Formation of performance characteristics of a Formula SAE sports vehicle based on calculations and simulations of systemic elements

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Abstract. The article presents a description of the research and development carried out by NNSTU employees as part of their participation in the international project Formula Student. The paper includes the calculated assessment of the operational properties of the vehicle being designed: the calculation of the center of mass coordinates, with the presentation of data based on the configuration features of the vehicle, the calculations of stability, the efficiency of the brake system, the input parameters of the steering simulation are given, on the basis of which the graphical dependences of the angles were obtained steering wheel rotation, as well as data on prototyping mechatronic transmission systems. The data obtained is compared with the technical regulations requirements of the international student competitions «Formula SAE».

1. Introduction
During developing a prototype of a vehicle, the performance characteristics of the vehicle are relevant indicators. These characteristics are evaluated on the basis of calculated dependencies, simulation processes, and a basis of experiments in the case of automation of existing systems. At the same time, it is necessary to take into account the features of the configuration and operational restrictions within which the system is being developed [1]. Researchers from Nizhny Novgorod State Technical University have many years of experience in the design of wheeled vehicles. Recently, there has been active research into the development of light commercial vehicles [2, 3] and off-road vehicles on ultralow pressure tires [4-6]. These vehicles are produced at industrial enterprises in Nizhny Novgorod. Also at the university, students have developed different vehicles. This is a mini all-terrain vehicle "Korsak" [7-10] and a car developed within the framework of the Formula Student project [11, 13-15]. We will consider the modeling and design process in the article using a racing car as an example. Some of the calculations are fundamental and are necessary for the subsequent formation of vehicle systems calculations: safety cage, energy absorbing device, suspension, braking system, steering.

2. Calculation of the center of mass coordinates and sports car stability
The calculation of the center of mass coordinates is based on a detailed study of the vehicle configuration in the form of detailed 3D models of parts and assemblies. A number of 3D models are based on 3D scan data. In order to obtain data on the vehicle systems' center of gravity coordinates, the resulting virtual three-dimensional model is located at the origin of coordinates [11]. An
additional parameter which involved in the calculations is the mass of elements. A systematized set of parameters for the mass and coordinates of the center of gravity of the sports car systems is presented in Table 1.

The value of the coordinate of the center of mass of the sports car is calculated by the ratios:

\[ x_c = \frac{\sum m_i x_i}{\sum m_i}, \quad y_c = \frac{\sum m_i y_i}{\sum m_i} \]  \hspace{1cm} (1)

where \( x_c \) is the coordinate of the center of mass, plotted along the abscissa, \( m \); \( y_c \) is the coordinate of the center of mass, plotted along the ordinate, \( m \); \( m \) is the mass of the element, kg; \( i \) is the ordinal number of the element.

Table 1. Systematized set of parameters

| System name                  | Weight \( m_i \) (kg) | Coordinates   |
|------------------------------|-----------------------|---------------|
|                              |                       | \( X_i \) (mm) | \( Y_i \) (mm) |
| Frame                        | 39.2                  | 1140,505      | 361,136       |
| Damping device               | 0.725                 | -99.1572      | 362.5         |
| Partition wall               | 2.449                 | -1           | 362.5         |
| Refractory walls             | 9.1                   | 1495.3795     | 368.9754      |
| Cockpit floor and front floor| 2.269                 | 852.146       | 161.671       |
| Pilot                        | 70                    | 1041,108      | 450.8812      |
| Armchair                     | 2.2                   | 1199.05       | 326.35        |
| Engine                       | 62.538                | 1780.05       | 417.2767      |
| Fuel system                  | 5.748                 | 1664.793      | 531.5054      |
| Exhaust system               | five                  | 1766,439      | 226.0459      |
| Steering                     | 5.807                 | 664.5163      | 293.2873      |
| Transmission                 | 12.136                | 2150.09       | 282.5409      |
| Pedal assembly               | 4.807                 | 108.0867      | 238.9938      |
| Front axle suspension        | 11.223                | 663,1569      | 272,1079      |
| Rear axle suspension         | 10.705                | 2206,987      | 295,4099      |
| Brake system. front axle     | 2.703                 | 733,7443      | 263.9293      |
| Brake system. rear axle      | 2.703                 | 2283.83       | 270,1543      |
| Wheels                       | 52.56                 | 1437,701      | 270,215       |

Calculation of the necessary products of the mass of an element by the corresponding coordinate along the abscissa axis \( m_i x_i \) (kg ⋅ m) and ordinates \( m_i y_i \) (kg ⋅ m), and determining the total values of the mass \( m = \sum m_i \) (the sum of the masses of the vehicle elements), on the basis of data table 1, produced by Excel.

Calculation result: The sum of the products over \( X = 412046.2 \) mm; The sum of products by \( Y = 89659.05 \) mm; Mass \( m = 301.8744 \) kg; wheelbase \( L = 1550 \) mm.

Determine the value of the coordinates of the center of mass of the car using the formula (1) we obtain: \( x_c = 1.364 \) m, \( y_c = 0.297 \) m

The determination of static loads on the axes of the chassis is based on the results of calculating the mass and calculating the coordinates of the center of gravity of a sports car. The parameter of the static load distributed along the vehicle axes used as input data for calculating the braking system, suspension elements, and the stability of a sports car as well.

Calculation layout (figure 1) of determine the static load is realized on the basis of the following algorithm:

1. Get a side image of the vehicle.
2. Enter the Cartesian coordinate system and the origin point into the resulting image. In this case, it is necessary to pre-select the direction of the coordinate system axes. The origin point must coincide with the origin entered during the finding the sports car center of gravity.

3. Apply the circuit location parameters of the center of mass coordinates, the distance from the coordinates of the center of mass to the front axis of the vehicle, the wheelbase and the force of gravity, which replaced by reactions based on the liberation axiom. It is necessary to control that the resulting system is statically determinate.

![Figure 1](image1.png)  
**Figure 1.** Calculation scheme for determining the normal load \( R_{k_2} \) and \( R_{k_4} \) of a stationary vehicle, where \( G \) – cars weight, \( H \); \( x_{c_1} \) - distance from the front axle of the car to the coordinates of the center of mass, \( m \); \( L \) is the cars wheelbase, \( m \).

After building calculation layout, it is necessary to consider the equilibrium condition of the car, thereby to write the equations of statics, which are equivalent to zero the amount of projection on the axis of the Cartesian coordinates of all the forces acting on the body and the sum of the moments of these forces about the arbitrarily selected points or axes.

From the equation of the moments of forces acting relative to the axis of the front wheels, we determine the normal load acting on the rear axle of the vehicle:

\[
R_{k_2} = \frac{G x_c}{L} = \frac{m g x_{c_1}}{L},
\]  
(2)

Based on formula (2), we obtain the value of the load acting on the rear axle of a sports car \( R_{k_2} = 1293,4578 \) Н.

We determine the normal load acting on the front axle of the car from the equation of the moments of forces acting relative to the axis of the rear wheels.

\[
R_{k_4} = \frac{G (L-x_{c_1})}{L} = \frac{m g (L-x_{c_1})}{L},
\]  
(3)

Based on the formula (3), we get the value of the load acting on the front axle of a sports car: \( R_{k_4} = 1667,9300 \) H.

Based on the results of calculations of the coordinates of the center of mass and static loads, the operational stability indicator of the car is formed. This indicator is characterized by a critical angle of deviation in the transverse direction. In calculating the critical angle of inclination, the calculation scheme is used (Figure 2). this scheme determines the normal reactions on the wheels of the car based on the loading condition of the car. In a static position, the car is affected by gravity.

Each of the corresponding forces from the rolling surface will cause normal reactions on the front and rear wheels \( R_{A,B} = R_{K_2} \) and \( R_{C,D} = R_{K_4} \), side on the left and right wheels \( S_{A,B} \) and \( S_{C,D} \). At the static position of the car, the bond reactions are equal in value and opposite to the corresponding active forces.

Normal reactions of rolling surface on the wheels of the front axle \( R_A \) and \( R_B \) and the rear axle \( R_C \) and \( R_D \):

\[
R_A = \frac{1}{2} \cdot G \cdot \zeta_A; \quad R_B = \frac{1}{2} \cdot G \cdot \zeta_B; \quad R_C = \frac{1}{2} \cdot G \cdot \zeta_C; \quad R_D = \frac{1}{2} \cdot G \cdot \zeta_D,
\]  
(4)
The coefficient of redistribution of normal reactions on the wheels of a car is determined by the expressions:

\[
\zeta_{AB} = \frac{t_2}{L} \cdot \cos \alpha_0 \cdot \cos \alpha_S \pm \frac{h_c}{b} \cdot \sin \alpha_0 + \frac{t_2}{L} \cdot \frac{h_c}{b} \cdot \cos \alpha_0 \cdot \sin \alpha_S, \\
\zeta_{CD} = \frac{t_2}{L} \cdot \cos \alpha_0 \cdot \cos \alpha_S \mp \frac{h_c}{b} \cdot \sin \alpha_0 \mp \frac{t_2}{L} \cdot \frac{h_c}{b} \cdot \cos \alpha_0 \cdot \sin \alpha_S, 
\]

where \( \zeta \) is the coefficient of redistribution of normal reactions.

The coefficient of redistribution of normal reactions on the wheels of a car is determined by the expressions:

\[
R_{K_1} = R_A + R_B = \frac{1}{2} \cdot G \cdot (\zeta_A + \zeta_B) = \frac{1}{2} \cdot G \cdot \zeta_1, \\
R_{K_2} = R_C + R_D = \frac{1}{2} \cdot G \cdot (\zeta_C + \zeta_D) = \frac{1}{2} \cdot G \cdot \zeta_2.
\]

The total value of normal reactions on the left and right wheels of the car \( R_{LW} \) and \( R_{RW} \) are equal to:

\[
R_{LW} = R_A + R_C = \frac{1}{2} \cdot G \cdot (\zeta_A + \zeta_C) = \frac{1}{2} \cdot G \cdot \zeta_L, \\
R_{RW} = R_B + R_D = \frac{1}{2} \cdot G \cdot (\zeta_B + \zeta_D) = \frac{1}{2} \cdot G \cdot \zeta_R.
\]

Equality to zero of the normal reaction on the front or rear wheels means their complete unloading, which is equivalent to overturning the car, then the condition for overturning the car is \( R_{K_1} = 0 \) or \( \zeta_1 = 0 \), then

\[
tg \alpha_0 = tg \alpha_{0_{\text{max}}} = \frac{t_2}{h_c} \cdot \cos \alpha_S. 
\]

Losing of lateral stability will occur when the right or left wheels located on the upper part of the slope are completely unloaded, in case of \( R_{R_W} = 0 \) or \( \zeta_R = 0 \), then

\[
tg \alpha_{\text{max}} = \frac{h_c}{h_i}, \\
\alpha_{\text{max}} = \arctg(tg \alpha_{\text{max}}).
\]

Angles of stability factor are the difference between the stability angles and the actual tilt angles of the car.

Based on the design layouts and dependencies, an indicator of vehicle stability was obtained, \( \alpha_{\text{max}} = 63°39'44'' \), while this indicator exceeds the established value of 60 degrees and makes it possible to pass competitions stability tests.

2. Calculation of indicators of the brake system

The braking performance parameter is determined by the design features of the braking system components. These features, specified by the requirements of the Formula SAE technical regulation [1], mean the presence of two independent brake circuits of the front and rear axle of the vehicle. The brake circuits are interconnected with the balance bar. The element that distributes the load on the rods of the brake master cylinders, set by the pilot during interacting with the brake pedal. Preparation of optimal braking performance index calculation is implemented by the brake actuator (figure 3).

The main parameters, determining the index of braking performance: the estimated brake fluid pressure in brake circuit \( P_{\text{hyd}} \) (10) and the magnitude of the brake fluid pressure, required to lock the front wheels of the car \( P_{\text{th}} \) (11);

\[
P_{\text{hyd}} = \frac{F_{pp}}{A_{plb}}, 
\]

where \( F_{pp} \) is the value of the force acting on the rods of the brake master cylinders, taking into account the balance bar, N; \( A_{plb} \) is the area of the one brake master cylinder piston, mm\(^2\).
where $R_{z1}$ is the load on the front axle of the vehicle, N; $A_1$ is the ratio of the shoulder $r_1$ to the shoulder $r_2$, m; $r_d$ is the dynamic radius of the wheel, m;

This type of vehicle implies movement on asphalt or cement surface, the dynamic radius $r_d$ of the wheel is taken approximately equal to the static radius of the wheel $r_{st}$ and is calculated using the following formula:

$$r_{st} = 0.5 \cdot d + \lambda_z \cdot \Delta_{tr} \cdot B_{tr},$$  

where $\lambda_z$ is the coefficient of normal tire deformation ($\lambda_z = 0.8... 0.85$); $B_{tr}$ is the tire profile width, m; $H_{tr}$ is the tire profile height, m; $\Delta_{tr} = \frac{H_{tr}}{B_{tr}}$ is the coefficient of the shape of the tire profile; $d$ is the rim landing diameter, m; [12].

An important stage in the calculation is the selection of the optimal geometric parameters of the brake pedal. The shoulders, formed depending on the attachment points of the brake pedal axis and the balance bar axis, where: $r_1$ is the distance from the attachment point of the balance bar to the point of interaction between the pilot and the brake pedal, m; $r_2$ is the distance from the attachment point of the brake pedal axis to the attachment point of the balance bar axis, m;

![Figure 3. Brake drive circuit](image)

Geometrical parameters $(r_1, r_2)$ directly influence on the brake pedal $S_p$ (13):

$$S_p = S_{p, h} \cdot \frac{r_1}{r_2},$$  

where $S_{p, h}$ is the piston movement in the brake master cylinder, mm;

$$S_{p, h} = j_b \cdot j_{p, w} \cdot \frac{A_{p, w}}{A_{p, h}} \cdot S_{p, w},$$  

where $j_b$ is the number of supports connected to the brake master cylinder; $j_{p, w}$ is the number of pistons in the brake calipers; $A_{p, w}$ is the caliper piston area, mm$^2$; $S_{p, w}$ is the movement of the piston in the support, mm;

Since the design of the brake drive implies the presence of a balance bar, the value of the force acting on the brake master cylinders’ rods $F_{pp}$ (15) is determined as follows:

$$F_{pp} = F_{pr} \frac{b_2}{b_2 + b_1},$$  

where $F_{pp}$ is the value of the total force acting on the rods of the brake master cylinders;
The distance from the plane of symmetry of the balance bar to the longitudinal axis of the brake master cylinders’ rod of the rear axle, m; $b_{fr}$ is the distance from the plane of symmetry of the balance bar to the longitudinal axis of the brake master cylinders’ rod of the front axle, m;

The variables included in the design formulas geometrically depend on the component base of the designed system.

$$F_{pr} = \frac{F_p r_1}{r_2}$$

where $F_p$ is the force acting on the brake pedal, N [5];

The parameters required for calculating the brake drive are presented in Table 2.

The results of the dependencies (10-16) of the performance indicators of the braking system of a sports car are shown in Table 3.

The calculated values given should correlate with the requirements of the Formula SAE technical regulation for car wheel locking. The value of the pressure reached in the brake circuit $P_{hyd}$ should exceed the value of the pressure required to bring the front brakes to the full blocking $P_1$, as a result of which the performance of the vehicle's braking system will be ensured and the vehicle's braking efficiency is optimal.

Table 2. Parameters required for calculating the brake drive.

| Parameters name | Parameters value |
|-----------------|------------------|
| 1 $F_p$ is the force acting on the brake pedal, N | 490 |
| 2 $b_{fr}$ is the distance from the plane of symmetry of the balance bar to the longitudinal axis of the brake master cylinder rod of the front axle, m | 0.024 |
| 3 $b_f$ is the distance from the plane of symmetry of the balance bar to the longitudinal axis of the brake master cylinder rod of the rear axle, m | 0.015 |
| 4 $a_{ph}$ is the brake master cylinder piston area, mm$^2$ | 283.4 |
| 5 $a_{ph,c}$ is the caliper piston area, mm$^2$ | 506.45 |
| 6 $m_a$ is the total mass of the vehicle, kg | 301.87 |
| 7 $r_{fr}$ is the load on the front axle of the vehicle, N | 1667 |
| 8 $S_{pl,b}$ is the piston movement in the support, mm | 1.4 |

Table 3. Brake system exploitation parameters results

| Vehicle calculation parameters | Estimated values of vehicle performance |
|-------------------------------|----------------------------------------|
| 1 $r_1$ is the distance from the attachment point of the balance bar to the point of interaction of the pilot with the brake pedal, m | 0.25 |
| 2 $r_2$ is the distance from the attachment point of the brake pedal axis to the attachment point of the axis of the balance bar, m | 0.03 |
| 3 $F_{pm}$ is the value of the total force acting on the brake master cylinder rods, N | 4083.3 |
| 4 $F_{pm,c}$ is the value of the force acting on the brake master cylinder rod of the front axle, N | 2512.8 |
| 5 $P_{hald}$ is the calculated value of the brake fluid pressure in the brake circuit, bar | 88.6 |
| 6 $r_d$ is the dynamic radius of the wheel, m | 0.253 |
| 7 $A_1$ is the ratio of the arm $r_1$ to the arm $r_2$, m | 8.3 |
| 8 $P_1$ is the amount of pressure of the brake fluid in the brake circuit required to block the front wheels of the car, bar | 50.08 |
| 9 $Sp$ is the brake pedal travel, mm | 83.4 |
| 10 $S_{ph,c}$ is the piston movement in the brake master cylinder, mm | ten |
3. Calculation of steering performance

The key parameters for calculating steering are the wheelbase \( (L = 1550 \text{ mm}) \), the front axle track width \( (B = 1200 \text{ mm}) \), the scrub radius \( (r_f = 0 \text{ mm}) \) and the distance between the points of intersection of the pivot axles of wheels with the supporting surface \( (l_0 = 1200 \text{ mm}) \), they were obtained during the development of the suspension. The value of the minimum radius of a vehicle by rotation of the outer wheel should not exceed 4,500 mm (requirement FSAE Rules [1]).

![Figure 4. Vehicle turning layout.](image)

To exclude the contact of the inner surface of the wheel disk and the suspension arms, as well as the vehicle entering the turn, the turning radius should be close to the value of 3700 mm.

In order for the wheels to roll without side slip, their axes must intersect at one point (Figure 4). This point \((O)\) is called the center of rotation [12].

The steering angles of the steered wheels are related by the ratio:

\[
\operatorname{ctg} \theta_{\text{out}} - \operatorname{ctg} \theta_{\text{in}} = \frac{l_0}{L},
\]

where \( \theta_{\text{out}} \) is the angle of rotation of the outer wheel; \( \theta_{\text{in}} \) is the angle of rotation of the inner wheel.

The swing angle of the outer wheel is calculated by the formula:

\[
\theta_{\text{out}} = \arctg \left( \frac{L}{(R-r_f)^2-L^2} \right) = 24^\circ 46\prime,
\]

Knowing the angle of rotation of the outer wheel, we calculate the angle of rotation of the inner wheel by transforming formula (17):

\[
\theta_{\text{in}} = \arctg \left( \frac{\operatorname{ctg} \theta_{\text{out}} - \frac{l_0}{L}}{\operatorname{ctg} \theta_{\text{out}}} \right) = 35^\circ 40\prime.
\]

Formulas (17,18,19) are valid only for a car with rigid wheels. In our case, the car moves on elastic tires, as a result of which there is a drift, since lateral forces act on the car during a cornering. Due to this, the slip angle must be taken into account when calculating. For sports cars, the slip angle of the outer wheel is close to the value \( \alpha_o = 3^\prime \) [15], and for the inner wheel, the slip angle is very small, because the main part of the load in the turn falls on the outer wheel.

The actual steering angle of the outer wheel for a car with elastic tires will be calculated using the formula:

\[
\theta_{\text{actual}}^{\text{out}} = \theta_{\text{out}} + \alpha_{\text{out}} = 27^\circ 46\prime.
\]

The obtained calculations must be put into the kinematic diagram of the suspension and steering system, the simulation model of which was formed in the Adams Car software by MSC Software (Figure 5), using the iterative selection method and taking into account such requirements as:

1. Coordination of steering kinematics and suspension kinematics;
2. Configuration requirements of individual units and parts of the car;
3. Preservation of the required angles of rotation of the steered wheels (18, 20).
On the basis of simulation, a graph of the dependence of the steering angles on the steering rack travel was obtained (Figure 6).

To minimize the decrease in the indicators of points 1 and 2, the values of the angles of rotation of the steered wheels were changed in relation to the required ones, and for the inner and outer wheels were 34°48´ and 29° respectively [13]. Comparison of required and obtained values is shown in the graph (Figure 7). The upper curve is the required dependence, the lower is obtained from the simulation.

**Figure 5.** Steering system and suspension elements in Adams Car software

**Figure 6.** Angles of rotation of the steered wheels depending on the travel of the steering rack

**Figure 7.** Dependence of the difference between the angles of rotation of the wheels on the angle of rotation of the inner wheel

### 4. Performance indicators of mechatronic systems

During developing the mechatronic system, experiments were carried out to measure the characteristics of the DC electric motor and the forces on the shift shaft and the clutch lever. Experimental data are shown in Table 4.
Table 4. Experimental data

| Parameter                                      | Value  |
|-----------------------------------------------|--------|
| **Shift shaft**                               |        |
| 1. Maximum shaft rotation angle required to change gear, ° | 25     |
| 2. Maximum torque required for gear shifting, N / m | 7,056  |
| **Clutch lever**                              |        |
| 3. Lever stroke, mm                           | 27     |
| 4. Lever arm, mm                              | 36.5   |
| 5. Maximum torque required to disengage the clutch, N / m | 10.55215 |
| 6. Lever rotation angle, °                    | 42.4   |
| **DC motor**                                  |        |
| 7. Torque, N / m                              | 3      |
| 8. Shaft rotation frequency, rpm              | 65     |
| 9. Starting torque, N / m                     | 14     |

Based on the data obtained, the development of mechanisms was made that converts and transmits the torque of electric motors for shifting gears and disengaging the clutch. After that, the development, prototyping, assembly and testing of fasteners were carried out: system of supports for attaching the clutch actuator, fastening the clutch lever position sensor, mounting plate for the gearshift actuator, supporting element of the gearshift actuator, fastening the gearshift sensor.

At this stage of development, the results of the work are the ability, using mechatronic systems, to disengage the clutch, change gear and engage the clutch in 0.94 seconds, which is comparable to the time of shifting gears in manual mode. The “quick shift” feature reduces gear change times to 0.36 seconds.

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