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ALGORITHM OF ANTI-LOCK BRAKING SYSTEM FOR TWO-AXLE VEHICLES WITH ONE DRIVING AXLE WITH ADAPTIVE REDISTRIBUTION OF BRAKING FORCES

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The main purpose of active vehicle safety systems is to prevent an emergency situation. If such a situation arises, the system independently (without the participation of the driver) assesses the probable danger and, if necessary, prevents it by actively intervening in the driving process.
One of the ways to increase the active safety of vehicles when braking is the use of anti-lock braking systems (ABS). The main problems in ensuring the operation of the ABS, built on different control principles and with different control parameters, are the impossibility of directly determining the vehicle speed and, as a result, the slip coefficient, as well as the inability to effectively respond to changing road conditions during braking. For example, when braking on a slippery supporting surface and trying to avoid an obstacle in front, there is a risk of losing traction and skidding. The algorithms of the ABS operation developed at present do not ensure the prevention of the occurrence and development of skidding under the conditions indicated above.
The aim of the work is to increase the stability and controllability of two-axle vehicles with one driving axle during braking due to the adaptive redistribution of braking forces on the wheels. An algorithm for the operation of an anti-lock braking system with adaptive redistribution of braking forces on the wheels of a vehicle is proposed. Thanks to this algorithm, when braking on a slippery surface of a two-axle vehicle with one driving axle, the absence of wheel blocking and also skid resistance are ensured. The efficiency and effectiveness of the proposed algorithm when braking a two-axle vehicle with one driving axle on a slippery supporting surface were proved by the methods of simulation.

Keywords: *anti-lock braking system of a vehicle; stability and controllability of the vehicle; skid resistance.*

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Introduction

The anti-lock braking system (ABS) is one of the solutions to the problem of increasing vehicle active safety during braking. Recognizing this fact, the legislators of several countries are encouraging vehicle manufacturers to implement the ABS. As a result, in Russia, all M2 buses with more than 8 passenger seats are required to have an ABS (in the European Economic Community, since 2004, every new vehicle has been equipped with an ABS). Simultaneously, the algorithms for controlling ABS operations are being improved, resulting in a higher level of control over vehicle movement parameters during braking.

Based on the control parameters, the ABS is categorized by the following [1–6]:

- the value of the wheel slip coefficient corresponding to the maximum wheel adhesion (s-regulation);
- the maximum interaction coefficient value (μ-regulation); and
- the value and sign of the dμ/ds parameter, which characterizes the degree of approach to the maximum adhesion (gradient regulation).

When using s-regulation, the following basic algorithms and their combinations are typically used [7–13]:

- the equality mode of angular wheel and linear decelerations of the vehicle;
- the wheel slip coefficient and its further maintenance within the specified limits; and
- the threshold deceleration of the braking wheel.

Most of the disadvantages of s-regulation are since neither wheel slip nor wheel deceleration provides sufficient information to determine the optimal braking force control. The impossibility of directly determining vehicle speed, and thus the slip coefficient, and the impossibility of effectively responding to changes in road conditions during braking are the main problems in ensuring ABS operation based on different principles and with different control parameters.

This work is aimed at increasing the stability and controllability of two-axle vehicles during braking owing to the adaptive redistribution of braking forces on wheels.

Algorithm for estimating vehicle movement parameters during braking

Wheels are known to slow down with an increased braking torque during braking. At a certain point, the wheel deceleration exceeds the value that the vehicle deceleration cannot physically exceed. As the braking torque increases, the wheel declaration (not the vehicle) also increases. The physical vehicle declaration determines wheel deceleration threshold $\dot{\omega}_n$, and can be approximately calculated as follows:

$$\dot{\omega}_n = \frac{\boldsymbol{a}_{OX_T}}{r_s},$$

where $\boldsymbol{a}_{OX_T}$ is the current linear vector of acceleration projection $\boldsymbol{a}_O$ of the wheel center O (Fig. 1) on the plane of its rotation; r_s is the static radius of the wheel.

To determine $\boldsymbol{a}_{OX_T}$, we consider the accelerat ion plan for the wheel center during curvilinear vehicle movement and assume that the rolling plane of the wheel is perpendicular to the flat support base.

The acceleration $\boldsymbol{a}_O$ (Fig. 1) of point O (wheel center) during plane motion is equal to the vector sum of acceleration $\boldsymbol{a}_C$ of the center of mass of the vehicle (point C) and acceleration $\boldsymbol{a}_{OC}$ of point O during rotational motion around pole C:

$$\boldsymbol{a}_O = \boldsymbol{a}_C + \boldsymbol{a}_{OC}. \quad (1)$$

In Figure 1, C is the center of mass of the vehicle; O is the center of the vehicle wheel; CXY represents axes of the coordinate system associated with the center of mass of the vehicle; OX_TY_T represents the coordinate system axes associated with the center of the vehicle wheel; $\boldsymbol{a}_C$ represents the vector of acceleration of the vehicle mass center; $\boldsymbol{a}_O$ represents the vector of acceleration of the vehicle wheel center; $\boldsymbol{a}^{\tau}_{OC}$ represents the vector of tangential acceleration; $\boldsymbol{a}^{n}_{OC}$ is the vector of normal acceleration; $\boldsymbol{a}_{OX}$ represents the current linear vector of acceleration projection, $\boldsymbol{a}_O$ of the center O of the wheel on the X_T axis; Θ is the angle of rotation of the controlled wheel; ω_Z is the angular speed of the vehicle rotation about the vertical axis.

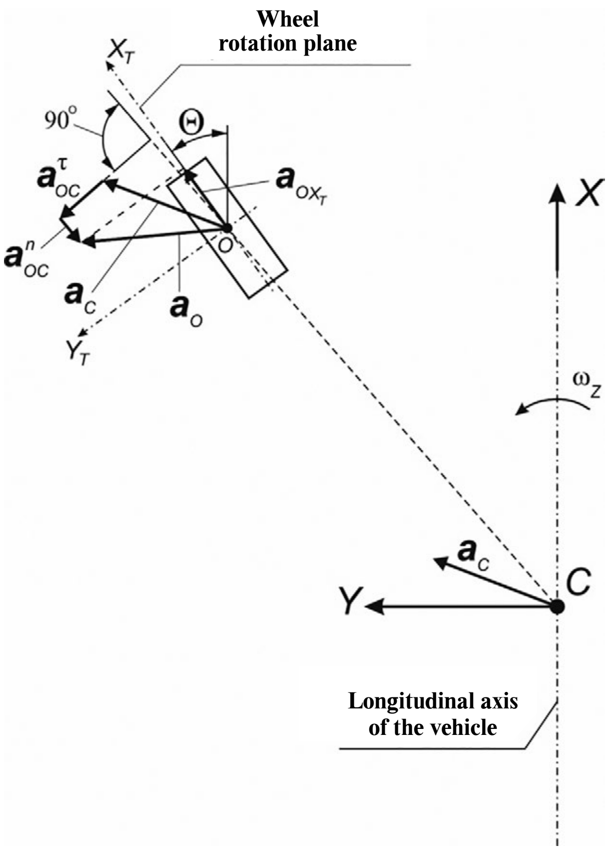

Fig. 1. Acceleration plan for the center of the wheel during curvelinear motion of vehicle

In the associated coordinate system, we take into account that the transfer velocity vector $\boldsymbol{V}_{OC}$ of point O relative to the pole C is as follows:

$$\boldsymbol{V}_{OC} = \boldsymbol{\omega} \times \boldsymbol{OC}, \quad (2)$$

where $\boldsymbol{\omega} = \left[\omega_x,\ \omega_y,\ \omega_z \right]$ is the vector of the angular velocity of point O relative to point C and $\boldsymbol{OC} = \left[x_o,\ y_o,\ z_o \right]$ is the radius vector from point O to point C in the axis of the associated coordinate system CXY projections.

Thus,

$$\boldsymbol{a}_{OC} = \boldsymbol{\varepsilon} \times \boldsymbol{OC} + \boldsymbol{\omega} \times \left(\boldsymbol{\omega} \times \boldsymbol{OC} \right),\ \ \boldsymbol{\varepsilon} = \frac{d\boldsymbol{\omega}}{dt}, \quad (3)$$

where $\boldsymbol{\varepsilon}$ is the vector of the angular acceleration of the vehicle.

It is noteworthy that the acceleration vector $\boldsymbol{a}_{OC}$ consists of tangent and normal components:

$$\boldsymbol{a}_{OC}^{\tau} = \boldsymbol{\varepsilon} \times \boldsymbol{OC},\ \ \boldsymbol{a}_{OC}^{n} = \boldsymbol{\omega} \times \left(\boldsymbol{\omega} \times \boldsymbol{OC} \right). \quad (4)$$

The vector of tangential acceleration $\boldsymbol{a}_{OC}^{\tau}$ is directed perpendicular to the CO ray. The normal acceleration vector $\boldsymbol{a}_{OC}^{n}$ is directed from the center of the wheel O to the center of mass C of the vehicle.

Thus, the vector modulus $\left| \boldsymbol{a}_{OX_T} \right| = a_{OX_T}$ can be defined as follows:

$$a_{OX_T} = a_{OX} \cos\Theta + a_{OY} \sin\Theta, \quad (5)$$

where a_{OX}, a_{OY} are the projections of the center O of the wheel acceleration vector $\boldsymbol{a}_O$ on the X and Y axes of the coordinate system associated with the center of mass of the vehicle.

The intended purpose of braking torques on wheels

The braking torque M_{Ti} on the i-th wheel can be determined as follows, taking into account the ABS operation:

$$M_{Ti} = h_{brake} h_{ABSi} h_{fbi} T_{max}, i = 1, \ldots, N, \quad (6)$$

where $h_{brake} = \left[0 \ldots 1 \right]$ is the degree to which the driver presses the brake pedal; $h_{ABSi} = \left[0 \ldots 1 \right]$ is the reduction degree of the effective braking torque on the i-th wheel due to the ABS; $h_{fbi} = \left[0 \ldots 1 \right]$ is the redistribution degree of the braking torque on the i-th wheel when braking on a straight line (taking into account the normal reaction redistributions between the front and rear axles); T_{max} is the maximum braking torque developed by the wheel brake mechanism; N is the number of wheels on the vehicle.

The value h_{ABSi} can be defined as follows:

$$h_{ABSi} = \left| \frac{\dot{\omega}_{ïîð}}{\dot{\omega}_i} \right| \frac{\omega_i}{\omega_{max}}, i = 1, \ldots, N,$$

$$\omega_{max} = max\left(\omega_i, i = 1, \ldots, N \right), \quad (7)$$

where ω_i is the current angular speed of rotation of the i-th wheel.

The cofactor $\left| \frac{\dot{\omega}_{nop}}{\dot{\omega}_i} \right|$ in Eq. (7) allows the braking torque on the i-th wheel to be reduced when its angular deceleration $\dot{\omega}_i$ exceeds the threshold value $\dot{\omega}_{nop}$. Using the fastest wheel of the vehicle as a reference, cofactor $2\ \frac{\omega_i}{\omega_{max}}$ allows for an adjustment in braking torque reduction.

An adaptive algorithm for braking force redistribution on the vehicle wheels

When a vehicle brakes on a straight-line section of motion, the vehicle "bounces" forward, the rear wheels are relieved from normal loads, and the front wheels take on additional load owing to inertial forces. Therefore, the dynamic normal load R_{1d} on the wheels of the front axle and R_{2d} on the wheels of the rear axle can be determined as follows for a two-axle vehicle:

$$R_{1d} = R_{1s} + \Delta R_1,\ \ R_{2d} = R_{2s} - \Delta R_2,$$

$$R_{1s} = \frac{Ml_1}{L},\ \ R_{2s} = \frac{Ml_2}{L},$$

where R_{1s}, R_{2s} are the normal reactions on the wheels of the front and rear axles, respectively in a static position; $\Delta R_1, \Delta R_2$ represent an increment of normal responses to the front and rear axles, respectively, during braking; M is the weight of the vehicle sprung parts; l_1, l_2

are the distances from the center of the vehicle mass to the front and rear axles, respectively; and $L = l_1 + l_2$ is the vehicle wheelbase.

On the assumption that the stiffness of the suspensions of all wheels is approximately equal, the increment of normal reactions to the front and rear axles ΔR_1 and ΔR_2, is defined as follows:

$$\Delta R_1 = M\frac{|\boldsymbol{a}_{Cx}|}{g}h_c\frac{l_1}{l_1^2 + l_2^2},\quad \Delta R_1 = M\frac{|\boldsymbol{a}_{Cx}|}{g}h_c\frac{l_2}{l_1^2 + l_2^2}, \tag{8}$$

where $|\boldsymbol{a}_{Cx}|$ is the projection module of the center of mass acceleration onto the *X*-axis of the associated coordinate system and h_c is the height of the vehicle center of mass.

We defined the value $h_{fbi} = \dfrac{R_{id}}{R_{is}}$. Finally, using Eq. (8), we obtained the following for braking a vehicle in a straight-line section of motion ($|\Theta| \le 3°$):

$$h_{fb1,3} = 1 + \frac{|\boldsymbol{a}_{Cx}|}{g}h_c\frac{L}{l_1^2 + l_2^2} \text{ – for the front axle wheels,}$$

$$h_{fb2,4} = 1 - \frac{|\boldsymbol{a}_{Cx}|}{g}h_c\frac{L}{l_1^2 + l_2^2} \text{ – for the rear axle wheels.} \tag{9}$$

If $|\Theta| \le 3°$, and $h_{fbi} = 1$.

Testing the performance and efficiency of the ABS algorithm

Theoretical vehicle braking studies were performed using simulation mathematical modeling. The aspects of the mathematical model of motion have been considered in previous studies [14–19].

Using simulation modeling methods in testing the performance and efficiency of the proposed algorithm, it was discovered that emergency braking on a slippery road (coefficient of adhesion at full slip 0.35) of a passenger vehicle with a gross weight of 6000 kg at an initial speed of 60 km/h with a simultaneous turn of the steering wheel (the driver's attempt to bypass the obstacle) causes front axle drift. The trajectory of the vehicle's motion during braking is presented in Figure 2.

To avoid this drift in the front axle, it is required to first recognize the occurrence and development of this process. For this purpose, we used previous data [20], where a parameter $\delta_V = \left\| \boldsymbol{V}_{C1} \right| - \left| \boldsymbol{V}_{C2} \right\|$ represents the difference in the estimate of the linear velocities of the center of the vehicle mass, first using the linear speed of the center of the front axle (vector $\boldsymbol{V}_{C1}$), and subsequently using the linear speed of the center of the rear axle (vector $\boldsymbol{V}_{C2}$), as a diagnostic sign of the onset of front axle drift or rear axle skidding. Figure 3 presents a graph of the change in time of the diagnostic sign δ_V while the vehicle is braking.

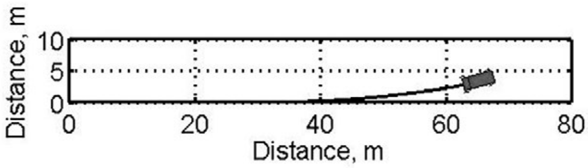

Fig. 2. Trajectory of movement of a vehicle with a gross weight of 6000 kg when braking with ABS without anti-skid function of the front axle

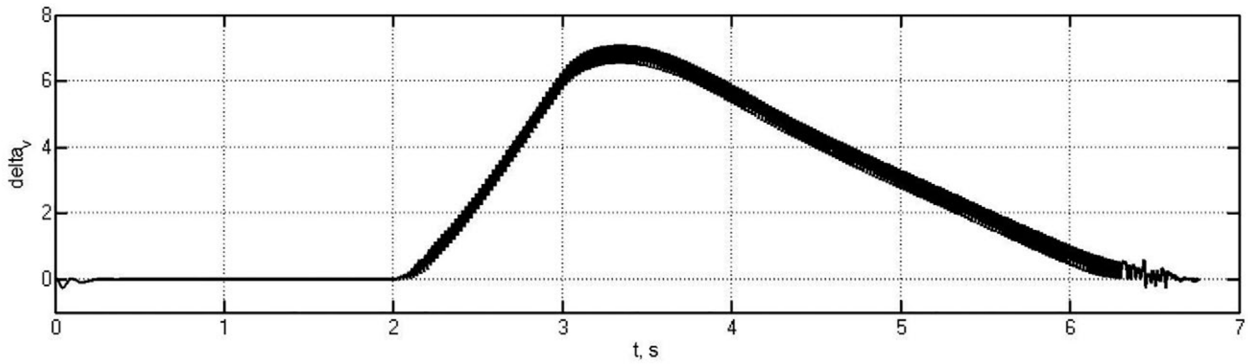

Fig. 3. The graph of the change in time of the diagnostic characteristics δ_V when braking the vehicle

The graph in Figure 3 shows a diagnostic sign that appears during braking $\delta_V > 0$, indicating the front axle drift occurrence.

A counter-rotation moment for skid resistance at the front axle is required owing to increased braking of the rear wheel inner concerning the direction of rotation. However, because more braking can cause the wheel to become stuck, it is necessary to release the brakes of all wheels, except for the rear wheel inner, concerning the rotational direction. Thus, Eq. (6) for determining the braking torque on each wheel is as follows:

$$M_{mi} = h_{brake} h_{ABSi} h_{fbi} h_{ESPi} T_{max}, i = 1,\ldots,N, \quad (10)$$

where $h_{ESPi} = [0\ldots1]$ is the degree of reduction of the effective braking torque on the *i*-th wheel due to the skid resistance algorithm at the front axle during braking (anti-skid function of the front axle).

Thus, considering the rule of signs adopted in the simulation, the algorithm for determining the value $h_{ESPi}, i = 1,\ldots,N$ should be as follows.

If $\Theta_1 > 0°$ (turn left) and $\delta_V > 0$ (front axle drift), then $h_{ESP1} = h_{ESP3} = h_{ESP4} = 1 - C_u \delta_V ; h_{ESP2} = 1$.

If $\Theta_1 < 0°$ (turn to the right) and $\delta_V > 0$ (front axle drift), then $h_{ESP1} = h_{ESP2} = h_{ESP3} = = 1 - C_u \delta_V ; h_{ESP} = 1$.

In the above equations, C_u is the controller's gain which is adjusted individually for each vehicle.

Using simulation modeling methods, the motion of a two-axle vehicle with a total mass of 6000 kg was simulated under the same conditions as described earlier to access the efficiency and performance of the proposed ABS operation during braking. Figure 4 presents the trajectory of the vehicle when braking with

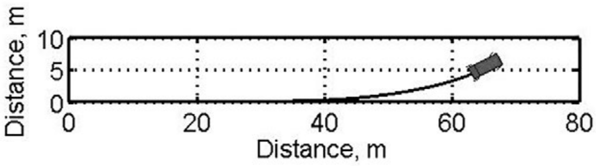

Fig. 4. Vehicle trajectory when braking with ABS and anti-skid function of the front axle

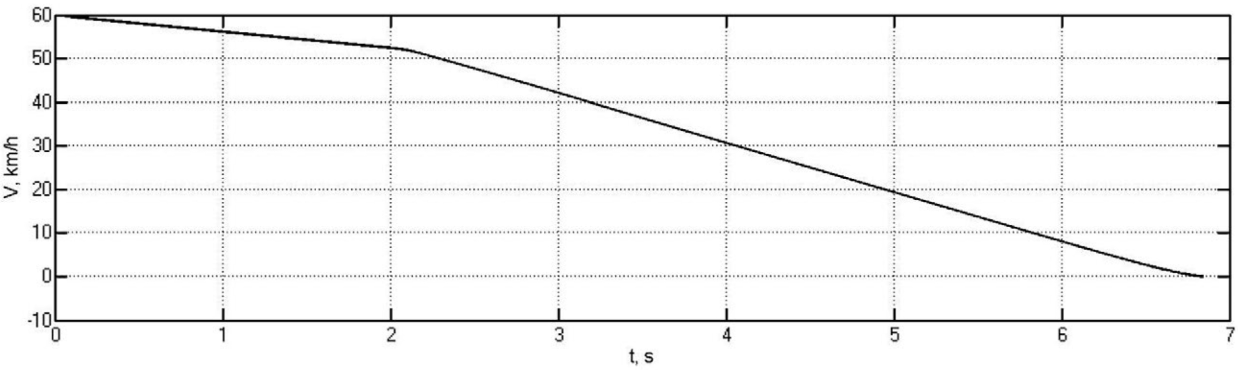

Fig. 5. Dependence of vehicle speed on time

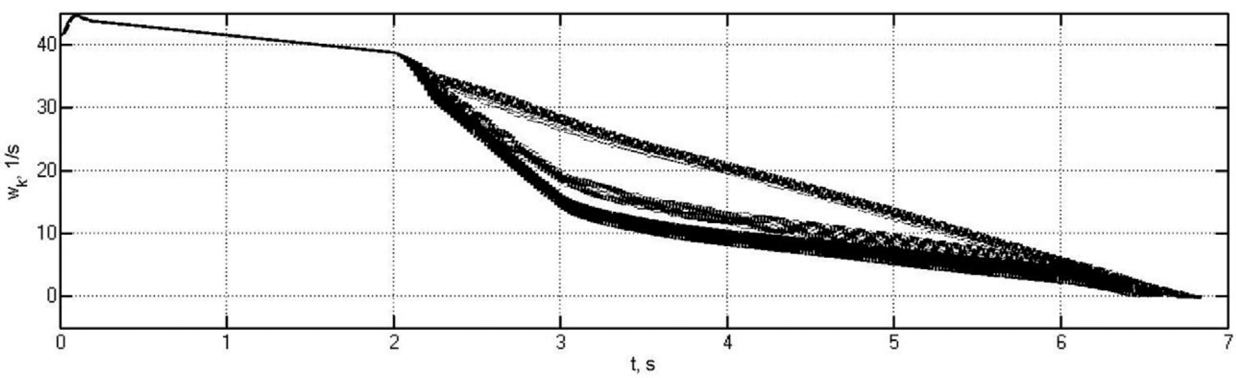

Fig. 6. Graphs of changes in angular speeds of wheels from time to time

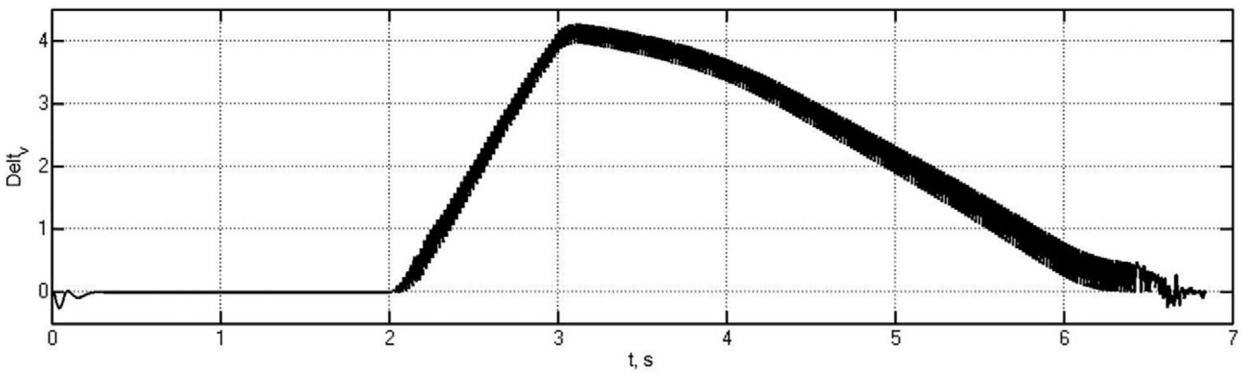

Fig. 7. The graph of the change in time of the diagnostic characteristics δ_V when braking a vehicle with ABS and with the function of countering the drift of the front axle

the ABS and the anti-skid function of the front axle, Figure 5 demonstrates the dependence of vehicle speed on time, Figure 6 presents graphs of changes in the angular velocities of the wheels on time, and Figure 7 presents a graph of the change in time of the diagnostic sign δV during braking.

Figures 4 to 7 illustrate that when braking with ABS and the anti-skid function of the front axle, the wheels do not lock, and the maximum value of the diagnostic sign δV decreases by 40%, indicating that the proposed algorithm for the operation of an ABS with an anti-skid function of the front axle is operable and efficient.

Conclusions

When braking a vehicle on a slippery supporting surface with a simultaneous steering wheel rotation, an algorithm for the operation of an ABS with an anti-skid function of the front axle for two-axle vehicles is proposed and characterized by not only the absence of wheel blocking but also an increase in vehicle controllability.

The operability and efficiency of the proposed algorithm for the operation of an ABS with an anti-skid function of the front axle have been proved using simulation modeling methods of braking a vehicle on a slippery supporting surface with a simultaneous steering wheel movement.

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АЛГОРИТМ РАБОТЫ АНТИБЛОКИРОВОЧНОЙ СИСТЕМЫ ДЛЯ ДВУХОСНЫХ АВТОМОБИЛЕЙ С ОДНОЙ ВЕДУЩЕЙ ОСЬЮ С АДАПТИВНЫМ ПЕРЕРАСПРЕДЕЛЕНИЕМ ТОРМОЗНЫХ УСИЛИЙ

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Главным предназначением систем активной безопасности автомобиля является предотвращение аварийной ситуации. При возникновении такой ситуации система самостоятельно (без участия водителя) оценивает вероятную опасность и при необходимости предотвращает ее путем активного вмешательства в процесс управления автомобилем.
Одним из способов повышения активной безопасности автомобилей при торможении является использование антиблокировочных систем (АБС). Основными проблемами в обеспечении работы АБС, построенных на разных принципах управления и с разными параметрами управления, являются невозможность прямого определения скорости автомобиля и, как следствие, коэффициента буксования, а также невозможность эффективно реагировать на изменяемые дорожные условия в процессе торможения. Например, при торможении на скользкой опорной поверхности и попытке объехать находящееся впереди препятствие существует опасность потери сцепления колес с дорогой и возникновения заноса. Разработанные в настоящее время алгоритмы работы АБС не обеспечивают предотвращение возникновения и развития заноса в указанных выше условиях.
Целью работы является повышение устойчивости и управляемости двухосных автомобилей с одной ведущей осью при торможении за счет адаптивного перераспределения тормозных усилий на колесах. Предложен алгоритм работы антиблокировочной системы с адаптивным перераспределением тормозных усилий на колесах автомобиля. Благодаря данному алгоритму, при торможении на скользкой опорной поверхности двухосного автомобиля с одной ведущей осью обеспечивается не только отсутствие блокировки колес, но и противодействие заносу. Методами имитационного моделирования доказаны работоспособность и эффективность предложенного алгоритма при торможении двухосного автомобиля с одной ведущей осью на скользкой опорной поверхности.

Ключевые слова: *антиблокировочная система автомобиля; устойчивость и управляемость автомобиля; противодействие заносу.*

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