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A CONCEPT OF THE AUTOMATIZATION OF DANGEROUS DRIVING MANOEUVRES

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Abstract

Automatic control is one of the main characteristic features of the present-day "electrified" motor vehicle. The avoidance of a suddenly appearing obstacle, including a rapid lane-change manoeuvre, is counted among the driving manoeuvres that are most dangerous and difficult to be automatized. Note that this seems to be a fundamental problem for autonomous cars. The authors have shown general vehicle automatization concepts derived from the control systems theory. Against this background, they have presented results of their own analytical studies and simulation research related to the system of automatic controlling of a rapid lane-change manoeuvre. A controller algorithm developed within this work, where a simplified reference model of the lateral dynamics of a motor vehicle was used (transmittance-type model based on well known "bicycle model" but transformed and linearized), was successfully examined in exhaustive simulation tests. The virtual object used for the simulation tests was an extensive simulation model (3D, nonlinear, MBS type) of a two-axle motor truck of medium load capacity, driven with quite a high speed. This detail model had been experimentally verified. In the article, only a fragment of a large research project has been described.

Keywords: *motor vehicle, active safety, lateral dynamics, avoidance automatization, obstacle voidance, model tests, simulation tests*

1. Introduction

Particular hazard to road traffic is posed by the motor vehicle driving manoeuvres where a vehicle rapidly changes its lane at a high speed. A manoeuvre like this calls for a high degree of driver's skill and is not always successful. It is also difficult for being automatized, because it requires the steering of an object whose dynamic behaviour is unstable and susceptible to changes in road conditions and whose trajectory is subject to tight restrictions. Therefore, the automatization of the rapid load-change manoeuvre is perceived as being fundamental for the development of mechatronic active safety systems to be used as driving aids as well as for the development of autonomous motor vehicles, steered without human intervention.

The issue of controlling the vehicle steering angle during the lane-change manoeuvre is addressed by numerous research works, e.g. [2, 3, 12, 13]. Usually, they are based on a vehicle steering concept that covers planning the desired vehicle path and following the predicted 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, in respect of the manoeuvre duration time or the smoothness of vehicle trajectory, for the assumed form of the function of shape of the desired vehicle path (sinusoid segment, composition of arcs, curves composed of representations of algebraic functions, etc.) and for a specific vehicle speed. The regulation systems proposed in such publications are usually based on the structures and algorithms that are known from theory.

Within authors' studies presented in many publications, e.g. [5-11], works were continued to seek for an algorithm that would enable efficient controlling of the rapid lane-change process in various vehicle operation conditions, with taking also into account possible errors in the measurements of process variables and imperfections in the identification of the reference model used. In the work reported herein, attention was focused on the selection of parameters of the reference control signal generated.

2. Concept of a system to control vehicle motion in case of a suddenly appearing obstacle

The standard scenario of the behaviour of an experienced vehicle driver may be briefly described as a two-stage process of controlling the vehicle motion. The first stage consists of hard vehicle braking without turning the steering wheel; at the second stage, a lane-change manoeuvre is rapidly carried out at a stable vehicle speed. Simultaneous operation of brake pedal and steering wheel can also be imagined, but such a method is extremely difficult and, at the same time, risky and only specially trained rally drivers may boast adequate skills of this kind.

In theoretical terms, the standard controlling of the motion of a fast-moving motor vehicle after an obstacle is noticed to have suddenly sprung up may be treated as a model realization of a two-stage task of optimum steering. A natural criterion of optimization of the vehicle control process is the time of duration of the dangerous road situation. The first issue is the optimization of the instant when the vehicle braking stage should turn into the lane-changing stage. This problem has been analytically solved by the authors in study [11]. A conclusion drawn from that analysis based on mathematical modelling was a seemingly surprising finding that in terms of the time of duration of the dangerous situation, the optimum would be starting the lane-change manoeuvre immediately after noticing the obstacle.

The concept of the automatic control system was prepared without excluding the possibility of braking within the vehicle motion control process. Quite the opposite, an assumption was made a priori that the vehicle motion control process would be run in two stages. It should be noticed that the braking stage would reduce the risk of tragic effects of a collision at a high impact speed and, at the same time, it may help to identify current parameters of the reference model used for controlling the lane-change process (e.g. to determine current parameters of the circumferential and lateral tyre slip characteristics). Thus, the system of automatic controlling of a motor vehicle in case of an obstacle having suddenly sprung up includes modules accountable for hard braking, assessment of the current road situation, identification of the model of vehicle motion dynamics, and lane-changing manoeuvre control process. The said system is based on typical mechatronic systems supplemented with new elements and combined into a single functional entity.

The general idea of the system may be described as follows.

- The motor vehicle is provided with braking and steering systems' digital control modules, electromechanical actuators, motion sensors such as those used in ABS, BAS, and ESP systems, as well as devices to monitor the road and vehicle surroundings, i.e. cameras, radars, and lidars, typical for ACC and LAS systems.
- The motion sensors provide data needed for determining instantaneous values of the variables that describe the dynamics of vehicle motion and the monitoring devices provide all the information necessary for the system to calculate the roadway width available.

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– The system processors carry out the processes of identifying the parameters present in the reference models of longitudinal and lateral vehicle dynamics and of assessing the current road situation and, in the case that the only way to avoid a collision is dodging of the obstacle, they determine the optimum instant when the vehicle braking stage should turn into the lane-changing stage, generate reference signals to control the steering wheel angle, vehicle trajectory, and vehicle yaw angle, and adjust the control function by implementing appropriate correcting algorithms.
– The reference models of vehicle motion dynamics have special, relatively simple mathematical forms. Thanks to this, analytical forms of the reference signals generated and analytical forms of the algorithms of operation of correcting regulators can be built, which is essential for effective real-time controller operation.
– When, in result of the automatic monitoring, the control system detects that the vehicle moves along a straight path in which an obstacle has appeared (at a distance shorter than the necessary vehicle stopping distance) then, at first, a process of hard braking, with using the ACC, BAS, and ABS systems, is started, during which the model of vehicle motion dynamics is identified and the possibilities of stopping the vehicle before hitting the obstacle are estimated. If a collision is found unavoidable in spite of the braking and if the adjacent lane is found to be clear, the automatic control of the steering wheel angle is started and the steering wheel is operated so that the vehicle passes by the obstacle with a constant speed (determined by the final conditions of the vehicle braking stage) along a path parallel to the initial vehicle path.
– The lane-change process controller operates in the structure of an optimum follow-up system [1]. The generator of a signal to control the steering wheel angle sends a predetermined "bang-bang" reference signal (according to the terminology used in automatic control engineering) to the steering system actuator. Before being sent, the signal is corrected by two Kalman regulators switched on in succession. In the first (transposition) phase, the transposition regulator corrects the reference control signal based on the deviation between the reference and measured lateral vehicle displacement curve, for the lateral vehicle displacement to be achieved as prescribed in the reference signal according to the overall vehicle and obstacle dimensions and the width of the lane available. In the second (stabilization) phase, the stabilization regulator corrects the reference control signal based on the deviation between the reference and measured yaw angle curve for the zero vehicle yaw angle to be finally achieved as prescribed in the reference yaw angle signal. The reference signals are determined from the reference model adopted. The essence of the two-phase (transposition plus stabilization) control of the lane-change process has been illustrated in Fig. 1 and a schematic diagram of the control system has been shown in Fig. 2.

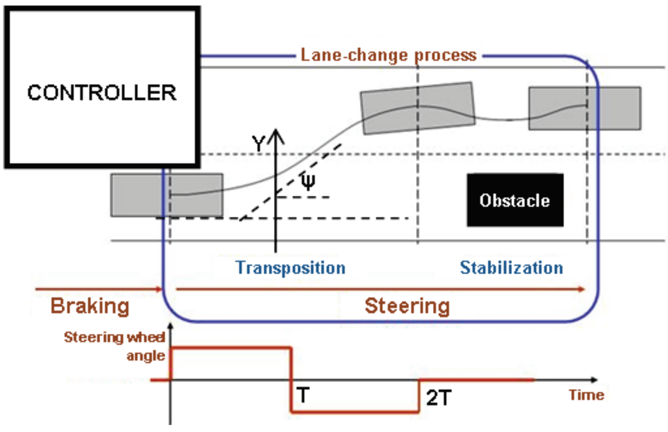

Fig. 1. Two-phase lane change control process

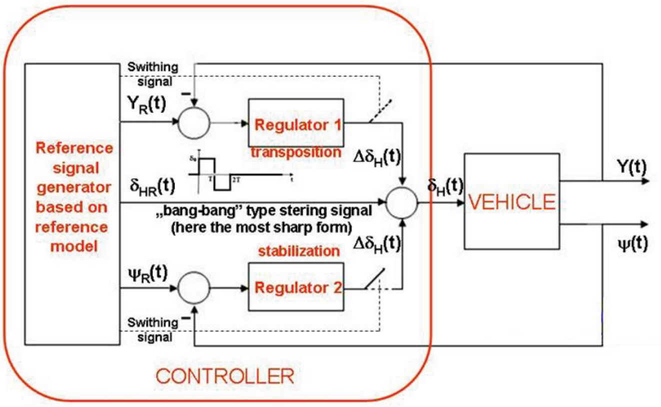

Fig. 2. Schematic diagram of the lane change control system

3. Lane-change model and its use to synthesize the controller algorithm

The lane change control system under consideration has already been described in detail in previous authors' publications, e.g. [5-7]. An important feature distinguishing it from other similar systems is the fact that both the reference signals and the regulator algorithms are determined with taking as a basis a specially prepared reference model of lateral dynamics of a motor vehicle at a "bang-bang" type input applied to the steering system. The reference model derives from the known "bicycle model", subjected to additional mathematical operations (transformation of variables from the local to global coordinate system, linearization, Laplace transformation, determining of transmittances, and reduction of transmittances). Thanks to this, its final form enables analytical determining of the necessary reference signals and regulator algorithms.

The theoretical deliberations are based on the known "bicycle model" describing the lateral dynamics of motion of a motor vehicle moving with a steady speed at not too big disturbances. Such a model is employed in many publications dedicated to the controlling of vehicle motion by means of the vehicle steering system. The idea of the model adopted has been illustrated in Fig. 3.

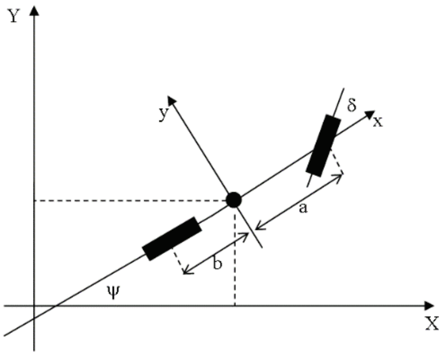

Fig. 3. Idea of the "bicycle model" of a motor vehicle

Notation of model variables and parameters:

- *T* – time (t = 0 means the instant when the steering system is switched on),
- δ(t) – time history of the front wheel steering angle,
- ψ(t) – time history of the vehicle yaw angle relative to the road centre line,
- Ω(t) – time history of the vehicle yaw (angular) velocity ($\Omega(t) = \dot{\psi}(t)$),
- *U*(t) – time history of the linear lateral vehicle velocity in the local coordinate system (*x, y*),
- *V* – linear longitudinal vehicle velocity (constant) in the local coordinate system,

X(t), Y(t) – time histories of the global coordinates of the vehicle mass centre location,
M – vehicle mass,
J – vehicle moment of inertia relative to the vertical axis going through the vehicle mass centre,
a, b – distances of the front and rear vehicle wheel axes, respectively, from the projection of the point representing the vehicle mass centre,
k_A, k_B – cornering stiffness of tyres for the centres of front and rear wheel axes, respectively.

The mathematical model of the lateral dynamics of a motor vehicle, taken here as a starting point ("initial model") for further deliberations, consists of linear equations of motion, representing equilibrium between the forces and moments of forces acting on the vehicle treated as a bicycle moving in the road plane. In the equations that constitute this model, the variables are U(t) and Ω(t), which represent the lateral and angular vehicle velocities in a moving coordinate system attached to the vehicle:

$$m\dot{U}(t)+\frac{k_A+k_B}{V}U(t)+\frac{mV^2+k_Aa-k_Bb}{V}\Omega(t)=k_A\delta(t), \tag{1}$$

$$J\dot{\Omega}(t)+\frac{k_Aa^2+k_Bb^2}{V}\Omega(t)+\frac{k_Aa-k_Bb}{V}U(t)=k_Aa\delta(t)\,. \tag{2}$$

For the equations to be transformed from the moving local coordinate system to the global system fixed to the road, the following equations transforming the velocity vectors are used:

$$\psi(t)=\int_0^t\Omega(\tau)d\tau\,, \tag{3}$$

$$\dot{X}(t)=V\cos(\psi(t))-U(t)\sin(\psi(t))\,, \tag{4}$$

$$\dot{Y}(t)=V\sin(\psi(t))+U(t)\cos(\psi(t))\,. \tag{5}$$

Hence, the trajectory of the vehicle mass centre Y(X) will be defined by the formulas:

$$X(t)=\int_0^t\dot{X}(\tau)d\tau=\int_0^t\left(V\cos(\psi(\tau))-U(\tau)\sin(\psi(\tau))\right)d\tau\,, \tag{6}$$

$$Y(t)=\int_0^t\dot{Y}(\tau)d\tau=\int_0^t\left(V\sin(\psi(t))+U(\tau)\cos(\psi(\tau))\right)d\tau\,. \tag{7}$$

Equations (1-7) may be treated as an "initial reference model".

For small (in terms of amplitude) and short-duration disturbances to the motion (and this is the case at the obstacle avoidance manoeuvre), the linearization of the transformation equations and presentation of the model in an operator-based form with transmittances may be allowed (for details see authors' publications [5, 8]). In consequence, the operator-based model describing the lateral dynamics of a motor vehicle may be expressed as follows:

$$\tilde{Y}(s)=G_{Y\delta}(s)\tilde{\delta}(s)\,, \tag{8}$$

$$\tilde{\psi}(s)=G_{\psi\delta}(s)\tilde{\delta}(s)\,, \tag{9}$$

where the transmittances and their parameters are defined by formulas

$$G_{Y\delta}(s)=\frac{VG_{\Omega\delta 0}\left(T_{Y\delta}^2s^2+2\xi_{Y\delta}T_{Y\delta}s+1\right)}{s^2\left(T_0^2s^2+2\xi_0T_0s+1\right)}\,, \tag{10}$$

$$G_{\psi\delta}(s)=\frac{G_{\Omega\delta 0}\left(T_{\Omega\delta}s+1\right)}{s\left(T_0^2s^2+2\xi_0T_0s+1\right)}\,, \tag{11}$$

$$G_{\Omega\delta 0} = \frac{k_A k_B (a+b)V}{k_A k_B (a+b)^2 - mV^2(k_A a - k_B b)}, \tag{12}$$

$$T_0 = V\sqrt{\frac{mJ}{k_A k_B (a+b)^2 - mV^2(k_A a - k_B b)}}, \tag{13}$$

$$\xi_0 = \frac{m(k_A a^2 + k_B b^2) + J(k_A + k_B)}{2\sqrt{mJ(k_A k_B (a+b)^2)}}, \tag{14}$$

$$T_{Y\delta} = \sqrt{\frac{J}{k_B (a+b)}}, \tag{15}$$

$$\xi_{Y\delta} = \frac{b}{2V}\sqrt{\frac{k_B (a+b)}{J}}, \tag{16}$$

$$T_{\Omega\delta} = \frac{maV}{k_B (a+b)}. \tag{17}$$

It should be noted that the value $W = k_A a - k_B b$ is decisive for the understeer or oversteer vehicle characteristics (if $W < 0$, the vehicle shows understeer; at $W = 0$, the vehicle is neutral; if $W > 0$, the vehicle shows oversteer). The value of this factor has a strong impact on transmittance parameters. In the case of a neutral vehicle, the formula of the amplification factor $G_{\Omega\delta 0}$ becomes much simpler and the value of this factor does not depend on k_A, k_B, and m. In such a case

$$G_{\Omega\delta 0} = \frac{V}{a+b}. \tag{18}$$

The model in its operator-based form is a very efficient tool for calculating the steady-state values of variables Y(t) and ψ(t), as the following known formula may be used in such a case:

$$\lim_{t\to\infty}(f(t)) = \lim_{s\to 0}(s\tilde{f}(s)). \tag{19}$$

At a step input applied (beginning of the “bang-bang” type process):

$$\delta(t) = \delta_0 1(t) \tag{20}$$

and

$$\tilde{\delta}(s) = \delta_0 / s. \tag{21}$$

Then, following the cessation of the transient process:

$$\lim_{t\to\infty}\Omega(t) = \lim_{s\to 0}(s\tilde{\Omega}(s)) = \lim_{s\to 0}(ssG_{\psi\delta}(s)\tilde{\delta}(s)) = G_{\Omega\delta 0}\delta_0 = \Omega_0. \tag{22}$$

This relation makes it possible to connect the values of coefficient $G_{\Omega\delta 0}$ and input δ_0 with the maximum acceptable vehicle yaw velocity value Ω_0 at which the model motion would securely remain stable.

If the steering signal is of the “bang-bang” type (Fig. 1), it may be expressed as a combination of step signals with lags. In the time-related and operator-based notation, we thus have, respectively:

$$\delta(t) = \delta_0(1(t) - 2\cdot 1(t-T) + 1(t-2T)), \tag{23}$$

$$\tilde{\delta}(s) = \delta_0(1/s - 2e^{-sT}/s + e^{-s2T}/s) = \delta_0(1-e^{-sT})^2/s. \tag{24}$$

Then

$$\lim_{t\to\infty}Y(t) = \lim_{s\to 0}(s\tilde{Y}(s)) = \lim_{s\to 0}(sG_{Y\delta}(s)\tilde{\delta}(s)) = T^2 V G_{\Omega\delta 0}\delta_0 = Y_0, \tag{25}$$

$$\lim_{t\to\infty}\psi(t) = \lim_{s\to 0}(s\tilde{\psi}(s)) = \lim_{s\to 0}(sG_{\psi\delta}(s)\tilde{\delta}(s)) = 0. \tag{26}$$

Relation (25) connects the value of lateral displacement Y_0 of the vehicle mass centre with the input parameters and vehicle's operational characteristics, while relation (26) confirms that at the end of the process, the vehicle will move with a zero yaw angle relative to the initial path.

Formulas (22) and (25) define limits for time t approaching infinity. Parameters T and δ_0 determined from a combination of formulas (22) and (25) may be treated as limit values, i.e.

$$\delta_0 \le \Omega_0 / G_{\Omega\delta 0}(m, k_A, k_B, a, b, V) , \tag{27}$$

$$T \ge \sqrt{Y_0 / V\Omega_0} . \tag{28}$$

The analytical relations (27) and (28) will facilitate the determination of parameters T and δ_0 for the reference "bang-bang" type control signal δ_R(t) if all the parameters necessary for this purpose, i.e. m, k_A, k_B, a, b, V, Y_0, and Ω_0, are known. The final determination of parameters T and δ_0 must be done by the system processor, based on a set of the simulation calculation results brought together before.

The coefficients present in formulas (27) and (28) depend on the vehicle speed V resulting from the vehicle braking process. Thus, the lane-change process is strongly parametrically connected with the braking process. Example $G_{\Omega\delta 0}(V)$ characteristic curves for three vehicle types (understeering, neutral, and oversteering) have been shown in Fig. 4.

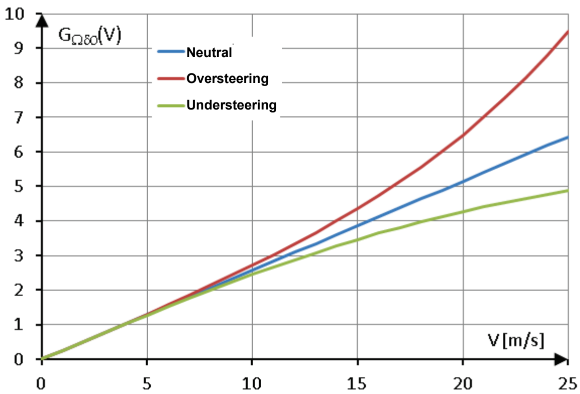

Fig. 4. Example $G_{\Omega\delta 0}(V)$ characteristic curves

The model described in an operator-based form by relations (8-17) is by no means a final form of the reference model the system controller operation is based on.

For control procedures to be possible that would be both simple and effective in terms of calculation accuracy and time, a reduced form of the transmittance-based model was used. Thus, the Kalman regulators used are based upon a transmittance-type model where the transmittances defined by formulas (10) and (11) have been replaced with reduced transmittances.

$$G_{Y\delta}(s) = \frac{VG_{\Omega\delta 0}}{s^2} , \tag{29}$$

$$G_{\psi\delta}(s) = \frac{G_{\Omega\delta 0}}{s} . \tag{30}$$

When a reference model is used to control the steering system, the steering gear ratio p must be taken into account. In the model adopted, the relation between the reference signal δ_{HR}(t) (of the "bang-bang" type), which controls the steering wheel angle, and the reference steering angle vs time curve δ_R(t) is described by a formula

$$\delta_R(t) = \delta_{HR}(t) / p . \tag{31}$$

4. Simulation testing of the control system

To validate the control algorithm and to examine the susceptibility of the model used to its simplifications, changes in individual parameters, and possible signal disturbances, extensive simulation tests had to be carried out. In these tests, a "complex" model of the dynamics of motion of a two-axle motor truck of medium load capacity, thoroughly experimentally verified [4], was used as the virtual steered object. In this model, all the important attributes of a real vehicle have been taken into account.

The physical vehicle model used here (Fig. 5) is a three-dimensional discrete dynamic system. It consists of seven rigid bodies, linked with each other by spring and damping elements having non-linear characteristics. The model has 20 degrees of freedom. The tyre-road interaction model adopted, based on the Dug off model, makes it possible to simulate the vehicle motion in the conditions of full tyre slip and, on the other hand, it handles the wheel lift-off effect.

The model of the vehicle steering system has been built with taking into account its kinematics as well as spring and damping characteristics. In the model, a planetary gear and an electric motor have been added to a conventional steering system, thanks to which the unit may function as an active servo, digitally controlled (e.g. in accordance with a controller algorithm proposed, where the current vehicle position measured, feedback, and regulators may be made use of). In the model of the steering system treated as a servo, not only the spring and damping effects but also the dynamic effects caused by system operation inertia are allowed for.

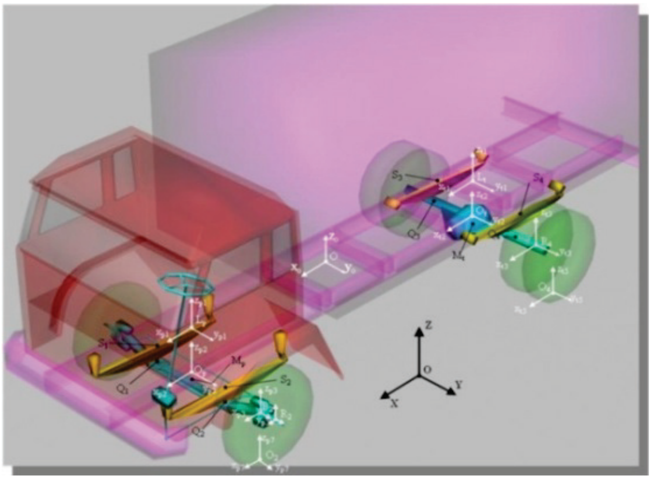

Fig. 5. Idea of the model of a motor truck treated as a virtual vehicle

With the use of the "complex" vehicle model, many simulations of the obstacle avoidance manoeuvre were carried out, which have already been reported in a number of authors' publications, e.g. [5-7, 9, 10]. In this article, some simulation results unpublished hitherto have been presented, with attention being focused on the selection of controller parameters T and δ_0.

The values of amplitude δ_0 and arrest duration time T of the reference control signal $\delta_{HR}(t)$ generated cannot be unequivocally determined. In accordance with the reduced reference model, the same Y_0 displacement effect may be obtained when input signals with low δ_0 and high T values or high δ_0 and low T values are applied. The lane-change process duration time (and, thus, the distance to the obstacle that would be sufficient for a collision with the obstacle to be avoided) cannot be minimized by unlimitedly raising the control signal amplitude because this may result in process destabilization. The analytical dependence of Y_0 on δ_0 and T, described by equations (25), (27), and (28), will only hold if, after all, the model linearization assumptions are true; hence, in practice, the instantaneous values of the derivatives of functions $Y(t)$ and $\psi(t)$ must be within acceptability limits. These limiting peak values may be determined in a simulation process.

To examine the possibilities of minimizing the necessary obstacle avoidance time, simulation tests were carried out where the virtual object, i.e. the vehicle unladen and, alternately, fully laden with the centre of vehicle mass being in its low position, rapidly performed a lane-change manoeuvre, with a speed of 70 km/h at the beginning of the manoeuvre, on a wet road characterized by a sliding adhesion coefficient value of $\mu_s = 0.3$. The two vehicle load states as mentioned above were chosen intentionally because previous tests carried out on the virtual object in the open-loop configuration had revealed [7] that the unladen vehicle would be far more difficult to be automatically steered in boundary conditions in comparison with the fully laden vehicle with a low position of the centre of mass. This was because with lateral acceleration growing to approach the limit value for a specific road surface, the unladen vehicle rapidly turns from understeering to oversteering. By contrast, the fully laden vehicle remains understeering over the whole range of lateral accelerations and, additionally, this understeer steeply increases when the lateral acceleration comes close to its limit value.

The limit yaw velocity values Ω_{0i} (i = 1, 2, 3, 4) at which the vehicle motion on a wet road ($\mu = 0.3$) with a speed of 70 km/h would securely remain stable was calculated by various methods. The following data options were used for this purpose:

– Acceptable value of the sliding adhesion coefficient μ_s on a wet road; then, $\Omega_{01} = \mu_s/V$.
– Maximum lateral acceleration value a_{ymax}, determined from a simulated steady-state circular driving test [10]; then, $\Omega_{02} = a_{ymax}/V$.
– Maximum yaw velocity value Ω_{03} determined from a simulated straightforward drive with a rapid turn of the steering wheel (with an arrest) by a maximum angle at which stable vehicle motion was still possible.
– Maximum yaw velocity value Ω_{04} determined from a simulated straightforward drive with a double rapid turn of the steering wheel, once to the left and once to the right (with an arrest for time T), by a maximum angle at which stable vehicle motion was still possible.

The Ω_{0i} values determined as described above for the vehicle being unladen and fully laden have been specified in Tab. 1. These values grew for the successive methods arranged in the order as above and this growth was markedly steeper when the vehicle was fully laden. Furthermore, values of amplitude δ_{HR} and arrest duration time T of the reference signal were calculated with using equations (27), (28), and (31) and with taking into account the fact that the centre of vehicle mass would have to displaced by Y_0 = 3 m for a collision with the obstacle to be avoided. Of course, the growth in Ω_{0i} values resulted in an increase in δ_{HR} and a decrease in T.

Tab. 1. Limit yaw (angular) velocity values and corresponding values of the amplitude and arrest duration time of the reference signals

Vehicle unladen				
	Ω01 = 0.151 rad/s	Ω2 = 0.159 rad/s	Ω03 = 0.161 rad/s	Ω04 = 0.171 rad/s
δHR [deg]	59.8	62.8	63.7	67.0
T [s]	1.01	0.99	0.98	0.95
So [m]*	42.4	41.4	41.1	39.8
Vehicle fully laden				
	Ω01 = 0.151 rad/s	Ω02 = 0.169 rad/s	Ω03 = 0.2 rad/s	Ω04 = 0.25 rad/s
δHR [deg]	55.1	61.7	72.8	90.9
T [s]	1.01	0.96	0.88	0.79
So [m]*	45.2	43.0	40.1	37.5

* So = minimum distance necessary for the vehicle to avoid a collision with the obstacle

The raising of δ_{HR} with simultaneous reduction in T would cause the reference signals to approach the red dashed curves, which would result in increasingly difficult following of the said reference signals at automatic performance of the obstacle avoidance manoeuvre. The reference signals presented in Fig. 6 were used to carry out simulations of the automatic obstacle avoidance manoeuvre.

The simulation results for the unladen and fully laden vehicle have been shown in Fig. 7 and 8. The colouring of the curves as that in Fig. 6. The additional notation has been used in the graphs:

β – vehicle mass centre sideslip angle,
a_y – lateral acceleration of the vehicle mass centre.

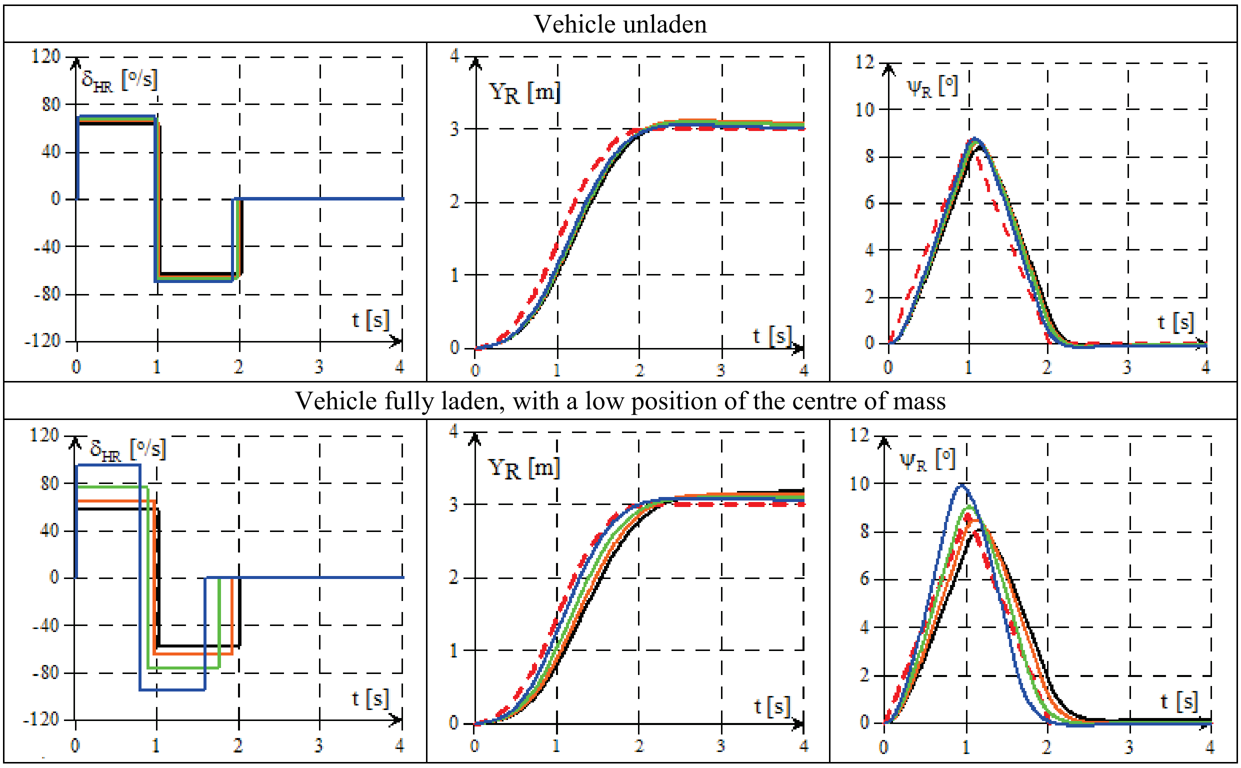

Fig. 6. Reference signals of the steering wheel angle $\delta_{HR}(t)$ lateral displacement of the vehicle mass centre $Y_R(t)$, and vehicle yaw angle $\psi_R(t)$

The graphs in Fig. 7 and 8 show that in result of the applying of reference steering wheel angle signals characterized by increasing values of amplitude δ_{HR} and simultaneously decreasing values of arrest duration time T in successive simulations, the yaw velocity and lateral acceleration values gradually approach the limit values not to be exceeded in the case of the road surface being wet. This becomes particularly well visible at the second jerk of the steering wheel. In spite of this, the vehicle still does not lose its stability and avoids a collision with the obstacle. Additionally, the minimum distance necessary for the vehicle to avoid a collision with the obstacle (*So*) is thus shortened by about 2.6 m (6%) for the unladen vehicle and by about 7.7 m (17%) when the vehicle is fully laden (see Tab. 1). Unfortunately, increasing corrections $\Delta\delta_H$ must be added by the regulator to the reference signal δ_{HR} for the prescribed vehicle path and direction of motion to be maintained and this will result in increasing values of vehicle yaw angle ψ and sideslip angle β. The vehicle moves with growing sideslip and any further increase in the amplitude of the reference signal of the steering wheel angle, in spite of a reduction of arrest duration time T, inevitably leads to a loss of directional stability. This can be particularly clearly seen in the curves plotted in blue in Figure 7 and 8, if taken as an example. Although a collision with the obstacle has been avoided in this case, the regulator still must cause the steering wheel to be turned by corrective angles.

The curves in Fig. 7 and 8 and the *So* values specified in Tab. 1 show that the raising of the limit values of vehicle yaw velocity Ω_0 (used for determining the reference signals) is not always effective. In the case of the unladen vehicle, the benefits obtained from this were insignificant, while the vehicle was simultaneously exposed to a serious hazard of a loss of directional stability. This means that in this case, the limit Ω_0 value must be determined with a great care, with taking the sliding adhesion coefficient as a basis (see the curves plotted in black). In the case of the vehicle being fully laden, the limit Ω_0 value should be determined with employing the method of rapid turn of the steering wheel during straightforward drive.

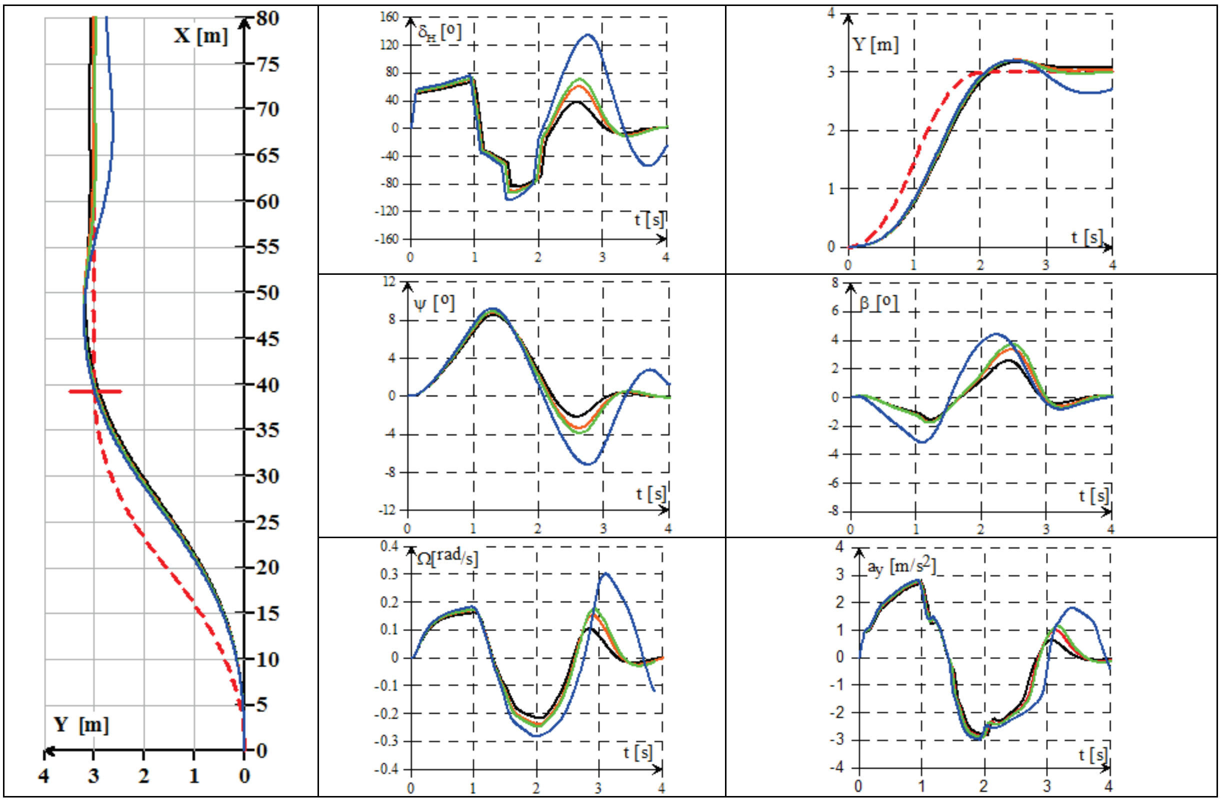

Fig. 7. Results of the simulation of an obstacle avoidance manoeuvre for the unladen vehicle

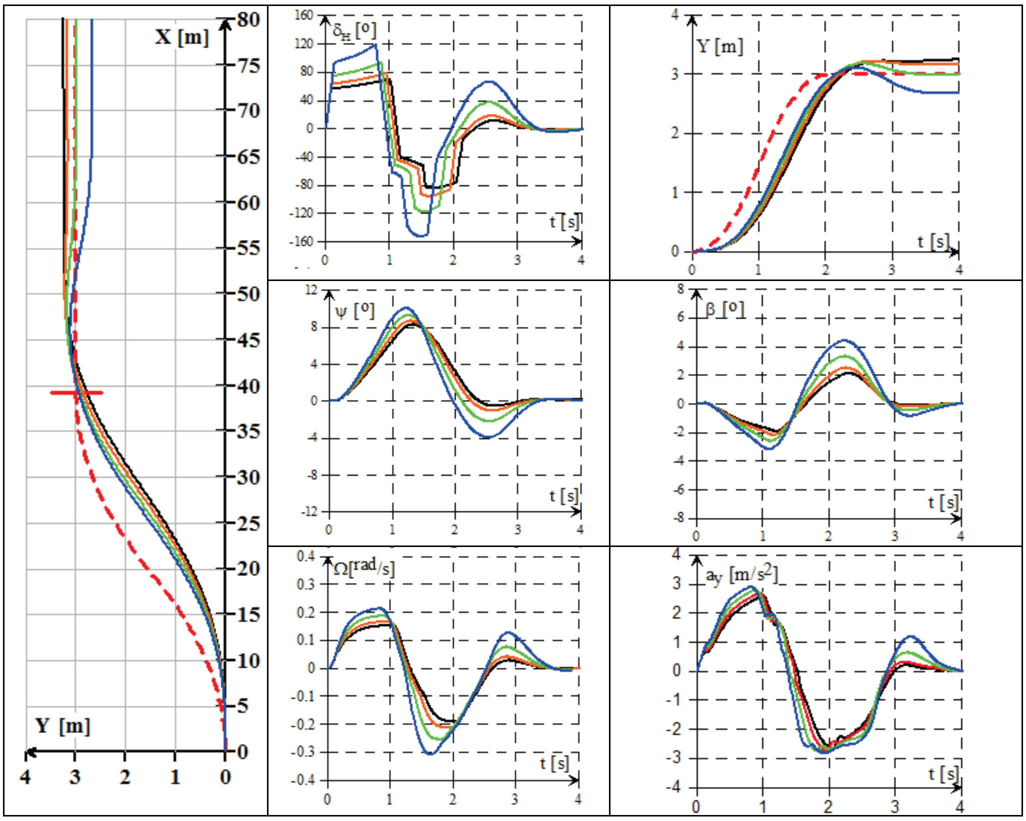

Fig. 8. Results of the simulation of an obstacle avoidance manoeuvre for the fully laden vehicle

5. Recapitulation and conclusions

This article presents a concept of the automatization of a particularly dangerous driving manoeuvre, where a motor truck moving with a high speed on a slippery road must suddenly change its lane. The idea of the system developed consists in planning the desired vehicle path and following the predicted trajectory in a tracking process (with the use of appropriate sensors and regulating devices) exclusively by turning the steering wheel. The lane-change process controller operates in the structure of an optimum follow-up system.

In the work, attention was focused on the selection of parameters, i.e. the amplitude and arrest time, of the reference steering wheel angle signal applied by the controller. The article presents results of simulation tests, where a "complex" model of the dynamics of motion of a motor truck, thoroughly experimentally verified, was used as the virtual steered object. Based on the test results, recommendations have been formulated as regards the method of selection of process parameters for the vehicle being unladen and fully laden. The findings include a statement that for the unladen vehicle, which is difficult to be automatically steered, the raising of the amplitude of the reference signal is not as effective as expected.

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