

# Analysis of nine-speed planetary change-gears through the torque method

Sanjin Troha<sup>1\*</sup>, Dimitar Karaivanov<sup>2</sup>, and Željko Vrcan<sup>1</sup>

<sup>1</sup>University of Rijeka, Faculty of Engineering, 58 Vukovarska St., Rijeka, Croatia

<sup>2</sup>University of Chemical Technology and Metallurgy, Department of Applied Mechanics, 8 Kl. Ohridski Blvd., Sofia, Bulgaria

**Abstract.** An alternative method for kinematic and power analysis of compound gear train is presented, which uses the torques and lever analogy of a given gear train. The method combines the accuracy of the Willis analytical method with the clarity of the graphic method of Kutzbach. Unlike these methods, the torque method allows determining not only the gear ratio, but the magnitude and direction of power flows, and hence the determination of efficiency of a given compound planetary gear train. The application of the method is illustrated with example of novel arrangement of complex nine-speed change-gear. Structural, kinematic, and power (efficiency) analysis are made. A software for these analysis facilitation is developed. A numerical example is presented and discussed.

## 1 Introduction

### 1.1 General

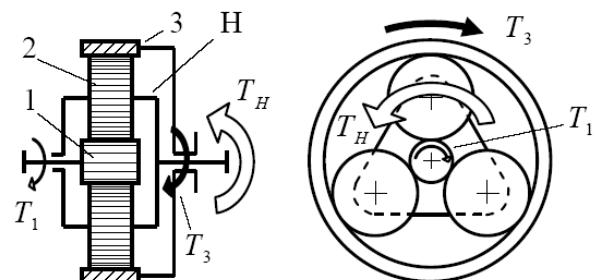
There are various methods for analysis of planetary gear trains (PGT) at all. Two, however, are most commonly used. One is the analytical method of Willis, the other is a graphical method of Kutzbach. Both methods have proved their worth over many years. The method of Willis (1841) [1] is used for almost two centuries and the method of Kutzbach (1927) [2] is there nearly a century.

An easy-to-use, eye-catching, and definitely engineering method for kinematic and power analysis is the torque method [3-6]. Compared to the above two methods, it has, in general, the following advantages, especially in the analysis of complex compound PGTs:

- accuracy;
- simplicity;
- maximum clarity;
- easy usability (relevance);
- easy check of the results;
- generally, the opportunity to achieve more goals than with previous methods, namely - not only determining the gear ratio, but the presence or absence of internal division or circulation of power, determining the efficiency and, in particular in gearboxes, determining the load spectrum of the elements of gear train (wheels, shafts, bearings), as a prerequisite for their reliable load capacity calculation;

The trend towards increasing the number of gears (speed ratios) in automotive planetary gearboxes (change-gears) [7] has been reinforced with the development of electric cars. One way to avoid increasing the number of gear stages is to develop

rational designs [8]. The authors consider that it is appropriate to develop such structures on the basis of modules of two-carrier PGTs with two compound and four external shafts [9-11]. Practice has shown that the most suitable building element for compound PGTs is the simple (single-carrier) PGT with one external, one internal mesh, and with one-rim planets [5] (Fig. 1).



**Fig. 1.** The most often used PGT (with one external meshing, one interna meshing, and one-rim planets) and it torques: 1 – sun gear; 2 – planets; 3 – ring gear; H – carrier.

### 1.2. Essence of the torque method

Unlike the methods of Willis and Kutzbach using angular velocity  $\omega$  (or rotational speed  $n$ ) and the peripheral velocities  $v$ , at the presented here method the torques  $T$  are used. The method is based on the following well-known principles of mechanics, illustrated with the simplest case of single-carrier PGT (from Fig. 1 for example) with three external shafts that has a very good lever analogy (Fig. 2):

- a.) Equilibrium of ideal (with no account of losses) external torques acting on the three shafts

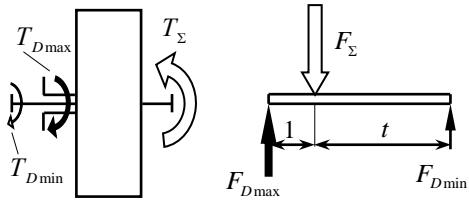
\* Corresponding author: [stroha@riteh.hr](mailto:stroha@riteh.hr)

$$\sum T_i = T_{D\min} + T_{D\max} + T_{\Sigma} = 0 \quad (1)$$

where

$T_{D\min}$  and  $T_{D\max}$  are the unidirectional external ideal torques (with different sizes), therefore are indicated by single lines on the figure, but with different thickness, according to their size;

$T_{\Sigma}$  is the largest torque, in absolute value equal to the sum of other two torques  $T_{D\min}$  and  $T_{D\max}$ , and therefore is marked with a double line.



**Fig. 2.** Planetary gear train with three external shafts, their torques and the lever analogy.

b.) Equilibrium of the real (considering the losses) external torques

$$\sum T'_i = T'_{D\min} + T'_{D\max} + T'_{\Sigma} = 0 \quad (2)$$

where

$T'_{D\min}$ ,  $T'_{D\max}$  and  $T'_{\Sigma}$  are the same external torques as above, but real now considering the losses in the gear train.

c) Sum of ideal (not considering the losses) powers

For the simpler case, when the gear train is with one fixed shaft, i.e. working with  $F = 1$  degree of freedom, the condition is

$$\sum P_i = P_A + P_B = T_A \cdot \omega_A + T_B \cdot \omega_B = 0 \quad (3)$$

where

$P_A$ ,  $T_A$  and  $\omega_A$  are the input power, torque and angular velocity;

$P_B$ ,  $T_B$  and  $\omega_B$  - the output power, torque and angular velocity;

The above condition is used to determine the speed ratio

$$i = \frac{\omega_A}{\omega_B} = -\frac{T_B}{T_A} \quad (4)$$

d) Sum of real (considering the losses) powers

$$\sum P'_i = \eta \cdot P'_A + P'_B = \eta \cdot T'_A \cdot \omega_A + T'_B \cdot \omega_B = 0 \quad (5)$$

where

$P'_A$ ,  $T'_A$  and  $\omega_A$  are the input power, torque and angular velocity;

$P'_B$ ,  $T'_B$  and  $\omega_B$  - the output power, torque and angular velocity;

From this follows the formula for the efficiency

$$\eta = -\frac{T'_B / T'_A}{\omega_A / \omega_B} = -\frac{i_T}{i_k} \quad (6)$$

In this formula  $i_T$  means the ratio of the real external torques  $T'_B$  and  $T'_A$  at the input A and output B of gear train, as signified more as dynamic [12], as a power ratio

or torque transformation [13, 14]. The other ratio  $i_k$  is the usual speed ratio and unlike the above, if necessary, the index k - from the "kinematic" can be used.

To determine the real external torques  $T'_i$  it is necessary that the directions of relative (rolling) powers  $P_{rel}$  are identified in each gear train according to the respective carrier, and each gear train has its own basic efficiency  $\eta_0$  [12-14].

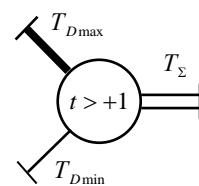
In most cases the direction of relative (rolling) power  $P_{rel}$  in one planetary gear train (from the sun gear 1 to the ring gear 3 and vice versa) can be determined, proceeding from the direction of the torque on the sun gear and its relative angular velocity toward the carrier  $\omega_{rel} = \omega_l - \omega_H$ , by the following simple criteria:

- when the directions of the torque  $T_l$  and the relative angular velocity  $\omega_{rel}$  coincide i.e.  $\omega_{rel} \cdot T_l > 0$  - the relative power is transmitted from the sun gear 1 through the planets 2 to the ring gear 3, or in other words, the sun gear appears to be the driving one in the relative motion;
- when  $\omega_{rel} \cdot T_l < 0$  - the relative power is passed from the ring gear 3 to the sun gear 1;

In relatively rare cases this situation is not that clear. Then it is appropriate for determining the direction of the relative power to use the method of the samples [15].

Along with the foregoing known principles of mechanics, the following two small but very useful innovations are made:

e) For the greatest possible illustration of the method the known symbol of the Wolf [16] (Fig. 3) is used, but modified [3-6]. The each simple (single-carrier) PGT, which has three external shafts is depicted with a circle, from which the three shafts are coming out, marked differently from the original - i.e. not with numbers, letters or inscriptions, but by the thickness of the lines corresponding to the size of the external torque, as already mentioned above. This little innovation concerning the manner of marking the shafts makes the method very clear and useful



**Fig. 3.** Modified symbol of Wolf (symbol of Wolf-Arnaudov) with three external shafts and their torques.

- f) Besides using the symbol of Wolf-Arnaudov, where appropriate another innovation is made. A new torque ratio is defined - the one of the unidirectional external torques [3-6, 17]

$$t = \frac{T_{D_{\max}}}{T_{D_{\min}}} > +1 \quad (7)$$

because  $T_{D_{\max}} > T_{D_{\min}}$  (Fig. 2).

This innovation is especially important. It is connected to the lever analogy.

g) Ideal external torques are always in a certain ratio, expressed with the torque ratio t

$$T_{D_{\min}} : T_{D_{\max}} : T_{\Sigma} = T_1 : +t \cdot T_1 : -(1+t)T_1 = 1 : +t : -(1+t) \quad (8)$$

in which always

$$T_{D_{\min}} < T_{D_{\max}} < |T_{\Sigma}| \quad (9)$$

however:

- How many degrees of freedom F (F = 1 or F = 2) the gear train is running with;
- Which shaft is fixed at F = 1 degree of freedom;
- What is the direction of power flow, respectively does the gear train works as a reducer or a multiplier at F = 1 degree of freedom, respectively such as a collecting or as a separating gear at F = 2 degrees of freedom, i.e. as a differential;
- Does the gear works as a single or as a component train along with others in a compound planetary gear train.

### 1.3. Application of the torque method

The application of the torque method is best illustrated in [4, 5]. Main points as determination of relative power direction in component planetary gear trains, circulation or division of power, etc. are presented and explained.

Coefficient of efficiency of the PGTs is in direct proportion of the coefficient of internal losses  $\psi_0$  and the basic (internal) coefficient of efficiency  $\eta_0$ . This is the coefficient of efficiency of the PGT at immovable carrier H ( $\omega_H = 0$ ), i.e. gear as normal, nonplanetary. When losses in the meshing prevail, the basic coefficient of efficiency  $\eta_0$  is calculated starting from the coefficients of overlapping  $\varepsilon_a$  of the two meshings - external and internal, the coefficient of friction  $\mu_z$  of the meshing, and other factors [18-23]. There is also a simplified formula for determining the basic (internal) coefficient of efficiency [24, 25]

$$\eta_0 = 1 - \psi_0 \approx 1 - 0,15 \left( \frac{1}{z_1} + \frac{1}{z_3} \right) + 0,2 \left( \frac{1}{z_2} - \frac{1}{z_3} \right). \quad (10)$$

For the coefficient of efficiency determination of the PGT according to the torque method the real external torques (considering losses) must be determined, for which you need to know the direction of power transmission with respect to the carrier H, i. e. the relative power  $P_{rel}$ . There are two options – it is transmitted from the sun gear 1 through the planets 2 of the ring gear 3 or vice versa. Using basic (internal) coefficient of efficiency  $\eta_0$  for these two cases, the following relations are obtained for the real external torques

$T'_3 = \eta_0 \cdot t \cdot T'_1$  - when the relative power is transmitted from 1 to 3;

$T'_1 = \eta_0 \frac{T'_3}{t}$  - when the relative power is transmitted from 3 to 1.

The very direction of the relative power  $P_{rel}$  transmission is defined by the following criteria - a match or mismatch of the direction of the torque  $T_1$  of the sun gear 1 with the direction of the relative angular velocity  $\omega_{rel} = \omega_1 - \omega_H$  with respect to the carrier H.

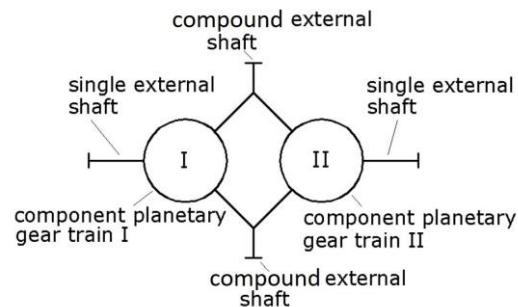
Problems of the power circulation are explained in [5].

This simple method is very suitable for studying of complex, multi-carrier PGTs [6, 26] such as the nine-speed change-gear discussed below. Simple formulae make it possible to compile process optimization software.

The aim of this article is to reveal the functions of the transmission and the degrees of utilization of the nine-speed transmission in order to create a basis for the analysis and synthesis of transmissions with any of the structurally achievable speed ratios of each of the component PGTs.

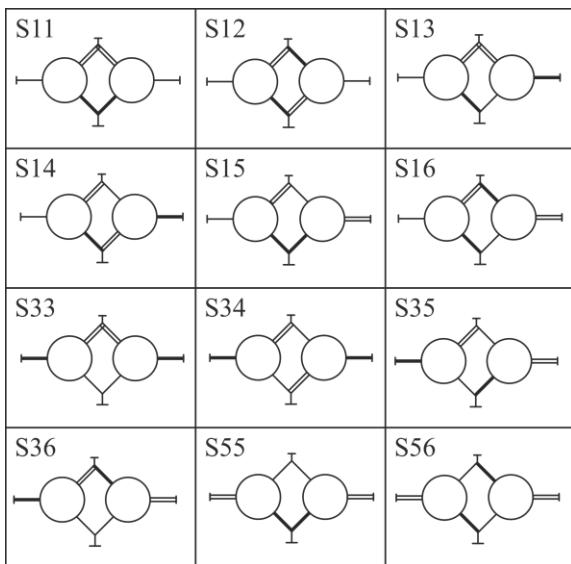
## 2 Structural schemes of the nine-speed planetary change-gear trains

In [5, 27], all two-carrier PGTs with two compound and four external shafts (Fig. 4) that can be physically arranged have investigated in detail.. That's a total of 120 different variants.



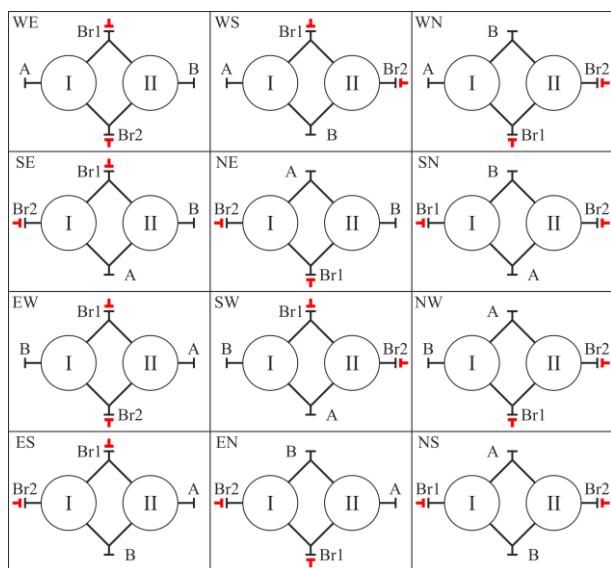
**Fig. 4.** Two-carrier PGT with two compound and four external shafts.

The analysis shows that out of this whole amount, only 12 structural schemes are non-repeating. These schemes are shown in Fig. 5. The designation from [5] is used, with the numbers indicating how the component PGTs shafts are located (Table 30.1 from [5]).



**Fig. 5.** Possible variants of two-carrier PGTs with two compound and four external shafts.

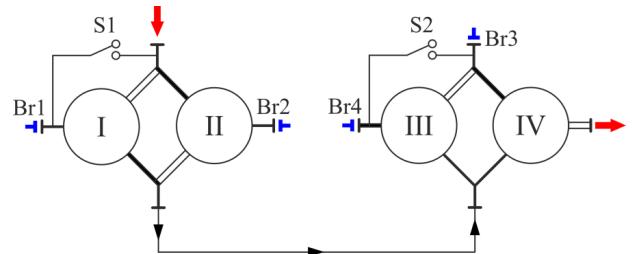
Each of the 12 variants from Fig. 5 has four external shafts on which are the power input, power output, and two brakes – Br1 and Br2. There are 12 ways for positioning of this shafts in each of the variants. It is appropriate to designate them according to the position of the input and output shafts with letters corresponding to the cardinal directions of the compass – north (N), east (E), south (S), and west (W), as it is shown in Fig. 6. In [28], the designation of a two-carrier PGT in this way is used to provides coded information about the structure of the PGT. This coding method is also used below in the article.



**Fig. 6.** Possible locations of both brakes at two-carrier PGTs with two compound and four external shafts.

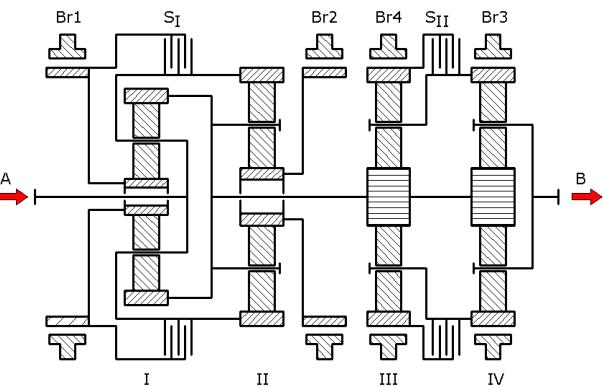
## 2.1 Nine-speed planetary change-gear trains development

If two couplings are added to the known change-gear with four speeds in one direction [5], as shown in Fig. 7, a nine-speed transmission is obtained.



**Fig. 7.** Structural scheme of the nine-speed planetary change-gear with four brakes and two couplings (clutches).

A possible arrangement of this four-carrier change-gear is shown in Fig. 8.



**Fig. 8.** A possible arrangement of nine-speed four-carrier planetary change-gear.

The presentation of the gear train as composed of two two-carrier PGTs allows to apply the methods developed for the two-carrier PGTs [8-11, 27] to its study.

## 2.2 Structural analysis of nine-speed planetary change-gear trains

The nine-speed PGT in question (Fig. 7) consists of two two-carrier PGTs connected in series, the first being S12NS with component gear trains I and II, and the second being S36SE with component gear trains III and IV.

Component gear trains I and II are connected through a shaft connecting planet carrier I with ring gear II and another shaft connecting ring gear I to planet carrier II. The power input (north of first symbol) is through the shaft connecting planet carrier I to ring gear II. Sun gears I and II are on single shafts on which brakes Br1 (west of first symbol) and Br2 (east of first symbol) can act. Clutch S1 can connect or disconnect the single shaft of sun gear I with the shaft connecting planet carrier I to ring gear II.

Power exits the first two-carrier PGT through the shaft (south of first symbol) connecting ring gear I to planet carrier II and enters the second two-carrier PGT

through the shaft (second symbol south) connecting sun gear III with sun gear IV. Planet carrier III is connected by a shaft to ring gear IV. Clutch S2 is used to connect or disconnect the single shaft of ring gear III to the shaft connecting planet carrier III and ring gear IV. Ring gear III can be stopped by brake Br4 (west of second symbol), and ring gear IV can be stopped by brake Br3 (north of second symbol). Planet carrier IV (east of second symbol) serves as the power output for the whole gearbox. It should be noted that clutch S2 may be used to cause the whole second two-carrier PGT to rotate as a block, while clutch S1 may be used to cause the whole first two-carrier PGT to rotate as a block.

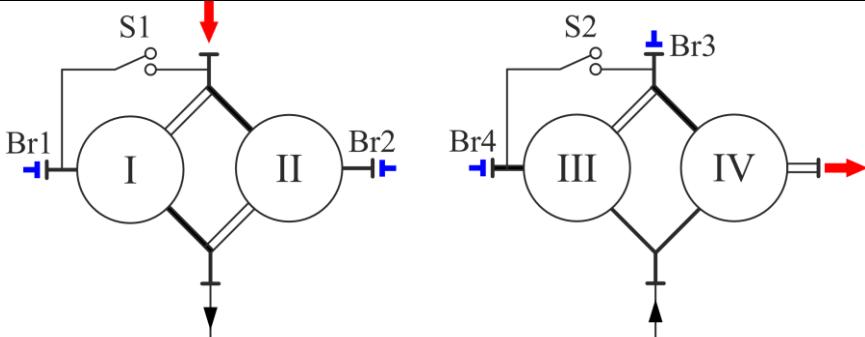
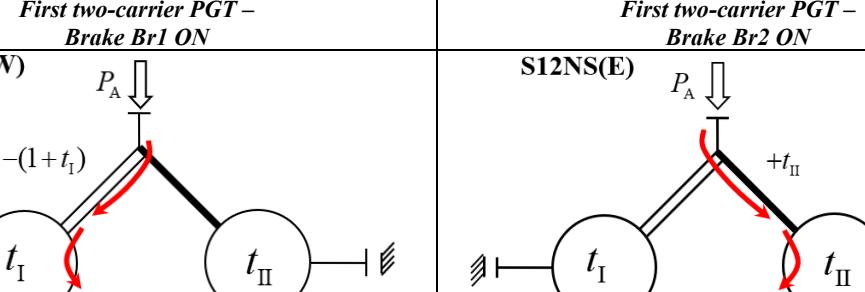
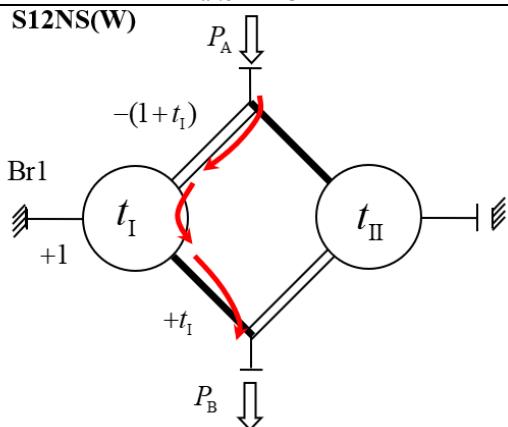
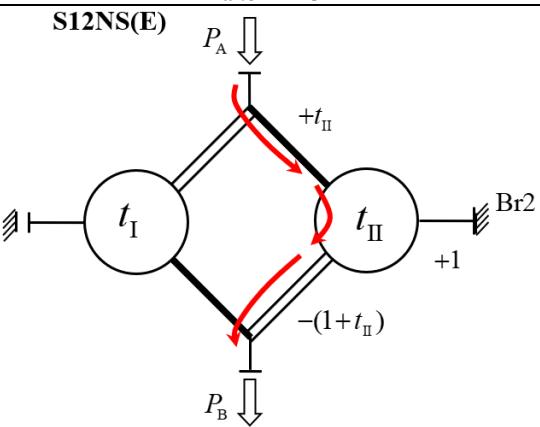
### 3 Kinematic analysis

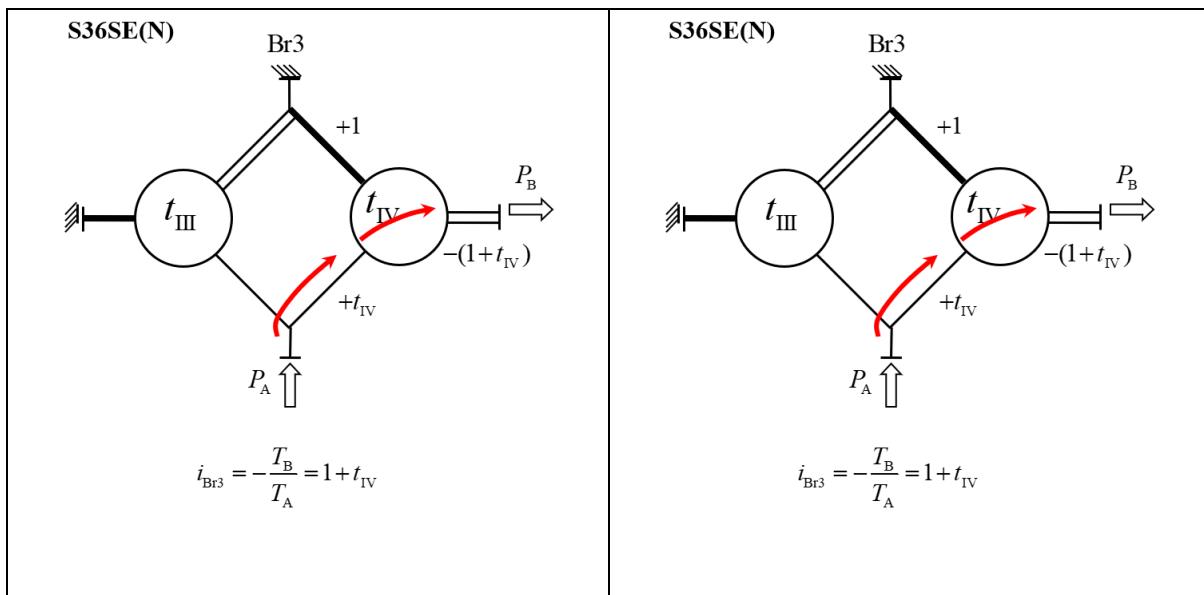
The kinematic analysis is performed by torque method analysis applied to the structural scheme of the transmission, with the results being laid out in Table 1. The speed ratios (determined through the torque ratios  $t_I$ ,  $t_{II}$ ,  $t_{III}$ , and  $t_{IV}$  of component simple PGTs) and power

flows (red arrows) for four different operating states are shown in Table 1, the first being with clutch S2 engaged and brake Br1 on (locked) (S12NS(W)). In the second operating state, clutch S2 is also on, but brake Br2 is now on (S12NS(E)). The third operating state is with clutch S1 engaged and brake Br3 on. (S36SE(N)). The fourth state is with clutch S1 engaged and brake Br4 on (S36SE(W)). The fifth state is achieved with both clutches S1 and S2 engaged, causing the gear train to rotate as a block with a transmission speed ratio equal to 1.

The sixth state is achieved with brakes Br1 and Br3 on, the seventh with brakes Br1 and Br4 on, the eighth with brakes Br2 and Br3 on, and finally the ninth state with brakes Br2 and Br4 on. The transmission ratios for states six to nine are obtained by multiplying the speed ratios of the respective two-carrier gear train from Table 1.

**Table 1.** Speed ratios and power flows for characteristic operating states of the 9-speed change-gears train.

<i>Planetary gearbox S12NS-S36SE</i>	
	
<i>First two-carrier PGT – Brake Br1 ON</i>	<i>First two-carrier PGT – Brake Br2 ON</i>
<b>S12NS(W)</b>  $i_{Br1} = -\frac{T_B}{T_A} = \frac{t_I}{1+t_I} < 1$	<b>S12NS(E)</b>  $i_{Br2} = -\frac{T_B}{T_A} = +\frac{1+t_{II}}{t_{II}} > 1$
<i>Second two-carrier PGT – Brake Br3 ON</i>	<i>Second two-carrier PGT – Brake Br4 ON</i>



The operating states and the respective control element activations are listed with the transmission speed ratio

formulae in Table 2. These formulae may be further used in the synthesis and analysis of a planetary gearbox.

**Table 2.** Locked elements (brakes and couplings) at each gear and corresponding speed ratio  $i_k$  as function of torque ratios of component PGTs.

Gear $n$	I two-carrier PGT			II two-carrier PGT			Speed ratio $i_k$
	Br1	Br2	S1	Br3	Br4	S2	
1		•		•			$\frac{1+t_{\text{II}}}{t_{\text{II}}} (1+t_{\text{IV}})$
2			•	•			$1+t_{\text{IV}}$
3		•			•		$\frac{1+t_{\text{II}}}{t_{\text{II}}} \left( 1 + \frac{t_{\text{III}} t_{\text{IV}}}{1+t_{\text{III}}+t_{\text{IV}}} \right)$
4	•				•		$\frac{t_1}{1+t_1} (1+t_{\text{IV}})$
5				•		•	$1 + \frac{t_{\text{III}} t_{\text{IV}}}{1+t_{\text{III}}+t_{\text{IV}}}$
6	•					•	$\frac{t_1}{1+t_1} \left( 1 + \frac{t_{\text{III}} t_{\text{IV}}}{1+t_{\text{III}}+t_{\text{IV}}} \right)$
7		•				•	$\frac{1+t_{\text{II}}}{t_{\text{II}}}$
8			•			•	1
9	•					•	$\frac{t_1}{1+t_1}$

#### 4 Power analysis

For gear train efficiency calculation (by formula (6)) the real torque determination is needed. Their values depend

on the direction of relative (rolling) power  $P_{\text{rel}}$  in component PGTs – see section c) of paragraph 1.3. In Table 3 by green dotted arrow this direction is shown. The table shows formulae for real torques and torque ratio  $i_T$  determination for four operating states of the

gear train in question. By this torque ratio the efficiency for all gears of the change-gear train may be calculated. A software for this purpose is developed. Values of the basic efficiencies of component PGTs may be taken as constants, as well as calculated by formulae (10) or by more complex way.

**Table 3.** Power flows, real torques, and efficiencies for characteristic operating states of the 9-speed change-gears train.

<i>Planetary gearbox S12NS-S36SE</i>	
<i>First two-carrier PGT – Brake Br1 ON</i>	<i>First two-carrier PGT – Brake Br2 ON</i>
<p><b>S12NS(W)</b></p> $\eta_{Br1} = \frac{\left(\frac{T_B}{T_A}\right)_{\text{with losses}}}{\left(\frac{T_B}{T_A}\right)_{\text{without losses}}} = \frac{t_I \eta_0 I}{1 + t_I \eta_0 I}$	<p><b>S12NS(E)</b></p> $\eta_{Br2} = \frac{\left(\frac{T_B}{T_A}\right)_{\text{with losses}}}{\left(\frac{T_B}{T_A}\right)_{\text{without losses}}} = \frac{1 + \frac{t_II}{\eta_0 II}}{1 + t_II}$
<i>Second two-carrier PGT – Brake Br3 ON</i>	<i>Second two-carrier PGT – Brake Br4 ON</i>
<p><b>S36SE(N)</b></p> $\eta_{Br3} = \frac{\left(\frac{T_B}{T_A}\right)_{\text{with losses}}}{\left(\frac{T_B}{T_A}\right)_{\text{without losses}}} = \frac{1 + \eta_0 IV * t_{IV}}{1 + t_{IV}}$	<p><b>S36SE(W)</b></p> $\eta_{Br4} = \frac{\left(\frac{T_B}{T_A}\right)_{\text{with losses}}}{\left(\frac{T_B}{T_A}\right)_{\text{without losses}}} = \frac{1 + \eta_0 III * t_{III} + \eta_0 IV * t_{IV} + \eta_0 III * \eta_0 IV * t_{III} * t_{IV}}{1 + t_{III} + t_{IV}}$ $T_A = + \frac{1 + \eta_0 III * t_{III} + \eta_0 IV * t_{IV}}{\eta_0 IV * t_{IV}}$

## 5. Numerical example

The data from Table 2 have been used to create a program (software) enabling the calculation of all speed ratios of a planetary gearbox for different combinations of ideal torque ratios ( $t$ ) or, alternatively, different gear tooth number combinations ( $z_3/z_1$ ) of component PGTs.

For all cases from Table 2 formulae for efficiency determination as function of torque ratios ( $t_I$ ,  $t_{II}$ ,  $t_{III}$ ,  $t_{IV}$ ) and basic efficiencies are developed.

b) For gear tooth numbers  $z_{1I} = 21$ ,  $z_{3I} = 63$ ,  $z_{1II} = 29$ ,  $z_{3II} = 101$ ,  $z_{1III} = 23$ ,  $z_{3III} = 103$ ,  $z_{1IV} = 17$ ,  $z_{3IV} = 53$ . It must be noted that the gear tooth numbers in this case have been selected to obtain values as close as possible to those from example a), resulting in the charts displayed in Fig. 10.



**Fig. 9.** The influence of control element activation on gearbox overall transmission speed ratios (left) and gearbox efficiency in relation to control element activation (right) for defined ideal torque ratios and average gearset efficiency values.



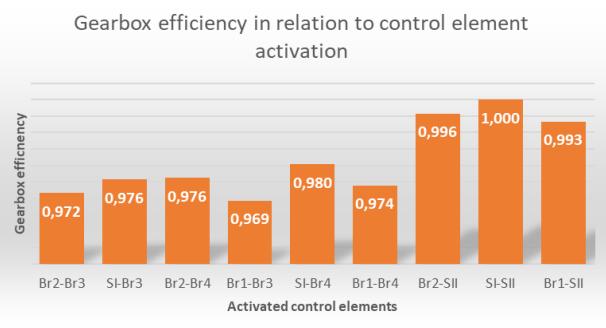
**Fig. 10.** The influence of control element activation on gearbox overall transmission ratios (left) and gearbox efficiency in relation to control element activation (right) for defined gear tooth numbers and calculated gearset efficiency values.

## 6 Conclusions

The purpose of this paper is to discover the transmission speed ratio functions that present the basis for the synthesis and analysis of four-carrier planetary gear trains composed of two series-connected two-speed, two-carrier PGTs equipped with two clutches, enabling the gearset to achieve nine operating states (gears). The analyses have shown that a gearbox suitable for electric vehicles may be obtained by selecting the appropriate torque ratios or gear tooth numbers.

Two programs (software) have been created to determine the transmission speed ratios and efficiencies for each operating state. It has been confirmed that close ideal torque ratios will not cause significant changes in the overall transmission speed ratios. Furthermore, it was established that the overall efficiency of the gearbox will be affected based on whether the efficiencies of the planetary gears are calculated based on the gear tooth numbers or by using the values recommended in literature ( $\eta_0 = 0,98$ ).

The analysis of the data in the chart shows that there is no significant difference in the transmission speed ratios, and there are no significant changes in overall gearbox efficiency.



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