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# Determination of the Tooling Module of the Gear Wheels for Wear Resistance of Gears Teeth

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**ABSTRACT**: The article discusses the definition of the gearing module of gear wheels on the wear resistance of gear teeth. The existing methods for calculating the engagement module do not take into account the wear resistance of teeth in the presence of abrasive particles in the oil. A gear made of a material with high mechanical properties may not always be sufficiently wear-resistant. From this we can conclude that the gears of the aggregates of machines operating in an abrasive environment must, along with sufficient bending strength and contact endurance, be provided with the necessary wear resistance of the teeth.

**KEY WORDS**: Wear Resistance, Gearing, Gear Teeth, Abrasive Particles.

## I.INTRODUCTION

Gears are widespread in industrial units and in household appliances. They act as an intermediate link between the source of rotational-translational motion and the node acting as the final consumer of this energy. Moreover, the transmitted power can be calculated as negligibly small units (watch movements and measuring instruments), as well as by enormous efforts (turbines of power stations).

**Modes of motion transmission.** The engine that generates energy, and the final unit that consumes it, often differ in characteristics such as rotation speed, power, angle of application of force. In addition, one source of rotational energy can be used to actuate several different nodes or aggregates at once. To ensure the delivery of torque in such conditions, intermediate modules are needed, which would transmit this force with minimal losses.

If, as a result of such a distribution or transformation, the revolutions of the drive shaft become greater than those of the follower, then it is customary to speak of a reduction gear. In this case, the loss of speed is compensated by an increase in the load on the driven axle and an increase in the power of the consuming node. In the event that an increase in the number of revolutions is eventually observed, such a transmission will increase. Accordingly, this will be accompanied by a decrease in the force on the driven shaft.

**Features gear mechanism.** The belt drive assumes the presence of an intermediate link between the pulleys on the connected shafts - a flexible belt. Gear mechanism from such a connection is characterized by the presence on the surface of the mating parts of the teeth of the gearing. They are identical in profile and size.

The head of the wheel's tooth engages with a repetitive hollow on the gear. When the drive shaft rotates, the driven shaft turns in the opposite direction. Between them, the minimum possible gap is provided constructively, providing sliding, thermal expansion and lubricant to prevent jamming. In this case, the leading part of the pair mechanism is called the wheel, and the driven part - the gear.

In the belt drive, the plane of engagement of the belt with the pulley is at least one third of the circumference. In a gear mechanism between the drive wheel and the driven gear under load, one pair of teeth is in constant contact. Wheels and gears on shafts are usually mounted on a keyed joint.

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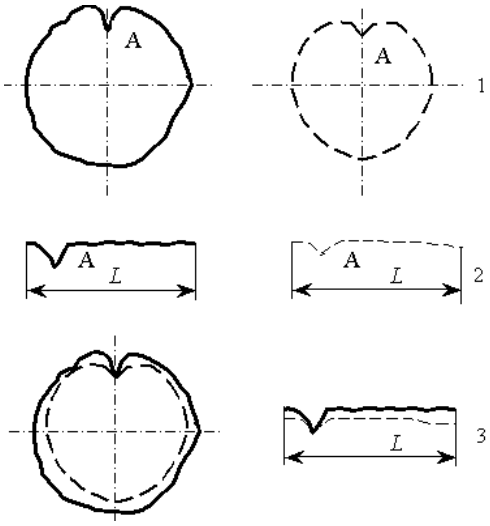
## II. EXPRESSION FOR THE GRAIN SENSOR CAPACITY

**Wear measurement methods.**

The wear of parts is measured by the method of direct geometric measurements using a measuring tool - micrometers, calipers, calipers, brackets with and without indicators, etc. When fine-tuning machines and their structural elements at the factory during their testing, more accurate methods are used.

One of the most practical is the method of artificial bases or the method of holes cut on the friction surface of the part. In the process of testing, the geometric characteristics of the wells are measured on a new part and after its dismantling at a certain operating time. The difference in measurements is judged on the actual wear of surfaces.




A - a label for registration; L is the length of the measured area; 1 - in cross section; 2 - in longitudinal section; 3 - combined profilogram: before wear; after wear.

Fig.1. Measurement of Wear by the Method of Superimposing Macro Profiles

The method of superimposing macroprofilograms has found a fairly widespread use for measuring the wear of the surfaces of parts at the machine manufacturers. It is most often used when measuring cylindrical surfaces and holes by removing and superimposing macro profiles on new and used parts (Fig. 1). For the correct combination of macro-profiles, taken from the new and worn parts, marks on the surface to be tested are pre-marked with carbide tools. The difference in sizes on the profilograms shows the values of the actual wear of the surface of the part in this section.

The above describes methods for measuring the wear of parts that require complete or partial disassembly of aggregates or machinery mechanisms. More practical for these purposes should be considered methods of measuring the wear of machine parts by indirect indicators.

This is, first of all, the method of spectral analysis, which in practice is called the “iron in oil” method. It consists in burning oil samples in a volt arc, collecting combustion products into a prism and obtaining, due to a beam of light, an interference picture in which impurities of each metal have their own color.

The most widespread in the world practice of mechanical engineering and operation found the method of radioactive isotopes. The essence of the method is based on the change in the amount of radioactive radiation from the rubbing surfaces at certain wear after opening the radioactive inserts of certain isotopes. Before testing, radioactive inserts with a known emissivity or, more often, different quantities of them are mounted in the volume of parts [1-2].

There are other methods for measuring wear of rubbing parts, which, however, have not found wide practical application.

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## III. EXPERIMENTAL RESULTS

Wear resistance, overall dimensions and metal consumption of transmission units of automobiles, tractors, road-building and other machines operating under high loads and in an abrasive environment, largely depend on the design parameters of gears, as well as on their loading. In most designs of machine units, there is a tendency to ensure noiselessness, smoothness and wear resistance of the gear teeth of machine transmission units. This is achieved by reducing the engagement module and the working width of the gear teeth. This condition in gears is provided with sufficient wear resistance of gear teeth in an abrasive environment, bending stress strength and resistance to contact stress. As a rule, the main cause of gear teeth breakage during the operation of machine aggregates is material fatigue under the action of bending stresses at the base of the tooth.

For transmission units of tractors with spur gears, the gearing gear module, on the basis of bending strength, is determined by the formula:

$$m = 1{,}435\sqrt[3]{\frac{2M}{C\psi z_w[\sigma_u]}},\ \text{m}, \qquad (1)$$

where C - a complex parameter of strength; M - the calculated torque on the drive gear, Nm; ψ - tooth width ratio; $z_w$ - is the number of teeth of the pinion gear; [σi] - is the allowable bending stress, MPa, the value of which is accepted for the gears most frequently operating under load, 250 MPa, for rarely working gears - 400 MPa.

The main causes of chipping are fatigue phenomena that occur on the contact surfaces of the gear teeth. The gearing engagement module for the contact endurance of the material is determined according to:

$$m = 30{,}5\sqrt[3]{\frac{M(i+1)}{[\sigma_H]z_w^2\psi_\alpha i\sin 2\alpha}},\ \text{m}, \qquad (2)$$

where $\psi_\alpha$ - is the ratio of the length of the tooth; $\psi_\alpha$ =L/m (here L is the length of the tooth, m); α - tool profile angle, $\alpha=22^o$; $[\sigma_н]$ - permissible contact stress, for steel with volumetric hardening with hardness HRC 38 ... 50 varies between 1200 and 1500 MPa.

With $[\sigma_i]$=250 MPa; $[\sigma_н]$=1200 MPa; C=0,75 and the unit efficiency is η=0,915, it can be concluded that increasing the load transmitted by the gear leads to an increase in the length of the gear teeth. The increase in the engagement module and the number of teeth on the gears, which absorb the load, leads to a decrease in the length of the tooth due to a decrease in the ratio of end overlap.

Reducing the engagement module also leads to an increase in the number of gear teeth, which absorb the load, by increasing the ratio of the end overlap and reducing the bend shoulder, as a result of which the bending stress at the base of the tooth decreases.

Table 1
The dependence of the ratio of the mechanical overlap of the gear transmission from the engagement module and the number of gear teeth

1) at $A_1$=0,450 m; $A_2$=0,462 m; $i_1$=2; $i_2$=2,0

| m, mm | 1,0 | 2,5 | 5,0 | 7,5 | 10,0 | 12,5 | 15,0 |
|---|---|---|---|---|---|---|---|
| $z_1$ | 300 | 120 | 60 | 40 | 30 | 24 | 20 |
| $\varepsilon_{\alpha 1}$ | 1,86 | 1,84 | 1,80 | 1,76 | 1,72 | 1,68 | 1,64 |
| $z_3$ | 370 | 148 | 74 | 49 | 37 | 29 | 25 |
| $\varepsilon_{\alpha 2}$ | 1,87 | 1,84 | 1,81 | 1,77 | 1,74 | 1,70 | 1,67 |

The ratio of the face of overlap gears is determined by the formula:

$$\varepsilon_{\alpha(1,2)} = \left[1{,}88 - 3{,}2\left(\frac{1}{z_{1,3}} + \frac{1}{z_{2,4}}\right)\right]\cos\beta,$$

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where β - is the angle of inclination of the teeth, hail; $z_{1,3}$ and $z_{2,4}$ - the number of gear teeth.

Thus, the existing methods for calculating the engagement module do not take into account the wear resistance of teeth in the presence of abrasive particles in the oil. A gear made of a material with high mechanical properties may not always be sufficiently wear-resistant. From this we can conclude that the gears of the aggregates of machines operating in an abrasive environment must, along with sufficient bending strength and contact endurance, be provided with the necessary wear resistance of the teeth. For the calculation of the gearing module gearing that meets these requirements, and in the presence of slippage between the teeth and active participation in the process of wear of abrasive particles, the dependence is obtained:

$$m = \frac{LiH_{w,k}{}^{2}\gamma_{aw,k}\sqrt{\gamma_a}\,n_{p(w,k)}z_k(z_k + z_w)}{22{,}33\sigma_a{}^{2}\Gamma_{w,k}\sqrt{\varepsilon_k{}^{3}}\sqrt{\gamma_m}\,d_{cp}n_{w,k}(i+1)(z_w \pm k)} * \text{, м,} \quad (3)$$

$$* \frac{1}{\sqrt{z_w{}^{2}\sin^2\alpha + 4kz_w \pm 4k^2} - z_w\sin\alpha}$$

where $\gamma_{aw,\,к}$ - is the wear rate of gear teeth with the active participation of abrasive particles of the driving (driven) gear, m/h; $n_{p(w,k)}$ - is the number of loading cycles, leading to the destruction of the deformed volume of the metal of the driving (driven) gear; $\gamma_a$ - is the density of the abrasive particle, kg / m$^3$.

The engagement module taking into account the contact strength of the material of gear wheels and the wear rate with the passive participation of abrasive particles is equal to [1]:

$$m = \frac{E_{np}\sqrt{i\gamma_{\partial w,k}z_w z_k n_{p(w,k)}c\sigma_{T(w,k)}L}}{22{,}9\sigma_{н}{}^{2}\sqrt{n_{w,k}}\sqrt{(z_w{}^{2}\sin^2\alpha + 4kz_w \pm 4k^2)\theta_{w,k}(z_w \pm 1)}} * \text{, m,} \quad (4)$$

$$* \frac{1}{\sqrt{\sqrt{z_w{}^{2}\sin^2\alpha + 4kz_w \pm 4k^2} - z_w\sin\alpha}}$$

where$\gamma_{aw,\,к}$- is the wear rate of the teeth with the passive participation of abrasive particles of the driving (driven) gear, m/h.

## IV. CONCLUSION

In order to ensure the wear resistance of gears with the active and passive participation of abrasive particles, the strength of gear teeth to bending stress and contact endurance is taken as the highest value of the engagement modules calculated from expressions (1), (2), (3).

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