TRANSPORT PROBLEMS 2016 Volume 11 Issue 1
PROBLEMY TRANSPORTU DOI: 10.20858/tp.2016.11.1.3

articulated vehicle; course stability; lane change maneuver; simulation

Rusi RUSEV, Rosen IVANOV*, Gergana STANEVA, Georgi KADIKYANOV
University of Ruse, Department of Engines and Vehicles
8, Studentska str., 7017 Ruse, Bulgaria
**Corresponding author*. E-mail: rgr@uni-ruse.bg

A STUDY OF THE DYNAMIC PARAMETERS INFLUENCE OVER THE BEHAVIOR OF THE TWO-SECTION ARTICULATED VEHICLE DURING THE LANE CHANGE MANOEUVRE

Summary. The course stability and the steerability of the two-section wheeled vehicle at high velocities during the lane change manoeuvre are studied in this paper. Mechanic-mathematical model is developed, on which base simulation programmes are elaborated. Simulation studies of the course stability at different velocities depend on some constructive and exploitation factors are made by these models. Based on the simulation results an idea for an algorithm and a semi-active control system of the resistance in the pulling-supporting device is proposed.

ИССЛЕДОВАНИЕ ВЛИЯНИЯ ДИНАМИЧЕСКИХ ПАРАМЕТРОВ НА ПОВЕДЕНИЕ ДВУХЗВЕННОГО СОЧЛЕНЕННОГО АВТОПОЕЗДА ВО ВРЕМЯ МАНЕВРА ПЕРЕМЕНЫ ПОЛОСЫ ДВИЖЕНИЯ

Аннотация. В этой работе исследованы курсовая устойчивость и управляемость двухзвенного колесного автопоезда во время маневра перемены полосы движения с высокими скоростями. На основании предложенной механико-математической модели, разработаны симуляционные программы. С помощью этой модели проведены симуляции движения, показывающие зависимость курсовой устойчивости и управляемости от некоторых конструктивных и эксплуатационных факторов. В результате симуляций предложены алгоритм и полуактивная система управления сопротивлением седельно-сцепного устройства.

1. INTRODUCTION

The strong development of the automobile transport put with significant sharpness some problems. One of them is the traffic safety. Car accidents bring to society big damages, destroy health and take lives of thousands of people. The question of improving the active and passive safety of vehicles got considerable actuality [1 - 5]. The complicated traffic conditions as a result of high velocities and the large number of participating automobiles and vehicles imposed more strict requirements to their course stability and steerability. The estimation of these characteristics currently is considered inadequate, resulting in limitation of the maximal velocities [3, 5]. The rise of the tractive and dynamic properties of vehicles and the travel intensity impose growing requirements to the course stability and their transverse overturn stability. In many cases of high-velocity motion in small-radius curves in stopping regime the vehicle show tendency towards knife-jacking.

For improving the traffic safety, definite requirements are demanded to the processes of steerability and course stability:

1. Steerability and course stability are characteristics of the system vehicle-driver. Many constructive parameters of the vehicle, occasional environmental factors and also the qualification of the driver exert influence on them. This dependence leads to complicating the studies in this area and that is why these characteristics are referred to as characteristics of the machine itself.
2. During the last 20-30 years the course stability and traffic safety have considerably improved due to introduction of technical means such as:

- automated projecting of units and their aggregates;
- improvement of tyre characteristics;
- introduction of antilock braking systems, stability systems, different dampers of transverse vibration at the connection point of the truck to the trailers;
- computerized control of the steering processes of the automobile systems, etc.

Despite of these improvements, the permitted exploitation velocities in most European countries do not exceed 70km/h. Many papers are devoted to investigations of vehicle behavior during transient motion and determined the influence of vehicle design factors and driver inputs to the steerability and stability of the vehicles on various roads [2, 3, 5, 7, 10 - 14].

Different devices are proposed for overcoming this phenomenon [6 - 12]. These devices exert as well positive influence on the travel stability of the vehicle during different manoeuvres, as the one of line change.

From the analysis and the conclusions done it is obvious that despite the introduction of modern devices and systems for improvement of the steerability and the course stability during vehicle motion, their safety is not significantly bettered. As a result of this the permitted velocities for these vehicles are not enhanced above 70-80 km/h in last years.

The purpose of this study is dedicated to improvement of the steerability and the course stability of a two-section vehicle via adaptive management of the characteristics of the link between its sections.

2. DEVELOPMENT OF A MECHANIC-MATHEMATICAL MODEL OF AN ARTICULATED VEHICLE

The mechanic-mathematical model of a vehicle, which point of articulation lies in the longitudinal basis of the truck (fig. 1), is plane and has four degrees of freedom, because of the applied links in the articulation of the two sections of the vehicle.

In the horizontal plane the model of the articulated vehicle includes masses m_b, m_p and the mass inertia moments J_b, J_p related to the vertical central axes of the two sections. The side slip coefficients of the front, middle and rear axle are $k_{\delta_1}, k_{\delta_2}, k_{\delta_3}$ respectively, and the resistant moment between the two sections during their relative angle displacement is M_{ξ}.

As coordinates, determining the movement of the systems, are choose the longitudinal and the lateral velocities of the center of the truck masses V_x, V_y and the angle velocities of the front and rear section ω_1, ω_2.

The relationship between the projections of the reactions and velocities in the point of articulation in the chosen moving coordination systems $x\,C_b\,y$ and $\tau\,C_p\,\eta$ for two sections of the vehicle (fig. 1) are:

$$T_x = -T_\tau.\cos\varphi - N_\eta.\sin\varphi \tag{1}$$

$$N_y = +T_\tau.\sin\varphi - N_\eta.\cos\varphi \tag{2}$$

where: $\varphi = \varphi_1 - \varphi_2 = \int \omega_1 dt - \int \omega_2 dt$

$$V_\tau = V_x.\cos\varphi - V_{by}\sin\varphi \quad (3)$$

$$V_{b\eta} = V_x.\sin\varphi + V_{by}\cos\varphi \quad (4)$$

where:

$$V_{by} = V_{oy} + \omega_1.l_{bt} \quad (5)$$

As l_{bt} is taken with its sign ("+" or "-") in the moving coordination system $X\,C_b\,Y$ (on fig.1. $l_{bt} < 0$). If point B of articulation is forward the mass center of the truck, $l_{bt} > 0$.

The motion of the two sections is described by the following system of differential equations:

$$\begin{aligned}
&-m_b.a_x + \sum F_x = 0\\
&-m_b.a_y + \sum F_y = 0\\
&-J_b.\dot{\omega}_1 + \sum M_b = 0\\
&-m_p.a_\tau + \sum F_\tau = 0\\
&-m_p.a_\eta + \sum F_\eta = 0
\end{aligned} \quad (6)$$

The accelerations of the mass centers of the truck and the trailer on the axes of their corresponding moving coordination systems are:

$$\text{v } a_x = \dot{V}_x - \omega_1 V_{oy};\, a_y = \dot{V}_{oy} + \omega_1 V_x$$

$$a_\tau = \dot{V}_\tau - \omega_2 V_\eta;\, a_\eta = \dot{V}_\eta + \omega_2 V_\tau, \quad (7)$$

The lateral reactions of the road on the tyres are proportional to the corresponding coefficients k_δ and side slip angles δ.

$$F_i = -k_\delta.\delta_i, \qquad \delta_i = arctg\left(\frac{V_{yi}}{V_x}\right) - \theta_i, \quad (8)$$

After rearranging and substituting of the forces with their corresponding expressions in case of $\theta_2 = 0$ and $\theta_3 = 0$, for the system (6) we will have:

$$\begin{aligned}
&m_b\left(\dot{V}_x - \omega_1 V_{oy}\right) = \left(F_{bk1} - F_{bf1}\right)\cos(\theta + \delta_1) + \left(F_{bk2} - F_{bf2}\right)\cos\delta_2 +\\
&+ F_{by1}\sin(\theta + \delta_1) + F_{by2}\sin\delta_2 + F_{ba}\cos\alpha_{ba} +\\
&+\cos\varphi\begin{bmatrix} m_p\left(\dot{V}_x\cos\varphi - V_x\omega_1\sin\varphi - \dot{V}_{oy}\sin\varphi - \dot{\omega}_1 l_{bt}\sin\varphi - \omega_1 V_{oy}\cos\varphi - \omega_1^2 l_{bt}\cos\varphi + \omega_2^2 l_{pt}\right) - \\ - F_{pk}\cos\delta_3 + F_{pf}\cos\delta_3 - F_{py}\sin\delta_3 - F_{pa}\cos\alpha_{pa}\end{bmatrix} +\\
&+\sin\varphi\begin{bmatrix} m_p\left(\dot{V}_x\sin\varphi + \omega_1 V_x\cos\varphi + \dot{V}_{oy}\cos\varphi + \dot{\omega}_1 l_{bt}\cos\varphi - \omega_1 V_{oy}\sin\varphi - \omega_1^2 l_{bt}\sin\varphi - \dot{\omega}_2 l_{pt}\right) - \\ - F_{pk}\sin\delta_3 + F_{pf}\sin\delta_3 + F_{py}\cos\delta_3 - F_{pa}\sin\alpha_{pa}\end{bmatrix}
\end{aligned} \quad (9)$$

$$\begin{aligned}
&m_b\left(\dot{V}_{oy} + \omega_1 V_x\right) = \left(F_{bk1} - F_{bf1}\right)\sin(\theta + \delta_1) + \left(F_{bk2} - F_{bf2}\right)\sin\delta_2 -\\
&- F_{by1}\cos(\theta + \delta_1) - F_{by2}\cos\delta_2 + F_{ba}\sin\alpha_{ba} -\\
&-\sin\varphi\begin{bmatrix} m_p\left(\dot{V}_x\cos\varphi - V_x\omega_1\sin\varphi - \dot{V}_{oy}\sin\varphi - \dot{\omega}_1 l_{bt}\sin\varphi - \omega_1 V_{oy}\cos\varphi - \omega_1^2 l_{bt}\cos\varphi + \omega_2^2 l_{pt}\right) - \\ - F_{pk}\cos\delta_3 + F_{pf}\cos\delta_3 - F_{py}\sin\delta_3 - F_{pa}\cos\alpha_{pa}\end{bmatrix} +\\
&+\cos\varphi\begin{bmatrix} m_p\left(\dot{V}_x\sin\varphi + \omega_1 V_x\cos\varphi + \dot{V}_{oy}\cos\varphi + \dot{\omega}_1 l_{bt}\cos\varphi - \omega_1 V_{oy}\sin\varphi - \omega_1^2 l_{bt}\sin\varphi - \dot{\omega}_2 l_{pt}\right) - \\ - F_{pk}\sin\delta_3 + F_{pf}\sin\delta_3 + F_{py}\cos\delta_3 - F_{pa}\sin\alpha_{pa}\end{bmatrix}
\end{aligned} \quad (10)$$

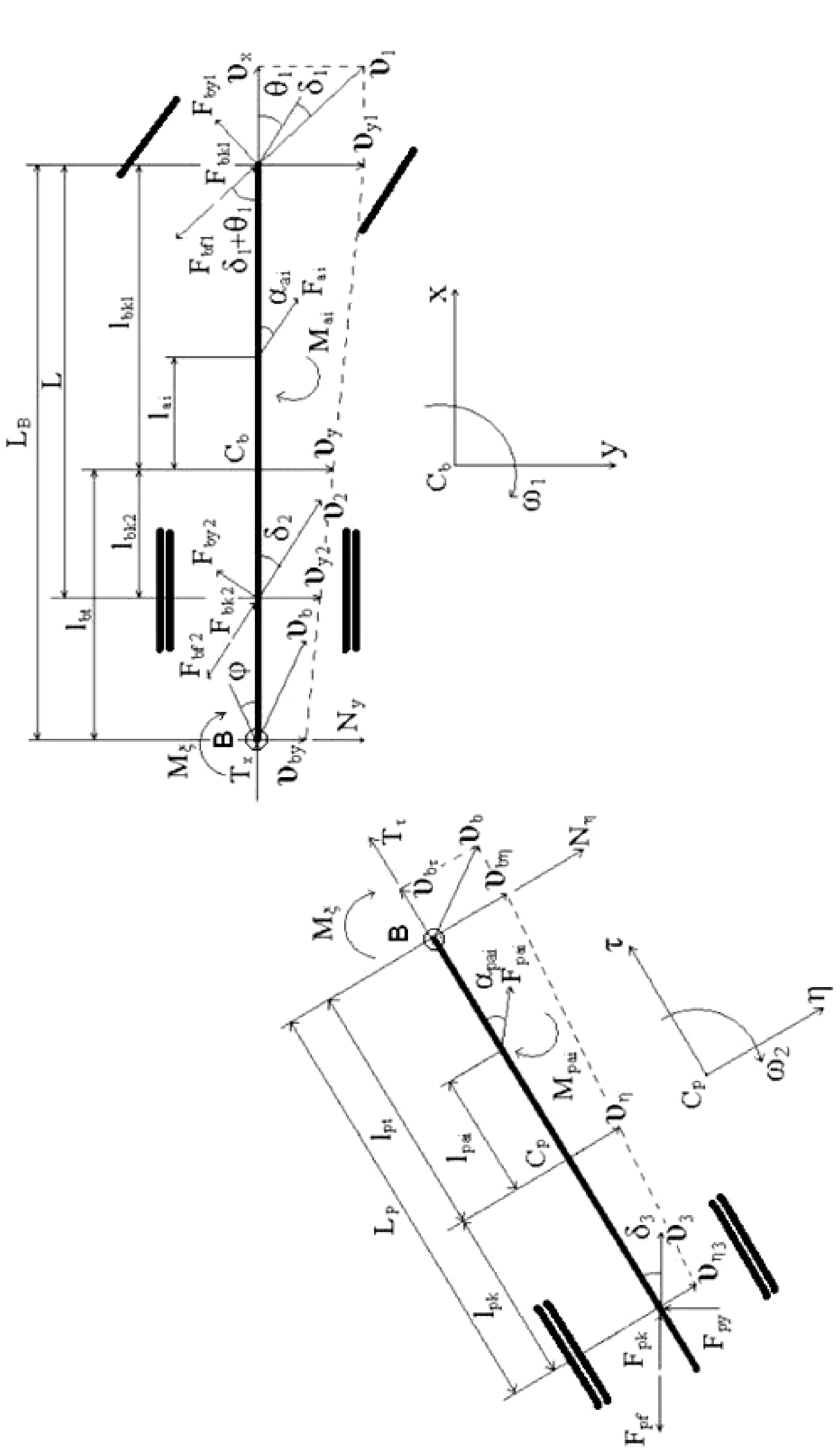

Fig. 1. Mechanic-mathematical model of an articulated vehicle
Рис. 1. Механико-математическая модель сочлененного автопоезда

$$\begin{aligned}
J_b\dot{\omega}_1 &= l_{bk1}\left(F_{bk1}-F_{bf1}\right)\sin(\theta+\delta_1)+l_{kb2}\left(-F_{bk2}+F_{bf2}\right)\sin\delta_2 - \\
&- l_{kb1}F_{by1}\cos(\theta+\delta_1)+l_{kb2}F_{by2}\cos\delta_2 + \\
&+ l_{ba}F_{ba}\sin\alpha_{ba}+M_{ba}+M_{rezist} + \\
&+ l_{bt}\begin{bmatrix} \sin\varphi\begin{bmatrix} m_p\left(\dot{V}_x\cos\varphi - V_x\omega_1\sin\varphi-\dot{V}_{oy}\sin\varphi-\dot{\omega}_1 l_{bt}\sin\varphi-\omega_1 V_{oy}\cos\varphi-\omega_1^2 l_{bt}\cos\varphi+\omega_2^2 l_{pt}\right)- \\ -F_{pk}\cos\delta_3+F_{pf}\cos\delta_3-F_{py}\sin\delta_3-F_{pa}\cos\alpha_{pa}\end{bmatrix} - \\ -\cos\varphi\begin{bmatrix} m_p\left(\dot{V}_x\sin\varphi+\omega_1 V_x\cos\varphi+\dot{V}_{oy}\cos\varphi+\dot{\omega}_1 l_{bt}\cos\varphi-\omega_1 V_{oy}\sin\varphi-\omega_1^2 l_{bt}\sin\varphi-\dot{\omega}_2 l_{pt}\right)- \\ -F_{pk}\sin\delta_3+F_{pf}\sin\delta_3+F_{py}\cos\delta_3-F_{pa}\sin\alpha_{pa}\end{bmatrix}\end{bmatrix}
\end{aligned} \tag{11}$$

$$\begin{aligned}
J_p\dot{\omega}_2 &= l_{pk}\left[\left(-F_{pk}+F_{pf}\right)\sin\delta_3+F_{py}\cos\delta_3\right]+F_{pa}\sin\alpha_{pa}+M_{pa}-M_{rezist} + \\
&+ l_{pt}\begin{bmatrix} m_p\left(\dot{V}_x\sin\varphi+\omega_1 V_x\cos\varphi+\dot{V}_{oy}\cos\varphi+\dot{\omega}_1 l_{bt}\cos\varphi-\omega_1 V_{oy}\sin\varphi-\omega_1^2 l_{bt}\sin\varphi-\dot{\omega}_2 l_{pt}\right)- \\ -F_{pk}\sin\delta_3+F_{pf}\sin\delta_3+F_{py}\cos\delta_3-F_{pa}\sin\alpha_{pa}\end{bmatrix}
\end{aligned} \tag{12}$$

$$\dot{\varphi} = \omega_2 - \omega_1 \tag{13}$$

One of the criteria for the course stability of the vehicle is the grip force of the composition by axles. As a value of this criterion 80% of the corresponding maximal value of the grip force of axles or exceeding the angle of side slip $\delta_i > 8^o$ is accepted [13]. After exceeding these values the vehicle motion is considered unstable, due to the elastic properties of the tyres this state is very close to the entire slipping of the tyres on the road. Only 80% of the maximal value of the grip forces are taken to guarantee work of the model just in the elastic part of the relationship between lateral reaction and the slip angle. A limiting criterion is the altitude of the lateral accelerations of the vehicle sections as well.

For solvation of the differential equations a SIMULINK program in MATLAB is developed, which does the calculations. It gives opportunity to examine the vehicle motion during making a turn as the road conditions, vehicle equipment and the law of changing the steering angle of the steerable wheels are varied. There is a possibility for visualizing the steering angle of the wheels, the side slip angles of the axles, the lateral reactions on the axles, the lateral accelerations of the vehicle sections as well as visualizing the position of the sections on the road plane. The model is developed in the presence of resistance moment M_ξ in the point of articulation.

The change of lane manoeuvre of the truck is chosen for studying the stability and steerability. The adequacy of the model, was verified via previous investigations [11, 12]. By natural experiments the changing of the angle of deviation of the steerable wheels is established as a sine form.

3. MODELING THE MOTION OF AN ARTICULATED VEHICLE WITH ONE SEMI-TRAILER AND TESTING ITS COURSE STABILITY

The influence of the travel velocity, the lateral tyre hardness, the grade/degree of loading and the grip coefficient on the travel stability of an articulated vehicle with one semi-trailer is examined. The study is done with the mechanic-mathematical model, shown on fig. 1. As a prototype a Russian vehicle has been used.

A linear regression analysis for establishing the degree of influence of the control factors on the trajectory, the angles of slipping, the lateral forces and the lateral accelerations of the centers of gravity of the two vehicle section is done. The results from it are shown in table 1.

A conclusion can be made on the basis of the tests and the regression analysis that for further investigations it is important to study the vehicle unstable motion above 70 km/h and the influence of its velocity along asphalt-covered roads on the travel stability. The carried out simulations show that traveling at higher velocities along curved trajectories with rapid change of direction is unstable and often the vehicle loses its stability. On its turn this worsens the travel safety at such speeds.

In order to establish the critical cases of motion during the line change manoeuvre series of simulations are done at velocity 70-95 km/h. The semitrailer load varies from empty to full-loaded trailer. The tyre hardness changes in the range 80000-120000 N/rad. The road conditions are changed only by the grip coefficient from $\psi = 0{,}65$ (wet asphalt) to $\psi = 0{,}85$ (dry asphalt). The model from the previous experiment is used. The time for manoeuvre accomplishment is taken as a condition. This is inevitable because of the impossibility of the driver to reduce the manoeuvre time with raising the speed so as to keep the manoeuvre distance constant at different velocities, especially greater than 70 km/h.

Table 1

Values of the regression equations coefficients

Equation for	Variable coefficients for				
	V	k_y	ψ	m	free term
Y1	-0,0452	-0,073	0,002237	0,02664	3,554487
Y2	0,02086	-0,014	-0,00249	0,03692	3,566749
Y3	0,0303	0,01706	-0,0262	0,00279	3,519358
F1	7528,979	688,6911	-732,515	560,2792	14768,68
F2	12989,48	4563,515	-6255,67	6496,019	22717,72
F3	12991,48	6936,027	-9170,53	8156,427	21241,10
a_y	1,192088	0,05359	-0,0832	0,05278	2,252825
a_η	1,125407	0,03088	-0,0697	0,03207	1,905868

For manoeuvre time that at 70 km/h is accepted. Criterion for evaluating the motion is the lost of stability and value of the side slipping of the axles' middles during and after the manoeuvre. The criteria for stability remain the same – limitation on grip, angle of slipping and lateral accelerations. For evaluation of the trajectories the term that the trajectories are in the corridor of the adjacent line after manoeuvre accomplishment with size $3{,}5 \pm \Delta$, where $\Delta = 0{,}25m$, is used. The simulations are done at complete combining of all control factors – velocity, loading, slipping coefficient of the tyres and grip coefficient of the tyres with the road. Fig. 1 shows the trajectories of three of the cases.

As it is seen with growing velocity the axle trajectory of the trailer starts to enlarge its swing. The analysis of the simulation results shows that at zero and medium loading even with enhanced swing of the axle trajectory of the trailer the vehicle does not lose stability and does not leave the pre-determined corridor. This behavior is preserved despite the values of the grip coefficient and the side slip coefficient. The latter influences just the amplitudes and the process of attenuation.

At greater values of the side slip coefficient of the tyres amplitude magnitude is reduced and oscillation process is attenuated faster, which is good for the travel safety.

With a fully loaded trailer the behavior changes. These simulations show that in any case the vehicle violates some of the limit terms. At equal other conditions the tyre hardness exert significant influence on the slip angles and the trajectory. In the cases with enough cohesion and small tyre hardness considerable violation of the corridor is seen. At low loading and with enhancing the tyre hardness the amplitudes are diminished and the oscillations are resolved. At speeds above 80 km/h these conditions are not enough for the vehicle to enter the corridor or to reduce the axle slip.

Two types of resistance of the articulation are examined in this paper: viscous friction; and mechanic dry friction with consideration over the elasticity of the link of the frictional. The results on fig. 3 shows the effect of two devices that increases the resistance in the point of articulation of the two sections on the stability of the vehicle in critical situations.

A device with purely viscose resistance and a device with dry friction are used. The simulations are repeated when the border terms were violated. They are done in two groups – one with the first device and another with the second device and the results are compared.

As a common conclusion from the analysis a good effect from the application of both devices is noted. Improvement of travel stability is realized in all simulations independently from the conditions. Fig. 3 shows the trajectories with both types of devices at 92 km/h.

The vehicle trajectories in cases of hydraulic or dry friction resistance have different character. The hydraulic resistance offers quick oscillation silencing. Increase in the resistance moment does not affect the pre-determined trajectory. At constant time for ruling the steering wheel by the driver, with increase of the resistance moment, the time and respectively the distance for manoeuvre completion are prolonged. However, if maximal approximation to the trajectory to the middle of the corridor 3,5 m is not seek, the prolongation of the manoeuvre time is not significant.

With mechanic dry friction considerably smaller in absolute value resistant moments are needed for reduction of trajectory amplitudes than with the previous device. It turns out that applying the friction moment changes the predetermined trajectory. This imposes the correction of the steering angle of the wheels so that the vehicle could travel in the predetermined corridor. Another effect is the slow silencing of the trajectory oscillations. Compared to the hydraulic resistance the silencing time here is much longer.

It is noticed that applying a dry friction moment provides the same manoeuvre time with the moments increase. Its increase leads to a more serious correction of the angle of steerable wheels deviation. This deviation angle enhancement is analogous to the manoeuvre time enhancement in the previous case with the hydraulic moment.

4. FUNCTION ALGORITHM OF THE SYSTEM FOR ACTIVE REGULATION OF THE RESISTANCE IN THE ARTICULATION DEVICE BETWEEN THE TRUCK AND THE TRAILER

The travel stability during the change of line manoeuvre depends on many factors. Some of them exerting strong influence can be used as entrance for a system for ruling the resistance device between the vehicle sections. Firstly, the machines velocity can be received from the system for management and control of the brakes. This system as well as the system for management of the engine and the gearbox can give information about the presence of a braking or driving force. Setting the machine load is registered by the system for sustaining a constant trailer level. The road cohesion can be set by the driver via a button on the control panel or can be taken from the memory of the block that controls the brakes (ABS). The tyre hardness can also be set by the driver or "the intelligent tyres" can be used, that continuously deliver data about the pressure in them, their load and hence their hardness, being connected to the board computer. A switching on point for the system is the increase in the speed of the steering wheel turning over a definite critical for the situation point. Then according to the turning speed optimal resistance in the articulation device between the two-vehicle sections is realized. There is also a following device that follows the angle and the speed of turning of the two sections one in relation to the other. Fig. 4 shows a block scheme of the system for active control of the resistant moment. When the steering wheel turning is terminated by the driver for straight motion of the vehicle, the system for stable articulated vehicle motion switches off.

It should be added that the results obtained can be used to simulate the deformation of a separate bus, for example, with the use of techniques [16].

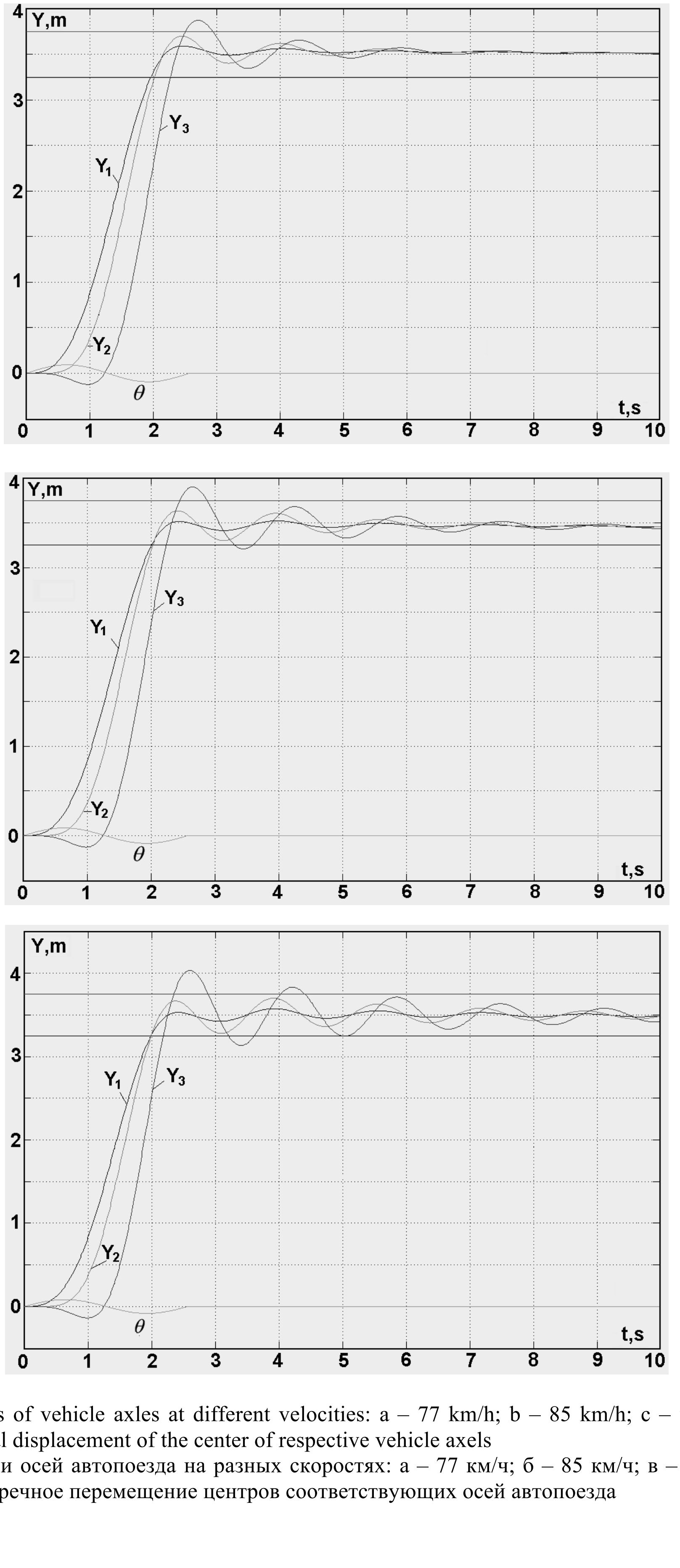

Fig. 2. Trajectories of vehicle axles at different velocities: a – 77 km/h; b – 85 km/h; c – 92 km/h; Y1, Y2, Y3 – lateral displacement of the center of respective vehicle axels

Рис. 2. Траектории осей автопоезда на разных скоростях: а – 77 км/ч; б – 85 км/ч; в – 92 км/ч; Y1, Y2, Y3 – поперечное перемещение центров соответствующих осей автопоезда

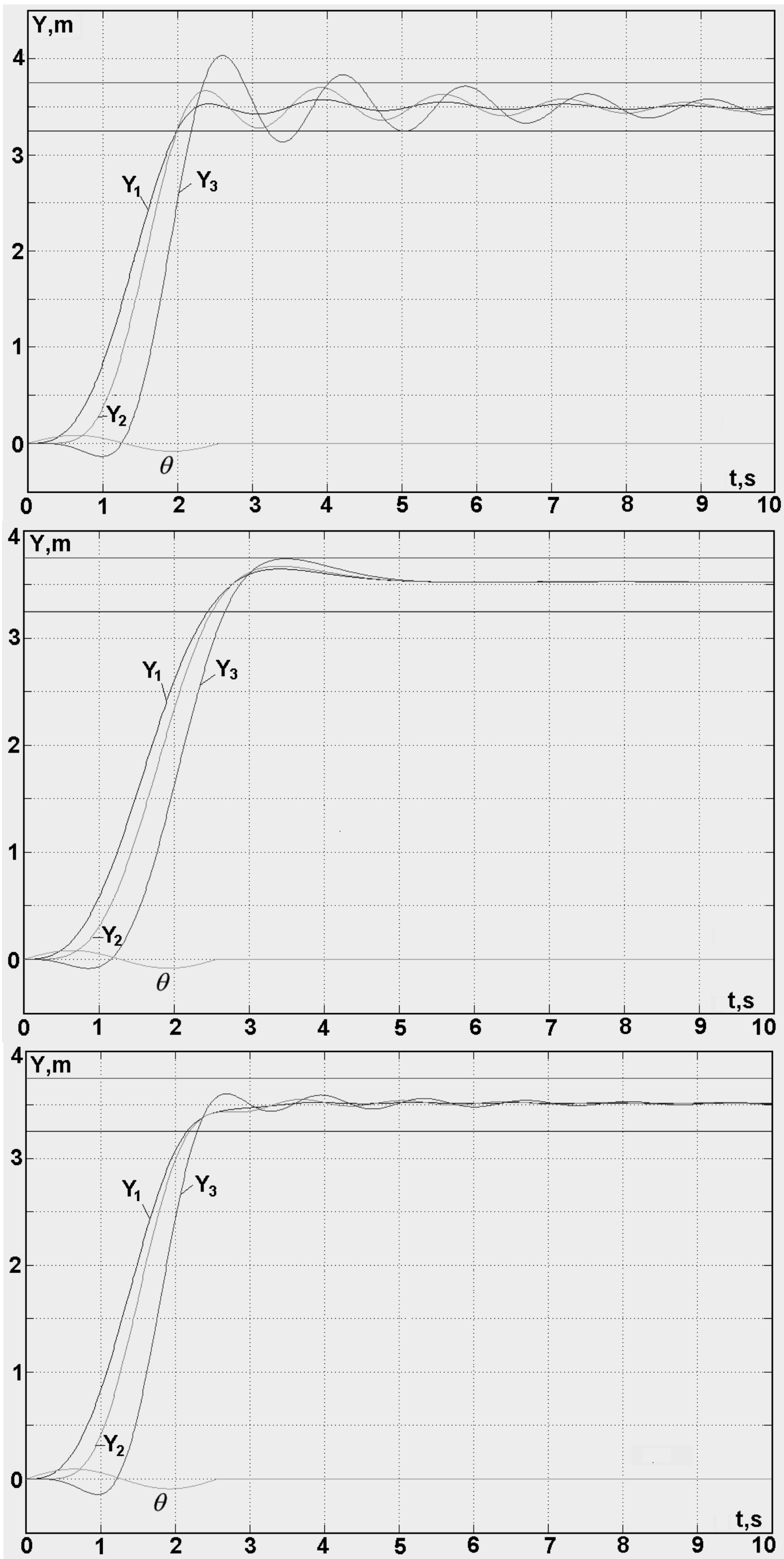

Fig. 3. Trajectories of the vehicle axles: a – no additional resistance; b – with viscous resistance; c – with dry friction resistance; Y_1,Y_2, Y_3 – lateral displacement of the center of respective vehicle axels

Рис. 3. Траектории осей автопоезда: а – без дополнительного сопротивления; б – с вискозным сопротивлением; в – со сопротивлением типа сухого трения; Y1,Y2, Y3 – поперечное перемещение центров соответствующих осей автопоезда

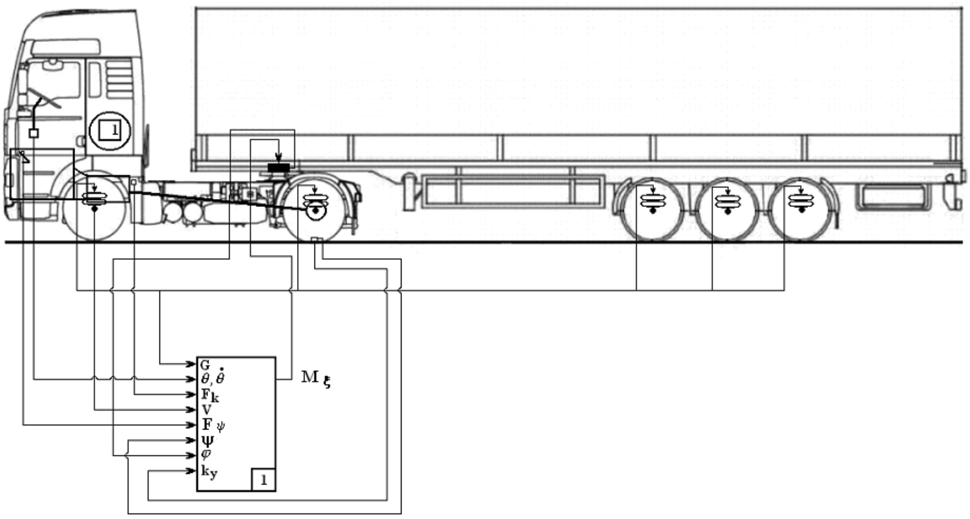

Fig. 4. A block scheme of a system for active control of the resistant moment
Рис. 4. Блок-схема системы активного управления моментом сопротивления

NOTATION

a_x, a_y - longitudinal and lateral accelerations of the center of the truck masses;

a_τ, a_η - longitudinal and lateral accelerations of the center of the trailer masses;

F_1, F_2, F_3 - lateral forces of the axles respectively;

F_{ki} – traction force;

F_ψ - grip force;

F_f - rolling resistance force;

F_{ba}, F_{pa} - external active forces acting on the truck and trailer respectively;

G - vehicle weight;

J_b - mass inertia moment of the truck, related to the vertical central axle;

J_p - mass inertia moment of the trailer, related to the vertical central axle;

$k_{\delta_1}, k_{\delta_2}, k_{\delta_3}$ - the side slip coefficients of the front, the middle and the rear axle respectively;

l_{pa} - distance along the $c_p\tau$ axis to the point of action of the external force F_{pa};

l_{ba} - distance along the $c_b x$ axis to the point of action of the force F_{ba};

l_{bt} - distance along the $c_b x$ axis to the point B of the coupling;

m_b - mass of the truck;

m_p - mass of the trailer;

M_{ba}, M_{pa} - external moments acting on the truck and the trailer, respectively;

M_ξ - resistant moment at the articulation point between the truck and the trailer during their relative angle displacement;

V_x, V_y - longitudinal and lateral velocities of the center of the truck masses;

V_τ, V_η - longitudinal and lateral velocities of the center of the trailer masses;

Y_1, Y_2, Y_3 - lateral coordinates of the respective axle trajectory;

α_{ba} - the angle between the longitudinal axis $c_b x$ and the vector of the external force F_{ba};

α_{pa} - the angle between the longitudinal axis $c_p\tau$ and the vector of the external force F_{pa};
$\delta_1, \delta_2, \delta_3$ - side slip angles of the front, middle and rear axle respectively;
ω_1, ω_2 - angular velocities of the front and rear section respectively;
θ_i - steering angles of the wheels;
ψ - grip coefficient.

Suffixes:
i - concerns the axle number;
b - concerns the truck;
p - concerns the trailer.

5. CONCLUSIONS

1. The results from the examination have allowed to establish the main exploitational and constructive factors, upon which the course stability of the two-section vehicles depends directly.
2. At equal other conditions by enhancing the lateral tyre slip coefficient K_y from 80 to 120 kN/rad the oscillation amplitude of the trajectories of the vehicle axle middles considerably drop and the silencing time is reduced by 30-40%.
3. It is not necessary for the driver to correct the angle of wheel deviation in a vehicle with a device with an hydraulic resistance moment. The damping properties of the system improve with adding this device, but the manoeuvre time is increased up to 3 times as the hydraulic resistance coefficient rises to $\zeta = 1200 kNms / rad$.
4. The damping properties of the dry friction device are much worse than those of the hydraulic device (the silencing process is about 60% longer). The angle of steerable wheel deviation must be corrected up to 10% to enable the truck to move along the same trajectory as in the case with no such moment. This imposes an adaptive period for the driver to the system. The manoeuvre time keeping is considered a positive characteristic of the device.
5. The development of a vehicle with an adaptive management of the resistance in the point of articulation gives opportunity for improvement of the course stability of the vehicle. The heightened velocities due to the adaptive device, will allow to enhance the productivity of the automobile transport. The positive effect from the adaptive device is improving the course stability. Its using leads also to enhancing the safety of motion of these articulated vehicles.

References

1. Scheibe, R. & Shields, L. Computer Modeling of Factors Significant to Electronic Stability Control Effectiveness. *SAE Technical Paper* 2009-01-0455. doi:10.4271/2009-01-0455.
2. Русев, Р.Г. & Стоянов, С.Г. & Иванов, Р.П. & Николов, В.С. Изследване влиянието на някои експлоатационни фактори върху курсовата устойчивост на движение на съчленен автовлак с полуремарке. Сп. *Машиностроителна техника и технологии*. 2004. Т. 2. Р.62-65 [In Bulgarian: Rusev, R.G.& Stoyanov, S.G. & Ivanov, R.P. & Nikolov, V.S. Research of the influence of some running factors on the course stability of the articulated vehicle with semitrailer. *Journal of Mechanical engineering and technologies*].
3. Nordstrom, O. Heavy duty vehicle dynamics related to braking, steering and tyres – Swedish research and proposals by VTI. *SAE Technical paper Series – Vehicle Dynamics related to Braking and Steering.* Paper 892502. Warrendale, USA. P.43-57.
4. Nikolov, V.S. at all. Моделиране на движението на автомобил при смяна на лентата за движение. *Сборник от МНК MOTAUTO'03. М.2 „Автомобилна техника и транспорт“.*

София. P. 53-56. ISBN 954-9322-02-5. [In Bulgarian: Modelling the vehicle movement when changing the traffic lane. *MOTAUTO'03. Proceedings.* Sofia. Vol. II. P.53-56].

5. Simiński, P. Safety and analysis of modern transport. *Journal of KONES Powertrain and Transport*, Vol. 18, No. 1. 2011.
6. US Pat. 5183283 *Apparatus for limiting lateral movement in trailer*. MKI 5 B62D13/02/ Jarlsson Assai. No 429931. Filed 01.11.89. 02.02.93. NKI 280/426.
7. Chen, L.K. & Hsu, J.Y. Investigation of jack-knife prevention in an articulated scaled vehicle. *International Journal of Vehicle Mechanics and Mobility*. 2008. Vol. 46. Supplement 1. P. 765-777.
8. Авт. Свидетелство на НРБ, No 24853. Попов, П.К., Й.Н. & Димитров, К.П. & Косев, К.Р. *Устройство, възпрепятстващо относителното завъртане между звената на седловия автовлак*. София, ИИР. 1978. [In Bulgarian: Patent No 24853. Popov, P.K. & Dimitrov, J.N. & Kosev, K.P. *Device, preventing relative turning of the articulated vehicle sections*. Sofia, IIR. 1978].
9. EP 2672547 France. *Dispositive du liason pour optimizer la traectorie d'une remorqué attelee derriere un tracteur.* MKI 5 B60D1/42, B62D53/00 onet. No 9101788.
10. Zhou, S.W. Jackknife Control on Tractor Semi-trailer during High Speed Curve Driving. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*. January 1, 2011. Vol. 225. P. 28-42.
11. Zhou, S.W. & Zhang, S.Q. Jackknife Control on Tractor Semi-trailer during Emergency Braking. *Advanced Materials Research.* 2011. Vol. 299-300. P. 1303-1306.
12. Azad, N.L. & Khajepour, A. & McPhee, J. Effects of locking differentials on the snaking behaviour of articulated steer vehicles. *International Journal of Vehicle Systems Modelling and Testing*. 2007. Vol.2. P.101-127.
13. Jun Ishio & Hiroki Ichikawa & Yoshio Kano & Masato Abe. Vehicle-handling quality evaluation through model-based driver steering behaviour. *International Journal of Vehicle Mechanics and Mobility.* 2008. Volume 46. Supplement 1. P. 549-560.
14. Nikolov, V.S. at all. Изследване на закона на отклонение на управляемите колела при извършване на маневрата „смяна на лентата за движение“. *Сборник от МНК MOTAUTO'03. том 2. „Автомобилна техника и транспорт“.* P. 50-52. ISBN 954-9322-02-515. [In Bulgarian: Investigation of the regularity of the steerable wheels deviation during the manoeuvre "change the traffic lane*". MOTAUTO'03. Proceedings.* Volume II. Sofia*.* P. 50-52].
15. Hidehisa Yoshida & Shuntaro Shinohara & Masao Nagai. Lane change steering manoeuvre using model predictive control theory. *International Journal of Vehicle Mechanics and Mobility.* 2008. Volume 46. Supplement 1. P. 669-681.
16. Sapragonas, J. & Dargužis, A. & Daya, E.M. & Daouadji, A. & Merzoug, B. Model of radial deformations of protector of vehicle tire. *Mechanika*. 2011. Vol. 17. No. 1. P. 21-29.

Received 02.09.2014; accepted in revised form 11.02.2016