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Motor Vehicles & Motors 2024

ECOLOGY - VEHICLE AND ROAD SAFETY - EFFICIENCY

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PREDGOVOR

U oktobru se na Fakultetu inženjerskih nauka Univerziteta u Kragujevcu tradicionalno održava skup istraživača i naučnika koji se bave proučavanjem motornih vozila, motora i drumskog saobraćaja. Od 1979. do 2004. godine održano je trinaest bienalnih MVM simpozijuma koji su 2006. prerasli u Međunarodni kongres MVM. Od tada je održano devet MVM kongresa, a oktobra 2024. godine Fakultet inženjerskih nauka je organizovao deseti međunarodni kongres MVM od 10. do 11. oktobra 2024. godine.

Na deseti kongres Motorna vozila i motori, MVM2024 dostavljen je veliki broj naučnih radova iz Srbije i inostranstva. Kongres tradicionalno podržavaju Ministarstvo za nauku, tehnološki razvoj i inovacije Republike Srbije, Univerzitet u Kragujevcu, Fakultet inženjerskih nauka i međunarodni časopis „Mobility and Vehicle Mechanics“.

Tema Kongresa MVM 2024 bila je „Ekologija – Bezbednost vozila i na putevima – Efikasnost“. Tokom ovog istraživačkog putovanja, učesnici su puno naučili kroz rad na različitim sekcijama, koje su pokrivale širok spektar tema u vezi sa inženjerstvom u automobilskoj industirji, od fundamentalnih istraživanja do industrijskih primena, naglašavaju interakciju između vozača, vozila i životne sredine i stimulišući naučnu interakcije i saradnju.

Međunarodni naučni odbor u saradnji sa organizacionim odborom izradio je podsticajan naučni program. Program je ponudio preko 54 prezentacije radova, uključujući predavanja po pozivu i radove u sekcijama. Prezentacije na ovom kongresu obuhvatile su aktuelna istraživanja u oblasti motornih vozila i motora sprovedena u 12 zemalja iz celog sveta.

Zadovoljstvo nam je bilo što su nam uvodničari bili profesor Emrulah Hakan Kaleli (sa Tehničkog univerziteta Yıldız, Turska), profesor Ralph Putz (sa Univerziteta Landshut UAS, Nemačka) i profesori Nenad Miljić i Slobodan Popović (sa Univerziteta u Beogradu, Srbija). Izazovi i rešenja u korišćenju vodonika kao goriva za motore sa unutrašnjim sagorevanjem, korišćenje aditiva nanoborne kiseline dodatog u motorno ulje, kao i evropska politika o budućoj mobilnosti na putevima su bile teme uvodnih predavanja.

Sigurni smo da je ovaj program pokrenuo živu diskusiju i podstakao istraživače na nova dostignuća.

10. Kongres MVM 2024. finansijski je podržalo Ministarstvo za nauku, tehnološki razvoj i inovacije Republike Srbije.

Zahvaljujemo se iskusnim i mladim istraživačima koji su prisustvovali i prezentovali svoju stručnost i inovativne ideje na našem kongresu.

Posebnu zahvalnost dugujemo članovima međunarodnog naučnog odbora i svim recenzentima za njihov značajan doprinos visokom nivou kongresa.

Naučni i organizacioni komitet Kongresa MVM2024

FOREWARD

In October, the Faculty of Engineering University of Kragujevac traditionally holds gatherings of researchers and academics who study motor vehicles, engines and road traffic. From 1979 to 2004, thirteen, biennal MVM Symposiums have been held and they grew into an International Congress MVM in 2006. Since then, ninth MVM Congresses have been held, and in October 2024, the Faculty of Engineering organized the tenth International Congress MVM from 10th to 11th October 2024.

A large number of scientific papers from the Serbia and abroad were submitted to the tenth Congress “MVM2024”. Congress is traditionally supported by the Ministry of Science, Technological Development and Innovation of the Republic of Serbia, University of Kragujevac, Faculty of Engineering and the International Journal “Mobility and Vehicle Mechanics”.

The theme of the Congress MVM 2024 was "Ecology - Vehicle and Road Safety - Efficiency". Along this journey we learned from the various sessions, which broadly cover a wide range of topics related to automotive engineering from fundamental research to industrial applications, highlight the interaction between the driver, vehicle and environment and stimulate scientific interactions and collaborations.

The International Scientific Committee in collaboration with the Organising Committee built up a stimulating scientific program. The program offered over 54 presentations, including key-note speakers and paper sessions. The presentations to this conference covered current research in motor vehicle and motors conducted in 12 countries from all over the world.

We were pleased to have professor Emrullah Hakan Kaleli (from Yıldız Technical University, Türkiye), professor Ralph Pütz (from Landshut University UAS, Germany) and professors Nenad Miljić and Slobodan Popović (from University of Belgrade, Serbia) as the keynote speakers, addressing Challenges and solutions in using hydrogen as a fuel for internal combustion engines, using nanoboric acid (nBA) additive added in engine oil, as well as European policy on future road mobility.

We are sure this program will trigger lively discussion and will project researchers to new developments.

The 10th Congress MVM 2024 was financially supported by the Ministry of Science, Technological Development and Innovation of the Republic of Serbia.

We would like to thank experienced and young researchers, for attending and bringing their expertise and innovative ideas to our conference.

Special thanks are due to the International Scientific Board Members and all reviewers for their significant contribution in the high level of the conference.

Scientific and Organizational committee of Congress MVM2024

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SUSPENSION RATIOS OF MACPHERSON STRUT SUSPENSION

ABSTRACT: Bearing in mind the important role of the suspension system in maintaining ride comfort, stability and safety of the car by transferring the forces and moments from the road to the car body in strictly determined manner, the aim of the research presented in this paper was to derive analytically the suspension force ratios. Two suspension ratios were defined – the first at the connecting point between the shock absorber and the car body and the second at the connecting point between the lower arm and wheel hub. The spatial geometrical model of the MacPherson strut suspension was used to establish the relationships between the defined suspension force ratios and main suspension geometry factors. The results of the analysis of the influencing factors on the suspension ratios were presented and discussed. Car designers can use the derived analytical expressions to check how geometrical layout of the suspension characteristic points influences the suspension ratio and use this knowledge to optimize the design.

KEYWORDS: MacPherson strut, suspension ratio, influencing factors

INTRODUCTION

General Motors automotive engineer Earl MacPherson had patented the initial design of the independent wheel strut suspension back in 1947. This design has been known for years as MacPherson strut suspension and it is still one of the most widely used front suspension systems in the range from compact cars to SUVs. The advantages of the MacPherson strut suspension are: simple and compact structure, cost-effectiveness, design flexibility and providing of decent ride comfort. Limitations of this type of suspension include: limited adjustability, tendency for camber changes during cornering, potential impact on the steering system and relatively low lateral stiffness (due to which this system can only be used on lower mass vehicles).

The MacPherson strut suspension has been widely studied through the years. It still occupies the attention of automotive engineers who apply the latest methods of design, control and testing to this suspension system in order to improve its performance in modern cars. The relevant research in the past decade includes the design modifications in order to enhance the steering stability, handling, ride comfort and damping efficiency, using kinematic, dynamic and fatigue analysis [1].

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Modification of geometry of the MacPherson strut in order to enhance ride quality was performed in [2] along with sensitivity analysis of modified MacPherson suspension geometry based on Pearson correlation coefficient. It was concluded that not all parameters should be considered in suspension analysis because of their small impact on suspension performance (for example, it was found that the most sensitive parameter is damping coefficient, while the top mount position could be ignored in geometry design). Since the most of the developed MacPherson suspension models take into account only strut and tyre, they have been convenient for ride analysis but not for car handling analysis. However, the model developed in [3], takes into account the influence of suspension geometry (particularly caster angle and track width) on car handling.

A comprehensive study of the MacPherson suspension spatial kinematics was performed in [4]. The development of kinematic equations was done using Rodrigue's parameters and computation of Gröbner basis for solving the system equations. Two independent system inputs were used – one from the steering system and the other from the road roughness.

The research of dynamics of the MacPherson strut is mainly focused on the means to reduce the influence of lateral loads on suspension components and on the system. Considering the lateral stability of the car equipped with the MacPherson strut, it is important to determine the optimal roll centre height [5] which leads to improvement of the lateral load transfer and handling behaviour. In [6], numerical simulations of deformations and effective Von Misses stresses of the strut and wheel were determined using finite element analysis, for different values of the applied tyre pressures and vertical wheel forces. To minimize shock absorber friction, king pin moment and side loads, coil spring force vector (force line position) was determined experimentally [7] or using Finite Element Method (FEM) analysis [8]. The aim was to obtain the ideal force line position or desired spring force vector. Analysis of side load forces acting on MacPherson suspension strut was the object of study of several research [9-13]. Detection of the side force magnitudes, which could damage the strut components using FEM analysis, can be used in failure prevention [9]. A way to minimize the side loads of suspension strut is to use the larger number of spring coils [10]. Minimization of the strut side load can be obtained using the Kringing model [11] or progressive meta-mode based on sequential approximate optimization [12] to determine optimal spring positions. Considerable reduction of side load can be done by using the C-type coil spring developed in [13] due to its centreline shape and the end coil angular position.

The MacPherson strut suspension exhibits nonlinear behaviour that should be taken into account when forming its mathematical models. Comparisons between the results produced by a two-dimensional mathematical model and a two-dimensional multibody ADAMS/View model for three simulation cases show that most differences occur in values for wheel camber angle due to simplifications in the calculations [14].

The most recent research of MacPherson strut suspension is focused on optimization [15-20] and control [21, 22], including the use of virtual sensors [23]. The optimization objectives are various:

- suspension mechanism kinematics, using the Random search method [15],
- variation in alignment parameters, using neighbourhood activation genetic algorithm [16],
- matching the combinations of parameters of front MacPherson strut suspension with parameters of the rear E-type multilink suspension using optimal matching by Non-dominated Sorting Genetic Algorithm (NSGA-II) at different car speeds [17],
- shock absorber top mount characteristics with experimental verification [18] and
- position of characteristic points from the aspect of kinematics and compliance, using neural network and genetic algorithm [19] or machine learning algorithms [20].

Modern MacPherson strut suspensions have elements that require some form of control. Control algorithms include, for example, implementation of fuzzy-based adaptive sliding mode controller for semi-active MacPherson strut suspension with magneto-rheological (MR) shock absorber [21] or development of a new adaptive moving sliding mode controller and comparison analysis of different control strategies for the same type of suspension [22]. Virtual sensors for virtual measurement of wheel centre loads that are developed using Augmented Extended Kalman Filter may be implemented [23], using information from the other car sensors and the system model.

In recent years, the use of well-developed software packages for multibody modelling in suspension design has overshadowed the research related to derivation of analytical expressions that follow the transfer of forces and moments from the road, through the MacPherson strut suspension to the car body. In [24], kinematic and static planar analysis of relationship between the "installation ratio" and geometrical parameters of the MacPherson strut suspension were conducted. The main research parameters were the instant centre and the installation ratio of the suspension system. The terms like "vertical installation ratio" and the new concept of "lateral installation ratio" were introduced as ratios between the vertical force at the wheel-ground contact and the shock absorber force or lateral shock absorber force, respectively.

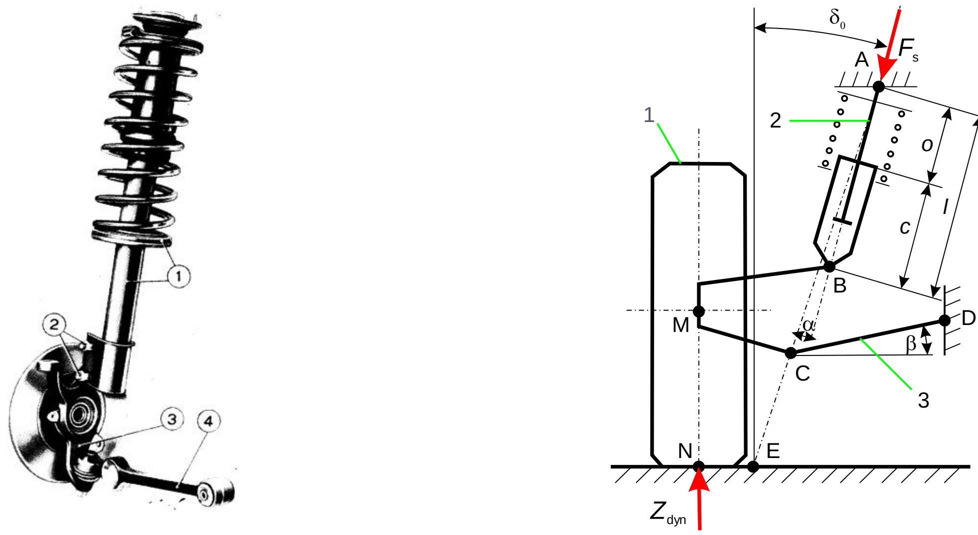
Since the role of the suspension system in transfer of forces and moments from the road to the car body and in reduction of impact loads to the car body is very important, the research presented in this paper was focused on analytical derivation of suspension ratios of the MacPherson strut suspension system. The analytical expressions were derived for the spatial model of the passenger car suspension. The results of the analysis of influencing design variables were presented and discussed.

MCPHERSON STRUT SUSPENSION RATIOS

The MacPherson strut suspension has a simple design, figure 1a. It consists of the strut, which is a combination of shock absorber and spiral coil spring and lower arm. The value of the force acting in the direction of the strut axis, F_s , which is induced by vertical reaction of the road, Z_{dyn} , figure 1, depends on the type of the used suspension system. Thus, the suspension force ratio, i_F , was introduced as a parameter of force transfer from the road to the car body:

$$i_F = \frac{F_s}{Z_{dyn}} \ . \tag{1}$$

In defining the force ratio, an assumption is made that the point N, representing the contact between the wheel and the road, has vertical motion. Also, in definition of kinematic characteristics of the suspension system of the passenger vehicles, usually the normal weight balance is assumed defined by the two-passenger load.



a) b)

Figure 1. *MacPherson strut suspension system: a) actual design (1 - strut, 2 - connection of the strut with wheel hub carrier, 3 - wheel hub carrier, 4 - lower arm), b) geometrical model (1 - wheel, 2 - strut, 3 - lower arm)*

During vehicle's straight-line drive with constant speed, tangential forces are also acting upon the wheels. These are the reactions to the vehicle traction forces and vehicle braking forces. In addition, rolling resistance forces are always present, but they can be neglected due to their comparatively smaller values.

Vertical dynamic reaction acting on the wheel, Z_{dyn} , is the result of dynamical redistribution of the forces during the vehicle motion:

$$Z_{\mathrm{dyn}} = Z_{0\mathrm{dyn}} - \frac{G_{\mathrm{uns}}}{2} \ , \tag{2}$$

where $Z_{0\mathrm{dyn}}$ is the total weight transferred to the road through the wheel during the straight line motion of the vehicle and G_{uns} is the unsuspended weight of the axle.

The force ratios of the McPherson strut suspension at the joints *A* (the connecting point between the shock absorber and the car body) and *C* (the connecting point between the lower arm and the wheel hub carrier) are defined as:

$$i_A = \frac{F_{As}}{Z_{dyn}} \ , \tag{3}$$

$$i_C = \frac{F_{Cs}}{Z_{dyn}} , \tag{4}$$

where F_{As} , F_{Cs} are the projections of the reactions at joints *A* and *C* in the direction of the strut axis (*AB*), respectively.

Figure 2 shows the adopted spatial model of the MacPherson suspension system and coordinate system O_{xyz} used to derive the force ratios. The origin of the used coordinate system, *O*, was located on the ground between the wheels, at the half track width, *s*, laterally from the wheel, in vertical lateral plane going through the axle wheel centres. Characteristic point *i* coordinates were denoted as x_i, y_i and z_i.

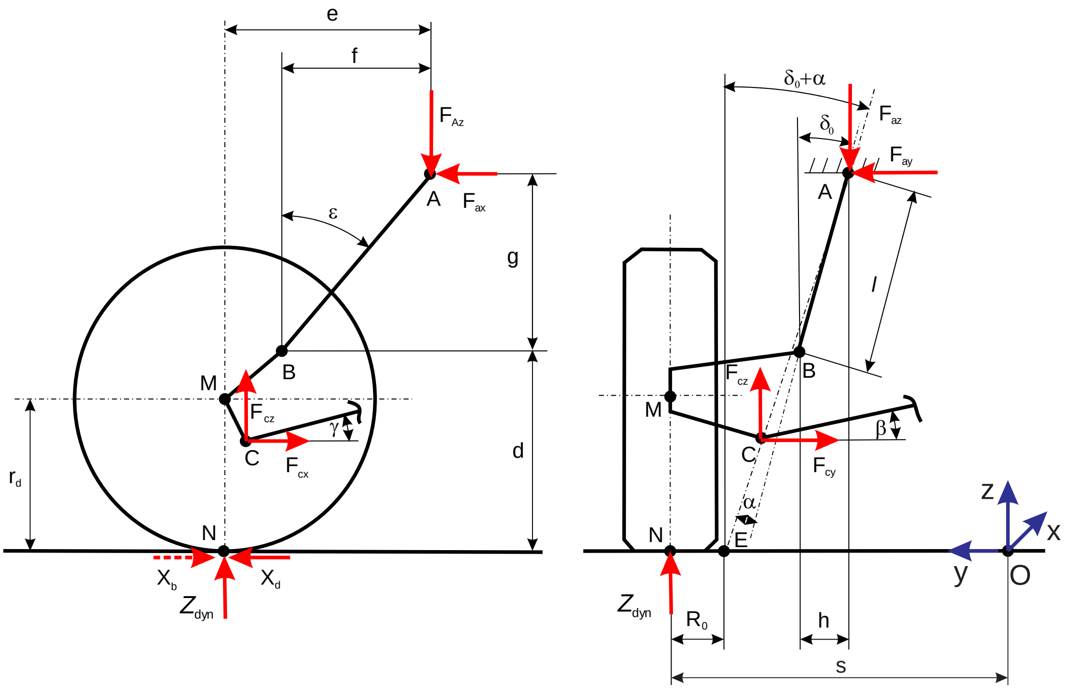


Figure 2. *Spatial model of the McPherson suspension*

The angular position of the strut axis (AB) is determined by the following angles:

- ε - angular displacement of the strut axis from the lateral vertical plane measured in the longitudinal direction and
- δ_0 - angular displacement of the strut axis from the longitudinal vertical plane, measured in the lateral direction.

The position of the lower suspension arm (CD) is determined by the following angles:

- γ - angular displacement of the lower arm axis from the horizontal plane measured in longitudinal direction and
- β - angular displacement of the lower arm axis from the horizontal plane measured in lateral direction.

In order to determine the components of the reactive forces at joints *A* and *C*, values for geometrical distances *e*, *f*, *g* and *h* must be determined. For this purpose, figure 3 presenting the spatial position of the strut can be used.

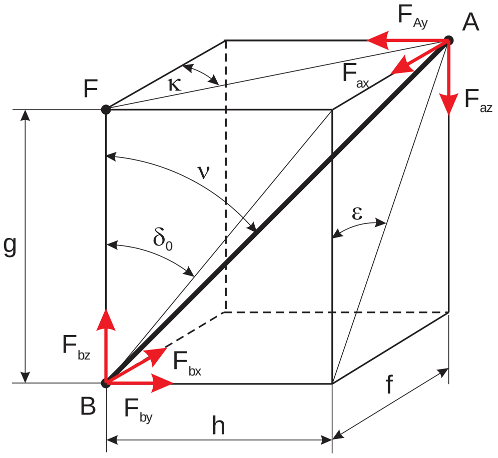


Figure 3. *Spatial position of the strut and components of the reaction forces at joints A and B of the strut*

The angular displacement of the strut, ν, from the vertical axis should be determined first, using the following relations between angles and distances:

$$tg\nu = \frac{\overline{AF}}{g} \ , \tag{3}$$

$$tg\delta_0 = \frac{h}{g} \ , \tag{4}$$

$$tg\varepsilon = \frac{f}{g} \ . \tag{5}$$

Now, the following is valid:

$$tg^2\delta_0 + tg^2\varepsilon = \frac{h^2 + f^2}{g^2} = \frac{\overline{AF}^2}{g^2} = tg^2\nu \ , \tag{6}$$

so the angular displacement, ν, can be calculated as:

$$\nu = arctg\left(\sqrt{tg^2\delta_0 + tg^2\varepsilon}\right) . \tag{7}$$

The geometrical values from figures 1 to 3 can be obtained in the following manner:

$$g = \left|z_A - z_B\right| = \overline{AB} \cdot \cos\nu = \left(c + o\right) \cdot \cos\nu \ , \tag{8}$$

$$h = \left|y_A - y_B\right| = g \cdot tg\delta_0 = \left(c + o\right) \cdot \cos\nu \cdot tg\delta_0 \ , \tag{9}$$

$$f = \left|x_A - x_B\right| = g \cdot tg\varepsilon = \left(c + o\right) \cdot \cos\nu \cdot tg\varepsilon \ , \tag{10}$$

$$e = \left|x_A - x_M\right| . \tag{11}$$

The angular displacement of the strut from the vertical longitudinal plane measured in the horizontal plane, κ, is:

$$tg\kappa = \frac{h}{f} = \frac{tg\delta_0}{tg\varepsilon} \ . \tag{12}$$

The reaction force at joint *C* has spatial direction of the lower arm axis. By following the same methodology for obtaining the geometrical parameters of the spatially located strut, the following relationship for components F_{Cy} and F_{Cz} of the reaction force at joint *C* in lateral and vertical direction, respectively, is obtained:

$$F_{Cz} = F_{Cy} \cdot tg\beta \ , \tag{13}$$

where $tg\beta = \dfrac{\left|Z_C - Z_D\right|}{\left|y_C - y_D\right|}$.

Using the equation of equilibrium of moments acting about the axis going through point *A* and parallel to *x*-axis, the following was obtained:

$$Z_{dyn} \cdot \left|y_A - y_N\right| + F_{Cz} \cdot \left|y_A - y_C\right| - F_{Cy} \cdot \left|z_A - z_C\right| = 0 \ . \tag{14}$$

By substituting F_{Cz} from (13) in (14), the reaction component F_{By} is gained:

$$F_{Cy} = Z_{dyn} \cdot \frac{\left|y_A - y_N\right|}{\left|z_A - z_C\right| - \left|y_A - y_C\right| \cdot tg\beta} = Z_{dyn} \cdot P \ , \tag{15}$$

where

$$P = \frac{\left|y_A - y_N\right|}{\left|z_A - z_C\right| - \left|y_A - y_C\right| \cdot tg\beta} \ . \tag{16}$$

Now, the reaction component F_{Cz} from (13) is:

$$F_{Cz} = Z_{dyn} \cdot P \cdot tg\beta \,. \tag{17}$$

Value of the reaction component F_{Cx} in longitudinal direction is gained from the equilibrium of moments acting about the axis parallel to y- axis and going through point *A*:

$$X_d \cdot z_A + Z_{dyn} \cdot |x_A - x_N| + F_{Cz} \cdot |y_A - y_C| - F_{Cx} \cdot |z_A - z_C| = 0 \,. \tag{18}$$

With substitution of values for F_{Bz} from (17) into (18), the following expression for F_{Bx} is gained:

$$F_{Cx} = X_d \cdot \frac{z_A}{|z_A - z_C|} + Z_{dyn} \cdot \frac{|x_A - x_N| + P \cdot |y_A - y_C| \cdot tg\beta}{|z_A - z_C|} \,. \tag{19}$$

In order to calculate the force ratio at the joint *C* relative to vertical road reaction, it is necessary to determine the value of the force F_{Cs} as the projection of the lower arm force F_C in direction of the strut axis:

$$F_{Cs} = F_{Cy} \cdot \sin\kappa \cdot \sin\nu + F_{Cz} \cdot \cos\nu + F_{Cx} \cdot \cos\kappa \cdot \sin\nu \;, \tag{20}$$

which gives:

$$F_{Cs} = Z_{dyn} \cdot Q + X_d \cdot \frac{z_A}{|z_A - z_C|} \cdot \cos\kappa \sin\nu \;, \tag{21}$$

where:

$$Q = P \cdot \sin\kappa \cdot \sin\nu + P \cdot tg\beta \cdot \cos\nu + \frac{|x_A - x_N| + P \cdot |y_A - y_C| \cdot tg\beta}{|z_A - z_C|} \cdot \cos\kappa \cdot \sin\nu \,. \tag{22}$$

Now, the force ratio at joint *C*, i_C , may be written as follows:

$$i_C = \frac{F_{Cs}}{Z_{dyn}} = Q + \frac{X_d}{Z_{dyn}} \cdot \frac{z_A}{|z_A - z_C|} \cdot \cos\kappa \sin\nu \;, \tag{23}$$

or, considering that $\phi_d = \dfrac{X_d}{Z_{dyn}}$ is the engaged adhesion coefficient:

$$\boxed{i_C = \frac{F_{Cs}}{Z_{dyn}} = Q + \phi_d \cdot \frac{z_A}{|z_A - z_C|} \cdot \cos\kappa \cdot \sin\nu \,.} \tag{24}$$

Components of reaction force at joint *A* are obtained from the equations of the equilibrium of forces:

$$F_{Ay} = F_{Cy} = Z_{dyn} \cdot P \;, \tag{25}$$

$$F_{Ax} = F_{Cx} - X_d = X_d \cdot \frac{z_A - |z_A - z_C|}{|z_A - z_C|} + Z_{dyn} \cdot \frac{|x_A - x_N| + P \cdot |y_A - y_C| \cdot tg\beta}{|z_A - z_C|} , \tag{26}$$

$$F_{Az} = F_{Cz} + Z_{dyn} = Z_{dyn} \cdot (P \cdot tg\beta + 1) \,. \tag{27}$$

By projecting them in the direction of the strut axis, the following is obtained:

$$F_{As} = F_{Ay} \cdot \sin\kappa \cdot \sin\nu + F_{Az} \cdot \cos\nu + F_{Ax} \cdot \cos\kappa \cdot \sin\nu \;, \tag{28}$$

or

$$F_{As} = Z_{dyn} \cdot R + X_d \cdot \frac{z_A - |z_A - z_C|}{|z_A - z_C|} \cdot \cos\kappa \cdot \sin\nu \;, \tag{29}$$

where:

$$R = P \cdot \sin\kappa \cdot \sin\nu + \left(P \cdot tg\beta + 1\right) \cdot \cos\nu + \frac{\left|x_A - x_N\right| + P \cdot \left|y_A - y_C\right| \cdot tg\beta}{\left|z_A - z_C\right|} \cdot \cos\kappa \cdot \sin\nu \ . \tag{30}$$

The force ratio at the joint *A* (where the shock absorber connects with the car body) is as follows:

$$i_{\mathrm{A}} = \frac{F_{\mathrm{As}}}{Z_{dyn}} = R + \frac{X_d}{Z_{dyn}} \cdot \frac{z_A - \left|z_A - z_C\right|}{\left|z_A - z_C\right|} \cdot \cos\kappa \cdot \sin\nu \ , \tag{31}$$

or, since $\phi_d = \dfrac{X_d}{Z_{dyn}}$ is the adhesion utilization (engaged adhesion coefficient) during driving, the final form is:

$$\boxed{i_{\mathrm{A}} = \frac{F_{\mathrm{As}}}{Z_{dyn}} = R + \phi_d \cdot \frac{z_A - \left|z_A - z_C\right|}{\left|z_A - z_C\right|} \cdot \cos\kappa \cdot \sin\nu \ .} \tag{32}$$

If vehicle brakes, then the brake force X_b (figure 2) must be used for calculation of force ratios. The procedure for calculation of force components at joints *A* and *C* remains the same. The following expressions for suspension ratios in case of braking are obtained:

$$\boxed{i_C = Q - \frac{X_b}{Z_{dyn}} \cdot \frac{z_A}{\left|z_A - z_C\right|} \cdot \cos\kappa \cdot \sin\nu = Q - \phi_b \cdot \frac{z_A}{\left|z_A - z_C\right|} \cdot \cos\kappa \cdot \sin\nu \ ,} \tag{33}$$

$$\boxed{i_{\mathrm{A}} = R - \frac{X_d}{Z_{dyn}} \cdot \frac{z_A - \left|z_A - z_C\right|}{\left|z_A - z_C\right|} \cdot \cos\kappa \cdot \sin\nu = R - \phi_b \cdot \frac{z_A - \left|z_A - z_C\right|}{\left|z_A - z_C\right|} \cdot \cos\kappa \cdot \sin\nu \ ,} \tag{34}$$

where ϕ_b is the adhesion utilization (engaged adhesion coefficient) during braking.

ANALYSIS OF MCPHERSON STRUT SUSPENSION RATIOS

Quasi-static calculation of possible spatial positions of the characteristic points of the suspension system was conducted for straight line drive of the car, with the usual assumptions that the suspended car mass is immovable and that the wheel is absolutely rigid. These assumptions were introduced in order to establish possible spatial configurations of the suspension system elements, bearing in mind constructive limitations. As a consequence, vector positions of point A (joint between shock absorber and the car body) and point D (joint between the lower arm and car body) remained unchanged ($\vec{r}_A = const, \quad \vec{r}_D = const$). Since the observed mechanisms consist of rigid body elements, the laws of motion of some points had to fulfil conditions of constant distances, like the constant length of the lower arm ($\left|\vec{r}_C - \vec{r}_D\right| = const$).

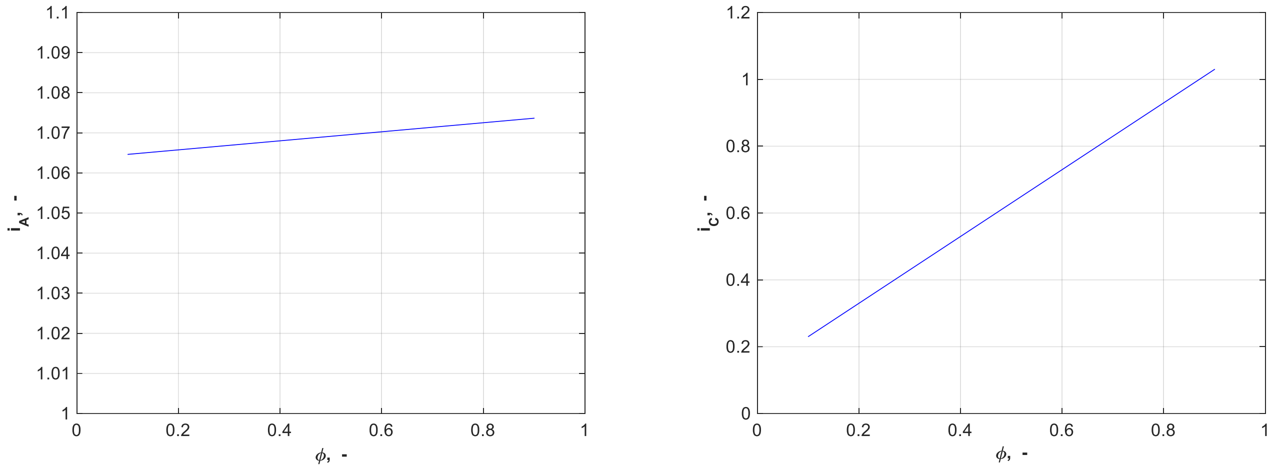
Coordinates of the characteristic points used for calculation are presented in Table 1. The coordinates were established from the constructive documentation of the observed passenger car with MacPherson strut suspension on its front wheels.

Table 1. *Coordinates of characteristic points of the suspension system in static condition*

Point	*x*, mm	*y*, mm	*z*, mm
A	-25.0	522.0	829.0
B	-6.0	612.0	340.0
C	0.0	640.0	189.0
D	-143.5	379.5	215.0
M	0.0	704.5	259.0
N	0.0	700.0	0.0

By analysis of the analytical expressions for suspension ratios from equations (24) and (32-34), the influence of the adhesion utilization (engaged adhesion coefficient) on the force ratios during driving can be established, figure 4. This means that, in the case of the spatial shock absorber position, the force ratios depend not only on design parameters but also on the conditions of exploitation (intensity of the force reactions on the wheels).

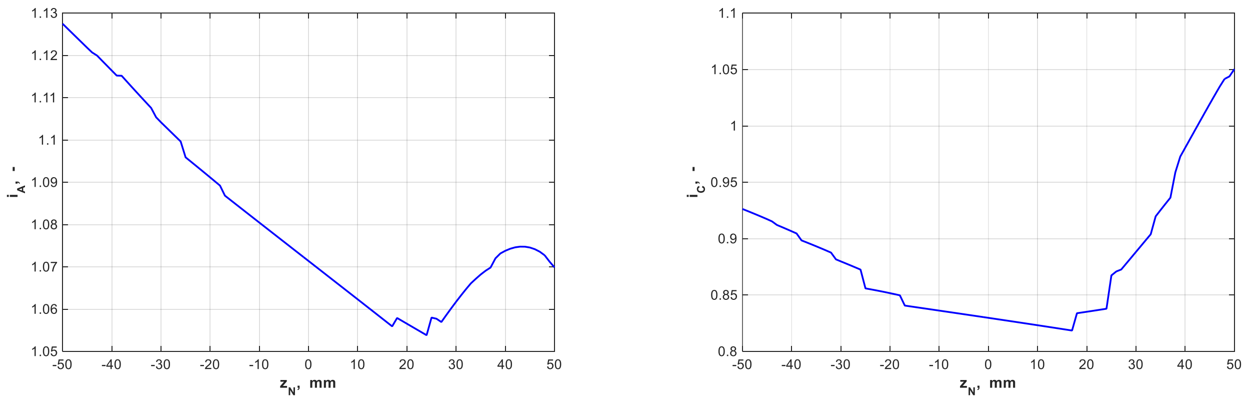
The influence of the tangential force sign (whether it is drive force or brake force) is also obvious from suspension ratio equations. Since the values for Q and P are positive for real design parameters, the force ratios during acceleration are higher than during braking. It is obvious that the suspension ratio at the connecting point A between the shock absorber and the car body is very close to 1 for a range of values for adhesion utilization.



a) b)

Figure 4. *Influence of the adhesion utilization during driving on suspension ratios: a) i_A , b) i_C*

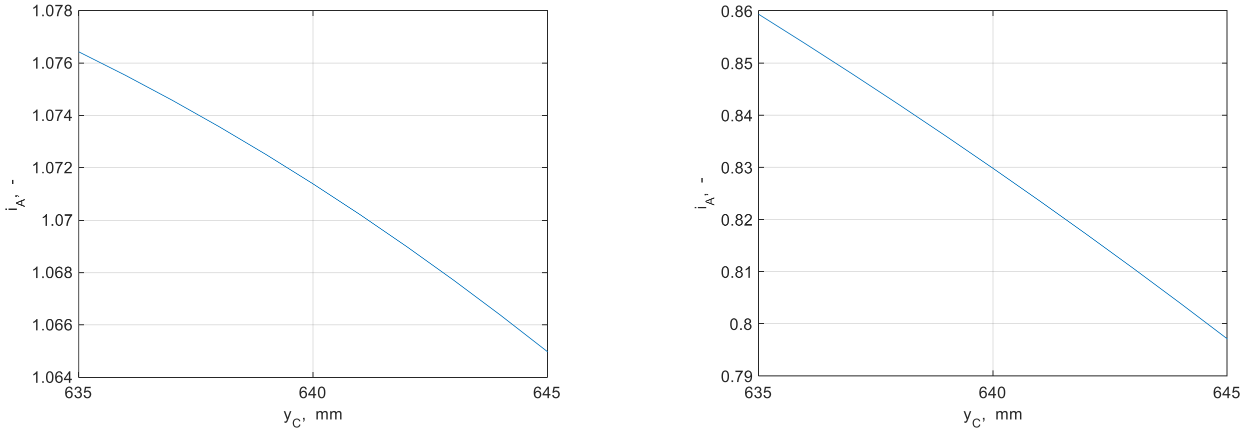
Influence of the road profile height, represented with vertical motion of the point N, z_N, on the values of suspension ratios is presented in figure 5. Compression of the suspension system ($z_N > 0$) generally induces the reduction of the values of suspension ratios i_A and i_C , while its extension ($z_N < 0$) generally induces the increase in suspension ratio. Presented curves of suspension ratios reflect the nonlinear behaviour of the system and also show the consequence of limited motion of the joint C. Namely, joint C (lower arm joint connected to the wheel carrier) can move only along the spherical surface that has diameter equal to lower arm length. This fact influences the motion of all other connected elements of the suspension system and it represents another constraint in suspension system kinematics. Hence, characteristic points exhibit complicated motions that result in constant changes in relevant angular positions of suspension elements transferring its influence to the suspension ratios.



a) b)

Figure 5. *Influence of the road profile height on suspension ratios: a) i_A , b) i_C*

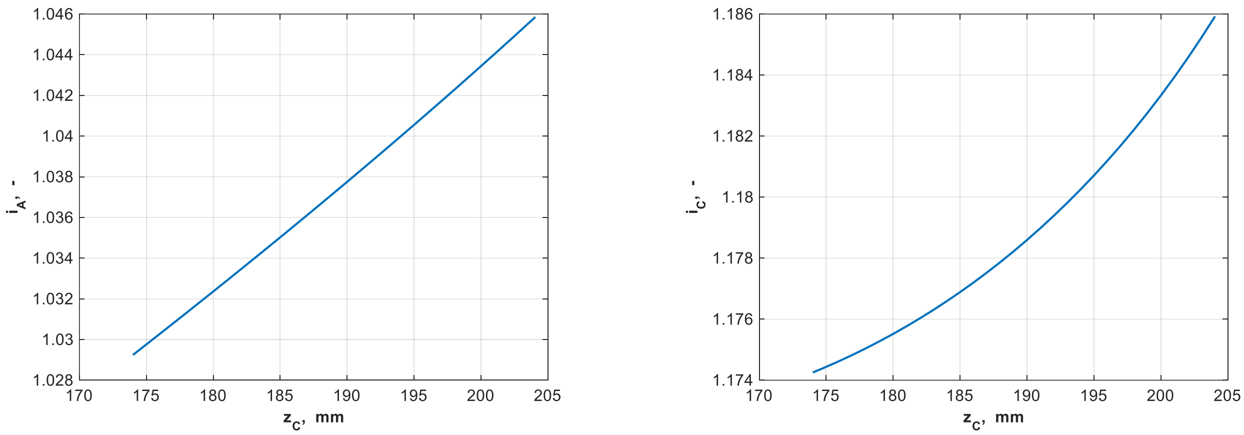
Figure 6 shows the influence of small lateral movement of the joint C on the observed suspension ratios. When joint C moves laterally closer to the wheel plane, suspension ratios i_A and i_C have decreased values, while the lateral distancing of the joint C from the wheel brings higher values of suspension ratios.



a) b)

Figure 6. *Influence of the lateral position of the joint C on suspension ratios: a) i_A , b) i_C*

Increase of vertical position of the joint C induces the increase in values of the suspension ratios, while decrease in vertical position induces the decrease in values of the suspension ratios, as can be seen in figure 7.



a) b)

Figure 7. *Influence of the vertical position of the joint C on suspension ratios: a) i_A , b) i_C*

It is obvious that spatial positions of characteristic points have direct influence on the values of the suspension ratios. These complicated, limited and mutually connected motions result in constant changes in relevant angular positions of the suspension elements and the wheels, which also induce the changes in suspension ratio. The obtained values for the suspension ratio i_A confirm the previous findings that MacPherson system suspension ratio values are close to 1.

CONCLUSIONS

The developed spatial model of MacPherson strut suspension was used to research the influence of various factors (design variables, type of driving, wheel force reactions) on suspension force ratios. The existing kinematic limitations were taken into account. The case of the straight line drive of the car was observed, so the influence of the steering system was not taken into account.

It was shown that the suspension ratio depends not only on initial geometric and kinematic properties of the wheel-suspension system, but also on adhesion utilization and the type of car motion (whether the car drives or brakes). The increase of adhesion utilisation during diving causes the small increase in values of the suspension ratio at joint A, but larger increase of the suspension ratio at the joint C. When the car breaks, the opposite occurs - the increase in adhesion utilization induces the decrease in suspension ratios.

The height of the road profile represented by the vertical motion of wheel-road contact point N has complex influence on suspension force ratios. The compression of the suspension system increases the values of suspension ratios, while rebound action brings more complex changes in suspension ratios influenced by limitations in motions of some characteristic points of the suspension system model.

Lateral positioning of the lower arm joint C closer to the wheel decreases the suspension ratio, while its moving away from the wheel brings lower values of the suspension ratios. Increase in vertical position of the joint C gives greater values of the suspension ratios.

The developed software can be used for variation of values of initial coordinates of characteristic points in order to obtain the optimum values in the phase of suspension system design. Obviously, the presented model can be improved by adding the elements of steering system to research the curvilinear motion of the car and the influence of the interaction between the steering and suspension systems on suspension force ratios.

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