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KINEMATICS OF TWO SIMPLE PLANETARY GEARS

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ABSTRACT

Multi-speed planetary gear transmissions (PGTs) have a limited application. Two and three-speed planetary gear transmissions were used in GGT (hydromechanical transmission); in recent years the number of gears in GGT has been growing. With the increase of the number of gear, the complexity of the PGT is growing significantly. The PGT with two degrees of freedom for 8 gears requires 24 gearwheels and for 16 gears it requires 48 gearwheels. This article describes the structural methods of realization of the kinematic capacities of two simple three-link planetary gears connected in series.

Keywords: simple planetary gear, sun gear, carrier, satellites, epicyclical wheel, kinematics, structure.

1. INTRODUCTION

The modern LTVs (Land Transport Vehicles) including road trains require the application of the multi-speed transmissions that provide the optimal operation mode of the engine in the wide range of operation conditions. 16-speed transmissions of the modern road trains have a range (reduction ratio of lower and upper gears) $D \approx 17$, interval (reduction ratio of the neighbouring gears) $q \approx 1.21$. The traditional methods of design lead to the increase of the dimensions and weight of the transmission units.

More often, the planetary gears are used in such transmissions. The planetary gears are installed on the output of the MGB (multi-speed gear box) in the wheel reduction gears or final reduction gears, in PGBs (planetary gear boxes) consisting of several rows of planetary gears.

Planetary gears and their variant - harmonic drives that are characterized by the small dimensions and weight satisfy more completely the requirement to the decrease of the specific amount of metal of the machines among all types of mechanical gears. This is connected to the multithreading effect and the application of the internal toothing [1].

2. KINEMATIC CAPACITIES OF TWO SIMPLE PLANETARY GEARS WITHOUT INTERFACE REDUCER

The increase of the use efficiency of gearwheels can be reached due to the more complete use of the kinematic capacities of planetary gears. A simple PG consisting of the three links: a sun gear (*a*), an epicyclic wheel (*b*) and a carrier (*h*) with satellites is characterized by the internal parameter $K = Z_b/Z_a = 1.5 - 5$, which is equal to the number of teeth Z_b of the epicyclic wheel and Z_a of the sun gear. It can provide 7 gears in the gear modes, 5 forward gears: 1) $U_{ah}^{b} = K+1$; 2) $U_{bh}^{a} = (K+1)/K$; 3) direct gear $U = 1.0$; 4) $U_{hb}^{a} = K/(K+1)$; 5) $U_{ha}^{b} = 1/(K+1)$; two reverse gears: 1R) $U_{ab}^{h} = -K$; 2R) $U_{ba}^{h} = -1/K$. A sign "-" points out the change of direction rotation of the link in the output of the planetary gear. The upper index points out the stopped link, the lower indices point out the links of input and output of the torsion torque. For example, the first slow gear U_{ah}^{b} is realized at the stopped epicyclic wheel (*b*), when the torsion torque is delivered to the sun gear (*a*) and when the torque is removed from the carrier (*h*). The PG can also provide 3 gears in the integration modes when the torsion torque is delivered to the two links with the different rotational rate and it is removed from the third one.

The kinematic capacities of the transmission unit increase significantly at the series connection of two planetary gears. At least 5 × 5 = 25 variants of the forward gears can be achieved and also the additional gears when the reverse gears are activated in series.

Figure-1 shows the kinematic scheme of the transmission unit type 10R4 (10 forward gears and 4 reverse gears), consisting of two simple planetary gears connected in series and its ray diagram with the tables of the clutches positions. The upper positions of the gear clutches correspond to the gear: 1R) $U_{ab}^{h} = -K$; the lower positions: 1) $U_{ah}^{b} = K+1$.

The kinematic capacities of the units are convenient to represent in a form of the ray diagrams. The horizontal scales are the value of the reduction ratio in the logarithmic form. On the low horizontal the upper scale is uniform in the logarithmic scale; the lower scale is a value of the reduction ratio in the physical terms. The rays emerge from the point 0. The rays directed to the right point out the decelerating gears, the flatter the ray, the bigger the reduction ratio. The vertical ray characterizes the gear with $U = 1.0$. The rays directed to the left point out the accelerating gears, the flatter the ray, the smaller the reduction ratio.

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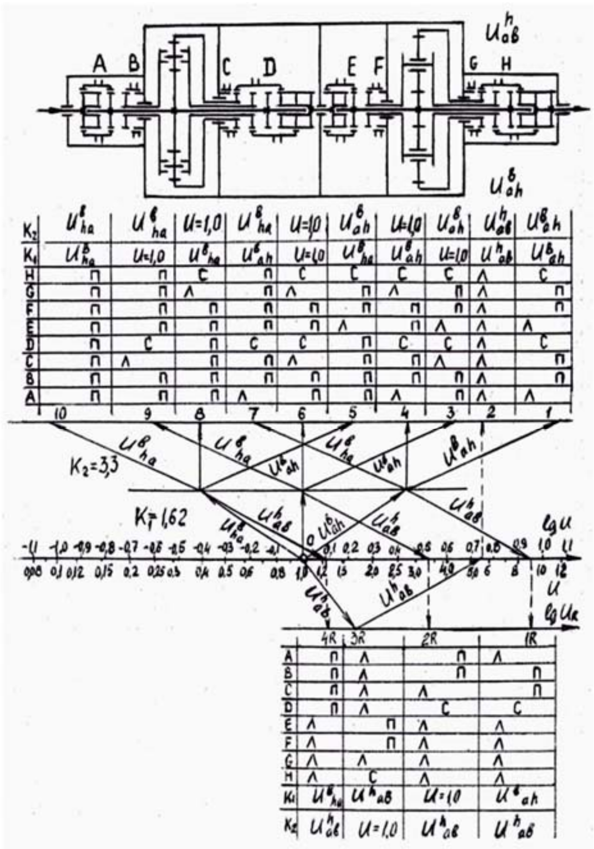

Figure-1. Kinematic scheme of the transmission unit type 10R4, consisting of the simple planetary gears connected in series and its ray diagram.

The work of the unit is clearly seen. For example, at the first gear (see the table above) the torsion torque from the input shaft is transmitted by the clutch A in the left (L) position to the ring gear, the central shaft and the sun gear (*a*) of the first PG with $K_1 = 1.62$. The clutches B and F are in the right (R) neutral position. The clutch C also in the right position stops the epicyclic wheel (*b*) regarding the casing. The clutch D in the middle (M) position delivers the increased torsion torque from the carrier (*h*) to the ring gear of the input shaft of the second PG with $K_2 = 3.3$. The clutch E in the left (L) position delivers the force to the central shaft and the sun gear (*a*) of the second PG. The clutch is in the right (R) neutral position. The clutch G also in the right position stops the epicyclical wheel (*b*) regarding the casing. The clutch H in the middle (M) position delivers the torsion torque from the carrier (*h*) to the ring gear of the output shaft.

On the ray diagram, two rays correspond to this gear U_{ah}^{b} from the point 0 to the right and up to the point 1 on the upper horizontal. The reduction ratio of the first gear will be $U_1 = (K_1 + 1)(K_2 + 1) = (1.62+1)(3.3+1) = 11.27$; lq $U_1 = 1.05$. For the 10th gear $U_{10} = 1/U_1 = 1/11.27 = 0.089$; lq $U_{10} = -1.05$. The range of the unit is very high $D = U_1/U_{10} = 11.27/0.089 = 126.6$.

A simple series connection of two planetary gears does not provide the necessary effect. As Figure-1 shows, every PG can realize only three forward gears (two gears at the stopped epicyclic wheel (*b*), one is a direct gear) and one reverse gear at the stopped carrier (*h*). As a result, 10 reverse gears were obtained (1–10 above the upper horizontal, 3 × 3 = 9 + 1 gear when connecting the reverse gears - 2nd gear) and 4 reverse gears (1R-4R under the lower horizontal). The gears are not optimally located, symmetrically to the point 0; this point should be closer to the gear 10. The intervals between the gears are too big and this makes the gear change more difficult. The direct connection of two planetary gears excludes the gears with the stopped sun gear (*a*), the interface reducer between the planetary gears is necessary.

The problems of coordination of two planetary gears arise. It is necessary to select the internal parameters *K* of the planetary gears in the way that the bigger number of gears can be realized at their rational distribution in the range *D*.

In Figure-2, the two ray diagrams are shown: *a*) $K_1 = 1.62$ and *b*) $K_1 = 2.0$; they characterize the work of two planetary gears connected in series.

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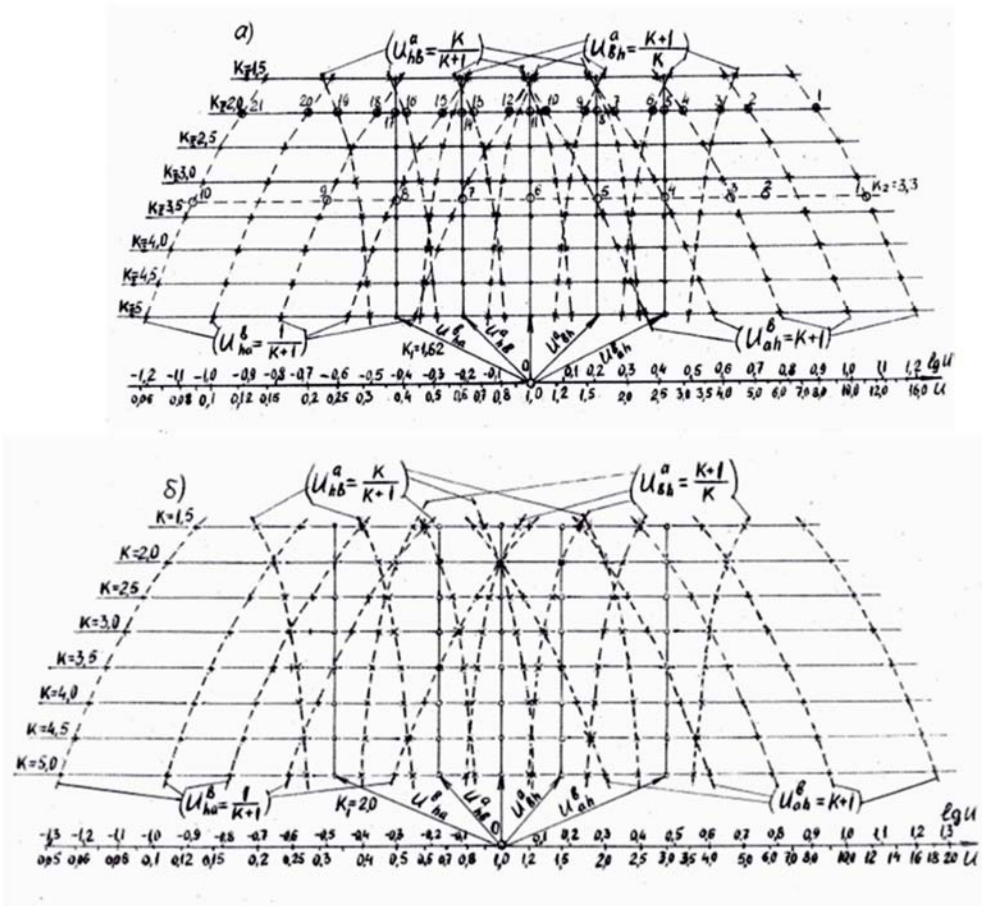

Figure-2. Distribution of values of reduction ratio in the range while connecting two planetary gears with different internal parameters: *a*) K_1 = 1.62; *b*) K_1 = 2.0.

The five rays between the lower horizontals reflect the work of the first planetary gear. For example, in Figure-2, *a* at K_1 = 1.62 the equal segments were obtained 0.21 in a logarithmical scale. From the points: -0.42; -0.21; 0.0; 0.21 and 0.42 the vertical lines emerge. The values of the reduction ratios for PG with different internal parameters are shown on the upper horizontals (top-down) from K_2 = 1.5 to K_2 = 5.0 at the various conditions of PG that are connected by the dash lines. The conditions of PG are shown: down to the right 1) $U_{ah}^{b} = K + 1$; to the left 5) $U_{ha}^{b} = 1/(K+1)$; top to the right 2) $U_{bh}^{a} = (K+1)/K$; to the left 4) $U_{hb}^{a} = K/(K+1)$; direct gears are marked by the vertical lines coming out from the ends of rays *a*) K_1 = 1.62 *b*) K_1 = 2.0. The segments between the points on every line point out the intervals between the neighbouring gears.

In Figure-2, it is clearly seen that the distribution of intervals according to the ranges is not uniform; some points coincide almost completely, the interval 1.62 is too big and makes the gear change difficult. It is preferable to have the distribution with small equal intervals; this will provide the comfort of gear changing. The analysis of Figure data 2 shows that the distribution from the upper part of the figure can be used as a basis. For example, the ray diagram in Figure-1 corresponds to the row in Figure 2, *a* at K_2 = 3.3.

3. KINEMATIC CAPACITIES OF TWO SIMPLE PLANETARY GEARS CONNECTED BY INTERFACE REDUCER

Figure-3 shows a kinematic diagram of the unit type 21R10 and a ray diagram with the table of position of gear clutches on the different gears for two planetary gears configured linearly - one by one. The first PG has K_1 = 1.62; the second PG has K_2 = 2.0; the interface reducer is located between them. The upper positions of gear clutches correspond to the gear 1) $U_{ah}^{b} = K + 1$; the lower positions - 1R) $U_{ab}^{h} = -K$.

In the ray diagram, the input of the torsion torque into the unit is marked by the point 0 on the third horizontal from the bottom. It shows the values of the reduction ratios in the logarithmic and natural form. The additional gear 1 *d*, shown in the lower horizontal, is formed by the two consequent reverse gears. 21 out of 25 gear variants can be realized, their distribution in the range is shown on the upper horizontal and corresponds to the second upper horizontal in Figure-2, *a*.

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The installation of the interface reducer between the planetary gears allowed to stop the sun gear wheels of the planetary gear by the clutches D and E, to realize the gears 2) $U_{bh}^{a} = (K+1)/K$ and 4) $U_{hb}^{a} = K/(K+1)$, and also to optimize the position of the row of the reduction ratios due to the displacement to the lower gears. When increasing the reduction ratio of the interface reducer the point 0 corresponding to the input shaft is shifted to the left increasing the reduction ratio of the 1st gear.

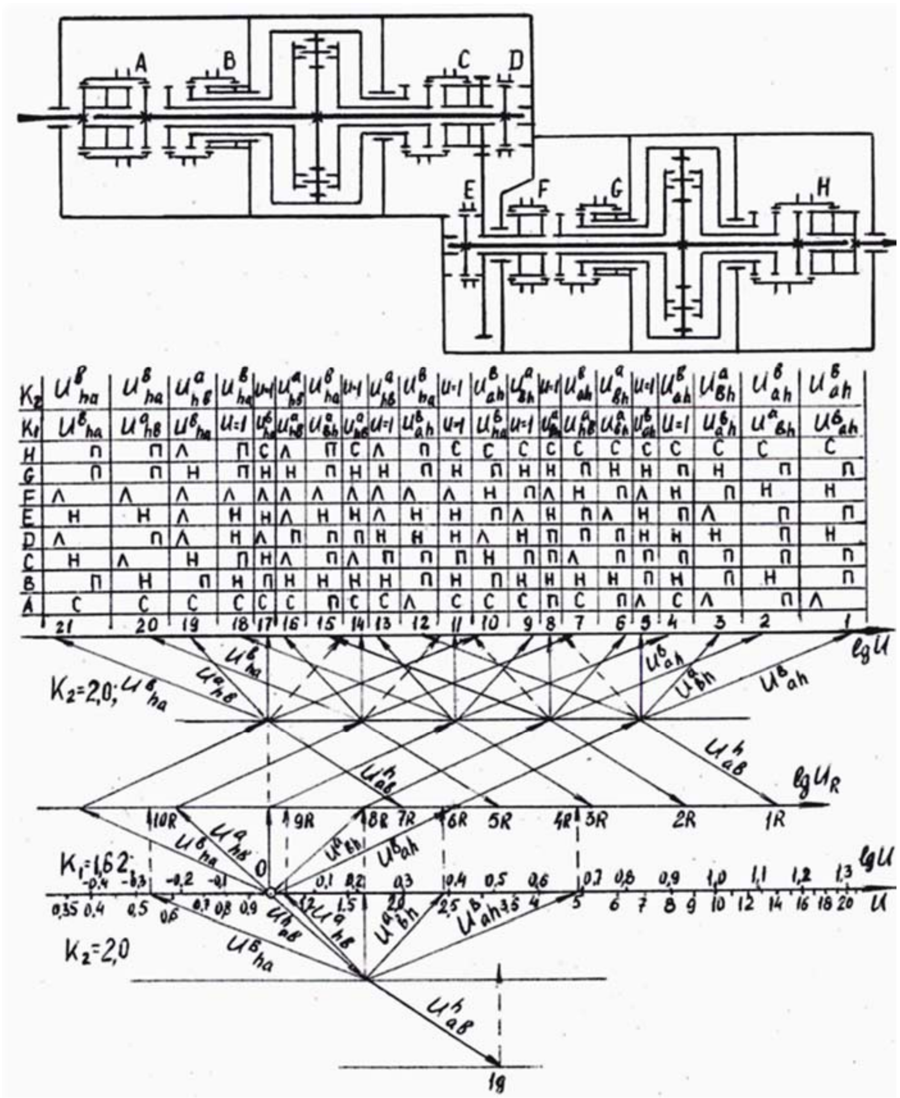

Figure-3. Kinematic diagram of the transmission unit type 21R10, consisting of two simpleplanetary gears with the interface reducer and its ray diagram.

On the first gear, the torsion torque from the input shaft is delivered by the clutch A in the left (L) position to the ring gear, the central shaft and the sun gear (a) of the first planetary gear with $K_1 = 1.62$. The clutches B and G in the right (R) condition stop the epicyclic wheels (b) regarding the casing. The clutch C also located in the right (R) position delivers the increased torsion torque from the carrier (h) to the ring gear of the driving gearwheel of the interface reducer. The clutches D and F are in the neutral (N) position. The clutch E in the right position delivers the torque increased by the interface reducer to the central shaft and the sun gear wheel (a) of the second PG with $K_2 = 2.0$. The clutch H in the middle (M) position delivers the increased torsion torque from the carrier (h) to the ring gear of the output shaft.

On the ray diagram, three rays from the point 0 to the right and up correspond to this gear: U_{ah}^{b} of the first PG, then the ray of the interface reducer, then U_{ah}^{b} of the second PG to the point 1 on the vertical horizontal.

The reduction ratio of the 1st gear will be $U_1 = (K_1 +1)\, U_{c.p.}\, (K_2 +1) = (1.62+1)\, 2.6\, (2.0+1) = 20.44$; lq $U_1 = 1.31$. For the 21st gear, it is $U_{21} = 1/(K_1 +1)\, U_{c.p.}\, 1/(K_2 +1) = (1/2.62)\, 2.6\, (1/3) = 0.33$; lq $U_{21} = -0.48$. The range of the unit is high $D = U_1/U_{21} = 20.44/0.33 = 61.9$.

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The distribution of gears in the range is close to the optimal. The 1st gear with U_1 = 20.44 provides the "creeping" motion mode, for example, when manoeuvering on the narrow site during the loading-unloading works.

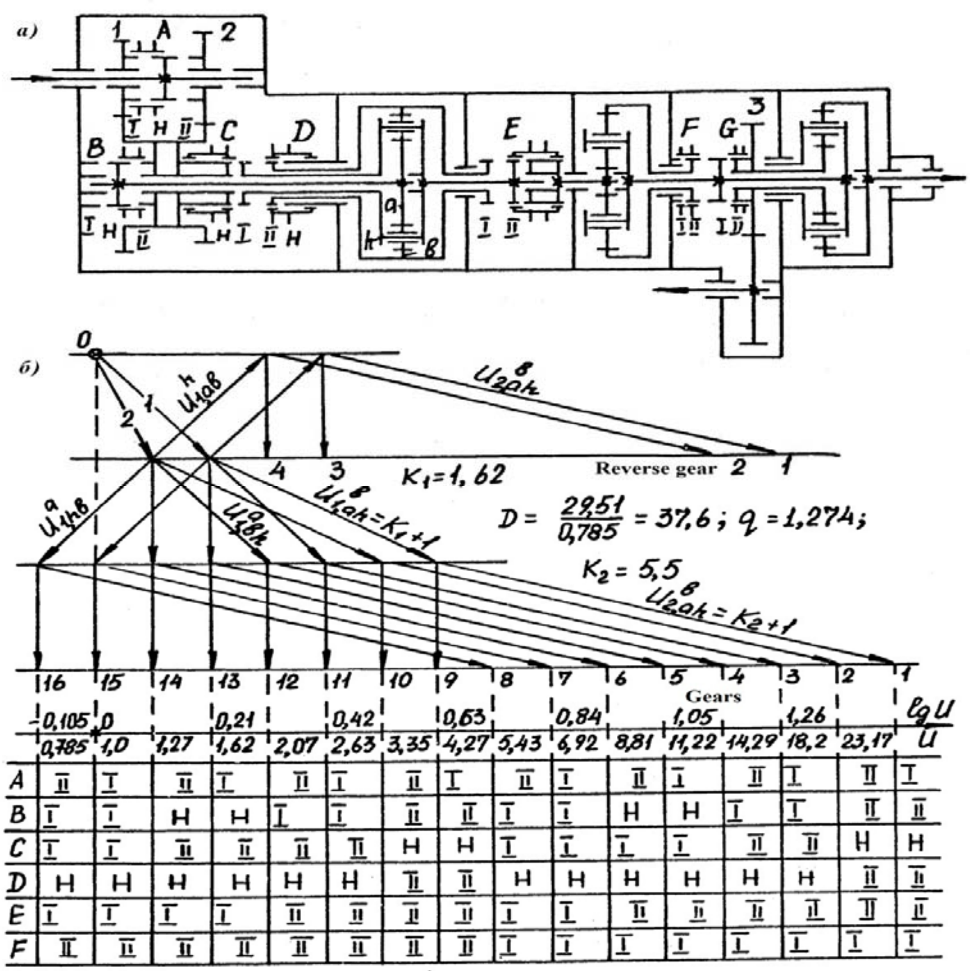
4. KINEMATIC CAPACITIES OF TWO SIMPLE PLANETARY GEARS WHEN INSTALLING ADDITIONAL GEAR BOX

The equal small intervals between the gears can be provided when installing the additional gear box in front of the planetary gear. In Figure-4, the kinematic diagram of the shaft-planetary MTGB (multistep transfer gear box) type 16R4 (16 forward gears and 4 reverse gears) and its ray diagram.

Two pairs of gearwheels marked by 1, 2, and the clutch A that is located between them on the main shaft are used in the mode of the front splitter. The first PG (left) with the internal parameter K_1 = 1.62 is used on 4 forward motion modes and one reverse gear. The second PG with K_2 = 5.5 is used as the rear additional reducer - auxiliary gear box. It realizes two variants of gears: first - direct gear - clutch F in the second (right) position, second - slow - clutch F in the first (left) position.

On the first gear of MTGB the torsion torque during the left position of the clutch A is delivered from the main shaft to the 1 pair of gearwheels, later along the clutch D in the right position to the drive axis of the sun gear (*a*) of the first PG. At the same time, the clutch C is located in the left (neutral) position, and the clutch D in the second (middle) position stops the epicyclic wheel (*b*) regarding the casing. The torque is removed by the clutch E in the second position from the carrier (*h*) to the drive axis of the sun gearwheel of the second PG. The clutch F in the left position of the epicyclic wheel (*b*) of this PG is also stopped regarding the casing of the unit, increased torsion torque from its carrier is delivered to the carrier (*h*) of the right PG in the mode of the nonequalizing differential.

On the ray diagram, three rays belong to this gear: from the point 0 on the left and up the ray 1 to the right and down, then the ray $U_{1,ah}^{b}$ = K_1 +1, and the ray $U_{2,ah}^{b}$ =K_2 + 1 to the point 1 on the lower horizontal with U_1 = 29/1. $D = U_1/U_{16}$ = 29.51/0.785 = 37.6.



Gears	16	15	14	13	12	11	10	9	8	7	6	5	4	3	2	1
lgU	-0,105	0		0,21		0,42		0,63		0,84		1,05		1,26		
U	0,785	1,0	1,27	1,62	2,07	2,63	3,35	4,27	5,43	6,92	8,81	11,22	14,29	18,2	23,17	
A	II	I	II	I	II	I	II	I	II	I	II	I	II	I	II	I
B	I	I	H	H	I	I	II	II	I	I	H	H	I	I	II	II
C	I	I	II	II	II	II	H	H	I	I	I	I	II	II	H	H
D	H	H	H	H	H	H	II	II	H	H	H	H	H	H	II	II
E	I	I	I	I	II	II	II	II	I	I	II	II	II	II	II	II
F	II	II	II	II	II	II	II	II	I	I	I	I	I	I	I	I

Figure-4. Kinematic diagram of MTGB type 16R4 with two planetary gears in the gear modes and the right PG in the nonequalizing differential mode.

5. TECHNICAL AND ECONOMICALASSESSMENT OF MORE COMPLETE USE OF KINEMATIC CAPACITIES OF TWO SIMPLEPLANETARY GEARS

The planetary gear boxes in comparison with the shaft-drive gear boxes possess several advantages: a little higher coefficient of efficiency, the big gear compactness, the decrease of the teeth module while changing from the simple gear boxes to the planetary gear boxes; it provides the increase of the lifetime, noiselessness of operation, the higher reduction ratio for the same diameters of the gearwheels can be obtained.

The increase of the number of gears makes the planetary gear boxes more complicated. Planetary gear boxes with two degrees of freedom for 8 gears require 24 gearwheels, and for 16 gears they require 48 gearwheels.

The number of degrees of freedom is one gear more than the number of the control elements activated on one gear. The number of gearwheels decreases significantly at three degrees of freedom of the PG - 12 and 16 gearwheels correspondingly [2].

Even at three freedom degrees of the planetary gear box the efficiency of use of the gearwheels is not big, the intensity coefficient of the use of the gearwheels equal to the ratio of the number of forward gears to the number of used gearwheels for it Ki = 8/12 = 0.67 and Ki = 16/16 = 1.0 is not high. The higher Ki, the smaller the dimensions and the specific amount of metal of the unit are [3].

The considered diagram of the MTGB in Figure-4 simplifies the transmission of the vehicle, reduces its dimensions and weight. For example, 16-stepped ratio gear transmission of the vehicle KrAZ-260 consists of two massive units (gear box 340 kg and transfer gear box 360 kg), 20 gearwheels and differential on 9 shafts and one axis controlled by seven clutches; distribution of gear is not optimal in the range equal to 14.08; the intervals are changing from 1.08 to 1.29. Ki = 16/20 =0.8 is also not high. At the diagram type 16R4, shown in Figure-4, at their optimal distribution in the increased range the same 16 gears are provided by six gearwheels of the external toothing, three planetary gears and seven clutches [3].

Transmission unit type 21R10, shown in Figure 3, has a high indicator of efficiency of the use of gearwheels. Taking into account that every PG consists of five gearwheels, we will get $Ki = 21/(2 \times 5 + 2) = 1.75$.

8. SUMMARY

The more complete use of kinematic capacities of two simple planetary gears connected in series decreases significantly the dimensions and the specific amount of metal of the transmission units, the effect intensifies due to the installation of interface reducer and additional gear box in front of the PG between the planetary gears.

9. CONCLUSIONS

The offered method of design allows creating the competitive compact transmission units of low specific amount of metal.

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