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# INFLUENCE OF CAR WHEEL SUSPENSION PARAMETERS ON IMPROVEMENT OF ACTIVE SAFETY AND RIDE COMFORT

# WPŁYW PARAMETRÓW ZAWIESZENIA KÓŁ SAMOCHODU NA POPRAWĘ BEZPIECZEŃSTWA CZYNNEGO I KOMFORTU JAZDY

A b s t r a c t

This paper deals with the simulation of car suspension parameter impact on the contact between the tyre and an uneven road. The simulation was based on the quarter car vehicle model used for the analysis of vertical vibrations of sprung and unsprung mass under the influence of discrete road unevenness. The parameters of the model relate to the front axle of a VW Passat B5. Suspension and damping characteristics were described by non-linear functions. The results included in the paper cover the influence of changing the rebound damping force, main spring stiffness coefficient and the unsprung mass on active safety and ride comfort.

*Keywords*: *vehicle vibrations*, *modelling*, *suspension*, *active safety*, *ride comfort*

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

W artykule przeprowadzono symulacyjną analizę wpływu parametrów masowych, sztywnościowych oraz tłumiennych zawieszenia koła samochodu na bezpieczeństwo czynne i komfort jazdy. Wykorzystano tzw. ćwiartkowy model pojazdu do analizy drgań pionowych masy nieresorowanej i resorowanej pod wpływem dyskretnych nierówności drogi. Parametry modelu dotyczą przedniej osi samochodu VW Passat B5. Charakterystyki resorowania i tłumienia opisano nieliniowymi funkcjami. W artykule zamieszczono wybrane wyniki analizy symulacyjnej dotyczące wpływu zmiany siły tłumienia amortyzatora w fazie odbicia, sztywności resorowania sprężyny głównej oraz masy nieresorowanej na wskaźniki bezpieczeństwa i komfortu jazdy.

*Słowa kluczowe*: *drgania pojazdów*, *modelowanie*, *resorowanie*, *bezpieczeństwo czynne*, *komfort*

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## 1. Introduction

The suspension and damping characteristics of car wheel suspensions are chosen as a compromise between a number of requirements, such as: vibration comfort (isolation) [4, 5], durability of wheel suspension, and minimizing the time of separation between the tyre and the road surface [1]. In order to obtain the specific characteristics of the suspension and damping, often time-consuming cycles of road tests are conducted, usually using trial and error methods. The use of computer modelling allows us to improve and speed up the design process of the suspension elements [3, 5].

The aim of the simulation is to analyse the impact of changes in mass, stiffness and damping parameters of car wheel suspension on active safety (related to tyre cooperation with uneven road) and ride comfort. The simulation was based on a quarter car vehicle model used for the analysis of vertical vibrations of sprung and unsprung mass under the influence of discrete road unevenness. The model parameters cover the front axle of a VW Passat B5. Suspension and damping characteristics are described as non-linear functions. The subject of the analysis is discussed in more detail in [1].

## 2. Definition of the quarter car vehicle model

### 2.1. Physical scheme of the model

The model used for the analysis of vertical vibrations of the wheel-body set is a two-mass, non-linear model known as a quarter car vehicle model [2–4]. The scheme of the model is shown in Fig.1.

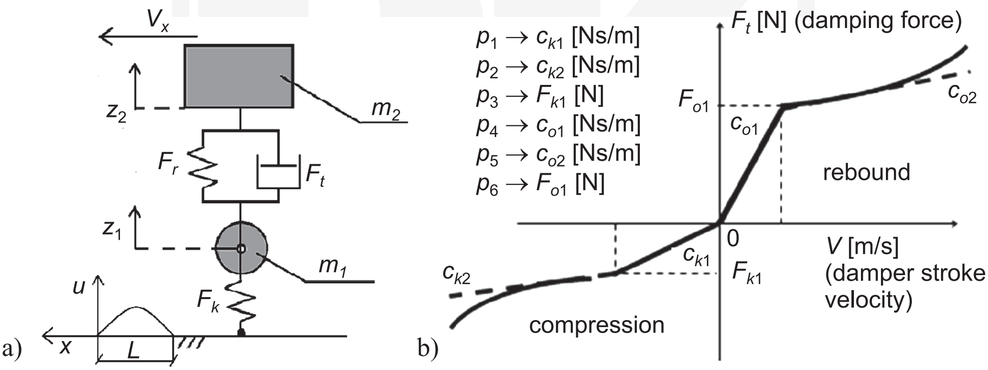


Fig. 1. a) Scheme of the quarter car vehicle model, b) damper characteristics used in the model

The model has two degrees of freedom (Fig. 1a), covering the vertical vibrations of the masses ($m_1$ and $m_2$). Coordinate $z_1$ is related to the vibration of unsprung mass ($m_1$). Coordinate $z_2$ is related to the vibration of the sprung mass ($m_2$). The model parameters are related to the damper which generates damping force $F_t$. Damper characteristics are described by parameters $p_1$ ... $p_6$, which refer to compression and rebound forces ($F_{k1}$, $F_{o1}$) and to coefficients which determine the slopes of damper characteristics ($c_{k1}$, $c_{k2}$, $c_{o1}$, $c_{o2}$). The description of these parameters is shown in Fig. 1b. Suspension elements which transfer the replacement force $F_r$, are described by, among others, the stiffness coefficient $c_2$. The tyre, which only carries force normal to the surface ($F_k$) by point contact, is described by stiffness coefficient $c_1$. The coefficient $c_1$ refers to a tyre of size 195/65 R15, with a pressure of 2.3 bar. Profile ($u$) of the unevenness (a hump), which has a certain length and height, is shown in Fig. 1a.

Table 1

**Base values of the model parameters involved with the front axle of VW Passat [3]**

| | |
|---|---|
| $m_1$ = 45 [kg] (unsprung mass) | $p_2$ ($c_{k2}$) = 1433 [Ns/m] (damp. coeff. (Fig.1b) |
| $m_2$ = 495 [kg] (sprung mass) | $p_3$ ($F_{k1}$) = 742[N] (damper compression force) |
| $c_2$ = 78 000 [N/m] (spring element stiffness) | $p_4$ ($c_{o1}$) = 3120 [Ns/m] (damp. coeff. (Fig.1b) |
| $c_1$= 260 000 [N/m] (tyre radial stiffness) | $p_5$ ($c_{o2}$) = 623 [Ns/m] (damp. coeff. (Fig.1b) |
| $p_1$($c_{k1}$) = 14070[Ns/m] (damping coefficient (look Fig. 1b)) | $p_6$ ($F_{o1}$) = 2053 [N] (damper rebound force) |

## 2.2. Assumptions for the analysis

The simulation model was based on the following simplifying assumptions [3]:
1) movement of the vehicle is a constant speed ($V_X$) and a straight line motion,
2) vertical oscillations with frequencies up to several Hz were analysed,
3) the road surface is non-deformable,
4) simulation runs were performed for road unevennesses with smooth and symmetric profiles,
5) the effects of aerodynamics and the influence of the brake system and powertrain were omitted,
6) the hysteresis of the damper and spring was omitted.

The base values of the model parameters equalled the front axle of a stock, unmodified VW Passat. The numerical data of the masses and the wheel suspension is included in Tab. 1.

## 2.3. Equations of the vehicle motion

In order to analyse the vertical vibrations of the wheel-body system the following equations were formulated describing the motion of the masses in the model (Fig. 1):

$$m_1\ddot{z}_1 + F_k(z_1 - u) + F_r(z_1 - z_2) + F_t(\dot{z}_1 - \dot{z}_2) = 0 \tag{1}$$

$$m_2\ddot{z}_2 + F_r(z_1 - z_2) + F_t(\dot{z}_2 - \dot{z}_1) = 0 \tag{2}$$

where:

$m_1$, $m_2$ – unsprung and sprung mass,
$z_1$, $z_2$ – vertical displacements (vibration components) of unsprung and sprung mass,
$F_k$ – the force transmitted through the tyre,
$F_r$ – the force transmitted through spring elements,
$F_t$ – damping force of the shock absorber,
$u$ – road profile described in relation to the distance travelled $x$.

## 2.4. Evaluation of ride comfort and active safety

The model's response under the influence of the uneven road was evaluated according to the criteria of comfort and active safety. Comfort criteria (K1 to K5) are associated with the vertical vibration of the sprung mass where the passenger seats are mounted (Fig. 2). The driving comfort criteria analysed in the paper are: maximum, minimum, and root mean square value of the body vertical acceleration ($a_z$), the RMS value of the body jerk (K4) and maximum vertical body displacement $z_2$ (K5).

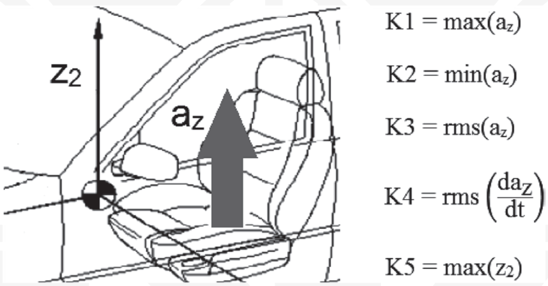


Fig. 2. Description of the driving comfort criteria

Criteria of active safety (B1 to B3) are associated with the normal tyre force ($F_k$) (unsprung mass element) with respect to the road (Fig. 3). Changes in that force causes changes in the tyre adhesion force to the road surface. In these circumstances, the vehicle response to acceleration, deceleration or change of direction forced by driver may be different than expected. Active safety criteria used in the analysis are: the RMS value and maximum of the force $F_k$ and the time when the wheel lost contact with the road surface (criterion B3).

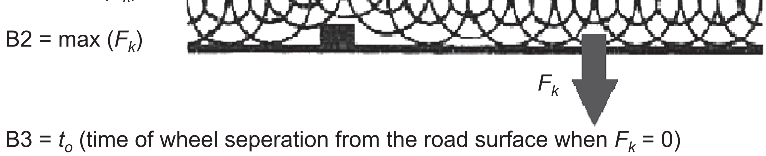


Fig. 3. Description of active safety criteria

## 3. Simulation analysis

For the analysis, Matlab simulation software was used. Five simulation scenarios were defined [1]. The results submitted in this section are compared with passing through a pothole (with a length of 1 m and a depth of 0.05 m) with a speed of 100 km/h and 50 km/h and also with passing through a bump (with a length of 1 m and a height of 0.05 m) with a speed of 100 km/h.

**Scenario 1:** bump, 100 km/h, influence of damper rebound force

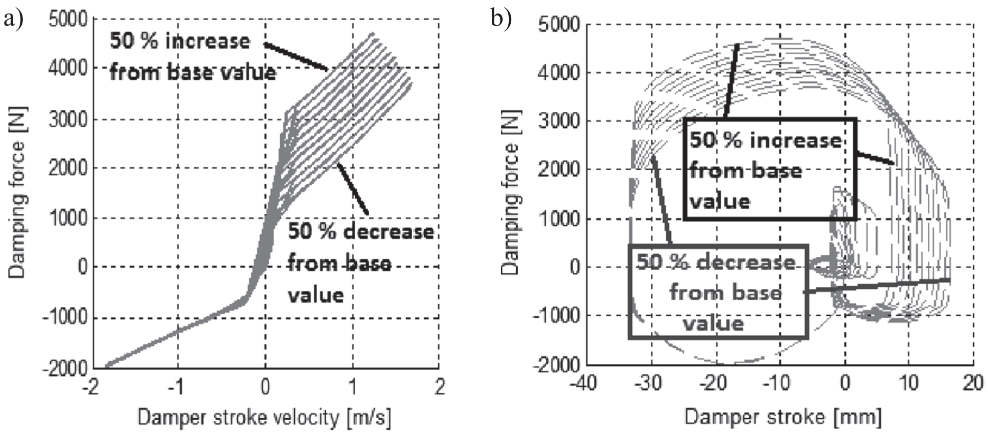


Fig. 4. Influence of damper rebound force change on damper characteristics: a) relation between damping force and damper stroke velocity, b) relation between damping force and damper stroke

The damper rebound force was varied within ± 50%, which was encoded in the range of –1 to 1, where 0 is the base value of the parameter. The base value refers to a brand new shock absorber, without regulation. Fig. 4a and Fig. 4b show the influence of the rebound damping force change on the established damper characteristics. The relationship between the damping force and the damper stroke velocity is shown in Fig. 4a. Positive ranges of the damping force and the damper stroke velocity refer to the rebound motion of the car's suspension. On the other hand, the negative ranges of the damping force and the damper stroke velocity refer to the opposite direction of car suspension motion (compression). The change of the parameter affects the part of characteristics shown in Fig. 4a which refers to the positive range of damper stroke velocity. The rebound damping force increased by 50% from the base value determines a maximum of damping force of about 4500 N and maximum of damper stroke velocity about 1.25 m/s. When the rebound damping force is decreased by 50% from the base value, the maximum damping force is about 3500 N and corresponds to a maximal value of damper stroke velocity of about 1.75 m/s. When the damper rebound force is increased, the maximum damper stroke velocity takes smaller values. It is provoked by increased suspension movement resistance caused by greater damper rebound force.

The relationship between the damping force and the damper stroke is shown in Fig. 4b. Positive ranges of the damping force and the damper stroke correspond to the rebound motion of the car's suspension and negative ranges refer to the suspension compression. The damper rebound force change affects the positive ranges of the values shown in Fig. 4b. Greater suspension movement resistance (caused by an increased damper rebound force) results in reduced damper stroke. The maximum damping force (about 4500 N) occurs when the damper parameter is increased by 50%. The line of the damping force at about 4500 N passes into the line of damper stroke about 8 mm (to the right of Fig. 4b.) The minimum damping force (about 3500 N) occurs when the damper rebound force is decreased by 50%. The line of the damping force at about 3500 N passes into the line of damper stroke at about 16 mm.

The driving comfort and active safety criteria were also normalized, so that a value of 1 corresponded to the base values of parameters. The reduction of that value corresponds to an improvement in the wheel-car body system qualities and an increase in that value refers to deterioration of the criteria under consideration. For example, the value Criteria K = 1.1 refers to the deterioration of the comfort criterion by about 10%. On the other hand, the value Criteria B = 0.95 shows an improvement in the ride comfort criterion by about 5%.

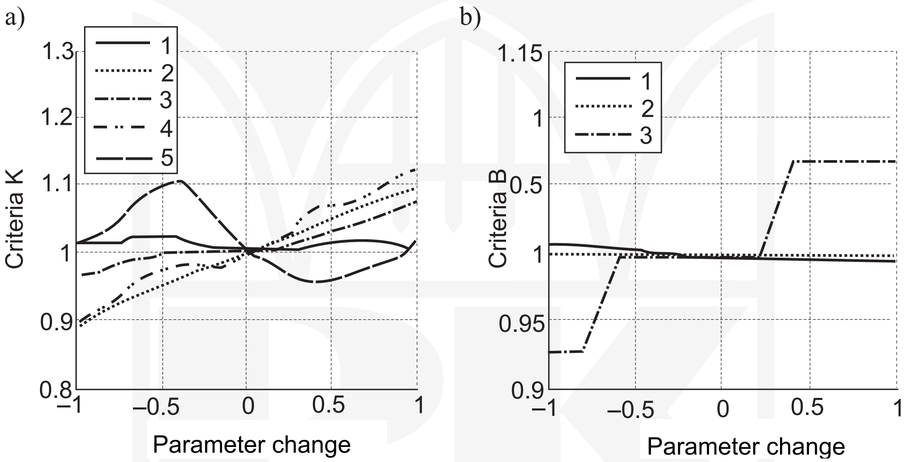


Fig. 5. Influence of the rebound damping force change in scenario no. 1 on the criteria: a) K-ride comfort, b) B-active safety

In this scenario, the increase in the rebound damping force provokes a deterioration in four of the five comfort criteria analysed (Fig. 5a). Only the maximum of body vertical displacement (criterion K5) shows irregular change. The maximum normal tyre force (criterion B2) does not depend on the change in rebound damping force, which is shown in Fig. 5 b. The RMS value of normal tyre force shows negligibly small changes. The criterion which depends on the parameter change most of all is the period of wheel separation from the road surface (criterion B3). When the rebound damping force is greater, the wheel separation time is longer (Fig. 5a).

**Scenario 2:** pothole, 100 km/h, influence of damper rebound force

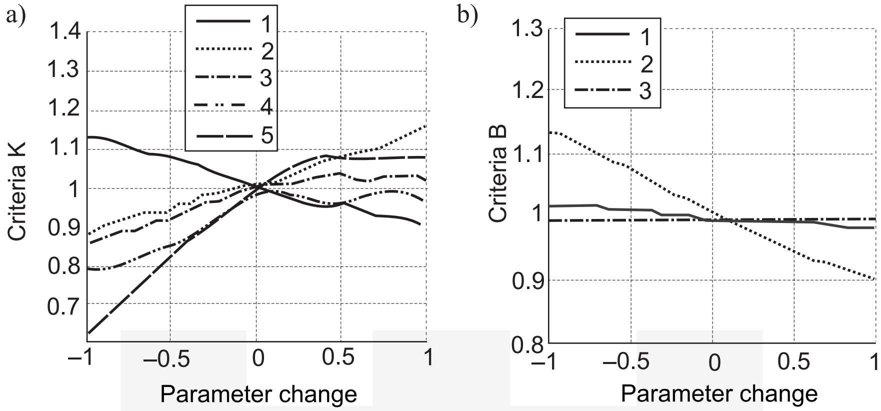


Fig. 6. Simulation scenario no. 2- influence of the rebound damping force change on the criteria: a) K-ride comfort, b) B-active safety

Increasing the rebound damping force provokes deterioration in four of five of the comfort criteria analysed (Fig. 6a). Only criterion K1 (maximum of body vertical acceleration) shows an improvement. The RMS value of normal tyre force (criterion B1) decreases when the rebound damping force is greater. However, the amount of change in this criterion is inconsiderable (Fig. 6b). The maximum normal tyre force also decreases, but change in this criterion is significant. According to Fig. 6b, the difference between maximum and minimum values of criterion B2 obtained is about 25%. In scenario 2, time of wheel separation from the road surface does not depend on a change in the rebound damping force.

**Scenario 3:** bump, 100 km/h, influence of unsprung mass

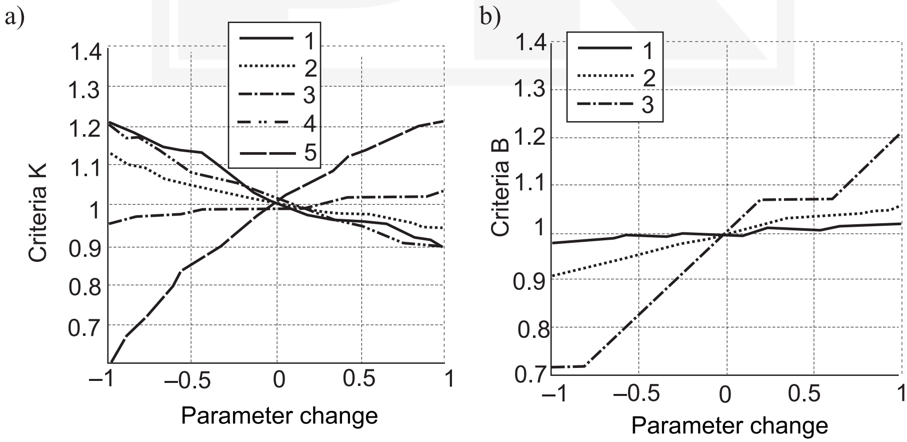


Fig. 7. Influence of the unsprung mass change in scenario no. 3 on the criteria: a) K-ride comfort, b) B-active safety

In the case of scenario 3, increasing the rebound damping force provokes improvement in the maximum and minimum vertical body acceleration (criteria K1 and K2 respectively) and the RMS value of body vertical jerk (criterion K4). This is shown in Fig. 6a. The RMS value of body vertical acceleration (criterion K3) and maximum of body vertical displacement (criterion K5) show deterioration (increase). The changes in criterion K3 are inconsiderable, but the increase in the maximum body vertical displacement is prominent. The difference between the maximum and minimum value of this criterion obtained is about 50%.

Increase in the rebound damping force provokes a deterioration in all the active safety criteria. The criterion which shows the most significant deterioration is the period of wheel separation from the road surface (B3). A greater rebound damping force provides resistance which slows damper rebound motion when the wheel is separated from the road surface. In these conditions, the period of wheel separation from the road surface is extended.

**Scenario 4:** pothole, 100 km/h, influence of unsprung mass

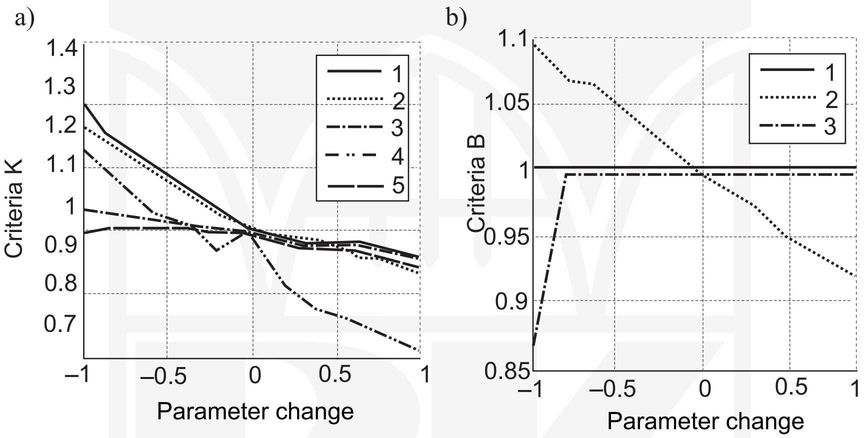


Fig. 8. Simulation scenario no. 4- change of the criteria: a) K-ride comfort, b) B-active safety in accordance with unsprung mass change

In this scenario, all the active safety criteria show improvement when the unsprung mass is increased (Fig. 8a). The criterion whose value is minimized the most is the RMS value of body jerk (K4). The difference between maximal and minimal values for this criterion is about 30%. In the case of maximum and minimum vertical body acceleration (criteria K1 and K2 respectively) this difference is about 15%. Criteria K3 and K5 (respectively the RMS value of body vertical acceleration and maximum of body vertical displacement) show comparatively small improvement.

When the unsprung mass is greater, the maximum normal tyre force (criterion B2) shows a significant decrease (improvement). This is shown in Fig. 8b. In scenario no. 4, the RMS value of this force does not depend on the unsprung mass change. When the unsprung mass is reduced by 50% compared to the base value, the period of wheel separation from the road surface (criterion B3) is comparatively beneficial. However, when the unsprung mass is further increased, this criterion does not depend on this parameter change.

**Scenario 5:** pothole, 50 km/h, influence of spring elements stiffness

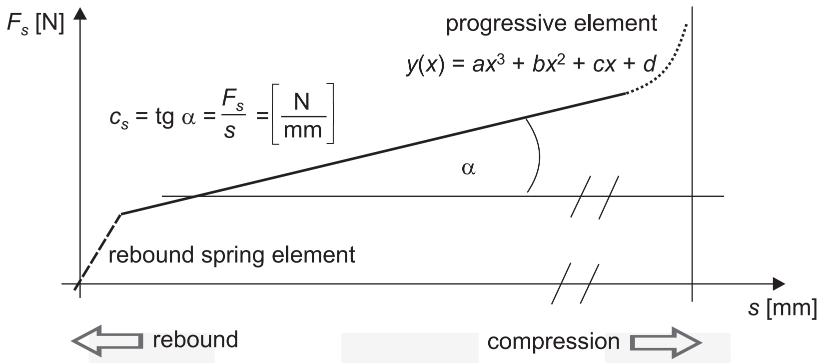


Fig. 9. The suspension characteristics

The suspension element in the front axle of a VW Passat B5 consists of three components, which act in different ranges of suspension movement (Fig. 9). In the largest part of the suspension travel (solid line in Fig. 9), the relation between the suspension force and suspension movement is linear. In this range of suspension travel the main spring element functions, described by its stiffness coefficient $c_s$. The suspension stiffness coefficient is equal to the tangent of the angle between the blue line of suspension characteristics and the horizontal axis. In the front suspension of a VW Passat B5, the main spring element is a coil spring. In the final phase of the rebound movement (dashed line in Fig. 9), the suspension stiffness coefficient achieves a larger value. This is provoked by the additional rebound spring element. In the final stage of the suspension compression (dotted line in Fig. 9), the progressive element acts. Its characteristics are described in the numerical model by a polynomial of the third degree. The model parameter changed in this scenario is the suspension stiffness coefficient $c_s$.

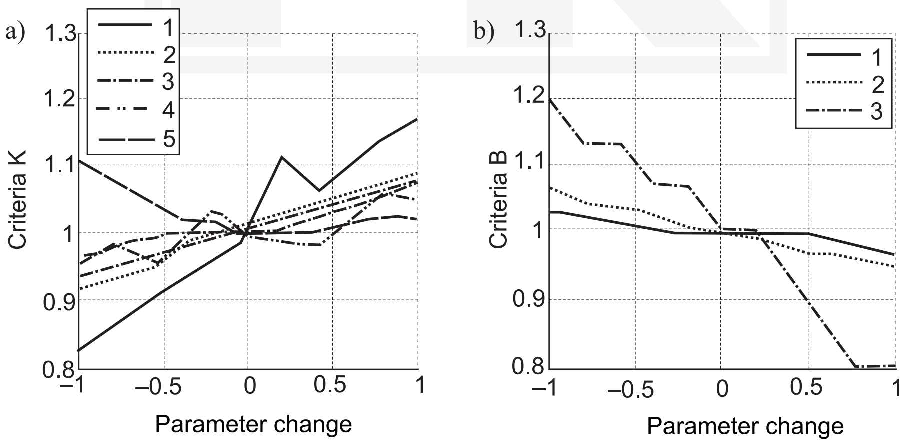


Fig. 10. Scenario 5: influence of spring elements stiffness change on criteria: a) K-ride comfort, b) B-active safety

An increase in spring element stiffness provokes deterioration in four of the five comfort criteria obtained (Fig. 10a). A large spring element stiffness causes resistance on wheel vertical movement under the influence of an uneven road. For this reason, the vibrations provoked by the uneven road are largely transferred to the car's body. In this simulation scenario, only criterion K5 (maximum of body vertical displacement) shows irregular changes.

A spring element which has a large stiffness has a tendency to rebound rapidly after taking the load off. This quality is confirmed in Fig. 10b and results in significant improvement in the period of wheel separation from the road surface (criterion B3). Criteria B1 and B2, involved with normal tyre force (its maximum and RMS value), are also improved. However, this improvement is less significant.

## 4. Conclusions

The results obtained confirm the contradiction between the analysed criteria of comfort and active safety which occurs in most of the simulation cases [1]. An increase in damper rebound force provokes deterioration in the considered ride comfort criteria in most simulation cases. At the same time, an increase in this parameter provides an improvement in the most active safety criteria [1]. An increase in the unsprung mass results in deterioration in the comfort criteria and active safety criteria. When the spring element stiffness is increased, comfort criteria take less beneficial values, but active safety criteria are improved.

The relationships between the criteria considered are not smooth because of the non-linearities that occur in certain model response conditions. Due to the large number of independent parameters and variety of simulation scenarios, the selection of specific settings (e.g. in order to maximize active safety) is a very complex issue. This problem is difficult to solve without appropriate optimization algorithms [3, 5].

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