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DESIGN FOR A BI-PLANETARY GEAR TRAIN

Summary. The article presents the design for a bi-planetary gear train. The project description is supplemented with calculations of kinematics, statics and meshing efficiency of the gear wheels included in the gear train. Excluded are calculations of strength and geometry of gears, shaft and rolling bearing, since they are similar to classical calculations for planetary gears. An assembly drawing in 2D and assembly drawings in 3D of the designed bi-planetary gear train are also shown. This gear train will form the main element of the research in hand.

Keywords: bi-planetary gear train, transmission ratio, meshing efficiency

1. INTRODUCTION

The structural diagram of a bi-planetary gear train, or double planetary gear train, is shown in Fig. 1a and Fig. 2 [1]. It consists of a main planetary gear set and an internal planetary gear set, which is known as an internal satellite system. The main planetary gear set comprises sun gear 1, satellites 2 and 6, external carrier arm h and ring gear 7, which is fastened to the frame, while the internal satellite system comprises sun gear 3, satellites 4 with internal carrier arm H, and ring gear 5, which is fastened to carrier arm h. In the bi-planetary gear

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train, the role of second toothed satellite 6 (Fig. 1b) fulfils the needs of the planetary gear train, known as the internal satellite system, with carrier arm H. H drives planetary gear 6, which already belongs to the main gear set. At the same time, rim gear 5 drives carrier arm h of the main planetary gear set.

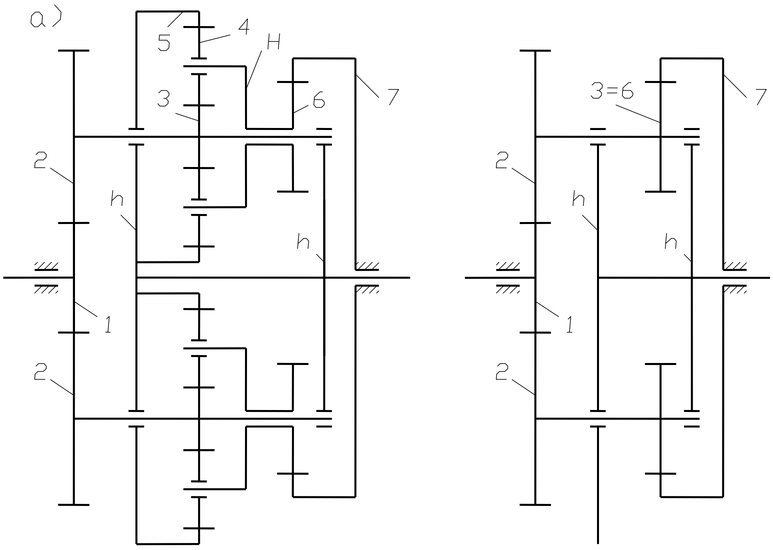


Fig. 1. Kinematic diagrams of the bi-planetary and planetary gear trains [1]

A characteristic feature of this gear train is that the satellites within the satellite mechanism rotate around three axes – those of its own central planetary gear set and the main central bi-planetary gear train. This particular property of bi-planetary gear train is mainly used in face milling cutters for combined mining.

2. DESIGN CALCULATIONS

2.1. Kinematic diagram of the gear

The structure of the transmission is selected depending on the kinematic ratio. The values of output torque are equal to $T_1 = 120,2\ \mathrm{N \cdot m}$ and $n_{out} = (24 \div 27)$ rpm, respectively, when the value of input revolution is equal to $n_N = n_1 = 1470$ rpm.

For the required range of output speed $(n_{out\,min}; n_{out\,max})$, the mean transmission ratio and the acceptable range of ratios are:

$$\left|i_{in,out}\right|_{mean} = \frac{n_N}{n_{out\,mean}} = \frac{1470}{25,5} = 57,647\,, \tag{1}$$

$$\left|i_{in,out}\right|_{\Delta} = \frac{n_N}{(n_{out\,min}; n_{out\,max})} = \frac{1470}{(24 \div 27)} = (61,250 \div 54,444)\,. \tag{2}$$

On basis that the demands correspond to the planetary gear, the kinematic diagram, as shown in Fig. 2, is according to [2, 3, 9].

This gear comprises the inner transmission (satellite planetary mechanism) of 3, 4, 5, H and of the outer transmission of 1, 2, 6, 7, h.

The degree of mobility of the transmission is:

$$W = 3 \cdot n + 2 \cdot p_5 - p_4 = 15 - 10 - 4 = 1 \tag{3}$$

where $n = 5$ (number of moving parts); $p_5 = 5$ (number of pairs of 5-class (arm h on the right is oversized or redundant)); and $p_4 = 4$ (number of pairs 4-class (engagement of teeth, counted only once)).

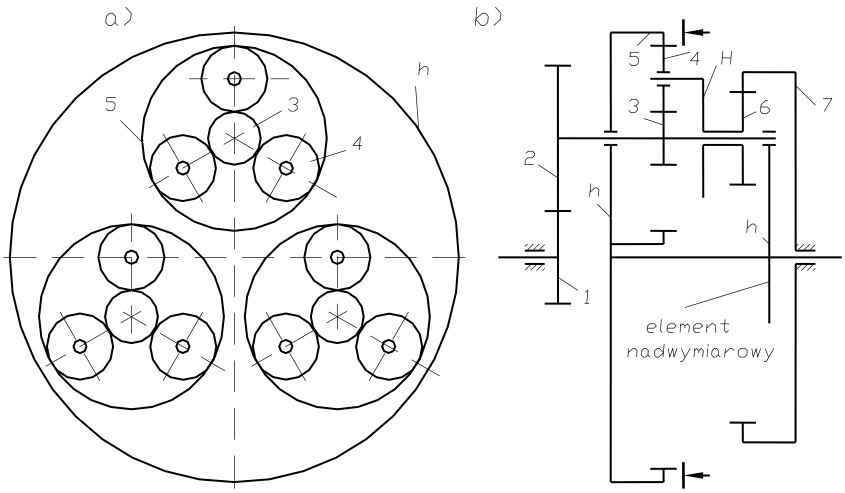


Fig. 2. Kinematic diagram of the bi-planetary gear train

As the degree of mobility is $W = 1$, only one value of a speed output or input is needed to determine the speed of each of the transmission components.

2.2. Gear transmission ratio

According to the definition of the kinematic gear ratio, $i_{1,h}^{7}$ (from pinion 1 to arm h, when wheel 7 is fixed) is equal to [1, 3, 4, 5, 6, 7]:

$$i_{1,h}^{7} = \left(\frac{n_1}{n_h} \right)_{n_7 = 0}, \tag{4}$$

where $n_1 = n_N = n_{in}$ (the rotational speed of pinion 1 or input rotational speed of transmission) and $n_h = n_{out}$ (the rotational speed of arm h, that is, the transmission output rotational speed) (Fig. 1). In order to determine the kinematic ratio, $i_{1,h}^{7}$ must first determine the base ratio of the inner transmission, assuming that the whole planetary gear rotational speed is equal to $-n_h$ (that is, to consider the kinematics of planetary transmission in relation to arm h). Then, the relative rotational speed of each wheel gear is equal to $n_j^h = n_j - n_h$ $(j = 1, 2, \ldots, 7)$ at the same time that $n_5^h = n_5 - n_h = 0$ and $n_7^h = n_7 - n_h = -n_h$ (because $n_5 = n_h$ and $n_7 = 0$).

Similarly, the relative rotational speed of inner arm H, in relation to external arm h, is $n_H^h = n_H - n_h$. The basic kinematic ratio of the inner transmission, which consists of wheels 3, 4, 5 and arm H with known relative speed n_3^h, n_4^h, n_5^h and n_H^h, can be determined using the

Willis formula (5) and expression (6) for the transmission ratio of wheels 3, 4 and 5, in relation to arm H as a function of the number of teeth (such as for the gears with fixed axes) [1]:

$$i_{3,5}^{H} = \frac{n_3^h - n_H^h}{n_5^h - n_H^h} \tag{5}$$

$$i_{3,5}^{H} = \left(\frac{n_3^h - n_H^h}{n_4^h - n_H^h}\right)\cdot\left(\frac{n_4^h - n_H^h}{n_5^h - n_H^h}\right) = \left(\frac{n_3^h - n_H^h}{n_5^h - n_H^h}\right) = \left(-\frac{z_4}{z_3}\right)\cdot\left(-\frac{z_5}{z_4}\right) = \left(\frac{z_5}{z_3}\right) \tag{6}$$

Hence:

$$i_{3,5}^{H} = \left(\frac{z_5}{z_3}\right) = \frac{n_3^h - n_H^h}{-n_H^h}\,, \tag{7}$$

because $n_5^h = n_5 - n_h = 0$.

To calculate the requested transmission ratio of the bi-planetary gear train, it is firstly necessary to determine unknown relative rotational speeds n_3^h and n_H^h using formula (7) above, as a function of n_h (9) and n_1 of the two conditions on the gear ratios $i_{H,7}$ (8) (the ratio from arm H to wheel 7) and $i_{3,1}$ (10) (the ratio from wheel 3 to wheel 1):

$$i_{H,7} = \frac{n_H^h}{n_7^h} = \frac{n_6^h}{n_7^h} = -\frac{z_7}{z_6}\,, \tag{8}$$

Hence

$$n_H^h = n_7^h \cdot \left(-\frac{z_7}{z_6}\right) = n_h \cdot \frac{z_7}{z_6}\,, \tag{9}$$

because $n_7^h = n_7 - n_h = -n_h$.

Similarly,

$$i_3^1 = \frac{n_3^h}{n_1^h} = \frac{n_2^h}{n_1^h} = -\frac{z_1}{z_2}\,, \tag{10}$$

Hence

$$n_3^h = n_1^h \cdot \left(-\frac{z_1}{z_2}\right) = \left(n_1 - n_h\right)\cdot\left(-\frac{z_1}{z_2}\right), \tag{11}$$

because $n_3 = n_2$ and $n_{3,h} = n_{2,h}$.

Thus, after the subsequent transformations, the quotient of the number of teeth z_5/z_3, according to formula (7), takes the following form (12):

$$\left(\frac{z_5}{z_3}\right) = \frac{\left(\frac{n_1}{n_h} - 1\right)\cdot\left(-\frac{z_1}{z_2}\right) - \frac{z_7}{z_6}}{-\frac{z_7}{z_6}}\,. \tag{12}$$

Hence the formula for the ratio of the bi-planetary gear train (4) is as follows:

$$i_{1,h}^{7} = \left(\frac{n_1}{n_h}\right)_{n_7=0} = 1 - \frac{z_7}{z_6}\cdot\frac{z_2}{z_1}\cdot\left(1 - \frac{z_5}{z_3}\right) \tag{13}$$

The required value of the ratio obtained for the numbers of teeth of the bi-planetary gear train – namely, $z_1 = 21,\ z_2 = 81,\ z_3 = 21,\ z_4 = 24,\ z_5 = -69,\ z_6 = 21,\ z_7 = -72$, is due to the following:

$$i_{1,h}^{7} = 1 - \frac{z_7}{z_6} \cdot \frac{z_2}{z_1} \cdot \left(1 - \frac{z_5}{z_3}\right) = 1 - \frac{-72}{21} \cdot \frac{81}{21} \cdot \left(1 - \frac{-69}{21}\right) = 57{,}676\,, \tag{14}$$

That is, $\left(i_{1,h}^{7} = 57{,}676\right) \in \left(54{,}444 \div 61{,}250\right)$.

The actual value of the output speed is $n_h = \dfrac{n_1}{i_{1,h}^{7}} = \dfrac{1470}{57{,}676} = 25{,}487\ \text{rpm}$.

2.3. Engagement efficiency of the transmission

Determining the efficiency of the gear train only takes into consideration the losses incurred due to friction in the gear engagements, i.e., it does not include the friction that occurs between the bearings and the resistance of the splashing oil. Thus, according to formula (15), the efficiency of the designed bi-planetary gear train only depends on the base kinematic ratio $i_{1,7}^{h}$ of the transmission and the base efficiency $\eta_{1,7}^{h}$ [1, 6, 7]:

$$\eta_{1,h}^{7} = \frac{1 - i_{1,7}^{h} \cdot \eta_{1,7}^{h}}{1 - i_{1,7}^{h}}\,. \tag{15}$$

The base efficiency is determined in relation to the serial sequence composed of the pair of wheels 1 and 2, planet gears 3, 4 and 5, and the last pair of wheels 6 and 7:

$$\eta_{1,7}^{h} = \eta_{1,2}^{h} \cdot \eta_{3,H}^{5} \cdot \eta_{6,7}^{h}\ . \tag{16}$$

The efficiency of the pair of outer wheels 1 and 2, when considered in motion relative to arm h, is determined by the following known formula [3, 6]:

$$\eta_{1,2}^{h} = \eta_{out(1,2)} \approx 1 - 2{,}3 \cdot \mu_z \cdot \frac{u_{1,2} + 1}{u_{1,2} \cdot z_1} = 1 - 2{,}3 \cdot \mu_z \cdot \frac{\dfrac{z_2}{z_1} + 1}{\dfrac{z_2}{z_1} \cdot z_1} \tag{17}$$

A similar formula determines the efficiency of the pair of inner wheels 6 and 7, when considered in motion relative to arm h [2, 3, 5, 6]:

$$\eta_{6,7}^{h} = \eta_{int(6,7)} \approx 1 - 2{,}3 \cdot \mu_z \cdot \frac{\left|u_{6,7}\right| - 1}{\left|u_{6,7}\right| \cdot z_6} = 1 - 2{,}3 \cdot \mu_z \cdot \frac{\dfrac{|z_7|}{z_6} - 1}{\dfrac{|z_7|}{z_6} \cdot z_6}\,. \tag{18}$$

The base efficiency for the internal epicyclic gearing of 3, 4, 5 and H is determined by the following formula [2, 3, 5, 6]:

$$\eta_{3,H}^{5} = \frac{1 - i_{3,5}^{H} \cdot \eta_{3,5}^{H}}{1 - i_{3,5}^{H}}, \tag{19}$$

where $i_{3,5}^{H}$ (see formula (7)) and $\eta_{3,5}^{H}$ are the base transmission ratio and the base efficiency of the inner epicyclic transmission of 3, 4, 5 and H, respectively:

$$\eta_{3,5}^{H} = \eta_{3,4}^{H} \cdot \eta_{4,5}^{H} = \eta_{out(3,4)} \cdot \eta_{int(4,5)} \tag{20}$$

where $\eta_{3,4}^{H} = \eta_{out(3,4)}$ (base efficiency of wheels 3 and 4).

$$\eta_{out(3,4)} \approx 1 - 2{,}3 \cdot \mu_z \cdot \frac{u_{3,4} + 1}{u_{3,4} \cdot z_3} = 1 - 2{,}3 \cdot \mu_z \cdot \frac{\frac{z_4}{z_3} + 1}{\frac{z_4}{z_3} \cdot z_3}, \tag{21}$$

where $\eta_{4,5}^{H} = \eta_{int(4,5)}$ (base efficiency of wheels 4 and 5).

$$\eta_{int(4,5)} \approx 1 - 2{,}3 \cdot \mu_z \cdot \frac{|u_{4,5}| - 1}{|u_{4,5}| \cdot z_4} = 1 - 2{,}3 \cdot \mu_z \cdot \frac{\frac{|z_5|}{z_4} - 1}{\frac{|z_5|}{z_4} \cdot z_4}. \tag{22}$$

Component values of the base efficiency for the assumed value of the coefficient of friction in engagement $\mu_z = 0{,}08$ are:

$$\eta_{1,2}^{h} = \eta_{out(1,2)} \approx 0{,}989, \qquad \eta_{6,7}^{h} = 0{,}994$$

$$\eta_{3,H}^{5} = 0{,}983, \qquad i_{3,5}^{H} = -3{,}286,$$

$$\eta_{3,5}^{H} = 0{,}978, \qquad \eta_{out(3,4)} = 0{,}984, \qquad \eta_{int(4,5)} = 0{,}994.$$

Thus, the efficiency of the planetary gear is:

$$\eta_{1,h}^{7} = \frac{1 - i_{1,7}^{h} \cdot \eta_{1,7}^{h}}{1 - i_{1,7}^{h}} = \frac{1 + 56{,}676 \cdot 0{,}966}{1 + 56{,}676} = 0{,}967, \tag{23}$$

because:

$$i_{1,7}^{h} = i_{1,2}^{h} \cdot i_{3,H}^{5} \cdot i_{6,7}^{h} = \left(-\frac{z_2}{z_1}\right) \cdot \left(1 - \frac{z_5}{z_3}\right) \cdot \left(-\frac{z_7}{z_6}\right) = \left(-\frac{81}{21}\right) \cdot \left(1 - \frac{-69}{21}\right) \cdot \left(-\frac{-72}{21}\right) = -56{,}676$$

And:

$$\eta_{1,7}^{h} = \eta_{1,2}^{h} \cdot \eta_{3,H}^{5} \cdot \eta_{6,7}^{h} = 0{,}989 \cdot 0{,}983 \cdot 0{,}994 = 0{,}966.$$

The value of efficiency of the bi-planetary gear unit is similar to the value of efficiency of the three-axes non-planetary gear train ($\eta_{1,6} \cong 0{,}99^3 = 0{,}970$), although it is greater than the efficiency value of the planetary gear, even when the transmission ratio is of less value.

2.4. Forces acting on the spur gear teeth

According to the principle of the operation of the bi-planetary gear train, the output torque T_h consists of three components:

a) Torque T_{h1} generated by $s_I = 3$ satellite 2 [1, 2]:

$$T_{h1} = s_I \cdot F_{h,2} \cdot r_h = s_I \cdot F_{o2,1} \cdot r_h = \frac{T_1}{r_1} \cdot (r_1 + r_2) = 583{,}83\ \mathrm{N \cdot m}, \qquad (24)$$

where $F_{h,2} = F_{o2,1} = \frac{T_1}{s_I \cdot r_1} = 1907{,}94\ \mathrm{N}$ (the force acting on the axis of satellite 2) (Fig. 3);
$r_1 = 21\,\mathrm{mm}$, $r_2 = 81\,\mathrm{mm}$ (the pitch radiuses of sun gear 3 and the satellite 2)'; and
$r_h = r_1 + r_2 = 102\ \mathrm{mm}$ (the radius of carrier arm h).

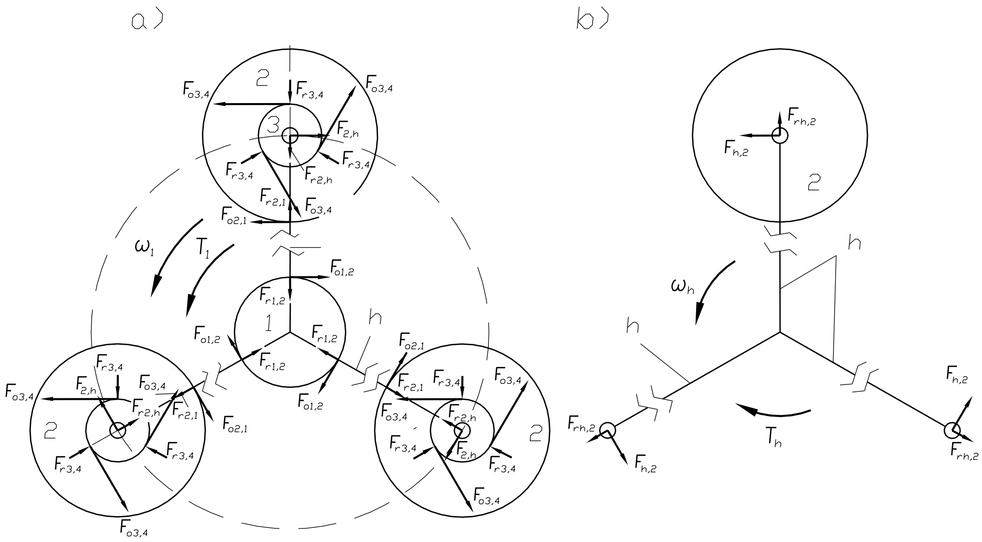


Fig. 3. The forces acting during the engagement of gears 1, 2, 3 and on left arm h

b) Torque T_{h2} generated by $s_{III} = 3$ rim gear wheel 5 (Fig. 3, Fig. 4):

$$T_{h2} = s_{II} \cdot s_{III} \cdot F_{o5,4} \cdot \frac{|d_5|}{2} = s_{III} \cdot \frac{T_1}{s_I} \cdot \frac{r_2}{r_1 \cdot r_3} \cdot |r_5| = 1523{,}35\ \mathrm{N \cdot m}, \qquad (25)$$

where $r_3 = 21\,\mathrm{mm}$, $|r_5| = 69\ \mathrm{mm}$ (the pitch radiuses of sun gear 3 and rim gear 5); and
$F_{o5,4} = F_{o3,4} = \frac{T_1}{s_I \cdot r_I} \cdot \frac{r_2}{s_{II} \cdot r_3} = 2453{,}06\,\mathrm{N}$ (the circumferential force acting on gear 5).

c) Torque T_{h3} generated by $s_{III} = 3$ carrier arm H and satellite 6 (Fig. 5):

$$T_{h3} = F_{h,6} \cdot r_h = F_{o6,7} \cdot r_h = \frac{T_1}{s_I} \cdot \frac{r_2}{r_1 \cdot r_3} \cdot s_{III} \cdot \frac{r_3 + r_4}{r_6} \cdot r_h = 4825{,}52\ \mathrm{N \cdot m}, \qquad (26)$$

where $r_6 = 21\,\text{mm}$ (the pitch radius of satellite 6); and

$F_{h,6} = F_{o6,7} = \frac{2 \cdot T_1}{s_I} \cdot \frac{r_2}{r_1 \cdot r_3} \cdot s_{II} \cdot \frac{r_3 + r_4}{r_6} = 15769{,}68\,\text{N}$ (the force acting on the axis of satellite 6).

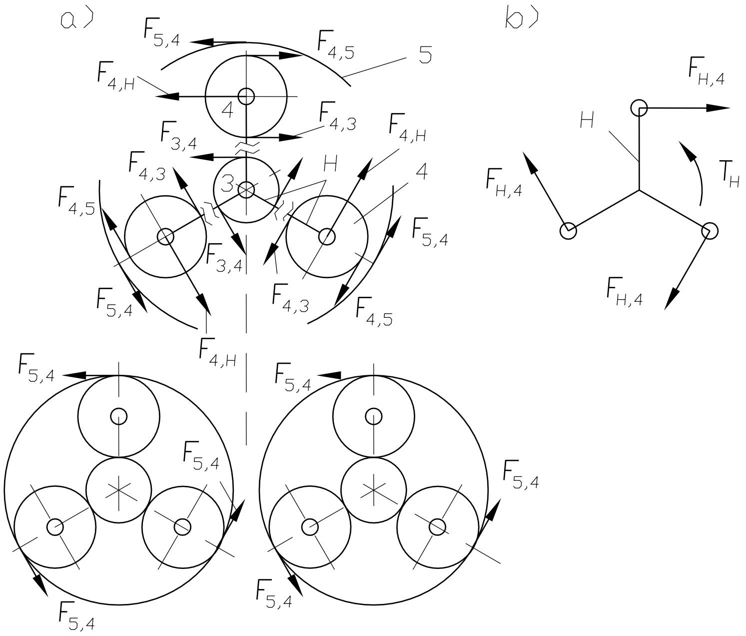


Fig. 4. The forces acting during the engagement of gears 3, 4, 5 and on arm H

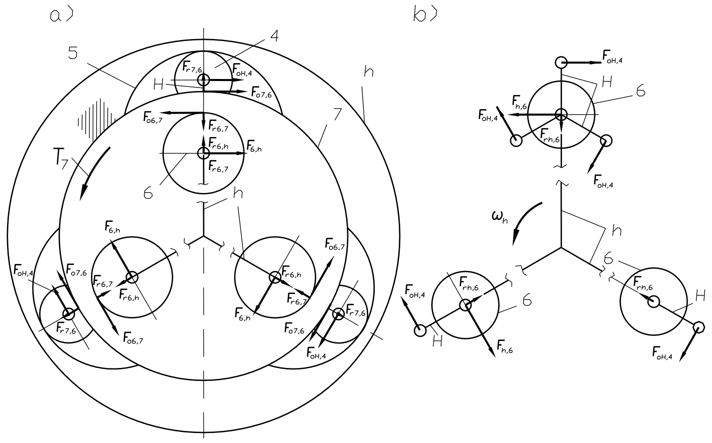


Fig. 5. The forces acting during the engagement of gears 6, 7 and on right arm h

As such, total output torque T_h is equal to:

$$T_h = T_{h1} + T_{h2} + T_{h3} = 6932{,}70\ \text{N} \cdot \text{m}. \tag{27}$$

Total output torque T_h, with regard to engagement efficiency $\eta^{7}_{1,h}$, is equal to:

$$T_h(\eta) = T_h \cdot \eta_{1,h}^{7} = 6932{,}70 \cdot 0{,}967 = 6703{,}92\ \text{N} \cdot \text{m}\,. \tag{28}$$

3. DESIGN BI-PLANETARY GEAR TRAIN

3.1. Description of the gear train

As previously mentioned in this paper, the bi-planetary gear train consists of two planetary drive systems, i.e., the outer gear set, also known as the main gear set, and the inner gear set, which is known as a satellite planetary system. The main planetary gear set consists of sun gear 3, which meshes with the three satellites 4 mounted on three shafts, namely, pinions 10 of the satellite planetary system (Fig. 6). Pinions 10 represent the sun gear of the planetary satellite system. It drives the three satellites 8, which are still engaged with the gear wheel due to internal teeth 9 being fitted to one of the three holes of arm 11. Each satellite is mounted rotationally on axle 12, on which is supported one of the three internal arms H (according to Fig. 1). Each arm H is a part of second satellite 13, which already belongs to the main gear set.

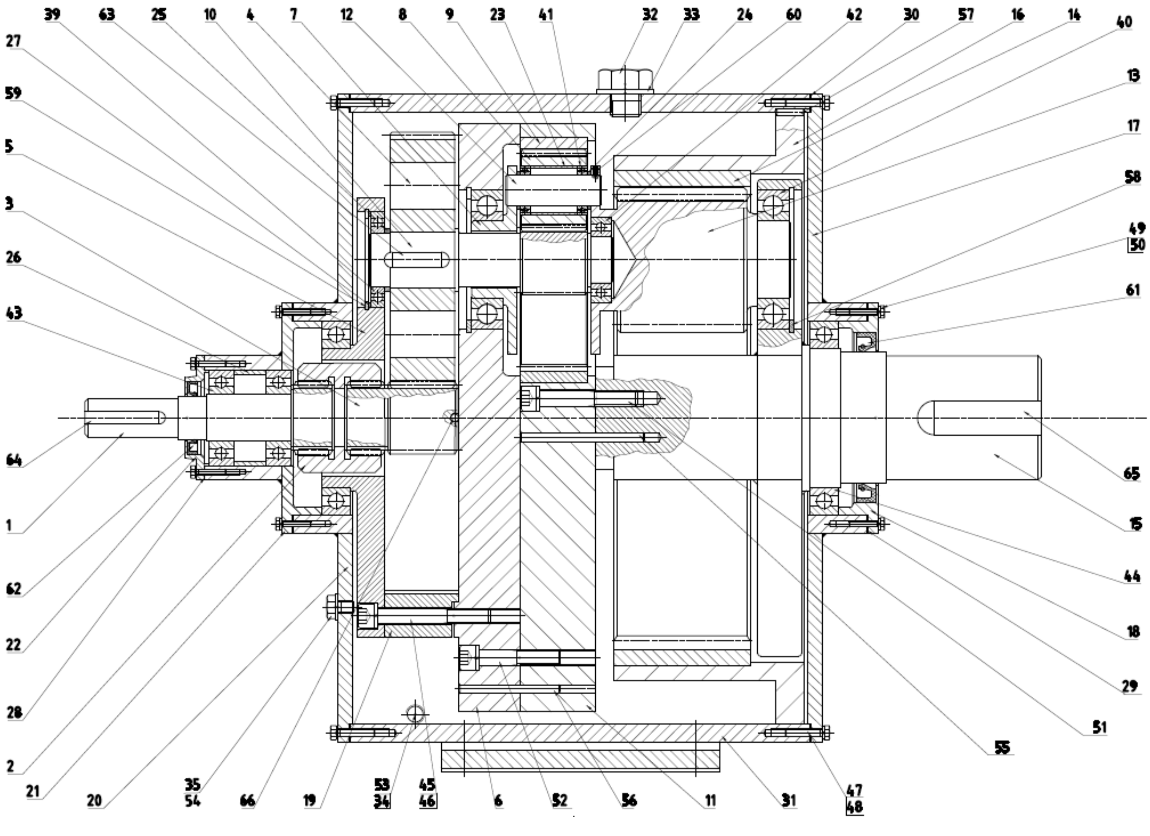


Fig. 6. Assembly engineering drawing of the bi-planetary gear train

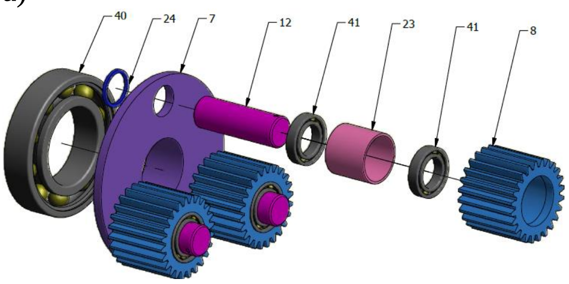
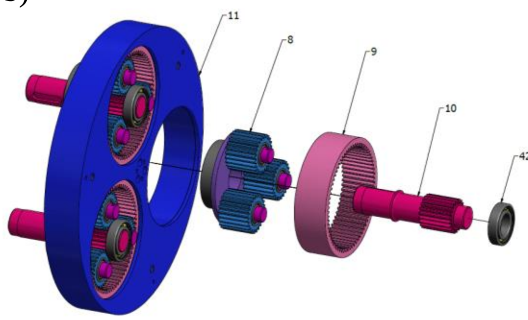
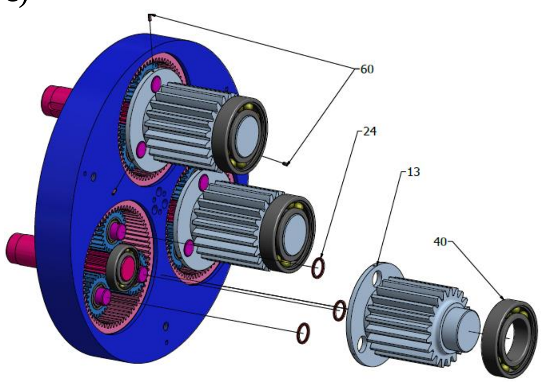
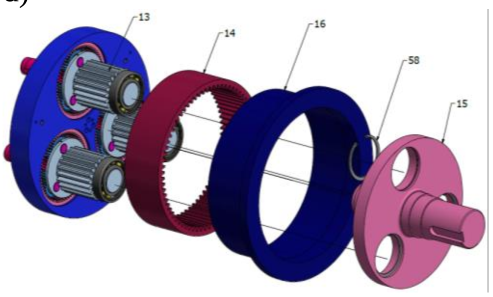
Satellite 13 is meshed with fixed internal gear wheel 14 and internally mounted on the right side of the hole in arm 11 of output shaft 15. Power is transmitted from the electric motor via shaft 1 and clutch 2 to sun gear 3, which is engaged with the three satellites 4 mounted on shaft 10 that transmits part of the torque along intermediate arm 11, then along output shaft 15, while the rest of the power is transmitted through gear wheel 9 to arm 11, then through internal arm 7 (satellite 13) to right arm 11.

3.2. The order of the assembly of the bi-planetary gear train elements

Engineering drawings of the assembly sequence, which allow for checking the correctness of the designed bi-planetary gear trains (Fig. 6) in terms of mounting options and possibly assessing its degree of difficulty, are included in Table 1.

Tab. 1.

Engineering drawings of the assembly sequence

a) 	b) 
c) 	d) 

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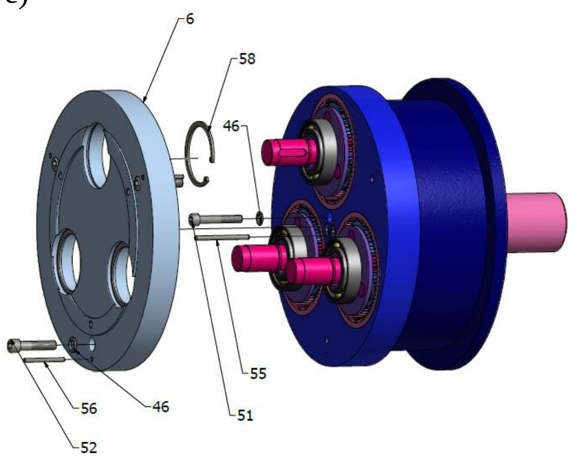

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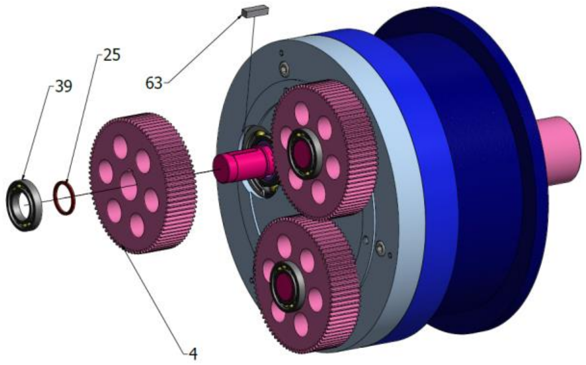

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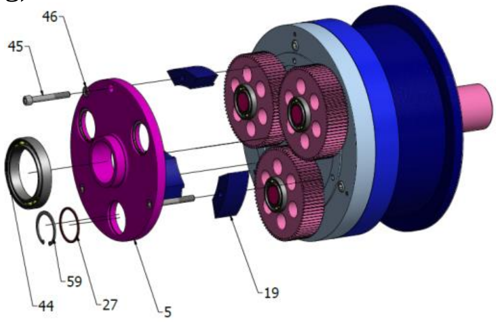

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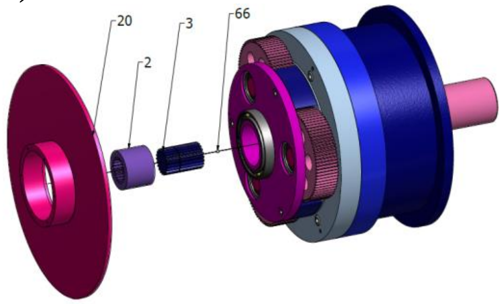

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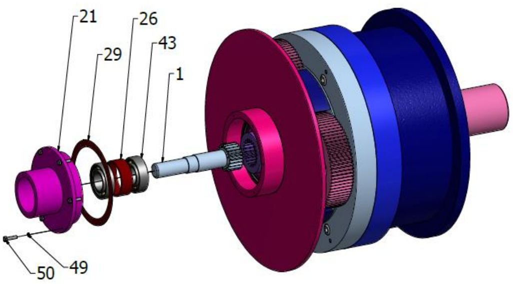

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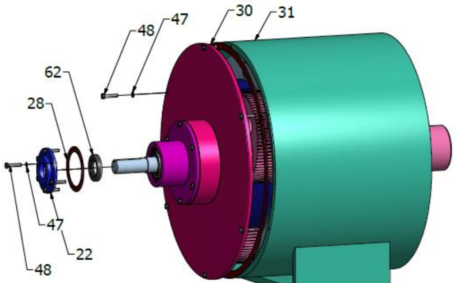

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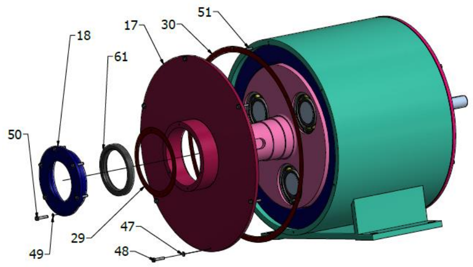

Note: not shown is the sequence of the assembly of simple elements, such as oil filler plugs, oil level and oil drain plug.

4. CONCLUSION

This paper presents the design of a bi-planetary gear train together with the calculation of the kinematics, statics and meshing efficiency of gear wheels. Omitted from the calculation are the geometry and strength of gears, shafts and rolling bearings, as these are recognized as being typical design calculations. 3D drawings of the order of assembly components have confirmed the correctness of the design of the bi-planetary gear train. The designed planetary gear is constructed as a prototype for transmission to the test bench in a laboratory for fundamental machine design. The main aim of this study is in comparing the dynamics of the transmission between fastened rim wheel 14 and the solution, when wheel 14 is suspended flexibly from the coupling gear (variant II of the solution).

References

1. Cyplakow Ju.S. 1968. *Biplanetarnye mechanizmy*. [In Russian: *Bi-planetary Mechanisms*]. Moscow: Mas'inostroenie.
2. Drewniak J. 2000. *Zbiór zadań z PKM III*. [In Polish: *The Set of Tasks from PKM III*]. Bielsko-Biała: ATH.
3. Kudriawcew W.N., Ju.N. Kirdiaszew (red.). 1977. *Planetarnye peredaczi*. [In Russian: *Planetary Peredaktsi]*. Leningrad: Mas'inostroenie.
4. Looman J. 1988. *Zahnradgetriebe*. [In German: *Gear Transmission*]. Berlin: Springer-Verlag.
5. Mueller H.W. 1971. *Die Umlaufgetriebe*. [In German: *The Epicyclic Gear*]. Berlin: Springer Verlag.
6. Mueller L., A. Wilk. 1996. *Zębate przekładnie obiegowe*. [In Polish: *Toothed Planetary Gears*]. Warsaw: PWN.
7. Rudenko W.N. 1980. *Planetarnye i wolnowye peredaci*. [In Russian: *Planetary and Wave Transmission*]. Moscow: Mas'inostroenie.
8. Homišin J. 2008. *Dostrajanie układów mechanicznych drgających skrętnie przy pomocy sprzęgieł pneumatycznych: kompendium wyników pracy naukowo-badawczych.* [In Polish: *Tuning Torsionally Vibrating Mechanical Systems using Pneumatic Couplings: A Compendium of Performance Research*]. Bielsko-Biała: ATH. ISBN: 978-83-60714-55-3.
9. Homišin J., P. Kaššay, P. 2009. „Comparison of tuning in shipping system using pneumatic tuners of torsional oscillations". *Machine modeling and simulations:* 147-154. ISBN 978-80-89276-18-9.
10. Grega R., J. Krajňák. 2012. „The pneumatic dual-mass flywheel". *Scientific Journal of Silesian University of Technology. Series Transport* 76: 19-24. ISSN: 0209-3324.
11. Grega R. et al. 2015 „The chances for reduction of vibrations in mechanical system with low-emission ships combustion engines". *International Journal of Maritime Engineering* 157 (A4): 235-240. ISSN 1479-8751.
12. Homišin J., P. Kaššay, M. Urbanský. 2011. „High-flexibility characteristics of pneumatic flexible shaft couplings". *Pneumatyka* 79(2): 26-29. ISSN 1426-6644.
13. Medvecká-Beňová S., J. Krajňák. 2015. „Modifications to a gearbox to increase the performance of a winding machine for steel sheets." *IT-strojár*: 1-3. ISSN 1338-0761.

14. Bigoš P., J. Kuľka, M. Mantič, M. Kopas. 2015. „Comparison of local stress values obtained by two measuring methods on a blast furnace shell”. *Metalurgija* 54(1): 101-104. ISSN 0543-5846.
15. Grega R., J. Krajňák. 2013. „The application of pneumatic flexible coupling in a conveyor drive”. *Technológ* 5(4): 51-54. ISSN 1337-8996.
16. Medvecká-Beňová S., J. Vojtková. 2014. „Application of an asymmetric profile in gearings with eccentric gears”. *Scientific Journal of Silesian University of Technology. Series Transport* 85: 89-93. ISSN: 0209-3324.
17. Kuľka J., M. Mantič. 2014 „Effect of operational condition changes on the durability of the rope for pusher of circular wagon tipper”. *Applied Mechanics and Materials* 683: 28-32. ISSN 1660-9336.
18. Homišin J., M. Urbanský. 2015. „Partial results of extremal control of mobile mechanical system”. *Diagnostyka* 16(1): 35-39. ISSN 1641-6414.

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