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Research Paper

Research into the Properties of Selected Single Speed Two-Carrier Planetary Gear Trains

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Abstract. A two-carrier planetary gear train (PGT) configuration, named S13WN, was developed for a specific application with a negative transmission ratio in the interval between -20 and -21. 72 valid combinations of component PGTs ideal torque ratios have been developed for this configuration, from which kinematically feasible combinations providing minimal dimensions and maximum efficiency within the required interval have been selected. The minimal dimensions of the PGT were achieved with a cylindrical case shape, i.e., with the ratio of the ring gears reference diameters close to unity. Nine other PGT configurations have been synthesized for the same transmission ratio, and their design parameters optimized. The two solutions that offer the most improvements over S13WN have been developed into design concepts.

Keywords: Two-carrier planetary gear train, transmission ratio, efficiency, planetary gear train radial dimensions, ring gear reference diameter ratio.

1. Introduction

Almost all machines require some form of mechanical power transmission, as it enables the transfer of mechanical energy from the driving machine to the driven machine. The transmission often provides other useful functions, such as changing the direction, frequency or magnitude of forces or torques acting on the driven machine [1]. Geared transmissions are some of the most used forms of power transmission. Because of that, they are often subject to the interest of numerous authors, who consider the various aspects of their modelling and simulation [2,3,4], design [5], thermal effects [6], overall efficiency [7], and optimization [8, 9], to name just a few.

Planetary gear trains (PGTs) are a special variant of geared transmission which offers several advantages in relation to conventional gear trains, the most notable being a compact design and improved durability and reliability due to the beneficial effect of power being split over several planet gears, which may be even further enhanced by vibration analysis [10] and customized bearing solutions [11]. This has enabled the design of PGTs having high power ratings combined with a wide range of transmission ratios, especially in an ever-expanding area of industrial applications. However, a large diversity of kinematic schemes and the need for relatively complex calculations in comparison to conventional gearboxes, means that systematic research must be undertaken to realize the full potential of planetary gearboxes.

Current research shows that industrial applications use transmission ratios in the range from 18 to 90 [12]. A basic (single carrier) PGT designated as 1AI is usually used for transmission ratios in the range from 3 to 8 and may be exceptionally used up to 12. This means that compound PGTs, created by combining two basic PGTs must be used to achieve the required transmission ratios [13-17].

Gearboxes using such compound PGTs have found a range of applications in cranes and transportation technology in general, for example as a replacement for worm and involute elevator main drive gearboxes. Machine tool gearboxes are another important application, which was covered in [18], and optimization of two-carrier two-speed PGTs with brakes on coupled shafts was covered. The possibility of optimization of two-carrier two-speed PGTs with brakes on single shafts for fishing boat main propulsion gearboxes is covered in [9].

Planetary gear trains have been widely used in the aviation industry, due to their small size and weight, quiet, smooth running, high loading capacity and a long service life. Unequal load sharing between gears in the PGT is a very common issue, so its impact on reliability of the PGT cannot be ignored, and a method for predicting the reliability of planetary gear train in partial load state is presented in [19].

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PGTs also have broad applications in the automobile and automation industries, notably in epicyclic automatic transmissions containing any number of basic, compound or complex-compound planetary gear sets are the subject of [20], where a generalized formulation for the analysis of speeds and forces acting on the gear components of planetary automatic transmissions is presented. Furthermore, the correct calculation of the internal power flows of a PGT is prerequisite to a successful design because the internal power flows may yield significant power losses at gear meshes and fail the concept. New concepts for the calculation of internal power flows such as the split-power ratio and the virtual split-power ratio are introduced and presented in [21].

PGTs are also used in electric vehicles because of their high power density and ability to be designed and operated as a multi speed transmission. Furthermore, PGTs are resistant to conditions that can be encountered in an electric vehicle drivetrain, e.g., high input speed, large input speed variations, and motor torque fluctuations. A hybrid dynamic model for helical PGTs that operate in conditions of high and variable input speed was proposed in [22].

The application of PGTs in epicyclic transmission has led to multilink planetary gear trains that are the subject of interests of many studies.

The conjoint analysis of the overall efficiency and transmission ratio ranges that can be achieved with all the possible constructive solutions of four-, five-, and six-link planetary gear trains was the objective of [23]. The range of transmission ratios that can be achieved with each train was obtained, and the overall efficiency as a function of the transmission ratios within that range was also calculated.

Similarly, seven speed PGTs based on six-speed PGTs for heavy-duty commercial vehicles have been proposed [24]. To reduce the design cost of PGTs using a torque converter as the power input, a systemic synthesis method of seven-speed PGTs was proposed, while a design approach to systematically synthesize feasible configurations for series–parallel and parallel hybrid transmissions subject to design constraints and required operation modes using a basic PGT was presented in [25].

The analysis of PGTs is a complex issue that involves the task of conceptualizing the configuration of a PGT, and the definition of its fixed, clutched and input-output interconnections between gear members to achieve a desired set of input-output ratios. Mathematical representation of a PGT is central to all such procedures. The generalized lever is a useful tool in gear train representation [26] and sees important usage in many studies.

There has been relatively little research into two-carrier PGTs, mostly sporadic, however some systematic research into multi-carrier PGTs has been carried out in the last decade or so. The structures or means of connection between the component gear trains have been systematically researched in [27] and methodology has been provided to determine the transmission ratios and overall efficiency by means of lever analogy.

The kinematic properties of the structures have been extensively researched in [28, 29] as well as the efficiencies of single and two-speed PGTs by means of the torque method from [30, 31]. Furthermore, within the research carried out in [12] the DVOBRZ software was developed, enabling the synthesis, analysis, and optimal selection of two-carrier PGTs. Some of the results obtained by means of this software package have been used in this paper.

2. The Researched Two-Carrier, Single Speed Planetary Gear Train

The subject of this paper is a single speed two-carrier PGT, schematically shown in Fig. 1. The application constraints require a kinematically negative transmission ratio between -21 and -20, with the component PGTs being of similar size. This will result in the casing having the simplest possible shape, which will reduce manufacturing costs.

This particular gear train is composed of component gear trains I and II. Input A is to sun gear I, while planet carriers I and II are connected together to output B. Ring gear I is connected to hollow sun gear II through which the shaft connecting planet carriers I and II is passing. A large rolling bearing supports the rotation of ring gear I, while ring gear II is locked to the gear train casing. The application also demands the use of this particular gearset, which is described in [32].

In this article, the way in which two-carrier PGTs are composed will be introduced first, and the layout variants explained as well. DVOBRZ software will be then applied to determine possible solutions which satisfy the design requirements. However, this will result in other two-carrier PGTs capable of fulfilling the same application being identified. Their efficiencies will then be determined, and dimensional analysis carried out to determine the optimal design solution. The analysis and comparison of all variants can be performed then to determine whether improved solutions are possible.

3. Component PGTs and Two-Carrier Single Speed PGTs

The most commonly used basic PGT, 1AI, is shown in Fig. 2 together with the specific torques on its shafts and its Wolf-Arnaudov symbol [16]. This PGT is also commonly used as a component of complex PGTs [33]. It is of relatively simple construction, its parts being the sun gear 1, the planet gears 2, the ring gear (annulus) 3 and planet carrier S.

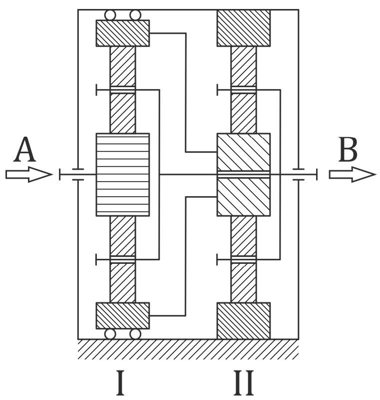

Fig. 1. Schematic overview of the researched two-carrier, single speed planetary gear train.

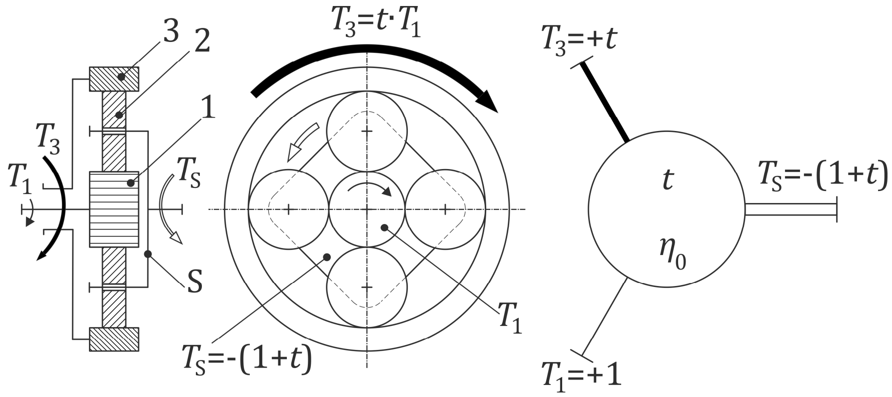

Fig. 2. 1AI, the most used basic PGT with the specific torques on its shafts indicated (left) and its Wolf-Arnaudov symbol (right).

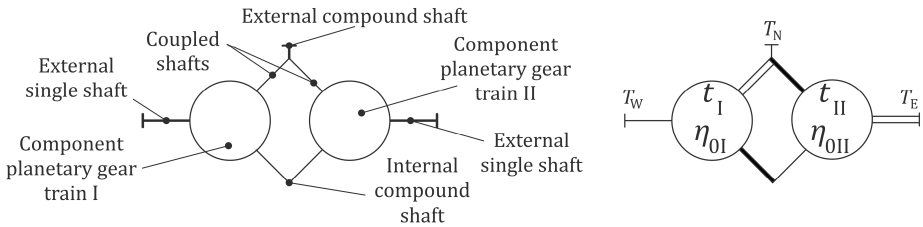

Fig. 3. General case of two-carrier PGT symbol with shafts labelled (left) and the symbol of an actual PGT with shaft torques marked (right).

The 1AI PGT has imposed itself over other PGTs as it offers most of the positive sides of PGT application. Internal gearing ensures lower contact loads and smaller dimensions as all other gears are placed inside the ring gear. Having a single row of planet gears provides for a compact build and ease of manufacture due to multiple equal parts. Due to all those benefits, the 1AI PGT is commonly used in gearbox manufacturing, and besides that it is commonly found in complex PGTs [14]. It should be noted that the ideal torque ratio t of the PGT is given by Eq. (1), while the shaft torque ratio is given by Eq. (2), where z_1 is the number of teeth of the sun gear, z_3 is the number of teeth of the ring gear, T_1 is the torque acting on the sun gear shaft, T_3 is the torque acting on the ring gear shaft, T_S is the torque acting on the planet carrier shaft, and T_D is the differential torque:

$$t = \frac{T_3}{T_1} = \frac{T_{D\max}}{T_{D\min}} = \frac{|z_3|}{z_1} > +1 \tag{1}$$

$$T_1 : T_3 : T_S = +1 : +t : -(1+t) \tag{2}$$

Multi-carrier PGTs are created by connecting the shafts of various basic PGTs, commonly known as basic types (1AI) [15]. As two-carrier PGTs are the subject of this paper, we shall consider one-speed, two-carrier PGTs with three external shafts composed of two basic PGTs. Of the three external shafts, two are single shafts and one is a compound shaft, as indicated in Fig. 3.

Some two-carrier PGTs are capable of relatively high transmission ratios while having a reasonably high overall efficiency. Fig. 3 schematically shows the connection between two basic component gear trains. A general case of the connection between component PGTs is shown in the left side of the image, complete with the names of the parts being connected, while the right side shows the symbol of an actual two-carrier PGT with torque markings on the external shafts. The torque markings of the external shaft torques (T_W, T_N and T_E) follow the cardinal directions (W, N, E), and are ordered from power input to power output. The symbol contains the markings of the ideal torque ratio (t_I and t_{II}) and efficiency (η_{0I} and η_{0II}) for every basic component PGT [9].

An overview of possible structures of two-carrier single speed PGTs has been given in Table 1 [16]. It shows that the basic component PGTs can be combined in 36 possible ways, giving 36 different PGT symbols. As some layouts are isomorphous, this is reduced to 21 practical layouts. Every PGT can provide six different operating modes, as the stationary member may be any of three external shafts, with the remaining two external shafts acting as input and output.

Therefore, it is possible to achieve a total of 126 (21 layouts x 6 operating modes = 126) different transmission variants [16]. The scheme and operating mode are noted with a matrix type designation (e. g. S15 – line 1, column 5), while the power input and output are marked by cardinal directions, the stationary element being placed in parentheses. Therefore, S15WN(E) points to layout 15 with power input being in the west, power output being in the north, and the eastern shaft being locked. However, as we explicitly state that the PGT has three external shafts, it is not necessary to state the stationary element, therefore it is enough to write just S15WN to fully designate a PGT.

4. Research of Single Speed Planetary Gear Train S13WN

After confronting the schematic of the researched single speed PGT already shown in Fig. 1 with the schematics from Table 1, it has been determined that according to the classification used in this paper, the PGT in question is layout S13, operating mode WN(E), hence S13WN (Fig. 4). Therefore, S13WN will be the first variant to be researched, after which the research will be extended to all other variants fulfilling the application conditions.

Table 1. Overview of existing PGT layouts.

	...1 t_{II}	...2 t_{II}	...3 t_{II}	...4 t_{II}	...5 t_{II}	...6 t_{II}
1... t_I	11	12	13	14	15	16
2... t_I	21≡12	22	23	24	25	26
3... t_I	31≡13	32≡23	33	34	35	36
4... t_I	41≡14	42≡24	43≡34	44	45	46
5... t_I	51≡15	52≡25	53≡35	54≡45	55	56
6... t_I	61≡16	62≡26	63≡36	64≡46	65≡56	66

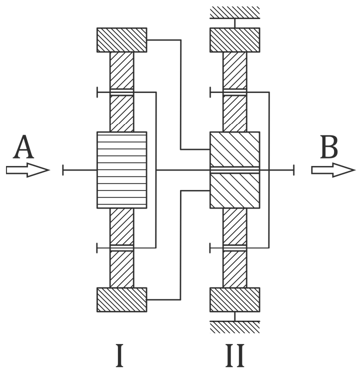

Fig. 4. Schematic overview of the two-carrier, single speed planetary gear train S13WN.

The design parameters required to manufacture the PGT were determined from constraints suggested in the literature, although it is expected that all gears will be made from the same material. Torque method analysis and the DVOBRZ software package will be used to identify valid solutions having a kinematically negative transmission ratio between -21 and -20, according to the criteria of minimal dimensions (cylindrical casing) and maximum efficiency (stepped casing).

4.1 Torque Method Analysis

Structural analysis has shown that PGT variant S13WN features internal power circulation [8]. Fig. 5 shows the ideal (white background) and real (gray background) specific torques on all PGT shafts. Red solid lines denote the absolute power flow, the green dashed lines denote the relative power flow within the PGT, P_A denotes input shaft power, while P_B denotes output shaft power.

The transmission ratio is given by Eq. (3), while the relation of the PGT efficiency to the ideal torque ratios (t_I, t_{II}) and internal efficiencies (η_{0I}, η_{0II}) is given by Eq. (4). Equations (3) and (4) have been integrated into the DVOBRZ package and present the basis for the determination of the transmission ratio and overall efficiency of the S13WN PGT.

The internal efficiencies η_{0I} and η_{0II} are determined according to the expressions in [28], which consider the number of teeth of the sun gear, ideal torque ratio of the component PGT, coefficient of churning losses k_c, coefficient of planet bearing losses k_b and coefficient of seal frictional losses k_s.

$$i = -\frac{T_B}{T_A} = -(t_I \cdot t_{II} - 1) \tag{3}$$

$$\eta = \frac{(T_B / T_A)_{\text{losses}}}{(T_B / T_A)_{\text{without losses}}} = \frac{\eta_{0I} \cdot \eta_{0II} \cdot t_I \cdot t_{II} - 1}{t_I \cdot t_{II} - 1} \tag{4}$$

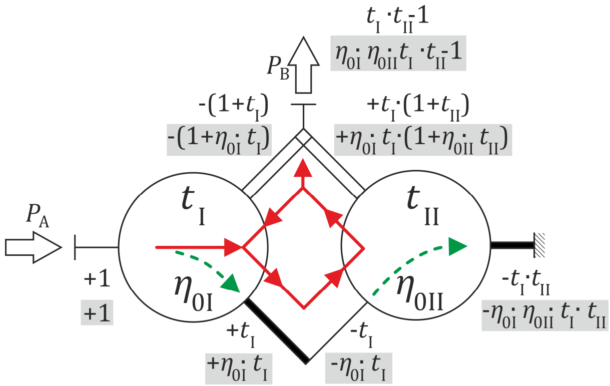

Fig. 5. Torque method analysis of S13WN PGT.

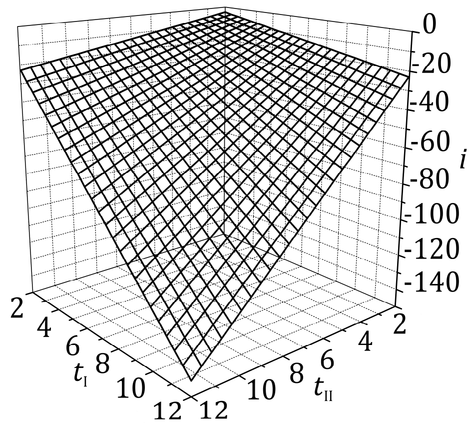

Fig. 6. Influence of ideal torque ratios t_I and t_{II} on the overall transmission ratio i.

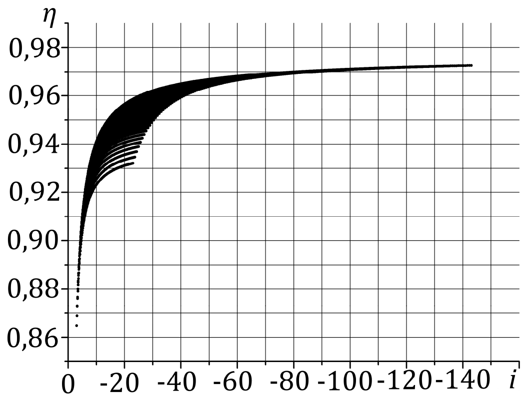

Fig. 7. Influence of the overall transmission ratio i on PGT efficiency η.

Equation (3) has been used to plot a graph of possible transmission ratios as a function of ideal torque ratios of the component PGTs t_I and t_{II} in Fig. 6. The largest negative transmission ratio can be achieved for ideal torque ratios $t_I = 12$ and $t_{II} = 12$. Equation (3) can be used to calculate the exact transmission ratio range, going from the maximum value of -143 to the minimum value of -3. This minimum value will be achieved for the smallest ideal torque ratios of $t_I = 2$ and $t_{II} = 2$. The ideal torque ratios for three coplanar planet gears cannot exceed 12 because of interference issues [18]. It is possible to determine the relation of the transmission efficiency to the overall transmission ratio, as visible in Fig. 7. It has been determined that the overall efficiency increases with the transmission ratio for this PGT type. Furthermore, it can be concluded from the diagram in Fig. 7 that PGTs with transmission ratios in the -21 to -20 interval have significant variations in efficiency. This enables optimisation for maximum PGT efficiency.

4.2 Principle of Operation of the DVOBRZ Software Package

As the DVOBRZ software was used to identify viable solutions under application constraints, the principle of operation of the software must be explained. The DVOBRZ program was originally developed to identify the variants of two-carrier PGTs and their parameters that fulfil the kinematic requirements, and list them in order of priority according to the selection criterion, e.g., maximal efficiency, minimal weight, or size. Support for multi-criteria optimisation has been provided as optimisation criteria do not have to be equally important. The weighted coefficient method can be used as the weight coefficients represent the importance of every optimisation criterion, as every two-speed drive may be realised with several combinations of basic component gear trains. The program can provide solutions for two-speed and single-speed gearboxes, depending on whether the actual gearbox will have a fixed transmission ratio or a user operated shifting mechanism. A flow diagram of the software can be seen in Fig. 8.

The program operates by checking the ideal torque ratios of every possible combination of basic component gear trains and discards those that cannot provide the required transmission ratios. The range of the ideal torque ratios is determined by design limits, notably the number of planet gears per component gear train. The transmission ratio for both gears is calculated for every possible combination of ideal torque ratios and is checked whether it is within the tolerance range for the desired transmission ratios. The ideal torque ratios are represented using the numbers of teeth on sun and ring gears for both component gear trains (Eq. 5 and Eq. 6):

$$t_{I} = \frac{|z_{3I}|}{z_{1I}} \tag{5}$$

$$t_{II} = \frac{|z_{3II}|}{z_{1II}} \tag{6}$$

The tooth numbers of the sun gears z_{1I} and z_{1II} must be set on program initialization. The program will then enlarge one ring gear (usually z_{3I}) by one tooth and check whether the ideal torque ratio is valid, which is achieved if the basic component gear train satisfies the assembly conditions. If it does not, the ideal torque ratio is discarded, and the ring gear enlarged by one more tooth. This procedure is repeated until a valid ideal torque ratio is found or the maximum allowable ideal torque ratio for that component gear train is reached. The same procedure will then be carried out for the second basic component gear, creating a set of possible transmission ratios.

The program calculates and stores the values of different parameters for each valid member of the set of ratios (basic geometry of component gear trains, component efficiency, transmission ratios, overall efficiency for each transmission ratio etc.) as a function of the ideal torque ratios of the component gear trains t_I and t_{II}. The resulting database is then used to select the best gearbox variant for the application, whether according to a single criterion (overall efficiency, minimal ratio of ring gear reference diameters, reference diameter of the largest ring gear etc.), or by multi-criteria optimisation. In the case of multi-criteria optimisation, the weighting coefficients for each optimisation criteria must be determined according to the application conditions, depending on how important each criterion is for the application demand.

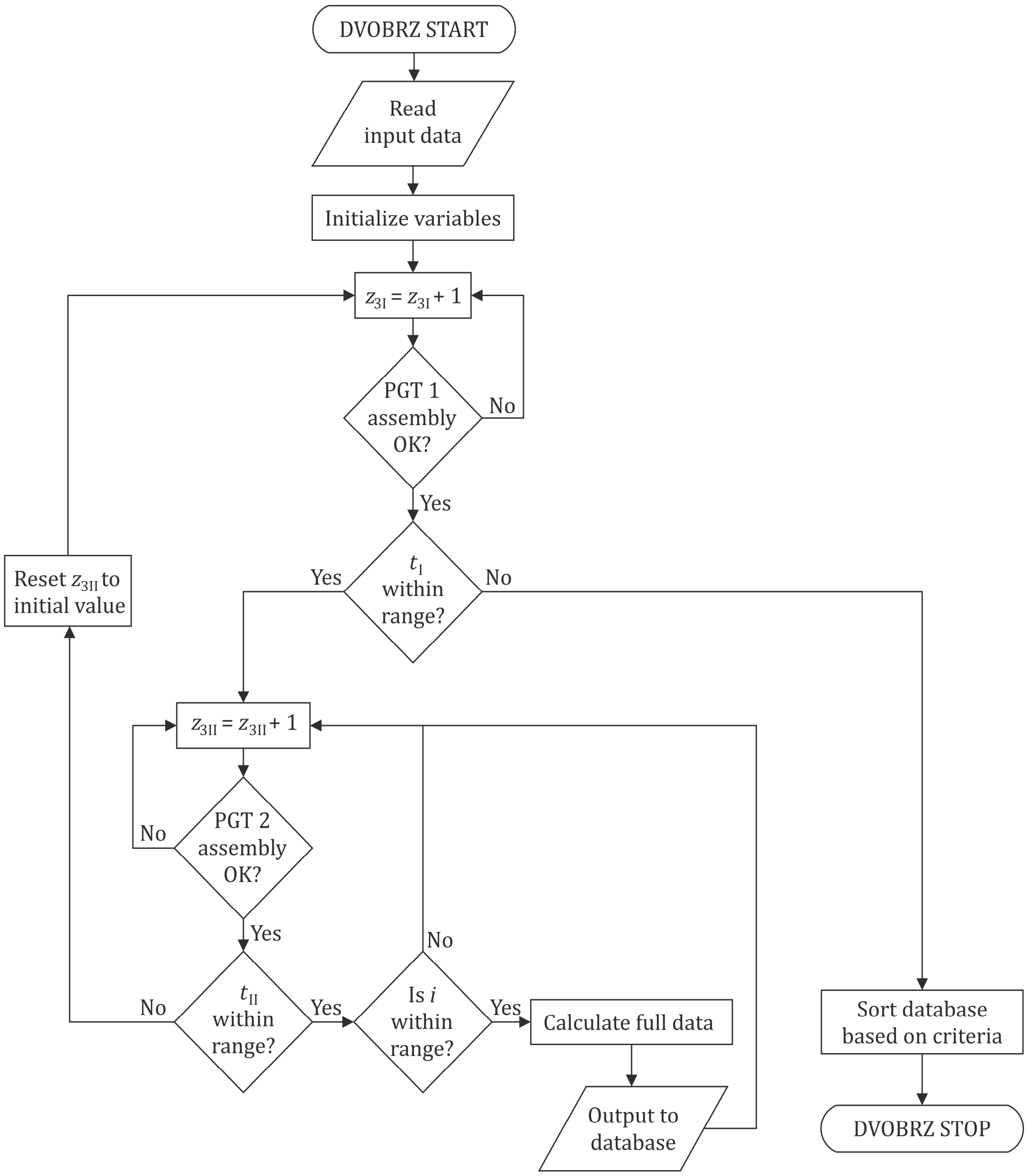

Fig. 8. Flow diagram of the DVOBRZ program.

However, a kinematic scheme must be created for any selected layout variant to check out whether the solution is kinematically valid, and that it meets the relevant design and technological criteria. This will result in the most appropriate variant being selected.

The program has been coded using Fortran 95 as a 32-bit console application. One full calculation cycle required to prepare data ready for kinematic validation takes about 10 minutes on a 7th generation i7 CPU. Generally speaking, the program running time significantly depends whether additional calculations have to be performed, e.g. whether the numbers of the teeth of the sun gears, and the numbers of the planet gears are being preset or varied. Most of the running time, however, is spent on post-processing and sorting of valid ideal torque pairs according to the optimization criteria. This could be somewhat improved by rewriting the code into C, or a more modern variant of Fortran.

4.3 Application of the DVOBRZ Software Package

The DVOBRZ software package was used to determine the basic parameters of transmissions fulfilling the application demands. The most important input data is summarised as follows:

- Number of teeth of the first sun gear z_{1I} = 21(selected value),
- Number of teeth of the second sun gear z_{1II} = 21 (selected value),
- Overall transmission ratio i $\in$(-21…-20) (application demand),
- Average value of planet bearing losses coefficient k_b = 0,065 [12,14],
- Average value of seal frictional losses coefficient k_s = 0,05 [12,14],
- Average value of churning losses coefficient k_c = 0,125 [12,14],
- Gear width to diameter ratio b / d_1 = 0,7 (selected value),
- Number of planets per PGT k = 3 (application demand),
- Gear material 16MnCr4 steel (application demand),
- Overall efficiency $\eta \geq$ 0,93 (application demand).

The analysis module finds 72 possible solutions or combinations of ideal torque ratios for layout S13WN which provide the required transmission ratio (Fig. 9 and Fig. 10).

Every point in the domain (horizontal x-y plane) presents a pair of ideal torque ratios enabling an overall torque ratio in the desired range. The vertical (z) axis on Fig. 9 presents the size ratio of the larger and smaller PGT ring gear. The chart shows that this ratio can variate between 1 and more than 4. Further analysis of the results has shown that PGTs with z - axis values equal or close to 1 will have minimal radial dimensions. The z – axis in Fig. 10 is used to represent the overall efficiency of the PGT in relation to the combination of ideal torque ratios. The chart shows that for torque ratios in the 2 to 10 range, efficiencies ranging from 0,944 to 0,964 can be achieved. The results are shown condensed in Table 2.

It can be seen in Table 2 that the optimal solution in accordance with the criteria of maximum efficiency provides a borderline increase in efficiency in relation to the solution for minimum dimensions, however the component PGTs will have different outside diameters. Therefore, it can be concluded that the better solution is to optimise for equal outside diameters, as the decrease in overall efficiency will be negligible.

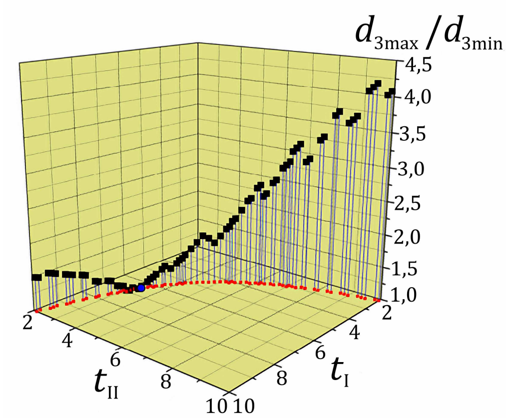

Fig. 9. The influence of ideal torque ratios on the size ratio of the larger and smaller ring gears diameters.

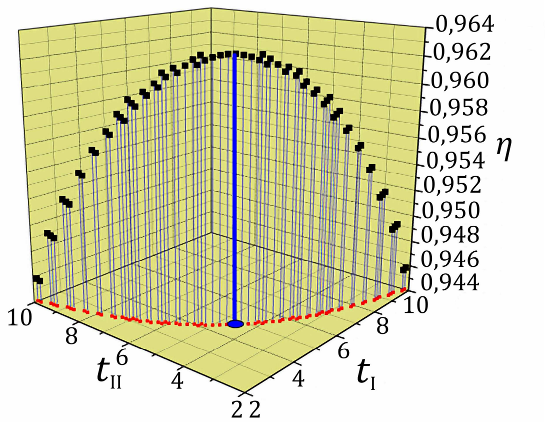

Fig. 10. The influence of ideal torque ratios on overall efficiency.

Table 2. Research results.

Minimal size – cylindrical casing	Maximum efficiency – stepped casing	Improved version	Percentage
Ideal torque ratio I t_I = 6,4286	Ideal torque ratio I t_I = 4,7143	n. a.	
Ideal torque ratio II t_{II} = 3,2857	Ideal torque ratio II t_{II} = 4,5714	n. a.	
Overall efficiency η = 0,960	Overall efficiency η = 0,963	Stepped	0,4%
Number of teeth of sun gear I z_{1I} = 21	Number of teeth of sun gear I z_{1I} = 21	n. a.	
Number of teeth of sun gear II z_{1II} = 21	Number of teeth of sun gear II z_{1II} = 21	n. a.	
Number of teeth of ring gear I z_{3I} = 135	Number of teeth of ring gear I z_{3I} = 99	n. a.	
Number of teeth of ring gear II z_{3II} = 69	Number of teeth of ring gear II z_{3II} = 96	n. a.	
Gear train I module m_I = 1,375 mm	Gear train I module m_I = 1,375 mm	n. a.	
Gear train II module m_{II} = 2,75 mm	Gear train II module m_{II} = 2,5 mm	n. a.	
Ring gear I reference diameter d_{3max} = 189,75 mm	Ring gear I reference diameter d_{3max} = 240 mm	Cylindrical	21%
Ring gear reference diameter ratio d_{3max} / d_{3min} = 1,022	Ring gear reference diameter ratio d_{3max} / d_{3min} = 1,763	Cylindrical	42%

Table 3. Priority list of PGTs according to the criteria of minimal radial dimensions.

Rank	Designation	t_I	t_{II}	i	η	d_{3max}, mm	d_{3max} / d_{3min}
1.	S66NE	2	1,8571	-20	0,417	136,5	1,083
2.	S33NE	3	2,8571	-20	0,463	150	1,058
3.	S26WN	5,1429	2,4286	-20,061	0,961	153	1,030
4.	S55NE	1,8571	2,1429	-20,429	0,594	156	1,156
5.	S16WE	5,4286	2,7143	-20,163	0,963	156,75	1
6.	S44NE	3,2857	2,7143	-20,357	0,775	172,5	1,009
7.	S23WN	6	2,8571	-20	0,960	180	1,039
8.	S13WN	6,4286	3,2857	-20,122	0,960	189,75	1,022
9.	S11NW	3	2,8571	-20	0,463	210	1,026
10.	S22NE	4,7143	5	-20	0,619	346,5	1,015

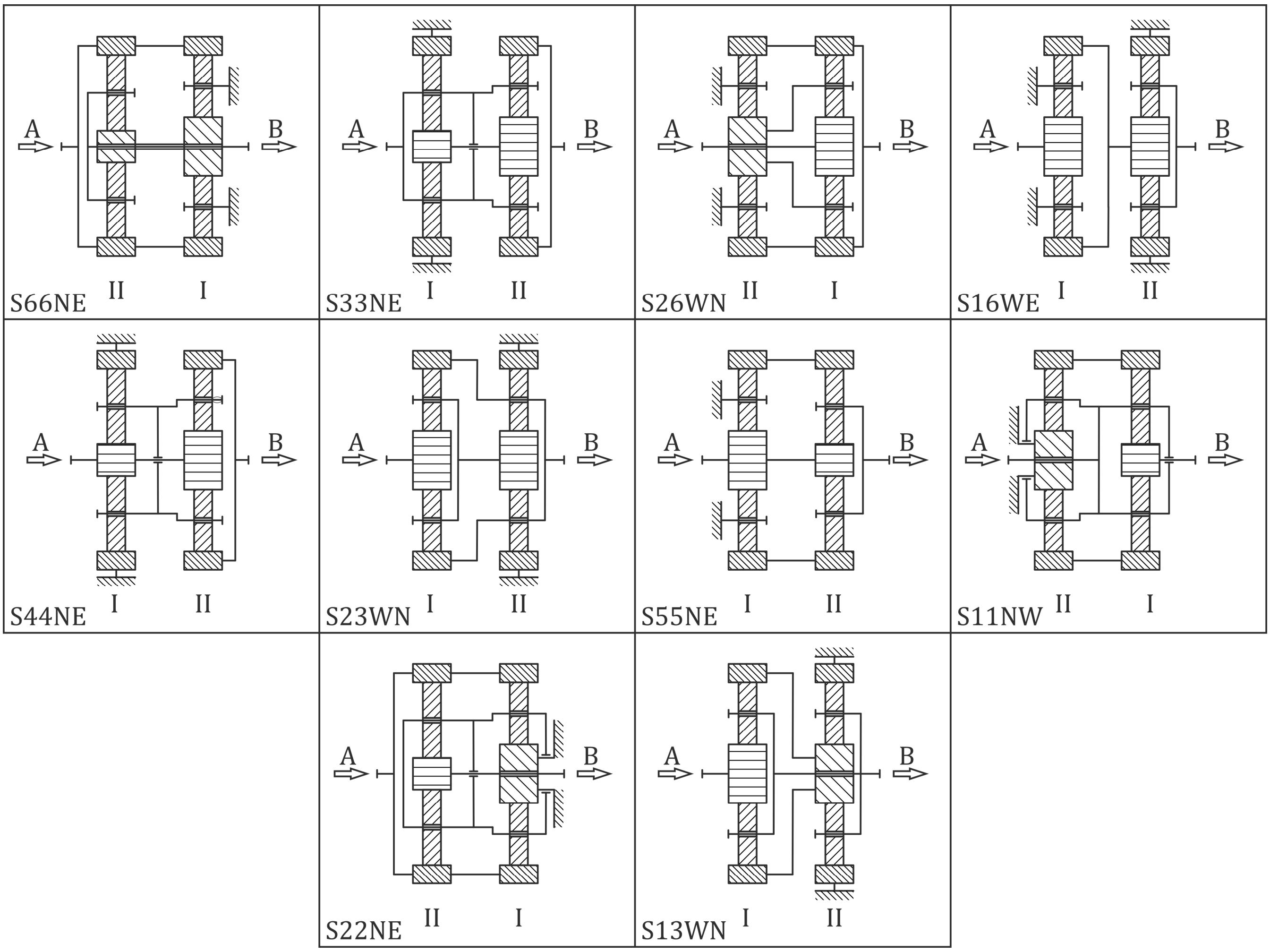

Fig. 11. Single speed transmission variants.

Table 4. Priority list of PGTs according to the criteria of maximum efficiency.

Rank	Designation	t_I	t_{II}	i	η	d_{3max}, mm	d_{3max}/d_{3min}
1.	S16WE	4,4286	3,7143	-20,878	0,965	195	1,398
2.	S23WN	3,7143	4,4286	-20,878	0,965	232,5	1,987
3.	S26WN	3,5714	3,7143	-20,551	0,965	195	1,733
4.	S13WN	4,7143	4,5714	-20,551	0,963	240	1,763
5.	S55NE	6,4286	9,8571	-20,357	0,937	362,25	1,342
6.	S44NW	6,4286	9,8571	-20,357	0,937	362,25	1,342

The remaining four solutions are not listed as they do not meet the efficiency criteria

5. Exploration of Alternate Solutions

The DVOBRZ software package can detect all PGT layouts capable of achieving the desired transmission ratio. Therefore, other two-carrier PGTs capable of fulfilling the same application were identified, and their efficiencies calculated. Dimensional analysis was performed to determine the optimal design solution. The overall efficiency of the identified PGTs was compared to the maximal possible efficiency, which depends on the ideal torque ratio distribution between the two component gear trains. This calculation has been made with the presumption that gears will be made from the same material. After processing the input data, it has been determined that the required transmission ratios can be achieved by the following ten transmission variants: S66NE, S33NE, S26WN, S55NE, S16WE, S44NE, S23WN, S13WN, S11NW and S22NE. The design concepts of these transmission variants are shown in Fig. 11.

Every variant has its specific mechanical, sizing, mass, and manufacturing properties. An analysis of the solutions (Table 3) will cause variant S66NE to be rejected immediately, as it provides minimal dimensions with an unacceptably low overall efficiency. Variants 3 and 5 provide much better solutions with regards to gearbox size. However, variants S66NE and S33NE would be acceptable if a gearbox with self-locking properties was required.

The data in Table 4. Shows that the S13WN variant ranks fourth according to the efficiency criteria, however it lags considerably behind variant S16WE (ranked first) and variant S26WN (ranked third) regarding the ring gear reference diameter d_{3max} criterion.

Based on the analysis provided in Table 3 and Table 4, it can be concluded that variant S13WN does not provide an optimal solution as it has a ring gear diameter larger than variants S16WE and S26WN.

Therefore, according to the application criteria, any of those two gearboxes would provide a better solution than variant S13WN. The conceptual schematics of variants S16WE and S26WN are provided in Fig. 11 and Fig. 12.

Fig. 12 (left) shows variant S16WE having the input and output shafts on opposite sides, while Fig. 12 (right) shows variant S26WN in an alternative configuration using the connecting outer ring gear shaft as the output. Variant S16WE is commonly used for marine propulsion and industrial applications, as the calculations for both component PGTs of S16WE are relatively simple and can be performed independently of each other. On the other hand, variant S26WN, in particular the alternate configuration described above, has recently found use as a replacement for elevator worm gear drives due to its high overall efficiency [34].

6. Conclusions

This paper deals with the analysis of a two-carrier PGT developed for a specific application, having a transmission ratio in the range from -21 to -20. The application conditions request PGT type S13WN to be used. The DVOBRZ software package was used to determine the variants and basic parameters of two-carrier drives fulfilling the application demands, while taking into consideration the design parameters such as gear geometry of the component gear trains, overall transmission ratio, average value of internal losses, gear material and overall efficiency. The identified variants were listed in order of priority according to the selection criteria. The values of different parameters for all valid PGTs were calculated and stored as a function of the ideal torque ratios of the component gear trains, and the resulting database was used to select the best gearbox variant for the application. The optimization was carried out to minimize the PGT dimensions, as well as to maximize efficiency. Analysis has shown that the overall efficiency of a PGT optimised for minimum size will be borderline smaller than of a PGT optimised for maximum efficiency, but the PGT optimised for minimum size will be considerably easier to manufacture, due to both ring gears being of the same size. Further analysis and comparison to all the other kinematically equivalent PGTs has shown that S13WN does not present the best solution according to either criterion. It has been determined that superior solutions would be provided by S16WE and S26WN according to both the criteria of minimum size and maximum efficiency. It must be also considered that those variants have a significant advantage over S13WN of not having internal power circulation, resulting in a considerably lighter build. Furthermore, solutions for single speed planetary gear trains have been determined for transmission ratios between -21 and -20, and most solutions have been found to be of lower mechanical efficiency. The DVOBRZ software could then be applied to extend the range of transmission ratios, for example between -20 and -25, providing a much greater number of solutions. Finally, selected transmission ratios from the whole range could be used to form a family of transmissions.

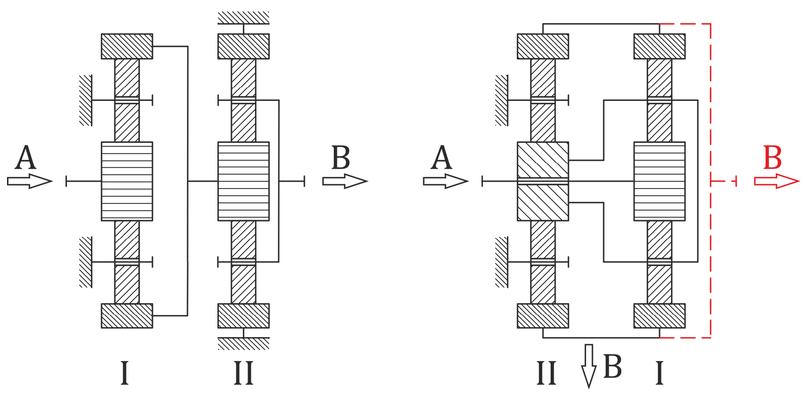

Fig. 12. Design concepts of PGTs S16WE (left) and S26WN (right). The original output of S26WN as in Fig. 11 has been added in dashed red lines for clarity.

Author Contributions

Ž. Vrcan prepared an overview of applications for the intended solutions, coordinated the research activities of the authors, prepared graphics and schematics, and assembled the contributions of all authors into the final manuscript; J. Stefanović-Marinović analyzed the solution set and developed procedures to select optimized solutions; M. Tica cross checked the optimized solution set and validated the theoretical assumptions; S. Troha initiated the research, implemented the theoretical assumptions into the software required to create the solution set, prepared the solution set and laid out the basic outline of the manuscript. The manuscript was written through contribution of all authors. All authors discussed the results, reviewed, and approved the final version of the manuscript.

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Conflict of Interest

The authors declare that they have no potential conflicts of interest concerning the research, authorship, and publication of this article.

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Data Availability Statements

The data generated and analyzed during the research covered in this paper is available from the corresponding author on reasonable request.

Nomenclature

b	Gear width, [mm]	z_{1I}	Number of teeth of sun gear I [-]
d_1	Gear pitch diameter, [mm]	z_{1II}	Number of teeth of sun gear II [-]
i	Overall transmission ratio [-]	z_{3I}	Number of teeth of ring gear I [-]
k	Number of planets per component gear train [-]	z_{3II}	Number of teeth of ring gear II [-]
k_b	Coefficient of planet bearing losses [-]	P_A	Input shaft power [W]
k_c	Coefficient of churning losses [-]	P_B	Output shaft power [W]
k_s	Coefficient of seal frictional losses [-]	T_D	Differential torque [Nmm]
d_{3max}	Ring gear I reference diameter [mm]	T_E	Eastern output shaft torque [Nmm]
d_{3min}	Ring gear II reference diameter [mm]	T_N	Northern output shaft torque [Nmm]
m_I	Module of component gear train I [mm]	T_W	Western output shaft torque [Nmm]
m_{II}	Module of component gear train I [mm]	η	Overall gear train efficiency [-]
t_I	Ideal torque ratio of component gear train I [-]	η_{0I}	Internal efficiency of component gear train I [-]
t_{II}	Ideal torque ratio of component gear train II [-]	η_{0II}	Internal efficiency of component gear train II [-]

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