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COMPARISON OF THE SIZE AND EFFICIENCY OF A TWO– CARRIER PLANETARY GEAR TRAIN AND KINEMATICALLY EQUIVALENT PLANETARY GEAR TRAIN

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Abstract: A two– carrier planetary gear train (PGT) developed for a specific purpose is discussed in this paper. This PGT is suitable for applications which require negative transmission ratios in the range from – 3 up to – 143. The mechanical and dimensional characteristics of the planetary gear train for nominal negative transmission ratios of – 30 and – 40 have been considered. Many combinations of ideal torque ratios of the PGT were obtained for the mentioned transmission ratios, from which only the combinations providing the minimum radial dimensions of the planetary gear units were selected. It was found that the minimum radial dimensions of the PGT will be obtained when the ratio of the reference diameters of the planetary unit ring gears of is close to unity, i.e., when the PGT housing is cylindrical rather than stepped. An introduction to single speed two– carrier planetary gear trains is given, in addition to an overview of the application of the DVOBRZ software package used to synthesize different gearboxes for the required transmission ratio. Acceptable gearboxes were selected from all PGTs according to the criteria of minimum dimensions and acceptable efficiency, and their construction concepts were created.
Keywords: two– carrier planetary gear train; transmission ratio; efficiency

INTRODUCTION

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. Besides that, the transmission 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, and planetary gear trains (PGTs) are a special variant of geared transmission which offers several advantages in relation to conventional gear trains. The most notable advantage is 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 and customized bearing solutions [2]. This has enabled the design of PGTs having high power ratings combined with a wide range of transmission ratios. 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 with ratios in the range from 18 to 90 [3]. A basic type of 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 PGTs of basic type must be used to achieve the required transmission ratios [4– 8]. Gearboxes using such compound PGTs have found a range of applications in cranes and transportation technology in general. Machine tool gearboxes are also an important area of application and optimization of two– carrier two– speed PGTs with brakes on coupled shafts which was covered in [9]. The possibility of optimization of two– carrier two– speed PGTs with brakes on single shafts for fishing boat main propulsion gearboxes is covered in [10].

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 long service life. A method for predicting the reliability of planetary gear train in partial load state was presented in [11].

The application of PGTs in the automobile and automation industries is also important, notably in automatic transmissions containing any number of simple, compound or complex– compound planetary gear sets [12]. New concepts for the calculation of internal power flows such as the split– power ratio and the virtual split– power ratio have been introduced and presented in [13].

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. A hybrid dynamic model for helical PGTs that operate in conditions of high and variable input speed was proposed in [14].

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 [15] and methodology has been provided to determine the transmission ratios and efficiency by means of lever analogy.

The kinematic properties of the structures have been extensively researched in [16,17] as well as the efficiencies of single and two– speed PGTs by means of the torque method from [18,19].

Furthermore, within the research carried out in [3] 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. The mechanical and dimensional characteristics of the planetary gear train for nominal negative transmission ratios of – 30 and – 40 have been considered. Several combinations of ideal torque ratios of the PGT were obtained, from which the combinations providing the minimum radial dimensions of the planetary gear units were selected. The best gearboxes were selected according to the criteria of minimum dimensions and acceptable efficiency, and their construction concepts were created.

THE RESEARCHED TWO– CARRIER, SINGLE SPEED PLANETARY GEAR TRAIN

The subject of this paper is a single speed two– carrier PGT. The application constraints require a kinematically negative transmission ratio of – 30 and – 40, with the component PGTs being of similar size. This will result in the casing having the simplest possible shape, which will in turn reduce manufacturing costs.

This type of casing can be achieved if the relation of maximum and minimal values of the ring gear diameters is close to one, and it is for this reason that the relation with ring gear diameters is considered.

Both component gear trains are the most commonly used simple PGT, 1AI, which is shown in Figure 1 together with the specific torques on its shafts and its Wolf– Arnaudov symbol [7]. 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.

The Wolf– Arnaudov symbol simplifies the representation of PGTs as the train shafts are represented by lines of different thickness and a circle. The sun gear shaft 1 is represented by a thin line, the ring gear shaft 3 by a thick line and the carrier shaft S by two parallel lines. The carrier shaft is the summary element because a negative transmission ratio is obtained by locking the planet carrier.

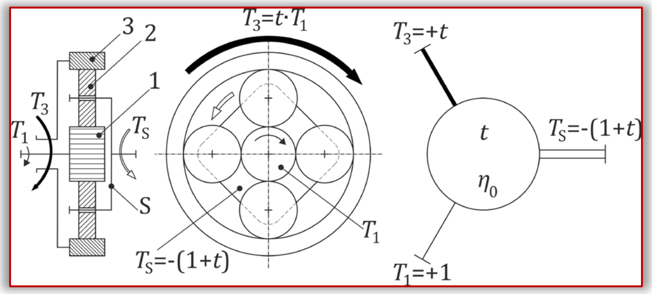


Figure 1. 1AI, the most used simple PGT with the specific torques on its shafts and its Wolf– Arnaudov symbol

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:

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

$$T_1 : T_3 : T_S = +1 : +t : -(1+t) \quad (2)$$

Multi– carrier PGTs are created by connecting the shafts of simple PGTs (1AI) [6]. 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 simple basic PGTs. Two of the three external shafts are single shafts and the third one is a compound shaft, as indicated in Figure 2.

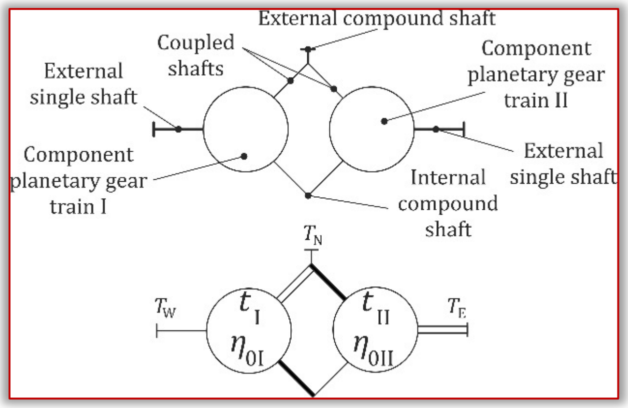
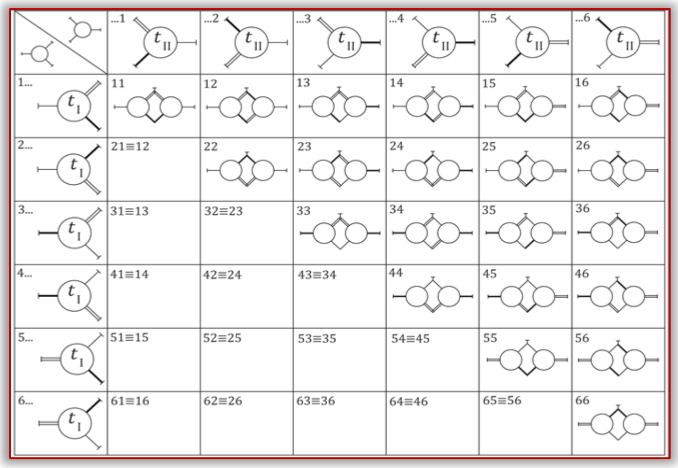


Figure 2. Two– carrier PGT symbol with shafts and respective torques marked

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 [10].

An overview of possible structures of two– carrier single speed PGTs has been given in Table 1 [7]. It shows that the simple 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.

Table 1. Existing PGT layout



Therefore, it is possible to achieve a total of 126 different transmission variants [7]. 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 enough to write just S15WN to fully designate a PGT.

DVOBRZ SOFTWARE PACKAGE

Since the DVOBRZ software was used to identify variable solutions under application constraints, the principle of operation of the software is explained in detail in [20] and will be briefly presented in this paper. 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. 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.

The program operates by checking the ideal torque ratios of every possible combination of simple component gear trains and discards those that cannot provide the required transmission ratios. 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. 3 and Eq. 4):

$$t_{\mathrm{I}} = \frac{|z_{3\mathrm{I}}|}{z_{1\mathrm{I}}} \qquad (3)$$

$$t_{\mathrm{II}} = \frac{|z_{3\mathrm{II}}|}{z_{1\mathrm{II}}} \qquad (4)$$

The tooth numbers of the sun gears $z_{1\mathrm{I}}$ and $z_{1\mathrm{II}}$ must be set on program initialization. The program will then enlarge one ring gear (usually $z_{3\mathrm{I}}$) by one tooth and check whether the ideal torque ratio is valid, which is achieved if the simple 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 simple component gear train.

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 optimization. In the case of multi– criteria optimization, the weighting coefficients for each optimization criteria must be determined according to the application conditions, depending on how important each criterion is for the application demand.

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.

APPLICATION OF THE DVOBRZ SOFTWARE PACKAGE

— Example 1

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

- Overall transmission ratio i=– 30,
- Number of teeth of the first sun gear $z_{1\mathrm{I}}$ = 21(selected value),
- Number of teeth of the second sun gear $z_{1\mathrm{II}}$ = 21 (selected value),
- Number of planets per PGT k = 3 (application demand),
- Gear material 16MnCr4 steel (application demand),
- Average value of planet bearing losses coefficient k_b = 0,065 [3,5],

- Average value of seal frictional losses coefficient k_s = 0,05 [3,5],
- Average value of churning losses coefficient k_c = 0,125 [3,5],
- Gear width to diameter ratio b / d_1 = 0,7 (selected value),
- Efficiency $\eta \geq$ 0,93 (application demand).

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.

The analysis module finds 16 variants of transmissions which require demands, however due to isomorphy there are only 10 different variants. The schematic review of these variants is shown in Figure 3.

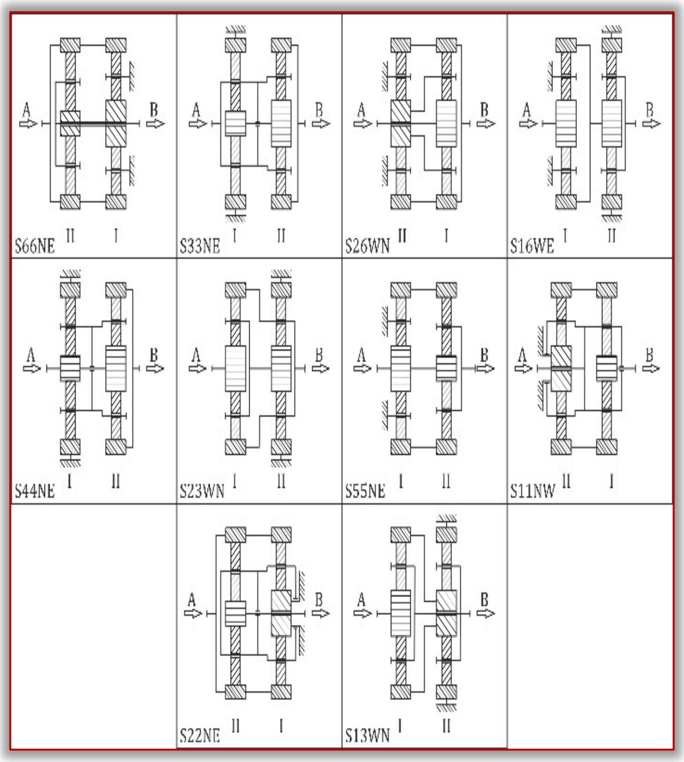


Figure 3. Single speed transmission variants

Analysis of the program results shows that the optimal solution in accordance with the criteria of maximum efficiency provides a borderline increase in efficiency in relation to the solution for minimal 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. The indicator of approximately equal outside diameters is the relation of maximal and minimal values of outside diameters of the two component gear trains which has to be close to unity.

The research results are condensed in Table 2 taking into consideration the following criteria: housing shape, efficiency and maximum transmission diameter.

Table 2. Research results in Example 1

Variant	t_I	t_{II}	d_{max}/d_{min}	d_{max}	η	m_I	m_{II}
S66NE	4	3,83	1,043	216	0,52	3	3
S55NE	3,17	3,67	1,081	231	0,77	3,75	3,5
S26WN	6,5	3	1,020	234	0,97	2	4,25
S16WE	6,83	3,33	1,025	246	0,97	2	4
S44NE	3,67	3,17	1,022	247,5	0,77	3,75	4,25
S23WE	7,33	3,5	1,014	267,7	0,97	2	4,25
S33NE	5,17	5	1,033	279	0,50	3	3
S13WN	7,83	3,83	1,022	282	0,97	2	4
S22NE	4	4,17	1,008	378	0,52	5,25	5
S11NW	5	4,83	1,017	435	0,50	4,75	5

The variant S13WN shown in Figure 4 is chosen as the optimal transmission, even though it has internal power circulation. However, it is insignificant [21], and the internal power flow may be seen in Figure 5. The reason for such a decision is the fact that this variant is easy for manufacturing and it has been already studied in literature [22].

This gear train layout is composed of component gear trains I and II. Input A is connected to sun gear I, while planet carriers I and II are connected to a common shaft leading 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.

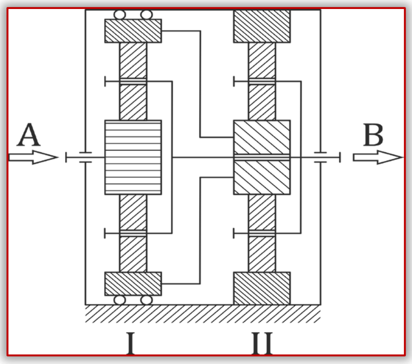


Figure 4. Schematic overview of the two– carrier, single speed planetary gear train S13WN

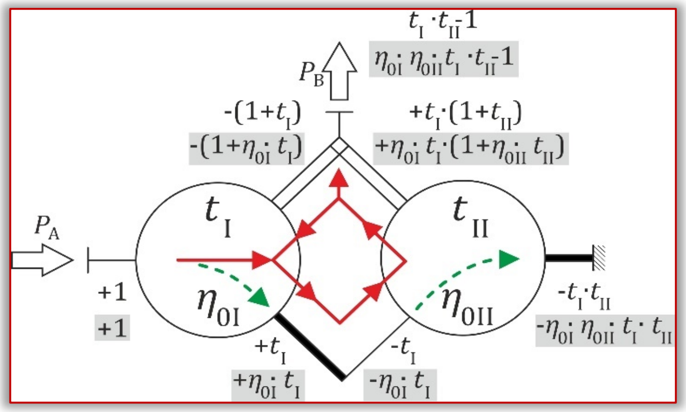


Figure 5. Torque method PGT analysis – ideal and real specific torques on all PGT shafts.

The analysis module finds possible solutions or combinations of ideal torque ratios for layout variants which provide the required transmission ratio. This data enables the creation of graphic representations of dependencies of different parameters.

The first graph is shown in Figure 6. A considerable connection of values at x– and y– axes can be seen to exist. This makes sense since it is about maximum of ring gear diameter and relation of maximum and minimum of ring diameter: an increase at the y– axis implicates an increase at the x– axis, and the points (x, y) form a cloud in the x– y plane.

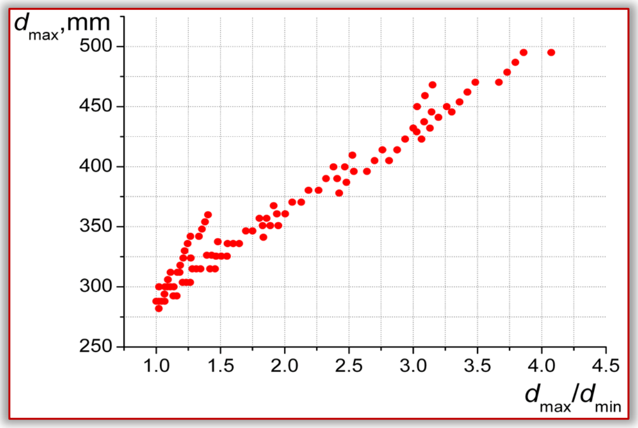


Figure 6. Correlation between maximal diameter of transmission and relation of ring gear reference diameters

The other dependencies from the analysis module are given in Figs. 7, 8 and 9.

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 Figure 7 presents the size ratio of the larger and smaller PGT ring gear diameters. 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 one will have minimal radial dimensions.

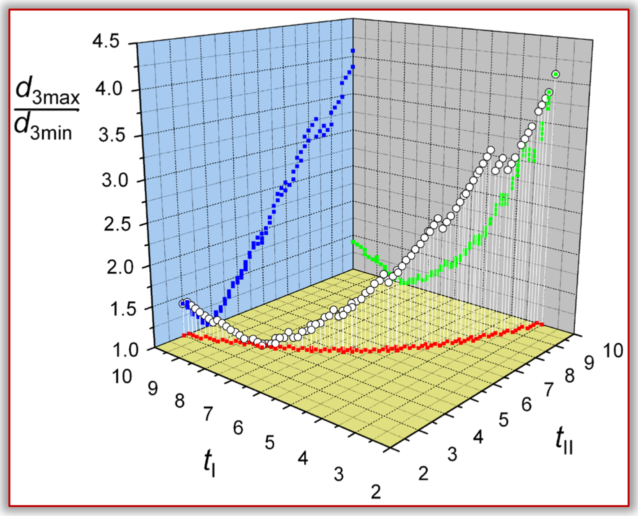


Figure 7. The influence of ideal torque ratios on the size ratio of of the larger and smaller ring gears diameters

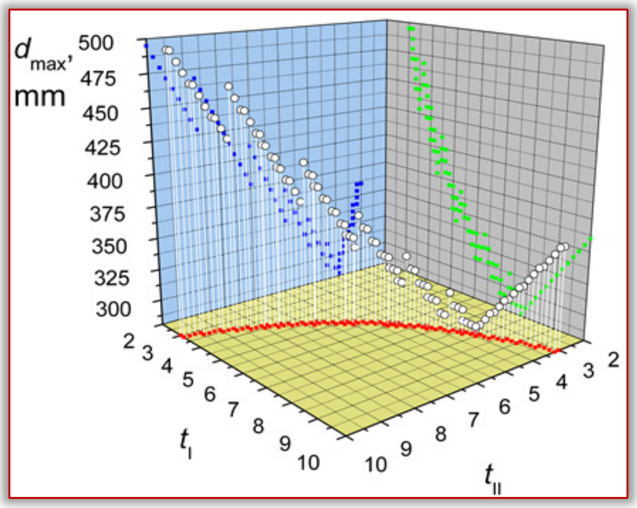


Figure 8. The influence of ideal torque ratios on maximal diameter of transmission in Example 1

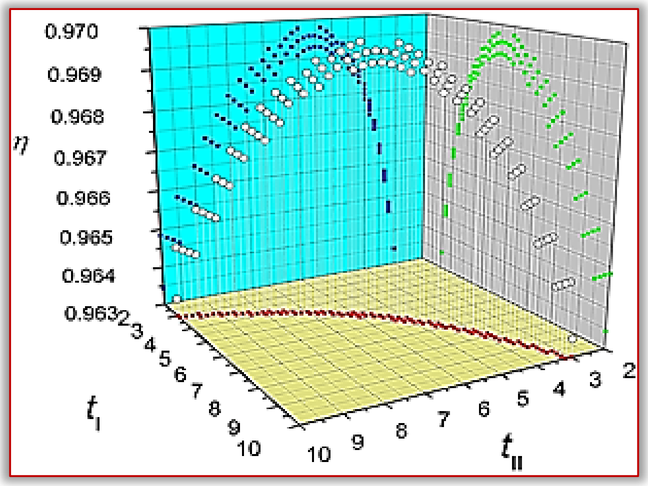


Figure 9. The influence of ideal torque ratios on overall efficiency

Every point in the x– y plane of Figure 8 also presents a pair of ideal torque ratios, while the z – axis presents the maximal diameter of the transmission.

The z – axis in Figure 9 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 0,969 can be achieved.

— Example 2

In this example of application of the DVOBRZ software, only the overall transmission ratio is different, i = – 40. The other input data is the same as in Example 1. The analysis module has also found 16 variants of transmissions which are in accordance with application demands. Due to isomorphy, there are only 10 different variants which is identical to Example 1.

The research results were processed in the same way as in Example 1 and are condensed in Table 3, taking into consideration all the criteria: housing shape, efficiency and maximum transmission diameter.

Table 3. Research results in Example 2

Variant	t_I	t_{II}	d_{max}/d_{min}	d_{max}	η	m_I	m_{II}
S26WN	7,67	3,67	1,016	280,5	0,97	2	4,25
S16WE	7,83	4	1,021	288	0,97	2	4
S66NE	5,83	5,67	1,03	288,7	0,49	2,75	2,75
S55NE	3,83	4,33	1,06	292,5	0,97	4	3,75
S44NE	4,33	3,83	1,064	312	0,76	4	4,25
S23WE	8,83	4	1,039	318	0,97	2	4,25
S13WN	8,67	4,67	1,077	336	0,50	2	4
S33NE	6,83	6,67	1,025	338,2	0,97	2,75	2,75
S22NE	5,67	5,83	1,029	551,2	0,52	5,25	5,25
S33NE	6,67	6,83	1,025	615	0,47	5	5

The variant S13WN was chosen in Example 2, for the same reasons as in Example 1.

There are 71 combinations of ideal torque ratios combination which satisfy the demands for *i* =– 40, and they enable different dependencies between design parameters to be shown in Figs. 10, 11, 12, and 13.

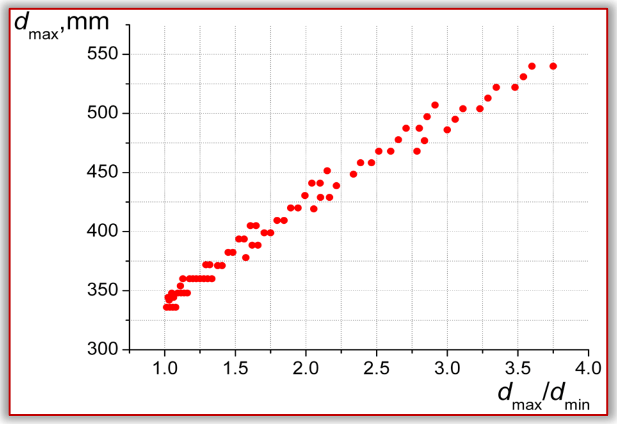


Figure 10. Correlation between maximal diameter of transmission and relation of ring gear reference diameters

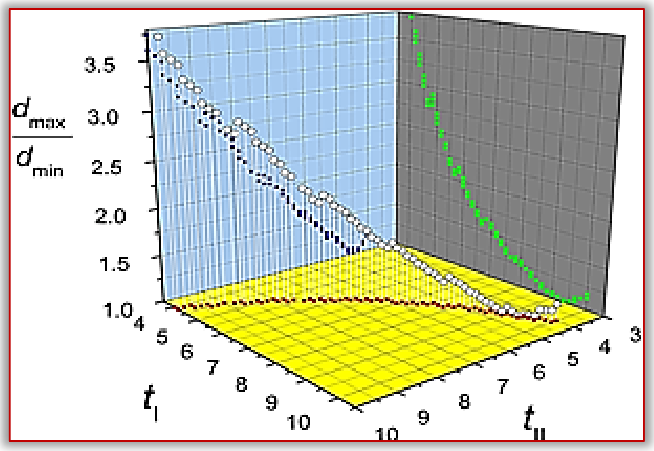


Figure 11. The influence of ideal torque ratios on the size ratio of of the larger and smaller ring gears diameters

First, the relation of the maximum transmission diameter and maximum to minimum ring gear diameter ratio is given in Figure 10. The graph resembles Figure 6, as the increase on y– axis implicates an increase on x– axis, and the points (x, y) form a cloud in the x– y plane. The difference is in the values at the x– axis and y– axis. The maximum value on the y– axis is higher than the value in Figure 6, and the value at the x– axis is smaller than the value in Figure 6.

Figure 11 presents the size ratio of the larger and smaller PGT ring gear at the z– axis, while the horizontal x– y plane presents a pair of ideal torque ratios of the component gear trains. The chart shows that this ratio can variate between 1 and 4, and that the ideal torque ratios are in the range from 3 to 10.

Further analysis of the results has shown that PGTs with z – axis values equal or close to one will have minimal radial dimensions.

In Figure 12, every point in the x– y plane presents a pair of ideal torque ratios (range from 3 to 10), while the z– axis presents the maximal diameter of transmission. The graph has a shape similar to the graph in Figure 8, with slightly larger values at the z– axis.

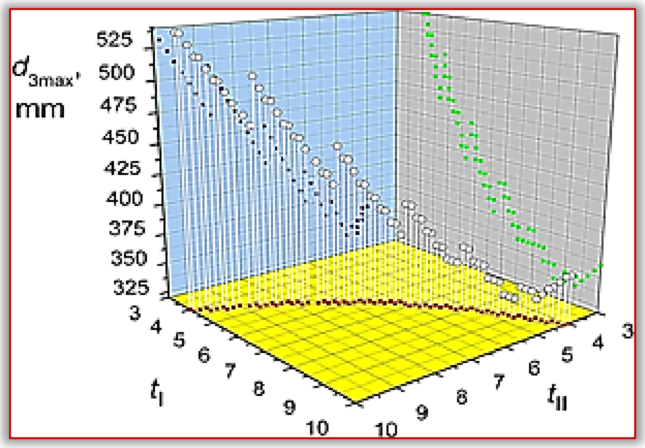


Figure 12. The influence of ideal torque ratios on maximal diameter of transmission in the example 2

Also, the ideal torque ratios are in relation with the overall efficiency of the PGT which is presented in Figure 13. The z – axis in Figure 13 is used to represent efficiency, while the x– axis and y– axis are used to represent the horizontal plane. The chart shows that for torque ratios in the 3 to 10 range, efficiencies ranging up to 0,972 can be achieved, which is a slightly larger value in comparision with the value in Example 1.

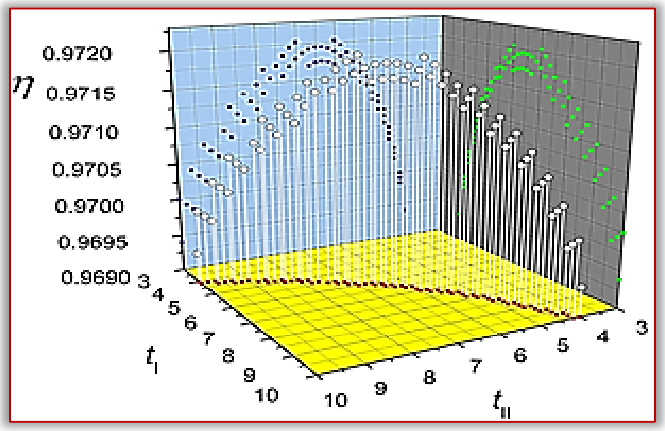


Figure 13. The influence of ideal torque ratios on overall efficiency

It is possible to determine the relation of the transmission efficiency to the overall transmission ratio, as visible in Figure 14. It has been determined that the efficiency increases with the transmission ratio for this PGT type. Furthermore, it can be concluded from the diagram in Figure 14 that PGTs with transmission ratios in the interval from – 30 to – 40 interval have very small variations in efficiency. The efficiency variations at transmission ratio of – 40 are less pronounced than at transmission ratio of – 30. This indicates that there is a very small window for PGT maximum efficiency optimization.

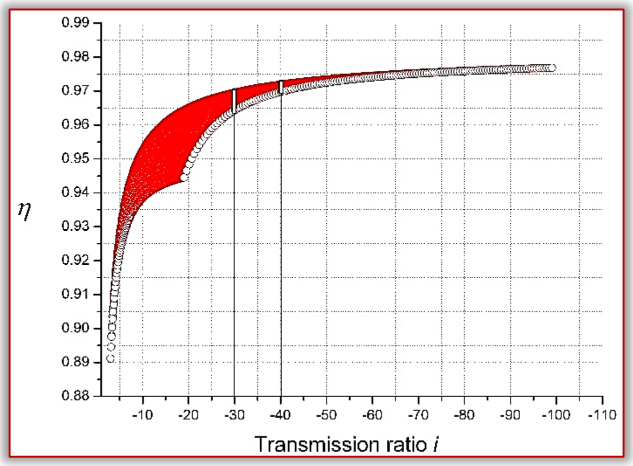


Figure 14. Influence of the overall transmission ratio on efficiency

Also, there is a slight increase in maximal diameter of the transmission at transmission ratio of – 40 in comparison to the maximal diameter of transmission at transmission ratio of – 30. The curves change in the same way in the both examples.

Based on the analysis provided in the paper it can be concluded that more two variants provide necessary conditions, but detailed analyse is required. It is about variants S26WN and S16WE, shown in Figure 15.

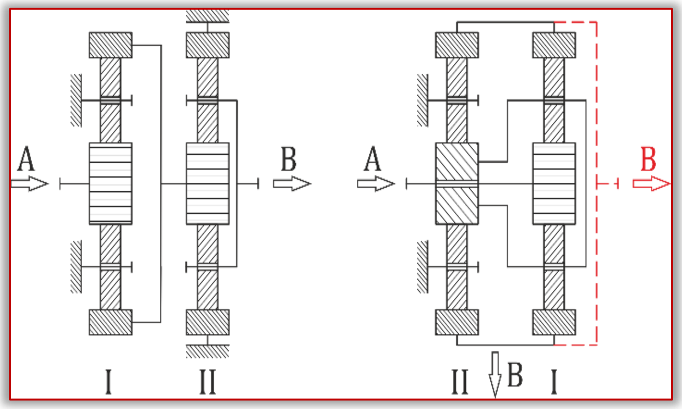


Figure 15. Schematic overview of the two– carrier, single speed planetary gear trains S16WE and S26WN (the original output located as in Figure 3 has been added in dashed red lines for clarity)

Variant S16WE having the input and output shafts on opposite sides, while 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 efficiency [23].

By comparing the parameters of variant S13WN and the other possible solutions, S26WN and S16WE it can be seen that solutions S26WN and S16WE have better values of maximal transmission diameters, while the efficiency remains in the same range of values. The relations of ring gear reference diameters are equal for the transmission ratio of – 30, while solutions S26WN and S16WE have values closer to one for the transmission ratio of – 40, providing a better solution than S13WN. However, S13WN remains the better solution if the selection criteria are limited to technological demands.

CONCLUSIONS

This paper deals with the analysis of a two– carrier PGT developed for a specific application, having a transmission ratio of – 30 and – 40. The application conditions request demand the type S13WN PGT to be used. The DVOBRZ software package was used to determine the variants and basic parameters of two– carrier drives within the application constraints, 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 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 according to two criteria. The first criterion was minimization of the PGT dimensions, while the second was maximization of PGTefficiency.

Analysis has shown that the 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 both S16WE and S26WN provide valid solutions according to both the criteria of minimum size and maximum efficiency. It must be also considered that those variants have no internal power circulation, resulting in a considerably lighter build in relation to S13WN. However, variant S13WN still retains precedence due to technological demands and the fact that this variant is thoroughly examined in literature.

Acknowledgment

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