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# PROBLEMS IN ASSESSING THE DURABILITY OF A SELECTED VEHICLE COMPONENT BASED ON THE ACCELERATED PROVING GROUND TEST

# PROBLEMY OCENY TRWAŁOŚCI WYBRANEGO ELEMENTU POJAZDU NA PODSTAWIE PRZYSPIESZONEGO TESTU PRZEBIEGOWEGO*

*The paper presents the results of the analysis of the durability of elastic elements occurring in the special-purpose 4x4 off-road truck suspension using data obtained during an accelerated proving ground test conducted during off-road driving. The limitations in access to material data present at the stage of the initial selection of the component (lack of fatigue strength data) are indicated and an alternative analytical method for fatigue strength estimation is given. The differences in the obtained results and their most important sources are pointed out. A method for using a generalized durability index d as a parameter independent of the subassembly material data is also described. The indicator can be used to assess the influence of resultant loads (recorded) appearing during the vehicle operation in the determined road conditions on the durability of the subassembly under study and to associate their value with the type of the test road section.*

***Keywords***: *spring, stabilizer, proving ground test, accelerated tests, durability, generalized durability index, off-road truck.*

*W artykule przedstawiono wyniki analizy trwałości elementów sprężystych występujących w zawieszeniu specjalnego terenowego pojazdu ciężarowego 4x4 wykorzystując dane uzyskane podczas przyspieszonego testu drogowego przeprowadzonego podczas jazdy off-road. Wskazano na występujące ograniczenia w dostępie do danych materiałowych jakie są obecne na etapie wstępnego doboru podzespołu (brak danych wytrzymałości zmęczeniowej) oraz podano alternatywną analityczną metodę szacowania wytrzymałości zmęczeniowej. Wskazano na powstające różnice w uzyskanych wynikach oraz na najważniejsze ich źródła. Przedstawiono również sposób wykorzystania uogólnionego wskaźnika trwałości d jako parametru niezależnego od danych materiałowych podzespołu, który można wykorzystać do oceny wpływu obciążeń wynikowych (rejestrowanych) powstających podczas ruchu pojazdu w ustalonych warunkach drogowych na trwałość analizowanego podzespołu i powiązać ich wartość z rodzajem testowego odcinka drogowego.*

***Słowa kluczowe***: *resor, stabilizator, test przebiegowy, badania przyspieszone, trwałość, uogólniony wskaźnik trwałości, ciężarowy samochód terenowy.*

## 1. Introduction

Vehicle durability evaluation is a very complex and challenging issue, and at the same time necessary in the process of achieving the series-production readiness of a vehicle structure [15]. In the case of complex objects, e.g., special-purpose off-road trucks that are required to be highly reliable and durable, the design and construction process is organized according to the appropriate management model. An example of such a model can be the V-model [12] developed by NASA. This model assumes that the transition to the next stage of the design and construction process is possible only when the previous step is rated positively. To have it evaluated, this is necessary to conduct appropriate tests, whose complexity and labor intensity depends on the determination of the impact degree of a given stage on the quality of the final product. Such an analysis can be performed using, for example, the Design for Six Sigma method [17], which makes it possible to indicate the accuracy that is necessary to assess individual stages of the design and construction process for the final product to have the required durability or reliability. Therefore, it is crucial to select and carry out appropriate tests that reflect with sufficient accuracy the influence of the loads predicted for the planned operating conditions on the product durability [11, 12, 14].

In the case of special-purpose off-road trucks, the selection of appropriate tests seems to be particularly tricky. These are vehicles produced in small series, designed to be driven in changing road conditions with variable load over a long period of operation (up to 30 years). It appears therefore necessary and essential to adopt several simplifying assumptions regarding, among other things, the location and conditions for testing.

Some vehicle manufacturers conduct their tests on parametrized road measurement sections, e.g., Tatra [22], which should be representative of the actual road conditions, where the degree of influence of the road profile and the vehicle traffic parameters on the value of the resulting loads and, ultimately, on the durability of the analyzed subassemblies are determined. Acceptance tests of a vehicle in running order are carried out by a designated certification body, while implementing an established testing program, on behalf of a future user. The examination results in the issuance or refusal to issue a certificate of conformity of the product with the requirements of the recipient. However, the results of these tests are available only when the vehicle is ready for production

There are also some vehicle manufacturers who do not have access to road testing centers. Thus, it is problematic to conduct tests.

(*) Tekst artykułu w polskiej wersji językowej dostępny w elektronicznym wydaniu kwartalnika na stronie ***www.ein.org.pl***

In such a case, they are performed on selected available road sections, including public roads. However, there is a problem of correlating the loads assumed as representative (occurring in test sections of the certification body) with those used by the manufacturer. Hence, vehicle makers are looking for different parameters that can be used to compare the test conditions of the certification body with their test conditions.

Due to the limited time and financial resources, but also for example because of data shortages, testing generally leads to the ultimate limit state of the subassembly under examination or to the moment when, based on the data collected, the relationship between the conditions of use and the sustainability of the component can be established. Different models of degradation processes are used to link the resulting loads (traffic conditions) and the life span of the element. They also include those whose use does not require the knowledge of detailed material data obtained through experimental bench studies, which are very time-consuming and pricey.

The obtained results of the durability of the tested subassemblies often refer to the values describing the utilized labor resource, e.g., in units of vehicle mileage (km), engine operating hours (EOH), and others according to the future user's requirements. On the adoption of simplifying assumptions that, e.g., road test sections and established traffic parameters are constant, the acquired outcomes, enable linking the unit mileage of a vehicle with the degree of its degradation. As a result, data are received which allow for the comparison of the influence of selected types of road test sections (those of the certification body with own sections) and established traffic parameters of the vehicle on the degree of degradation of a selected component. Examples of estimating the durability of vehicle subassemblies can be found in the literature, among others in [2, 5, 10, 20]. The problem remains, however, the identification of a parameter, the determination of which could be used as a comparative indicator for the initial estimation of the component durability in connection with the selected road test section.

## 2. The aim and scope of the research

The research aimed to estimate the durability of selected components of an off-road truck under specified traffic conditions and to check whether it is possible to apply a generalized durability index to the initial assessment of the suitability of these components for the vehicle. The setting of the indicator mentioned above does not require full knowledge of the strength of the material of which the elements were made, which is a typical problem occurring during accelerated mileage tests. Choosing a generalized durability index and determining its value in acceptance test conditions of a certification body would allow the similar test program to be determined based on the road sections available to the manufacturer. The detailed characteristics of the generalized durability index used in the tests are not presented in this paper but are described in the publication [6].

The subject of accelerated mileage tests were elastic elements (parabolic springs, stabilizers) occurring in the suspension of a special-purpose 4x4 off-road truck. The testing was carried out under off-road traffic conditions with limited data concerning the strength of the material of which these elements were made. The vehicle manufacturer specified the limitation of the test to one type of road section.

Parabolic springs, which allow for relative movement of wheels and body in the vertical axis, and at the same time remove the freedom of movement in the other axes, and stabilizers, which reduce the lateral tilt of the body, thus improving the stability of the vehicle motion, proved to be the susceptible elements in the suspension of the analyzed vehicle [16]. The subassemblies operate in a complex stress state, but in order to simplify the tests, it is often assumed that the springs are subjected to bending and the stabilizer rods are twisted [1]. Figures 1 and 2 show the stiffness characteristics of the springs and the deflection ranges at different vehicle loads.

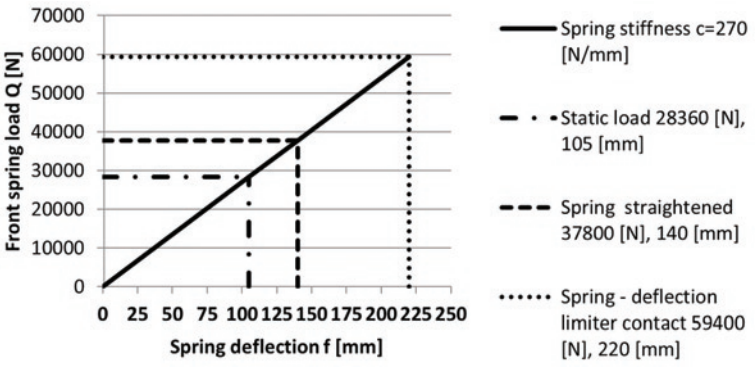


*Fig. 1. Stiffness characteristics of the front axle parabolic spring*

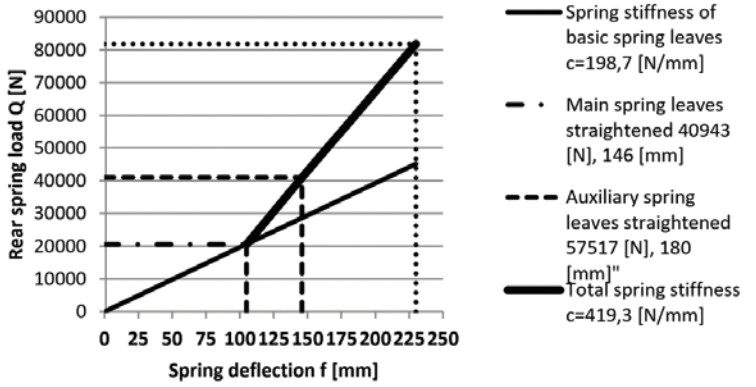


*Fig. 2. Stiffness characteristics of the rear axle parabolic spring*

*Table 1. Summary of basic characteristic dimensions of stabilizers*

| | Front axle stabilizer | Rear axle stabilizer |
|---|---|---|
| Length of the element subject to torsion [mm] | 730 | 820 |
| Arm of torsional force [mm] | 520 | 340 |
| Diameter of the element subject to torsion [mm] | 40 | 50 |
| Torsional strength index of the cross-section [cm3] | 6,28 | 12,27 |

The stabilizers were made of rods with a circular cross-section. The basic characteristic dimensions are shown in Table 1.

The material used in the production of these components was 51CRV4 steel (Rm=1350 MPa). The manufacturer's declaration in-

*Table 2. Summary of number of samples, load levels and percentage of repetitions*

| Number of samples $n_s$ | Load levels SL | Percentage of repetitions LP |
|---|---|---|
| 12 | 2 | 83,3 |
| 12 | 3 | 75,0 |
| 24 | 3 | 87,5 |
| 24 | 4 | 83,3 |
| 24 | 5 | 79,2 |
| 24 | 6 | 75,0 |

dicated that the spring leaves were heat treated, and according to the standard [15], the core hardness should be in the range between 363 and 460 HB. In addition, the spring feathers on the stretched side were shot peened. With such a procedure, normal compressive stresses, which significantly reduce the values of tensile stresses arising during the component's operation, were introduced on this surface [3, 18]. Due to the lack of data on the values of those stresses as well as the depth of their introduction into the material structure, the available data, presented in, e.g., [9, 13, 15], were used to estimate them. Based on the data, it was assumed that in the unloaded state, compressive stresses might reach the value from 300 to 400 MPa, and the depth of the introduced strains can be 15 - 25 μm.

## 3. Fatigue strength model of the analyzed subassemblies

Testing for fatigue-limited durability of the subassemblies is a complex and time-consuming task. The sample size for experimental tests depends on the stage of the design and construction process, the number of analyzed load levels and test repetitions. In the initial step of selecting a component, the number of samples from 6 to 12 is usually enough and increases to 24 for reliability tests [1]. The number of test repetitions can be determined from the following dependency [10]:

$$LP = 100\left(1 - \frac{SL}{n_s}\right) \tag{1}$$

where: LP - percentage of repetitions, SL - number of load levels, ns - number of samples.

The percentage of repetitions in the pre-test phase is between 17 and 33. Table 2 summarizes the number of samples, load levels and repetitions for 12 and 24 samples respectively.

The presented data illustrate the time-consuming experimental tests of a subassembly performed to determine its fatigue strength characteristics. The conducted studies, which were preliminary tests of ready-made components checked by the manufacturer, were restricted to assessing the correctness of their selection for the vehicle. The tests were limited to one truck. Due to the lack of detailed data concerning fatigue strength (the experimentally determined S-N curve), it was necessary to determine the curve through theoretical calculations and to link the obtained results with the parameter connecting the component durability with the type of road test section [6].

The dependencies, which allowed to determine the fatigue graph based on a limited set of data, were used to calculate the fatigue strength of the spring. It was assumed that the determination of fatigue strength in the high cycle range, i.e., within the range from $10^3$ to $10^6$ cycles, was of crucial importance. The method for the determination of individual values was taken from available publications, among others [4,10].

The fatigue strength for $10^3$ number of cycles was determined using the following relationships:

$$A_{NC,R} = A_{1000} \cdot C_R \tag{2}$$

where: $A_{NC,R}$ – stress amplitude for low cycle loads including reliability factor $C_R$, $A_{1000}$ –stress amplitude for low cycle loads, $C_R$ – reliability factor.

The value $A_{1000}$ can be determined from the relationship:

$$A_{1000} = \alpha_{NC} \cdot R_m \tag{3}$$

where: $R_m$ – limit of material strength determined in the static tensile test, $\alpha_{NC}$ - load type dependent coefficient for $10^3$ cycles; 0.9 for bending, 0.72 for torsion.

The value of the reliability factor $C_R$ depends on the expected operating reliability of the component. In the tests it was initially assumed that $C_R$=1.

The fatigue strength for $10^6$ cycles was derived from the relationship in which corrective factors were taken into consideration:

$$A_{WC,R} = A_{WC} \cdot C_L \cdot C_S \cdot C_D \cdot C_R \tag{4}$$

where: $A_{WC,R}$ – stress amplitude for high cycle loads including the reliability factor $C_R$, $A_{WC}$ –stress amplitude for high cycle loads, $C_L$ – load type factor, $C_S$ – surface condition factor, $C_D$ – size dependent coefficient, $C_R$ – reliability factor.

The value $A_{WC}$ can be determined from the dependencies:

$$A_{WC} = \alpha_{WC} \cdot R_m \tag{5}$$

where: $R_m$ – limit of material strength determined in the static tensile test, $\alpha_{WC}$ – coefficient depending on the type of material for $10^6$ cycles; for steel ($R_m$ <1400 MPa) it is 0.5.

The value of load type factor $C_L$ was assumed according to data available in literature [10]. For bending $C_L$=1 and for twisting $C_L$=0,58.

The value of the surface condition factor $C_S$ can be determined from the surface roughness measurement and the material strength value $R_m$. The components supplied by the manufacturer were factory protected with protective paint against the harmful effects of weather conditions. The measurement of the actual surface roughness would require the effective removal of this layer. Because of the existing limitations, the roughness was not measured and the available literature data [10] were used to determine the factor $C_S$. The springs were rolled and shot peened, and in this way compressive stresses were introduced into the structure of the material, thereby partially compensating the tensile stresses arising during the operation of the subassembly. The value of factor $C_S$ equal to 0.76 was used in the calculations.

The value of the coefficient depending on the size of the $C_D$ element was calculated from the following dependencies [10]:

$$C_D = 1{,}189 \cdot d^{-0{,}097} \tag{6}$$

where: d – element diameter, mm.

For a rectangular section element (a leaf spring), the equivalent diameter can be derived from [10]:

$$d_z = \sqrt{0{,}65 \cdot s \cdot w} \tag{7}$$

where: s - section width, w - section height.

The calculated $C_D$ values is shown in Table 3.

Figure 3 presents the diagrams of fatigue strength of springs and stabilizers were prepared based on the determined data, which is presented in Figure 3. The determined fatigue strength values of the front and rear stabilizers are comparable, and the difference occurring in the area of unlimited fatigue strength is slight and amounts to 5 MPa. The strength values identified for this area are 247 MPa for the front stabilizer and 242 MPa for the rear stabilizer.

*Table 3. The $C_D$ values determined for the analyzed elements*

| | Front spring | Rear spring | Front stabilizer | Rear stabilizer |
|---|---|---|---|---|
| $C_D$ | 0,84 | 0,84 | 0,83 | 0,81 |

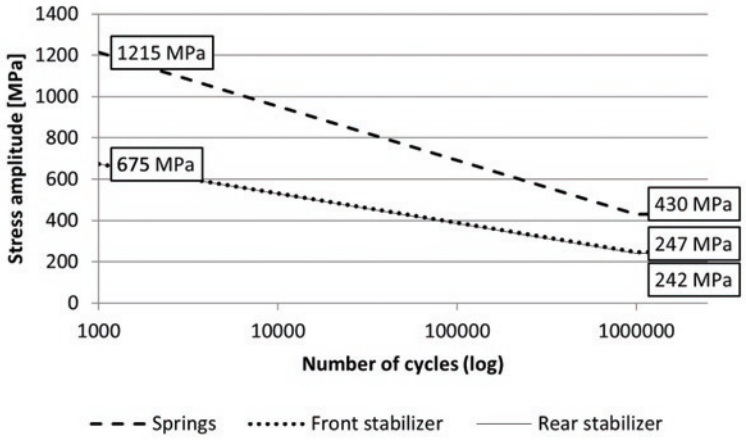


*Fig. 3. Determined diagrams of fatigue strength of springs and stabilizers*

The obtained diagrams of fatigue strength of the subassemblies were used to analyze the durability of the tested components.

## 4. The course of the tests

The examination was carried out in the training ground conditions at the University of Land Forces in Wrocław. The selected sandy off-road section was an approximately 1 km long measuring loop. Due to the nature of the unevenness, the average driving speed was about 7 km/h. It was determined from previous trips and conclusions from prior studies [7,8]. The selected road measurement section corresponded to the testing ground conditions, which are taken into consideration when designing the vehicle to the expected traffic conditions described in the vehicle exploitation profile [11]. However, that section was not parameterized. A test driver of the manufacturer drove the vehicle. The test vehicle was loaded evenly, using the total payload.

Motor vehicle springs operate in a complex stress state [1,18]. However, in accelerated mileage tests, it is difficult to record all the occurring loads and assess their influence on the fatigue life of a spring. Therefore, it is assumed that the dominant load is bending, which causes normal stresses in the cross-sections of spring leaves. In the case of stabilizers, they are designed to be torqued. Adoption of the simplifications presented causes the collection of data necessary for further analysis to be reduced to the recording of emerging stresses caused by bending of springs and torsion of stabilizers. The data reduction achieved in this way is a thoughtful step resulting from the economics of time and available resources as well as limited data on the analyzed components. Table 4 presents a set of characteristics that were available at the stage of the initial selection of subassemblies.

The measuring system used in accelerated tests of elastic vehicle components consisted of strain gauge sensors glued to the prepared surfaces of spring leaves and stabilizers (Fig. 4÷5). The strain train gauges were glued in places where the highest stress values were expected to be obtained (around the yoke fixing the spring leaves, and in the case of a stabilizer in the middle of the section subject to torsion). The choice of locations was additionally confirmed based on the FEM model of springs [19, 20], which is not a standard step.

During road tests, load courses were recorded and then filtered through Rainflow to specify load cycles. Figure 6 shows an exemplary load course of a rear axle leaf spring. Rainflow filtration was performed by determining and counting the load cycles from the recorded load course. The method is now widely used and standardized. When mounted on a vehicle, the springs are initially loaded with the vehicle's weight and freight, which affects the asymmetry of the loads generated when bending and unbending these elements while driving (shifting the mean value). The Goodman model [21] was used to take this effect into account. The Palmgren-Miner hypothesis, which assumes the linear accumulation of damages up to the limit value considered as 1, was harnessed to sum up fatigue damages. It is a model commonly used in fatigue calculations.

*Fig. 5. Stabilizer with glued-on strain gauge*

## 5. The analysis of the results obtained

The durability of the tested subassemblies was estimated from recorded mileage and theoretically determined fatigue strength, and given in units of vehicle mileage. Under the assumption that the loads occurring during the tests are representative for future predicted operating conditions, the results obtained are preliminary information used to assess the appropriateness of the choice of components for the vehicle. A significant scattering of the received values to the individual subassemblies can be observed in the summary of the results collected in Table 5. The reason for this scattering is the lack of accurate data on the actual value of pre-stresses introduced into the spring leaves, which had to be estimated.

The data presented in Table 5 indicate that the calculated spring durability is strongly influenced by the correctly assumed value of compressive pre-stress, which can be identified based on e.g., the as-

*Table 4. Basic data on the analyzed subassemblies*

| | |
|---|---|
| **Springs** | stiffness characteristics, dimensions, weight, material, type of heat treatment and plastic processing, declared hardness on the surface, declared durability, |
| **Stabilizers** | stiffness characteristics, dimensions, weight, material, type of heat treatment and plastic processing, |

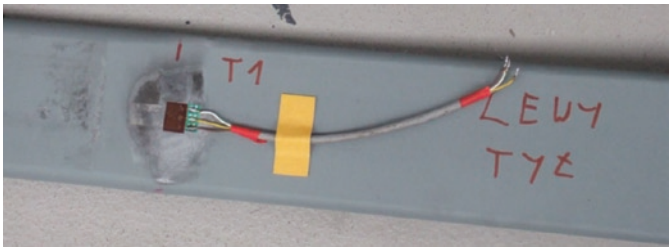


*Fig. 4. Leaf spring with glued-on strain gauge*

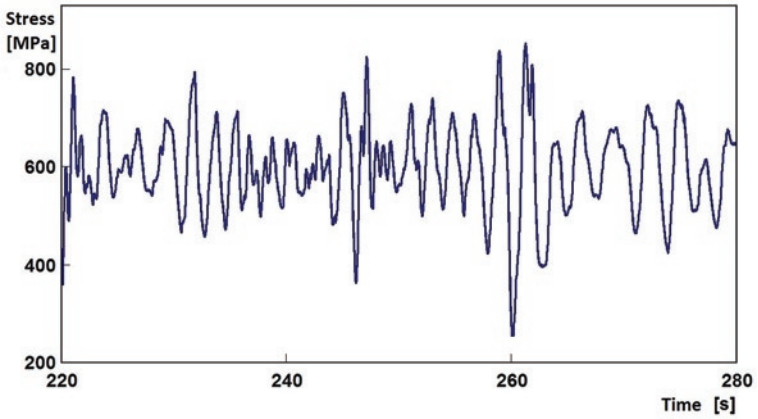


*Fig. 6. Example of the stress pattern of a leaf spring installed on the rear axle of a vehicle (values do not include preliminary stresses due to peening)*

sessment of the depth of changes in the microstructure of the material, which is the result of shot peening. Such an evaluation may be carried out by, among others, performing material destructive tests of a component [9, 13]. The general information provided by the manufacturer about the plastic processing, without detailed data, is insufficient for correct calculation of the component durability.

Table 6 shows how the calculated durability of the components is affected by the reduction of loads directly or indirectly influenced by the driver's driving style. From the data provided it is clear that a 5% load reduction (e.g., speed reduction, rerouting, tire pressure adjustment, etc.) can extend the life cycle of a component by approximately 50% and a 10% load reduction can increase it by ca. 100%.

The data presented show that the attempt to determine the component durability limited by fatigue strength in an accelerated mileage test poses many difficulties and may be subject to material error, e.g., due to the adoption of approximate intermediate volumes. Significant limitations in establishing the exact values include the lack of data concerning the experimentally determined fatigue strength of the subassembly, which requires approximate theoretical calculations to be made, the lack of detailed material data of the component (real value Rm, value of introduced compressive stresses and their depth) and parameters describing the condition of the top layer (roughness). Moreover, in preliminary mileage tests, when there is no access to parameterized test tracks, there is a need to compare the effects of the application of new structural solutions of subassemblies in relation to those previously used and to evaluate their work in connection with the type of road test section used by the certification body. A useful parameter in solving this type of problem may be the quantity called a generalized durability index *d* [6], which expresses numerically the overall impact of parameters describing the vehicle motion (e.g., speed, type of test section) on the durability of the component, but without reference to the material characteristics of the element.

*Table 5. Summary of predicted durability of elements for different pre-stress values*

| Volume | Spring LP | Spring PP | Spring LT | Spring PT | Front stabilizer | Rear stabilizer |
|---|---|---|---|---|---|---|
| Range (excluding compressive pre- stresses) [km] | 44 | 35 | 12 | 10 | 555 | 600 |
| Range (initial compressive stresses 300 MPa) [km] | 56698 | 45708 | 13878 | 11338 | - | - |
| Range (initial compressive stresses 350 MPa) [km] | 121229 | 100819 | 30231 | 24357 | - | - |
| Range (initial compressive stresses 400 MPa) [km] | 273596 | 236196 | 71713 | 56939 | - | - |

*Table 6. Effects of load values on component durability*

| Subassembly | Durability at the registered load (without taking compressive pre-stresses into account) | 5% reduced load durability | 10% reduced load durability |
|---|---|---|---|
| Right front spring | 35 | 48 | 68 |
| Right rear spring | 10 | 15 | 21 |
| Front stabilizer | 600 | 776 | 1016 |
| Rear stabilizer | 555 | 718 | 941 |

*Table 7. Summary of the generalized durability index values for the vehicle's front and rear springs on the left and right, respectively*

| Value of the generalized durability index *d* | Left front spring | Right front spring | Left rear spring | Right rear spring |
|---|---|---|---|---|
| $d_{100\%}$ | $6{,}74*10^{16}$ | $8{,}48*10^{16}$ | $2{,}42*10^{17}$ | $2{,}80*10^{17}$ |
| $d_{95\%}$ | $4{,}71*10^{16}$ | $6{,}08*10^{16}$ | $1{,}72*10^{17}$ | $1{,}99*10^{17}$ |
| $d_{90\%}$ | $3{,}33*10^{16}$ | $4{,}30*10^{16}$ | $1{,}18*10^{17}$ | $1{,}40*10^{17}$ |
| $d_{300MPa}$ | $5{,}19*10^{13}$ | $6{,}43*10^{13}$ | $2{,}12*10^{14}$ | $2{,}59*10^{14}$ |
| $d_{350MPa}$ | $2{,}43*10^{13}$ | $2{,}92*10^{13}$ | $9{,}73*10^{13}$ | $1{,}21*10^{14}$ |
| $d_{400MPa}$ | $1{,}07*10^{13}$ | $1{,}24*10^{13}$ | $4{,}10*10^{13}$ | $5{,}19*10^{13}$ |

The concept of using a generalized durability index $d$ is described in [6] and is based on the determination of the value of the expression:

$$d = \sum n_i A_i^{\beta} \tag{8}$$

where: $d$ - generalized durability index (pseudo damage), $A_i$ - load amplitude determined by, e.g., Rainflow method, $n_i$ - number of load cycles with $A_i$ amplitude, $\beta$ - fatigue curve slope coefficient (it may be assumed that for elements performed without special finishing operations (e.g., grinding, polishing) the coefficient β=5).

The described generalized durability index $d$ was used to present the differences in the loads of the same components on the left and right respectively. Examples of the calculation results are given in Table 7.

The values of the generalized durability index $d$ presented in Table 7 apply to cases where the values of measured stresses (d100%), stresses reduced by 5% and 10% (d95%, d90%) and initial compressive stresses (300 MPa, 350 MPa, and 400 MPa, respectively) were considered. The increasing value of the parameter $d$ indicates a more destructive course of loads. The data presented in Table 7 show that the front right spring, which is the same as the front left one, was subjected to more destructive loads during the tests. Similarly, the right rear spring was more fatigue loaded than the rear left one. One can also see that the elastic components in the front axle suspension are more durable than those in the rear axle. However, the received values for the generalized durability index $d$ of the subassembly do not represent the actual life cycle of the component, but only constitute a numerical representation (easy to compare) of whether the loads acting on the element are more or less destructive under given traffic conditions compared to another component of the same type.

## 6. Conclusion

The primary objective of the research was to identify the loads acting on the spring components of the suspension and to estimate their durability limited by fatigue strength, as shown in Table 5 and used as preliminary data to check the suitability of these subassemblies for the vehicle. An additional aim was to indicate a parameter, the use of which would allow for the assessment of the extent to which the traffic conditions and the type of road measuring section influence the value of loads on the selected elements, thus limiting their durability.

The durability of the analyzed components is crucial for the estimation of vehicle reliability, which is understood as a technical system whose loads resulting from traffic conditions vary widely (from driving on hard-surfaced roads with no cargo to off-road driving with freight). The analyses presented were based on limited data available at the stage of the initial selection of a new subassembly for the vehicle. The obtained results of component durability are presented concerning the theoretical driving range of the car, which is an effective comparison parameter. Due to limited data, different load values (as the result of possible changes in the driver's driving style) and initial compression stresses of spring leaves were used for calculations, thereby showing how they affect the vehicle mileage being analyzed.

The tests were limited to only one vehicle (one set of analyzed subassemblies) moving at a set speed in selected road conditions. Therefore, the results obtained are only a preliminary material for further analysis. However, it is worth noting that the use of the proposed generalized durability index $d$ makes the initial comparison of the durability of individual vehicle springs possible. The distinction of the degradation degree of the same springs, but differently loaded (which stems from the non-identical shape of the ground under each wheel during the journey) indicates that the values of parameter $d$ determined for the same component (spring) in various road conditions (test sections) can also be compared. If the parameter $d$ is additionally normalized and its value is reduced to the unit length of the measurement distance (e.g., to 1 km), it will be possible to estimate the degradation degree of the same component in different traffic conditions and on varying test sections. This gives reason to believe that it is possible to reproduce the effect of the loads recorded on one test section (e.g., of a certification body) with another available test section (available from the vehicle manufacturer), which would be an innovative use of the parameter $d$ identified on the basis of the transformed Basquin equation. Confirmation of this assumption will, however, require additional testing.

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