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PLANETARY GEARBOX 5R2

Oksana Olegovna Gorshkova, Vladimir Ivanovich Nekrasov
Tyumen Industrial University
Tyumen, 625000, Russia

ABSTRACT

This article discusses the existing designs of planetary gearboxes, their advantages and disadvantages. The invention relates to transport engineering, to vehicle transmissions, land vehicles, including transporting vehicles, cars, tractors. The essence of the proposed device is as follows: the planetary gearbox is comprised of a housing, simple three-link planetary mechanism consisting of sun gear, epicyclical wheel and pinion carrier with satellites engaged with the sun gear and the epicyclical wheel; the pinion carrier and the housing of the epicyclical wheel are equipped with shafts with gear rings and stopping clutch of epicyclical wheel; herewith, the input shaft has three-position gear ring with clutch. The proposed device makes it possible to decease sizes and metal consumption of vehicle transmission due to more complete use of kinematic capabilities of simple three-link planetary mechanism. The device expands vehicle specifications due to increase in the range and number of gears.

Keywords: Gearbox, Planetary Mechanism, Planetary Gearbox, Transmission, Vehicle.

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1. INTRODUCTION

The existing transmission assemblies, gearboxes with stepwise ratio variation comprised of gears, shafts supports, switchgears installed on Russian and foreign vehicles [1], [2], [3], are characterized by large dimensions and high metal consumption. Five-stage gearboxes (GB) are widely applied both upon longitudinal and transversal arrangement. They can be either three-shaft or two-shaft assemblies [4]. Disadvantages of these assemblies also consist of high metal consumption and complicated design as well as of the fact that each forward gear requires for individual gear pair and reverse gear requires for three or four gears. Gear banks increase longitudinal sizes and specific metal amount of GB. Cylindrical gears are characterized by limited load capacity due to single-pair (single-thread) engaging, and upon increase in transferred torque, it is required to increase center-to-center distance, which increases GB transversal sizes.

2. METHODS

The methods used in the article were as follows: theoretical methods (studying, systematization, analysis, synthesis of publications, patent search on the considered issue; analysis of the analyzed subject; comparison and grouping of theoretical material on the considered issue; simulation and designing of the device; generalization of the obtained results); experimental methods (mathematical data processing).

3. RESULTS AND DISCUSSION

3.1. Planet gears in GB

Decrease in specific amount of metal in vehicles can be provided by planetary transmissions characterized by small sizes and weight. This is attributed to multithreading and application of internal engagement [5].

Simple planetary mechanism (PM) is comprised of three links: sun gear (a), epicyclical wheel (b), and pinion carrier (h) with satellites. PM is characterized by the internal parameter $K = Z_b/Z_a = 1.5–5$, which equals to the ratio of the teeth number of epicyclical wheel Z_b to the teeth number of sun gear Z_a. The minimum K value is limited by the minimum sizes of satellites, and the maximum value – by that of the sun gear.

Simple PM in addition to differential is used mainly as one-stage wheel reduction gear of hub-reduction final drive (FD) of vehicle drive axle or two-stage gear–splitter at output of multistage gearbox (MGB) [4], the most closest to the proposed device. A disadvantage of such PM is that it performs only two gears: the forward one at blocking of PM due to engagement of two links: epicyclical wheel (b) and pinion carrier (h); and the slow one in the mode $U_{ah}^{b} = K+1$, where U is the ratio, superscript is the stopped link and subscripts are the links of torque input and output. When the epicyclical wheel (b) is stopped, the torque is transferred to the sun gear (a) and taken off from the pinion carrier (h).

The planetary GB (PGB) splitter is available [6] proposed for using in power drives of heavy vehicles in order to improve cross country ability under heavy road conditions. The design of GB planetary splitter excludes excessive freedom of floating link of crown gear since upon activation of lower range it is pressed to splitter case wall [7]. The splitter design makes it possible to decrease dynamic impacts and vibrations. This is achieved as follows: the splitter with planetary gear set is assembled in internal housing cavity, which is connected to GB case with sun gear installed on output shaft and linked cinematically with crown tooth wheel; it is capable to move in axial direction by means of fork which is installed in its external groove and connected with shaft of pneumatic cylinder piston upon air supply under pressure into one of sub-piston spaces and alternate engagement of wheel teeth via blocking rings which are located at both sides of satellites mounted in pinion carrier with gear rings clutches; one of which is fixed to stationary housing wall and the other is installed on continuously rotating link of the set; in the groove of housing wall, shock absorber is installed with a stud which limits movement of teeth wheel upon holding it by the fork in the position of activated reducing gear; and a damper is deformed into ring groove on external surface of teeth wheel [6]. Such design does not decrease sizes and specific metal amount of vehicle transmission assembly.

The developed transmission design, used on a vehicle instead of transmission and clutch, is simpler and more cost efficient since the transmission unit requires lower stages of tooth gear in order to provide numerous various ratios. This system is characterized by the fact that the first clutch is provided by brake, and the clutch of tooth gear stages is located between the input and the first output and formed by the planetary tooth gear comprised of at least three rotating elements, the first of which is linked with the input, the second rotating element is connected with the first output, and the third rotating element is connected with the brake. An additional variant of implementation of this transmission variant is different in that the first clutch is formed by the first clutch, and the tooth gear shift is located between the input and the first output and is formed by the main stage of the tooth gear [8].

The existing designs of MGB have been described in [9], [10], [11], [12]. The closest variant to the proposed MGB is the GB according to the RF Patent RU №2349816 [13] comprised of coaxial three-shaft GB. Seen pairs of engaged gears, a bank of reverse gears with intermediate gear on separate axis, four three-position clutches, and one two-position clutch provide 16 forward gears and 4 reverse gears. The drawbacks of the described transmission unit are in limited vehicle operation and arrangement specifications due to insufficient number of gears and ratio range, high ratios of outer gears, which leads to increase in center-to-center distance and transversal dimensions; decrease in accuracy of tooth engagements due to shaft bending at their significant length, this results in noise, decrease in efficiency and lifetime of MGB.

The design of 5R1 GB [14] is similar to that of the proposed device but provides only one reverse gear. The design of 5R2 PGB has two reverse gears.

3.2. Design 5R2 GB

Figure 1 illustrates layout of planetary gearbox (PGB) with simple three-link PM (planetary mechanism) equipped with five clutches, shafts of input and output, and stopping of three links of PM with gear rings. The clutches at the top are shown in the position of the 1-st gear, at the bottom – of the 2-nd reverse gear [15].

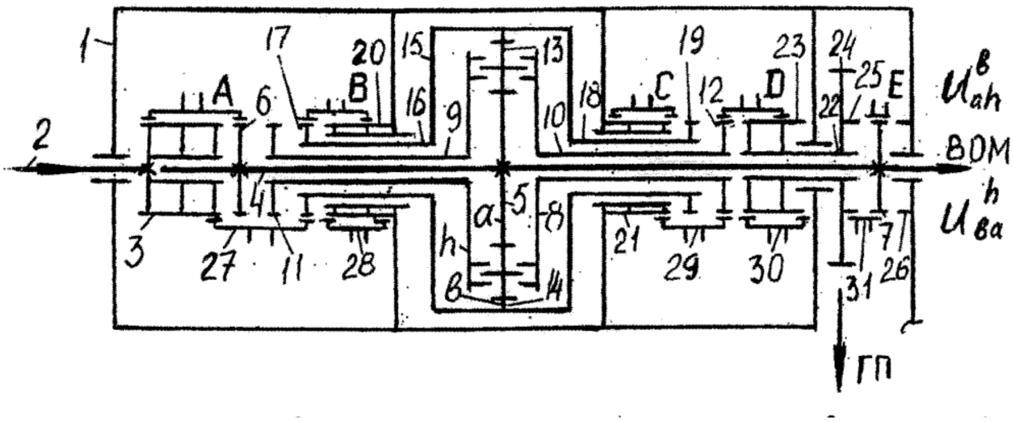

Figure 1 Schematic view of gearbox with simple three-link PM [15]

[a]1 - PGB housing; 2 - PGB input shaft; 3 - three-position gear ring of the input shaft 2; 4 - drive shaft of the sun gear 5 (a); 5 (a) - sun gear; 6 - gear ring at input of the shaft 4; 7 - gear ring at shaft output of the shaft 4; 8 (h) - pinion carrier of PM; 9 - tubular input shaft of the pinion carrier 8 (h); 10 - tubular output shaft of the pinion carrier 8 (h); 11 - gear ring of the shaft 9 at output of the pinion carrier 8 (h); 12 - gear ring of the shaft 10 at output of the pinion carrier 8 (h); 13 - satellites on axes of the pinion carrier 8; 14 (b) - epicyclical wheel of PM; 15 - housing of the epicyclical wheel 5; 16 - input coaxial shaft of the housing 15 of the epicyclical wheel; 17 - gear ring of the input coaxial shaft 16; 18 - output coaxial shaft of the housing 15 of the epicyclical wheel; 19 - gear ring of the output coaxial shaft 18; 20 - three-position gear ring of the front internal wall of the housing 1; 21 - three-position gear ring of the rear internal wall of the housing 1; 22 - primary tubular shaft of pinion gear drive; 23 -

three-position gear ring of the tubular shaft 22; 24 - main pinion gear; 25 - gear ring at output of the primary tubular shaft 22; 26 - gear ring of rear wall of the PGB housing 1 near the gear ring 7; 27 (A) - three-position clutch on the gear ring 3 of the input shaft 2; 28 (B) - clutch of teeth stopping of PM on the gear ring 20 of the front internal wall of the housing 1; 29 (C) - clutch on the gear ring 21 of the rear internal wall of the housing 1; 30 (D) - clutch at input of the primary tubular shaft 22; 31 (E) - clutch on the gear ring 7 at output of the shaft 22.

Figure 2 illustrates ray path diagram of PGB operation for K = 1.62 [15].

	U^{b}_{ha}	U^{a}_{hb}	**U = 1,0**	U^{a}_{bh}	U^{a}_{ah}	U^{h}_{ab}	U^{h}_{ba}
E	L	R	N	R	N	N	L
D	N	L	R	R	R	L	N
M	N	N	N	N	N	N	R
B	R	N	N	N	R	L	N
A	M	M	M	R	L	L	R
	5	4	3	2	1	1R	2R

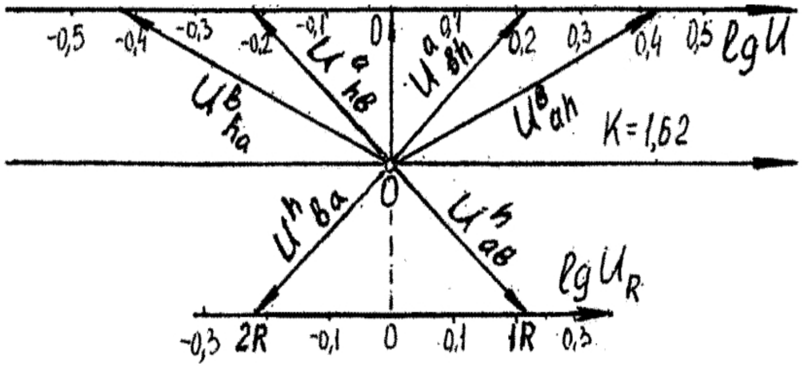

Figure 2 Ray path diagram of PGB operation [15].

Modification of the internal parameter K leads to variation of kinematic properties of PGB. At K = 1.62 we have equal segments between gears on horizontal logarithmic scale where PM ratios are plotted. The vertical ray indicates at direct transfer of torque from engine to FD gear. The ray to the right characterizes decelerating mode of PM operation: the steeper is the ray, the higher is the ratio. The ray to the left upwards characterizes accelerating mode of PM operation: the steeper is the ray, the lower is the ratio. The ray path diagrams are useful for understanding of MGB operation, since the value of each ray is constant at all diagram segments. Above the ray path diagram, the table summarizes positions of clutches at various gears. The PM states at each gear are shown at the top of the table.

In the supports of the PGB assembly 1, the input shaft 2 is located coaxially with the mounted three-position gear ring 3 and the shaft 4 of sun gear drive 5 (a) with the gear rings 6 and 7 at input and output of this shaft. The output of the shaft 4 can be used as power take-off shaft. Tubular shafts of the pinion carrier 8 (h) are installed on the shaft 4: the input shaft 9 and the output shaft 10 with the gear rings 11 and 12 at the input and output of these shafts. On the axes of the pinion carrier 8, the satellites 13 are mounted engaged with the sun gear 5 (a) and the epicyclical wheel 14 (b). At the input of the casing 15 of the epicyclical wheel 14 (b) the input coaxial shaft 16 is mounted with the gear ring 17, and at the output – the output coaxial shaft 18 with the gear ring 19. Above the input coaxial shaft 16 on the front internal wall of the housing 1 near the gear ring 17, the three-position gear ring 20 is located. Above the output coaxial shaft 18 on the rear internal wall of the housing 1 near the gear ring 19 the three-position gear ring 21 is located. On the shaft 4 near the gear ring 12 the drive tubular shaft 22 is positioned with the three-position gear ring 23 at the input of this shaft and with the FD gear 24 with the gear ring 25 at the output of the drive shaft 22. The FD can be cylindrical, conical, hypoid. Against the gear ring 25 near the gear ring 7 of the shaft 4 on the rear wall of the housing 1 the gear ring 26 is located. The three-position clutch 27 (A) is mounted on the gear ring 3 of the input shaft 2 for selective engagement between the gear ring 6 and the drive of the sun gear 5 (a), between the gear ring 11 and the drive of the pinion carrier 8 (h), between the gear ring 17 of the drive and the epicyclical wheel 14 (b). The clutch 28 (B) is mounted on the gear ring 20 of the front internal wall of the housing 1 for stopping the pinion carrier 8 (h) by the gear ring 11 or the epicyclical wheel 14 (b) by the gear ring 17, neutral (N) – deactivated state is provided by placement of the clutch between the gear rings 11 and 17 or its transfer to the wall. The clutch 29 (C) is mounted on the gear ring 21 of the rear internal wall of the housing 1 for stopping the pinion carrier 8 (h) by the gear ring 12 or the epicyclical wheel 14 (b) by the gear ring 19, neutral (N) – deactivated state is provided by placement of the clutch between the gear rings 12 and 19 or its transfer to the wall. The clutch 30 (D) is mounted at the input of the primary tubular shaft 22 for selective engagement with the gear ring 19 of the output coaxial shaft 18 of the epicyclical wheel 14 (b) or with the gear ring 12 of the output tubular shaft 10 of the pinion carrier 8 (h), neutral (N) is provided by placement of the clutch between the gear rings 19 and 12. The clutch 31 (E) is mounted on the gear ring 7 at the output of the shaft 4 of the sun gear 5 (a) for the drive of the pinion gear 24 upon engagement with the gear ring 25 or stopping the sun gear 5 upon engagement with the gear ring 26 of the rear wall of the housing 1.

Simple three-link PM can provide five gears without variation of rotation direction, two reverse gears and three gears in integral (summing) modes when the torque is supplied to two PM links at various rotation frequency and is taken-off from the third link.

3.3. Operation of 5R2 PGB

The device operates as follows.

1st gear. In the ray path diagram (Fig. 2) this mode is shown by flat ray from point 0 upwards to the right to point 1. Above this point the clutch states are shown (clutch upper position): A - left (L), B and D - right (R), C and E - neutral (N). The clutches B and C are partially duplicated. Possible states are as follows: clutch B – neutral (N); clutch C – left (L). In the top of the table the PM state Ua_h^b = K+1 is shown .

The epicyclical wheel 14 (b) is stopped by the clutch 28 (B) which engaged the gear rings 20 of the front internal wall of the housing 1 and 17 of the tubular shaft 16 of the housing 15 of the epicyclical wheel 14. The torque (Fig. 1) from the gear ring 3 of the input shaft 2 by the shaft 27 (A) is transferred to the gear ring 6 and by the shaft 4 to the sun gear 5 (a). The sun gear 5 (a) rotates the satellites 13, which, rolling across the epicyclical wheel 14 (b), rotate the pinion carrier 8 (h) at decreased frequency but with increased torque. Via the tubular shaft 10, the gear ring 12, the clutch 30 (D), the torque is transferred to the gear ring 23, to the drive tubular shaft 22 and the drive gear 24. At K = 1.62 the ratio is U_{ah}^b=K+1=2.62; lq 2.62=0.42.

2nd gear. In the ray path diagram this mode is shown by steep ray from point 0 upwards to the right to point 2. Now the clutch A is switched from the left position to the right position (R), the clutch B - from the right position to the neutral (N), deactivated, position to the wall of the housing 1, and the clutch E - from the neutral to the right (R) position. In the top of the table the U_{bh}^a state of PM is shown. In comparison with the 1st gear the links are changed: the sun gear 5 (a) is stopped, torque is transferred to the epicyclical wheel 14 (b). The sun gear 5 (a) is stopped by the clutch 31 (E) which engaged the gear rings 26 of the rear wall of the housing 1 and 7 of the shaft 4 (Fig. 1, right side). The torque from the ring gear 3 of the input shaft 2 via the clutch 27 (A) is transferred to the ring 17 and via the tubular shaft 16 via the housing 15 to the epicyclical wheel 14 (b) which rotates the satellites 13, they rolling across the sun gear 5 (a) rotate the pinion carrier 8 (h), then, as in the case of the 1st gear with reduced frequency but with increased torque. At K = 1.62 the ratio is $U_{bh}{}^{a} = (K + 1)/K = 2.62/1.62 = 1.62$; lq 1.62 = 0.21.

3rd forward gear. In the ray path diagram this gear is shown by vertical ray from point 0 to point 3. The clutch A is switched from the right position to the middle (M) position, the clutch E - from the right to neutral (N) position. The torque from the gear ring 3 via the clutch 27 (A) is transferred to the gear ring 11 of the tubular shaft 9 of the pinion carrier 8, via the pinion carrier 8 and further as for the 2nd gear.

4th gear. In the ray path diagram this mode is shown by steep ray from point 0 upwards to the left to point 4. The clutch D is switched from the right to the left (L) position, the clutch from the neutral to the right (R) position, herewith, in comparison with the 2nd gear the teeth of torque input and output are changed. The sun gear 5 (a) is stopped by the clutch 31 (E). The torque to the pinion carrier 8 (h) is transferred as in the 3rd gear which rotates the satellites 13, they rolling across the stopped sun gear 5 (a) rotate the epicyclical wheel 14 (b), via the housing 15 to the coaxial shaft 18, the gear ring 19, via the clutch 30 (D) to the gear ring 23 and then as in the previous gears but with increased frequency and reduced torque. At K = 1.62 the ratio is $U_{hb}{}^{a}= K/(K+1) = 1.62/2.62 = 0.62$; lq 0.62 = –0.21.

5th gear. In the ray path diagram this mode is shown by flat ray from point 0 upwards to the left to point 5. The clutch B is switched from the neutral to the right (R) position, the clutch D - from the left to the neutral (N) position, the clutch E - from the right to the left (L) position; in comparison with the 1st gear, the teeth of torque input and output are changed. The epicyclical wheel 14 (b) is stopped by the clutch 28 (B) which engaged the gear rings 20 of the front internal wall of the housing 1 and 17 of the coaxial shaft 16 of the housing 15 of the epicyclical wheel 14. The torque (Fig. 1) from the gear ring 3 of the input shaft 2 via the clutch 27 (A) is transferred to the pinion carrier 8 (h) as in the 3rd gear which rotates the satellites 13, they rolling across the stopped epicyclical wheel 14 (b) rotate the sun gear 5 (a), and then via the shaft 4 to the gear ring 7, via the clutch 31 (E) to the gear ring 25 and the drive wheel 24, but with increased frequency and decreased torque. At K = 1.62 the ratio is $U_{ha}{}^{b} = 1/\{K + 1) = 1/2.62 = 0.38$; lq 0.38 = –0.42.

1st reverse gear 1R. In the ray path diagram (Fig. 2) this mode is shown by steep ray from point 0 downwards to the right to point 1R. The clutch states are shown in the second right column of the table: A, B and D – left (L), C – neutral (N). In the top of the table the $U_{ab}{}^{h} = -K$ state of PM is shown. The minus sign indicates at variation of rotation of the pinion gear 24 in comparison with the drive shaft 2. In comparison with the 1st forward gear the states of output and stopped links of PM were changed. The pinion carrier 8 (h) was stopped by the clutch 28 (B) which engaged the gear rings 20 of the front internal wall of the housing 1 and 11 of the tubular input shaft 9. The torque (Fig. 1) from the gear ring 3 of the input shaft 2 via the clutch 27 (A) is transferred to the gear ring 6 and via the shaft 4 to the sun gear 5 (a). The sun gear 5 (a) rotates the satellites 13 which rotating on the axes of the pinion carrier 8 (h) rotate the epicyclical wheel 14 (b) in reverse direction with decreased frequency but increased torque. Then, as in the 4th forward gear, via the coaxial shaft 18, the gear ring 19, the clutch 29 (D) the torque is transferred to the gear ring 23, to the drive tubular shaft 22 and the drive gear 24. At K = 1.62 the ratio is $U_{ab}{}^{h} = K = 1.62$; lq 1.62 = 0.21.

2nd reverse gear 2R. In the ray path diagram this mode is shown by steep ray from point 0 downwards to the left to point 2R. The clutch states are shown in the outmost right column of the table: A and C – right (R), B and D – neutral (N), E – left (L). In the top of the table the $U_{ba}{}^{h} = -K$ state of PM is shown. The minus sign indicates at variation of rotation of the pinion gear 24 in comparison with the drive shaft 2. In comparison with the 1st reverse gear the states of input and output links of torque were changed. The pinion carrier 8 (h) is stopped by the clutch 29 (C) which engaged the gear rings 21 of the rear internal wall of the housing 1 and 12 of the tubular output shaft 10. The torque from the gear ring 3 of the input shaft 2 via the clutch 27 (A) is transferred to the gear ring 17 and via the coaxial shaft 16 to the housing 15 and the epicyclical wheel 14 (b). The epicyclical wheel 14 (b) rotates the satellites 13 which, rotating on the axes of the pinion carrier 8 (h), rotate the sun gear 5 (a) in reverse direction with increased frequency but with decreased torque. Further, as in the 5th forward gear, via the shaft 4, the gear ring 7, the clutch 31 (E) the torque is transferred to the gear ring 25 and the drive gear 24. At K = 1.62 the ratio is $U_{ba}^{h} = 1/K = 1/1.62 = 0.62$; lq 0.62 = –0.21. The PGB range will be $D = U_1/U_5 = 2.62/0.38 = 6.9$. Variation of K will vary the kinematic properties of PGB.

4. CONCLUSION

PGB consists of the housing of gearbox, simple three-link PM comprised of the sun gear, the epicyclical wheel and the pinion carrier with satellites engaged with the sun gear and the epicyclical wheel; the pinion carrier and the epicyclical wheel housing have shafts with gear rings and stopping clutch of epicyclical wheel characterized in that the input shaft has three-position gear ring on which the clutch is installed; the sun gear shaft with gear rings at the edges is installed in the support of pinion gear ring and in the support of rear wall of gearbox housing with gear ring; the pinion carrier is fixed on input and output tubular shafts with gear rings; the epicyclical wheel housing is fixed on input and output coaxial shafts with gear rings; on internal walls of gearbox housing near the gear rings of the input shafts the gear rings are fixed with stopping clutches of pinion carrier or epicyclical wheel; near the gear rings of the output shafts the primary tubular shaft is installed with gear ring at shaft input and driving gear with gear ring at output of this shaft near gear ring of the sun gear shaft with clutch.

The proposed device relates to transport engineering, vehicle transmissions, land vehicles, including transporting vehicles, cars, tractors, etc.

The obtained advantage is comprised of decreased sizes and metal consumption of vehicle transmission assembly due to more complete use of kinematic capabilities of simple three-link PM.

Therefore, we obtained compact transmission assembly with low specific metal amount due to significant decrease in longitudinal dimension, thus creating better conditions for transversal arrangement of engine and transmission. The proposed variant provides five forward gears and two reverse gears with sufficient range (2.62/0.38=6.9) together with decrease in sizes and specific metal amount in comparison with typical designs.

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