ANALYTICAL APPROACH FOR VEHICLE BODY STRUCTURES BEHAVIOUR UNDER CRASH AT ASPECTS OF OVERLOADING AND CRUMPLE ZONE LENGTH

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Abstract: Road safety problem is still topical, especially since the number of vehicles and the volume of traffic are increasing. It is possible to increase the safety of road users through systemic changes in many areas related to transport. The deformation of the vehicle body during an accident has an impact on the loads acting on the passengers. Vehicle body deformation depends on complex parameters, and knowledge of these parameters is essential for designing crumple zones and the accident reconstruction process. Knowledge of the mechanical parameters of the vehicle structure during deformation is also a reference to passenger injury indicators assessment. This paper reports results from the analytical approach for determining the protection level of personal vehicles. The proposed conception is based on the results from the static stiffness characteristic of the Ford Taurus, which gives the possibility of phenomenological and simple body crumple analytical description at a speed equal to 10 km/h, 40 km/h, 56 km/h and 60 km/h, which is an original part of the work. The approach enables us to describe the vehicle crash by focusing on variations of deformation in time, stiffness, vehicle collision time (duration), deceleration and dynamic crash force. Basing on the body stiffness data of the personal vehicle, the length of the deformation zone in the front of the car and the maximum values of force at the crash for a speed of 60 km/h are presented. Results obtained by the authors show that is possible to estimate the overloading level during the crash time of a vehicle based on the stiffness characteristic of the car body. The proposed methodology can be developed and the advantage of the presented procedure is an uncomplicated useful tool for solving complex problems of a vehicle crash.

Key words: crash test, car body stiffness, overload, dynamic force

1. INTRODUCTION

Road safety is constantly considered by engineers and researchers' groups for the design of modern cars at increasing the protection level against a crash. This becomes also more significant because the number of vehicles is increasing. The importance of this problem can be proved by the development of passive safety protection systems, gaining its expression among others in the number of published research results. They focus not only on the assessment of the impact of seat belts functioning on the passenger's body [1–6], the concepts of its improvement [6], manufacturing, testing [7,8,9], minimizing the risk of injury [10] but also the overloads and zone size crush [11].

Regardless of the passive safety systems used, the body structure and the resulting rigidity play a key role. Sauders et al. [12] analysed the rear impact resistance of seats used in passenger cars. Advances in the design of car seats have been noted, however, there are cases where seats aren't sufficiently resistant to accident overloads. Of course, there exists a correlation between the dynamic crumple stiffness of the car's body and dynamic forces operating on the seats and passengers. Sugimoto et al. [13] pointed out the problem of the conducted crash tests to real road accidents. The work presents the results of vehicle-to-vehicle crash tests, especially in the situation when cars have different weights. The question was asked about the reasons for the differences between a crash test and a real accident. Attention was also paid to the problem of cars hitting energy-consuming barriers. Citing the results of research conducted in the USA, it was noted that there is a strong correlation between the mass and the stiffness of the vehicle. The authors defined car body stiffness as the slope of the loading on the chassis as derived from an accelerometer attached to the cabin floor on the chassis. Witteman [14] also researched the weight influence of the crash car process. Pointed out the event of a vehicle with a mass >1,500 kg colliding with a honeycomb structure, the test results are different from real collisions. Jawad and Saad [15] described a problem of different mass of vehicle compatibility during a crash. The phenomenon of deformation of the body structure during frontal collisions was investigated. Simulation result for the dynamic behaviour of the crumple zone was presented. Also investigated relation between car crash parameters and passengers' injury criteria. Sadeghipour in his doctoral dissertation [16] considered the problem of the compatibility of cars in Europe during a collision. The author noted that the crash tests of the 1980s and 1990s are not effective in testing modern cars. Due to the high cost of crash tests, the author has conducted extensive finite element method analysis. Barbat et al. [17] conducted research also related to the compatibility of vehicles during a crash. Authors paid attention that geometric interaction, vehicle mass and body stiffness are decision parameters in the vehicle deformation process. The FEM method was used for model simulations. Subramaniam et al. [18] investigated the compatibility of crumple zones and according to the authors, the structural compliance of the crush zones of various cars can improve road safety. In road accident crash analysis and vehicle deformation, the popular method is energy-based analysis. Żuchowski [19] researched the dependence of the impact force on the deformation of the car's body based on energy methods. The case of a car hitting an energy-consuming barrier was considered and for applied simplified linear model identification parameters, the results of crash tests of three different cars were used. Differences in the design of the front body parts of cars manufactured today and 20 or 30 years ago were noted, which is important in assessing the speed of the collision. The author also commented on the issue related to the car's body dynamic stiffness during an impact. The described method was analyzed for three popular cars: Toyota Echo, Honda Accord and Ford Escape. Leibowitz's PhD dissertation [20] is thematically related to the analytical and experimental methodology of car crash energy calculation based on body deformation stiffness. Based on NCAP NHTSA tests data frontal body structure was modelled. Attention was drawn to the necessity of experimental tests to determine the stiffness characteristics of vehicle bodies. The author used the 3D scanning method for more accurate measurement of the deformed body. The author drew attention to the current industry standard for crash energy calculation which is based on the assumed linear relationship of energy and vehicle crush. The body stiffness calculation was based on NHTSA's Full Frontal NCAP. Neades in his doctoral dissertation [21] described the mechanics of the vehicle movement in the aftermath of the crash. A new methodology was proposed to describe the total work of the deformation zone to a particular vehicle. The advantage of the proposed method is the incorporation of restitution effects and the obtained results are identical to those obtained based on the momentum. Khattab's PhD thesis [22] described the problem of controlling energy dissipation of passive and adaptable energy absorbers during crumple zone deformation depending on the vehicle speed during a collision. Different mechanisms of impact energy dissipation have been highlighted. Tests of the damping elements used in the construction of the car body were carried out. McCoy et al. [23] considered the problem of stiffness of additional steel elements attached to the front of the car body. It was noted that additional elements or covers attached to the body result in a change in the frontal body stiffness. The authors determined the stiffness coefficients of the described additional body elements and compared them to frontal body stiffness. Chen [24] tested the dynamical properties, and the damping behaviour of structures filled with aluminium foams was carried out. Hollowell et al. [25] described and analysed several crash test procedures. Brell [26] described factors that affect crumple zone deformation and response. Impact on passengers during a collision is also presented. The author described the instantaneous stiffness of the crumple zone. Research is also focused on collision modelling. Lukoševičius et al. [27] presented three and four mass dynamic models of crumple zone for passenger cars during a frontal impact. The linear stiffness in the initial stage of the vehicle body deformation and then the ideal plastic deformation of the crush zone were assumed resulting in the body stiffness which then increases to infinity. The necessity to use nonlinear relations for stiffness at higher collision velocities was pointed out. Pahlavani and Marzbanrad [28] presented a 12-degree model of a vehicle for a frontal crash, and composed model based on basic rheological material models such as spring, dumper and Maxwell body. The result obtained by model analysis was compared to experimental crash tests. Munyazikwiye et al. [29] presented a mathematical model of the frontal crash of vehicle-to-vehicle. The model is based on the mass of the crashed cars and linear Kelvin elements which represent the mechanical properties of the crumple zones. Wiacek et al. [30] studied the impact of modern materials such as high-strength steel and aluminium on a car's body structure stiffness. The authors used four methods for car body stiffness calculation based on cars produced between 2002 and 2014. Data were taken from NCAP frontal crash experimental. A very interesting problem was raised by Sungho and HaengMuk [31] who investigated the problem related to vehicle bodies after crash repair. Using the technical data of a popular personal car, simulation research was carried out. The authors noted the reduction in the strength of the repaired body and the change in stiffness to the vehicle that did not crash. Prochowski et al. [32] investigated side-impact vehicle crash in terms of accident reconstructions. Based on experimental results, deformation and stiffness parameters were estimated.

Sahraeia et al. [11] presented results of the impact that linked the stiffness of the frontal structure of the car to the risk of injuries of passengers in the rear seats. The stiffness of the structure can be improved by applying high-strength steel or by increasing the thickness of the sheets which was investigated by Wiacek et al. Vehicle stiffness can be also controlled by profiles shape which was described by Obst et al. [33, 34].

Car accidents are also a problem of passenger injuries the scope of which also depends to a large extent on the mechanical properties of the vehicle crush zone.

The simulations on the MADYMO mannequin showed an increase in the incidence of head injuries from 4.8% (at 1,000 N/mm stiffness) to 24.2% (at 2,356 N/mm stiffness). Additionally, the risk of chest injuries increased from 9.1% to 11.8%. According to the authors, additional measures should be introduced to protect passengers seated on the rear seats when the stiffness of the vehicle increases. Such safeguards may be applied using additional airbags or active seat belts. The comparison of the injuries of passengers seated in the front seats with the injuries of passengers seated in the rear seats was investigated by Mitchell et al. [35].

Bunketorp and Elisson [36] pointed out that further analysis of the resulting injuries with particular attention to angular deformation and translation should be followed.

A reduction of consequences due to vehicles crash requires employing more energy-absorbing elements which was described by Szeszycki [37]. However, the practical use of these components can be achieved at low speeds such as no more than 30 km/h. At higher values of the physical quantity, absorbers in the body structure of the front of the vehicle play an important role because they carry the impact energy by cracking, deforming and collapsing thereby saving passengers.

Because all the aforementioned test bench methods sometimes show far-reaching and difficult-to-estimate discrepancies with the actual impact effects, which in terms of initial conditions may significantly differ from those assumed experimentally, and the only reliable characteristic is obtained in the static longitudinal compression test of the car body, so the dependence of the force

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on the length of the crumple zone is reasonable to describe the behaviour of the vehicle based on these characteristics.
Additionally, the widely used definitions of mean values of car body stiffness may not be an optimal approach because it does not take into consideration the local values of stiffness and their influence on instant acceleration.

The aim of this paper is focused on analysing the differences between the equivalent quasi-constant dynamic stiffness of the front vehicle body structure and the instant stiffness, in terms of the length of the crumple zone and the maximal overloading occurring during the frontal impact of the car into a stiff obstacle. It is also an attempt at a simple, based on a static characteristic, mathematical description of the deceleration generated during the collision process, which is also the original part of the work. The proposed simple analytical methodology can be of course developed in parallel to the crash test of real cars. Based on the proposed model, more advanced analysis is possible if for example we have data on the dynamic stiffness of the car's crumple zone. Presented solutions can be adopted for analysis of rear crash problems, in case of facial impact. It is worth noting that the proposed simple analytical model is handy which in the case of engineering practice is an excellent analytical tool.

To the fact that detailed experimental data on the static vehicle front body stiffness are not generally available, it was decided to use the characteristics of Ford Taurus presented by Sahraeia et al. As a result, the lengths of the frontal vehicle body deformation zones and the maximal overloading values at impact for the car speed of 10 km/h, 40 km/h, 56 km/h (Euro NCAP "The European New Car Assessment Programme" test) and 60 km/h were determined analytically and numerically.

2. THE PHENOMENOLOGICAL METHOD

A used method of investigation was the phenomenological approach, applied to experimental characteristics, taken from Ref. [11]. The experimental results of the static compression of the Ford Taurus whole car body structure, provided the relation between force and displacement. The approximation of discussed relation lets to formulate the differential equation of the car body motion, during face collision with the rigid obstacle. The solution of the discussed equation, complemented by the boundary conditions (like mass and initial velocity) gives the functions describing the length of the crash zone, deceleration and in consequence impact force.

Let us take into consideration the collision process of a car with mass m [kg], initial velocity V_0 [m/s] and body stiffness k [kN/mm], with an infinitely stiff undeformable obstacle. In the case of a small number of protective elements, the impact energy is directly absorbed by the vehicle's superstructure at an unknown and difficult-to-estimate force. It is influenced by both, the construction and speed of the car. Protective components must determine the maximum values of overloading at the moment of the collision, and the length of the crumple zone. Each of these quantities is closely related to the probability of the survival chance of the driver and passengers. Nevertheless, the amount of experimental data on the vehicle body characteristics is still not enough. However, in Ref. [11], the behaviour of the Ford Taurus during the collision is discussed in details.

In Fig. 1, we can see that the car's body response to the crash is in a form of an increasing relationship between force and displacement with oscillations, indicating variations in a range of deformation zones. This makes it difficult to elaborate in detail on the car's behaviour during the crash. Therefore, this relationship can be presented in a straight line. The characteristic wave of the graph is associated with the deformation of the car body resulting in complex blocking and destruction of its overlapping elements. Besides the knowledge of this stage is well represented by the relationship between force and displacement as well as the deformed components as a function of the observed behaviour is irregular. Therefore, the force versus displacement under a crash can be approximated linearly as follows:

$$F(b) = kb, \tag{1}$$

where: k is the stiffness coefficient (slope of the straight line on Fig. 1, $k \approx 1\ [\mathrm{kN}/\mathrm{mm}]$), $b\ [\mathrm{mm}]$ is the depth of the crash zone from the point of contact between the front bumper and the obstacle.

The discussed situation is illustrated in Fig. 2.

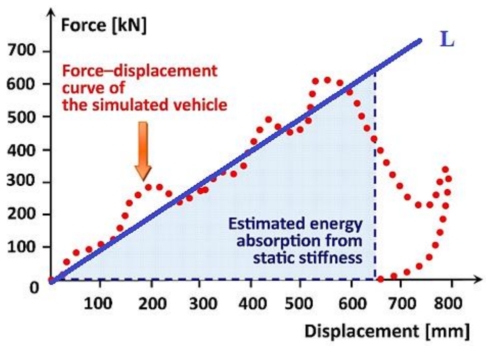

Fig. 1. Variations of force versus displacement for the Ford Taurus crash test. Data taken from Ref. [11]

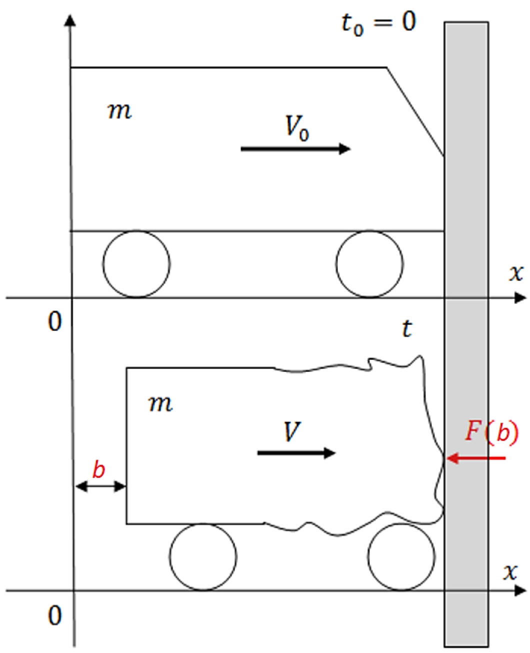

Fig. 2. The car body in two stages of a crash test on an undeformable obstacle: (a) in the initial stage without any deformation and (b) in the further stage at elastic and plastic deformation

Using Newton's second law, we receive:

$$ma = -F(b), \tag{2}$$

or equivalently in differential form:

$$m\frac{d^2b}{dt^2} = -F(b), \tag{3}$$

$$mV\frac{dV}{db} = -F(b), \tag{4}$$

where $V\left[\frac{m}{s}\right]$ and $a\left[\frac{m}{s^2}\right]$ are the velocity and deceleration during the collision, respectively.

Using Eq. (1) and making some transformations the following equation can be obtained:

$$V^2 = \frac{-k}{m}b^2 + C. \tag{5}$$

Velocity $V_0\left[\frac{m}{s}\right]$ at the beginning of the collision process satisfy the following assumption: $V_0 = V(0)$, therefore $C = {V_0}^2$, which in turn means:

$$V(b) = \sqrt{{V_0}^2 - \frac{k}{m}b^2}. \tag{6}$$

Eq. (6) specifies the speed of oncoming movement as a function of path s destroying the deformable compartment of the body of the vehicle.

From a practical point of view, a more meaningful equation is one that shows the dependence of the path of destruction of the deformable compartment of the body of the vehicle as a function of time.

From the fact that $V(b) = \frac{db}{dt}$ the following relationship can be written:

$$\frac{db}{dt} = \sqrt{{V_0}^2 - \frac{k}{m}b^2} \tag{7}$$

and after transformations, as follows:

$$\arcsin\left(\sqrt{\frac{k}{m{V_0}^2}}b\right) = \sqrt{\frac{k}{m}}t + D. \tag{8}$$

Because in time of the contact of the front part of the body of the car and undeformable obstacle, the body deformation equals zero, therefore $b(0) = 0$. After transformations $D = 0$, and finally as below:

$$b = V_0\sqrt{\frac{m}{k}} \cdot \sin\left(\sqrt{\frac{k}{m}}t\right). \tag{9}$$

Differentiating the obtained equation after time we get:

$$V = \frac{db}{dt} = V_0 \cdot \cos\left(\sqrt{\frac{k}{m}}t\right), \tag{10}$$

and hence:

$$a = \frac{dV}{dt} = -V_0 \cdot \sqrt{\frac{k}{m}} \cdot \sin\left(\sqrt{\frac{k}{m}}t\right), \tag{11}$$

The vehicle stops during the crash when $V = 0$. The time of the collision process can be written as:

$$t = \frac{\pi}{2}\sqrt{\frac{m}{k}}. \tag{12}$$

In that time the maximal deceleration occurs if:

$$a_{\max} = -V_0 \cdot \sqrt{\frac{k}{m}}, \tag{13}$$

and the impact force can be expressed as follows:

$$F_{\max} = m \cdot a_{\max} = -\sqrt{mk} \cdot V_0. \tag{14}$$

The deceleration value Eq. (13) is only an approximation and may significantly differ from the maximum peak, which is significantly influenced by the detailed characteristics of the body deformation (body structure and used components, types of engineering materials, operational conditions), impact speed and the associated effect of dynamic stiffness oscillation and many other minor factors. To get a result that is closer to the realistic model of the destruction of the body while crashing on a rigid obstacle the static characteristics of the body of the Ford Taurus car were used [11].

Using the differential equation of the motion during a vehicle collision with a rigid obstacle and making the transformations the following formula is obtained:

$$\frac{1}{2}mV^2 = -\int_0^b F(l)dl, \tag{15}$$

where the right side of the above equation contains everything, which is responsible for longitudinal car body static reaction. Introduced this way the phenomenological conception eliminates the necessity of knowledge about experimental car body parameters like stiffness and damping coefficients. Those parameters are neglected, when someone decides to use the multimas model in analytical crumple zone modelling, like Lukoševičius et al. [27] and can be determined in the numerical way (FEM "Finite Element Method" investigations), like work of Lankarani and McCoy [23]. Applying the initial condition $V(0) = V_o$ the following equation can be received:

$$V = \sqrt{{V_o}^2 - \frac{2}{m}\int_0^b F(l)dl}. \tag{16}$$

The total depth $\hat{b}$ of the zone (crumple) of controlled destruction of the front part of the body of the vehicle is obtained by solving the following equation:

$$V(\hat{b}) = 0, \tag{17}$$

Eq. (16) allows us to determine the speed as a function of the current depth of the deformed vehicle body compartment. However, it does not provide information about the variability of discussed velocity in time, and thus makes it impossible to estimate the delay.

Applying the left side of Eq. (16) to the definition of instantaneous speed we get:

$$\frac{db}{dt} = \sqrt{{V_o}^2 - \frac{2}{m}\int_0^b F(l)dl}, \tag{18}$$

and finally:

$$t = \int_0^b \frac{db}{\sqrt{{V_o}^2 - \frac{2}{m}\int_0^b F(l)dl}}. \tag{19}$$

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As we can see that Eq. (19) is different compared with Eq. (12) because $F(t)$ is non-linear.
Marking the right side of equality Eq. (19) by $G(b)$ we get:

$$t = G(b) - G(0), \tag{20}$$

and finally after some transformations:

$$b = G^{-1}(t). \tag{21}$$

Thus, the change in delay and speed as a function of time is:

$$\tilde{a}(t) = \frac{-F\left(G^{-1}(t)\right)}{m}, \tag{22}$$

$$\tilde{V}(t) = V_o + \int_0^t \tilde{a}(l)dl. \tag{23}$$

3. RESULTS OF COMPUTATIONS

The authors have decided to present two previously shown approaches of $F(t)$, i.e. the first based on approximation Eq. (1) of the experimental characteristics of static body rigidity and the second as an exact reconstruction without referring to the approximating function relying only on the experimental data set.

The depth of the stopping distance, speed and deceleration (during a collision) as a function of time was determined in two ways: using Eqs (10–12) and (21–23).

The approximate value of the static body characteristic was assumed at the level $k = 1\left[\frac{\mathrm{kN}}{\mathrm{mm}}\right]$, when we used the experimental characteristics, the corresponding velocity value during the impact was determined from Eq. (10).

Based on Fig. 1, the body stiffness of the Ford Taurus car (year 2004, weight 1,740 kg) was reproduced and then the relationship between the speed V during the collision and its duration was numerically elaborated using Eq. (16). The elaborated plots are illustrated in Figs. 3 and 4.

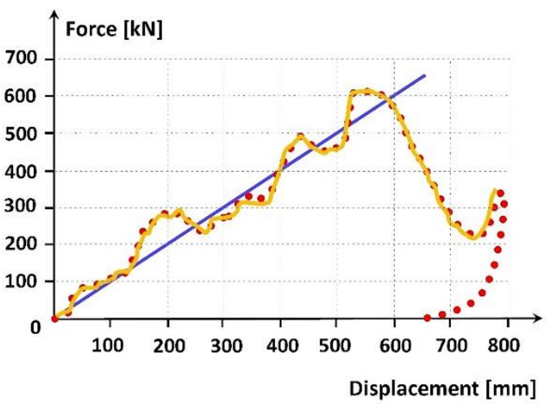

Fig. 3. Superimposition of the reading graph (orange line) on the original body characteristics – data taken from Ref. [11] (red dots represents continuous data) and its linear approximation

Variations of the deformation zone depth were determined at two approaches, expressing the sensitivity of the relationship on the proposed method (Fig. 4). Differences between these results were clearly visible at a speed exceeding 50 km/h. In the case of the relationship between the depth of a deformation zone and time, the differences in the two approaches were not significant Fig. 5. At the speed of 60 km/h the first and second iteration expressed not the same values in s(t) and time (Fig. 5). Variations of the speed versus time during the crash were sensitive to the approach stage (Fig. 6). This was expressed by differences in the value of time, taking 30%.

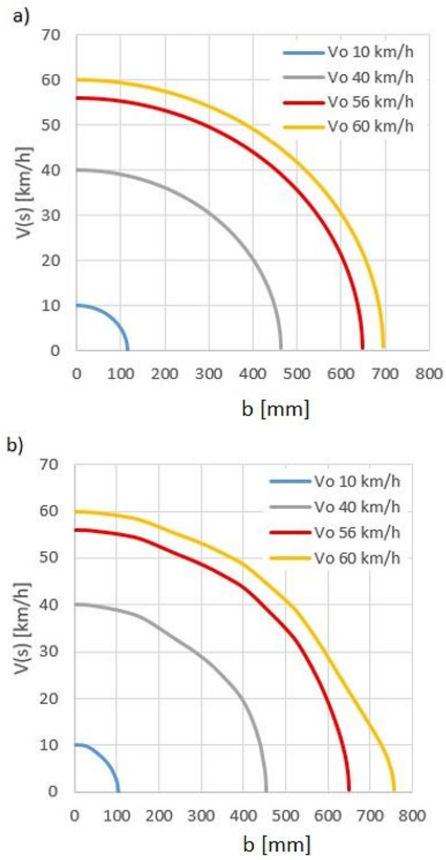

Fig. 4. Car speed as a function of the current crumple depth determined using the first approach (a) and the second approach (b) at the four values of initial speed: 10 km/h, 40 km/h, 56 km/h and 60 km/h

Summarizing the results shown in Figs. 4–7 is worth emphasizing the approximation of the static characteristics of the car body using a straight line:
– does not significantly affect the depth of the crumple zone (Fig. 5),
– influences the time (duration) of the collision, which is independent of the initial speed (Fig. 6),
– significantly changes the relationship of deceleration-time (Fig. 7).

The local maxima, presented in Fig. 7(b), are strongly correlated with the course of the static characteristic versus time shown in Ref. [11].

The results obtained on the basis of the static characteristics and the formulas expressed by Eqs (15)–(23) as well as the relationship versus time are compared with data from the experiment (Fig. 8).

It is observable, that approximation by the use of static vehicle body stiffness characteristics, gives results comparable with the experiment (the model year 2000). The plot of acceleration value (red line determined by the authors) is similar to over plots. The non-linearity and pulsating value of deceleration is a direct consequence of the changeable static and dynamic stiffness of the vehicle body, which is a very accomplished structure. It can be explained based on structural features of the components (manufacturing state) against crash because their mechanical parameters and geometry follow the static stiffness at a not significant speed in a car accident while in the case of a crash at dynamic conditions, significant differences occurred in cross-sections and length of elements leading to a local concentration of permanent deformation, fracturing and collapsing as a final stage. Therefore, different values of force and deformation at crashing can be evidenced. From the practical point of view, it means the crumple zones at the initial state and after an accident at permanent deformation will express significantly different behaviour if a vehicle was repaired in a process with drawing and straightening. It can be connected with a vehicle history and operating condition such as a small collision, i.e. crumple zones of pure cars can be more deformable absorbing a higher level of energy and saving passengers compared to the exploited elements, at the same crush zone design. This means that the zone exhaustion state with deformation will occur earlier than its counterpart without loading history. As a result, the occupied zone of the exploited vehicle will be subjected to loading faster. Additionally, the behaviour of older cars is different from younger ones to the application of modern structural materials such as high-strength steel [38, 39]. Worth noticing that dynamic stiffness is very demanding because the experiment is very expensive due to the permanent deformation of the tested object which does not enable it to use again. Therefore, capturing details from this kind of test is very difficult.

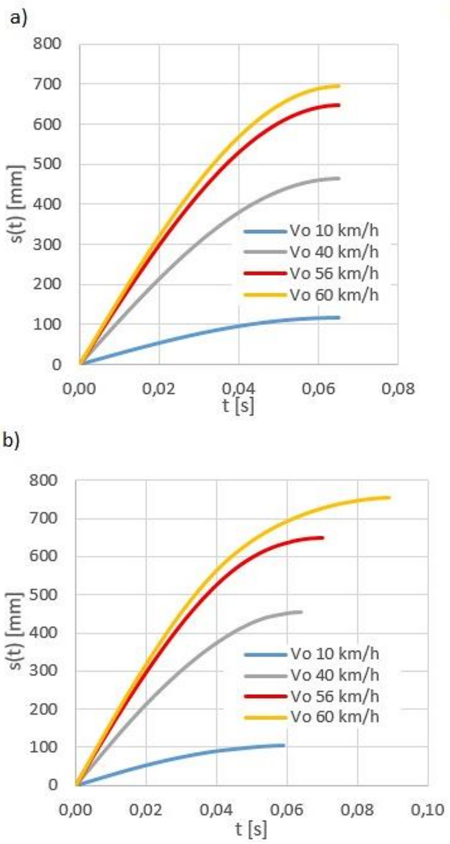

Fig. 5. Depth of the crumple zone as a time function for first approach (a) and second approach (b) captured at the initial speed of car equal to 10 km/h, 40 km/h, 56 km/h and 60 km/h

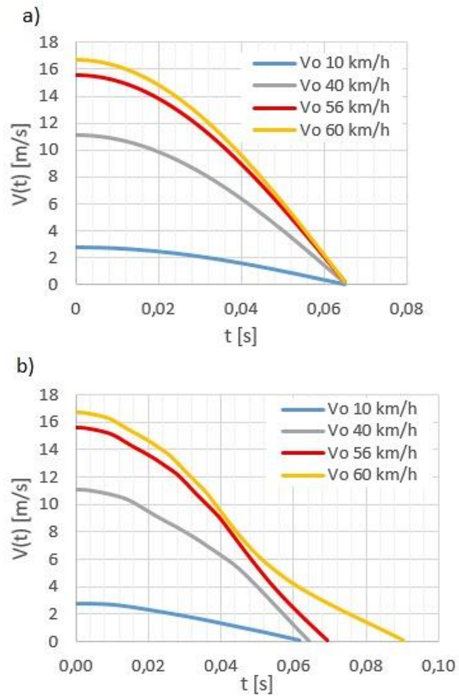

Fig. 6. Car speed versus time during a collision for the first approach (a) and second approach (b) at the initial speed of the car: 10 km/h, 40 km/h, 56 km/h and 60 km/h

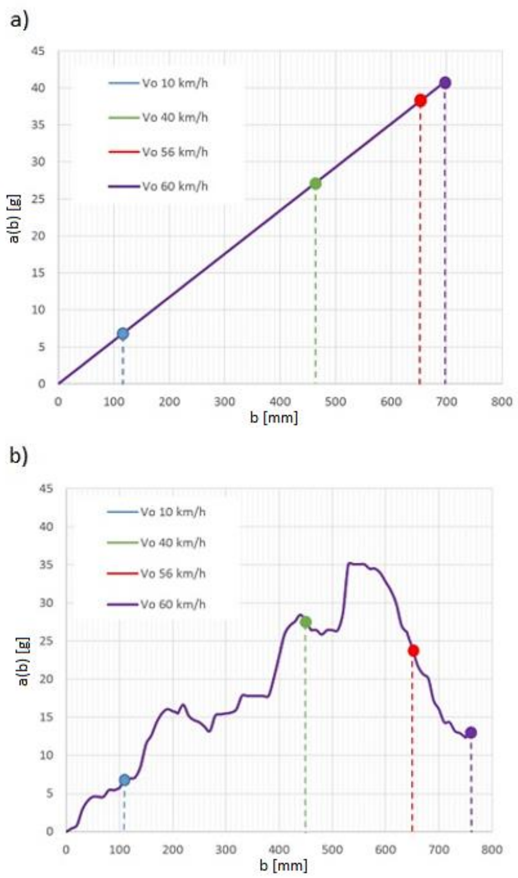

Fig. 7. Deceleration for the first approach (a) and the second approach (b) at the initial speed of the car taking off at 10 km/h, 40 km/h, 56 km/h and 60 km/h

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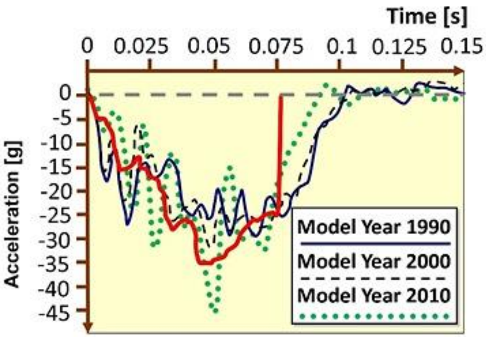

Fig. 8. Comparison of the results determined by means of the authors's approach (red line) and data taken from Ref. [11]

To clarity the comparison between static and dynamic stiffness (determined based on Fig. 8) was shown in Fig. 9. It is based on patterns:

$$\begin{cases} F(t) = m \cdot a(t) \\ b(t) = \int_0^t a(l)dl \end{cases} \Rightarrow F(b) \tag{24}$$

where points of characteristics' a(t) are directly taken from Fig. 8.

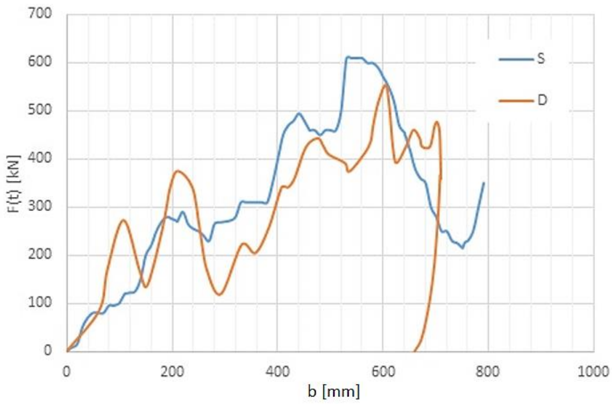

Fig. 9. Comparison of static (S) and dynamic (D) stiffness (impact velocity 56 km/h) of Ford Taurus

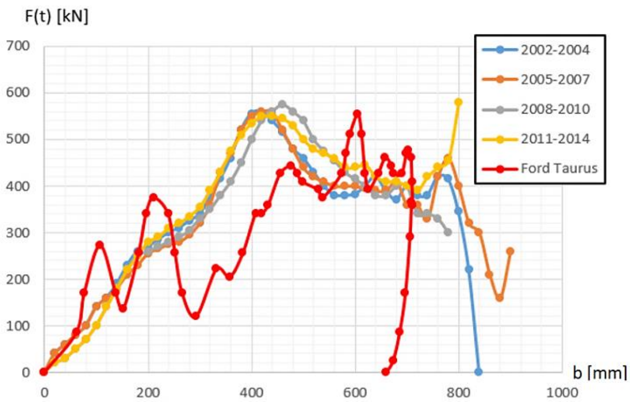

Fig. 10. Comparison of dynamic (D) stiffness (impact velocity 56 km/h) of Ford Taurus and experimental stiffness of series personal cars produced in years 2002–2014 according to Ref. [37]

It is important to note, that the static stiffness is a little bit large compared to dynamic stiffness, which is a known phenomenon described by Wiacek et al. [37]. In consequence, deceleration calculated by the use of static characteristics describes possibly the worst situation. Hence the assessment based on such type of approach is much more conservative.

The relation's shape and maximum force value on the discussed plot are comparable to those mentioned in Ref. [37] (see Fig. 10).

4.CONCLUSIONS

The conducted analytical modelling of the process of the frontal collision of a car with a rigid obstacle allows concluding that the quasistatic characteristic of the body stiffness is a sufficient tool to describe the phenomena occurring during dynamic loading, in the basic scope. It gives results comparable to the experimental ones (Fig. 8), which takes place when we use variable static stiffness in the modeling. Carrying out an analogous line of reasoning for the quasistatic averaged stiffness leads to a difference in the range of instantaneous deceleration values, with the results obtained for the actual stiffness (Fig. 7), although on the other hand, it does not cause a significant change in the length of the crush zone and the duration of the collision.

The lower values of dynamic stiffness, compared to the static one, can be justified by the immediate degradation of some elements of the body during dynamic loading.

Although the effects of a collision cannot be predicted with accuracy, the use of the phenomenological concept of the analytical description of this phenomenon allows us to satisfactorily determine the maximum value of the instantaneous delay, which is critical due to possible injuries to passengers (overloading effect).

The influence of the difference between vehicle body static and dynamic stiffness on the values of deceleration during the crash event is still not sufficiently presented, therefore this stage is selected for further investigations.

The presented way of approach can be used for any other car. However, knowledge about car body stiffness, is strictly recommended. The discussed Ford Taurus is only an example, illustrating the proposed method.

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