Sanjin Troha
Neven Lovrin
Miloš Milovančević

ISSN 1333-1124

SELECTION OF THE TWO–CARRIER SHIFTING PLANETARY GEAR TRAIN CONTROLLED BY CLUTCHES AND BRAKES

UDC 621.833.65-027.231

Summary

In this paper, the methodology of establishing two-carrier planetary gear trains (PGT) equipped with a speed-changing system which contains breaks and clutches is considered. The reactive shaft is changed by means of breaks whereas the input and output shaft of the gear train are changed by clutches. Compound multi-speed planetary gear trains are obtained by a combination of two-carrier two-speed planetary gear trains controlled by only two breaks. The coding system of the two-speed planetary gear trains, which defines the connection between shafts of planetary gear sets and the shaft layout equipped with breaks, a driving and an operating machine is shown. It is established which variants of two-speed planetary gear trains are possible to combine by means of corresponding clutches in order to get two-variant or three-variant PGT. The methodology of choosing two-carrier PGT which are created by a combination of two or three two-speed variants within the frame of the same scheme is described as well as the example of finding an optimal three-speed PGT by means of the developed computer program 6-*brz* is shown.

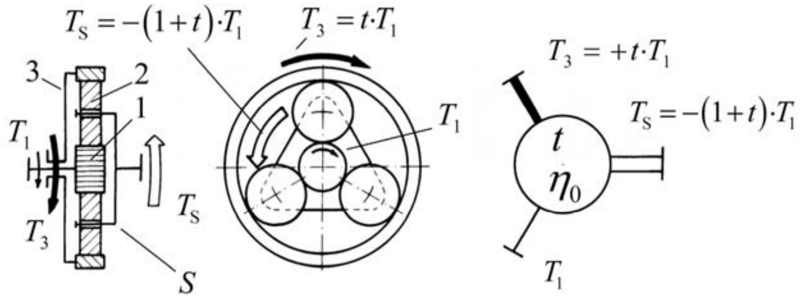
Key words: *shifting planetary gear train, planetary gear set, transmission ratio*

1. Introduction

Gearboxes in machines are frequently of a planetary design [1]. Mechanisms with varied change capabilities are created by connecting a few planetary gear sets in a whole with breaks and clutches on specific planetary gear set shafts. There are a number of scientific papers regarding shifting planetary gear trains (PGTs) of various structural schemes, equipped with several brakes and couplings that are activated in order to change speed. Some of these papers are [1-14]. There are also some patents that consist of different solutions of shifting PGTs [15-17]. These papers provide analyses of kinematic, geometric, energetic and structural characteristics of shifting PGTs. In papers [3, 8, 11, 18], a practical engineering method for determining the transmission ratios of the compound PGTs is presented. In paper [9], two-carrier PGTs with three external shafts that allow single-speed drives are systematically analyzed and the methodology and software system for the selection of the optimal PGT are presented. The available literature lacks the methodology and the software system for the selection of the optimal variant for multi-speed multi-carrier PGTs, although some companies have probably developed or are developing such programs. The aim of this paper is to present the methodology for the optimal selection of two-carrier shifting PGTs with two coupled and four external shafts stocked with brakes and clutches. For this purpose, a software system 6-*brz*, which allows the selection of the optimal variant of the aforementioned multi-speed PGT, is developed. The software system will be described in this paper.

The considered mechanisms having an appropriate control system can realize up to seven speeds by connecting two planetary gear sets. It is very important to systematically research into the kinematic change capability of such PGTs and to compose their synthesis methodology and optimal selection methodology.

The basic element of shifting PGTs which are investigated in this paper is the planetary gear set 2k-h variant A [2]. Such planetary gear set is conveniently presented by the *Wolf-Arnaudov symbol* [5, 11, 18, 19]. Therefore, each shaft is marked in a different way. The sun gear shaft is marked with one thin line, the crown gear shaft is marked with one thick line and the carrier shaft is marked with two parallel thin lines. Planetary gear set shafts are loaded with torques whose ratios are shown in Fig. 1. Shaft torques T_3 and T_S are given as functions of the ideal torque ratio t and the sun-gear shaft torque T_1.



Prerequisite: $\eta_0 = \eta_{13(S)} = \eta_{31(S)} = 1$

Ideal torque ratio: $t = \dfrac{T_3}{T_1} = \left|\dfrac{z_3}{z_1}\right| > +1$

Torques: $T_1 : T_3 : T_S = +1 : +t : -(1+t)$

Fig. 1 The most often used single-carrier planetary gear train and its torques [3]

2. Two-speed planetary gear train

Characteristic variants of two-carrier PGT (made up of two planetary set 2k-h variant A) with two coupled and four external shafts and breaks placed on two external shafts are analyzed in detail in [11]. Driving and operating machines are connected on the remaining two shafts. By alternating the activation of breaks, reactive shafts are changed and, by this, the energetic flow through PGT which, as a consequence, leads to a change in the transmission ratio. The analysis has shown that such PGT can give different combinations of two transmission ratios; therefore, they can be applied in industry for the realization of different two-speed drives (two-speed transport machines, tool machines, ships, etc.).

Twelve possible ways of connecting planetary gear sets (structural schemes) by a two-carrier PGT with two coupled and four external shafts are presented in Fig. 2. Markings are in accordance with those in [11]. Possible variants of break layouts (Br1 and Br2) and driving and operating machines on the external shafts are shown in Fig. 3 (layout variants). Any of these layout variants can be applied to every scheme shown in Fig. 2. Therefore, it is possible to get 144 variants of PGT. Some of these variants are identical so the number of different variants is reduced to 120. Each of these 120 variants has specific changeable capabilities, that is, each can realize a different set of transmission ratio combinations.

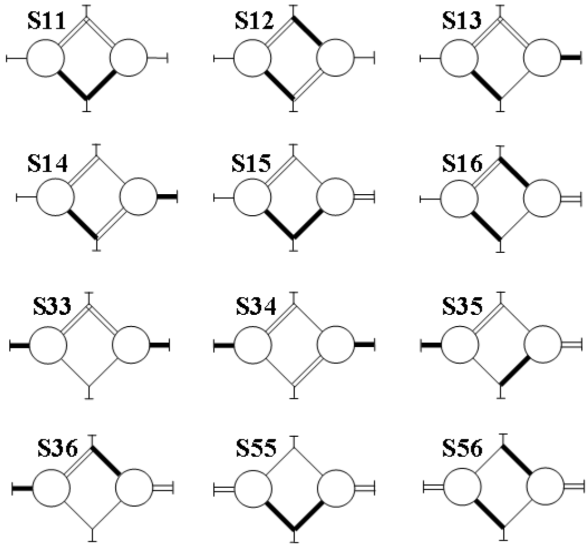
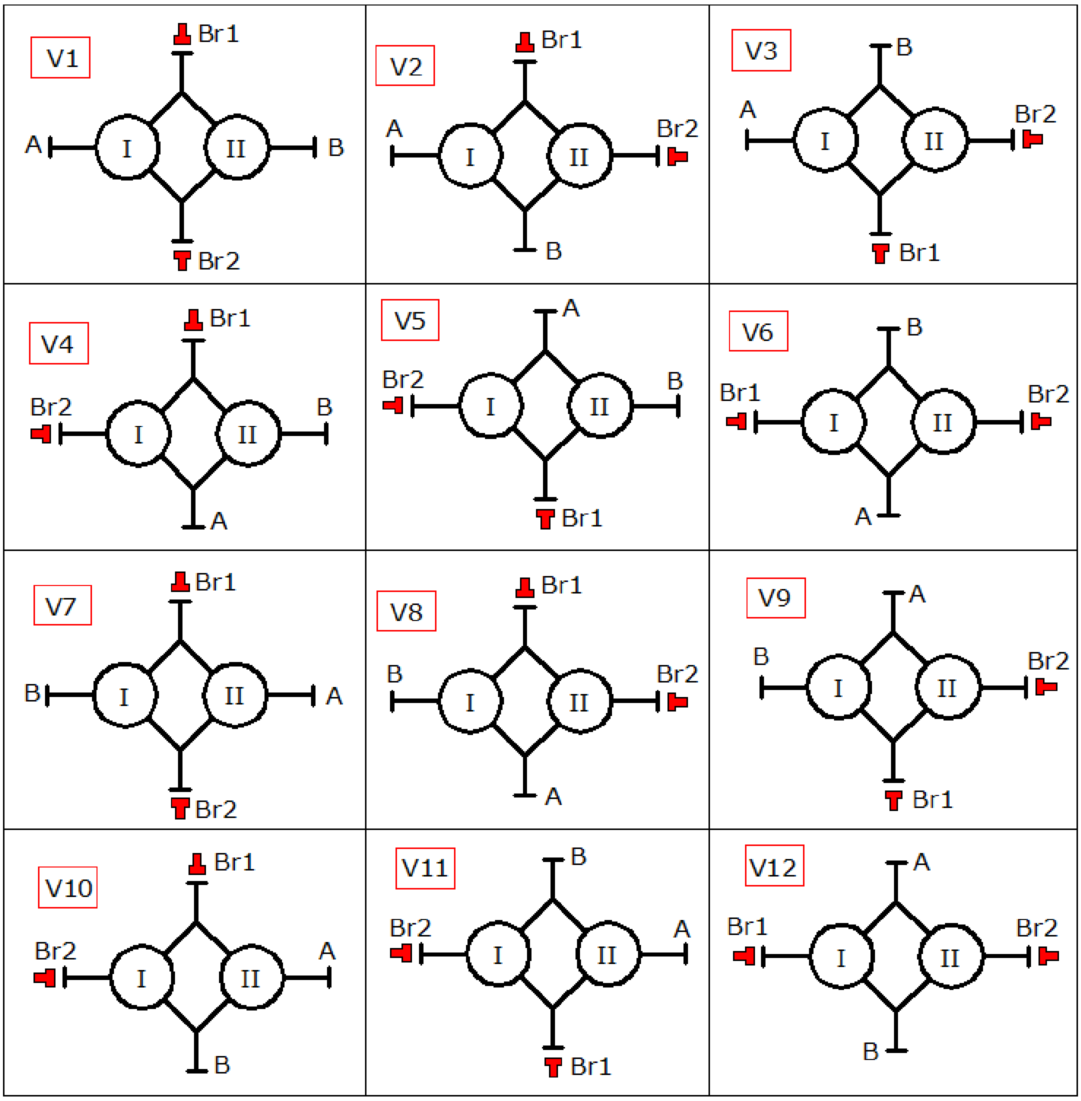


Fig. 2 Systematization of all schemes of the two-carrier PGT with two coupled and four external shafts



A – input shaft ; B – output shaft ; Br1, Br2 – brakes ; V1…V12 – layout variants

Fig. 3 Systematization of all layout variants of the two-carrier PGT with two coupled and four external shafts

Structural analyses of the PGT according to the structural scheme S12 and layout variant V2 (S12V2) are presented in Fig. 4. In Fig. 4a the operation of this PGT is symbolically shown with activated break Br1 (S12V2Br1) and in Fig. 4b with activated break Br2 (S12V2Br2). The ideal torques on the shafts are determined and from them transmission ratios i_{Br1} and i_{Br2} as functions of ideal torque ratios of planetary gear sets t_{I} and t_{II}.

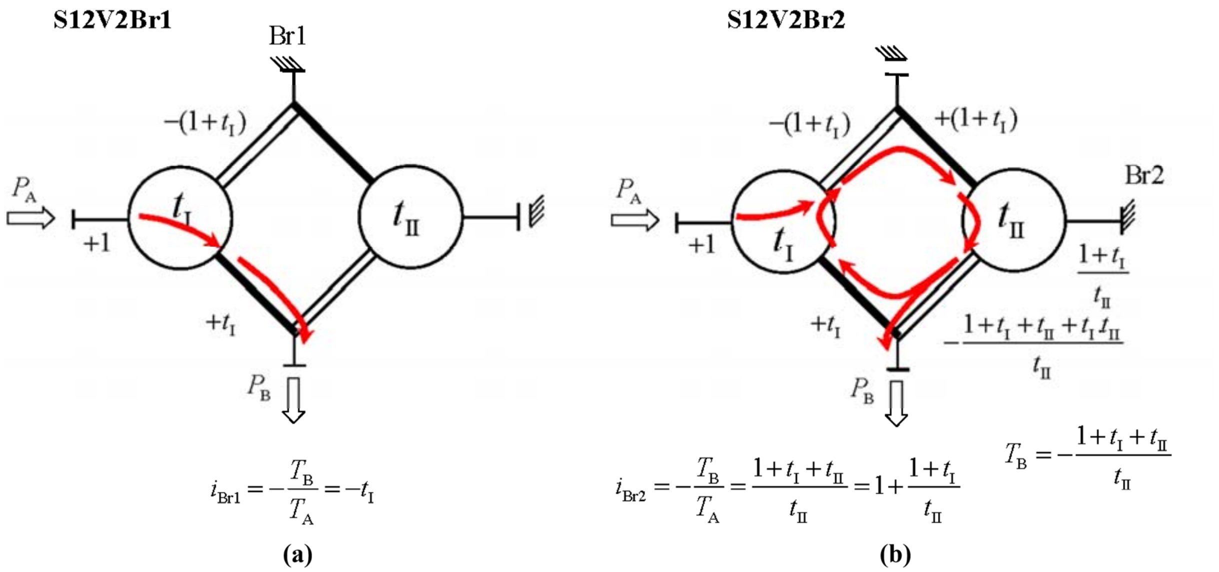


Fig. 4 Determination of the transmission ratios for PGT S12V2 with the activated breaks Br1 (a) and Br2 (b)

Based on the functions i_{Br1} and i_{Br2} of transmission ratio it can be concluded that in the speed Br1 (activated break Br1), the PGT operates as a reducer with the opposite rotational direction of the output and input shafts. While operating in speed Br2 (activated break Br2) the PGT operates as a reducer with the same rotational direction of the output and input shafts. The kinematic scheme of this PGT is presented in Fig. 5.

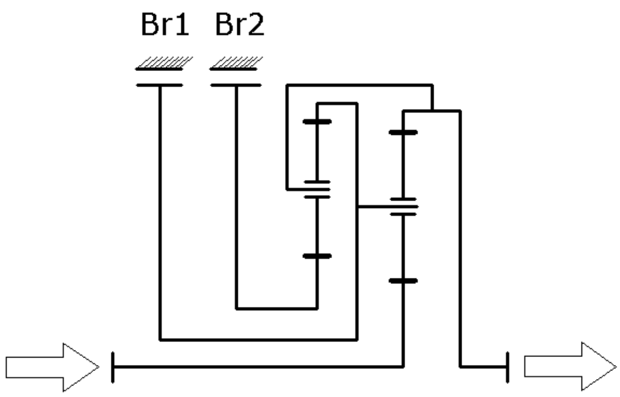
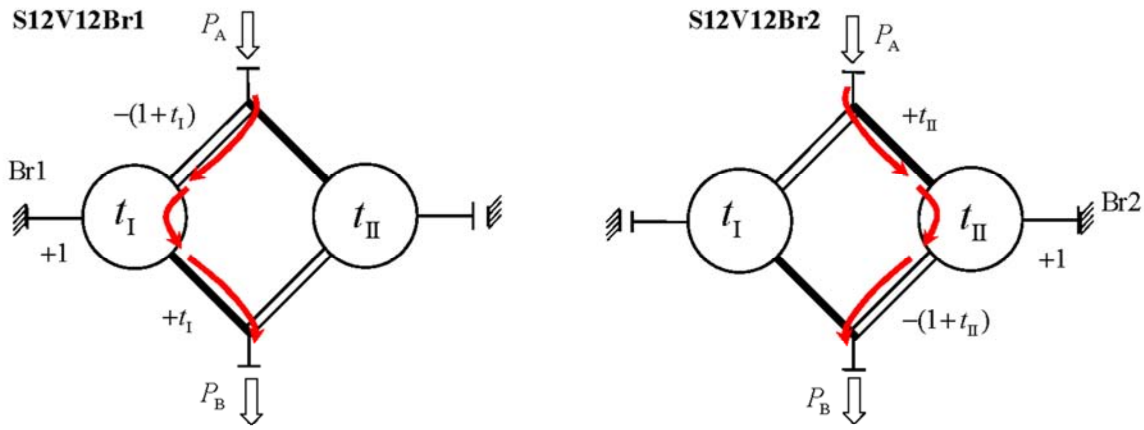


Fig. 5 Kinematic scheme of PGT S12V2

Structural analysis of PGT S12V12 is shown in Fig. 6. The operating mode of the PGT with activated break Br1 (S12V12Br1) is symbolically shown in Fig. 6a, and with activated break Br2 (S12V12Br2) in Fig. 6b.



$$i_{Br1} = -\frac{T_B}{T_A} = \frac{t_I}{1+t_I} < 1$$

(a)

$$i_{Br2} = -\frac{T_B}{T_A} = +\frac{1+t_{II}}{t_{II}} > 1$$

(b)

Fig. 6 Determination of transmission ratios of PGT S12V12 with activated breaks Br1 (a) and Br2 (b)

Based on the transmission ratio function it can be concluded that in the speed Br1, the PGT operates as a multiplier with the same rotational direction of the output and input shafts. It can also be concluded that in the speed Br2 the PGT operates as a reducer, also with the same rotational direction of the output and input shafts. The kinematic scheme of this PGT is shown in Fig. 7.

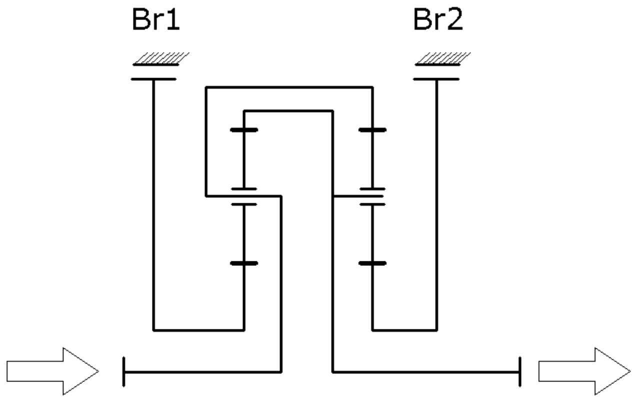
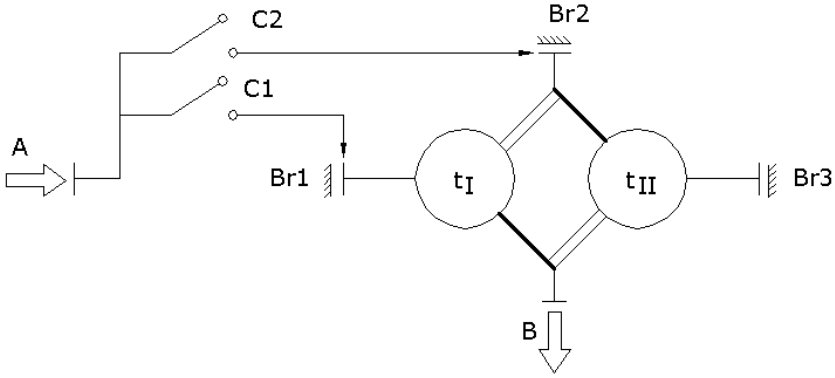


Fig. 7 Kinematic scheme of PGT S12V12

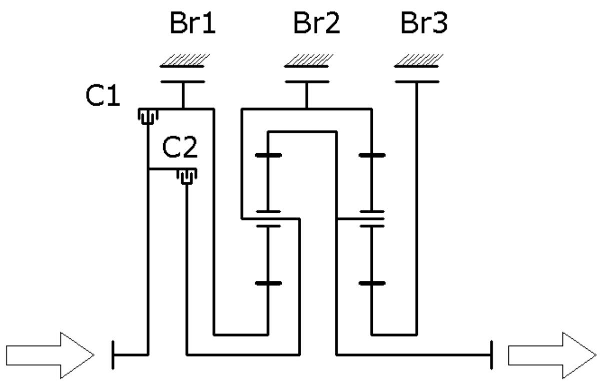
For all 12 layout variants (V1…V12), for all 12 schemes (S11…S56) transmission ratio functions are carried out as well as the operating regime (reduction, multiplication, rotational direction of output in relation to input shaft). These functions are integrated into an algorithm of the computer program 6-*brz* which enables the synthesis of shifting PGTs with one, two, or even three layout variants within the same structural scheme. Kinematic schemes of all 120 PGT variants are created and thus the starting of their design (procedure) is made possible [11].

3. Multi-variant planetary gear trains

If a two-carrier PGT is designed with a control system which enables the shift of input or output shaft through the change of the reactive shaft, the number of possible speeds is increased. The shift of input shaft or output shaft means the change of layout variant within the same scheme. Since it is possible to combine two two-speed variants (two-variant two-carrier PGT) or three two-speed variants (three-variant two-carrier PGT) by using the clutches, the highest number of possible speeds is four or six (without including the speed at which only the clutches are activated so that the transmission ratio is equal to 1).



(a)



(b)

Fig. 8 Structural scheme (a) and kinematic scheme (b) of two-variant PGT S12V2/V12

The two-variant PGT S12V2/V12 created by a combination of variants as shown in Figs 4 and 6, as well as in Figs 5 and 7, is shown in Fig. 8.

By a combination of control devices this PGT can realize 5 speeds which are given in Tab. 1 together with their corresponding transmission ratio functions.

Table 1 Possible speeds with corresponding activated control devices and transmission ratio functions

speed	*activated control devices*	*transmission ratio functions* i_k
1	C1-Br2	$-t_{\mathrm{I}}$
2	C1-Br3	$1+\frac{1+t_{\mathrm{I}}}{t_{\mathrm{II}}}$
3	C2-Br1	$\frac{t_{\mathrm{I}}}{1+t_{\mathrm{I}}}$
4	C2-Br3	$\frac{1+t_{\mathrm{II}}}{t_{\mathrm{II}}}$
5	C1-C2	1

The analysis of Fig. 3 leads to the conclusion that it is not possible to mutually combine all layout variants within the frame of the same scheme. The only variants that can be combined are those which have a common input shaft or output shaft. That common shaft is always connected to the driving or operating machine. All the possible combinations of two layout variants are given in Tab. 2. There are 24 of them all together. By different combinations of layout variants from the same row, the position of the driving and the operating machine is changed. The combinations of layout variants in which both variants have a common output shaft are shown in the first column of the table, while the combinations in which both variants have a common input shaft are given in the second column.

Table 2 Theoretically realizable combinations of two different layout variants within the same scheme

combination of layout variants with a common output shaft	*combination of layout variants with a common input shaft*
V1, V2	V7, V8
V1, V3	V7, V9
V7, V10	V1, V4
V7, V11	V1, V5
V2, V3	V8, V9
V8, V10	V2, V4
V8, V12	V2, V6
V9, V11	V3, V5
V9, V12	V3, V6
V10, V11	V4, V5
V4, V6	V10, V12
V5, V12	V6, V11

The analysis of Fig. 3 confirms that there are eight possible combinations of three layout variants (Tab. 3). No other combinations are possible because the variants within them do not have a common input or output shaft. The first column of the table shows the combinations in which all three layout variants have a common output shaft, and the second column shows the combinations in which all three layout variants have a common input shaft. In different combinations of layout variants from the same rows, the position of the driving and the operating machine is changed.

Table 3 Theoretically realizable combinations of three different layout variants within the same scheme

combination of layout variants with a common output shaft	*combination of layout variants with a common input shaft*
V1, V2, V3	V7, V8, V9
V7, V10, V11	V1, V4, V5
V4, V6, V8	V2, V10, V12
V5, V9, V12	V3, V6, V11

4. The methodology of synthesis of multi-variant planetary gear trains

The kinematic synthesis of multi-variant PGT is based on the knowledge of transmission ratio functions of two-speed PGT.

The graphical representation of transmission ratio functions of one two-variant PGT which can realize up to four transmission ratios is given in Fig. 9a.

The intervals of the required transmission ratios I_1, I_2, I_3, I_4 for which $i_{\mathrm{k}1} \in \mathrm{I}_1$, $i_{\mathrm{k}2} \in \mathrm{I}_2$, $i_{\mathrm{k}3} \in \mathrm{I}_3$, $i_{\mathrm{k}4} \in \mathrm{I}_4$, are shown on the z-axis in Fig. 9b.

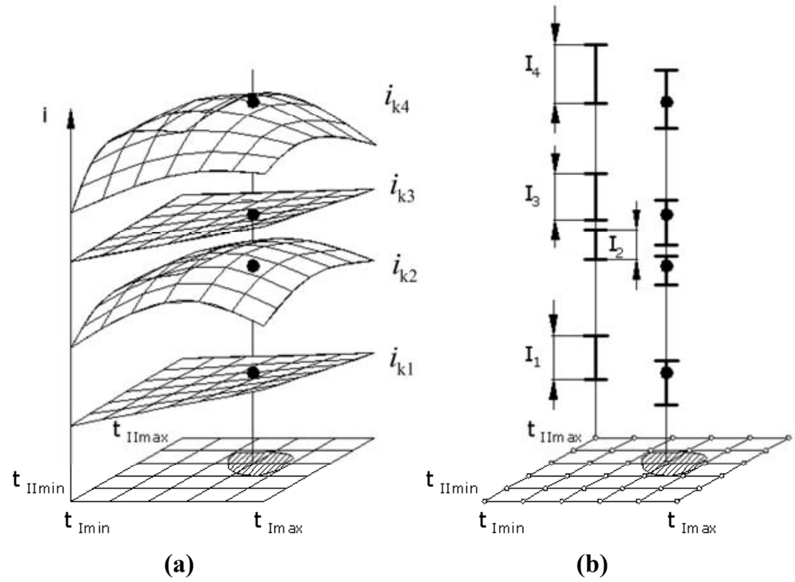


Fig. 9 Search of the domain with the aim of finding the combinations of variants which enable the realization of required transmission ratios

The developed computer program *6-brz* in the frame of each multi-variant PGT for each possible combination of ideal torque ratios determines the values of transmission ratio functions and checks whether these values are found in the required intervals I_1, I_2, I_3, and I_4 (Fig. 9b).

If there are combinations of ideal torque ratio, in which the values of transmission ratio are found in the required intervals, then the program separates them as solutions.

After finding the solutions, the program compares them according to relevant criteria (minimal radial dimensions of planetary gear sets, maximal equivalent efficiency, etc.) [20-23].

In order to design an appropriate PGT, the 6-*brz* software system is developed. The software system is written in Visual Fortran Professional Edition 6.6.0. The program allows the determination of the structure and the basic geometric, kinematic and energy parameters of multi-variant PGTs (from two-speed up to seven-speed). Deducted expressions for the transmission ratios, efficiency and crown-gears pitch diameter of all 12 structural schemes for all 12 layout variants are integrated into the program. The basic input data for the program are: the required number of speeds, intervals of the given transmission ratios, share of operating time at different speeds, the maximum torque of the input shaft, the rotational speed of the input shaft, the number of teeth of sun-gears, intervals of ideal torque ratio of planetary sets, data of the material, etc.

After data are fed into the input file, the program 6-*brz* determines the values of transmission ratio functions for each speed at every scheme, as well as for every possible combination of layout variants (Fig 2, Fig 3), and for all combinations of ideal torque ($t_{\mathrm{I}}, t_{\mathrm{II}}$). The ideal torque ratio of planetary gear trains with three satellites is limited by the design and varies in the range from at least 2 up to 12. The program checks which of the schemes and combinations of layout variants has the required combination of transmission ratios. If there is such a combination, the program returns: the PGT label, the label of the layout variant and brakes for a certain speed, the ordered pair of ideal torque ratios that corresponds to the required combination of transmission ratios, an equivalent efficiency, the teeth numbers of the crown-gears, modules, and crown-gear pitch diameters. The results are recorded in the output file and can be displayed in tables in a suitable program such as the Origin 7.5. Several solutions can be ranked according to the relevant criteria of PGT quality (minimum overall radial dimension, the maximum efficiency, etc.). Such criteria are given in [20-23].

For the purpose of showing the computer program operation, the procedure of getting the solutions of three-speed PGT is shown. Some of the input data are required:

- transmission ratio intervals $2.4 \le i_{\mathrm{k1}} \le 2.6$; $1.35 \le i_{\mathrm{k2}} \le 1.45$; $-2.7 \le i_{\mathrm{k3}} \le -2.6$;
- number of teeth of sun gears of the first and second planetary gear sets $z_{1\mathrm{I}} = z_{1\mathrm{II}} = 18$;
- torque on the input shaft of PGT $T_{\mathrm{A}} = 50\,\mathrm{Nm}$.

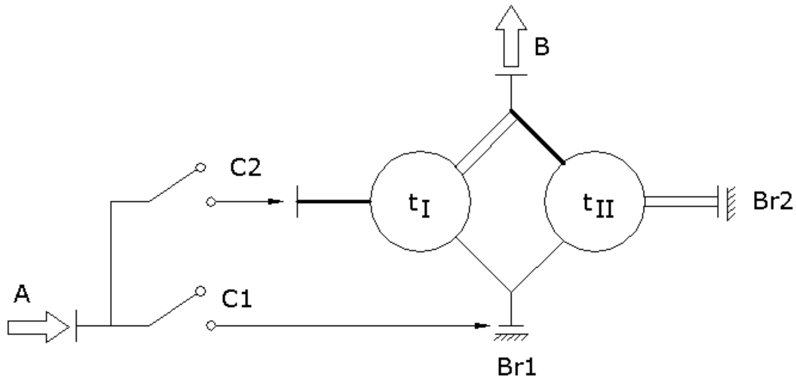
After the input data have been entered and the program system 6-*brz* has been started, three conceptions of PGT are found, each of which has two solutions within the required intervals (Tab. 4).

With the acquired data of the ideal torque ratio t_{I} and t_{II}, transmission ratios i_{k1}, i_{k2} and i_{k3}, the number of sun gears $z_{3\mathrm{I}}$ and $z_{3\mathrm{II}}$, and the pitch diameter of the ring gears $d_{3\mathrm{I}}$ and $d_{3\mathrm{II}}$, the computer program system determined the variant and breaks with which each transmission ratio is achieved. The PGT S36V3/V6 (t_{I}=2.5 and t_{II}= 2.6667) is the optimal solution according to the criterion of minimal radial dimensions of planetary gear sets.

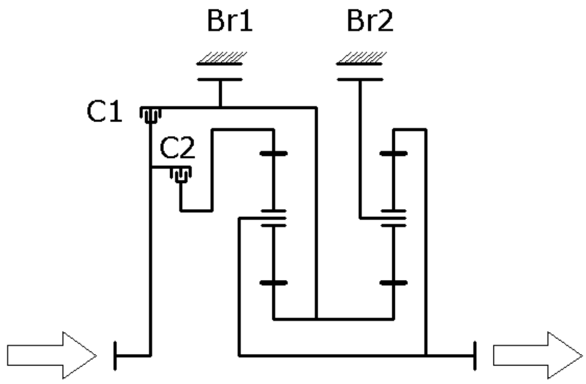
Table 4 Solutions which comply with the given requirements

Mark of PGT	t_{I}	t_{II}	i_{k1}	i_{k2}	i_{k3}	$z_{3\mathrm{I}}$	$z_{3\mathrm{II}}$	$d_{3\mathrm{I}}$ /mm	$d_{3\mathrm{II}}$ /mm
S12V6/V11	2.3333	2.6667	2.571 (V11Br2)	1.428 (V6Br1)	-2.666 (V11Br1)	42	48	94.5	96
	2.5	2.6667	2.466 (V11Br2)	1.4 (V6Br1)	-2.666 (V11Br1)	45	48	101.25	96
S12V2/V12	2.6667	2.3333	2.571 (V2Br2)	1.428 (V12Br2)	-2.666 (V2Br1)	48	42	96	94.5
	2.6667	2.5	2.466 (V2Br2)	1.4 (V12Br2)	-2.666 (V2Br1)	48	45	96	101.25
S36V3/V6	2.3333	2.6667	2.571 (V3Br2)	1.428 (V3Br1)	-2.666 (V6Br2)	42	48	63	96
	2.5	2.6667	2.466 (V3Br2)	1.4 (V3Br1)	-2.666 (V6Br2)	45	48	61.87	96
The optimal solution of PGT with the basic parameters is given in the gray shaded row									

The structural scheme is determined on the basis of markings of optimal PGT obtained by the computer according to Figs 2 and 3 (Fig. 10a). On the basis of the structural scheme, the kinematic scheme is determined (Fig. 10b).



(a)



(b)

Fig. 10 Structural scheme (a) and kinematic scheme (b) of two-variant PGT S36V3/V6

Therefore, after the solution is determined by means of the presented methodology, it is necessary to symbolically show the structure of PGT. Then, it should be determined whether it is possible to form the kinematic scheme using the symbolically shown structure. It has been established that it is not always possible to physically connect the shaft of PGT as it is shown in the symbolical representation. In [11, 12, 13, 14, 22], some examples of kinematic schemes exist, and they can be useful in creating other schemes.

5. Conclusion

All possible structural schemes of a two-carrier PGT with two coupled and four external shafts are shown as well as layout variants of breaks and driving and operating machines on the PGT external shafts. By the analysis of structural schemes and layout variants, it is established that there are 120 different variants of the two-speed PGT. It is determined which two-speed variants can be theoretically combined in order to design two-variant and three-variant PGT. The method to design one two-variant PGT from two two-speed PGTs by using the clutches is presented. The computer program *6-brz* which enables the determination of optimal variant multi-speed PGT by means of demonstrated methodology is developed. The example of selection of one three-speed PGT by the computer program *6-brz* is given and its structural and kinematic schemes are made. The shown methodology and computer program

6-brz enable quick determination of structure and base parameters of shifting PGT which can hypothetically satisfy transmission requirements.

REFERENCES

[1] Lechner, G., Naunheimer, H.: Automotive Transmissions, Springer-Verlag, Heidelberg, 1999.

[2] Planetary gears. Handbook. Edit by V. N. Kudryavtsev and Iu. N. Kirdyashev. Leningrad: Mashinostroenie, 1977. (in Russian)

[3] Arnaudow, K., Karaivanov, D. Systematik, Eigenschaften und Möglichkeiten von zusammengesetzten Mehrsteg-Planetengetrieben. In: Antriebstechnik (2005) 5, S. 58-65.

[4] Velicu, R.: On the mechanical efficiency of 1 DOF two-planetary groups, Machine design (2008), 48, pp. 69-74

[5] Arnaudov, K., Genova, P., Dimitrov, L.: For an unified and correct IFToMM terminology in the area of gearing, Mechanism and Machine Theory, 40 (2005) pp. 993-1001

[6] Ciobotaru, T.: A Method for the Analysis of Epicyclic Gearboxes, Seoul 2000 FISTA World Automotive Congres, June 12-15, 2000, Seoul, Korea

[7] Joksimović, S.,Vukuević, N.: Resarch of working regime of gear of heavy caterpillars for relevant estimation of technical condition, Tractors and power machines 11 (2006), pp. 50-55.

[8] Arnaudow, K, Karaivanov, D.: Die zusammengesetzten Mehrstegplanetengetriebe, ihre Systematik, Eigenschaften und Moeglichkeiten, International Conference "Power Transmissions '03" (2003)

[9] Gupta, K, Sharma, A.: Technique for EXAMING Epicyclic Gear Box, International Journal of Scientific & Engineering Research, Volume 2, Issue 11, November-2011

[10] Csoban, A., Determining the heat capacity of planetary gearboxes, Budapest University of Technology and Economics, Dissertation, 2011.

[11] Troha, S., Analysis of a planetary change gear train's variants. Dissertation. Engineering Faculty, University of Rijeka (Croatia), 2011. (in Croatian)

[12] Looman, J. Zahnradgetriebe. Berlin: Springer Verlag, 1996.

[13] Müller, H. W.: Die Umlaufgetriebe. Berlin: Springer Verlag, 1998.

[14] Orlić, Ž., Orlić, G.: Planetni prijenosi, Zigo, Rijeka, 2006.

[15] Kraynev, A, F.,Astashev, V.,K., Salamandra, K., B., Raghavan, M.: Multi-speed planetary transmission with three brakes and four clutches, Unated States Patent US 2011/0218074 A1, Sep. 8, 2011

[16] Tiesler, P., Gumpoltsberger, G., Multiple gear stage automatic transmission, Unated States Patent US 2005/0054478 A1, Mar. 10, 2005

[17] Heitmann, A., Dreibhilz, R., Gumpoltsberger, G., Automatic transmission for motor vehicles, Unated States Patent US 2004/0180751 A1, Sep. 16, 2004

[18] Karaivanov D. Structural Analysis of Coupled Multi-Carrier Planetary Gear Trains – from Lever Analogy to Multi-Objective Optimization. In: Proceedings of 3rd International Conference on Manufacturing Engineering (ICMEN), 1-3 October 2008, Chalkidiki (Greece), pp. 579-588.

[19] Karaivanov D.: Theoretical and experimental studies of the influence of the structure of the coupled two-carrier planetary gear trains on its basic parameters [dissertation], Univ. of Chemical Technology and Metallurgy, Sofia, 2000.

[20] Stevanović-Marinović, J.: Višekriterijumska optimizacija zupčastih parova planetarnih prenosnika, doktorska disertacija, Mašinski fakultet u Nišu, Niš, 2008.

[21] Troha S., Petrov P., Karaivanov, D.: Regarding the optimization of coupled two- carrier planetary gears with two coupled and four external shafts, Machinebuilding and electrical engineering, 2009, Nr. 1, pp. 49 – 56.

[22] Pantić, M.: Gubici snage u ozubljenju kao parametar za formiranje optimalne kinematske šeme planetarnih prenosnika primenjenih u sistemu za prenos snage motornih vozila, doktorska disertacija, Mašinski fakultet Univerziteta u Beogradu, 1997.

[23] Stefanović-Marinović, J., Milovančević, M.:The Optimization Possibilities at the Planetary Gear Trains, Journal of Mechanic Engineering and Automation 2 (2012), pp. 365-373.

Submitted: 20.12.2011

Accepted: 13.9.2012

Dr. sc. Sanjin Troha
Faculty of Engineering, University of Rijeka, Croatia, stroha@riteh.hr

Prof. dr. sc. Neven Lovrin
Faculty of Engineering, University of Rijeka, Croatia, neven.lovrin@riteh.hr

Assist. prof. dr. sc. Miloš Milovančević
Faculty for Mechanical Engineering, University in Niš, Serbia, milovancevic@masfak.ni.ac.rs