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OPTIMIZATION OF 5-ROD CAR SUSPENSION FOR ELASTOKINEMATIC AND DYNAMIC CHARACTERISTICS

The paper presents optimization of 5-rod (5-link) suspension mechanism used in passenger cars for independent guiding of the wheels. Selected stiffness coefficients defined for five elastomeric bushings installed in joints of the suspension rods are considered as design variables. Two models with lumped parameters (i.e. elastokinematic and dynamic) of wheel-suspension-car body system are formulated to describe relationships between the design variables and the performance indexes including car active safety and ride comfort, respectively. The multi-criteria goal function is minimized using a deterministic algorithm. The suspension with optimized bushings rates fulfils desired elastokinematic criteria together with a defined dynamic criterion, describing the so-called rolling comfort. An event of car passing over short road bump is considered as dynamic conditions. The numerical example deals with an actual middle-class passenger car with 5-rod suspension at the front driven axle. Estimation of the models parameters and their verification were carried out on the basis of indoor and outdoor experiments. The proposed optimization procedure can be used to improve the suspension design or development cycle.

1. Introduction

Guiding mechanism for vehicle wheels can be considered as a multi-body system composed of rigid links (wheel carriers, rods, subframe/bogie, chassis) with elastic elements (main spring, tire) and damping components (shock absorber), linked with each other by kinematic joints and compliant joints. The latter joints, made in a form of some elastomeric bushings (silent-blocks) or mounts, are used in the suspension rods and subframe to

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vibroisolate the car body and to properly guide the wheels under exerted load. In that way vehicle performance in terms of ride and acoustic comfort, handling and design/assembly economy is influenced.

In order to better meet conflicting criteria, a tendency of taking the advantage of suspension mechanisms with more links and elastic joints can be currently observed [11]. More links means more design freedom, but at the expense of need to use of more complicated computational approaches. Five-rod (also known as five-link or five-point) suspension mechanism of the car wheels can be considered as an example of an in-parallel mechanism, like Stewart platform [7, 8]. Other structures of the independent wheel suspensions (e.g. double wishbone, McPherson) can be treated as a simplified case of 5-link mechanism [11]. A lot of studies were carried out concerning kinematic and elastokinematic analysis of suspension mechanism, including 5-link type as well [4, 7]. Fewer known papers are devoted to dynamic analysis [3, 10]. Problem of multi-link suspension synthesis is very rare in available literature [6, 8].

Main dimensions of the wheel suspension mechanism are usually completely determined by design specifications concerning gross motions of the linkage with rigid links and kinematic joints. Various characteristics are taken into account in the suspension dimensional synthesis for desired wheel bounce and steer displacements, like for example: variation of toe and camber angles of the wheel, lateral track; anti-dive/squat properties and location of the virtual steering axis [9]; workspace requirements; link interference and joint-limits; force specifications and others [11].

In a successive step of the suspension synthesis, stiffness parameters of the elastic joints are usually selected, based on some specifications concerning car handling, ride, durability, and costs [6]. The spatial compliance of the wheel guiding mechanism as function of its position and bushings compliance coefficients can be studied by using elastokinematic characteristics [4, 7], i.e. elastic change of wheel position and orientation with respect to the car body under quasi-static load applied to the wheel. The optimization problem of the bushings rates (in linear case) for desired elastokinematic characteristics, like toe angle vs. braking force or camber angle vs. lateral force, was solved by the authors [7].

However, the elastomeric bushings, considered as an intermediate vibroisolation level of the car body, should guarantee a proper dynamic performance of the suspension system [5, 10], besides the elastokinematic features. The dynamic loads imposed on suspension components during vehicle driving are transferred through bushings and joints. A ride comfort can be evaluated, i.e. during a car maneuver of passing over single road unevenness. Vertical acceleration of the sprung mass and variation of the tire normal force are usually taken in such cases for evaluation of car comfort and active safety (road holding), respectively [2]. These criteria are influenced mainly by tire properties, suspension compliance and damping in direction of the wheel bounce motion [2, 11]. Besides of the described vertical dynamics, the suspension mechanism transmits longitudinal loads to the car body, responsible for generation of longitudinal acceleration and jerk acting on vehicle occupants and cargo. The longitudinal compliance of the wheel guiding mechanism plays here the most important role and it is mainly influenced by the elastomeric bushings compliance and damping [5, 13]. This feature is directly connected with the so-called suspension rolling comfort responsible for attenuation of road-wheel harshness and noise [11].

In this paper, the authors introduce a novel algorithm for solving the synthesis problem of 5-rod (5-link) wheel guiding mechanism with a known kinematic model. Five stiffness coefficients of elastomeric bushings installed in joints of the suspension rods are considered as design variables. The design objectives contain the so-called elastokinematical specifications, what was considered by the authors already [8], but a defined dynamic criterion, concerning the so-called rolling comfort, is now taken into account additionally. Two models, i.e. elastokinematic and dynamic, of wheel-suspension-car body system are formulated to describe relationships between the design variables and the defined performance indexes. Next, the experimentally verified models are implemented into an optimization routine. The multi-criteria goal function is minimized using a deterministic algorithm. The numerical example deals with an actual middle-class passenger car with 5-rod suspension at the front driven axle. An event of car passing over a road bump with short length and sharp edges is considered. The proposed optimization procedure can be used in the suspension design or development cycle.

2. Model description

The model of road-wheel-5-rod suspension-car body system, presented in Fig. 1, is intended to use for simplified analysis of the vehicle passing over a single road unevenness. This model (H_1) is dynamic, deterministic, nonlinear with physical and functional characteristics, described by lumped parameters. The model is implemented into an optimization algorithm, therefore it should strike a balance between computational accuracy and effectiveness.

The following assumptions for the H_1 model are made:

– model describes a quarter of the vehicle due to the vehicle and the excitation symmetry;
– the front wheel response is concerned in frequency domain up to ca. 20Hz;

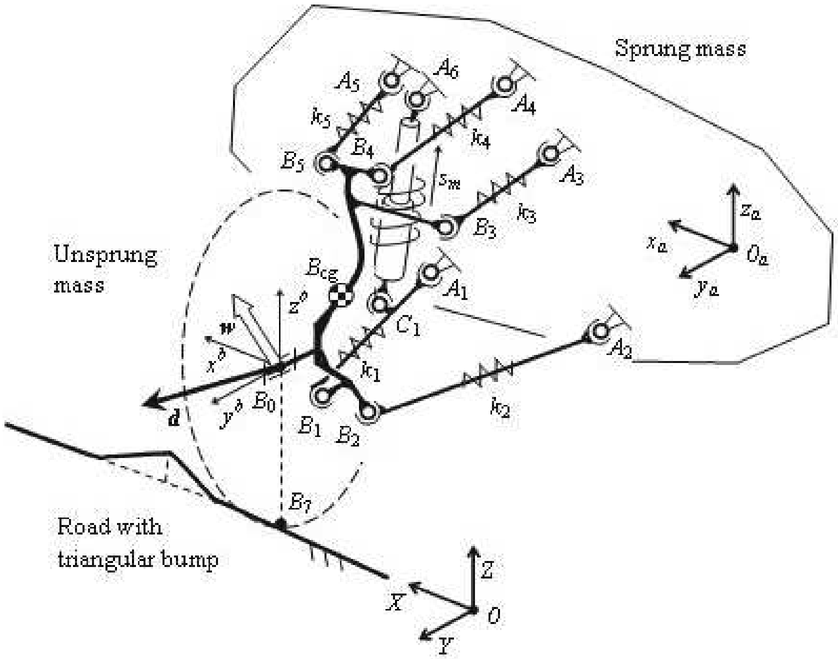

Fig. 1. Quarter-car model (H_1) of wheel-5rod suspension-sprung mass system (front left wheel)

- steering, power train and brake system are neglected;
- the wheel load excited by the rigid road bump is transmitted to the knuckle hub;
- the wheel knuckle undergoes spatial vibrations, constrained by five rods of the suspension and spring-damper module, which transmit forces to the car body;
- the car body (sprung mass) undergoes in-plane (rigid) vibrations without pitch angle;
- the suspension rods include an equivalent longitudinal compliance [7, 10] as a consequence of radial compliance of the elastomeric bushings installed in the rod joints; thus each rod is described by linear stiffness rates ($k_i, i = 1 \ldots 5$) considered as the design variables;
- the shock absorber (telescopic, hydraulic damper) model is formulated as a rheological one [1, 15], including nonlinear spring-damper elements in serial and parallel connections;
- elastic components of the shock absorber assembly (C_1A_6 in Fig. 1), like main coil spring, jounce bumper, internal rebound spring and top mount, are described by static force-deflection characteristics;
- radial-interradial spring model [12] of the pneumatic tire is used to describe its in-plane characteristics like filtering of short length road profile

and one-sided constraints; the tire model generates vertical and horizontal forces at the wheel hub; tangential force in tire contact patch are not included.

In the goal function describing purely elastokinematic criteria for the front wheel suspension mechanism, a simplified version of the H_1 model is used. The following assumptions for this elastokinematic model (H_2) are made:

– the model is described by geometrical and stiffness parameters only;
– the wheel is supported on the base platform with bearings, which is loaded in a defined direction in a quasi-static fashion; and the car body is fixed, what corresponds to conditions of measurements on Kinematic@Compliance test stand [7, 10].

3. Problem formulation

In order to describe a spatial displacement and a quasi-static change of the external load reduced to the wheel center (B_0 in Fig. 1), the following spatial vectors (twist and wrench) are used, respectively [7]:

$$[\boldsymbol{d}]_{6x1} = [\boldsymbol{q}^T \boldsymbol{\varepsilon}^T]^T; \qquad [\boldsymbol{w}]_{6x1} = [\boldsymbol{f}^T \boldsymbol{m}^T]^T \tag{1}$$

where: the respective vectors (three-by-one) describe: $\boldsymbol{q}$ – translation, $\boldsymbol{\varepsilon}$ – infinitesimal rotation, $\boldsymbol{f}$ – force increment, $\boldsymbol{m}$ – torque increment.

The design variables, i.e. stiffness coefficients for each rod (Fig. 1), are collected in the vector:

$$[\boldsymbol{k}\,]_{5x1} = [k_1\; k_2\; ...\; k_5]^T \tag{2}$$

The rest of the model parameters are kept as constant.

Input data for the elastokinematic optimization includes selected p-points on the desired characteristic of the wheel elastic displacement ($\boldsymbol{d}_d$) under quasi-static load ($\boldsymbol{w}_d$) change. Response of the model H_2 for the given load ($\boldsymbol{w}_d$), is a spatial displacement ($\boldsymbol{d}$).

The following elastokinematic (vector) residua should be minimized:

$$r_{e,i} = (d_{d,i} - d_i)^2; \qquad i = 1\; ...\; p \tag{3}$$

where: $d_i = h_2(\boldsymbol{k})$ for given $\boldsymbol{w}_d$, and h_2 stands for a function of the model H_2.

The additional criterion included in the optimization procedure concerns a vibratory response (up to ca. 20 Hz) of the dynamic model H_1 (Fig. 1) after passing over a single road bump with a given initial velocity (v_x), what excites damped in time (t) vibrations in the wheel-suspension-sprung mass system.

Each rod (with bushings) of the suspension mechanism transmits force to the car body. A longitudinal component of their net force contributes to longitudinal vibrations of the car occupants. Dispersion of the longitudinal acceleration (a_x), measured e.g. by variance function, should be minimized in order to improve ride comfort [2]. The considered dynamic (scalar) criterion can be therefore written as:

$$r_d = variance[a_x(t)] \tag{4}$$

where: $a_x(t) = h_1(\boldsymbol{k})$ for a given bump and velocity, and h_1 stands for a function of the model H_1.

The multi-criteria optimization problem is formulated as the following:

$$min\ [\boldsymbol{r}] \tag{5}$$

where: $[\boldsymbol{r}]_{mx1} = [\boldsymbol{r}_e^T\ r_d]^T; \quad 5 \leq m.$
under inequality constraints: $\boldsymbol{k}_{min} < \boldsymbol{k} < \boldsymbol{k}_{max}$.

It is assumed that the vector goal function (5) is continuous in the decision space. The design variables (2) and the goal function (5) were normalized before optimization in order to improve the problem conditioning. The multi-objective problem (5) is solved using a goal attainment algorithm (Matlab [14]), which belongs to classical gradient based methods. In order to find a global minimum this deterministic algorithm was launched a few times from randomly chosen starting values.

4. Estimation of the model parameters and verification

The numerical example deals with an actual middle-class (ca. 1400 kg) passenger car with 5-rod suspension at front driven axle and twisted beam rear axle. Parameters (concerning geometry, mass, stiffness and damping) of the model were estimated on the basis of results of lab experiments with individual components, mainly using MTS test rig [16]. Selected parameters are given in Appendix.

The characteristics of the bushings radial force as function of radial deflection were linearised in the vicinity of working point adequate to load state in the suspension mechanism in the design position [9].

The parameters of elastic components of the shock absorber assembly (C_1A_6 in Fig. 1) were estimated on the basis of determined quasi-static characteristics of force as function of deflection, which is presented in Fig. 2, where positive forces and deflections correspond to the extension of the assembly, and negative – correspond to the compression.

The model with estimated parameters fits very well to the measured characteristics, Fig. 2, which can be divided into ranges where the main

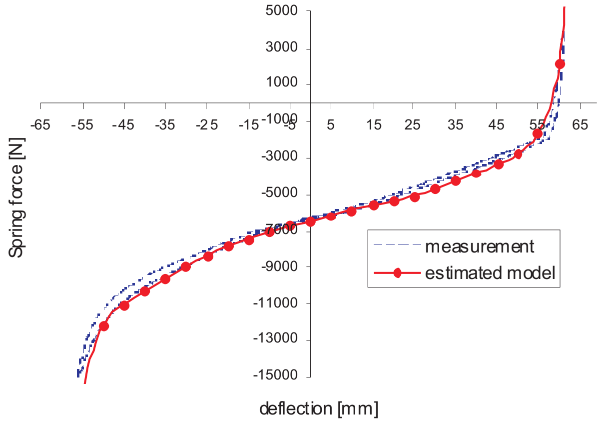

Fig. 2. Spring force as function of deflection of the shock-absorber assembly

coil spring, jounce bumper and internal rebound spring can be identified [2, 11]. Damping parameters of the shock absorber assembly were estimated on the basis of damping force vs. velocity characteristics (Fig. 3) obtained for sinusoidal excitation on MTS test stand.

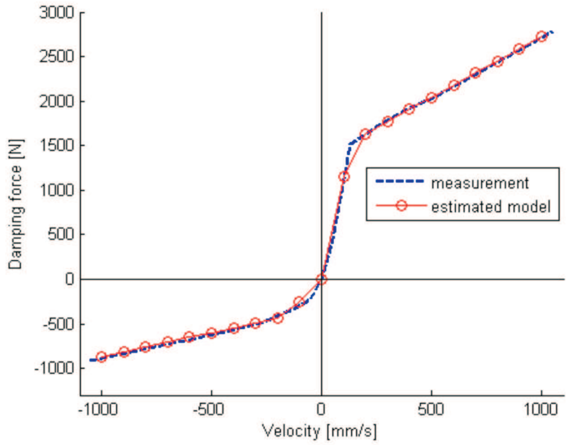

Fig. 3. Steady-state damping force as function of shock-absorber velocity

Parameters of the radial-interradial spring model of the pneumatic tire (205/55 R16 91V) were estimated on the basis of the experimental results obtained by pressing the wheel against road with various normal load and tire pumping pressure. Exemplary characteristics of normal force as function of the tire radial deflection in the case of a flat support and different pumping pressures (2.3 and 2.8 bar) are presented in Fig. 4. Results from the measurements are compared here with the results obtained by using the tire model with estimated parameters. Similar characteristics were obtained in the case of pressing the wheel against the road bump, which was used during the road tests.

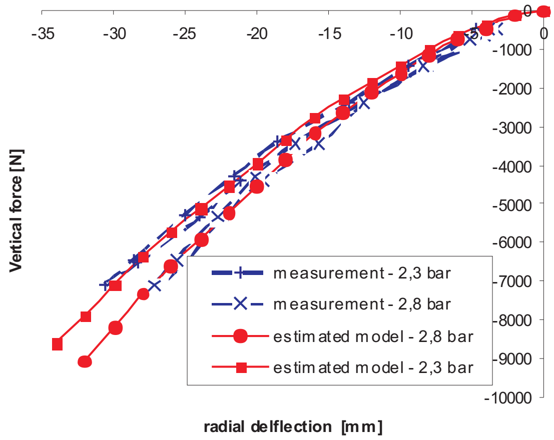

Fig. 4. Tire vertical force acting on a flat support as function of tire radial deflection for different pumping pressures. Comparison of the results obtained from measurements and simulation using tire model

Verification of the prepared models, with the parameters already estimated, was carried out based on several lab and road tests. Kinematic and elastokinematic characteristics of the 5-rod front suspension were studied by the authors in detail, using Kinematic&Compliance test rig [7], where position and orientation of the wheel carrier can be measured under various conditions. Kinematic characteristics were considered when the mechanism was guided virtually without load (the spring-damper assembly was dismounted). Exemplary results concerning a relationship between y and z coordinates of the wheel center (Fig. 1) for the wheel bounce motion are presented in Fig. 5a. Elastokinematic characteristics were considered when the mechanism was loaded through the tire contact patch in a quasi-static fashion. The determined relationship between x coordinate of the wheel center (Fig. 1) and longitudinal force is presented in Fig. 5b.

During the road tests, the considered car was passing over a single bump (the same for left and right wheels) with section of isosceles triangle (5 cm height and 20 cm base). In order to estimate the car motion state, the car body and the wheel knuckle were instrumented with acceleration and gyro sensors. Stroke (s) of the shock absorber (C_1A_6 in Fig. 1) was estimated using wire displacement sensor. Additionally, longitudinal forces in four rods

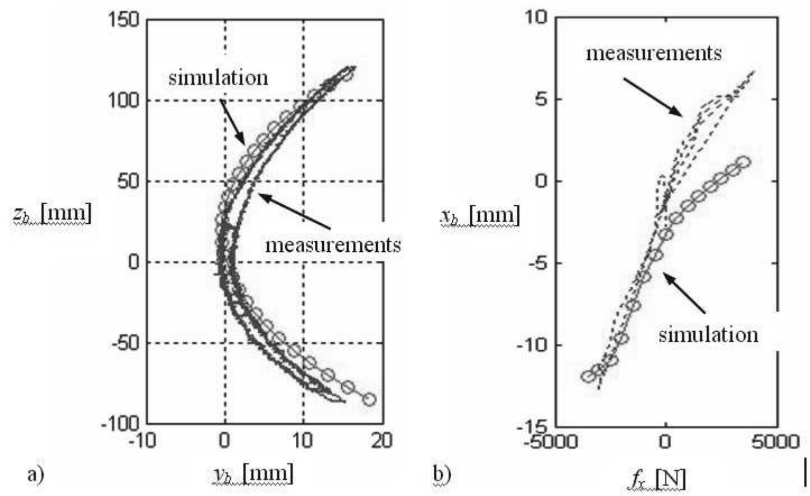

Fig. 5. a) Relationship between y and z coordinates of the wheel center for the bounce motion during kinematic test. b) Change x coordinate of the wheel center under quasi-static longitudinal force during elastokinematic test

of the front suspension and in the shock absorber-spring assembly were measured using strain gages to estimate the load state. The measured and simulated time response of stroke of the spring-damper module in the case when the front wheel crosses over the bump with velocity v_x = 60 km/h is presented in Fig. 6a. The wheel hop frequency (14 Hz) and the primary ride frequency (1.3 Hz) can be noticed in the considered output. The response of the longitudinal acceleration (a_x) of the car body (Fig. 6b) is affected by longitudinal vibrations of the wheel-carrier system, excited by the road bump. This output is described by higher frequency and smaller damping than the wheel hop response.

Table 1.

Load-displacement (***w-d***) specifications for synthesis of front driven wheel (LH-side)

Maneuvers	M1: pure braking	M2: pure driving	M3: pure cornering
Load change at the wheel centre	f_x, m_y	f_x	f_y, m_x, m_z
Recommended ranges of displacement	(1) $-0.1 \leq \varepsilon_z \leq 0[°]$ for $f_x = -1$ kN	(2) $0 \leq \varepsilon_z \leq 0.15[°]$ for $f_x = 1$ kN	(3) $-0.05 \leq \varepsilon_z \leq 0[°]$ (4) $-0.3 \leq \varepsilon_x \leq 0[°]$ for $f_y = -1$ kN
where: $m_x = f_y r_w$; $m_y = -f_x r_w$; $m_z = -f_y t$; $r_w = 0.31$ m; $t = 0.03$ m			

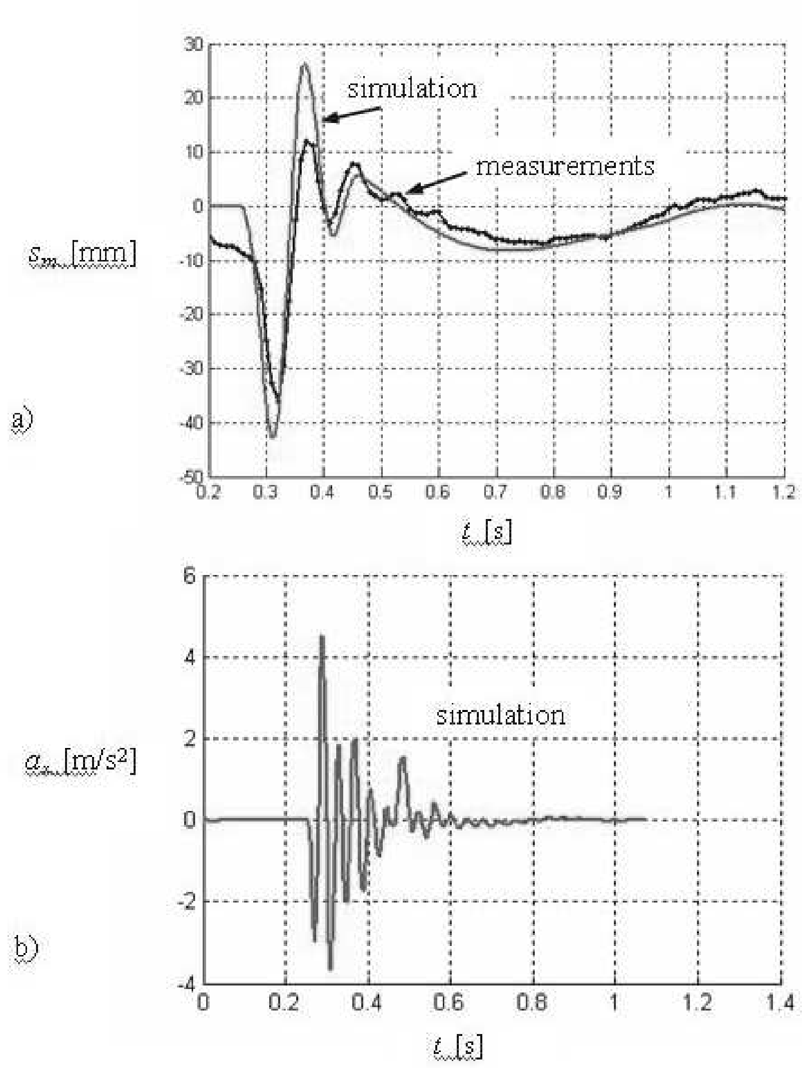

Fig. 6. Time responses of: a) s_m – stroke of the spring-damper module and b) a_x – longitudinal acceleration of the car body, when the front wheel crosses over the road bump with velocity v_x = 60 km/h

5. Results of the stiffness optimization

The verified models were implemented into an optimization algorithm in order to find stiffness rates of the rods bushings meeting the defined criteria in the best way.

The desired elastokinematical characteristics can be achieved by an ideal wheel guiding mechanism with stiffness properties which are strongly diversified and often contradictory. Recommended ranges [8, 11] of selected components of the wheel carrier displacement ($\boldsymbol{d}$), produced by an external load change ($\boldsymbol{w}$) at the wheel centre corresponding to some typical car maneuvers are contained in Tab. 1. The considered maneuvers are the following: (M1) straight-line braking, (M2) straight-line driving/acceleration, and (M3) pure cornering. In order to meet the synthesis assumptions, magnitudes of the horizontal forces (f_x and f_y, reduced to the wheel centre) are chosen at the level of 35% of the wheel normal load (f_z), corresponding to the mechanism design position. In this range of the load change the bushings are well approximated by constant spring-rates.

In order to ensure exact and quick reactions of a car during cornering (M3), the lateral stiffness determined at the point of contact between the wheel and the road should be high enough. The most responsible for that is a change of the wheel camber (ε_x) under side force (f_y), and it should be kept at low level (criterion 4, Tab.1). For all the considered maneuvers the wheel should always be steered (ε_z) accurately with desired magnitude (usually the lesser the better) and sense (toe in, when $\varepsilon_z < 0$). These requirements are defined as criteria 1, 2 and 3 in Tab. 1. On the other hand, in order to achieve appropriate roll-off comfort, evaluated for instance when passing over a road projection, the wheel guidance should demonstrate a sufficient degree of longitudinal compliance. Under the action of negative longitudinal force (f_x) the wheel center should displace backward (q_x) as far as possible. This criterion was additionally utilized in [7]. However in this paper the dynamic criterion is used instead.

In order to find the solution of (5) with a practical meaning, the design variables are bounded ($0.14 < k_i < 14$ [kN/mm]). All the displacement components are small when compared with the lengths of the suspension rods (from about 250 to 450 mm). This in part justifies the use of the relations based on first-order kinematics for the mechanism.

The values of the optimized design (C1) variables (2), i.e. stiffness coefficients (k_i) of the bushings, are presented in Fig. 7 together with the initial values (C0 – base line) estimated for the 5-rod suspension. The rates of the bushings no 3, 4 and 5 tend to much higher than in the base line.

The values of the goal function (5) components obtained for the optimized version of the 5-rod suspension are presented in Fig. 8 in relation to the results of the base line (base line criteria are normalized to unity). The optimized suspension mechanism is described by ca. 80% decrease of braking force compliant steer (r_1), ca. 70% decrease of driving force compliant steer (r_2), ca. 30% decrease of lateral force compliant steer (r_3), and ca. 20% decrease of longitudinal vibrations of the car body (r_1) for the considered event. However, some trade off is noticeable for lateral force compliant camber (r_5), which increased with respect to the base line by ca. 10%.

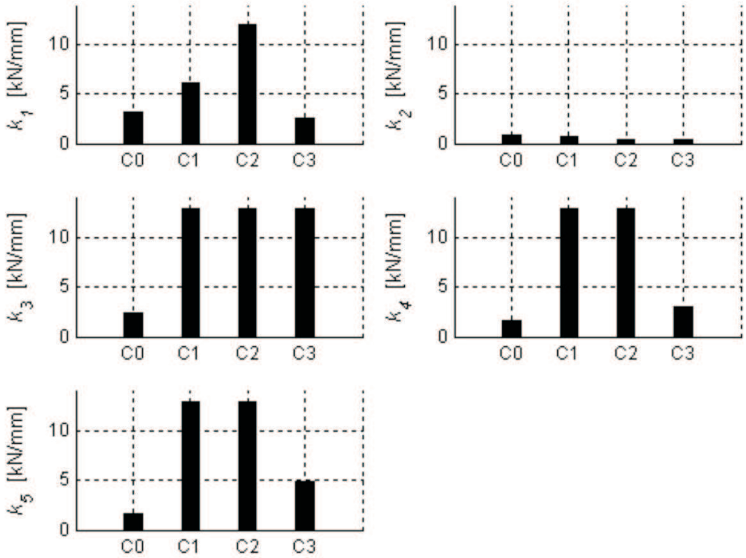

Fig. 7. Comparison of the base line (C0) stiffness coefficients with the optimization results (C1) for the 5-rod suspension

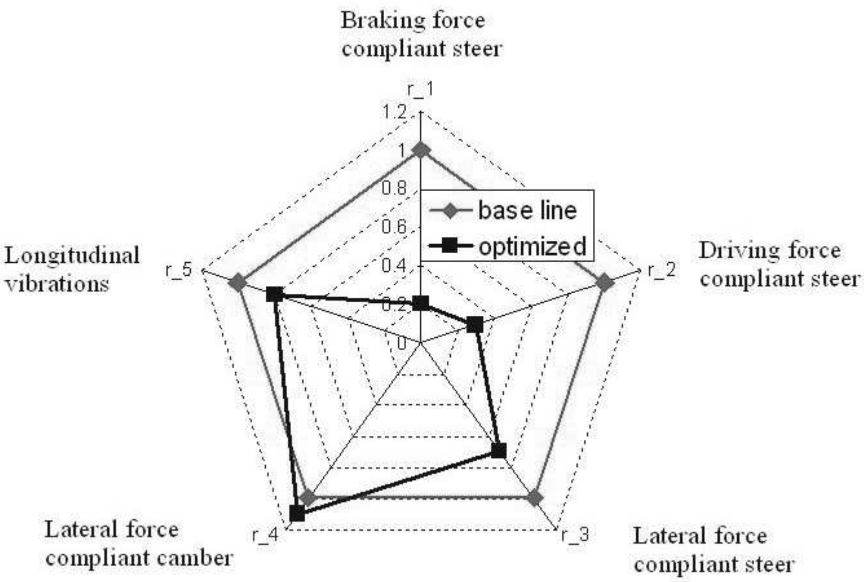

Fig. 8. Comparison of the goal function relative components for the base line and the optimized version of the 5-rod suspension

6. Conclusions

The suspension with optimized bushings rates fulfils the desired elastokinematic criteria together with a defined dynamic criterion, describing the so-called rolling comfort. The obtained results of the optimization procedure used in numerical example show potential improvements of the actual car performance, what is planed to confirm in road tests.

A correlation between the so-called suspension longitudinal compliance (static criterion) and variation of the longitudinal force (f_x) transmitted to the car body (dynamic criterion) was noticed. Other dynamic criteria can also be included, like: proper cooperation of the wheel-suspension with systems generating variable brake torque (ABS, ESP) [10], or maximization of the bushings durability [6].

Limitations for applications of the presented method result from linear range of stiffness characteristics of bushings. A proper selection of models of tire and shock absorbers as a compromise between computational effectiveness and accuracy is an important matter. Evolutionary algorithms for the optimization could be used for more effective solution in the case of nonlinear characteristics. The similar analysis should be conducted for another typical forms of road unevenness (like: step bump, pothole).

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Appendix

Table A1.

Coordinates of the joint centers of the 5-rod mechanism (Fig. 1) in the design position (at nominal load)

[mm]	a_1	a_2	a_3	a_4	a_5	a_6	c_1	
x	-103.2	309.3	200.8	205.8	-1.7	-96.0	0.0	
y	-490.6	-204.6	-283.6	-269.6	-349.6	-52.9	5.6	
z	-65.6	-49.6	-11.6	95.4	118.4	156.5	-302.0	
	b_1	b_2	b_3	b_4	b_5	b_6	b_7	b_{cg}
x	-47.8	34.9	139.1	76.9	6.3	6.3	6.3	6.3
y	-50.1	-77.3	-47.4	-59.7	-52.2	-52.2	-52.2	-52.2
z	-87.9	-133.8	-42.1	85.9	119.9	119.9	119.9	119.9

Table A2.

Masses of the wheel-suspension-car body system (Fig.1)

[kg]	m_a (sprung mass)	m_w (tire+rim)	m_b (wheel carrier)	m_1 (rod 1)	m_2 (rod 2)	m_3 (rod 3)	m_4 (rod 4)	m_5 (rod 5)
	450	16	18	3	3	1.8	1	1

Table A3.
Moments of inertia of the wheel and its carrier with respect to centers of gravity (Fig.1)

[kgm^2]	$I_{xx} = I_{zz}$ (wheel)	I_{yy} (wheel)	I_{xx} (wheel carrier)	I_{yy} (wheel carrier)	I_{zz} (wheel carrier)	I_{xy} (wheel carrier)	I_{xz} (wheel carrier)	I_{yz} (wheel carrier)
	0.7	1.4	0.25	0.3	0.2	0.01	0.015	0.05
Other moments of inertia are neglected								

Optymalizacja elastokinematycznych i dynamicznych charakterystyk 5-wahaczowego zawieszenia kół samochodu

S t r e s z c z e n i e

W artykule przedstawiono wyniki optymalizacji 5-wahaczowego mechanizmu zawieszenia wykorzystywanego w samochodach osobowych do niezależnego prowadzenia kół. Wybrane współczynniki sztywności, zdefiniowane dla pięciu przegubów elastomerowych instalowanych w wahaczach zawieszenia, są rozpatrywane jako zmienne decyzyjne w optymalizacji. Sformułowano dwa modele o dyskretnych parametrach opisujące właściwości odpowiednio elastokinematyczne i dynamiczne układu koło-zawieszenie-nadwozie. Modele wykorzystano do zapisania związku pomiędzy zmiennymi decyzyjnymi a kryteriami uwzględniającymi wymagania co do komfortu podróży i aktywnego bezpieczeństwa pojazdu. Wielokryterialna funkcja celu jest minimalizowana przy ograniczeniach wykorzystując deterministyczny algorytm. Wyznaczone z optymalizacji podatności przegubów zawieszenie kół przednich spełniają kryterium elastokinematyczne związane z właściwym prowadzeniem koła względem nadwozia przy określonym obciążeniu zewnętrznym. Dodatkowo, wprowadzono kryterium dynamiczne rzutujące na komfort podróży podczas przejazdu samochodu po pojedynczej nierówności typu garb. Przykład numeryczny dotyczy samochodu średniej klasy o przednim napędzie. Estymację parametrów i weryfikację modeli przeprowadzono podczas badań na stanowiskach oraz na drodze.