Sanjin Troha
Assistant Professor
University of Rijeka
Faculty of Engineering
Croatia

Jelena Stefanović-Marinović
Associate Professor
University of Niš
Faculty of Mechanical Engineering
Serbia

Željko Vrcan
Assistant Professor
University of Rijeka
Faculty of Engineering
Croatia

Miloš Milovančević
Associate Professor
University of Niš
Faculty of Mechanical Engineering
Serbia

Selection of the Optimal Two-Speed Planetary Gear Train for Fishing Boat Propulsion

Planetary gear trains (PGTs) offer numerous benefits when compared to conventional gear trains, e. g. multi-carrier planetary gear trains may be built by linking the shafts of different component planetary gear trains. A two-carrier two-speed PGT consisting of two coupling shafts and four external shafts is a special type of multi-carrier PGT. This type of compound gear train has many important characteristics, the most notable being the possibility to change the transmission ratio under load. This paper presents a method for quick determination of the structure and important basic parameters of two-speed planetary gear trains that meet the predefined transmission requirements. A computer program developed for the examination of two-speed planetary gear trains DVOBRZ is used for this purpose. A numerical example of the procedure is provided, dealing with the application of a two-speed PGT as a fishing boat reduction and reversing gearbox.

Keywords: *planetary gear trains, two speed, structure, parameters, fishing boat propulsion, reduction and reversing gearbox.*

1. INTRODUCTION

Planetary gear trains (PGTs) offer numerous advantages when compared to conventional gear trains, the most notable of which is a reduction in mass and size for the same power rating. Because of that, their use has been significantly expanding in a variety of branches in mechanical engineering, e. g. in the vital parts of helicopters, tracked vehicles, and all kinds of agricultural, mining and other earth-moving machinery. For example, planetary reduction gears may be used to operate a two rope grab as pointed out in [1]. Furthermore, harmonic drives in humanoid robots may be successfully substituted by a low backlash planetary gear box [2]. PGTs as a whole, and especially compound multi-carrier PGTs represent a very extensive area of technical knowledge [3], and therefore compound PGTs are usually divided into simple and higher [3]. Two-carrier PGTs are considered simple compound gear trains, while three-carrier, four-carrier, and multi-carrier PGTs in general, are considered higher compound gear trains.

Multi-carrier PGTs are built by linking the shafts of different component gear trains. A special multi-carrier PGT type is a two-speed two-carrier PGT consisting of two coupling shafts and four external shafts. This type of compound PGT has many important characteristics, the most observable of which is the possibility to change the transmission ratio under load.

Two-speed two-carrier PGTs have not been investigated systematically until now. The review of 15 schemes of reversible transmissions of this type was presented in [4] with approximate values of the transmission ratio and efficiency. Some sporadic reviews of selected schemes have been carried out in [5-8], while [9] presents the characteristics of some schemes, without including the methodology for the selection of the optimal scheme. The torque method developed by Arnaudov is referred to as the principal tool of systematic analysis of multi-carrier PGTs in [10-12], and an explanation of the method is also given there. The torque method is definitely universal and contributes to clarity, since it holds not only for two-carrier PGTs, but also for multi-carrier PGTs as the authors pointed in [11]. Complex PGTs may also be analysed using classical methods, the two most common of which are the Willis (analytical) method and the Kutzbach-Smirnov (graphical) method. The torque method combines the accuracy of the Willis analytical method with the clarity of the graphic method of Kutzbach-Smirnov. The torque method can be used to determine not only the transmission ratio, but also the magnitude and direction of power flows, and therefore to determine the efficiency of a given compound PGT.

The significant systematic research carried out between 2006 and 2011 is the basis of the methodology for the selection of optimal PGT compound structures, which was implemented in the *DVOBRZ* computer program [13].

All the possible schemes of these transmissions and their working regimes included in the program *DVOBRZ* have been explored by Troha in [14] and [15]. The choice of a variant of a two-speed PGT intended for use as a fishing boat transmission was discussed in [16].

The purpose of this paper is to demonstrate the capabilities of the computer program *DVOBRZ* that has been developed for the analysis and optimization of two-speed PGTs, followed by an example in which the optimal solution determined by the structure and

Received: December 2019, Accepted: February 2020
Correspondence to: Dr Sanjin Troha
University of Rijeka, Faculty of Engineering,
Vukovarska 58, 51000 Rijeka, Croatia
Email: stroha@riteh.hr
doi:10.5937/fme2002397T

important parameters of the two-speed two-carrier PGTs is obtained.

2. TWO-SPEED PLANETARY GEAR TRAINS

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.

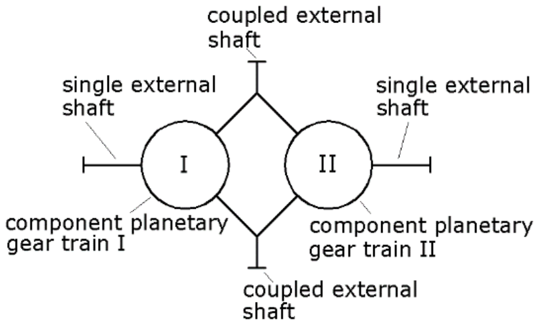


Fig. 1 Planetary gear train with four external shafts (compound train)

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). These symbols simplify the representation of PGTs as the train shafts are represented by lines of different thickness and a circle. The sun gear shaft 1 is represented by a thin line, the ring gear shaft 3 by a thick line and the carrier shaft *h* by two parallel lines. 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. Because of its characteristics, it is applied in mobile and stationary machines without limitations in shaft speed and torque.

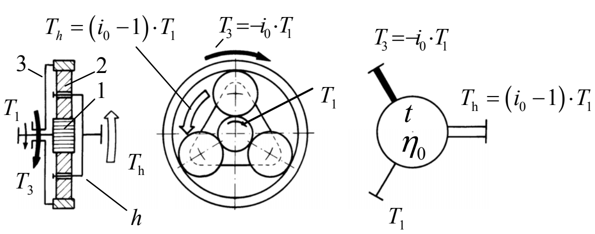


Fig. 2 Wolf-Arnaudov symbol and torque ratios of the basic type of PGT [3]

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

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

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. A simple PGT offers three transmission ratios. The first one is trivial, with the planet gears locked to the planet carrier, causing the PGT to rotate as a solid block. The second transmission ratio is achieved by locking the planet carrier and transmitting power from the sun to the ring gear, causing the input and output elements to rotate in the opposite direction with the transmission ratio $i = z_3/z_1$. The final transmission ratio is achieved by locking the ring gear and causing the power to be transmitted from the sun gear to the planet carrier with the ratio $i = 1\text{-}(z_3/z_1)$. In this case, the input and output shaft rotate in the same direction.

Those transmission ratios will be substantially different for any simple PGT, therefore another solution, such as the compound PGT must be used for a reversing gearbox. The two-stage compound PGT is an obvious choice as it is the simplest form of compound PGT.

Two component trains can be joined in a total of 36 possible ways [13], however due to isomorphism, there are only 12 different ways which result in PGTs with four external shafts, Fig. 3.

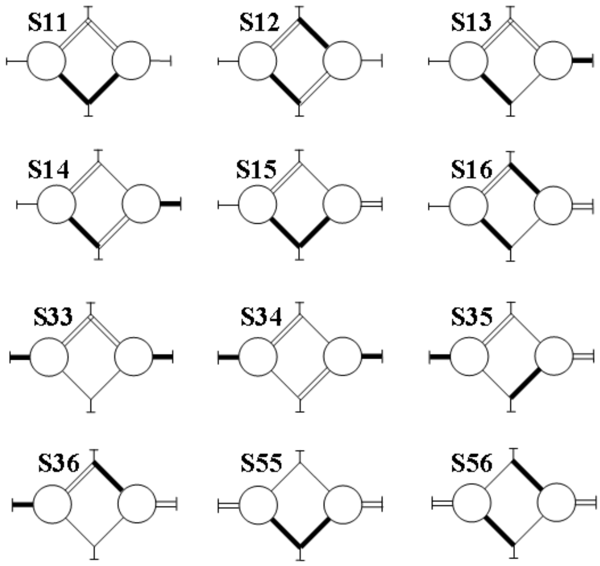


Fig. 3 Systematization of all schemes of two-carrier planetary gear trains with four external shafts [12, 13]

In every presented scheme it is possible either to place a brake, or connect the prime mover, or the powered 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. This is an important advantage as these transmissions can be used as multiple-speed gearboxes.

DVOBRZ, a computer program used to select the optimal variant from similar multi-speed PGTs is described in detail in [13]. The program is based on a principle of synthesis of a two-speed PGT briefly described in section 3.

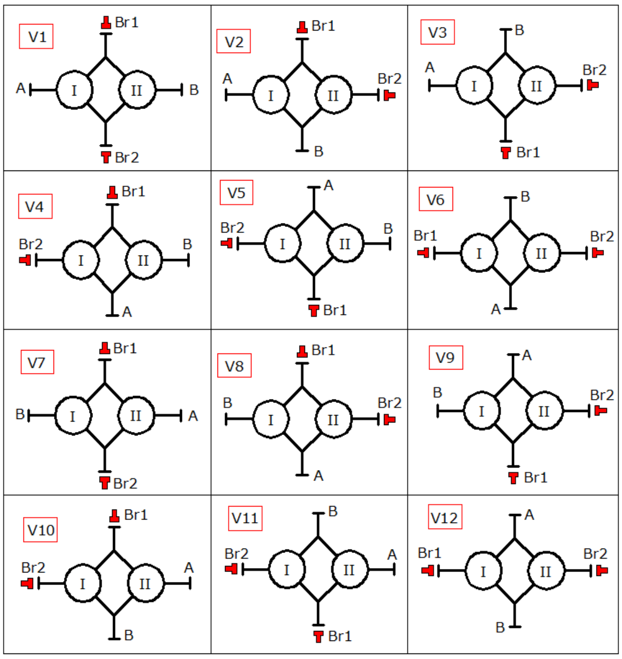


Fig. 4 Layout variants of two-carrier planetary gear trains with four external shafts [14, 15]

3. THE PRINCIPLE OF A TWO-SPEED PLANETARY GEAR TRAIN PROGRAM SYNTHESIS

The required transmission ratios are labelled as i_{k1} and i_{k2}. All possible variants that meet the imposed constraints are searched for by the program, and a set of feasible solutions is created. First of all, it is necessary to determine the intervals of ideal torque ratios $t_{Imin}\dots t_{Imax}$ and $t_{IImin}\dots t_{IImax}$. Also, the intervals of the required transmission ratios I_1 and I_2 should also be known. A graphical explanation of this principle is shown in Figure 5. The relationship of the transmission ratio and the ideal torque ratios representation (thus forming surface areas) is also given in Fig. 5. The corresponding transmission ratio intervals I_1 and I_2 ($i_{k1} \in I_1$ and $i_{k2} \in I_2$ must be fulfilled) are given on the vertical axis in Figure 5. This data corresponds to an unfounded two-speed PGT used as an example.

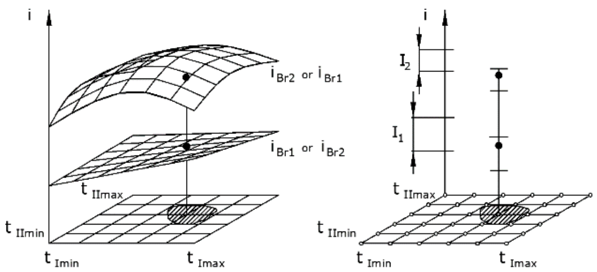


Fig. 5 The principle of synthesis of a two-speed planetary gear train [15]

The principle of PGT program synthesis can be explained in several steps. First the values of the transmission ratio functions for each design structure of a two-speed PGT are calculated for every possible combination of ideal torque ratios. The program then checks if the calculated transmission ratios fall within the user-defined I_1 and I_2 intervals, and if such torque ratio combinations exist, the program extracts them as possible solutions. 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. [13].

4. NUMERICAL EXAMPLE AND DISCUSSION

A fishing boat transmission will be used as a practical example for the selection of two-speed PGTs. The prime mover is a high speed, four-stroke diesel engine with a working speed range of 1800÷2100 min^{-1}. The engine is run in heavy duty application conditions with unlimited working hours during the year. The reduction gear should provide a transmission ratio of i=1,5…7.

The two-speed PGT is applicable in this case as a main propeller drive gearbox situated between the engine and the driveshaft [8]. Therefore, the transmission should operate with the transmission ratio i=5 in one direction of rotation, and i=-5 in the other direction of rotation.

Based on the requirements and assumptions listed above, the *DVOBRZ* program lists six possible solutions for two-speed PGT. After taking into consideration that the required transmission ratios are i_1= 5 and i_2= -5, solutions have been found with transmission ratios in the ranges i_1= 4,9…5,1 and i_2= -5,1…-4,9.

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 four with the planet tooth numbers written in bold (cells in Table are shading) and three where the number of planets is written in regular. 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 [18], and presented in Table 1.

Table 1 Main parameters of both component gear trains

Mark	t_I	t_{II}	z_{1I}	z_{2I}	z_{3I}	z_{1II}	z_{2II}	z_{3II}
S36V6	4,053	5	19	**29**	77	16	**32**	80
S16V1	2	1,553	24	**12**	48	47	13	73
S33V4	2	5	24	**12**	48	16	**32**	80
S13V3	2	4,053	24	**12**	48	19	**29**	77
S12V2	5	1,52	16	**30**	80	50	13	76
S55V5	4,053	2	19	**29**	77	24	**12**	48

The tooth numbers respect the assembly conditions (conditions of coaxiality, adjacency and conjunction). The optimal solution is selected by the designer according to technological and economical demands. However, in order to choose a set of appropriate solutions, it is necessary to analyse the work regimes of all acceptable solutions.

A detailed description of the first presented variant was given in [16] and will be briefly discussed here. After a brake is activated, only one stage remains in active mode, while the other goes to idle mode. A symbolic view with the power flow during the activation of Br1 and respectively Br2, is shown in Fig.6.

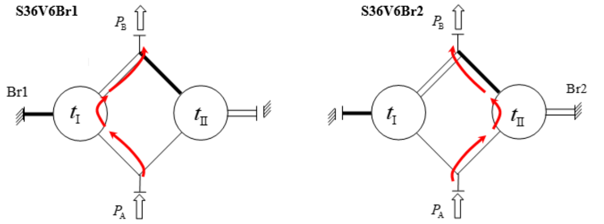


Fig. 6 Symbolic view of the power flow for solution 1: a) Br1 is activated, b) Br2 is activated.

With solution 2, S16V1, both planetary stages work in two-shaft active mode in both regimes, regardless of which brake is activated, as can be seen in Fig. 7.

Solution 3, S33V4 is specific, as with the activation of the brake Br1 only one component gear train works in active mode, while activation of the brake Br2 results in internal power circulation in the compound gear train has. The symbolic view with the power flow for both cases is shown in Fig. 8.

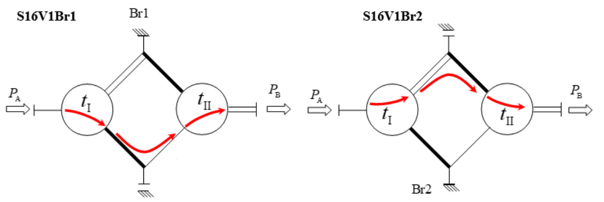


Fig. 7 Symbolic view of the power flow for solution 2: a) Br1 is activated, b) Br2 is activated.

Solution 4, marked as S13V3, has similar characteristics: with activation of the brake Br1 the ring gear of the first stage is locked and only this component gear train operates in active mode, while the second gear train is idling. Activation of the brake Br2 locks the ring gear of the second component gear resulting in internal power circulation in the compound gear train. Symbolic view with the power flow for both cases of this solution is shown in Fig. 9.

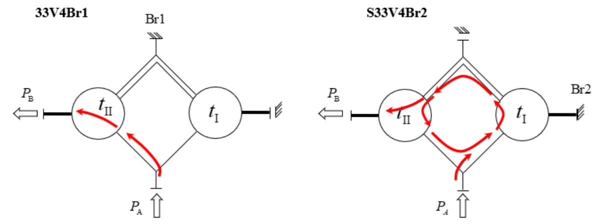


Fig. 8 Symbolic view of the power flow for solution 3: a) Br1 is activated, b) Br2 is activated.

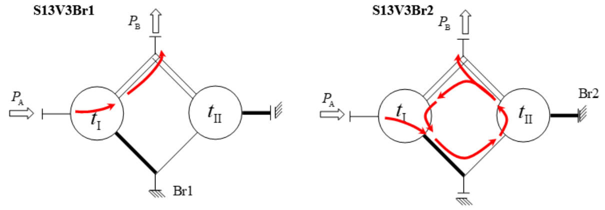
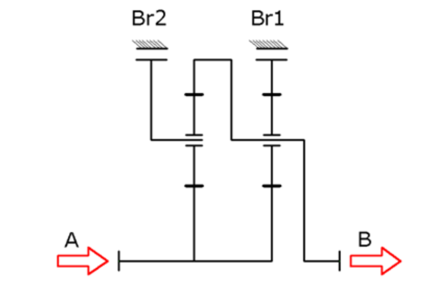
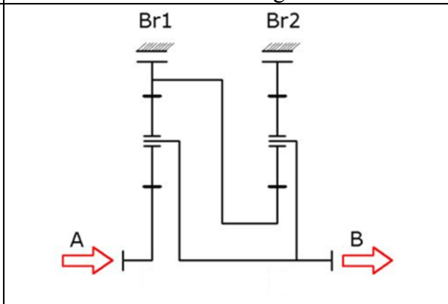
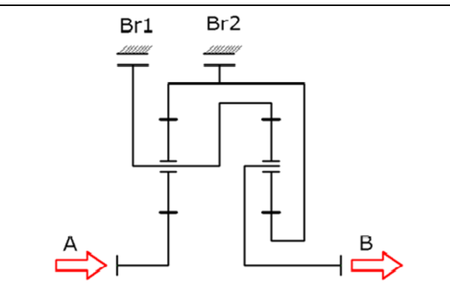
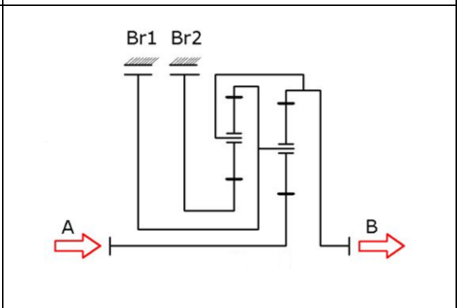
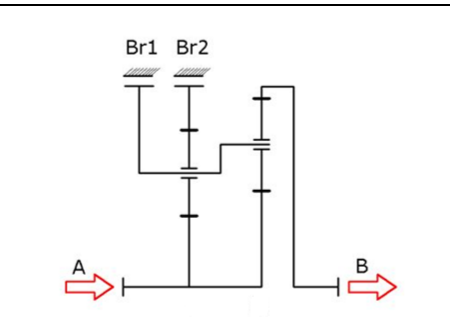
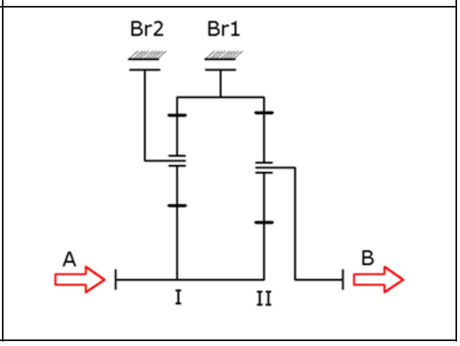


Fig. 9 Symbolic view of the power flow for solution 4: a) Br1 is activated, b) Br2 is activated.

Table 2 Acceptable solutions (A-input shaft; B-output shaft)

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

The same occurs with solution 5, S12V2, as can be seen in Fig. 10.

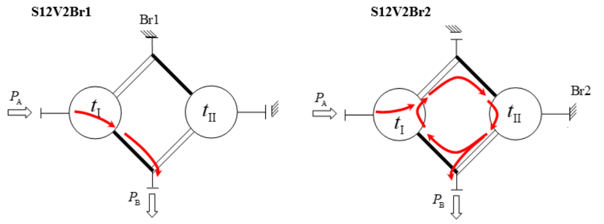


Fig. 10 Symbolic view of the power flow for solution 5: a) Br1 is activated, b) Br2 is activated.

The sixth solution, S55V5, also manifests internal power circulation through compounded planetary gear trains when the brake Br2 is activated. A symbolic view of the power flow in both cases for this solution is given in Figure 11.

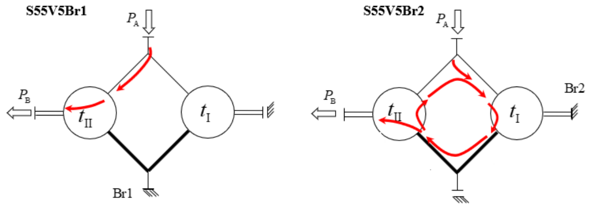


Fig. 11 Symbolic view with the power flow for solution 6: a) Br1 is activated, b) Br2 is activated.

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 brake Br1 active i_{Br1} and transmission ratio with brake Br2 active 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 [13, 19], and presented in Table 3.

The efficiency with brake Br1 active η_{Br1} and the efficiency with brake Br2 active η_{Br2} was calculated as a function of ideal torque ratios and basic efficiencies [13].

The optimal solution in these circumstances may be chosen by analysing the data given in Table 3. It can be expected that the gearbox will be used in the forward direction of travel for most of the time, therefore the mode in which the PGT achieves its greatest efficiency should be used for forward travel.

Therefore, the brake that gives the greatest efficiency will be activated for forward travel, resulting in greater fuel efficiency. Except for solution 2, S16V1, all other solutions achieve greater efficiency with brake Br1 active.

The first solution, S36V6 is chosen as the optimal solution according to the efficiency criterion. This transmission has the added bonus of being the best possible solution from a manufacturing standpoint, as an examination of the kinematic sheme shows that no drilled shafts (shafts going through each other) are needed for this transmission to operate.

This solution has been already discussed in [20] without defining the tooth numbers. Because of that, the basic efficiency in [20] was not defined, but a standard value of $\eta_0 = 0.98$ was adopted. In this paper, the tooth numbers of all gears from all acceptable solutions are indicated, so the basic efficiencies can be calculated. The power flow through the optimal transmission solution is presented in Fig. 12.

Table 3 Transmission ratios and efficiencies

Mark	i_{Br1}	i_{Br2}	$\eta_{0\mathrm{I}}$	$\eta_{0\mathrm{I}}$	η_{Br1}	η_{Br2}
S36V6	5,053	-5	0,983	0,982	0,986	0,982
S16V1	-5,106	4,932	0,969	0,973	0,953	0,969
S33V4	-5	5	0,969	0,982	0,982	0,948
S13V3	5,053	-4,923	0,969	0,983	0,986	0,922
S12V2	-5	4,947	0,981	0,973	0,981	0,966
S55V5	5,053	-7,102	0,983	0,969	0,986	0,945

Both brakes are acting on single external shafts. It is obvious that in both situations, i.e. by activating any brake, only one component gear train works in the active operating mode (two-shaft operating mode), while the other operates in idle mode. Because of that, power losses occur in only one component PGT and there is only one power exit.

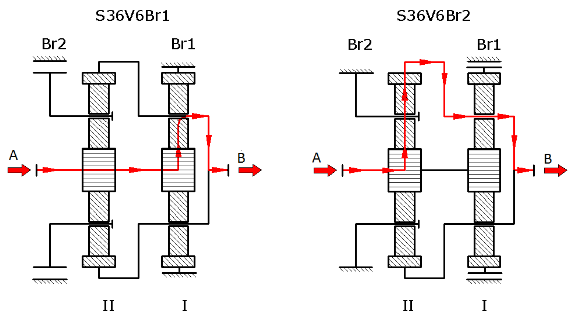


Fig. 12 Power flow through the optimal transmission solution: a) with brake Br1 activated, (b) with *brake Br2 activated*

It is interesting that solution 1, transmission S36V6 is already being used as a reversing gearbox in heavy tracked earth-moving machines [21], while solution 2, S16V1 is mentioned as a fishing boat propulsion gearbox in [22], however S36V6 has been proven to be a better solution.

In order to select the optimal transmission solution for heavy mobile machines and for marine applications, it is necessary to consider the conditions under which they are expected to operate.

Heavy mobile machines usually do not need to travel at high speeds, however considerable torque must be developed in order to move them. Therefore, multistage PGTs built to a constant transmission ratio are used in conjunction with a hydrostatic or electrical drive from the prime mover. Such drives can benefit from a compound PGT as described in this paper to be used as an additional reversing gearbox, as it results in a simplified hydrostatic or electrical drive system and reduced manufacturing costs. Heavy machinery is expected to receive regular mainteinance by specialists, so a design with clutch type brakes would not pose any maintenance problems in this case.

On the other hand, a marine transmission, especially one intended for a heavy-duty application like a fishing boat main gearbox, must be designed with simplicity, reliability and serviceability in mind. For a compound

PGT as described in this paper, this can be achieved only by using band brakes in order to effect transmission ratio changes. Band brakes are of relatively simple design, and what is most important, band brakes can be changed or repaired even at sea without having to lift the gearsets from the transmission casing, which is not the case with clutch type brakes.

It should be pointed out that clutch brakes require less space and maintenance, however in the event of a breakdown, the whole gearset has to be lifted out from the casing in order to service the brakes. However, the design with clutch type brakes could be viable for pleasure craft where it is expected that the craft will be regularly serviced by specialist mechanics.

In addition to fishing boats and heavy mobile machinery, reversing PGTs could be also applied to robotics. PGTs are usually used in robotics with an absolute encoder for positioning, however such encoders limit the movement to 360 degrees or less. An application of the compound PGTs described in this paper could lead to the use of incremental encoders, enabling a movement in excess of 360 degrees, which could be advantageous for some applications. Furthermore, a reversing PGT could enable the use of a more economical drive motor and motor conntroller as the motor would be turning in one direction only. In this particular case it would be recommended to build reversing PGTs with clutch-type brakes due to their smaller dimensions and faster engagement time.

5. CONCLUSION

This paper deals with two-speed PGTs with four external shafts. Due to their characteristics, they are particularily suited to applications which require the transmission ratio to be changed under load, such as fishing boat main reduction gearboxes.

This paper presents a procedure for the rapid determination of the structure and important basic parameters of two-speed PGTs. This is enabled by means of a computer program developed for the examination of two-speed two-carrier PGTs *DVOBRZ*. By considering the working conditions, requirements can be defined for the *DVOBRZ* program to list possible solutions for two-speed PGTs. The acceptable solutions are presented with kinematic schemes, main parameters and symbolic view complete with power flow diagrams. The optimal solution was selected according to the criterion of maximum efficiency, as fuel effciency is a vital requirement for fishing boats. When selecting optimal solutions for other applications, other criteria may be applied.

The procedure for two-speed two-carrier PGT structural and parametric optimization implemented in the *DVOBRZ* program is original and enables a quick selection of optimal solution for predefined transmission requirements.

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ИЗБОР ОПТИМАЛНОГ ДВОСТЕПЕНОГ ПЛАНЕТАРНОГ ПРЕНОСНИКА ЗА ПОГОН РИБАРСКОГ БРОДА

С. Троха, Ј. Стефановић-Маринoвић, Ж. Врцан, М. Милованчевић

У поређењу са класичним зупчастим преносницима, планетарни зупчасти преносници имају низ предности, на пример могућност добијања сложених планетарних преносника повезивањем више вратила различитих планетарних јединица. Двостепени планетарни преносник са два носача сателита који има два спојна вратила и четири спољашња је специјални тип сложеног планетарног зупчастог преносника. Овај преносник има значајне карактеристике, међу којима је најбитнија могућност промене преносног односа под оптерећењем. У овом раду је представљен метод за брзо одређивање структуре и основних параметара двобрзинског планетарног зупчастог преносника при задатим условима преноса. То је омогућено применом компјутерског програма развијеног за истраживање двостепених зупчастих преносника ДВОБРЗ. Нумерички пример који илуструје овај метод се бави избором двостепеног планетарног зупчастог преносника за примену на рибарским броду, као редуктора и мењача.