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SIMULATION INVESTIGATION OF THE DYNAMICS OF THE PROCESS OF SUDDEN OBSTACLE AVOIDING BY A MOTOR VEHICLE

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Summary

Extensive analytical work was undertaken on using the active steering system EPS (Electric Power System) for automatic driving of a motor vehicle (a two-axle motor truck of medium load capacity, provided with typical ESC - Electronic Stability Control system components as well as obstacle detectors and road monitoring systems) in the situation when an obstacle has suddenly sprung up. This paper presents a fragment of this work, which includes results of simulation investigation of dynamics of the lane-change process in an "open" and "closed-loop" system, i.e. with the automatic control function of the active steering system (EPS) being used in the latter case. The theoretical deliberations and simulation tests were based on mathematical models of the controlled system and the controller. The model of the controlled system includes a complex and detailed representation of the dynamics of motion of a motor truck, where the non-linearities and three-dimensionality of the vehicle motion are taken into account. The controller model is based on a reference model, which is considerably simplified and, in consequence, very effective in the carrying out of the necessary real-time calculations. The automatic controller operates like a Kalman regulator in a closed-loop system.

Keywords: active safety of a motor vehicle, model and simulation investigations, electrically controlled active steering systems, avoiding of an obstacle that has suddenly sprung up

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1. Introduction

At many world's research centres, works are carried out on systems that are to automatize road manoeuvres of motor vehicles at high driving speeds, which include e.g. obstacle avoidance and overtaking [14, 15]. The manoeuvres of this kind are difficult for being automatized, because the automatization requires in this case the controlling of an object whose dynamics is unstable and susceptible to changes in many parameters and whose trajectory is subject to strict limitations. Usually, a manoeuvre of this kind constitutes a sequence of elementary lane-change manoeuvres [2, 13]. Therefore, the automatization of the lane-change manoeuvre is perceived as fundamental for the automatization of the motor vehicle driving process.

The issue of controlling the steering angle during a lane-change manoeuvre is addressed by numerous scientific works. Usually, they are based on a vehicle steering concept that covers planning the vehicle path and following the planned trajectory in a tracking process where appropriate sensors and regulating devices are employed. The vehicle path planning is sometimes treated in such cases as a problem of parametric optimization for the assumed form of the function of shape of the desired vehicle path (sinusoid segment, composition of arcs, algebraic spline curve, etc.). At the vehicle path optimization of this kind, not only minimum distance to the obstacle, short manoeuvre duration time, smoothness of the path, or reduction of the feeling of jerks should be sought, but also the possibilities of good realization of the manoeuvre in a tracking process should be taken into account (the shape of the vehicle path affects the regulation errors). The regulation systems proposed in result of such works are based on the structures and algorithms that are known from theory.

Within Project No. N N509 568439 [4, 5, 6, 7, 10, 12], extensive work was done on the issues of automatic driving of a motor vehicle in situations of accident hazard caused by an obstacle having suddenly sprung up. In this paper, the concepts of the models and control systems developed have been described, with focusing on the interesting results of simulation tests. A complex motor vehicle model, which will be discussed in Section 2, represents a two-axle motor truck of medium load capacity. Such a model was used in the Project as a virtual controlled system. A simplified reference model, discussed in Section 3, was taken as a basis for the algorithm of automatic controlling of the lane-change manoeuvre performed within the obstacle avoidance process. This innovative simple algorithm will be discussed in Section 4. The simulation tests presented in Sections 5 and 6 have the nature of sensitivity tests, as they cover different variants of road surface, driving speed, and current vehicle load. The investigation results show the complexity of the dynamic properties of the system under test and seem to confirm the reasonability of the automatic control solutions adopted.

2. Vehicle model

The typical medium-capacity two-axle motor truck under consideration has a superstructure integral with the vehicle frame, to which the front and rear axle beams are fastened. Each of the axles is guided by two longitudinal leaf springs, which simultaneously perform the function of spring-damping elements in individual wheel suspension systems.

The physical model of the vehicle (Fig. 1) is a three-dimensional discrete dynamic system, in which all the major degrees of freedom of the real vehicle are taken into account. It consists of seven rigid bodies having mass (vehicle body, front axle, rear axle, and four road wheels). These bodies are linked with each other by flexible elements having non-linear spring and damping characteristics.

The model has 20 degrees of freedom, where the vehicle body solid has six of them, the front and rear axle beams have four degrees of freedom each, and the front and rear wheels have two and one degrees of freedom each, respectively (the rear wheels being assumed as dual wheels).

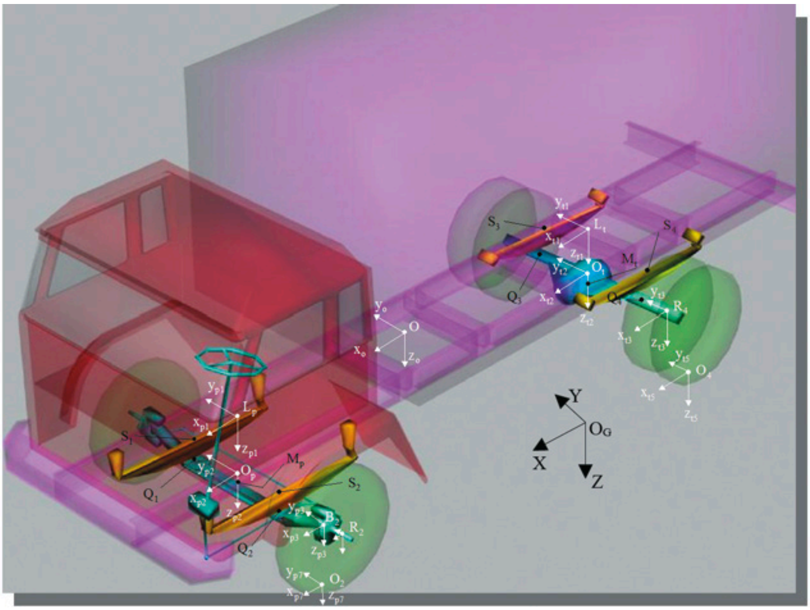

Fig. 1. Idea of the model of a motor truck

The model of the vehicle steering system has been built with taking into account the geometry, kinematics, as well as spring and damping characteristics of the system. The wheels have been given the design toe-in and camber angle; the kingpin inclination and castor angles are also as provided in the vehicle design specifications. When the vehicle model includes an "electrical steering system", a planetary gear and an electric motor is added to the conventional steering system model, with the steering shaft being divided into two separate parts coupled with each other by the planetary gear, which in turn is driven by the electric motor. Thanks to the application of the planetary gear for coupling the upper and lower part of the steering shaft, the steering gear input shaft can be rotated (and, in consequence, the steered wheels can be turned) by both the electric motor and the steering wheel operated independently of each other.

The model of a road wheel with a pneumatic tyre describes the interaction between the tyre and the even road surface, with taking into account the spring and damping characteristics of the tyre in the radial, lateral, and circumferential direction. The model was built with making use of the Dugoff, Fencher, and Segel model [3] extended by adhering to the guidelines arising from the works done by Prof. Mitschke. An important good point of this model is the fact that in spite of relatively simple notation, it makes it possible to take into account the impact of a number of parameters characterizing the motion of a pneumatic tyre on the values of the tangential reactions acting in the tyre-road contact plane. The tyre model adopted makes it possible to simulate the vehicle motion in the conditions of full tyre slip.

The vehicle model was extended by adding simplified models of the ABS, ASR, and ESC systems to it. These additional models may be activated and deactivated, depending on needs.

The inputs applied to the motor vehicle include aerodynamic forces and moments, excitations caused by road surface irregularities, braking or driving moments applied to individual vehicle wheels, as well as forces and moments induced by steering angles. The controlling variables are the steering wheel angle and the accelerator and brake pedal positions.

The equations of motion were formulated with employing a method presented in detail in publication [8] (the method, implemented in a two-stage procedure, was based on the Boltzmann-Hamel equations, written in "quasi-coordinates"). The equations of motion were derived for the vehicle model treated as a free object, with taking into account the constraints that confine the motion of the free object and arise from the tyre-road interaction model adopted and from the impact of the existing steering linkage that determines individual wheel steering angles depending on the angle of rotation of the steering gear lever. As a result, a system of twenty second-order differential equations was obtained, supplemented with two equations representing the constraints.

For the simulation purposes, a special computational program was prepared in the FORTRAN language. At the first stage, the program solves a system of 22 algebraic equations and determines the values of individual second derivatives of the generalized coordinates and the values of two reactions of the constraints imposed on the steered wheels. Afterwards, the program solves the system of the twenty second-order differential equations with the use of the Runge–Kutta fourth-order method with Gill's modification.

Based on this main program, a package of computational programs was prepared, capable of carrying out different variants of simulation calculations for various system control modes (where the steering wheel angle or individual wheel driving and braking moments were used as the controlling variable) and for different external disturbance types (road surface roughness or wind gusts). Variations in the inputs applied to the vehicle may be represented by any functions of time. The programs can handle problems with various road surface types, including the case that the left and right vehicle wheels move on surfaces with different tyre-road adhesion. The wheel lift-off effect is also taken into account in the simulation.

The computational programs prepared require the loading of more than 200 vehicle model parameters. In this work, the values of these parameters were determined from design specifications of the STAR 1142 motor truck and from results of experimental testing of such a vehicle and its components. Particular attention was paid to rig tests of the tyres used [11, 12].

A noteworthy feature of the vehicle model prepared is the fact that it has passed a very extensive and thorough experimental verification during road tests [9, 11], which were carried out in both steady-state conditions (uniform motion along a circle) and transient conditions (rapid turn of the steering wheel during straightforward drive or braking during straightforward drive or drive along a circle, with applying brakes of selected or all vehicle wheels). During the tests, time histories of several dozens of quantities, describing the parameters of motion of individual vehicle solids, were measured. A comparison of results of road tests with results of model tests, carried out in the same conditions of motion, made it possible to specify the values of uncertain parameters of the vehicle model with better accuracy. The very good conformity obtained between the calculation and experimental test results [9, 11] has provided grounds for a statement that the vehicle model prepared adequately describes the vehicle properties in steady and transient conditions of motion. Therefore, the model is suitable for the simulation of motion of a vehicle, even if automatically controlled, in various traffic conditions and situations.

3. Simplified reference model of the vehicle

The reference model used in the automatic control algorithm was developed from the known "bicycle model" of the lateral dynamics of a motor vehicle moving with a constant velocity.

In classic approach, the bicycle model describes the vehicle motion in a local coordinate system attached to the vehicle body solid. This model consists of two linear equations of motion representing the time histories of the linear lateral and angular velocities and accelerations of the vehicle body solid $\dot{y}(t)$, $\ddot{y}(t)$, $\dot{\psi}(t)$, and $\ddot{\psi}(t)$ in response to a prescribed time history of the steering angle $\delta(t)$ being applied as an input. There are seven parameters in this model: vehicle velocity V, mass m, moment of inertia I_z, coefficients k_A and k_B of tyre resistance to sideslip of the front and rear axle wheels, respectively, and distances L_A and L_B between the front and rear wheel axes, respectively, and the projection of the centre of vehicle mass. For the motion to be described in the global coordinate system (OXY), simple trigonometric transformations are usually made.

It has been shown [6] that for small and short-duration disturbances $\delta(t)$ (this condition being met by the control signals that occur during the obstacle avoidance manoeuvre), the transformation equations may be linearized. Thus, a model has been obtained, which describes the vehicle motion in the global coordinate system, as well as the trajectory Y(X): this is the simplified reference model. In this model, all the relations are linear; therefore, they may be subjected to the Laplace transformation and appropriate transmittances may be obtained that would define the relations between the transforms of the corresponding time functions.

$$Y(s) = G_{Y\delta}(s)\delta(s); \quad \psi(s) = G_{\psi\delta}(s)\delta(s) \tag{1), (2}$$

The transmittance equations provide a basis for analysing the vehicle behaviour following a rapid turn of the vehicle wheels, made once to the one side and then to the other side. Since the analytic form of the input $\delta(t)$ (and its transform $\delta(s)$ as well as the transmittances are known, the relations known from the operational calculus may be used to analyse the response.

The simplified transmittance-based reference vehicle model in a reduced version was used for designing the control system regulators and for generating the prescribed reference time function signals.

4. Control system

The control strategy is based on decomposition of the lane-change process into time-related phases and is consistent with the vehicle driving practice followed by experienced drivers. In the first phase of the process, the controlling is done in a partly-open system ("blindly", "just quickly") by generating an appropriate turn of the steering wheel $\delta_H(t)$. The precision of this phase of the manoeuvre is ensured by the previously identified reference model. During this phase of the control process, corrective controlling also takes place. Within this correction, the steering wheel angle is adjusted, based on comparing the signal representing the variable that describes the vehicle's lateral transposition according to the reference model $Y_R(t)$ with the signal that represents this variable being actually measured $Y(t)$. Simultaneously, the reference signals $\delta_{HR}(t)$ and $Y_R(t)$ are generated. The reference signal $Y_R(t)$, which represents the transposition ensuring the obstacle avoidance, is determined by generating a controlling (input) signal according to the equation:

$\delta(t) = \delta_0\big(1(t) – 21(t–T) + 1(t–2T)\big)$, where $1(t)$ is the Heaviside step function, (3)

and by determining the response of the simplified reference model to this input. The values of the amplitude of the steering wheel angle δ_0 and the time T of holding this angle at its maximum value are so selected that the transient functions meet the requirements $|\ddot{y}(t)| \leq \ddot{y}_{dop}$ and $|\dot{\psi}(t)| \leq \dot{\psi}_{dop}$ and that the steady state is reached within a time not exceeding the value defined by the vehicle velocity and the minimum distance to the obstacle at which the manoeuvre could be safely performed on the road surface characterized by a specific coefficient of adhesion μ.

In the second phase, the control process is run in a closed-loop system, on the principle of regulation based on comparing the signals that represent the angular position of the vehicle, i.e. $\psi R(t)$ and $\psi(t)$. In this case, the reference signals generated take a trivial form:

$\psi_R(t) = 0$ and $\delta_R(t) = 0$ (4), (5)

According to the concept adopted, the two-phase control process (phase I – transposition and phase II – angular stabilization) is carried out in one switchable control system (Fig. 2). In the first phase of the control process, the transposition regulation system is on (activated) and the angular stabilization system is off (deactivated); in the second phase, these connections are reversed. The switching over takes place when the centre of vehicle mass reaches a position where the obstacle avoidance is ensured.

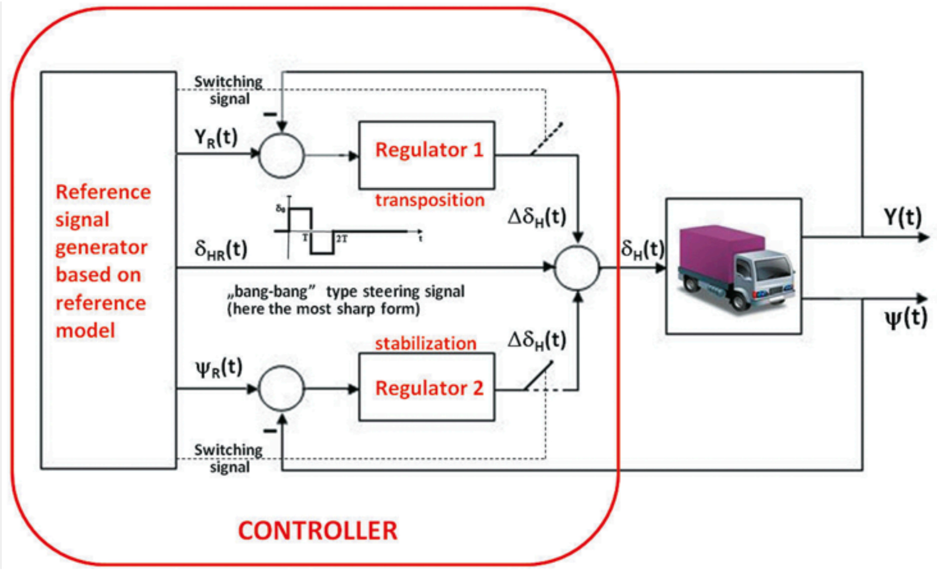

Fig. 2 Block diagram of the system to control the lane-change process [6]

The algorithms of the regulating devices are based on the reference model. They were founded [5, 6] on the Kalman theory applied to the "linear-quadratic problem" [1]. The regulators that have been devised are effective and can be implemented in a relatively simple way. As an example, the model of the regulator for the transposition phase, describing the dependence of the steering angle correction signal on the deviation signals, is represented by the formula

$$\Delta\delta(t) = -\frac{1}{G_{\dot{\psi}\delta 0} V r}\left(\sqrt{\frac{p_{11}}{r}}\Delta Y(t) + \sqrt{p_{11}\left(p_{22} + 2\sqrt{\frac{p_{11}}{r}}\right)}\Delta\dot{Y}(t)\right) \tag{6}$$

where: r, p_{11}, p_{22} – regulator tuning parameters [1];
$\Delta Y(t), \Delta\dot{Y}(t)$ – regulation deviation signals;

$$G_{\dot{\psi}\delta 0} = \frac{k_A k_B (L_A + L_B) V}{k_A k_B (L_A + L_B)^2 - mV^2 (k_A L_A - k_B L_B)} \text{ – transmittance parameter} \tag{7}$$

The simplified reference model is extremely important for the functioning of the control system. For good results to be obtained by using such a model, it must be identified on-line in the time preceding the activation of the control system.

The algorithm prepared for the automatic control system constitutes a basis for the controller of an active steering system. In the simplest design, the time histories of the steering wheel angle $\delta_H(t)$ and the correcting signals $\Delta\delta_H(t)$ may be treated as scaled curves $\delta(t)$ and $\Delta\delta(t)$, where the scaling translates into a constant multiplication ratio, treated as a system parameter. For the dynamics of the steering system to be taken into account, additional corrective members may be introduced.

5. Testing of vehicle's running characteristics

With the use of the complex model of the dynamics of motion of a motor truck, preliminary simulation tests were carried out in the open system at various types of the steering wheel angle input (constant steering wheel angle, rapid turn by a predefined angle with an arrest, jerk of the steering wheel in both directions pursuant to equation (3)). Moreover, a few variants of the vehicle load (vehicle unladen, half-laden, and fully laden, with low and high position of the centre of the sprung mass), different road surface types (with the adhesion coefficient value within the range of μ = 0.1-0.7), and different driving speeds were taken into account.

Each of the tests in specific predefined conditions of motion was repeated several times, with trying to bring the vehicle motion to the state of instability. The tests made it possible to define the scenarios of loss of vehicle stability and to determine the critical process parameter values at which the instability occurred.

Table 1 presents, in a synthetic form, the results of steady-state circular driving tests with a constant steering wheel angle, showing the reasons for the stability loss, for different vehicle loads and road surface types (the tests were carried out in the same vehicle speed ranges, with changing the value of the steering wheel angle for each road surface type). Changes in the load carried on the motor truck load bed considerably affected the vehicle behaviour. The unladen vehicle lost its directional stability in result of the loss of tyre grip at the rear axle wheels regardless of the road surface type. The fully laden vehicle, in turn, lost its directional stability on every road surface type due to the loss of tyre grip at the front axle wheels. The half-laden vehicle, with the centre of the sprung mass being in the low position, lost its directional stability in result of the loss of tyre grip at the rear axle wheels when being driven on slippery road surfaces; in the case of road surfaces with higher adhesion coefficient values, the loss of stability was caused by the loss of tyre grip at the front axle wheels. The higher position of the centre of the sprung mass, combined with higher values of the tyre-road adhesion coefficient, resulted in the fact that the vehicle stability was lost due to the unloaded vehicle wheels being lifted off rather than a tyre slip.

Table 1. Reasons for the vehicle stability loss

μ	Vehicle unladen	Vehicle half-laden		Vehicle fully laden	
		Low centre of mass	High centre of mass	Low centre of mass	High centre of mass
0.1	Full slip of rear axle wheels	Full slip of rear axle wheels	Full slip of rear axle wheels	Full slip of front axle wheels	Full slip of front axle wheels
0.2	Full slip of rear axle wheels	Full slip of rear axle wheels	Full slip of rear axle wheels	Full slip of front axle wheels	Full slip of front axle wheels
0.3	Full slip of rear axle wheels	Full slip of front axle wheels	Full slip of rear axle wheels	Full slip of front axle wheels	Full slip of front axle wheels
0.4	Full slip of rear axle wheels	Full slip of front axle wheels	Full slip of front axle wheels	Full slip of front axle wheels	Full slip of front axle wheels
0.5	Full slip of rear axle wheels	Full slip of front axle wheels	Rear wheel lift-off	Full slip of front axle wheels	Rear wheel lift-off
0.6	Full slip of rear axle wheels	Full slip of front axle wheels	Rear wheel lift-off	Full slip of front axle wheels	Rear wheel lift-off
0.7	Full slip of rear axle wheels	Full slip of front axle wheels	Rear wheel lift-off	Full slip of front axle wheels	Rear wheel lift-off

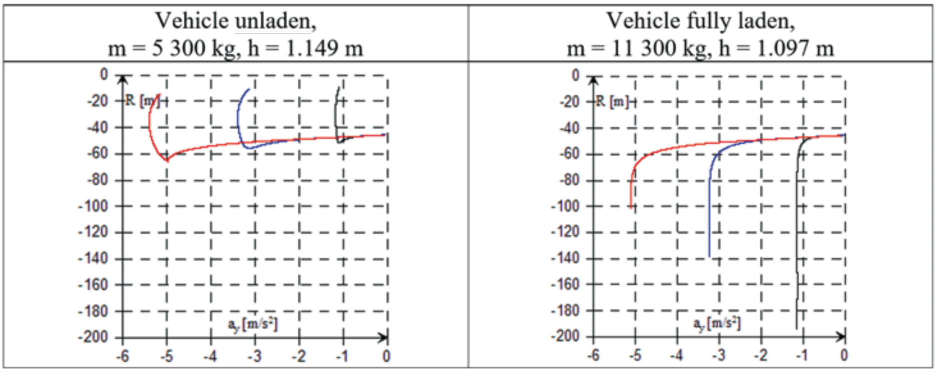

Fig. 3 Motor truck steerability characteristics for 2 variants of vehicle load and 3 road surface types ($\mu = 0.1$; $\mu = 0.3$; $\mu = 0.5$)
R – radius of the circular path; a_y – lateral acceleration

Fig. 3 shows the steerability characteristics of the vehicle being unladen and fully laden, obtained on roads with three different surface types, with low position of the centre of vehicle mass (the tests were carried out with an identical value of the steering wheel angle having been set for all the road surface types). As it can be seen in the graphs, all the vehicles were slightly understeering at low lateral acceleration values, regardless of the vehicle load and road surface type. When the lateral acceleration rose to approach the limit value for the specific road surface type, the unladen vehicle rapidly began to oversteer. The fully laden vehicle was understeering over the whole range of lateral accelerations, with its understeering characteristics rapidly growing when the limit lateral acceleration value was approached.

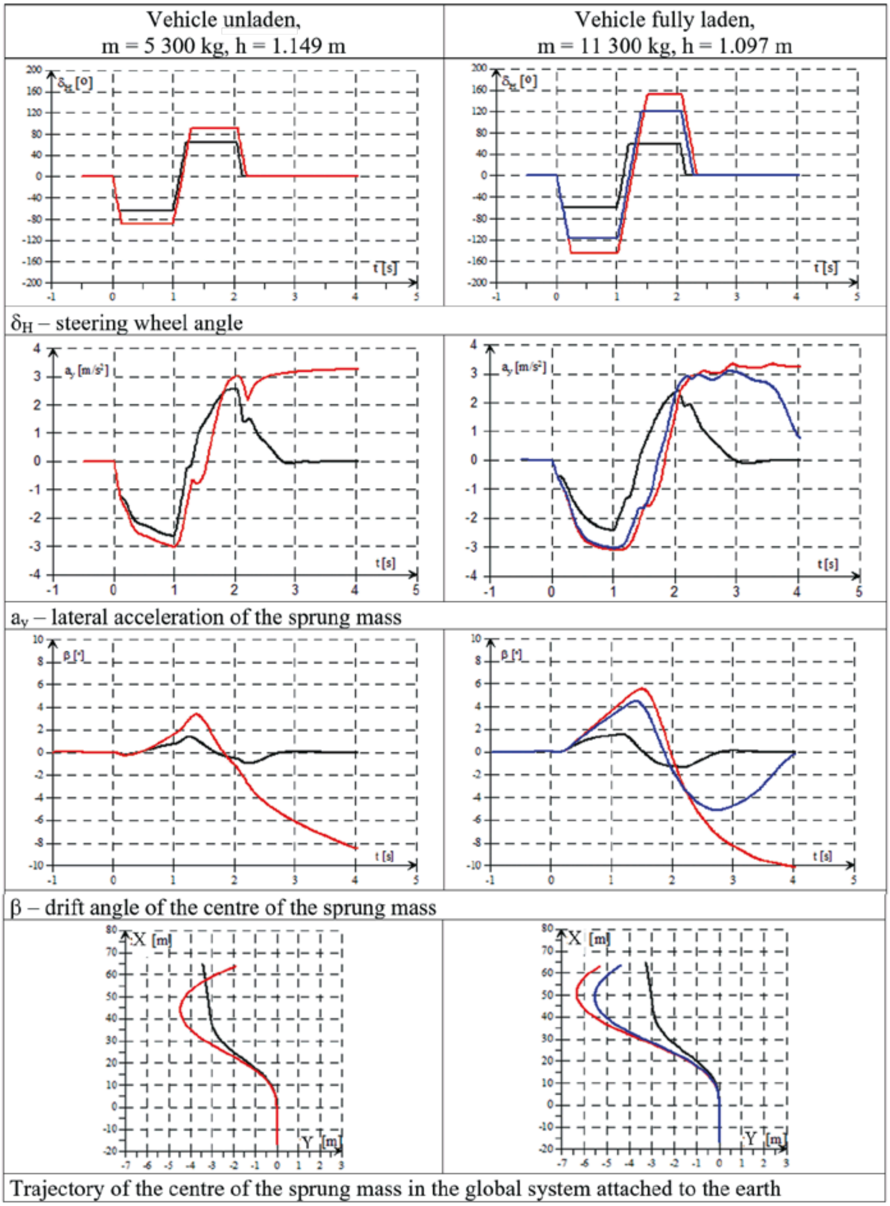

Fig. 4. Jerk of the steering wheel in both directions during straightforward drive

Fig. 4 presents selected results of examination of the complex model response to a jerk of the steering wheel in both directions during straightforward drive. The tests procedure was as follows: the vehicle was accelerated to a prescribed speed, the driving system was disengaged, the steering wheel was rapidly turned by an angle of δ_0 to the left (with a rate of 600 deg/s), arrested in this position to a time of $t_1 = T$, rapidly turned by an angle of $2\delta_0$ to the right, arrested in this position to a time of $t_2 = 2T$, again rapidly turned by an angle of δ_0 to the left, and arrested in this position, which was equivalent to setting the steered wheels straight forwards.

The values of the steering wheel angle δ_0 and time T were determined with using the simplified model in such a way that the vehicle motion at the boundary conditions was obtained as a result. This was done with taking into account the road surface type, vehicle load variant, and driving speed. In the next tests, the steering wheel angle was increased without changing the values of time T and other input parameters in order to achieve the loss of vehicle stability.

The simulation test results that have been presented as an example were obtained for the unladen and fully laden vehicle being driven with a speed of V = 60 km/h on a wet road (μ = 0.3). Individual graphs show a comparison between the curves representing selected quantities, obtained for different values of the steering wheel angle (the black curves represent the stable vehicle motion while the red and blue curves represent the unstable vehicle motion caused by excessive values of the steering wheel angle). The graphs presented confirm the conclusions that have already been formulated on the grounds of test results obtained previously. The stability loss was most easily achieved for the unladen vehicle and the bringing of the vehicle to the loss of stability was most difficult when it was fully laden. For the fully laden vehicle, the directional stability was not lost when the steering wheel angle was even twice as large as that taken as the reference basis. Although gross exceedances of the basic steering wheel angle, at turning the steering wheel leftwards, resulted in the lateral transposition of the centre of vehicle mass being much bigger than that taken as the reference basis (even up to 3 m), they did not cause the vehicle to lose its directional stability. In all the tests, the vehicle did not lose its directional stability before the steering wheel was turned backwards to return the steered wheels to the straightforward position.

The results of three different tests show that the vehicle was most susceptible to the loss of directional stability when it was unladen. Therefore, this was the vehicle load variant to which particular attention had to be paid when planning and preparing the automatic lane-change manoeuvre at the boundary conditions.

6. Obstacle avoidance simulation test results

To verify the vehicle steering concept adopted and the correctness of operation of the regulators, extensive simulation tests were carried out in the closed-loop system; the tests consisted in dodging an obstacle that had suddenly sprung up (by a single lane-change manoeuvre) within as short a distance as possible.

The simulation test conditions were as described below:

- The vehicle under test was moving rectilinearly with a constant speed along a one-way road having two lanes per direction, in the middle of the right-hand lane.
- The width of a single lane was 3 m and the vehicle width was 2.4 m).
- The vehicle speed and road surface type (the tyre-road adhesion coefficient value) were known.
- An obstacle occupying the whole width of the right-hand lane suddenly sprang up in front of the vehicle.
- The distance between the obstacle having suddenly sprung up and the centre of vehicle mass was insufficient for the vehicle to stop before the obstacle but was equal to the minimum distance needed for the obstacle to be avoided.
- For the obstacle to be safely avoided, the centre of vehicle mass had to be laterally displaced by 3 m; in result of this, the vehicle was to move in the middle of the left-hand lane, parallel to the carriageway centreline, with a clearance of Δd = 0.3 m, measured across the carriageway, being left between the vehicle and both the obstacle and the left-hand carriageway edge.
- At the preset vehicle load variant and the predefined vehicle operation conditions, the automatic obstacle avoidance manoeuvre was started, the reference signals representing the required lateral transposition of the centre of vehicle mass and steering wheel angle were generated with the use of the simplified vehicle model and for the time of avoiding the obstacle, the vehicle steering function was taken over by the control system having been designed.

The model tests were preceded by a regulator tuning operation. The regulators were tuned with the use of the trial-and-error method.

A number of simulations of the obstacle avoidance manoeuvre were carried out for different vehicle operation conditions, with changing the vehicle speed (within the range of V = 40 80 km/h), tyre-road adhesion coefficient value (within the range of μ = 0.1-0.7), and vehicle load in successive tests. In total, over 100 obstacle avoidance tests were carried out. The basic criterion of considering the solutions adopted as satisfactory was successful performance of the predefined vehicle's manoeuvre of avoiding the obstacle in all the tests without a need to change the predetermined values of the regulator tuning parameters.

Example results of the simulation of the obstacle avoidance manoeuvre have been presented in Figs. 5 and 6 for the vehicle being unladen and fully laden, respectively.

In the graphs, the curves having been plotted should be interpreted as follows:

- The red dashed line represents the theoretical trajectory of the centre of vehicle mass, based on a sequence of two circular arcs with minimum radius, whose value was determined as the limit at which the lateral tyre grip is lost or a vehicle rollover occurs.
- The black dashed line represents the trajectory of the centre of vehicle mass, whose motion was a response of the complex model of the vehicle to the predefined steering wheel angle input (with the control system being off).

- The blue solid line represents the reference trajectory of the centre of vehicle mass, whose motion was a response of the simplified and linearized reference model of the vehicle to a jerk of the steering wheel in both directions (3) with prescribed values of δ_0 and T.
- The black solid line represents the actually obtained trajectory of the centre of vehicle mass, whose motion was determined as a result of simulation calculations carried out for the "electrical steering wheel" control system being on.

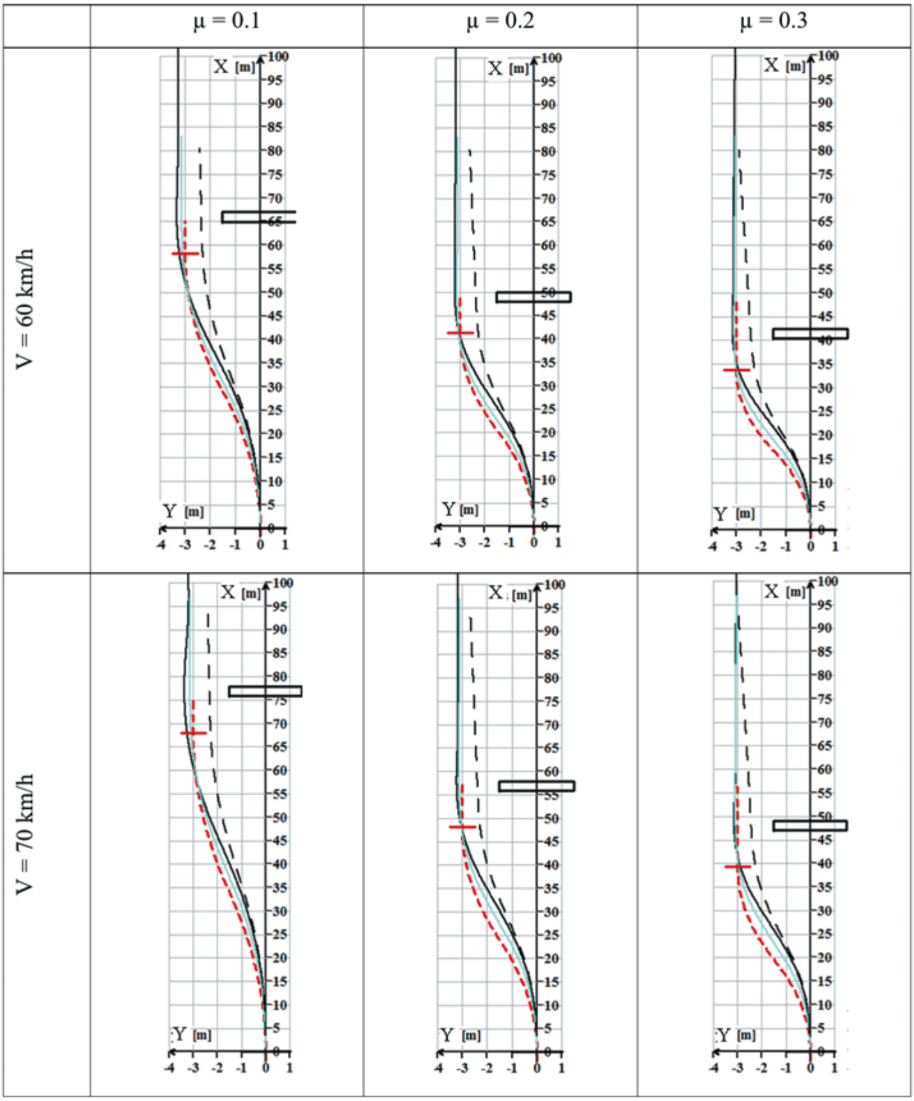

Fig. 5. Results of the simulation of the obstacle avoidance manoeuvre for the unladen vehicle

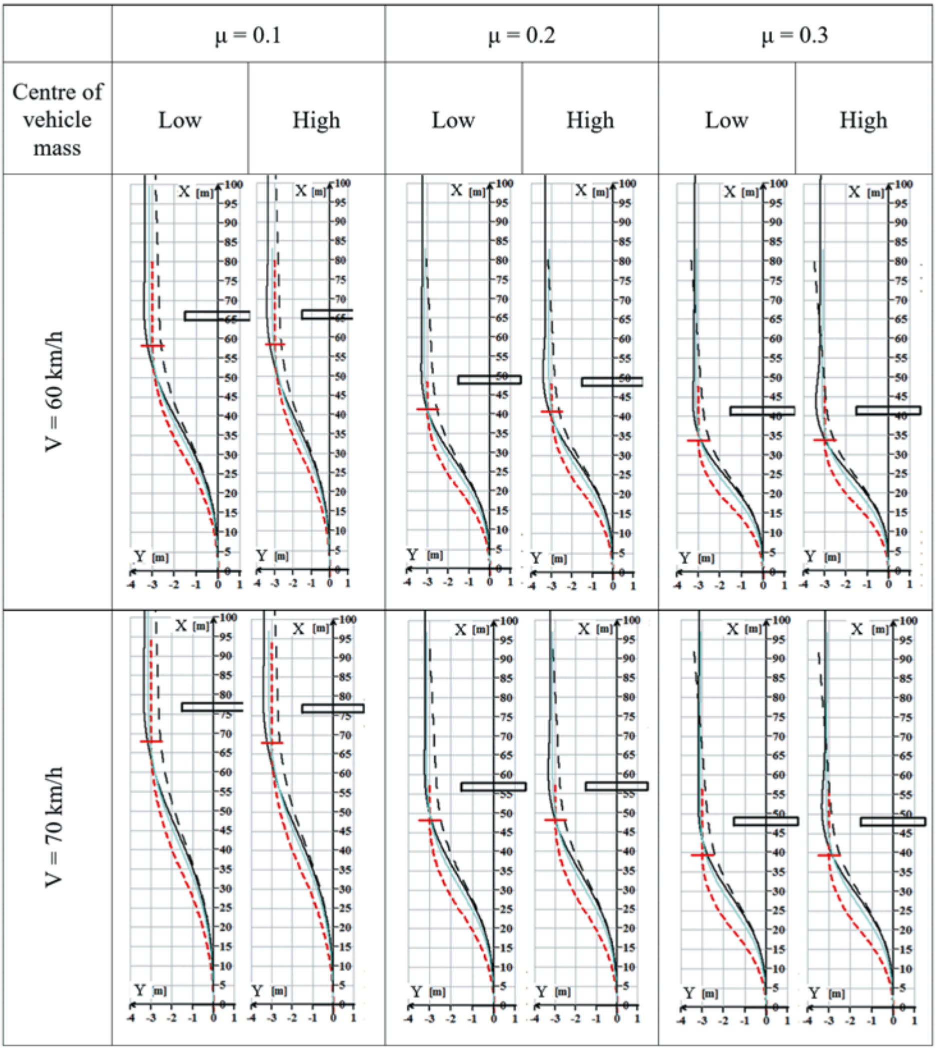

Fig. 6. Results of the simulation of the obstacle avoidance manoeuvre for the fully laden vehicle

Additionally, the theoretical minimum distance between the centre of vehicle mass and the obstacle, determined for the trajectory based on a sequence of two circular arcs with minimum radius, has been marked in the graphs as a short red line. The black rectangle in the graphs represents the obstacle so situated that the distance to the centre of vehicle mass was equal to the realistic minimum distance between the vehicle and the obstacle (i.e. the minimum distance at which the obstacle could be safely avoided in the predefined vehicle operation conditions).

The simulation results showed that the obstacle avoidance manoeuvre was performed correctly in all the tests carried out. Although the simulated vehicle operation conditions were changed over a wide range, the vehicle in the automatic control mode every time avoided the obstacle with remaining within a corridor of prescribed length and width, without losing its stability. Hence, a statement may be made that the vehicle steering concept proposed and the regulators developed proved to be insusceptible to variable conditions of motion and, thanks to that, effective in carrying out the lane-change manoeuvre at the boundary conditions of motion.

7. Recapitulation

The simulation investigation of the process of sudden obstacle avoiding by a motor vehicle was carried out with employing:

- a complex model of a two-axle motor truck of medium load capacity; the model was subjected to very extensive and thorough experimental verification and then it was used as a virtual controlled system;
- a simplified reference model developed from the "bicycle model" by transposition to the global coordinate system and linearization; this model was then used for designing the control system regulators and for generating the prescribed reference time function signals;
- an algorithm of automatic controlling of the lane-change manoeuvre performed within the sudden obstacle avoidance process; the algorithm was founded on the Kalman theory applied to the "linear-quadratic problem".

With the use of the complex model of the dynamics of motion of a motor truck, preliminary simulation tests of vehicle behaviours were carried out in the open system when performing selected vehicle driving manoeuvres at various vehicle operation conditions, for different variants of vehicle load. The tests made it possible to define the scenarios of loss of vehicle stability and to determine the critical vehicle motion parameter values at which the instability occurred. The test results were utilized for the planning and preparation of the automatic lane-change manoeuvre at the boundary conditions.

To verify and validate the solutions adopted for the automatic controlling of motor vehicles, extensive simulation tests were carried out in the closed-loop system (with using the automatic control function of the active steering system EPS), during which an obstacle having suddenly sprung up was dodged by the motor vehicle under test within as short a distance as possible. The test results have shown the automatic control concept adopted to be correct and the regulators devised to function effectively in various vehicle operation conditions.

The solution proposed for the automatization of the process of sudden obstacle avoiding by a motor truck may be said to have the hallmarks of originality and innovativeness on the grounds of the following facts:

- The performance of the control function is based on a simplified reference model describing the vehicle motion in the global coordinate system (a "bicycle model" fully linearized and, moreover, with far-reaching simplification of the structures obtained from its transmittances).

- The controlling is done by generating a predefined signal that represents the current steering wheel angle (rapid jerks applied to the steering wheel in one and the other direction with arresting the wheel in between, with the amplitudes and times of duration of individual phases determined on the grounds of the simplified reference model) and by correcting this signal in result of the operation of two regulation systems, which provide correction signals based on the processing, in the regulators, of appropriate signals of deviations between the signals measured in the real system and those generated from the simplified reference model. The generation of the correction signals is accomplished with the use of Kalman regulators, whose algorithms and operating parameters have been determined from a simplified transmittance-based reference vehicle model in a reduced version.
- The reference signals representing the lateral transposition of the centre of vehicle mass $Y_R(t)$ and the vehicle yaw angle $\psi_R(t)$ are responses of the simplified reference model to the predetermined steering wheel angle $\delta_{HR}(t)$ applied as an input.
- The controlling of the obstacle avoidance process is done in two phases (lateral transpositioning of the vehicle to a new parallel path and stabilization of vehicle's angular position on the new path so that the vehicle yaw angle is brought to zero); therefore, the correction operations are sequentially carried out in the regulation systems (at first, with the use of the regulator of the lateral transpositioning regulator, and afterwards, with the use of the regulator of the angular position stabilization process).

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