Application of Two-Carrier Planetary Gear Trains to Shifting Gearboxes

Sanjin Troha, Željko Vrcan, Kristina Marković, Milan Tica, Miroslav Milutinović

Abstract— **This article deals with two-carrier planetary gearsets having two connecting and two external shafts, which may operate as two-speed gearboxes when brakes are placed on two external shafts. Analysis of the S16WN(S/E) gearbox is performed using the torque method, the specific torques are determined on all shafts, and then used to calculate both transmission ratios. The kinematic scheme of the gearbox has been synthesized, and the relative power flows have been identified. Efficiency functions are calculated for both gear ratios. The graphical representation of the transmission ratio functions for both gear ratios was used to obtain the diagram of the shifting capabilities of the S16WN(S/E) gearbox. The importance and utility of such diagrams and the relevant equations for the determination of ideal torque ratios of the component gear trains is explained. The size of the transmission ratio domain suggests that the S16WN(S/E) gearbox has a wide area of possible applications.**

Index Terms— **planetary gear train, gearbox, two-carrier, two-speed, ideal torque ratio, transmission ratio, efficiency**

I. Introduction

Recent developments in mechanical power transmission and the transition to greener technologies result in conventional gearboxes being replaced with planetary gear train (PGT) based gearboxes. PGTs type 2k-h variant A (according to Kudryavtsev) are most suitable as building blocks for such transmissions due to their compact size, coaxiality of input and output shafts, and division of torque to several planets in mesh, resulting in a considerably lighter design. Simple component PGTs are built for transmission ratios i = 3…8 (…12) and must be combined for other transmission ratios [1-6]. Such simple PGTs have been thoroughly researched, however focus has moved to multi-carrier PGTs which can provide more than one transmission ratio [1-10].

Two-speed PGT based transmissions have not been sufficiently explored. For example, in [1] the author provides corrected transmission ratio intervals for reversing PGT transmissions and the procedure for determining the intervals for all such two-speed PGTs [11,12]. Two-carrier PGTs with fixed transmission ratios are successfully used as replacements for worm drives, i.e., as belt conveyor or elevator main gearboxes. The possibilities of applicaton of shifting two-speed PGTs in machine tools and marine propulsion units have been discussed in [13-18].

Multiple PGTs may be used to build automatic planetary gearboxes that contain any number of component PGTs [19-21]. PGTs are also very convenient for electric and hybrid vehicle applications due to high input RPM combined with large variations of input shaft speed and torque [22-24].

The purpose of this article is to introduce the methodology for obtaining diagrams of shifting capabilities based on the selected variant of the S16WN(S/E) PGT, as its broad shifting interval enables various practical applications. The most important parameters for different combinations of transmission ratios will also be discussed.

II. 2. Two-Carrier PGT with Two Connecting and Four External Shafts

A simple PGT of the 2k-h type, variant A is displayed in Fig. 1. The primary parts of this PGT are the sun gear 1, ring gear 3 and planet carrier h on which the planets 2 rotate. The sun gear, ring gear and planet carriers rotate on concentric shafts, and every one of them may be locked. The torque ratio between them remains constant regardless of whether the PGT operates in two-shaft or three-shaft mode [5,9]. The PGT in Fig. 1 is also shown using a Wolf-Arnaudov symbol which uses lines of different thickness to encode PGT components. The sun gear is represented by a thin line, the ring gear by the thick line, and the planet carrier is represented by a thick line as it is the summary member (locking it provides a negative transmission ratio). These symbols considerably simplify the analysis of complex PGTs.

A two-carrier PGT with two connecting and four external shafts may be obtained by connecting the shafts of two 2k-h PGTs (Figure 1). By connecting alternate members of the first and second component PGT, twelve possible structures can be achieved (Fig. 2) [1]. By installing brakes on two of the four external shafts, a gear-changing structure is achieved, capable of obtaining two different transmission ratios by alternate engagement of the brakes [15], thus

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S. Troha is with the University of Rijeka, Faculty of Engineering, Vukovarska 58, 51000, Croatia (sanjin.troha@riteh.uniri.hr).
Ž. Vrcan is with the University of Rijeka, Faculty of Engineering, Vukovarska 58, Rijeka, Croatia (zeljko.vrcan@riteh.uniri.hr).
K. Marković is with the University of Rijeka, Faculty of Engineering, Vukovarska 58, Rijeka, Croatia (kristina.markovic@riteh.uniri.hr).
M. Tica is with Faculty of Mechanical Engineering, University of Banja Luka, Bulevar vojvode Stepe Stepanovića 71, Banja Luka, Bosnia and Herzegovina (milan.tica@mf.unibl.org).
M. Milutinović is with Faculty of Mechanical Engineering, University of East Sarajevo, Vuka Karadžića 30, East Sarajevo, Republic of Srpska, Bosnia and Herzegovina (miroslav.milutinovic@ues.rs.ba).

changing the direction of the power flow inside the gear train.

It is possible to brake both single and coupled shafts, resulting in a total of 12 different brake configurations, which are named according to cardinal directions (Fig. 3). Knowing that there are 12 different structures, and that 12 different brake layouts may be applied to each structure, the theoretical number of two-carrier PGTs is 144, however isomorphous properties reduce this to 120 [11]. It is assumed that power flows from input shaft A to output shaft B. Brakes are marked with Br1 and Br2. The naming convention will be explained on the S13WS(N/E) gearset. S13 means structure 13, WS means that input is from the western shaft with output to the southern shaft, and (N/E) means that shafts N or E may be braked, resulting in gear trains S13WS(N) or S13WS(E).

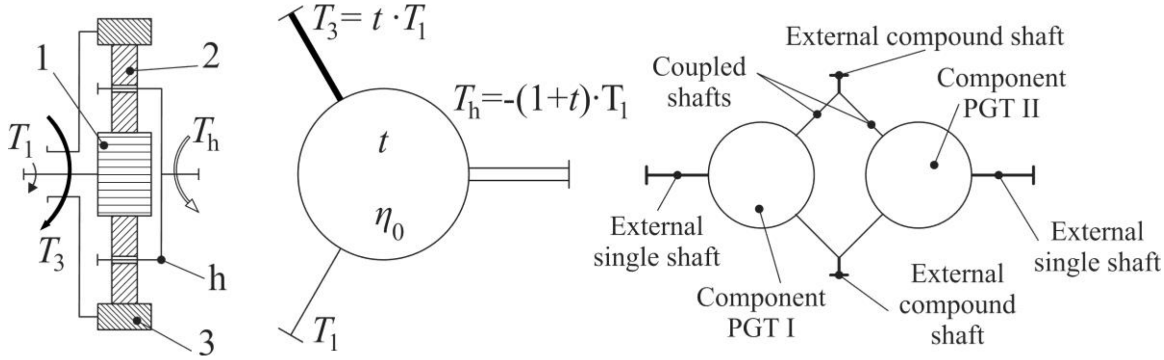

Fig. 1. Simple PGT type 2k-h variant A (left); Wolf-Arnaudov symbol (center); General arrangement of two-carrier PGT with two coupled and four external shafts (right).

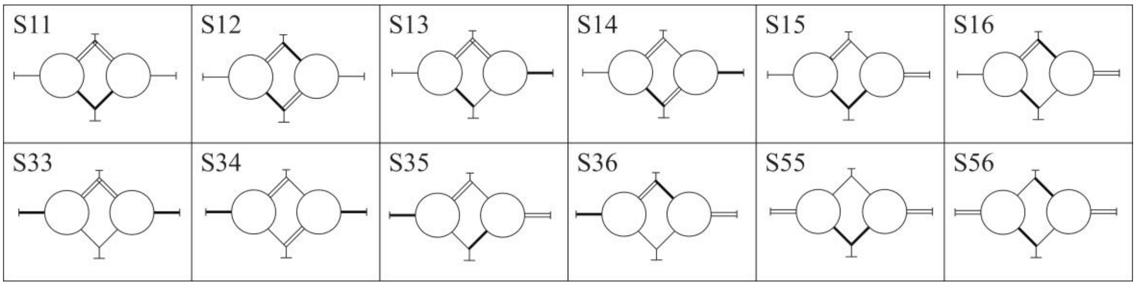

Fig. 2. Systematization of all schemes of two-carrier planetary gear train with four external shafts

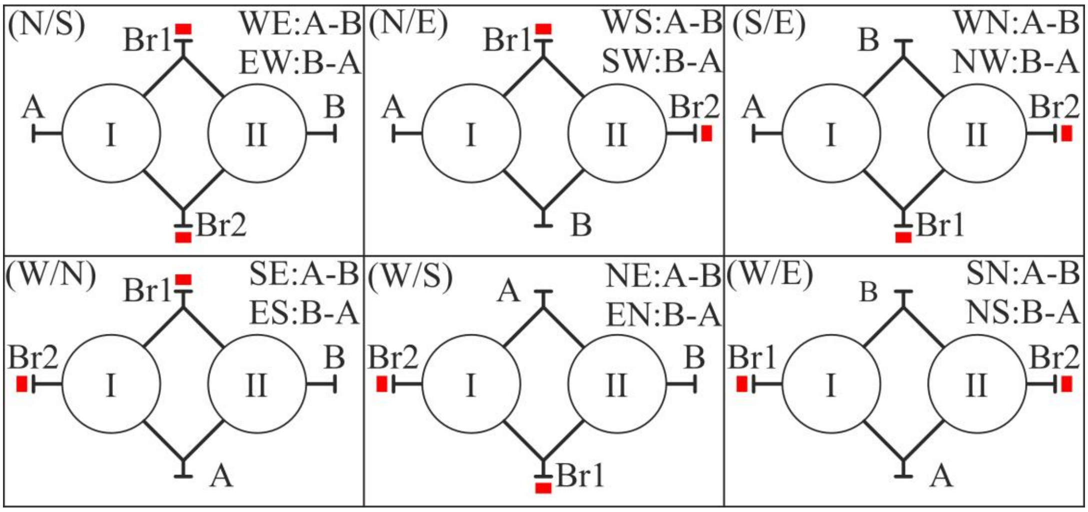

Fig. 3. Listing of brake layouts for two-carrier planetary gear trains with four external shafts.

III. Torque Method Analysis of Planetary Gear Trains

PGT analysis is a complex issue that demands a conceptual solution defining the input and output links between the internal components to achieve the desired transmission ratio range, and lever analogy is used for these purposes [10,13,21] The S16WN(S/E) PGT has been selected to demonstrate the analysis process, with its kinematic scheme shown in Fig. 4.

Input is through the sun gear of component PGT I, while ring gear II and planet carrier I are connected to the output shaft. Ring gear I is connected to sun gear II and both can be locked by brake Br1. Planet carrier II is connected to brake Br2. The power flow for both brake engagements is also shown in Fig. 4.

The torque method described in [10] was used to determine the specific torques on the PGT shafts, comprising the ideal and real specific torques which account for internal losses by considering the relative efficiencies of the component PGTs. The functions of the appropriate torques are given next to its respective shafts on the structural symbol (Fig. 5). Torque analysis is very important as it is necessary to accurately calculate the internal power flows in the planetary gearbox to determine power losses in meshing or the presence of power circulation, discarding an otherwise valid solution at the concept stage.

When operating with brake Br1 on (Figure 5, left), only gear train I is active, while gear train 2 runs idle. Active power (red line) enters at the sun wheel I and exits through the connected shafts of carrier I and ring gear II. The relative power (green line, meshing losses) [5] is directed towards ring gear I. With brake Br2 on (Figure 5, right), both gear trains are active, however power is split through two paths, the second going through ring gear I to re-join through ring gear II. The relative power path follows from sun gear to ring gear in both component PGTs. The specific torques were used to determine the transmission ratio functions and efficiency ratios for both gear ratios.

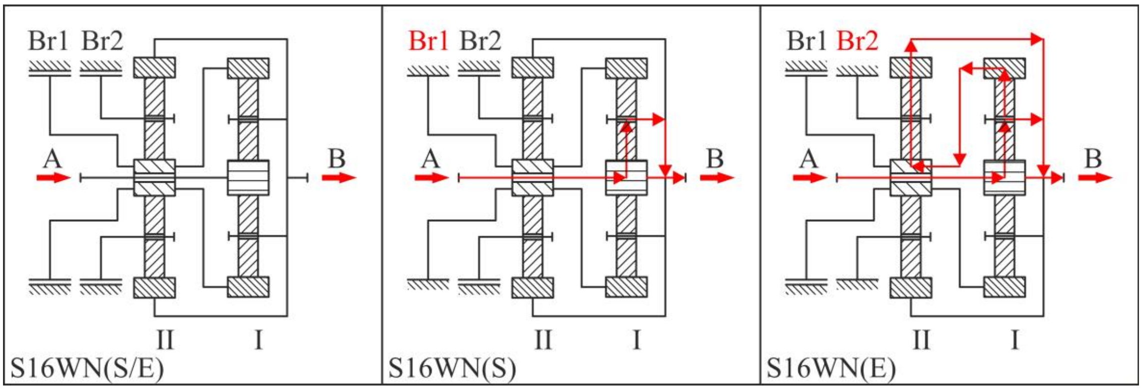

Fig. 4. Kinematic scheme and power flow of S16WN(S/E) PGT

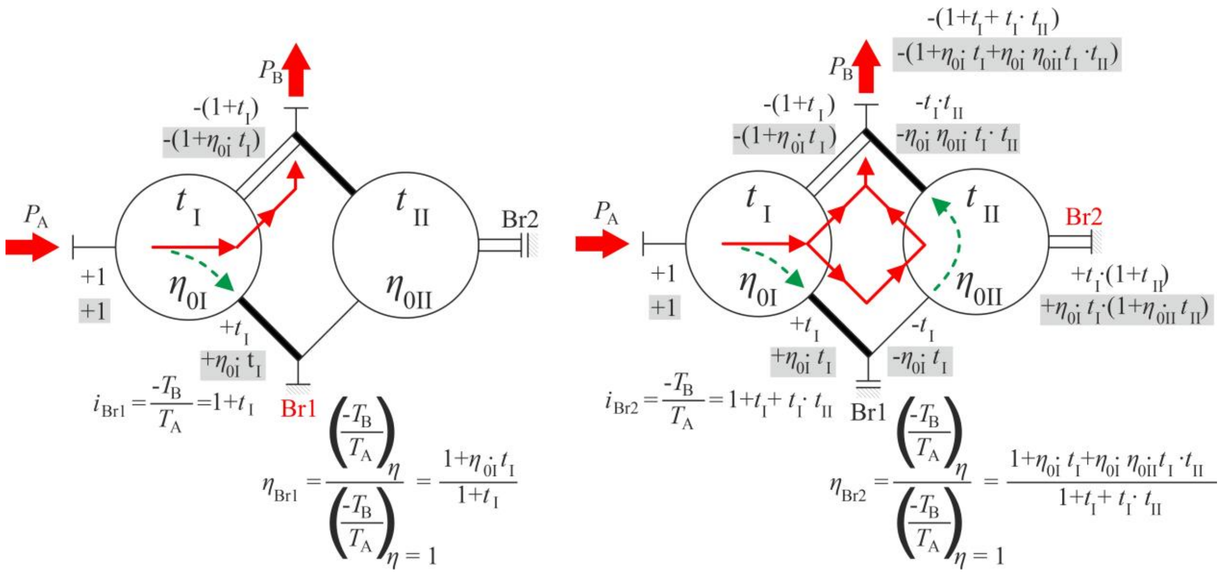

Fig. 5. Torque analysis of S16WN(S/E) PGT

IV. 4DIAGRAMS OF TRANSMISSION RATIOS AND SHIFTING CAPABILITIES

Graphic representations of the transmission ratio functions were generated in the domains of ideal torque ratios t_I and t_{II} as surfaces (Figure 6). The values of t_I and t_{II} are in a design limited interval for PGTs with three planet gears from 2 to 12.

The values are calculated by rising vertical lines from all points of the discrete domain (t_I, t_{II}) to the intersections with the surfaces of the transmission ratio functions $i_{Br1}(t_I, t_{II})$ and $i_{Br2}(t_I, t_{II})$, with three such lines shown in Figure 6. The vertical lines are applicates the values of which are transferred to the x and y axis of the calculated diagram, creating a set of all valid combinations of i_{Br1} (x-axis) and i_{Br2} (y-axis) for PGTs having three planet gears per component geartrain. The shaded area in the x-y plane clearly shows which transmission ratio pairs may be achieved by the gear train, and which brake is used to achieve that transmission ratio. These diagrams are called shifting capability diagrams for two-speed planetary gear trains [11]. Every point of these diagrams presents a combination of valid transmission ratios, and the solution set may take any polygonal shape. Therefore, it is not correct to say that the shifting set is in the interval i_{Br1min} to i_{Br1max} and i_{Br2min} do i_{Br2max} as additional limitations may appear. These diagrams should be accompanied by equations for the calculation of t_I and t_{II} using the transmission ratio functions by expressing t_I and t_{II} as functions of i_{Br1} and i_{Br2}. It should also be mentioned that these diagrams show that PGTs may achieve both negative and positive transmission ratios, and that they might shift from reduction gears to multiplicator gears with specific combinations of t_I and t_{II}.

A systematic search of a set of such diagrams yields all the PGT configurations that can provide the required transmission ratio pair, and this provides a major assistance to the gear train designer. This can be performed using the DVOBRZ software which can provide an optimal PGT configuration within minutes.

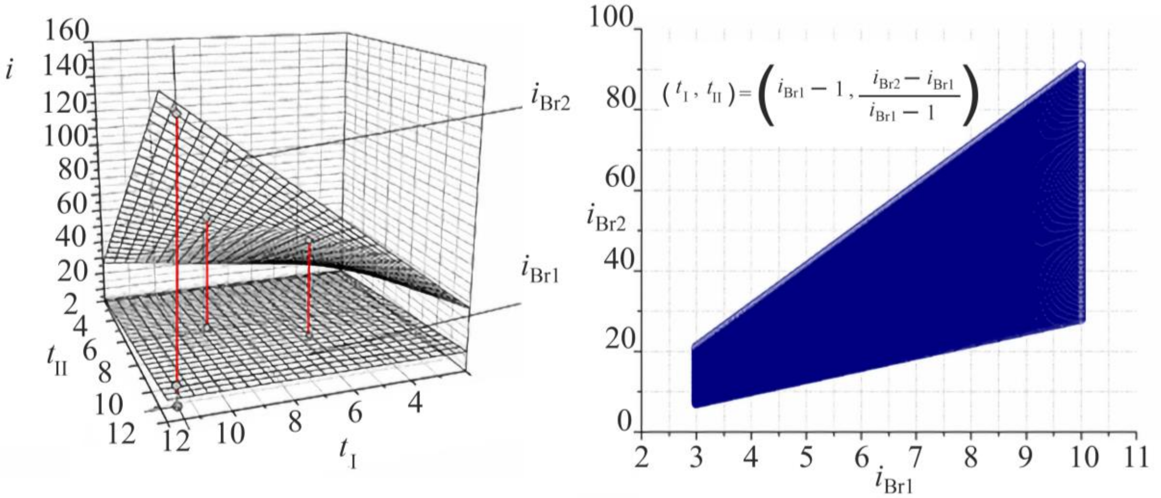

Fig. 6. Graphic representations of the transmission ratio functions in the domain of ideal torque ratios t_I and t_{II} for S16WN(S/E) PGT, Diagram of shifting capabilities for S16WN(S/E) PGT Torque analysis of S16WN(S/E) PGT

V. Conclusion

An overview of the methods for the analysis of two-speed, two-carrier PGT transmissions with four external and two connecting shafts, and two brakes was provided in this article, together with a naming convention system and analysis procedure. The analysis procedure is demonstrated on the S16WN(S/E) gear train, together with the shaft torque and efficiency calculation. The results are then used to create the transmission ratio diagram and diagram of shifting capabilities for the gearset. The procedure for the creation of the diagram of shifting capabilities is laid out, and the possibilities for the application of the diagram of shifting capabilities are discussed. Finally, the advantages of its application for planetary gear train designers are explained.

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