How to improve steerability and stability of a bus using constructive factors

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A 3-D mathematical model of four-wheeled vehicle is presented, it reveals the influence of structural factors on its critical speed when moving along the trajectory of "turning radius of 35 meters."

The present stage of development of the world and domestic automobile industry is characterized by a constantly growing number of cars on the roads, and increase of their average speeds at performance of transport work. Under the circumstances active safety improving becomes the major reason for vehicle’s construction refining. In curvilinear motion, the best way to attain active safety is to preserve a car’s steerability and stability, and this depends on traffic conditions and its design parameters.

Normative documents [1…5] provide for testing of the vehicle for control and stability in the performance of maneuvers "turning circle". At movement on standardized marked trajectories define critical speeds when a car loses controllability and stability. The values of critical speeds are standardized for all categories of vehicles. The ability to fit in a given trajectory of motion with the normalized maximum speed depends on the size and ratio of vehicle design parameters. At the design stage of the car designer should choose its design parameters taking into account the requirements of normative documents for the indicators of steerability and stability.

Taking this into account the given research on this problem is actual and timely. The urgency of problems of control and stability of automobiles caused a large number of publications on the subject [6…15].

The purpose of the work is to determine the influence of the design parameters of the bus related to the M3 category vehicles to the critical speed when performing the maneuver "turn radius 35 m". The spatial (3-D) mathematical model of the vehicle [15], based on the concept of introduction of the sprung mass in the form of a solid body (Figure 1), was used for the study of the control and stability of the bus.
One stationary and several movable coordinate systems were chosen to display the equations of movement of the car. The following generalized coordinates are entered, which uniquely define the system state:

- $\alpha, \beta$ – angles of longitudinal and transverse rolls of sprung mass (body);
- $\theta$ – the course angle formed by the longitudinal axis of the vehicle $O_1x_1$ with the axis $Ox$ of the stationary coordinate system;
- $z_i$ – distance from the center of the $i$ wheel to the support surface;
- $\gamma$ – the angle formed by the longitudinal main axis of the inertia sprung of the mass (body) with the plane $O_2x_2y_2z_2$;
- $\phi$ – the angle of the right front steering wheel;
- $v$ – is the speed of the vehicle's translational motion.

When describing the interaction of an elastic car wheel loaded with longitudinal and lateral forces, with a solid support surface used hypothesis of longitudinal pseudo-slip (creep) and transverse creep or drift [16] and nonlinear dependence of lateral Forces $F_y$ from the angle of drift $\delta = \frac{u}{v}$, where the transverse $U$ and longitudinal $V$ components of the speed of the center $i$ - th wheel, was approximated expression [10]:

$$F_y = -k_y \delta \frac{\phi F_z}{\sqrt{(\phi F_z)^2 + (k_y \delta)^2}},$$

where $k_y$ is the drift coefficient on the linear part of the dependence $F_y = F_y(\delta)$; $\phi$ – coefficient of traction of the wheel with a supporting surface; $F_z$ – normal load of the wheel.

At small values of an angle of drift this dependence expresses linear drift $F_y = -k_y \delta$, and at large values of a angle of drift - transition from rolling to a sliding at which the limit value of lateral force is equal to force of a coupling $F_{y,\text{max}} = -\phi F_z$. 

Figure 1. Mathematical model of the vehicle.
Equations of motion are obtained on the basis of Lagrange’s Equations [16]:

– Equation on the vertical coordinate $z$ of the center of the sprung mass:

$$m\ddot{z} + mg + \sum_{i=1}^{4} R_i = 0.$$  

– Equation on the angle $\alpha$ of the longitudinal roll of the sprung mass:

$$B_i\ddot{\alpha} = B_i\ddot{\alpha} + (mh^2 + B_i - A)\ddot{\alpha} + A\dot{\alpha}^2 + A\gamma\dot{\alpha}^2 - mgh\alpha + a_i(R_i + R_2) - a_2(R_3 + R_4)$$

– Equation on the angle $\beta$ of the transverse roll of the sprung mass:

$$A_1\ddot{\beta} + \beta[mh^2\alpha - A(\alpha + \gamma)]\dot{\alpha}\dot{\beta} + mh(\dot{u} + v\dot{\theta}) + mg\dot{h}\beta + b(R_1 - R_2 + R_3 - R_4) = 0.$$  

– Equation for the course angle $\theta$:

$$[C + mh^2\alpha^2 + B_i\beta^2 + A(\alpha + \gamma)^2]\ddot{\theta} = \Pi_4 - \dot{\alpha}\dot{\theta}[2mh^2\alpha + A(\alpha + \gamma)^2] - 2B_i\beta\dot{\alpha}\dot{\beta} + (A + B)\ddot{\alpha} + mh(\alpha\ddot{u} - \beta\dot{v} + \alpha\dot{v} + B\dot{\theta}u) + B_i\ddot{\alpha} - \ddot{\theta}[mh^2\alpha - A(\alpha + \gamma)].$$

The equation of transverse displacement of the sprung mass with the speed $u$:

$$m\ddot{u} + m\dot{v}\dot{\theta} + mh(\ddot{\beta} + 2\ddot{\alpha} + \alpha\ddot{\theta} - \beta\dot{\theta}^2) = \Pi_5.$$  

In the given formulas the following symbols are accepted:

$m$ – sprung mass; $g$ - acceleration of free fall; $h$ – distance from the center of the sprung mass to the plane passing through the top points of fastening of suspension; $A, B, C$ are the main central moments of inertia of the sprung mass respectively $(A_1 = A + mh^2; B = B_i + mh^2)$; $R_i$ – forces acting on the part of the suspension on the sprung mass $(i = 1, 4)$; $\Pi_4, \Pi_5$ – generalized forces by coordinates $\theta$ and $u$ accordingly; $a_i$ and $a_2$ – distances from the center of the sprung mass to the front and back suspension respectively.

The object of the study is a vehicle of category M3, the structural parameters of which are close to the parameters of the bus Paz-4234: distance from the center of mass to the front axle $a_1 = 2.3\ m$; distance from the center of the mass to the rear axle $a_2 = 2.22\ m$; height of the center of mass above the plane of fastening of suspension $h = 0.1\ m$; the main moments of inertia sprung mass $A = 25000\ kg\ m^2$; $B = 17000\ kg\ m^2$; $C = 17000\ kg\ m^2$; sprung mass $m$ in the loaded state 6315 kg, total 9875 kg; rigidity of the front suspension = 126.4 kN/m and rear suspension = 161.7 kN/m; the coefficient of resistance of shock absorbers of the front suspension $k_{1f} = 4.0\ kN/s/m$ and rear suspension $k_{2f} = 4.0\ kN/c/m$; the angle of the static inclination of the sprung mass $\gamma = 0.08\ rad$; the drift coefficient front wheel $500\ kN/rad$ and rear wheel $700\ kN/rad$.

In the study wide range of the sprung mass, the height of the center of sprung mass, the rigidity front and rear suspension, the resistance of shock absorbers front and rear suspension, the drift coefficients for the front and rear axles.

The sprung mass of the bus varied from 4 to 12 tons in 1 t increments. The results of the calculations are shown in Fig. 2. It follows from the figure that when the sprung mass increases from 4 to 12 tons, the critical speed varies from 15.33 to 15.35 m/s. Hence the conclusion follows that the value of the sprung mass has a very insignificant effect on the critical speed of the bus entering the turn.
Figure 2. The effect of changing the sprung mass on the critical speed of the entry into the rotation with a radius of 35 m

In calculations, the height of the center of the sprung mass varied with respect to the floor level in the range from -0.1 m to +0.4 m. Figure 3 shows the effect of the change in the height of the center of the sprung mass on the critical speed of the entrance to the turn. As the height of the center of mass in the indicated range increased, the critical speed of the entrance to the turn decreased from 15.37 to 15.31 m/s. Consequently, it is practically impossible to significantly affect the critical speed of the entrance to the turn, by changing the height of the center of the sprung mass above the road support surface within the limits permitted by the autobus parameters.

Figure 3. The influence of the change in the height of the center of the sprung mass of the bus on the critical speed of entry into the turn with a radius of 35 m

The effect of the rigidity of the suspension on the critical speed of the entry of the autobus into the turn is shown in Figure 4.
Figure 4. The influence of the rigidity of the bus suspension on the critical speed of entry into the turn of a radius of 35 m: a - front suspension; b - rear suspension

With the increase in the rigidity of the springs of the front suspension (Fig. 4, a) from 80 to 200 kN/m, the critical speed varies insignificantly within the range of 15.36 ... 15.32 m/s, and further variation of the rigidity from 200 to 220 kN/m leads to a sharp decrease in the critical speed to 15.2 m/s. The change in the stiffness of the rear suspension (Figure 4, b) from 120 to 300 kN/m practically does not affect the value of the critical speed.

Figure 5. Influence of the drag coefficient of shock absorbers of bus suspensions on the critical speed of entry into the turn of a radius of 35 m: a - the front suspension; b - a rear suspension

Thus, an increase in the resistance coefficient of the shock absorbers of the front suspension (Figure 5, a) from 1 to 15 kN s/m increases the critical speed from 15.32 to 15.4 m/s. With an increase in the resistance coefficient of the shock absorbers of the rear suspension (figure 5, b) from 1 to 35 kN s/m, the critical speed of the entrance to the turn increases from 15.34 to 15.39 m/s. In this case, unlike all previous graphs, the dependence of the critical velocity on the resistance coefficient of the shock absorbers is of a stepped nature. With an increase in the resistance coefficient of shock absorbers of the rear suspension, over 36 kN s/m, a bus skidded.

Simultaneous increase in the resistance coefficients of shock absorbers from 18 kN s/m to 36 kN s/m leads to an increase in the critical speed to 15.42 m/s.

The change in the resistance coefficients to the drift of the front and rear axles of the bus (Figure 6) affects the critical speed of the entrance to the turn in different ways.
An increase in the drift coefficient for the front axle (Figure 6, a) from 100 to 200 kN/rad leads to an increase in the critical speed from 15.3 to 15.38 m/s, and with a change in the drift coefficient from 200 to 3000 kN/rad the critical velocity first increases and then decreases. With an increase in the drift coefficient to more than 3000 kN/rad, a bus skidded.

The dependence of the critical velocity on the drift coefficient on the displacement of the rear axle (Figure 6, b) is rather complicated. As the drift coefficient increases from 100 to 200 kN/rad, the critical speed increases from 15.3 to 15.41 m/s, and at its subsequent increase from 300 to 3000 kN/rad decreases to 15.32 m/s.

Conclusions
1. The developed spatial mathematical model of the four-wheeled car allowed investigating the process of curvilinear movement of the bus PAZ-4234 along the trajectory "turning with a radius of 35 m".
2. The influence of the sprung mass, the height of the sprung mass center, the stiffness of the front and rear suspensions, the resistance coefficients of the shock absorbers of the front and rear suspensions, the drift coefficients for the leading and trailing axles, is shown on the critical speed of the entrance of the bus to the turn of a radius of 35 m.
3. It is established that the greatest impact on the critical speed of the entrance of the bus in the turn is due to changes in the stiffness of the front springs, the drift coefficients of the rear and front axle drive train and the resistance coefficient of the front suspension shock absorbers. To a lesser extent, the critical speed of the entrance of the bus to the turn of the height change of the center of the sprung mass and the resistance coefficient of the shock absorbers of the rear suspension is less affected, and the changes in the rigidity of the rear springs and sprung mass practically do not affect it.
4. The results of the research can be useful to designers and testers involved in the design, testing and finishing of bus structures, to provide standardized indicators of their handling and stability, as well as developing guidelines for working with undergraduates and graduate students.

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