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Compact Transmission Assembly of Low Metal Intensity

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ABSTRACT

This article discusses the existing designs of planetary gearboxes, their functional peculiarities, fields of application; comparative analysis of the existing units is presented, the advantages and disadvantages of each design are determined. The design of compact transmission assembly of low metal intensity is presented in the form of 5R1 planetary gearbox. Operation of the compact transmission assembly of low metal intensity in various modes is described; kinematic flowchart of the assembly is presented. It has been proved that the proposed gearbox design reduces its metal intensity, decreases its weight and sizes due to application of three-link planetary gear train allowing to expand kinematic capabilities of the design. The assembly increases the number and range of gears aiming at improvement of its specifications, the assembly can be used in transport facilities and vehicles.

Key words: compact transmission assembly, low metal intensity, planetary gear train, gearbox.

1. INTRODUCTION

Nowadays numerous designs of gearboxes are developed with planetary reduction gears which are installed on various vehicles. A drawback of the existing designs is high metal intensity, large dimensions [1]. Modern design of gearbox, based on sufficient ratios, should improve vehicle efficiency in the preset range of its traction forces as well as its economic efficiency by rational selection of ratios.

Five-stage three- or two-shaft gearboxes (GBs) are widely applied with longitudinal and transversal arrangement of transmission [4]. They are free from such disadvantages as complicated design when each forward gear requires for individual gear pair and reverse gear requires for several gears (3-4), which leads to high dimensions of GB and high metal intensity. Single-pair (single-thread) engagement of cylindrical gears determines low load capacity, which, in its turn, leads to increase in center-to-center distance during increase in transferred torque. In order to eliminate the mentioned drawbacks, it would be reasonable to apply planetary gears, which facilitate multithreading and application of internal engagement [2].

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 considered 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 Planetary Gears in GBs

Simple planetary gear train is comprised of three links: sun gear (a), epicyclical wheel (b), and pinion carrier (h) with satellites. Planetary gear train is characterized by the internal parameter equaling to the ratio of the teeth number of epicyclical wheel Zb to the teeth number of sun gear Za: K = Zb/Za = 1.5–5. The minimum value of K is limited by the minimum sizes of satellites, and the maximum value – by that of the sun gear.

Simple planetary gear train is used mainly as one-stage wheel reduction gear of hub-reduction final drive of vehicle drive axle or two-stage gear–splitter at output of multistage GB [5]. This design is the most similar to the proposed design. A disadvantage of such planetary gear train is that it performs only two gears: the forward one during blocking of planetary gear train due to engagement of two links: epicyclical wheel (b) and pinion carrier (h); and the slow one in the mode Uahb = 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 existing planetary splitter of GB [6] applied in heavy duty vehicles as power drive improves cross-country ability under heavy road conditions. The design of the planetary splitter excludes excessive freedom of floating link of crown gear since upon activation of lower range, it is pressed to splitter case wall. The planetary set used in the splitter decreases dynamic impacts and vibrations; however, it does not decrease dimensions and metal intensity of vehicle transmission assembly [6].

The existing designs of multistage GBs have been described in [7], [8], [9], and others. The GB according to the RF Patent RU №2349816 [10] is the most similar variant to the proposed design. It is comprised of coaxial three-shaft GB, seven pairs of engaged gears, a bank of reverse gears with intermediate gear on separate axle, four three-position clutches, and one two-position clutch that provide 16 forward gears and 4 reverse gears. The drawbacks of the described transmission assembly are 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, as well as to decrease in accuracy of tooth engagements due to shaft bending at their significant length, this results in noise, decrease in efficiency and lifetime of multistage GB.

The 5R1 planetary GB has one reverse gear. The 5R1 design of compact transmission assembly of low metal intensity is similar to the 5R2 GB described in RF Patent № 2621214 [11], which provides two reverse gears [3]. Herewith, the 5R1 planetary GB provides five forward gears and one reverse gear with sufficient range (D = 2.62/0.38 = 6.9) together with decreased dimensions and metal intensity in comparison with typical designs.

3.2 5R2 Planetary GB

The 5R1 planetary GB is equipped with simple three-link planetary gear train (PGT) comprised of four clutches, the shafts of input, output, and stopping of three PGT links with ring gears. The simple three-link PGT, comprised of sun gear, epicyclical wheel, and pinion carrier with satellites engaged with the sun gear and 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. The flowchart of 5R1 planetary GB is illustrated in Fig. 1. The clutches at the top are shown in the position of the 1-st gear, at the bottom in the position of the 4-th gear [12].

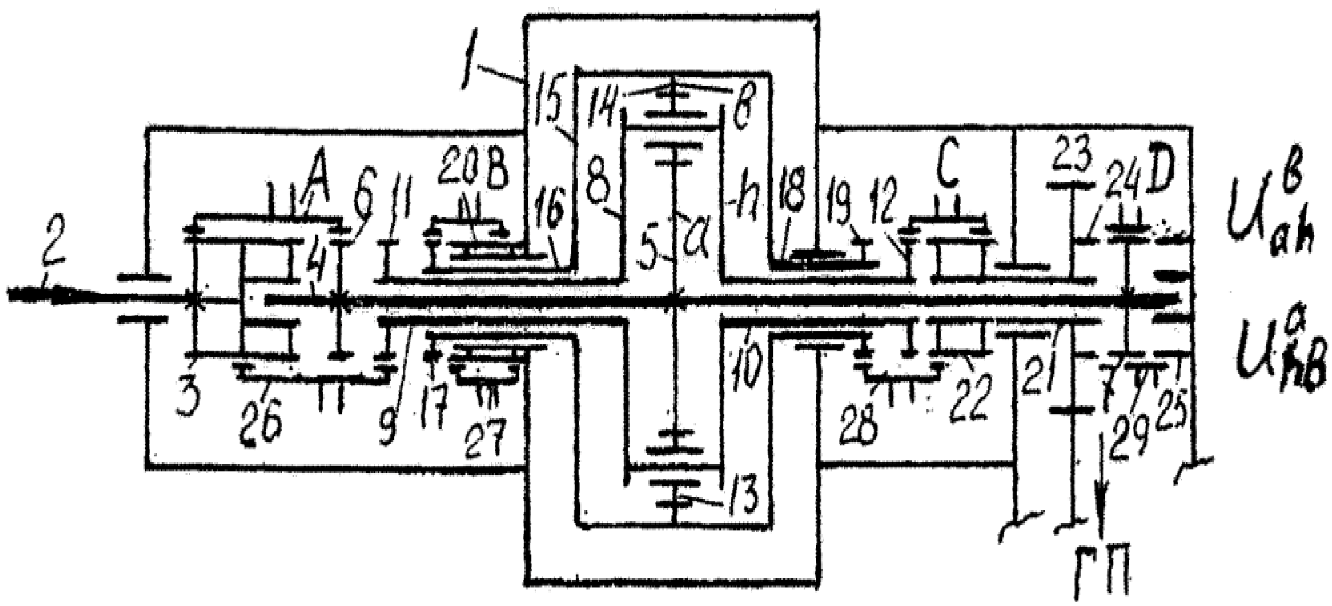

Figure 1. Schematic view of GB with simple three-link 5R1 PGT[12]

1 - assembly housing of PGB (planetary GB); 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 output of the shaft 4; 8 (h) - pinion carrier of planetary gear train; 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 axles of the pinion carrier 8; 14 (b) - epicyclical wheel of PGT; 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 - tubular drive shaft of driving gear; 22 - three-position gear ring of tubular drive shaft 21; 23 - driving gear of MG (main gear); 24 - gear ring at output of the tubular drive shaft 21; 25 - gear ring of the rear wall of the PGB housing 1 near the gear ring 7; 26 (A) - three-position clutch on the gear ring 3 of the input shaft 2; 27 (B) - stopping clutch of PGT links on the gear ring 20 of the front internal wall of the housing 1; 28 (C) - clutch on the input of the tubular drive shaft 21; 29 (D) - clutch on the gear ring 7 at the output of the shaft 21.

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 axles 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 16 on the rear internal wall of the housing 1 near the gear ring 17, the three-position gear ring 20 is located. On the shaft 4 near the gear ring 12, the drive tubular shaft 21 is positioned with the three-position gear ring 22 at the input of this shaft and with the MG 23 with the gear ring 24 at the output of the drive shaft 21. The MG can be cylindrical, conical, hypoid. Against the gear ring 24 near the gear ring 7 of the shaft 4 on the rear wall of the housing 1 the gear ring 25 is located. The three-position clutch 26 (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 27 (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 28 (C) is mounted at the input of the tubular drive shaft 21 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) state is provided by placement of the clutch between the gear rings 19 and 12. The clutch 29 (D) is mounted on the gear ring 7 at the output of the shaft 4 of the sun gear 5 for the drive of the pinion gear 23 upon engagement with the gear ring 24 or stopping the sun gear 5 upon engagement with the gear ring 26 of the rear wall of the housing 1.

Simple three-link PGT 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 links of PGT at various rotation frequency and is taken-off from the third link.

3.3 Operating Principle of 5R1 PGB

First gear. The epicyclical wheel 14 (b) is stopped by the clutch 27 (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 26 (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 28 (C), the torque is transferred to the gear ring 22, to the drive tubular shaft 21 and the drive gear 23. At K = 1.62, the ratio is Uahb=K+1=2.62; lq 2.62=0.42.

Second gear. 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). In comparison with the first 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 29 (D) which engaged the gear rings 25 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 26 (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 Ubha = (K + 1)/K = 2.62/1.62 = 1.62; lq 1.62 = 0.21.

Third gear – forward gear. The clutch A is switched from the right position to the middle (M) position, the clutch D – from the right to neutral (N) position. The torque from the gear ring 3 via the clutch 26 (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.

Fourth gear. The clutch D is switched from the right to the left (L) position, the clutch E – from the neutral to the right (R) position; herewith, in comparison with the second gear, the teeth of torque input and output are changed. The sun gear 5 (a) is stopped by the clutch 29 (D). 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 28 (C) to the gear ring 22 and then as in the previous gears but with increased frequency and reduced torque. At K = 1.62, the ratio is Uhba= K/(K+1) = 1.62/2.62 = 0.62; lq 0.62 = –0.21.

Fifth gear. The clutch B is switched from the neutral to the right (R) position, the clutch C – from the left to the neutral (N) position, the clutch D – from the right to the left (L) position; in comparison with the first gear, the teeth of torque input and output are changed. The epicyclical wheel 14 (b) is stopped by the clutch 27 (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 26 (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 29 (D) to the gear ring 24 and the drive wheel 23, but with increased frequency and decreased torque. At K = 1.62, the ratio is Uhab = 1/{K + 1) = 1/2.62 = 0.38; lq 0.38 = –0.42.
1R Reverse gear. The clutches A, B, and C are in the left (L) position, the clutch D is in neutral (N) position, Uabh=-K is the state of PGT. The minus sign indicates at variation of rotation of the pinion gear 23 in comparison with the drive shaft 2. In comparison with the first forward gear, the states of output and stopped links of PGT change. The pinion carrier 8 (K) is stopped by the clutch 27 (B) which is engaged by 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 26 (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 fourth forward gear, via the coaxial shaft 18, the gear ring 19, the clutch 28 (C) the torque is transferred to the gear ring 22, to the drive tubular shaft 21 and the drive gear 23. At K = 1.62, the ratio is Uabh = K = 1.62; lq 1.62 = 0.21. The GB range is D = U1/U5 = 2.62/0.38 = 6.9. Variation of K will vary the kinematic properties of PGB.

4. CONCLUSION

The proposed design of 5R1 PGB is a compact transmission assembly with reduced sizes and weight (due to significant decrease in longitudinal size), thus creating conditions for better transversal arrangement of engine and transmission. The 5R2 PGB design reduces metal intensity of vehicle transmission assembly. This is achieved by more complete use of kinematic capabilities of simple three-link PGT, which provides five forward gears and one reverse gear with sufficient range.

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