APPLICATION OF REVERSING OUTPUT DUAL BRAKE PLANETARY TRANSMISSIONS

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Abstract *– A transmission for internal combustion powered railway vehicles must be rugged and reliable. There are two categories of transmissions for the low to mid-high power range of railway vehicles: transmissions derived from ordinary highway truck or bus automatic transmissions and hydromechanical transmissions. Unlike steam locomotives, modern railway vehicles are expected to be capable of moving in both directions at the same speed, which is particularly important when units from the same series are connected together in multiple. Hydrodynamic and mechanical transmissions are known for use in these vehicles, but planetary gear trains are also suitable for the application due to their characteristics. An output planetary gearbox providing two equal transmission ratios, however with the output shaft rotating in different directions. Priority should be given to designs using clutch-type brakes for compactness and reliability, however band brakes should be considered if ease of maintenance is a priority. To sum up, the application of this design of planetary gearbox would simplify the design and manufacture of both hydrodynamic and mechanical transmissions for railway vehicles, which is the subject of the paper.*

Keywords – Railway vehicle, transmissions, planetary gear box

1. INTRODUCTION

Planetary gear trains (PGTs) offer numerous advantages when compared to conventional gear trains. Because of that, their use has been significantly expanded in a variety of applications in mechanical engineering. Examples of PGT applications at fishing boats and machine tools are shown in [1,2,3].

Their application is also possible at railway vehicles, as they are required to move both forwards and backwards with the same speed. Both hydrodynamic and PGT transmissions are commonplace in these vehicles, however reversing requires an additional mechanical stage. This stage can be easily replaced by an output planetary gearbox providing two equal transmission ratios, however with the output shaft rotating in different directions. Compound two-speed PGTs which are obtained from component planetary gear trains (stages) by linking shafts of different component planetary gear trains may be used in the design of such output gearboxes. Two speed PGTs are the subject of this paper and we shall consider two-speed two-carrier PGTs with four external shafts composed of two PGTs of the basic type.

The internal structures of compound gear trains are laid out. The significant number of all possible schemes requires a systematization of the variants, as well as their labelling.

The procedure of structure and important basic parameters determination is followed by a numerical example in which the optimal two-speed planetary gear train that meet predefined transmission requirements is selected, defined by the numbers of teeth of the central gears, modules and transmission ratios. The position of the transmission and working conditions of the railway vehicles determine the input data for the computer program that define structure and important parameters of the component planetary gear trains.

The basic part of the paper is the review of acceptable solutions of the transmission for the selected application by using a specially developed computer program. The choice between the program obtained variants is then made by comparative analysis of the solutions.

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2. PLANETARY GEAR BOXES

By connecting two shafts of one component PGT to two shafts of the other component PGT, a mechanism with four external shafts is obtained (Fig. 1). Among these four external shafts, two are coupled shafts and two are single external shafts. The component PGTs will be referred to as the component trains and the obtained mechanism with four external shafts will be referred to as the compound train. The two-component compound PGT is the simplest form of compound PGT

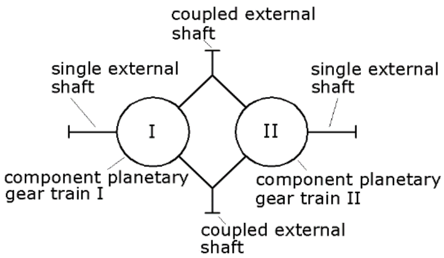


Fig. 1. Planetary gear train with four external shafts (compound train)

Both component trains are planetary gear trains of the basic type consisting of a sun gear 1, planet gear 2, ring gear 3 and planet carrier h, as shown in Fig. 2.

The simple and compound PGTs discussed in this paper will be described by means of Wolf-Arnaudov symbols (Figure 2) [1,3]. This is the most common type of PGT, and it has seen the broadest use in mechanics. It is most commonly used as a single stage transmission, or as a building block for higher compound planetary gear trains. Its primary advantage over other PGT types lies, first of all, in its efficiency. The overall dimensions and mass of this type of PGT are small, and its manufacturing costs are relatively low because of the relatively simple production process.

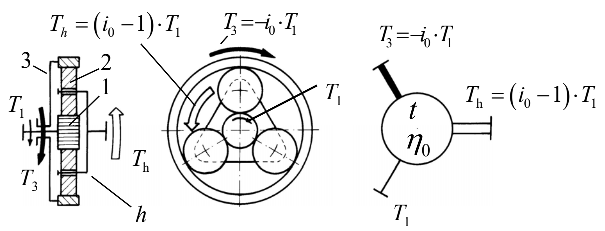


Fig. 2. Wolf-Arnaudov symbol and torque ratios of the basic type of PGT [4]

Planetary gear train shafts are loaded with torques as indicated in Fig. 2. The torque on the ring gear shaft T_3 and the torque on the carrier shaft T_h are given as functions of the ideal torque ratio t and the torque acting on the sun gear shaft T_1.

The ideal torque ratio is

$$t = \frac{T_3}{T_1} = \left|\frac{z_3}{z_1}\right| = -i_0 > +1 \qquad (1)$$

where i_0 is the basic transmission ratio, z_1 is the number of teeth of the sun gear and z_3 is the number of teeth of the ring gear.

Transmission ratio depends on locked element: sun gear, ring gear or carrier.

Two component trains can be joined in a total of 36 possible ways [5], however due to isomorphism, there are only 12 different ways which result in PGTs with four external shafts, Fig. 3.

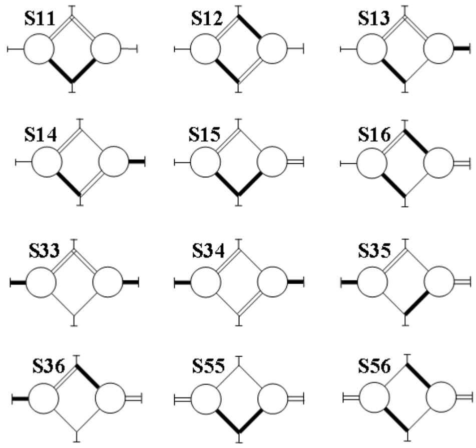


Fig. 3. Systematization of all schemes of two-carrier PGTs with four external shafts [1,6]

In every presented scheme it is possible to put brakes as well as the driving or the operating machine on external shafts in 12 different ways (V1…V12), which are referred to as the layout variants (Fig. 4). By placing brakes on different shafts it is possible to influence the power flow and kinematic characteristics. It is an important advantage, since these transmissions can be used as multiple-speed gearboxes.

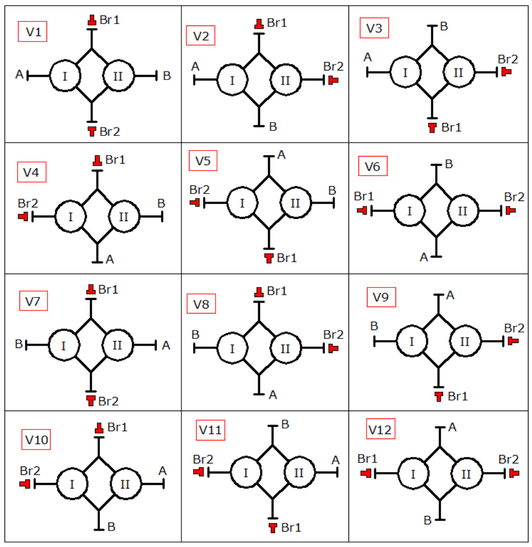


Fig. 4. Layout variants of two-carrier planetary gear trains with four external shafts [6,7]

3. NUMERICAL EXAMPLE AND DISCUSSION

A computer program *DVOBRZ* is based on a principle of synthesis of a two-speed PGTs [5]. It is used to select the optimal variant from similar multi-speed PGTs

The transmission ratio achieved by the activation of brake Br1 is marked as i_{Br1} and the transmission ratio realized by the activation of brake Br2 is marked as i_{Br2}, and all valid solutions are finally stored. The program is able to compare them according to the defined relevant criteria, e.g. minimal radial dimensions, maximum equivalent efficiency etc. [5].

A railway vehicle transmission will be used as a practical example for the selection of two-speed PGTs.

After taking into consideration that the required transmission ratios are i_1= -4.5 and i_2= 4.5, solutions have been found with transmission ratios in the ranges $-4{,}6 \le i_1 \le -4{,}4$ and $4{,}4 \le i_2 \le 4{,}6$. The important outher input datum is also frequency of working with transmission ratios: α_{i1}=0,5 (50%) and α_{i2}=0,5 (50%). The task is choice of optimal solution according to efficiency taking into consideration condition under which railway vehicle works. Based on the requirements and assumptions listed above, the *DVOBRZ* program lists six possible solutions for two-speed PGTs.

The main parameters are summarized in Table 1 while the kinematic schemes of acceptable solutions are shown in Table 2. The main parameters include the numbers of teeth of all gears and ideal torque ratios for both component gear trains. The number of planets is three exept bolded solution where the number of planets is four. The program *DVOBRZ* gives the ideal torque ratios for both gear trains. The tooth numbers of all gears were adopted on the basis of the ideal torque ratios [8], and presented in Table 1.

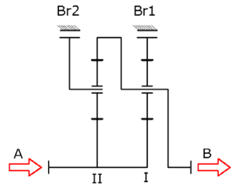
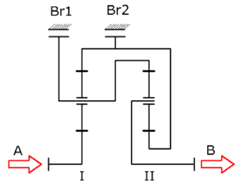
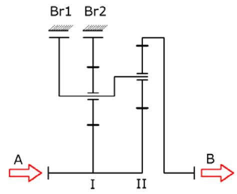
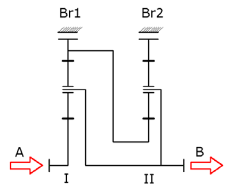
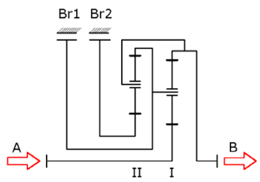
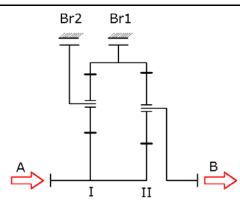
Tab. 1. Main parameters of both component gear trains

Mark	t_{I}	t_{II}	$z_{1\mathrm{I}}$	$z_{2\mathrm{I}}$	$z_{3\mathrm{I}}$	$z_{1\mathrm{II}}$	$z_{2\mathrm{II}}$	$z_{3\mathrm{II}}$
S36V6	3.5	4.555	14	17	49	**20**	**35**	**92**
S16V1	1.714	1.604	42	15	72	53	16	85
S33V4	1.714	4.437	42	15	72	16	27	71
S13V3	3.5	1.553	14	17	49	47	13	73
S12V2	4.55	1.553	20	35	91	47	13	73
S55V5	1.714	3.4	42	15	72	15	18	51

The tooth numbers respect the assembly conditions (conditions of coaxiality, adjacency and conjunction).

Kinematic diagrams of acceptable solutions are shown in the Tab. 2.

Tab. 2. Kinematic diagrams of acceptable solutions

No	Desig nation	Kinematic diagram
1.	S36V6	
2.	S16V1	
3.	S33V4	
4.	S13V3	
5.	S12V2	
6.	S55V5	

The transmission ratios and efficiencies have been calculated for all acceptable solutions in both cases: with brake Br1 active, and with brake Br2 active. The results are presented in Table 3.

The transmission ratio with active brake Br1 i_{Br1} and transmission ratio with active brake Br2 i_{Br2} are defined by using the adopted tooth number (Table 3). Also, the basic efficiency η_0 was calculated as a function of the tooth numbers of all gears [5, 9], and presented in Table 3.

The efficiency with active brake Br1 η_{Br1} and the efficiency with active brake Br2 η_{Br2} were calculated as a function of ideal torque ratios and basic efficiencies [5].

Tab. 3. Transmision ratios and efficiencies

Mark	i_{Br1}	i_{Br2}	η_{0I}	η_{0I}	η_{Br1}	η_{Br2}
S36V6	4,5	-4,55	0,973	0,985	0,979	0,985
S16V1	-4,464	4,407	0,976	0,978	0,962	0,976
S33V4	-4,438	4,422	0,976	0,980	0,980	0,959
S13V3	4,5	-4,436	0,973	0,973	0,979	0,934
S12V2	-4,55	4,573	0,985	0,973	0,985	0,969
S55V5	4,4	-4,475	0,976	0,974	0,980	0,924

All solutions provide the required transmission ratios and also present high efficiency values in both directions of output shaft rotation.

The optimal solution is then selected by the designer according to technological and economical demands, i.e. manufacturing costs.

This is achieved by analysis of kinematic diagrams, Tab. 2. Priority is given to a design which does not require drilled shafts, and this is achieved with layout S36V6. The power flow through this transmission solution is presented in Fig. 5 and in Fig. 6.

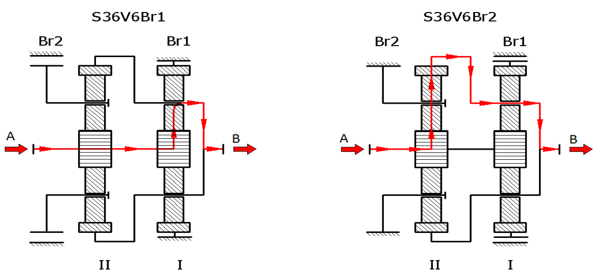


Fig. 5. Power flow through the transmission when the brake Br1 is activated (a) and when the brake Br2 is activated (b)

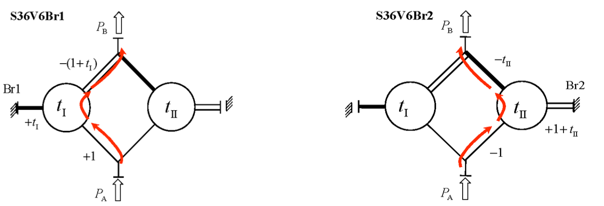


Fig. 6. Symbolic view with the power flow for solution S36V6: a) Br1 is activated, b) Br2 is activated

In this design. both brakes are acting on single external shafts. It is obvious that in both situations, i.e. by activating any brake, only one stage works is operational (two-shaft operating mode), while the other revolves idly. Because of that, power wastage occurs in only one PGT stage and there is only one power sink.

4. CONCLUSION

Two-speed planetary gear trains with four external shafts, composed of two simple planetary gear trains are presented in this paper, including a systematization of their kinematic structures and layout variants. Due to their characteristics, they are applicable in systems which require transmission ratio change under load.

This paper presents a quick determination of the structure and important basic parameters of two-speed planetary gear trains. This is enabled by using DVOBRZ, a computer program developed for examination of two-speed planetary gear trains. The procedure is followed by a numerical example dealing with the application to a railway vehicle where two directions of rotation at the same speed are necesary.

All possible schemes obtained by program are realized and main parameters are defined. The most appropriate scheme is chosen by analyzing obtained schemes according to technological demands.

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