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Identification of the ESP sensors condition during the vehicle service life

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Highlights

- Extension of periodic tests of the ESP selected sensors is proposed.
- The idea assumes using a universal diagnostics tester and a wheel play detector unit.
- The flat model of a vehicle placed on a wheel play detector unit is presented.
- The results of simulation are compared with the ESP sensor signal.

Abstract

The paper presents the proposals of extension of the periodic tests of the selected ESP system sensors: angular velocity sensor and lateral acceleration sensor using a universal diagnostics tester and a plate stand (a wheel play detector unit). The idea of this approach is to evaluate the signals from the above sensors in terms of their amplitude and frequency in the case of known forcing at the plate stand. Knowledge of the amplitude and frequency of the plates excitation and the model of tested vehicle allows for predicting the response of vehicle. On this way the verification of sensors indications is possible. This article presents the flat model of a vehicle placed on the plate stand, simulation tests and the results of its validation for three different vehicles. The results of the investigation show that the wheelbase of vehicle has a significant impact on the steady-state vibration amplitude. This conclusion is important in the practical application of this method to test the vehicle yaw rate sensor in the ESP system.

Keywords

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ESP sensors diagnostics, yaw rate sensor, testing of the mechatronic safety systems, integrated diagnostic.

1. Introduction

Modern motor vehicles are equipped with a number of systems responsible for reducing the likelihood of an undesirable road incident (e.g. collision). One of the most important and the most intensively developed by automotive engineers is Electronic Stability Program (ESP). The statistical research shows that ESP system can decrease the number of crash situations, associated with defensive maneuvers even about 8% [13]. The effectiveness of the track stabilization system increases by the recently developing integrated systems combining ESP and AFS (Active Front Steering) [3] or ESP and TVD (Torque Vectoring Differential) [9]. Furthermore, the concepts of using ESP for diagnosing automotive damper defects appears in the literature [19]. Active car safety is particularly dependent on the correct operation of systems that affect the operation of the braking system and the stabilization of the drive track. Currently, these tasks are included in the scope of duties of the mechatronic ESP system, whose operation is based on the analysis of signals from sensors located in the vehicle, which include among others: wheel speed sensors, yaw rate (vehicle angular velocity) sensor and lateral acceleration sensor, steering angle sensor. Assessment of the efficiency of these sensors during the vehicle service life is therefore important from the point of view of road safety. Thus, in last years there are papers deal with sensors diagnosis and estimation their bias under normal driving conditions [16,

17, 18]. Considering the fact that the role of mechatronic systems in vehicles is growing very rapidly, it seems natural to state that periodic testing of vehicles should carefully take into account the control of these elements. Now the tests of these systems consist on verifying whether the on-board diagnostics system informs a possible malfunction via the MIL lamp. This supervision concerns the efficiency of electrical and electronic systems. However, the condition of sensors during lifetime may also change in the mechanical field. Therefore, the control of the operation of the system as a whole is recommended especially in vehicles with extended service life and crashed.

The research and development works conducted in the field of extending vehicle inspection tests take into account the fact that vehicle assemblies have become mechatronic systems. Their operation depends both on the efficiency of the mechanical part of these systems, tested on stationary stands, and on the efficiency of sensors and actuators. New test methods should take into account the need to test these elements. The proposed solutions in the field of safety system control include the use of computer testers [2] and external measurement tools for periodic tests at the PTI (Periodic Technical Inspection) [11]. The basic problem related to the direct application of diagnostic stands is the difficulty in obtaining data from vehicle controllers, sensors and actuators. This is due to the need to interfere with the car's electrical system and driver software. In addition, car manufacturers

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do not provide information on both the location of the sensors and their characteristics (scale factors). Therefore, it becomes necessary to use specialized diagnostic testers connected to the vehicle's IT system via the OBD socket. This facilitates and speeds up the process of acquiring data from sensors. On the other hand, the modification of the testers software requires the proper sampling frequency of signals, because too low frequency hinders the qualitative assessment of the results. The above problems are currently being undertaken by research centers in the European Union [1, 2, 8]. It is proposed to modify the currently used diagnostic programs in order to standardize procedures, facilitate access to other systems and accelerate the performance of diagnostic tests [8]. The scope of obligatory control tests of active and passive safety systems is being developed as well as the requirements in this regard for diagnostic testers used at PTI stations [1, 2, 12]. This applies to testing brakes (roller stations) and engines (chassis braking) [4,5].

One of the proposals to extend the scope of periodic tests is to check the operation of the angular velocity and lateral acceleration sensors (usually built in an integrated form) of the ESP system. The determination of vehicle angular velocity is based on the Coriolis effect acting on sensor's vibrating element [10]. The lateral acceleration is calculated on the basis of an electric signal proportional to the mass displacement in the sensor [10]. Internal elements of the sensor are subject to forced vibrations. Its characteristics may change during the period of use, e.g. due to loosening of the fastening, overload during a collision of the vehicle, repairs of the body. These circumstances justify the need to test the sensors during the vehicle service life.

The article presents the method of testing the operation of the sensors of angular velocity and lateral acceleration of the ESP system (usually built in an integrated form) in bench conditions. A method of forcing a vehicle rotation on a plate stand was proposed, with the use of devices used so far in periodic technical tests. The idea of the proposed method is to evaluate the signals from the above sensors in terms of their amplitude and frequency, with the known rotation of the vehicle being forced on a plate stand for checking the looseness in the suspension (a wheel play detector unit). The course of the signals from the sensors is monitored in real time with a diagnostic tester and evaluated after the test.

The aim of the study was to show that the knowledge of the amplitude and frequency of the excitation of the plate movement, i.e. the excitation acting on the vehicle and appropriate model of the vehicle allows predicting the vehicle's response and verifying the values measured by the sensors. This article presents the flat model of a vehicle with the front wheels placed on the diagnostic stand with two coaxially moving plates and the results of its validation for three different vehicles.

The hypothesis that needed to be proved is as follows: a flat model of the vehicle, taking into account the characteristics of the tires, subjected to excitation from the station's plates with a known amplitude and excitation frequency can be used to determine body vibrations on the plate stand and to assess the correctness of the ESP rotational speed sensor indications.

The proposed test method is a new solution, not used so far [15]. It is applicable in Periodic Technical Inspections.

2. Mathematical model

The developed model adopts a flat model of the vehicle whose body rotates relative to the instantaneous center of rotation. This is illustrated schematically in Figure 1. The assumption of the vehicle's rotational movement on the plate stand is justified because:

- the plates are located only under the front axle wheels,
- the rear axle wheels are free to roll during the measurement,
- diagnostic plates reciprocate movement, whose amplitude is small (up to 100 mm) relative to the distance of the plates from the center of rotation of the vehicle body.

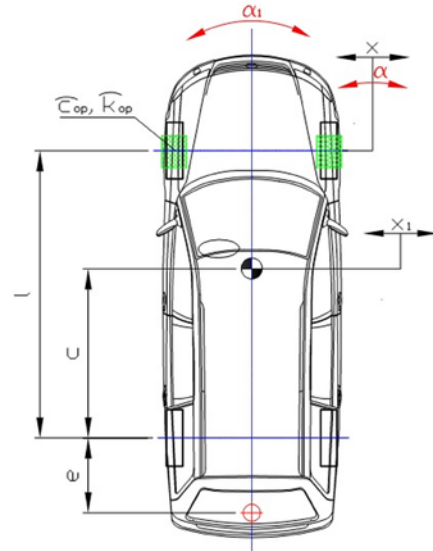


Fig. 1. Flat model of the vehicle at the plate bench (diagnostic plates under the front axle wheels), c - distance between the center of mass and the rear axle, e - distance between the center of rotation and the rear axle, l - wheelbase

The hydraulically driven diagnostic plates reciprocate in a direction perpendicular to the longitudinal axis of the vehicle. Under ideal conditions, the x coordinate associated with the plate can be described by the equation:

$$x = A \sin(\omega t) \quad (1)$$

where:

A - one-sided amplitude of the plate,

ω - circular frequency of plate movement ($\omega = 2\pi f$, f - frequency).

Considering the fact that the amplitude of the plate movement is small in relation to their distance from the instantaneous center of rotation of the vehicle and the wheels on the plates are unbraked, the model uses the angular coordinate α describing the kinematic forcing acting from the plates on the wheels as:

$$a = \frac{x}{l+e} = \frac{A}{l+e} \sin(\omega t), \quad (2)$$

where:

x - displacement of plates,

l - wheelbase of the vehicle,

e - distance between the center of rotation of the body and the rear axle (e - in front of or behind the rear axle).

The angular velocity of excitation will be then:

$$\dot{\alpha} = \frac{\dot{x}}{l+e} = \frac{A\omega}{l+e} \cos(\omega t). \quad (3)$$

It is kinematic forcing on the vehicle's wheels. The movement of the plates is transferred to the body through flexible tires and suspension. Considering the above assumptions, the equation of the vehicle body movement will take the form:

$$I\ddot{\alpha}_1 + \hat{k}_{op}(\dot{\alpha}_1 - \dot{\alpha}) + \hat{c}_{op}(\alpha_1 - \alpha) = 0 \quad (4)$$

where:

I - moment of inertia of the vehicle relative to the instantaneous center of rotation,

α_1 - angular coordinate associated with the vehicle body,

\hat{k}_{op} - equivalent coefficient of lateral damping of tire for rotary motion,
 \hat{c}_{op} - equivalent coefficient of lateral stiffness of tire for rotary motion.

The analyzed model takes into account the stiffness and damping of the front axle tires (the impact of rear axle tires was omitted). Considering the direction of loading resulting from the plate forcing, the flexibility of suspension elements was omitted. The subject literature provides information on the damping coefficients and lateral stiffness of a tyre for linear motion [6, 7]. Due to the fact that rotational motion is considered in this analysis, the following approximate relationships have been adopted for the above coefficients:

$$\hat{k}_{op} = 2k_{op}(l + e)^2, \quad (5)$$

$$\hat{c}_{op} = 2c_{op}(l + e)^2, \quad (6)$$

where:

k_{op} - tyre lateral damping coefficient,

c_{op} - tyre lateral stiffness coefficient.

They result from the following relationships for forces and moments from the elasticity and damping of tires:

$$\begin{aligned} F_{cop} &= c_p \cdot (l + e) \cdot (\alpha_1 - \alpha) & M_{cop} &= c_p \cdot (l + e) \cdot (\alpha_1 - \alpha) \cdot (l + e) = \hat{c}_p \cdot (\alpha_1 - \alpha) \\ F_{kop} &= k_p \cdot (l + e) \cdot (\alpha_1 - \alpha) & M_{kop} &= k_p \cdot (l + e) \cdot (\alpha_1 - \alpha) \cdot (l + e) = \hat{k}_p \cdot (\alpha_1 - \alpha) \end{aligned} \quad (7)$$

The constant '2' in equations (5) and (6) results from the fact that the two wheels of the front axle are treated as a parallel combination of elastic and damping elements.

The moment of inertia I relative to the instantaneous center of rotation was determined on the basis of Steiner's theorem:

$$I = I_o + m(c + e)^2, \quad (8)$$

where:

I_o - moment of inertia about the vertical axis passing through the vehicle's center of mass,

m - vehicle mass,

c - as in Fig. 1.

The solution of equation (4) is angle α_1 , i.e. the angular coordinate associated with the vehicle body as a function of time. On its basis, the angular velocity (yaw rate) usually designated in the literature as Ψ will be:

$$\psi = \dot{\alpha}_1. \quad (9)$$

The lateral acceleration a_y of the center of mass can be written as:

$$a_y - \ddot{x}_1 = \ddot{\alpha}_1(c + e). \quad (10)$$

3. Parameters of test vehicles with particular emphasis on tyre characteristics

Simulation analyzes and validation of the developed model were carried out for three passenger cars with different inertial and geometric parameters. Table 1 contains a list of values significant from the model's point of view.

The lateral stiffness of the tyre was determined on a special stand for testing tyres under static conditions (Fig. 2). The movable plate under the rigidly mounted wheel can be moved perpendicular to the wheel disk.

Table 1. Selected mass and geometrical parameters of the tested cars (size symbols in accordance with the markings in the text)

	Fiat Panda II	Opel Astra G	Renault Kadjar
l [m]	2,30	2,61	2,60
c [m]	1,14	1,56	1,3
m [kg]	1050	1165	1545
I_o [kgm ²]	1085	1586	2260

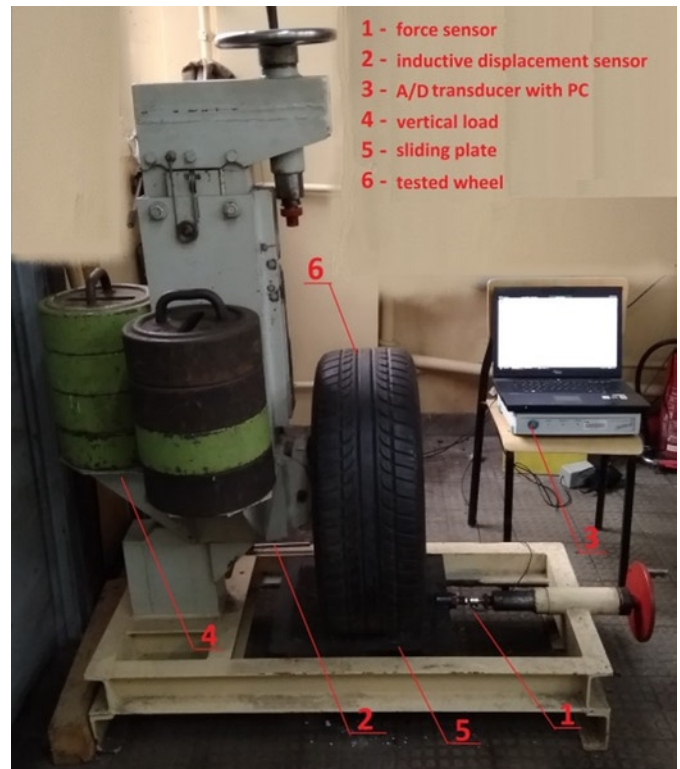


Fig. 2. Stand for determining the transverse stiffness of tyre

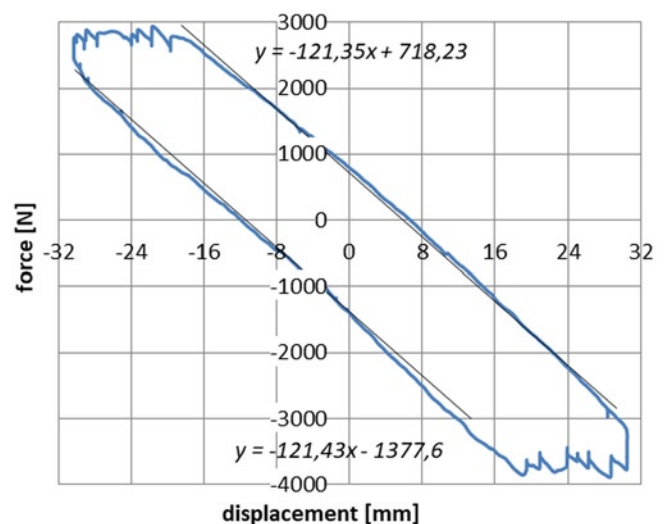


Fig. 3. Transverse stiffness characteristics of 205/55R16 (2.2 bar) tyre

This motion is carried out through a screw mechanism connected to the load plate by a force sensor. An inductive sensor is used to measure displacement. Data is saved to the hard disk via an A/D converter. The construction of the stand allows applying any vertical load. The test was performed for the 205/55 R16 radial tyre with a pumping pressure of 2.2 bar. The specified vertical load was 3 kN, which cor-

responds to a typical normal reaction for a front wheel car. The characteristics of the lateral stiffness of the tyre obtained in this way are presented in Fig. 3. The visible hysteresis loop results, among others, from exceeding the limit force of adhesion between the wheel and the plate. Regardless of the return of the applied force, the linear nature of the relationship between force and deformation was recorded. This allowed easy determination of the lateral stiffness coefficient of the tested tyre, whose value was:

$$c_{op} = 121 \text{ N/mm.}$$

The presented stand does not allow for obtaining high plate speeds, which means that the possibilities of determining the transversal damping factor are very limited. The value of this parameter was taken from the literature. The papers [6] and [7] contain numerous simulations and studies on lateral dynamics of tires. According to these data, the value of the lateral damping coefficient for tires of the same size as the tested tyre, loaded with a normal force of 3600 N and with a pump pressure of 2.75 bar is:

$$k_{op} = 1770 \text{ kg/s.}$$

In turn, the lateral stiffness coefficient then takes the value of 126 N/mm. Bearing in mind similar (compared to the considered) tyre parameters and almost identical values of the stiffness coefficient, in the course of further calculations the above value of the coefficient k_{op} was adopted.

4. Simulation tests and model validation

Fig. 4 presents the influence of the position of the center of rotation on the body deflection speed for sinusoidal forcing. The position of the center of rotation is represented by the parameter e (Fig. 1). Negative values e correspond to the shift towards the vehicle's center of mass.

Shifting the center of rotation toward of the rear axle increases the amplitude of yaw angular velocity. For the center of rotation shifted by 1 m towards the center of mass, the amplitude increases almost twice (relative to the center of rotation on the rear axle). Shifting the center of rotation by the same value in the opposite direction results in a slight decrease in the amplitude from the initial value.

The position of the instantaneous center of rotation of the vehicle was verified during tests by measuring lateral linear accelerations at various points in the longitudinal axis of the vehicle. The measurements showed that it is located at the intersection of the longitudinal axis and the rear axle of the vehicle (with the accuracy of measurements made). Therefore, $e = 0$ was assumed in further analysis. Simulation tests and model verification were carried out for the results obtained on the test stand with a modified hydraulic control system [14]. The stand together with the vehicle prepared for testing is shown in Fig. 5. The stand control system allowed for changing the number of jerking cycles and the plate pitch.

The vehicle body was set in a vibrating motion through stand plates moving in the same phase. The pitch of the plates during the tests was 100 mm. Recording the position of the plate using an inductive sensor



Fig. 5. Modified test stand for checking the yaw rate and lateral acceleration sensors (Unimetal Złotów) together with the test vehicle

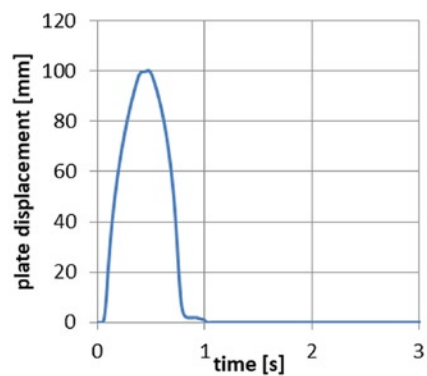


Fig. 6. Single forcing pulse

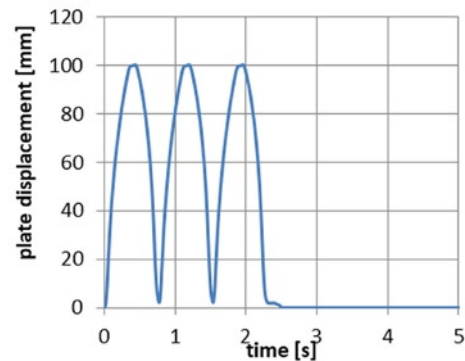


Fig. 7. Multiple forcing pulse

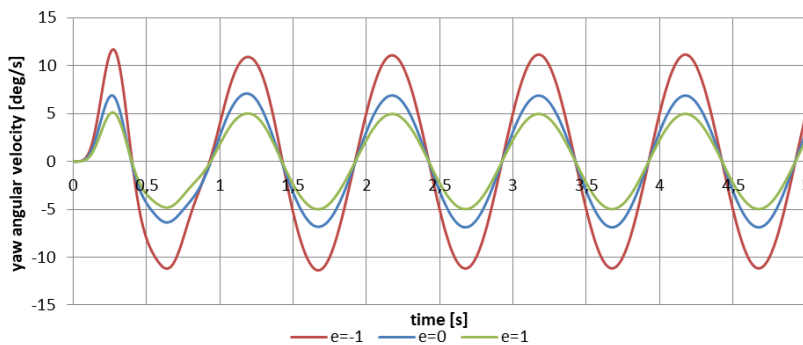


Fig. 4. Impact of the temporary center of rotation position on the body deflection speed for Opel Astra G - simulation according with dependence (1)

enabled the precise definition of the forcing function $x(t)$. Examples of single and multiple forcing impulses are illustrated in Figs. 6 and 7.

The obtained results $x(t)$ after differentiation were used to calculate the velocity $\dot{x}(t)$ and $\dot{\alpha}_1$ to solve the equation (4) and simulation.

Due to the introduction of real forcing velocity course into equation (4), calculations were made in the Matlab R2015b software. An integrated PIC DAQ triaxial gyroscope was used as a reference sensor for measuring the angular velocity of the body vibration ($\dot{\alpha}_1$), Fig. 9.

Figures 11-13 contain comparisons of angular body speeds ($\dot{\alpha}_1$) calculated according to the developed model and measured with a reference sensor for each of the

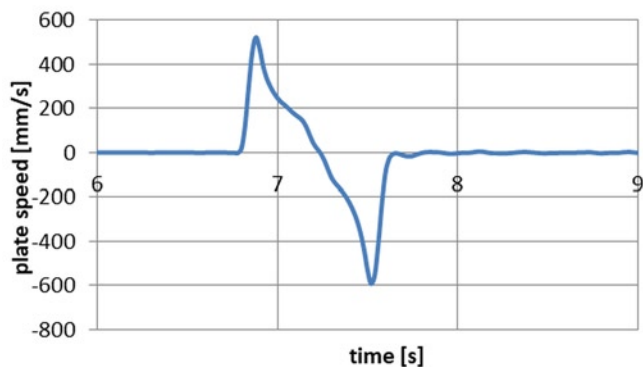


Fig. 8. Plate speed $\dot{x}(t)$

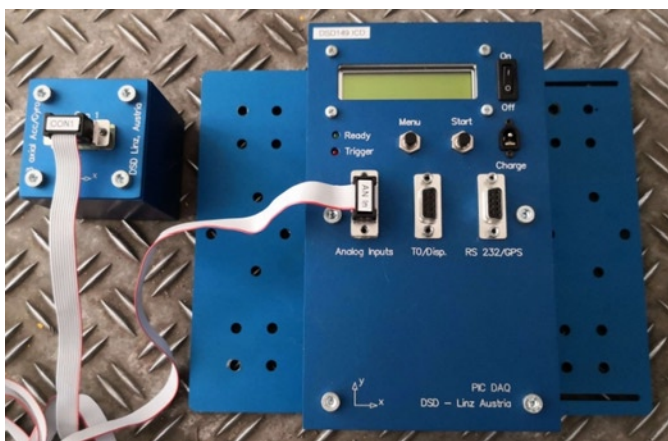


Fig. 9. A reference sensor together with a data recording system used to measure the angular velocity of the deflection

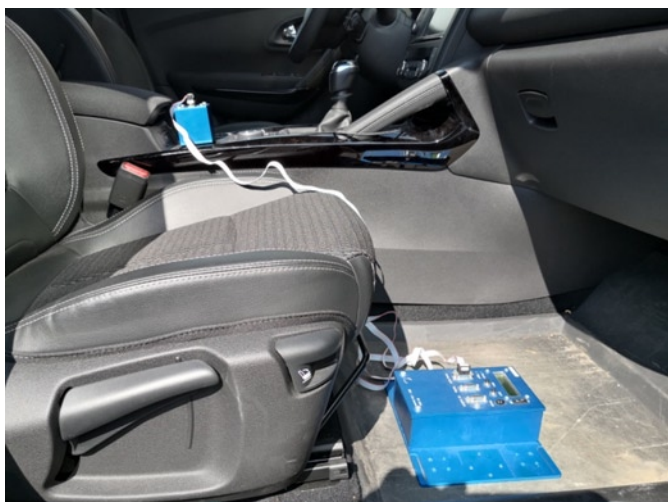


Fig. 10. Example of placing the reference sensor in the tested vehicle

research vehicles. The acting forcing was a single plate stroke (Fig. 6). For each car, satisfactory compliance of the amplitude and frequency of vibrations was obtained. The visible differences in the phase of decreasing angular velocity ($\dot{\alpha}_1$) result from the adopted flat vehicle model, which does not take into account the lateral slope of the body. The differences in the vibration damping phase are due to the fact that the body subjected to transverse tilting also performs minimal revolutions relative to the longitudinal axis. They result from different stiffness of the front and rear suspension of the vehicle. The sensor of the angular speed of the body used in the tests, as well as the sensor of the ESP system record these vibrations - Fig. 15. The flat model does not take it into account. Nevertheless, according to the authors, this model can be predestined for the applications referred to diagnostics tests,

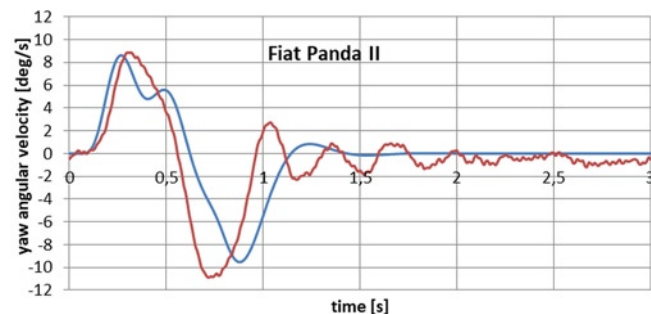


Fig. 11. Comparison of the angular velocity (yaw velocity) ($\dot{\alpha}_1$) of the body according to the model and according to the measurements with the reference sensor, enforcement by single impulse - Fiat Panda II; blue line – simulation, red line – measurement

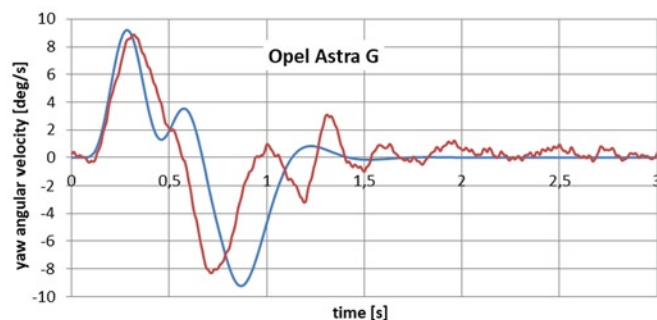


Fig. 12. Comparison of the angular velocity (yaw velocity) ($\dot{\alpha}_1$) of the body according to the model and according to the measurements with the reference sensor, enforcement by single impulse - Opel Astra G; blue line – simulation, red line – measurement

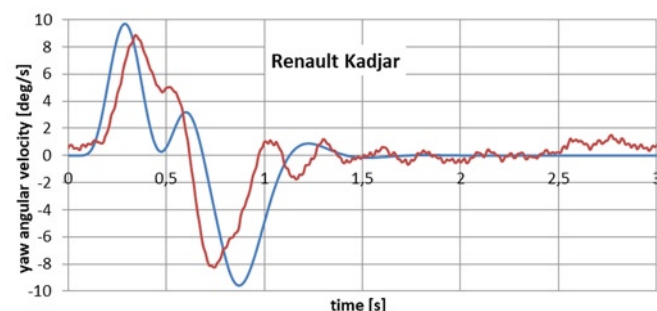


Fig. 13. Comparison of the angular velocity (yaw velocity) ($\dot{\alpha}_1$) of the body according to the model and according to the measurements with the reference sensor, enforcement by single impulse - Renault Kadjar; blue line – simulation, red line – measurement

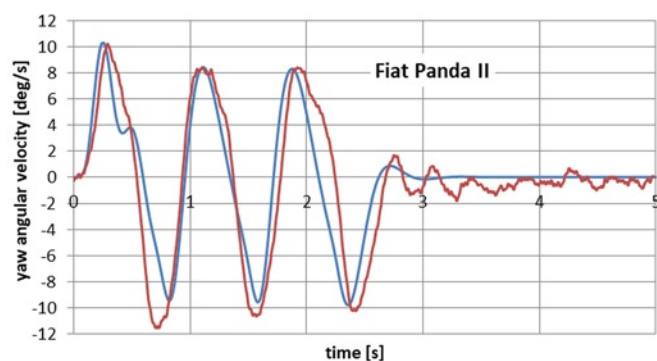


Fig. 14. Comparison of the yaw velocity of the body in simulation model to the measurement with the reference sensor for multiple impulse - Fiat Panda II; blue line – simulation, red line – measurement

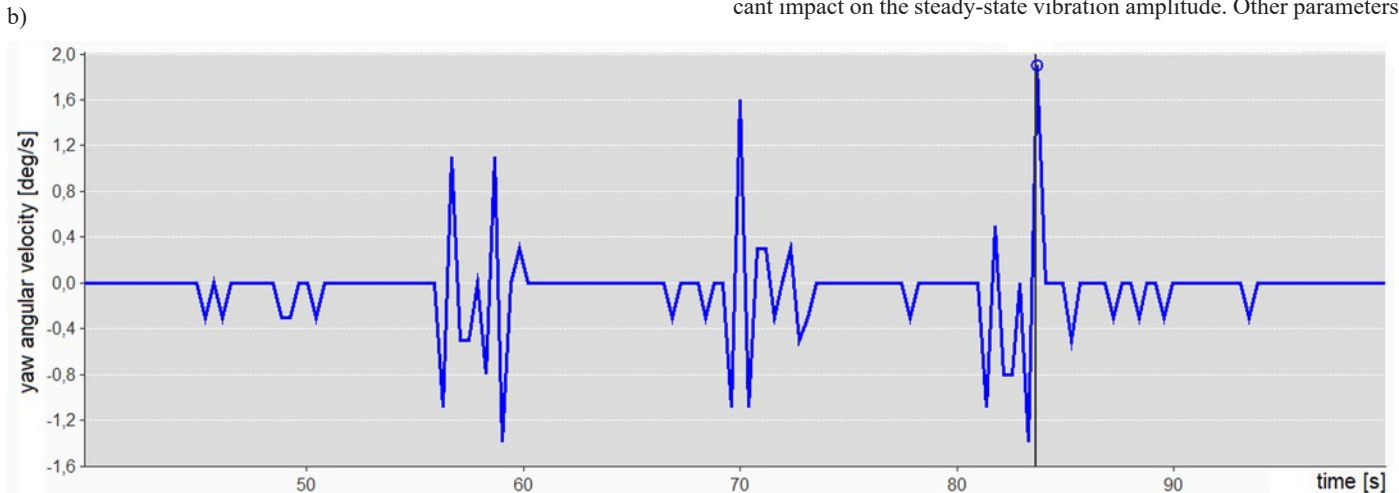
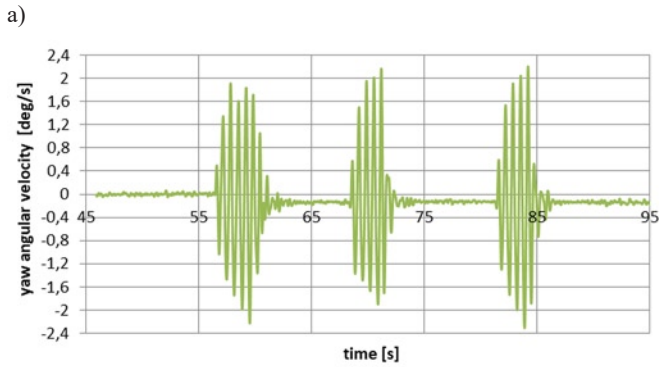


Fig. 15. Recording of angular velocity ($\dot{\alpha}_1$) during body vibrations (tilts) relative to the longitudinal axis of the vehicle: a) reference sensor, b) ESP sensor

5. Impact of selected vehicle parameters on simulation results

The following charts (Fig. 16-19) show the impact of changes in selected vehicle parameters on the angular velocity of the body ($\dot{\alpha}_1$). Simulations were made for the Opel Astra G. One parameter was changed in each case. The forcing were a five impulses for the vehicle body.

Tyre lateral stiffness affects the amplitude of transient vibration - Fig. 16. The lateral damping of the tyre affects the amplitude of transient vibrations and the time after which the determination of vibration amplitude occurs - Fig. 17.

Based on the Fig. 18, it can be seen that the wheelbase has a significant impact on the steady-state vibration amplitude. Other parameters

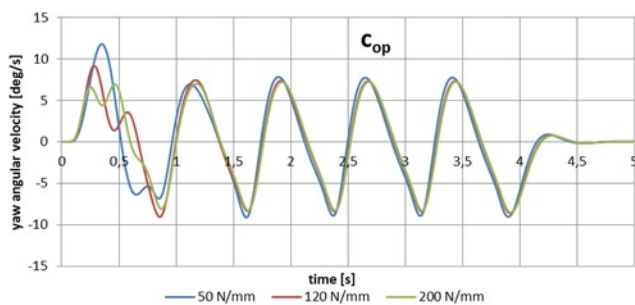


Fig. 16. Impact of tyre lateral stiffness on simulation results

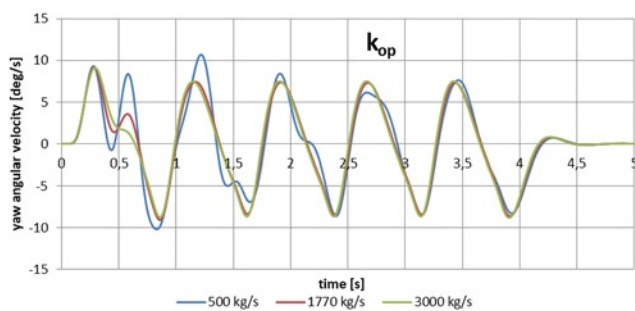


Fig. 17. Impact of lateral tyre damping on simulation results

especially taking into account the measurement accuracy of sensors used in ESP systems.

Figure 14 presents an analogous comparison to the above for one of the research vehicles, where the excitation was a triple impulse (Fig. 7). Also in this case, an acceptable correlation was observed between simulation and measurement, which confirms the usefulness of the proposed model.

are responsible for the amplitude of the transient vibrations and the time after which they are determined. Equation (4) describing the vibrational motion of a vehicle is a heterogeneous linear equation. The general integral of such an equation is the sum of the general integral of a homogeneous equation and the special integral of a non-homogeneous equation. The general integral of the homogeneous equation

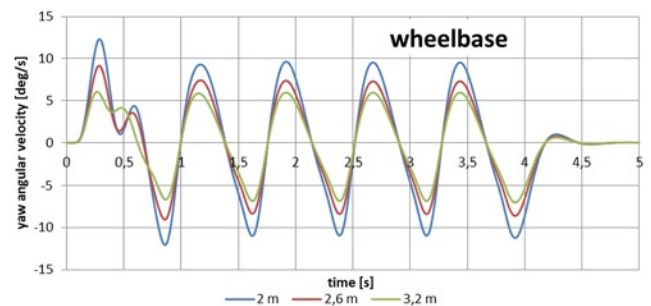


Fig. 18. Impact of vehicle wheelbase on simulation results

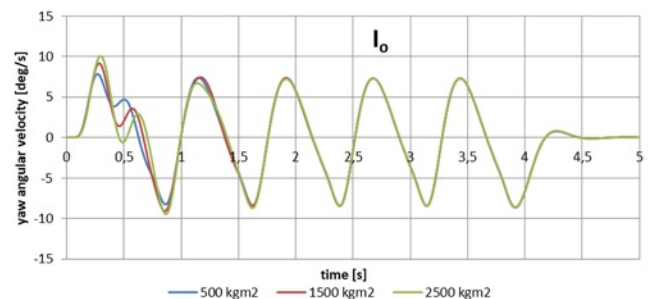


Fig. 19. Impact of the moment of inertia relative to the vertical axis of the vehicle on the simulation results

describes free (unforced) vibrations, so in the considered case their amplitude drops to 0 (due to tyre damping). As a result, specific body vibrations (α_j) will tend to introduce α from the movement of the plates (the model assumes that the movement of the plates x forces the body to rotate through the wheels of the vehicle). According to the relationship (3), the vehicle parameter affecting the angular excitation speed ($\dot{\alpha}_1$) is the wheelbase. Assuming that the center of rotation is on the rear axle, it can be concluded that the amplitude of the angular speed of the body (yaw rate) will be a function of the wheelbase and the parameters of the plate movement (amplitude and frequency). This conclusion is important in the practical application of this method to test the vehicle angular velocity sensor in the ESP system. It should be noted that with known plate motion parameters, to evaluate the sensor operation, it is sufficient to know the wheelbase of the vehicle. For lateral acceleration, the position of the vehicle's center of mass (parameter c in equation (10)) will also be relevant.

6. Conclusion

The analysis shows that the efficiency of the angular velocity sensor and the ESP lateral acceleration sensor can be assessed using a diagnostic test stand for suspension tests. The experimental tests confirmed the postulated hypothesis that the developed model of the vehicle movement on the plate stand can be used to determine the body vibrations. The results of the vehicle angular velocity obtained computationally on the basis of the model can be a reference for the operation evaluation of the yaw rate sensor and the lateral acceleration sensor.

The vibratory motion of the vehicle body in the solid phase can be calculated on the basis of knowledge of the plate movement (amplitude and frequency), vehicle wheelbase, position of the lateral acceleration sensor (to determine lateral acceleration) and the position of

the instantaneous center of rotation. The values of the amplitude and frequency of the vehicle body angular velocity and lateral acceleration obtained in this way should coincide with the values measured by the sensors of the ESP system.

The conducted tests have shown that the position of the instantaneous center of rotation of the body is on the rear axle, or in its immediate vicinity ($e \approx 0$, Fig. 1). This determination will allow the practical application of this method for diagnosing the signal from the vehicle angular velocity sensor and the lateral acceleration sensor during the vehicle service life.

The extension of the periodic tests scope of used vehicles on elements of the mechatronic systems (ABS, ESP), especially for vehicles with high mileage and repaired after accidents, has an impact on active safety in road traffic. The tests of the operation of these sensors in stand conditions allow for detecting mechanical malfunctions in the systems, e.g. loosening of sensor mounting, changes in their characteristics due to vibrations or weather conditions. These failures are not signalled by the on-board diagnostic system OBD but have an impact on the active safety systems of the vehicle. The costs reduction of such tests can be achieved by adapting stationary stands (a set of wheel play detectors) to cooperation with diagnostic testers.

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References

1. Buekenhoudt P. ECSS testing: concept and implementation of a wider interrogation of the electronic controlled safety system via OBD. CITA Conference, Dubai: 2015. [<https://citainsp.org/wp-content/uploads/2016/01/1.-Workshop-B2-Final-Presentation.pdf>].
2. CITA. ECSS: Study on a new performance test for electronic safety components at roadworthiness tests – Final Report: 2014. [https://www.researchgate.net/profile/Pascal-Buekenhoudt/publication/344654916_ECSS_-_Study_on_a_new_performance_test_for_electronic_safety_components_at_roadworthiness_tests_-_final_report/links/5f87078b458515b7cf7fc262/ECSS-Study-on-a-new-performance-test-for-electronic-safety-components-at-roadworthiness-tests-final-report.pdf].
3. Fan X, Zhao Z. Vehicle dynamics modeling and electronic stability program/active front steering sliding mode integrated control. Asian Journal of Control 2019; 21(5): 2364–2377, <https://doi.org/10.1002/asjc.1822>.
4. Gajek A. Directions for the development of periodic technical inspection for motor vehicles safety systems. The Archives of Automotive Engineering 2018; 80(2): 37-51, <https://doi.org/10.14669/AM.VOL80.ART3>.
5. Gajek A, Strzpek P, Dobaj K. Algorithms for diagnostics of the hydraulic pressure modulators of ABS/ESP systems in stand conditions. MATEC Web of Conferences 2018; 182: 1-9, <https://doi.org/10.1051/mateconf/201818201020>.
6. Hackl A, Hirshberg W, Lex C, Rill G. Experimental validation of a non-linear first-order tyre dynamics approach. The Dynamics of Vehicles on Roads and Tracks: 24th Symposium of the International Association for Vehicle System Dynamics 2016; 24: 443-452.
7. Hackl A, Hirshberg W, Lex C, Rill G. Tire dynamics: model validation and parameter identification. In Andreescu C, Clenci A. (eds): Proceedings of the European Automotive Congress EAEC-ESFA 2015. Springer, Cham: 2016: 219-232.
8. IDELSY. Initiative for Diagnosis of Electronic Systems in Motor Vehicles for Periodic Technical Inspection (PTI) - Final Report: 2006. [https://ec.europa.eu/transport/road_safety/sites/roadsafety/files/pdf/projects_sources/idelsy_management_summary.pdf].
9. Jaafari S, Shirazi K. Integrated Vehicle Dynamics Control Via Torque Vectoring Differential and Electronic Stability Control to Improve Vehicle Handling and Stability Performance. ASME Journal of Dynamic Systems, Measurement and Control 2018; 140(7): 1-13, <https://doi.org/10.1115/1.4038657>.
10. Kraft M, White N. MEMS for automotive and aerospace applications. Cambridge, UK, Woodhead Publishing Limited: 2013: 29-53.
11. Pieniżek W, Janczur R, Gajek A, Wolak S. Verification of sensors for yaw rate and lateral acceleration in car ESP system. The Archives of Automotive Engineering 2020; 88(2): 61-76, <https://doi.org/10.14669/AM.VOL88.ART5>.
12. Taracido E. Capability analysis of different scanning tools to check ECSS. CITA Conference, Dubai: 2015. [<https://citainsp.org/wp-content/uploads/2016/01/1.-Workshop-B2-Final-Presentation.pdf>].
13. Tumasov A, Vashurin A, Toropov E, Moshkov P, Trusov Y. Estimation of influence of ESP on LCV active safety in condition of curvilinear movement. In Proceedings of the International Conference on Vehicle Technology and Intelligent Transport Systems (VEHITS 2016), 118-123.
14. Unimetal Sp. z o.o. - information materials and technical specifications. [<https://unimetal-moto.com/>].
15. Unimetal Sp.z o.o. The diagnostic method of controlling and checking the operation of the ABS or ESP/ESC pressure modulator for motor vehicles. The patent application No P.436848 with Gajek A and Strzpek P. The Patent Office of the Republic of Poland 2021.

16. Wu Y, Ahmed Q, Chen W, Tian W, Chen Q. Model-Based Fault Diagnosis of an Anti-Lock Braking System via Structural Analysis. *Sensors* 2018; 18(12): 1-23, <https://doi.org/10.3390/s18124468>.
17. Yongqiang Z, Kaicheng Z, Chang L, Xiang L. Development of the safety diagnosis system for VCU of pure electric vehicle. *Journal of Physics: Conference Series* 2020; 1605(012033): 1-10, <https://doi.org/10.1088/1742-6596/1605/1/012033>.
18. Zhang G, Yu Z, Wang J. Correction of contaminated yaw rate signal and estimation of sensor bias for an electric vehicle under normal driving conditions. *Mechanical Systems and Signal Processing* 2017; 87: 64-80, <https://doi.org/10.1016/j.ymssp.2016.05.034>.