

# **THE GEOMETRIC AND CONSTRUCTIVE SOLUTION FOR VEHICLE STEERING GEAR BOX WITH VARIABLE TRANSMITTING RATIO AND TESTING STAND**

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**Abstract:** This paper presents a gear with ascending variable transmission ratio designed for steering gear boxes of vehicles. The proposed solution uses a translational wheel and a geared segment that is eccentrically articulated. The numerical simulations were performed on a virtual model by establishing the motion function, the geometrical and functional parameters and the rolling radius variation of the geared sector, which results in a variable transmission ratio. The stand consists of a translational wheel (1), mating with an eccentrically geared segment (2); the translational wheel (1) is locked by a ball nut (3) articulated with a ball screw (4) coming from a moto-reducer (5). The moto-reducer is driven by the controller (6) and the power supply (7). The supporting frame (8) is made by welded beams.

## **1. INTRODUCTION**

The paper proposes to achieve a steering box with increasing variable transmitting ratio, and a testing stand for the steering box. The gear “translational wheel - geared segment eccentrically articulated” that achieves the variable transmitting ratio, is based on the classical gear, with rack and geared segment, which is widely used in automotive due to its advantages, such as higher mechanical efficiency and fewer linkage parts.

Usually the method of achieving variable ratio of a gear with rack and geared segment [2] is by using a rack that has variable shapes of the teeth from the center to both ends[4], but technologically difficult.

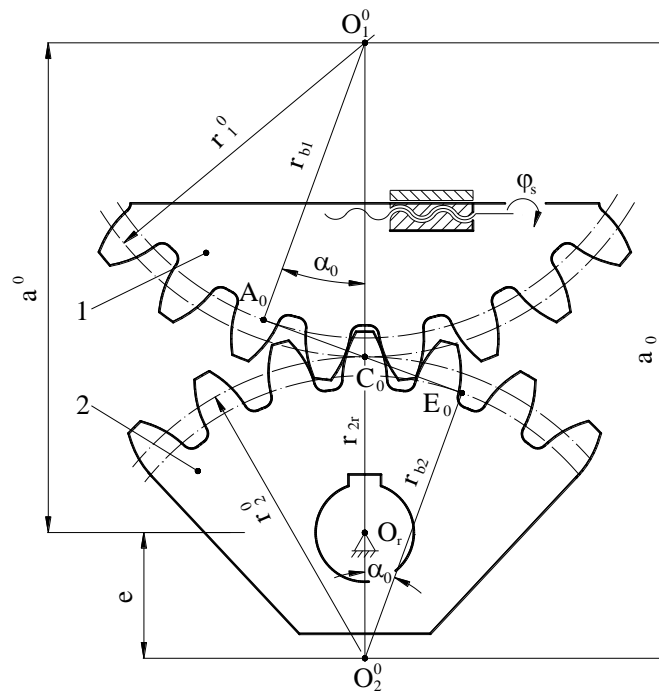
The objective of this paper is that the gear used for achieving the increasing variable ratio, to achieve an increasing transmitting ratio from center toward the ends, using a simple technology.

## **2. THE GEOMETRIC AND CONSTRUCTIVE SOLUTION OF THE STEERING GEAR BOX.**

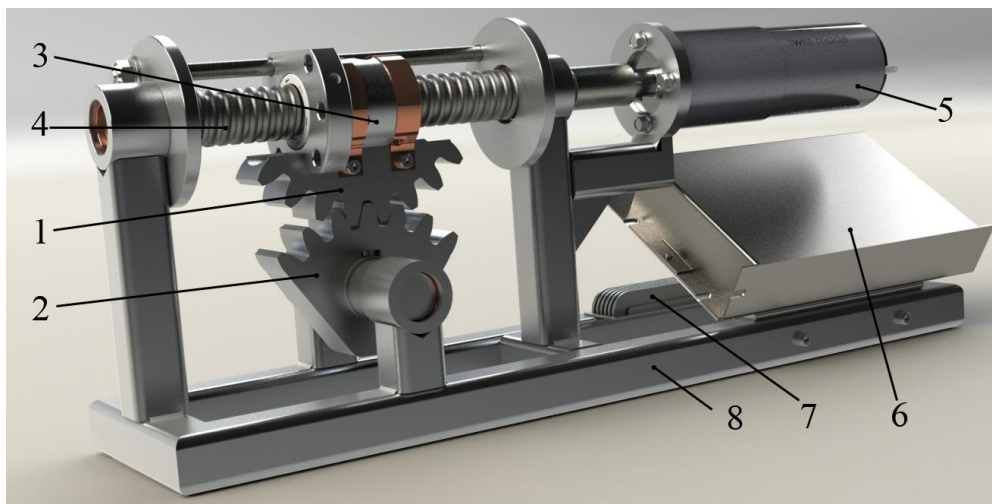
The idea for this steering box came from article [1] which has a briefly presented version of a steering box, the one with a driving rack, which can perform variable transmitting ratio by using a circular “curved rack” and a driven geared wheel which is eccentrically articulated.

In the present paper, for achieving the circular “curved rack” and the driven geared segment, parts of a classical involute wheels were used, without the need for a special technology, the rack being transformed into a translational wheel (Fig.1,a).

Numerical simulations were performed for different parameters (number of teeth of wheels –  $z_1/z_2$ , eccentricity –  $e$ , module –  $m_0$ ), assigning different values. After the evaluation of numerical simulations, the optimal version has been chosen for achieving the constructive solution.



**a.**



**b.**

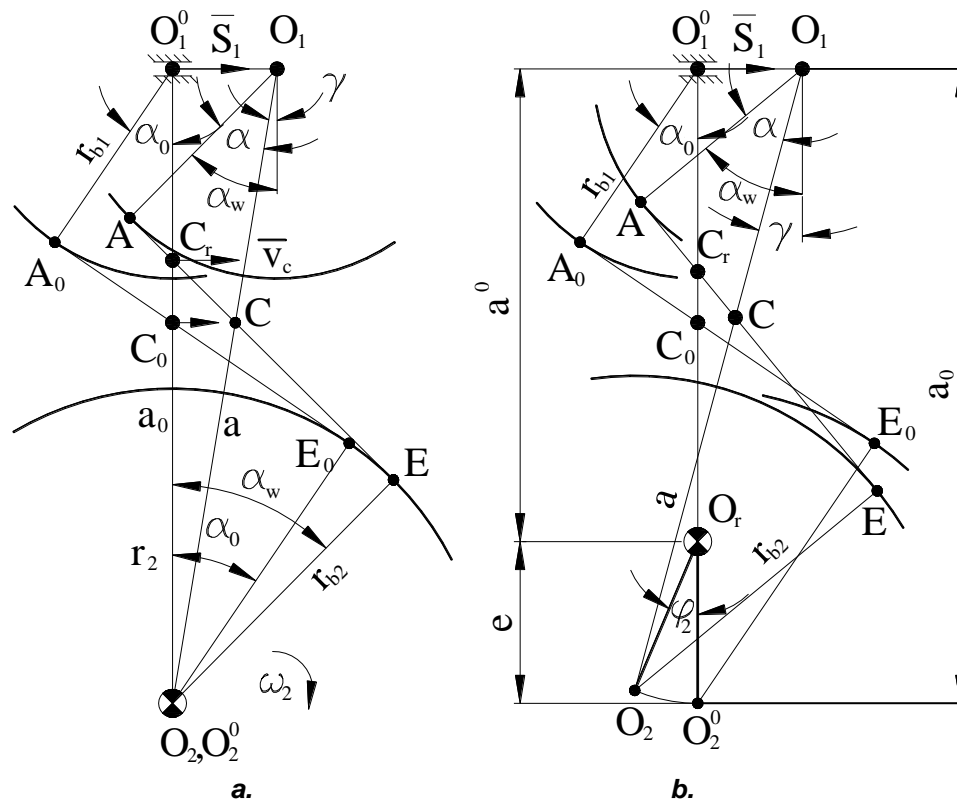
**Figure 1.a. Translational gear wheel – geared sector eccentrically articulated  
 b. Virtually 3D modeled stand to test steering gear box with variable ratio**

The stand consists of a translational wheel (1) (sector of an involute wheel), matting with an eccentrically geared sector (2); the translational wheel (1) is locked by a ball nut (3) – type FSI 20 – ST3 articulated with a ball screw (4) coming from moto-reducer (5) type Maxon - motor RE40/148867; reducer GE42/203115. The moto-reducer is driven by the controller (6) and the power supply (7). The supporting frame (8) is made by welded beams.

The purpose of the testing stand is to compare the gear functionality with the numerical simulations and observing the behavior of the gear under different situations of load and lost motion.

The stand is equipped with devices to measure various distances and angles (translational wheel displacement, angular displacement of geared segment, gear clearance, and input and output forces).

### 3. ACHIEVING THE VARIABLE TRANSMITTING RATIO.



**Figure 2. A cylindrical gear with driving „translational wheel”. a) Classic, b) Eccentric sector.**

The notations from figure 2 are:  $r_{b1} / r_{b2}$  - radii of the base circles;  $\alpha_0 / \alpha_w$  - pressure angles;  $C_0 / C_r$  - gearing poles;  $A_0E_0 / AE$  - lines of action;  $v_c$  - velocity of the pole  $C_r / C_0$ ;  $\omega_2$  - the angular speed of wheel 2;  $\alpha$  - the angle of the fictive gear with centers  $O_1 / O_2$ .

Considering the cylindrical gear with a driving translational wheel and centric sector (Fig.2,a) with centers  $O_1^0 / O_2^0$ , if there is applied a translation  $S_1$  to the driving wheel 1 on a direction which is normal to the centers' line, then the center  $O_1^0$  of the wheel will change its position in  $O_1$ . Therefore the angle  $\alpha_0$  will be  $\alpha_w$ , and by modifying the position of tangent plane from  $A_0E_0$  to  $AE$ , the contact point  $C_0$  will be moved in  $C_r$ .

As a result the variable transmitting ratio  $i_t$  will be given by changing the pole position  $C_0 \rightarrow C_r$ . Variable ratio is defined as  $v_c / \omega_2$ , and it varies according to radius  $r_2$ , which becomes  $r_{2r} = O_2^0C_r$ :

$$i_t = \frac{v_c}{\omega_2} = r_2, \quad (r_2 \rightarrow r_{2r}); \quad (1)$$

as radius  $r_{2r} = \overline{O_2^0C_r}$  varies according with displacement  $S_1$ ,  $r_2 \rightarrow r_{2r}(S_1)$ , therefore the ratio is variable.

It is obvious that the gear presented in figure 1.a is not functional, the cause being the clearance between the teeth and extremely small moving range. To assure the functionality of the gear, the wheel 2 will be eccentrically articulated [4], therefore the eccentricity will be  $e = \overline{O_rO_2^0}$ , so the center  $O_2$  of the wheel will no longer correspond with the revolution center, and becomes  $O_r$ . Thus, with the movement of the translational

wheel 1 with  $S_1$ , the wheel 2 will rotate with angle  $\varphi_2$ , therefore center  $O_2^0$  becomes  $O_2$ , and  $C_0 \rightarrow C_r$ . For a proper functionality of this gear, an optimal correlation value of geometrical parameters  $r_{b_1}, r_{b_2}, e$  is needed, so that the movement function  $\varphi_2(S_1)$  and clearance  $\bar{j}_n$  between the teeth will have acceptable variation fields.

The movement function has the form (2):

$$f(S_1, \varphi_2) = a_0 \sin \alpha_0 + r_{b_1} (\alpha + \gamma - \alpha_0) - r_{b_2} (\alpha_0 + \varphi_2 - \alpha - \gamma) - a \sin \alpha = 0, \quad (2)$$

the clearance between the teeth (3) respectively sector radius  $r_{2r}$  (4),

$$j_n = \overline{AE} - (\overline{AA'} + \overline{EE'}) = a \sin \alpha - (r_{b_1} + r_{b_2}) (\alpha + \theta_0), \quad \theta_0 = \operatorname{tg} \alpha_0 - \alpha_0. \quad (3)$$

$$r_{2r} = \overline{O_r C_r} = \overline{O_r O_1^0} - \overline{O_1^0 C_r} = a^0 - \left( S_1 \operatorname{tg} \psi + \left( r_{b_1} - \frac{S_1}{\cos \psi} \right) \frac{1}{\sin \psi} \right), \quad \psi = 90^\circ - \alpha_w. \quad (4)$$

#### 4. NUMERICAL SIMULATIONS

Steering box demands are referring to:

- Implementing the prescribed rotation  $\varphi_2$  for the geared segment, usually  $\varphi_{2\max} \cong 40^\circ$ ;
- Implementing the variation field of the transmission ratio  $\Gamma = r_{2r\max} / r_{2r}^0 \cong 1.3$ .

These functional requirements must be carried out in good gearing conditions, namely:

- assuring the acceptable clearance  $j_n$  between the teeth, excluding the negative values,
- maintaining the difference in distance between the axes " $a - a_0$ " to acceptable values, while avoiding negative ones.

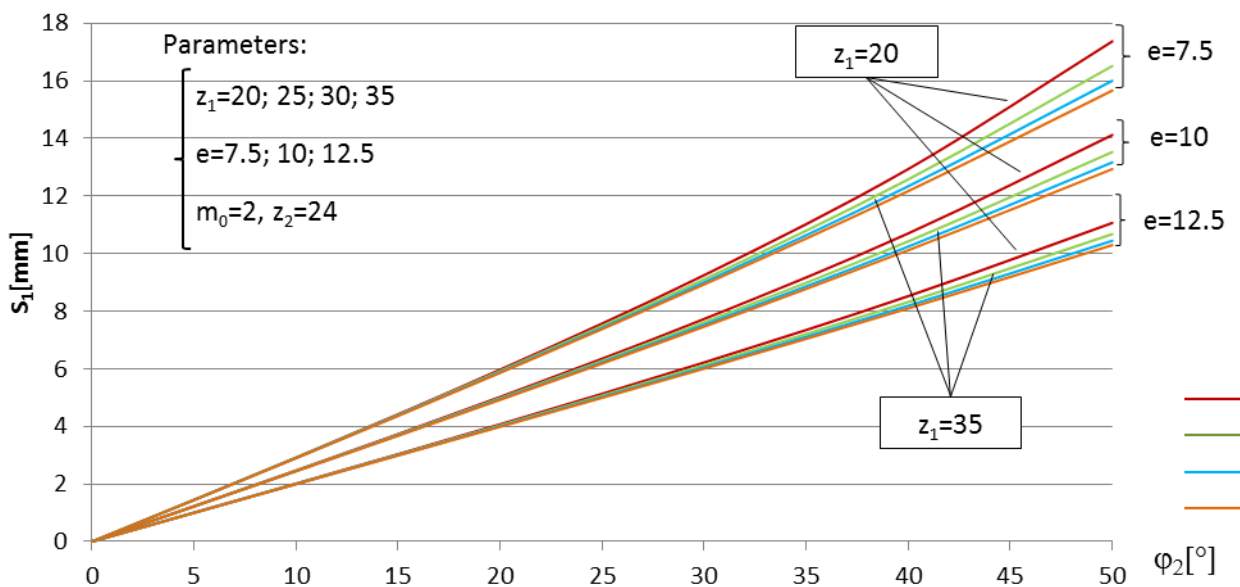
The numerical application aims to highlight how the functions  $\varphi_2(S_1)$ ,  $r_{2r}(S_1)$ ,  $j_n(S_1)$  vary for a range of values of geometrical parameters (e.g.  $z_1$  and  $e$ ), the goal being to establish the optimal values of these parameters.

To perform the numerical simulation, the following geometrical parameters were used: the module  $m_0 = 2 \text{ mm}$ , number of teeth  $z_2 = 24$ , number of teeth  $z_1$  and the eccentricity  $e$  is acceptable for  $z_1 = 20, 25, 30, 35$ , and  $e = 7.5, 10, 12.5$ . The results were shown in Table 1, starting from impose values  $\varphi_2 = 10, 20, 30, 40, 50^\circ$ .

**Table 1**

		$\varphi_2 [^\circ]$																	
		0°	10°	20°	30°	40°	50°	0°	10°	20°	30°	40°	50°	0°	10°	20°	30°	40°	50°
$z_1$	$e$	$S_1$ [mm]					$j_n$ [mm]					$r_{2r}$ [mm]							
20	7.5	0	2,92	5,97	9,26	12,94	17,38	0	0,03	0,13	0,34	0,71	1,41	16,50	17,04	18,03	19,75	22,77	28,88
	10	0	2,47	5,02	7,73	10,70	14,12	0	0,02	0,07	0,19	0,39	0,75	14,00	14,34	15,01	16,15	18,06	21,46
	12.5	0	2,02	4,08	6,23	8,53	11,07	0	0,00	0,02	0,04	0,10	0,22	11,50	11,64	12,01	12,68	13,76	15,51
25	7.5	0	2,91	5,93	9,11	12,58	16,52	0	0,02	0,10	0,25	0,51	0,95	16,50	16,92	17,68	18,94	20,98	24,48
	10	0	2,46	4,98	7,61	10,43	13,53	0	0,01	0,04	0,11	0,22	0,42	14,00	14,23	14,71	15,53	16,82	18,87
	12.5	0	2,01	4,04	6,13	8,32	10,68	0	0,00	-0,01	-0,03	-0,04	-0,03	11,50	11,54	11,77	12,23	12,97	14,11
30	7.5	0	2,90	5,89	9,01	12,34	16,01	0	0,02	0,07	0,18	0,36	0,66	16,50	16,82	17,43	18,40	19,90	22,25
	10	0	2,45	4,95	7,53	10,25	13,17	0	0,00	0,02	0,05	0,10	0,20	14,00	14,14	14,50	15,12	16,09	17,53
	12.5	0	2,00	4,01	6,06	8,19	10,45	0	-0,01	-0,04	-0,08	-0,14	-0,19	11,50	11,46	11,60	11,93	12,52	13,41
35	7.5	0	2,90	5,86	8,93	12,17	15,67	0	0,01	0,05	0,13	0,26	0,46	16,50	16,75	17,24	18,02	19,20	20,94
	10	0	2,45	4,92	7,47	10,12	12,94	0	0,00	0,00	0,00	0,01	0,05	14,00	14,07	14,34	14,84	15,61	16,74
	12.5	0	2,00	3,99	6,01	8,10	10,30	0	-0,01	-0,06	-0,13	-0,21	-0,31	11,50	11,40	11,47	11,73	12,23	13,02

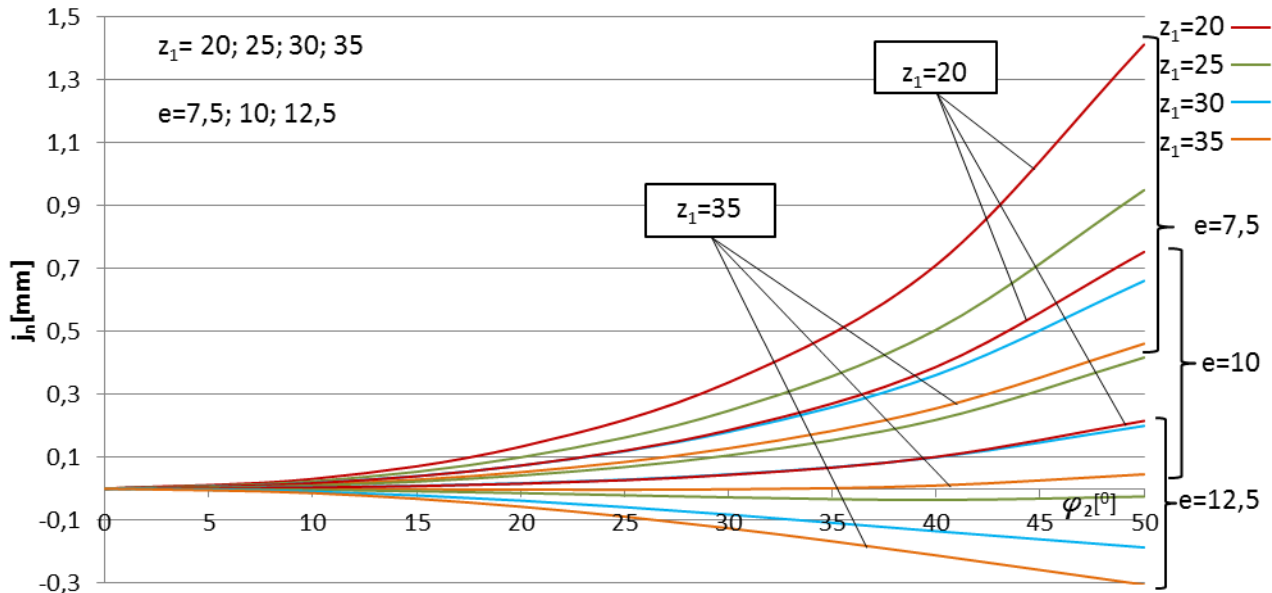
Figures 3 – 6 show the visualization of the results, through the charts of the functions  $S_1(\varphi_2)$ ,  $j_n(\varphi_2)$ ,  $r_{2r}(\varphi_2)$ . The charts are displayed according to  $\varphi_2$  of the geared segment because the values of  $\varphi_2$  are important for steering box's feature.



**Figure 3. Motion function  $S_1(\varphi_2)$ , for the considered parameters.**

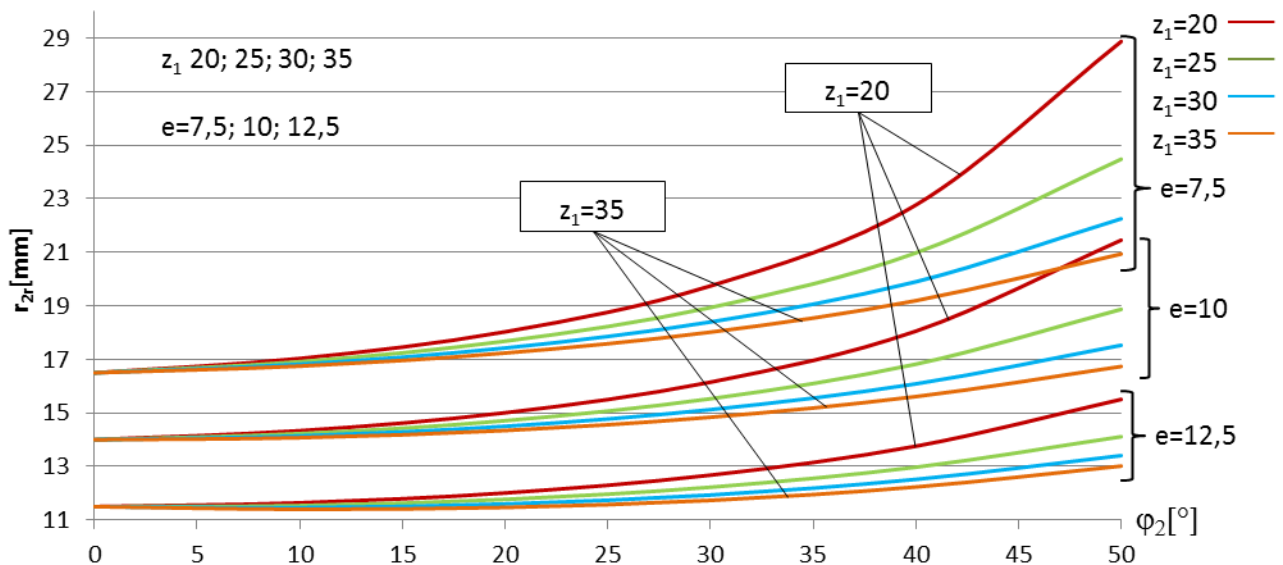
The results shown in Table 1 and related charts lead to important conclusions for establishing the optimal / acceptable values of imposed gear.

Therefore in figure 3 – function  $S_1(\varphi_2)$ , it is found that the influence of the number of teeth  $z_1$  of translating wheel is relatively small, while eccentricity “e” having significant influence. For values of  $\varphi_2 = 50^\circ$ , the displacement  $S_1$  is in the range  $S_1 = 15.7 - 17.4 \text{ mm}$  for  $e = 7.5 \text{ mm}$ , respectively  $S_1 = 10.3 - 11.1 \text{ mm}$  for  $e = 12.5 \text{ mm}$ . It is found that the eccentricity  $e = 12.5 \text{ mm}$  (although it is constructively disadvantageous) requires smaller rotation angle of the steering wheel.



**Figure 4. The variation of the clearance between the teeth  $j_n(\varphi_2)$**

From Figure 4 – function  $j_n(\varphi_2)$  is inferred that both the eccentricity  $e$  and number of teeth  $z_1$  are significantly influencing the clearance  $j_n$ . Generally,  $j_n > 0$  increases with decreasing number of teeth  $z_1$ , namely the radius  $r_1$ . It appears that for small eccentricities ( $e = 7.5 \text{ mm}$ ) the clearance  $j_n$  grows  $0.5 \dots 1.5 \text{ mm}$ , and for large eccentricities ( $e = 12.5 \text{ mm}$ ) the clearance  $j_n$  becomes negative – the gear locks. We consider that the value of  $e = 10 \text{ mm}$  is convenient.



**Figure 5. The variation of the sector radius  $r_{2r}(\varphi_2)$**



including the increasing variation of the rolling radius of the sector  $r_{2r}$  as a result of moving the contact pole  $C_r$  on centers' line.

The variation of the rolling radius  $r_{2r}$  of the sector has, as a direct result, the increasing variation of the transmitting ratio from the central position toward ends of the translational wheel race (functional motivation of this gear).

The nominated gear, simulated for the required conditions of a car steering box ( $\varphi_2 < 50^\circ$ ,  $\Gamma < 1,35$ ), is corresponding to the functional requirements imposed.

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### Refereces:

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