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STRUCTURAL-KINEMATIC COMPONENTS OF A STEERING SYSTEM FOR VEHICLES WITH INTEGRAL STEERING

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Abstract: *The paper presents a mechanical system for turning the rear wheels of a vehicle with four-wheel drive, under integral steering conditions. The rear steering box – from the author's perspective – includes (contains) a cam with double profiles that drives the slide block, or the rocking lever of the steering mechanism. The drive of the cam is performed by the steering rack of the front axle. The kinematic correlations of the steering system components of a vehicle with integral steering are given as a purely mechanical structure, driven by the steering wheel.*

Key words: *CAM, mechanism, steering box, integral steering*

1. INTRODUCTION

For vehicles with four-wheel drive, traditionally, the rear wheels are turned in the opposite direction to the front ones, in order to increase the maneuverability, i.e. to reduce the steering radius. By turning the wheels in such a way, the vehicle stability – at high speeds - will be compromised.

To ensure both a good maneuverability and stability, solutions are explored for the so-called "integral steering", where, at the beginning of a turn –when vehicle travelling at high speeds – the rear wheels will be turned in the same direction to the front ones (ensuring a good stability), and as the curvature of the trajectory increases, the rear wheels to return in normal position (for going straight) and then will be turned in the opposite direction to the front ones (obtaining small turning radius).

A number of researches have proposed various mechanisms for the steering box of the rear axle to fulfill the requirements of the integral steering, highlighting papers (Fraukawa, 1985) – for steering box with 4R linkage,– driven by two cranks; as well as paper (Kido, 1990) – for steering box with cam mechanisms (cam with a pair of cam followers which are hold by a follower support).

The authors' papers (Alexandru et al., 2011, Macaveiu & Alexandru, 2011)– with the idea of a pure mechanical structure – develop and underlie variants of linkages and cam mechanisms for the steering box of the rear axle which will drive the rear wheels and turn them under the integral steering requirements, respectively: at the beginning of a turn (at high speeds) the rear wheels are turned in the same direction to the front ones, return in normal position and then turn in the opposite position.

2. STRUCTURAL-KINEMATIC COMPONENTS

The paper presents the correlation among characteristics of steering transmission components of vehicles with two steering axles for a purely mechanical structure under the integral steering condition.

The purely mechanical structure will be driven only by the car's steering wheel. Therefore, the transmission from the steering wheel to the wheels is following the kinematic chain (fig.1.a): the front steering box (e.g. rack and pinion), the front steering mechanism (e.g. with central rack) and (in parallel) a device for driving the longitudinal shaft (e.g. rack and pinion gear); reducer (e.g. spur-gear), the rear steering box (e.g. cam mechanism), the rear steering mechanism (e.g. translational slide block).

The kinematical parameters from figure 1 are:

V_a – vehicle speed (direction of travel),

$\theta_{e,i}$ –turning angle of the front wheels (*e,i* – exterior/interior to the vehicle path),

φ_v – steering wheel rotation angle,

S_c – the displacement of the front steering rack,

φ_t – the rotation angle of the rear translation shaft,

φ_c – the rotation angle of the cam of the rear steering box,

S_t – the displacement of the translational slide block of the rear steering mechanism,

$\sigma_{e,i}$ –turning angle of the rear wheels (*e,i* – exterior/interior to the vehicle path),

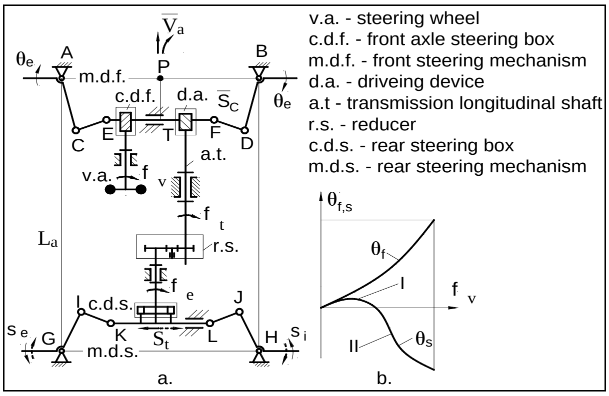

Fig. 1. Transmission components of the steering system

By turning the steering wheel with the angle φ_v, the displacement S_c of the rack is obtained:

$$S_c = r_p \varphi_v = \frac{m_t z_p}{2} \varphi_v = \frac{m_n}{2\cos\beta} z_p \varphi_v, \qquad (1)$$

where: r_p – the pinion radius; m_t – teeth module; z_p – number of teeth of the pinion; m_n – nominal module, β - helix angle.

The rack displacement S_c causes the turning of the front wheels with the angle θ_e / θ_i, the function $\theta_e (S_c / \varphi_v) / \theta_i (S_c / \varphi_v)$ being determined by the geometry ACEFDB of the steering mechanism. The correlation $\theta_i (\theta_e)$ shows the turning law of the front wheels.

The steering angle θ_f of the front axle, is considered to be

$$\theta_f = (\theta_e + \theta_i)/2\ , \qquad (2)$$

thus, is dependent on the geometry of the steering system, i.e. the length of the bars,

$a = \overline{AC} = \overline{BD}$, $b = \overline{AB}$, $l = \overline{CE} = \overline{DF}$, $c = \overline{EF}$, $e = \overline{PT}$, as well as on the gear characteristics of the steering box m_n, β, z_p, based on $\theta_f (\varphi_v)$ (fig.1.b).

For the device that drives the longitudinal shaft, the rotation of the transmission shaft is obtained from the rack,

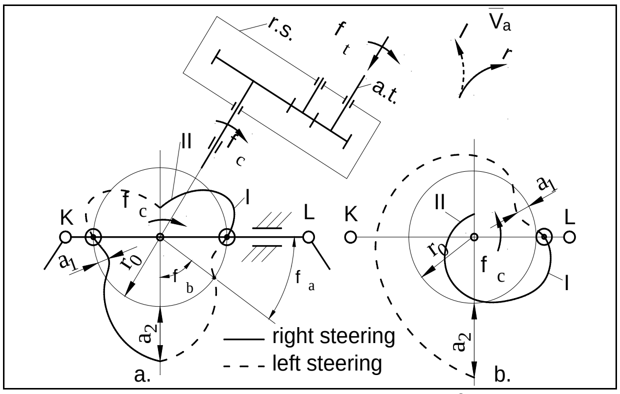

Fig. 2. Cams arrangement in the steering box of the rear axle

through a gear of the same module m_n and number of teeth z_r.

$$\varphi_t = \frac{S_c}{r_r} = \frac{r_p \varphi_v}{r_r} = \frac{z_p}{z_r} \varphi_v . \qquad (3)$$

The transmission reducer, positioned before the steering box of the rear axle, serves to transmit to the central element of the steering box a rotational movement φ_c, through which the two axles are correlated. Thus, for its transmission ratio i_t, the rotation angle of the central element of the steering box is:

$$\varphi_c = \frac{\varphi_t}{i_t} = \frac{z_p}{z_r i_t} \varphi_v . \qquad (4)$$

ratio i_t can be positive or negative.
The steering box of the rear axle, depending on its geometry, will cause the displacement S_t of the central cam follower/slide block according to the cam profile – fig. 2.a, for example, within a sine law of amplitude a_1, respectively a_2,

$$S_t = a_1 \sin \frac{\pi}{\varphi_a} \varphi_c , \text{ respectiv } S_t = a_2 \sin \frac{\pi}{2\varphi_b} \varphi_c . \qquad (5)$$

where, φ_a and φ_b are the maximum rotation angles of the cam, for going through the profile of the cam.
For steering to the right, if the ratio i_t is positive, the rotation φ_c has the same direction as φ_v, and the cam profile will be as in figure 2.a (the profile with continuous line). For the negative ratio i_t, the direction of the rotation φ_c is opposed to φ_v (fig.2.b), the cam profile being symmetrical (inverted) to the previous case. In both cases from figure 2, the cam follower was considered with a follower support that holds a pair of rollers. The profile with dotted line refers to steering to the left.
With the value of the ratio i_t, the maximum rotation angle of the cam is determined: for the one with follower support with tow roller being 90^o, with the angle φ_a (for the profile I) and $\varphi_b = \pi/2 - \varphi_a$ (for the profile II).

For the follower with one roller, the rotation angle of the cam can reach up to 270^o, in one way or the other (fig.2.c), in this case $\varphi_b = 3\pi/2 - \varphi_a$. Thus, we obtain the dependency

$$S_t = a_1 \sin \frac{\pi}{\varphi_a}\left(\frac{z_p}{z_r i_t}\varphi_v\right), \text{ resp. } S_t = a_2 \sin \frac{\pi}{2\varphi_b}\left(\frac{z_p}{z_r i_t}\varphi_v\right). \qquad (6)$$

According to the displacement S_t of the central slide block and to the steering mechanism geometry GHIJKL of the rear axle, which usually corresponds with the one of the front axle mechanism, rear wheels turnings $\sigma_e(\varphi_v)$ and $\sigma_i(\varphi_v)$are obtained. The turning angle θ_s of the rear axle

$$\theta_s = (\sigma_e + \sigma_i)/2, \qquad (7)$$

is depending on the rotation φ_v of the steering wheel - θ_s (φ_v).
Both angles of turning, θ_f (φ_v) and θ_s (φ_v) define the turning type/stage of the vehicle with two steering axles.
If the steering box of the rear axle is built from a linkage mechanism, e.g. the mechanism with the rocker, it needs, as seen above, two driving elements.

3. NUMERICAL APPLICATION

For reals values of a vehicle:
r_0=60 mm; a=116.8 mm; b=1299 mm; c=639 mm; l=326.2 mm e=131.3 mm; m_n=2; z_p=7; β=10^o→r_p=7.11 mm; L_a=2475 mm.
results the values from table 1

S_c[mm]	θ_f [^{o}]	φ_v [^{o}]	φ_t [^{o}]	φ_c [^{o}]			$S_{t\,I,II}$
0	0	0	0	i_t=3	i_t=1.5	i_t=1	0 mm
10	5^{o}01’	80.5	40.2	13.4	26.8	40.2	10
20	10^{o}05’	161	80.5	26.8	53.6	80.5	0
30	15^{o}14’	241.5	120.7	40.2	80.5	120.7	14.71
40	20^{o}33’	322	161	53.7	107.3	161	27.90
50	26^{o}06’	402.5	201.2	67.1	134.2	201.2	37.89
60	32^{o}02’	483	241.5	80.5	161	241.5	43.74
67	36^{o}33’	540	270	90	180	270	45

Tab. 1 Numerical results for the steering system

S_t representing the displacement of the slide block of the rear steering mechanism.

$$S_{t_I} = a_1 \sin \frac{\pi}{\varphi_a} \varphi_c = 10 \sin \frac{180}{26.8} \varphi_c , \qquad (8)$$

$$S_{t_{II}} = a_2 \sin \frac{\pi}{2\varphi_b} \varphi_c = 45 \sin \frac{90}{63.2} \varphi_c . \qquad (9)$$

Respectively the polar radius of the came according to table 2:

$$r_I = r_0 \pm S_{t_I} , \; r_{II} = r_0 \mp S_{t_{II}} . \qquad (10)$$

$r_{I,II}$ [mm]	60	67	70	67	60	45.3	32.1	22.1	16.2	15
	60	52.9	50	52.9	60	74.7	87.9	97.8	103.7	105

Tab. 2. Values of the cams polar radius

And the obtained function from figure 3

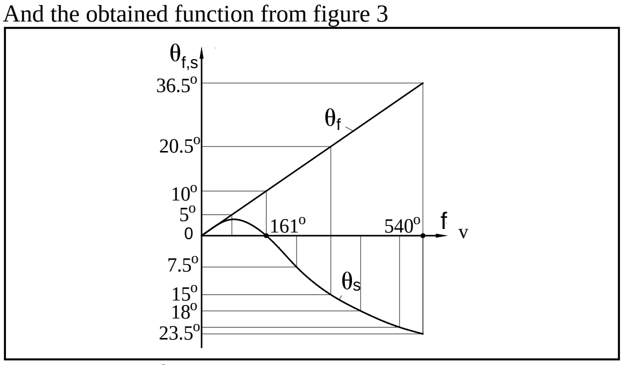

Fig. 3. Steering function charts within the integral steering

Conclusion: the steering system with the cam mechanism meet the requirements of the integral steering (fig. 3 related to fig.1,b)

4. ACKNOWLEDGEMENTS

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