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Sobirov Bekzodbek Akhmadjonovich
PhD Student, Materials Science Department, Tashkent State Technical University Named After Islam Karimov, Uzbekistan

Djumabaev Alijon Bakishevich
Professor, Materials Science Department, Tashkent State Technical University Named After Islam Karimov, Uzbekistan

Sobirov Mamurjon Marufjanovich
Senior Lecturer, Automative Engineering Department, Andijan Machine-Building Institute, Uzbekistan

Corresponding Author:
Sobirov Bekzodbek Akhmadjonovich
Phd Student, Materials Science Department, Tashkent State Technical University Named After Islam Karimov, Uzbekistan

Choosing modern methods of modelling when operating Isuzu buses suspension mechanisms

Sobirov Bekzodbek Akhmadjonovich, Djumabaev Alijon Bakishevich and Sobirov Mamurjon Marufjanovich

Abstract
The work presents the results of the experimental research into modelling of operation of Isuzu buses suspension mechanisms using Solid Edge ST and ANSYS 12.0 computer programs. Shock absorbers were calculated by the method of finite elements using Solid Edge ST and ANSYS 12.0 programs.

Keywords: Automobile, carriage spring, bending, suspension, stress, stiffness, shock absorber, bowing, acceleration, deformation

Introduction

The operating modes and conditions of the suspension parts are influenced by many factors. These include: road surface, car acceleration, various loads, suspension design, etc.

When designing the Isuzu SAZ NP 37 car, special attention was paid to the compliance of the new car design with the operating conditions. This compliance is essential for cost-effective car operation. It was taken into account that the car should work reliably during operation in various climatic conditions on roads with improved pavement.

The spring suspension of the Isuzu SAZ NP 37 car was very stiff, and the significant vertical accelerations associated with it created unfavorable conditions for the driver to work, tiring him. The longevity of the springs was insufficient. On the Isuzu SAZ NP 37, the spring suspension is much softer. Measures have been taken to increase its durability. The reduction of vertical accelerations in the front suspension, as well as the effective damping of vibrations achieved by the installation of telescopic shock absorbers, significantly improved the working conditions of the driver. The mass of the spring suspension at the same time, of course, increased. The working conditions of the driver were also significantly improved due to the introduction of a steering mechanism equipped with a hydraulic booster, a mechanism for adjusting the driver's seat, an effective ventilation system for the cab, etc. Obviously, all these improvements contributed to the increase in the mass of the car.

The Isuzu SAZ NP 37 bus uses dependent suspension with leaf springs. The widespread use of such suspensions is due to the simplicity of their manufacture and maintenance, as well as the fact that they provide the car with steady movement.

In the suspension, where the semi-elliptic leaf spring acts as a guiding device, the correct choice of the design of mounting the springs to the car frame is of great importance. This is due to the fact that the leaf leaves of the springs are exposed to a complex of forces and moments that increase significantly during the operation of cars in difficult road conditions. If you underestimate the effect of these loads, the operational reliability of the suspension will drop dramatically. Therefore, when choosing the type of attachment of the springs to the frame, a number of the most common designs on automobiles were considered and analyzed, taking into account their reliability, convenience and ease of maintenance, as well as economic feasibility [1, 2].

The main types of fastening of the ends of the spring to the frame or car body are as follows:

- Fixing the end of the spring (i.e. the end of the spring, which receives all the forces acting on the suspension) with a twisted or detachable eye or on a rubber support;
- The free end of the spring (i.e. the end of the spring, perceives all forces, except the longitudinal ones that arise when the car is moving) on the earring, on a rubber or sliding support.
- The combination of fastenings of the ends of the spring can be very different.

In practice, fixing the fixed end of a spring with a twisted eye and the free end on an earring or sliding support is most often used.

Fastening the fixed end of the spring with a twisted eye is distinguished by its simplicity of construction, low cost and lower weight compared to other types of mounts. When operating in difficult road conditions, there are a number of difficulties associated with providing the necessary strength of the eye.

Research method

The most common and simple way to increase the strength of the eye - increasing the thickness of the root sheet - does not always give a positive result. If you increase the thickness of only one root leaf, leaving the thickness of the remaining sheets unchanged, this can lead to a significant decrease in the durability of the spring due to premature fatigue failure of the thickened root leaf. If the thickness of the root and other sheets is simultaneously increased, then to preserve the deflection and average design stress specified in the calculation, it will be necessary to lengthen the spring, which is not always possible for layout reasons, and, in addition, can lead to an irrational increase in the spring mass due to a decrease in the number of sheets.

Table 1 shows the average characteristics of the measurement cycle in a diesel bus.

Table 1: Average characteristics of the measurement cycle in a diesel bus

Parameters	The average	Maximum
Speed, m/s	17,9	25
Acceleration, m/s^2	0,21	1,42
Deceleration, m/s^2	-0,72	-1,88

The load depends on the reaction R_z on the wheel and the weight of the unsprung masses $G_{n.m}$.

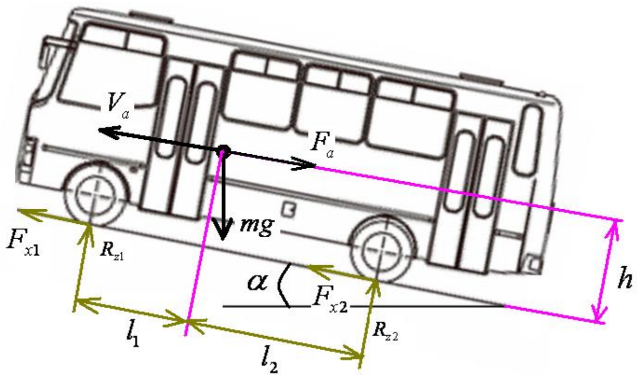

Fig 1: The forces acting on the car in the general case of movement

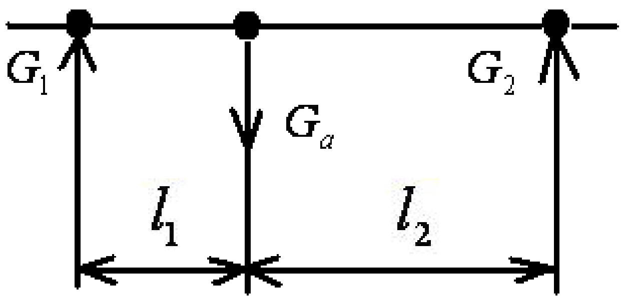

Fig 2: Scheme of the forces of formation of moments of forces

$$G_1 + G_2 = G_a$$
$$l_1 + l_2 = L$$
$$(l_1 + l_2) \cdot G_2 = G_a \cdot l_1$$
$$l_1 = \frac{G_2 \cdot L}{G_1 + G_2} = \frac{5000 \cdot 3.8}{3100 + 5000} = 2.346\, m$$
$$l_2 = L - l_1 = 3.800 - 2.346 = 1.454\, m$$
$$F_a = \frac{\rho_a}{2} \cdot c_x \cdot B \cdot H \cdot v^2 = \frac{1.225}{2} \cdot 0.65 \cdot 2.8 \cdot 1.665 \cdot 25^2 = 1160\, H$$
$$F_{x1} + F_{x2} - mg \sin\alpha - F_a = m\dot{v}$$
$$R_{z1} + R_{z2} - mg\cos\alpha = 0$$
$$R_{z1} \cdot 0 - R_{z2} \cdot (l_1 + l_2) + mg\cos\alpha \cdot l_1 + (F_a + m\dot{v} + mg\sin\alpha) \cdot h = 0$$
$$R_{z2} \cdot 0 - R_{z1} \cdot (l_1 + l_2) + mg\cos\alpha \cdot l_2 + (F_a + m\dot{v} + mg\sin\alpha) \cdot h = 0$$
$$R_{z1} = \frac{1}{L} \cdot (mg\cos\alpha \cdot l_2 - (F_a + mg\sin\alpha + m\dot{v}) \cdot h) \qquad (1)$$
$$R_{z2} = \frac{1}{L} \cdot (mg\cos\alpha \cdot l_1 + (F_a + mg\sin\alpha + m\dot{v}) \cdot h) \qquad (2)$$

Where G_1 and G_2 - load on the front and rear axle; R_{z1} - reaction force to the front wheels; R_{z2} - reaction force on the rear wheels; F_a - aerodynamic forces; ρ_a - air density; c_x - dimensionless coefficient of total aerodynamic force (buses: wagon layout - 0.6...0.75); B - track; H - overall height; m - is the loaded mass of the vehicle; g - acceleration during free fall.

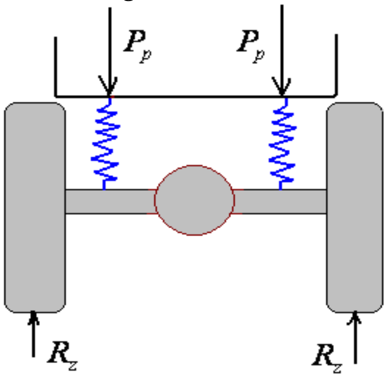

Fig 3: Design scheme for determining the loads on elastic elements of the suspension

The estimated spring load is determined as follows [2, 3]:

$$P_p = R_z - 0.5 \cdot G_{\text{Н.М.}} \quad (3)$$

Where $G_{\text{Н.М.}}$ – weight reactions of unsprung masses.

Table 2 shows the parameters of the acceleration of the car when driving on the rise.

Table 2: The parameters of the acceleration of the car when driving on the rise

α, ⁰	R_{z1}		R_{z2}		P_{p1}		P_{p2}	
	Average acceleration	**Max acceleration**	**Average acceleration**	**Max acceleration**	**Average acceleration**	**Max acceleration**	**Average acceleration**	**Max acceleration**
0	29620	27041	49760	52339	17120	14541	37260	39839
3	28486	25906	50786	53365	15986	13406	38286	40865
5	27684	25105	51394	53973	15184	12605	38894	41473

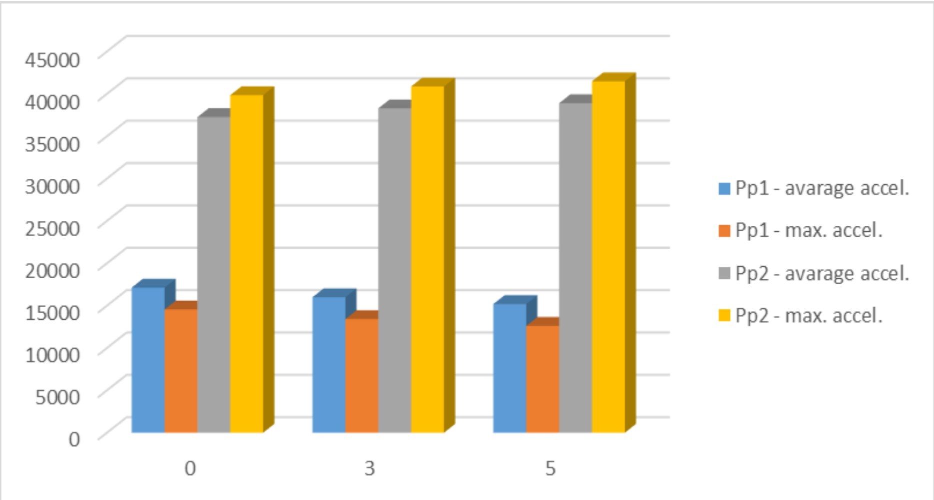

Fig 4: Distribution of loads on springs during movement on an inclined road

Static suspension design

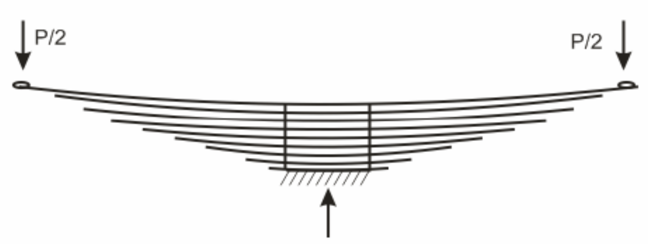

Fig 5: Design scheme of the suspension

The results of experimental studies and their discussion. The circuit suspension (Figures 3, 5) can be considered a Beam. It has a length of L = 1350 mm, a width of b = 70 mm and a wall thickness of 10 mm. At the centers, it is attached to rigid axes embedded in walls. Loading is performed by a transverse force P = 39839 N.

Returning to the design of the beam, we consider what happens to it under the influence of static load. From the geometry of the beam, one can expect a distribution of displacements and stresses similar to that realized in a classical beam with pinched ends loaded at the boundary. As a result of calculations using standard engineering formulas, we obtain that the calculated characteristics of the leaf spring are bending stress σ_u, deflection f_p, stiffness c_p.

For a symmetric semi-elliptic multi-leaf spring, we calculate the maximum deflection and stress by the formulas [2, 3].

$$\sigma_u = \frac{1.5 P_p L}{nbh^2} \qquad \sigma_{перед} = \frac{1.5 \cdot 14541 \cdot 1350}{7 \cdot 70 \cdot 10^2} = 601\ \frac{Н}{мм^2}$$

$$\sigma_{u.з.} = \frac{1.5 \cdot 39839 \cdot 1300}{10 \cdot 70 \cdot 10^2} = 1110\ \frac{Н}{мм^2}$$

$$f_p = \frac{\delta P_p L^3}{4Enbh^3} \qquad f_p = \frac{1.25 \cdot 14541 \cdot 1350^3}{4 \cdot 2.05 \cdot 10^6 \cdot 7 \cdot 70 \cdot 10^3} = 11.13\ мм$$

$$f_p = \frac{1.25 \cdot 39839 \cdot 1350^3}{4 \cdot 2.05 \cdot 10^6 \cdot 10 \cdot 70 \cdot 10^3} = 19.06\ мм$$

$$c_p = \frac{P_p}{f_p} = \frac{Enbh^3}{4\delta L^3} \qquad c_p = \frac{14541}{11.13} = 1306.5\ \frac{Н}{мм}$$

$$c_p = \frac{39839}{19.06} = 2090.1\ \frac{Н}{мм}$$

Where L - is the full length of the spring; n - is the number of sheets; E - Modulus of elasticity (2.05 MPa); b - sheet width; h - sheet thickness; δ is the deflection coefficient (1.25).
Estimation of maximum displacements and stresses: 19.06 mm, 1110 N/mm^2.

The finite element method (FEM) is today the universally recognized main method of structural analysis in a number of areas of science and technology. Therefore, in this study, we calculated the spring in the FEM using the Solid Edge ST program and ANSYS 12.0.
First, we measured the leaf springs and drew using Solid Edge ST (Figure 6).

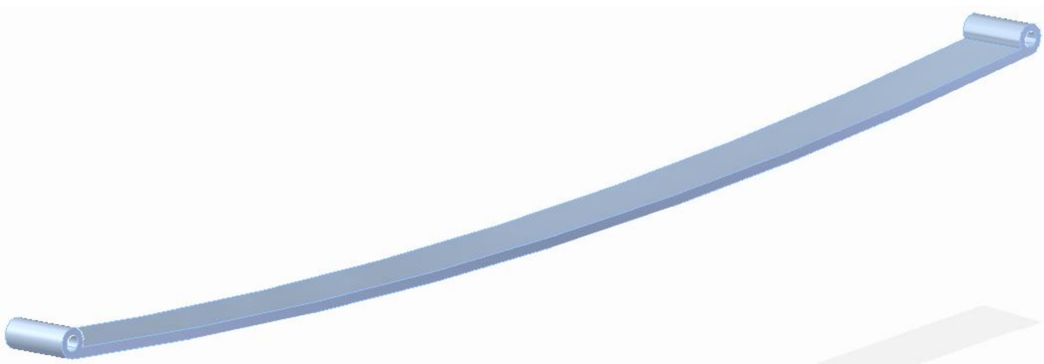

Fig 6: The root sheet

Then we drew the remaining leaf sheets in stages and then assembled the leaf sheets using Solid Edge ST (Figure 7).

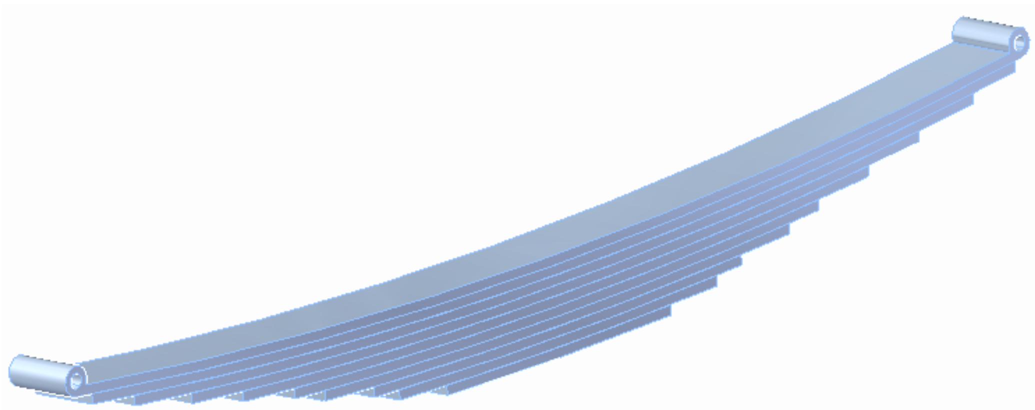

Fig 7: Assembled leaf spring.

Then we got a drawing (Figure 8)

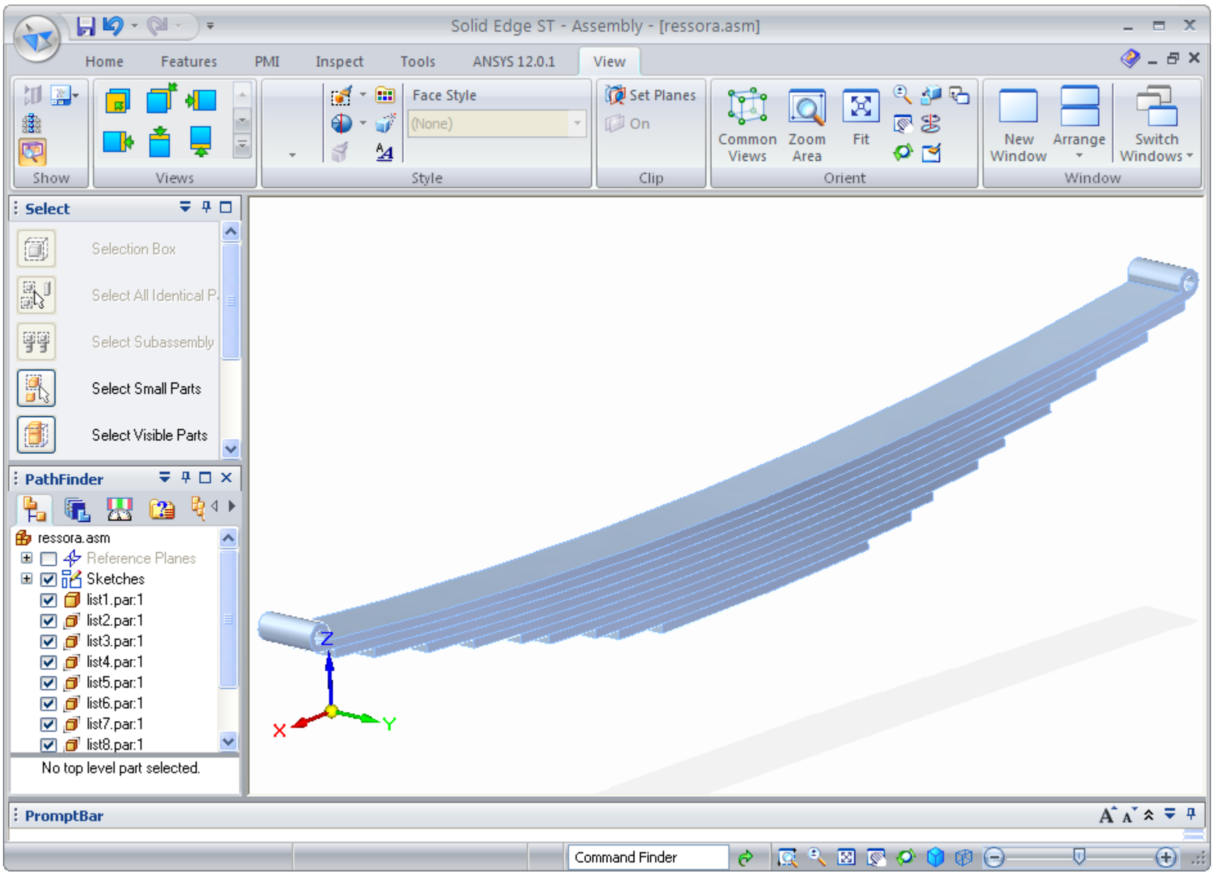

Fig 8: General drawing of the spring in the program Solid Edge ST

Then this model was exported to the ANSYS 12.0 program (Figure 9).

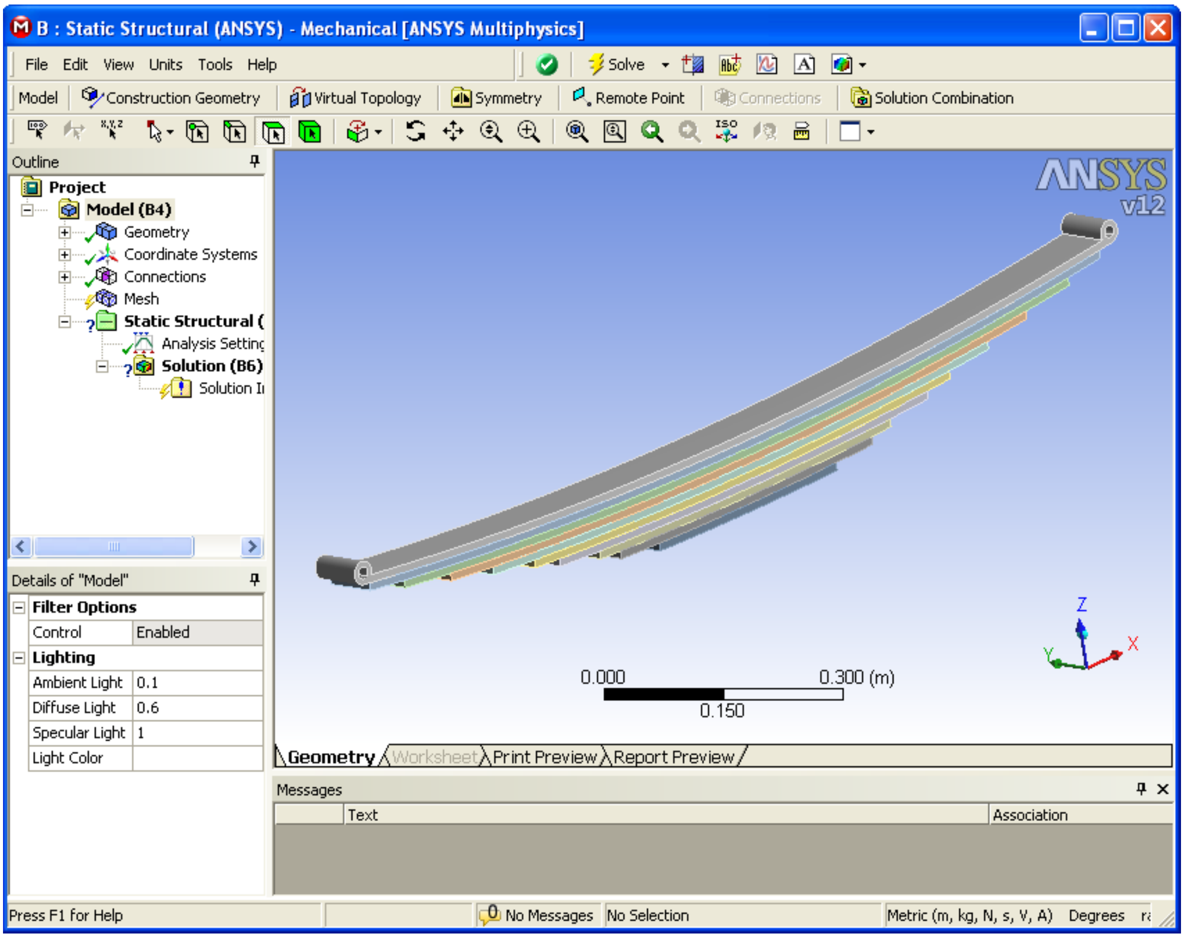

Fig 9: Exported spring view in ANSYS 12.0

After exporting the model to ANSYS 12.0. Click Supports and select Fixed Support functions (Figure 10). Using this function, we fix the lower part of the spring.

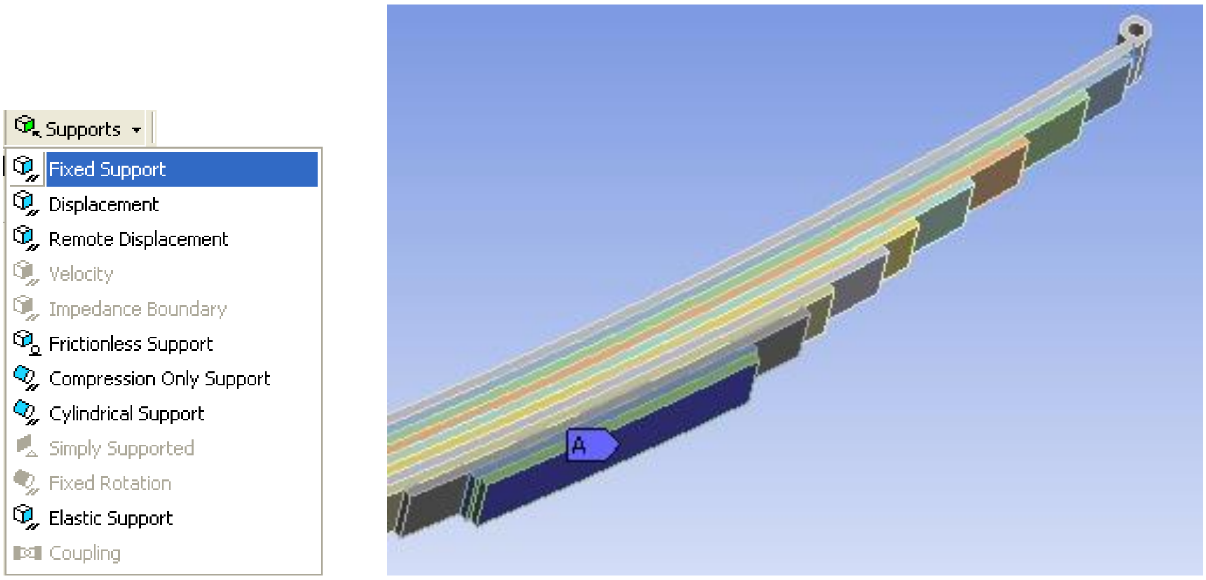

Fig 10: Fixed Support function and fixing the bottom of the spring.

After that, we establish the force factors and perform the simulation (Figure 11).

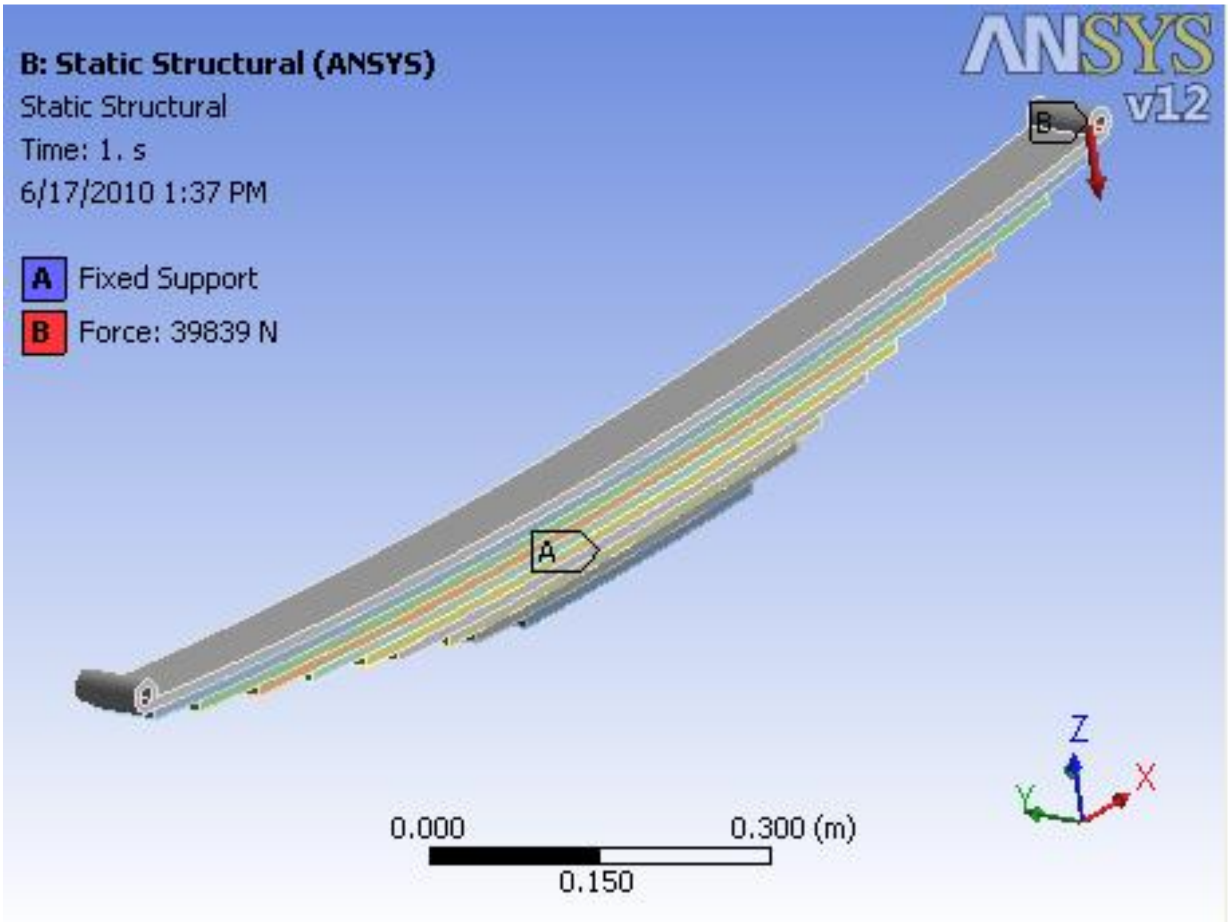

Fig 11: The delivered forces and the fixed type of springs

In the simulation of simulation (Simulation) springs using ANSYS 12.0. We get equivalent stress (Equivalent stress), total deformation (Total deformation) and margin of safety (Safety Factor). As a result of modeling, we determine the maximum loads that do not destroy the spring (Figure 12).

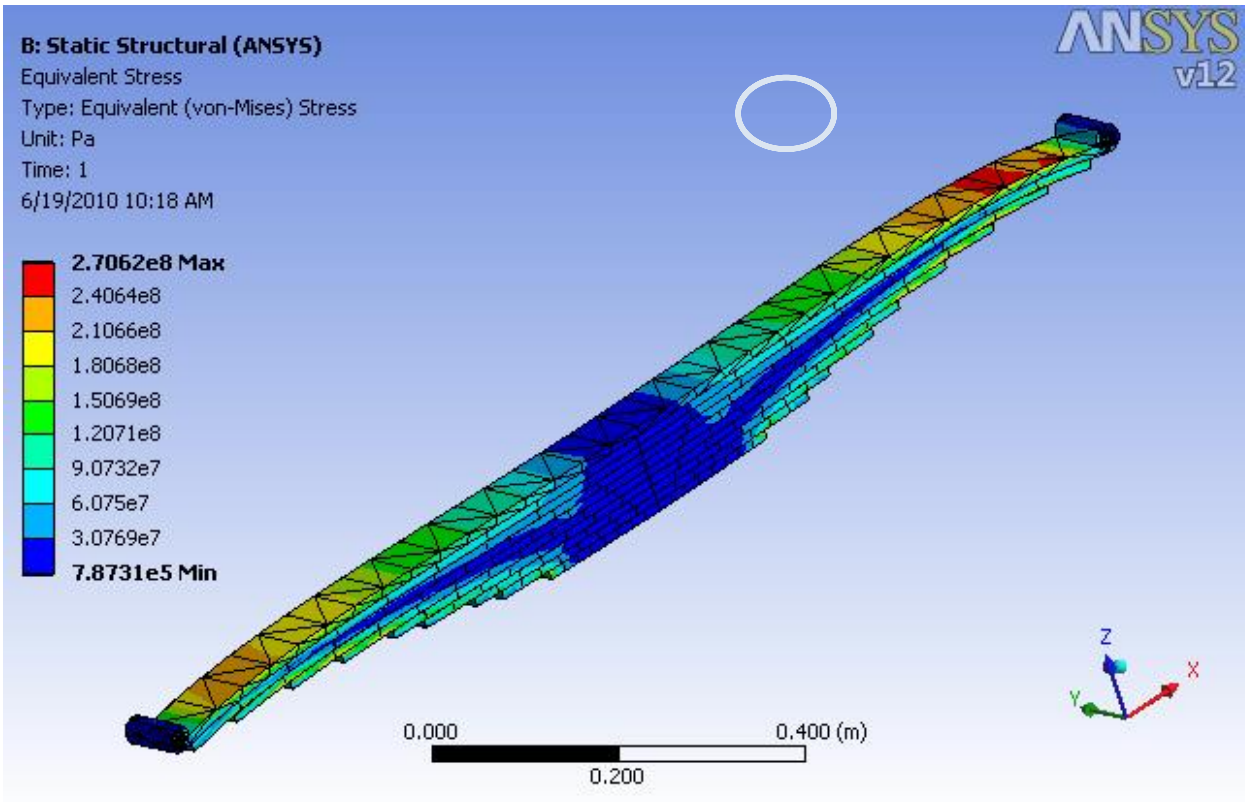

Fig 12: View after simulation of the spring

Conclusion: The calculations showed that a selection of optimization modeling methods is required when operating the Isuzu bus suspension mechanisms.

Fig 13: Damage to springs

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