A NONLINEAR SIMULATION OF CAR VIBRATION BY MSC/NASTRAN

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ABSTRACT

Setting the task of creating a modern competitive vehicle, the designers need the more exact estimation of the degree of their design optimum at the early stages of the work. According to this, besides the traditional methods of analysis (static analysis, normal modes analysis and frequency response analysis), the transient response analysis of the car real loading becomes more interesting. This type of analysis provides the data, which are the close analogue to the test results and allows, later on, to go over to the fatigue analysis.

In this article the example of the nonlinear simulation of the car vibration in time domain by MSC/NASTRAN is considered.

INTODUCTION

OJSC "GAZ" is one of the largest companies in Russia. Founded at 1930, at present time it produces a wide range of all types of automobiles: cars, trucks, buses and specials (fig. 1). Gorky Automobile Plant ("plant" is "zavod" in Russian) is one of the leaders in Russian automobile industry. It is also one of the leaders among other enterprises in the field of science and technology, none of which anew the model row of its production as frequently, as GAZ does. Every year OJSC "GAZ" launches a new model into production, which is a record for Russia at present time.

According to western standards OJSC "GAZ" test department is not developed enough. OJSC "GAZ" realizes that the modern test center is a necessary component of such company, as Gorky Automobile Plant, and it does it's best to develop it. Nevertheless the computer simulation aimed at decreasing the period of new car designing and developing is used by OJSC "GAZ" more and more nowadays.

At present time OJSC "GAZ" is solving the problem of possessing the methodologies of the simulation the car loading taking into consideration the greater number of features of this process, if possible. It tries to make the computer simulation results as close as possible to the results of similar tests. The final aim of these efforts is to possess the methodology of durability analysis of car parts and units.

It is well-known that the loading of the car during its running is the nonlinear process. The simulation of this process, in principle, is possible by such codes as MSC/DYTRAN and LS-DYNA3D. But these codes use the explicit method of integration. It leads to very small time step which results in *hours* or even *days* of CPU time for the simulation of car loading during about 0.2 second of real time, if the size of the finite-element model is about 100 thousand elements. At the same time, for the correct estimation of the random loading process it is necessary to analyse the stress/strain time realizations of the accessing process within the duration of a *few seconds* (or even *dozens seconds* better). The run time for this simulation in some cases may be not acceptable.

To solve the above mentioned problem the practical method of the simulation of the car vibration with nonlinear approach is proposed. It allows to get the stress and strain time realizations in the car body parts with the duration of a few seconds within dozens minutes or some hours (wall clock) of workstation computer using the modal method of solution.

DESCRIPTION OF PROBLEM

Generally the equation of the dynamical system vibration is stated as:

$$\|M\| \cdot \ddot{u}(t) + \|B[\dot{u}(t)]\| \cdot \dot{u}(t) + \|K[u(t)]\| \cdot u(t) = P(t), \quad (1)$$

with $\|M\|, \|B[\dot{u}(t)]\|, \|K[u(t)]\|$ - mass, damping and stiffness matrixes respectively;

$u(t)$ - grid points displacement vector;

$P(t)$ - the vector of the loads.

As it can be seen from the equation (1), the damping and stiffness matrixes depend on the grid points displacements. Taking into consideration, that the size of the vector $u(t)$ in real problems mounts to hundreds of thousand, it takes much CPU time for the *direct* solution of this equation.

It is known that the *modal* method provides the essential decrease of the CPU time. But it may be used for solving linear problem only.

METHOD OF SOLUTION

To avoid the problem of applicability of the modal method for solving the task under consideration, let's extract the linear part from the assessing dynamical system. In case of the application to the car, it may be done quite easy. For example, in the simulation of car vibration the nonlinear shock absorber rate is used. Let's present this rate as the sum of the linear and nonlinear fractions (fig. 2).

Presenting all internal non elastic forces as the sum of the linear and nonlinear fractions, the second addendum in the left part of the equation (1) will be stated as

$$\|B[\dot{u}(t)]\| \cdot \dot{u}(t) = \|B^*\| \cdot \dot{u}(t) + N_1[\dot{u}(t)],$$

with $\|B^*\|$ - the damping matrix of the linear part of the dynamical system;

$N_1[\dot{u}(t)]$ - nonlinear fraction of the internal non elastic forces of the dynamical system.

The same can be done with the internal elastic forces of the dynamical system. After that the equation (1) will be restated as:

$$\|M\| \cdot \ddot{u}(t) + \|B^*\| \cdot \dot{u}(t) + \|K^*\| \cdot u(t) = \\ = P(t) - N_1[\dot{u}(t)] - N_2[u(t)], \qquad (2)$$

with $\|K^*\|$ - the stiffness matrix of the linear part of the dynamical system;

$N_2[u(t)]$ - nonlinear fraction of the internal elastic forces of the dynamical system.

The left part of the equation (2) satisfies the condition of linearity, therefore this equation may be solved by modal method.

It is necessary to note, that the use of the modal method and the constant time step of integration is the reason which makes the estimation of the current size of the internal nonlinear forces N_1 and N_2 impossible. Therefore the calculation of these forces is performed based on the vectors $\dot{u}(t)$ and $u(t)$, which were obtained as the result of the previous time step of integration. Thus the final form of the dynamical system vibration equation will be stated as:

$$\|M\| \cdot \ddot{u}(t) + \|B^*\| \cdot \dot{u}(t) + \|K^*\| \cdot u(t) = \\ = P(t) - N_1[\dot{u}(t - \Delta t)] - N_2[u(t - \Delta t)],$$

with Δt - the time step of integration.

This feature of the method enforces to use a smaller time step in comparison with the case of completely linear problem to ensure the stability and acceptable accuracy of result.

The MSC/NASTRAN code gives the possibility to apply the nonlinear forces to the finite-element model. If the modal method (SOL 112) is used, it is achieved by the use of EPOINT, TF and NOLINi entries [Reference]. Thus, the above mentioned method is no more than the "widened" application of already existing MSC/NASTRAN possibilities.

STRUCTURAL MODEL

In development and tests of the proposed method the finite-element model of car, showed in fig. 3, was used.

The loading of the model is performed by moving the nodes, which correspond to the points of contact between tires and road surface, in vertical direction. A special code is used for input data deck building which allows to define the required motion of the aforesaid nodes in MSC/NASTRAN run. This code includes the random digital generators and digital filters, which provide producing the digital sequences with the required power spectral density.

Grid points location in the finite-element model corresponds to the car loaded by the gravity. If special measures are not be taken, the applying of the gravity will cause additional deformation of the structure, first of all the suspensions of the wheels. To avoid this the forces, equal to the constant loads, arising because of the gravity, are applied to the corresponding nodes of the finite-element model (fig. 4).

According to the analysis, the deformation of the finite-element model under the gravity and the forces showed in fig.4, is simulated not enough correctly if only the eigenvectors extracted by the "usual" normal modes analysis are used. As a result the stress in the finite-element model parts is not correct either. MSC/NASTRAN code allows to solve this problem. By the use of the alternative solution (versions 68 and 69) or by defining the special solution parameters (version 70 and the following: PARAM,RESVEC,YES [Reference]) the calculating of the residual vectors is initialize. These vectors are added to the already calculated eigenvectors and provide correct simulation of the car deformation under the constant loads.

RESULTS

The tests of the proposed method were conducted under the following conditions.

The finite-element model consisted of about 10 thousand elements and most of them were shell elements. The range of the frequencies, where the eigenvectors for modal analysis were found, was from 0 Hz to 50 Hz. There were three nonlinear effects simulated by the tested model: the devices which constrained the relative displacement of the wheels and the body (i.e. jounce bumpers), the features of the shock absorbers rates and the possibility of the jumping the wheels over the road surface. The power spectral density of the road profile in the model corresponded to the profile of the non-smooth cobblestone track of the Gorky automobile plant proving ground. The velocity of the car motion was 10 m/s (36 km/hour). The duration of the car loading process was 10 seconds. The time step of integration was 0.001 second.

For the tests the computer IBM RISC/6000 mod. 591 was used and the working storage space was 150 megabytes.

The run time (wall clock) for the considered model was about 1 thousand seconds. It is necessary to note, that for correct simulation of the car static deformation under the constant loads MSC/NASTRAN added two residual vectors to the primary eigenvectors.

In fig. 5 the vertical displacements of the road surface under the left front wheel, the left front wheel itself and the left body rocker (zone near the B-pillar) are depicted. The vertical displacements of the bottom and top parts of the spring strut of the left front wheel and the force, acting on the left front jounce bumper, are shown in fig. 6 and fig. 7 respectively. From the last two figures it can be seen, that if the deformation of the suspension exceeds the certain amount, the jounce bumper acts and a force arises.

At present time the proposed method is being tested with utilization of the already existing calculation results and experimental data, obtained during the tests of a new car GAZ-3111 "Volga" (fig. 8), which is planning to be produced at the end of 1999.

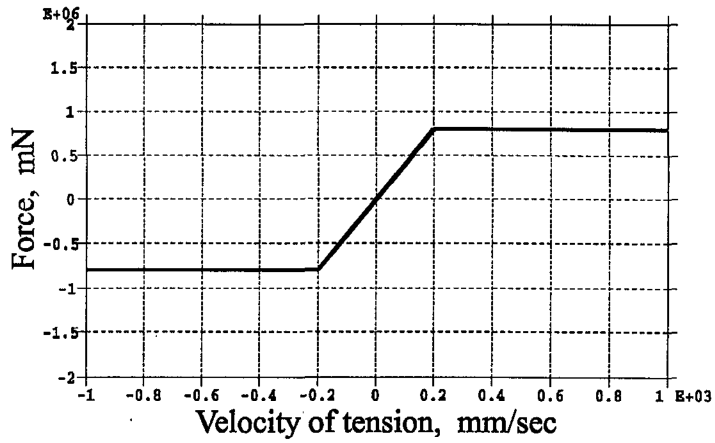
CONCLUSION

The conducted studies with the use of considered finite-element model have shown the high efficiency of the proposed method of a nonlinear simulation of car vibration. The possibility to get the durable realization of results, in our opinion, gives new ways to the almost real computer simulation of the vibration and stress/strain analysis of the car parts and units and, in the future, to the estimation of car durability based on pseudo random loading in time domain with nonlinear effects.

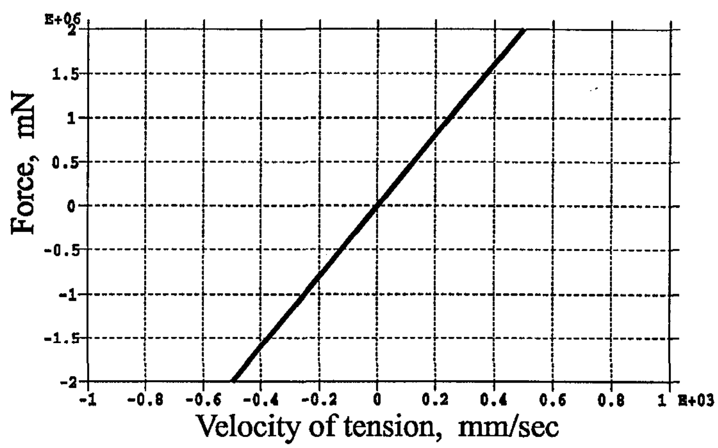
REFERENCE

MSC/NASTRAN Quick Reference Guide. Version 70. The MacNeal-Schwendler Corporation, Los Angeles, CA.

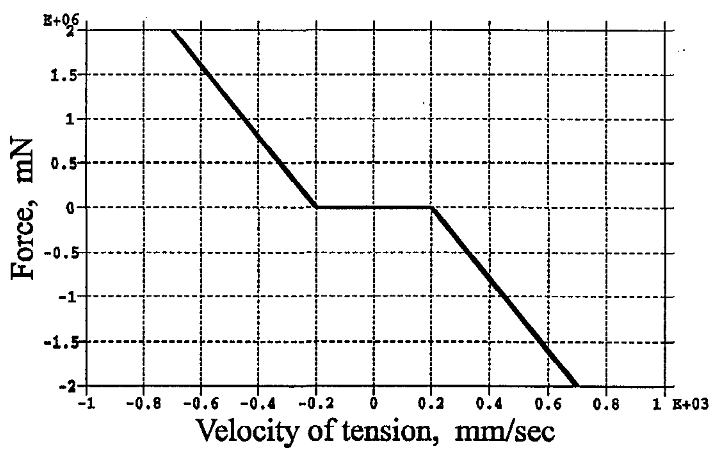
Figure 1. Automobiles of OJSC “GAZ”



a) The complete rate



b) The linear fraction



c) The nonlinear fraction

Figure 2. Decomposition of the shock absorber rate

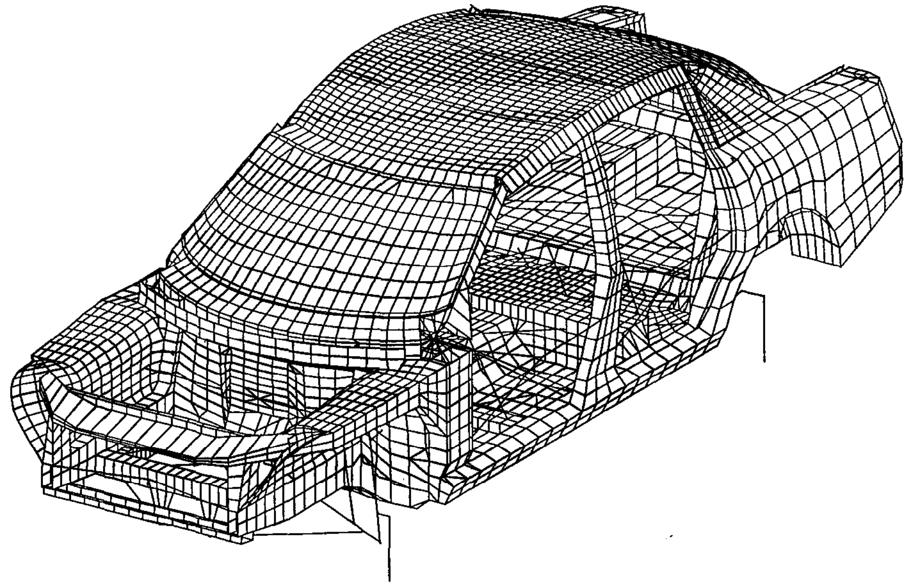


Figure 3. Finite-element model

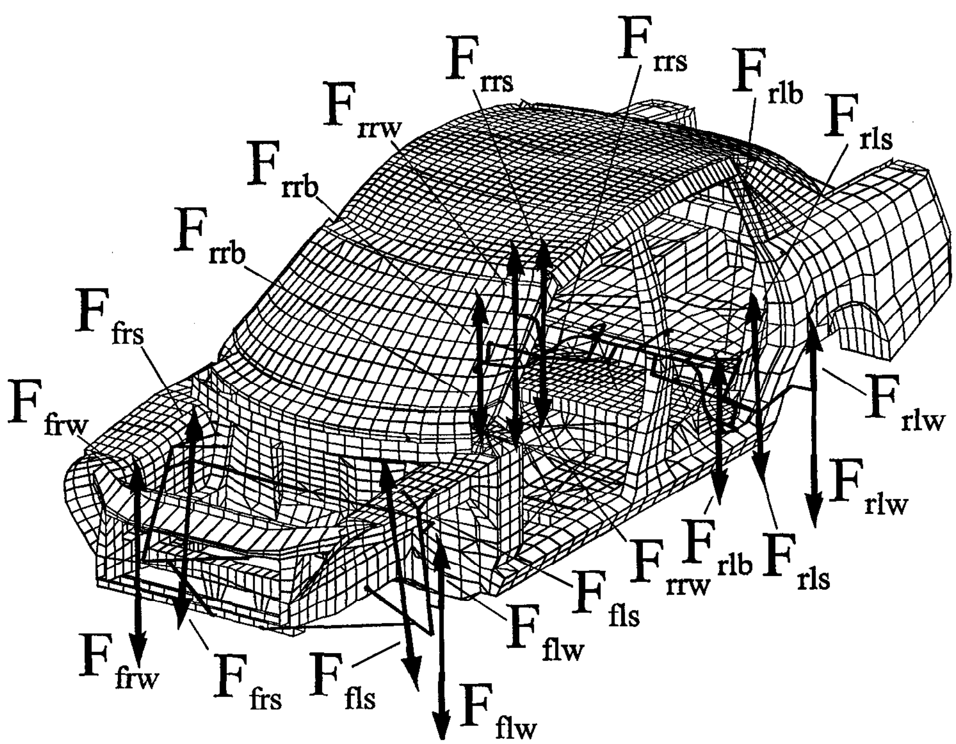
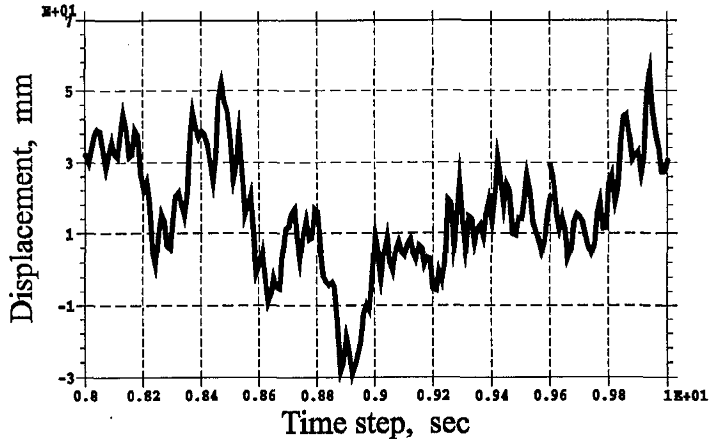
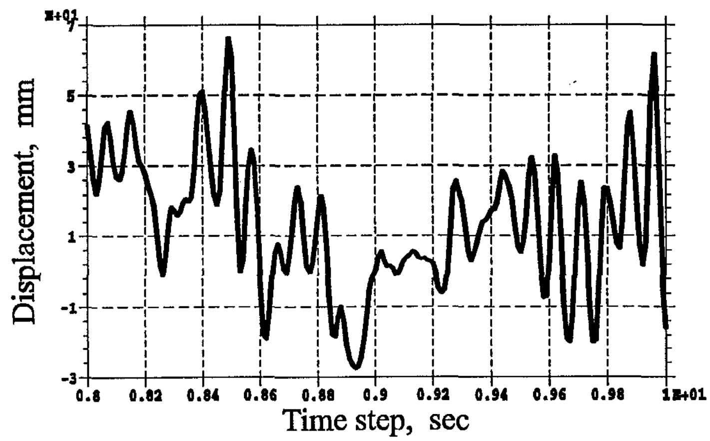


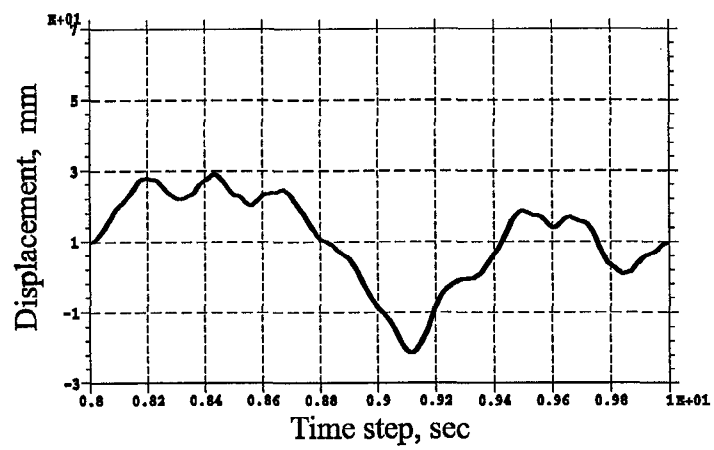
Figure 4. The static forces in the elastic parts of the suspensions



a) The “displacement” of the road surface



b) The displacement (T3) of the wheel



c) The displacement (T3) of the rocker

Figure 5. The displacement of the model parts

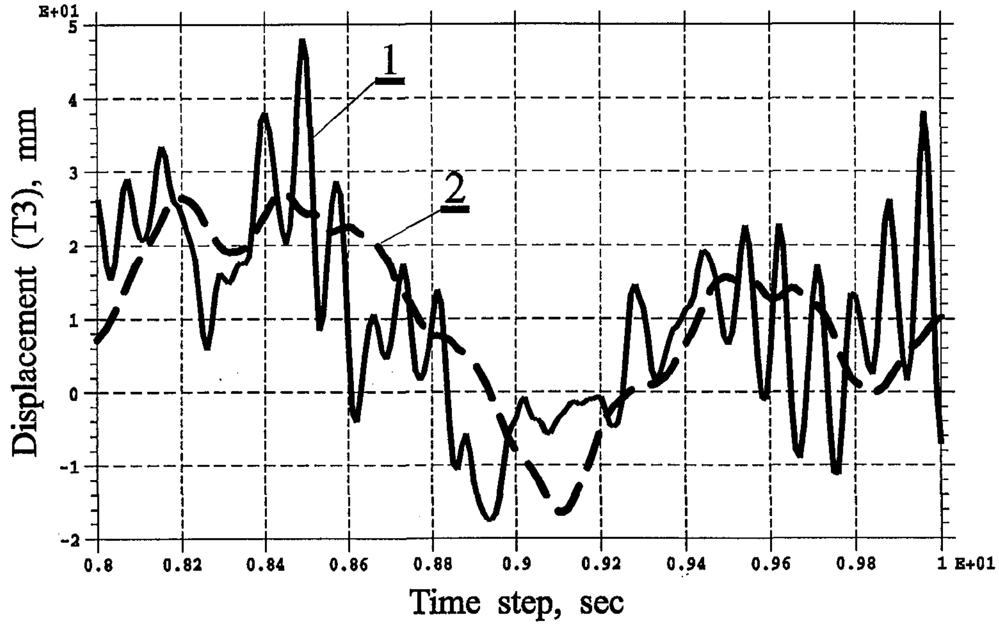


Figure 6. The displacement (T3) of the lower (1) and upper (2) parts of the spring strut of the left front wheel

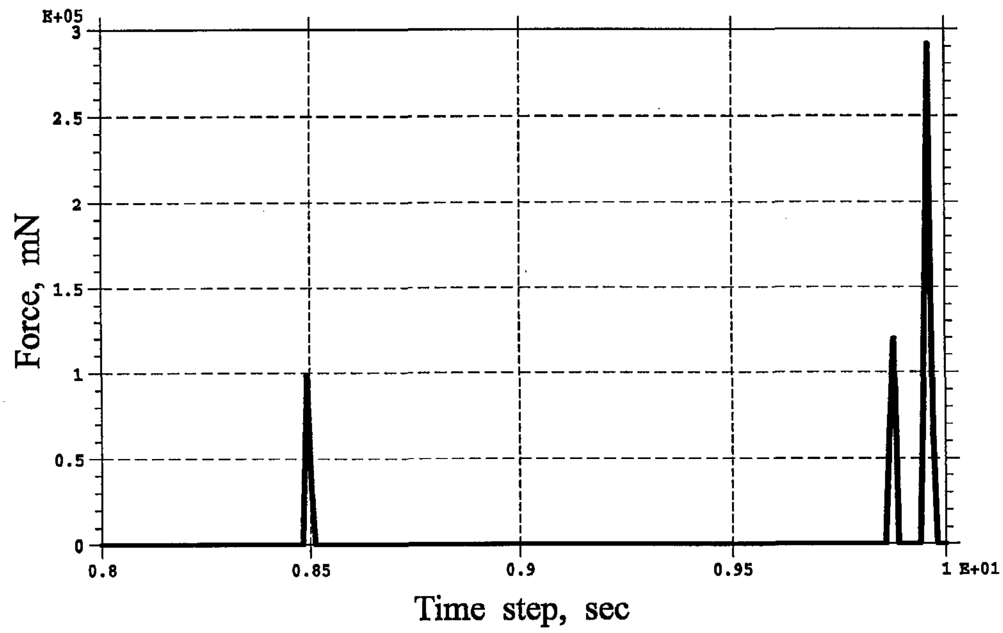


Figure 7. The force, acting on the left front jounce bumper

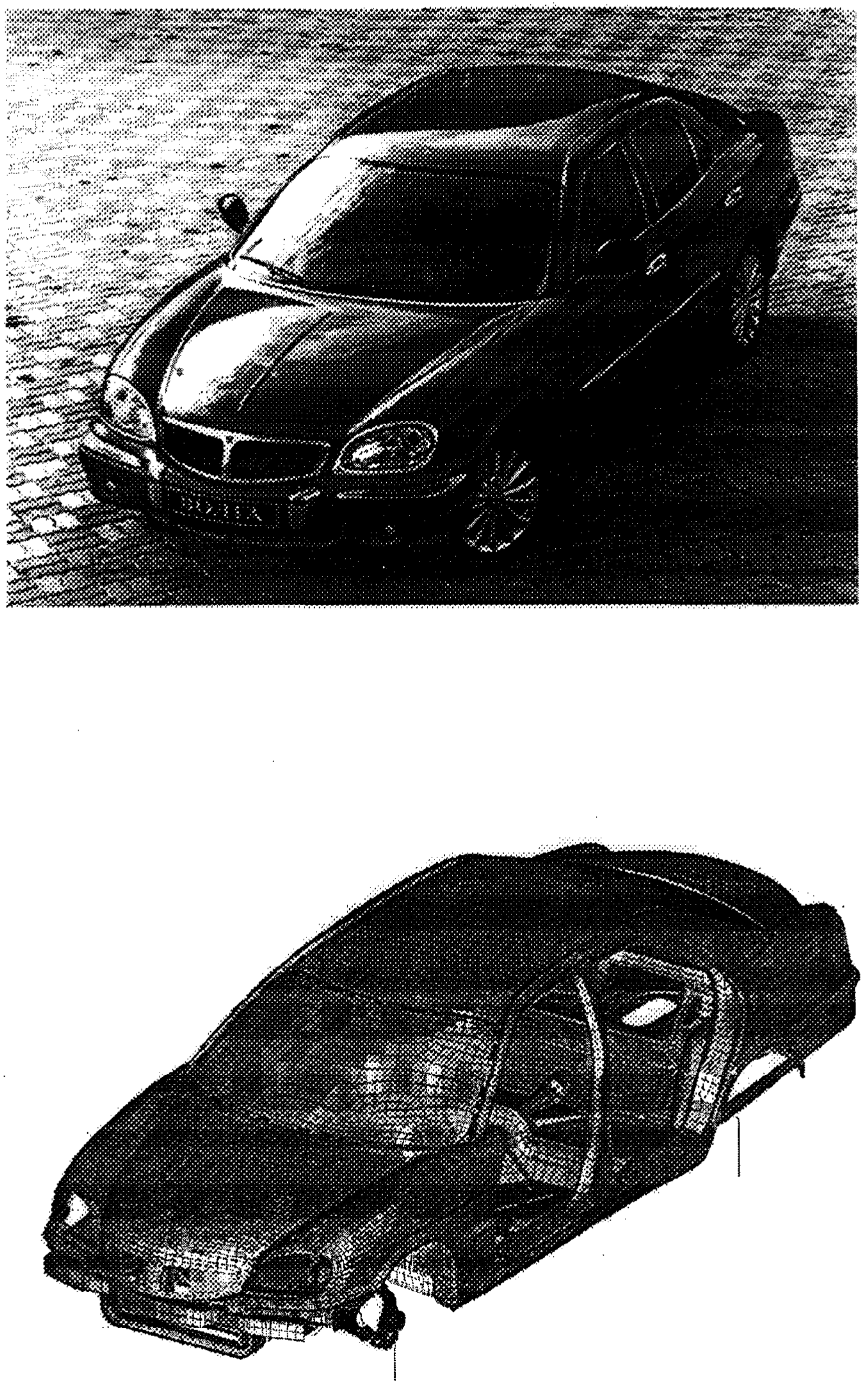


Figure 8. New GAZ-3111 "Volga"