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Research on the Influence of the Mounting Configuration on the Elastic Characteristics and Energy Dissipation Capacity of Multi-Leaf Springs for Truck Vehicles

Wiesław Krason ^{1,*} , Paweł Bogusz ¹  and Józef Wysocki ²

¹ Institute of Mechanics and Computational Engineering, Military University of Technology, ul. Gen. Sylwestra Kaliskiego 2, 00-908 Warsaw, Poland; pawel.bogusz@wat.edu.pl

² Faculty of Security, Logistics and Management, Military University of Technology, ul. Gen. Sylwestra Kaliskiego 2, 00-908 Warsaw, Poland; jozef.wysocki@wat.edu.pl

* Correspondence: wieslaw.krason@wat.edu.pl

Abstract: This paper presents selected results of experimental tests on multi-leaf springs used in the dependent suspensions of trucks. The main objective of this research was to determine the dissipation energy in the actual spring structure for various methods of mounting the spring in the vehicle supporting system, with quasi-static vertical loading. Two basic methods of spring cooperation in a dependent suspension were tested: a free spring and a spring with a hanger. The experimental test conditions took into account different loading (unloading) speeds and changes in the angle of the hanger in relation to the frame and changes in its length. Based on the results of experimental tests, the elastic characteristics of springs with a hysteresis loop were developed. Load energy and dispersion values were calculated from the curves using the numerical integration method. The test results are presented in the form of elastic characteristics of springs with a hysteresis loop and comparative diagrams of load and dissipation energy. The original aspects of this work include the design, the construction of the stand and the methodology of experimental tests of the multi-leaf springs together with a comparative analysis of the test results and conclusions.

Keywords: dependent suspension; multi-leaf spring; hanger; experimental tests; damping; hysteresis loops; research methodology



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1. Introduction

The main problem in research on the dynamics of means of transport is the assessment of the level of inelastic resistance forces generated in a suspension. The phenomenon of the interaction of a road or railway vehicle with the road (rail) surface is an important design problem from the standpoint of mechanical and hydraulic solutions and geometric parameters of the vehicle suspension [1,2].

The geometric, elastic and damping properties of a suspension have a decisive influence on its dynamics during steady, curvilinear and accelerated (delayed) motion [3,4].

Improper selection of stiffness and damping causes an excessive increase in dynamic loads (acceleration amplitudes) of the vehicle [5,6], which negatively affects the comfort of transported passengers and loads, the smoothness of vehicle movement and, consequently, the effectiveness of its passive and active protection.

Selecting the appropriate stiffness and damping becomes particularly important for the suspension of trucks [7–9].

Multi-leaf springs [10,11], coil springs [4,7] and suspension mounting components, including hangers and other solutions [12], e.g., components for damping control and shaping selected suspension parameters [1,7,13], play an important role in shaping both the elastic and damping characteristics [14] of a truck suspension [15–17].

The influence of elastic characteristics on suspension performance is well documented in numerous works relating to multi-leaf springs [3,8,10,11,18]. Unfortunately, the impact of damping provided by multi-leaf springs is much less frequently determined, and the impact of this type of damping on suspensions and sprung vehicles, especially those intended for the transport of heavy loads (trucks, work machines and freight rail vehicles) is much less frequently studied. Dealing with suspensions of heavy goods vehicles is discussed, for example, in works [7,9,15]. There are technically no studies that discuss the impact of spring damping depending on the method of mounting in the suspension. The issue of using hangers cooperating in a suspension with multi-leaf springs is rarely described in the literature. They often have the nature of commercial guides [9], which unfortunately do not provide parameters regarding the hangers themselves, but it is also difficult to find the operational parameters of the complete suspension assemblies presented.

The issue of using additional equipment, including hangers, very rarely appears in the context of numerical analyses.

When analyzing the operating conditions of a multi-leaf spring, it can be clearly stated that they change continuously during operation [18–20]. In the initial state, there is usually a layer of graphite grease between the leaves, which is successively removed as the leaves rub against each other. Changing the lubricant layer causes a change in the cooperation conditions of the leaves, similar to dry friction. Additionally, a layer of oxides appears on the leaves' surface. Variable operating conditions of multi-leaf springs clearly influence changes in parameters describing the operation of the suspension, in particular, friction and drivers' working conditions [20].

Paper [21] presents a multi-aspect research problem with emergency transport of high-risk newborns on a typical route between primary and secondary care hospitals. Experimental research was conducted to assess the impact of mechanical vibrations and vibration shocks on the newborn's entire body. It was found that, regardless of the type of road, the daily vibration dose limits (VDVs—vibration dose values) were exceeded for all measurements. It has been proven that the suspension system of the ambulance and the equipment for transporting newborns should be radically redesigned by introducing vibration damping in the bands from 1 to 3 Hz and intensively damping vibrations from road unevenness and vehicle suspension in the range from 5 to 18 Hz.

Also noteworthy is paper [22], presenting the results of experimental research, including three-axial accelerations on the driver's seat and vertical accelerations on the floor of the operator's cabin, as well as vehicle speeds and trajectories during a typical work shift, on exposure to vibrations of a group of drivers of various types trucks equipped with lifting equipment. The results were analyzed in accordance with the ISO standard [23,24]. In all analyzed work cycles, the daily vibration limit (VDV) was exceeded, and impulsive vibrations constituted frequency dominants. In such situations, the working conditions of drivers/operators of lifting equipment should be radically changed by introducing damping that will limit the effective values of acceleration amplitudes (RMS—root mean square) along the x, y and z axes.

In work [19], a computational model developed using commercial software was used to analyze the dynamics of the rear suspension system of a passenger car (weight approx. 1t). Selected vibration parameters for suspension modeling were determined experimentally. Simulation tests on the dynamics of the rear suspension, in which the time courses of vertical accelerations and their frequency characteristics were calculated, were used to assess passenger driving comfort, and the daily vibration exposure (VDV) was determined. It was found that a small change in the damping coefficient significantly affects the dynamic loads of the suspension and causes an intensive increase in the VDV index.

During operation, suspension parts are subject to wear or accidental failures due to excessive dynamic loads. There is an urgent need to identify damage during routine inspections and inspections of the technical condition of suspension. Some of them are easy to diagnose, e.g., broken or corroded spring leaves, cracked springs, oil leakage from the hydraulic shock absorber or other visible mechanical damage. Sometimes we are dealing

with hidden failures and their identification is only possible during experimental tests at specialized laboratory stations [25,26], which enables the determination of the current elastic or damping characteristics of the failed suspension assemblies (parts). For this purpose, test stands are most often used, e.g., for elastic or damping components having standard characteristics. The actual characteristics of the tested suspension element are compared with the reference characteristics.

In the reviewed source literature related to elastic elements and dampening suspensions of road (rail) transport, various approaches resulting from current needs are represented [27,28].

For example, in work [29], a neural network was used to detect damage to a hydraulic shock absorber in the suspension of a rail vehicle. The impact of changing damping on the dynamics of a rail vehicle was investigated. The values of acceleration signals recorded during homologation tests were introduced as input values. Condition monitoring accuracy was achieved at a level of less than 63%. With the current, advanced designs of rail vehicle suspensions, the obtained effects of technical condition monitoring are ineffective and require further supplementation to improve the accuracy of diagnosis. The authors confirm that further research on the use of neural networks to detect technical failures in suspensions should focus on expanding the set of potential faults, various vehicle load states, different speeds and modifications of the neural network. It should be emphasized that the question of using neural networks in diagnosing the technical condition of mechanical devices, including suspension systems of various means of transport, is very topical and raises great interest among research teams.

There are few works dealing with the issue of mounting springs in suspensions. The few works found by query, e.g., [3,5], concern research issues indicating the importance of spring mounting from a practical, operational point of view, e.g., determining the impact of friction on the operational parameters of the suspension [25], describing the strength and deformations of the most stressed suspension components [10,20] and components of the vehicle's supporting structure cooperating with the suspension components [10,28]. Most often, these are works in which the results of numerical tests of separate springs [10,30] or complete load-bearing components of trucks are discussed [26,30]. Some works describe bench tests, the results of which were used to determine the characteristics of elastic suspension components of various types of vehicles [18,20] and to validate and assess the quality of the numerical models used [10].

There are technically no works that discuss the influence of spring damping on the operation of the suspension of vehicles transporting heavy loads. No works have been found that discussed experimental research on the damping properties of multi-leaf springs for trucks cooperating with hangers and indicated the possibility of shaping the damping amount by selecting different lengths of hangers and the influence of their positioning in relation to the spring leaves, e.g., by changing the inclination angles of the hanger in an unladen truck suspension.

The possibility of increasing suspension damping in the case of railway bogies [29] and freight wagons for transporting heavy loads at high speeds (above 100 km/h) becomes particularly important. Railway industry companies are trying to modernize standard bogies in order to adapt their structure to high speeds.

The authors draw attention to the possibility of using multi-leaf springs to develop such suspensions. In this type of structure, the internal damping of the spring itself and energy dissipation between the individual leaves can be used [20]. There is no answer in the available literature as to what effect the method of mounting springs in the vehicle suspension has on the energy dissipation in motion.

Therefore, this study presents the results of laboratory bench tests and selected results of numerical simulations for two basic methods of mounting multi-leaf springs in a vehicle's dependent suspension in order to determine the elastic characteristics and dissipation energy. A free spring and a spring with a hanger were tested. The tests were carried out for selected loading (unloading) speeds, three hanger lengths and appropriate angles in

relation to the frame. The free spring was tested with vertical quasi-static and impulse excitation. However, the spring with a hanger with vertical forcing was only considered. The test results were presented in the form of elastic characteristics of springs with a hysteresis loop and comparative diagrams of load and dissipation energy.

2. Tests of Freely Mounted Multi-Leaf Springs

The subject of this research is a double-leaf spring designed for the rear suspension of a truck with a maximum permissible weight of 3.5 t. This is a prototype spring designed specifically for laboratory testing and not utilized in any vehicle applications. It consists of a four-leaf main spring and a two-leaf auxiliary spring. This design solution is a parallel connection of elastic elements (with different geometries), which enables obtaining bilinear elastic characteristics. A schematic diagram of free mounting and loading of the tested multi-leaf spring and a view of the unloaded spring in the guide rail are presented in Figure 1.

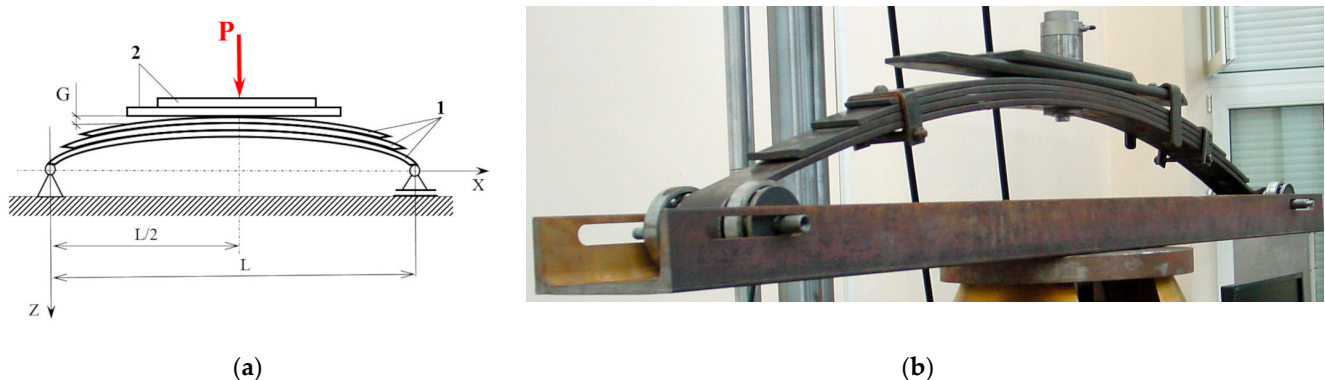


Figure 1. The concept of free mounting of a double-leaf spring and the implementation of the impact of external loads that change over time: (a) diagram of free mounting and loading (without hanger); (b) view of the spring freely supported in the guide rail.

In the presented stage of research on the influence of the method of mounting a multi-leaf bilinear spring on its characteristics, the spring and station/test setup/testing bed shown in Figures 1 and 2 were used. In bench tests with the same spring mounted freely, its current elastic characteristics and energy dissipation were determined. For this purpose, a laboratory stand for free mounting of a multi-leaf spring was used, as presented in Figure 2. For weighting the quasi-static double-leaf spring, a testing machine with a vertical loading capacity of 1200 kN was used (Figure 2a).

Two series of tests of a freely supported multi-leaf spring were performed to confirm the repeatability of the results. These series and the corresponding results are marked as follows: A and B. In both series, the conditions for carrying out comparative tests with free mounting of the ends of the double-leaf spring leaves were identical. The tested spring (Figures 1 and 2) consists of a four-leaf main spring (Figure 2(1)) and a two-leaf auxiliary spring (Figure 2(2)). The specificity/characteristic of this design solution is the presence of play/clearance/tolerance (Figure 1a, marking G) between the main and auxiliary springs [18]. The spring can move on the surface of the guide channel (Figures 1b and 2(3)), on four ball bearings (Figure 2(4)). A U-section with appropriate longitudinal milled spots on the side walls allows for mounting and guiding both ends of the spring in variable straight-line movement.

The main goal of the experimental studies with the free mounting of the double-leaf spring presented in this part was to demonstrate the influence of loading rate (quasi-static speed range) on the behavior of the elastic characteristics and to estimate the amount of energy losses during a single load–unload cycle of the spring. For experimental tests, a testing machine with a hydraulic drive was used, as shown in Figure 3a, which implemented a

constant speed of loading/unloading the double-leaf spring (Figure 1) for one forcing cycle. The tests assumed three different loading rates: 50, 100 and 150 mm/min. Hysteresis loops were obtained for such loading velocities in the “load–unload” cycles of the double-leaf spring. Nominally, the maximum force value was set at 8 kN. The conditions for carrying out comparative tests are described in detail in Table 1. The following loading conditions were adopted during tests: loading rate and maximum deflection of the tested spring.

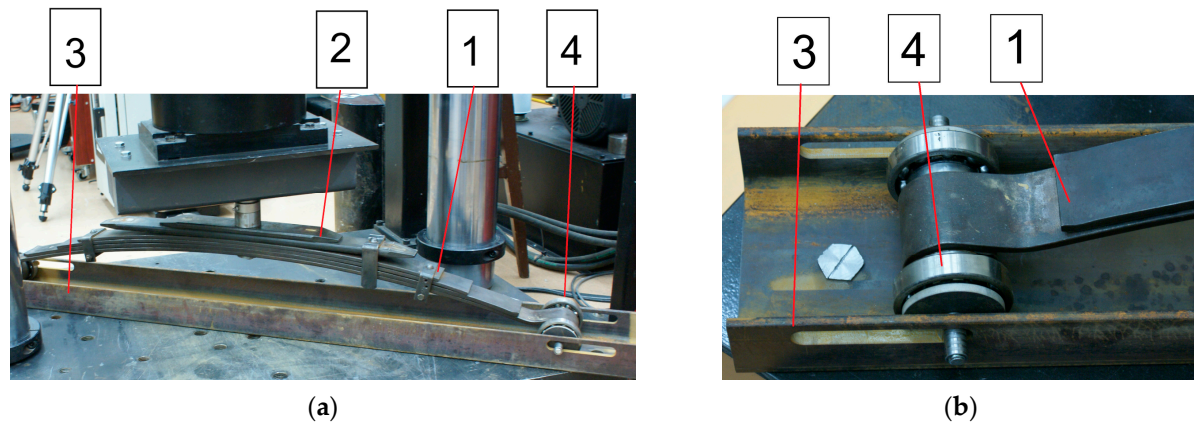


Figure 2. The method of free mounting of the ends of a leaf spring in quasi-static loading/unloading tests: (a) view of the spring test stand loaded with a head with a force sensor on a testing machine; (b) enlarged view of the support of one free end: 1—main spring; 2—secondary spring; 3—guide channel; 4—ball bearings.

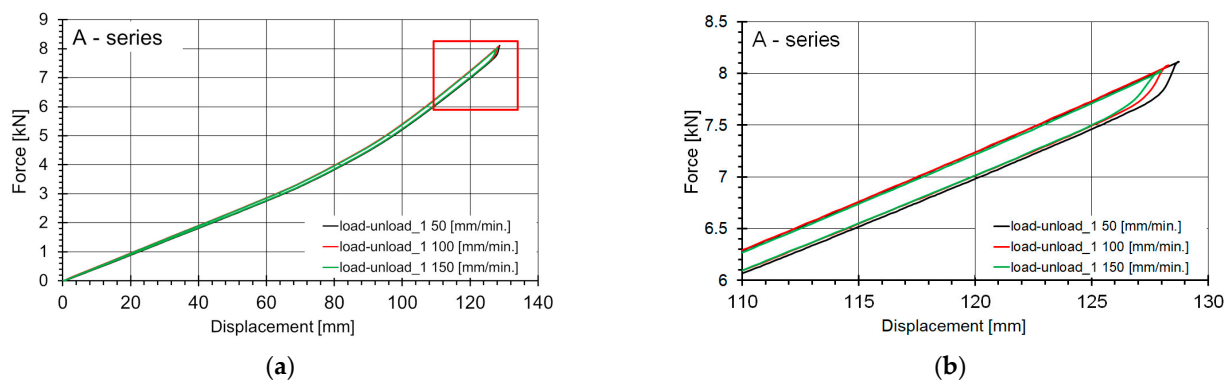


Figure 3. Summary of elastic characteristics of the prototype spring for tests with different loading/unloading rate values—series A results: (a) complete curves; (b) close-up of the final segments.

Table 1. Basic load parameters and measurement sampling frequency for a single quasi-static “load/unload” cycle of the free spring [9].

Test No.	Load Parameters			Sampling Frequency [Hz]
	Loading Rate [mm/min]	Maximum Deflection [mm]	Peak Force [kN]	
1	50	128.0	8.04	10
2	100	128.0	8.05	10
3	150	128.0	8.03	10

Based on the results of the bench tests, force–displacement diagrams were prepared in which hysteresis loops were identified for individual tests. Figures 3 and 4 show force–displacement diagrams developed on the basis of loading/unloading tests of the double-leaf spring with the tested excitation velocities.

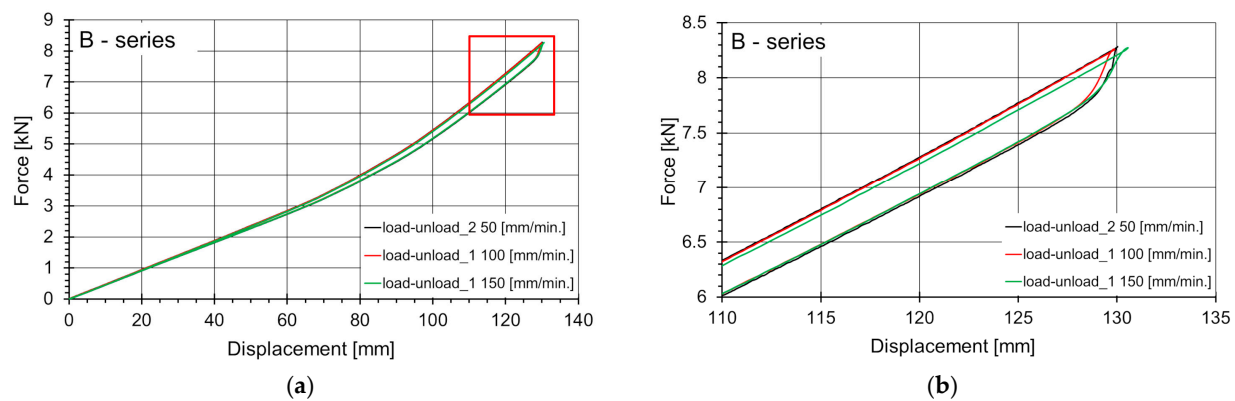


Figure 4. Summary of elastic characteristics of the prototype spring for tests with different loading/unloading rate values—series B results: (a) complete curves; (b) close-up of the final segments.

Figure 3 presents force–displacement diagrams prepared on the basis of the results obtained in series A; Figure 4 presents diagrams prepared on the basis of the results obtained in series B.

Table 2 presents a summary of detailed results of quasi-static tests of archived and updated tests, based on the example of the values of maximum load forces and the corresponding maximum displacements for three different loading/unloading speed values. The spring loading energies, the energies dissipated during spring unloading and the corresponding relative shares of the dissipated energy in the spring loading energy are also included in the table. The results of historical tests obtained for various loading speed values are given in Table 2 in normal italic font, and in the Item column, they are marked as follows:

- 1A—speed equal to 10 mm/min;
- 2A—speed equal to 50 mm/min;
- 3A—speed equal to 100 mm/min;
- 4A—speed equal to 150 mm/min.

Table 2. List of parameters for quasi-static tests of loading a freely mounted spring, developed on the basis of the results of test series A and B.

Item	Description	Cross-Beam Piston Speed	Max. Force	Max. Displacement	Spring Load Energy	Energy Dissipated When the Spring Is Unloaded	Share of Dissipated Energy in Relation to the Load Energy
		[mm/min.]	[kN]	[mm]	[J]	[J]	[%]
1A	Pressing	10	8.38	131.2	462.4	-	-
2A	Single load–unload cycle	50	8.11	128.7	439.8	17.20	3.9
2B		50	8.30	130.1	453.5	20.43	4.5
2B		50	8.28	130.0	451.7	21.11	4.7
3A		100	8.08	128.3	434.8	14.40	3.3
3B		100	8.26	129.9	449.2	17.84	4.0
3B		100	8.27	129.8	449.8	18.07	4.0
4A		150	8.03	128.0	431.6	12.00	2.8
4B		150	8.27	130.6	451.7	15.42	3.4
4B		150	8.30	130.6	453.2	14.59	3.2

The test results in series B obtained for three different loading speeds are given in Table 2 in a simple bold font, and in the No. column, the following symbols are used: 2—speed equal to 50 mm/min; 3—speed equal to 100 mm/min; 4—speed equal to 150 mm/min. Updated tests for the freely mounted spring were performed twice for the same loading/unloading conditions. Therefore, all results reported in Table 2 for these cases are double.

Based on the comparison of the diagrams presented in Figures 3 and 4, a slight enlargement of the hysteresis loop from the updated tests shown in Figure 4 was observed. This observation is confirmed by the analysis of detailed data from the comparison of results presented in Table 2.

Updated quasi-static tests showed the following:

- Several percent share of dissipated energy in relation to the spring load energy,
- A slight percentage decrease in dissipated energy as the loading rate increases,
- Energy dissipation is increased with larger spring deflection arrows. For deflection of 60 mm, the value of dissipated energy can be neglected.
- No significant impact of the loading rate of 50–150 mm/min on the hysteresis loops.
- Further findings are as follows:
- Differences in the final loop courses and energy values between the archived and updated tests may result from the difference in the spring deflection, which in test A was about 2 mm smaller than in B.
- The shapes of the hysteresis loop in measurements A and B are very similar throughout the course.
- The repeatability of results recorded for the same object in identical laboratory conditions of site tests was confirmed.

3. Tests of a Multi-Leaf Spring Mounted with a Hanger

Laboratory tests were performed on a double-leaf spring (Figure 1) mounted using brackets with a hanger. In the tests, the length of the hanger and its initial setting in relation to the main spring were changed. This approach examined the impact of different configurations of multi-leaf spring supports in the design of dependent suspensions for trucks. The elastic characteristics and energy dissipation in the process of quasi-static loading and unloading of the spring with vertical force were determined. The same test conditions were adopted as for the free spring (as shown in Section 2).

3.1. Test Setup for Quasi-Static Testing of the Spring with a Hanger

To carry out experimental tests of a multi-leaf double-leaf spring (Figure 1) with a hanger, a prototype laboratory stand was built with the original design for mounting on a testing machine. One end is fixed to the bracket with a cylindrical joint, and the other end is connected to the hanger with a double cylindrical joint. The result is a specific crank system consisting of a spring and a hanger. This design solution allows for various spring mounting configurations. Details are presented in Figure 5.

The base of the stand (Figures 2 and 5) was made of a C250 channel with appropriate holes drilled through its side walls. Two mounting holes (6) were drilled in the node for fixed mounting of the bracket to the base of the testing bed, and three holes were drilled in the mobile node of the testing bed to fasten the hanger plates with the base channel (7). The holes (7) are placed at equal distances $\Delta x = 20$ mm from each other, in a horizontal row next to each other. The fixed bracket, placed on the left side of the stand (Figure 5), consists of two plates installed vertically in relation to the base of the stand (2). Three holes (8) were drilled in each plate of the fixed support, placed at equal distances of $\Delta z = 20$ mm (Figure 6), one above the other, for mounting the lower leaf of the double spring (1). The holes (8) are located in one vertical row above the two holes (6) for connecting to the base (2). Similarly, three holes (9) were drilled in both plates (4) of the movable hanger, placed at equal distances Δz (Figure 6), one above the other, and used to attach the other end of the lower leaf of the spring (1).

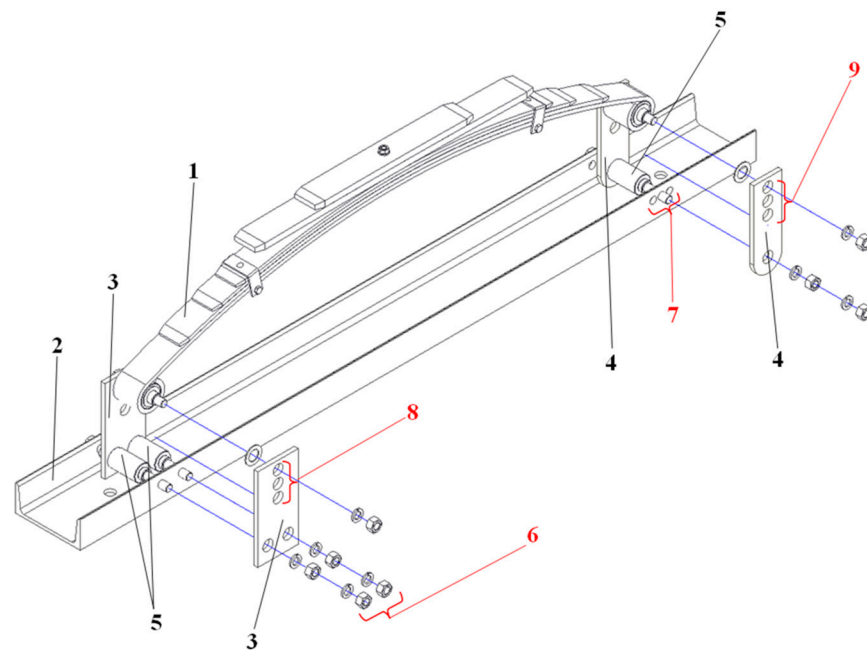


Figure 5. Testing bed for laboratory tests of dependent suspensions equipped with a multi-leaf spring and hanger: 1—main spring; 2—stand base; 3—spring support with a cylindrical joint; 4—spring hanger (with a double cylindrical joint); 5—spacer sleeves; 6—screw connections; 7—three horizontal holes; 8 (9)—vertical holes in the bracket (hanger).

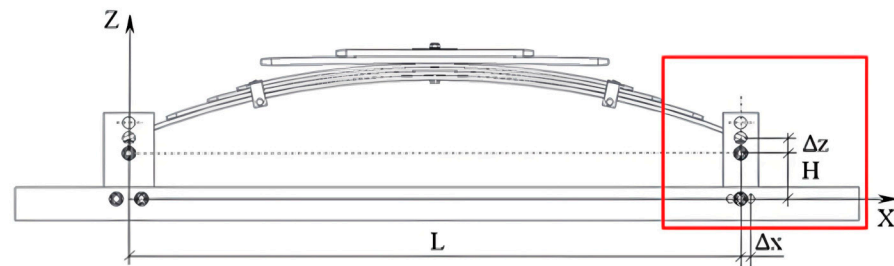


Figure 6. Diagram explaining changes in the configuration of mounting the double-leaf spring on the stand using a hanger. The movable mount, highlighted with a red square, is presented in Figure 7.

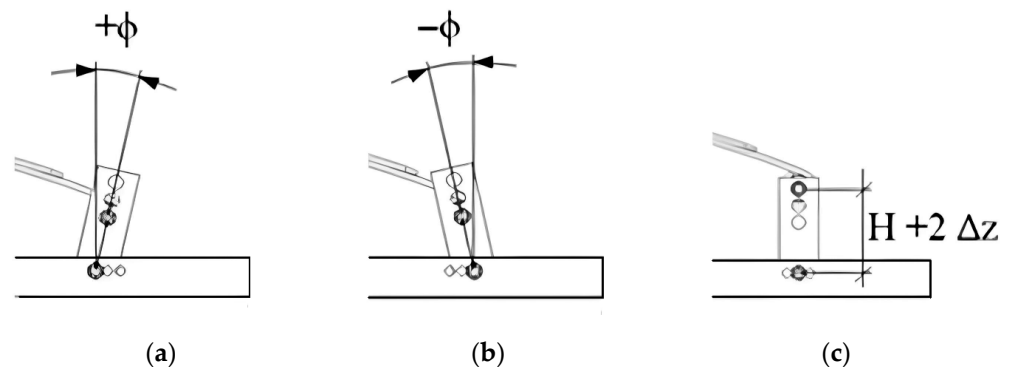


Figure 7. Diagram explaining changes in the configuration of mounting the double-leaf spring on the stand using a hanger: (a,b) change in angle ($\pm\phi$) initial deflection of the hanger by changing the position of the rotation axis (X); (c) changing the length (H) of the hanger.

The holes (9) are located in one vertical row above the single hole (7) for connecting the hanger with the base (2). By mounting the spring at different levels of holes (8) and (9), the height and position of the spring in relation to the hanger mounting points in the C-section

were changed. Mounting the spring at different levels of holes (8) and (9) changed its height and position relative to the hanger mounting points in the C-section. This design solution for mounting the components in the hanger movable node (on the right side of the stand base in Figures 5 and 6) enables independent rotations of the hanger plates (4) in relation to the right end of the spring and in relation to the stand base in the process of loading the spring. The movable mount is highlighted with a red square, with close-ups in different configurations shown in Figure 7.

The design solutions for mounting the leaf spring presented in Figure 5 with the testing bed base via a bracket and a hanger enable three configurations of the initial positions of the hanger in relation to the vertical.

A row of three horizontal holes (Figures 5 and 6) in the right bracket allows you to adjust the angle of the hanger relative to the vertical. Setting the screw connection between the hanger and the base in the middle hole (7) allows the initial vertical positioning of the hanger ($\phi = 0$), as shown in Figure 7c. The remaining hanger settings are illustrated in Figure 7a,b. The length of the hanger can be changed by adjusting the position of the spring end screw connections in the vertical holes (8 and 9 in Figure 5). It is possible to change the length of the hanger in the range from $H_{min} = H$ to $H_{max} = H + 2\Delta z$ (see Figure 7c). The location of the hanger rotation axis is defined by the coordinate $X = L \pm \Delta x$ (Figure 6). Nine spring mounting combinations can be realized in the designed test setup.

3.2. Laboratory Research Methodology

Experimental tests of the double-leaf spring with a hanger were performed at the laboratory station presented in Figure 8. Its construction and principle of operation are described in detail in Section 3.1 of the present study. A testing machine Instron Satec (Illinois Tool Works Inc., Norwood, MA, USA) with a permissible vertical load of 1200 kN was used to load the quasi-static spring [31]. The base of the stand (2 in Figure 8), with a hanger (3) and a bracket (4), was mounted to the base (5) of the testing machine using two bolts. The spring was loaded at a constant rate up to a maximum force of 8 kN, with a vertical force applied at the midpoint of the spring, at half of its length, using a movable transverse beam (6 in Figure 8) equipped with a force head.

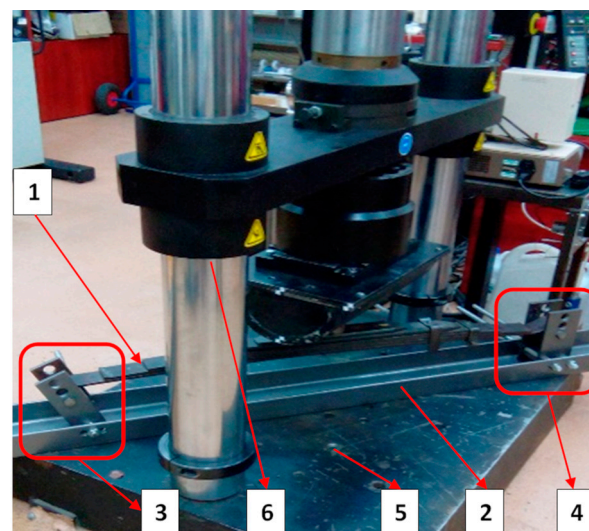


Figure 8. Double-leaf spring test stand with a hanger mounted on a testing machine during the trial quasi-static loading/unloading: 1—double-leaf spring (deformed) in the final phase of loading; 2—base of the spring mounting setup; 3—movable spring mounting node with a displaced hanger; 4—bracket; 5—base of the testing machine with mounting hole; 6—column of the testing machine with a movable transverse beam and loading head.

In laboratory tests of a free double-leaf spring, a very small influence of the velocity of quasi-static vertical loading on the course of its elastic characteristics (hysteresis loops) was observed. Based on this conclusion in double-leaf spring tests with the hanger, the same vertical loading speed was assumed, i.e., 100 mm/min. A maximum deflection of 130 mm was assumed.

The main goal of the presented experimental research on the rear dependent suspension equipped with a double-leaf spring and a hanger was to demonstrate the impact of changing the hanger configuration (angle and length) on the elastic characteristics and to estimate the amount of energy loss during each spring load–unload cycle, at a fixed loading rate.

The functional and kinematic characteristics and stiffness changes of the suspension system equipped with a double-leaf spring a hanger were examined in the range of quasi-static forcing, with the inclination angle of the initial hanger position being changed in various positions: front (Figure 7a), neutral (vertical initial hanger position (Figure 7c) and in the rear deflection position (Figure 7b). Similar tests were also performed for three different hanger lengths (Figure 7). The tests were affected by time-varying external loads from $P(t) = 0$ to $P(t) = P_{\max}$ (Figure 1a).

The designations of the test configurations and settings of the initial deflection of the hanger in the spring mounting movable node (Figures 3 and 8), used in the individual test variants, are explained in Figure 9. The example test configuration shown in Figure 9 is marked in the results as variant KN/P_II.

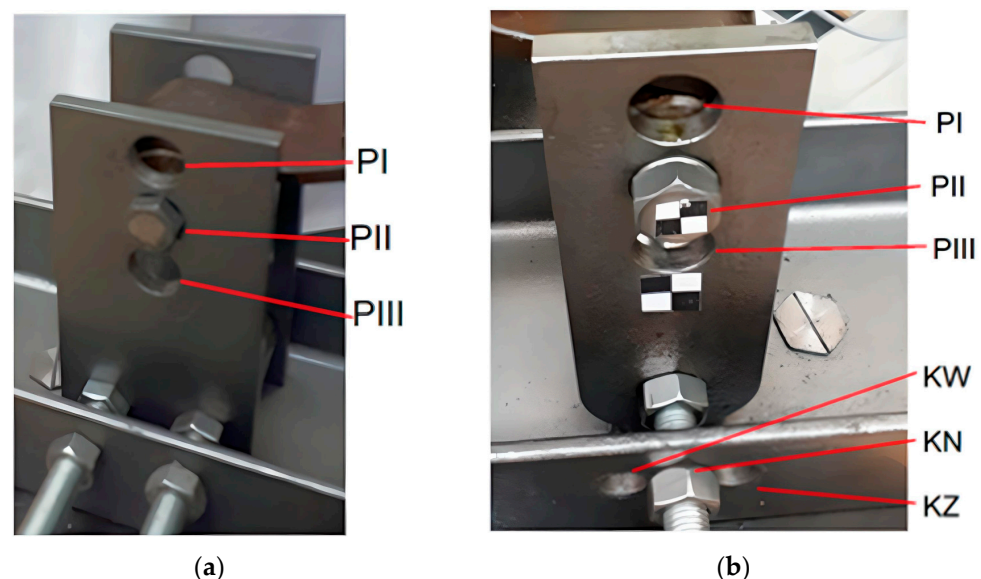


Figure 9. Markings of test configurations and initial hanger deflection settings: (a) spring bracket for variants with different hanger lengths (H); (b) spring hanger for variants with different lengths (H) and angle (ϕ) initial deflections.

The PI, PII and PIII designations shown in Figure 9a,b correspond to test variants with three hanger lengths ranging from $H_{\min} = H$ to $H_{\max} = H + 2\Delta z$ (Figure 7).

The markings KW, KN and KZ in Figure 9b define the configuration of mounting the spring in the movable node for variants with different angle ϕ initial deflections of the hanger, corresponding to different positions of the hanger rotation axis defined by coordinates in the range $(L - \Delta x) \leq x < (L + \Delta x)$ (Figure 7).

The values of parameters describing the configuration of test hanger lengths and initial hanger deflection settings are listed in Table 3.

Table 3. List of parameter values describing test configurations corresponding to individual variants of bench tests of the spring with a hanger (marked in Figure 7).

Configuration Designation	Parameter	Value [mm]
I	$H_{\max} = H + 2\Delta z$	150
II	$H_{II} = H + \Delta z$	130
III	$H_{\min} = H$	110
KW	$X = L - \Delta x$ (angle ϕ)	1200
KN	$X = L$ (angle $\phi = 0$)	1220
KZ	$X = L + \Delta x$ (angle $-\phi$)	1240

The height (H) variants can be freely associated with the ϕ variants of the initial deflection angles of the hanger. In total, this gives nine variants of setting the spring mounted via a hanger (Table 3).

3.3. The Influence of Mounting a Multi-Leaf Double-Leaf Spring on Elastic Characteristics

Multi-variant quasi-static bench tests were performed on the loading–unloading cycle of a double-leaf spring mounted via a bracket with a hanger in the configurations described in Sections 3.1 and 3.2.

Based on the results of measurements of the vertical compressive force and displacements of the head loading the spring, elastic characteristics were developed. Figures 10–13 compare the obtained graphs for selected configurations of spring mounting using a bracket and a hanger. In the legends of each graph, the name of the tested configuration is provided according to the description in Table 3. The curves for the measurements of the spring at levels I, II and III are presented.

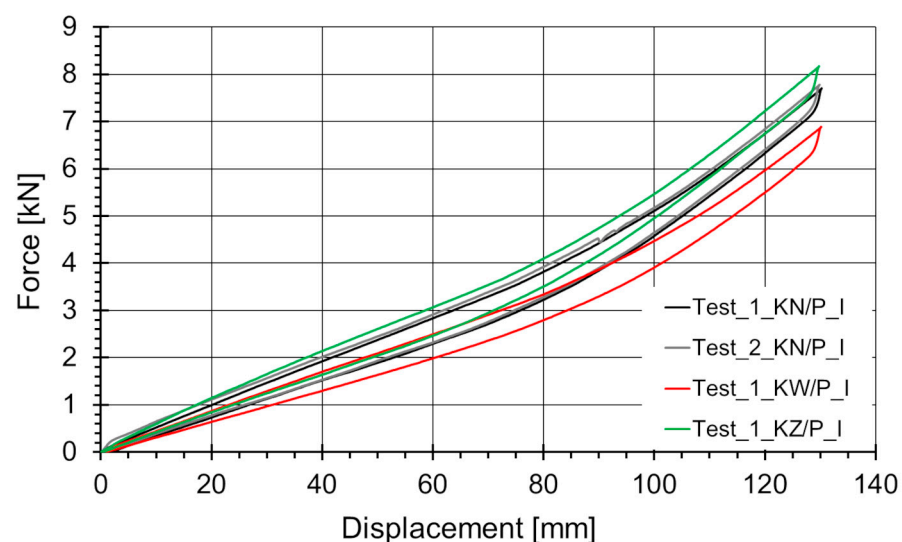


Figure 10. Elastic characteristics of a double-leaf spring with a hanger with the greatest length $H_{\max}$ in variants of different angle (ϕ) initial deflections.

Figure 10 presents the result diagrams of loading/unloading tests of a double-leaf spring mounted with a hanger of length $H_{\max}$ (configuration I), for all variants of setting the angle of initial deflection of the hanger, i.e., $<-\phi, 0, \phi>$, which corresponds to the KW, KN, KZ configurations (Figure 7).

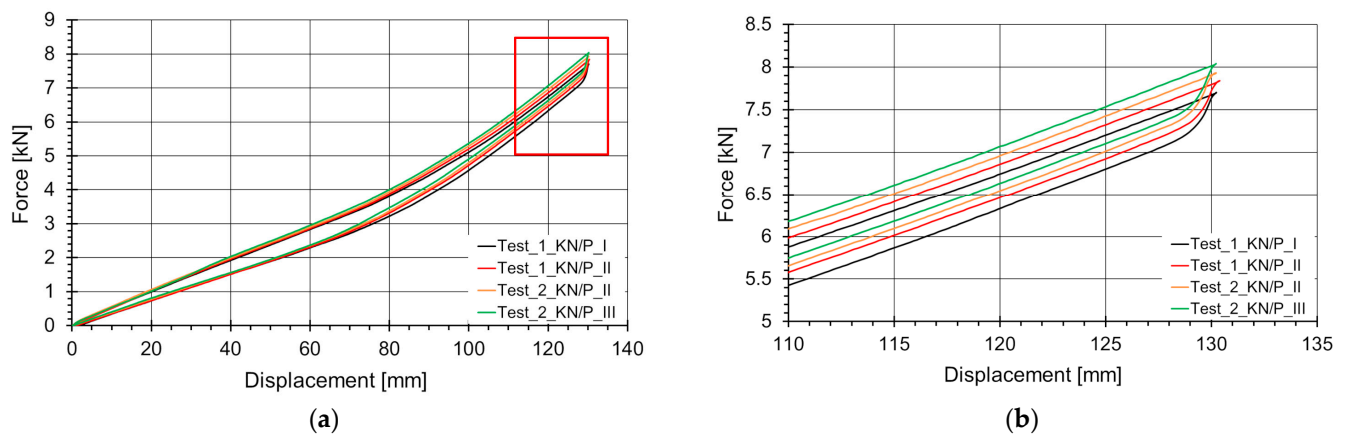


Figure 11. Spring characteristics of the double-leaf spring for various hanger lengths changed from Hmin to Hmax, with the angle of inclination $\phi = 0$ (Figure 7c): (a) complete curves; (b) close-up of the final segments.

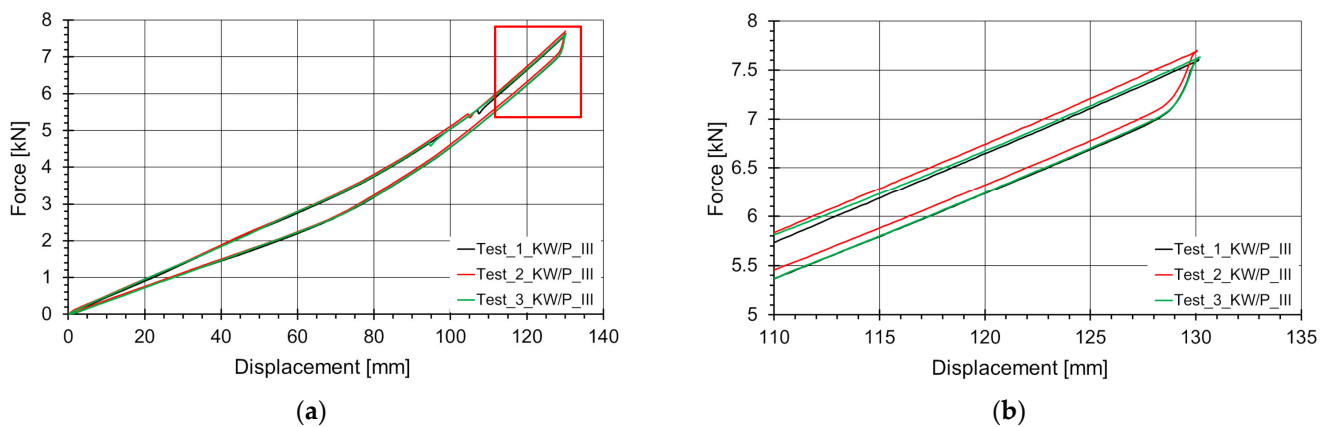


Figure 12. Elastic characteristics of a double-leaf spring for a hanger with the shortest length Hmin, in a fixed initial deflection of the hanger and a constant angle of $+\phi$ (Figure 7a): (a) complete curves; (b) close-up of the final segments.

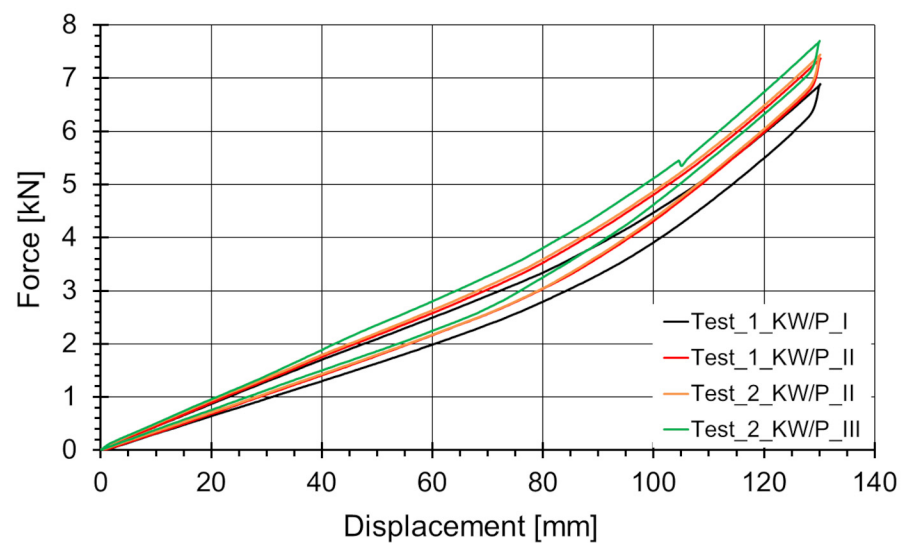


Figure 13. Elastic characteristics of the double-leaf spring for various hanger lengths from Hmin to Hmax in a fixed variant of the initial deflection setting with a constant angle $+\phi$ (Figure 7a).

The middle diagram (black, Figure 10) presents two tests for the KN/P_I variant, which illustrate the high repeatability of the experiment.

For a negative initial deflection angle ($-\phi$ for the KZ configuration), an elastic characteristic with increased stiffness was obtained (the hysteresis loop is located at the highest). Changing the hanger position for the KN and KW configuration variants reduces the stiffness of the elastic characteristics almost in the entire range of spring deflections (Figure 10).

Figure 11 shows the diagrams of the elastic characteristics (hysteresis loops) of the double-leaf spring for various hanger lengths, varying from H_{min} to H_{max} (configurations III-I), in the neutral variant of setting the initial angle of the hanger $\phi = 0$ (hanger positioned vertically, Figure 7c). Changing the length of the hanger in a perpendicular position has little effect on the hysteresis loop cycle. It was found that level III (the shortest hanger) has a minimally raised hysteresis loop and a higher maximum force than the other configurations.

The results of three hysteresis loops of a double-leaf spring mounted with hangers of the shortest length H_{min} in the variant of setting the initial deflection of the hanger with an angle ϕ (Figure 7a) are presented in Figure 12. The elastic characteristics differ in the course of the hysteresis loop, and the value of the maximum force is slightly below 0.5%. It can be assumed that they are identical. They illustrate the very good repeatability of the experiment. The high consistency of the course of subsequent loads also proves that the cyclic loading tests took place in the elastic range. The spring and hanger mounting did not undergo plastic deformation.

It is also worth noting that the maximum relative differences in the maximum load obtained at the maximum spring displacements for the elastic characteristics (hysteresis loops) presented in Figures 11 and 12 are less than 4.5%.

In the next stage, the test results of the double-leaf spring in variable hanger length configurations ranging from H_{min} to H_{max} were analyzed.

In the comparison presented in Figure 13, the initial angle of the hanger is constant and even ϕ (Figure 7a). Similarly to the previous analyses, in the example of configuration II, the repeatability of tests with identical configurations is presented. The hysteresis loop of test III (Figures 7 and 9, Table 3), corresponding to the length H_{min} of the hanger, is the highest. Increasing the length of the hanger resulted in a gradual lowering of the load loop and, consequently, the value of the recorded maximum force. This relation coincides with the analysis presented in Figure 11.

The greatest influence on the changes in the elastic characteristics of the double-leaf spring with a hanger, obtained in quasi-static cyclic load tests, is exhibited by the initial setting angle of the hanger (from $-\phi$ to $+\phi$). This relation, for example, based on the longest length of the hanger equal to $H_{max} = H + 2\Delta z$, is illustrated in Figure 10. The maximum relative difference in the maximum load force obtained for configuration I of the tested spring support is approximately 18.5%.

The change in the length of the hanger in the tested range from H_{min} to H_{max} has a slightly lower impact on changes in elastic characteristics. This is illustrated in Figure 13, and the results of the hysteresis loop with a constant setting of the initial hanger deflection angle equal to $+\phi$ are analyzed. The maximum relative difference in the maximum load force obtained for the configuration KW of the tested spring support is approximately 10%.

The elastic characteristics shown in Figure 12 were determined for the spring mounted with hangers of the shortest length H_{min} in the variant of setting the initial hanger deflection with a constant angle $+\phi$ (Figure 7a). The maximum force values differ by approximately 1%.

4. Energy Dissipation Tests for Cyclic Loading of a Double-Leaf Spring with Hanger

4.1. Methodology for Determining Energy Dissipation Based on Experimental Data

In the process of quasi-static loading/unloading, elastic characteristics were determined for all spring mounting configurations (Figures 10–13) with hysteresis loops. The area under the loading part of the compression diagram (upper part of the hysteresis loop) was taken as a measure of the energy absorbed when loading the spring (Figure 14).

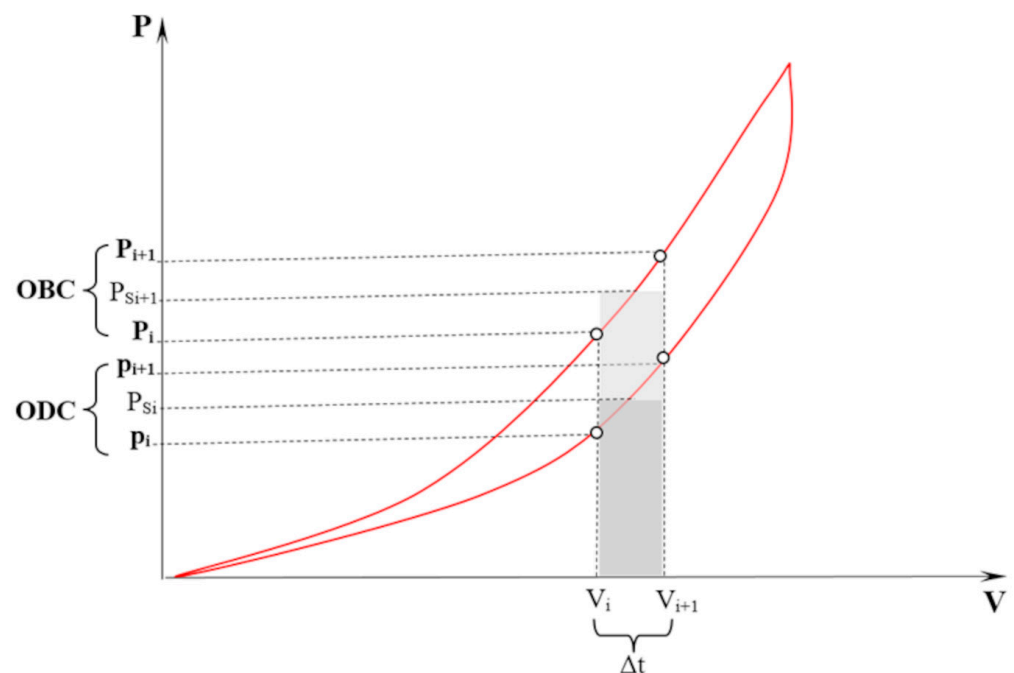


Figure 14. Diagram of the differential procedure used in numerical integration to identify hysteresis loops.

The integration of the area under the graph was performed numerically using the finite difference method [32]. Formula (1) was used.

$$L_{OBC} = \int_0^{v_{max}} P dV = \sum_{i=1}^{n-1} \frac{(P_{i+1} + P_i)}{2} (v_{i+1} - v_i) \quad (1)$$

The integration step Δt (Figure 14) was constant and derived from a sampling frequency of 10 Hz, applied consistently across all tests.

The energy released during unloading was determined using a similar method. The area under the load-relieving part of the graph (lower part of the loop in Figure 14) was calculated according to Equation (2).

$$L_{ODC} = \int_{v_{max}}^{v_{min}} p dV = \sum_{i=1}^{n-1} \frac{(p_{i+1} + p_i)}{2} (v_{i+1} - v_i) \quad (2)$$

$$|L_{ODC}| < |L_{OBC}| \quad (3)$$

$$\Delta L = |L_{OBC}| - |L_{ODC}| \quad (4)$$

This value is negative relative to the previously calculated loading work and is lower in terms of modulus (3). The difference ΔL of these quantities constitutes the energy dissipated into the environment, mainly as a result of the heat generated during friction [12]. This energy is measured by the area enclosed within the hysteresis loop (Figure 14).

In this study, the ratio of dissipated energy (ΔL) to loading energy (L_{OBC}) was taken as a measure of the damping properties of the spring (5).

$$\frac{\Delta L}{L_{OBC}} \cdot 100 [\%] \quad (5)$$

4.2. The Influence of Mounting on Energy Dissipation in the Process of Loading and Unloading a Multi-Leaf Spring

Multi-variant tests of quasi-static loading/unloading were conducted for a multi-leaf double spring with a hanger, using the mounting configurations presented in Section 3.2. This resulted in obtaining the numerical values necessary to determine the elastic characteristics and energy dissipation of the spring. Selected research results are summarized in Table 4.

Table 4. Summary of selected test results of elastic characteristics and dissipation energy in the process of loading and unloading a multi-leaf double-leaf spring with a hanger for various mounting options.

Test	Mounting Variant	Max. Displacement [mm]	Force for Maximum Displacement [kN]	Loading Energy L_{obc} [J]	Dissipated Energy ΔL [J]	$\frac{\Delta L}{L_{obc}} \cdot 100$ [%]
1	KN/P_I	130.23	7.70	433.84	55.97	12.90
2	KN/P_I	129.87	7.78	443.10	61.37	13.85
Mean value			7.74	438.47	58.67	13.38
1	KN/P_II	130.40	7.84	442.02	54.64	12.36
2	KN/P_I	130.23	7.93	448.57	55.23	12.31
Mean value			7.89	445.30	54.93	12.34
1	KN/P_III	130.20	7.96	445.65	50.59	11.35
2	KN/P_I	130.22	8.04	454.27	54.42	11.98
Mean value			8.00	449.96	52.50	11.67
1	KW/P_III	130.14	7.61	422.79	51.96	12.29
2	KW/P_I	130.04	7.69	429.46	52.14	12.14
3	KW/P_I	130.20	7.63	426.21	53.54	12.56
Mean value			7.64	426.15	52.55	12.33
1	KW/P_II	130.22	7.37	404.23	47.98	11.87
2	KW/P_II	130.16	7.44	410.94	51.03	12.42
Mean value			7.41	407.59	49.50	12.14
1	KW/P_I	130.18	6.89	381.32	54.67	14.34
2	KW/P_II	130.42	6.98	390.03	58.35	14.96
Mean value			6.93	385.67	56.51	14.65
1	KZ/P_I	129.77	8.17	464.62	59.85	12.88
2	KZ/P_I	130.13	8.25	472.05	61.84	13.10
Mean value			8.21	468.33	60.84	12.99
1	KZ/P_II	130.05	8.20	463.92	54.96	11.85
2	KZ/P_II	129.96	8.26	469.20	57.02	12.15
Mean value			8.23	466.56	55.99	12.00
1	KZ/P_III	130.33	8.22	464.34	53.88	11.60
2	KZ/P_III	130.19	8.25	465.53	51.86	11.14
Mean value			8.24	464.93	52.87	11.37

At least two load–unload tests were performed for each variant of supporting the double-leaf spring with a hanger (Table 4). Only in the case of the KW/P_III configuration, i.e., for the spring with the hanger of the shortest length H_{min} in the variant of setting the initial deflection of the hanger with a fixed angle $+\phi$ (Figure 7a, Table 3), three tests

were performed. Therefore, the analysis of the results of bench tests of the spring with a hanger, for various design solutions, was performed for the average values of parameters determined in individual tests (Table 4).

Based on the average values presented in Table 4, bar charts were prepared to illustrate the impact of spring mounting on selected parameters describing the spring load and changes in energy dissipation in the process of quasi-static loading/unloading. They are presented in Figures 15–17.

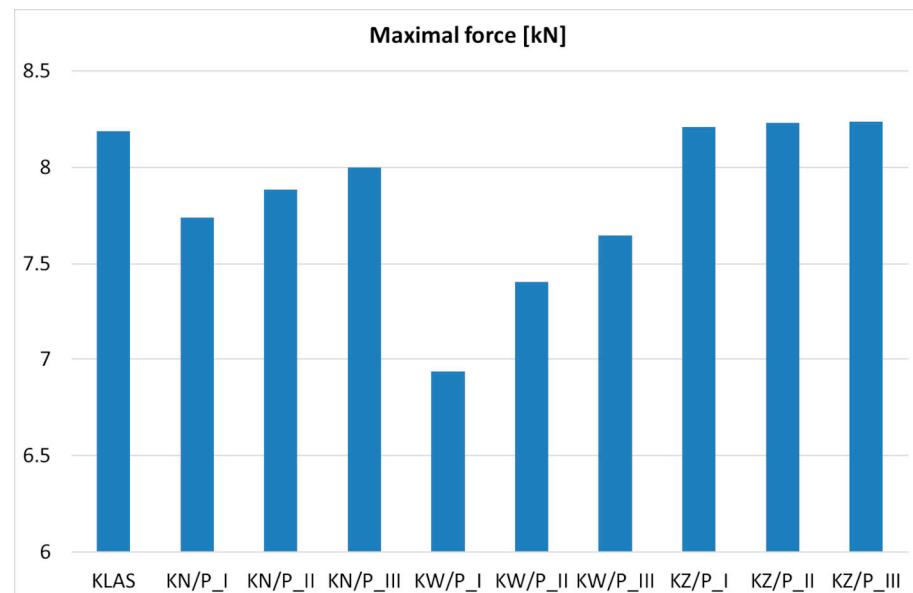


Figure 15. Maximum force values obtained at the maximum spring deflection for all tested design solutions.

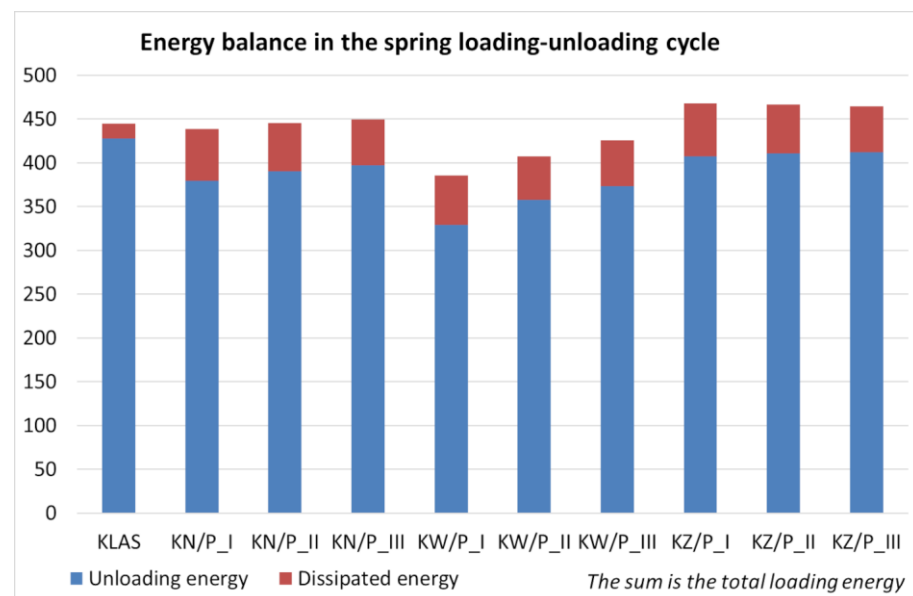


Figure 16. Energy balance in the spring loading–unloading cycle for all tested spring mounting configurations.

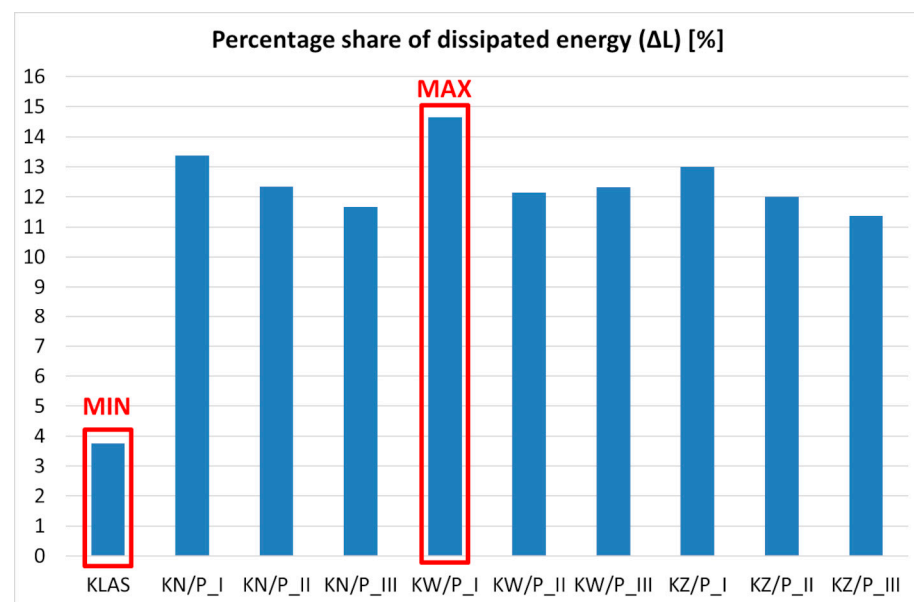


Figure 17. Percentage of dissipated energy (ΔL) in relation to the loading energy (L_{OBC}).

The variant of spring mounting is described on the abscissa of the diagrams. The first bar in the charts in Figures 15–17 concerns the results obtained for a multi-leaf double-leaf spring freely supported (‘KLAS’ designation). The remaining variants on this axis are marked according to the description in Tables 3 and 4 and refer to the tests of the spring with a hanger.

A tabular summary of the categories used on the abscissa axes of the graphs is provided to compare the results of the energy dissipated in the individual design variants of the spring attachment from Figures 15–17.

Table 5 presents the categories used on the abscissa axes of the graphs from Figures 15–17, presenting a comparison of the results of the dissipated energy in the individual design variants of the spring mounting. Detailed data and descriptions for each of these categories are given in Table 4.

Table 5. A tabular summary of categories used on the abscissa axes of the graphs from Figures 15–17.

Categories Used on the Abscissa Axes of the Graphs from Figures 15–17									
KLAS	KN/P_I	KN/P_II	KN/P_III	KW/P_I	KW/P_II	KW/P_III	KZ/P_I	KZ/P_II	KZ/P_III

Figure 15 presents changes in the value of the maximum force obtained in the loading process for all variants of spring mounting. The lowest value of the maximum excitation force of 6.89 kN (the average value of two tests is 6.93 kN) was measured during tests of the spring with a hanger for the KW/P_I mounting variant (hangers with the greatest length H_{max} , and its initial setting angle is $+\phi$) (Figure 15 and Table 4). It constitutes approximately 83% of the load force value determined during testing of the freely supported spring (Diagram ‘KLAS’ in Figure 15).

However, for the KZ/P_III mounting variant (double-leaf spring mounted with a hanger of the shortest length H_{min} with the initial angle of the hanger being equal to $-\phi$) (Figure 15 and Table 4), the maximum value of the excitation force was measured at 8.25 kN (the average value of two tests is 8.24 kN). The average value of the maximum force of 8.24 kN measured for the KZ/P_I, II and III mounting variants is comparable to the value of the loading force determined during the test of the freely supported spring (Diagram ‘KLAS’ in Figure 15).

Figures 16 and 17 demonstrate the energy balance in the loading–unloading cycle for all tested double-leaf spring mounting variants and the percentage of dissipated energy (ΔL) in relation to the total load energy (L_{OBC}).

The sum of the heights of the red and blue bars in the diagram in Figure 16 corresponds to the spring load energy.

A summary of laboratory tests with quasi-static forcing of a multi-leaf double-leaf spring with a hanger in the dependent suspension of a truck with medium load capacity for various geometric variants of the hanger arrangement is as follows:

1. Due to the very small influence of the loading/unloading speed demonstrated in the free mounting of the spring, tests of the spring with a hanger for all variants were performed at the same speed of 100 mm/min.
2. As a result of bench tests, the basic elastic characteristics of the bilinear spring in terms of quasi-static excitations were identified for two basic methods of spring cooperation in dependent suspension: a free spring and a spring with a hanger; the data necessary to support design work were determined, and a universal laboratory station was constructed.
3. In the test variant of the spring mounted with a hanger, in the KW/P_I configuration, the lowest value of load energy L_{OBC} was determined at the level of 381.32 J (the average value of two samples is 385.67 J).
4. The maximum load energy of 472.05 J (average of 468.33 J from two tests) was observed for the KZ/P_I configuration (the spring mounted with the hanger with the longest length H_{max} in the variant of setting the initial deflection of the hanger with an angle $-\phi$) (Figure 16 and Table 4). The average load energy value of 468.33 J measured during tests of the spring mounted with a hanger in the KZ/P_I configuration is slightly higher (the relative difference is less than 5%) than the average load energy $L_{OBC} = 449.5$ J determined during testing of the freely supported spring (Diagram 'KLAS' in Figure 16).
5. The average value of the dissipated energy determined in the KZ/P_I variant, corresponding to the maximum load energy of the spring with a hanger, is 60.84 J and is significantly higher than the average dissipated energy of the freely supported spring, equal to 17.93 J (Table 2), determined on the basis of the results of two tests with identical loading/unloading speed 100 mm/min (Diagram 'KLAS' in Figure 16). The average dissipated energies (ΔL) in all variants of the spring mounted with a hanger are larger than those determined for the free spring ($\Delta L = 17.93$ J). Their average values change in the range of 49.50 J to 60.84 J (Table 4). The average energy dissipated in the KZ/P_I variant is higher than the average dissipated energy of the freely supported spring by almost 340%.

These findings are confirmed by a comparison of the appropriate bar charts in Figure 17, which indicate the minimum and maximum shares of dissipated energy in load energy.

5. Summary of Research and Comparative Analysis

Standard testing machines were used to build a test stand for quasi-statics of multi-leaf springs. The setups were assembled in two variants: for testing a free multi-leaf double-leaf spring and for testing a multi-leaf double-leaf spring with a hanger that cooperates in the vehicle's dependent suspension in various variants in terms of setting geometry.

Bench tests identified the elastic characteristics of the bilinear spring under quasi-static excitations for two cooperation methods in a dependent suspension: a free spring and a spring with a hanger. Additionally, the data necessary for supporting design work and constructing a universal laboratory station were established.

As part of the analysis of historical and updated results of the tests performed in the initial phase for quasi-static tests of a multi-leaf double-leaf spring in free mounting, the following was proven (as shown in Table 2):

1. The share of dissipated energy (ΔL) in relation to the spring load energy (LOBC) is several percent ($<5\%$).
2. A slight decrease in dissipated energy was observed as the loading rate increased.
3. No signs of degradation of the spring's elastic properties were observed over time, as evidenced by similar hysteresis loop courses in both old and new measurements.

Based on the analysis of the quasi-static test results of a multi-leaf double-leaf spring with a hanger in various operational configurations (Table 4 and Figures 10–17), the following was found:

4. The greatest influence on changes in elastic characteristics with a constant hanger length ($H_{\max}$) is attributed to adjustments in the initial deflection angle of the hanger within the range of deflection angles from $-\phi$ to $+\phi$.
5. Altering the length of the hanger from $H_{\min}$ to $H_{\max}$, while keeping the initial deflection angle (ϕ) constant, also affects the elastic characteristics. Changes in hanger length in a perpendicular position have little effect on the hysteresis loop cycle. It was found that level III (the shortest hanger) achieved the highest force value for maximum displacement (Table 4).
6. For the KW/P_I mounting variant, the lowest value of the maximum force at the maximum spring deflection (Figure 15), the lowest load energy (Figure 16) and the highest share of dissipated energy in the energy balance (Figure 17) were obtained.
7. The values of the maximum forces obtained at the maximum spring deflection for the KZ configuration and the freely supported spring are at a similar level (Figure 15), while the average value of the dissipated energy corresponding to this configuration is significantly higher than the average dissipated energy of the freely supported spring, more than three times.
8. The average dissipated energies (ΔL) in all variants of mounting the spring with a hanger are higher than those determined for the free spring.
9. The share of dissipated energy in relation to the load energy for free support of the double-leaf spring does not exceed 5% (Table 2).

6. Conclusions

1. In conclusion, this research highlights the significant impact of hanger configurations on the elastic characteristics and energy dissipation of multi-leaf springs, demonstrating the importance of design considerations in suspension systems.
2. The proposed research stand and the developed methodology for experimental testing of multi-leaf springs can be successfully used to assess dissipation energy in testing the elastic components of trucks of various categories (N1, N2, N3) and special railway bogies intended for intermodal transport.
3. The presented results concern the operating conditions of suspensions of various vehicles, where multi-leaf springs are only used, without additional components, e.g., shock absorbers and friction dampers, which have a significant impact on the damping properties of the suspension, mainly of trucks, work machines, railway rolling stock equipment and other devices that carry extremely heavy loads.
4. The presented test results identify the level of inelastic resistance in the dependent suspensions of various vehicles caused by the deflection of the multi-leaf spring with fastening elements (hanger, joints, etc.) without additional damping components. Knowledge of the level of inelastic resistance at the suspension design stage is necessary because it is a reference level for introducing additional damping in the vehicle suspension by using a listed damper, e.g., a hydraulic shock absorber. Such damping components (e.g., hydraulic shock absorbers or friction dampers) have a decisive impact on the damping properties of the suspension of trucks, work machines, railway equipment and other devices that carry extremely heavy loads.
5. The novelty in the presented work is the original design, construction of the stand and methodology of experimental tests of the multi-leaf springs together with a comparative analysis of the test results and an answer to the question of the influence

- of the spring mounting conditions on the mechanical characteristics of the multi-leaf spring and the ability to dissipate energy under quasi-static load conditions.
6. The use of hangers for mounting multi-leaf springs in the suspension of medium-duty vehicles significantly increases the ability to dissipate the energy of external loads.
 7. The presented results and conclusions are original and practical data regarding the operation of multi-leaf springs because the bench tests reproduced the design conditions of their mounting in the suspensions of various vehicles and the operation of the springs.
 8. The authors anticipate using the presented research results in work on a universal laboratory stand for testing various types of multi-leaf springs used in suspensions of road, rail and other means of transport, paying attention to those means of transport used to transport heavy loads.

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