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AN INFLUENCE OF THE INTERNAL COMBUSTION ENGINE CHARACTERISTICS UPON THE TRAFFIC SAFETY IN THE REGIMES OF OVERTAKING [1]

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UDC: 621.43:656.052.48]:656.08

Abstract

The overtaking vehicles' manoeuvre is a very often cause of traffic accidents resulting in serious consequences. The influence of the engine as a vehicle's drive unit is rarely evaluated within the analyses of the causes leading to those accidents. On the other hand, a necessary condition for overtaking which is to be carried out in a safe manner is given through the vehicle's motion due to the specified acceleration values which primarily depend on the characteristics of engines and transmission. Therefore, what has been elaborated in the work itself is a model including the fundamental regularities of vehicles' motion in the regimes of overtaking, complete with the characteristics of engines, transmission and vehicles. The emphasis of the analysis has been placed on the engine as a drive unit, owing to which it has been necessary to develop adequate non – steady models that represent the only models able to define the characteristics of engines in the regimes of overtaking.

Modeling has been performed by using the *Simulink* and *SimDriveline* program from the *MathLab* program package. The developed model provides a complex analysis of all the factors influencing the overtaking manoeuvre: a driver, the engine, the transmission, a vehicle, positions and speeds of the vehicles involved in the overtaking process.

Key words: traffic safety, overtaking, drive unit, modeling.

UTICAJ KARAKTERISTIKA MOTORA SUS NA BEZBEDNOST SAOBRAĆAJA U REŽIMIMA PRETICANJA

UDC: 621.43:656.052.48]:656.08

Rezime: Manevar preticanja vozila je veoma čest uzrok saobraćajnih nezgoda sa teškim posledicama. U analizama uzroka takvih nezgoda retko se ocenjuje uticaj motora kao pogonskog agregata vozila. S druge strane, za bezbedno izvođenje preticanja neophodno je kretanje vozila određenim vrednostima ubrzanja koja u prvom redu zavise od karakteristika

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motora i transmisije. Otuda je u radu razvijen model koji obuhvata osnovne zakonitosti kretanja vozila u toku preticanja, karakteristike motora, transmisije i vozila. Težište analize je bilo na motoru kao pogonskom agregatu, zbog čega je bilo potrebno razviti odgovarajuće nestacionarne modele koji jedino mogu da definišu karakteristike motora u režimu preticanja vozila.

Modeliranje je vršeno uz korišćenje programa *Simulink* i *SimDriveline* iz programskog paketa *MathLab*. Razvijeni model omogućava kompleksnu analizu svih uticajnih faktora na manevar preticanja: vozač, motor, transmisija, vozilo, položaji i brzine vozila koja ušestvuju u preticanju.

Ključne reči: bezbednost saobraćaja, preticanje, pogonski agregat, modeliranje.

AN INFLUENCE OF THE INTERNAL COMBUSTION ENGINE CHARACTERISTICS UPON THE TRAFFIC SAFETY IN THE REGIMES OF OVERTAKING

Dragoljub Radonjić*[1], *Aleksandra Janković

UDC: 621.43:656.052.48]:656.08

1. INTRODUCTION

The intensive progress of traffic, as a logical consequence of the technological progress of the society in general, produces series of undesired effects upon the environment: the increase in the number of traffic accidents with fatal consequences, the pollution of the environment as a result of the exhaust gases of engines, the noise, non – recycled materials of parts of transport means, complete with a transformation of natural surfaces into asphalt and concrete surfaces of traffic roads, parking space and the like. As regards the traffic safety, a particular influence is shown by: the increase in the number of transport means (vehicles), the improvement of their performances (maximum speed, acceleration), and also, by the inadequate development of the road network based on the traffic capacity and the way of traffic regulation. Considering the fact that the influential factors in this area are connected to the man – vehicle – environment system, what is going to be analyzed in this paper is the influences of vehicles, that is, of their drive units, although the other two factors cannot be completely excluded because of the evident interactions.

A driver performs the overtaking manoeuvre on the basis of his/her own evaluation of the position and speed of not only his/her own vehicle but of the other vehicles passing by as well, incessantly endeavouring to drive the vehicle along the optimal distance with the maximum acceleration, so that he could perform the operation itself in the shortest amount of time. The vehicle acceleration is defined through the tractive balance equation and it represents the most significant parameter of the vehicle longitudinal dynamics [4]:

$$a = \frac{1}{m\delta}\left(F_O - \sum R\right)$$

where: m represents a vehicle mass, δ – the coefficient of a participation of a vehicle's rotating masses, F_o – a driving (propelling) force, $\sum R$ – the sum of the tractive resistance.

A driving force of a vehicle's driving wheels is defined through the characteristics of the drive unit (T_e) and a vehicle transmission ratio ($i_m i_o$) according to the following equation [4]:

$$F_O = \frac{T_e i_m i_O \eta}{r_d}$$

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The following symbols are used in the previously given equation : T_e – the torque, i_m and i_o – the transmission ratio and the rear axle ratio, η – the transmission efficiency and r_d –the rolling radius. The foregoing equations practically define the influence of the drive unit's characteristics (in the shape of the torque's curve) upon the vehicle's acceleration values, and therefore, upon the overtaking vehicle's manoeuvre as well.

2. GENERAL CHARACTERISTICS OF OVERTAKING

Overtaking practically represents going round a vehicle along the same carriageway at a certain speed. Considering the fact that this kind of a manoeuvre entails an alternation of the carriageway, the risk of the possible crash with other vehicles moving along the carriageway where this overtaking is taking place – is significantly being increased. Frontal or back impacts usually occur in those situations and their consequences are the most serious ones.

Characteristics of the overtaking process differ depending on whether it is being done along the roads with one – way or two – way traffic. These two cases are schematically presented in the Figure 1.

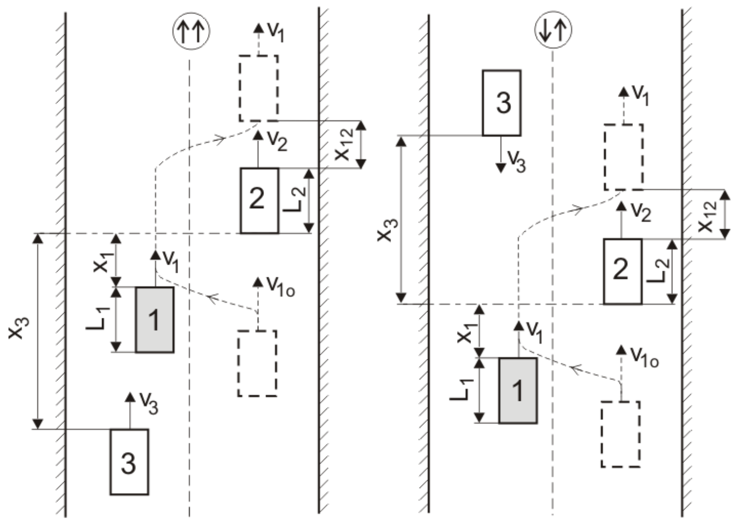

Figure 1.

According to the chart given in the Figure 1, the vehicle 1 is overtaking the vehicle 2, and during this process it is supposed to avoid a possible crash into the vehicle 3 moving in the same direction, in the first case, and in the second case, it is moving in the opposite direction. The vehicles' speed values are the following: v_1, v_2 i v_3, and the distances between the foregoing vehicles are defined by the following values: of the x_1 and x_3 frontal areas of the vehicles 1 and 3 of the rear side of the vehicle 2. The length values of the vehicles 1 and 2 are: L_1 and L_2. The overtaking process is being terminated at the moment when the rear of the vehicle 1 occurs in front of the front part of the vehicle 2.

Monitoring the positions of particular vehicles during their motion can be realized by displaying their distances which have been illustrated through the diagrams presented in the Figures 2 and 3. Because of the assumption relating to vehicles going at a constant speed at the initial moment, the law referring to the alternation of a distance is linear. The characteristic position presented in the diagram is a point *b* which represents the moment (time t_b), when the frontal areas of the vehicles 2 and 3 are found going in the same direction (S_2=S_3=S_b). It is evident that the vehicle 1 is supposed to terminate the overtaking process before that particular moment. Variants in relation to the initial speed of a vehicle that can have the following values: $v_{1o}= v_2$ i $v_{1o}> v_2$ _ are also presented in these diagrams. In the first case, overtaking is performed after a vehicle has previously been going in a weaving lane and it requires a vehicle to be accelerated. In the second case, if there is enough difference between the vehicles' speeds: $v_1 – v_2$, the overtaking process can be performed even at a constant speed v_1; on the contrary the vehicle has to be accelerated. The alternation of a distance of the vehicle 1 during the acceleration has been displayed in the diagrams by using a curve S_1*(accel.)*. If the overtaking process is to be performed in a safe manner, what is necessary is to realize a vehicle's acceleration values according to which the distance of the vehicle 1 at the moment t_b is: $S_1(t_b) \geq S_b$, which represents the basic criterion taken into account within the analysis of factors influencing the overtakaking process itself.

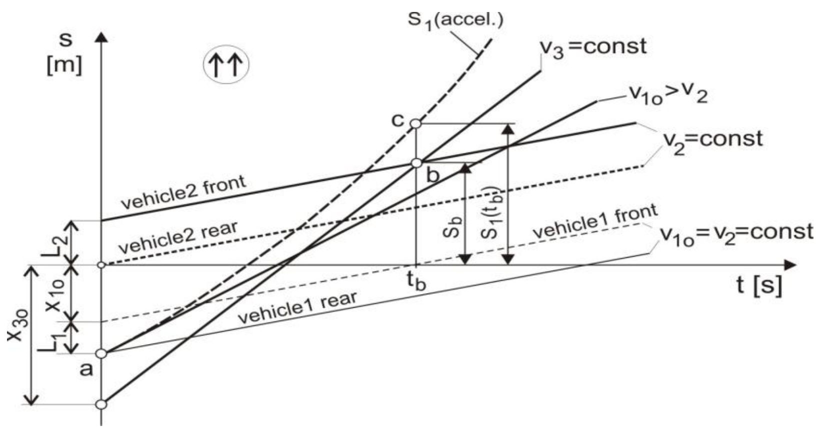

Figure 2.

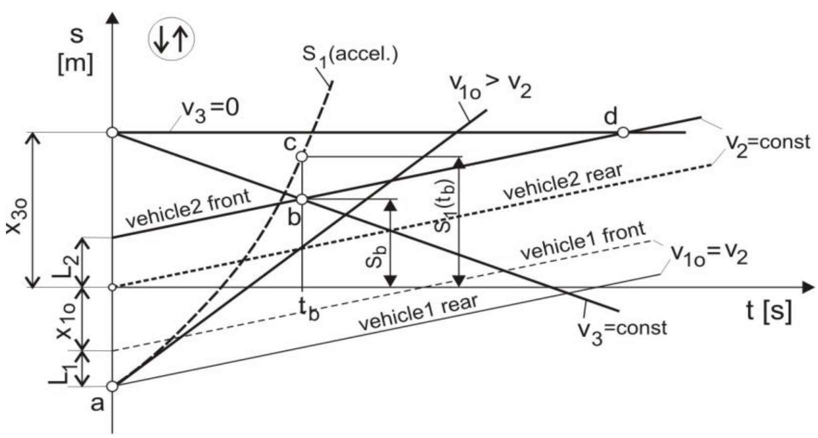

Figure 3.

While defining the simulation model the following assumptions have been made:

- what is being neglected is the carriageway alternation manoeuvre of the vehicle 1 – at the initial moment the vehicle 1 is going along the carriageway used for overtaking,
- it is not predicted to use the braking system with the purpose of changing one's mind as regards overtaking the vehicle 1 or the speed alternation of the vehicles 2 and 3, the values of which are always constant,
- what is used in this model is a minimum distance x_{1o} which is equal to the safe following distance and its value according to [4] is: $x_{1o}=(v_1^{\,2}- v_2^{\,2})/2g\cdot 3.6^2(\varphi+f)$, where: g represents the acceleration of gravity, φ – the coefficient of adhesion, f – the rolling resistance force coefficient. The following value: $x_{1o}=10\ m$ is taken in the case when: $v_1 = v_2$.

3. AN ENGINE MODEL

Engines as drive units of motor vehicles operate in extremely varying modes of operation. Existing (drive) engine characteristics are defined and determined in the steady (steady–state) modes of operation and the same characteristics cannot be used for an examination of vehicles' overtaking manoeuvre which is a typical non–steady process. Therefore, what is required is to form an engine model which is to generate the fundamental drive values (power and torque) in the rapidly varying modes of operation, such as a vehicle acceleration during the overtaking process. While establishing such kinds of models, it is possible to use the following two approaches:

- a model based on the calculation of the engine cycle in varying modes of operation,
- a model established due to the use of empirical and semiempirical formulae.

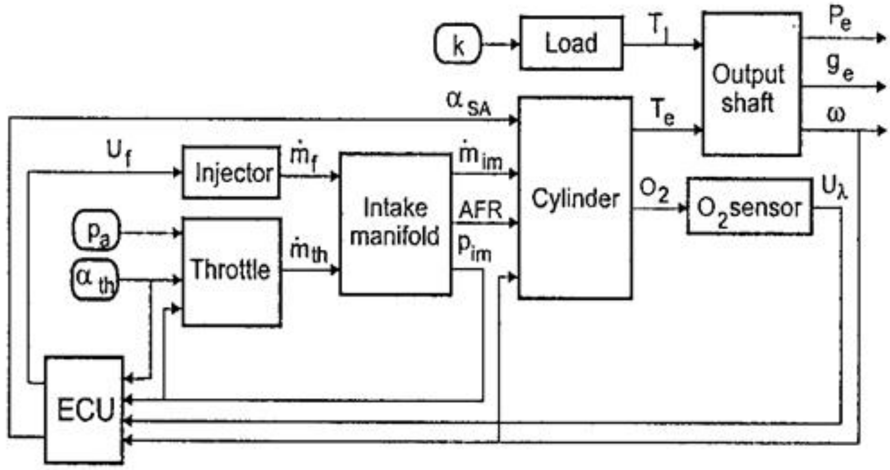

Figure 4.

The first method entails a calculation of an actual work cycle according to the block diagram presented in the Figure 4, [2], taking into consideration actual processes of an alternation of working substance and fuel combustion. The fundamental control variables are: a position of the throttle α_{th} and the load intensity k. The results of a simulation performed while using such a model are presented in the Figure 5, [2].

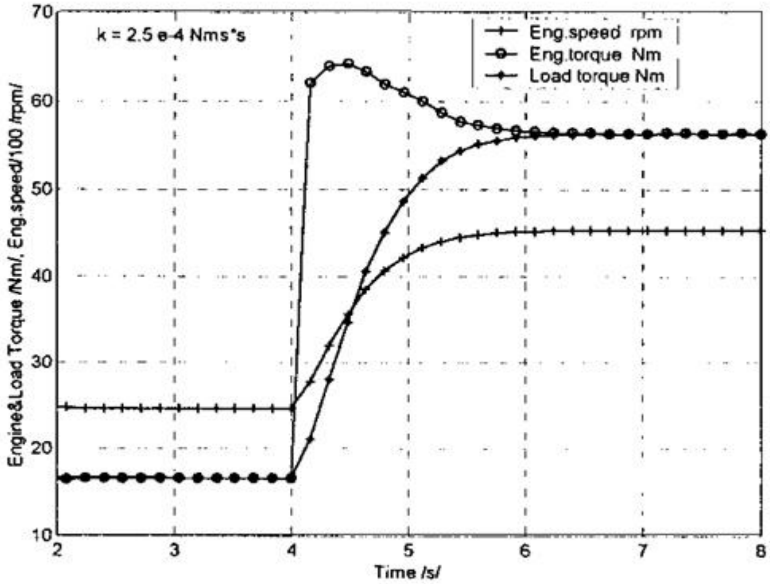

Figure 5 [2.]

The alternations of output values: the torque, load and a number of revolutions have been obtained during a sudden (step-function) opening of the throttle from 20% to 60% and the load factor: $k=2.5e\text{-}4\ Nms^2$.

The second method is based on the use of empirical power and torque formulae. One of the often used formulae is given through the third degree polynomial (1) showing power dependance P_e on the angular speed of the crankshaft ω. By using the well–known connection between the engine power and torque T_e ,(2), an appropriate torque formula is obtained (3).

The values of the coefficients *a, b, c* and *d* given in these formulae are being defined based on the conditions which are valid for the characteristic points on the engine performance and torque curves presented in the Figure 6: the maximal engine torque (n_T ,T_{emax}); the maximal engine power (n_P ,P_{emax}); the moment value if the number of revolutions is n_P, ($T_e(n_P)$). These conditions are given within the following formulae: (4), (5) and (6).

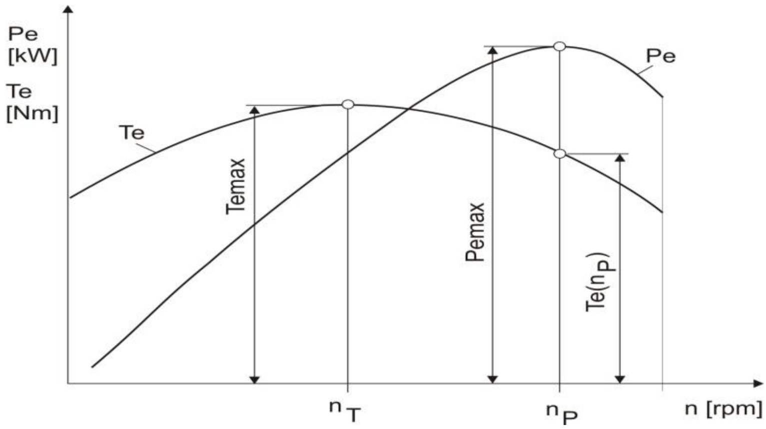

Figure 6.

$$P_e = a\omega^3 + b\omega^2 + c\omega + d \tag{1}$$

$$T_e = P_e / \omega \tag{2}$$

$$T_e = a\omega^2 + b\omega + c \tag{3}$$

$$\frac{dP_e}{d\omega} = \omega\frac{dT_e}{d\omega} + T_e \tag{4}$$

$$\frac{dP_e}{d\omega} = 0 \quad \Rightarrow \quad P_e = P_{e\max} \tag{5}$$

$$\frac{dT_e}{d\omega} = 0 \quad \Rightarrow \quad T_e = T_{e\max} \tag{6}$$

$$e_n = \frac{n_T}{n_P} \tag{7}$$

$$e_T = \frac{T_{e\max}}{T_e\left(n_P\right)} \tag{8}$$

$$d = P_{eo} \ge 0 \quad \Rightarrow \quad e_T \ge 1.5 - 0.5 e_n \qquad\qquad c = T_{eo} \ge 0 \quad \Rightarrow \quad e_n \le 0.75$$

The shape of the curves presented in the Figure 6 is often expressed through the compliance coefficient: according to the number of revolutions e_n, (7) and the moment e_T, (8); therefore, it is convenient to express the coefficients given in the empirical power and torque formulae in the function of these values.

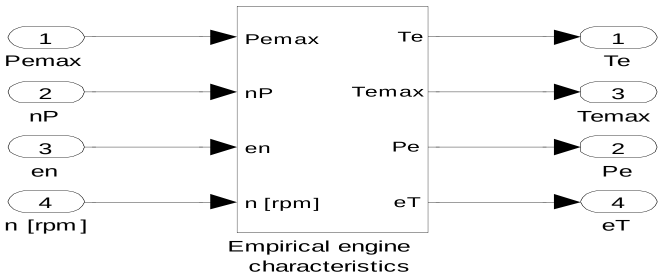

Figure 7.

Commencing with the conditions given by the foregoing formulae, the values of the coefficients *a, b, c* and *d* are defined and also, by using the *Simulink* program from the *MathLab* program package, the process of modeling the power and torque empirical formulae is performed. The model is depicted by the block presented in the Figure 7, in which it is shown how, based on the given inputs: P_{emax} ,n_P ,e_n and n, the output values: $T_e = f(n)$, T_{emax} , $P_e = f(n)$ and e_T are being defined, where n represents the number of revolutions of the crankschaft which has been adopted for the value of the independently variable.

What has been done with the purpose of checking the empirical model is a comparison of the characteristics obtained by its use (the full line drawn in the Figure 8) and the experimental results obtained after running inspection based on the engine of *1.4 l* – for the *Florida* vehicle (the dotted line). The appropriate curves are obtained during the full throttle. Deviations are minimum, as it can be seen from the diagram depicted in the Figure 8.

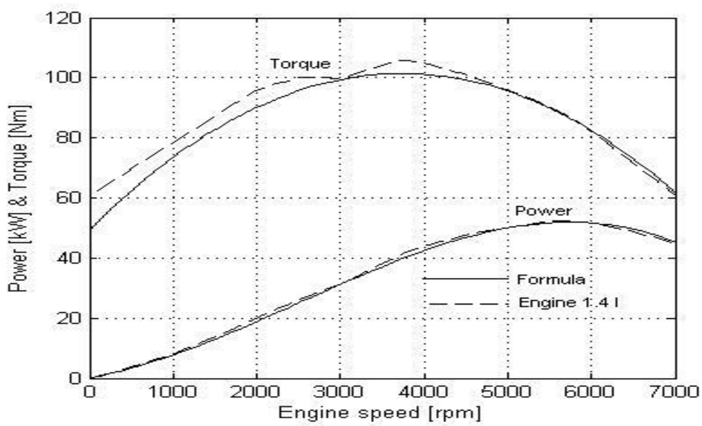

Figure 8.

By using the same model it has been possible to evaluate an influence of the compliance coefficient, according to the number of revolutions, upon the shape of the engine performance and torque curves. The curves presented in the Figure 9 are obtained for the engine with the following characteristics: $P_{emax}=60\ kW$, $n_P=6000\ o/min$ and $e_n=0.1-0.75$, with the maximum (100%) full throttle.

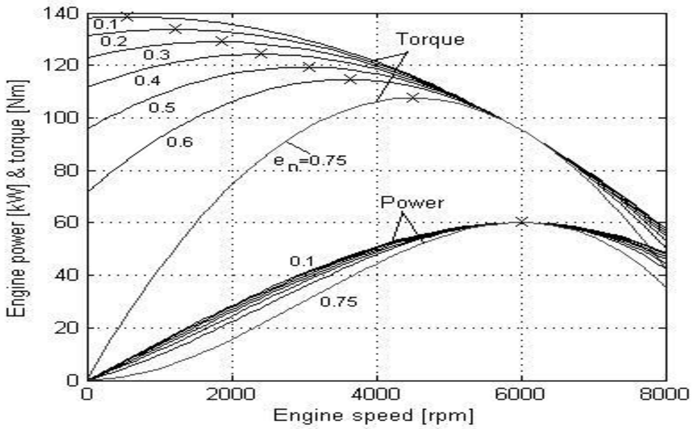

Figure 9.

The compliance coefficient's alternation e_n within the given boundaries influences the position and value of the maximum moment, complete with the shape of the engine performance and torque curves, as it is shown in the diagram presented in the Figure 9.

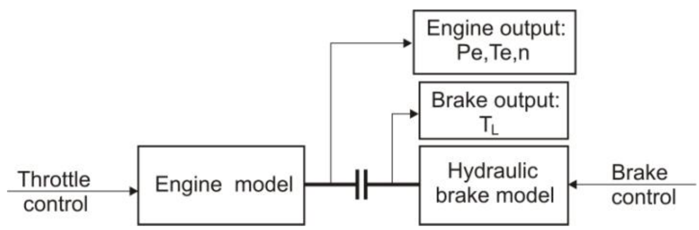

Figure 10.

Modeling of the engine non–steady characteristics is performed, in this case, by introducing a characteristic of the throttle position K_{th}, the value of which ranges within the boundaries: $0-1$, which corresponds to the angle of the throttle position: $\alpha_{th}=0-90^o$, that is, to the throttle openness: $0-100\%$:

$$[T_e(n)]_{unsteady}=K_{th}\,[T_e(n)]_{steady}$$

The control of the output characterisrics of this model has required the same model to be realized within the *Simulink* program according to the block diagram shown in the Figure10. Except for the engine model (the empirical one), the hydraulic brake model has also been installed with the purpose of the load torque simulation. The simulation results are displayed in the Figure 11, in which steady characteristics are also displayed, the ones corresponding to the throttle openness: *20*, *40*, *60*, *80* and *100%*, complete with the non–steady characteristic occuring during an alternation of the throttle position according to the step function from *20* to *60%*.

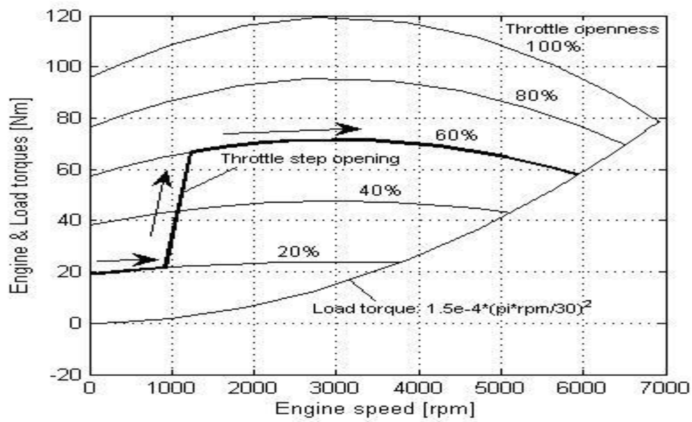

Figure 11.

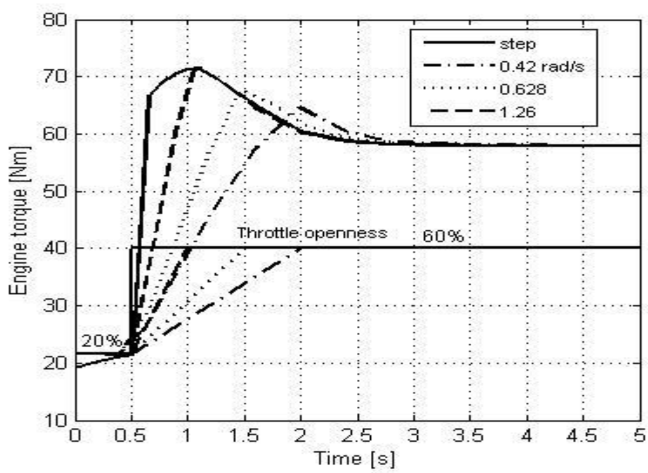

Figure 12.

Non–steady engine characteristics for various speeds of the throttle opening (intensities of effects upon the accelerator pedal) have been simulated by using the same model, commencing with the step function. The results are displayed by means of a diagram given in the Figure 12, and they are considerably in accordance with the results obtained by the use of the model displayed in the Figure 4, which are presented in the Figure 5.

4. VEHICLE MODEL

A model has been formed, by using the *SimDriveline* program, for the purpose of simulating the output values of vehicles (a vehicle distance *S*, vehicle speed *v* and vehicle acceleration *a*) in the regimes of overtaking, and its block diagram is displayed in the Figure 13.

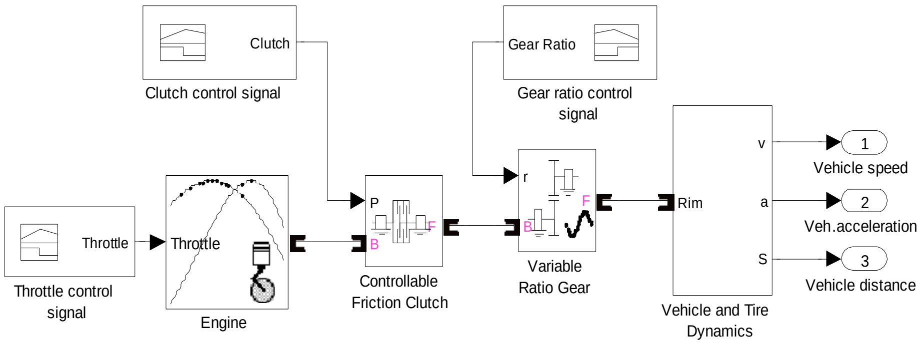

Figure 13.

The model's structure consists of the following: the engine model (which refers to the previously depicted empirical model) with the throttle control signal, the friction clutch with a control signal for its connection and disconnection, the steplessly variable ratio gear with the gear ratio control signal with the purpose of defining the law of the ratio gear alternation and a model of pneumatic tyres for vehicle simulations.

For the purpose of a verification of the output values, the model has been used for a calculation of performances of a *Florida 1.4 l* vehicle, with the input values characterized as it follows: the full throttle (100%), the clutch which is always connected (the time required for the ratio gear alternation is being neglected), the ratio gear alternation is being done according to the diagrams given in the Figure 14.

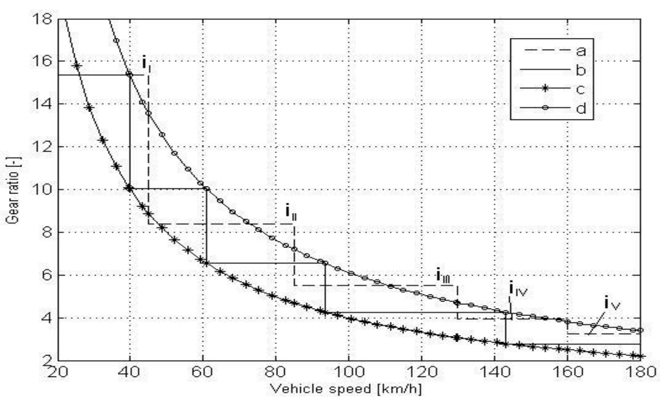

Figure 14.

Diagrams shown in the Figure 14 present the law of the ratio gear alternation of various types of a gear (a gearbox). The line *a* (the dotted one) corresponds to the step– shaped gearbox of the *Florida 1.4 l* vehicle, the line *b* corresponds to the step–shaped gearbox in which the ratio gear alternation is being done with T_{emax} and P_{emax}, the line *c* corresponds to the variable ratio gear which enables the engine running in the T_{emax} regime and the line *d* corresponds to the variable ratio gear which enables the engine running in the P_{emax} regime.

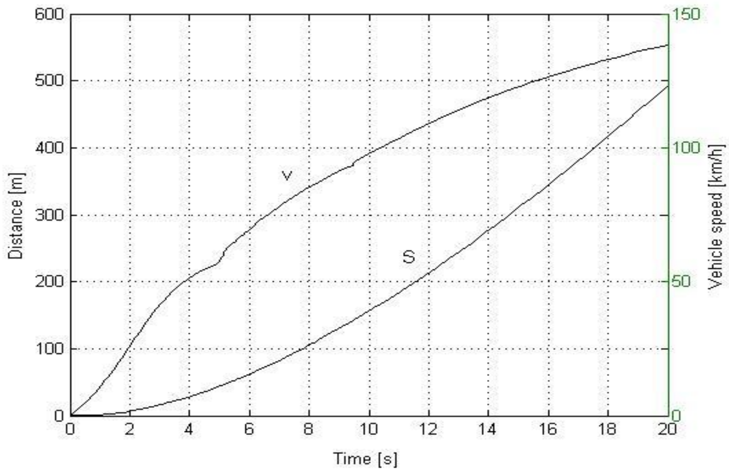

Figure 15.

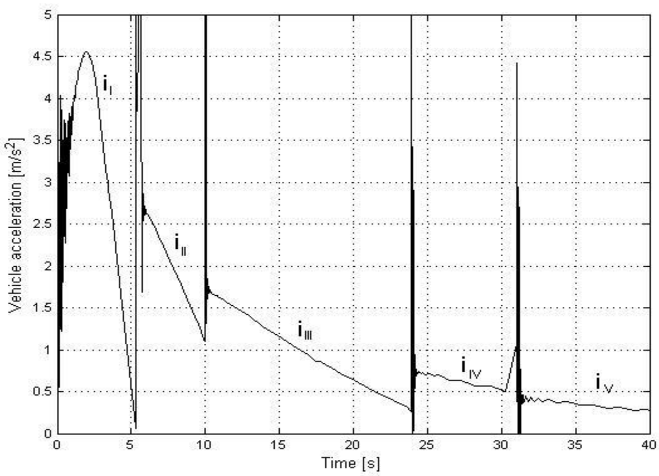

Figure 16.

Comparing the simulation results obtained for the presented laws of the ratio gear alternation, it can be discerned that the most convenient output parameters of the vehicle are obtained while using the law which is defined by the line *d* (P_{emax}), and then by the line *a*. The simulation results for the *Florida 1.4 l* vehicle with the step–shaped gearbox, a characteristic of which is displayed by the line *a* complete with the ratio gear alternation at maximum speeds in the particular ratio gears according to the recommendations given by the producer – are presented in the Figures 15 and 16. The alternations of the vehicle distance *S* and speed *v* are given in the Figure 15, and the Figure 16 represents the vehicle acceleration *a* in the particular ratio gears realized during the acceleration process.

5. SIMULATION RESULTS

A complete model, developed in order to simulate the overtaking process of vehicles, is included in the block diagram given in the Figure 13 complete with the model which, being in accordance with the relations given in the Figures 1, 2 and 3, determines the time of the encounter t_b of the vehicles 2 and 3, complete with the appropriate distances up to that moment: S_b and $S_1(t_b)$. Also, it should be emphasized that the distance indicated in the Figures 2 and 3 with S_1 *(accel.)*, represents the output value from a vehicle model block given in the Figure 13.

What has been made is a research of an influence of the internal combustion engine characteristics upon the traffic safety in the regimes of overtaking, based on the example of a road with two – way traffic (the Figure 3), by using the depicted model presented with the input values characterized in the following manner: $L_1 = L_2 = 4\ m$; $x_{3o} = 150\ m$; $v_{1o} = 90\ km/h = const.$, $v_2 = 80\ km/h = const.$, $v_3 = 80\ km/h = const.$, the coefficient of adhesion: $\varphi = 0.75$ and the rolling resistance force coefficient: $f_o = 0.018$.

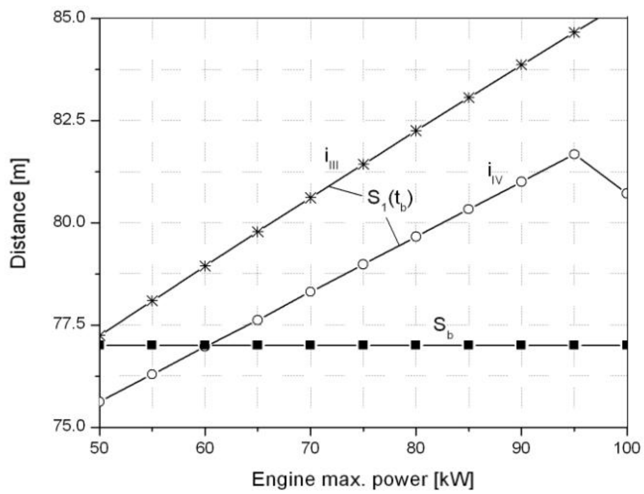

Figure 17.

The time of the encounter of the vehicles 2 and 3 and the appropriate distance, based on the predetermined input values, are characterized in the following manner:

$t_b = 3.285\ s, \qquad S_b = 77\ m.$

An influence of the maximal engine power has been presented by diagrams given in the Figure 17, in order to show the case of an overtaking manoeuvre performance within the third and fourth ratio gears. The character of these dependencies confirms the well–known fact related to the engine characterized by the greater maximal power enabling the same vehicle to achieve greater accelerations, owing to which the time of an overtaking performance is being shortened. Considering the previously introduced criterion for a safe overtaking performance, given within the relation $S_1(t_b)>S_b$, the simulation results presented in the Figure 17, are displaying the fact that in the fourth ratio gear – overtaking can be performed in a safe manner exclusively by means of the engine, the maximal power of which is greater than *60 kW*. In other words, it means that vehicles provided with engines producing less power have to perform the overtaking process throughout the third ratio gear or to commence the same process at the point when:

$x_{3o} > 150\ m.$

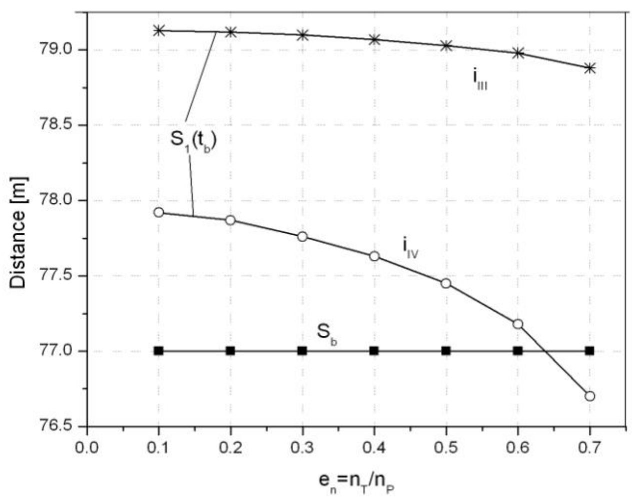

Figure18.

The diagram given in the Figure 18 displays an influence of the compliance coefficient e_n upon the overtaking manoeuvre. This influence reflects the character of the torque alternation depending on the value of the coefficient e_n which has been presented in the Figure 9. Namely, an increase in values of the coefficient e_n, leads to a decrease in values of the maximal engine torque within the empirical model.

An influence of the maximum throttle opening has been presented in the Figure 19. For the predetermined input values, a safe overtaking process can be performed only if the throttle openings are greater than 80%.

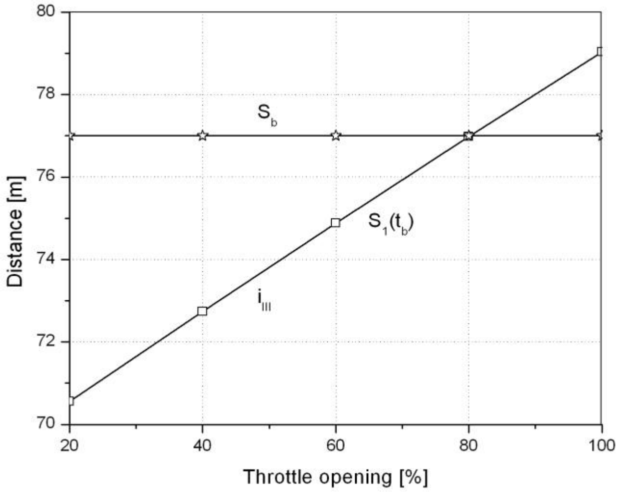

Figure 19.

For the purpose of a verification of the simulation results obtained by using the developed simulation model, characteristics of modern medium class passenger vehicles have been analyzed according to the criterion of maximum accelerations which can be achieved in the third ratio gear at speeds ranging from 60 to 100 km/h and in the fifth ratio gear at speeds ranging from 80 to 120 km/h. The results have been presented in diagrams given from the Figure 20 to the Figure 24. As it has previously been emphasized, vehicles with greater acceleration values have better performances regarding the overtaking manoeuvre process.

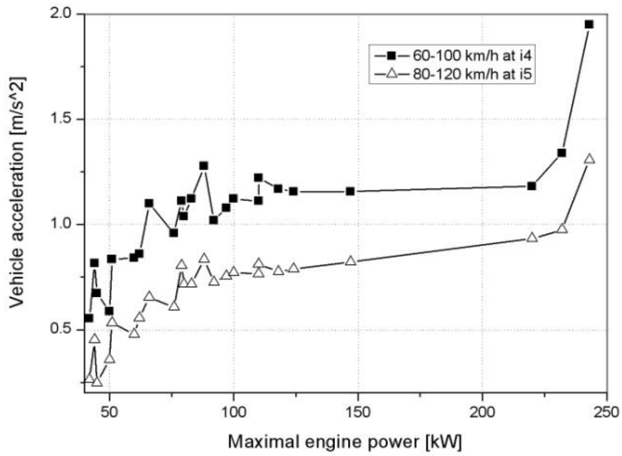

Figure 20.

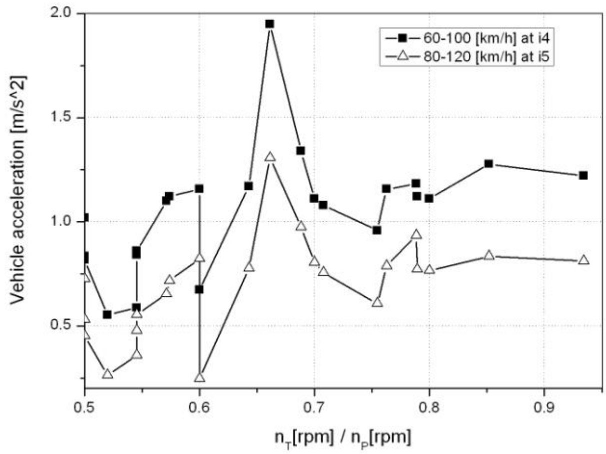

Figure 21.

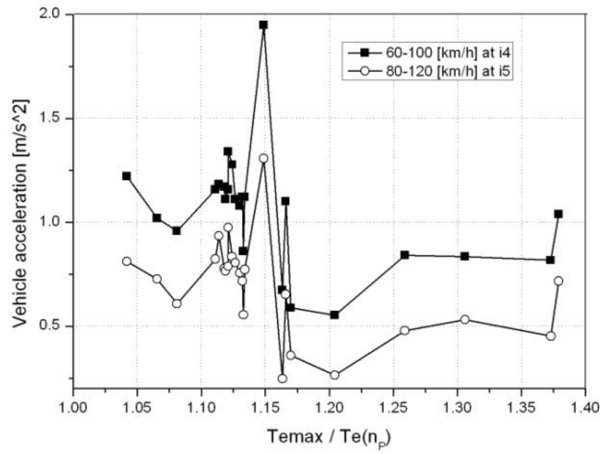

Figure 22.

Diagrams given in the Figures 20, 23 and 24, display a vehicle acceleration dependency on the maximal engine power, that is, on the maximal engine torque. These dependencies show the same direction of the influence just like the results obtained owing to the simulation models: an increase of the maximal engine power, that is, the maximal engine torque leads to the increase of acceleration values, and therefore, the time required for the overtaking manoeuvre performance is being shortened.

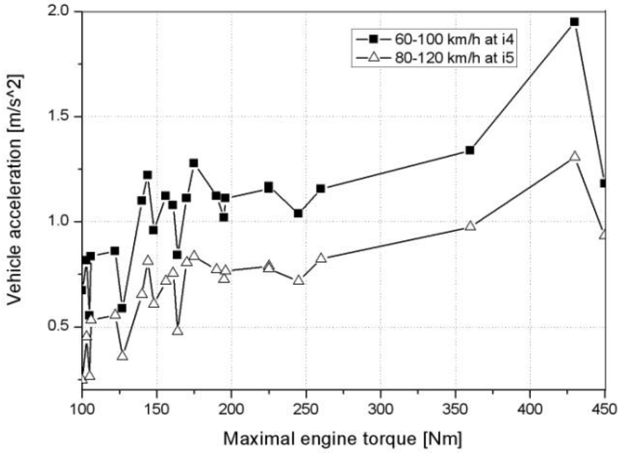

Figure 23.

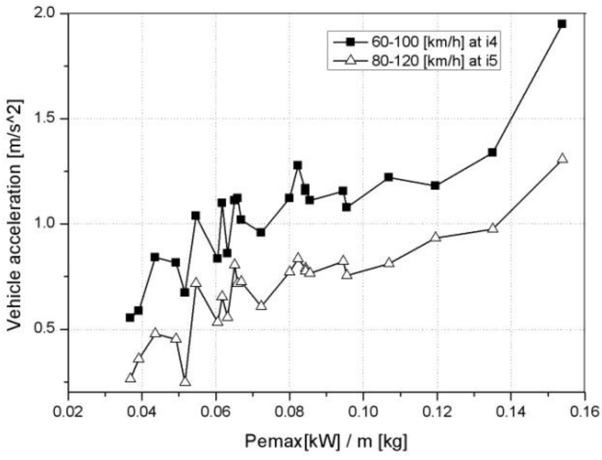

Figure 24.

The influence of the compliance coefficients displayed within diagrams given in the Figures 21 and 22 primarily depends on the maximal engine torque value which is equal to the given value of the coefficient itself. Therefore, this influence has a stochastic character. As regards modern engines used for passenger vehicles' drive units, and owing to the application of variable systems, the engine torque curve is characterized by the constant value of the maximum in a wide area of a number of revolutions (1000-5000 o/min), which renders the influence of the compliance coefficients invariant .

6. CONCLUSIONS

The developed model of the overtaking manoeuvre of vehicles enables an assessment of influences of a greater number of parameters, commencing with the engine characteristics, the intensity of its effect upon the throttle control linkage, the friction clutch connection process, transmission characteristics and vehicle characteristics. The engine influence as a vehicle's drive unit has been analyzed in detail by presenting a concrete case.

Obtained results confirm the fact that the engine characteristics are the ones having a crucial influence upon the safety of vehicles' overtaking manoeuvre performance. The fundamental parameter significant for the overtaking process is, by all means, a vehicle acceleration which primarily depends on the intensity of its effect upon the throttle control linkage. All of these influences have been analyzed by using the above presented model and interpreted by displaying appropriate diagrams.

Having a certain knowledge of the engine characteristics regarding the maximum acceleration values which can be achieved by means of the given vehicle – is significant particularly when the overtaking process is performed in cases when it is required to avoid a critical situation owing to the misjudgement of speeds and following distances. Except for the analysis of influences of the engine, transmission and vehicle's characteristics, the model indirectly enables the assessment of a driver's reaction regarding his/her evaluation of the following distances (x_{1o} ,x_{3o}) and speeds (v_2 ,v_3). By varying these values, the boundary valves can be established, the same ones owing to which the overtaking manoeuvre can be performed in a safe manner by using the given engine and vehicle.

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[1]VEHICLE STEERING CONTROL

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UDC: 621.43:681.515.2/.8

Abstract

In this paper, the problem of vehicle unstable motion related to desired path is considered. A model to study vehicle lateral dynamics with respect to steering control is developed. Three possibilities to solution of vehicle directional unstability are analysed, as follows, driver steering control, optimal controller application, combined cotrol - as driver steering action supported by a technical controller. But, only optimal controller application is further focused in this paper. In this sense a procedure to design a vehicle steering controller based on the optimal control theory is proposed. For input data a passenger car midlle class the structure and parameters of optimal controller are determined. Then, simulation researchs are conducted and some results of controled vehicle behaviour by acting specified disturbance are presented and discussed.

Key words: vehicle, unstability, driver, controller, simulation.

KONTROLA UPRAVLJANJA VOZILA

UDC: 621.43:681.515.2/.8

Rezime: U radu je razmotren problem nestabilnog kretanja vozila u odnosu na željenu putanju. Razvijen je model za proučavanje bočne dinamike vozila sa aspecta kontrole upravljanja. Analizirane su tri mogućnosti za rešavanje problema nestabilnosti pravca vozila i to, uz pomoć vozača, zatim uvođenjem optimalnog regulatora, i konačno kombinovanom kontrolom – dejstvo vozača podržano tehničkim regulatorom. U radu je dalje, razmotrena samo primena optimalnog regulatora. U tom smislu, predložena je prcedura za projektovanje optimalnog regulatora bazirana na teoriji optimalnog upravljanja. Za ulazne podatke putničkog automobila srednje klase određeni su struktura i parametri optimalnog regtulatora. Zatim su sprovedena simulacioana istraživanja i neki rezultati upravljanja ponašanjem vozila pri dejstvu specificiranih poremećaja su priloženi i discutovani.

Ključne reči: vozilo, nestabilnost, vozač, regulator, simulacija.

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VEHICLE STEERING CONTROL

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UDC: 621.43:681.515.2/.8

INTRODUCTION

The wheeled motor vehicle do not possesses own stability of lateral and angular motion along desired path. From this reason, some form of driver steering control or other controller must be included to vehicle stabilization and guidance. Many researchs have been made to study of driver behaviour by different control tasks in different driving conditions, [1], [12]. Numerous mathematical models developed to describe of driver control action are named as, ideal, quasi-linear, optimal, predictive, supervisory and so on. [2], [3], [4], [13], [14]. The term driver model is a synonym for a mathematical description driver control with models of vehicle dynamics in interaction with road and environment.

Driver models were developed, before all, in order to get information about how a vehicle and its design changes influence the handling quality and driver efforts, also to accident causations. Recently, theoretical and experimental researchs of driver – vehicle – environment interaction have great importance with aspect to above pointed out problems but with aspect design, optimisation and application automated vehicle control, [9]. According to above mentioned problems in this paper some phenomenon of vehicle unstability are studied and any solution to assessment vehicle steering performances are proposed, in order to continue our earlier researchs, [8], [15].

1. VEHICLE LATERAL DYNAMICS

The vehicle lateral dynamics can be modelled with different complexity, from a simple two – wheeled model to a numerous degree of freedom model. The complex vehicle model requires using of numerical integration which makes it difficult to form general conclusion. In many currently researches developed vehicle steering controllers for automated vehicles have focused on low lateral acceleration conditions, [9]. Under these conditions linear vehicle and tire models are suitable for the controllers developing and design. However, most emergency situation require complex vehicle model, which today can be successfully realized by means of advanced simulation methods, [10].

In this paper, a simplified model is used, which simulates vehicle motion over a flat and level road surface, when the forward speed is kept constant. The equations of motion are written in relation to vehicle lateral Δy, and heading deviation $\Delta\varepsilon$, from desired path, which represent absolute lateral displacement y, and heading angle ε, on the straight-line road, according to presentation in Fig. 1a, vehicle model and Fig. 1b, relevant inputs to vehicle steering control. As can see from implicit expresions (1) and (2), output variables, y and ε,

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are mutually coupled by means of the system matrices coefficients, a_{ji}, b_j. Input variable is steering wheel angle, denoted with, β. The initial equations (1) and (2) are transformed into state space form suitable to control problems solution, (3), (4), (5).

Equation (3), as matrix expresion, presents state space form of basic vehicle model in open- loop with state vector (4). On the other hand, matrix equation (5), describes closed loop system in state space form for alternative control defined by state vectors: (5.1) – driver steerig control, (5.2) – optimal controller application, (5.3) – combined control, driver action supproted by technical controller for given condition, so called vehicle stabilty augmentation systems, which modifies the driver steering command, as presented in Fig. 2, internal loop with input variables, y, ε, and output variable, β_2.

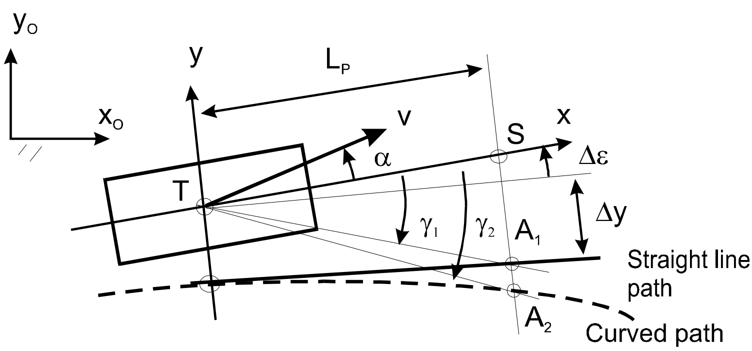


Figure 1a: *Model of vehicle lateral dynamics*

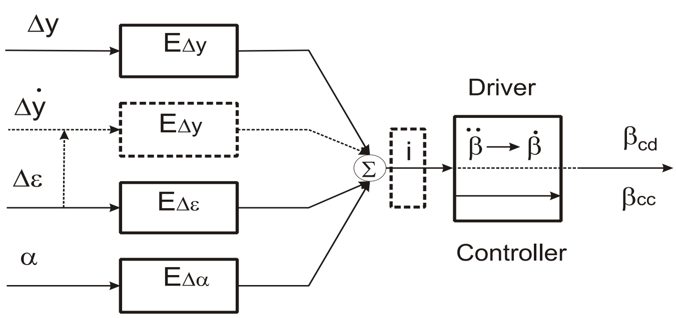


Figure 1b: *Real and estimated inputs to vehicle steering control, according to Fig. 1a, a/ driver, b/ optimal controller*

$$\ddot{y} = \ddot{y}(a_{22}\dot{y}, a_{23}\varepsilon, a_{24}\dot{\varepsilon}, b_2) \tag{1}$$

$$\ddot{\varepsilon} = \ddot{\varepsilon}(a_{42}\dot{y}, a_{43}\varepsilon, a_{44}\dot{\varepsilon}, b_4) \tag{2}$$

$$\dot{x}_1 = Ax_1 + Bu_1 \tag{3}$$

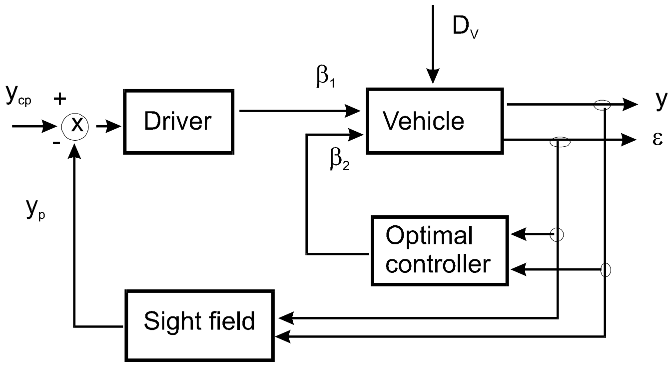


Figure 2: *Possible variants of vehicle steering control*

$$x_1 = [y, \dot{y}, \varepsilon, \dot{\varepsilon}] \tag{4}$$

$$u_1 = [\beta] \tag{5}$$

$$\dot{x}_c = Ax_c + Bu_c \tag{6}$$

$$x_c = [y, \dot{y}, \varepsilon, \dot{\varepsilon}, \beta(....)] \tag{7}$$

$$\beta_{cv} = \beta_{cv}(y, \varepsilon, \alpha, L) \tag{7.1}$$

$$\beta_{co} = \beta_{co}(y, \dot{y}, \varepsilon, \dot{\varepsilon}, ...) \tag{7.2}$$

$$\beta_{cc} = \beta_{cc}(\beta_{cv}, \beta_{co}) \tag{7.3}$$

$$u_c = [y_0(\varepsilon_0)] \tag{7.4}$$

The more attention in this paper is dedicated to design an optimal controller to vehicle steering control according to theoretical considerations given in chapter number 3, but with previous hypothetic coments about possible driver steering control strategies given in next chapter, 2.

2. DRIVER STEERING CONTROL STRATEGIES

Driver control action by a single – loop compensatory task can be described by means of differential equation derived from driver convencional quasi – linear model, [2], [5], [8], and presentation in Fig. 1b, as relationship between steering wheel movement, β and vehicle lateral deviation, Δy:

$$\ddot{\beta}[(\tau + T_N)T_i] + \dot{\beta}(\tau + T_N + T_i) + \beta = -KT_L\Delta\dot{y} - K\Delta y \qquad (8)$$

where, τ - driver time delay, T_N – driver neuromuscular system time lag, T_L, T_i – driver time lead and time lag equalization, respectively, K – driver gain factor.

Driver control strategy in a comprising multi – loop system, shows in Fig. 3, is based on the combined control of the lateral deviation, Δy, into outer loop, and heading deviation $\Delta\varepsilon$, into internal loop. So, driver compensate directly lateral deviation and indirectly angular deviation based on the modification of his mode control in outer loop.

By two level control strategies, [6], [7], [8], driver used visual cues of sight field derived from vehicle position relative to road, for compensatory action in equvalent single – loop, advanced in time, and parameters of reference path, as curvature, focused curve segments and so on , to guidance tasks. These three driver strategies, hier presented , can help by design optimal controller and by interpretation results in next chapter.

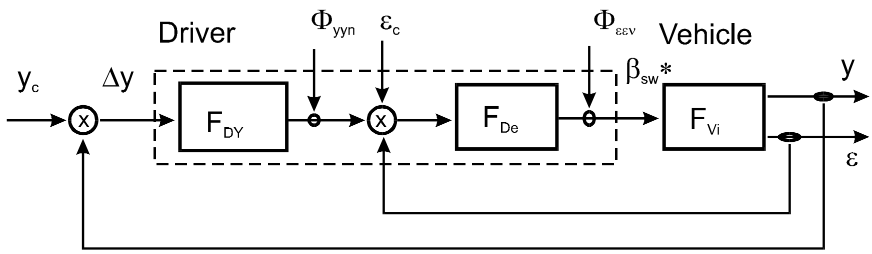


Figure 3: *Driver steering control strategy in multi-loop systems*

3. DESIGN OF VEHICLE STEERING CONTROLLER

The vehicle optimal controller has been designed using linear quadratic regulator method based on the optimal control theory, [11]. Therefore, some basic design phase and ideas of the theory should be presented. The dynamic equations of the closed loop system in Figure 4 are present in a state space form:

$$\dot{x} = Ax + Bu \qquad (9)$$

where, x = $[x_1\ x_2\ .....x_n]$, is the state vector in general form, u = $[u_1\ u_2\ ...u_m]$, the input vector determined by controller. The output vector, y, can be defined in different forms as a linear combination of the state variables:

$$y = Cx \qquad (10)$$

depending on the chosen form of matrix C. The optimal control concept is formulated as a problem to minimize a functional of general form:

$$J = \frac{1}{2} y^T (t_f) S y(t_f) + \frac{1}{2} \int_{t_o}^{t_f} \left[y^T (t) Q(t) y(t) + u^T (t) R(t) u(t) \right] dt \tag{11}$$

where Q, R, S are symmetric weighting matrices, u(t) input, y(t) output variables. With Hamilton – Jacobi equation

and by used

$$H\left[y(t), u(t), \lambda(t), t \right] = \frac{1}{2} y^T Q y + \frac{1}{2} u^T R u + \lambda^T A y + \lambda^T B u \tag{12}$$

maximum principle:

$$\begin{aligned} \frac{\partial H}{\partial u} &= 0 = R(t) u(t) + B^T (t) \lambda(t) \\ \frac{\partial H}{\partial y} &= -\dot{\lambda} = Q(t) y(t) + A^T (t) \lambda(t) \end{aligned} \tag{13}$$

at constraint condition:

$$\lambda(t_f) = \frac{\partial \theta}{\partial y(t_f)} = S y(t_f) \tag{14}$$

needed solution of (9) is :

$$u(t) = -R^{-1}(t) B^T (t) \lambda(t) \tag{15}$$

with presumed partial solution:

$$\lambda(t) = P(t) y(t) \tag{16}$$

where P(t) is the solution of Riccati matrix equation:

$$\dot{P} = -P(t) A(t) - A^T (t) P(t) + P(t) B(t) R^{-1}(t) B^T (t) P(t) - Q(t) \tag{17}$$

by constraint condition $P(t_f) = S$.

With P(t) solution from equation (13) can be synthesized optimal controller structure in time domain:

$$u(t) = K(t)y(t) = -R^{-1}(t)B^{T}(t)P(t)y(t) \tag{18}$$

In this paper optimization control problem is solved with respect to algorithm from equation (9) to equation (18) for defined control task variants in internal loop,on the Fig 2. The results of the Riccati matrix equation solutions are used of line in simulation procedure. Vehicle handling characteristics without control and with optimal controller was idetified in simulation procedure for input data a typical passenger car and some results presented in next chapter.

4. RESULTS

Simulation results for typical passenger car are shown in Figures 4 to 25. Possible simulation variables are, system structure, matrix format C as relation system outputs and system state, format and values of weighting matrices, S, Q, R, denoted in previous chapter, also, optimization criterion, further, forward vehicle velocity, different combination of controller/vehicle system functional and design parameters, shape of reference path on the roadway, etc. On the presented Figures in this chapter, are used following variables name or/and signs: latac – lateral acceleration, y'', latvel – lateral velocity, y', latdev – lateral deviation, y, yawac – yawing acceleration, ε'', yawvel – yawing velocity, ε', yawan – yawing angle, ε.

The results in Figure 4, 5, 6, 7, present vehicle behaviour by fixed steering wheel, initial lateral velocity of 5m/s as vehicle disturbance of initial conditions, longitudinal vehicle velocity, 20 m/s. In Fig. 8, are compared the change of variables from Fig. 4, 5, 6, 7, but by lateral disturbance of 3m/s, and longitudinal velocity 10 km/h. At same condition, Fig. 9, shows time change of vehicle lateral deviation by tunning steering wheel according sinus change with 0.1 rad magnitude and 0.1 Hz frequency.

Figure 10, 11, 12, 13, show vehicle behaviour by impulse disturbance on the front steering wheels, at a constant speed of 20 m/s, in a straight-line path.

The influence of the optimal controller on the vehicle steering in straight-line direction for initial condition disturbance type, a/ initial lateral displacement of 1m, and b/ initial lateral velocity of 5 m/s, are presented in Figures 8 and 9, respectively.

The vehicle time response to lateral disturbance, presented in Fig. 4, 8, as well as, to sinus and impulse disturbance on the steering wheel, presented in Fig. 9 and 10,, illustrate examples of unstable motion releted to lateral motion. Namely, after an instantaneous disturbance at zero initial time the vehicle lateral deviation, shorter name ″ latdev″ , presented in mentioned Figures, increases incessantly with time. Similar behaviour vehicle exhibits releted to angular deviation, as presented in Fig. 5, 8, 11, but with different deviation curve slope and settling time.

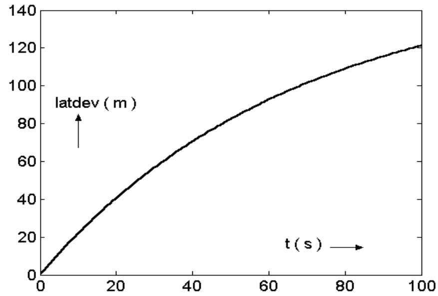


Figure 4: *Vehicle lateral deviation versus time*

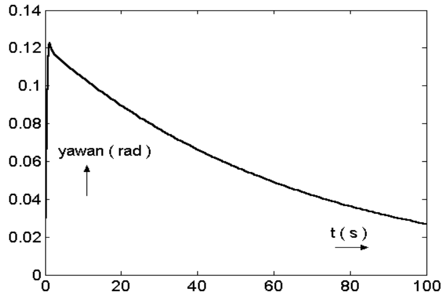


Figure 5: *Vehicle angular deviation versus time*

Lateral disturbance 5m/s, longitudinal velocity 20 m/s

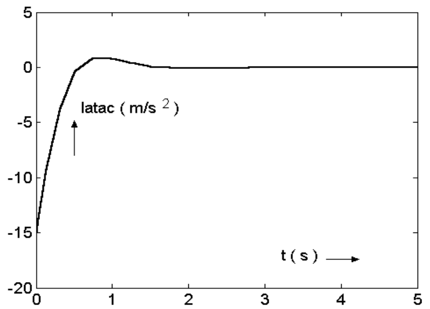


Figure 6: *Vehicle lateral acceleration versus time*

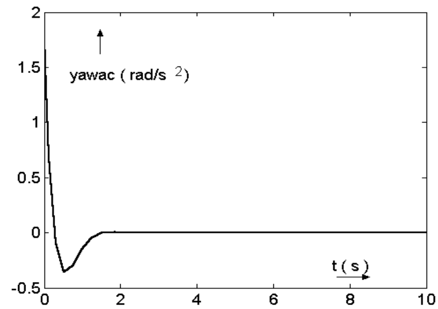


Figure 7: *Vehicle yaw acceleration versus time*

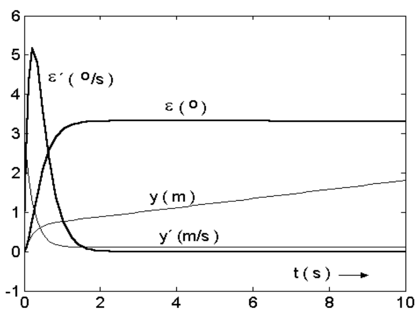


Figure 8: *Variables from Fig. 4–7, at lateral disturbance of 3m/s, longitudinal velocity of 5m/s*

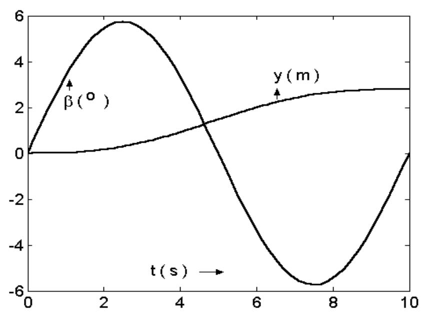


Figure 9: *Lateral deviation versus time at sinus tunning steering wheel. Lateral disturbance of 3m/s, longitudinal velocity 5m/s*

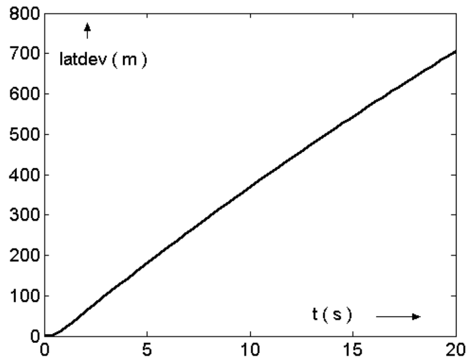


Figure10: Vehicle lateral deviation versus time

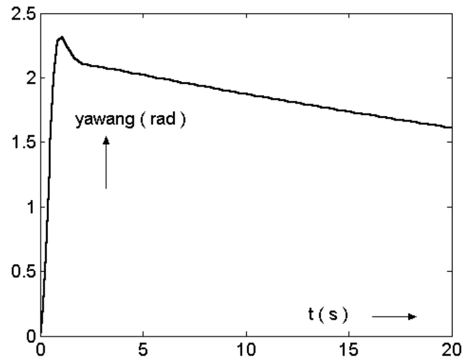


Figure 11: *Vehicle angular deviation versus time*

Impulse disturbance on steering wheel, longitudinal velocity 20m/s.

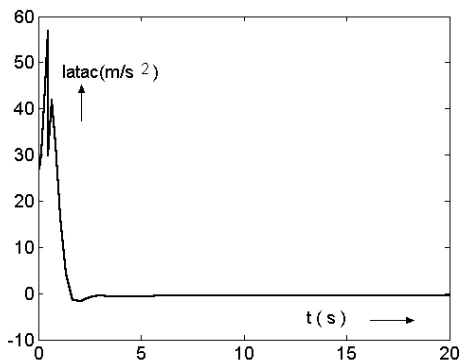


Figure 12: *Vehicle lateral acceleration versus time*

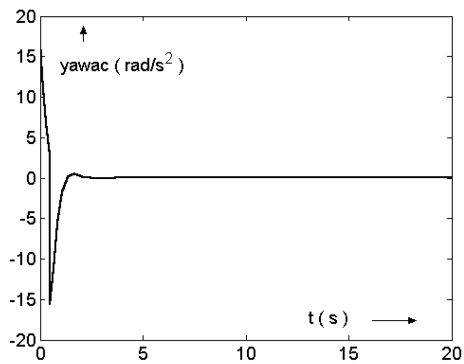


Figure 13: *Vehicle angular acceleration versus time*

The simulation results in Figs. 14 - 25, present vehicle behaviour in the case when optimal controller is acting. The optimal controller structure is determined according to described mathematical relations (9) to (18) with full state vector used as output vector to be minimized. Fig. 14 – 19, show effects optimal controller by initial lateral displacement disturbance of 1m at initial zero time. During optimal controller action, this displacement as lateral deviation from desired path is corrected and vehicle is returned back to its straight-line steady cruise condition.

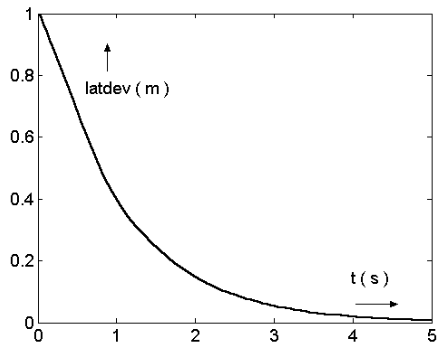


Figure14: *Lateral deviation versus tim.*

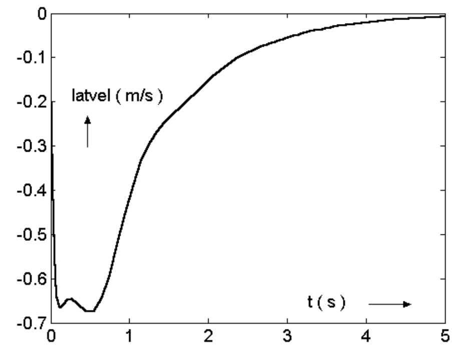


Figure 15: *Lateral velocity versus time*

Optimal controller, lateral disturbance 1m.

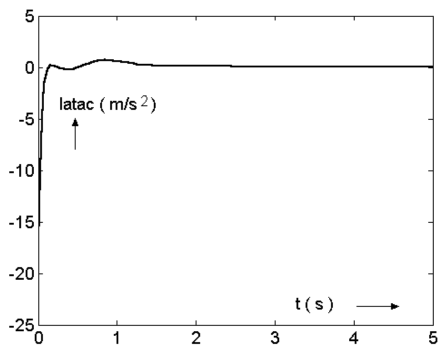
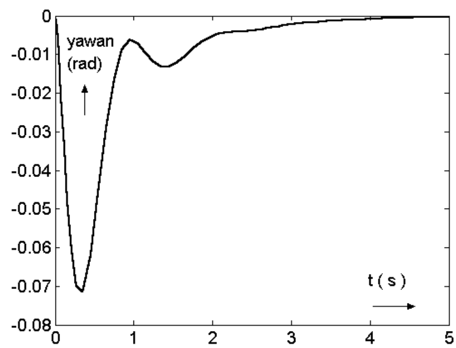


Figure16: *Lateral acceleration versus time*



Picture17: *Angular deviation versus time*

Optimal controller, lateral disturbance 1m.

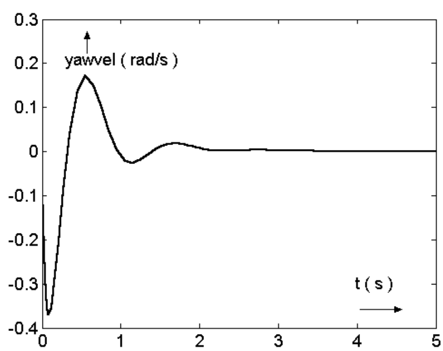


Figure 18: *Yaw velocity vesus time*

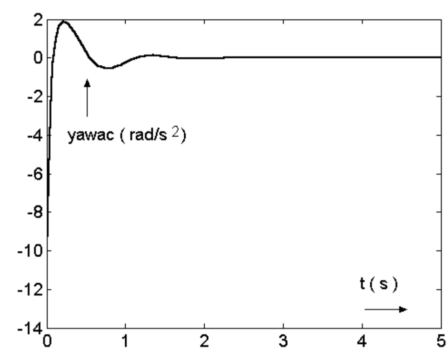


Figure 19: *Yaw acceleration versus time*

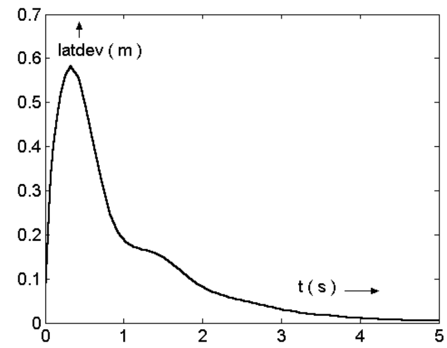


__Figure 20:__ Lateral deviation versus time

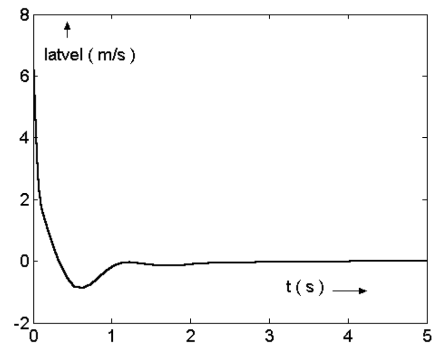


__Figure21:__ Lateral velocity versus time

Optimal controller, lateral disturbance 5m/s.

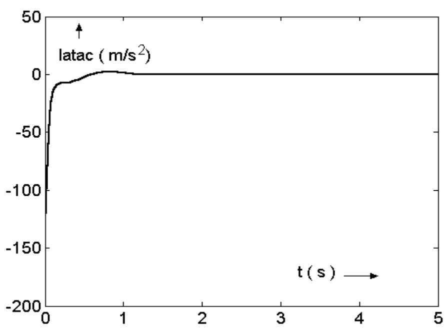


__Figure 22:__ Lateral acceleration versus time

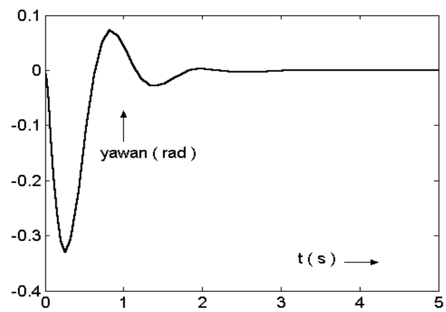


__Figure 23:__ Aangulardeviation versus time

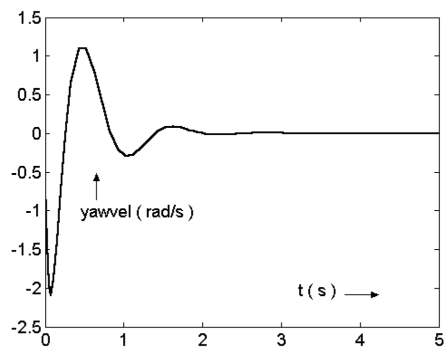


__Figure 24:__ Yaw velocity versus time

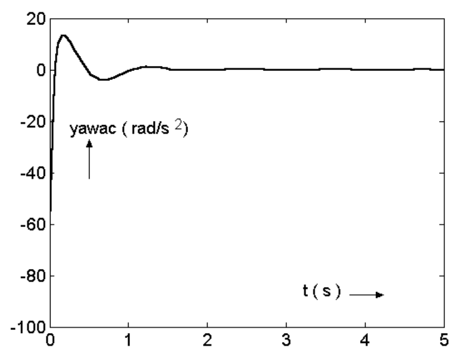


__Figure 25:__ Yaw acceleration versus time

In Figure 14 - 19, the lateral deviation, latdev, is a well damped process, with relatively shorter settling time, but time response of the lateral velocity and yawing angle shown any sharper peaks, unsymmetrical in relation to zero levels. On the other hand, lateral acceleration, yawing acceleration and yawing velocity are symmetrical well damped processes. These effects can be influenced to vehicle ride quality and reduced by means an appropriate choice of the performance index in optimization procedure, equation (11).

In Figure 20 - 25, results of the vehicle optimal control by lateral disturbance of 5m/s are presented. In this case, control process of the lateral deviation is more difficult task, while the rest time responses are symmetrical damped.

5. CONCLUSIONS

The problem of vehicle directional unstability play important role by design new as well as by improvment existing vehicle types. The useful solutions can be obtained by consideration of vehicle as part of closed-loop system with various variants of steering control : (a) by driver, (b) by technical controller, (c) by driver and technical controller simultaneously. The previous studies of driver used control strategies give a good basis to better understanding requirements by choice structure and parameters of optimal controller. On the other hand, possible driver strategies and estimated controller design concept can help to vehicle optimal design according to, today all severer of traffic safety requirements. The proposed methodology and obtained results in this paper can contribute in this direction, before all, to assessment of vehicle handling performance quality with respect to above mentioned requiremets and driver effort to steer vehicle.

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[1] MEASUREMENT OF DECELERATION ON THE ACCELERATOR PEDAL CONNECTION

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UDC: 629.113:62-592.11

Abstract

By using the recorded characteristic perceived through the shift - time in the area of the connection between the bulkhead of a passenger car's shell and the accelerator rod, and also, by using the numerical methods, the time characteristics of acceleration have been defined in this particular area. The time alteration of acceleration is known to represent the basis for making conclusions and precausion measures as regards the safety of a driver or passenger. This curve is modified in a number of ways throughout the paper itself, owing to which the following numerical values have been obtained: the values related to the shift concerning the connection between the bulkhead of the shell and the accelerator rod and all of this is placed within the connection between the lower left screw and the pedal adapter.

The shift optimization can be used by the constructor for a modification of the above mentioned connection with the purpose of obtaining the optimal acceleration, that is, the acceleration that would reduce the possible leg injuries of a driver to the minimum. The well- known formula defined in the AIS charts of biomechanical endurance has been used as an injury criterium.

Key words: the alteration of acceleration, shell, impact, injury.

MERENJE USPORENJA NA VEZI PEDALE GASA

UDC: 629.113:62-592.11

Rezime: Koristeći snimljenu karakteristiku pomeranje-vreme na mestu veze pregradnog zida školjke putničkog automobila i poluge pedale za gas, korišćenjem numeričkih metoda, došlo se do vremenske karakteristike ubrzanja na ovom mestu.

Poznato je da je vremenska promena ubrzanja baza za donošenje zaključaka i mera o bezbednosti vozača ili putnika. U radu je ova kriva modifikovana na više načina, tako da su dobijene numeričke vrednosti pomeranja veze pregradnog zida skoljke i poluge pedale za gas i to na mestu veze donjeg levog zavrtnja za priključak pedale.

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Optimiranje ovih pomeranja može da posluži konstruktoru za modifikaciju pomenute veze u cilju dobijanja optimalnog ubrzanja, odnosno ubrzanja koje bi dovelo do najmanjih povreda noge vozača. Kao kriterijum povrede je korišćena opšta formula biomehaničke izdržljivosti.

Ključne reči: promena ubrzanja, udar, pedala, car body, impact, gas accelerator.

MEASUREMENT OF DECELERATION ON THE ACCELERATOR PEDAL CONNECTION

Đorđe Antonijević[1], Milan Lončar, Aleksandra Janković, Rade Đukić

UDC: 629.113:62-592.11

1. EXPERIMENTAL RESEARCHES

1.1. Experimental measurements of acceleration pedal joint points displacement conducted on a domestic vehicle

In a laboratory of "Zastava Automobili" plant, before it evanesced as the result of the economy transition, a certain number of researches have been performed in order to reach one goal - car safety improvement. Those researches have been conducted in order to reach and to fulfill certain criteria which are obliged in high developed countries, above all to fulfill ECE regulation in order to secure driver's and passenger's safety which will be in accordance with demands of a market.

One of conducted researches is recording of joint point displacement among the acceleration pedal and a car body, figure 1. and details 1.a) and 1.b).

Figure 1: *Figure of examined part*

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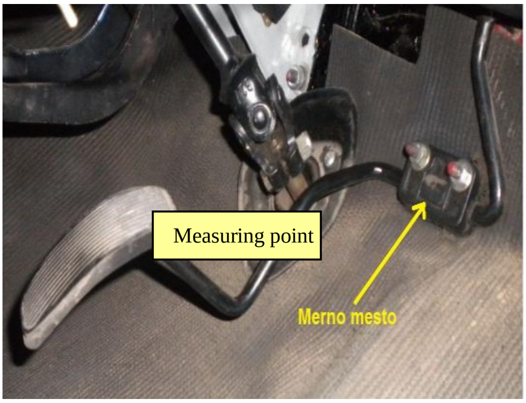


Figure 1a): *Measuring point "A"* ***Figure 1.b)****: Detail*

Recorded diagram, here enclosed, is taken from a manufacturer's data base, figure 2.

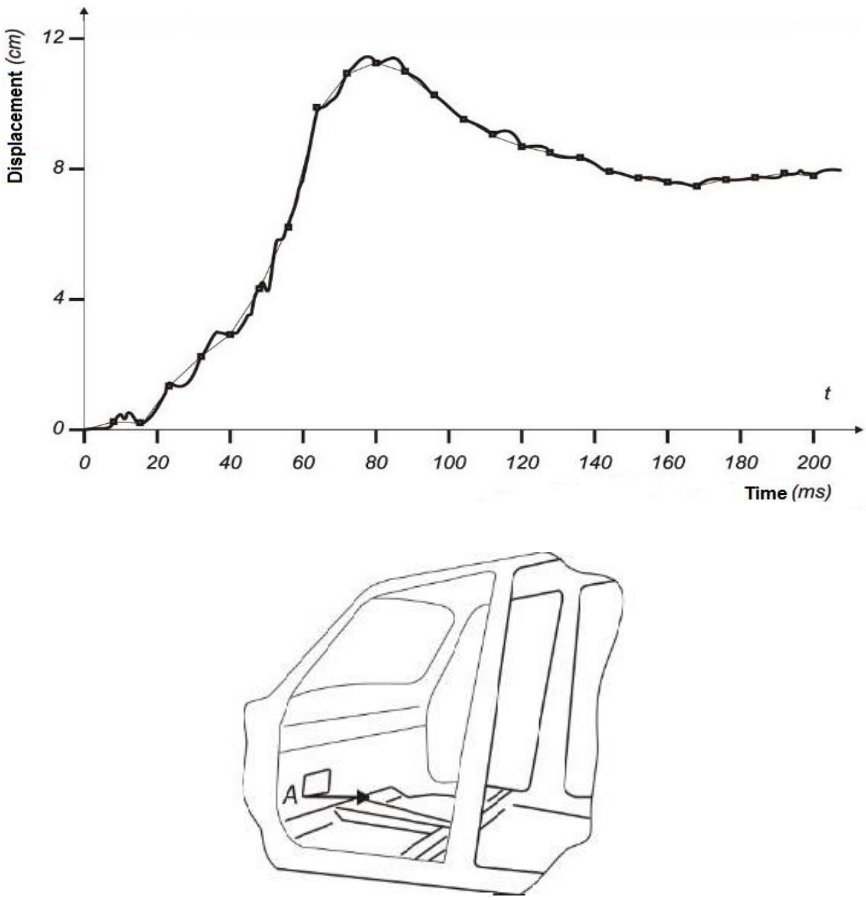


Figure 2: *Recorded diagram and a scheme of the location where the emitter was placed*

In a diagram a displacement of car body partition wall, in the area of its connection with the acceleration pedal, is shown depending on time.

By means of numerical differentiation, are obtained speed and acceleration values in "A" point, as shown in table 1.

Table 1: *Displacement, speed and acceleration values in "A" point (Figure 2.)*

No.	Time t (ms)	Displacement x (mm)	Speed (m/s)	Acceleration (g)
1.	0	0.0	0.32	-4.9
2.	8	2.5	-0.07	19.9
3.	16	2.0	1.52	-6.0
4.	24	14.1	1.04	-2.3
5.	32	22.4	0.86	10.7
6.	40	29.3	1.72	9.1
7.	48	43.0	2.45	24.0
8.	56	62.5	4.36	-36.4
9.	64	97.4	1.45	-12.6
10.	72	109.1	0.45	-10.1
11.	80	112.6	-0.37	-7.1
12.	88	109.7	-0.93	1.3
13.	96	102.2	-0.83	2.1
14.	104	95.6	-0.66	3.1
15.	112	90.3	-0.42	2.0
16.	120	87.0	-0.26	0.6
17.	128	84.9	-0.21	-3.5
18.	136	83.2	-0.49	2.9
19.	144	79.4	-0.25	0.7
20.	152	77.3	-0.19	0.8
21.	160	75.8	-0.13	5.0

In this case, beginning of plastic deformation will occur approximately after 16 ms, during that time deformation of car body is 2 mm. Deformation is being progressively increased up to its maximum value, 112.6 mm in this case for the period of 80 ms. After the highest value is reached, partition wall deformation is being decreased due to elastic features of the material approximately to 75 mm, after that a permanent plastic deformation occurs (approximately 80 mm).

By differentiation of recorded displacement values the speed and acceleration values of the joint point among the acceleration pedal lever and a car body are obtained. Acceleration diagram is shown in figure 3.

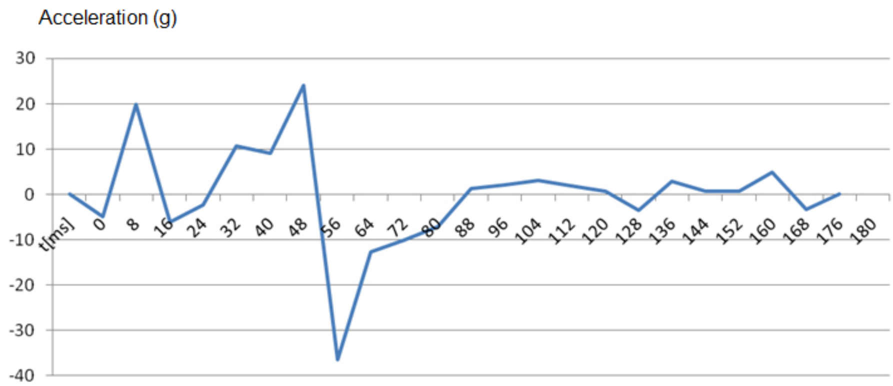


Figure 3: *Acceleration diagram (g) depending on time (t)*

Based on a diagram (figure 3.) it might be concluded that in the area of maximal displacements, acceleration modify its sign. From the aspect of driver's safety the most suitable case is when the acceleration values have the same sign. The "maximum" on a displacement curve, in figure 2. is characteristic for so called "snapp-through" problems, because the examined section of a car body is convex, observed along the force direction. Based on a thin shell theory it's familiar that for smaller displacements a curve has to be reduced and the thickness of the metal sheets has to be increased.

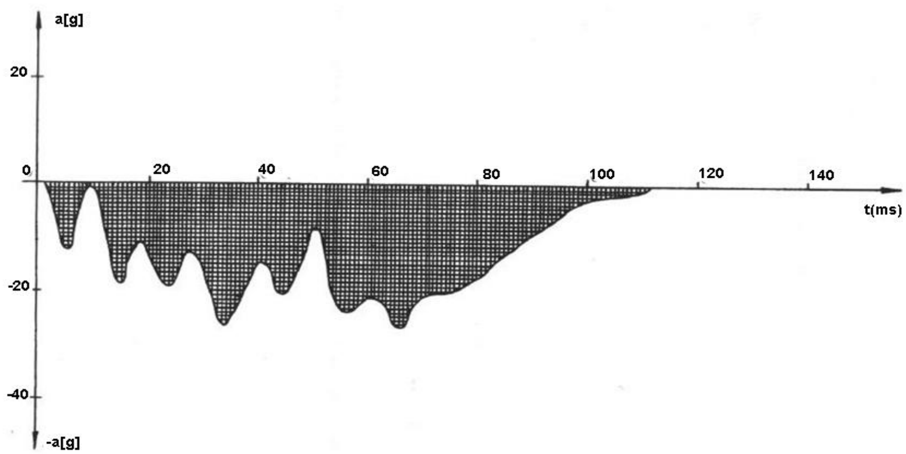


Figure 4: *Diagram of deceleration alteration for the frontal impact at the speed v=13,42 [m/s], measuring point is in passenger compartment based on ECE 33*

A diagram shown in figure 4. presents vehicle's deceleration in a function of time, which is obtained by direct crash to a fix barrier. Deceleration value "U" is about 30 g, such a value might be considered as a satisfactory result from the aspect of driver's safety within a cabin compartment. Boundary deceleration values, which will guarantee drivers safety and preserve him from the injuries, are various for different parts of driver's body and these values are depending on acceleration effect during the time. Observing the available data, for medium class passenger cars, measured acceleration mostly do not exceed 40 g. Also, it should be notified that acceleration values should have the same sign, all the time during the impact interval.

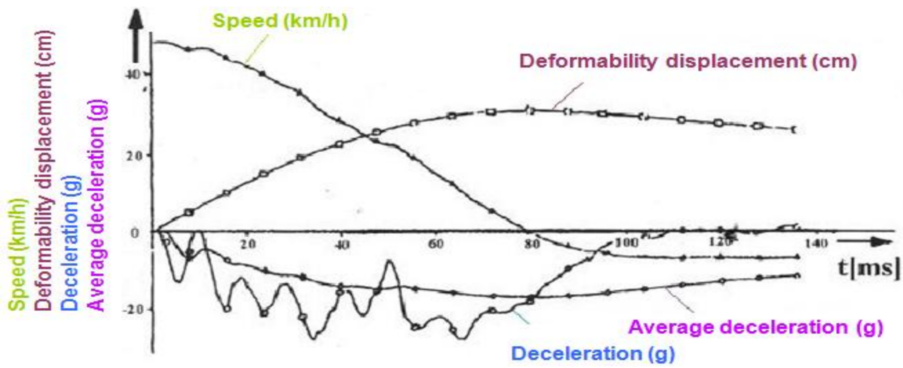


Figure 5: *Speed, deceleration and deformability displacement alterations of the same passenger car during direct impact to a fix barrier*

Based on a car deceleration diagram (figure 5.) it's possible to calculate impulse of the impact force per mass unit, same as the value of the impact coefficient [k]. Based on deformability displacement curve it's possible to calculate maximum deformation value same as the permanent deformation value.

1.2 Measurements of some other manufacturers

Examination results are depending great deal on material features same as on vehicle's weight, i.e. on weight/mass distribution along the vehicle. On the following examples will be shown the effect of a vehicle's weight to acceleration (g) which is being transmitted to a driver. Values presented in figures 6, 7, 8 and 9 are obtained by the measuring conducted inside the passenger compartment (cabin) according to ECE 33 regulation.

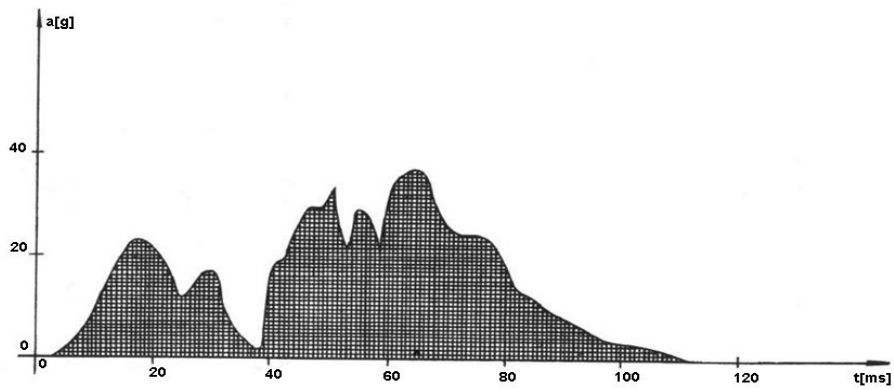


Figure 6: *Deceleration diagram of a light passenger car, [1]*

The effect of a vehicle's mass on to maximal deceleration is shown in figures 6. and 7. while the effect of the weight/mass distribution i.e. the power-train position is shown in figures 8. and 9.

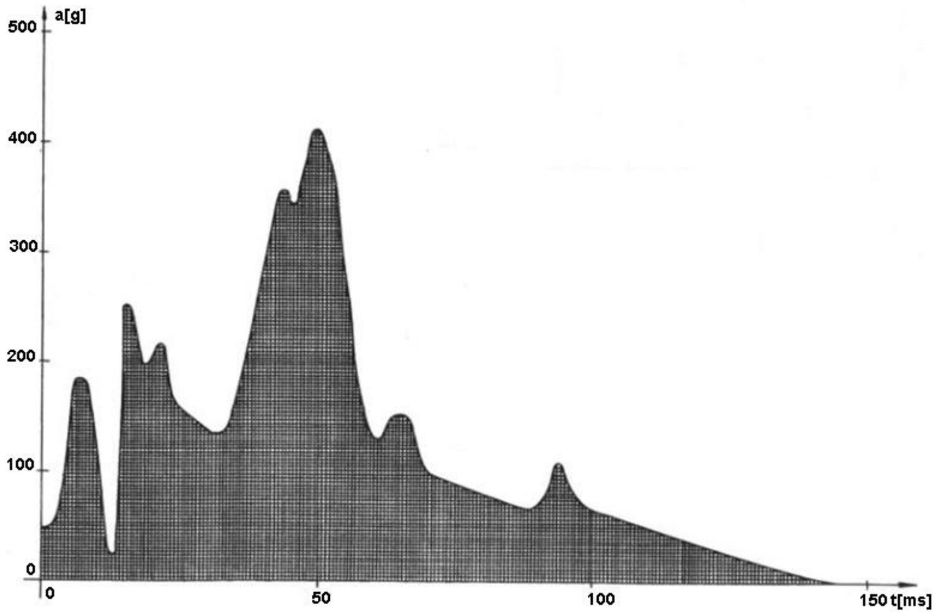


Figure 7: *Deceleration diagram of a heavy passenger car, [1]*

Examples of mass distribution effects to deformation and acceleration (g) values:

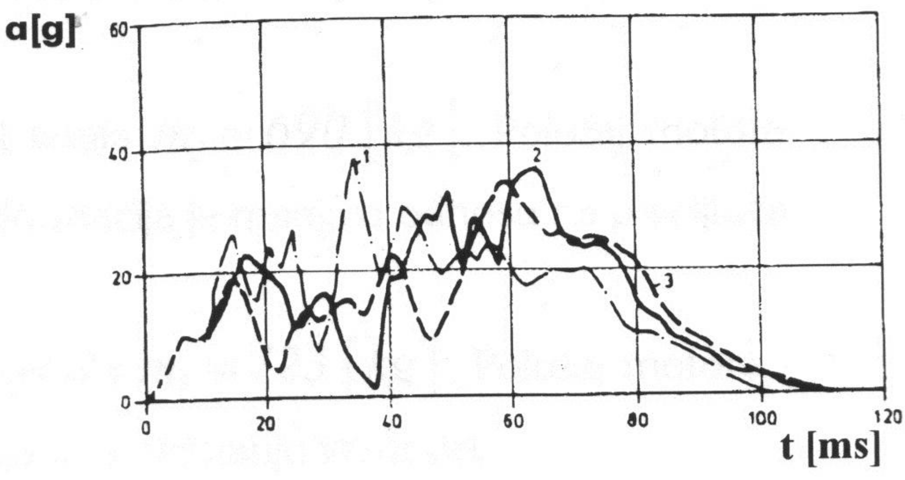


Figure 8: *Deformation characteristic for the vehicles m≤800kg, V=50km/h at the frontal impact, [2]*

Curve 1 - characteristic of a vehicle 640kg weight with the front engine

Curve 2 - characteristic of a vehicle 690kg weight with the rear engine

Curve 3 - characteristic of a vehicle 723kg weight with the rear engine

Curve 1 – If in a “acceleration-time” and “force-road” diagram the values of maximum deformation are observed it is obvious that in this case deformation is the highest.

Curve 2 – maximum deformation diagram in this case is lower than in a first case.

Curve 3 – maximum deformation in this case is the lowest.

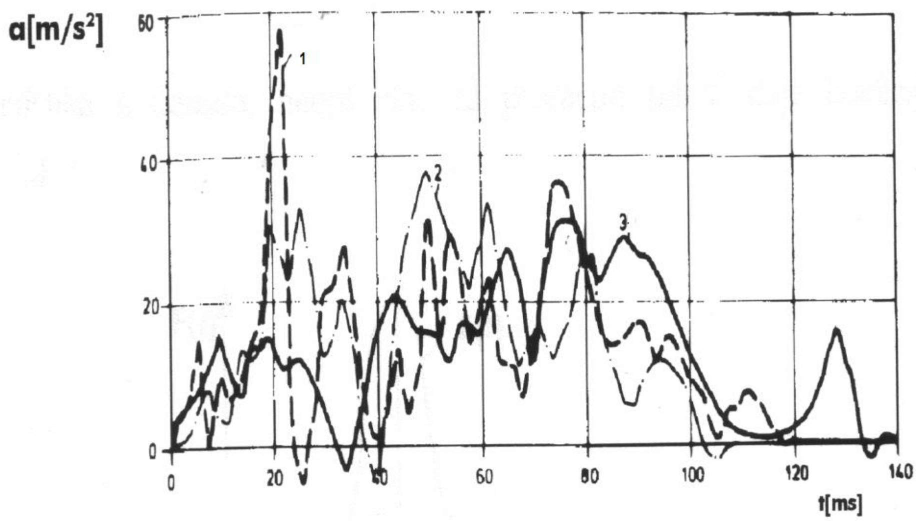


Figure 9: *Deformation characteristic for the vehicles m≤1300kg, V=50km/h at the frontal impact, [2]*

Curve 1 – Vehicle 1250kg weight, with the front engine

Curve 2 – Vehicle 1215kg, weight, with the front engine

Curve 3 – Vehicle 1252kg, weight, with the rear engine

Curve 1 – the value of maximal deformation in this case is the highest.

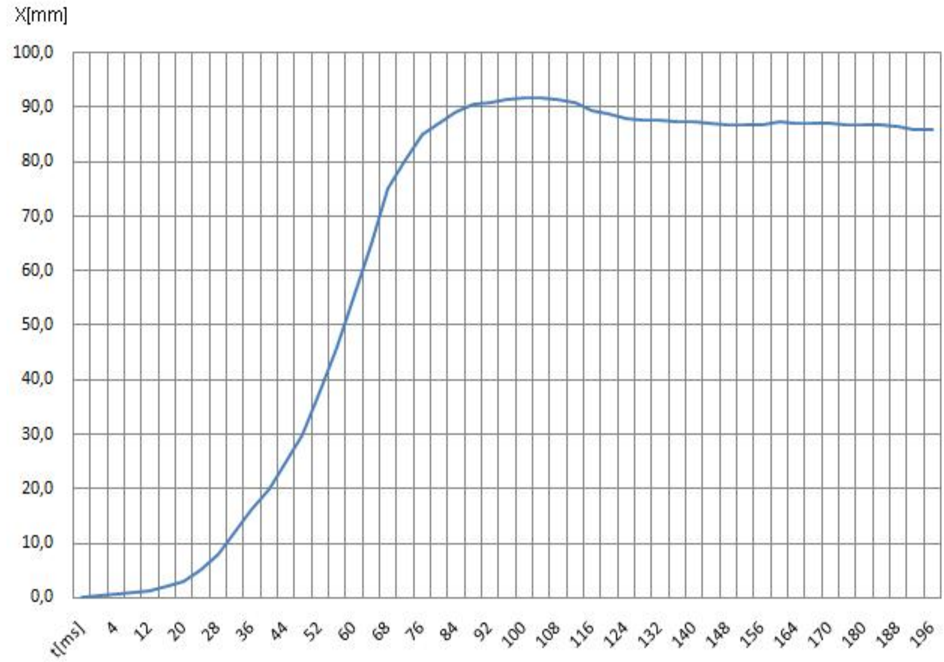
Curve 2 – for the same position of the engine (front) with vehicle’s weight reduction and maximal deformation values are also decreased.

Curve 3 – Vehicle with the highest weight, with the rear engine, the value of maximal deformation is the lowest.

2. ACCELERATION ANALYSIS IN ACCELERATION PEDAL JOINT POINTS DEPENDING ON DIFFERENT STIFFNESS OF THE JOINTS

By deformation displacing variation x, the features of different joints were simulated and numerically compared accelerations.

2.1 The case of stiffener joint than the initial one



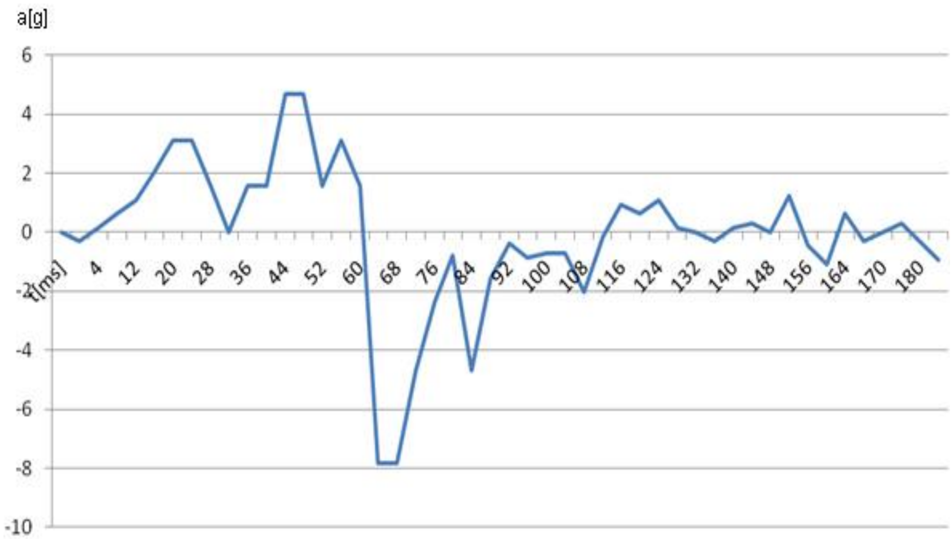


Figure 10: *Displacement and acceleration (g) dependency on time*

2.2 The case of initially stiff characteristic with big relaxation

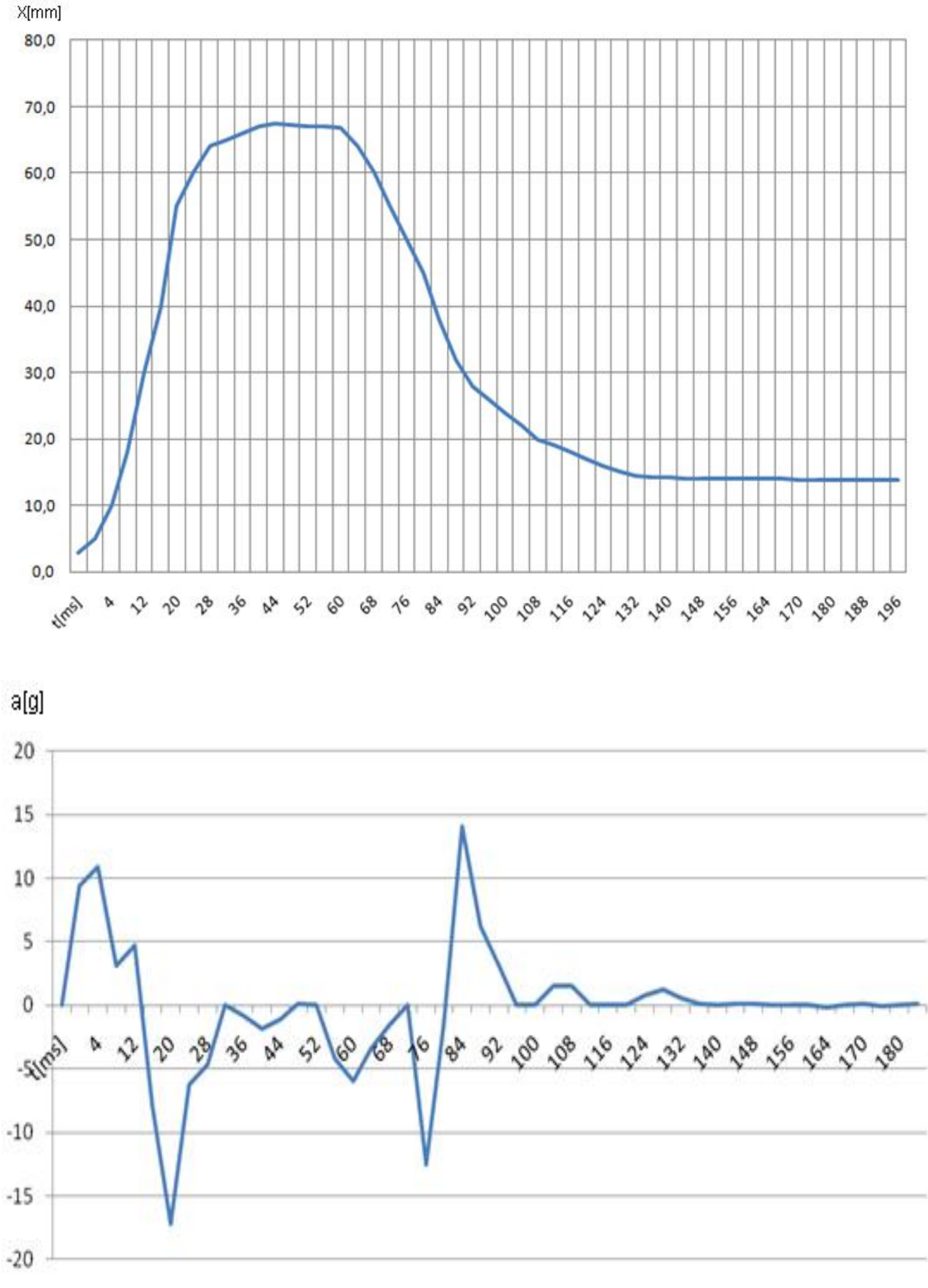


Figure 11: *Acceleration (g) dependency on elastic material deformations*

2.3 The case of congealed characteristics

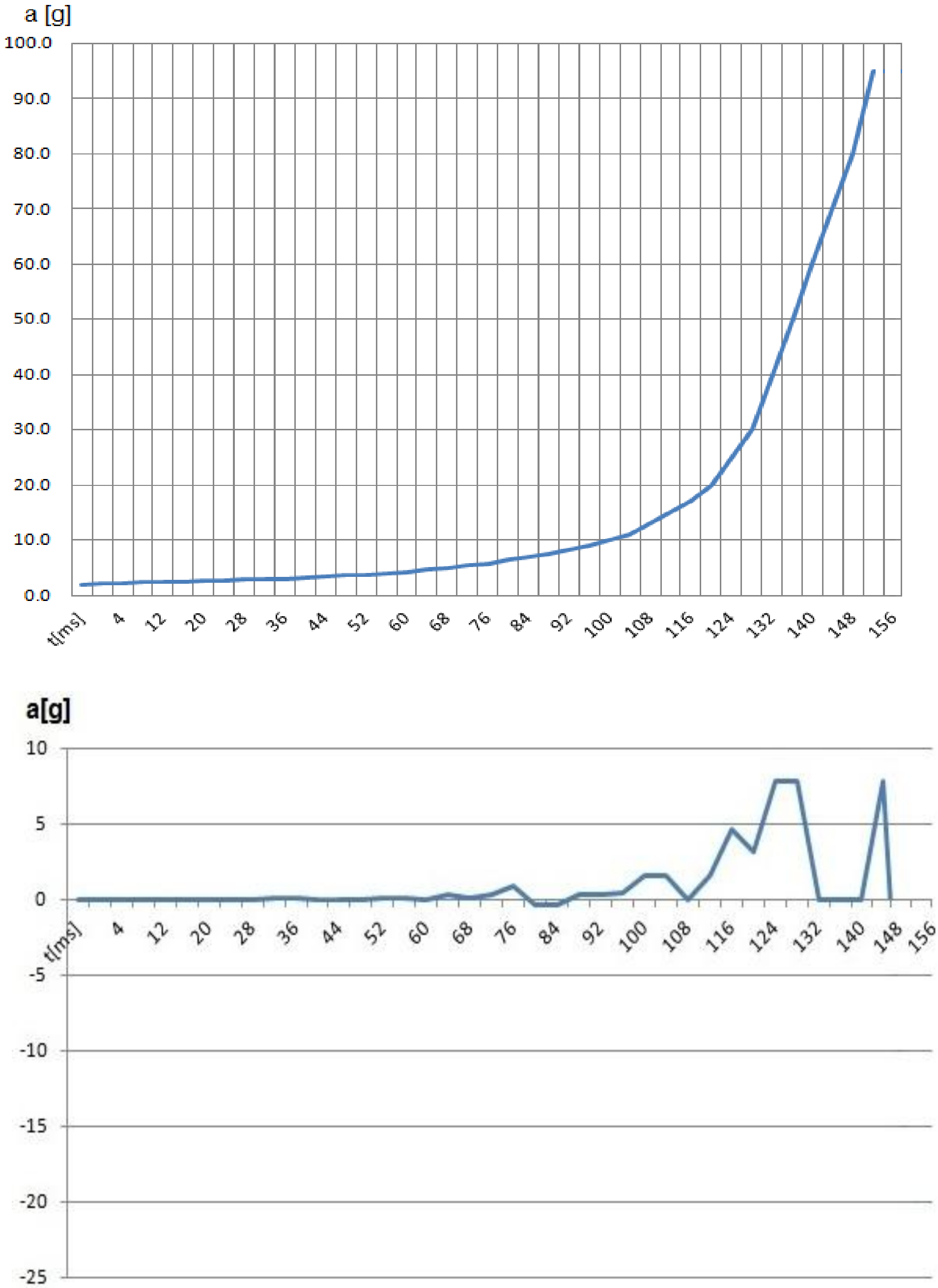


Figure 12: *Acceleration (g) dependency on fragile material deformation*

It's quite obvious that the latter characteristic is the most convenient both from the aspect of acceleration level and from the aspect of the moment, when the maximum is reached. Postponed period required for reaching the maximum value gives the possibility for safety system action.

CONCLUSION

Based on this paper it can be concluded that accelerations created during the deformation of the partition wall in the connection point with the acceleration pedal has to be reduced to its lowest intensity, and their maximum to be postponed to the up most moment. Impact interval should be the shortest it's possible. In this way the impact force F_U is also reduced in order to avoid driver's leg injuries during the crash.

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