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 vikw 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 moved 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 graduate 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 bath of rotation sense; they transmit the big moment of torsion; they have a small dimensions; they ashore a high durability; a relative simple construction; a high cut-in speed and synchronizing speed independent of position and relative rotational 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 chancel a from driving 2 between polyhedral hub 12 and elastic ring 6, the last one being distancing.

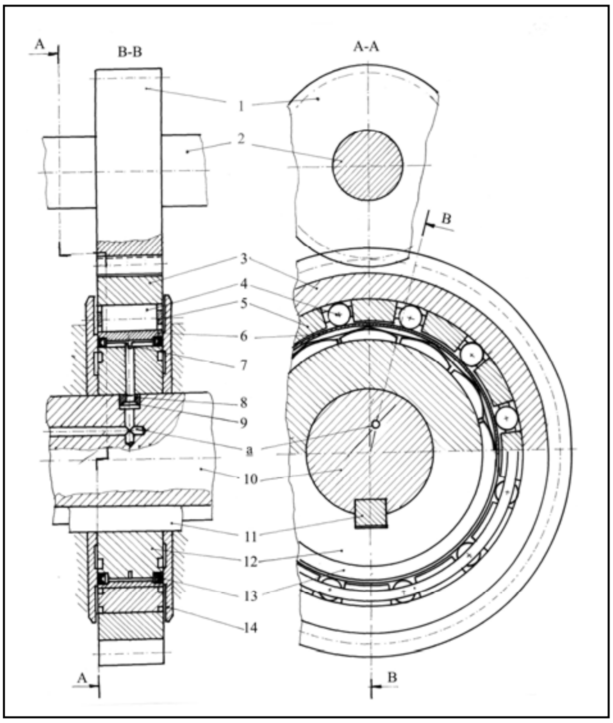

Figure 1. Freewheeling clutch and polyhedral ring. 1- pinion; 2- driving; 3- driven wheel; 4- roller; 5- cage; 6- casting ring with polyhedral ring; 7- the spacer; 8- seal; 9- disc spring; 10- driven shaft; 11- key; 12-hub with polyhedral exterior; 13-seal; 14-collar.

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

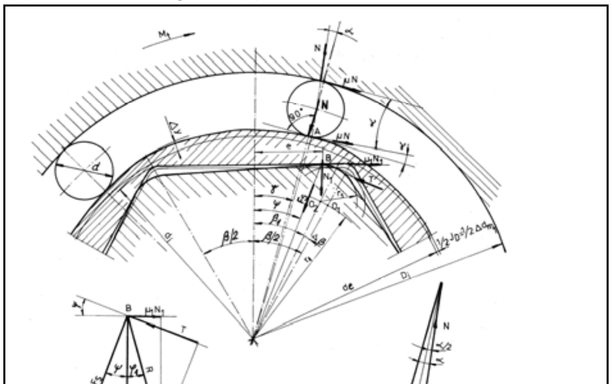

Figure 2. Forces and deformations in coupling.

motions being impossible, the moment of torsion will be transmitted to drive shaft (10).The wedding 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 ş12ş) and polyhedral interior of elastic ring (6) .

2. CALCULATUION ELEMENTS

The force system with reacting on rollers, in deformed state 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.

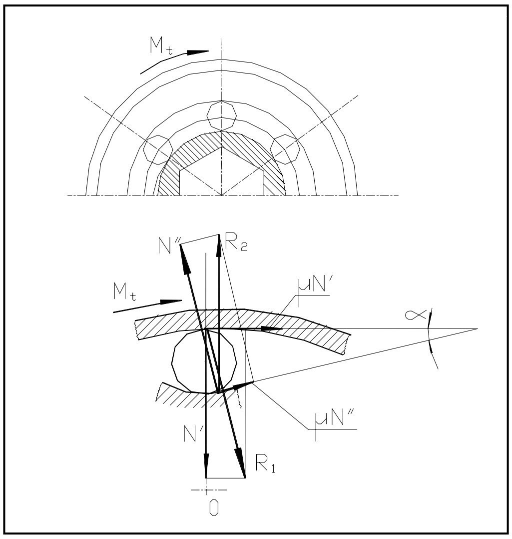

Figure 3. The forces which react on the rollers in deformed state of elastic ring.

Using the notation from figure 4, is possible to write the relation:

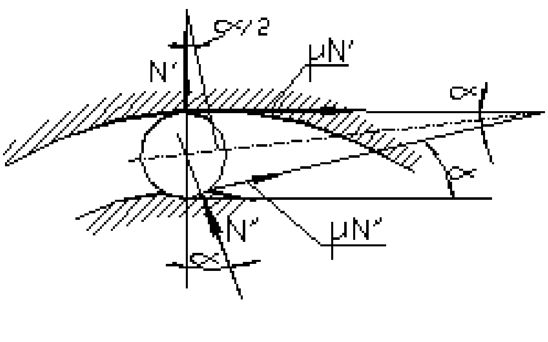

Figure 4. The determination of wedging condition.

$$N' \sin\frac{\alpha}{2} + N'' \sin\frac{\alpha}{2} < \mu N' \cos\frac{\alpha}{2} + \mu N'' \cos\frac{\alpha}{2} \quad (1)$$

from which result :

$$tg\frac{\alpha}{2} < \mu = tg\varphi \quad (2)$$

where : φ is angle of friction.

The moment of torsion which will be transmitted will be:

$$M_t = \mu \cdot N' \cdot z \cdot \frac{D_i}{2} \quad (3)$$

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

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

$$N = \frac{2M_t}{\mu \cdot z \cdot D_i} \quad (4)$$

Replacing in the last relation the initial condition from relation 2, are obtaining the value for nominal force:

$$N < \frac{2M_t}{z \cdot D_i \cdot tg\frac{\alpha}{2}} \quad (5)$$

The angle of wedging α which appear in case of moving with Δβ angle of elastic ring (6) in ratio of polyhedral hub, were β is the angle between rollers .

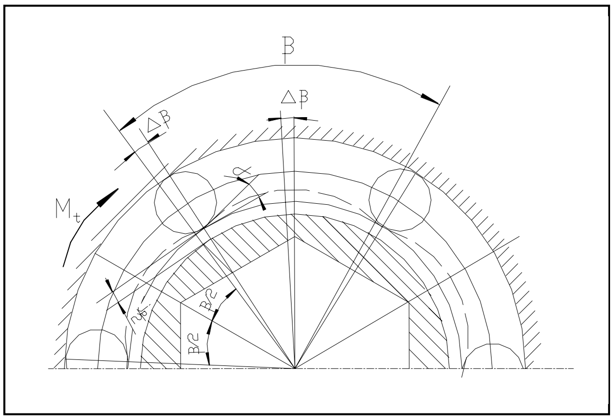

Figure 5. Crating mode for angle of wedging.

Using the notation from figure 6, the medium diameter of elastic ring and medium diameter of rim have the following values:

$$d_m = \frac{d_i + d_e}{2} ; D_m = \frac{D_i + D}{2} \quad (6)$$

Diametral gaming between rollers and elastic ring and the framing, is:

$$j_D = D_i - \left(d_e + 2 \cdot d\right) \quad (7)$$

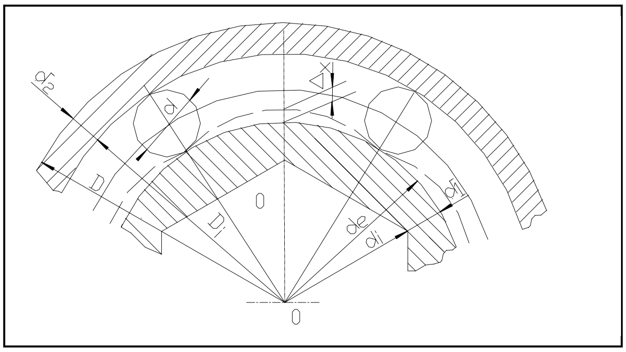

Figure 6. Polyhedral elastic ring in deformed state.

were: d – roller diameter.

Under command pressure action, the medium diameter of elastic ring growth $\Delta d_m = j_D$, and l circumference of ring growth with $\Delta \ell$ value.

Is possible to write the following relation:

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

Tension stress 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 şp$_{1r}$) is the theoretical pressure adjusted with K (adjustment coefficient) which is dependent of number of rollers.

Is possible to write:

$$p_{1r} = K \cdot p_1 = K \cdot \frac{2\delta_1 \cdot \sigma_t}{d_i} = K \cdot \frac{2\delta_1 \cdot E \cdot j_D}{d_i \cdot d_m} \quad (11)$$

and:

$$p_{1r} = \frac{2\delta_1 \cdot \sigma_{rr}}{d_i} ; \frac{\sigma_{tr}}{\sigma_t} = K \cdot \frac{p_{1r}}{p_1} \quad (12)$$

were: δ_1 is minimum thickness of elastic ring in the corner of internal polygon.

For dependence relation establish between geometrical, constructive and functional parameters for coupling will be determine, first, the F, F_s and N forces.

Using the notation from figure 2, the F resultant of p_2 pressure, steady distributed, will be determine with the next relation:

$$F = \frac{\pi \cdot d_i \cdot b \cdot p_2 \cdot \beta}{360^0} \quad (13)$$

were: p_2 is blocking pressure; b – the width of ring .

From balance equation of forces in vertical plan, we obtaining the F_2 concentrated load, which's resultats from polyhedral hub pressing:

$$F_s = \frac{T}{tg\left(\psi + \varphi_1\right)} = \frac{M_t}{z\left(r + r_1\right) tg\left(\psi + \varphi_1\right)} \quad (14)$$

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

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

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

$$N = F + R \cdot cos\left(\gamma + \varphi_1\right) = = F + \frac{F_s \cdot cos\left(\gamma + \varphi_1\right)}{cos\left(\psi + \varphi_1\right)} \quad (16)$$

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

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

The total necessary pressure for coupling function is:

$$p_t = p_{1r} + p_2 \tag{18}$$

from which :

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

With this p_2 value, relation no.17 because:

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

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:

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

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.

Bibliography

1. **Belcin , O.** *Cercetări asupra cuplajelor cu role comandate hidromecanic şi hidrostatic direct şi posibilităţilor de implementare în construcţia de maşini . Contract 1081/B/93, M.C.T., Bucureşti .*
2. **Belcin, O.** *Variatoare discrete de turaţie acţionate hidrostatic. Teză de doctorat. Universitatea Tehnică din Cluj - Napoca, 1996.*
3. **Belcin, O.** *Organe de maşini. Rulmenţi, cuplaje, roţi de fricţiune, angrenaje. Editura RISOPRINT, Cluj - Napoca, 2000.*
4. **Bogdan, M., Belcin, O., Şandor, L.** *Cuplaje şi variatoare hidrostatice de turaţie. Lito I.P.C - N, 1990.*
5. **Chişu, E. ş.a.** *Cuplaje mecanice intermitente şi cu contacte mobile. Editura Lux Libris, Braşov, 1999.*

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