

FREEWHEELING CLUTCH AND POLYHEDRAL RING

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INTRODUCTION

Freewheeling clutch and polyhedral ring resets in mechanical coupling category, intermittent, controlled, and asynchronous and are similar from functional point of view with unisense coupling.

Freewheeling clutches hydraulically controlled allowed coupling and releasing for two rotating axels with different rotational speed. The turning movement transfer and coupling transfer are made with some rollers which, hydraulically controlled, are self-locking between two mating surfaces which are in relative rotating motion.

The roller blocking phenomenon between two mating surfaces it is possible to be realized gradually as instantaneous with adequate adjustments of a regulation liquid pressure from hydraulic circuit.

The advantages of this coupling categories are: allowed the coupling and releasing during the function time, in both of rotation sense; they transmit the big moment of torsion; they have a small dimensions; they assure a high durability; a relative simple construction; a high cut-in speed and synchronizing speed independent of position and relative rotative speed of coupling pieces.

To this couplers, the moving energy are dissipated the hydraulically way. In this way is not possible to produce the coupler heating in case of repeated operation, with high frequency, for instance in case of vehicles gearing.

1. CONSTRUCTION. FUNCTIONING.

In figure 1 are presented a freewheeling clutch and elastic polyhedral ring. The basic elements are: the driving 2, the pinion 1, the conduction crown gear, the rollers 4, the elastic ring with rounded peak side 12, pressing a driven shaft 10, from key 11. In releasing position, the crown gear are on free rotation over the 4 roller and elastic ring.

The command hydraulic liquid are introduced inside of a channel a from driving 2 between polyhedral hub 12 and elastic ring 6, the last one being distancing.

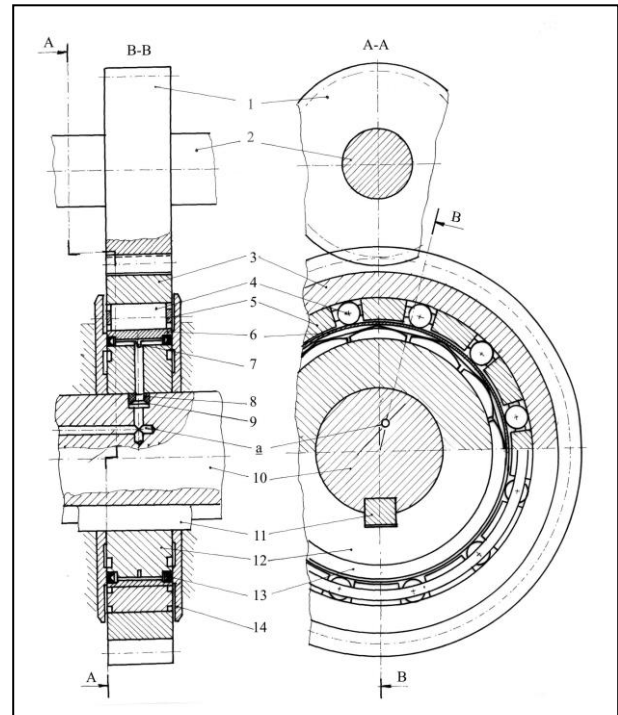


Figure 1. Freewheeling clutch and polyhedral ring.
 1- pinion; 2- driving; 3- driven wheel; 4- roller; 5- cage; 6- casting ring with polyhedral ring; 7- the spacer; 8- seal; 9- disc spring; 10- driven shaft; 11- key; 12- hub with polyhedral exterior; 13- seal; 14- collar.

The elastic ring wall 6, being with variable thickness, the deformation they have a polyhedral rate (fig. 2), rollers will be wedging between ring (6) and crown gear (3). The elements 3, 4, 6 relative motions being impossible, the moment of torsion will be transmitted to driven shaft (10).

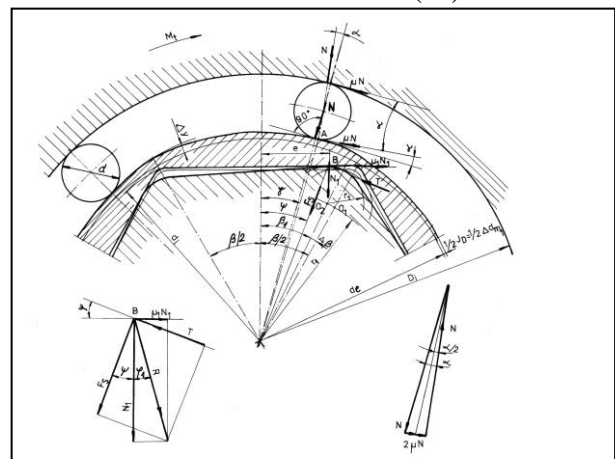


Figure 2. Forces and deformations in coupling

The wedging effect of rollers are assured and amplified by the reacting force F_s (fig.2) which appear in the contact point B between rounded peak of the polyhedral hub (12) and polyhedral interior of elastic ring (6) .

from which result :

$$\operatorname{tg} \frac{\alpha}{2} < \mu = \operatorname{tg} \varphi \tag{2}$$

where : φ is angle of friction .

The moment of torsion which will be transmitted will be:

$$M_t = \mu \cdot N' \cdot z \cdot \frac{D_i}{2} \tag{3}$$

were: z is numbers of roller ; D_i – interior diameter of framing (fig.2) .

Because: $N' = N'' = N$, from relation (3) results :

2. CALCULATUION ELEMENTS

The force system with reacting on rollers, in deformed start of elastic ring is presented in figure 3. Because the motion and the moment of torsion is needed to be transmitted, the roller will not drop between two surfaces, with the following condition: the resultant of force projection in the same direction with α angle bisector to be directed in clockwise direction.

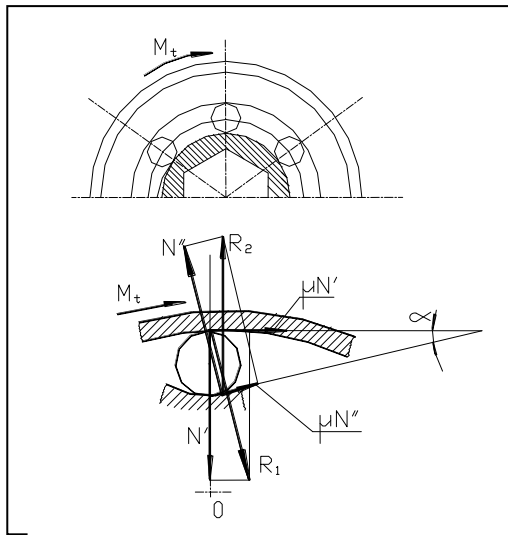
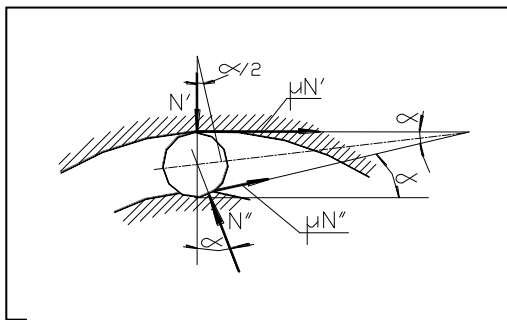


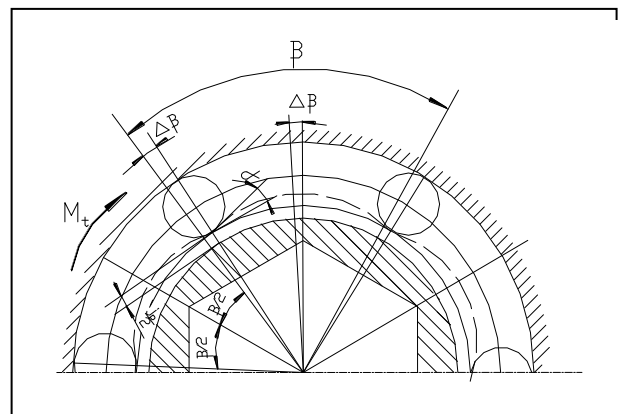
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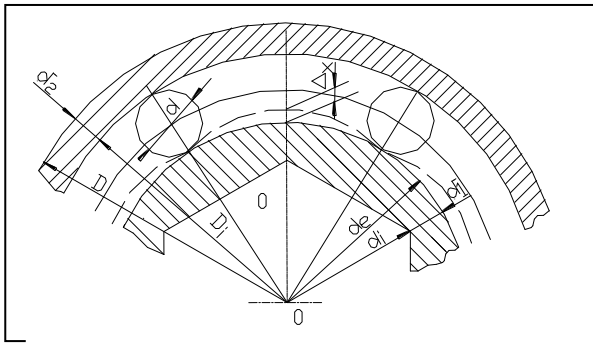
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$$\Delta d_m = \frac{\pi}{\pi} \approx \frac{\pi \cdot m}{E} = j_D \quad (8)$$

Tension stress adequate with j_D deformation is :

$$\sigma_t = \frac{E \cdot j_D}{d_m} \quad (9)$$

Hydrostatical pressure which is necessary for j_D deformation realization and σ_t tension is :

$$p_1 = \frac{2\delta_1 \cdot \sigma_t}{d_i} \quad (10)$$

were: p_1 is theoretical initial pressure for clearance between ring and rollers realization, if the ring is not polyhedral.

The real pressure for polyhedral ring deformation (p_{1r}) is the theoretical pressure adjusted with K (adjustment coefficient) which is dependent of number of rollers.

Is possible to write :

If we consider: $r + r_1 \approx \frac{d_i}{2}$ the last

relation became:

$$F_s \cong \frac{2M_t}{z \cdot d_i \cdot \text{tg}(\psi + \varphi_1)} \quad (15)$$

were: $\psi = \text{arctg} \frac{e}{\frac{d_i}{2} \cos \frac{\beta}{2}}$ -tangential force

between the ring and the hub transmission angle ;

φ_1 - angle of friction between the ring and the hub ;

$\psi > \varphi_1$ - for self blocking avoided.

The entire pressing force on the roller is deduced from forces diagram, making the projections in oblique direction with γ angle in comparison with vertical direction. It is possible to write:

$$\begin{aligned}
 N &= F + R \cdot \cos(\gamma + \varphi_1) = \\
 &= F + \frac{F_s \cdot \cos(\gamma + \varphi_1)}{\cos(\psi + \varphi_1)} \quad (16)
 \end{aligned}$$

Replacing F force (rel.13), F_S force (rel.15) and N force (rel.4) in the last relation, we obtaining:

$$\begin{aligned}
 \frac{\pi \cdot \beta \cdot d_i \cdot b \cdot p_2 \cdot z}{360 \cdot M_t} &\geq \frac{2}{\mu \cdot D_i} - \\
 &- \frac{\cos(\gamma + \varphi_1)}{(r + r_1) \sin(\psi + \varphi_1)} \quad (17)
 \end{aligned}$$

The total necessary pressure for coupling function is:

$$p_t = p_{1r} + p_2 \quad (18)$$

from which :

$$p_2 = p_t - p_{1r} = p_t - K \cdot \frac{2\delta_1 \cdot E \cdot j_D}{d_i \cdot d_m} \quad (19)$$

With this p₂ value, relation no.17 because:

$$\begin{aligned}
 \frac{\pi \cdot d_i \cdot b}{M_t} \left(p_t - K \cdot \frac{2\delta_1 \cdot E \cdot j_D}{d_i \cdot d_m} \right) &\geq \\
 \geq \frac{2}{\mu \cdot D_i} - \frac{\cos(\gamma + \varphi_1)}{(r + r_1) \sin(\psi + \varphi_1)} \quad (20)
 \end{aligned}$$

On predimensioning, is possible to assume:

$$r + r_1 \approx \frac{d_i}{2}$$

$$\psi \geq \varphi_1$$

$$\gamma = \psi$$

$$K \approx 1$$

and relation no.20 because:

$$\begin{aligned}
 \frac{\pi \cdot \beta \cdot d_i \cdot b}{360 \cdot M_t} \left(p_t - 2 \frac{\delta_{1 \cdot E \cdot j_D}}{d_i \cdot d_m} \right) &\geq \\
 \geq \frac{2}{\mu \cdot D_i} - \frac{2}{d_i \cdot \operatorname{tg}(\psi + \varphi_1)} \quad (21)
 \end{aligned}$$

being useful in coupling dimensioning from this tip.

In real lifting capacity establish for coupling, we make a correlation between obtaining

data from relation no.21 with resistance consideration establish in contact fatigue strength.

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