Reversible Planetary Gearsets Controlled by Two Brakes, for Internal Combustion Railway Vehicle Transmission Applications

Milan Tica[1], Željko Vrcan[2], Sanjin Troha[2], Dragan Marinković[3,4]

[1]University of Banja Luka, Faculty of Mechanical Engineering, Vojvode Stepe Stepanovića 71, BA-78000 Banja Luka, Bosnia and Herzegovina
milan.tica@mf.unibl.org

[2]University of Rijeka, Faculty of Engineering, Vukovarska 58, HR-51000 Rijeka, Croatia
zeljko.vrcan@riteh.hr, sanjin.troha@riteh.hr

[3]University of Niš, Faculty of Mechanical Engineering, Aleksandra Medvedeva 14, RS-18000 Niš, Serbia

[4]Technische Universität Berlin, Department of Structural Mechanics and Analysis, Strasse des 17. Juni 135, 10623 Berlin, Germany
dragan.marinkovic@tu-berlin.de

Abstract: Internal combustion powered railway vehicles require extremely rugged and reliable transmissions. While high-powered applications use electric or complex hydrodynamic transmissions, in the low to medium power range of railway vehicles, transmissions derived from ordinary highway truck or bus automatic transmissions and hydromechanical transmissions, are used. Modern railway vehicles must be able to operate in both directions at the same speed, which is particularly important when units from the same series are connected in a multiple unit lash-up. Simple hydrodynamic and mechanical transmissions are commonly used in such vehicles, but planetary gear trains are also suitable for the application, either as an output gearbox or as the main transmission gearbox in the case of simpler vehicles. This planetary gearbox is designed to provide two equal transmission ratios, however with the output shaft rotating in different directions. Design priority should be given to clutch-type brakes for compactness and reliability, however band brakes should have priority for ease of maintenance is a priority. Additionally, the gearbox design should give priority to boxes that do not experience power circulation, and do not require hollow shafts or complex planet carrier arrangements. The application of planetary gearbox designed according to the guidelines laid out in this paper would simplify the design and manufacture of hydrodynamic, hydromechanical and mechanical transmissions for railway vehicles.

Keywords: Railway vehicle; mechanical power transmission; reversible planetary gearbox

1 Introduction

Internal combustion engines (ICEs), whether using spark or compression ignition, have been in railway use since the early 1930s as an improvement over steam technology. However, as the crankshaft of an ICE cannot be connected directly to the wheels as in the case of a steam engine, it requires a transmission which will transfer power to the wheels while matching the speed and torque range of the engine to the requirements of the wheels [1]. It must also enable the ICE to disconnect from the wheels when the vehicle is stationary, so that the ICE can idle or be started up from standstill.

As the ICE is a constant power machine, the transmission must provide an amplification of the output torque in the range of 10:1 to 5:1, depending on the application, with an adequate number of transmission ratios to enable the whole power range of the engine to be used. Furthermore, the internal efficiency of the transmission must be as high as possible to meet environmental demands and reduce operating costs [2-8]. The transmission must also change gears without reducing or interrupting the engine power output. It must also be reversible, and in some special applications (e.g., shunting) it must be even able to reverse under load. The transmission should be robust, reliable, and low maintenance [9]. Finally, the transmission should be unaffected by climatic extremes and exposure to snow.

Currently, mechanical, hydrostatic, hydrodynamic, electrical, and combined hydro-mechanical transmissions are in use, depending on the application. Modern electrical transmissions are highly reliable, but their weight and size make them applicable only to power ranges beyond 500 kW. Hydrodynamic transmissions have been successfully applied up to 2 MW, although the transmissions tend to become large and heavy in the higher power ranges. Hydrostatic transmissions are simple, reliable and offer continuously variable transmission ratios, however oil cooling issues limit them to approximately 150 kW. Mechanical transmissions use simple, robust, and reliable automotive-derived solutions. Designs using friction and dog clutches are applicable up to 200 kW, however they are usually limited to lightweight maintenance vehicles. More robust designs using planetary gearsets must be coupled to a torque converter and can be used up to 600 kW.

Modern railway vehicles must be able to operate in both directions at the same speed, especially when operated in multiples, as they might end up coupled “head to head” or “tail to tail” and still need to move in the same direction. Furthermore, the transmission should have the same efficiency in both directions. Electrical and hydrostatic transmissions achieve this easily, however on hydrodynamic, mechanical, and hydro-mechanical transmissions a separate gearbox or gearbox stage is required, however this solution creates a slightly different transmission ratio in one direction of travel.

This can be resolved with the application of a two-carrier reversible planetary gearbox controlled by two brakes, which will have the same transmission ratio in each of its directions of rotation, preferably with the same or very similar efficiency [10-18]. This gearbox can be used on its own with a torque converter for low powered applications, as the output box for a main planetary gearbox in medium powered applications, or as the output stage of high-powered hydrodynamic transmissions.

It is known that the application of planetary gear trains (PGTs) offers considerable advantages in relation to conventional gear train solution, resulting in expanded possibilities for application in mechanical engineering solutions. Some areas of application have been mentioned in [19-22], with railway vehicles being another area of application as they are required to move both forwards and backwards and in multiple at the same speed. Hydrodynamic and PGT transmissions are commonly used in such vehicles, however an additional mechanical stage is required to reverse. This mechanical stage can be easily replaced by an output planetary gearbox consisting of a compound two-speed PGT created by appropriately linking the shafts of the elements of its component planetary gear trains (stages). For the purposes of the research presented in this article, two-speed two-carrier PGTs with four external shafts composed of two PGTs of the basic type were considered. The internal structures of the researched gear trains were laid out. As there is a considerable number of all possible schemes and layout variants, a systematization was performed, and appropriate labelling was devised. A software program for numerical simulation and calculation of PGT parameters was developed to determine the structure and important basic parameters of the component gear trains and the whole gear train, based on the application constraints. The explanation of the operation of this software is followed by a numerical example in which the optimal two-speed planetary gear train that meets predefined transmission requirements is selected and then defined by the numbers of teeth of the component PGT sun gears, gear modules and transmission ratios. The position of the PGT in the transmission chain and the operating conditions of railway vehicles determine the input data for the computer program that defines the structure and important parameters of the component planetary gear trains. The acceptable transmission solutions for the selected application were generated using this specially developed computer program. The final selection between the solutions generated by the program, is performed by comparative analysis [23] [24].

2 Planetary Gear Boxes

The two-carrier compound PGT (compound train) is the simplest form of compound PGT (Fig. 1).

This simplest form of compound PGT has two component trains and four external shafts (Fig. 1).

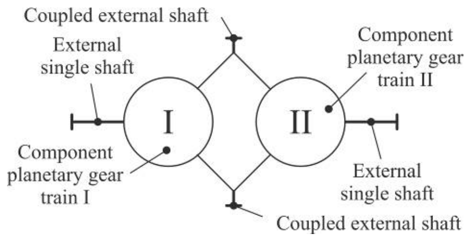

Figure 1
Planetary gear train with four external shafts (compound train)

It is created by connecting two shafts of one component PGT with two shafts of the other component PGT. The resulting four external shafts can then be subdivided to two coupled shafts and two single external shafts. Both component PGTs 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) [12, 15, 18]. This is the most common type of PGT, and it is commonly used in engineering applications as a single stage transmission, or as a building block for higher compound planetary gear trains. The application of this component PGT offers several advantages over other types, notably its efficiency, small overall dimensions and mass, and relatively low manufacturing costs due to the relatively simple production process.

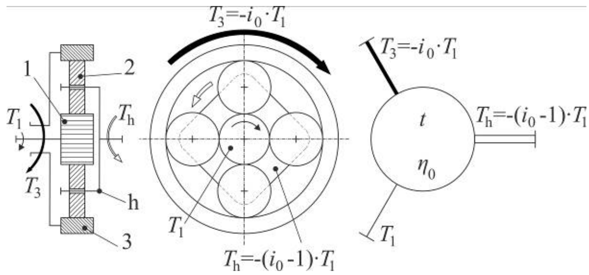

Figure 2
Wolf-Arnaudov symbol and torque ratios of the basic type of PGT [8]

The torque loads on the planetary gear train shafts are 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 defined as

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

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. The transmission ratio depends on whether the sun gear, ring gear or carrier is the locked element. It is possible to connect two component trains in a total of 36 possible ways (schemes) [12, 15], however this is reduced by isomorphism to only 12 different schemes resulting in PGTs with four external shafts, Fig. 3. 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 configurations (layout variants), the cardinal directions of the input and output shafts being used for naming (Fig. 4). The power flow and kinematic characteristics of the gearbox are influenced by the placement of brakes on different shafts, enabling their use as multiple-speed gearboxes.

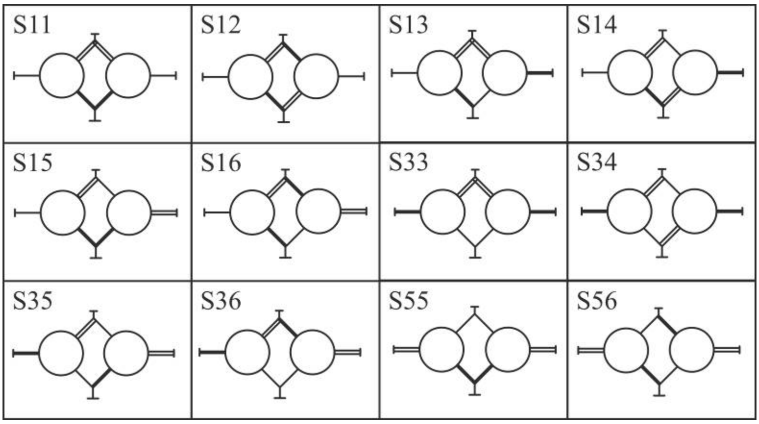

Figure 3

Systematization of all schemes of two-carrier PGTs with four external shafts [10, 15]

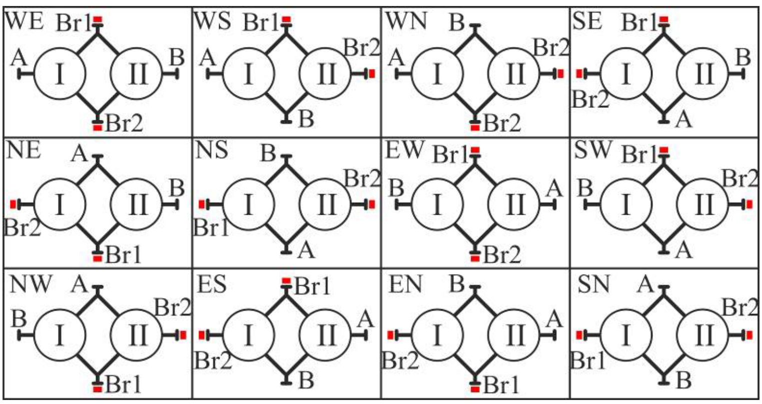

Figure 4

Layout variants of two-carrier planetary gear trains with four external shafts [15] [16]

3 Numerical Example and Discussion

The computer program DVOBRZ is used to select the optimal variant from similar multispeed PGTs, and it operates by synthesizing two-speed PGTs [14]. The program can determine the values of the parameters of valid component PGTs, such as basic gearset efficiency η_0, gear pitch diameters $d_{1,2,3}$, gear modules m etc. for every valid combination of gear tooth numbers $z_{1,2,3}$, as functions of the ideal torque ratios $t_{\mathrm{I,II}}$. Furthermore, for every valid combination of component PGTs for the selected scheme and layout variant, the parameters of the component PGT, such as i_{Br1}, i_{Br2} (transmission ratios with the respective brake Br1 or Br2 activated), η_{Br1} and η_{Br2} (efficiencies with the respective brake activated), are also calculated as functions of the ideal torque ratios t_{I} and t_{II} and then stored (Fig. 5). Every layout variant has two separate functions relating the transmission ratios as i_{Br1}, i_{Br2} to the ideal torque ratios t_{I} and t_{II} (Fig. 5, left). The program effectively seeks the ideal torque ratios that will place the overall transmission ratios of the compound PGT into the desired range (Fig 5, right).

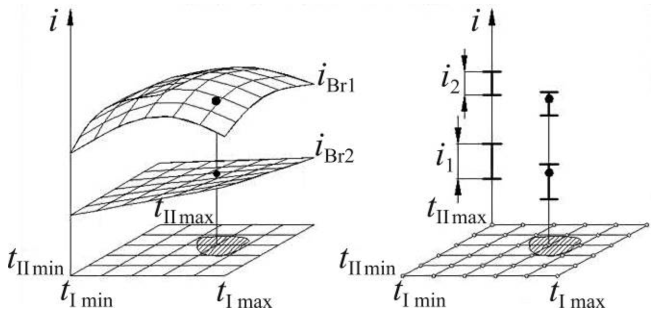

Figure 5

Operating principle of the DVOBRZ program

All valid solutions are compared according to the defined relevant criteria, such as minimal radial dimensions, maximum equivalent efficiency etc. [9]. In this case, a railway vehicle transmission will be used to demonstrate the selection of two-speed PGTs. Considering 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 outer input datum is the frequency of operation of each transmission ratio: $\alpha_{1\mathrm{i}}$ = 0.5 (50%) and $\alpha_{2\mathrm{i}}$ = 0.5 (50%). The program is now tasked with finding the optimal solution according to efficiency, while considering the operating conditions of the vehicle. The DVOBRZ program then lists six possible solutions for two-speed PGTs, using the previously listed requirements and assumptions. The main solution parameters are summarized in Table 1 while the kinematic schemes of acceptable solutions are shown in Figs. 5-10. The main parameters include the numbers of teeth of all gears and ideal torque ratios for both component gear trains.

The program DVOBRZ determines the ideal torque ratios for both gear trains. The tooth numbers of all gears were adopted based on the ideal torque ratios [17] and presented in Table 1. All tooth numbers respect the assembly conditions of coaxiality, adjacency and conjunction. Component gearsets have either 3 or 4 (z_2 marked in bold in Tab. 1) planets. The transmission ratios and efficiencies have been calculated for all acceptable solutions for cases of either brake being active. The results are presented in Table 1, with transmission ratios i_{Br1} and i_{Br2} defined by means of the using the adopted tooth number. Also, the basic efficiency η_0 was calculated as a function of the tooth numbers of all gears [14] [18]. 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 [13]. It was determined that all solutions provide the required transmission ratios and present high efficiency values in both directions of output shaft rotation.

Table 1
Main parameters of both component gear trains

Mark	t_I	t_{II}	i_{Br1}	i_{Br2}	z_{1I}	z_{2I}	z_{3I}	z_{1II}	z_{2II}	z_{3II}	η_{Br1}	η_{Br2}
S36SN	4.053	5	5.053	-5	19	**29**	77	16	**32**	80	0.986	0.982
S16WE	2	1.553	-5.106	4.932	24	**12**	48	47	13	73	0.953	0.969
S33SE	2	5	-5	5	24	**12**	48	16	**32**	80	0.982	0.948
S13WN	2	4.053	5.053	-4.923	24	**12**	48	19	**29**	77	0.986	0.922
S12WS	5	1.52	-5	4.947	16	**30**	80	50	13	76	0.981	0.966
S55NE	4.053	2	5.053	-5.102	19	**29**	77	24	**12**	48	0.986	0.945

The valid solutions are presented in Figs. 6-11, showing the general kinematic layout and power flow for cases of brake Br1 or brake Br2 being on. The power flow is marked by the red line, while A denotes power input and B denotes power output. The kinematic layouts shown in Figs. 6-11 have been obtained using computer simulation and torque method analysis combined with appropriate transformations. A diagram linking the ideal torque ratios t_I and t_{II} of the component PGTs (ring to sun gear tooth number ratio) to the required transmission ratio i_{rev} is provided for each valid solution. The torque ratios can then be used to determine the required tooth numbers for both component PGTs by just inverting the assembly conditions. Design constraints (bearing solutions, planet gear rotational speed, noise etc.) place the acceptable values of i_{rev} into an area where the ideal torque ratios exceed 1.5.

The operation of the S36SN gearbox is shown in Figure 6. The gearbox is idle with both brakes off (Fig. 6a). With brake Br1 on, a positive transmission ratio is obtained with only component PGT I active, while component PGT II is idle (Fig. 6b). With brake Br2 on, a negative transmission ratio is obtained, only component PGT II is active, while component PGT I is idle (Fig. 6c). The ideal torque ratios for transmission ratio i_{rev} = +/- 5, are t_I = 4 and t_{II}= 5 (Fig. 6d). The acceptable transmission ratio range is i_{rev} = +/- 2.5…+/- 12.

The operation of the S16WE gearbox is displayed in Figure 7. The gearbox is idle with both brakes are off (Fig. 7a). With brake Br1 on, a negative transmission ratio is obtained with both component PGTs in active operation (Fig. 7b). With brake Br2 on, a positive transmission ratio is obtained, also with both component PGTs in active operation (Fig. 7c). The ideal torque ratios for transmission ratio i_{rev} = +/- 5, are t_I = 1.5 and t_{II} = 2 (Fig. 7d). The acceptable transmission ratio range is i_{rev} = +/- 4…+/- 5.

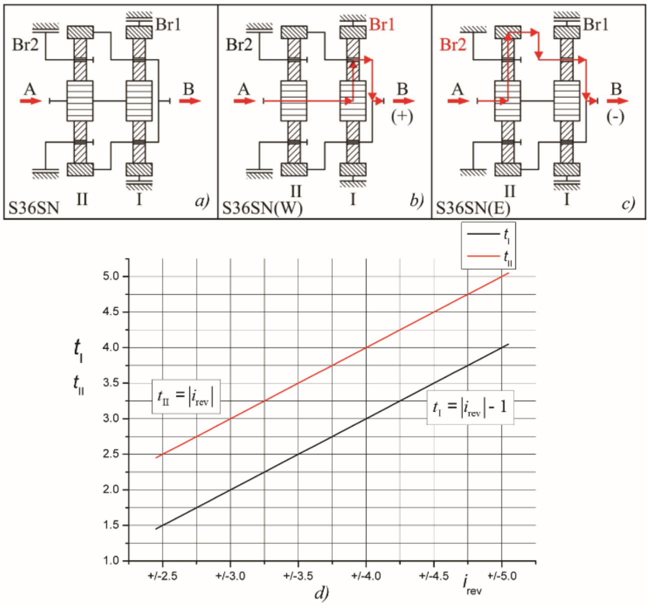

Figure 6
Kinematic arrangement and power flow (top row), and relation of transmission ratio i_{rev} to ideal torque ratios t_I and t_{II} (bottom) for S36SN gearbox

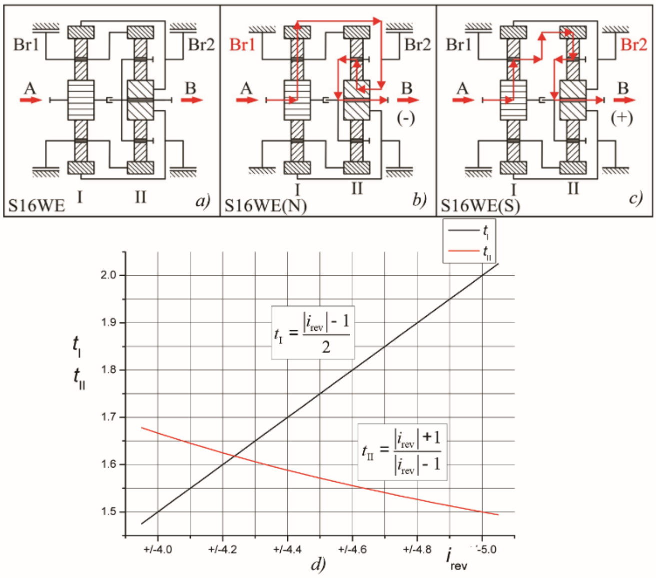

Figure 7

Kinematic arrangement and power flow (top row), and relation of transmission ratio i_{rev} to ideal torque ratios t_I and t_{II} (bottom) for S16WE gearbox

The operation of the S33SE gearbox is seen in Figure 8. The gearbox is idle with both brakes off (Fig. 8a). With brake Br1 on, a negative transmission ratio is obtained with only component PGT II in active operation, while component PGT I remains idle (Fig. 8b). With brake Br2 on, a positive transmission ratio is obtained, with both component PGTs in active operation (Fig. 8c). The ideal torque ratios for transmission ratio i_{rev} = +/- 5, are t_I = 2 and t_{II}= 5 (Fig. 7d). The acceptable transmission ratio range is i_{rev} = +/- 4…+/- 12.

The operation of the S13WN gearbox is shown in Figure 9. The gearbox is idle with both brakes off (Fig. 9a). With brake Br1 on, a positive transmission ratio is obtained with only component PGT I active, while component PGT II is idle (Fig. 9b). With brake Br2 on, a negative transmission ratio is obtained with both component PGTs active. The power flow goes from A to B, with power circulation inside the gearbox as shown in the figure (Fig. 9c). The ideal torque ratios for transmission ratio i_{rev} = +/- 5, are t_I = 4 and t_{II}= 1.5 (Fig. 19d). The acceptable transmission ratio range is i_{rev} = +/- 2.5…+/- 5.

The S12WS gearbox is shown in Figure 10. With both brakes off, the gearbox is idle (Fig. 10a). With brake Br1 on, a negative transmission ratio is obtained with only component PGT I active, while component PGT II idles (Fig. 10b). With brake Br2 on, a positive transmission ratio is obtained with both component PGTs in active operation. Power flows from A to B, with power circulation inside the gearbox as shown in the figure (Fig. 10c). For example, for a transmission ratio of ±5, the ideal torque ratios are t_I=5 and t_{II}=1.5. The ideal torque ratios for transmission ratio i_{rev} = +/- 5, are t_I = 5 and t_{II}= 1,5 (Fig. 10d). The acceptable transmission ratio range is i_{rev} = +/- 1.5…+/- 5.

The operation of the S55 NE gearbox is seen in Figure 11. The gearbox is idle with both brakes off (Fig. 11a). With brake Br1 on, a positive transmission ratio is obtained with only component PGT II active, while component PGT I is idle (Fig. 11b). With brake Br2 on, a negative transmission ratio is obtained with both component PGTs active. The power flows from A to B, with power circulation inside the gearbox as shown in the picture (Fig. 11c). The ideal torque ratios for transmission ratio i_{rev} = +/- 5, are t_I = 2 and t_{II}= 4 (Fig. 11d). The acceptable transmission ratio range is i_{rev} = +/- 4…+/- 5.

The optimal solution is then selected by the designer according to technological and economical demands, such as manufacturing costs. This is achieved by analyzing the kinematic diagrams, (Figs. 6a to 11a). Priority is given to designs which do not require drilled shafts or complex planet carrier arrangements, and layout S36SN satisfies both conditions. In this design, both brakes are acting on single external shafts, and it is obvious that in both situations, i.e. with any of two brakes activated only component PGT is operational (two-shaft operating mode), while the other remains idle. Because of this, power wastage occurs in only one PGT stage and there is only one power sink.

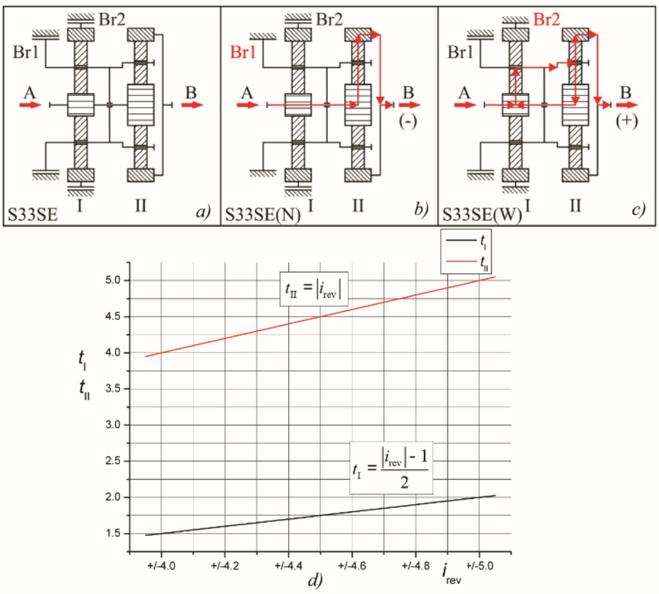

Figure 8
Kinematic arrangement and power flow (top row), and relation of transmission ratio i_{rev} to ideal torque ratios t_I and t_{II} (bottom) for S33SE gearbox

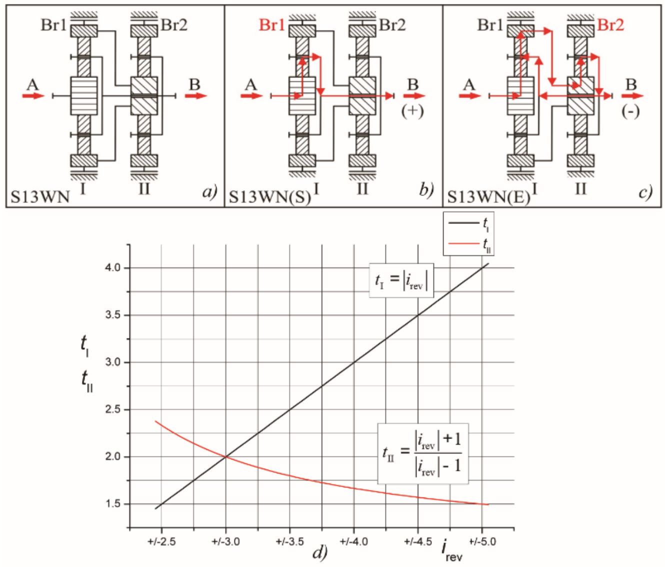

Figure 9

Kinematic arrangement and power flow (top row), and relation of transmission ratio i_{rev} to ideal torque ratios t_I and t_{II} (bottom) for S13WN gearbox

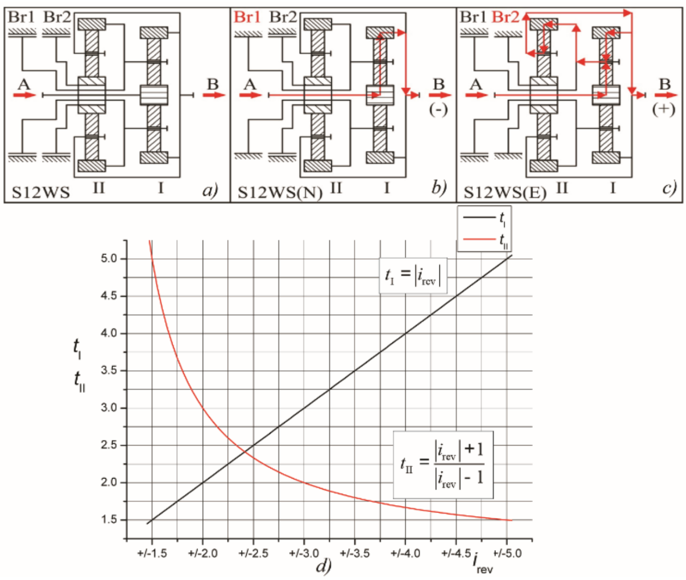

Figure 10

Kinematic arrangement and power flow (top row), and relation of transmission ratio i_{rev} to ideal torque ratios t_I and t_{II} (bottom) for S12WS gearbox

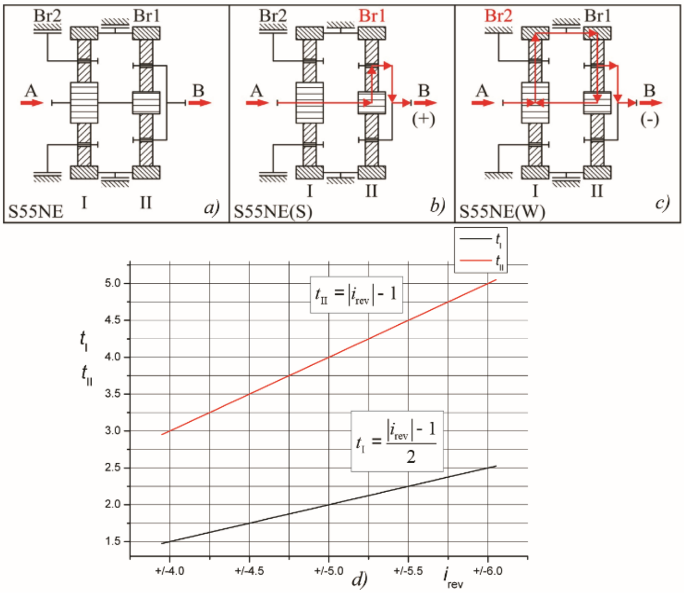

Figure 11
Kinematic arrangement and power flow (top row), and relation of transmission ratio i_{rev} to ideal torque ratios t_I and t_{II} (bottom) for S55NE gearbox

Conclusions

This paper covers two-speed planetary gear trains, with four external shafts controlled by two brakes, composed of two simple component PGTs, complete with a systematization of their kinematic structures and layout variants. Due to their characteristics, such gear configurations are applicable in systems which require the transmission ratio to be change under load, or without disconnecting the prime mover from the transmission. A concise determination of the structure and important basic parameters of two-speed planetary gear trains is also presented, enabled by the application of DVOBRZ, a computer program developed for the research of two-speed planetary gear trains. The procedure is explained by a numerical example dealing with the application for a railway vehicle, where two directions of rotation, at the same speed are necessary. All possible schemes obtained by program are then analyzed and the main parameters are defined. The most appropriate scheme is selected by ranking the obtained systems according to technological demands.

References

[1] Syzrantsev, V., Syzrantseva, K.: The Arc Teeth Semi-rolled Cylindrical Gear Meshing Geometry, Acta Polytechnica Hungarica, Vol. 19, No. 2, 2022, pp. 173-192

[2] Milovančević, M., Milčić, D., Andjelkovic, B., Vračar, Lj.: Train Driving Parameters Optimization to Maximize Efficiency and Fuel Consumption, Acta Polytechnica Hungarica, Vol. 19, No. 3, 2022, pp. 143-154

[3] Tigh Kuchak A. J., Marinkovic D., Zehn M.: Finite element model updating - Case study of a rail damper, Structural Engineering and Mechanics, Vol. 73, No. 1, 2020, pp. 27 -35

[4] Bhagat, P. R., Naz, F., Magda, R.: Role of Industry 4.0 Technologies in Enhancing Sustainable Firm Performance and Green Practices, Acta Polytechnica Hungarica, Vol. 19, No. 8, 2022, pp. 229-248

[5] Kuchak, A. J. T., Marinkovic, D., Zehn, M., 2021, Parametric Investigation of a Rail Damper Design Based on a Lab-Scaled Model, Journal of Vibration Engineering and Technologies, 9(1), pp. 51-60

[6] Fischer, S.: Geogrid reinforcement of ballasted railway superstructure for stabilization of the railway track geometry – A case study, Geotextiles and Geomembranes, Vol. 50, 2022, pp. 1036-1051

[7] Macura, D., Laketić, M., Pamučar, D., Marinković, D.: Risk Analysis Model with Interval Type-2 Fuzzy FMEA – Case Study of Railway Infrastructure Projects in the Republic of Serbia, Acta Polytechnica Hungarica, Vol. 19, No. 3, 2022, pp. 103-118

[8] Jover, V., Fischer, S.: Statistical analysis of track geometry parameters on tramway line no. 1 in Budapest, The Baltic Journal of Road and Bridge Engineering, Vol. 17, No. 2, 2022, pp. 75-106

[9] Jovanović, V., Janošević, D., Pavlović, J: Analysis of the Influence of the Digging Position on the Loading of the Axial Bearing of Slewing Platform Drive Mechanisms in Hydraulic Excavators, Facta Universitatis-Series Mechanical Engineering, Vol. 19, No. 4, 2022, pp. 705-718

[10] Müller, H. W.: Die Umlaufgetriebe. 2. Auflage. Springer Verlag, Berlin, 1998

[11] Kudriavtsev, V. N., Kirdyiashev, I. N.: Planetary Gears Handbook. Leningrad: Mashinostroenie, Leningrad, 1977

[12] Tkachenko, V.: Planetary Mechanisms. Optimal Design, Harkov: HAI, 2003 (in Russian)

[13] Arnaudov, K., Karaivanov, D.: The torque method used for studying coupled two-carrier planetary gear trains, Transactions of FAMENA, Vol. 37, No. 1, 2013, pp. 49-61

[14] Troha, S.: Analysis of a planetary change gear train's variants (in Croatian), PhD Thesis. Rijeka: Faculty of Engineering – University of Rijeka, 2011

[15] Jelaska, D. T.: Gears and Gear Drives. Wiley & Sons, Chichester, 2012

[16] Looman, J. Zahnradgetriebe. 3. Auflage. Berlin: Springer Verlag, 1996

[17] Troha, S., Vrcan, Ž., Karaivanov, D., Isametova, M.: The Selection of Optimal Reversible Two-speed Planetary Gear Trains for Machine Tool Gearboxes, Facta Universitatis-Series Mechanical Engineering, Vol. 18, No. 1, 2020, pp. 121-134

[18] Arnaudov, K., Karaivanov, D.: Planetary Gear Trains, 1st edition, CRC Press, Boca Raton, 2019

[19] Stefanović-Marinović, J., Vrcan, Ž., Troha, S., Milovančević, M.: Optimization of two-speed planetary gearbox with brakes on single shafts, Reports in Mechanical Engineering, Vol. 3, No. 1, 2022, pp. 94-107

[20] Pavlovic, A., Fragassa C.: Geometry optimization by FEM simulation of the automatic changing gear, Reports in Mechanical Engineering, Vol. 1, No. 1, 2020, pp. 195-205

[21] Vrcan, Ž.; Stefanović-Marinović, J., Tica, M., Troha, S.: Research into the Properties of Selected Single Speed Two-Carrier Planetary Gear Trains, Journal of Applied and Computational Mechanics, Vol. 8 No. 2, 2022, pp. 699-709

[22] Karaivanov, D., Troha, S.: Optimal Selection of the Structural Scheme of Compound Two-Carrier Planetary Gear Trains and their Parameters in: Recent Advances in Gearing: Scientific Theory and Applications, Radzevich, S. P. (ed.). Springer International Publishing, Basel, 2021, pp. 339-403

[23] Kurhan D., Fischer S.: Modeling of the Dynamic Rail Deflection Using Elastic Wave Propagation, Journal of Applied and Computational. Mechanics, Vol. 8, No.1, 2022, pp. 379-387

[24] Franulović, M., Marković, K., Trajkovski, A.: Calibration of Material Models for the Human Cervical Spine Ligament Behaviour Using a Genetic Algorithm, Facta Universitatis-Series Mechanical Engineering, Vol. 19, No. 4, 2021, pp. 751-765