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EFFECT OF A TORSEN DIFFERENTIAL MECHANISM ON CAR TYRE WEAR

WPŁYW MECHANIZMU TORSEN NA ZUŻYCIE OPON SAMOCHODOWYCH

Key words: 4WD, Torsen differential, tyre friction work, tyre wear.

Abstract 4WD systems include various constructions aimed at optimal distribution of torque between drive axles. In these systems, it is necessary to apply mechanisms separating the power provided from the engine to drive wheels. Many systems operate on the basis of wet multi-disc clutch or/and dog clutch control with the use of mechatronic devices. However, in view of their reliability and construction simplicity, the most valuable 4WD systems under field conditions are mechanical solutions. Torsen is an inter-axle differential mechanism, where the emerging internal friction moment of this mechanism causes that all kinematic discrepancies between transmission shafts of the front and rear axle of a vehicle produce a change in the distribution of torque, helping to level the rotational speeds of transmission shafts of the front and rear axles. As a result of the phenomenon described, the drive wheels of both axles of the vehicle move with different peripheral speeds, which causes energy losses and wear of tyres on both vehicle axles. This paper presents an analysis of the results obtained in tests on a vehicle with the Torsen system and describes phenomena occurring in the examined system as a result of emerging kinematic discrepancies, which may be caused by such factors as vertical force acting on the wheels of individual drive axles and tyre pressure.

Słowa kluczowe: napęd 4WD, mechanizm Torsen, praca tarcia opon, zużycie opon.

Streszczenie Wśród układów napędowych 4WD wyróżnia się wiele konstrukcji mających na celu w optymalny sposób rozdzielać moment obrotowy pomiędzy osiami napędowymi. W tych układach niezbędne jest zastosowanie mechanizmów rozdzielających moc doprowadzaną z silnika do kół napędowych. Działanie wielu układów oparte jest na sterowaniu sprzęgłami wielotarczowymi mokrymi lub/i sprzęgłami kłowymi z wykorzystaniem urządzeń mechatronicznych. Jednakże, ze względu na swoją niezawodność oraz prostotę konstrukcji, w warunkach terenowych najbardziej cenionymi układami 4WD są rozwiązania mechaniczne. Mechanizm Torsen jest to międzyosiowy mechanizm różnicowy, gdzie powstający moment tarcia wewnętrznego tego mechanizmu sprawia, że wszelkie niezgodności kinematyczne pomiędzy wałami napędowymi przedniej i tylnej osi pojazdu są powodem zmiany rozdziału momentu napędowego w wyniku dążenia do wyrównania prędkości obrotowych wałów napędowych osi przedniej i tylnej. W wyniku opisanego zjawiska koła napędowe obu osi pojazdu w wielu sytuacjach poruszają się z odmiennymi prędkościami obwodowymi, co powoduje powstawanie strat energetycznych i zużywanie opon obu osi pojazdu. W pracy przedstawiono analizę wyników podczas badań pojazdu z układem Torsen oraz opisano zjawiska zachodzące w badanym układzie na skutek powstających niezgodności kinematycznych, których przyczyną mogą być takie czynniki jak obciążenie pionowe kół poszczególnych osi napędowych oraz ciśnienie w ogumieniu.

INTRODUCTION

In vehicles with 4WD systems using various types of inter-axle differentials **[L. 1, 2]**, changes in torque distribution affecting the tyre wear characteristics can be observed **[L. 3–10]**.

Another factor affecting tyre wear is differentiated wheel surface grip, expressed by the value of the longitudinal grip coefficient **[L. 1, 8, 10,** and **11]**. An examination of the effect of this coefficient on traction properties of vehicles has been the focus of research by multiple authors **[L. 6–8]**. This results from the need to ensure the proper level of driving safety, as well as to reach the highest vehicle movement efficiency. Consequently, the effect of grip for various road surfaces and tyres on traction properties is described in **[L. 7, 9, 10]**.

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The results of those works were used to improve calculation models, usually using empirical data **[L. 6–10]**, the development of methods for drive transmission control **[L. 6]**, and developing technical specification for improving the surface and tyre quality **[L. 3–5]**. At the same time, the literature pays little attention to the effect of the 4WD system on tyre wear, which is the subject of this paper.

Tyre wear is unavoidable during operation, and this is affected by the tyre operation conditions, environment, tyre structure, and the properties of the mix of which it is produced. This has an effect, not only on the functional performance of the tyre, such as vibrations, noise, and grip, but primarily on driving safety. In recent years, many researchers have devoted a lot of attention to analysing micro- and macro-mechanisms of interactions occurring between the tread and the surface to explain various forms of tyre wear.

In this paper **[L. 8]**, the dependency used for determining tyre wear is based on the effect of temperature and dynamic characteristics of the vehicle. Additionally, the following factors are analysed: the effect of speed, environmental temperature, tyre pressure, and the sprung mass on tyre wear.

There are several models used for quantitative determination of wear **[L. 4, 5]**. The Archard wear model expresses abrasive wear through the following equation:

$$Z = \frac{Q_i \cdot l \cdot k}{H} \quad (1)$$

where Z – is the wear expressed in m³, Q_i – is the vertical force on the wheel [N], l – is the distance covered [m], K – is a dimensionless wear coefficient assuming different values depending on the materials applied on which the slip occurred, and H – is the hardness factor expressed in N/m².

The Schallamach model originated from the research focusing on the wear of slipping wheels. Schallamach used an equation of a structure similar to the Archard model. However, the Schallamach expression focused on wear of tyres made of various types of rubber, and the derived equation for the general wear model has the following form:

$$Z = \gamma \cdot l \cdot Q_i \quad (2)$$

where Z – is the amount of worn material [m³], γ – is wear per unit of dissipated energy [m³/J], l – is the sliding distance covered [m], and Q_i – vertical force [N] acting on the wheel **[L. 4]**.

The approach of Bulgin and Walter uses the combination of fatigue and abrasive wear of material, which results in an empirical relationship of tyre wear. An advantage of this approach is the fact that it takes into account the value of the tyre slip. However, it is the only parameter taken into account to determine the value of wear **[L. 5].**

To analyse the effect of operation of inter-axle differential on vehicle tyre wear, on the basis of dependencies presented in works **[L. 3–5, 7]** and the hypothesis of linear dependency between wear and a vertical force acting on the wheel and the value of slip, the following equation was used:

$$Z = a \cdot b^s \cdot s^c \quad (3)$$

where a, b, c are empirical coefficients, s – slip.

Using partially or entirely worn tyres in a vehicle leads to changes in the relation between the grip coefficient and the vehicle speed. The zone of a sudden increase in the coefficient value moves towards higher values of vehicle speeds. The paper **[L. 3]** analysed factors affecting tyre wear. The focus should be placed on the dominant role of slip, sprung masses, and ambient temperature with regard to changes in the tyre wear rate.

RESEARCH OBJECT

The research object was an Audi A6 Quattro. The drive transmission system for the front and rear axles of the vehicle **(Fig. 1)** was separated by the inter-axle symmetric differential mechanism **(Fig. 2)** or by increased internal friction of the Torsen type **(Fig. 3)**. Detail technical and operational specifications of the vehicle are presented in **Tab. 1**.

A symmetric differential **(Fig. 2)** is the simplest inter-axle differential mechanism applied. In this mechanism, if the values of grip coefficient for wheels of both axles are identical, torque is equally split between drive axles. However, in deteriorated grip conditions

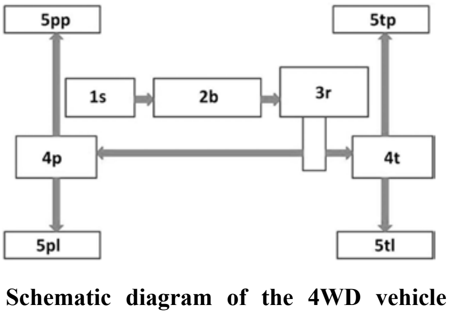

Fig. 1. Schematic diagram of the 4WD vehicle drive system model, where: 1s – vehicle engine, 2b – transmission box with a clutch, 3r – distribution box with inter-axle differential, 4p, 4t – differentials on vehicle axles, 5pp, 5pl, 5tp, 5tl – vehicle wheels (own study)

Rys. 1. Schemat ideowy modelu układu napędowego pojazdu 4WD, gdzie: 1s – silnik pojazdu, 2b – skrzynka przekładniowa ze sprzęgłem, 3r – skrzynka rozdzielcza z międzyosiowym mechanizmem różnicowym, 4p, 4t – mechanizmy różnicowe na osiach pojazdu, 5pp, 5pl, 5tp, 5tl – koła pojazdu (opracowanie własne) for one axle wheels, such a mechanism is not able to transmit higher torque to the other wheel than the one provided to the axle of worse wheel grip to the surface. The use of this type of mechanism is often supported by additional locks, aimed at levelling the rotational speeds of drive shafts in deteriorated grip conditions.

Application of the Torsen differential as the inter-axle differential mechanism affects the improvement of 4WD vehicle performance **[L. 11, 13]**. This mechanism is able to distribute torque between driven axles without additional control systems (e.g., electronic) under deteriorated grip conditions for one of the axles. The principle of Torsen operation (**Fig. 3**) is based on self-locking properties of a worm gear, which makes it possible to differentiate rotational speeds of axle drive shafts.

Table 1. Characteristics of the examined vehicle
Tabela 1. Charakterystyka badanego pojazdu

Parameter	Value
Maximum power	142 kW at 6,000 rev/min^{-1}
Maximum torque	280 Nm at 3,200 rev/min^{-1}
Tyre size	205/60 R15
Length / Width / Height of the vehicle	4.795 m / 1.785 m / 1.430 m
Wheelbase; front/rear track width	2.685 m; 1.520 m / 1.525 m
Gross vehicle mass / front axle / rear axle	2,100 kg / 1,000 kg / 1,100 kg
Drag coefficient C_x	0.32
Frontal area	2.1 m^2

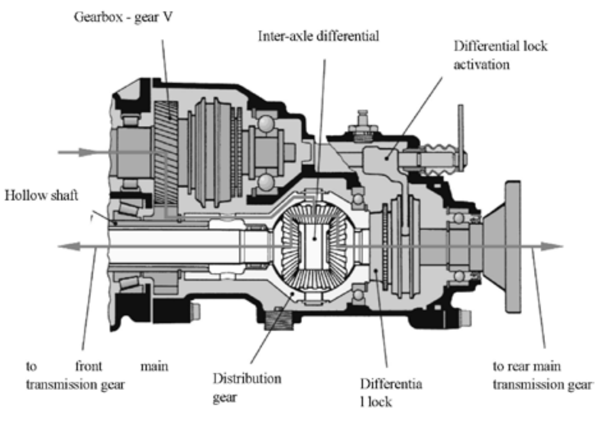

Fig. 2. Scheme of inter-axle symmetric differential applied in the Audi Quattro drive system [L. 12]
Rys. 2. Schemat międzyosiowego mechanizmu różnicowego symetrycznego zastosowanego w układzie napędowym Audi Quattro **[L. 12]**

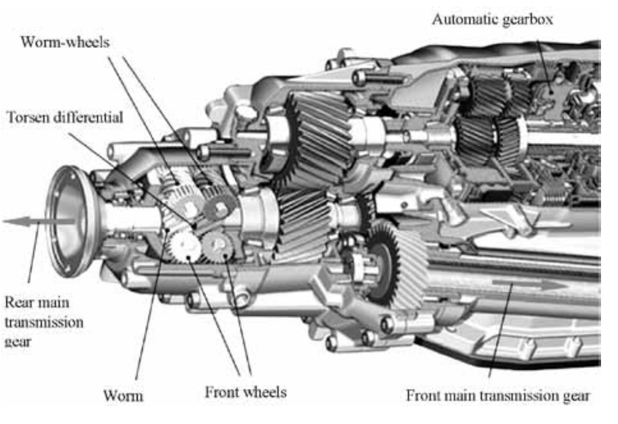

Fig. 3. Torsen T-2 mechanism used as inter-axle differential [L. 12]
Rys. 3. Mechanizm Torsen T-2 wykorzystany jako międzyosiowy mechanizm różnicowy **[L. 12]**

RESEARCH METHOD

The proposed method for examining wheel tyres in a 4WD system applies a verified and validated **[L. 16]** simulation model in which, among others, the relationships presented below are used.

The sum of the traction forces of a vehicle moving with constant velocity *v* = *const* is balanced with the sum of resisting forces and amounts to the following:

$$\sum F_t = F_r + F_w + F_g \tag{4}$$

where rolling friction $F_r = m \cdot g \cdot f \cdot \cos\alpha$, aerodynamic drag $F_w = 0,5 \cdot \rho \cdot A \cdot C_X \cdot v^2$, and grade climbing resistance $F_g = m \cdot g \cdot \sin\alpha$, while m – mass of the vehicle, g – acceleration due to gravity, α – road angle, f – rolling resistance coefficient, ρ – air density, A – frontal area of the vehicle, C_X – drag coefficient, v – linear velocity of the vehicle.

The sum of forces (4) is balanced with a sum of quotients obtained by dividing wheel torques by wheel radiuses on respective axles:

$$\sum F_t = \frac{T_{WF}}{r_F} + \frac{T_{WR}}{r_R} \tag{5}$$

where T_{WF} – front wheel torque, T_{WR} – rear wheel torque, r_F – front wheel dynamic radius wheel, r_R – rear wheel dynamic radius.

The sum of forces from the wheel grip condition is described with the following equation:

$$\sum F_t = Q_F \cdot \mu_F + Q_R \cdot \mu_R \qquad (6)$$

where Q_F – front wheel pressure, Q_R – rear wheel pressure, and the values of grip coefficients $\mu = \mu_F = \mu_R$ are related by the function dependence to the slip value s **[L. 14]**.

The slip value is determined from the equation:

$$s = 1 - \frac{v}{v_t} \qquad (7)$$

where v_t – theoretical velocity as a product of angular velocity and wheel radius, v – driving velocity.

Friction work of tyre thread W_{fr} is the sum of the product of wheel pressure Q_i, the coefficient μ, and friction path L_{fr}, where the friction path is the difference between the theoretical path resulting from the angular velocity and the wheel radius and the actual distance covered by the vehicle.

$$W_{fr} = \sum Q_i \cdot \mu \cdot L_{fr} \qquad (8)$$

Based on the relationships presented, a simulation model was designed (**Fig. 4**) using AMESim software **[L. 15]**. This program enables simulation tests that reflect the effect of the inter-axle differential mechanism on the car tyre wear, depending on the applied type of the inter-axle differential. The model contains a detailed description of drive wheel grip conditions **[L. 14]**.

The simulation assumed a linear movement of the vehicle with a set constant speed of 20 m/s. The vehicle moved along the path of the road angle $\alpha = 5°$.

The values of the vehicle mass, the distribution of masses between drive axles, dynamic radiuses, and the course of changes in the grip coefficient μ were introduced to the simulation software. The design of the simulation assumed changes in the grip coefficient as a function of the distance covered by the vehicle. The coefficient value was assumed as $\mu = 1$ from the moment of simulation start for the first 1,500 meters. Next, the value of the coefficient μ decreased to the value of $\mu = 0.4$ for the path of 200 m from the beginning of the simulation and was maintained at this level until the vehicle covered the distance of 3,500 meters. Next, the value of the grip coefficient increased again, to again reach the value of $\mu = 1$ at the distance of 4,000 meters and remained at this level to the end of the simulation. To carry out the analysis of the effect of kinematic discrepancies on tyre wear, simulations were conducted for three variants of dynamic radius values:

– $r_F - r_R = 0$ (*Front = Rear*),
– $r_F - r_R < 0$ (*Front < Rear*),
– $r_F - r_R > 0$ (*Front > Rear*).

For simulation variants, a difference was assumed between the values of dynamic radiuses at the level of 2%. This would correspond to a situation when worn tyres, but still permitted on roads (the value of the thread depth min 1.6 mm) are installed on wheels of one axle, while new tyres (thread depth about 8 mm) are fitted on the wheels of the other axle. Such a value differentiating dynamic radiuses of wheels is the limit value acceptable by systems controlling the drive system of the vehicle. The simulation tests were carried out for the 4WD system equipped with the inter-axle Torsen differential, and – for comparison purposes – with the symmetric differential.

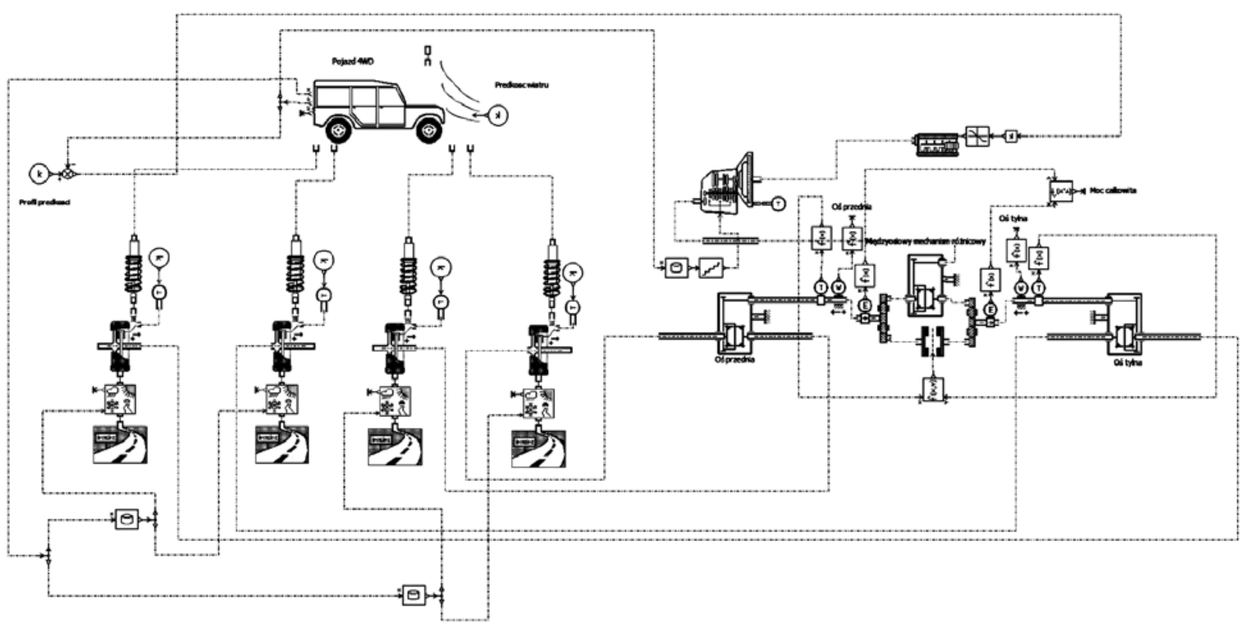

Fig. 4. Scheme of the simulation model of a vehicle with a 4WD system, constructed in the AMESim software

Rys. 4. Schemat modelu symulacyjnego pojazdu z układem napędowym 4WD zbudowanego w programie AMESim

THE RESULTS OF SIMULATION TESTS

The simulation test results demonstrated a significant effect of kinematic discrepancies introduced to the system on the differentiation of torque distribution between drive axles (**Fig. 5**). It should be emphasized that, when Torsen was applied in the drive system as the inter-axle differential mechanism, all differences in values of dynamic radiuses of front and rear axle wheels result in significant distortions of torques and, consequently, the traction forces obtained on drive wheels. This phenomenon occurs both on the surface of a higher value of coefficient μ = 1, as well as on the surface with coefficient μ = 0.4. When applying the symmetric inter-axle differential in the drive system, torques transmitted to drive axles were of similar values, and the system was not sensitive to introduced kinematic discrepancies. This is related to the characteristics of both solutions. For the symmetric mechanism, the system aims at levelling the values of drive torques on both axles. For the Torsen mechanism, the system aims towards levelling the values of rotational speeds of front and rear drive shafts.

Consequently, with kinematic discrepancies introduced in simulation tests, it can be observed that drive shafts at the Torsen differential have very similar velocity values, while the drive torque on both axles in the same conditions significantly differs, and, for the situation presented in **Fig. 5**, it can be observed that the values of drive torque on the front axle are negative, which proves that front wheels brake the vehicle, and only rear wheels power it. The obtained values of drive torques also translate to the values of the drive wheel slip. **Figure 6** presents the slip values for two types of road surface (I: μ = 1, II: μ = 0.4) for Torsen (T) and the symmetric differential (O) with three variants of dynamic radiuses of vehicle drive wheels. These values are presented separately for front axle wheels (F) and rear axle wheels (R). It was observed that the slip values of individual drive wheels at levelled dynamic radiuses of both axles are similar. However, when kinematic discrepancies emerge in the system as a result of differentiating the value of dynamic radiuses of front and rear axle wheels, for the system using the Torsen differential, the slip values of front and rear axle wheels are significantly differentiated. The wheels featuring the lowest radius value even cause the vehicle to brake (slip value s<0).

As a result, the wear ratio of tyres in the vehicle is increased, and (as the result of higher energy losses (**Fig. 7**) the demand of the drive system for power is higher, and consequently, fuel consumption increases.

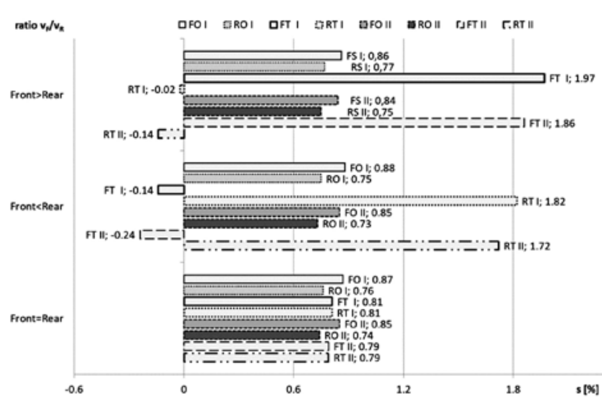

Fig. 6. Changes in the slip value of front and rear axle wheels

Rys. 6. Zmiany wartości poślizgu kół osi przedniej i tylnej

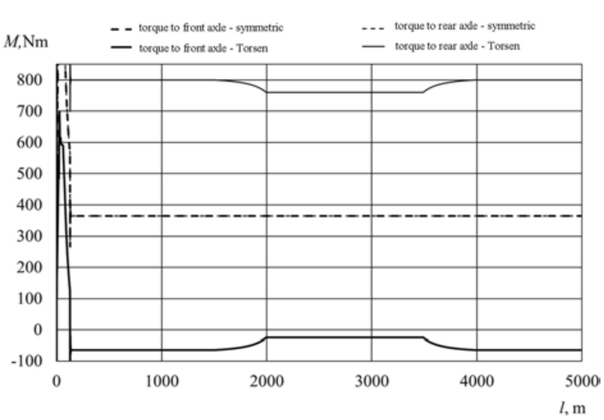

Fig. 5. Courses of changes in torque values on shafts of both drive axles (front axle – thick line, rear axle – thin line) for Torsen (continuous lines) and the symmetric differential (broken lines) for $r_F - r_R < 0$

Rys. 5 Przebiegi zmian wartości momentu obrotowego na wałach obu osi napędowych (oś przednia – linia gruba, oś tylna – linia cienka) dla mechanizmu Torsen (linie ciągłe) oraz symetrycznego (linie przerywane) w sytuacji, gdy $r_F - r_R < 0$

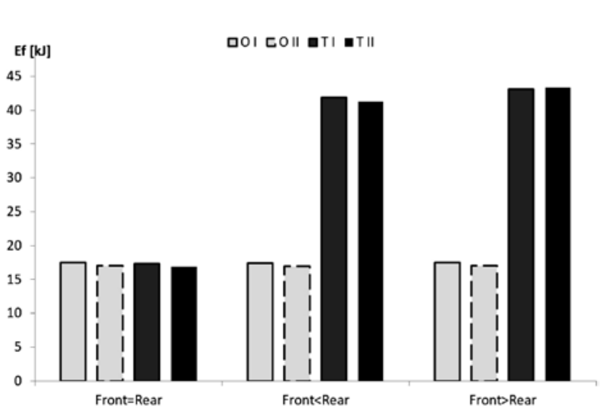

Fig. 7. Summary work of tyre friction forces during the operation of various differentials and ratios of dynamic radiuses of drive wheels at the distance of 1 km

Rys. 7. Sumaryczna praca sił tarcia opon podczas funkcjonowania poszczególnych mechanizmów różnicowych i stosunkach promienni dynamicznych kół napędowych na odcinku drogi 1 km

During the simulation tests of the drive system with the Torsen differential mechanism, with kinematic discrepancies introduced to the system (r_F>r_R), higher slip values of the front axle wheels were observed during the movement on good grip surface (μ = 1). As a result of higher slip values, the wear of tyres on the front axle is also increased. In comparison to the results of the same variant of the simulation for the vehicle equipped with the symmetric inter-axle differential, an increase in front axle tyre wear for the Torsen differential was 89.22%. The absolute value of tyre wear for the Torsen differential amounted to 7.7 mm^3 per 100 km of vehicle mileage. For simulating the movement along the surface with a lower grip coefficient (μ = 0.4) for the same simulation variant (r_F>r_R), front axle tyre wear increases in relation to the vehicle with the symmetric differential mechanism by 90.2%. Similarly, changes in tyre wear can be observed for the simulation variant where r_F<r_R, but in that case, the wear of rear axle tyres increases. At levelled values of dynamic radiuses for vehicle movement on various road surfaces, an increased wear of rear axle wheel tyres by 16% was observed for the Torsen mechanism and a reduction in the wear of front axle wheel tyres by 22% in relation to the system equipped with a symmetric mechanism. This is related to the characteristic features of the Torsen differential operation and the distribution of masses in the vehicle, and consequently, a higher load on the rear axle wheels.

CONCLUSIONS

In vehicles with a 4WD system using various types of inter-axle differentials, changes in the distribution of torque affecting the tyre wear characteristics can be observed. Based on previous studies **[L. 3, 4, 5, 7]**, a hypothesis of linear relationship between car tyre wear and the vertical force acting on the wheels and the slip value was assumed. Consequently, during the analysis of simulation test results, particular emphasis was placed on slip values obtained on individual drive wheels.

As results from the simulation tests conducted, when applying Torsen in the drive system as the inter-axle differential mechanism, all differences in values of dynamic radiuses of front and rear axle wheels cause significant distortions of torques and, consequently, the traction forces obtained on drive wheels.

The obtained values of drive torques also translate to the values of the drive wheel slip. For instance, at levelled values of dynamic radiuses for vehicle movement on various road surfaces, the tests demonstrated for the Torsen differential mechanism an increased wear of rear axle wheel tyres by 16% and a reduction in the wear of the front axle wheel tyres by 22% in relation to the system equipped with a symmetric differential. This is related to the characteristic features of Torsen operation and the distribution of masses in the vehicle and, consequently, a higher load on the rear axle wheels. Regardless of the surface type, with kinematic discrepancies introduced to the system, a significant (about 90%) increase in the wear of axle wheels characterized by a higher value of dynamic radiuses was observed for the Torsen differential used in the drive system compared to the symmetric differential mechanism.

Based on the obtained results of simulation tests, it can be seen that the application of Torsen as the differential mechanism in the drive system results in higher wear ratio of vehicle tyres than in case of the symmetric differential. The Torsen differential mechanism aims at levelling the rotational speeds of front and rear axle wheels, while the symmetric mechanism in all grip conditions operates towards an equal distribution of torque between the wheels of front and rear drive axles.

REFERENCES

1. Naunheimer H., Bertsche B., Ryborz J., Novak W.: Automotive Transmissions: Fundamentals, Selection, Design and Application. Second Edition. Springer: Heidelberg, Dordrecht, London, New York. 2011. 715 p.
2. Dębowski A. Modelling of centre differential control. Journal of KONES Powertrain and Transport, Vol. 18, No. 1 2011.
3. Yong Li, Shuguang Zuo, Lei Lei, Xianwu Yang and Xudong Wu: Analysis of impact factors of tire wear. Journal of Vibration and Control 18(6) 833–840 DOI: 10.1177/1077546311411756.
4. Mehdi Davari M., Jerrelind J., Stensson Trigell A., Drugge L.: Extended brush tyre model to study rolling loss in vehicle dynamics simulations International Journal of Vehicle Design Volume 73, Issue 4 DOI: 10.1504/IJVD.2017.083418.
5. Zibo Chen, Sharan Prathaban: Modeling of Tyre Parameters' Influence on Transport Productivity for Heavy Trucks. Chalmers University of Technology, Gothenburg, master thesis 2013.
6. Kanwar Bharat Singh & Saied Taheri (2015) Estimation of tire–road friction coefficient and its application in chassis control systems, Systems Science & Control Engineering, 3:1, 39-61, DOI: 10.1080/21642583.2014.985804.
7. Svendenius J. Tire: Modeling and Friction Estimation. Department of Automatic Control, Lund University, Lund, April 2007, p. 194.

8. Park K.-S., Oh C.-W., Kim T.-W., Leong H.-Y.: An improved friction model and its implications for the slip, the frictional energy, and the cornering force and moment of tires. Journal of Mechanical Science and Technology (KSME Int. J.), Vol. 20, No. 9, pp. 1399–1409, 2006.
9. Bechtoff J., Koenig, Isermann R.: Cornering Stiffness and Sideslip Angle Estimation for Integrated Vehicle L. Dynamics Control FAC-Papers On Line 49-11 (2016) 297–304.
10. Villagra J., Andre-Novel B., Fliess M., Mounier H.: A diagnosis-based approach for tire–road force sand maximum friction estimation Control Engineering Practice 19 (2011) 174–184.
11. Heißing B., Bleck U., Bensinger J., Müller E.: The Influence Of A Torsen Centre Differential On The Handling Of Four-Wheel Drive Vehicles. SAE Technical Paper 885140, 1988, https://doi.org/10.4271/885140.
12. Fischer R., Gscheidle R., Heider U., Hohman B., Keil W., Mann J., Pichler W., Schlogl B., Siegmayer P., Wimmer A., Wormer G.: Podwozia i nadwozia pojazdów samochodowych. Podręcznik do nauki zawodu Technik pojazdów samochodowych, Warszawa 2012.
13. Steinwart J., Bauder A., Ohrnberger G.: Design and Development of the New Audi 3.6 litre, 32 Valve V8 Engine, SAE Technical Paper 900650, 1990, https://doi.org/10.4271/900650.
14. Pacejka H.: Tire and vehicle dynamics. Elsevier, 2005.
15. AMESim® User's Guides. 2013. p. 596.
16. Janulin M.: Wybrane aspekty efektywności układu przeniesienia napędu typu 4WD pojazdu kołowego, Praca doktorska, Olsztyn 2017.