ARCHIWUM MOTORYZACJI
**2,** pp. 127-144 (2010)

# Power steering adjustment considering lateral acceleration influence

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The paper presents an idea of power steering modification. The idea aims at providing the driver with favourable road feel which is considered to be essential during a driving process. The authors of the paper point out that lateral acceleration should be included in power steering regulation algorithm in order to adjust its characteristics. Lateral acceleration can be included by application of the parameter called steering effort which is closely connected to that value. Experimental and theoretical research of dynamic characteristics of steering systems was carried out. An opportunity of steering effort application to power steering algorithm was studied on the basis of the results obtained. The most important results are presented and discussed in the paper.

## 1. Introduction

Power steering application has contributed to comfort and safety of driving improvement in vehicles. The main aim of introducing power steering was to decrease steering torque which had to be applied by the driver to perform the desirable manoeuvre. To achieve that aim particular constructional solutions were found and improved.

Recently the tendency to improve power steering regulation has grown. That improvement aimed at providing the driver with favourable road feel. Road feel has been defined by the authors of this paper as a process of receiving information about the condition of a car motion by means of the driver's senses.

The information may concern e.g. forces and torques which act in contact area of tyres and road surface. Good road feel requires an undisturbed flow of information about the condition of a car motion to the driver. As a driver gains an experience of driving he or she creates in memory some sort of dynamic model of the particular car. The model provides comfort and safety of driving because on the basis of that model the driver can easily predict car's response to every steering wheel turn.

The relation between input applied by the driver such as steering wheel angle, steering wheel turn velocity and steering torque and output such as yaw velocity and lateral acceleration should not change suddenly. It makes car control easy for the driver and provides favourable road feel [1].

In commonly used power steering regulation gain level is adjusted basing on steering torque and velocity values. It is insufficient to guarantee good road feel. During cornering not only velocity but also lateral acceleration influences information about the vehicle motion condition.

The authors of this paper suggest a need to make a modification of power steering. Lateral acceleration should be included in regulation algorithm by application of parameter called steering effort. That parameter was defined by Jaksch [2] as a product of steering wheel angle gradient and steering torque gradient:

$$E' = \frac{d\delta_H}{d\tilde{a}_y} \cdot \frac{dM_H}{d\tilde{a}_y} \tag{1}$$

where:

$\delta_H$ – steering wheel angle

$M_{\delta H}$ – steering torque

$\tilde{a}_y$ – standarized lateral acceleration - lateral acceleration divided by acceleration of gravity

Jaksch [2] carried out experimental tests that proved good correlation between steering effort defined above and driver's subjective rating of steering control quality.

In this paper results of experimental and theoretical research are presented. The relation of steering effort to velocity and the relation of steering effort to lateral acceleration were examined. An opportunity of steering effort application to power steering algorithm in order to adjust its characteristics was studied on the basis of the results.

## 2. Steering effort

As it was emphasized in the introduction, the influence of lateral acceleration on steering system characteristics is very significant. Lateral acceleration is one of the most important values that provide the driver with information concerning the state of car motion during driving process and it influences the driver due to its connection to yaw velocity and sideslip angle. The authors of this paper suggest that lateral acceleration should be included in power steering algorithm by including steering effort defined above by the formula (1).

The aim of application of that parameter to adjust the power steering regulation and to improve road feel can be justified by its good correlation with driver's subjective rating proven by Jaksch [2].

Various definitions of steering effort can be found in literature. It is very often identified with steering torque [1, 3÷6). The analysis [7÷9] made by the authors of this paper revealed that steering effort identified with steering torque is insufficient for power steering regulation adjustment.

Some authors [10] show that steering effort is related to steering wheel movement which a driver performs during driving process. According to that paper, steering effort is closely related to good visibility during the driving process and the visual assessment of motion conditions performed by the driver. However, it is difficult to measure them during tests.

## 3. Experimental research of dynamic characteristics of steering systems

Road tests were carried out in order to determine steering system characteristics in various road conditions. Selected manoeuvres enabled an assessment of steering system properties connected with steering effort.

### 3.1. Programme of road tests

The research project consisted of the following manoeuvres:

- step input [11],
- double lane change [12],
- slalom [13].

Manoeuvres were executed at velocities: 40, 50, 60 and 70 km/h on hard, even and dry road surface with adhesion coefficient greater than 0,75. Wind velocity did not exceed 7 m/s.

### 3.2. Examined vehicles

There were four vehicles examined:

- M 1.2 class vehicle [14], with front-wheel drive, without power steering system - called vehicle A in this paper.
- M 1.3 class vehicle [14], with front-wheel drive, with an electric power steering system with two gain levels called vehicle B. In this case two settings of power steering system were examined: standard level and City function.
- M 1.5 class vehicle [14], with rear-wheel drive, with a hydraulic power steering system with constant gain level called vehicle C. In this case power steering was switched on and off during road tests.
- M 1.7 class vehicle [14], with front-wheel drive, with an electrohydraulic power steering system with constant gain level, called vehicle D.

All cars, that were examined, were equipped with steering system with rack-and-pinion gear.

### 3.3. Measured values and measuring equipment

During the tests the following values were measured and recorded:

- steering wheel angle, $\delta_H$,
- steering torque, $M_{\delta H}$
- lateral acceleration, $a_y$
- yaw velocity of a vehicle, $\dot{\psi}$.

The AD12 set was applied to record, process and display signals. A computer was used for on-line controlling and displaying the recorded values.

At first signals were recorded and processed by computer. Data processing was started by converting them to physical form, expressed in voltage units. The scaling, filtering, and shifting to zero were performed. Those operations aimed at changing data form to the form suitable for further processing i.e. for numerical calculation. Data processing was made in Matlab and Excel.

For velocity measurement the S-CE head of a Correvit system was used. This device was designed and applied to a non-contact measurement of velocity vector components of a selected car body point. An example of the method of assembling is shown in Fig. 1.

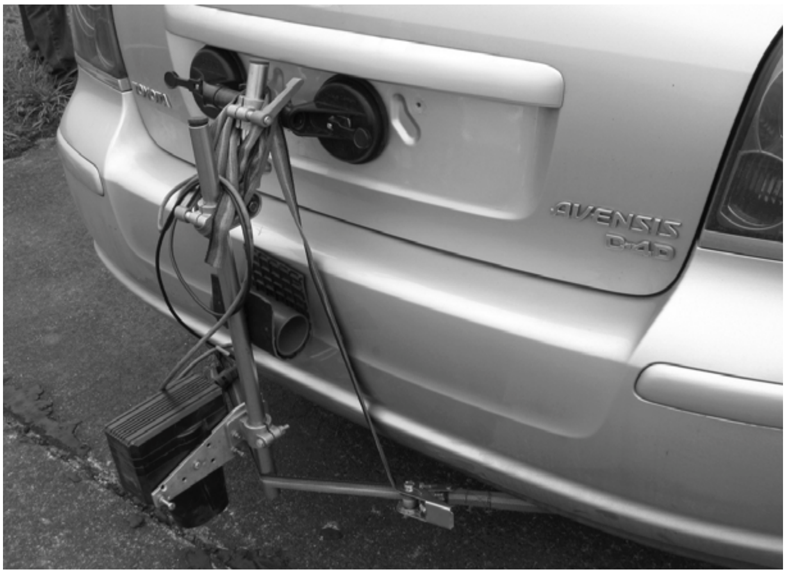


Fig. 1. Tested vehicle with Correvit S-CE head mounted.
Rys. 1. Badany pojazd z zamontowaną głowicą Correvit S-CE.

Measurements of steering wheel angle and steering torque values were made using steering wheel constructed in the laboratory of Cracow University of Technology. That steering wheel is presented in Fig. 2.

Fig. 2. Equipment used for measurements of steering wheel angle and steering torque.
Rys. 2. Urządzenia użyte do pomiaru kąta obrotu kierownicą i momentu obrotowego na kierownicy.

Steering wheel was connected to converters of steering torque and steering wheel angle.

## 4. Analysis of road test results

Steering effort values were calculated for all examined vehicles on the basis of data recorded during road tests. Gradients of steering torque and steering wheel angle were designated for each performed manoeuvre in the entire range of tested velocities. To designate those gradients from one test performed with a particular velocity a relation between steering torque and time or a relation between steering wheel angle and time were analyzed. Calculated values were estimated considering the range of favourable steering effort values specified in literature [2]. Moreover, steering system evaluation for all examined vehicles in regard to road feel was made.

Some exemplary results were presented in Fig. 3. As an example a double lane change was chosen.

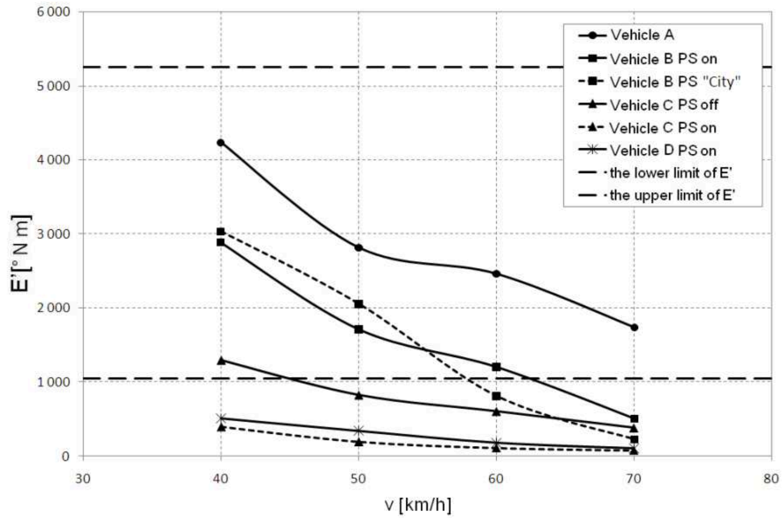


Fig. 3. Steering effort values for examined vehicles.
Rys. 3. Wysiłek kierowania badanych pojazdów.

The meaning of designations used in Fig. 3 is as follows:
*Vehicle A* - steering effort calculated for vehicle A
*Vehicle B PS on* - steering effort calculated for vehicle B, standard gain level
*Vehicle B PS "City"* - steering effort calculated for vehicle B, „City" function engaged
*Vehicle C PS off* - steering effort calculated for vehicle C, power steering switched off
*Vehicle C PS on* - steering effort calculated for vehicle C, power steering switched on
*Vehicle D PS on* - steering effort calculated for vehicle D
*The lower/upper E' limit* - the lower/upper limit of the range of favourable steering effort.

Regarding vehicle A, it can be stated on the basis of the range of favourable steering effort [2] that steering system of that vehicle ensures maintaining of steering effort level in the range of favourable values apart from the velocity. For that vehicle steering effort values are close to the lower limit of that range.

For vehicle B, the analysis showed that the differences between steering effort values of two considered power steering gain levels are relatively small. Regarding the range of favourable steering effort it was stated that when the velocity was smaller than 55 km/h steering effort values are favourable for both power steering gain levels.

At the velocities higher than 60 km/h, the steering effort values are smaller than the lower limit of the range of favourable steering effort and are gradually decreasing. Driving a vehicle B at those velocities might probably be unpleasant for a driver.

Similarly to the results of vehicle A and vehicle B, the results achieved for vehicle C showed that steering effort value decreases when velocity increases. Such tendency was observed for both configurations of power steering system with different intensity. That intensity was smaller when the power steering was on. For both power steering configurations steering effort was smaller than the lower limit of the range of favourable steering effort. It is possible that, both power steering configurations of vehicle C will not ensure pleasant road feel.

The results obtained for vehicle D showed that steering effort values were smaller than the lower limit of the range of favourable steering effort. Those values were similar to values of vehicle C with power steering switched on. It is possible that, steering system of vehicle D will be assessed by the driver as too sensitive and road feel will be unfavourable.

On the basis of performed tests it was stated that runs of steering effort versus velocity had the same tendency for all examined vehicles i.e. steering effort decreases as velocity increases. For particular vehicle steering effort maintains the same level irrespective of the type of performed manoeuvre. It is the additional argument for application of this quantity in power steering algorithm in vehicles in order to provide the driver with favourable road feel.

## 5. Theoretical research

### 5.1. Car model

The programme of road tests and the range of measured quantities were considered for selecting the mathematical model of a vehicle. An application of a particular mathematical model should make calculations of wheels' longitudinal and lateral velocities and normal forces for various velocities and manoeuvres possible. Considering those requirements a quasi-static car model [15] with 11 degrees of freedom was adopted. The model was examined and verified by the authors of this paper. A comparison of calculation results with experimental test results was made and good correlation between them was proved. It justifies the application that model.

A vehicle was defined as a set of rigid bodies connected by elastic and damping elements.

It was assumed that the road surface is ideally flat and horizontal and that the suspension and tyre stiffness characteristics are linear. Tyre damping was disregarded because of its small values caused by car body tilt movements.

Considering results previously received by the authors of this paper and analysis results presented in literature [15], the influence of power transmission system considered in calculations referred only to the driven wheels and was expressed as driving torque.

To describe forces and moments acting in the contact area of a tyre and road surface a model evaluated by Dugoff and modified by Uffelmann [16] was applied.

An additional input caused, for example, by unbalancing of wheels was disregarded. The adopted vehicle model is shown in Fig. 4. The most important quantities are presented.

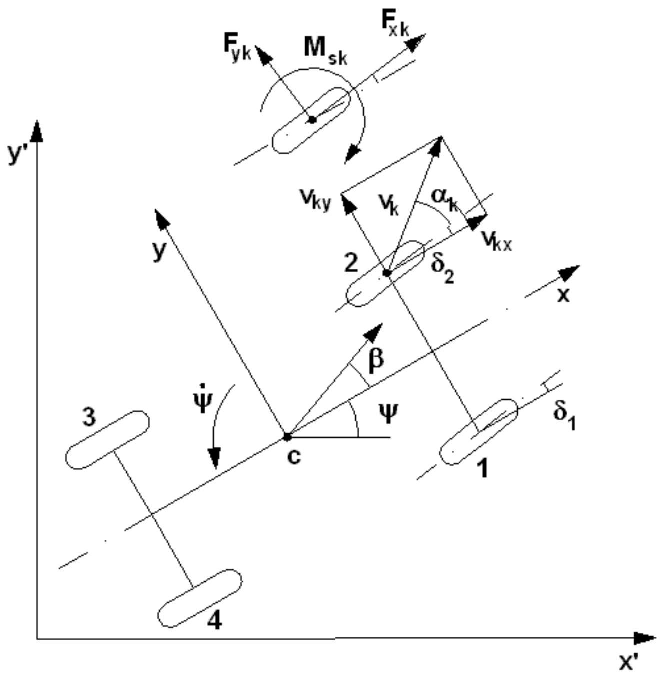


Fig. 4. Vehicle physical model [15].
Rys. 4. Model fizyczny pojazdu [15].

The vehicle movement is described by the following system of the second order differential equations with constant coefficients:

$$m \cdot \ddot{x}'_c = \sum_1^4 F_{xk} \cdot \cos(\psi + \delta_k) - \sum_1^4 F_{yk} \cdot \sin(\psi + \delta_k) - F_w \cdot \cos\psi \tag{1}$$

$$m \cdot \ddot{y}'_c = \sum_1^4 F_{xk} \cdot \sin(\psi + \delta_k) + \sum_1^4 F_{yk} \cdot \cos(\psi + \delta_k) - F_w \cdot \sin\psi \tag{2}$$

$$I \cdot \ddot{\psi} = \sum_1^4 \left\{ -l_{yk} \cdot \left[ F_{xk} \cdot \cos\delta_k - F_{yk} \cdot \sin\delta_k \right] + l_{xk} \cdot \left[ F_{xk} \cdot \sin\delta_k + F_{yk} \cdot \cos\delta_k \right] - \sum_1^4 M_{sk} \right\} \tag{3}$$

$$I_k \cdot \dot{\varphi}_k = M_{nk} - M_{hk} - F_{xk} \cdot r_{dk} + F_{zk} \cdot f'_k \tag{4}$$

For which the following designations were applied:

| | |
|---|---|
| $c_{op}$ | - tyre circumferential stiffness coefficient (identical value for all wheels) |
| $f'_k$ | - equivalent coefficient of rolling resistance |
| $f_k$ | - rolling resistance coefficient (identical value for all wheels) |
| $F_w$ | - aerodynamic resistance force |

| | |
|---|---|
| $F_{xk}$ $F_{yk}$, $F_{zk}$ | - components of force at tire-road surface contact (k = 1, 2, 3, 4) |
| I | - moment of vehicle inertia about vehicle vertical axis |
| $I_k$ | - wheel moments of inertia about axis of rotation, (k=1,2,3,4) |
| $l_{xk}$ | - distance of the centre of wheel trace to centre of inertia along axis x |
| $l_{yk}$ | - distance of the centre of wheel trace to centre of inertia along axis y |
| m | - vehicle mass |
| $M_{hk}$ | - braking torques affecting the wheels, (k=1,2,3,4) |
| $M_{nk}$ | - driving torque affecting the wheels, (k=1,2,3,4) |
| $M_{sk}$ | - aligning torques of tires |
| $r_{dk}$ | - dynamic wheel radius, (identical value for all wheels) |
| $\delta_k$ | - steer angles (for back wheels it is assumed that $\delta_3$=$\delta_4$=0), |
| $\varphi_k$ | - car wheels' angle of rotation (k = 1, 2, 3, 4) |
| $\psi$ | - yaw angle |

A car model was completed by a steering system model shown in Fig. 5. The idea of describing dynamics model of a steering system as an separated autonomous model coupled with separately assigned dynamics model of a vehicle was acknowledged as a correct one in [17].

The steering system was defined as a two dimensional system. It transmits steering from a steering wheel to wheels by elements that rotate on the plane that is parallel to the road surface and have fixed axis of rotation.

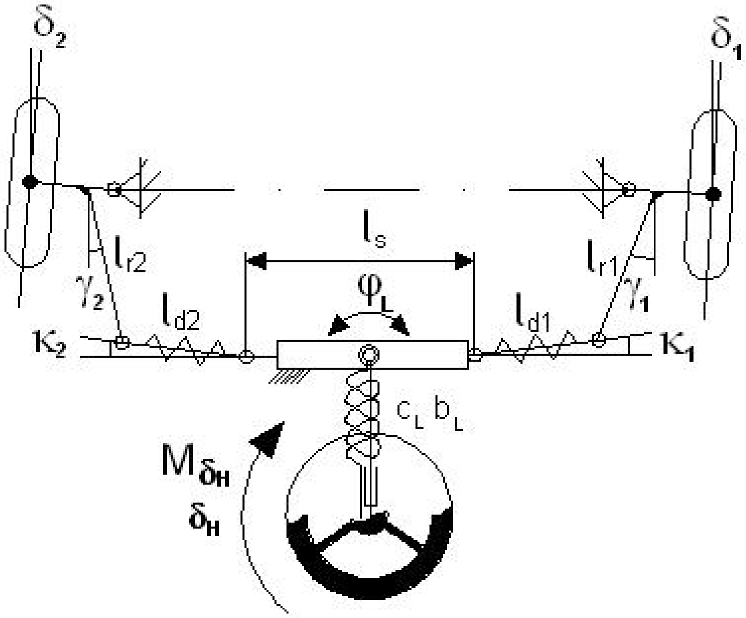


Fig. 5. Steering system model [15].
Rys. 5. Model układu kierowniczego [15].

An application of two dimensional model may be justified by a relatively small mass of elements transmitting steering movements in space in comparison to mass of turned wheels.

Elements that have considerable influence on dynamics characterized by inertia (turned wheels), backlash (gears) and friction (steering knuckle bearing) are moving on the planes parallel to the road surface [17]. That approach is commonly used in literature [17].

It was assumed that steering system model consists of rigid body that represents the rack and four weightless arms that represent the steering knuckle system (Fig. 5).

In the adopted model stiffness of the steering column shaft was represented by an elastic damping element described by $c_L$ and $b_L$. An equivalent viscous friction in steering column bearing was also considered. It was assumed that dry friction occurs in the sliding joint of the rack and in the bearing of the left and the right steering knuckle. It was assumed that the value of dry friction moment in the bearings of steering knuckles is proportional to normal forces affecting steering knuckles [15]. It was also assumed that steering knuckle axis is perpendicular to the road surface.

The applied steering system model makes possible to determine the relationship of angles $\delta_1$, $\delta_2$ and rack displacement $x_L$, and to determine values of moments $M_{zwr,1}$ and $M_{zwr,2}$, acting along the axes of the left and the right steering knuckle.

Rack movement can be expressed by the following equations:

$$-m_L \cdot \ddot{x}_L + F_1 \cdot \cos\kappa_1 - F_2 \cdot \cos\kappa_2 + c_L \cdot i_L \cdot (\delta_H - \varphi_L) + b_L \cdot i_L \cdot (\dot{\delta}_H - \dot{\varphi}_L) - F_t = 0 \tag{5}$$

Steering wheel movement can be computed using equation:

$$I \cdot \ddot{\delta}_H = M_{\delta H} - c_L \cdot (\delta_H - \varphi_L) - b_L \cdot (\dot{\delta}_H - \dot{\varphi}_L) \tag{6}$$

Turn of the wheels is described by the following formulas:

$$I_1 \cdot \ddot{\delta}_1 = M_{zwr1} - F_1 \cdot l_{r1} \cdot \cos(\gamma_1 + \kappa_1) \tag{7}$$

$$I_2 \cdot \ddot{\delta}_2 = M_{zwr,2} - F_2 \cdot l_{r2} \cdot \cos(\gamma_2 + \kappa_2) \tag{8}$$

$m_L$ - rack mass
$b_L$ - steering column dumping coefficient
$c_L$ - steering column torsional stiffness coefficient
$F_k$ - steering rods axial forces (k=1,2)
$F_t$ - rack friction force
$i_L$ - steering column ratio
$l_{dk}$ - steering rod length, ($k$=1,2)
$l_{rk}$ - knuckle arm length, ($k$=1,2)

$l_s$ - spacing of rack ball pins
$M_{zwr,k}$ - moments affecting the left and the right steering knuckle
$M_{\delta H}$ - steering torque
$r_z$ - pinion radius
$\gamma_k$ - angle between direction perpendicular to rack movement and knuckle arm, $l_{rk}$,
$\kappa_k$ - angle between steering rod and rack longitudinal axis, ($k$=1,2)
$\varphi_L$ - pinion rotation angle

The presented steering system model made possible to describe relative movements of the particular elements of that system and to describe turn angles of the particular wheels as independent of each other.

#### 5.2. Results of simulation tests

Computer simulation results for vehicle A are presented in this chapter. Step input results were analyzed because that manoeuvre did not require application of a driver model.

The analysis of an influence of velocity and lateral acceleration on steering torque $M_{\delta H}$ and steering effort *E'* was performed. In Fig. 6 a curve of steering torque maximum values versus velocity is presented.

To examine an influence of velocity on steering torque and steering effort values it was assumed that lateral acceleration is constant The authors of this paper verified that assumption in experimental research. While carrying out road tests they developed a particular method of manoeuvres performance. The method guaranteed gaining of similar values of lateral acceleration for manoeuvres performed with different velocities.

In simulation test it was assumed that lateral acceleration equals 4,9 m/s$^2$. That value was reached by lateral acceleration of vehicle A during road tests.

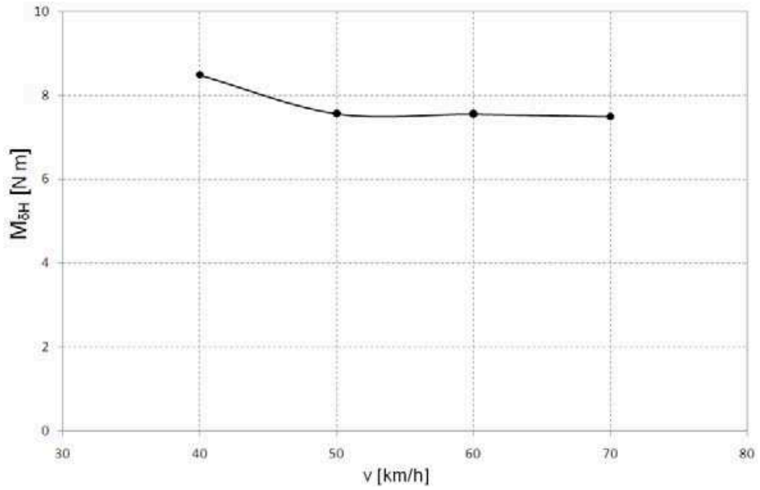


Fig. 6. Steering torque versus velocity, vehicle A.
Rys. 6. Zależność momentu obrotowego na kierownicy od prędkości jazdy, pojazd A.

The steering torque curve had smooth shape within the considered velocity range. At 40 km/h steering torque reached 8,5 N m. When velocity incerased from 50 km/h to 70 km/h changes of steering torque did not exceed 1%. Therefore it can be assumed that within the considered velocity range steering torque was constant.

The results of analysis of lateral acceleration influence on steering torque are shown in Fig. 7. The relation of steering torque to lateral acceleration was determined at constant velocity. The velocity of 60 km/h was chosen as most of the power steering systems are inactive at this velocity. Vehicle A was not equipped with power steering.

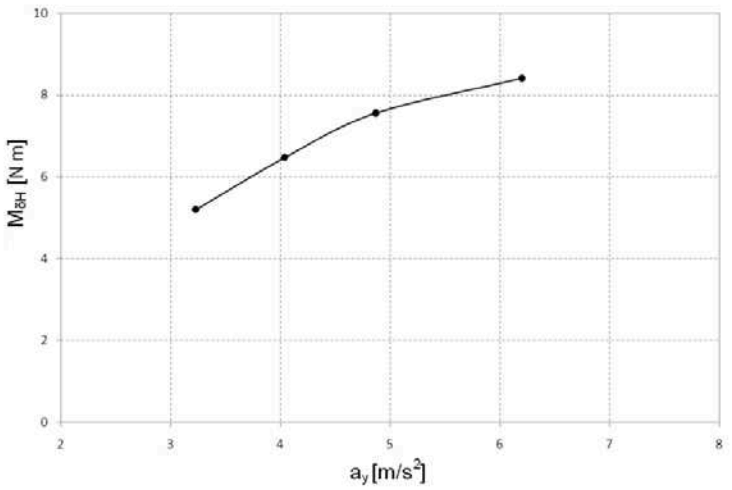


Fig. 7. Steering torque versus lateral acceleration, vehicle A.
Rys. 7. Zależność momentu obrotowego na kierownicy od przyśpieszenia poprzecznego, pojazd A.

It was stated that when lateral acceleration of the analyzed vehicle increased from the value 3,2 m/s$^2$ to 6,2 m/s$^2$, steering torque increased by about 62%.

Taking into account the fact that the analyzed lateral accelerations belong to the range typical for the average driving conditions for that vehicle during step input, it can be stated that such a considerable increase of torque can be badly tolerated by the driver.

Comparing the relation of steering torque to lateral acceleration with the relation of steering torque to velocity (Fig. 6), it can be stated that the observed increase of steering torque is noticeable. Therefore two important conclusions can be stated:

- the relation of steering torque to lateral acceleration is significant
- the relation of steering torque to velocity is noticeable however changes of steering torque to velocity are small in comparison to changes of that quantity to lateral acceleration. The curves of particular steering torque relations have different shapes. The curve of steering torque to lateral acceleration increases monotonically while the curve of steering torque to velocity is nearly constant.

As it was mentioned a significant increase of steering torque to lateral acceleration occurs and power steering regulation based on velocity only seems to be insufficient, especially at velocities greater than 60 km/h. Simulation results showed that both velocity and lateral acceleration should be included in power steering regulation algorithm.

Taking into account different tendencies of curves of steering torque to velocity and to lateral acceleration, the relation of steering effort with those quantities ($v$, $a_y$) was examined.

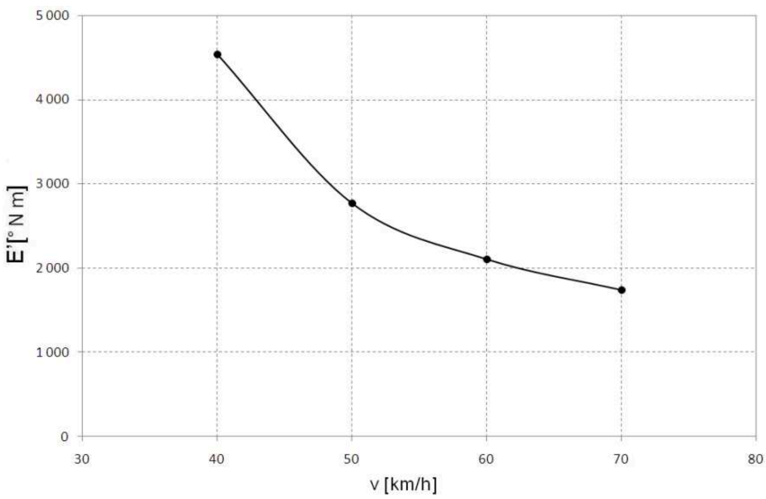


Fig. 8. Steering effort versus velocity, vehicle A.
Rys. 8. Zależność wysiłku kierowania od prędkości jazdy, pojazd A.

An exemplary run of steering effort to velocity is presented in Fig. 8. In order to examine an influence of velocity on steering effort it was assumed that lateral acceleration is constant and equals 4,9 m/s$^2$ as it was in the previous chapter.

It was stated that steering effort decreases when velocity increases. Analysis revealed that the velocity increase causes small changes of steering torque gradient, which is one out of steering effort factors. Steering angle gradient, which is the other steering effort factor, decreases noticeably and causes steering effort decrease. Such tendency was confirmed not only by theoretical research results but also by experimental research results. Within the range of considered velocity values, steering effort decreases by 62%.

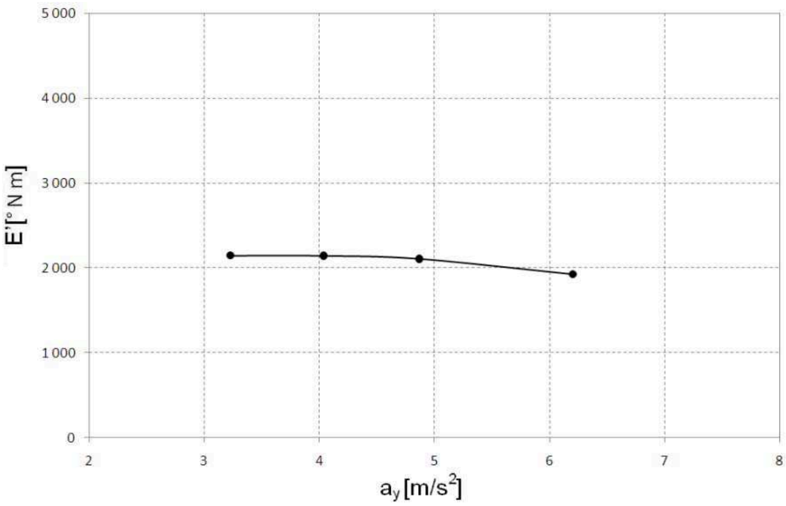


Fig. 9. Steering effort versus lateral acceleration, vehicle A.
Rys. 9. Zależność wysiłku kierowania od przyśpieszenia poprzecznego, pojazd A.

Relation of steering effort to lateral acceleration was also analyzed (Fig. 9).Only small changes of steering effort values in relation to lateral acceleration were observed. Within the range of considered velocity values steering effort decreases by10%.

#### 5.3. Verification of theoretical research results

The adopted models had some simplifications in comparison to real objects. Verification of the obtained results seems to be necessary.

Simulation tests were performed using steering wheel angle values recorded during road tests. Comparison of simulation results with road tests results was made on the basis of steering torque runs recorded during step input and double lane change. Vehicles A and C were selected from the examined vehicles to be examined in simulation tests. Exemplary comparisons of simulation results with road test results made for vehicle C for double lane change are presented below.

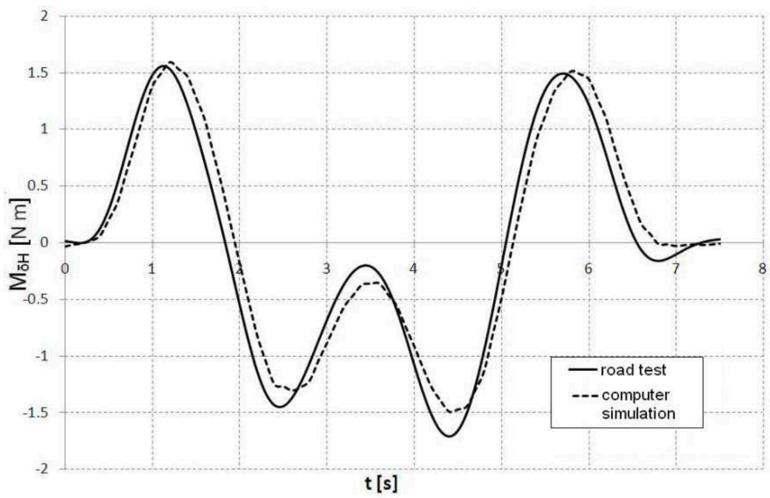


Fig. 10. Comparison of steering torque runs, vehicle C.
Rys. 10. Porównanie przebiegów momentu obrotowego na kierownicy, pojazd C.

In Fig. 10 an exemplary comparison of runs of steering torque versus time is presented. Curves at 50 km/h are plotted as an example.

Exemplary comparison of runs of yaw velocity versus time at 50 km/h recorded during road tests and received by simulation is shown in Fig. 11.

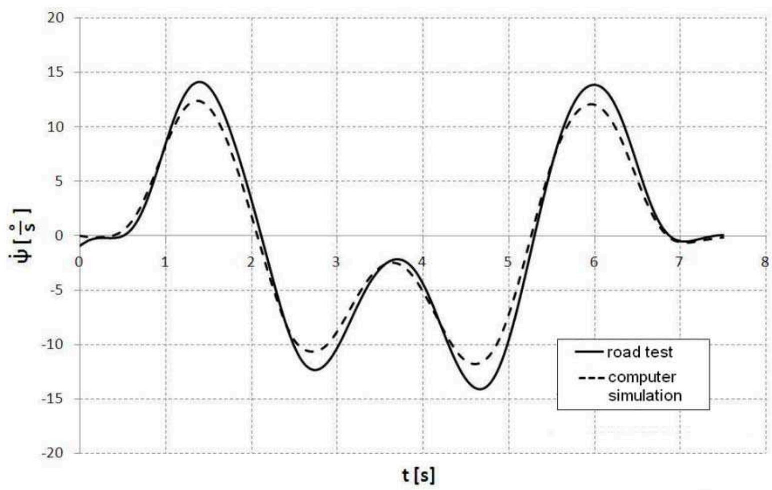


Fig. 11. Comparison of yaw velocity runs, vehicle C.
Rys. 11. Porównanie przebiegów prędkości odchylania, pojazd C.

The comparison of simulation results with road test results showed that differences between them are small and do not exceed 20%. Having compared the accuracy of received results with the accuracy received in the other research centres and presented in literature it was stated that simulation reflects the performance of examined vehicles in reliable way.

## 6. Summary and conclusions

Favourable road feel is very important. It ensures not only comfort but also safety of driving. Road feel is favourable when relation of input data and output data is predictable for a driver and does not change suddenly. Moreover, favourable road feel requires an acceptable steering effort value.

The application of power steering made to control and to change relation of input and output in steering system possible. Nowadays such change is realized by regulation of steering torque value which depends on steering system load and velocity. That kind of regulation may sometimes cause such decrease of steering torque that steering control quality will be assessed as very low by the driver. Forces and torques affecting steering system depend not only on velocity but also on lateral acceleration. It seems to be very important to define a relation between the driver's subjective evaluation of steering control quality and lateral acceleration.

The results of theoretical research confirmed that steering effort that has been defined in the paper can be applied to power steering adjustment.

It was found that steering effort values calculated using formula (1) are similar to values designated by Jaksch [2] as the lower limit of favourable steering effort range. To a certain degree steering effort tends to decrease when velocity is increasing. Considering good correlation between steering effort defined above and driver's subjective rating of steering control quality, it can be stated that power steering adjustment should ensure the decrease of steering effort to a value from the range of favourable steering effort and maintaining that value. Thus, power steering regulation considering steering effort, would guarantee good road feel apart from different motion conditions.

The results of theoretical research revealed that the relation of steering torque to velocity is different than the relation of steering effort to lateral acceleration. These results are similar to the results of experimental research. The relation of steering torque and lateral acceleration is significant. It seems to be insufficient to apply only velocity to adjust power steering gain.

Additionally, the analysis revealed that steering effort may be applied to steering system evaluation. That value refers to both steering torque and steering wheel angle and can be used for evaluation of every car and every manoeuvre. The application of steering effort optimal value and the range of favourable steering effort values are essential.

The approach of steering effort presented in the paper can be applied in further experimental and theoretical research that aims at improvement of power steering regulation. It would be advisable to carry out more experimental tests in order to define the relation of the driver's subjective rating of steering control quality and objective rating of steering control quality.

It would be advisable to perform additional road and simulation tests in order to find and describe the relation of steering effort and velocity that would be recommended in terms of favourable road feel. That relation could be used in power steering adjustment.

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# Kształtowanie charakterystyk wspomagania układu kierowniczego przy wykorzystaniu przyśpieszenia poprzecznego pojazdu


## Streszczenie

W pracy została przestawiona pewna koncepcja modyfikacji wspomagania układu kierowniczego. Ma ona na celu zapewnienie kierowcy korzystnego odczucia kierowania, które jest bardzo istotne podczas kierowania pojazdem. Autorzy pracy wykazują, że przyśpieszenie poprzeczne powinno zostać uwzględnione w algorytmie sterowania regulacją wspomagania układu kierowniczego w celu poprawienia charakterystyki tego układu. Uwzględnienie przyśpieszenia poprzecznego w regulacji wspomagania układu kierowniczego byłoby zrealizowane przez zastosowanie wielkości ściśle z nim związanej, zwanej wysiłkiem kierowania. Zostały przeprowadzone badania eksperymentalne i teoretyczne układów kierowniczych. Na podstawie otrzymanych wyników sprawdzono możliwość zastosowania wysiłku kierowania do regulacji wspomagania układu kierowniczego. Najważniejsze wyniki zostały zaprezentowane i omówione w pracy.