VEHICLE SLALOM PASSAGE ANALYSIS

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Abstract. The paper is a continuation of the authors' research dealing with a sensitivity study of the wheel camber variation of the experimental vehicle Alfa Romeo 156. A numerical comparison of the wheel camber change versus vertical axle position has already been performed and published for the vehicle in question. The same methodology was applied to four other selected axle types (for the purpose of comparison with each other), represented by vehicles in which these axles are actually implemented. As a result of the research, the values of the camber and the forces acting on the wheels during the simulated cornering of the vehicle were determined. Subsequently, the influence of the kinematic parameters on the dynamic response of the vehicle was evaluated. The current work of the authors focuses on experimental runs performed on the real vehicle (Alfa Romeo 156). Accordingly, the data from the experimental runs are compared with a numerical simulation of the passage of the modelled vehicle. The aim of this research is to evaluate the magnitude of body roll and body pitch for the vehicle slalom manoeuvre. The results indicate an excellent correlation between the numerical results and the measured values for this manoeuvre. On the basis of these results, it can be concluded that the numerical simulations are predictive, a finding that is crucial for developers pursuing equivalent kinematic parameters. In addition, the predictive nature of the simulation program allows to simulate diverse manoeuvres (e.g. vehicle braking) with variable input parameters, which would be problematic to implement in the case of experimental runs. Reducing the development time and tuning time of a prototype vehicle is one of the key factors to the competitiveness of companies.

Keywords: slalom traversing, numerical simulation, stability, drive, body roll.

Introduction

In most cases, students get to handle extreme situations in driving schools only from a theoretical point of view. It is essential to remember that a car weighing more than a thousand kilograms and travelling at a non-zero (not insignificant) speed is purely in contact with the road at four points. Moreover, these four points are not greater than a human palm. Therefore, it is advisable to drive the car in a way that corresponds to the optimal behaviour of these four points [1-3]. The only point of contact of a vehicle with the substrate is through its tires, which determine whether a vehicle can continue in its current direction or halt before a barrier [4]. Road transport safety is significantly impacted by the technical state of the vehicle's safety systems [5; 6].

Nevertheless, sometimes the driver is confronted with a situation where the vehicle needs to be driven more dynamically. These situations are unpredictable with a probability bordering on certainty (for example, a child or an animal ran onto the road) [7; 8]. Accordingly, the car needs to perform a driving manoeuvre based on a test method – the so-called moose test (avoiding animals weighing up to 600 kg by means of a sharp avoidance manoeuvre is crucial). This method is mainly used by inhabitants of Sweden. However, the term moose test was made famous in 1997 by the affair with the launch of the Mercedes-Benz A-Class by the German carmaker. The editorial team of the illustrious Swedish motor magazine Teknikens Värld (World of Technology) borrowed the new Mercedes for the test. In the course of driving, the wheel rims touched the road and the overturning of the vehicle inevitably followed. At the carmaker's headquarters, the test results were analysed in detail, inasmuch as the test drivers covered several million kilometres with the new car; a number of test drives were devoted to slalom with cones. However, under no circumstances were they able to overturn the car, despite the gradual loading of the roof. The carmaker had to invest 250 million euros in redesign and another 1.5 billion in development [9-11].

Therefore, the authors of this paper attend to analyse the vehicle slalom passage and perform a comparison of the experimental results with the results of the numerical analysis in order to demonstrate the predictability of the program. The objective of the slalom passage will be to observe and compare the vehicle body roll when passing between the cones at a safe speed, in other words, in order that the vehicle is not in danger of slipping. The safe speed will be derived based on the release method and the creating of formulae respecting the slip boundary condition, i.e. larger frictional forces. Subsequently, the experimental passage of the vehicle will be compared with the passage built in MotionView. The

simulated passage was based on the standardised test (Fig. 1). This test was designed by the Spanish car magazine KM77. This discipline tests the driving characteristics of vehicles at a speed of 77 km.h^{-1}, while its chief task is to check the stability and behaviour of the chassis of the tested vehicle [12;13]. In the interest of safety and to prevent damage to the experimental vehicle (Alfa Romeo 156), the maximum speed at which there is still no risk of slipping will be determined analytically [14;15].

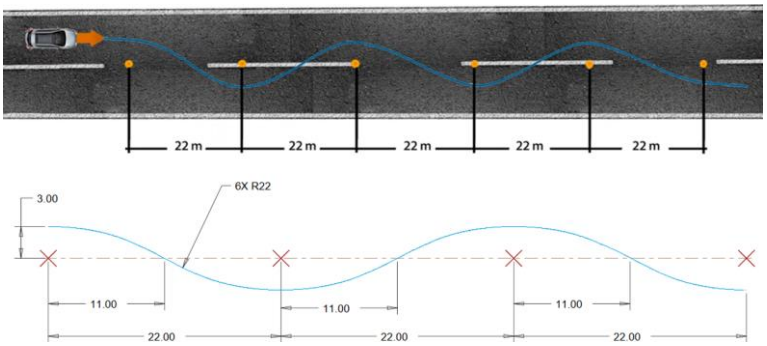

Fig. 1. **Slalom trajectory curve of the tested vehicle**

Within the limits of the test track, a real vehicle will be driven through a curve with radius R = 17 m. The turn radius was determined from a curve representing the vehicle's path using Creo Parametric 4.0. The actual vehicle path is depicted in Fig. 1.

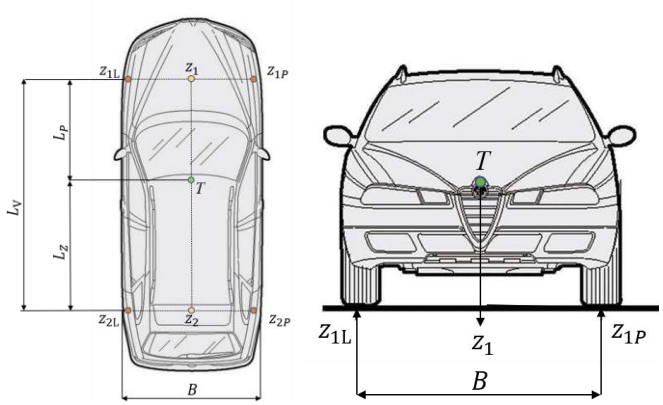

Fig. 2. **Wheel forces acting on the vehicle – top view (left) and axle load (right)**

The friction coefficient f = 0.9 (dry asphalt) [16] and the gravitational acceleration g = 9.81 m·s^{-2} were considered in the calculations. Fig. 2 shows the individual wheel loads and the dimensions required for their calculation.

Materials and methods

Prior to traversing a slalom course by the real vehicle, a calculation was made of the vehicle speed at which the vehicle would analytically slip. The vehicle has a wheel track B = 1.52 m, wheelbase L_v = 2.59 m and mass m = 1 466 kg. The distance of the centre of gravity of the vehicle from the front axle, L_p = 0.945 m, was determined by means of MotionView software based on the simulated vehicle. For the sake of calculation of the load on the rear axle (Fig. 3), it was first necessary to develop a moment equilibrium equation to the front axle (1), from which the load was expressed (2). By substituting the vehicle parameters, the rear axle load Z_2 = 5 247.29 N can be determined. On the basis of this calculation, it was subsequently feasible to calculate the load for the front axle Z_1 = 9 134.17 N, which was expressed from the equation of the forces acting in the Y direction (3), (4):

$$\sum M_{iZ1} = 0: \; -G_v \cdot L_p + Z_2 \cdot (L_p + L_z) = 0, \tag{1}$$

$$Z_2 = \frac{G_v \cdot L_p}{(L_p + L_z)}, \tag{2}$$

$$\sum F_{iy} = 0: \; Z_1 - G_v + Z_2 = 0, \tag{3}$$

$$Z_1 = G_v - Z_2, \tag{4}$$

where G_v – weight of the vehicle, N;
L_p – distance of the centre of gravity of the vehicle from the front axle, m;
L_z – distance of the centre of gravity of the vehicle from the rear axle, m;
Z_1 – front axle load, N;
Z_2 – rear axle load, N.

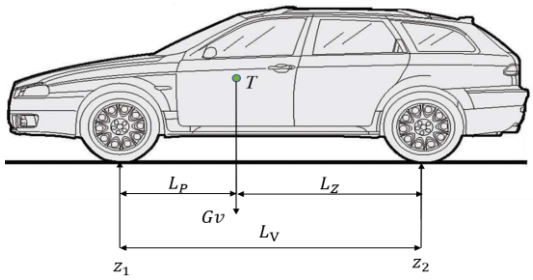

Fig. 3. **Wheel forces acting on the vehicle – side view**

From the equation of moment equilibrium to the left front wheel (5), the load on the right front wheel (6) can be determined. The calculated value of the load $Z_{1L} = Z_{1P} = 4\,565.09$ N is identical for both wheels, inasmuch as the centre of gravity of the vehicle is located at its centre (Fig. 2, right):

$$\sum M_{iZ\,2L} = 0: \; Z_{1P} \cdot B - Z_1 \cdot \frac{B}{2} = 0, \tag{5}$$

$$Z_{1P} = Z_1 \cdot \frac{1}{2}. \tag{6}$$

Moreover, the same calculation described by formulae (7) and (8) is applied to the rear axle, after substituting values $Z_{2L} = Z_{2P} = 2\,623.65$ N is obtained:

$$\sum M_{iZ\,2L} = 0: \; Z_{2P} \cdot B - Z_2 \cdot \frac{B}{2} = 0, \tag{7}$$

$$Z_{2P} = Z_2 \cdot \frac{1}{2}. \tag{8}$$

The wheel force amounts to the normal force acting on the contact patch of the wheel (Fig. 4, left). It is necessary to multiply the normal force by the friction coefficient f in order to calculate the friction force on the wheels. After substituting the normal forces acting on the wheels into formula (9), a frictional force of $F_T = 12\,946$ N was determined.

$$F_T = (F_{N1L} + F_{N1P} + F_{N2L} + F_{N2P}) \cdot f, \tag{9}$$

where F_T – frictional force, N;
F_{N1L} – normal force acting on the front left wheel, N;
F_{N2L} – normal force acting on the rear left wheel, N;
F_{N1P} – normal force acting on the front right wheel, N;
F_{N2P} – normal force acting on the rear right wheel, N;
f – friction coefficient.

Because of a smooth slalom passage of the vehicle without slipping, the frictional force F_T must be greater than the centrifugal force F_{Ce} (10), (11). Ultimately, the maximum vehicle speed in slalom traversing v is obtained (Fig. 4, right) by dint of expressing formula (12) and substituting the crucial values.

$$F_T \geq F_{Ce}. \tag{10}$$

$$F_T \geq \frac{m \cdot v^2}{R}. \tag{11}$$

$$v = \sqrt{\frac{F_T \cdot R}{m}}. \quad (12)$$

The calculations considered the slalom traversing of a perfectly rigid body without the following properties:

- a single contact point at the contact patch between the tyre and the road was considered;
- the stiffness of the car body suspension has not been considered;
- tire stiffness has not been considered.

On the basis of acquired knowledge, a real slalom traversing was performed at speeds of 40 and 50 km·h^{-1} during which body roll of the vehicle was observed.

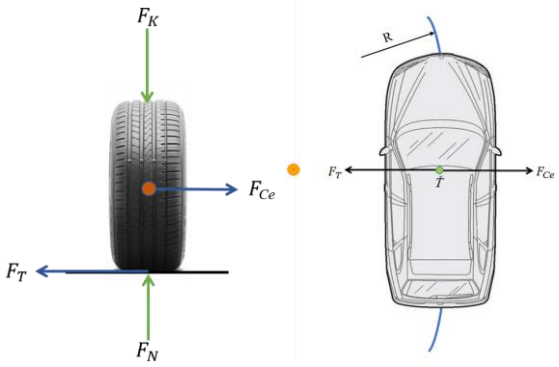

Fig. 4. **Considered forces acting on the wheel (left) and forces acting on the vehicle during slalom traversing (right)**

For creating the simulation, it was necessary to create a text file in the program. The file contained the trajectory of the vehicle with utilisation of spatial coordinates. This corresponded most closely to the real traversing. After entering all indispensable parameters, the program will be able to perform a simulation, where the body roll (output) could be monitored.

Results and discussion

The intended simulation is shown in Fig. 5. In order to measure the body roll angle during a real passage (slalom traversing), it was necessary to create a sequence of photos from the videos at selected vehicle positions. The photos (Fig. 6) were subsequently imported into Creo Parametric 4.0. In this software, the angle subtended by a straight line passing through the contact points of the tires with the road and a straight line passing through the center of a pair of vehicle distinctive points (fog lights) was measured. This angle represented the body roll angle.

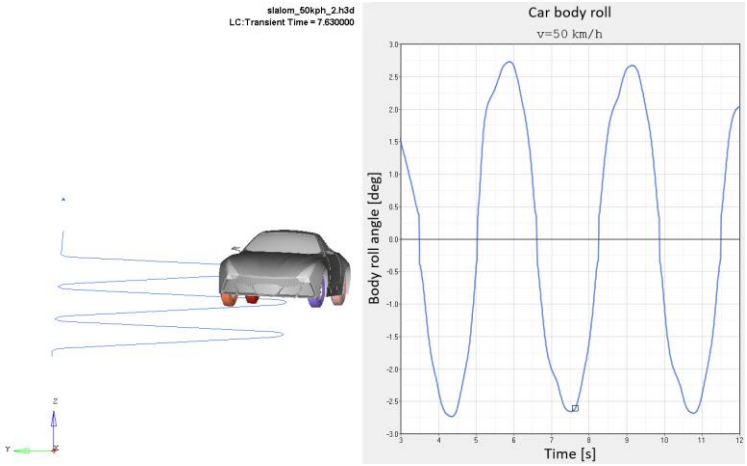

Fig. 5. **Vehicle body roll analysis in MotionView software**

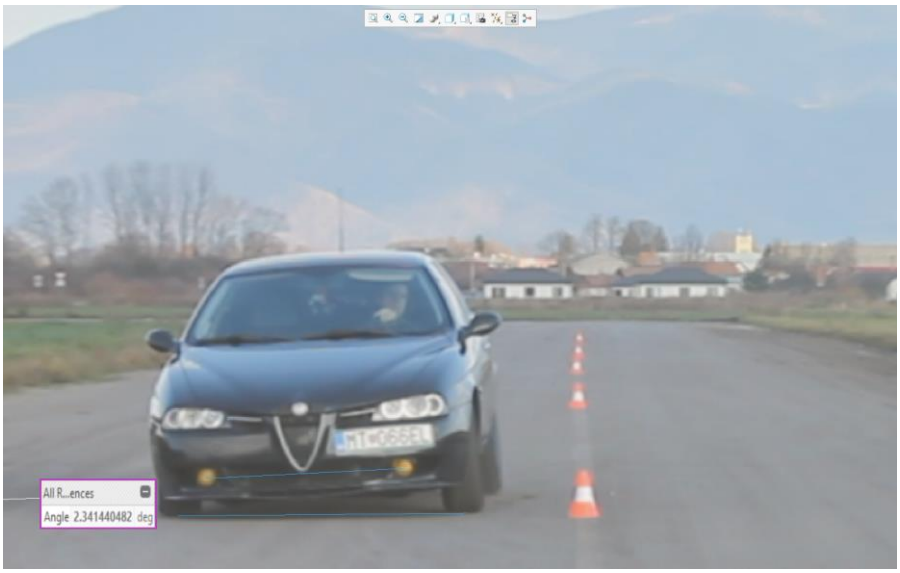

Fig. 6. **Vehicle body roll angle measurement in Creo Parametric 4.0 computer program**

Fig. 6 shows that the vehicle body roll angle was approximately 2.5°. This value appears to be commensurable in the comparison of conventional vehicles. Fig. 7 contrasts the simulation with the real slalom traversing of the vehicle.

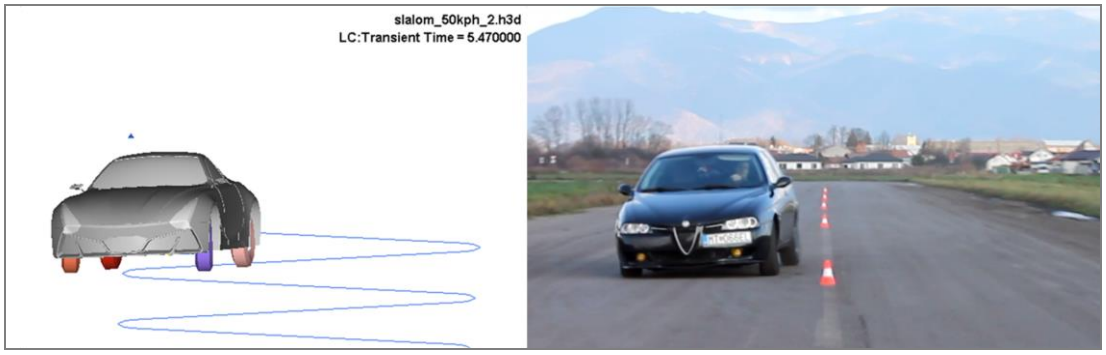

Fig. 7. **Comparison of simulation with real vehicle passage**

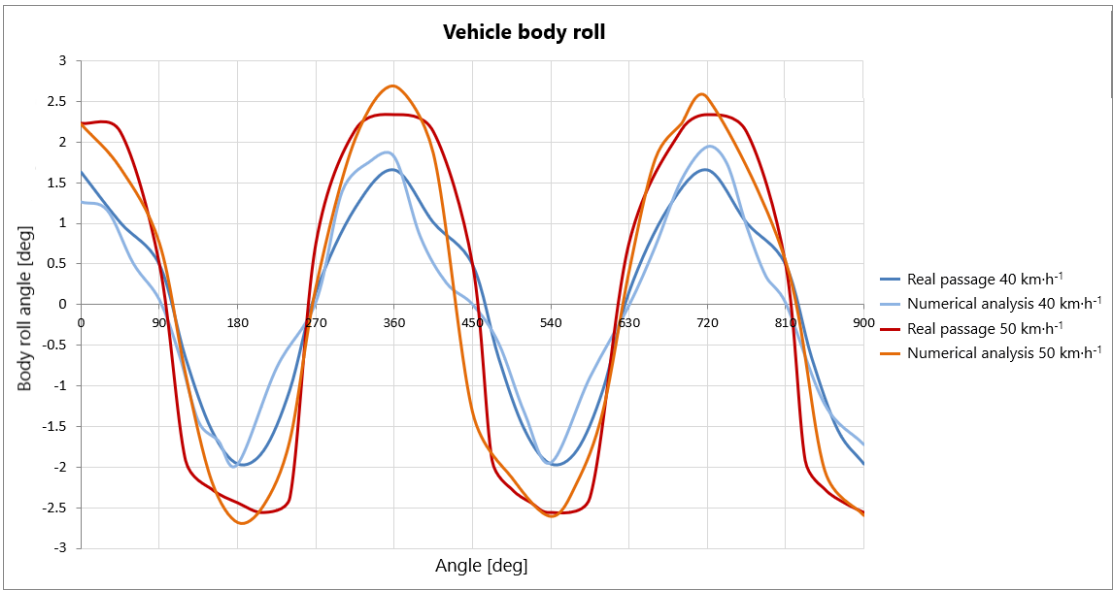

Fig. 8. **Graph of body roll depending on vehicle position**

Graphical dependencies of the body roll angle versus time were created from the measured values, either from MotionView or from videos of real driving. For the sake of a better comparison of the graphs

of two different speeds (40 km·h^{-1} and 50 km·h^{-1}), the time axis was replaced by angular coordinates. The graph evidently shows (Fig. 8) that the values measured using Creo Parametric are extremely close to the values from the MBD simulation. Ultimately, it can be concluded that these numerical simulations are predictive (essential information for development engineers). The average deviation for the 40 km·h^{-1} speed of slalom traversing was 0.24 ° and for the 50 km·h^{-1} slalom traversing the deviation was 0.36 °.

In terms of slalom manoeuvres, the Evasive Manoeuvre Assist (EMA) function can be suitably employed in the case of under-reactive and over-reactive drivers. The real-time vehicle testing of EMA function utilization demonstrates that EMA can be adapted to a defined set of driver reactions. It aims to provide the effective torque assistance or resistance intervention with adequate robustness [7;8]. From the point of view of improving the vehicle stability a method of determining the three-dimensional stability region of 'lateral speed–yaw rate–roll angle' can be employed. The results of this method demonstrated that the control strategy considering the body roll angle under disparate working conditions can enhance the vehicle stability [17]. The body roll angle of the vehicle is closely linked to the hazardous rollover phenomenon. If the speed or the distance between the center of gravity and the roll axis increases, respectively, the roll angle will rise. Moreover, if the steering angle is more significant, the body roll angle will be greater [18;19].

Conclusions

The main objective of this work was to carry out simulations and experimental runs on a real vehicle and to compare the evaluated kinematic parameters with the obtained numerical simulation of the vehicle. The authors scrutinised the body roll in the transverse and longitudinal directions for the slalom driving manoeuvre. The results of the analysis demonstrated a favourable correlation of the values with the experiment. Accordingly, it can be concluded that the numerical simulations of the issue in question are predictive. This insight is indispensable for the development engineers in the course of observing the equivalent kinematic parameters. The predictive nature of the program will enable in the future to simulate vehicle braking with different (but controllable) deceleration input values. This step would be arduous to implement in the case of real braking.

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Author contributions

Conceptualization, M.B., D.M, J.D., V.I.; methodology, M.B. and J.D.; software, J.D., D.M.; validation, M.B. and J.D; formal analysis, M.B and D.M.; investigation, V.I., M.B., D.M. and J.D.; data curation, V.I., D.M. an J.D.; writing—original draft preparation, M.B.; writing – review and editing, D.M. and J.D.; visualization, D.M. and V.I.; project administration, J.D. and M.B.; funding acquisition, J.D. All authors have read and agreed to the published version of the manuscript.

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