The Method of Designing the Superstructure of the Car Body Based on the Requirements of Low-Speed Collisions

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Abstract. This article discusses the method of designing the superstructure of the car, based on the requirements of insurance companies, on the example of the simulation of low-speed collisions. The experimental and calculated data of the stress-strain state of the automobile body are compared. The method is based on the use of discrete models. The method takes into account the requirements for passive safety and rigidity of the car body at the first stages of design. This is the reason for reducing the time spent on designing a new car.

Introduction

The car is characterized by technical parameters and consumer properties. One of these properties is the ratio of the cost of restoring the car after the accident to the total cost of the car.

To estimate the cost of restoring the car, the vehicle is tested at low-speed collision. The damages of the car are received as a result of test defines the volume of recovery work and the list of the details which are subject to replacement.

The cost of spare parts and car repairs is calculated based on the analysis of damage to the car. The parameter of the recovery costs relative to the initial price is used as the estimated parameters. These indicators are used by insurance companies to determine the cost of the insurance amount of the car. The low cost of restoring the car makes it more attractive for the consumer.

In Europe, the USA and in Canada, there are requirements for low speed collisions of the vehicle with another vehicle or a non-deforming barrier. As a result of these tests, the vehicle shall be able to maintain the functionality of the bodywork, critical interior systems and lighting equipment in a condition sufficient for safe driving after a collision.

All currently known requirements for testing methods of light vehicles at low speeds are divided into legislative and consumer requirements (requirements of insurance companies) [1, 2], presented in table 1.

Tests of insurance companies impose more stringent requirements to the car design than the requirements of the legislation. Let's consider the case of testing the car taking into account the requirements for low-speed collisions on the example of the Dunner tests used by European insurance companies [1, 2, 3].

Main part

The paper describes the method of testing vehicles for Dunner tests, analyzes the vehicles of different classes and proposes recommendations for the use of bumper beams and special energy-absorbing structural elements.
Insurance companies use a class of insurance to determine the amount of the premium, which is defined as a function of two variables: the value of the cost of repairing a damaged car and the index of the frequency of damage (a statistical value of the Association of insurance companies, indicating how often a particular type of car falls into an accident).

### Table 1. Requirements for low-speed collisions

| The test schema | IIHS tests | Dunner tests |
|-----------------|------------|--------------|
| The test schema | Front impact | Front impact with overlap |
| IHS tests       | A front impact on the barrier | The rear-end impact |
| Dunner tests    | The rear-end impact |

| Offset  | 100% | 100% | 40% | 40% |
|---------|------|------|-----|-----|
| Angle of inclination | 0° | 30° | 0° | 0° |
| Barrier | Not deformable | Not deformable | Fixed barrier | Mobile deformable barrier with a mass of 1000 [kg] |
| Speed   | 7.96 ±0.24 [km/h] | 7.96 ±0.24 [km/h] | 7.96 ±0.24 [km/h] | 15 ±1.0 [km/h] |
| The condition of the vehicle | Unloaded the car. The tank is filled on 90%. | The mass in running order plus 75 [kg] |

The main evaluation criterion:
- deformation of the body shall not exceed 9.5 [mm];
- the total skew / displacement shall not exceed 19 [mm] of the original contour line.
- holes and gaps should not exceed a diameter of 10 [mm], and placed in the critical area;
- gouging or cracking of the material should not be in length (diameter) of more than 20 [mm] and placed in the critical zones.
- The cost of repair, replacement, adjustment or finishing operations should be taken into account when assessing damage. The results should be published to consumers.

The following method of determining the insurance class is used.

The first step is to determine the average $SD$ damage value. Based on the data obtained after the vehicle tests at low speeds [4], determined by substituting the data in the formula (1).

$$SD = \frac{(54\% D + 16\% F + 30\% C) \cdot 100 \cdot 1.13}{AC},$$

(1)
where $D$ is the cost of repairing damage to the front of the car; $F$ is the cost of repairing damage to the side of the car; $C$ is the cost of repairing damage to the rear of the car; $AC$ is the average cost of repairing all damage recorded by insurance companies; 1.13 is the coefficient (index) of the estimated overhead costs as a result of an accident (for example, the cost of renting a car or towing).

The second stage is determined by the index of the frequency of damage to $SH$. The damage frequency index is a statistical value of the Association of insurance companies, indicating how often a particular type of vehicle (with reference to the trademark) falls into the accident. These figures are not disclosed to the public, and each manufacturer’s information is available only on the values of their own brands.

At the third stage it is necessary to determine the demand for damage $SB$ [5]. According to the data obtained during the first and second stages, the demand for damage is determined by the formula (2):

$$SB = \frac{SD \cdot SH \cdot 0.94}{100},$$

where 0.94 is the correction for the index of the frequency of damage (increasing the factory recoverability of vehicles).

In the fourth stage, the $VK$ insurance class is defined. The decision to assign a certain class of insurance to a car is made on the basis of table 2.

| $SB$  | 10-39 | 40-49 | 50-59 | . . . | 130-139 | 140-149 | etc. |
|-------|-------|-------|-------|-------|---------|---------|------|
| $VK$  | 10    | 11    | 12    | . . . | 20      | 21      | etc. |

The average cost of restoring the car is the highest for small-class cars, the lowest for $SUV$ cars. The reason is that for different classes, the prices of cars differ in ascending order, the lowest for small cars, the highest for $SUV$ class. The nature of different class cars damages is different.

Reducing the cost of repairing damage to the car in a low-speed frontal collision can be achieved with the help of special energy-intensive removable elements, which would prevent deformation of non-
removable parts of the car body [6, 7]. The projection begins with the determination of the power structure of the front of the vehicle for efficient energy quenching during a frontal collision using the Dunner test method. It is necessary to minimize the deformation of the fixed units of the car. Figure 1 shows two characteristic schemes of energy dissipation. In the first scheme, the damping of the impact energy occurs due to the deformation of the bumper beam. In the second scheme, the damping of the impact energy occurs due to the deformation of the crash box. The main parameter for determined is the length $L$ of the energy-intensive element.

Determination of the length of the energy-intensive element for schemes of energy dissipation is the same. The effective deformation is determined from the equality of the deformation energy of the energy-intensive element of the impact energy and is 70% of the total length of the element.

Figure 2 shows the results of calculating the length of the energy-intensive element depending on the vehicle mass and the critical force of the front spar.

![Figure 2](image-url)

**Figure 2.** The total length of the energy-intensive element, depending on the weight of the vehicle and the critical load of the front spar

One of the first stages of the synthesis of the power circuit of the perspective car is the moderation of low-speed collision and the choice of the optimal design of the bumper beam [8, 9].

Traditionally used material for the manufacture of the bumper beam is steel. The main advantages of using steel are low cost and high energy consumption. But such weeks of steel as a specific weight, and the complexity of the process lead to the need to search for new materials.

Some manufacturers for the production of bumper beams are used aluminum, which has a relatively low specific weight, ease of manufacturing technology (molded beam) and sufficient in comparison with the steel specific energy consumption [10].

In this work a study was made of the energy-absorbing properties of aluminum bumpers. As the base was adopted by a power circuit of a passenger car with typical governmental power elements of the frame.

The impact of the vehicle on the non-deformable barrier at a speed of 15 km/h is carried out according to the requirements of the Dunner test with a 40% overlap.

The analysis was carried out in several stages:

1. Dynamic calculation for the six aluminium bumper beams with different cross sections (Fig. 3). This step made it possible to evaluate the dependence of the energy absorbing properties on the cross-sectional configuration.

2. Dynamic calculation of the bumper beam type "a" (Fig. 3) with its division into characteristic sections (longitudinal zones – horizontal and vertical ribs and four transverse zones, fundamentally different in terms of energy absorption). This stage allowed to reveal the areas of the beam having the greatest energy-absorbing ability and to estimate energy intensity of each part of the beam.
3. Dynamic calculation of the bumper beam type "a" (Fig. 3) with a round neckline opposite the spar. This variant of calculation allowed to estimate losses in energy absorption which need to be compensated by input of an additional element-crash-box.

4. Dynamic calculation of the bumper beam type "a" (Fig. 3) with inserted aluminum crash box in the area opposite the spar. The calculation was made for several types of crash boxes.

5. Dynamic calculation of the bumper beam type "a" (Fig. 3) with different set of vertical and horizontal fins thicknesses. At this stage, a search was made for the best variant of the structure with optimal rigidity, which on the one hand is the most energy, on the other hand is small enough to, completely deformed, not to reduce the force on the side of the body spar.

6. Dynamic calculation of a straight bumper beam with a cross section similar to a bumper beam type "a" cross section. At this stage, the influence of the beam on its energy-absorbing properties was assessed.

The criterion for assessing the moment of loss of stability of the spar was the load increase in the cross section of the spar. Vehicle weight is taken equal to 1150 kg the vehicle Speed is 4.17 m/s. Thus, the impact energy is 10 kJ.

Figure 3. Types of beam cross section of the bumper

The thickness of the ribs of the beams of all types was set to 2 mm. All types of bumper beams have a height of 100 mm. Dynamic calculation of the bumper beams of different cross sections allowed to determine:

1. The moment of loss of stability of the front spar (the moment of end of effective energy absorption of the bumper beam).
2. The energy absorbed by the bumper before the deformation of the spar.
3. The residual kinetic energy and vehicle speed at the time corresponding to the end of the effective energy absorption of the beam.

The results of the calculations are summarized in table 3.

| The beam type | The moment of buckling of the spar, [ms] | The energy absorbed by the bumper, [kJ] | The residual kinetic energy, [kJ] | Residual vehicle speed, [ms] |
|---------------|----------------------------------------|----------------------------------------|---------------------------------|----------------------------|
| a             | 26                                     | 5.81                                   | 4.15                            | 2.69                       |
| b             | 25.5                                   | 2.40                                   | 7.60                            | 3.64                       |
| c             | 24                                     | 3.51                                   | 6.49                            | 3.36                       |
| d             | 23                                     | 4.67                                   | 5.33                            | 3.04                       |
| e             | 24                                     | 3.85                                   | 6.15                            | 3.27                       |
| f             | 25                                     | 5.54                                   | 4.46                            | 2.79                       |

As a result of the calculations, it becomes obvious that with a wall thickness of 2 mm, the bumper beam with a section of type "a" is the most energy-intensive of the considered.

To assess the energy intensity of each part of the beam, a dynamic calculation was made [11] with the beam split into 16 parts. Each rib, located in the zone of 40% overlap, was divided into three parts: the inner (closest to the axis of symmetry), the middle (opposite the spar) and the last part of the beam.
The part of a beam which is not in an overlap zone makes is other group. The outer horizontal ribs were grouped in pairs (Fig. 4). The results of the calculation are given in table 4.

![Figure 4](image.png)

**Figure 4.** The separation of beams on the characteristic

| № parts | Absorbed energy, [J] | % of the total energy |
|---------|-----------------------|-----------------------|
| 1       | 19                    | 0.33                  |
| 2       | 262                   | 4.50                  |
| 3       | 172                   | 2.96                  |
| 4       | 67                    | 1.15                  |
| 5       | 1035                  | 17.81                 |
| 6       | 1585                  | 27.27                 |
| 7       | 340                   | 5.85                  |
| 8       | 493                   | 8.48                  |
| 9       | 710                   | 12.21                 |
| 10      | 235                   | 4.04                  |
| 11      | 141                   | 2.42                  |
| 12      | 133                   | 2.28                  |
| 13      | 22                    | 0.38                  |
| 14      | 271                   | 4.66                  |
| 15      | 260                   | 4.47                  |
| 16      | 66                    | 1.14                  |
| All     | 5811                  | 100                   |

**Table 4.** The distribution of internal strain energy in parts I-beam bumper

Analysis of the obtained results shows that the main share of energy absorbs part of the beam, located opposite the spar (middle).

The distribution of internal energy of deformation along the ribs of the bumper beam is shown in table 5.

Thus, it becomes obvious that the most energy-intensive are the outer horizontal ribs.

| Rib name            | Part number | Absorbed energy, [J] | % of the total energy of the middle part |
|---------------------|-------------|-----------------------|-----------------------------------------|
| External vertical   | 3           | 172                   | 6.01                                    |
| External horizontal | 6           | 1585                  | 55.42                                   |
| Internal horizontal | 9           | 710                   | 24.83                                   |
| Internal vertical   | 12          | 133                   | 4.65                                    |
| Rear                | 15          | 260                   | 9.09                                    |

**Table 5.** Distribution of internal energy of deformation along the ribs of the bumper beam
For inclusion in the power scheme of a direct beam of bumper geometry of spars has been changed. Therefore, it is difficult to make an objective comparative assessment of energy absorption. The moment of completion of effective energy absorption of the beam was determined by changing the energy absorption of spars [12, 13]. Energy absorbed by the curved beam – $E (29 \text{ ms}) = 6.4 \text{ kJ}$, energy absorbed by a straight beam – $E (31.5 \text{ ms}) = 6.6 \text{ kJ}$.

Figure 5 and figure 6 show the diagrams of the energy absorbed by the beams depending on the time and deformation of the beam. The total energy intensity is higher in the curved beam, useful energy intensity is higher in the straight line. This is the reason why it is impossible to give an unambiguous assessment of the change in energy intensity.

This curvature of the bumper beam has no significant effect on the energy-absorbing properties.

**Summary**

According to the results of the analysis, the following conclusions can be drawn:
1. The highest energy absorption when the wall thickness of 2 mm has a beam type "a" (Fig. 3).
2. The most energy-intensive part is the outer horizontal ribs;
3. The hole in front of the spar affects the energy absorption 17%;
4. The use of an additional energy-extinguishing element within the framework of the considered beam design is ineffective;
5. The best ratio of the thickness of the ribs of the beam type "a" – horizontal ribs are 2.5 mm, the vertical ribs 2 mm;
6. The curvature of the beam in the considered power scheme has no effect on the energy-intensive properties.

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