IMPACT OF LONGITUDINAL SLOPE, LAYOUT AND LOADING ON THE BRAKING PROCESS OF AN ARTICULATED VEHICLE

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Abstract: *Calculation of braking properties has always been one of the most important stages in the design of any vehicle. With the advent of various active safety systems, such as anti-lock braking systems, collision avoidance systems, automatic emergency braking systems, mathematical models have become much more complicated, but the approach to studying the braking process remains almost unchanged. The authors of the article note the imperfection of modern methods of calculating the braking properties and offer a number of refinements that allow for taking into account factors such as the longitudinal slope of the road, as well as the layout and loading of the vehicle in relation to articulated vehicles with a semi-trailer and a full trailer.*

Keywords: *special purpose vehicles; braking efficiency; pneumatic brake drive; active safety; adhesion coefficient; weight redistribution; iterative method*

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I. INTRODUCTION

Evaluation of braking properties is one of the most important steps in calculating not only traditional vehicle control systems [1], but also modern active safety systems, which include an anti-lock system [2–8], electronic stability control [9–11], collision avoidance systems [12–19] and automatic emergency braking systems [20–21]. When calculating the algorithms for the systems mentioned above, they try to take into account the influence of the friction coefficient φ on the braking properties of a vehicle, which has its own characteristics in the presence of studded tires [22–23]. However, the currently existing methods and mathematical models, as a rule, do not take into account the asynchrony of reaching the grip limit by wheels of different axles, which is especially important for articulated vehicles [24–25]. An illiterate assessment of the inhibitory properties leads to the adoption of erroneous decisions in the organization of traffic [26] and, as a result, an increase in the mental tension of drivers [27].

In the traditional method of calculating the braking parameters, the balance of forces is considered under separate action on the tractor and trailer [28], and therefore the influence of factors such as loading from full mass m_{full} to curb mass m_{curb} and the effect of redistribution of gravity under the deceleration action d, which increases with increasing center of mass h of the links of the road train, the real ratio of the maximum braking force P_{bm} developed by the braking mechanisms, and the adhesion limit Φ, which can lead to a decrease in the specific forces γ_i on some axes, as well as to the transfer function of the brake mechanism K_b.

The brake system is designed for the full mass m_{full}, therefore, with a decrease in load, to achieve the maximum steady-state deceleration d_m, less braking force P_{bm} will be required. When calculating the braking efficiency for curb mass m_{curb} or partial load m_{part}, it is necessary to take into account the decrease in the center of mass, which can be very significant for trailed links.

Previously, the authors proposed refinements to take into account the redistribution of gravity, but that did not take into account the longitudinal reaction in the coupling device [29–32]. In this article, the authors propose refinements that take into account the influence of the longitudinal inclination angle, as well as the layout and loading of the articulated vehicle on its braking process, in particular, on wheel lock and longitudinal reaction in the coupling device.

II. FORCE SCHEME AT THE BRAKING PROCESS

In the general case, when braking, the following forces act on the links of the articulated vehicle (Fig. 1-2):

– the gravity G and the normal reactions opposing it on the wheels N_i and on the fifth wheel N_{fw};

– the braking forces P_{bi} and the inertia force m·d opposing them;

– the resulting force in the coupling device $P_{fw/ht}$;

– rolling resistance forces on wheels P_{fi};

– air resistance force P_w.

When calculating the braking process, the forces P_{fi} and P_w can be neglected, since they create additional resistance and positively affect the braking efficiency. In addition, during braking, the speed decreases, and the influence of these forces decreases. During emergency braking, all wheels are theoretically reduced to sliding, therefore, the force is P_f=0, and rotating masses (wheels, transmission and engine crankshaft) can be ignored.

Assumptions:

1) the dynamic radius of the wheel r_w is considered equal to the static one; the change in r_w with increasing weight acting on the wheel is not taken into account;

2) the weight along the axes of the trolley is distributed evenly;

3) the longitudinal displacement of the center of gravity of the sprung masses under the action of inertia is neglected due to its insignificance;

4) the coefficient of grip is constant in time and is evenly distributed over all the wheels of

the articulated vehicle (φ=const);

5) after reaching the maximum clamping force of the friction surfaces, the braking forces on the wheels have a constant value of P_{bm}, although in reality the braking process is accompanied by heating of the friction surfaces, as a result of which the coefficient of friction between the friction elements changes.

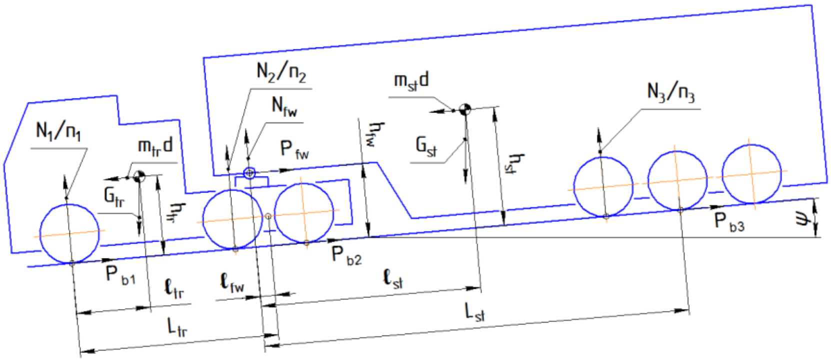

Fig. 1. *Articulated vehicle with a semi-trailer. Force scheme*

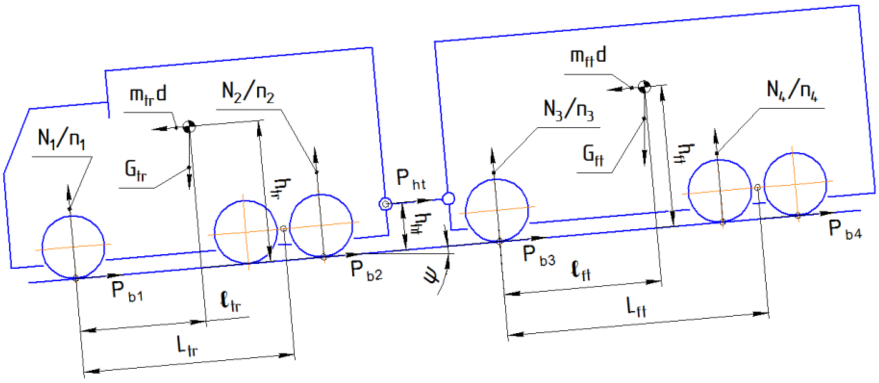

Fig. 2. *Articulated vehicle with a full trailer. Force scheme*

Initial data for calculation are given in the table 1, where:

– m_{tr}, m_{st}, m_{ft} – mass of tractor, semi-trailer, full trailer, respectively;

– $m_{1,2,3,4}$ – mass attributable to this axis;

– m_{fw} – mass attributable to the fifth wheel coupling device;

– L_{tr}, L_{st}, L_{ft} – wheelbase of the tractor, semi-trailer, full trailer;

– ℓ_{fw} – longitudinal displacement of the fifth wheel coupling device;

– h_{tr} – height of the center of mass of the tractor;

– h_{fw} – height of the fifth wheel coupling;

– h_{ht} – drawbar height of the towbar;

– r_w – radius of the wheel;

– ℓ_{exp}, ℓ_{lev} – shoulder of expanding force and length of the drive lever of the brake mechanism, respectively;

– f_{fr} – friction coefficient of the brake mechanism;

– S_{Dtr}, S_{Dst}, S_{Dft} – working area of the brake chambers of the tractor, semi-trailer, full trailer, respectively;

– $n_{1,2,3,4}$ – number of axles in the trolley.

Table 1. *The given data*

Parameter	GAZON NEXT C47R13 + Chajka-Servis 938410	URAL NEXT 7470 + PPO 22-23D UST 94651	KAMAZ-4310 + SZAP-8305	
			curb mass	full mass
m_{tr}, kg	8700	16500	8745	15205
m_1, kg	2300	6000	4315	5020
m_2, kg	6400	10500	4430	10185
$m_{st/ft}$, kg	8500	22050	4500	18000
m_{fw}, kg	3240	8050	–	
m_3, kg	5260	14000	1500	6000
m_4, kg	–	–	3000	12000
L_{tr}, m	4,5	5,5	4	
ℓ_{fw}, m	0	0,12	–	
$L_{st/ft}$, m	8	12	4,327	
$h_{fw/ht}$, m	1,1	1,32	0,96	
h_{tr}, m	0,835	1,03	1,04	1,16
r_w, m	0,419	0,6	0,59	
ℓ_{exp}, m	0,27	0,38	0,38	
ℓ_{lev}, m	0,205	0,18	0,18	
f_{fr}	0,4	0,3	0,25	
S_{Dtr}, mm^2	9032	12903	15484	
$S_{Dst/ft}$, mm^2	9032	19355	15484	
n_1	1	1	1	
n_2	1	2	2	
n_3	1	2	1	
n_4	–	–	2	

In the design diagrams, the force acting in the coupling device is directed backward, since this case is more favorable: firstly, the stability of movement increases (the likelihood of folding the articulated vehicle decreases), and secondly, the effect of weight redistribution decreases. However, due to the significant delay of the pneumatic brake drive of the trailed link relative to that of the tractor, the vector of the mentioned force will be directed forward. In order to avoid folding the articulated vehicle, a brake valve and a trailer brake control valve are used with special characteristics that provide some force ahead of the braking of the trailer link relative to the tractor. Most effectively the folding problem is solved by an electro-pneumatic brake drive [33].

The equations of the resulting forces for the tractor, semi-trailer and full trailer, respectively:

$$\sum F_{tr} = G_{tr} \sin\psi + m_{tr} d_{av} - P_{b1} - P_{b2}\,; \tag{1}$$

$$\sum F_{st} = G_{st} \sin\psi + m_{st} d_{av} - P_{b3}\,; \tag{2.1}$$

$$\sum F_{ft} = G_{ft} \sin\psi + m_{ft} d_{av} - P_{b3} - P_{b4}\text{, N.} \tag{2.2}$$

The equation of forces for determining the direction of the force acting in the fifth wheel coupling of an articulated vehicle with a semi-trailer:

$$P_{fw} = \sum F_{st} - \sum F_{tr}\text{, N.} \tag{3.1}$$

The same for determining the direction of the force acting in the towing device of an articulated vehicle with a full trailer:

$$P_{ht} = \sum F_{ft} - \sum F_{tr}\text{, N.} \tag{3.2}$$

Equations of moments for an articulated vehicle with a semi-trailer:

$$M_{fw} = 0 = N_3 L_{st} + G_{st}\left(h_{st} \sin\psi - \lambda_{st} \cos\psi\right) + m_{st} dh_{st} - P_{fw} h_{fw}\,;$$

$$M_3 = 0 = N_{fw} L_{st} - G_{st}\left(h_{st} \sin\psi + \left(L_{st} - \lambda_{st}\right)\cos\psi\right) - m_{st} dh_{st} + P_{fw} h_{fw}\,;$$

$$M_1 = 0 = N_2 L_{tr} + G_{tr}\left(h_{tr} \sin\psi - \lambda_{tr} \cos\psi\right) + m_{tr} dh_{tr} - P_{fw} h_{fw} - G_{fw}\left(L_{tr} - \lambda_{fw}\right);$$

$$M_2 = 0 = N_1 L_{tr} - G_{tr}\left(h_{tr} \sin\psi + \left(L_{tr} - \lambda_{tr}\right)\cos\psi\right) - m_{tr} dh_{tr} + P_{fw} h_{fw} - G_{fw} \lambda_{fw}\,.$$

Equations of moments for an articulated vehicle with a semi-trailer:

$$M_3 = 0 = N_4 L_{ft} + G_{ft}\left(h_{ft} \sin\psi - \lambda_{ft} \cos\psi\right) + m_{ft} dh_{ft} - P_{ht} h_{ht}\,;$$

$$M_4 = 0 = N_3 L_{ft} - G_{ft}\left(h_{ft} \sin\psi + \left(L_{ft} - \lambda_{ft}\right)\cos\psi\right) - m_{ft} dh_{ft} + P_{ht} h_{ht}\,;$$

$$M_1 = 0 = N_2 L_{tr} + G_{tr}\left(h_{tr} \sin\psi - \lambda_{tr} \cos\psi\right) + m_{tr} dh_{tr} - P_{ht} h_{ht}\,;$$

$$M_2 = 0 = N_1 L_{tr} - G_{tr}\left(h_{tr} \sin\psi + \left(L_{tr} - \lambda_{tr}\right)\cos\psi\right) - m_{tr} dh_{tr} + P_{ht} h_{ht}\,.$$

III. STATIC REACTIONS

The articulated vehicle, which is stationary on a horizontal surface (ψ=0), is affected only by gravity G and normal reactions on N_i wheels, id est d=0, P_{bi}=0, consequently, $P_{fw/ht}$=0. To determine the weight per axis, it is necessary to solve the system of equations of moments relative to each axis. Part of the gravity force G_{fw}=–N_{fw} from the semi-trailer is transmitted through the fifth wheel coupling to the tractor, therefore it is advisable to first solve the system of moment equations for the semi-trailer, and then to the tractor.

Thus, normal static reactions on the semi-trailer:

$$N_{03} = \frac{G_{st}\lambda_{st}}{L_{st}};\; N_{0fw} = \frac{G_{st}\left(L_{st} - \lambda_{st}\right)}{L_{st}}, \text{N}.$$

Normal static reactions on the full trailer:

$$N_{04} = \frac{G_{ft}\lambda_{ft}}{L_{ft}};\; N_{03} = \frac{G_{ft}\left(L_{ft} - \lambda_{ft}\right)}{L_{ft}}, \text{N}.$$

Normal static reactions on the tractor:

$$N_{02} = \frac{G_{tr}\lambda_{tr} + G_{fw}\left(L_{tr} - \lambda_{fw}\right)}{L_{tr}};\; N_{01} = \frac{G_{tr}\left(L_{tr} - \lambda_{tr}\right) + G_{fw}\lambda_{fw}}{L_{tr}}, \text{N},$$

where G_{fw}=0 for an articulated vehicle with a full trailer.

Knowing the distribution of mass along the axes, you can determine the location of the center of gravity along the longitudinal axis of each link:

$$\lambda_{tr} = L_{tr}\frac{m_2}{m_{tr}};\; \lambda_{st} = L_{st}\frac{m_3}{m_{st}};\; \lambda_{ft} = L_{ft}\frac{m_4}{m_{ft}}, \text{m}.$$

The results are given in the table 2.

Table 2. *Coordinates of gravity centers*

Parameter	GAZON NEXT C47R13 + Chajka-Servis 938410	URAL NEXT 7470 + PPO 22-23D UST 94651	KAMAZ-4310 + SZAP-8305	
			curb mass	full mass
ℓ_{tr}, m	3,31	3,5	2,03	2,68
$\ell_{st/ft}$, m	4,95	7,62	2,88	2,88

IV. BRAKING FORCES

The gear ratio of the brake mechanism of this axis:

$$K_{bi} = \frac{2 \cdot \lambda_{exp} \cdot f_{fr} \cdot \lambda_{lev}}{r_w \cdot A \cdot d_{cam}},$$

where 2 is the number of brake pads;

ℓ_{exp} – expanding shoulder, m;

f_{fr} – coefficient of friction;

ℓ_{lev} – length of drum brake drive lever, m;

r_w – radius of the wheel, м;

A=0,82 – design coefficient (depending on the type of mechanism);

d_{cam}=0,04 m – initial diameter of expansion brake cam.

Maximum braking force on this axis:

$$P_{bmi} = k_i K_{bi} S_{Di} \left(p_{fin} - 0{,}065\right),\ N,$$

где $k_i=2\cdot n_i$ – number of brake chambers per axle;

S_{Di} – working area of brake chamber diaphragm, mm^2;

p_{fin} – final pressure (in this case, maximum p_m), MPa.

V. WEIGHT REDISTRIBUTION

In addition to the gravity G and normal N_i reactions, the articulated vehicle during braking on a slope is affected by the braking forces on the P_{bi} wheels, the inertia force m·d and the resulting force in the coupling device $P_{fw/ht}$. Normal reactions on the axes during braking are found from the equations of moments.

For an articulated vehicle with a semi-trailer:

$$N_3 = \frac{G_{st}\left(\lambda_{st}\cos\psi - h_{st}\sin\psi\right) - m_{st} d h_{st} + P_{fw} h_{fw}}{L_{st}}; \tag{4.1}$$

$$N_{fw} = \frac{G_{st}\left(\left(L_{st} - \lambda_{st}\right)\cos\psi + h_{st}\sin\psi\right) + m_{st} d h_{st} - P_{fw} h_{fw}}{L_{st}}; \tag{5.1}$$

$$N_2 = \frac{G_{tr}\left(\lambda_{tr}\cos\psi - h_{tr}\sin\psi\right) - m_{tr} d h_{tr} + P_{fw} h_{fw} + G_{fw}\left(L_{tr} - \lambda_{fw}\right)}{L_{tr}}; \tag{6.1}$$

$$N_1 = \frac{G_{tr}\left(\left(L_{tr} - \lambda_{tr}\right)\cos\psi + h_{tr}\sin\psi\right) + m_{tr} d h_{tr} - P_{fw} h_{fw} + G_{fw}\lambda_{fw}}{L_{tr}}. \tag{7.1}$$

For an articulated vehicle with a full trailer:

$$N_4 = \frac{G_{ft}\left(\lambda_{ft}\cos\psi - h_{ft}\sin\psi\right) - m_{ft} d h_{ft} + P_{ht} h_{ht}}{L_{ft}}; \tag{4.2}$$

$$N_3 = \frac{G_{ft}\left(\left(L_{ft} - \lambda_{ft}\right)\cos\psi + h_{ft}\sin\psi\right) + m_{ft} d h_{ft} - P_{ht} h_{ht}}{L_{ft}}; \tag{5.2}$$

$$N_2 = \frac{G_{tr}\left(\lambda_{tr}\cos\psi - h_{tr}\sin\psi\right) - m_{tr} d h_{tr} + P_{ht} h_{ht}}{L_{tr}}; \tag{6.2}$$

$$N_1 = \frac{G_{tr}\left(\left(L_{tr} - \lambda_{tr}\right)\cos\psi + h_{tr}\sin\psi\right) + m_{tr} d h_{tr} - P_{ht} h_{ht}}{L_{tr}}. \tag{7.2}$$

VI. TRACTION LIMIT

The grip limit on this axis is determined:

$$\Phi_i = N_i \varphi, \text{ H}. \tag{8}$$

The values of P_{bmi} and Φ_i on each axis are compared:

1. If $P_{bmi}>\Phi_i$, then the wheels on this axis during braking reach the traction limit and are blocked, which means that braking will be performed with maximum efficiency ($\gamma_i=\varphi$), and the actual braking force $P_{biF}=\Phi_i$.

2. If $P_{bmi}<\Phi_i$, then it will not be possible to achieve wheel lock on this axis. Therefore, the coupling properties of the tires will not be fully realized ($\gamma_i<\varphi$), while $P_{biF}=P_{bmi}$. In other words:

$$P_{biF} = \min(P_{bmi}, \Phi_i). \tag{9}$$

If wheel locks cannot be achieved over the entire range of φ, we can conclude that the brake mechanisms on this axis are underloaded, and to increase the braking force, the size of the brake chambers should be increased. If the wheels are locked over the entire range of φ, the brake mechanisms are overloaded and the brake chambers should be reduced.

Next, the actual slowdown of the articulated vehicle is determined:

$$d_F = \frac{\sum P_{biF}}{m_{av}}, \text{ m/s}^2, \tag{10}$$

where m_{av} is the mass of the articulated vehicle.

Then P_{biF} and d_F are substituted into expressions (1), (2.1), (2.2).

For an articulated vehicle, the task is solved only by the **iterative** method [34–35]: in order to make up the equation of moments, it is necessary to determine the direction of the force in the coupling device; to do which, you need to know the actual braking forces and the actual deceleration of the articulated vehicle; for this, it is necessary to determine the grip limits; and to determine those, it is necessary to solve the equation of moments.

At the initial stage, it is assumed that the actual braking forces on all axes are equal to the limiting $P_b=P_{bm}$, respectively, the articulated vehicle slows down with the maximum deceleration, possible according to the conditions of grip of the wheels to the road:

$$d = d_m = g \cdot \varphi, \ m/s^2.$$

The number of iterations is determined based on the required calculation accuracy.

From the equality $P_{bmi}=N_i \cdot \varphi$, we can determine the grip coefficient φ_{lock}, at which wheels locking will begin on this axis. The lower the φ_{lock}, the less the tendency of the wheels to lock, so the range on which there is no slipping increases. However, this task has a simple solution only for a single vehicle. Consider a tractor unit ($P_{ht}=0$):

$$P_{bm2} = \frac{m_{tr}g(\lambda_{tr}\cos\psi - h_{tr}\sin\psi) - m_{tr}g\varphi_{lock2}h_{tr}}{L_{tr}} \cdot \varphi_{lock2};$$

$$P_{bm1} = \frac{m_{tr}g((L_{tr} - \lambda_{tr})\cos\psi + h_{tr}\sin\psi) + m_{tr}g\varphi_{lock1}h_{tr}}{L_{tr}} \cdot \varphi_{lock1}.$$

It turns out a system of quadratic equations:

$$-h_{tr} \cdot \varphi_{lock2}^2 + (\lambda_{tr}\cos\psi - h_{tr}\sin\psi) \cdot \varphi_{lock2} - \frac{P_{bm2}L_{tr}}{m_{tr}g} = 0;$$

$$h_{tr} \cdot \varphi_{lock1}^2 + ((L_{tr} - \lambda_{tr})\cos\psi + h_{tr}\sin\psi) \cdot \varphi_{lock1} - \frac{P_{bm1}L_{tr}}{m_{tr}g} = 0.$$

We are interested in a root with a positive root of discriminant. If the discriminant is equal to zero, this means that the wheels lock will occur on the entire range of φ.

The calculation results are shown in Fig. 3.

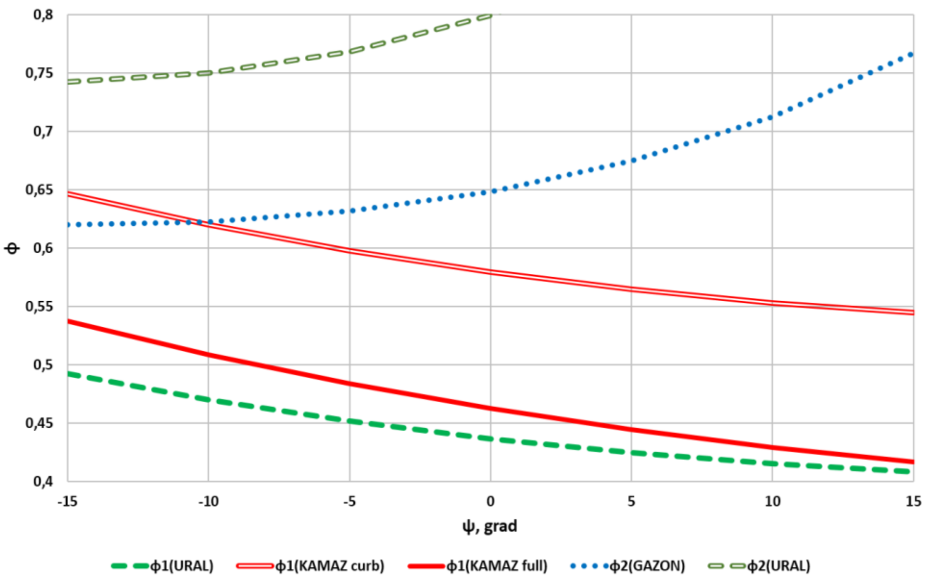

Fig. 3. *Dependence of the grip coefficient by locking on the slope and load*

To solve the task mentioned above for an articulated vehicle, it is necessary to apply an iterative method.

VII. CONCLUSION

The technique proposed by the authors allows to estimate the grip coefficient at which the axles of the vehicle begin to lock, taking into account the longitudinal slope, load and reaction in the coupling device. Analysis of the results shows the following:

1. With a decrease in load, the tendency of the wheels to lock increases.

2. On the downhill, the effect of the redistribution of gravity increases the tendency of the front wheels to lock decreases, and in the rear – vice versa.

3. For tractors with a fifth wheel coupling, the effect of the redistribution of gravity is leveled, since the rear carriage has the weight transmitted from the semi-trailer through the coupling device.

References

1. Pavlov V.V. *Proektirovochnye raschjoty transportnyh sredstv special'nogo naznachenija* [*Design calculations of special vehicles*]. Moscow, MADI, 2014, 116 p.

2. Dygalo V.G., Revin A.A. *Trudy NGTU im. R.E. Alekseeva*, 2011, no. 3, pp. 146-155.

3. Kristal'nyj S.R., Popov N.V., Fomichjov V.A. *Vestnik MADI*, 2012, issue 2, pp. 10a-17.

4. Dygalo V.G., Kotov V.V., Revin A.A. *Izvestija Volgogradskogo gosudarstvennogo tehnicheskogo universiteta. Serija: Nazemnye transportnye sistemy*, 2013, vol. 6, no. 10, pp. 13-16.

5. Gladov G.I., Zajcev S.V., Kotovich S.V. *Konstrukcii transportnyh sredstv special'nogo naznachenija* [*Construction of special vehicles*]. Moscow, MADI, 2014, 164 p.

6. Kristal'nyj S.R., Popov N.V., Fomichev V.A. *Avtotransportnoe predprijatie,* 2014, no. 6, pp. 50-53.

7. Dygalo V.G., Kotov V.V., Dygalo L.V., Revin A.A. *Izvestija Volgogradskogo gosudarstvennogo tehnicheskogo universiteta. Serija: Nazemnye transportnye sistemy*, 2014, vol. 9, no. 19, pp. 16-20.

8. Revin A.A., Dygalo V.G., Judina A.A., Dygalo L.V. *Jenergo- i resursosberezhenie: promyshlennost' i transport*, 2016, no. 4, pp. 7-11.

9. Kristal'nyj S.R., Toporkov M.A., Fomichjov V.A., Popov N.V. *Avtomobil'. Doroga. Infrastruktura*, 2015, No. 2, p. 2.

10. Ivanov A.M., Kristal'nyj S.R., Popov N.V., Fomichjov V.A., Sitnikov F.V. *Avtomobil'. Doroga. Infrastruktura*, 2017, no. 1, p. 5.

11. Ivanov A.M., Kristal'nyj S.R., Popov N.V., Shadrin S.S. *Intellektual'nye sistemy pomoshhi voditelju. Tehnicheskie trebovanija i metody ispytanij* [*Intelligent driver assistance systems. Technical requirements and test methods*]. Moscow, MADI, 2019, 100 p.

12. Gladov G.I., Malinovskij M.P. *Zhurnal avtomobil'nyh inzhenerov*, 2009, no. 2, pp. 30-33.

13. Malinovskij M.P., Gladov G.I. *Avtomobil'naja promyshlennost'*, 2009, no. 9, pp. 21-26.

14. Popov A.I., Kotenko I.V., Fransis O.O. *Vestnik MADI*, 2011, no. 3, pp. 12a-16.

15. Spinov A.R., Popov A.I., Kotenko I.V., Fransis O.O. *Avtotransportnoe predprijatie,* 2011, no. 10, pp. 42-45.

16. Popov A.I., Spinov A.R., Kotenko I.V., Fransis O.O., Naddennaja E.A. *Zhurnal*

avtomobil'nyh inzhenerov, 2011, no. 6, pp. 28-31.

17. Ivanov A.M., Solncev A.N., Spinov A.R., Shadrin S.S. *Zhurnal avtomobil'nyh inzhenerov*, 2013, no. 6, pp. 36-39.

18. Elistratov V.V., Bezrukov S.I., Stenin P.G. *Avtomobil'naja promyshlennost'*, 2014, no. 1, pp. 25-28.

19. Elistratov V.V., Bezrukov S.I., Stenin P.G., Klimakov V.S. *Sovremennye problemy nauki i obrazovanija*, 2014, no. 2, p. 12.

20. Ivanov A.M., Kristal'nyj S.R., Popov N.V., Toporkov M.A., Isakova M.I. *Trudy NGTU im. R.E. Alekseeva*, 2018, no. 2, pp. 146-155.

21. Ivanov A.M., Kristal'nyj S.R., Popov. N.V. *Sistemy avtomaticheskogo jekstrennogo tormozhenija* [*Automatic emergency braking systems*]. Moscow, MADI, 2018, 180 p.

22. Shadrin S.S., Ivanov A.M. *Izvestija Moskovskogo gosudarstvennogo tehnicheskogo universiteta MAMI*, 2014, vol. 1, no. 3, pp. 65-68.

23. Ivanov A.M., Kristal'nyj S.R., Popov N.V., Fomichjov V.A. *Zhurnal avtomobil'nyh inzhenerov*, 2017, no. 6, pp. 14-21.

24. Godzhaev Z.A., Izmajlov A.Ju., Miholap L.A. *Sel'skohozjajstvennye mashiny i tehnologii*, 2017, no. 6, pp. 3-8.

25. Godzhaev Z.A., Izmajlov A.Ju., Miholap L.A. *Fundamental'nye i prikladnye problemy tehniki i tehnologii*, 2017, no. 4-1, pp. 104-110.

26. Malinovskij M.P., Zhuravleva A.Ju. *Avtomobil'. Doroga. Infrastruktura*, 2016, No. 4, p. 10.

27. Malinovskij M.P. *Avtomobil'. Doroga. Infrastruktura*, 2018, No. 4, p. 3.

28. Petrenko A.M. *Metodicheskie ukazanija k laboratornym rabotam po discipline «Teorija special'nyh transportnyh sredstv»* [*Guidelines for laboratory work in the discipline "Theory of special vehicles"*]. Moscow, MADI, 2003, Part 1, 54 p.

29. Gorelov V.A., Padalkin B.V., Chudakov O.I. *Nauka i obrazovanie: nauchnoe izdanie MGTU im. N.Je. Baumana*, 2016, no. 12, pp. 1-17.

30. Gladov G.I., Petrenko A.M. *Special'nye transportnye sredstva: Teorija* [*Special vehicles: Theory*]. Moscow, Akademkniga, 2006, 215 p.

31. Smolko E.S. *Avtomobil'. Doroga. Infrastruktura*, 2019, no. 4.

32. Petrenko A.M. *Ustojchivost' special'nyh transportnyh sredstv* [*Stability of special vehicles*]. Moscow, MADI, 2013, 41 p.

33. Malinovsky M.P. *Sistemy upravleniya kolesnyh mashin* [*Control systems of wheeled machines*]. Moscow, MADI, 2018, 100 p.

34. Malinovskij M.P. *Avtomobil'naja promyshlennost'*, 2011, No. 5, pp. 33-35.

35. Dygalo V.G., Revin A.A. *Izvestija Volgogradskogo gosudarstvennogo tehnicheskogo universiteta. Serija: Nazemnye transportnye sistemy*, 2013, Vol. 7, No. 21, pp. 10-16.