USING OF CYCLOIDAL GEAR IN CAR SELFSTARTER

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Abstract

Nowadays the car selfstarters of new generation differ from those used so far, which are serial electromotors. The biggest manufacturers of these selfstarters are: Bosch, Valeo, Delco and Hitachi. Their main characteristic property is using of magnetic field from sintered magnets for excitation of selfstarter. The other charecteristic property is using of planetary gear. During initial researches the rub off of teeth of ring gear has been often met in investigated selfstarters. This observation motivated the authors of the paper to investigate possibility of replacing of planetary gear by cycloidal gear in selfstarter. The influence of replacement of planetary gear by cycloidal gear upon operating conditions, such as loading and relative velocity in selfstarter bearing and for teeth of pinion - flying wheel gear, is considered in this paper. Also the loading of elements of cycloidal gear has been calculated. It is difficult to find characteristics of such selfstarters in known literature - so it is necessary to obtain such characteristic by measurement or calculation approximated characteristic, for example, with FEM method. In the paper the approximated characteristic has been obtained for sample Bosch selfstarter. The knowledge of geometry of selfstarter, material properties and of approximated characteristic of selfstarter allows to obtain value of loading and relative velocities of mating parts with use of elaborated FEM model. For calculation the helping mathematical models have been used. The results of computation and the picture of wear of any selfstarter parts are presented in the paper.

Keywords: *combustion engines, car selfstarter, cycloidal gear, planetary gear, bearing, wear. loading*

1. Introduction

Nowadays the car selfstarters of new generation are used in inner combustion engines. The biggest manufacturers of these selfstarters are: Bosch, Valeo, Delco and Hitachi. Their main characteristic property is using of magnetic field from sintered magnets for excitation of selfstarter. The second charecteristic property is using of planetary gear. The sun gear is made of steel and mates with planets, which are made of steel also. The planets mate with ring gear made from steel or polimer composite. During initial researches made in one of diagnostic station, it has been observed that the rub off of teeth of ring gear has been often met in investigated selfstarters. This observation motivated the authors of the paper to investigate possibility of replacing of planetary gear by cycloidal gear in selfstarter. The aim of the paper is to research on influence of such replacement upon loading of elements in selfstarter, when the most of elements are the same. The scheme of the such selfstarter with planetary gear and with cyclo gear is presented in Fig. 1.

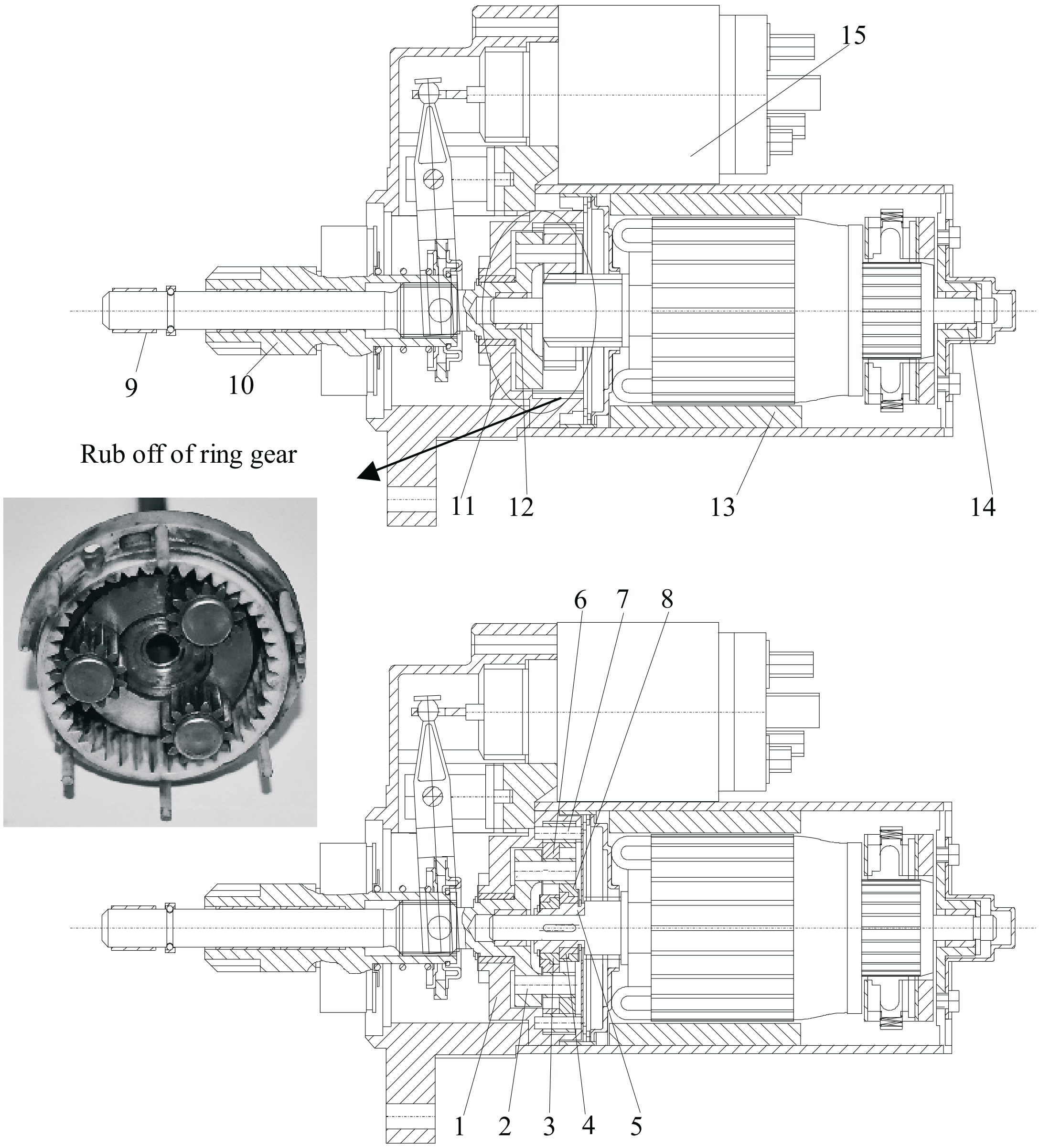

Fig. 1. Selfstarters with planetary gear and with cycloidal gear 1 - housing of cycloidal gear , 2 - the inner set of rolling bolts in straight-line mechanism, 3, 4 - central bearings of planetary wheels, 5 - eccentric ring, 6, 8 - planetary wheels, 7 - the outer set of rolling bolts in straight-line mechanism, 9, 12, 14 - bearing of selfstarter, 10 - pinion, 11 - housing of planetary gear, 13 - permanent magnets

2. Cycloidal gear in selfstarter

Cycloidal gear (Fig. 2) is compound of three kinematic couples: the central cylindrical sliding bearing, the set of 8 rolling bolts in straight-line mechanism and the special meshing of the planetary wheel. The gear ratio is $i = 7$. The meshing is created by 2 planetary wheels, displaced against each other with π angle, mating with stationary set of rollers. Planetary wheels have external teeth with the shape of equidistant of curtate epicycloid [1, 2]. Drive moment $M_h = M_{wn}$ is transmitted on planetary wheel from high-speed shaft at speed n_h by means of eccentric. The torque M_{wn} is described in the next part of the article. Straight-line mechanism is used for transmission the torque moment M_c from planetary wheel onto output shaft. That mechanism consists of 6 bolts rolling away internal holes of planetary wheel. The bolts are fixed in the disk connected with output shaft. The third of moments, M_2, loads rollers of mating wheel.

Dimensions of planetary gear and the whole gear are determined by three parameters (Fig. 2): eccentric $e = 2$ mm, curtailment coefficient of epicycloid $m_e = e \cdot z_k / r = 0.53$, offset of epicycloid equidistant equal to radius of the roller of mating wheel D_e / 2 = 3 mm.

Cycloidal gear should have relatively low spacing radius r of the rollers of mating wheel (Fig. 2). The value of radius r, at required transmission ratio, power and rotational speed of drive shaft, is the result of proper combination of values e, m_e and D_e - equation (1):

$$r = e \cdot (i+1) / m_e . \tag{1}$$

The loading of elements of cycloid gear has been calculated numerically by program RCYCLO [2]. The maximal contact pressure p_{imax} - in Hertz contact model - has been calculated by program ZCYCLO [2]. The value and direction of loading forces $F_2, F_3, F_4, Q_1, Q_2, Q_3, Q_4$, reaction force R and p_{imax} and geometrical parameters of cycloidal gear are presented in Fig. 2.

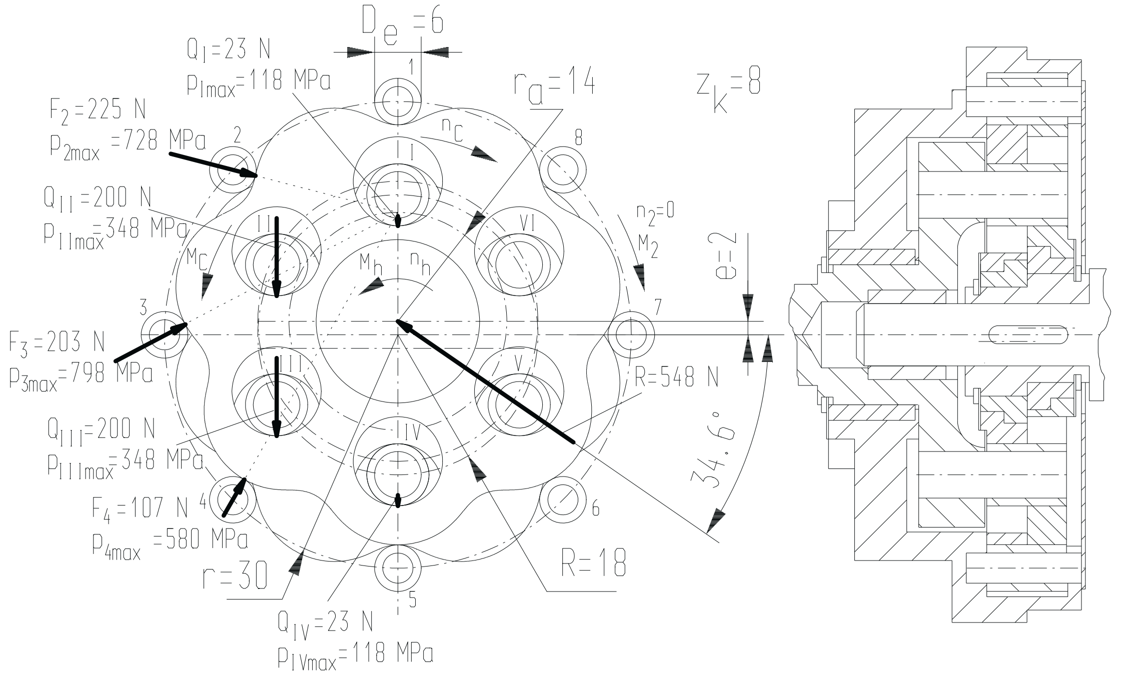

Fig. 2. The cycloidal gear in selfstarter

Because of great values of p_{imax}.the elements of such cycloidal gear should be made of steel.

3. Characteristics of selfstarter

In the literature it is difficult to find characteristics for selfstarters of new generation - so it is necessary to get it from measurement or to obtain approximated characteristics from computation with help of Finite Element Method. In the article the approximated characteristics of sample Bosch selfstarter is presented. During operating of such selfstarter the rub of frictional joints take place, especially for bearing. The value of rub of depends on operating conditions, in which mate movable elements of selfstarter, especially on loading, sliding velocity and temperature. Value of temperature can be assumed as constant and smaller than 340 K [3]. The knowledge of geometry, material properties, obtained for example from literature, and of approximated characteristics of selfstarter allows to obtain values of loading and sliding velocities of mating elements with use of elaborated model. The model has been obtained with Finite Element Method.

The value of torque acting on the rotor of selfstarter has been calculated from equation (1) [3]. It has been assumed, that value of battery resistance equal R_B=0.01 Ω, the decreasing of voltage on the brashes and commutator equals ΔU_S=1.5 V, the number of bars for rotor coil is equal N=56, value of battery voltage equals U_B = 12 V [3].

$$M_w = \frac{1}{2 \cdot \pi (R_a + R_B)} [(U_B - \Delta U_S) \cdot N \cdot \Phi]. \tag{1}$$

The value of torque for fully stopped selfstarter has been calculated from equation (2) [3]:

$$M_{wn} = \frac{\eta_{mn}}{2 \cdot \pi \cdot (R_a + R_B)} \cdot \left[(U_B - \Delta U_S) N\Phi - \frac{n_n}{60} \cdot (N\Phi)^2 \right]. \tag{2}$$

It has been assumed, that value of mechanical efficiency, for maximal (nominal) power conditions has been equal $\eta_{nm} = 0.8$. The value of electric resistance of rotor coil has been equal $R_a = 0.01\ \Omega$. The area of cross section of permanent magnet has been equal $S_M = 0.001\ m^2$ [3]

The value of magnetic flux has been calculated from equation (3) [3].

The calculated value of magnetic flux Φ has been equal 0.00044 Tm^2.

The obtained approximated operating characteristics of selfstarter has been shown in the Fig. 3.

$$\Phi = B_M \cdot S_M . \tag{3}$$

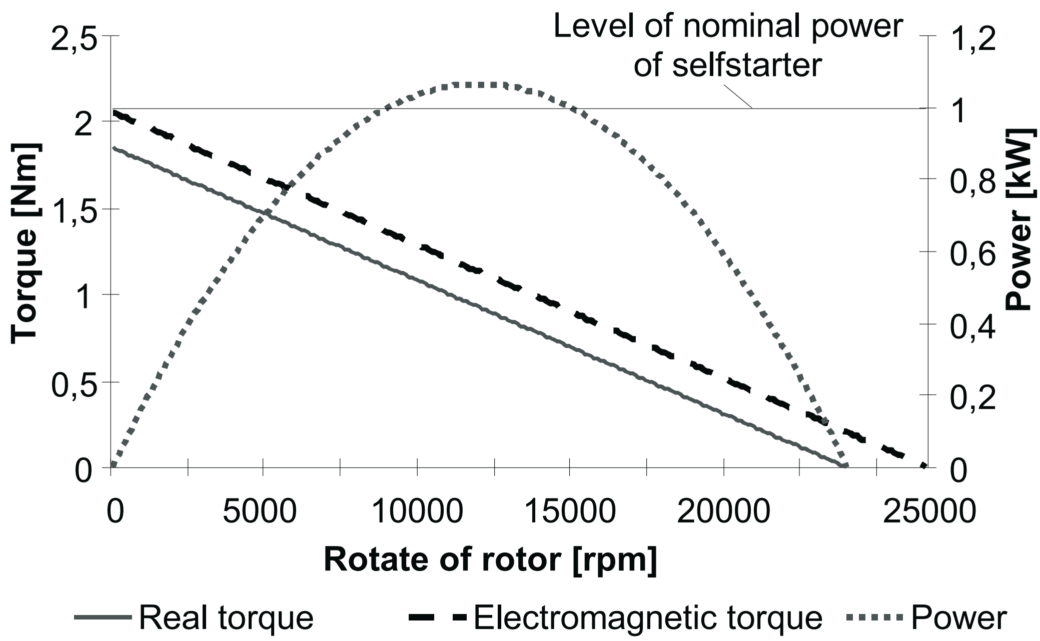

Fig. 3. The approximated operating characteristic of selfstarter

The mean value of magnetic flux density B can be estimate with use of Finite Element Method.

The values of loading F acting on rotor coil of selfstarter, versus time have been obtained analytically, with help of Finite Element Method, either [3]. As has been mentioned the Fig. 1 contains geometrical model of selfstarter. The rotor of this selfstarter together with the rotor coil moves in magnetic field of magnetic circuit. Such circuit is connect of rotor, air gaps, permanent magnets and body of selfstarter. The used ferrit magnets Fe have got near linear shape of demagnetization curve $B(H)$ [3, 4]. Their residual magnetic flux density has been equal $B_r = 0.4$ T; their coercive magnetic field intensity has been equal $H_c = 220$ kA/m.

The body of selfstarter has been made of stainless steel, the characteristics *B(H)* for such steel has been given by manufacturer [3, 4]. The rotor of selfstarter has been made of the packet of shells and this packet has been situated on the mandrel. It has been assumed, that rotor shells and the spigot have been made of steel, which characteristics is very similar to the characteristic of material for selfstarter body. The rotor coil has been made of copper.

The calculation of gradient of magnetic flux density in the magnetic field originated from permanent magnets has been made for cross area of selfstarter, in region of operating length of rotor coil (Fig. 1.). To simplify calculation, the influence of rotor coil movement relative to permanent magnets and selfstarter body upon generation of electromotive forces in rotor coil has been neglected. The mentioned electromotive forces, in real set, influence upon values of current in the coil and can change these values only about a few percents.

The influence of eddy currents generation in selfstarter body and rotor upon electromagnetic flux density in magnetic circuit connected of rotor, radial air gaps, selfstarter body and permanent magnets has been neglected, either.

For the calculations the finite element of type PLANE 53 [3] has been used.

The basic degrees of freedom have been values of the vector magnetic potential in the Z axis direction. The loading of finite element, with the uniform loading value in each node, has been current density.

The value of such current density has been calculated as the ratio of current in coil and value of cross area of coil winding.

The additional degrees of freedom have been magnetic flux density *B* and magnetic field intensity *H*.

The mesh of finite elements (Fig. 4) has been generated automatically by commercial program ANSYS [4].

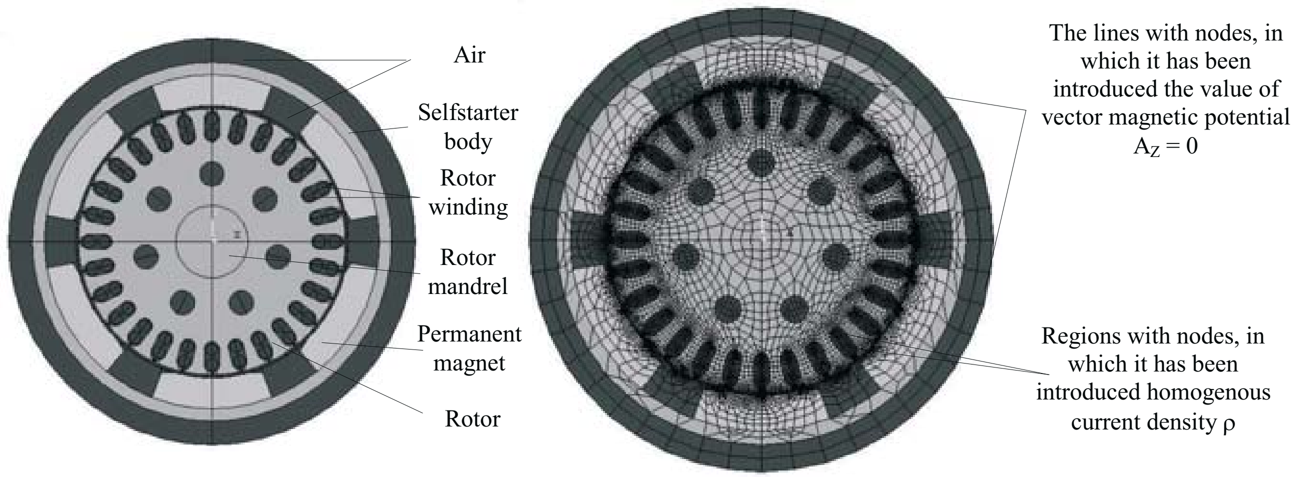

Fig. 4. Thegeometry for crossing section of selfstarter, the mesh of finite elements and used boundary conditions

It has been used the following boundary conditions (Fig. 4):

- it has been introduced the same values of current density ρ, in the nodes of finite elements, positioned in the crossing area of rotor coil winding,
- it has been introduced the values of the vector magnetic potential $A_Z = 0$, in the nodes of finite elements positioned on the outer lines limiting the air region.

The nodal values of vector magnetic potential *A* have been calculated numerically, in the region of each finite element. The values of output degrees of free have been calculated by program either.

The gradient of magnetic flux density in cross segment of selfstarter is presented in Fig. 5.

The mean value of magnetic flux density in permanent magnets region has been equal 0.44 T.

The passage of current in rotor coil (the passage by 56 bars) has resulted in the change of mean value of magnetic flux density smaller than 1% and, because of it, such change can be omitted.

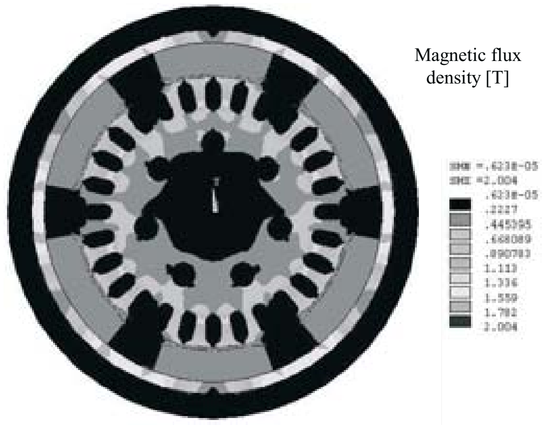

Fig. 5. The magnetic flux density in crossing segment of selfstarter

4. Loading of bearing and teeth of pinion in selfstarter

The scheme of loading of bearing and teeth of pinion in selfstarter is presented in Fig. 6.

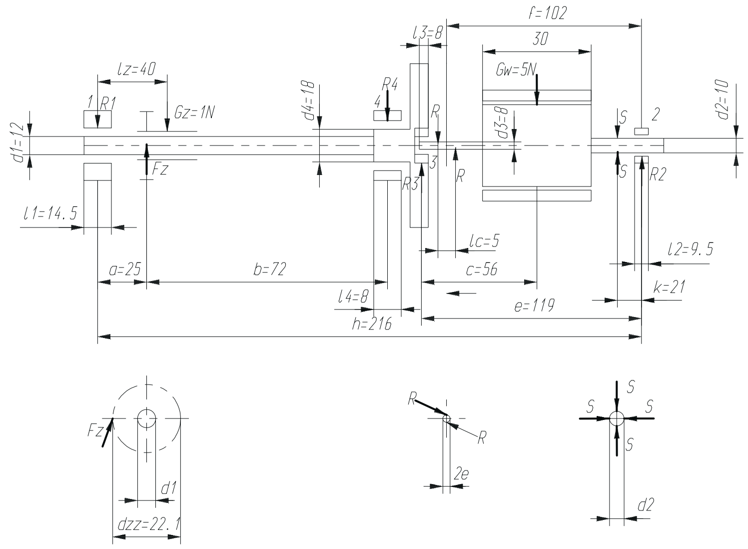

Fig. 6. The scheme of loading for bearing and teeth of pinion in selfstarter and geometrical parameters of selfstarter

The values of sliding velocity in bearing have been calculated from equation (4):

$$v_i = \pi \cdot n_s \cdot i_z \cdot d_c . \tag{4}$$

The calculated values of sliding velocity v_i in bearing are presented in Tab. 1.

The value of force loading the teeth of pinion can be calculated from equation (5) [3]:

$$F_Z = 2 \cdot M_{wn} \cdot i_c / d_{zz} , \tag{5}$$

where:

n_s - rotate of crank shaft of combustion engine,

$n_s = 175$ rpm when the pinion is turned off.

The values of v_i are smaller than limit value equal 15 m/s for such sintered self lubricated bearing.

The value of force loading the teeth of pinion can be calculated from equation (6) [3]:

$$F_Z = 2 \cdot M_{wn} \cdot i_c / d_{zz} . \tag{6}$$

Tab. 1. Loading of bearing in selstarter

i - numer of bearing	Total ratio gear i_z	Radius of pivot r_{ci} / radius of bearing busch r_{1i} [mm]	Length of bearing l_i [mm]	v_i for n_s=175 rpm and i_c=7 for cycloidal gear [m/s]	v_i for n_s=175 rpm and i_c=4.6 for planetary gear [m/s]
1	15	6/6.05	14.5	1.65	1.65
2	7·15	5/5.035	9.5	9.61	6.32
3	7·15-15	4/4.025	8	6.03	3.96
4	15	9/9.025	8	2.47	2.47

For selfstarter with cycloidal gear ($i_c = 7$) the maximum value of F_{Zmax} is equal 1140 N for stopped selfstarter, when $M_{wn} = 1.8$ Nm. It is greater of about 52% than the maximum value of $F_{Zmax} = 750$ N for selfstarter with planetary gear ($i_c = 4.6$).

To obtain contact pressure gradients in bearing, the following assumption have been used:

- pivot mates with bearing bush in the conditions of their parallel axes,
- only elastic strains in material of pivot and of bearing bush take place,
- diameter of inner cylindrical surface equals hole diameter of bearing bush,
- gradient of contact pressure p_i and contact area of pivot and bearing bush, determined by angle $2\ \varphi$, are calculated from Hertz model for cylinder pushed into inner cylindrical surface.

The scheme of contact pressure gradient in bearing is presented in Fig. 7.

The mean value of contact pressure p_i in i - bearing has been calculated from equation (7) [3]

$$p_i = R_i \Bigg/ \left[2 \cdot r_1 \cdot l_i \cdot \arcsin\frac{0.8}{r_1} \cdot \sqrt{\frac{R_i}{l_i} \cdot \frac{r_{i1} \cdot r_{ci}}{r_{1i} - r_{ci}} \cdot \left(\frac{1-\nu_{ci}^2}{E_{ci}} + \frac{1-\nu_{1i}^2}{E_{1i}} \right)_i} \right]. \tag{7}$$

The maximal value of contact pressure p_{imax} has been calculated from equation (8) [3].

$$p_{i\max} = 0.798 \cdot \sqrt{\frac{R_i}{l_i} \cdot \frac{r_{1i} - r_{ci}}{r_{1i} \cdot r_{ci}} \cdot \left(\frac{1-\nu_{ci}^2}{E_{ci}} + \frac{1-\nu_{1i}^2}{E_{1i}} \right)^{-1}}\ . \tag{8}$$

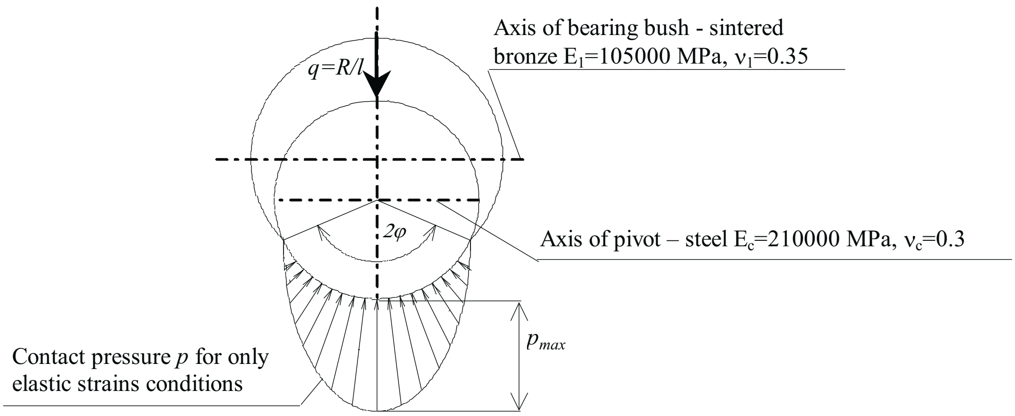

Fig. 7. Contact pressure gradient in bearing - parallel axes of pivot and bearing bush

The value of reaction force have been calculated from set of equations (9) [3]:

Calculated values of reaction force and contact pressure are presented in Tab. 2.

$$
\begin{cases}
R_1 = \sqrt{\begin{array}{l}\{[F_z \cdot \cos 20° \cdot b - G_z \cdot (l_z - a) - (R \cdot \sin 34.6° \cdot l_c + G \cdot (e-c)) \cdot (h-e)/e]/(a+b)\}^2 \\ + \{[F_z \sin 20° \cdot b - (R \cdot \cos 34.6° \cdot l_c / e) \cdot (e-c)]/(a+b)\}^2\end{array}} \\
R_2 = \sqrt{[(-G \cdot c - R \cdot \sin 34.6° \cdot l_c)/e]^2 + [R \cdot \cos 34.6° \cdot l_c / e]^2} \\
R_3 = \sqrt{[(G \cdot (e-c) + R \cdot \sin 34.6° \cdot l_c)/e]^2 + [R \cdot \cos 34.6° \cdot l_c / e]^2} \\
R_4 = \sqrt{\begin{array}{l}\{[F_z \cdot \cos 20° \cdot a - G_z \cdot l_z - (R \cdot \sin 34.6° \cdot l_c + G \cdot (e-c)) \cdot (h-e)/e]/(a+b)\}^2 \\ + \{[F_z \cdot \sin 20° \cdot a - (R \cdot \cos 34.6° \cdot l_c / e) \cdot (h-e)]/(a+b)\}^2\end{array}}
\end{cases}
\tag{9}
$$

Tab. 2. Reaction force and contact pressure for bearing in selstarters with planetary and cycloidal gear

i - numer of bearing	Reaction force R_i planetary gear [N]	Reaction force R_i cyclo gear [N]	Mean contact pressure p_i planetary gear [MPa]	Mean contact pressure p_i cyclo gear [MPa]	Max contact pressure p_{imax} planetary gear [MPa]	Max contact pressure p_{imax} cyclo gear [MPa]
1	554	828	40	51	49	63
2	2.4	24.4	3.2	4.2	10	13.4
3	2.6	24.6	4	5	12	15.5
4	190	274	14	19	18	23

The loading of bearing (i=1 or 4) for pinion mandrel is greater about 50%, but loading of rotor bearing (i = 2 or 3) is 10 times greater in case of cycloidal gear relative to one of planetary gear. Contact pressure values are greater than recommended limit value 20 MPa, but are still smaller than breaking stress in radial crushing 150-200 MPa - for such bearing.

5. Conclusions

1. The greater value of torque on pinion is one of advantage of replacement planetary gear by cyclo gear. It allows to start cold engine easier.
2. Such replacement can decrease durability of elements of selfstarter. The remedy for it can be using of magnets of smaller value of H_c resulting in smaller electromagnetic torque M_{wn}.
3. The values of slide velocity in bearings of pinion mandrel in selfstarter with cycloidal gear are the same as for one with planetary gear. For rotor bearing such values are 50-100% greater.
4. The reaction forces and contact pressure in bearing for selfstarter with cycloidal gear are greater than for one with planetary gear. The loading of bearing for pinion mandrel is greater about 50%, but loading of bearing for rotor is 10 times greater - so it can be dangerous.
5. Elements of cyclo gear should be made of steel, as contact pressure value can equal 800 MPa.

References

[1] Chmurawa, M., *Obiegowe przekładnie cykloidalne z modyfikacją zazębienia,* Mechanika, Vol. 140, Zeszyty Naukowe Politechniki Śląskiej, Gliwice 2002.

[2] Warda, B., *Wyznaczanie sił międzyzębnych w obiegowej przekładni cykloidalnej*, XXIII Sympozjon PKM, Materiały Vol. 2, pp. 436-444, Rzeszów - Przemyśl 2007.

[3] Siczek, K., The operational conditions and wear of elements in selfstarter of new generation for combustion engine, Combustion Engines, pp. 212-226, SC1 2007.

[4] Release 11.0 Documentation for ANSYS.

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