The study of the fatigue strength characteristics of welded joints in car components

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Abstract. The computational study of the fatigue life of car body components and its spot welds was carried out using two methods in this article. Besides, there analyzes the impact of spot welds failure on the stiffness characteristics of the car and a comparison of the sites of breakdown that arose during computer modeling using a digital twin with the sites of a breakdown in a real test. The study resulted in a significant influence of fatigue life on the stiffness characteristics of the car body. This noticeable effect is explained by most of the failed spot welds are located in the areas of the front and rear suspension fastenings, experiencing significant stress both during the car operation and during bench tests for bending and torsional stiffness of the body.

1. Introduction
The geometric parameters of the body structure and the quality of technological and welded processes determine the reliability of the vehicle design, as well as its service life. At the early stages of the car development, under a large number of changes made to the design, engineers do not have a full set of information about the car's wheel loads and their impact on its performance. This leads to the potential need for a constant recalculation of the target characteristics, including fatigue strength of car components.

Most of the changes and the impossibility of studying their influence using real prototypes can be compensated by creating a digital twin of the car [1, 2]. Moreover, a simplified assessment technique for rapid evaluation of fatigue life can be used, with complete or partial absence of data on wheel loads. According to this method, the loads applied to the front axle are calculated based on the design characteristics of the car, which in most cases are known and unchanged from the very beginning of the development process: the full curb weight of the car and the width of its track, and the passage of various geometrically shaped obstacles is modeled by a test bench in the form of applying loads with different frequency and amplitude. This technique has significantly lower accuracy compared to a full virtual fatigue test and compared to the site or full bench tests. However, at the early stages of the car development, critical areas of the structure are quickly identified and compare the impact of changes, introduced into the design in terms of fatigue life, evaluating their effectiveness.
At later stages of development, in order to understand not only the qualitative but also the quantitative characteristics of fatigue life, it is possible to test a digital twin of a car at a virtual testing ground, using real loads coming to the body during various types of operation and road surface [3, 4].

In this paper, two approaches to the study of the fatigue life of a car body were considered. These approaches can significantly reduce time and increase the speed of development of the vehicle design, allowing to abandon some of the expensive and resource-intensive real tests.

2. Determination of fatigue life of a car

The durability of a structure can be calculated using two methods that are radically different in approach way. The development of defects can be simulated, taking into account the boundary conditions in a particular structural element, as well as the rebuilding of the mesh in the model with a gradual increase in the defect size [5]. In addition, one can evaluate the degree of damage to the material in the car components without explicitly considering defects and their growth [6]. Since the goal of this work is to evaluate the fatigue properties and not the sequence of defects development, a method with damage assessment was chosen as providing all the results necessary for the development and, besides, significantly faster in terms of computing resources.

As initial data, there needs information on the elastic and fatigue properties of structural materials, including the SN curve [6]. A model is needed that describes the geometry of the structure with detailed modeling of welded joints, as well as information on the points of attachment of the suspension to the car body, namely, the stress-strain state of the structure under the action of loads applied at the corresponding interface points [7, 8].

Static strength analysis of the car body structure under the action of unit loads applied to the interface points of the front suspension is carried out based on the available initial data. Subsequently, based on the analysis data, areas of the structure can be identified that bear heavy loads and require special attention when fatigue life is analyzed. Besides, it seems possible to identify structural components that are likely to cause residual deformations during operation. Such areas should be the subject of low-cycle fatigue studies. When describing the process of multi-cycle fatigue, the SN curve is used, which explains the ability of a material to resist multi-cycle fatigue [9]. The SN curve describes the dependence of the peak voltage in the cycle on the total number of cycles to the complete failure of the element at the maximum voltage level, which is constant during the entire process of loading the element. For various materials, the SN curve has a different form in the area of multi-cycle fatigue.

A. Model Problem

Before solving the main problem, the model problem was considered to adjust the parameters of the solver and assess the adequacy of the approximation of the fatigue properties of materials according to known mechanical and physical properties. To solve the model problem, the design shown in Fig. 1, consisting of two pipes welded together from steel of the FE510 grade, was considered, the characteristics of which are presented in Table I, one of which was applied along the edges with a harmonic law force Wohler A 1858 Report on test of the Royal Niederschlesich-Märkischen Railway made with apparatus for the measurement of the bending and torsion of railway axles in service (in German) Zeitsch.Bauwesen (N8) pp 642-651 [10]. In total, several tests were performed at different values of the amplitude of the applied load.

![Figure 1. Dimensions of the test specimen](image)
To solve the model problem, a finite element model was constructed that takes into account the elastic properties of the material. The weld between the two pipes was modeled using shell finite elements. The appearance of the model is shown in [11].

Using the MSC Nastran software package, the problem was solved in a linear formulation for studying the stress-strain state under the action of a load of 1 kN. The stresses arising in the structure when such a load is applied are shown in [12].

Using the obtained stress-strain state and the fatigue properties of the material, the life expectancy of the structure under the action of periodic loads at three different loading amplitudes: 80 kN, 60 kN, and 40 kN were obtained.

The comparison of the number of cycles before fracture obtained in a numerical solution and during the test are given in Table II.

Table 1. Characteristics of the material FE510

| Material                  | Fe510 |
|---------------------------|-------|
| Yield strength            | 345 MPa |
| Strength limit            | 500 MPa |
| Fatigue limit             | 250 MPa |
| The number of fatigue limit cycles | 1.0e+5 |
| Elastic modulus           | 210000 MPa |

Table 2. The comparison of numerical modeling with experimental data

| Load, kN | Numerical solution, number of repeats | Test solution, number of repeats | Difference % |
|----------|--------------------------------------|---------------------------------|--------------|
| 80       | 37500                                | 35000                           | 7            |
| 60       | 86000                                | 80000                           | 7            |
| 40       | 315000                               | 300000                          | 5            |
The difference between the results of numerical modeling and experimental data was less than 7 percent. Thus, the calculation parameters and the general approach to solving the model problem can be used further under assessing the fatigue life of a car body, and the fatigue properties of materials can be approximated based on physical and mechanical properties.

B. The computational model of the car

To conduct a numerical assessment of fatigue life, a digital twin of the car under study was developed, which includes the following structural components that can influence the parameters under study: body, front suspension frame, dashboard reinforcement, front bumper reinforcement, rear bumper reinforcement, car front, roof, windshield and rear window. The calculated model of the car body is shown in [13].

![Figure 4. The computational model of the car body.](image)

The presented body model was prepared for static calculation in the MSC Nastran finite element analysis package and calculation of fatigue life in the MSC Patran/Fatigue package and contains:

- 1034640 shell elements
- 67234 volumetric elements
- 644 absolutely rigid connections (finite elements RBE2)
- 77603 interpolation links (finite elements RBE3)
- 28 beam elements

Shell elements are used to model structural parts of the car body, for example for stamped metal parts, as well as for windshields and rear windows.

Volumetric elements are used in the task of spot welds, adhesive joints, and cast parts of the car. Elements of an absolutely rigid connection are used to specify bolted connections between parts, since when an absolutely rigid connection is used, the relative location of the nodes included in it does not change, they move as a whole. For the simulation of bolted joints, the properties of an absolutely rigid connection are set so that to transfer only translational degrees of freedom of the nodes, but not rotational ones, providing the connection model more adequate to the real object. In addition, around bolted joints, an additional element area is added to an absolutely rigid connection for reducing the edge effect.

The disadvantage of absolutely rigid connections is the stiffness they introduce into the model. If in the case of bolted joints this modeling method is quite physical, then for modeling spot welds where relative movement of different sections of the joint is possible, modeling using an absolutely rigid connection can introduce a discrepancy in compared to reality [11].

To simulate spot weld, a volumetric finite element is used, connected to the mesh using elements with interpolation coupling.

Interpolation link distributes balanced the load of the control node between the nodes managed, while not introducing additional non-physical rigidity into the connection model itself, resulting in a positive effect on the accuracy of the model.

Elements of the beam type are used in the model to specify bolted connections in the front and rear axle boxes. Due to these bolted connections experience the highest loads when connecting the subframe to the body, they need to be set using deformable elements, and not an absolutely rigid connection.
By analogy with spot welds, the connections to the parts to be glued were also modeled using interpolation links. Modeling of welded lines of linear arc welding was carried out using shell elements. The materials used in the car body and their distribution are shown in 0. In the future, the developed digital twin of the car body can be used both for assessing fatigue life by a simplified method and for using a virtual test site.

Figure 5. Breakdown of materials in the car body model

C. Determination of durability by stress level

To explain multi-cycle fatigue, it is generally accepted to describe durability as a function of the characteristics of a stress cycle. The fatigue curve is called the Weller curve, and the approach in foreign literature is called the SN approach, by the name of the abscissa and ordinates on the Weller curve (Stress vs. Number of cycles) [12].

Moreover, the Weller curve has a different characteristic form for different materials and is described by the corresponding dependence proposed by Baskvin.

Therefore, the fatigue curve equation of materials with a physical fatigue limit has the form:

\[
N = \begin{cases} 
\frac{\sigma_{R}^{m}N_{G}}{\sigma_{\text{max}}^{m}}, & \sigma_{\text{max}} \geq \sigma_{R} \\
\frac{\sigma_{R}^{m}N_{G}}{\sigma_{\text{max}}^{m}}, & \sigma_{\text{max}} < \sigma_{R} 
\end{cases}
\]  

(1)

The equation of the fatigue curve of materials with two inclined sections in the fatigue curve takes the form presented in formula 2.

\[
N = \begin{cases} 
\frac{\sigma_{R}^{m}N_{G}}{\sigma_{\text{max}}^{m}}, & \sigma_{\text{max}} \geq \sigma_{R} \\
\frac{\sigma_{R}^{m}N_{G}}{\sigma_{\text{max}}^{m}}, & \sigma_{\text{max}} < \sigma_{R} 
\end{cases}
\]  

(2)

If the fatigue curve does not have an inflection when the number of cycles is increased, the Baskvin dependence takes the form presented in formula 3:

\[
N = \frac{\sigma_{R}^{m}N_{b}}{\sigma_{\text{max}}^{m}}
\]  

(3)

where \( m \) – the slope of the first section of the fatigue curve, \( m_{1} \) – the slope of the second section of the fatigue curve, \( N_{G} \) – the abscissa of the inflection point of the fatigue curve, \( \sigma_{R} \) – fatigue limit based on \( N_{G} \), and \( N \) – the number of cycles to failure. [13]

The above relations describe the SN curve only for a given asymmetry coefficient of the loading cycle. Thus, a complete description of the behavior of the material requires a set of SN curves obtained for different values of the asymmetry coefficient.

This approach may be inconvenient because when studying the structure fatigue resistance loading cycles can occur with a wide range of asymmetry coefficient values, in addition, the SN curve is an experimental curve and an experiment is required for each asymmetry coefficient value. To simplify this approach, most of the experimental data are presented for a symmetric loading cycle, after which the loading history is reduced to a symmetrical stress cycle with respect to damage using relation 4, constructed on the basis of the Goodman dependence, after which relations 1 - 3 are used to directly evaluate the durability.
\[
\sigma_{\text{equiv}} = \begin{cases} 
\frac{\sigma_{pp}^r}{\sigma_m}, & \sigma_m \geq 0 \\
1 - \frac{\sigma_m^{pr}}{\sigma_m}, & \sigma_m < 0 
\end{cases}
\] (4)

D. Material properties used in the car body
Since the S-N method based on the stress amplitude is used to calculate fatigue life, to set the properties of the material requires knowledge of the S-N curve characterizing the number of cycles to failure that the test material specimen bears at a given stress amplitude [12]. High-cycle fatigue breakdown can be predicted with high accuracy using the S-N method, which is ideal for analyzing the durability of structural elements of a car body under operational loads. However, using this technique, it is impossible to reliably predict low-cycle fatigue failure caused, for example, by suspension breakdown. So, during the breakdown of the suspension, the stresses in the body structure can reach the yield strength and complete failure of the structure is possible after several loading cycles. Table III shows the elastic and fatigue characteristics of the materials used in the car body model. Used fatigue properties of materials were obtained on the basis of physical and mechanical properties and the relationships between them and fatigue properties.

At the later stages of designing a car, it makes sense to use the fatigue properties obtained using fatigue tests of material samples [14]. This will level out the inaccuracies in the results caused by the discrepancy between the used, approximated properties of materials with real brands of metals and batches that are used in the manufacture of the body.

| Material       | DX56D/10X | HX260LAD | HX420LAD | HCT780X | ALU  |
|----------------|-----------|----------|----------|---------|------|
| Yield strength | 210 MPa   | 227 MPa  | 481.6 MPa | 500 MPa | 130.2 MPa |
| Strength limit | 270 MPa   | 372 MPa  | 598.8 MPa | 800 MPa | 200 MPa |
| Fatigue limit | 121 MPa   | 155.5 MPa | 307.2 MPa | 360 MPa | 60 MPa |
| The number of fatigue limit cycles | 2.0e+6 | 7.54e+5 | 2.06e+5 | 2.0e+6 | 1.0e+7 |
| Elastic modulus | 210000 MPa | 210000 MPa | 210000 MPa | 210000 MPa | 70000 MPa |

E. A simplified method for assessing fatigue life
Evaluation of fatigue life by a simplified method is based on the results of static strength analysis under the action of unit loads applied to the attachment points of the front suspension struts.

![Figure 6. Boundary conditions and loads in static analysis](image)

The translational degrees of freedom of the car body are fixed in the attachment points of the rear suspension according to 0. An analysis of the stress-strain state in two design cases of application of a single static loading is carried out: alternately, at the points of attachment of the left and right pillars of the front suspension to the body, a unit force is applied in the direction of the z-axis.
When analyzing the durability, the force along the $z$ axis calculated by the formula below is applied to the left and right struts of the front suspension with a phase difference $\pi$ according to the harmonic law:

$$ P = \frac{R_{st} \cdot B_f}{4 + L} $$  \hspace{1cm} (5)

where: $R_{st}$ - the static load on the front axle, $B_f$ – front wheel track equal to 1700 mm, $L$ – the meter mounting lever. (4)

$$ R_{st} = 0.49 \cdot P_{calc} \cdot g = 2588 \text{ kg} $$  \hspace{1cm} (6)

The loading scheme used in the analysis of durability by a simplified method consists of four stages and, taking into account the meter-mounting lever, is presented in Table IV.

### Table 4. Loading by the simplified method

| Loading stage | Amplitude loading, N | Frequency of loading cycles, Hz | Number of loading periods |
|---------------|----------------------|---------------------------------|--------------------------|
| 1             | 0.4*P=2125           | 5                               | 50000                    |
| 2             | 0.8*P=4251           | 4                               | 100000                   |
| 3             | 1.2*P=6377           | 4                               | 100000                   |
| 4             | 1.6*P=8503           | 3                               | 50000                    |

The sequence of test loads presented in Table IV reflects the full cycle of operation of the vehicle under study, corresponding to 200,000 kilometers. Thus, all fatigue breakdown occurring in the first test cycle is undesirable.

When conducting a virtual durability test, the fatigue module of the MSC Patran software package calculates the stresses that occur in the structure under the action of forces applied to the interface points, superposing the results of the stress-strain state under the action of unit loads. This method, using a superposition of stress-strain states, can be applied in cases where a linear-elastic formulation of the problem is used. In this case, this is so, since we are talking about multi-cycle fatigue and the effects that lead to stresses significantly lower than the yield strength of the materials used.

Based on these data, and taking into account the fatigue properties of the material, it becomes possible to indicate the areas where failure is most likely to occur due to cyclic exposure.

**F. Advanced fatigue life assessment methodology**

Fatigue life using an advanced method accurately estimates the fatigue life of a body structure and welded joints in comparison with the simplified method.

As in a simplified technique, stress-strain states from the action of unit forces and moments are used as input data. However, if using the simplified methodology, unit loads were applied only to the attachment points of the front suspension struts, and then in the advanced methodology, all interface points are exposed to single impacts through which loads are applied to the body during vehicle operation. So in the advanced methodology for assessing durability, interface points are used:

- Front and rear connection points of the upper arm of the front suspension with the support of the front pillar of the car;
- The point of connection of the front suspension strut with the front support cup of the car body;
- Front and rear connection points of the silent block with the rear subframe bolt;
- The connection point of the rear suspension strut with the rear support cup of the car body;
- The attachment point of the anti-roll bar to the subframe;
- The front mounting point of the lower front suspension arm to the subframe;
- The front point of attachment of the lower arm of the front suspension to the subframe;
- Steering rack mounting point;
- The point of contact with the support of the transfer gearbox (TGB).

For all the indicated points, including symmetrical points, load profiles were used that correspond to the real road tests of the prototype car for an advanced assessment of the fatigue life of welded joints of a digital twin of a car using a virtual test site. These load profiles correspond to ten different types of maneuvers for five different types of pavement. This type of loading can be considered forced with an equivalence coefficient equal to 4.36. Thus, this loading can be considered forced, since 1 kilometer passed by this method corresponds to 4.36 kilometers of actual use.

A complete list of loading profiles and their parameters is presented in Table V.

Given the equivalence coefficient, approximately 9.2 repeats of the loading cycle shown in Table V are required to evaluate fatigue life after 2,000,000 kilometers.

**G. Methodology for assessing changes in body stiffness**

Body stiffness was chosen as a characteristic used to assess the consumer properties of the car. This value influences both the driving performance of the car and its further deterioration [15].

Spot welds failed at this stage of fatigue loading were sequentially removed to assess the dynamics of changes in body stiffness from a finite element model of a car. After that, torsional and bending stiffness were evaluated. To calculate torsional stiffness, the digital twin of the car was fixed in the areas of attachment of shock absorbers of the rear suspension, and multidirectional force was applied along the z-axis to the right and left supports of the shock absorbers of the front suspension.

A diagram of the application of forces and boundary conditions for assessing torsional stiffness is presented in 0, where the prohibition of translational degrees of freedom along the x and y axes is introduced at points 1 and 2, and the ban on movement over all translational degrees of freedom is introduced at points 3 and 4.

**Table 5. Types of roads for fatigue life calculation**

| Road type              | Load type                                      | The required duration, km | One cycle length, km |
|------------------------|------------------------------------------------|---------------------------|----------------------|
| High-speed mountain road| Acceleration; Braking; Double rebuild          | 3700                      | 0.74                 |
| Smooth pavement        | Sinus load; Hit a curb; Hit a curb in reverse; Exit from the curb during braking; Exit from the curb with one wheel during braking; | 600                       | 0.12                 |
| Profied pavement       |                                                 | 350                       | 0.07                 |
| Access road            |                                                 | 350                       | 0.07                 |
| **Total**              |                                                 | **5000**                  |                      |

A diagram of the application of forces and boundary conditions for assessing torsional stiffness is presented in 0, where the prohibition of translational degrees of freedom along the x and y axes is introduced at points 1 and 2, and the ban on movement over all translational degrees of freedom is introduced at points 3 and 4.

**Figure 7.** The calculation of the torsional stiffness of the car body.

Torsional stiffness is evaluated according to formula 7:
where \( F \) – applied force, \( l \) – half the distance between points 1 and 2, \( w_1 \) and \( w_2 \) – vertical displacements at points 1 and 2 of the application of forces, respectively.

Similarly, the body flexural rigidity is estimated: fixation is carried out according to the translational degrees of freedom at points 1, 2, 3, 4, and at the attachment points of the seats 5, 6, 7, 8, loading along the z axis is performed. The layout of loading points and boundary conditions is shown in [5].

The value of bending stiffness is estimated by the formula 8:

\[
c = \frac{4F}{<w>}
\]

where \( F \) – force intensity applied at points 5, 6, 7, 8; and \(<w>\) - average vertical displacement at these points.

\[ c = \frac{2Fl}{\arctg\left(\frac{w_1-w_2}{2l}\right)} \]  

\[ (7) \]

**Figure 8.** The calculation of the bending stiffness of the car body.

### 3. Tests results of fatigue life of the car body

**A. Car body test results using a simplified technique**

The implementation of the virtual tests described in the work was carried out on the basis of the platform CML-Bench\(\text{TM} \) (digital platform for the development of digital twins and an activity management system in the field of digital design, mathematical modeling and computer engineering, development of the engineering center CEC SPbPU). The calculations were performed using the supercomputer of the center "Polytechnic" SPbPU.

According to the results of a calculation check of the fatigue life of body structural elements using a simplified method, the integrity of most parts is maintained. However, minor fatigue breakdown zones are observed in motor shield reinforcements.

The analysis results showed from the entire cloud of spot welds of the digital twin of the car, shown in Fig. 9 and containing 9281 points, the main sites of the breakdown of the spot welds were in the area of attachment of the front and rear struts of the shock absorber and the area of reinforcements of the motor shield. In total, during the evaluation of the operation period, 175 spot welds were failed, which corresponds to less than two percent of their total number.

**Figure 9.** Cloud of all spot welds of the car.
[15] shows the main sites of the breakdown of welded joints are located around the most highly loaded areas, which are used to apply loads when calculating by a simplified method.

![Figure 10](image1.png)

**Figure 10.** The main site of the breakdown of weld points in the areas of attachment of the front and rear shock absorbers using a simplified modeling technique.

In addition, the sites of the breakdown of spot welds were located in areas that were used to set the boundary conditions: rear shock absorber supports. Additionally, there is a breakdown of the welded joints in the parts connecting the motor shield with the fastening areas of the front shock absorbers. The image of failure of spot welds in this area is shown in [12].

![Figure 11](image2.png)

**Figure 11.** Fatigue breakdown of spot welds in the area of the motor shield.

**Expanded body test results**

Compared to the results of the simplified method, the advanced method shows more areas of failure of the body parts themselves, rather than welded joints. So, the failure is observed in not only the areas of application of loads and boundary conditions (the area of attachment of the front and rear struts of shock absorbers, subframes, suspension arms) but also relatively lightly loaded parts of the car, such as the bracket for attaching a high-voltage battery. However, the main site of the breakdown is still in the areas where the struts are mounted. What is shown in [9].
Figure 12. The main site of the breakdown of spot welds in the areas of attachment of front and rear shock absorbers in the advanced modeling method.

In addition, single sites of the breakdown were identified in the front and rear spars.

B. Assessment of the drop in body stiffness during fatigue breakdown of welded joints

A comparison of the torsional and bending stiffness of the body was made for the entire production cycle, with a step equal to 10% of the cycle duration. Figure 13 shows a comparison of the stresses arising in the car body during the bending stiffness test before and after passing the fatigue test and Fig. 14 a similar comparison is presented for the body torsional stiffness test. As can be seen from the presented comparison, the fatigue fracture of weld points during the first operating cycle does not significantly influence the location of the centers of high stresses during testing for bending and torsional stiffness. However, it significantly influences the magnitude of stresses: for a body that has not undergone fatigue, the maximum stresses in a bending stiffness test are 110 MPa, 200 MPa - in torsional stiffness, and after a durability test, the maximum stresses are 120 MPa for bending testing and 350 MPa for torsional, thus showing the growth of 10% and 75%, respectively.

There is a significant increase in stress values, especially in the areas of fastening of the front struts of shock absorbers that experience significant loads during operation. This can lead to accelerated fatigue breakdown of the car body structure during further operation.

Figure 13. Comparison of stresses arising in the body when bending stiffness is tested.
**Figure 14.** Comparison of stresses arising in the body when torsional stiffness is tested.

The maximum displacement along the z axis during the bending stiffness test was 0.77 mm for a body without a fatigue breakdown of welded joints and 1 mm for a car body after the first operating cycle. [6] presents a comparison of movements along the z axis that occur in a car body when bending stiffness is tested, before and after passing the test for fatigue life.

**Figure 15.** Comparison of movements along the z axis during the test for bending stiffness of the body.

The maximum displacement along the z axis during the torsional stiffness test was 1.25 mm for a body without fatigue breakdown of welded joints and 2.6 mm for a car body after the first operating cycle. [4] shows a comparison of the displacements along the z axis that occur in a car body during a torsional stiffness test before and after passing the fatigue test.
Figure 16. Comparison of movements along the z axis when torsional stiffness of the body is tested. Based on the results of a numerical calculation of the bending and torsional stiffness of the body, graphs of changes in these parameters were constructed taking into account the fatigue breakdown of the spot welds of the body. These graphs are shown in [1] in comparison with the fracture schedule of spot welds.

In total, the bending stiffness of the car body decreased by 30 percent during one operation cycle: from 11,600 N/mm to 8,100 N/mm. Torsional stiffness decreased by more than 45 percent: from 26,700 Nm/deg to 14,600 Nm/deg.

Figure 17. Change in body stiffness during the failure of spot welds.
C. Comparison of the areas of failure obtained using simplified and advanced techniques with real-life tests

The results obtained in the previous paragraphs showed the main site of fatigue breakdown according to the assessment of the digital twin of the car are:
- In the area of fastening the front shock absorbers;
- In the area of the motor shield;
- In the area of mounting the rear shock absorbers.

Resource tests conducted for a real prototype of the car under development allow comparing the results obtained for the digital twin using a simplified technique and a virtual test site and the results obtained in reality. This comparison enables to assess the applicability of the used methods for further development and subsequent projects.

During the inspection of the prototype body, the failure of the body structure elements and welded points were found in the following areas of the structure:
- Area of fastening the front shock absorbers;
- Spot welds of the motor shield reinforcement.

A comparison of the areas of actually occurring and predicted fatigue breakdown is shown in Fig. 18 [2].

![Figure 18. Comparison of fatigue breakdown areas.](image)

At the same time, the area of fatigue breakdown was not found in the supports of the rear shock absorbers during real tests.

4. Conclusion

Various approaches to solving the problem of assessing the fatigue life of the car body and its welded joints were considered in this paper. Despite the conservatism in the results obtained using these techniques, the main site of fatigue breakdown coincide with those occurring during a real test of a car. At the same time, the considered methods can significantly reduce the cost and speed up the process of developing the car in comparison with real tests, especially in the initial stages. This helps to assess the location of potentially problematic areas of the body in terms of fatigue life and the decrease in consumer characteristics of the car during its operation.

The obtained results suggest the prospect of using these methods for assessing fatigue life in car development.
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