MODELLING AND SIMULATION OF A DISTURBANCE TO THE MOTION OF A MOTOR VEHICLE ENTERING A SKID PAD AS USED FOR TESTS AT DRIVER IMPROVEMENT CENTRES

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Summary

The paper concerns the modelling of a disturbance caused to vehicle motion on a skid pad during tests carried out at Driver Improvement Centres (DIC). The disturbance is induced with the use of a dynamic "kick plate", which is a mandatory element of the equipment of driver improvement centres of the basic and higher degree, pursuant to the Regulation of the Polish Minister of Transport, Construction and Maritime Economy of 16 January 2013 on the improving of driving techniques.
The disturbance caused by a lateral displacement of the kick plate (relative to the vehicle path) forces the driver to undertake defensive manoeuvres such as turning the steering wheel, braking, or accelerating, separately or in combinations.
The paper presents a model of motion and dynamics of a motor vehicle as well as a method of practical implementation of such a disturbance. The simulation results, showing the scale of the disturbance to vehicle motion to which the driver being trained must respond, have also been included. The methodology and results of choosing the speed with which the vehicle is to be driven during the test have been presented as well.

Keywords: simulation, motor vehicle dynamics, Driver Improvement Centres

1. Introduction

At Driver Improvement Centres (DIC), tests are carried out where, before the vehicle under test enters a skid pad characterized by reduced tyre-to-road adhesion, the vehicle motion is disturbed by means of a dynamic "kick plate". Such a kick plate must be available at every Centre of this kind, according to the Regulation of the Polish Minister of Transport, Construction and Maritime Economy of 16 January 2013 on the improving of driving techniques [12].

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In practice, the said disturbance is induced in most cases by causing a lateral displacement (jerk) of the kick plate (relative to the vehicle path) in the instant when the front vehicle wheels have just left the kick plate and the rear wheels are still moving on it. The disturbance caused by such a movement of the plate forces the driver to undertake defensive manoeuvres, usually to turn the steering wheel. However, other driver's actions, even most surprising, are also possible, e.g. pressing the brake or accelerator pedals or various combinations of the said reactions, which translate into various vehicle manoeuvres.

The paper presents a model of motion and dynamics of a motor vehicle as well as a method of causing the said disturbance to vehicle motion. The simulation results, showing the scale of the disturbance to vehicle motion to which the driver being trained must respond, will also be included. Only the behaviour of the vehicle as such is subject to the assessment, without the impact of driver's actions on it being taken into account.

2. Construction of the track on which the tests with disturbing the vehicle motion are carried out

Pursuant to the applicable legal instrument [12], the infrastructure of every Driver Improvement Centre of the basic or higher degree should include a facility for intentional destabilization of the vehicle travel path. Before the entrance to a "rectangular skid pad" (an area with reduced tyre-to-road adhesion, representing road surface covered with a water layer), there should be "a mechanical device destabilizing the vehicle travel path, situated immediately at the skid pad (...), provided with a device other than trolleys, making it possible to change the adhesion of the front or rear vehicle wheels to the ground". Such a destabilizing device may be a "dynamic plate", also referred to as "kick plate" [16]. In most cases, the motion-disturbing action is applied to the rear wheels [13, 15, 16, 17]. A rectangular skid pad with safety zones, being an element of the infrastructure of a Driver Improvement Centre of the higher degree [12], has been shown in Fig. 1.

3. Simulation models of motion and dynamics of a passenger car

The model of motion and dynamics of a two-axle road vehicle corresponds to a medium-class passenger car KIA cee'd SW. For this model, very good results of experimental verification were obtained at the tests recommended by ISO.

3.1. Main simplifying assumptions

First of all, the basic vehicle motion is analysed, represented by the coordinates of position of the vehicle body solid and their derivatives. Selected disturbances to vehicle motion, related to the fact that the vehicle moves on an uneven road surface and to the phenomena that occur within the tyre-road contact patch are also taken into account. The vehicle is treated as a set of rigid bodies and material particles linked with each other by

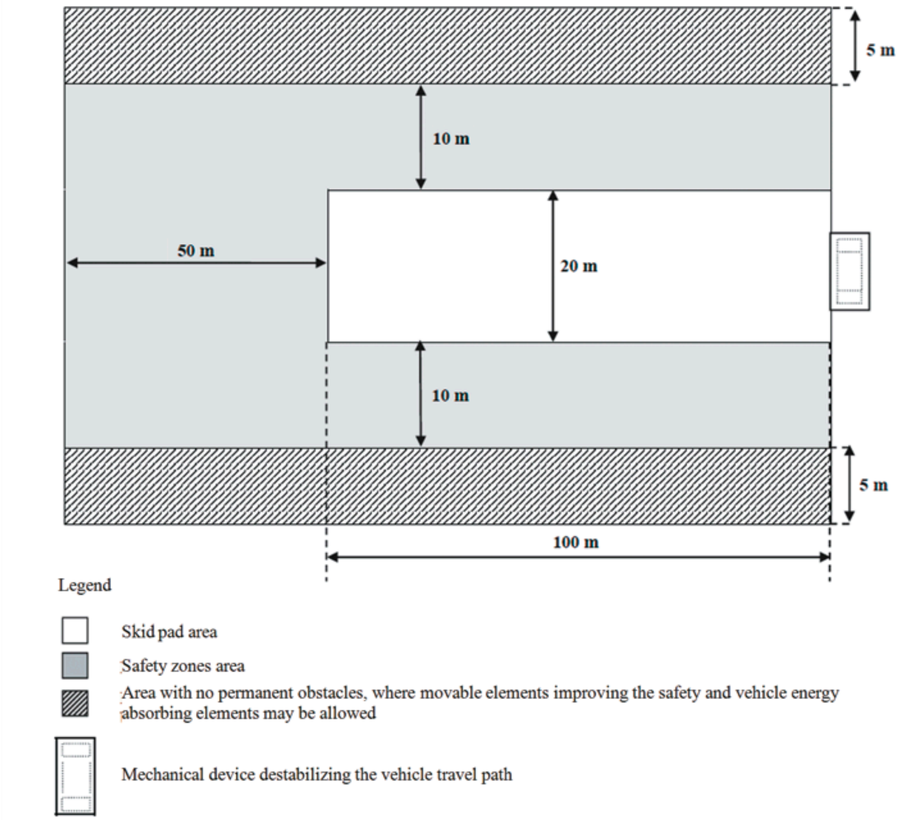

Fig. 1. Rectangular skid pad with safety zones, being an element of the infrastructure of a Driver Improvement Centre of the higher degree [12]

spring and damping leading elements. The relative movements of passengers, driver, load, and powertrain are ignored. These elements are included in the rigid body representing the vehicle body. The vehicle body has a longitudinal symmetry plane. The inputs originate from the driver, who acts on vehicle controls and thus changes the steering wheel turning angle, brake pedal force, throttle opening or, more generally, position of the control of the engine fuel feed system. The road surface is considered as undeformable (perfectly rigid). Longitudinal slope (grade) and transverse slope (cross slope) as well as rough road surface are allowed. The vehicle acts on the road through flexible wheels (pneumatic tyres).

3.2. Physical model of the vehicle

The KIA cee'd SW car (Fig. 2) has a McPherson strut front suspension system with an antiroll bar. The suspension systems of the left and right rear wheels are independent from each other (apart from being coupled by an antiroll bar) and each of them consists of spring element (a coil spring), shock absorber, transverse arms, and trailing arm.

A physical model of a two-axle motor vehicle with independent front and rear wheel suspension systems, together with the coordinate systems adopted, has been shown in Fig. 3.

Fig. 2. The KIA cee'd SW car [14]

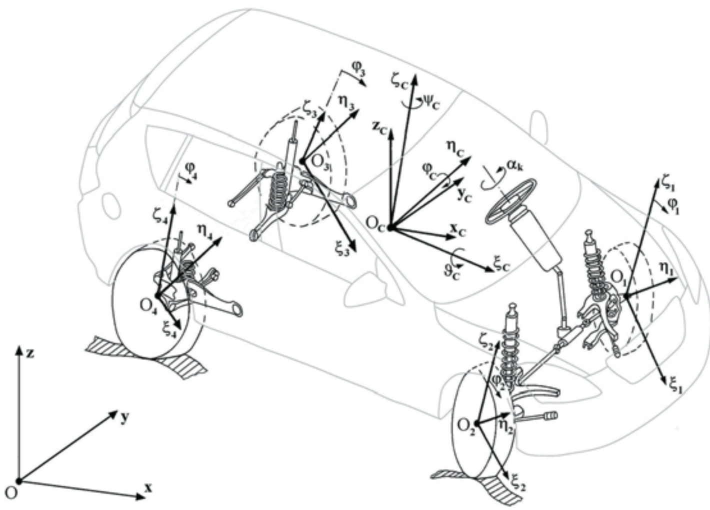

Fig. 3. Physical model of a passenger car, i.e. a two-axle vehicle with independent front and rear wheel suspension systems, together with the coordinate systems adopted [10]

3.3. Inertial properties of the vehicles modelled

The model of a two-axle passenger car presented in Fig. 3 consists of nine mass elements: vehicle body solid (treated as a rigid body), four material particles O_1, O_2, O_3, and O_4, where the vehicle "unsprung masses" have been concentrated (including road wheels in their translational motion), and four solids representing the rotating road wheels (exclusively in their rotational motion) [9].

3.4. Coordinate systems adopted

The following coordinate systems have been adopted (Fig. 3) [7, 8, 9, 10, 11]:

- $Oxyz$ – an inertial system, fixed to the road, with the Ox and Oy axes being horizontal and the vertical axis Oz pointing upwards;
- $O_Cx_Cy_Cz_C$ – a non-inertial system, with its axes being respectively parallel to axes Ox, Oy, and Oz and with its origin situated at the centre of mass of the vehicle body solid O_C;
- coordinate systems fixed to the rigid bodies of the model, i.e. body solid ($O_C\xi_C\eta_C\zeta_C$) and four road wheels ($O_1\xi_1\eta_1\zeta_1$, $O_2\xi_2\eta_2\zeta_2$, $O_3\xi_3\eta_3\zeta_3$, $O_4\xi_4\eta_4\zeta_4$);
- auxiliary systems, facilitating the defining of transformation matrices.

To describe the translational motion of the solids and material particles of the model, the positions of the centres of mass (O_C, O_1, O_2, O_3, O_4) of the said solids are used. The axes $O_i\xi_i$, $O_i\eta_i$, $O_i\zeta_i$ (i = 1, 2, 3, 4) are treated as the principal central axes of inertia of the corresponding rigid bodies. The rotation of the vehicle body solid about the fixed point O_C has been described with the use of "aircraft angles", also referred to as "quasi-Euler angles" [7, 8, 9, 11]. The axes of individual rotations are treated as the principal central axes of rotation of the vehicle body solid.

A more detailed description of the family of models of motion and dynamics of two-axle vehicles and a description of the tyre-road interaction may be found in reference literature items [8, 9, 10] published by the author of this paper.

3.5. Equations of motion

The equations of motion have been derived with the use of Lagrange equations of the second kind (e.g. [6, 9]). Prior to this, the following 14 generalized coordinates were adopted:

$q_1 = x_{OC}$, $q_2 = y_{OC}$, $q_3 = z_{OC}$ – coordinates defining the position of the centre of mass of the vehicle body solid (O_C) in the inertial reference system $Oxyz$;

$q_4 = \psi_C$, $q_5 = \varphi_C$, $q_6 = \vartheta_C$ – coordinates describing the rotation of the vehicle body solid about its centre of mass O_C; these are the quasi-Euler (aircraft) angles, i.e. heading angle, pitch angle, and bank angle, respectively;

$q_7 = \zeta_{CO1}$, $q_8 = \zeta_{CO2}$, $q_9 = \zeta_{CO3}$, $q_{10} = \zeta_{CO4}$ – coordinates describing the motion of points O_1,

O_2, O_3, O_4 relative to the vehicle body solid in the direction of axis $O_{C\xi C}$ of the $O_C\xi_C\eta_C\zeta_C$ coordinate system; to these points, the "unsprung masses" of the suspension system are reduced;

$q_{11} = \varphi_1$, $q_{12} = \varphi_2$, $q_{13} = \varphi_3$, $q_{14} = \varphi_4$ – angles of rotation of road wheels (front left and right and rear left and right wheel, respectively).

3.6. Geometric and spring characteristics of the steering system

The actual characteristic of the steered wheel turning angle as a function of the steering wheel turning angle has been implemented in the model and the flexibility of the steering system has been taken into account. Additional steered wheel turning angles have been introduced as functions of the stabilizing moments, torsional flexibility of the steering column with the steering gear, and flexibility of the left and right side of the steering linkage [9].

3.7. Forces and torques occurring within the tyre-road contact patch

The tyre-road contact forces have been described with the use of the HSRI-UMTRI model [4, 5] supplemented with the IPG Tire model of transient states of tyres [9]. The forces developing within the tyre-road contact patch can be modelled with taking also into account the operation of driving aids referred to as ABS (Antilock Braking System), ASR (Anti-Spin Regulation system), and ESP (Electronic Stability Program), in accordance with guidelines and algorithms proposed by Bosch [2, 3] and with the available documentation provided by the manufacturer of the vehicle being modelled.

4. Adding of a disturbance, caused by a lateral displacement of the kick plate put before the entrance to the skid pad, to the vehicle motion model

Fig. 4 shows a schematic diagram taken as a basis for the modification to the model and to the program simulating the vehicle motion on a test track used to examine vehicle drivers. For the simulation, the most popular variant has been chosen [13, 15, 16, 17], with disturbing the motion of rear vehicle wheels when the front wheels have already left the kick plate, as illustrated in Fig. 4. The length and width of the kick plate is l_p [m] and s_p [m], respectively.

During the process of disturbing the vehicle motion, the lateral velocity of the kick plate (relative to the initial vehicle travel path) is v_{yp} [m/s]. At the present stage of the work, when only general specifications of the kick plate are known, this velocity has been assumed as constant during the test. If the extreme displacement of the kick plate is s_{yp} [m], then the kick plate displacement time is t_{yp} [s] and may be calculated from equation (1) as the absolute value of the quotient of lateral displacement of the kick plate and its lateral velocity. This is the first limitation on the time of duration of the disturbance to vehicle motion.

$$t_{yp} = |s_{yp} / v_{yp}| \tag{1}$$

Another limitation is the time t_{np} [s] of the rear wheels being present on the kick plate. Its maximum value t_{npmax} [s] may be expressed by formula (2), where v [m/s] is the vehicle velocity in the instant when the disturbance begins and l [m] is the vehicle wheelbase.

$$t_{npmax} = l / v \quad \text{for} \quad l \le l_p \qquad \text{and} \qquad t_{npmax} = l_p / v \quad \text{for} \quad l > l_p \tag{2}$$

An assumption has been made here that the changes in vehicle heading angle ψ_C [rad] and in vehicle velocity v [m/s] during time t_{np} [s] are so small that for the phase when the vehicle moves on the kick plate, the approximation according to (3) may be considered satisfactory.

$$v_x = v_\xi = v \tag{3}$$

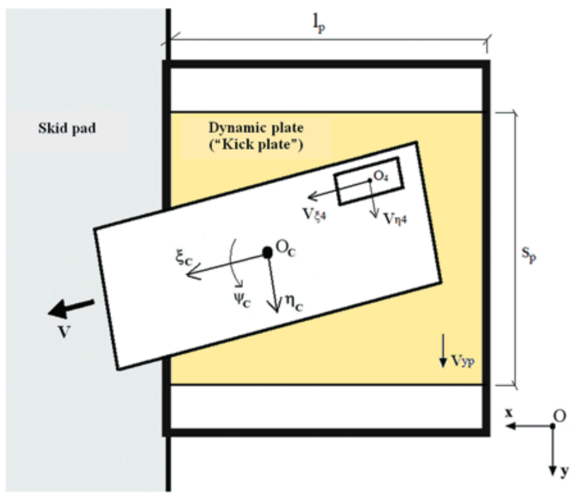

Fig. 4. Schematic diagram taken as a basis for the modification to the model and to the program simulating the vehicle motion on a DIC's test track provided with a skid pad and a dynamic plate ("kick plate")

The said modification to the vehicle motion model [9, 10] by adding a disturbance in the form of a lateral displacement of the kick plate has been implemented by modifying the values of the longitudinal and lateral velocities ($v_{\xi k}$ [m/s] and $v_{\eta k}$ [m/s], respectively) of the projection of the velocity of the k^{th} wheel's centre O_k on the road surface plane. These values are calculated in the local coordinate system $O_k\xi_k\eta_k\zeta_k$ fixed to the road wheel. Such a modification, described by equations (4) and (5), reflects the effects of the lateral movement of the kick plate, placed before the entrance to the skid pad, with a velocity of v_{yp} [m/s]. In Fig. 4, these quantities have been shown for the rear right wheel (k = 4).

$$v_{\xi kw} = v_{\xi k} - v_{yp} \cdot \sin \psi_C \tag{4}$$

$$v_{\eta kw} = v_{\eta k} - v_{yp} \cdot \cos \psi_C \tag{5}$$

where $v_{\xi kw}$ [m/s] and $v_{\eta kw}$ [m/s] represent the longitudinal component and lateral component, respectively, of the projection of the velocity of the k^{th} wheel's centre O_k on the road surface plane, measured in relation to the movable kick plate.

5. Data assumed for the vehicle under test, disturbance to, and conditions of the vehicle motion

The vehicle data correspond to the nominal specifications of the Kia cee'd SW car in running order, loaded with a driver and a driving instructor. The total mass of the vehicle under test was 1570 kg, wheelbase was l = 2.655 m, distances between the centre of mass and the front and rear axle planes were l_1 = 0.976 m and l_2 = 1.679 m, respectively, and static height of the centre of mass was z_{Oc} = 0.516 m.

The kick plate parameters were assumed in accordance with the specifications of the facility made by a Polish company UNIMETAL [17], i.e. length l_p = 3.0 m, width s_p = 2.7 m, lateral displacement velocity v_{yp} = 1.5 m/s, extreme lateral displacement s_{yp} = ±0.3 m (assumed here as s_{yp} = 0.3 m). The plate is coated with a material for which the maximum value of the coefficient of adhesion is 0.8. The author interprets this figure as the value of the coefficient of adhesion for a tyre slip velocity close to zero. For the positive kick plate displacement s_{yp}, formula (1) gives t_{yp} = 0.2 s. Therefore, the maximum value of the coefficient of adhesion was assumed as 0.5. The road, skid pad, and kick plate surfaces were assumed as being horizontal and even.

The simulation tests were carried out for seven values of the vehicle drive speed V, i.e. 20, 30, 40, 50, 60, 70, and 80 km/h. The relationship between the vehicle velocity v [m/s] in formulas (2) and (3) and the vehicle driving speed V [km/h] is defined by the generally known equation ($v = V$ / 3.6).

It was assumed that at the beginning of the test, the vehicle moved rectilinearly with a constant speed V, in parallel to axis Ox of the coordinate system fixed to the road (Fig. 4), and the vehicle travel path went through the centre of the moving part of the kick plate. The test was carried out according to a typical "open" test procedure, i.e. with no feedback in the driver-vehicle-environment-driver vehicle handling system [9]. During the test, the driver does not react, i.e. he/she does not change the steering wheel turning angle or the accelerator pedal position and does not press the brake pedal or the clutch pedal during the test. The driver assistance systems (ABS, ASR, and ESP) are deactivated (off), which often takes place when drivers undergo training at a higher (than basic) level.

6. Example results of the simulation in the time domain

Figs. 5, 6, and 7 show results of simulation calculations of selected quantities as functions of time for a vehicle speed value of V = 60 km/h. The time scale has been modified for the time value t = 0 to represent the instant when the front vehicle wheels are leaving the kick plate (the rear wheels are then on the kick plate, according to the parameters of the vehicle under test and the kick plate); at this instant, the lateral movement of the kick plate begins.

The following values have been presented in Fig. 5:

- lateral displacement of the centre of vehicle mass, y_{Oc} (denoted by "yOC" in the graph);
- vehicle yaw (heading) angle ψ_C ("psiC" in the graph);
- angular velocity of vehicle yaw $d_{\psi C}/dt$ ("psiCp" in the graph);
- lateral acceleration of the vehicle $a_{\eta h}$ ("aetah" in the graph).

In Fig. 6, the following values have been presented:

- moment of forces on the steering wheel EMK;
- sum of the lateral forces on the rear wheels F_{boczt} ("Fboczt" in the graph);
- necessary effective power of the kick plate driving system P_{boczt} ("Pboczt" in the graph); it is calculated as a product of F_{boczt} and kick plate velocity v_{yp};
- indicator of contact of the k^{th} wheel (k = 1, ..., 4) with the kick plate (indicator of the presence of the k^{th} wheel on the kick plate) INAP(k) ("0" means that the k^{th} wheel is out of the kick plate and "1" means that the k^{th} wheel is on the kick plate).

Fig. 7 shows time histories of the normal reactions N_k of road wheels ("ENC(k)" in the graph), where k = 1, ..., 4 indicates individual wheels.

The driver senses a disturbance to the vehicle motion through the following quantities:

- lateral vehicle acceleration $a_{\eta h}$ and angular velocity of vehicle yaw $d_{\psi C}/dt$ (these define the force of inertia acting on driver's body and detected by the balance organs in driver's inner ear and by sensory receptors in his/her limbs and trunk);
- lateral displacement of the centre of vehicle mass y_{Oc} and vehicle yaw (heading) angle ψ_C, detected by the sense of sight;
- moment of forces on the steering wheel EMK, detected by hands (more precisely: by sensory receptors in the upper limbs.

It is the time histories and extremums of these quantities that have a decisive impact on driver's reactions.

The results presented in Figs. 5 and 6 show the great importance of the disturbance to vehicle motion applied in the form of a lateral displacement of the kick plate. Within four seconds from the disturbance, the lateral vehicle displacement (in terms of absolute values) exceeds 15 m, the yaw angle stabilize at a level of more than 0.3 rad, the angular velocity of vehicle yaw reaches an extremum at a level of 0.45 rad/s, and the maximum of the lateral acceleration is 4.5 m/s^2. The moment of forces on the steering wheel (calculated from the road wheel stabilizing moments and from the steering linkage and steering gear ratios, i.e. without the effect of operation of the power steering system being taken into account) exceeds 10 Nm. The necessary effective power of the kick plate driving system reaches a value of about 5 500 W. Significant diversification can be observed in the normal road reactions on the vehicle wheels (Fig. 7). These results show a strong impact of the disturbance applied on the motion and dynamics of the vehicle in which the driver being examined or trained is present.

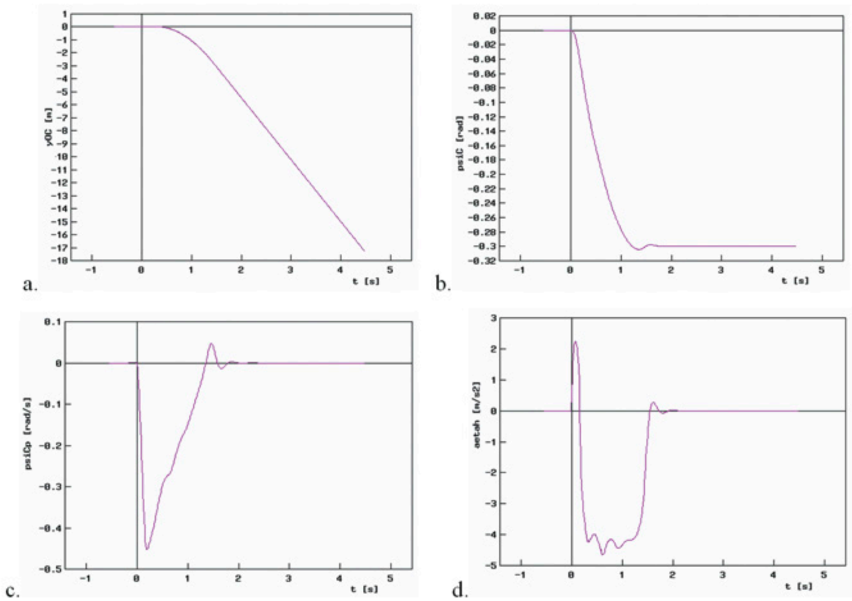

Fig. 5. Results of simulation calculations of selected quantities vs. time; V = 60 km/h:
a – lateral displacement of the centre of vehicle mass, y_{Oc} ("yOC" in the graph);
b – vehicle yaw (heading) angle ψ_C ("psiC" in the graph);
c – angular velocity of vehicle yaw $d_{\psi C}/dt$ ("psiCp" in the graph);
d – lateral acceleration of the vehicle $a_{\eta h}$ ("aetah" in the graph)

7. Criteria adopted to select the vehicle speed at the test with disturbing the vehicle motion on a kick plate

To select the vehicle speed at the test with disturbing the vehicle motion on a kick plate, the following nine criteria were adopted.

Criterion 1: Absolute value of the extremum of the lateral displacement y_{Oc} [m] of the centre of vehicle mass, i.e. extremum of "yOC" in the graph, within the first second from the disturbance.

Criterion 2: Absolute value of the extremum of the vehicle body solid yaw (heading) angle

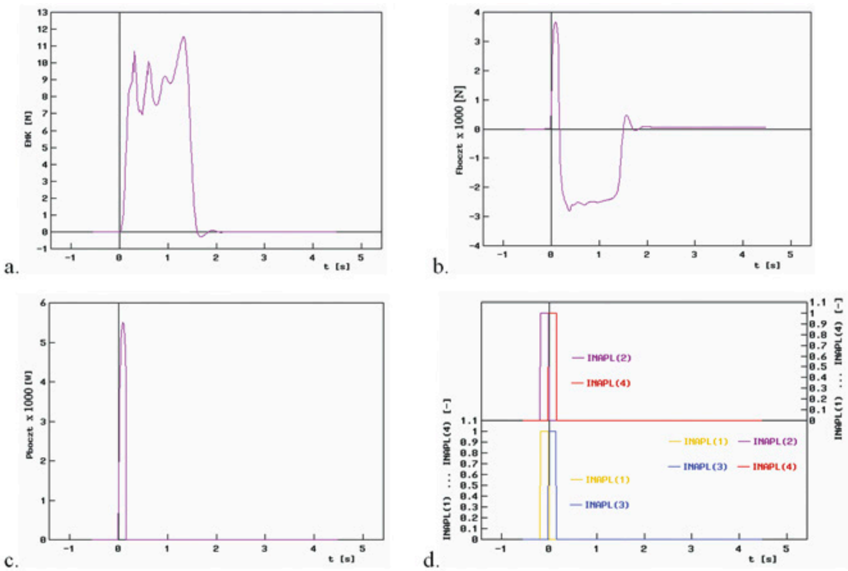

Fig. 6. Results of simulation calculations of selected quantities vs. time; V = 60 km/h:
a – moment of forces on the steering wheel EMK;
b – sum of lateral forces on the rear wheels F_{boczt} ("Fboczt" in the graph);
c – power related to the sum of lateral forces on the rear wheels P_{boczt} ("Pboczt" in the graph);
d – indicator of the presence of the k^{th} wheel on the kick plate INAP(k)

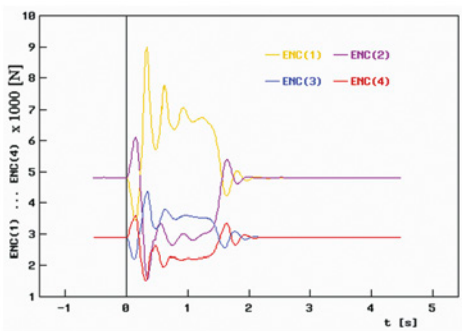

Fig. 7. Results of simulation calculations of selected quantities vs. time; V = 60 km/h.
Normal reactions N_k of road wheels ("ENC(k)" in the graph), where k = 1, ..., 4 indicates individual wheels:
1 – front left wheel; 2 – front right wheel; 3 – rear left wheel; 4 – rear right wheel

ψ_C [rad], i.e. extremum of "psiC" in the graph, within the first second from the disturbance.

Criterion 3: Absolute value of the extremum of the angular velocity of vehicle yaw $d_{\psi C}/dt$ [rad/s], i.e. extremum of "psiCp" in the graph, within the first second from the disturbance.

Criterion 4: Absolute value of the extremum of the lateral acceleration of the vehicle $a_{\eta h}$ [m/s^2], i.e. extremum of "aetah" in the graph, within the first second from the disturbance.

Criterion 5: Absolute value of the extremum of the moment of forces on the steering wheel EMK [Nm], i.e. extremum of EMK in the graph, within the first second from the disturbance.

Criterion 6: Absolute value of the extremum of the sum of the lateral forces on the rear wheels F_{boczt} [N], i.e. extremum of "Fboczt" in the graph, within the first second from the disturbance.

Criterion 7: Absolute value of the extremum of the power of resistance P_{boczt} [W] (put up by the sum of the lateral forces F_{boczt} on the rear wheels) on the surface of the moving part of the kick plate (the necessary effective power of the kick plate driving system), i.e. extremum of "Pboczt" in the graph, within the first second from the disturbance.

Criterion 8: Time [s] of the rear vehicle wheels being in contact with the kick plate (i.e. the time during which the rear vehicle wheels are present on the kick plate); this time is separately determined for the left and right wheel.

Criterion 9: Time [s] during which the rear vehicle wheels are present on the kick plate while the kick plate performs its lateral movement; this time is separately determined for the left and right wheel.

In the criteria from 1 to 7, the absolute value of the extremum of a specific quantity, observed during the first second from the disturbance, is taken into account. This is a consequence of an assumption made that during the typical driver reaction time, estimated at 1 s [1], the driver is unable to undertake any action (i.e. to operate any of the vehicle controls) that might change the state of vehicle motion. The driver's reactions can only affect the vehicle motion during the next seconds, and only these reactions may be taken as a basis to assess the driver at the examination or training carried out.

8. Results of the calculations carried out to select the vehicle speed at the test with disturbing the vehicle motion on a kick plate

Figs. 8 and 9 show results of simulation calculations of the quantities used as criteria 1–9.

In criteria 1–6, higher absolute values of specific quantities, i.e. y_{Oc}, ψ_C, $d_{\psi C}/dt$, $a_{\eta h}$, EMK and F_{boczt}, translate into better assessment of the disturbing system and vehicle speeds.

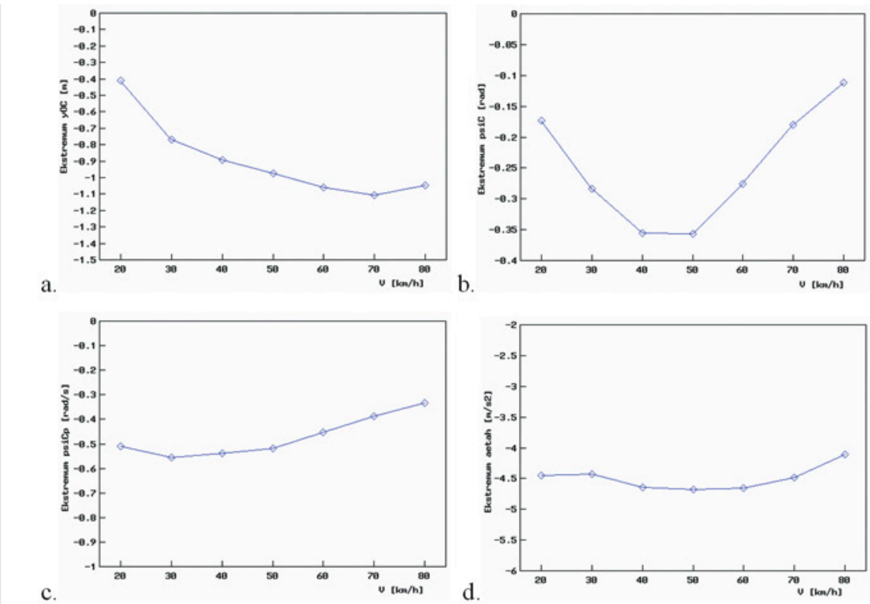

Fig. 8. Results of simulation calculations of the quantities used as individual criteria: a – criterion 1; b – criterion 2; c – criterion 3; d – criterion 4

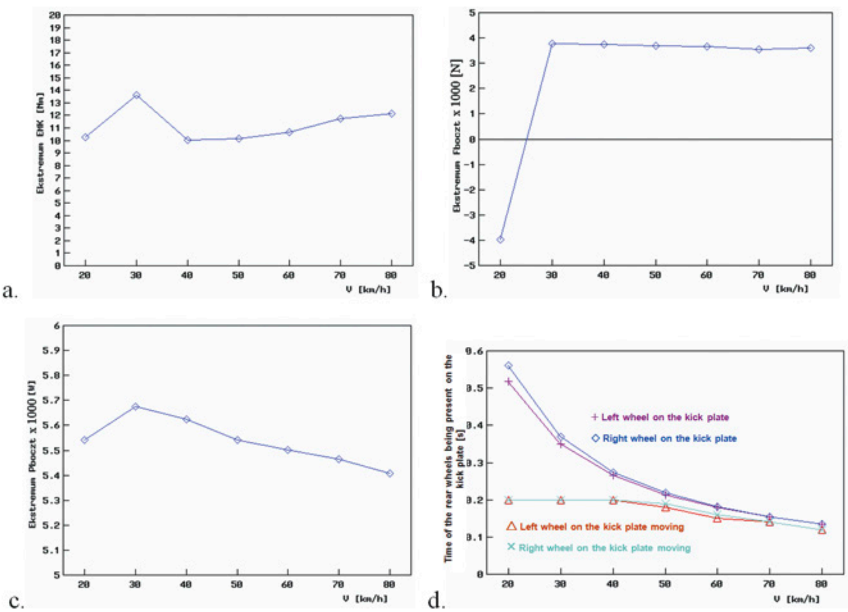

Fig. 9. Results of simulation calculations of the quantities used as individual criteria: a – criterion 5; b – criterion 6; c – criterion 7; d – criteria 8 and 9

In criterion 7, lower absolute values of P_{boczt} translate into lower requirements for the kick plate driving system.

In criterion 9, it is important that the time of the rear wheels being present on the kick plate should be close to the time of this plate being in motion. High vehicle speeds reduce this time below the value of t_{yp} (in the case of the kick plate design adopted for these tests, t_{yp} = 0.2 s), because this time is dictated by the vehicle wheelbase (l = 2.655 m).

Based on the calculation results presented in Figs. 8 and 9, the vehicle speed ranges preferred in consideration of the above selection criteria were defined.

For individual criteria, the preferable speed ranges were found to be as follows:

For criterion 1, 50–80 km/h;

For criterion 2, 40–50 km/h;

For criterion 3, 20–50 km/h;

For criterion 4, 40–60 km/h;

For criterion 5, 20–80 km/h;

For criterion 6, 20–80 km/h;

For criterion 7, 20 km/h and 50–80 km/h;

For criterion 8, 20–50 km/h, because within this range, the criterion value is ≥ 0.2 s;

For criterion 9, 20–40 km/h, because within this range, the criterion value is equal to 0.2 s.

When all the above sets were taken together and an intersection was sought that could be considered common to the greatest possible extent, **a speed range of 40–50 km/h was obtained**, because not more than two criteria were not met for it (criteria 1 and 7 for the speed of 40 km/h and criterion 9 for the speed of 50 km/h).

The result of this assessment may also be presented in a tabular form. Table 1 shows the preferences of the judging person (the author of this paper in this case). It visualizes the result of the process of searching for a vehicle speed range where the assessment criteria adopted would be met to the greatest extent. Number 0 (zero) has been assigned to the vehicle speed value for which the result of measurement of a quantity taken as a specific criterion was found unsatisfactory; on the other hand, number 1 (one) has been assigned when the measurement result met judging person's expectations. In the bottom row of the table ("Score"), the assessment results for all the criteria have been summed up. The highest scores have been achieved at the speeds of 40 and 50 km/h mentioned above.

Fig. 10 illustrates the result of the process of searching for a vehicle speed range where the assessment criteria adopted would be met to the greatest extent.

9. Recapitulation

At Driver Improvement Centres, tests are carried out during which, before the vehicle under test enters a skid pad characterized by reduced tyre-to-road adhesion, the vehicle motion

Table 1. Table of preferences of the judging person (the author of this paper)

Criterion	Vehicle speed during the test V [km/h]						
	20	30	40	50	60	70	80
1	0	0	0	1	1	1	1
2	0	0	1	1	0	0	0
3	1	1	1	1	0	0	0
4	0	0	1	1	1	0	0
5	1	1	1	1	1	1	1
6	1	1	1	1	1	1	1
7	1	0	0	1	1	1	1
8	1	1	1	1	0	0	0
9	1	1	1	0	0	0	0
Score:	**6**	**5**	**7**	**8**	**5**	**4**	**4**

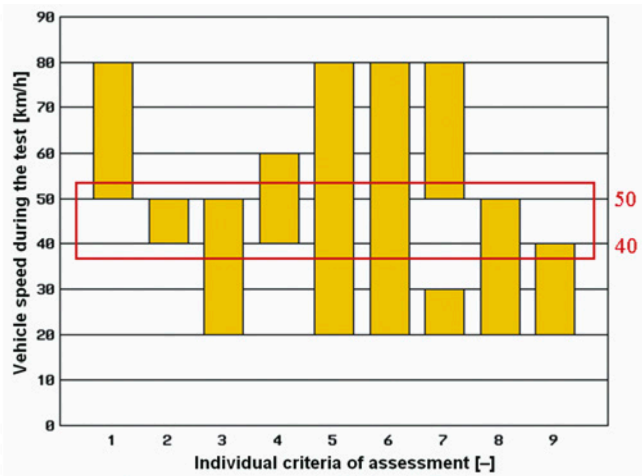

Fig. 10. Illustration of the result of the process of searching for a vehicle speed range where the assessment criteria adopted would be met to the greatest extent

is disturbed by means of a dynamic "kick plate" and the driver is thus forced to undertake defensive manoeuvres. In this paper, a model of motion and dynamics of a motor vehicle as well as a method of practical implementation of such a disturbance have been presented. The simulation results, showing the scale of the disturbance to vehicle motion to which the driver being trained must respond, have also been included. The methodology and results of choosing the speed with which the vehicle is to be driven during the test have been presented as well.

The simulations carried out with the use of an advanced and experimentally verified mathematical model of motion of a passenger car make it possible to determine critical parameters of the kick plate design, such as the force required to disturb the vehicle motion and the effective power of the system that is to cause a jerk of the kick plate, as well as the optimum initial vehicle speed during a test, depending on the assessment criterion adopted. Only the vehicle behaviour is assessed, without taking into account the impact of driver's actions on it. Therefore, the results presented will only be applicable to the period corresponding to the driver reaction time.

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