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THE DESIGN OF THE PINION-RACK STEERING GEAR WITH VARIABLE RATIO

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ABSTRACT

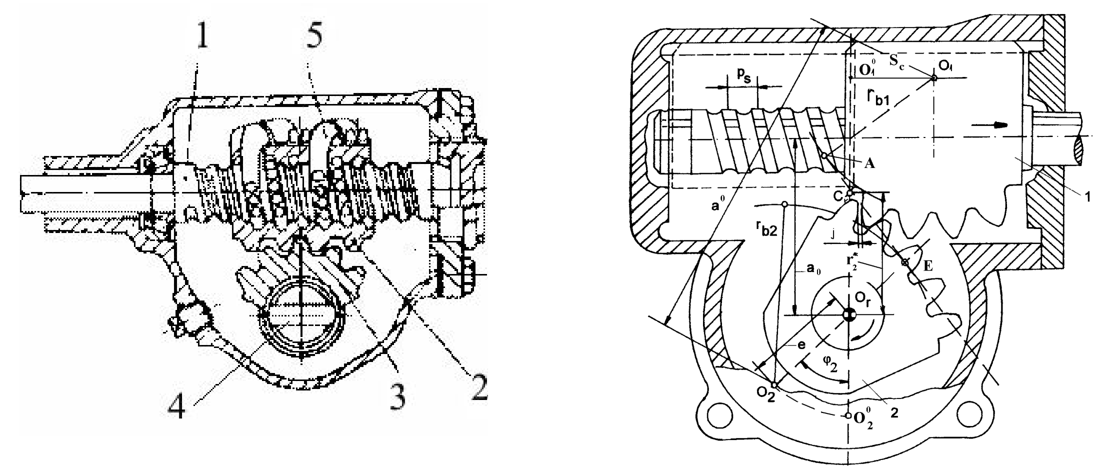
We consider the constructive solution of a gear box for cars, type bolt nut on bile's - segment gear. For getting a variable reduction ratio, we consider the nut / gear rack will be bent and the segment gear will be cam gear articulated.

There are obtained the kinematic and the geometrical parameters of the rack gearing with the advanced variable reduction ratio and these are optimal correlations in the functional conditions for the gear box.

Keywords: variable ratio, variable pith, curved rack, gear.

1. PRESENTATION OF THE STEERING BOX

Within the transmission from the vehicle's wheel to the steering wheels, the steering box has an essential role in multiplying the force. Many reasons [2] lead to the justification of the steering box solution whose transmission ratio – i_c - is variable, with an increasing factor from the straight line trajectory to the maximal turning positions, so that the driver should not perform an increased effort in curves or in parking lots.



a) b)

Figure 1. Gear rack – sector from steering box

An usual type of steering box is the one with bolt and nut on balls – geared segment, where the nut is provided on its exterior side with flanks making it a rack thus achieving the rack – wheel gear (fig.1a) – with evident constant ratio:

$$i_c = \frac{\omega_S}{\omega_r} = \frac{\omega_S}{v_1} \cdot \frac{v_1}{\omega_2} = \frac{2\pi}{p_S} \cdot r_2. \quad \ldots(1)$$

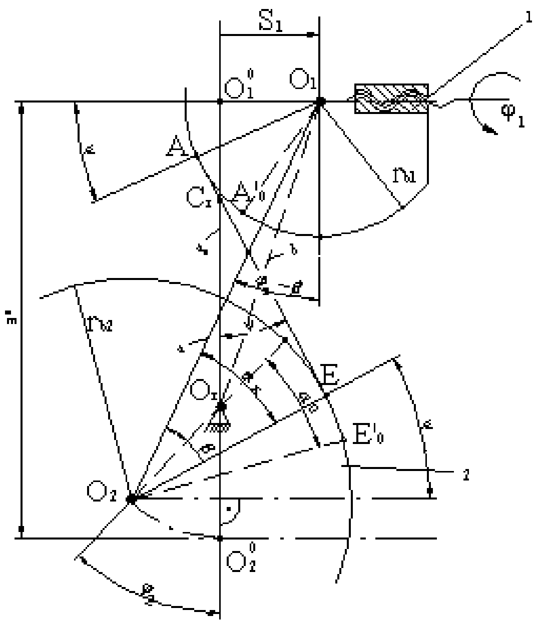


Figure 2. The kinematic scheme

p_s – pitch of screw and r_2 – radius of the geared segment.

In order to obtain a variable ratio, r_2 radius should be variable, respectively the gearing pole and implicitly the gearing angle to be variable.

Such a solution is possible if the rack is given a curved shape, transforming it into a shifting wheel, while the sector should be eccentrically articulated, as shown in figure 1b. [3, 4].

Such a gear pair has the initial position corresponding to the straight motion of the vehicle, according to figure 2, the rotation center of the sector being O_r, different from its geometric center O_2.

For the S_c movement of the rack's nut (from O_{10} in O_1), the sector will turn with the angle φ_2 around O_r, the O_{20} center reaching in O_2, respectively the C_{r0} gear pole takes the position Cr, while the gear angle α_{w0} takes the value α_{wx} *fig.2(

2. GEOMETRIC – KINEMATIC PARAMETERS

The geometric parameters of the gear/ mechanism are: $a^0 = O_1^0 O_2^0$ - distance between the wheels' centers in initial position; a= O_1O_2 current distance between axes; a_0= reference distance between the steering box axes; e= OrO2 eccentricity of the geared segment; $\alpha w_0/\alpha w_x$= gearing angles;rb_1/rb_2= radius of the basic circles; r_1/r_2 = radius of the dividing circles of the flanks;$r_{w1}=O_1Cr$ and $r_{w2}=O_2Cr$ – radius of the gear rolling circles; r_2^* = OrCr = current rotation radius of the sector; S_c = linearly movement of the rack (shifting wheel) - independent variable; φ_2 – current rotation angle of the sector; j = j_b – the allowance between the flanks of the teeth on the base circle (implicitly on the gearing line).

According to the deduction made by the authors [1, 3], the expressions of the motion function and of the other variable parameters are given by:

$$f(S_c, \varphi_2) = a \sin \alpha w_x + r_{b_2}(\alpha w_0 + \beta - \alpha w_x) - a^0 \sin \alpha w_0 - r_{b_1}(\alpha w_x + \varphi_2 - \beta - \alpha w_0), \quad \ldots(2)$$

$$a = r_{w_1} + r_{w_2} = \frac{\cos \alpha w_0}{\cos \alpha w_x} \cdot \frac{m}{2}(z_1 + z_2),\ p_w = p\frac{\cos \alpha w_0}{\cos \alpha w_x}, \quad \ldots(3)$$

$$\alpha w_x = \arccos \frac{r_{b_1} + r_{b_2}}{a},\ \alpha w_0 = \arccos \frac{r_{b_1} + r_{b_2}}{a^0}, \quad \ldots(4)$$

$$j_b = j_w \cos \alpha w_x,\quad j_w = p_w - S_{w_1} - S_{w_2}, \quad \ldots(5)$$

$$S_{w_{1,2}} = dw_{1,2}\left(\frac{S_{1,2}}{d_{1,2}} + \operatorname{inv}\alpha w_0 - \operatorname{inv}\alpha w_x\right),\ S_{1,2} = m\left(\frac{\pi}{2} + 2X_{1,2}\operatorname{tg}\alpha w_0\right), \quad \ldots(6)$$

$$i_a = \frac{v_1}{\omega_2} = r_2^* = a^0 - e - r_{b_1} \sin \psi - \left(r_{b_1} \cos \psi - S_c\right)\cos \psi, \quad \ldots(7)$$

$$\psi = 90^\circ - \alpha w_S - \varphi_2 + \beta, \quad \ldots(8)$$

in which $S_{1,2}/S_{w1,2}$ represent the thickness of the teeth on the dividing circles, respectively rolling, with $x_{1,2}$ - the profile movement coefficients.

3. NUMERICAL SIMULATIONS FOR OPTIMIZING THE GEAR

The optimization of the gear curved rack – eccentric sector will be made based on the following criteria:

- achieving the prescribed deflection angles of the wheel (aprox. φ_2= 50º);
- achieving the ratio variation range $\Gamma=r^*_{2max}/r^0_2$ (aprox. Γ=1,3);
- avoiding negative allowances between teeth and acceptable in extreme positions;
- achieving minimal differences $a-a^0$ in extreme positions.

The calculation program – developed in Ms Excel - which allows the determination and graphical representation of the foreseen variations, led us to the conclusion that the sizes of the two wheels must be very closed, so that for the numerical simulations there was decided that the sector to have the number of teeth Z_2= 24, while the rack wheel Z_1= 20, 24, 30 teeth.

A second extremely important construction parameter is the size of *e,* eccentricity, which must lead to acceptable allowances, in the given example (fig. 3), the eccentricities e=31-38 mm being considered good, and therefore for testing there are chosen the values for *e* as follows: e= 31, 33, 35 mm.

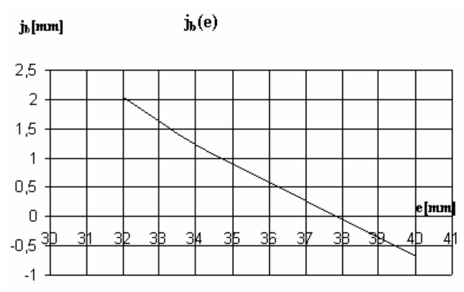


Figure 3. The dependence j_b - e

Out of the performed simulations, there are presented below on the parameters set $Z_1=Z_2$=24 and e=31, 33, 35 mm, the graphs for:

- motion function $S_c(\varphi_2)$ - figure 4a;
- variation of distance between axes $(a-a^0)(\varphi_2)$ - figure 4b;
- variation of allowance $j_b(\varphi_2)$ - figure 5a;
- variation of the gearing angle $\alpha_{wx}(\varphi_2)$ - figure 5b.

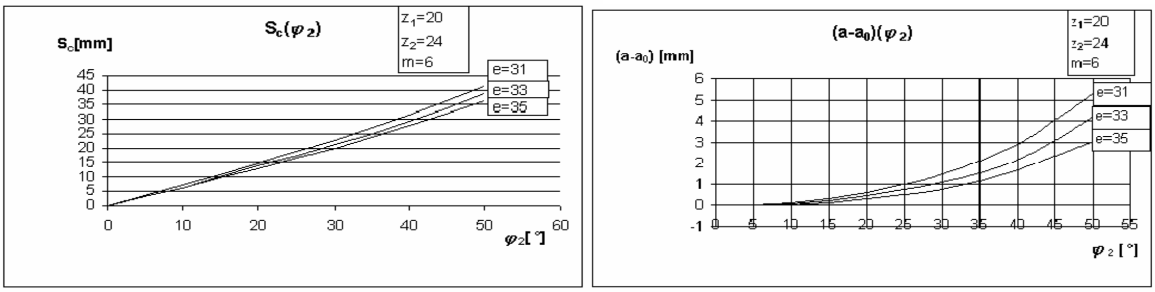


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Figure 4. The motion function and variation of distance axes

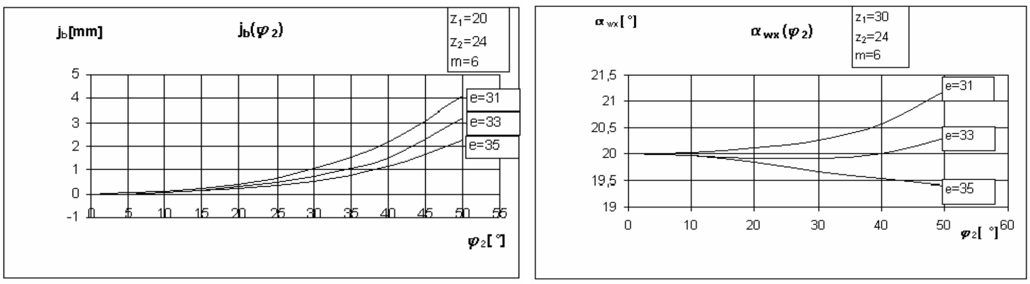

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Figure 5. The variation of allowance end gearing angle

The results indicate that the variations of the functions are kept in the acceptable range, decreasing with the eccentricity increasing, while the obtained domain can be considered as optimal.

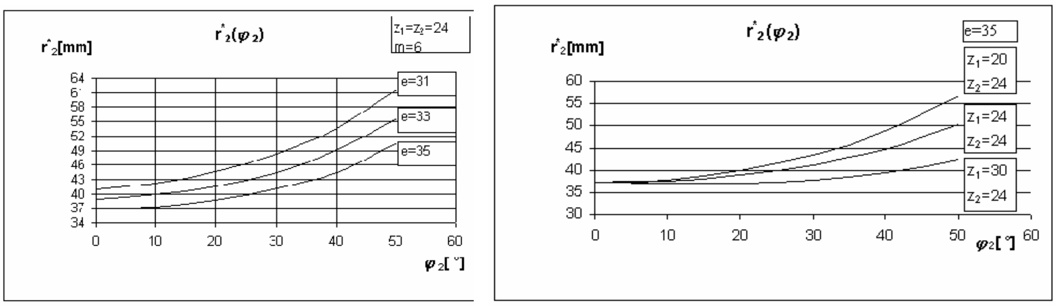

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Figure 6. The variation radius parameter e and z

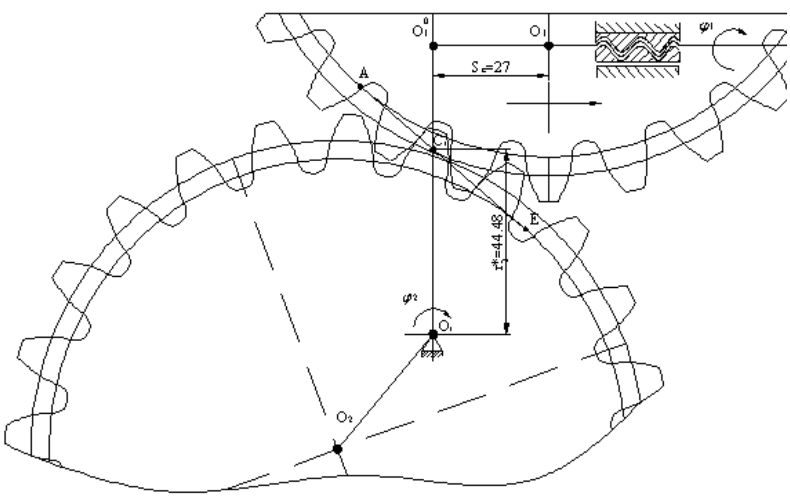

Figure 7. The gear limit position

The variation of the rotation radius $r_2^*(\varphi_2)$ for e = 31, 33, 35 mm is given in figure 6a, and for different values of Z_1= 20, 24, 30 is given in figure 6b, resulting also the variation of the Γ range of the transmission ratio.

It is noticed that the variation of the radius, and implicitly of the transmission ratio, increases with the rotation angle φ_2, but decreases with eccentricity *e*, decreasing also with the increase of the number of teeth belonging to the rack wheel.

In figure 7, there is presented the mechanism of the box bolt – nut/ shifting wheel – eccentric sector, with the gear at the imposed gear limit, S_c = 35 mm and φ_2 = 50°.(e=35 mm, Z_1=30, Z_2=24)

There can be concluded that:

- it was proven the functionality of a gear with shifting wheel - eccentric sector for the steering boxes of the vehicles;
- the obtained relations for the geometric and kinematics parameters achieve a correct description of the box functionality;
- the method and optimizing program lead to the obtaining of functional/ geometric acceptable parameters;
- the proposed solution can be considered for product implementation, since there are no technological difficulties.

4. REFERENCES

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