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FLOW RATE OF GEAR PUMPS WITH CYCLOID MESHING

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Diesel engine and gearbox lubrication systems are normally supplied by gear pumps which are independently driven for large slow speed engines and stand-by duties but usually shaft driven for medium and high speed engines. Gear pumps are also used for fuel and oil transfer, boiler combustion systems and other duties. Most often they are used in the field of small flow rate (before 10 m^3/h) and medium and high pressure. The pump parameters are almost completely determined by the gear train parameters. Analysis of the known methods of increasing the gear pump flow rate showed that today almost all known methods should be classified as extensive. Increased flow rate is provided by a proportional increase in the geometric dimensions of the gears and the pump.

Advances in the gear manufacturing technology have led to decrease in the amount of gears, manufactured through cutting. Modern machinery utilizes gears manufactured through casting, stamping and sintering. Thus, one may avoid the otherwise obligatory link between the tooth profile and the parameters of the cutting tool – rack cutter, gear-shaper cutter and gear hobbing cutter. The use of milling machines with CNC (computer numerical control) enables the manufacturing of a random tooth profile. Therefore, the technological limitations that have led to the general use of the involute tooth system vanish [1-3].

In this regard, the topic of using epicycloids and hypocycloids for tooth profiling is brought forth. Confirmation is obtained for increase of 20% of the fluid volume in the tooth space of the gear pump with combined involute-cycloid tooth profile compared to involute tooth profile [4-7].

A characteristic defect of the gear pump is an increase in the diametrical clearance between the housing and the tops of the gear teeth due to their wear. During the repair, bore the pump housing and install new gears with an increased outer diameter. Thus, when repairing a pump, it is possible to replace the involute pinions on the cycloidal profile gears.

It is necessary to develop criteria for cycloidal gears which installing instead of involute ones. And also give an estimate of the gains in pump performance that can be achieved.

The main parameters of cycloidal tooth system are: ratio of radii supporting and base circle r_i/R_i; number of gear teeth z_i; fillet radius of dedendum flank ρ_f; base circle radius R; supporting radius of the circle r; radial clearance c (Fig.1). The cycloidal profile of a tooth has the addendum contoured by an epicycloid and the dedendum outlined by a hypocycloid. Thus, the addendum has a convex surface, and the dedendum has a concave surface. Due to the concave surface of the dedendum, the area of the t tooth space in the cycloidal gearing is always greater than in the similar involute. The tooth space area of the cycloidal gearing is maximum with the ratio of the auxiliary circle radius to the pitch circle radius is equal 0,5. In this case, the hypocycloid degenerates into a straight line [8-9].

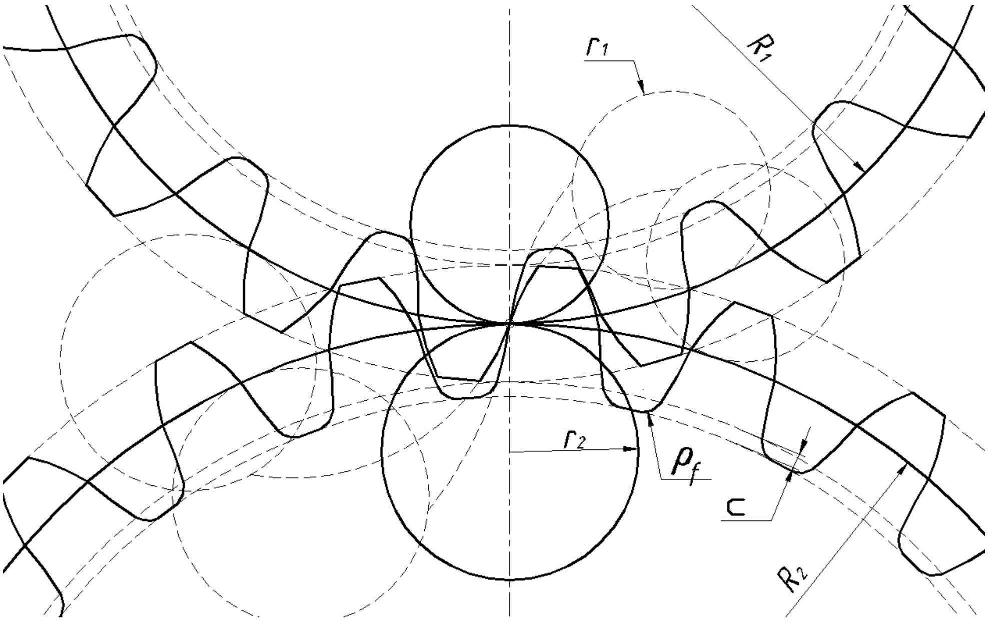

Fig.1 - The profiling scheme of cycloidal tooth system

A method for estimating the area of the tooth space was developed. With the help of the developed program "Gearing", the process of cutting teeth is modelled. The found profile of the tooth space is transmitted to the software program Mechanical Desktop 6.0. The configuration of the tooth space was approximated by a closed broken line and the area was determined using the command "AREA". The dependence of the tooth space area on the ratio of the circles radii - a dedendum auxiliary and a pitch r_2/R_2, was found (Fig. 2). The tooth space area at $r_2/R_2 = 0.2$ is

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taken as 100%, the increase in area with an increase in the ratio is indicated as a percentage of the base. The increase in pump performance by increasing the ratio r_2/R_2 can be 7.5%.

The influence of the circles radii ratio r_1/R_2 - an addendum auxiliary and a pitch was also investigated. The effect of changing the radius of the auxiliary circle on the tooth space area is much less. The increase in pump flow rate due to an increase in the ratio r_1/R_2 is less than 2%. Usually, the radii ratio of the addendum auxiliary and pitch circles should be equal to the radii ratio of the dedendum auxiliary and pitch circles of the gear. This imposes certain restrictions, for example, the ratio $r_2/R_2 = 0.45$ is acceptable and provides a large area of the tooth space and, consequently, higher pump flow rate, but with a ratio of $r_1/R_2 = 0.45$, the gear teeth may be wedged. Or if the ratio $r_1/R_2 = 0.15$ is acceptable and provides a large area of the tooth space, but the ratio $r_2/R_2 = 0.15$ cannot be recommended [6].

The dependence of the tooth space area on the radial clearance coefficient is investigated. Considering the decrease in the bending strength of the tooth due to the large value of the r_2/R_2 ratio, a narrower range of values radial clearance coefficient $c^* = 0.25–0.3$ was investigated. In these narrow variation limits of the radial clearance coefficient, the dependence is almost linear. The dependence of the tooth space area on the tooth root fillet radius coefficient ρ_f^* was also studied. The difference between the area of the tooth space with the standard value $\rho_f^* = 0.38$ and the value of $\rho_f^* = 0.3$ is only 0.3% with a significant decrease in bending strength. Therefore, the factor of a root fillet radius should not be changed for the purpose of increase the tooth space area.

Changing the profile to increase pump performance, at the same time leads to a decrease in the bending strength of the teeth. Therefore, the bending strength of the cycloidal tooth requires additional research. The strength of the cycloidal teeth was studied for various r_2/R ratios of radii. The safety factor for the considered involute gear is assumed to be 3.0. The safety factors for the cycloid gear are calculated relative to it for various values of the r_2/R ratios of radii (Fig. 10).

The tooth root stress was determined using the finite element method. In the software Mechanical Desktop 6.0, a three-dimensional solid model of the gear sector has been created. Since bending stress arise not only in the root fillets of a tooth that is in meshing, but also in the root fillets of the adjacent teeth, the model includes a tooth under load and adjacent teeth. The model represents a sector of the gear that is bounded by the surface of the shaft sleeve and the radial straight lines that are the axes of the teeth (Fig. 4). It is assumed that a surface of the shaft sleeve is constraint. The load (concentrated force) is applied to the tooth tip along the normal to the involute.

In order for the involute gear to be replaced in the pump with cycloidal profile gear, it is necessary that two conditions be performed: the equal center distance of

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both gears and the equal diameter of the addendum gear circle. We use the system of two equations

$$\begin{cases} d_a = mz + 2h_a^* m \\ a_w = m\dfrac{z_1 + z_2}{2} = mz \end{cases}$$

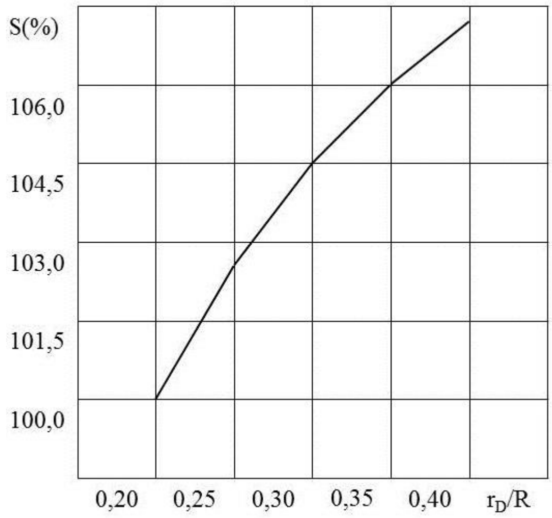

Fig.2 - Dependence of the tooth space area on the ratio circles radii - a dedendum auxiliary and a pitch.

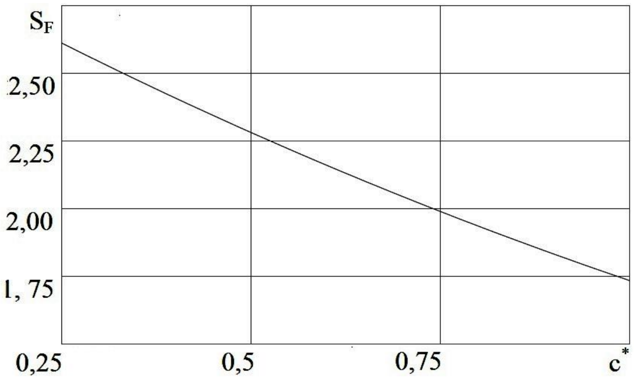

Fig. 3 - The dependence of the safety factor on the size of the radial clearance coefficient.

To compare the advantage of using an involute and cycloidal profile in gear pumps, gears with an involute and cycloidal tooth profile having the same parameters were selected. Involute gear parameters: the number of teeth z = 8, module m = 10, profile shift coefficient x = 0.624, radial clearance coefficient c* = 0.25, root radius factor ρ_f^* = 0.38, tooth addendum coefficient h_a^* = 1.

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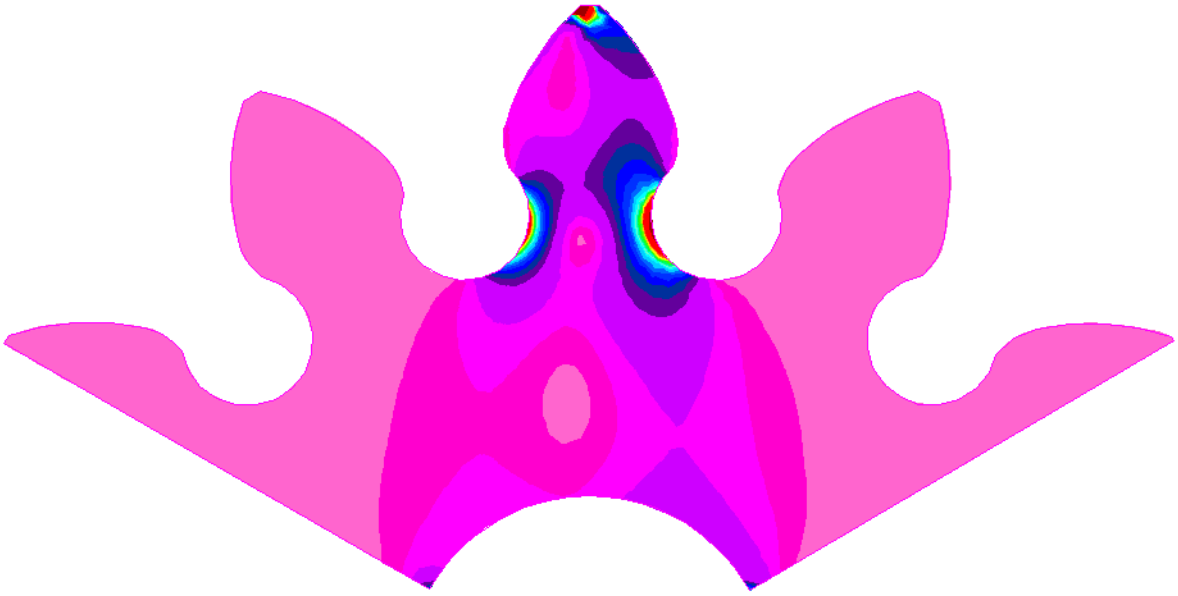

Fig. 4 - Solid model and stress distribution

Cycloidal gear parameters: the number of teeth z = 8, radial clearance coefficient c* = 0.25, root radius factor ρ_f* = 0.38, the circles radii ratio of an auxiliary and pitch r/R = 0.4 (for the addendum and the dedendum of a tooth). Solving these equations, we find the cycloidal gear train module m = 11.25 mm and the tooth addendum coefficient h_a* = 0.9991. It has been established by calculation that with the above gear parameters, the tooth space area of the cycloidal gear is 27% larger than the involute one (the tooth space area of the involute gearing is taken as the baseline).

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