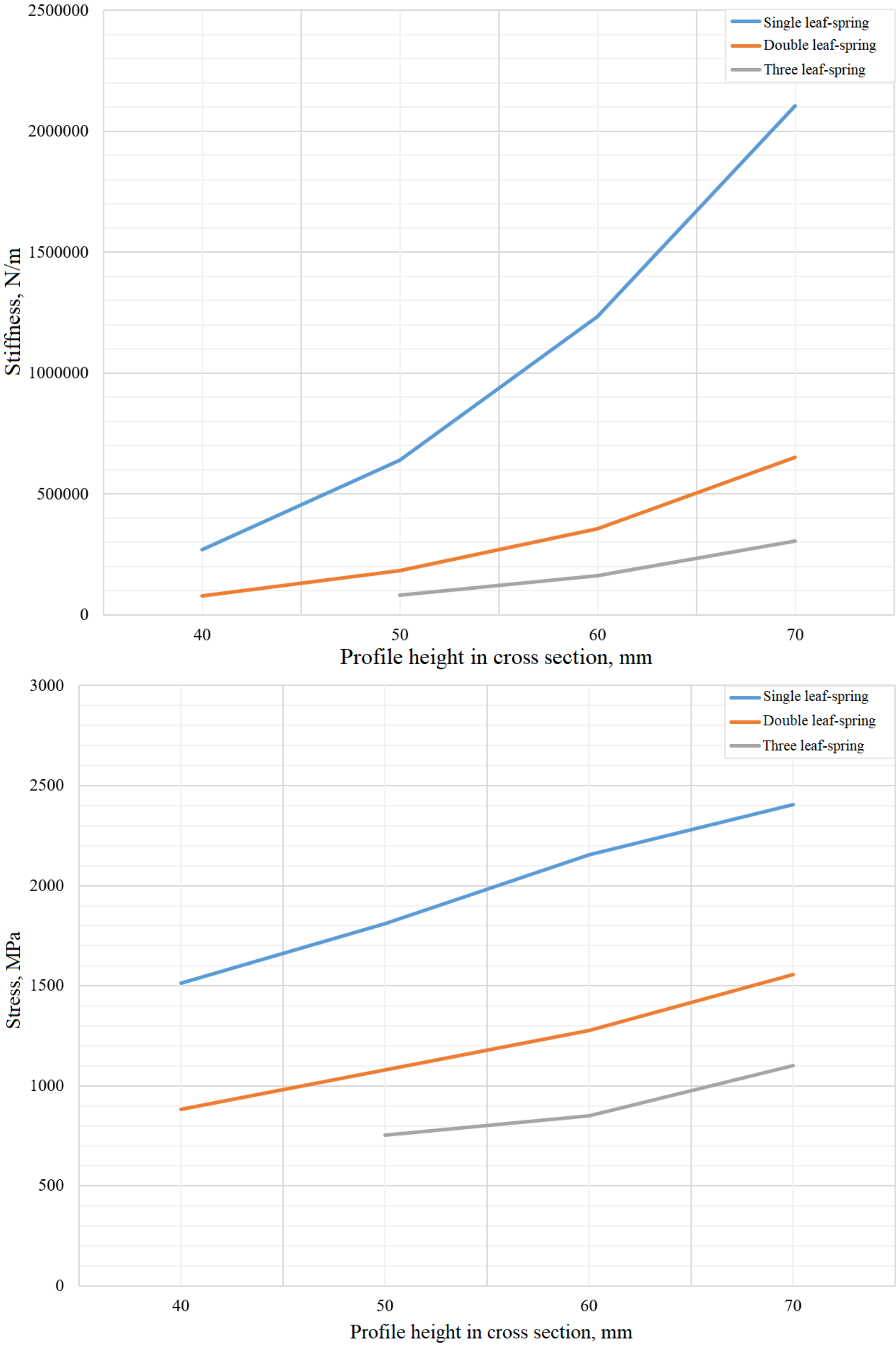

Figure 8. The dependencies between the stiffness and durability of few-leaf springs and the profile heights in the central cross-section

3.3. Methodology Efficiency Verification by the Vehicle Dynamics Simulation Results

To confirm the developed methodology efficiency, we performed the dynamics modeling using the simulation model of a vehicle in different weight modes [35]. The comparison was performed for the primary suspension standard configuration and for suspension system configurations synthesized in accordance with the developed methodology. We used these suspension parameters to compare the configurations in terms of comfort at different movement speeds on two road irregularities types: highway road and cobble road. The virtual modeling results for comfort tests are shown in Table 1 and Figure 9.

3.4. Methodology FEM Validation by Researching the Transverse Stabilizer Bars Stiffness

To confirm the analytical calculation correctness of transverse stabilizer bar stiffness, we performed FEM calculations [10], [11], [12], [27]. We used standard and randomly shaped stabilizer bars as the research objects. The stabilizer stiffness calculations results are shown in Figure 10.

The numerical values of transverse stabilizer stiffness calculated analytically using the developed method and the FEM are shown in Table 2. It can be seen that the discrepancies between the analytical calculations and the FEM results are below 5%, which means that the calculations are correct and the developed method for transverse stabilizer bar stiffness is functional. A modeling error within 5% is acceptable in vehicle suspension design processes.

3.5. Discussion

The results of applying the developed methodology for synthesizing the vehicle leaf-spring suspension characteristics to provide a high level of in-motion comfort demonstrate its significant contribution. It is known [5], [8], [28] that improving vehicle comfort indicators by 10% or more in various ways is always of scientific and

Table 1. The vehicle comfort improvement with suspension parameters synthesized using the developed methodology

Road type	Vehicle weight type	Vehicle comfort improvement
Highway road	Curb weight	up to 25%
	Total weight	up to 19%
Cobble road	Curb weight	up to 17%
	Total weight	up to 10%

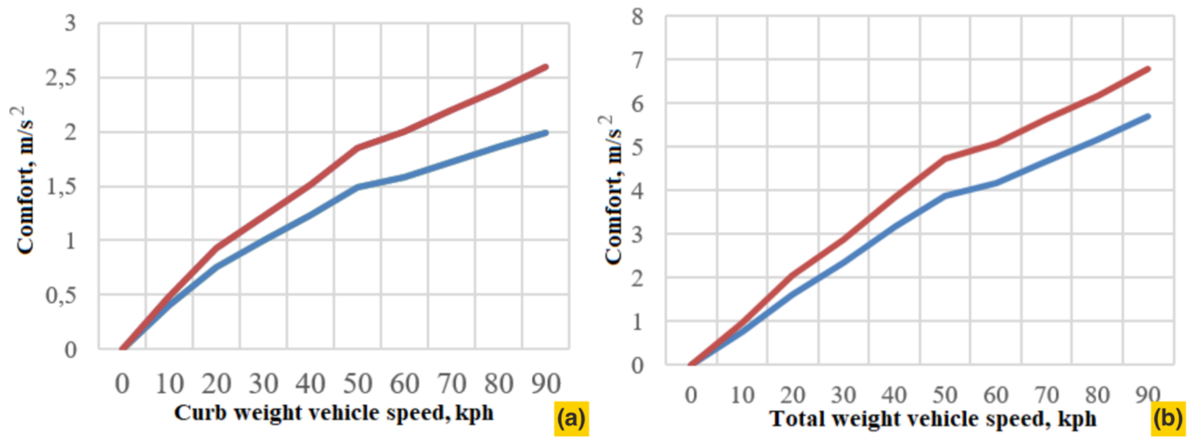

Figure 9. Comparison of vehicle comfort with a standard (red) suspension and a suspension generated using the developed methodology (blue): (a) on highway road; (b) on cobble road.

Table 2. The results of transverse stabilizer bar stiffness calculations.

	Analytical calculation using the developed method	FEM
The stiffness of a standard-shape stabilizer in Nm/°	203	211
The stiffness of a randomly shaped stabilizer in Nm/°	220	215

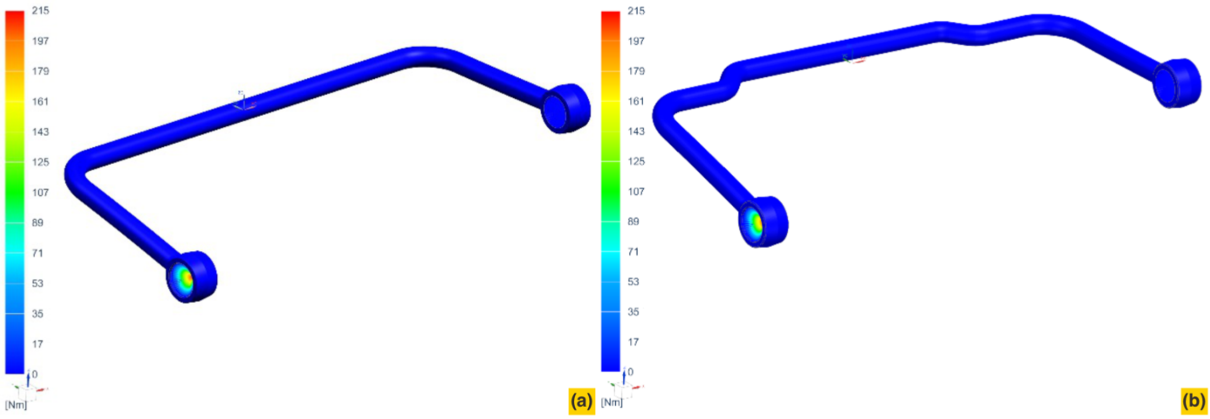

Figure 10. Stabilizer bars force factors distribution: (a) standard shape; (b) random shape

practical interest. The developed methodology corresponds to the current development level [4], [34], [35] and includes the use of both analytical calculation methods and the use of modern software and numerical methods [6], [24]. Meanwhile, the simulation results and the FEM validation accuracy meets the modern requirements of engineering calculations and modeling in software systems [5].

4. Conclusion

During this research, the authors developed a new leaf-spring suspension characteristics synthesis methodology for high-comfort vehicles. This includes the non-linear loading characteristic generation method for vehicle primary leaf-spring suspension based on its specifications and operating conditions. The authors proposed a new method to synthesize the leaf-spring optimal longitudinal profile geometry and determine the characteristics required to provide the high comfort and durability of the vehicle suspension system. They developed the new damping element loading characteristic generation method for the vehicle primary suspension system. The authors demonstrated an analytical randomly shaped transverse stabilizer stiffness calculation method. The discrepancies between the results obtained using the new methodology and the results obtained with the numerical calculations and FEM validation are below 5%, which confirms the applicability of the suggested methodology. Vehicle dynamics simulation methods were used to demonstrate that the vehicle in-motion comfort in question improved by 10-25% depending on the vehicle motion mode when the developed synthesis methodology was used compared to a primary leaf-spring suspension standard configuration. This fact emphasizes the significance of the developed methodology for future research in the field of vehicle suspension design.

Author's Declaration

Authors' contributions and responsibilities

The authors made substantial contributions to the conception and design of the study. The authors took responsibility for data analysis, interpretation and discussion of results. The authors read and approved the final manuscript.

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Availability of data and materials

All data are available from the authors.

Competing interests

The authors declare no competing interest.

Additional information

No additional information from the authors.

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