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DESIGN METHODOLOGY FOR A PLANETARY MULTIPLICATOR WITH SYNCRONUOUS BELTS OR CHAINS

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Abstract: The subject of the paper is a planetary transmission using synchronous belts or chains instead of gears. Starting from a planetary unit with four external gear wheels, the transmission has been developed as a speed multiplicator in order to be used for wind turbines. Since the first solution with syncronuous belts showed a lack of power capacity, the paper is presenting the design methodology for both constructions (syncronuous belts and chains). Based on the numerical results of design, the main conclusions on the influences on the power transmission capacity and also on mechanical efficiency can be drawn.

1. INTRODUCTION

Planetary gear mechanisms have a large use on the construction of motor vehicles, machine tools and industrial robots transmissions. A planetary unit is defined [3, 4, 6] as a 2 DOF gear mechanisms with three central elements: two gear wheels (central elements 1 and 4) with the same rotational axis (central axis of the planetary unit); one carrier H, with the same rotational axis. The planetary gear wheels (2 and 3) have their own rotational axis on the carrier. Figure 1, a presents the symbolical scheme and the structural scheme of a 2DOF planetary unit. In order to obtain a 1DOF planetary transmission, one of the three central elements (1, 4 or H) must be blocked.

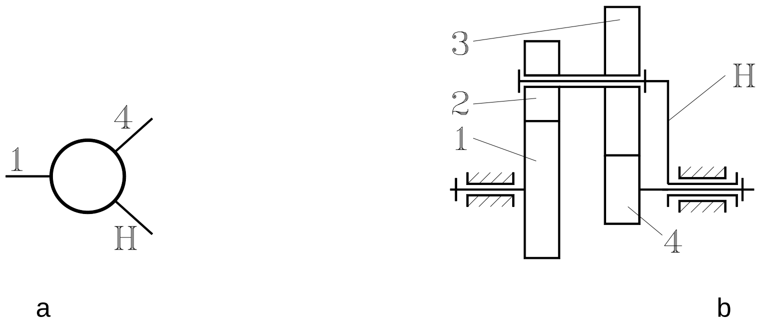

Figure 1. 2DOF planetary unit: a – symbolical scheme, b – structural scheme;

This kind of planetary unit can use wheels with small differences between their number of teeth (z1 ≈ z4, z2 ≈ z3) in order to obtain an interior transmission ratio i_0, very close to 1. In this case they can be used as speed reducer with the carrier H as input element, blocking the central wheel 4, and with the central wheel 1 as the output element, reccomended in order to reach very high transmission ratios [4].

$$i_0 = \frac{z_2}{z_1}\frac{z_4}{z_3} \tag{1.1}$$

As a speed multiplcator it can be used with the central wheel 1 as the input element, blocking the central wheel 4 and with the carrier H as output element. In this way, the transmission ratio of the planetary transmission is

$$i = i_{1H}^{4} = 1 - i_0 \tag{1.2}$$

and can take very small values.

The main disadvantage of the transmission is the relatively low value of the mechanical efficiency [4], which usually imposes its use only for kinematic transmissions (transmitting very small torques).

2. DESCRIPTION OF THE PLANETARY BELT OR CHAIN TRANSMISSION

The planetary belt (chain) transmission [2] which is the subject of this paper has been developed starting from the planetary gear unit presented in Figure 1, b. Instead of gears, the proposed planetary transmission is using synchronous belt or double chain (structural scheme presented in Figure 2, a). The axis of the planetary wheels (wheels 2 and 3) is moved towards the central axis of the planetary unit so that the overall dimension of the transmission is diminished.

As it can be seen in Figure 2, the belt (chain) planetary transmission is using two pairs of planetary wheels (satellites). Smaller center distances and smaller planetary wheels (wheels 2 and 3) lead to better contact between belt (chain) and the central wheels (wheels 1 and 4).

Figure 3 is presenting the assemble drawing of the belt planetary transmission. The two synchronous belts have contact on half of their width with each of the central wheel and planetary wheel. The transmission has been developed as a speed multiplicator in order to be used for wind electric generators. It wasn't based on a full strength calculus and so the capable transmitted power of he transmission wasn't established before detail design.

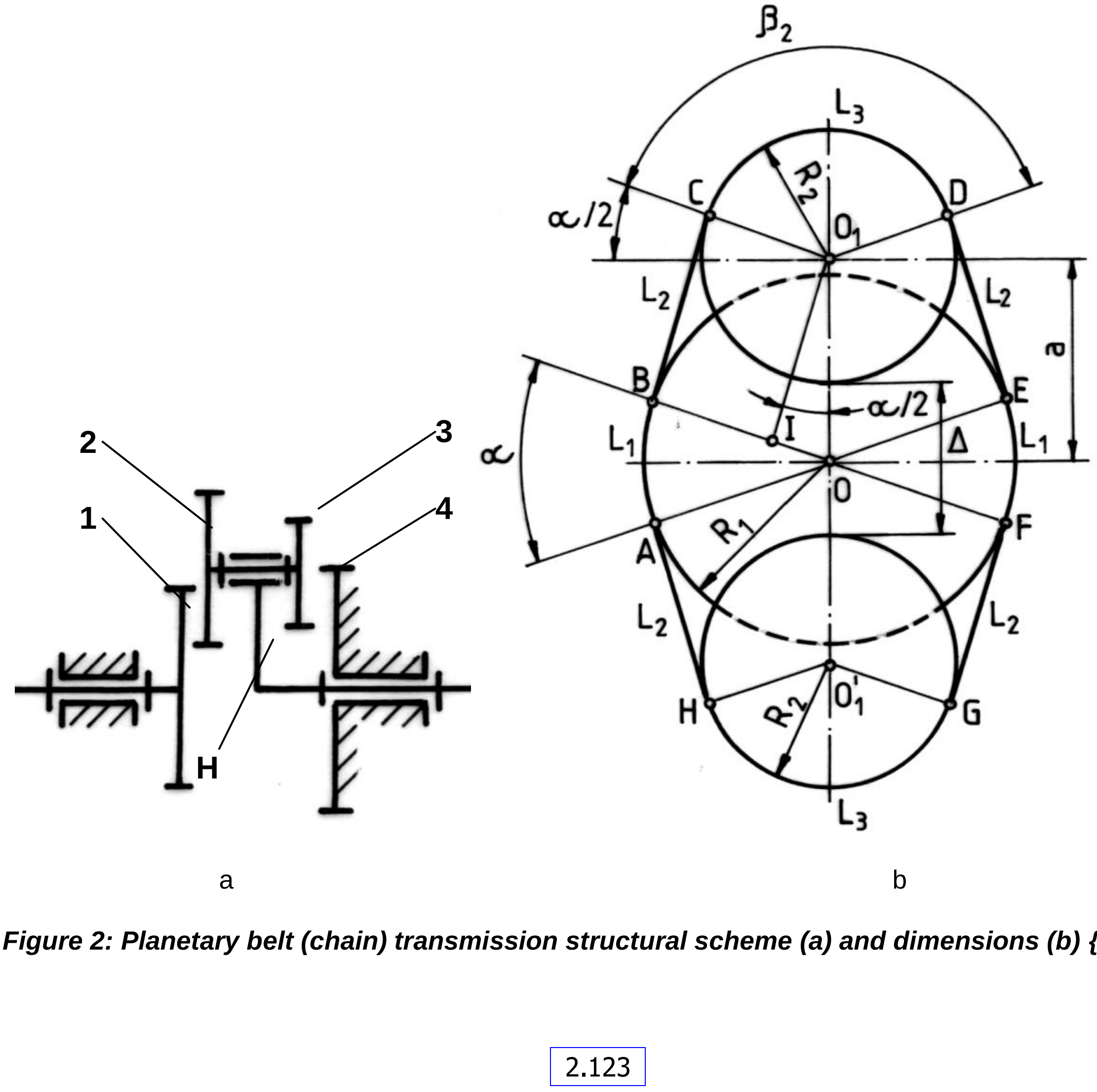

Figure 2: Planetary belt (chain) transmission structural scheme (a) and dimensions (b) {2}

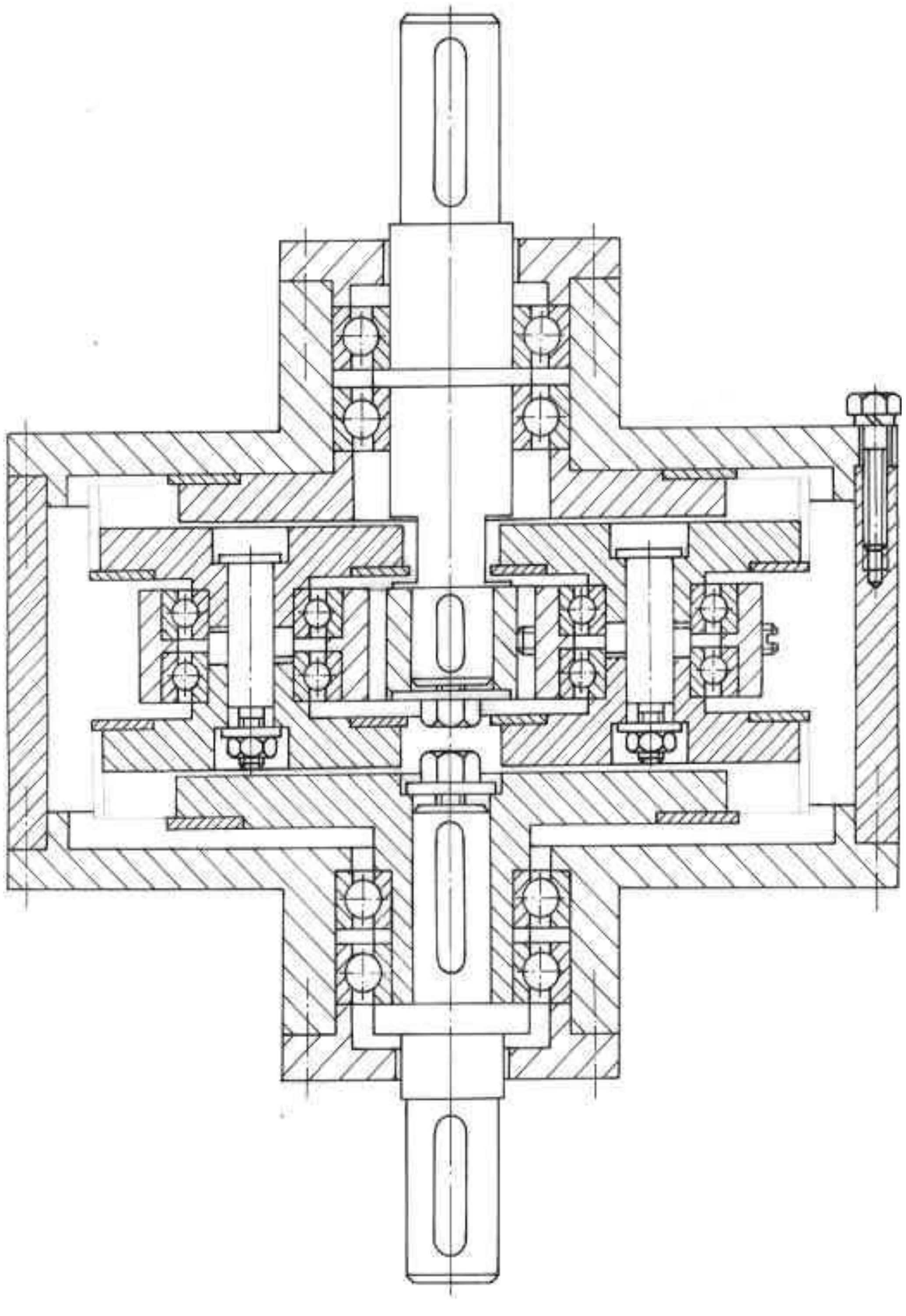

Figure 3. Assemble drawing of belt planetary transmission [2]

3. DESIGN METHOD. CONCLUSION

The design method was developed in order to establish the capable power of the transmission. Table 1 is presenting the design steps and the numerical results for the existing transmission with synchronous belts. It is based on reducing the real planetary transmission to a fixed axes transmission, on which the torques on elements can be easily established and are similar to the case of the real transmission.

Table 1. Design methodology for planetary belt transmission

No.	Parameter	Calculus relations. Recommendations	Existing transmission
1	Number of teeth of the belt wheels	z_1, z_2, z_3, z_4	$z_1 = 70$; $z_2 = 38$; $z_3 = 40$; $z_4 = 68$;
2	Number of satellites	s	s = 2
3	Belt type and dimensions (pitch, number of teeth, length, width)	p, z, L, b, mm	T5x420, p = 5 mm; z = 84; L = 420 mm; b = 16 mm
4	Input rotational speed (central wheel 1)	n_1 , rpm	$n_1 = 50$ rpm
5	Interior ratio of the planetary unit	$i_0 = \frac{z_2}{z_1}\frac{z_4}{z_3}$	$i_0 = 0.923$
6	Transmission ratio	$i = i_{1H}^4 = 1 - i_0$	$i = 1:13$
7	Output rotational speed (carrier H)	$n_H = n_1 \cdot i$, rpm	$n_H = 650$ rpm
8	Rotational speeds of the elements of the fixed axes mechanism (after the movement inversion)	$n_{1H} = n_1 - n_H$; $n_{2H} = n_{1H}\frac{z_1}{z_2}$; $n_{3H} = n_{3H}$; $n_{4H} = n_H$, rpm	$n_{1H} = 600$ rpm $n_{2H} = n_{3H} = 1105$ rpm $n_{4H} = 650$ rpm
9	Mechanical efficiency	$\eta = \frac{1 - {i_0}/{\eta_0}}{1 - i_0}$	$\eta = 0.369$, for $\eta_0 = 0.95$
10	Capable power per unit of width, for the synchronous belt	P/b = f(belt type, number of teeth of the smallest wheel, rotational speed of the smallest wheel), kW/mm [1]	P/b = f(T5x420, z_2, n_{2H}) = 0.0238 kW/mm
11	Capable power of the transmission	P = (P/b)·b , kW	P = 0.38 kW
12	Capable torques on elements (on the fixed axes mechanism)	$M_{t1} = 9{,}55 \cdot 10^6 \frac{P}{n_{1H}}$; $M_{t2} = M_{t3} = 9{,}55 \cdot 10^6 \frac{P}{n_{2H}}$; $M_{t4} = 9{,}55 \cdot 10^6 \frac{P}{n_{4H}}$, N mm	$M_{t1} = 6050$ Nmm $M_{t2} = M_{t3} = 3285$ Nmm $M_{t4} = 5580$ Nmm
13	Capable power of the transmission	$P = s\frac{M_{t1} n_1}{9{,}55 \cdot 10^6}$, kW	P = 0.064kW = 64 W

As it can be seen, the resulted capable power is relatively low. In order to improve the capable power, the only pertinent action was to change the type of the belt, keeping the same center distance between wheels and adjusting the number of teeth. Table 2 is presenting the capable power resulted for different belt types with higher pitches. Bigger pitches cannot be used due to decrease of the number of teeth.

Table 2: Capable power for different types of syncronous belts

Pitch, mm	5	9.525	12.7	22.225
Capable power, kW	0.064	0.120	0.529	0.655

Another solution for improving the capable power of the planetary multiplicator is the use of two double row bushing chain instead of the synchronous belts. Table 3 is presenting the design methodology for the planetary chain transmission.

Table 3. Design methodology for planetary chain transmission

No.	Parameter	Calculus relations. Recommendations	Existing transmission
1	Number of teeth of chain wheels	z_1, z_2, z_3, z_4	$z_1 = 70$; $z_2 = 38$; $z_3 = 40$; $z_4 = 68$
2	Number of sattelites	s	s = 2
3	Chain type and dimensions (pitch, width and length of link)	p, a_1, d_2, mm	05B, p = 8 mm; $a_1 = 4.77$ mm; $d_2 = 2.31$ mm
4 … 9	Input rotational speed (central wheel 1) Interior ratio of the planetary unit Transmission ratio Output rotational speed (carrier H) Rotational speeds of the elements of the fixed axes mechanism (after the movement inversion) Mechanical efficiency	Same as in the case of planetary belt transmission, see Table 1	
10	Admissible pressure	p_a , MPa	$p_a = 22.5$ MPa
11	Admissible useful force	$F_{ua} = a_1 d_2 \frac{p_a}{K_e}$, N [1]	$F_{ua} = 248$ N
12	Chain speed	$v = \frac{z_1 n_{1H}}{60000}$, m/s	v = 4.4 m/s
13	Chain capable power	$P = \frac{F_{ua} v}{1000}$, kW	P = 1.05 kW
14	Capable torques on elements (on the fixed axes mechanism)	Same as in the case of planetary belt transmission, see Table 1	$M_{t1} = 16600$ Nmm $M_{t2} = M_{t3} = 9076$ Nmm $M_{t4} = 15410$ Nmm
15	Capable power of the transmission	Same as in the case of planetary belt transmission, see Table 1	P = 0.176kW = 176 W

As it can be seen from Tables 2 and 3, the capable power of planetary chain transmission is bigger than the capable power of planetary synchronous belt transmission (0.176kW for a pitch of 8 mm, for belt transmission, in comparison with 0.120kW for a pitch of 9.525 mm, for chain transmission).

One of the most important issues in designing such a transmission is in assuring a competitive mechanical efficiency. As it can be seen from Table 1, the calculated mechanical efficiency is very low, depending on an approximated relatively low value of the mechanical efficiency of the transmission with blocked carrier (input element – 1, output element – 4, blocked carrier – H). Improving the mechanical efficiency of the belt or chain drive gives important improvement of the global mechanical efficiency of the multiplicator. As an example, calculus with η_0 = 0.97 (2% improvement) gives a global mechanical efficiency η = 0.629 (70 % improvement).

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