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CONDITIONS FOR TOOTHING IN TWO-STAGE MULTI-OUTPUT TRANSMISSIONS

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Abstract: *The paper deals with the necessary conditions which are to be satisfied when designing and constructing the two-stage multi-output transmissions. The issue is closely connected with the previous study on the transmissions.*
Key words: *involute teeth, internal wheel, gearings, cycloidal teeth.*

1. INTRODUCTION

The paper describes necessary conditions for designing of gearing for two-stage multi-output transmissions solved at the Department of Technological Systems Design of the Faculty of Manufacturing Technologies of the Technical University in Košice with the seat in Prešov as well as it refers to the works [1] [2]. Presented necessary conditions relate to involute and cycloidal teeth by means of which the mentioned gearing can be stepped. At the same time, one of the possible principal constructional solutions of such a gearing stepped by involute teeth is presented in the paper.

2. NECESSARY CONDITIONS FOR INTERCOUPLING OF TEETH

Kinematical and force ratios on the two-stage transmission with one internal toothed wheel were described in [1], [2]. The presented gearing, in fact, is the two-stage transmission of the two gearings with internal and external teeth where the internal wheel meshes with its toothed rims concurrently with the two wheels at the same time – with the wheel 1 and the second stage gearing. At the same time it rotates on its own axis on the eccentric (crank) of the input shaft. The presented solution eliminates the complicatedness of transmission of epicyclical motion into centric motion used with high-precision gearings described in [4]. The mentioned gearing enables to achieve great gear ratios in wide range. It depends on appropriate selection of number of teeth of the four toothed rims when the necessary conditions for their intercoupling are satisfied. By means of appropriate design it is possible to achieve the transmission in forward or opposite direction of rotation contrary to input revolutions.

In this case, gear ratios can be achieved much better than with the well-known high-precision gearings (TEIJIN SEIKI,SUMITOMO CYCLO etc). From the point of view of their performance, these gearings can be dimensioned to analogical values of performance and torques as with the well-known high-precision gearings. It follows from the principle of their construction. Even though, this design, with regard to simplification of transmission of epicyclical motion into centric motion, supposes the increase of their working life and as well stability of loading during their working life. It is supposed that their working life will be the same as working life of processing equipment, i.e. about 10 years in one-shift or two-shift operation under the condition that maintenance of equipment is carried out (eventually bearings or another parts dimensioned for shorter working life are changed). Disadvantages relating to pitting creation are of the same character as cycloidal teeth described in the previous chapters.

The principle of transmission as well as the kinematical scheme, which describes ratios at the second stage of transmission related to the solution presented in [3], is illustrated in Figure 1.

Range of gear ratios:

Real gear ratios with involute teeth can reach the values u_{max} = up to 1 000.

With cycloidal teeth, the values of maximum gear ratio can reach up to 10 times higher values than with involute teeth, thus u_{max} = up to 10 000.

<u>Necessary conditions when determining the number of teeth</u>

Regarding the solution [1], [2], the following necessary conditions when determining the number of teeth have been derived:

1. The same eccentricity for the teeth of the first and the second stage of transmission both with involute and cycloidal teeth. Thus, the following must have been valid:
   a) For the output shaft stepped with internal teeth: $r_e = r_1 - r_2 = r_4 - r_3$ or
   - for the involute: $r_e = \dfrac{(m_1 z_1 - m_1 z_2)}{2} = \dfrac{(m_2 z_4 - m_2 z_3)}{2}$, (1)
   - for the cycloidal: $r_e = \dfrac{k(d_{c1} z_1 - d_{c1} z_2)}{2} = \dfrac{k(d_{c2} z_4 - d_{c2} z_3)}{2}$. (2)

   b) For the output shaft stepped with external teeth: $r_e = (r_1 - r_2) = |r_4 - r_3|$ or
   - for the involute: $r_e = \dfrac{(m_1 z_1 - m_1 z_2)}{2} = \dfrac{|m_2 z_4 - m_2 z_3|}{2}$, (3)

- for the cycloidal: $r_e = \dfrac{kd_{c1}(z_1 - z_2)}{2} = \dfrac{kd_{c2}|z_4 - z_3|}{2}$. (4)

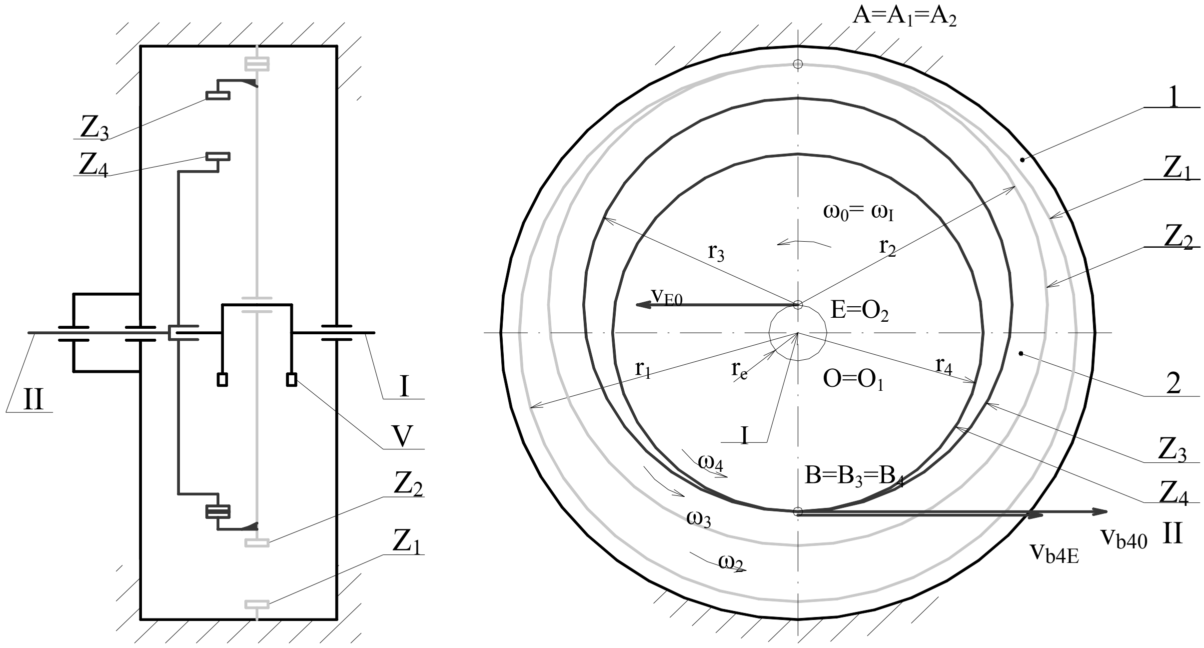

Fig. 1. Kinematical scheme of the second stage of transmission. Output shaft is stepped by the wheel with external teeth

2. With the involute teeth, the requirements in order to avoid the primary and secondary interference must be satisfied. With the internal teeth, the condition for the diameter of the tip circle r_a is valid:

$$r_a \geq m\sqrt{\frac{1}{tg^2\alpha} + \left(\frac{z_i}{2} - 1\right)^2}\,, \qquad (5)$$

where

α - angle of the mesh,

z_i - number of internal teeth

Condition of the secondary interference for the number of teeth:

$$(z_1 - z_2) = |z_4 - z_3| \geq 8 \text{ - for axial assembly of teeth,} \qquad (6)$$

$$(z_1 - z_2) = |z_4 - z_3| \geq 12 \text{ - for radial assembly of teeth.} \qquad (7)$$

With the cycloidal gears, conditions presented in (2) are not valid.. Difference in number of teeth z_1, z_2 resp. z_4, z_3 can be minimal, thus 1 tooth i.e. $(z_1 - z_2) \geq 1$ or $|z_4 - z_3| \geq 1$.

Dimensions of lantern wheel and epicycloidal wheel when stepping the mechanism of transmission with cycloidal teeth are presented in the table1. Solution of cycloidal teeth for the mentioned mechanism was presented in [5].

Calculation of the basic dimensions is presented in the following table:

Table 1 Calculation of the basic dimensions

Value	*Wheel - lantern*	**Wheel - epicycloidal**
Number of teeth	-	z_2
Number of lanterns (rollers)	z_1	
Diameter of lantern (roller) [mm]	d_c	
Spacing [mm]	$p = \pi d_c k$	$p = \pi d_c k$
Angle of spacing [°]	$\varphi_{p1} = 2\pi / z_2$	$\varphi_{p2} = 2\pi / z_1$
Diameter of root cylinder[mm]	$d_{f1} = (z_1 k - 1) d_c$	$d_{f2} = (z_2 k - 1) d_c$
Average spacing of the cylinder [mm]	$d_1 = d_c z_1 k$	$d_2 = d_c z_2 k$
Diameter of tip cylinder [mm]	$d_{a1} = (z_1 k + 1) d_c$	$d_{a2} = (z_2 k + 2k - 1) d_c$
Eccentricity r_e [mm]	$r_e = (d_1 - d_2)/2$	
Height of tooth top	$h_{a1} = d_c / 2$	$h_{a2} = d_c (2k - 1)/2$
Dedendum height	$h_{f1} = d_c / 2$	$h_{f2} = d_c / 2$
Width of teeth	Will follow from stress calculation	

From the Table 1 it follows that dimensions of the gears can change with the given gear ratio with the same number of teeth z_1 and z_2 and the given diameter of the roller (lantern) d_c, by changing the mutual eccentricity. This can be influenced by the coefficient k. For effective utilisation of gears with power transmission we select $k \geq 1$ (usually $k = 1$)

Figure 2 represents the principal construction solution of the solved transmission stepped by involute teeth.

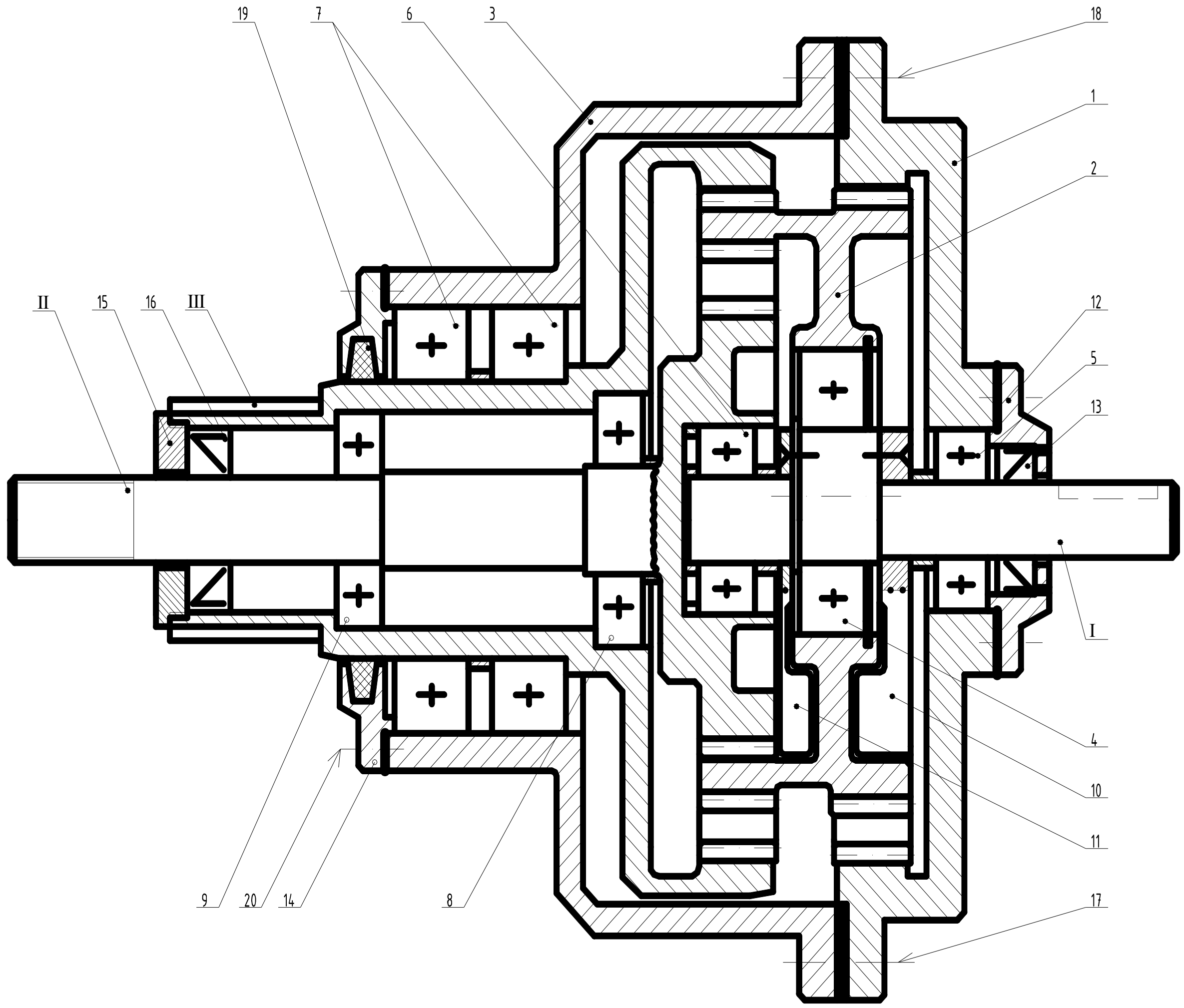

Fig. 2. Basic construction of the two-stage transmission with one internal impeller toothed wheel with the fixed balancing on input shaft

The legend to Figure 2

Basic parts:

I – input shaft,	8, 9 – bearings of the shaft II,
II, III – output shafts,	10, 11 – balancing with fixed gripping,
1 – body with internal teeth,	12, 14 – flanges,
2 – toothed wheel with 3 toothed rims,	13, 16 – sealing,
3 – body of the case,	15 – nut,
4 – bearing of the wheel 2,	17, 20 – screws,
5, 6 – bearings of the input shaft,	18 – sealing,
7 – bearing of the shaft III,	19 – sealing.

3. CONCLUSION

Particular characteristics of the gear box depends on the particular required solution for the required purpose concerning both the revolutions of transmission, accuracy as well as performance characteristics. In cooperation with the company REGADA in Prešov, the functional model of the transmission will be made in order to test the functions, possibilities of performance loading, characteristics of torque stiffness, efficiency and wear. Based on the analysed measuring and findings, further requirements to improve structural, functional and performance characteristics of the mentioned transmissions will be specified. The proposed solution is protected by the Certificate of Authorship number 3937 of 11 August, 2004: the authors: Jozef Haľko, Eng, PhD and Prof. Vladimír Klimo, Eng, PhD.

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