J. Appl. Comput. Mech., xx(x) (2024) 1-12
DOI: 10.22055/jacm.2024.46280.4490

ISSN: 2383-4536
jacm.scu.ac.ir

Shahid Chamran
University of Ahvaz

Journal of
Applied and Computational Mechanics

Research Paper

Selection of Optimal Planetary Transmission for Light Electric Vehicle Main Gearbox

Sanzhar Armanuly Kalmaganbetov[1], Madina Isametova[1] Sanjin Troha[2], Željko Vrcan[2], Kristina Markovic[2], Dragan Marinkovic[3,4]

[1] Department of Mechanical Engineering, Standardization, Certification and Metrology, Institute of Energy and Mechanical Engineering, Satbayev University, Satpaev Street 22, Almaty, KZ-050000, Kazakhstan, Email: sanzhar.kalmaganbetov@mail.ru (S.A.K.); isametova69@mail.ru (M.I.)

[2] Faculty of Engineering, University of Rijeka, Vukovarska 58, Rijeka, HR-51000, Croatia, Email: sanjin.troha@riteh.uniri.hr (S.T.); zeljko.vrcan@riteh.uniri.hr (Ž.V.); kristina.markovic@riteh.uniri.hr (K.M.)

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

[4] Department of Mechanical Engineering, Berlin Institute of Technology (TU Berlin), Strasse des 17. Juni 135, Berlin, D-10623, Germany, Email: dragan.marinkovic@tu-berlin.de

Received March 04 2024; Revised April 12 2024; Accepted for publication April 17 2024.
Corresponding author: K. Markovic (kristina.markovic@riteh.uniri.hr)

Abstract. This paper deals with the selection of optimal layout variants of planetary transmissions designed for operation as Light Electric Vehicle main gearboxes. The required transmission ratio was calculated, and the optimal two-carrier gear train with two connecting and three external shafts selected. A computer program was used for structural synthesis and to determine the required basic parameters of the component gear trains. The ring gear reference diameter was used to rank the layout variants according to size, with the smallest variants also ranked according to the calculated efficiency ratio. Finally, an optimal variant of the planetary gear train for light electric vehicle application is proposed. The procedure proposed in this paper may be expanded to other applications.

Keywords: Planetary gear train, two-carrier planetary gear train, main gearbox, light electric vehicle, gear train optimization.

1. Introduction

Recent increases in air pollution and restricted oil availability have resulted in rapid advances in electric vehicle (EV) technology [1], especially in plug-in electric vehicles (PEVs). Such vehicles have a propulsion system integrating an electrical and a mechanical subsystem. The electrical subsystem uses battery power to supply one or more electric motors by means of a power electronics-based motor controller. The mechanical subsystem must have a single or multi-speed reduction gearbox, while a clutch and differential gear [2] may also be included.

The electric motor of the EV is used to convert electricity into mechanical energy. The torque from the electric motor shaft is transmitted through the mechanical transmission to the wheels of the vehicle, allowing the vehicle forward or backward movement [3]. The transmissions commonly used by EVs are single speed, although multi-speed transmissions may be applied to increase the efficiency of the powertrain and reduce energy consumption, thus extending the vehicle's driving range [4]. The installation of multi-speed gearboxes in electric vehicles has been the subject of recent research [5-9], as these gearboxes are considered very promising when dealing with the demands of wider torque capabilities and improved energy efficiency, surpassing single-speed gearboxes [10, 11].

The application of planetary gear trains (PGTs) is common in power-efficient electric vehicles. Planetary gear trains offer a highly compact design and have an improved torque-to-mass ratio when compared to conventional gear trains. Furthermore, their higher efficiency in comparison to conventional gear trains makes them increasingly popular in the automotive industry [12-16].

PGTs offer numerous advantages over conventional gear trains, such as [17-19]:

- coaxial input and output shafts,
- gears of reduced size and mass due to the distribution of power to multiple meshings of the planet gears with the sun and ring gears,
- compact cylindrical or truncated cone shaped casings,
- a wide range of transmission ratios in a relatively small and compact gearbox etc.

Published online: April 27 2024

Table 1. Electric vehicle design constraints.

Number of seats	1
Vehicle mass	200 kg
Autonomy	100 km
Top speed	45 km/h
Slope climbing ability	20%
Wheelbase	1750 mm
Track width	1200 mm
Wheel size	R12

Table 2. Electric motor technical characteristics.

Nominal voltage	72 V
Rated power	10 kW
Rated torque	50 Nm
Nominal shaft speed	2000 min^{-1}
Maximum shaft speed	4000 min^{-1}

PGTs also have some disadvantages when compared to standard gear trains [17-19]:

- complex design and manufacture,
- significant centrifugal load on the planet gears and their bearings,
- difficult inspection and maintenance due to their compact design,
- lower lubricant capacity requiring more frequent replacement etc.

A solution for electric vehicle propulsion consisting of an electric motor and a single-speed planetary gear train is proposed in this paper. The key criteria that will be used in the selection of the optimal kinematic scheme of the PGT and its parameters will be the minimal possible volume of the gearbox and the highest possible efficiency. Minimal volume will result in minimized gear train mass, a lighter construction of the entire EV and reduced power consumption. The torque method [19-23] was used for calculations performed in this paper. This method is implemented into the dedicated software that was used for synthesis and selection of the optimal layout variant of the gearbox.

2. Research Methodology

The research presented in this paper aims to propose an optimal planetary gear train solution for an existing electric vehicle (Fig. 1), based on the characteristics of the vehicle itself and its electric motor. The torque method was used in the synthesis and selection of the optimal kinematic scheme and its parameters. The torques on the shafts of the component planetary gear trains were determined for each structural scheme and used to estimate the size of the gearbox and its efficiency [24, 15]. It is necessary for the solution to meet certain requirements, such as reduced vehicle weight and improved transmission efficiency while maintaining the existing autonomy of the vehicle and considering the required performance criteria and physical limitations [25] which may be seen in Table 1.

The vehicle has an electric motor of 10 kW rated power (Table 2) that operates at the nominal voltage of 72 V DC. As it is common for DC vehicle propulsion motors to have relatively high shaft rotational speeds to reduce the motor size and weight, it is necessary to use a reduction gearbox to match the motor speed with the wheel speed [26-28].

To begin with, the characteristics of the gearbox must be defined, such as the output torque, output shaft rotational speed and the transmission ratios. To determine the overall transmission ratio, expression (1) was adapted from the expression for the maximum speed that electric vehicles can develop on level ground [1]:

$$i_o = \frac{\pi n_{max} r_t}{30 \upsilon_{max}} \tag{1}$$

where n_{max} = 4000 min^{-1} is the electric motor rotational top speed, r_t = 254 mm is the vehicle wheel radius, υ_{max} = 12,5 m/s is the vehicle top speed. This resulted in an overall transmission ratio of i_0 = 10,6.

It is common for electric vehicles of this type to have a chain transmission because of the well-known characteristics of chain drives – their simplicity, and ease of maintenance and replacement. However, the main drawback of this type of vehicle is that the chain remains exposed to the elements and dirt, and, in vehicles with suspension systems, the chain might even get thrown off the sprockets when travelling at speed over rough terrain. Furthermore, in cases requiring large transmission ratios a two-stage chain and sprocket system must be used. The basic mechanical layout of such a vehicle, as previously shown in (Fig. 1), may be observed in (Fig. 2, left). On the other hand, the introduction of planetary gear trains into consideration enables the design of a smaller and more compact transmission, with the motor and gear train unit combined into a single casing with the rear axle (Fig. 2, right), enabling easier design of vehicles using suspension systems.

Fig. 1. Light electric vehicle.

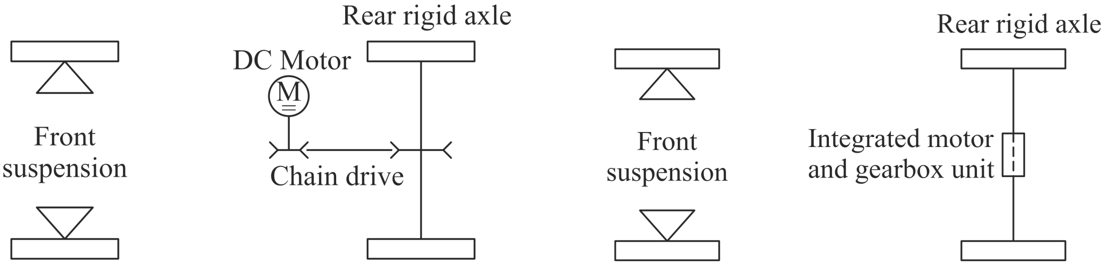

Fig. 2. Mechanical layout of a chain-driven (left) and of a compact drive (right) light electric vehicle.

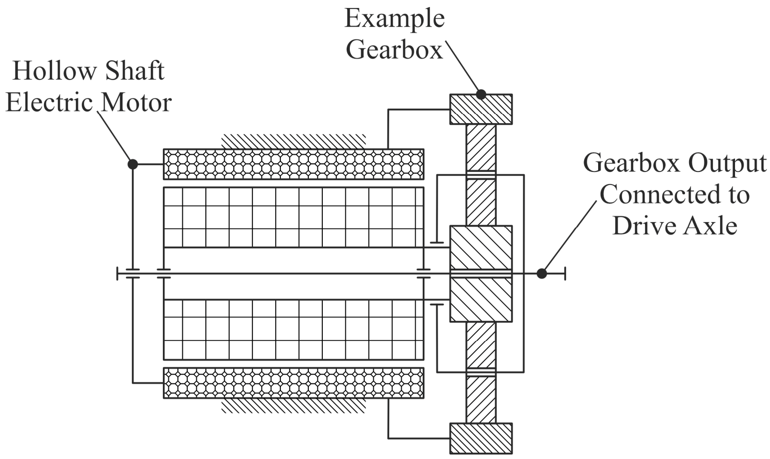

Fig. 3. Schematic layout of a hollow shaft motor and gearbox unit.

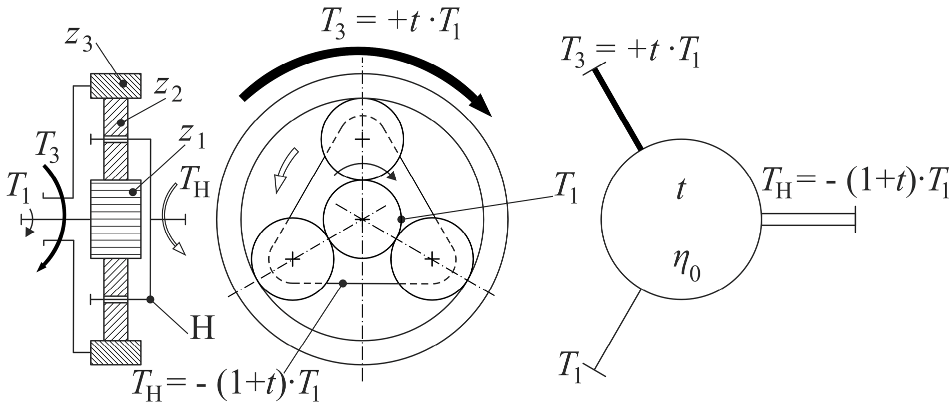

Fig. 4. Cross-section (left) and frontal view (center) of simple component gear train with Wolf-Arnaudov symbol and specific torques acting on the shafts (right) [18].

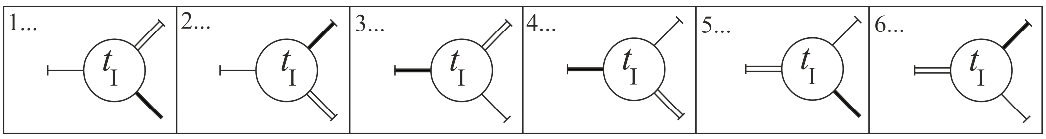

Fig. 5. Symbolic presentation of component planetary gear train connectivity options.

With integrated motor and gearbox units, the motor and gear train may be placed parallel to the axle, or, for the most compact design, the motor and gear train may be designed with hollow shafts and the rear axle passing through them (Fig. 3).

The planetary gear train discussed in this paper is a two-carrier PGT composed of two simple 2K-H, variant A planetary gear trains (Fig. 4). Due to its characteristics, this gearbox will be smaller than the equivalent conventional gearbox [10].

The torque method provides a convenient tool for kinematic analysis and synthesis of gearboxes and for the calculation of the efficiency of complex planetary gear trains [24].

Each component gear train within a two-carrier (complex) planetary gear train provides six different connectivity options, depending on the shaft layout (Fig. 5).

A total of 36 PGT schemes may be created by interconnecting two component gear trains in this manner. However, due to isomorphy, the actual number of schemes is reduced to 21 [18]. An overview of these schemes is given in Table 3.

Table 3. Schematic overview of two-carrier PGTs [23].

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

Each complex PGT shown in Table 3 can be operated in six different ways with one degree of freedom [18] by locking one external shaft while the other two are used as input or output. This yields 126 different layout variants, each of which has different geometric or mechanical characteristics [29, 30]. Selection of the appropriate scheme from 126 possible and the choice of the optimal basic structural parameters is not trivial, and requires appropriate methodology complemented by a dedicated algorithm.

It is very important to explain the nomenclature used to name the layout variants in this paper. All gear trains that are the object of the research performed in this paper have three external and two connecting shafts. The first external shaft is used to connect the prime mover, the second external shaft is used to connect the powered machine, while the third external shaft is permanently locked stationary (connected to the gearbox casing) as the reactive member. Cardinal points may be applied to the gear train symbols from Table 3. In this manner, the three external shafts correspond to the western (W), northern (N) and eastern (E) cardinal point, while the southern cardinal point (S) corresponds to an internal connecting shaft and is not used in the nomenclature (Fig. 6). The western and eastern points correspond to single shafts, while the northern cardinal point corresponds to the external connecting shaft.

Assuming that one shaft is always permanently connected to the gearbox casing, there are six different ways in which power can be transferred through a gearbox, e.g. if the external connecting shaft (N) acts as the power input, and the external single shaft (W) acts as the power output, the remaining single shaft (E) must be permanently locked stationary by a connection to the gearbox casing. In this case, the power flow would be encoded as NW(E). The first letter represents the cardinal direction of the power input, the second letter represents the direction of the power output, and the locked shaft is represented by the letter in the brackets. This nomenclature may be used to list all six modes of operation (layout variants), i.e. WE(N), WN(E), NW(E), NE(W), EW(N) and EN(W).

Every layout variant may be used on any of 21 different schemes, resulting in a total of 6 x 21 = 126 structurally different gear trains, and it must be noted that several design solutions may be applied to achieve a single structural solution [23]. The reader should keep in mind that the cardinal points do not correspond to the locations of the shaft on the actual gearbox, as the actual gearbox will always be set up to have the input and output shaft at opposing ends. For example, any of the six layout variants may be applied to scheme S33, and if the WE(N) layout variant is selected, the gear train will be labelled as S33WE(N).

The 2-BRZ software was developed to facilitate the selection of the appropriate scheme and layout variant, combined with its optimal design parameters (number of gear teeth of both component gearsets, gear modules, etc.) [21, 31]. The software requires three subsystems to operate: a subsystem for the analysis of the variant characteristics, a subsystem for the synthesis of all solutions meeting the set requirements, and a solution evaluation subsystem. The program operates by systematically determining the ideal torque ratios for all possible schemes that provide the required transmission ratio. For each potential combination of ideal torque ratios, the program calculates the total gear ratio of the gearbox, and then checks whether the calculated values are within the required interval. Furthermore, the program determines the reference diameters of the component gear train ring gears to check the sizing, while also calculating the theoretical efficiency ratios. Even though all structural details are not known at this early design stage, this calculation is highly useful for the comparative analysis of possible conceptual solutions. This data is then used by the synthesis and solution evaluation subsystems to generate a priority list of two-carrier single-speed gear train solutions.

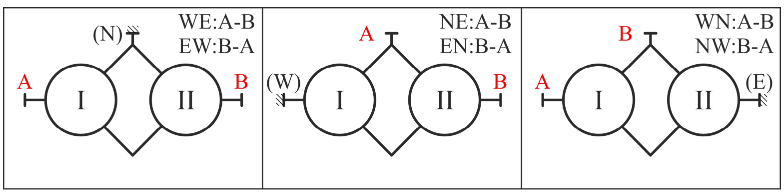

Fig. 6. Schematic overview of layout variant nomenclature according to cardinal points. The red letters A and B are used to denote the external shafts that are used to transfer power to or from the geartrain. The actual layout variant depends on the direction of the power flow being from A to B or from B to A.

2.1. Determination of the Optimal Ideal Torque Ratios

The software module with its integrated logic that enables the identification of variants with parameters that can achieve the required kinematic transmission ratios was developed based on the torque method described in [22,31].

The module checks all layout variants of the gear train in order. For each variant, a set of ordered pairs of ideal torque ratios is generated, which allows the realization of the required gear ratio. If a layout variant is unable to achieve the required gear ratios, an empty set is returned.

The limits of the interval in which the module searches for the ideal torque ratios t_I and t_{II} are selected by the user. The actual limits of the interval are determined by design constraints. Specifically, in the case of planetary gear trains with three planets per component gearset, they are $2 \le t_{Imin} \le t_I \le t_{Imax} \le 12$ and $2 \le t_{IImin} \le t_{II} \le t_{IImax} \le 12$ [18,22].

The transmission ratio is obtained for each possible combination of ideal torque ratios, and checked whether the value is within the tolerance interval of the required transmission ratio. As a more accurate efficiency calculation will be performed further downstream in the program, the step in which the ideal torque ratios are increased is determined regarding the number of gear teeth.

The following procedure is carried out in the computer program to determine the ideal torque ratios t_I and t_{II}:

- The required numbers of sun gear teeth of both component gear trains z_{1I} and z_{1II} are entered into the program,
- The minimum and maximum numbers of gear teeth of the ring gears are calculated,
  $z_{3Imin} = -\, t_{Imin}\, z_{1I}$,
  $z_{3Imax} = -\, t_{Imax}\, z_{1I}$,
  $z_{3IImin} = -\, t_{IImin}\, z_{1II}$,
  $z_{3IImax} = -\, t_{IImax}\, z_{1II}$.
- The current values of the ideal torque ratios $t_I = |z_{3I}| / z_{1I}$ and $t_{II} = |z_{3II}| / z_{1II}$ are determined, starting with z_{3Imin} and z_{3IImin},
- The number of teeth of one ring gear (usually the first) is increased by one, and the synthesis conditions of the gearset are checked [30]. If the condition of assembly is not met by this tooth number combination, the ideal torque ratio is discarded, the ring gear is increased by one more tooth, and the check is repeated. The user has an option to avoid the synthesis check, as some gear trains may be assembled with appropriate structural modifications even if the conditions of assembly are not met [32]. This procedure is repeated until the number of ring gear teeth reaches the maximum permissible value z_{3Imax},
- The number of gear teeth of the other ring gear is increased by one, and the procedure is repeated as for the first gearset until the maximum permissible value z_{3IImax} is reached. All conditions of planetary gearset synthesis also apply here, and the assembly check may be omitted if required.

The two program loops calculate the current values of several component planetary gearsets (η_{0I}, η_{0II}, d_{3I}, d_{3II}, m_I, m_{II}, n_{2rsI}, n_{2rsII} etc.) and the whole gear train (i_{Br1}, i_{Br2}, η_{Br1}, η_{Br2}, d_{3max}, d_{3max} / d_{3min} etc.) for each gear train variant as a function of the ideal torque ratios t_I and t_{II}.

The database created in this manner can be used in the analysis of the parameters of the gear train, and it may be used to generate graphical representations of these parameters as a function of the ideal torque ratios t_I and t_{II}, enabling the designer to quickly select the most appropriate variant of the gear train. It can also be used to determine which values of the ideal torque ratios t_I and t_{II} will provide the desired overall transmission ratio. In cases of multiple valid solutions, it is possible to determine the combination of t_I and t_{II} which fulfils one or more special optimisation requirements, such as minimal dimensions, minimal gear train mass, maximum efficiency ratio etc.

When dealing with single speed gear trains, each applicable optimisation parameter reaches its optimal value in the general case of a different combination of t_I and t_{II}. Selecting a layout variant by a single optimisation parameter gives a solution that is invalid from the perspective of multi-criteria optimisation. For example, optimisation for efficiency requires a maximum, while optimisation for dimension and mass requires a minimum. Therefore, multi-criteria optimisation is always a compromise solution as improving the solution by one criterion deteriorates the solution by one or more other criteria.

2.2. Selection of the Most Suitable Layout Variant

In case of single-criteria optimisation, the most suitable layout variant is selected in multiple stages. For each variant, the optimal value of the selected criterion and the corresponding combination of t_I and t_{II} is determined. The values of the optimal solutions according to each criterion are compared, and the most appropriate variant of the gear train is selected with its corresponding combination of t_I and t_{II}.

3. Results

After the input data was entered, the software system successfully sorted out the layout variants that met the specified criteria for the required overall transmission ratio of $i = \pm 10{,}6$.

When searching for the transmission ratio of $i = 10{,}6$, the number of non-isomorphic (different) variants is nine, while for the reverse transmission ratio of $i = -10{,}6$, the number of variants is ten. An overview of the valid kinematic schemes with their scheme and layout designation will be discussed later. The software was limited to layouts using three planet gears per component geartrain as this offers the broadest possible range of ideal torque ratios as well as the best possible load distribution between the meshings of planet gears with the sun and ring gears. Otherwise, the number of planets may be any valid positive integer.

Visual analysis of the kinematic schematics has revealed several valid solutions, as their input and output shafts are situated at the opposite ends of the gearbox. Therefore, quantitative analysis of the solution quality will be required for the selection of the optimal solution.

3.1. Planetary Gear Trains with Positive Overall Transmission Ratios

When dealing with gear trains having a positive transmission ratio according to the requirements, it was observed that the best solutions in the form of S16WN(E), S26WE(N), S23WN(E) and S13WE(N) have a calculated overall efficiency of about 0,96, while the others (S55NE(W), S66NE(W), S44NE(W), S11NE(W) and S22NE(W)) have significantly lower efficiency ratios.

For example, an efficiency scatter chart is shown in Fig. 7 for the S16WN(E) gearbox for the interval in which t_I, t_{II} = 1,5…6. It is possible to observe from the chart that with higher overall transmission ratios, there is less fluctuation of the efficiency ratio. the degree of utilization fluctuates less. For the gear ratio of $i = +10{,}6$, the efficiency can change to any value along the red line, depending on the corresponding combination of ideal torque ratios.

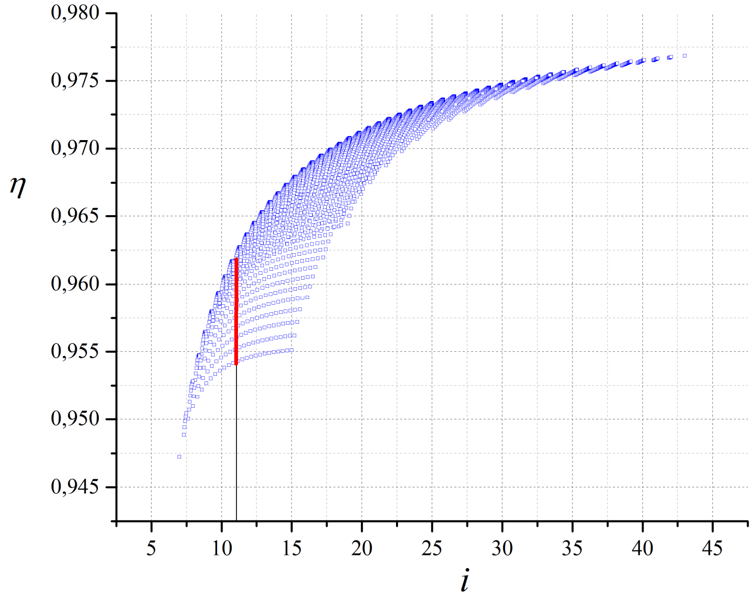

Fig. 7. Scatter chart showing the possible values of the efficiency ratio as a function of the overall transmission ratio i for the S16WN(E) PGT in the interval t_I = 1,5…6 and t_{II} = 1.5…6.

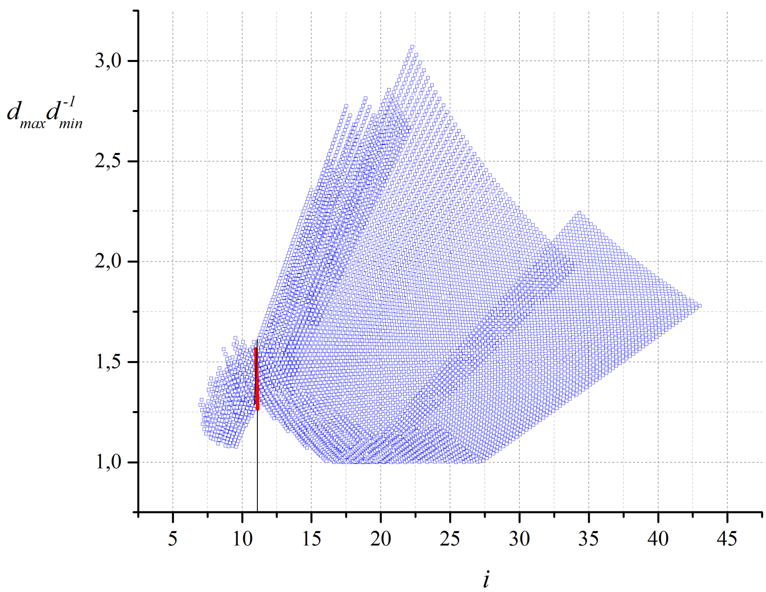

Fig. 8. Diagram of the ratio of the larger and smaller ring gear reference diameter as a function of the transmission ratio for the S16WN(E) PGT in the interval t_I = 1,5…5 and t_{II} = 1.5…5.

A diagram of the ratio of the larger and smaller ring gear reference diameter as a function of the overall transmission ratio for planetary gear train S16WN(E) in the interval t_I = 1,5…6 and t_{II} = 1.5…6 is provided in Fig. 8. From the results it may be observed that there are fewer fluctuations of the ring gear reference diameters with smaller transmission ratios. In case of the gear ratio i = 10,6, the ratio of the ring gear reference diameters may change depending on the corresponding combination of ideal torque ratios, as in the interval marked by the red line (1,25 to 1,6).

It is known that [33, 34] as the ratio of the ring gear reference diameters approaches unity, the greater ring gear diameter approaches its minimal value, and therefore the gear train reaches its minimal volume.

The calculations performed for the S16WN(E) gear train have resulted in five variants with two different combinations of ideal torque ratios ratios t_I and t_{II}. The data is listed in Table 4 and displayed in charts in Figs. 9 and 10.

Table 4. Five variants of ideal torque ratio combinations with the associated characteristics of the resulting planetary gear train S16WN(E).

Var.	t_I	t_{II}	i	d_{3I}, mm	d_{3II}, mm	d_{max}/d_{min}	η	z_{3I}	z_{3II}
1	2	3,8	10,6	87,5	130,625	1,49286	0,954	50	95
2	2,36	3,08	10,6288	81,125	115,5	1,42373	0,960	59	77
3	2,96	2,24	10,5904	83,25	126	1,51351	0,961	74	56
4	3,08	2,12	10,6096	86,625	119,25	1,37662	0,960	77	53
5	3,2	2	10,6	90	125	1,38889	0,959	80	50

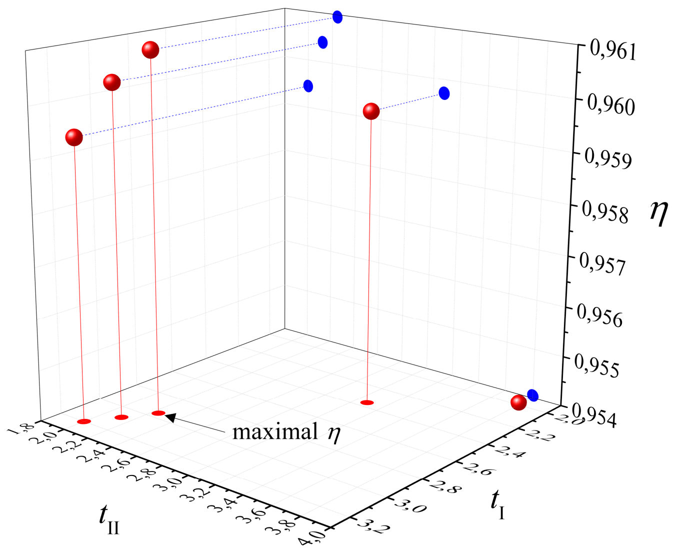

Fig. 9. Influence of numerically determined ideal torque ratios on the efficiency ratio of the S16WN(E) gear train.

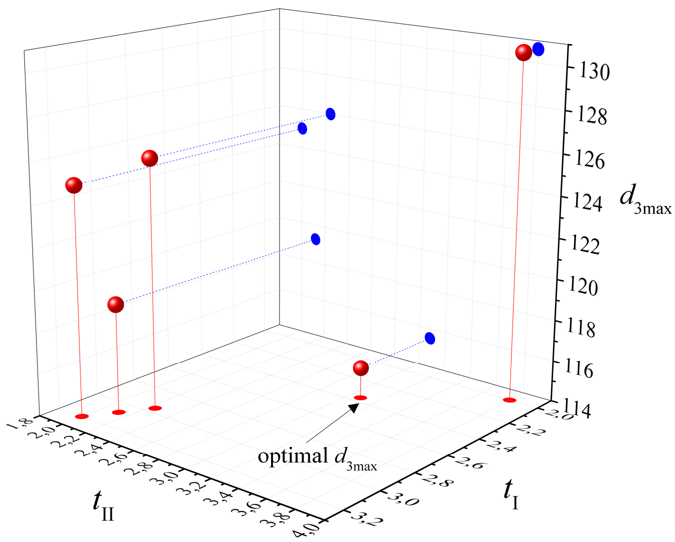

Fig. 10. Influence of numerically determined ideal torque ratios on the maximum ring gear reference diameter of the S16WN(E) gear train.

Based on the data from Table 4 and displayed in Figs. 9 and 10, it can be observed that the minimal size of the larger ring gear reference diameter d_{3IImin} = 115,5 mm is achieved in Variant 2 with the combination of t_I = 2,36 and t_{II} = 3,08. In this case, the efficiency ratio is η = 0,960. The maximum efficiency ratio η_{max} = 0,961 is obtained in Variant 3 with the combination of ideal torque ratios t_I = 2,96 and t_{II} = 2,24, however in this case the larger ring gear reference diameter equals d_{3II} = 126 mm. At this point it must be pointed out that any ring gear might be the larger one, depending on the gear train configuration.

Variant 3 clearly achieves the highest efficiency ratio, however the overall gearbox size is larger. As the increase in efficiency in relation to Variant 2 is less than 0,001, it can be concluded that Variant 2 is the optimal solution for layout variant S16WN(E), with ideal torque ratios t_I = 2,36 and t_{II} = 3,08.

When calculating for a positive overall transmission ratio according to the criterion of minimal radial dimensions of the gear train (minimal reference diameters of the component geartrains), it was observed that the gear trains S16WN(E), S26WE(N), S23WN(E) and S13WE(N) have larger ring gear reference diameters ranging from 115,5 mm to 134,75 mm and that the most compact solution is S16WN(E).

The program was used for the synthesis of all the solutions for each layout variant. After this, the best solutions for each variant were singled out and ranked into a priority list according to the criterion of the smallest size of the larger ring gear reference diameter (Tables 5 and 6).

3.2. Applicable Variants of PGTs with Positive Overall Transmission Ratios

Conceptual (kinematic) schemes were created for each computer-generated variant. All schemes were set up to have the input and output shaft at the opposite sides of the gearbox, and coaxiality was ensured. It must be noted that some layouts require drilled shafts. The kinematic schemes of all variants with a positive overall transmission ratio are given in Fig. 11, while the respective transmission ratio functions are provided in Table 7. In addition, the first component gear train has been marked with the Roman numeral I, and the second component gear train with the Roman numeral II, as the first gear train on the power transmission path does not have to be the one closest to the input shaft. Also, the red arrows denote the power flow, with A being the power input and B being the power output.

Table 5. Gear train parameters for i = +10,6.

SchV	t_I	t_{II}	i	d_3 [mm]	d_3 max/ d_3 min	η
S16WN(E)	2,36	3,08	10,6288	115,5	1,42373	0,959
S26WE(N)	1,52	3,2	10,584	133	1,10833	0,964
S23WN(E)	2	3,2	10,6	120	1,37143	0,959
S13WE(N)	3,44	3,08	10,5952	134,75	1,39276	0,960
S55NE(W)	5	3,08	10,625	156,25	1,35281	0,931
S66NE(W)	4,76	5,36	10,6	167,5	1,12605	0,815
S44NE(W)	3,08	5	10,625	171.875	1,27551	0,931
S11NE(W)	3,8	3,44	10,55556	193,5	1,01842	0,744
S22NE(W)	5,36	4,76	10,6	301,5	1,12605	0,815

Table 6. Component gear data for i = +10,6.

SchV	z_{1I}	z_{2I}	z_{3I}	m_I [mm]	z_{1II}	z_{2II}	z_{3II}	m_{II} [mm]
S16WN(E)	25	17	59	1,38	25	26	77	1,5
S26WE(N)	50	13	76	1,75	25	27	80	1,5
S23WN(E)	25	12	50	1,75	25	27	80	1,5
S13WE(N)	25	30	86	1,125	25	26	77	1,75
S55NE(W)	25	50	125	1,25	25	26	77	1,5
S66NE(W)	25	47	119	1,25	25	54	134	1,25
S44NE(W)	25	26	77	1,75	25	50	125	1,38
S11NE(W)	25	35	95	2	25	30	86	2,25
S22NE(W)	25	54	134	2,25	25	47	119	2,25

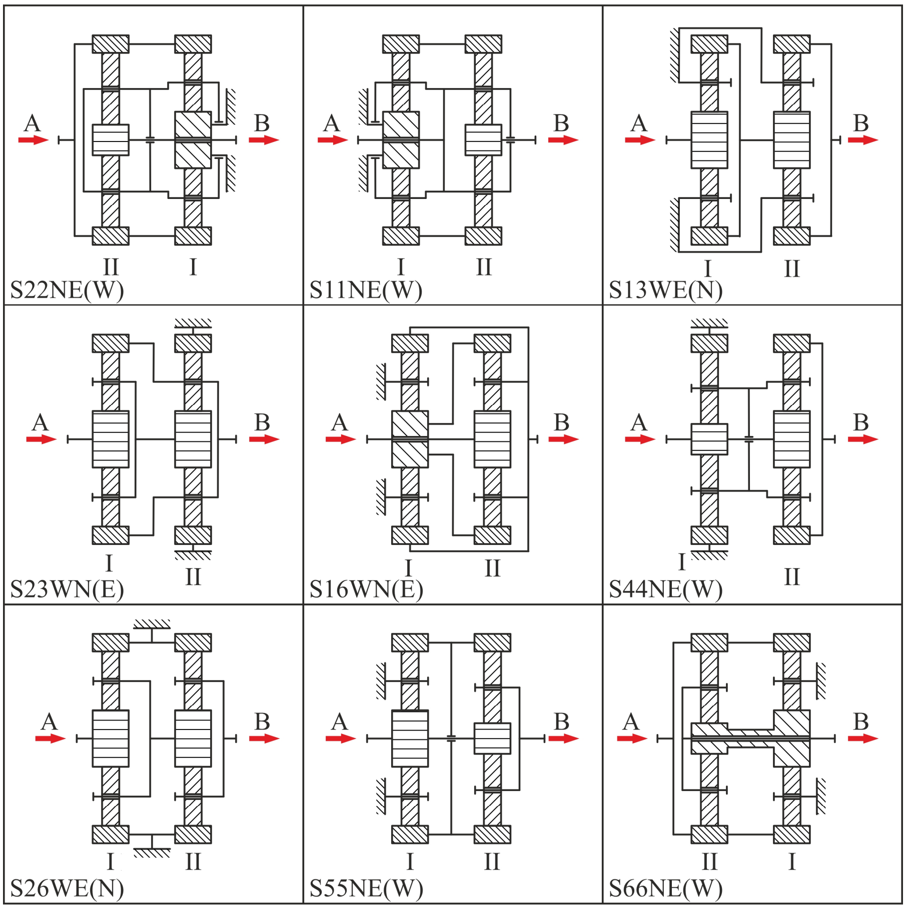

Fig. 11. Kinematic schemes of valid PGT variants with positive transmission ratios.

Table 7. Transmission ratio functions of valid PGT variants with positive transmission ratios.

SchV	i	SchV	i
S16WN(E)	$1+t_I+t_I t_{II}$	S66NE(W)	$\frac{1+t_{II}}{t_{II}-t_I}, t_I < t_{II}$
S26WE(N)	$1+t_I+t_{II}+t_I t_{II}$	S44NE(W)	$\frac{t_{II}+t_I t_{II}}{t_{II}-t_I}, t_I < t_{II}$
S23WN(E)	$1+t_{II}+t_I t_{II}$	S11NE(W)	$\frac{t_I}{t_I-t_{II}}, t_{II} < t_I$
S13WE(N)	$t_I t_{II}$	S22NE(W)	$\frac{1+t_I}{t_I-t_{II}}, t_{II} < t_I$
S55NE(W)	$\frac{t_I+t_I t_{II}}{t_I-t_{II}}, t_{II} < t_I$		

Table 8. Gear train parameters for i = -10,6.

SchV	t_I	t_{II}	i	d_3 [mm]	d_3 max/ d_3 min	η
S16WE(N)	2,6	3,08	-10.608	115,5	1,42154	0,958
S55NE(W)	2,36	3,32	-10,62	118	1,03395	0,864
S26WN(E)	2,12	2,72	-10,6064	119	1,28302	0,955
S23WE(N)	2,72	2,84	-10,5648	124,25	1,46176	0,959
S13WN(E)	3,92	2,96	-10,6032	129,5	1,1746	0,957
S33NE(W)	4,16	3,8	-10,5556	143	1,09474	0,715
S44NE(W)	3,32	2,36	-10,62	145,25	1,09416	0,864
S66NE(W)	5,96	5,36	-10,6	186,25	1,11194	0,789
S11NE(W)	3,8	4,16	-10,5556	234	1,09474	0,715

Table 9. Gear train parameters for i = -10,6.

SchV	z_{1I}	z_{2I}	z_{3I}	m_I [mm]	z_{1II}	z_{2II}	z_{3II}	m_{II} [mm]
S16WE(N)	25	20	65	1,25	25	26	77	1,5
S55NE(W)	25	17	59	2	25	29	83	1,38
S26WN(E)	25	14	53	1,75	25	21	68	1,75
S23WE(N)	25	21	68	1,25	25	23	71	1,75
S13WN(E)	25	36	98	1,125	25	24	74	1,75
S33NE(W)	25	39	104	1,38	25	35	95	1,38
S44NE(W)	25	29	83	1,75	25	17	59	2,25
S66NE(W)	25	62	149	1,25	25	54	134	1,25
S11NE(W)	25	35	95	2,25	25	39	104	2,25

3.3. Planetary Gear Trains with Negative Overall Transmission Ratios

When calculating for a negative overall transmission ratio according to the criterion of maximum efficiency, it was observed that gear trains S13WN(E), S16WE(N), S26WN(E) and S23WE(N) have an efficiency ratio slightly below 0,96, while the others have significantly lower efficiencies.

When the calculation was switched to the criterion of minimal radial dimensions of the gear train (minimal reference diameters of the component geartrains), it was observed that the gear trains S13WN(E), S16WE(N), S26WN(E), S23WN(E) and S55NE(W) have larger ring gear reference diameters ranging from 115,5 mm to 129,5 mm, meaning that the most compact solution is S16WN(E).

As in the case of PGTs with positive overall transmission ratios, all solutions were synthetized by means of the developed software program. The best solutions for each variant were singled out and ranked into a priority list according to the criterion of the smallest size of the larger ring gear reference diameter (Tables 7 and 8).

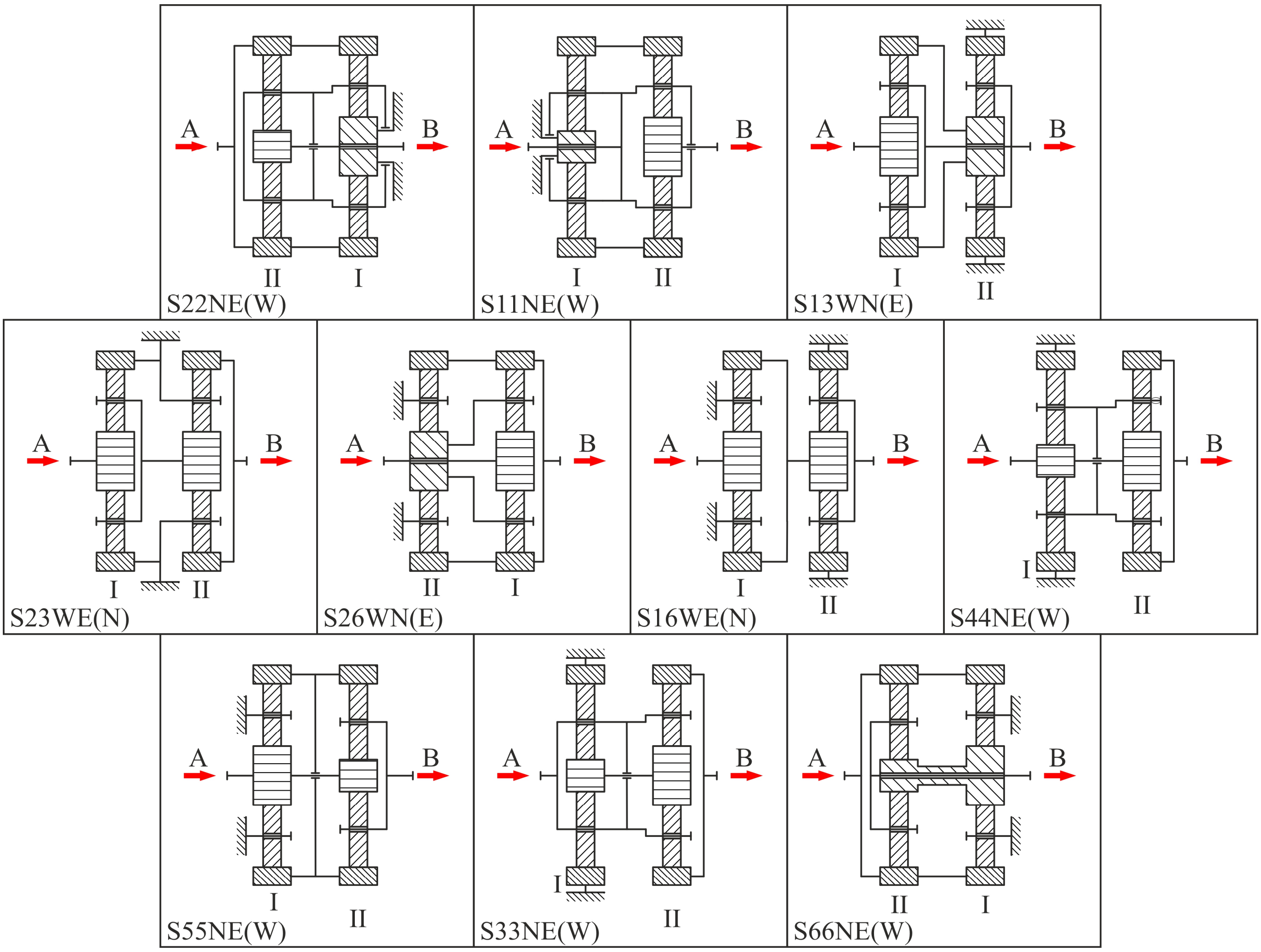

Fig. 12. Kinematic schemes of valid PGT variants with negative transmission ratios.

3.4. Applicable Variants of PGTs with Negative Gear Ratio

Conceptual (kinematic) schemes were created for each computer-generated variant. All schemes were set up to have the input and output shaft at the opposite sides of the gearbox, and coaxiality was ensured. It must be noted that some layouts require drilled shafts. The kinematic schemes of all variants with a negative overall transmission ratio are given in Fig. 12, while their respective transmission ratio functions are provided in Table 10. As in the case of Fig. 11, the first component gear train has been marked with the Roman numeral I, and the second component gear train with the Roman numeral II, as the first gear train on the power transmission path does not have to be the one closest to the input shaft, while the red arrows denote the power flow, with A being the power input and B being the power output. The same principles apply to the gear trains in Fig. 13.

Table 10. Transmission ratio functions of valid PGT variants with negative transmission ratios.

SchV	i	SchV	i
S16WE(N)	$-(t_I + t_I t_{II})$	S66NE(W)	$\frac{1+t_{II}}{t_{II}-t_I}, t_{II}<t_I$
S26WN(E)	$-(t_I + t_{II} + t_I t_{II})$	S44NE(W)	$\frac{t_{II}+t_I t_{II}}{t_{II}-t_I}, t_{II}<t_I$
S23WE(N)	$-(1 + t_{II} + t_I t_{II})$	S11NE(W)	$\frac{t_I}{t_I-t_{II}}, t_I<t_{II}$
S13WN(E)	$1 - t_I t_{II}$	S22NE(W)	$\frac{1+t_I}{t_I-t_{II}}, t_I<t_{II}$
S55NE(W)	$\frac{t_I+t_I t_{II}}{t_I-t_{II}}, t_{II}<t_I$	S33NE(W)	$\frac{t_{II}}{t_{II}-t_I}, t_{II}<t_I$

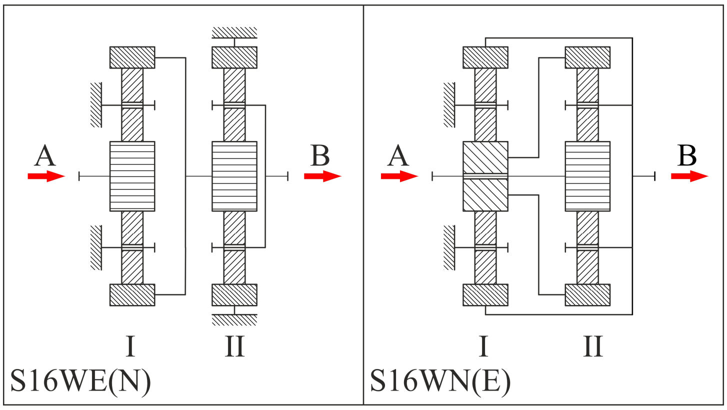

Fig. 13. Kinematic schemes of the S16WN(E) and S16WE(N) planetary gear trains.

The results of the analysis presented in this paper indicate that two options are acceptable, notably S16WN(E) and S16WE(N) (Fig. 13), however deeper analysis is required.

From the quality indicators in Tables 5, 6, 8 and 9 it can be concluded that both variants have a negligible difference in efficiency and are virtually identical in size, however preference should be given to the 16WN(E) variant from the technological standpoint, as it is easier to manufacture and is already used in a broad range of industrial applications. Therefore, this is the preferred option for an electric vehicle gearbox.

4. Conclusion

This article deals with a two-carrier PGT designed as a transmission for compact electric vehicles. The transmission must operate with an overall transmission ratio of $i = \pm 10{,}6$. Both positive and negative transmission ratios are acceptable as the prime mover is an electric motor. The gear train variants researched in this article are marked by means of a numerical designation for the schematic, and by a system using cardinal directions to designate the shafts of the layout variant. A software program was developed, using the torque method and appropriate algorithms to enable the synthesis of gear variants and their comparative analysis according to the criterion of minimal dimensions of the larger ring gear reference diameter, as well as to the criterion of the largest calculated efficiency ratio. The software program for the synthesis, analysis and optimal selection generates all possible solutions in accordance with the user input data. Each synthesized gear train can have multiple valid combinations of ideal torque ratios in the general case, so the software has selected the best combination of ideal torque ratios that provide minimal gearbox size with a satisfactory efficiency ratio at the same time. The program can select the appropriate variants and internal parameters of the gear train, while considering constraints such as gear geometry, overall transmission ratio, internal losses, gear material and efficiency limits. The transmission ratio functions of the compound gear trains analyzed in this article have been calculated using the torque method and are provided for ease of data comparison. Analysis has revealed that optimization for minimal gearbox size simplifies production and gives a minimum volume gearbox at the expense of a minimal reduction in gearbox efficiency. An exhaustive study and comparison of kinematically equivalent gearboxes has shown that 16WN(E) fulfils both the criteria of size and efficiency, and that it is justified to recommend this scheme for the application as a small electric vehicle transmission.

Author Contributions

S.A. Kalmaganbetov initiated the article, suggested the original idea and assembled the initial draft; M. Isametova provided planning oversight and research coordination; S. Troh developed the mathematical model and performed the numerical simulation; Ž. Vrcan performed the selection and kinematic validation of the simulation results; K. Markovic theoretically examined the mathematical background and performed sanity checking of the simulation results; D. Marinkovic performed an independent numerical validation of the simulation results. The manuscript was written through the contribution of all authors. All authors discussed the results, reviewed, and approved the final version of the manuscript.

Acknowledgments

This work has been supported by the University of Rijeka under projects number uniri-iskusni-tehnic-23-298 and uniri-iskusni-tehnic-23-168.

Conflict of Interest

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

Funding

The authors received no financial support for the research, authorship, and publication of this article.

Data Availability Statements

The datasets generated and/or analyzed during the current study are available from the corresponding author on reasonable request.

References

[1] Ehsani. M., Gao, Y.M., Emadi, A., *Modern electric, hybrid electric, and fuel cell vehicles: fundamentals, theory, and design*, 3rd ed., CRC Press, Boca Raton, 2018.
[2] Olabi, A. et al., Battery electric vehicles: Progress, power electronic converters, strength (S), weakness (W), opportunity (O), and threats (T), *International Journal of Thermofluids*, 16, 2022, 100212.
[3] Ravindra Jape, S., Thosar, A., Comparison of electric motors for electric vehicle application, *International Journal of Research in Engineering and Technology*, 6(9), 2017, 12–17.
[4] Wu, J., et al., Efficiency comparison of electric vehicles powertrains with dual motor and single motor input, *Mechanism and Machine Theory*, 128, 2018, 569–585.
[5] Zhang, W., Yang, J., Zhang, W., Influence of a new type of two-speed planetary gear automatic transmission on the performance of battery electric vehicles, *Energies*, 15(11), 2022, 4162.
[6] He, B., et al., Performance comparison of pure electric vehicles with two-speed transmission and adaptive gear shifting strategy design, *Energies*, 16(7), 2023, 3007.
[7] Mantriota G., Reina, G., Dual-motor planetary transmission to improve efficiency in electric vehicles, *Machines*, 9(3), 2021, 58.
[8] Bang, Y., Multi-speed transmission mechanism using a compound planetary gear set and brakes, *International Journal of Automotive Technology*, 20(4), 2019, 739–748.
[9] Fischer, S., Kocsis Szürke, S., Detection process of energy loss in electric railway vehicles, *Facta Universitatis-Series Mechanical Engineering*, 21(1), 2023, 81-99.
[10] Naunheimer, H. et al., *Automotive Transmissions: Fundamentals, Selection, Design and Application*, 2nd ed., Springer, Heidelberg, 2010.
[11] Han, J.O., Shin, J.W., Kim, J.C., Oh, S.H., Design 2-speed transmission for compact electric vehicle using dual brake system, *Applied Sciences*, 9, 2019, 1793.
[12] Patel, T., Dubey, A., Bhaskara Rao, L., Design and analysis of an epicyclic gearbox for an electric drivetrain, *International Journal of Recent Technology and Engineering*, 8(3), 2019, 6834–6842.
[13] Tomovic, A., Damjanovic, M., Tomovic, R., Jovanovic, J., The annulling of the sudden appearance of an unbalance in rotary machines by using active magnetic bearings, *Engineering Review*, 43(2), 2023, 117-134.
[14] Miltenovic, A. et al., Wear load capacity of crossed helical gears, *Facta Universitatis-Series Mechanical Engineering*, 22(1), 2024, 125-138.
[15] Vrcan, Ž., Stefanovic-Marinovic, J., Tica, M., Troha, S., Research into the Properties of Selected Single Speed Two-Carrier Planetary Gear Trains, *Journal of Applied and Computational Mechanics*, 8(2), 2022, 699-709.
[16] Abdali, S.H., Esmail, E.L., The structural synthesis of non-fractionated, three-degree-of-freedom planetary gear mechanisms, *Journal of Applied and Computational Mechanics*, 10(1), 2024, 205-223.
[17] Tica, M., Vrcan, Ž., Troha, S., Marinkovic, D., Reversible planetary gearsets controlled by two brakes, for internal combustion railway vehicle transmission applications, *Acta Polytechnica Hungarica*, 20(1), 2023, 95-108.
[18] Karaivanov, D., *Theoretical and experimental studies of the influence of the structure of the coupled two-carrier planetary gear trains on its basic parameters*, Ph.D. Thesis, University of Chemical Technology and Metallurgy, Sofia, 2000.
[19] Karaivanov, D., Petrova, A., Ilchovska, S., Konstantinov, M., Analysis of complex planetary change-gears through the torque method, *Machines. Technologies. Materials.*, 10(6), 2016, 38–42.
[20] Arnaudov, K., Karaivanov, D., Engineering analysis of the coupled two-carrier planetary gearing through the lever analogy, *Proceedings of the International Conference on Mechanical Transmissions*, Chongqing, China, 2001.
[21] Troha, S., *Analysis of a planetary change gear train's variants*, Ph.D. Thesis, University of Rijeka, Faculty of Engineering, Rijeka, 2011.
[22] Karaivanov, D.P., Troha, S., *Optimal selection of the structural scheme of compound two-carrier planetary gear trains and their parameters,* in Recent advances in gearing: scientific theory and applications, Radzevich, S.P. (Ed.), Springer, Cham, 2022.
[23] Arnaudow, K., Karaivanov, D., Systematik, Eigenschaften und Möglichkeiten von zusammengesetzten mehrsteg-Planetengetrieben, *Antriebstechnik*, 5, 2005, 58–65.
[24] Arnaudov, K., Karaivanov, D., *Torque method for analysis of compound planetary gear trains*, LAP Lambert Academic Publishing, London, 2017.
[25] Rodrigues, C., *Design of a high-speed transmission for an electric vehicle*, Ph.D. Thesis, Department of Mechanical Engineering, University of Porto, Faculty of Engineering, Porto, 2018.
[26] Larminie, J., Lowry, J., *Electric Vehicle Technology Explained*, John Wiley & Sons, New York, 2003.
[27] Hashemnia, N., Asaei, B., Comparative study of using different electric motors in the electric vehicles, *18th International Conference on Electrical Machines*, Vilamoura, Portugal, 1–5, 2008.
[28] Leitman, S., Brant, B., *Build Your Own Electric Vehicle*, McGraw Hill Professional, New York, 2013.
[29] Arnaudov, K., Karaivanov, D., Alternative method for analysis of complex compound planetary gear trains: Essence and possibilities, *International Conference on Power Transmissions*, Sinaia, Romania, 3-20, 2012.
[30] Arnaudov, K.B., Karaivanov D.P., *Planetary gear trains*, Taylor & Francis, New York, 2019.
[31] Karaivanov, D., Bekzhanov S., Saktaganov B., Torque method investigation of Wolfrom gear trains, *Engineering Review*, 43(1), 2023, 126-137.
[32] Troha, S., Milutinovic, M., Vrcan, Ž., *Characteristics and Capabilities of Two-Speed, Two-Carrier Planetary Gearboxes*, University of Rijeka, Rijeka, 2024.
[33] Vrcan, Ž., Ivanov, V., Alexandrov, A., Isametova, M., Size and efficiency based comparison of kinematically equivalent two-carrier planetary gear trains, *Engineering Review*, 42(3), 2022, 17-31.
[34] Mackic, T. et al., Analysis of Power Losses in Constrained Cycloid Drive, *Technical Gazette*, 30(4), 2023, 1104-1111.

ORCID iD

Sanzhar Armanuly Kalmaganbetov https://orcid.org/0009-0001-9248-4086
Madina Isametova https://orcid.org/0000-0003-4630-271X
Sanjin Troha https://orcid.org/0000-0003-2086-372X
Željko Vrcan https://orcid.org/0000-0002-7005-4130
Kristina Markovic https://orcid.org/0000-0003-1569-7464
Dragan Marinkovic https://orcid.org/0000-0002-3583-9434

How to cite this article: Kalmaganbetov S.A. et al., Selection of Optimal Planetary Transmission for Light Electric Vehicle Main Gearbox, *J. Appl. Comput. Mech.*, xx(x), 2024, 1–12. https://doi.org/10.22055/jacm.2024.46280.4490