ANALYSIS OF THE DYNAMICS OF PASSENGER CARS WITH MCPHERSON STRUT SUSPENSION SYSTEMS IN THE MSC.ADAMS SOFTWARE ENVIRONMENT WITH TAKING INTO ACCOUNT THE FRICTION IN SHOCK ABSORBERS

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Summary

The paper presents a method to analyse the dynamics of passenger cars with McPherson strut suspension systems, which will make it possible to consider the friction that takes place between major components, i.e. piston rod and cylinder, of front shock absorbers. The analysis was carried out with making use of modern tools to support the engineering design process, i.e. the Autodesk Inventor program (used to work out geometric models of the car under consideration), and the MSC. ADAMS software (in the environment of which the dynamics of the structural model of the car under consideration was analysed). The analysis included investigation of the tendency exhibited by a car to have variations in the distribution of car body weight among road wheels at sudden lane-change manoeuvres, especially in the case of previous severe defect (oil outflow) of one of the car shock absorbers as a result of which the shock absorber would fail to function fully. In such a situation, dry friction may occur between mating parts of the defective shock absorbers, which would impede (or even prevent) relative translational motion of the parts. In the authors' opinion, the study presented may be interesting for designers of McPherson strut suspension systems for automotive vehicles.

Keywords: dynamics, passenger car, McPherson strut car suspension, friction in shock absorbers, MSC. ADAMS software.

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1. Introductory information

The McPherson strut suspension system (Fig. 1) is a design used widely, chiefly used in popular medium-class cars with engines of small cubic capacity.

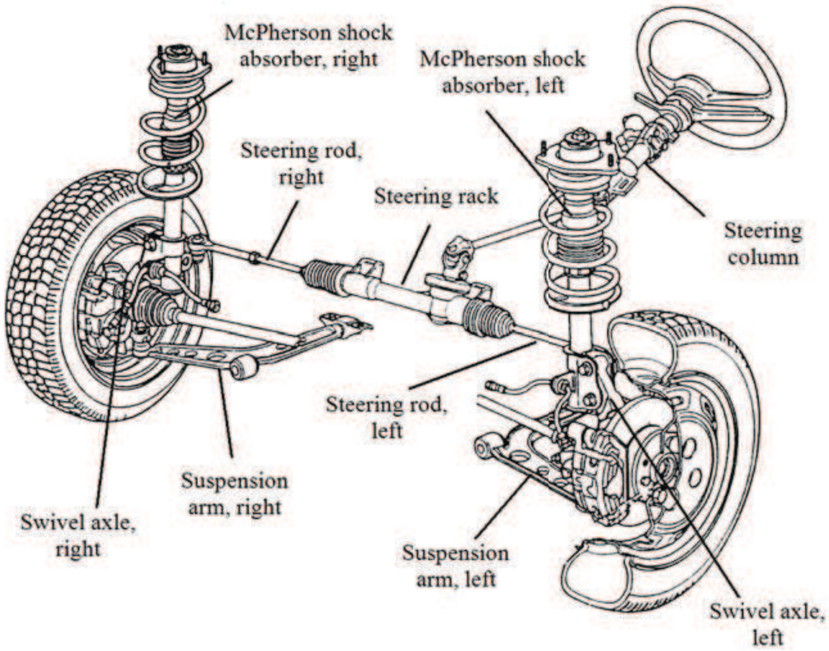

Fig. 1. Front McPherson strut suspension system, overall view.

An important advantage of such a suspension system is the fact that it takes up relatively small space in the car and is easy to assemble and install. However, it also has some disadvantages. One of them is related to the absence of upper suspension arm, resulting in which the major components of the shock absorber, taking over the suspension arm's functions (Fig. 2), i.e. piston rod and cylinder, are loaded in road conditions not only with axial forces but also with significant transverse forces and bending moments. As it can be seen in the illustration, the piston rod moves in a brass guide inside the cylinder and the piston, forming the end of the piston rod, mates with the inner cylinder surface through a plastic sealing ring.

The transverse forces and bending moments cause significant deformation (bending) of the mating shock absorber components. This in turn results in an increase in the values of the normal reaction forces occurring in the areas of contact between the parts and, in consequence, in the values of the friction forces. These forces affect the damping of vibrations of front suspension components; thus, they change the physical properties of the suspension system and significantly impair the springing quality. The significant

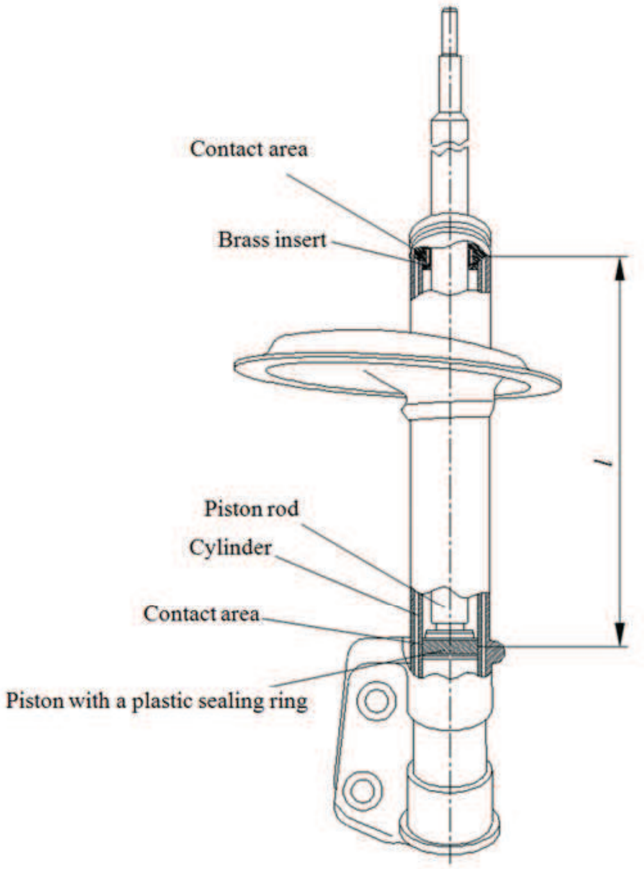

Fig. 2. Construction of a shock absorber in the McPherson strut suspension system.

friction may even result in temporary self-jamming of the piston rod in the cylinder. In such a case, the value of the axial component of the force acting on the piston rod may become insufficient for the shock absorber parts "detached" after having been jammed (i.e. for their relative motion to be initiated). This may happen when the vehicle turns on a road with almost perfectly even surface and the vertical forces acting on front wheels of the car are insufficient to initiate the relative motion of the mating parts of the "defective" shock absorber (this is referred to as "Boulevarde's effect [10]). The self-jamming may occur whatever the relative position of the shock absorber parts involved; it may also take place in only one shock absorber or in both at the same time. This may result in variations in the distribution of car body weight among road wheels, with the actual position of the car body being difficult to predicting in a specific case because of an infinite number of possible combinations (thus, the car body position is completely random). The randomisation of the body weight distribution jeopardises the stability of vehicle motion because, even if the front wheels are positioned "straight ahead," the loads transmitted to the left and right wheel from the road surface are not equal to each other; in consequence, the values of the forces applied by the individual steering rods to the steering rack, connected through a pinion with the steering column and steering wheel, are unequal as well. The unsymmetrical loading of the steering rack may result in "skewing"; if the driver tries to counteract this tendency, steering wheel wobbling (in German referred to as "Lenkradwobbeln" [10]) may develop. The problem of variations in the distribution of car body weight among road wheels induced by friction in front shock absorbers was previously thought to apply predominantly to commercial vehicles (i.e. vehicles with high centre of gravity), but more thorough research work has shown that this problem may also be important in the case of passenger cars, according to [10].

The mating shock absorber components operating in increased friction conditions are exposed to faster wear, especially in the extreme situation when oil has flown out from the unit. Fig. 3 shows the appearance of the surface of a considerably worn part (piston rod) of a shock absorber that was installed in a car used for amateur sport competitions, i.e. subjected to particularly severe working conditions, often without adequate servicing.

According to [10] and [12], all sorts of efforts were made in the past to minimise the values of the friction forces that develop in both the areas of contact between shock absorber components.

Included in this minimization, attempts were made to select appropriate materials with low values of the coefficients of friction for the sliding surfaces in both the areas of contact (e.g. the use of materials with Teflon admixture was proposed). However, materials of

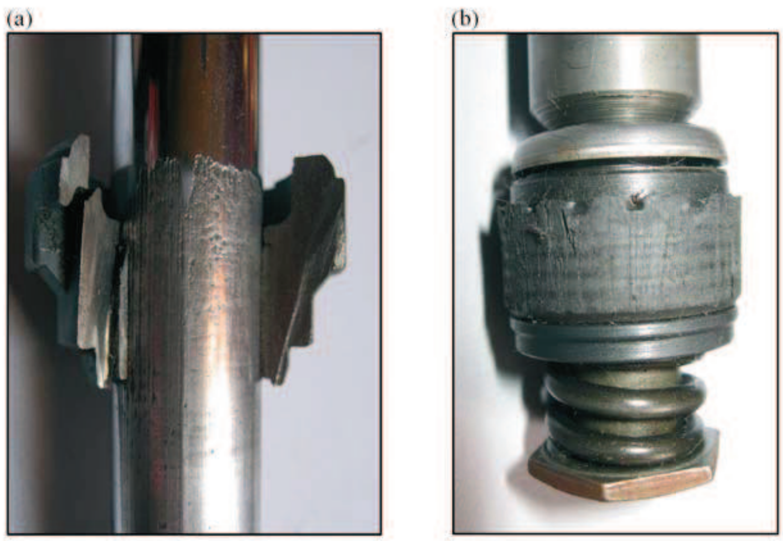

Fig. 3. Damaged parts of the piston rod [15]:
(a) The area of contact with a brass guide; (b) Piston sealing ring.

this kind also exhibit poor abrasion resistance; for this reason, they could not find wider application.

Another idea was to minimise the values of the normal reaction forces that occur in both the friction contact areas in the shock absorber.

According to monograph [18], the values of the friction forces (i.e. the normal reaction forces as well) in both the areas of contact between the piston rod and the cylinder may be minimised by reducing the distance between the centre of the ball joint connecting the suspension arm with the swivel axle and the plane of action of the vertical road surface reaction onto the wheel (denoted by "*b*" in Fig. 4).

Another method was to increase the distance between the areas of contact between the two major shock absorber components (denoted by "l" in Fig. 2), i.e. to increase the cylinder length. Furthermore, an attempt was made to change the angle of inclination of shock absorber springs in relation to shock absorber centrelines, i.e. to make the springs more inclined in the lateral plane. These two methods are now most popular among car manufacturers. Less often used method include, an offset providing between spring and

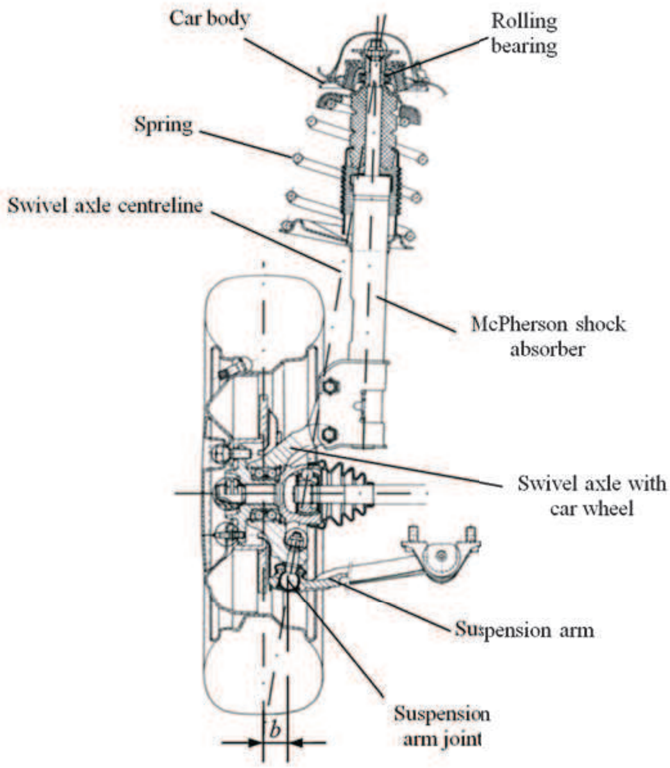

Fig. 4. Suspension system on the left front wheel of a car.

shock absorber centrelines; sometimes, springs with curved centrelines are installed (or the springs are made stiffer on the outer side). In certain solutions, the springs are installed in flexible seats. There are also designs where an additional horizontal spring is introduced between the shock absorber and the car body, with the spring centreline being situated transversely in relation to the car drive direction. In some other designs, an offset is also proposed to shift the bearing that connects the piston rod with the car body towards the centre of gravity of the car (thus to introduce a stress-releasing bending moment).

Apart from the above, concept works have also been undertaken to enable shock absorber components to be releated from the jamming state.

As an example, a proposal was made in one of the concept designs to excite resonance transverse vibrations of shock absorber components, as a result of which the values of the normal reaction forces and, in consequence, of the friction forces in both the friction contact areas would be reduced. In practice, the excitation of vibrations of this kind was found to be too impractical; besides, the values of the first "effective" resonance frequencies proved to be too low (they were 90 Hz and 270 Hz), adversely affecting the car ride comfort. Another method prepared to get shock absorber components out of the "self-jamming" state was the ultrasonic excitation of axial vibrations in the piston rod. In practice, however, the high frequency vibrations thus excited were found to undergo significant damping due to the properties of oil and material of seals between shock absorber components. Other, quite costly, methods to reduce the "self-jamming" effect were also tried. In one of the concepts proposed problem of the blocking of the relative axial motion of both the shock absorber components was to be overcome by causing the components to rotate in relation to each other (the unblocking of the parts in one direction resulted in liquidation of the blocking of movement in any other direction, inclusive of the axial movement). In another method, the bearing that connects the piston rod with the car body was to be elastically twisted under the influence of the axial force and, thanks to this, it was to apply a torsional moment to the piston rod, of a value sufficient to overcome the moment of static friction forces between both the shock absorber components (the purpose of overcoming of the friction moment was to initiate the relative motion of the parts, inclusive of the axial movement). One more method consisted of the mounting of an additional part, playing the role of a "flywheel," on the piston rod through a torsion spring. The vibrations of this weight were to facilitate the bringing of the shock absorber components out of the "self-jamming" state (in practice, however, this method proved insufficiently effective). An interesting method, although never implemented on a larger practical scale, was which the piston rod was to be continuously rotated by a small electric motor [7]. Such a solution, where permanent relative motion of mating shock absorber components would be produced, would definitely eliminate the "self-jamming" of these parts.

2. Friction modelling

2.1. Modelling of kinetic friction; the Stribeck curve

The authors of publications dealing with tribology issues often refer in their studies to the

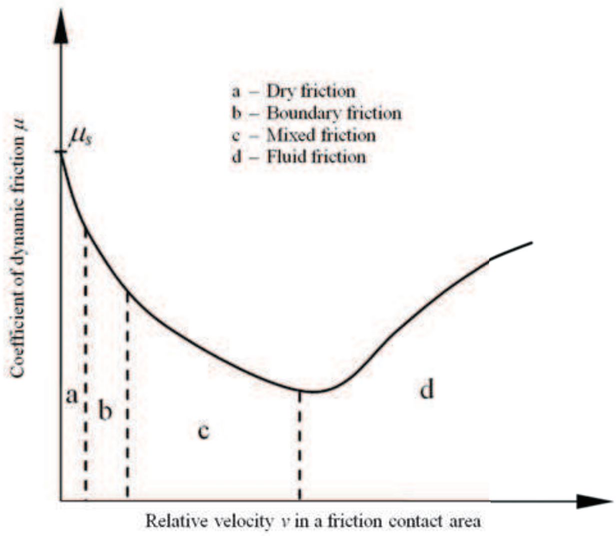

Fig. 5. The general form of the Stribeck curve.

"Stribeck curve" described for the first time in [21] as early as in 1902. One version of this curve, representing the value of the coefficient of kinetic friction μ vs. the value of the relative velocity v in a specific friction contact area, has been shown in Fig. 5 (according to the graph, if $v = 0$ then $\mu = \mu_s$, where μ_s is the value of the coefficient of static friction).

An analysis of this curve shows that the fluid friction in a specific friction contact area may only take place if the parts involved move in relation to each other with relative velocity of an adequate value. If the value of the relative velocity is below the threshold then the friction may become "mixed" (the intermediate state between "dry" and "liquid" friction), "boundary" (where the thickness of the lubricating layer is of the order of several tenth of one micrometre), or even "dry" (at very low rubbing speeds, where adequate lubrication conditions are difficult to be achieved).

The basic numerical difficulties encountered when determining the values of the kinetic friction force in a specific friction contact area, defined by the known Coulomb formula $F_{fric} = \mu N$ (where N is the value of the normal reaction force in the contact area and μ is, as previously stated, the value of the coefficient of kinetic friction), arise from the necessity to carry out calculations in the field of analysis of the dynamics of mechanical systems while taking into account the discontinuous function that represents the values of the coefficient of kinetic friction at zero value of the relative velocity (Fig. 5). A direct consequence of this discontinuity is discontinuity of the values of generalised accelerations of the structural model of the mechanical system under consideration, which results in numerical difficulties in determining the acceleration values when solving the equations of motion. To avoid this problem, a simplified model of the Stribeck curve is sometimes employed (Fig. 6), where the existence of friction in the zone close to relative rest of the mating parts is "neglected."

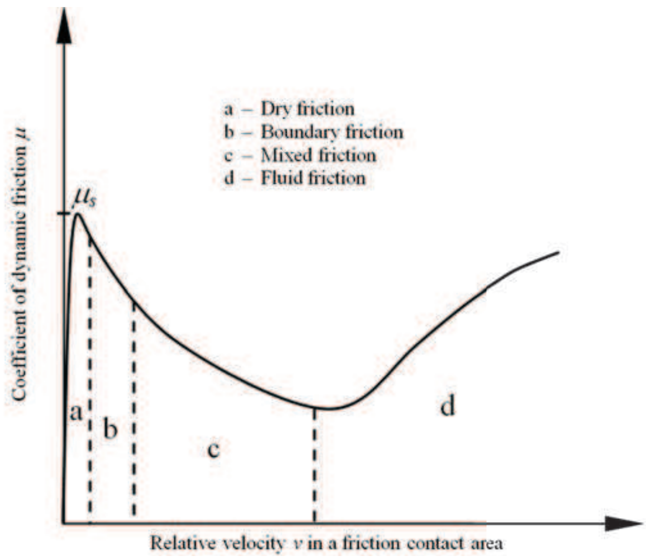

Fig. 6. A simplified form of the Stribeck curve.

If such a model of the Stribeck curve is adopted for the specific friction contact area, an assumption is simultaneously made that if the motion in this contact area is "stopped," i.e. when the value of the relative velocity of the mating parts drops to zero, the value of the coefficient of kinetic friction μ becomes zero as well.

The adoption of such a form of the curve means that the static friction phases, traditionally defined as the states where the relative velocity value remains zero for a prolonged time, are inapplicable to the contact area under consideration. For the same reason, this form of the curve cannot be used to analyse the dynamics of the mechanical systems whose movement play an important role (the "stick-slip" phenomenon occurring in some mechanical systems, e.g. in machine tools or robot manipulators are important examples). Nevertheless, the simplified model of the Stribeck curve may be successfully used to analyse the dynamics of cyclic movements of bar linkage members. The members and actuators of such mechanisms are sufficiently rigid and, in practice, only kinetic friction states occur in their joints (the phases with "zero value of relative velocity" phases are negligibly short). The oldest publication known to the authors where the use of the simplified form of the Stribeck curve was suggested is paper [3]. Since then, many reports have been published where the use of the curve in this form an analysis of selected problem areas of different mechanical systems has been described.

When writing about the behaviour of bodies remaining at rest in the static friction phase, we should mention the "creep" phenomenon [19], i.e. the occurrence of a "micro-slip" of a body to which an active force of a value smaller than that of the fully developed static friction force (i.e. incapable to cause actual slip, referred to as "macro-slip") has been applied. This phenomenon was thoroughly investigated and described for the first time in 1920s by two researchers, who carried out their studies independently in the UK and in the former Soviet Union, i.e. Rankin [17] and Verchowskij [22]. Nevertheless, the oldest publication known to the authors where test results confirming the occurrence of "micro-slip" preceding the actual slip were presented is paper [20], published as long ago as in 1899. The "creep" consists of two phases: reversible (elastic deformation) and irreversible (plastic deformation). Due to the high practical impact of the "creep" process for various questions related to mechanical systems, experimental and theoretical works on this issue are ongoing. The results of these works include a mathematical algorithm developed by an American researcher Dahl. The algorithm, referred to as Dahl model, makes it possible to define the dependence of the static friction force value on the value of the micro-displacement that develops in the friction contact area between elements remaining at rest in relation to each other [4]. It is frequently used by researchers who model the phenomenon of friction at the studying of problems related to the analysis of the dynamics and control of different mechanical systems (including, above all, the manipulators of robots).

2.2. Modelling of friction in the MSC.ADAMS software environment

The authors of this paper have decided that the proposed method of analysing the dynamics of passenger cars should be based on the MSC.ADAMS software [23], which is an advanced tool for the investigation of the dynamics of different mechanical systems. The friction modelling method proposed, where the simplified form of the Stribeck curve has been adopted and the "creep" process has been accounted for, has been formulated in the said software environment with the use of an algorithm having the form of a procedure defined by an acronym FRD (Friction Regime Determination), which is used to determine the coefficient of friction in the static and kinetic friction phases. This procedure is depicted in the form of an appropriate block diagram as presented in Fig. 7. Traditionally, the coefficient of friction is a scalar quantity; in the MSC.ADAMS software environment, however, it is expressed in the form of a vector $\boldsymbol{\mu}$, according to the diagram (in the subsequent part of this paper, it will be referred to as a "vector" coefficient of friction).

In the procedure adopted, the static and kinetic (or "dynamic") friction phases are handled separately, with the "threshold velocity" value Δv being taken as a boundary between them.

The "static friction" phase (see the lower part of the diagram) corresponds to the range of changes in the values of relative velocity in a specific friction contact area from 0 to Δv. It is assumed that the "vector" coefficient of friction related to one or other element of the friction pair (Fig. 8) can be described in this phase by the following equation:

$$\boldsymbol{\mu} = -(1-\beta)\mu_1\hat{\boldsymbol{\Delta}} - \beta\mu_s\hat{\mathbf{v}} \;. \tag{1}$$

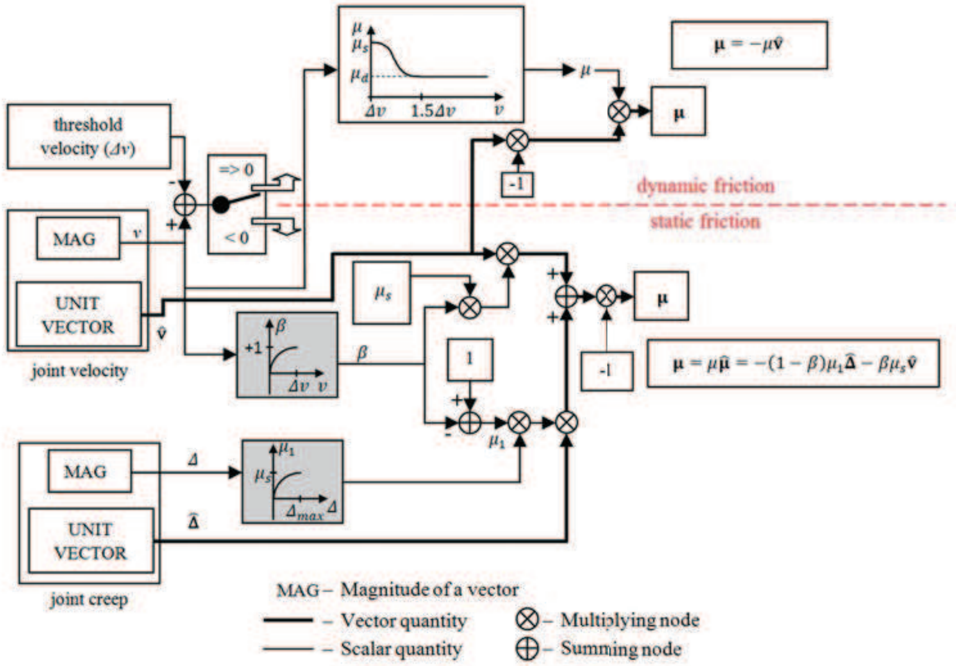

Fig. 7. Block diagram of the FRD procedure to determine the "vector" coefficient of friction (prepared by the authors, based on [23]).

According to the notation presented, vector $\boldsymbol{\mu}$ is a vector sum of two component vectors $-(1-\beta)\mu_1\hat{\Delta}$ and $-\beta\mu_s\hat{\mathbf{v}}$. The former represents the influence of the "creep" process taking place in the contact area under consideration ($\hat{\Delta}$ represents the unit vector of this "creep") and the latter defines the impact of the relative velocity of slip in this contact area ($\hat{\mathbf{v}}$ represents the unit vector of this velocity). The unit vectors $\hat{\Delta}$ and $\hat{\mathbf{v}}$ have the same direction and sense. The magnitudes of both component vectors depend on the current values of the β and μ_1 factors taken into account in the FRD procedure. As can be seen in the upper and lower shaded windows of the diagram, respectively, the β value is an increasing function of the value v of the relative velocity in the contact area under consideration (if $v = \Delta v$ then $\beta = 1$) and the μ_1 value is an increasing function of the value Δ of the "creep" in the same contact area, if the condition is met that $\Delta \leq \Delta_{max}$, where Δ_{max} is the limit (maximum) value of the "creep" (if $\Delta = \Delta_{max}$ then $\mu_1 = \mu_s$) is met.

The transition from the static to kinetic (or "dynamic") friction phase will take place when the value v of the relative velocity in the friction contact area under consideration exceeds the "threshold velocity" value Δv. In general, an assumption is made that when the transition to the new phase is completed then $\mu_1 = 0$.

In the kinetic friction phase (referred to as "dynamic friction" in the upper part of the

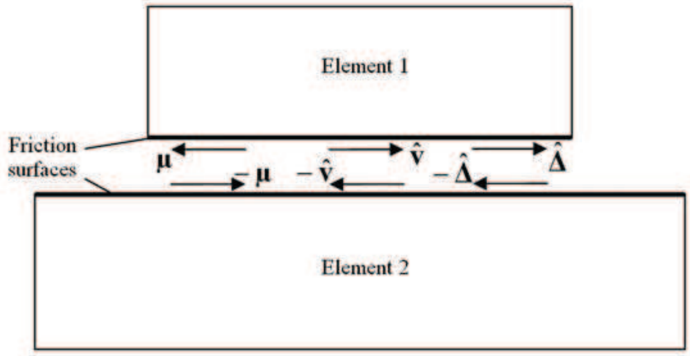

Fig. 8. Mutual interaction of elements of a friction pair in the static friction phase.

diagram, where the handling of this phase is illustrated), the "vector" coefficient of friction is expressed by the following equation:

$$\boldsymbol{\mu} = -\mu \hat{\mathbf{v}} \,. \tag{2}$$

The value of the coefficient of kinetic friction μ depends on the value of the relative velocity of slip in the specific friction contact area. This dependence has been illustrated in the window provided in this part of the diagram. As can be seen, it enables the use of an approximation of a jump function. The range of occurrence of kinetic friction may be divided into two sub-ranges: from Δv to 1.5 Δv and above 1.5 Δv. In the former one, the value of the coefficient of kinetic friction smoothly declines from the level corresponding to the coefficient of static friction μ_s to a value of μ_d, referred to as "dynamic coefficient of friction" in the terminology used in the MSC.ADAMS software environment.

Finally, the value of the coefficient of kinetic friction is defined by the following formula:

$$\mu = \begin{cases} \mu_s, & \text{for } v = \Delta \text{v}, \\ \mu_s + (\mu_d - \mu_s)k^2(3-2k), & \text{for } \Delta \text{v} < v < 1.5\Delta \text{v}, \\ \mu_d, & \text{for } v > 1.5\Delta \text{v}, \end{cases} \tag{3}$$

where: k is a variable parameter defined as $k = \dfrac{v - \Delta \text{v}}{1.5\Delta \text{v} - \Delta \text{v}} = \dfrac{v - \Delta \text{v}}{0.5\Delta \text{v}}$.

The transition from the kinetic to static friction phase will take place when the value v of the relative velocity in the friction contact area under consideration drops below the "threshold velocity" value Δv. Then, the unit vectors $\hat{\Delta}$ and $\hat{\mathbf{v}}$ will change their sense and the "vector" coefficient of friction will begin to be expressed by formula (1).

In recapitulation of the argument presented above, we may state that the basic inputs for the procedure illustrated by the diagram are the current values of the relative velocity in the friction contact area under consideration (v) and of the "relative displacement" in the same area (Δ). The "vector" coefficient of friction, determined from formula (1) or (2) depending on the current friction state, will be an output of this procedure.

To supplement the statements presented, it should be added here that a similar method of modelling the coefficient of friction, where the simplified form of the Stribeck curve was employed and the "creep" phenomenon was taken into account, was proposed in the Polish monograph [5].

3. Structural model of a car, developed in the MSC.ADAMS software environment

The modelling procedure was started with preparation of geometric models of major components of the vehicle under consideration in the Autodesk Inventor software environment. In particular, the models, which were to be used for building a structural model of the car in the MSC.ADAMS software environment, represented components of the steering system and the front and rear suspension systems (Fig. 9). The car body model was simplified, i.e. the body was assumed to be solid with its overall dimensions being approximately equal to those of the original. Attention was also given to satisfying the requirement that the mass of the solid, the location of its centre of gravity, and the loads of both car axles should be approximately identical to the actual data. The arrangement of the prepared geometric models of individual car components was planned simultaneously as the stage of building the models in the Autodesk Inventor software environment. Thanks to this, the geometric models of individual components read in succession into the MSC. ADAMS software environment, in which a structural model of the car under consideration was subsequently built, were, so to say, "automatically" placed in the coresponding positions in the car structure modelled when specific files saved in the *.stl format were opened. Afterwards, having already been placed in the MSC.ADAMS software environment, the car tyre models adopted in compliance with the formalism proposed by Pacejka [13, 14] were introduced with the use of the "Tire" module into the structural model of the car. At the final stage of the modelling procedure, appropriate joints were introduced to connect all the modelled components into a self-contained whole. The structural model built, was then subjected to a series of motion simulation tests within the analysis of its dynamics at the next stage of the study and has been presented in Fig. 10.

The analysis was reduced to the investigation of motion of front shock absorbers. In the analysis, the possibility of friction occuring in the areas of contact between piston rods and cylinders in the shock absorbers was taken into account. To this end, a model of a "cylindrical joint with friction" (Fig. 11) offered in the MSC.ADAMS software environment [23] was used. A joint of this kind provides the possibility of two types of relative motion of the parts involved, i.e. translation with a linear velocity value of w and rotation with an angular velocity value of ω. The drawing shows vectors of linear velocity $\mathbf{w}$ and angular

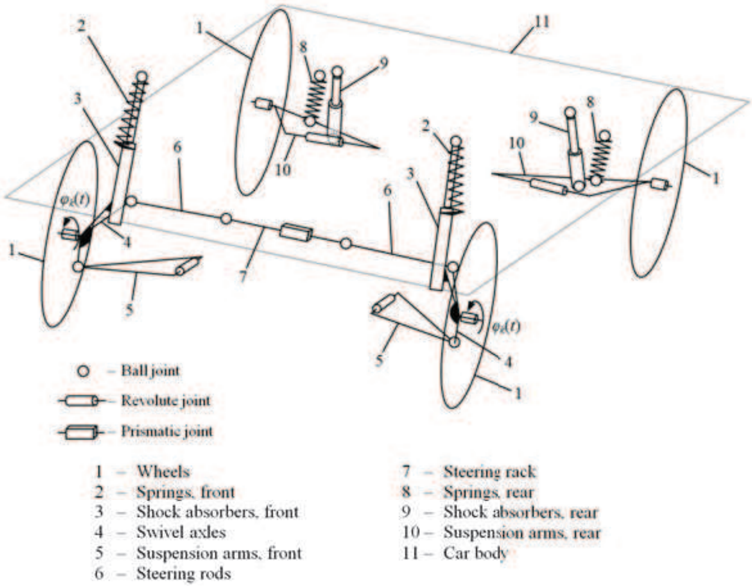

Fig. 9. Major car components taken into account in the structural model of the car.

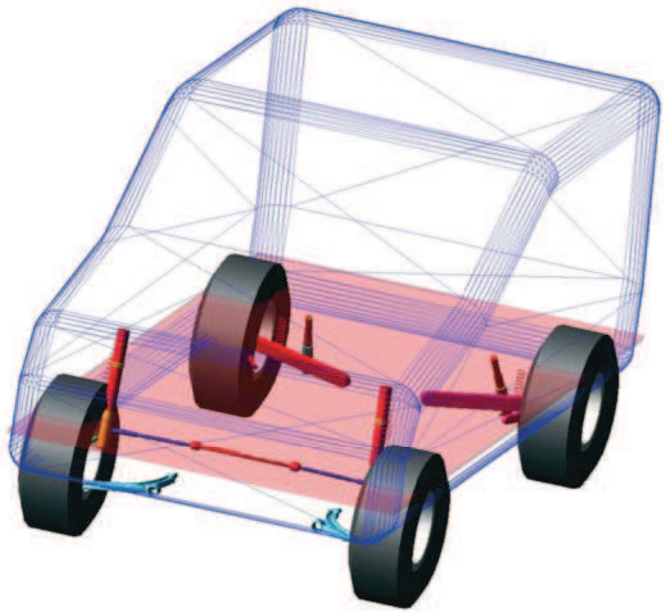

Fig. 10. Structural model of a car, built in the MSC.ADAMS software environment.

velocity $\boldsymbol{\omega}$ of the piston rod in its translational and rotational motion in relation to the cylinder, respectively. The discussion given in the preceding section about the "vector" coefficient of friction $\boldsymbol{\mu}$ vs. the relative velocity value v in the friction contact area in the general sense of this term applies in the case of cylindrical joint under consideration to the values of relative linear velocity w and relative angular velocity ω.

As can be seen in the drawing, right-handed coordinate systems $\mathbf{x}_i\ \mathbf{y}_i\ \mathbf{z}_i$ and $\mathbf{x}_j\ \mathbf{y}_j\ \mathbf{z}_j$, formed by unit vectors, the piston rod and the cylinder, respectively, in the model adopted. The position of the piston rod in relation to the cylinder is defined by two generalised coordinates: linear displacement d and angular displacement φ; hence, the relations $w = |\dot{d}|$ and $\omega = |\dot{\varphi}|$ are true. Simultaneously, displacement d defines the length of the "overlap" between the cylinder and piston rod. The interaction between the cylinder and piston rod may be reduced to forces with values of F and couples of forces bending the two parts and producing moments with values of M. The vectors of these forces and moments are situated in a plane that is perpendicular to the axis of the joint and is situated in the middle of the distance between the origins of the coordinate systems; a fragment of this plane has been shaded in the drawing. The drawing shows vectors representing the loads applied

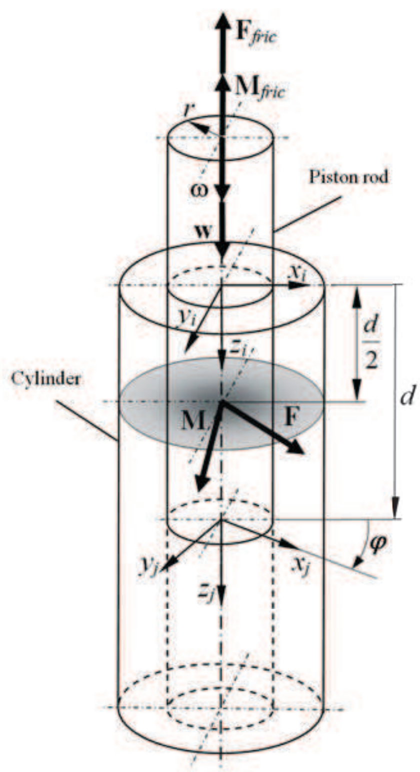

Fig. 11. Model of a "cylindrical joint with friction" (prepared by the authors, based on [23]).

to the piston rod by the cylinder and having the form of force $\mathbf{F}$ and moment $\mathbf{M}$ produced by a couple of forces. As a result of the interaction between the components of the joint, friction forces and moments with values of F_{fric} and M_{fric}, respectively, are generated. The drawing shows the vectors of loads of this kind, applied to the piston rod by the cylinder, i.e. friction force $\mathbf{F}_{fric}$ pointing opposite to the vector of relative linear velocity $\mathbf{w}$ of the piston rod and friction moment $\mathbf{M}_{fric}$ pointing opposite to the vector of relative angular velocity $\boldsymbol{\omega}$. To determine these vectors, a procedure offered in the MSC.ADAMS software environment and represented by the schematic diagram shown in Fig. 12 is used (i.e. these vectors are outputs of the procedure). The inputs of this procedure are linear displacement d and vectors of force $\mathbf{F}$ and moment $\mathbf{M}$ produced by the coupling of force. They are determined for specific points of time during the solving of equations of motion. Other inputs for the procedure are the constant values F_0 and M_0 of the force and moment of kinetic friction, respectively, resulting from the initial grip that holds the joint parts together.

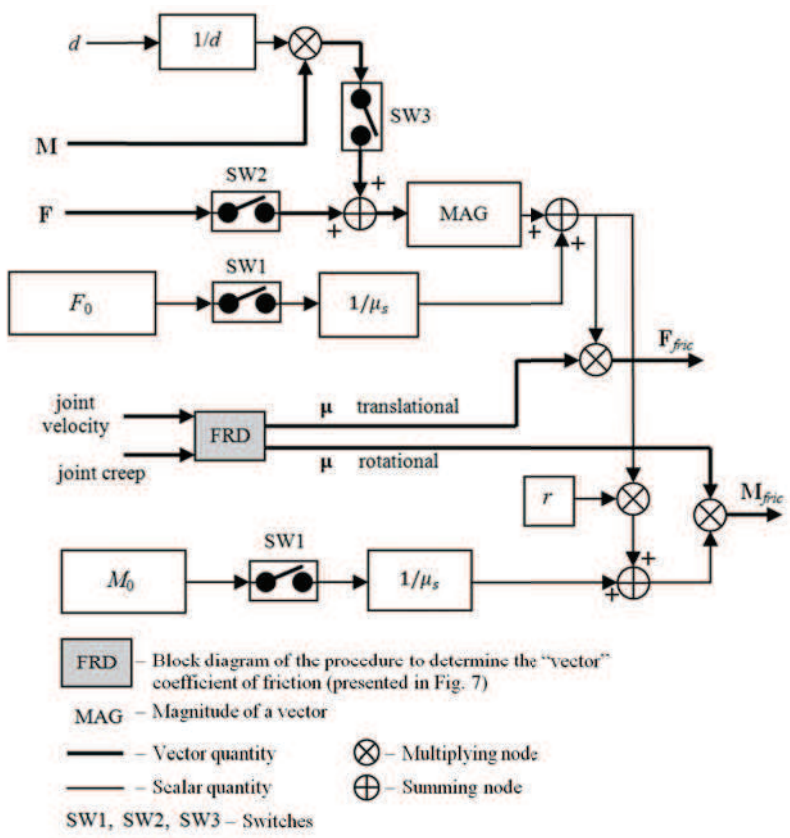

Fig. 12. Block diagram of the procedure to determine the force and moment of friction in a cylindrical joint (prepared by the authors, based on [23]).

As it can be seen in the diagram, the value of the normal pressure force in the joint is calculated from an equation based on the magnitude of a vector sum:

$$N = \left| \frac{1}{d}\mathbf{M} + \mathbf{F} \right| + \frac{F_0}{\mu_s} \tag{4}$$

The vectors of the force and moment of kinetic friction in the joint under consideration are determined from the following formulas, respectively:

$$\mathbf{F}_{fric} = \mu N, \tag{5}$$

$$\mathbf{M}_{fric} = \mu \left(Nr + \frac{M_0}{\mu_s} \right), \tag{6}$$

where: r is the piston rod radius (inner radius of the cylinder) as shown in Fig. 11.

The value of the coefficient of static friction μ_s was estimated from results of experimental tests carried out for a brand-new shock absorber [16]. Before starting the tests, oil was removed from the shock absorber and the mating piston rod and cylinder surfaces were carefully cleaned and pickled for dry friction contact zones to be obtained. Based on the information provided in publication [2], the boundary "creep" value was determined as Δ_{max} = 5×10^{-6} m and Δ_{max} = 5×10^{-6} rad for the relative translational and rotational motion of shock absorber components, respectively. Furthermore, the "threshold velocity" values for the simplified Stribeck curves adopted were assumed as Δv = 10^{-4} m/s and Δv = 10^{-4} rad/s for the relative translational and rotational motion of shock absorber components, respectively, based on the suggestions given in [8] and [9]. Apart from this, the calculations were made for the values of F_0 = 0 and M_0 = 0, i.e. an assumption was made that no initial grip existed between the mating parts.

4. Results of the analysis of dynamics of the structural model in the MSC.ADAMS software environment

When simulating the movement of a car, the car behaviour during lane-change manoeuvres starting with left turn and right turn was analysed.

For the purposes of the analysis, the input parameters were assumed as follows:

- During an initial period of 3 s, the car speed was increased to 60 km/h through pre-programming the values of the front wheel rotation angle to change with time according to appropriate curves $\varphi_k(t)$ (Fig. 9); during the next part of the simulation process, the car moved with a constant speed with a value as above.
- Afterwards, the value of the steering rack displacement was set to change with time according to a curve (Fig. 13) created with the use of splines determined in accordance with the algorithm proposed by Akima [1].

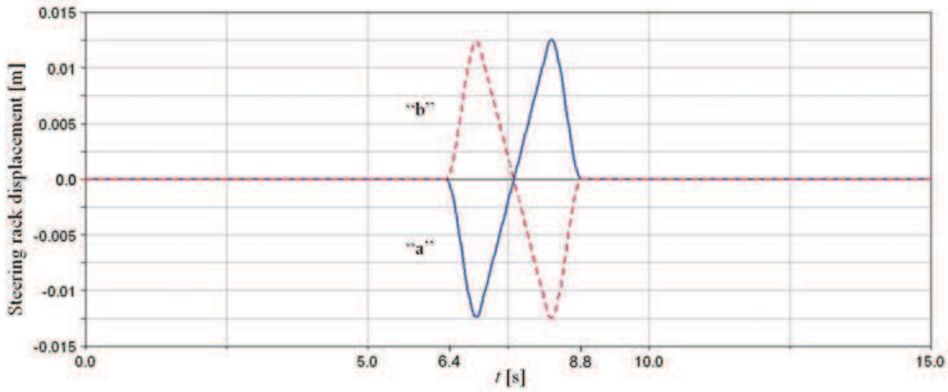

Fig. 13. Steering rack displacement vs. time during a lane-change manoeuvre started with left turn (case "a") or right turn (case "b").

At the input programmed, the distance between the tracks of wheels of the car model before and after the line-change manoeuvre was 3.7 m and the manoeuvre lasted for 2.4 s, as can be seen in Fig. 13 (which means that the manoeuvre time may be considered as reliably realistic).

The simulation was carried out for three sets of values of the coefficient of static and dynamic friction: μ_s = 0 and μ_d = 0, μ_s = 0.2 and μ_d = 0.15, and μ_s = 0.4 and μ_d = 0.3. For the first set of these values, an assumption was made that the friction contact areas in the shock absorber were suitably lubricated by the oil filling the unit, i.e. that fluid friction took place between the mating parts (according to suggestions made in [6] and [11], a damping coefficient value of 1250 Ns/m was then adopted for the model of a cylindrical joint). For the second and third sets of values of the coefficient of friction, which may be assessed as practically realistic and very high (i.e. rather unlikely), respectively, dry friction was assumed to take place in the contact areas. This may happen when oil has flown out of the shock absorber as a result of a breakdown; therefore, a zero value of the damping coefficient was adopted in both cases for the model of a cylindrical joint.

Figs. 14 ("a" and "b") and 15 ("a" and "b") show time histories of increments Δd of linear displacements d of piston rods in relation to cylinders in the left and right shock absorbers, measured from the displacement values corresponding to the state of static equilibrium, for the two cases of input ("a" and "b") presented in Fig. 13. Three types of the time histories have been shown in each of these figures: "1" for μ_s = 0 and μ_d = 0, "2" forμ_s = 0.2 and μ_d = 0.15, and "3" for μ_s = 0.4 and μ_d = 0.3. In each of the time histories presented, four recurring phases, denoted with symbols "I," "II," "III," and "IV," can be distinguished.

Phase I corresponds to the time interval when the speed of the car modelled increased to a value of 60 km/h. This phase began with a period lasting about 0.8 s, during which the position of the car modelled was stabilised and the spring elements included in the structural model of the car were quite rapidly deformed due to the impact of gravitational forces. In consequence, the shock absorber piston rods were momentarily displaced in relation to their cylinders and they then began to return. The acceleration of the centre of mass of the geometric model of the car body initially increased to the maximum and then dropped to zero. In result of this, the piston rods moved in relation to their cylinders in the second part of the phase under consideration as shown in the drawing; however, these return movements only occurred in the case of lower values of the coefficient of friction (curves "1" and "2"). For the values of the coefficient of friction adopted as μ_s = 0.4 and μ_d = 0.3, the piston rod return displacements in relation to the cylinders did not take place (curves "3").

Phase II corresponds to the period of the car being driven "straight ahead" with a constant speed.

During phase III, a lane-change manoeuvre was performed. The input was started with left turn (case "a" in Fig. 13), the car body modelled tilted first to the right and then to the left. When the input was started with right turn (case "b" in Fig. 13), the car body modelled tilted in reverse order. In result of these tilts, unsymmetrical relative displacements of the mating parts of the shock absorbers took place.

The curves presented in Figs. 14 (a) and (b) and 15 (a) and (b) show that during phase IV ("calming down of the motion"), the shock absorber components, in principle, only did not return to their position observed before start of the lane-change manoeuvre (place "a" in the graphs) but stopped in another position (place "b") in the case of significant friction in the shock absorber contact areas have been assumed (i.e. when the coefficients of friction were μ_s = 0.4 and μ_d = 0.3). If places "b" of curves "1," "2," and "3" presented in Fig. 14 (a) are compared with the same places of the corresponding curves in Fig. 14 (b) then the difference in the displacements of piston rods of both shock absorbers resulting from the turn, which determines the magnitude of variations in the distribution of car body weight among road wheels, may be estimated as being practically equal to zero. The same applies to the situation illustrated in Figs. 15 (a) and (b). Thus, the results obtained show that even if significant values of the coefficients of friction in the contact areas of shock absorber components are taken into account then no variations in the distribution of car body weight among road wheels will take place.

As could be expected, the curves shown in Figs. 14 (a) and (b) are almost identical to those of Figs. 15 (b) and (a), respectively. This confirms the method proposed to be appropriate: behaviour of this type should obviously be expected from the structural model of the car under consideration, which is an almost symmetrical system. The insignificant differences that can be observed in the shapes of the mutually corresponding curves may result from the asymmetry of the geometric model of the steering rack, where the teeth cut in the left part of the rack (viewed by the driver) as it is in the actual assembly (Fig. 1) have been taken into account.

Figs. 16 (a) and (b) show time histories of the values of the β and μ_1 factors (illustrated in the diagram in Fig. 7) that were determined when simulating the manoeuvre of changing the lane to the left for the case of μ_s = 0.2 and μ_d = 0.15, which corresponds to the curves

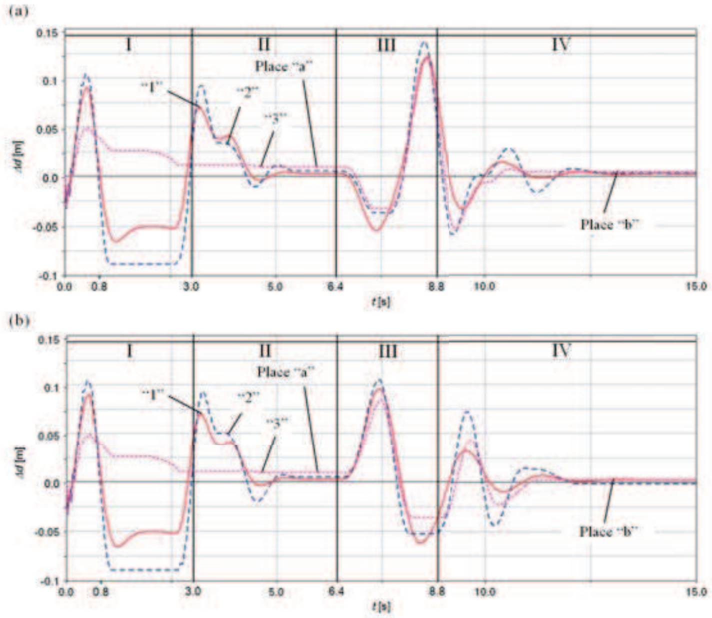

Fig. 14. Lane-change manoeuvre started with left turn.
Time histories of increments Δd of linear displacements d of piston rods in relation to cylinders in the left (a) and right (b) shock absorbers.

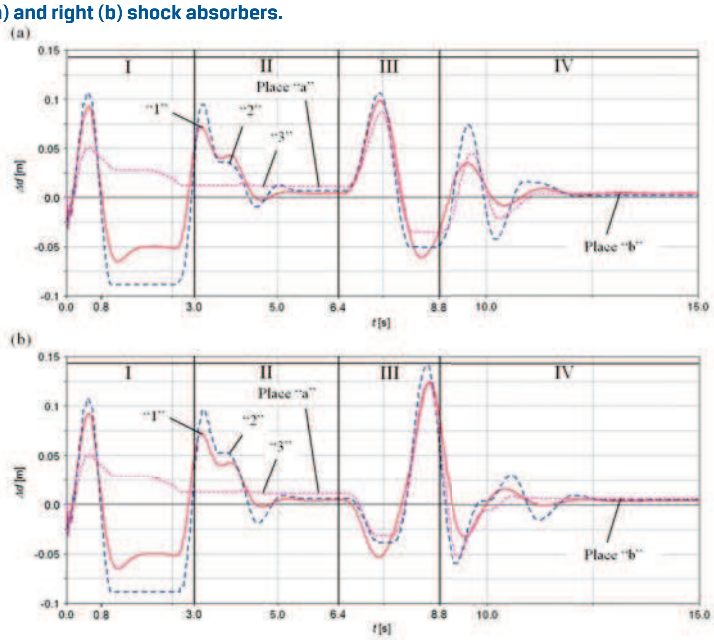

Fig. 15. Lane-change manoeuvre started with right turn.
Time histories of increments Δd of linear displacements d of piston rods in relation to cylinders in the left (a) and right (b) shock absorbers.

denoted by "2" in Figs. 14 (a) and (b). The phases of kinetic friction (where $\beta = 1$ and $\mu_1 = 0$) and static friction (where $\beta < 1$ and $\mu_1 > 0$) can be observed in the said Figs. 16 (a) and (b). According to the time history of the μ_1 value, only twice did the this coefficient reached its boundary value of $\mu_1 = \mu_s = 0.2$, which corresponded to the situation that $\Delta = \Delta_{max}$ (lower shaded window in Fig. 7). This means that the actual "creep" reached its boundary value of $\Delta_{max} = 5\times10^{-6}$ m twice, too.

Calculations carried out have shown that the rotational motion of shock absorber piston rods in relation to their cylinder takes place in a very small angular range, not exceeding 3°. Therefore, this motion has been ignored in the analysis of dynamics of the structural model of the car under consideration, with analysis exclusively focused on the investigation of the relative translational motion of the mating shock absorber parts.

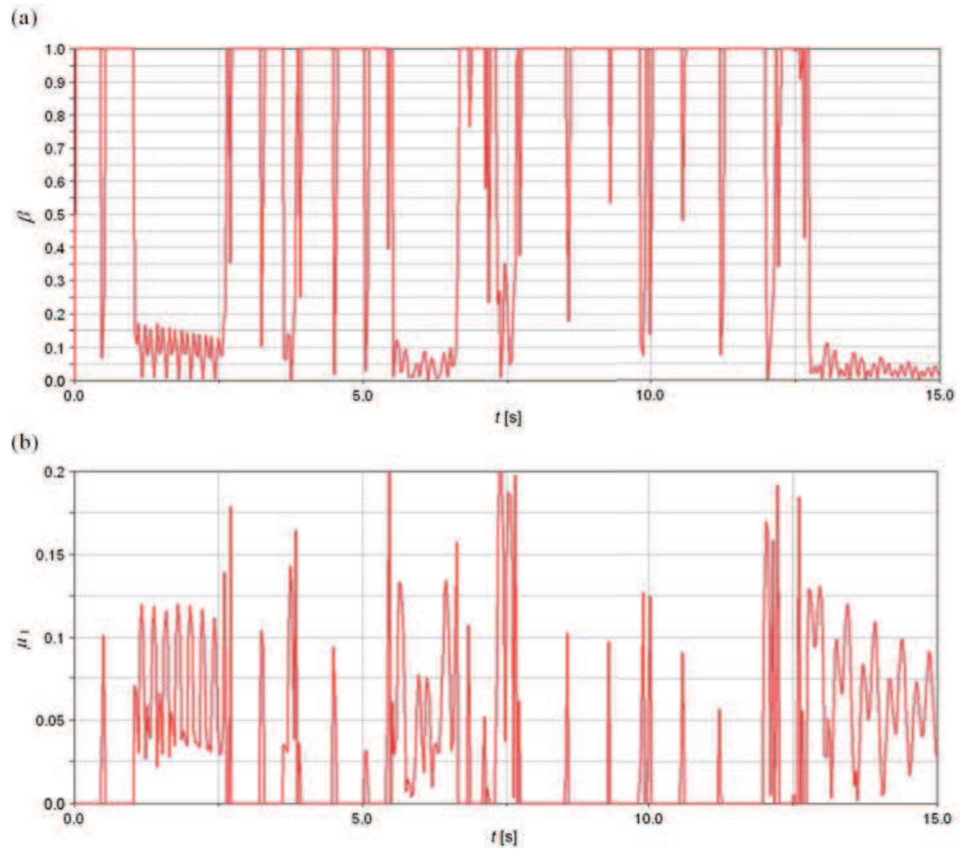

Fig. 16. Time histories of the β and μ_1 values, shown in the (a) and (b) parts of the drawing, respectively.

5. Recapitulation

The paper constitutes an attempt to analyse the dynamics of passenger cars with a McPherson strut suspension system taking into account the complex state of friction occurring between the mating parts of front shock absorbers of such cars. In particular, the discussion was focused on investigating the impact of increasing values of the coefficient of friction in the friction contact areas in shock absorbers during the self-jamming states that may result in variations in the distribution of car body weight among road wheels. As a result of calculations carried out, no tendency for such variations to occur has been revealed, even though significant values of this coefficient were taken into account. In the authors' opinion, this may indicate suitable of the construction of the front suspension system of the modern car under consideration, which already exhibits low susceptibility to the possibility that significant friction might develop in friction contact areas in the front shock absorbers in result of any failure. This means that the construction examined may be considered completely safe.

In authors' opinion, the presented method of analysing the dynamics of passenger cars with McPherson strut suspension systems, make it possible to take into account the complex state of friction in front shock absorbers of such cars. This analysis could be used during cardesign stage to check the cars behaviour in failure situations. When conditions might be created for variantions car the distribution of car body weight among road wheels.

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