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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), matting 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.

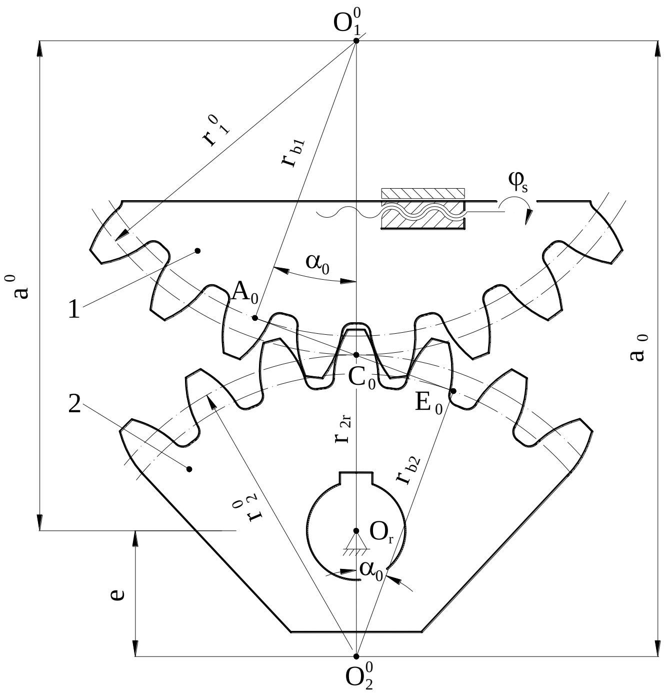
The objective of this paper is that the gear used for achieving the increasing variable ratio, to achieve an increasing transmitting ratio from center toward the ends, using a simple technology.

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

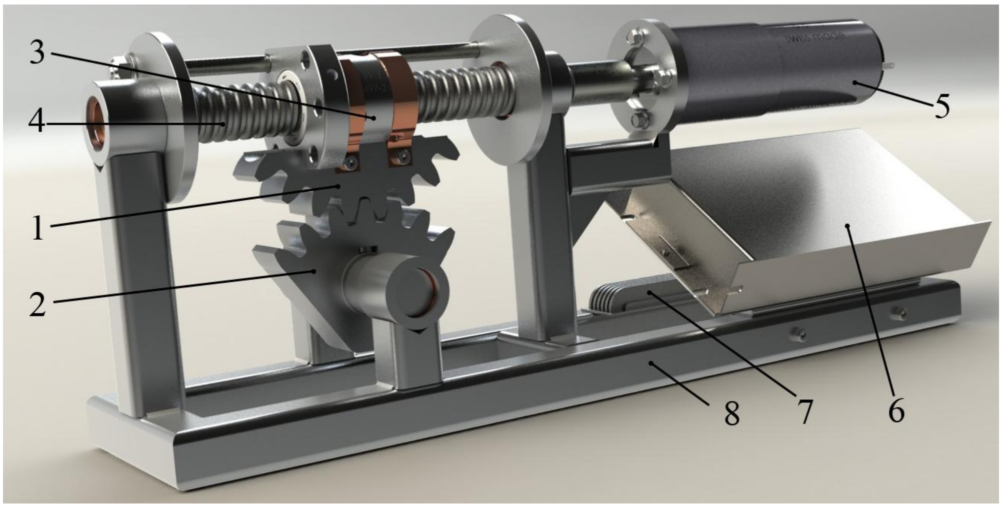
The idea for this steering box came from article [1] which has a briefly presented version of a steering box, the one with a driving rack, which can perform variable transmitting ratio by using a circular “curved rack” and a driven geared wheel which is eccentrically articulated.

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.

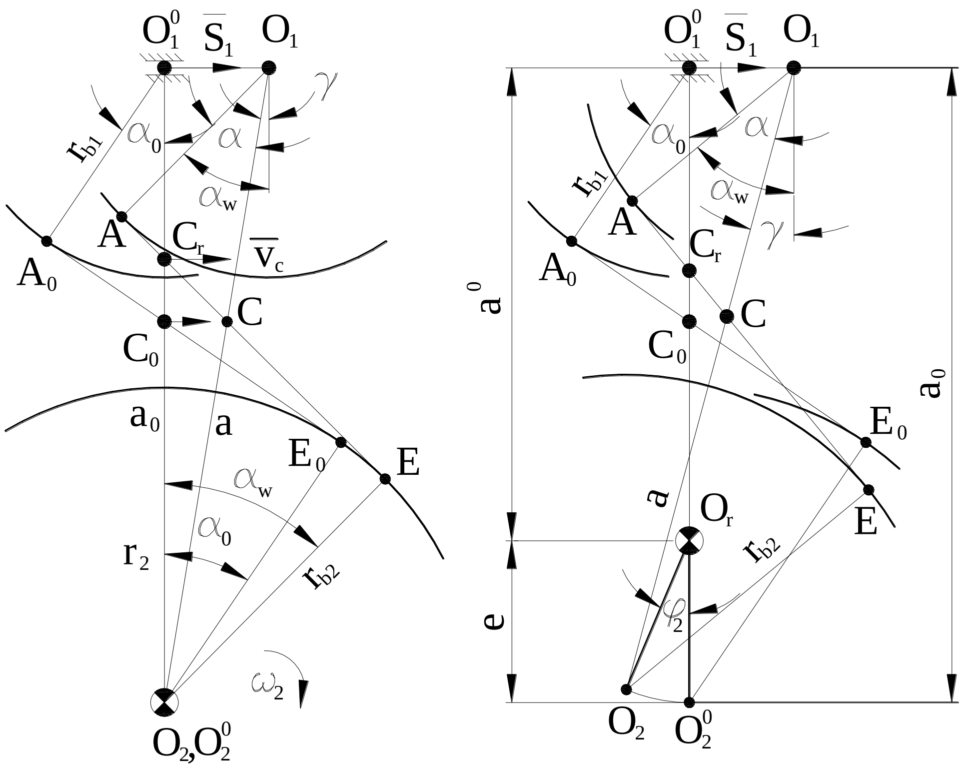
Figure 1.a. Translational gear wheel – geared sector eccentrically articulated
b. Virtually 3D modeled stand to test steering gear box with variable ratio

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.



a. *b.*

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

The notations from figure 2 are: r_{b_1} / r_{b_2} - radii of the base circles; α_0 / α_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; ω_2 - the angular speed of wheel 2; α - 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 α_0 will be α_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 $C_0 \to C_r$. Variable ratio is defined as v_c / ω_2, and it varies according to radius r_2, which becomes $r_{2r} = O_2^0 C_r$:

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

as radius $r_{2r} = \overline{O_2^0 C_r}$ varies according with displacement S_1, $r_2 \to r_{2r}(S_1)$, 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 $e = \overline{O_r O_2^0}$, 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 φ_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 $\overline{j_n}$ between the teeth will have acceptable variation fields.

The movement function has the form (2):

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

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

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

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

4. NUMERICAL SIMULATIONS

Steering box demands are referring to:

- Implementing the prescribed rotation φ_2 for the geared segment, usually $\varphi_{2\max} \cong 40°$;
- 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 $m_0 = 2mm$, number of teeth $z_2 = 24$, number of teeth z_1 and the eccentricity $\underline{e}$ 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 $\varphi_2 = 10, 20, 30, 40, 50°$.

Table 1

		φ_2[°]																	
		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 φ_2 of the geared segment because the values of φ_2 are important for steering box's feature.

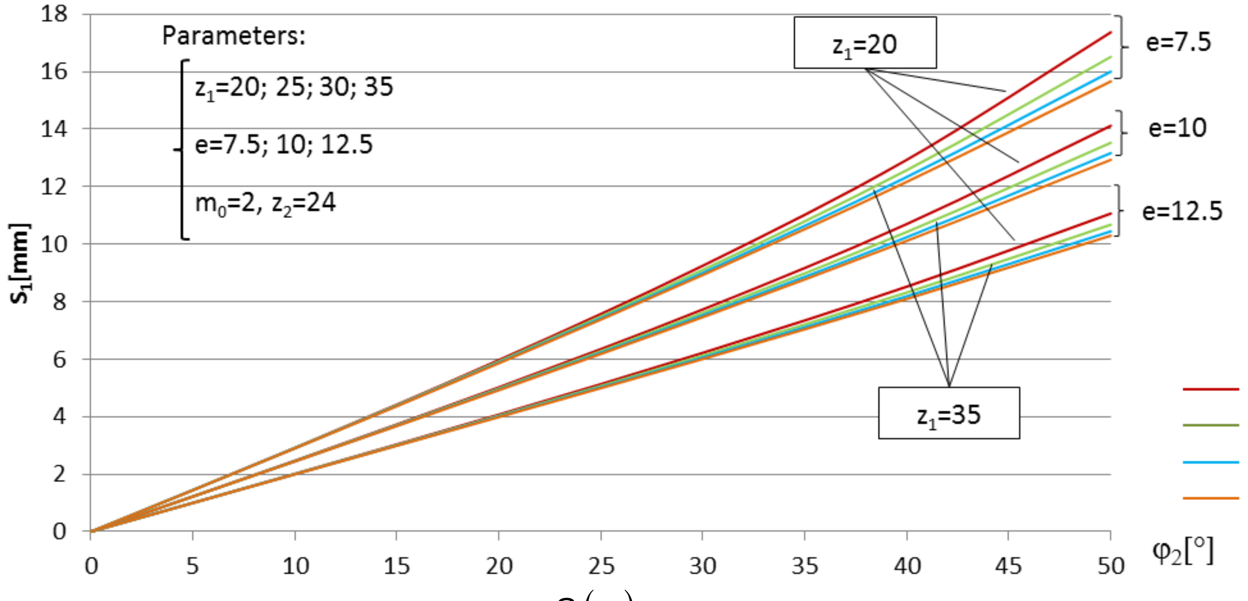


Figure 3. Motion function $S_1(\varphi_2)$, for the considered parametrers.

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 $\varphi_2 = 50°$, the displacement S_1 is in the range $S_1 = 15.7 - 17.4mm$ for $e = 7.5mm$, respectively $S_1 = 10.3 - 11.1mm$ for $e = 12.5mm$. It is found that the eccentricity $e = 12.5mm$ (although it is constructively disadvantageous) requires smaller rotation angle of the steering wheel.

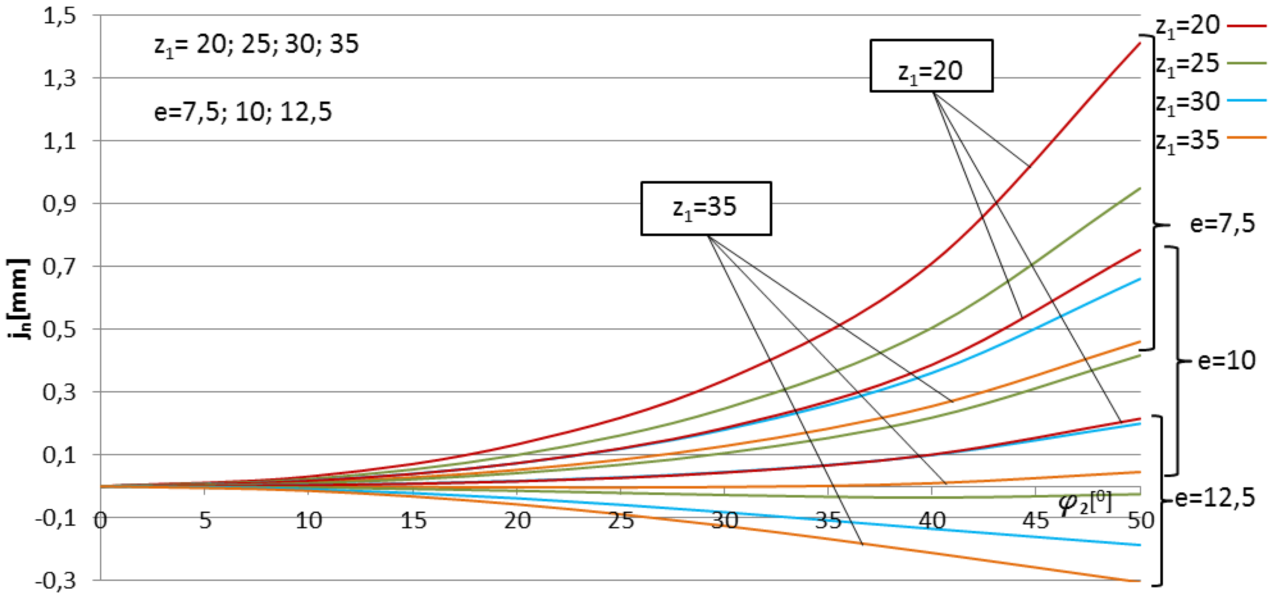


Figure 4. The variation of the clearance between the teeth $j_n(\varphi_2)$

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 ($e = 7.5mm$) the clearance j_n grows $0.5\ldots1.5\,mm$, and for large eccentricities ($e = 12.5mm$) the clearance j_n becomes negative – the gear locks. We consider that the value of $e = 10\,mm$ is convenient.

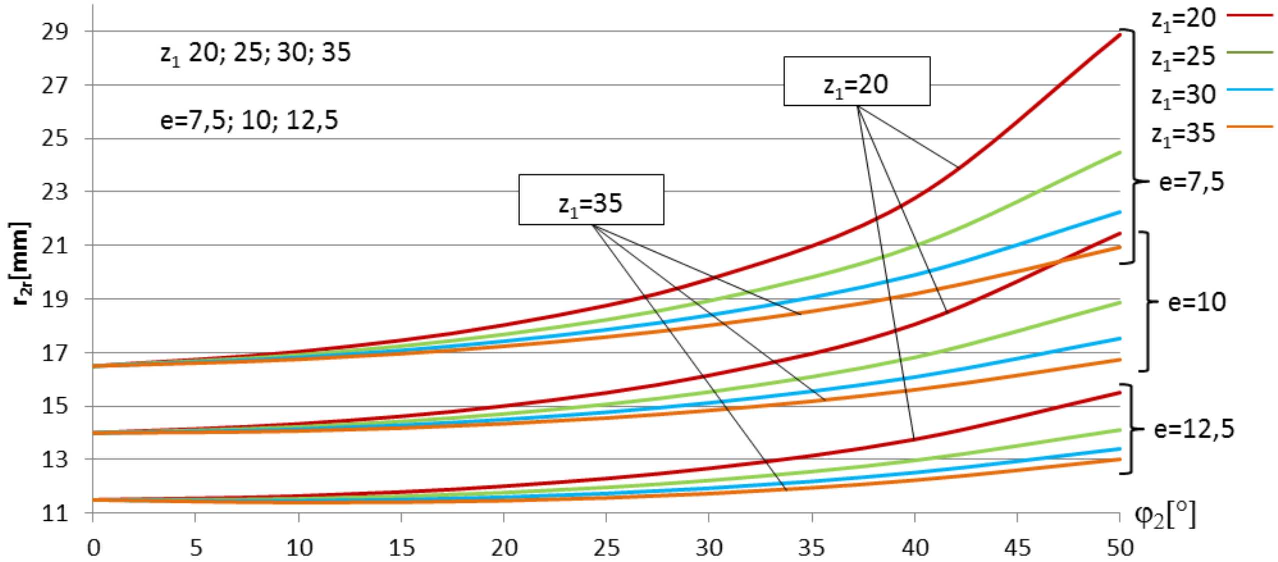


Figure 5. The variation of the sector radius $r_{2r}(\varphi_2)$

Figure 5 – function $r_{2r}(\varphi_2)$ gives response to the requirement for obtaining a variable ratio with up to 30%, both the number of teeth z_1 and eccentricity "e" having significant influence. The icrease the radius r_{2r} should be bigger for smaller values of the number of the teeth z_1.

By eliminating variations that have previously been considered unacceptable ($e = 12.5mm$ and $e = 7.5 / z_1 = 20$), ratio ranges is obtained (see tab. 1):

$$\Gamma = r_{2\max} / r_{2r}^0 \cong \frac{24.48; 22.25; 20.94}{16.5} = 1{,}48;\ 1{,}34; 1.26 \text{ for } e = 7.5mm,$$

$$\Gamma = r_{2\max} / r_{2r}^0 \cong \frac{21.46;\ 18.87;\ 17.53; 16.74}{14} = 1{,}53;\ 1{,}34;\ 1{,}25;\ 1{,}19 \text{ for } e = 20mm.$$

Finally, we can conclude that pairs $z_1 / e = 30 / 7.5$ and $z_1 / e = 25 / 10$ that obtained $\Gamma = 1{,}34$ and respectively $\Gamma = 1{,}34$, are closest to the desired purpose. Considering the resulted clearances ($j_{n\max} = 0{,}66$, respectively $j_{n\max} = 0{,}42$), the optimal solution is: $z_1 = 25$ and $e = 20mm$ (or closed values).

Figure 6 shows the gear considered to be optimal ($z_1 = 25, e = 20mm, z_2 = 24$, $m_0 = 2mm$) in the maximum mashing position (end of run $S_1 = 27mm$, $\varphi_2 = 50°$). With the change of the direction of movement, a small lost motion will appear for the cancellation of the clearance between the teeth.

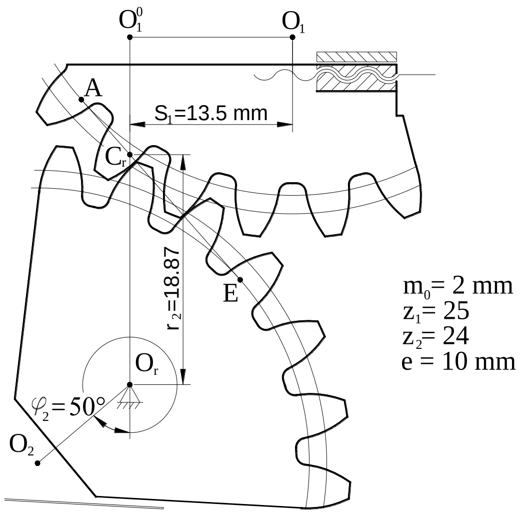


Figure 6. Visualization of the gear considered as opimum

5. CONCLUSION

From the graphs presented above, it is inferred that the number of teeth z_1 and z_2 should be fairly close, namely the translational wheel and the geared segment to have approximately equal radii, and eccentricity "e" to be somewhat less than half their radius.

The gear translational wheel - eccentrically geared segment is characterized by a nonlinear motion function $\varphi_2(S_1)$, respectively variable geometric - functional parameters, 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 $(\varphi_2 < 50°,\ \Gamma < 1{,}35)$, is corresponding to the functional requirements imposed.

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