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DETERMINATION OF THE MULTIAXIAL MOTOR VEHICLES STEERING QUALITIES CHARACTERISTICS

WYZNACZANIE CHARAKTERYSTYKI STEROWNOŚCI WIELOOSIOWYCH POJAZDÓW SAMOCHODOWYCH

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Summary

An elementary indicator of a car steerability, the understeering gradient, is determined on the basis of dependency of the steering wheel turn angle changes in the vehicle lateral acceleration function. So called comparative dynamic wheel turn angle is a component of an appropriate calculation formula, in which—for multi-axle vehicles—there exists so called equivalent wheelbase L_e of a substitute two-axle vehicle. This paper analyses methods for determining this parameter. There was used a substitute bicycle model of the vehicle with an equivalent wheelbase L_e, to which a flat model of a multi-axle vehicle can be compared. There have been presented and compared two analytical methods of calculation that were illustrated with concrete examples of figures. Deliberations presented in this paper can be used to support calculations when analysing the steerability tests of three-axle and four-axle special vehicles, including the High Mobility Wheeled Platform (HMWP) – a universal purposes prototype vehicle.

Keywords: multi-axle commercial vehicles, vehicle steerability, steerability characteristics, equivalent wheelbase, understeering gradient, High Mobility Wheeled Platform

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Streszczenie

Podstawowy wskaźnik kierowalności samochodu, gradient podsterowności, jest wyznaczany na podstawie zależności zmian kąta obrotu kierownicy w funkcji przyspieszenia poprzecznego pojazdu. Członem odpowiedniego wzoru obliczeniowego jest tzw. porównawczy, dynamiczny kąt skrętu koła, w którym, dla pojazdów wieloosiowych, występuje tzw. ekwiwalentny rozstaw osi L_e, zastępczego pojazdu dwuosiowego.

W pracy przeanalizowano metody wyznaczania tego parametru. Wykorzystano zastępczy, rowerowy model pojazdu, z ekwiwalentnym rozstawem osi L_e, do którego można sprowadzić płaski model pojazdu wieloosiowego. Przedstawiono i porównano dwie, analityczne metody obliczeń, które zilustrowano konkretnymi przykładami liczbowymi. Rozważania przedstawione w pracy mogą być wykorzystane do wspomagania obliczeń przy analizie badania kierowalności trzyosiowych i czteroosiowych pojazdów specjalnych, w tym także kołowej Kołowej Platformy Wysokiej Mobilności (KPWM) – prototypowego pojazdu uniwersalnego zastosowania.

Słowa kluczowe: wieloosiowe pojazdy samochodowe, kierowalność, charakterystyki kierowalności, ekwiwalentny rozstaw osi, gradient podsterowności, Kołowa Platforma Wysokiej Mobilności

1. Introduction

To some extent this paper was inspired by the requirement to test steerability of the High Mobility Wheeled Platform that was designed by the Team of Experts from the SZCZĘŚNIAK Pojazdy Specjalne sp. z o. o. company from Bielsko-Biała, in cooperation with the Military University of Technology and Military Institute of Armoured and Vehicle Technology. The prototype was produced by the Szczesniak Special Vehicles Ltd. company from Bielsko-Biała, within the Project No. WND-DEM-1-325/00 KoPlatWysMob.

The platform is a three-axle vehicle 6×6 of the N3G category. Among its many characteristics, it can be distinguished by its ability to achieve high speed (120 km/h), which imposes an obligation on the producer to perform steerability tests in order to check and assure that this vehicle can safely move in the traffic.

One of the aims of this paper is to present and compare methods for calculating the equivalent wheelbase of multi-axle vehicles that are replaced by a simplified equivalent two-axle vehicle during the steerability analysis.

2. Assessment of the vehicle steerability on the basis of experimental tests on a circular path for fixed movement states

When testing heavy motor vehicles, the results obtained from tests conducted on a circular path in fixed movement states depend on the radius of the path. For automobiles and light trucks (so called delivery trucks) this dependency usually exists for the higher lateral accelerations of the vehicle ($a_y > 4.5$ m/s^2), which is mainly connected with an exceeded linear range cooperation between the tyre and road. This is mirrored in basic steerability characteristics—graphical dependency of the steering wheel turn angle changes in the vehicle lateral acceleration function. For motor trucks, boundary of the courese linearity varies and often the course is linear for the entire range of the obtained lateral accelerations.

Drawing 1 shows sample courses of the motor trucks steerability characteristics that were taken from tests performed by Christian von Glasner and used in [3]. These trials were conducted on a path with a fixed radius of R = 42 m.

In view of domestic conditions, it is impossible to carry out a determined full circle drive test, the radius of which is $R \geq 50$ m, as there is not any sufficient military training area present. For this reason, these vehicles undergo a determined not full circle drive test (on the circle arc length) with the use of a method of "fixed speed, changeable turn angle of the steering wheel" [8, 9]. The test performance methodology is described in these publications in details. The trial includes consecutive drives with a fixed speed (according to the ISO norm [8] 50 ± 3 km/h and GOST [9], 60 ± 3 km/h). Trials, in which this method is used, can also be conducted with an adequate speed for vehicles of other categories (e.g. for M1 category vehicles 80 ± 3 km/h). Each drive starts with driving straight ahead. When the speed is fixed, the steering wheel is turned and the applied angle is kept for the time necessary to determine the measured circle movement parameters (not less than 3 sec.). The steering wheel turn angle must be increased in the following drives δ_H, until the limit value of the vehicle obtained lateral acceleration ay that results in a slip of the rear axle or limit border of the lateral tilt. It is recommended by the ГОСТ Р 52302-2004 Russian Federation norm [9] that the $a_y \geq 2{,}5$ m/s^2 acceleration is obtained.

To ensure security during the conducted trials, it is recommended that suitable security supports are used.

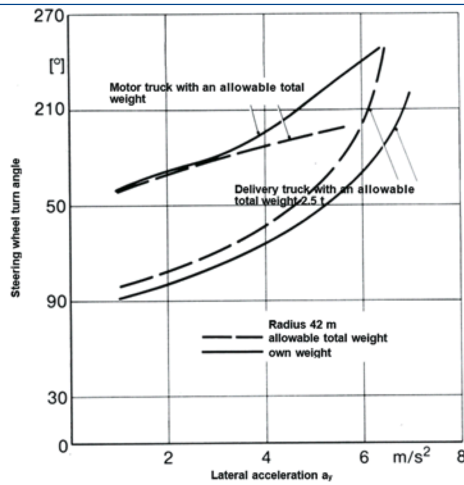

Fig. 1. Comparative characteristics of the truck vehicles steerability [3]

2.1 Measure of controllability

The measure of controllability constitutes gradient of the steering wheel turn angle with respect to the lateral acceleration [2, 8]. This gradient is also known as understeering gradient and defined by the formula:

$$Grad_p = \frac{\partial \delta_H}{\partial a_y}\frac{1}{i_s} - \frac{\partial \delta_D}{\partial a_y} \tag{1}$$

where: δ_H – steering wheel turn angle, i_s – steering system kinematic ratio, δ_D – comparative dynamic wheel turn angle, a_y – lateral acceleration.

The kinematic ratio i_s, that is present in the (1) formula, can be determined during tests on stations [5].

For a two-axle vehicle, the comparative dynamic wheel turn angle is determined from the formula:

$$\delta_D = \frac{L}{R} \tag{2}$$

where: L – is the two-axle wheelbase, R – is the masses center trajectory radius.

The δ_D angle does not exist, when the trial has place on a path with a fixed radius.

There can be distinguished three cases of vehicle steerability:

$Grad_p > 0$ – understeer vehicle
$Grad_p = 0$ – neutral vehicle
$Grad_p < 0$ – oversteer vehicle

From the active security point of view, the third case is unacceptable, because in practice it means that the turn is tightened, which relates to the tendency of self-acting change of lane that causes obvious danger for vehicles approaching from the opposite side.

2.2 Substitute models of multi-axle vehicles. An equivalent wheelbase

For multi-axle vehicles, the (2) formula contains an equivalent wheelbase Le of a substitute two-axle vehicle, instead of L. Drawings 2 and 3 show outlines of these vehicles.

The T value (so called *tandem factor*) is calculated from the formula:

$$T = \frac{\sum_1^N \Delta_i^2}{N} \tag{3}$$

where N – is a number of non-steering axles (in three-axle vehicle N = 2), Δ_i, i = 1,2 – half of the rear tandem axle wheelbase.

In the paper [7], the Le equivalent wheelbase of a two-axle substitute vehicle is determined by formula Δ:

$$L_e = L\left[1+\frac{T}{L^2}\left(1+\frac{C_{\alpha 2}}{C_{\alpha 1}}\right)\right] \tag{4}$$

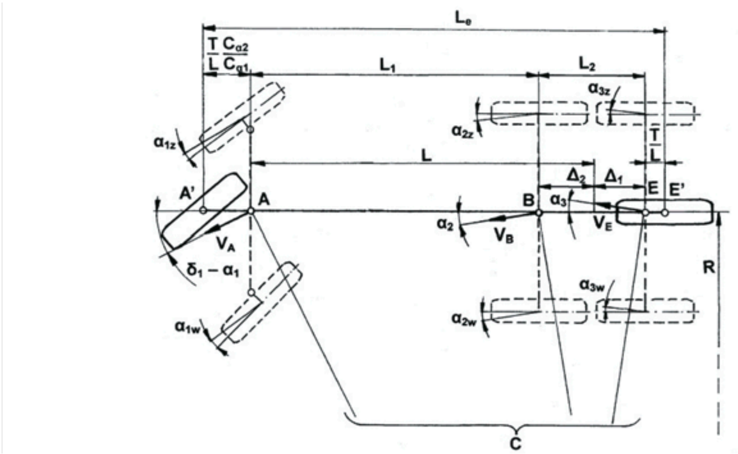

Fig. 2. Three-axle vehicle and its simplified two-axle model. Signs: $\mathbf{L}$ – distance from the 1st steering axle to the rear axle, α_1 – average substitute drift angle of the 1st steering axle, α_2, average substitute drift angle of the 2nd steering axle, α_3, α_4 – average drift angles of the rear axles, $\mathbf{T}$ – "tandem factor" – indicator of a tandem axle, $C_{\alpha 1}$, $C_{\alpha 2}$ – substitute coefficients of the front axle resistance to drifting, L_e – equivalent wheelbase, δ_1 – average turn angle of the 1st axle wheels

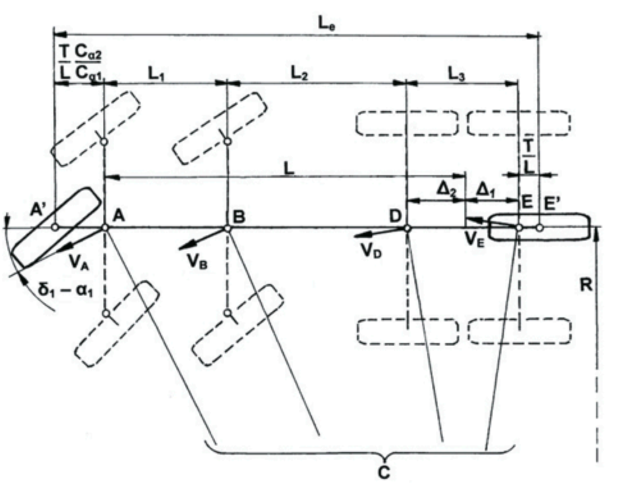

Fig. 3. Four-axle vehicle and its simplified two-axle model. Signs like in Drawing 2.

Usually, in the understeer and neutral vehicles $C_{\alpha 2} \leq C_{\alpha 1}$.

Substitute drift resistance coefficients $C_{\alpha 1}$, $C_{\alpha 2}$ that exists in this relation can be determined on the basis of the tyres characteristics specified on the special testing stations. For the steering axle, also the steering system stiffness must be considered. The stiffness can also be determined during testing on stations that do not require any specialized equipment [5]. It results from the (6) formula that $L_e > L$.

Drawing 4 shows tyres characteristics for various loads of the tested wheel

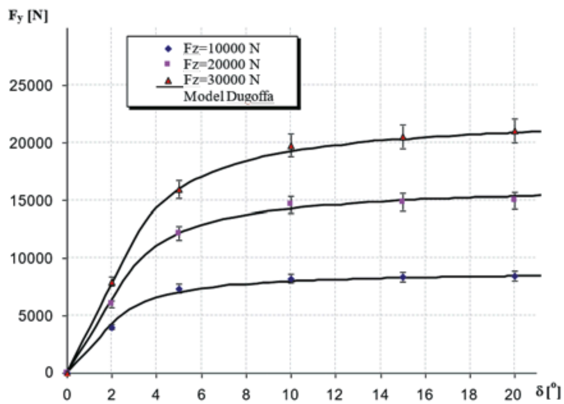

Fig. 4. Example of the tyres characteristics approximation determined when rolling a wheel with lateral drifting (tyre 255/70R22,5) [1]

Equivalent wheelbase can be determined on the basis of data taken from experimental tests, following formula [8]:

$$L_e = \frac{\partial \delta_H}{\partial \kappa} \cdot \frac{1}{i_s}, \ dla \ a_y \approx 0 \tag{5}$$

For this reason, from a fixed drive circle trials at a very low speed $V = 3 - 5$ km/h (then $a_y \approx 0$) one must determine the $\kappa = (1/R)$ track curvature for drives with different steering wheel angles δ_H. The L_e equivalent wheelbase is a slope of a straight line that approximates points with coordinates (κ, δ_k) in the chart $\delta_k = f(\kappa)$, where $\delta_k = \delta_H/i_s$. In the [9] norm, this straight line is called the Ackermann line (the Ackermann angle, as known in the PN-ISO 8555 terminology norm).

3. Another method for analytical determination of the multi-axle vehicles wheelbase

The (4) formula allows to determine L_e by analytical means. In the [3] paper, there have been presented formulas for the turn radius of a three-axle and four-axle vehicles that move at a low speed. It must be noticed that these formulas are created from "a purely" kinematic relations.

For a three-axle vehicle, in which the front axle is the steering one, the formula looks as follows:

$$R = \frac{L - l_2 \dfrac{l_1 - l_2}{(L + l_1)}}{tg\delta_1} = \frac{C_3}{tg\delta_1} \tag{6}$$

For a four-axle vehicle with two front steering axles the vehicle turn radius is (Drawing 3):

$$R = \frac{L - \dfrac{l_2 l_3}{2(l_2 + l_3)}}{tg\delta_1} = \frac{C_4}{tg\delta_1} \tag{7}$$

In the above formulas (Drawings 2 and 3): L is a distance between far axles of the vehicle, l_1 is the wheelbase for the 1st and 2nd axles (steering axles wheelbase in the four-axle vehicle), l_2 is a distance between the 2nd and 3rd axles (tandem in a three-axle vehicle), l_3 is a distance of the 3rd and 4th axles (in a four-axle vehicle), δ_1 – is the angle average of the 1st axle turn. To shorten the notation, the obvious C_3, C_4 fixed coefficients were introduced.

Using the (6) and (7) formulas, the $\delta_1 = f_1\ (1/R)$, $\delta_1 = f_2\ (1/R)$ dependencies for three-axle and four-axle vehicles can be determined. In an analogous method to the experimental tests (see chapter 2.2., formula 5), the slope of a straight line that approximates points on the obtained charts is the sought equivalent wheelbase L_e. Further in the paper, there will be made comparisons with values obtained on the basis of formula (4).

To assess accuracy of the applied calculation methods of L_e, one should compare the results with values determined on the basis of the experimental test data, as the most reliable reference values. In the paper, there will be made comparison only for a four-axle vehicle, for which L_e was also determined in an experimental way.

4. Three-axle vehicles equivalent wheelbase calculation examples

4.1 Three-axle motor truck with all-wheel drive (6x6), further marked with the A letter.

Calculations were made in accordance with (6).

Allowable total weight 7570 kg (distribution of masses between the front/rear axles 3810/3760 kg), wheelbase of first and second axles l_1 = 2.99 m, second and third (tandem) axles wheelbase l_2 = 1.25 m, distance from the front steering axle to the symmetry axis of the tandem rear axle) is l = 3.615 m.

Assuming the "bicycle" model (Drawing 3) [11], the rear axle shift value T/l = 0.106 m (Drawing 3) and the shift value $(T/l)\ (C_{\alpha r}/C_{\alpha f})$ = 0.212 m.

When making proper reductions of the tyres lateral stiffness, in accordance with the chosen model, and doing simple calculations using the (6) formula, we receive Le = 3.928 m.

4.2 Three-axle special vehicle with all-wheel drive (6x6), further marked with the B letter

Allowable total weight 13900kg (distribution of masses between the front/rear axles 4100/9800kg), first and second axles wheelbase $l_1 = 3.315$ m, second and third (tandem) axles wheelbase $l_2 = 1.370$ m, distance from the front steering axle to the symmetry axis of the tandem rear axle) is L = 3.995 m.

Doing calculations similar to those in point 3.1, we receive L_e = 4.243 m.

4.3 High Mobility Wheeled Platform—off-road vehicle (6x6), further marked as HMWP

Allowable total weight 34000 kg (distribution of masses between the front/rear axles 8000/26000 kg), $l_1 = 3.5$ m, $l_2 = 1.39$ m, $L = 4.125$. For this vehicle $T/l = 0.102$ m, $(T/l)\ (C_{\alpha r}/C_{\alpha f})$ = 0.204 m.

After calculations we receive L_e = 4.431 m.

This dependency for the A, B and HMWP vehicles is shown in Drawing 5.

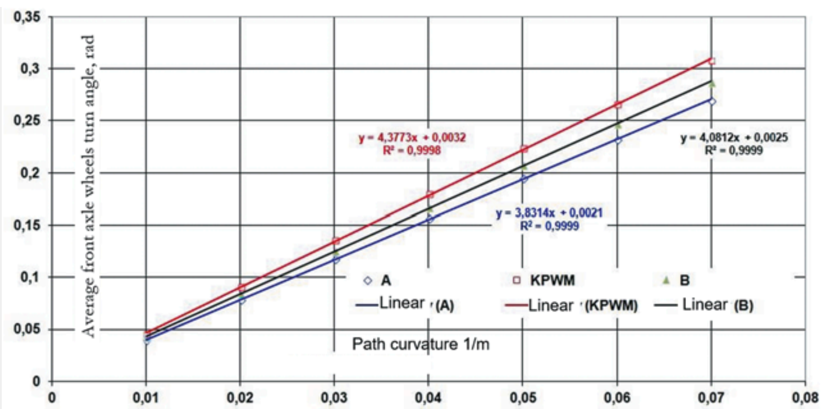

Fig. 5. Angular dependency of the δ_1-front steering axle on the track curvature 1/R for three-axle vehicles. The slop of the straight line constitutes the equivalent wheelbase $y = L_e$ of the substitute two-axle vehicle (instead of L (2))

Table 1 contains results of the equivalent wheelbase calculations Le for three-axle vehicles that were performed with the use of various methods.

Table 1. Results of the L_e equivalent wheelbase calculations for three-axle vehicles

No.	Vehicle	Equivalent wheelbase, L_e, m			Relative error (I–II)/I, %
		Method I (formula 4)	Method II (formulas 6 and 7)	Absolute error I–II	
1	A	3.928	3.831	0.097	2
2	B	4.243	4.081	0.162	4
3	HMWP	4.431	4.377	0.054	1.2

4.4 Four-axle vehicle 8x8, two front steering axles

For this vehicle L_e was determined also on the basis of the experimental tests data (chapter 2, formula (5)).

Table 2 contains results for a 8×8 special vehicle with the test mass of approximately 20 000 kg. Appropriate values for the (4) and (7) formulas are as follows: $L = 4.55$ m, $l_1 = 1.4$ m, $l_2 = 1.7$ m, $l_3 = 1.45$ m.

Table 2. Results of the L_e calculations for a 8×8 four-axle vehicle

Equivalent wheelbase, L_e, m					
Method I (formula 4)	Method II (formula 7)	III Experiment	Relative error* of the Ist method, %	Relative error* of the IInd method, %	Relative error of the IInd method ** %
4.100	4.074	3.96	3.5	2.7	0.6

*relative to III, **relative to method I

Drawing 6 shows a graphical comparison of Ledetermined with the use of method II (7) and experimental method (cf. chapter 2.2.).

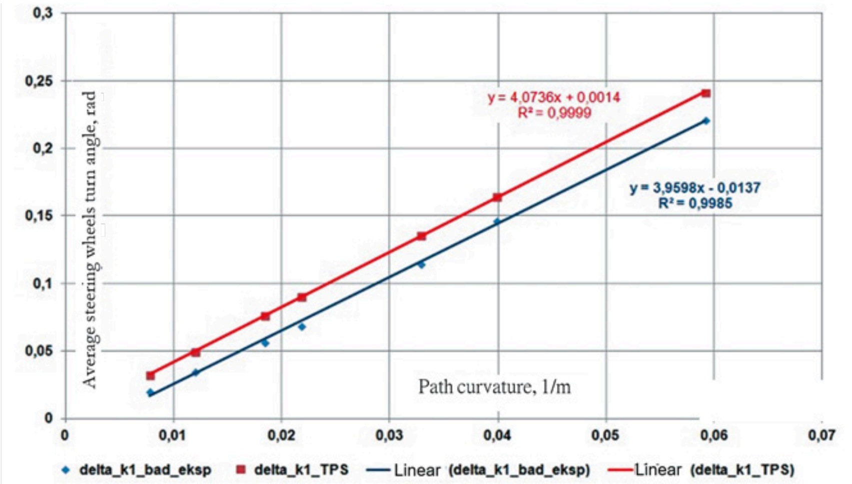

Fig. 6. Comparison of the L_e equivalent wheelbase for the 8×8 four-axle vehicle. Red line - course determined with the use of formula (7). Navy blue line - course determined on the basis of the experimental tests data

3. Conclusions

Comparison of the L_e equivalent wheelbase calculation methods for multi-axle vehicles with the use of analytical methods with reference values allowed assessing the relative error and usefulness of the used methods.

In the case of a three-axle vehicle with a front steering axle, this error does not exceed 4%. The L_e value was taken as a reference value. It was calculated in accordance with [7], taking under consideration resistance to the tyres drifting – formula (4) in chapter 2.2 of this paper. These vehicles were not tested in the military training area environment with the use of the "fixed speed, changeable steering wheel turn angle" method.

In case of a four-axle vehicle with two front steering axles, this error exceeds 1% with the reference value that is in accordance with the (4) formula and 4% with the reference value determined on the basis of the military training area tests data.

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