STUDY ON THE EFFICIENCY OF PLANETARY REDUCTION GEARS WITH A DOUBLE WHEEL CARRIER

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Abstract: *Among all the mechanical drives currently in use, the planetary reduction gears hold an important place, being characterized by a compact structure and high transmission ratios. The study of the planetary mechanisms displays two main aspects: the analysis, who settles the main issues and the synthesis, who determines the type and the diagram of the mechanism. This paper intends to perform an analysis both from the point of view of the overall dimensions and of the efficiency displayed by a planetary mechanism with a double wheel carrier.*

Key words: *planetary reduction gears with a double wheel carrier; efficiency*

1. INTRODUCTION

The planetary driving with toothed wheels covers a wide application field due to the benefits provided compared to the driving with fixed driving gears:

1. high kinematical opportunities for reaching a wide rage of values in the transmission ratios, with a diminished number of toothed wheels;
2. small overall dimensions;
3. high or at least acceptable output;
4. very compact structure;
5. it can be typified in a suitable manner;
6. there is the possibility for producing a wide range of powers up to very high powers;
7. a relatively simple construction.

In spite o all these, the planetary transmission was used on a small scale up to the latest decades, and this was because of certain disadvantages, such as:

1. an uneven distribution of the load in the situation when the transmission of motion is split through several identical wheel carriers;
2. the need for observing the conditions of coaxiality, vicinity and mounting

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3. the obligation for avoiding the phenomenon of interference for the case of the internal driving gear;
4. technical problems for producing toothed wheels with internal denture and an wheel carrying arm.

Today, these disadvantages have been removed especially by the use of highly expert tool machines. The observance of the conditions of coaxiality, vicinity and mounting may generate a disadvantage that can be balanced by combining the number of teeth of the component wheels. As for the low efficiency attained by certain types of planetary driving gears, this one is valid only for high power driving gears. As a result, the design of such a planetary driving gear and the use of adopted constructive solutions shall always be made considering the efficiency that can be attained.

2. STUDY ON THE PLANETARY MECHANISMS

The study performed on the planetary mechanisms includes two main aspects, i.e.: analysis, who settles the main problems and synthesis, who determines the type and the diagram of the mechanism, starting from the given data that cover the kinematical characteristics and to some extend the dynamic ones. The planetary mechanisms are reduced to a mechanism equivalent to predetermined axes and this procedure lays the foundation for a general study method of planetary mechanisms.

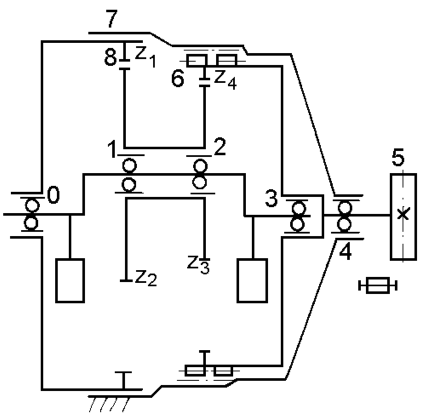

Figure 1. The kinematic diagram of the double satellite planetary reduction

The synthesis of a plain planetary mechanism includes:

- the kinematical synthesis which consists in determining the basic mechanism, the equivalent mechanism and transmission ratio *i;*

- the dynamical synthesis which consists in determining the mechanism so that is efficiency should be satisfactory to the purpose it has been created

This paper intends to present such an analysis, both from the point of view of the overall dimensions and from the point of view of the efficiency supplied by a planetary mechanism with a double wheel carrier. Figure 1 shows the kinematical diagram.

The preliminary data (Table 1) refer to the incoming power in the reduction gear, the total transmission ratio and the incoming rotation in the reduction gear. The values reached by these parameters are covered on one hand in the domain for which electric motors exist and on the other hand for which an applicative interest is being shown.

Table 1. Preliminary data

P_i, kW	1.8	5.8	13	2	6.7	10	20	3	7.5	17
n_i, rot/min	1500	1500	1500	1000	1000	1000	1000	1500	1500	
i_{total}	67	67	67	80	80	80	80	106	106	106
η	0.541	0.606	0.669	0.551	0.663	0.656	0.707	0.579	0.655	0.708

With the help of these values, there have been determined the geometrical elements for the two driving gears with internal denture and which form the reducer, there have been determined the sizes of the incoming shaft and there have been performed the calculations necessary for the selection of bearings.

3. CALCULATING THE EFFICIENCY

The determination of the efficiency for the pre-settled data considered the calculation of the specific power losses. The efficiency of the equivalent mechanism, η_e, shall be determined in relation to the specific losses:

$$\eta_e = 1 - \psi \tag{1}$$

where: ψ is the specific power loss, the sum of all the partial specific losses which impose in the equivalent mechanism; accordingly it results:

$$\psi = \psi_a + \psi_l + \psi_m \tag{2}$$

where: ψ_a is specific power loss produced by the gearing operation; ψ_l is specific power loss produced by friction inside bearings; ψ_m is specific power loss produced by friction in the place where movement occurs.

The specific power loss produced by the gearing operation, ψ_a, is the sum of the specific power losses produced by each gearing of the mechanism. The specific power loss that corresponds to one distinct gearing made up by the toothed wheels 1 and 2 is:

$$\psi_a = K_a\left(\pm\frac{1}{z_{e1}} \pm \frac{1}{z_{e2}}\right) \tag{3}$$

where: K_a is a coefficient that takes into consideration the friction ratio η among the gearing teeth, the rolling height of the upper part of teeth a_r, the gearing modulus m and the normal gearing angle α_{rn} of this one; z_{e1} an z_{e2} represent the number of corresponding teeth of the two toothed wheels in gearing; the plus sign (+) corresponds to the external toothed wheel and the negative sign (-) corresponds to the internal toothed wheel.

The specific power loss produced by friction inside bearings, ψ_l, weighs relatively small, its values being in relation to the type of bearings. For the case of sliding bearings, the specific power loss shall depend on their main sizes, on the lubricant used, on the type and the loading value and on the relative clearances. The specific power loss for roller bearings shall be calculated in relation to the friction momentum, M_f, occurred in the bearings:

$$\psi_l = \frac{M_f \omega_e}{P_e} \tag{4}$$

where: ω_e is the equivalent angular velocity reached by the bearings; P_e is the equivalent power of the mechanism.

The total specific power loss produced by the friction inside the bearings is the sum of all the specific losses that correspond to each bearing apart.

The specific power loss produced by the friction occurred between the moving parts and the atmosphere inside which this mechanism operates ψ_m, shall depend on the atmosphere, on the shape of the moving parts and on the relative velocity born between these moving parts and the surrounding atmosphere. An exact calculation of these losses cannot be made, this determination being made experimentally for each situation apart. Based on the researches carried out by different researchers, there has been concluded that these losses are below 10% of the total of all the other specific power losses whether the level of the lubricant is adequately selected.

The efficiency of the equivalent mechanism, determined in relation to the specific power losses, allowed a further calculation of the efficiency reached by the reduction gear and an analysis of the situations when this one attains acceptable values:

$$\eta = 1 - \frac{1-\eta_e}{i_0 - \eta_e} \tag{5}$$

where: η_e is the efficiency of the equivalent mechanism; i_0 is the transmission ratio of the basic mechanism.

4. EXEMPLE OF CALCULATION

Considering the features of the hopper wagons, the rotation speed reduction gears shall meet the following conditions:

- their size shall match the clearance from beneath the hopper wagon body (ϕ400, l=400) from the discharge end, allowing the rotation of the bogie with an angle of 30º to enter the curve;
- high transmission ratio of 30...35;
- the outlet shaft of the reduction gear shall take over high rotation moment and radial strain values, since the chain transmission driving wheel is mounted through it;
- they shall form a block construction along with the hydraulic motor, they shall be easily mounted on and dismounted from the wagon and possibilities of extending the transmission chains shall be available;
- they shall be able to perform normally in underground conditions (dust, water, mud);
- maintenance performed on the wagon with no dismounting.

For VSAH type hopper wagons, the planetary reduction gear shall transmit a power of P=7.5 kW for an input rotation speed of n=340rot/min with a transmission ratio $i\approx 34$.

The size condition and the load of the tooth system points out that an as large number as possible of teeth are necessary, so that the outside diameter of the satellite be as close as possible to the outside diameter of the inner tooth system, in the point opposite to putting into gear the tendency should be towards the tangency of the two circles, so that the condition z_1-$z_2 \geq 4$ should be met. In the simplest case of the reduction gear, where the two gears have the same modulus, the condition of coaxiality gives: z_1-z_2=z_4-z_3.

Knowing that: $z_1=z+x$, $z_2=z-y$, $z_3=z-x$, $z_4=z+y$, $z \gg x$, $z \gg y$, $|x|<y$, x=0.5k (k – total number), at the limit of gearing possibility of the satellite with the inner tooth wheel (z_1–z_2=z_4-z_3=4) the following relation results:

$$z = \sqrt{i_{hb}^{2} \cdot (16 - 8 \cdot x) + x^{2}} \qquad (6)$$

By the relation (6), for $i_{hb}^{a} = 35$ and a 400 mm maximum diameter, for a 7 mm modulus and x=-3.5, y=7.5 and z=39.4, the following number of teeth resulted: z_1 =47, z_2 =43, z_3 =32, z_4 =36.

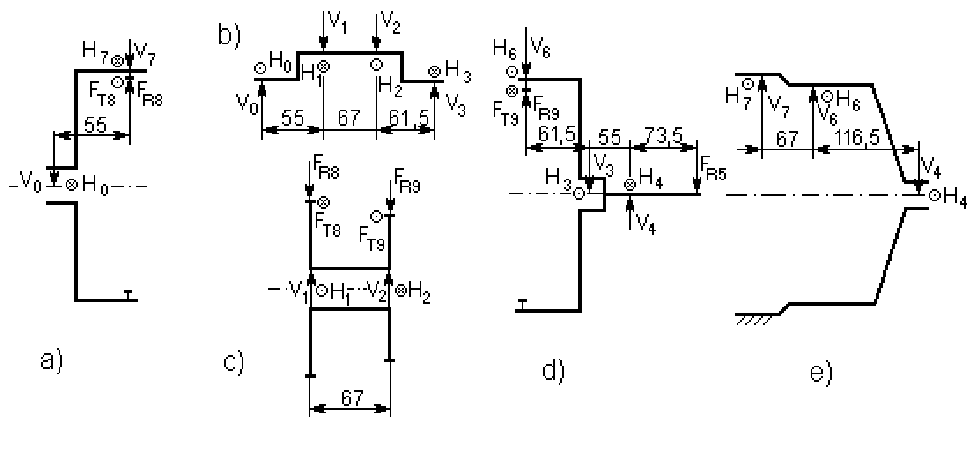

Figure 2. Loading schemes of the component parts of the reduction gear

Determining the equivalent power P_E =243.5 kW the reactions were calculated in the reference points of the reduction gear resulting in accordance with the loading schemes in figure 2 (**a** - cover; **b** - shaft; **c** - satellite; **d** - toothed wheel shaft; **e** - body) R_0 =33.25 kn.; R_1 =69.71 ;R_2 =68.16 kN; R_3 =32.44 kN; R_4 =139.46 kN; R_6 =112.96 kN; R_7 =40.92 kN. A comparison of these values shows that the loads in bearings *4* and *6* are smaller in the new construction by 1.75 times as to the traditional constructions, where higher dynamic loading capacity balls are necessary.

Knowing the loads in the bearings and gears, the specific power losses in reduction gear are calculated function of which the efficiency of the mechanism is calculated equivalent to the relation (1), where: ψ_a=6.05·10^{-4}; ψ_l=4.975·10^{-3}/16.775·10^{-3}; ψ_m=6.99·10^{-3}. The efficiency of the reduction gear is with relation (5) η=0.7.

The reduction gear thus carried out (figure 3) is made up of a housing *14* wherein an eccentric shaft is mounted *11* by two ball *7* and *8*. By balls *5* and *6*, the double satellite with two gear wheels is fitted.

The gear wheel *2* mates the fixed inner-toothed crown wheel *3*, and the gear wheel *3*, the inner-toothed crown wheel in the end of the outlet shaft *4*.

The outlet shaft is supported on ball *9*, and the inner gear wheel on ball *10.* Inside the reduction gear, on the eccentric shaft representing from a kinematics point of view the satellite carrier arm, two counter-balance weights are mounted *12* and *13*.

5. CONCLUSIONS.

Figure 4 shows the variation curve for the efficiency reached by the reduction gear depending on the incoming power P_i; Figure 5 shows the variation curve for the efficiency reached by the reducer depending on the total transmission ratio, i_{total}.

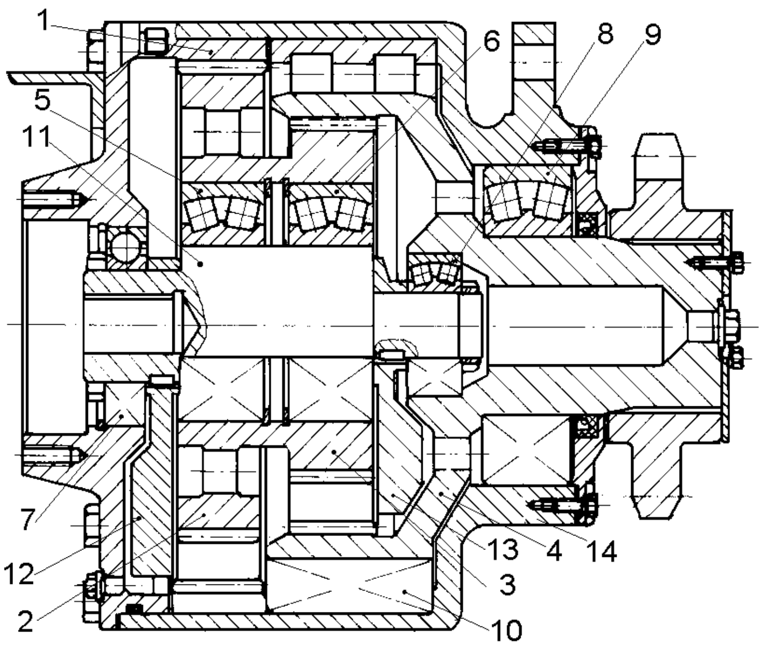

Figure 3. Double satellite planetary reduction gear

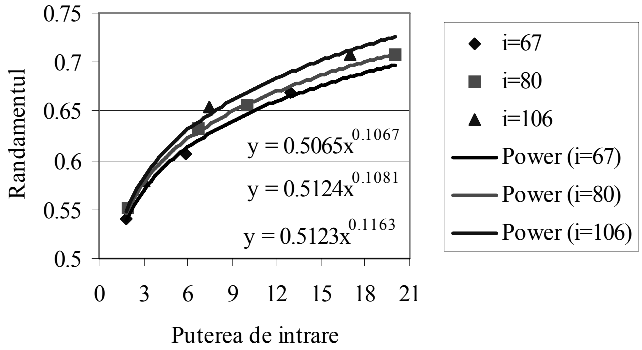

Figure 4. Dependence of the efficiency η by the incoming power P_i

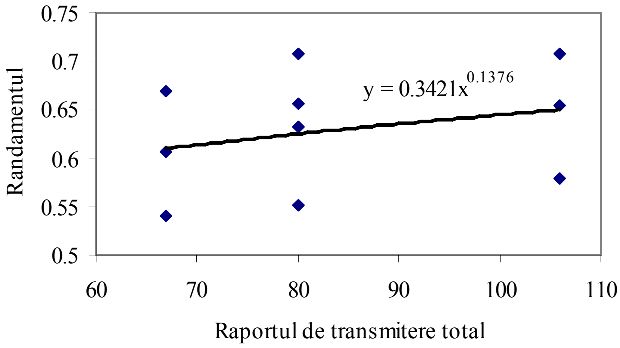

Figure 5. Dependence of the efficiency η by the total transmission ratio i_{total}

The law of variation related to the efficiency shown in Figure 5 emphasizes the existence of dependence between this one and the total transmission ratio for the range of values i_{total} = 60…120. For other values reached by the transmission ratio, these types of dependence could not be emphasized. For these transmission ratios, the most significant overall dimensions are those who make reference to the central wheel l.

These analyses can be extended for wide ranges of values, both with respect to the transmission ratio and to the power incoming the reduction gear.

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