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Review

THE INFLUENCE OF THE INTERNAL PARAMETER OF THE SIMPLE PLANETARY GEAR ON ITS CHARACTERISTICS

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ABSTRACT

This paper considers the influence of internal parameter K of a simple three-link planetary gear (PG) on its kinematic characteristics-the distribution of gear ratios over the range. The authors propose the methodology of selecting the value of K parameter when designing gear ratios of multi-stage gearboxes (MSG) with low specific quantity of metal.

INTRODUCTION

Planetary gears (PG) are widely used in engineering, including the shaft line of land transport vehicles (LTV) (Nekrasov, 2001). Design of power planetary multi-stage gearboxes is associated with difficulties when selecting internal parameter (K) of planetary gear, which will ensure the optimal distribution of the gear ratios within the range of multi-stage gearbox.

A simple planetary gear

A simple three-link planetary gear (PG) consists of pinion carrier (h) (1) with satellites (2), which are engaged with a sun gear (a) (3) and ring gear (b) (4) (Fig. 1) (Nekrasov, *et al*., 2016).

The ratio of the number of teeth of the ring gear Zb (4) to the number of teeth of sun gear Za (3) characterizes the internal parameter K=Zb/Za=1.5-5. The planetary gear (PM) with K=1.5 is presented in Fig. 1 to the left-the limitation is caused by the minimum sizes of satellites (2), the PG with K=5 is shown in Fig. 1 to the right-the limitation is caused by the small diameter of sun gear (a) (3).

Raypath plots (Fig. 2-4) provide a visual representation of the gear ratio values of the PG at different gears (Nekrasov, *et al*., 2016). When changing the value of the internal parameter K, the position of the half-lines in the plot changes noticeably (Nekrasov, 2001).

The gear ratio values are shown for three raypath plots with different K (from top to bottom: 1.62; 1.5, and 5.0) on three horizontal scales in Fig. 2: in natural form (U)-in the upper part, and in logarithmic form (lg U)-in the lower part. Half-lines directed to the right indicate reducing gears; the more sloping is the half-line the greater the gear ratio. The vertical half-line characterizes the gear with U=1.0. Half-lines directed to the left indicate overdriving gears; the more sloping is the half-line the less gear ratio.

Planetary gear statuses are indicated on the half-lines: U_{ah}^{b}, U_{bh}^{a}, U_{hb}^{a}, U_{ha}^{b}, U_{ab}^{h}, U_{ba}^{h}. The superscript refers to the stopped link, while subscripts refer to the input and output links of the driving torque. For example, at the first gear U_{ah}^{b}, ring gear (b) (4) is stopped, the driving torque is applied to the sun gear (a) (3) and is taken off from the pinion carrier (h) (1).

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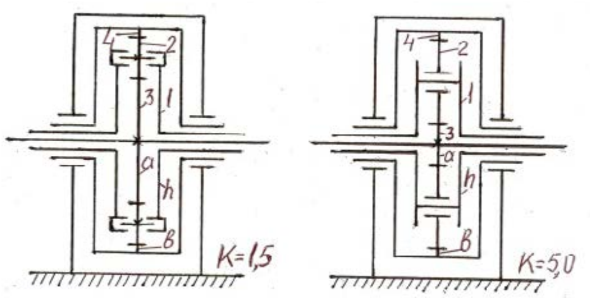

Fig. 1 Schematics of simple three-link planetary gears with different internal parameters: left K = 1.5; right K=5: 1-opinion carrier (h); 2-satellites; 3-sun gear (a); 4-ring gear (b) Raypath plots of the planetary gears

Maximum gear ratio is provided in the mode U_{ah}^{b}=K+1. For the lower raypath plot Uahb=5+1=6; lg 6=0.78; gear is presented by a sloping half-line directed upwards to the right towards the point 1. Raypath plot for K=1.5 is inverted and the half-line U_{ah}^{b}=1.5+1=2.5; lg 2.5=0.4 is directed downwards to the right. A dashed line for different K is plotted to the right between the values of 0.4 and 0.78.

The minimum gear ratio is provided in the mode of U_{ha}^{b}=1/(K+1), where compared to the first gear, the links of input and output torques are replaced. For the lower raypath plot U_{ha}^{b}=1/(5+1)=1/6=0.167; lg 0,167=- 0.78, and gear is presented by a sloping

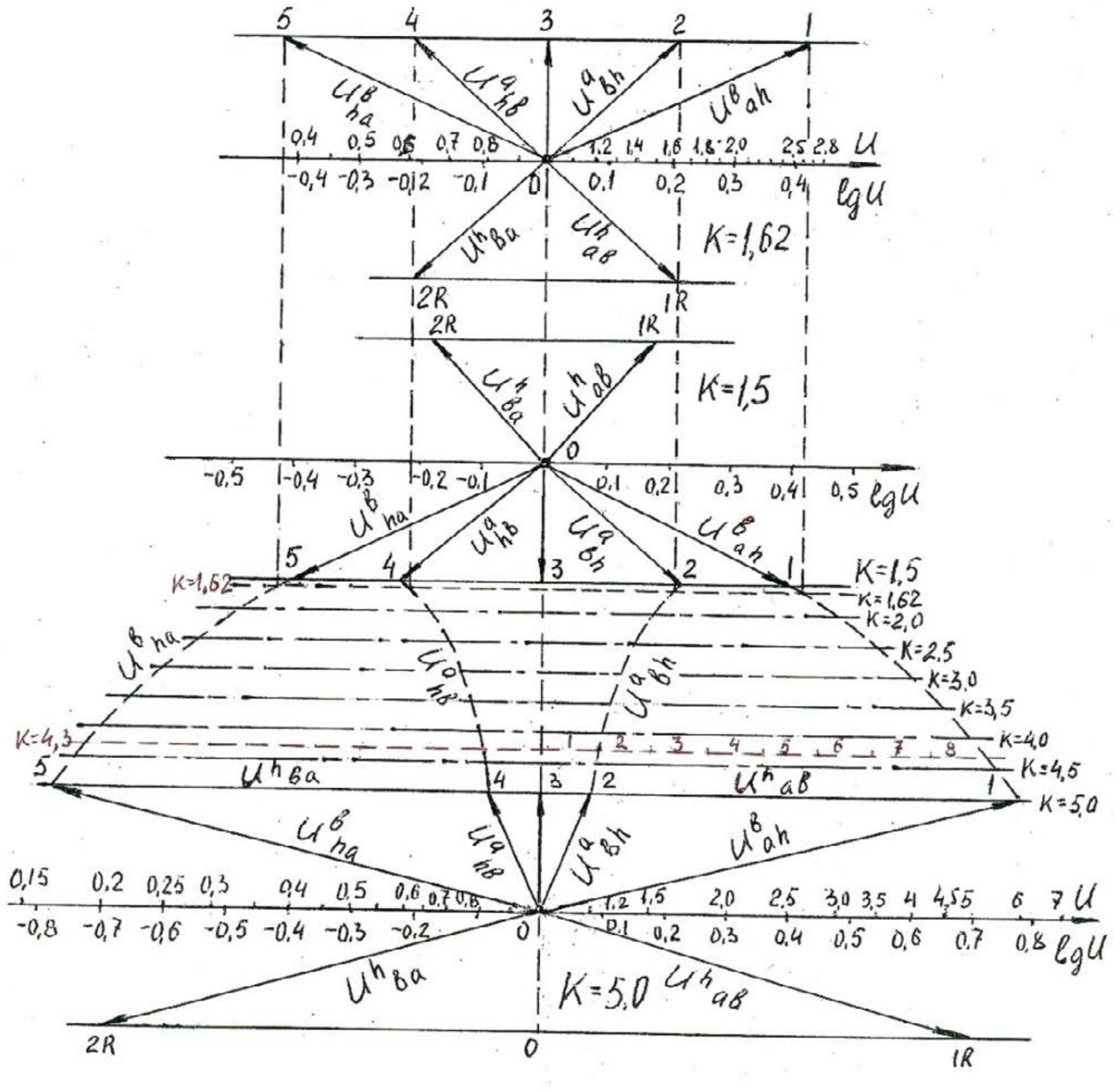

Fig. 2 Raypath plots for different internal parameters of the PG

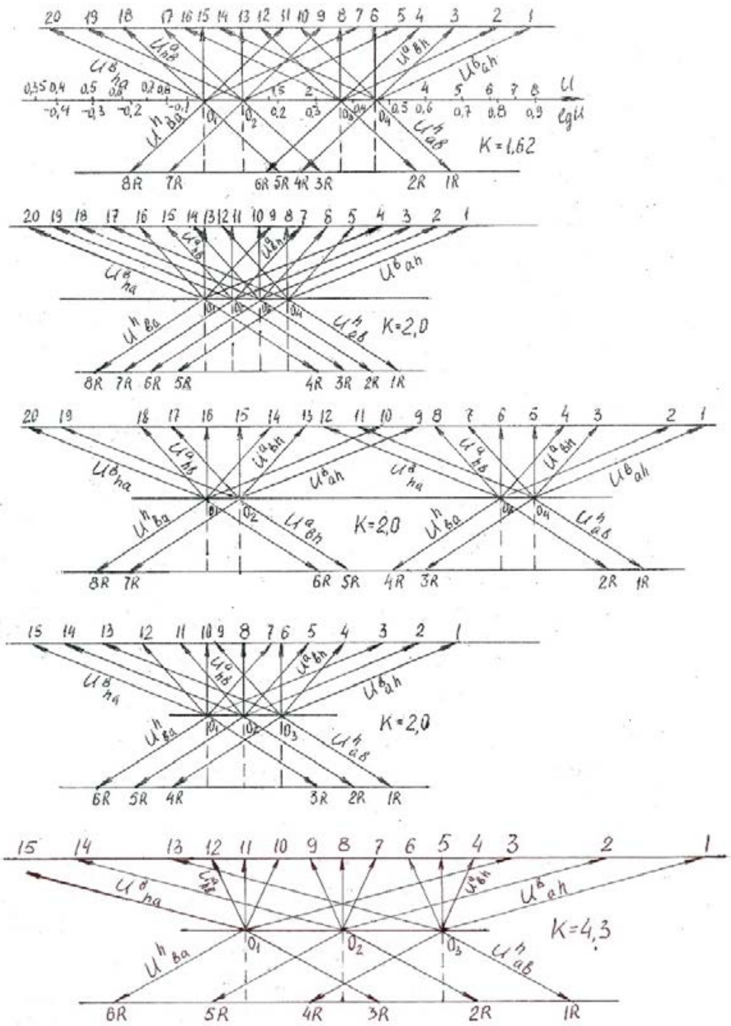

Fig. 3 Raypath plots of the MSG for different internal parameters of the PG.

half-line directed upwards to the left towards the point 5. The raypath plot for K=1.5 is as follows: half-line $U_{ha}^{b}=1/(1.5+1)=½.5=0,4$; lg 0.4=-0.4 is directed downwards to the left towards the point 5. The dotted line for different K is plotted in the left between values of -0.4 and -0.78. We obtained the mirroring line for U_{ah}^{b} and domed cross-section for the two considered conditions of the PG.

Steep half-line upwards to the right towards the point 2 represents condition $U_{bh}^{a}=(K+1)/K$. Sun gear (a) (3) is stopped, input is applied on ring gear (b) (4), while output is taken off from pinion carrier (h) (1). For K=5, the gear ratio $U_{bh}^{a}=6/5=1.2$; lg 1.2=0.08. For K=1.5, the gear ratio $U_{bh}^{a}=2.5/1.5=1.67$; lg1.67=0.22.

Replacing the input and output driving torque links we receive a following PG status $U_{hb}^{a}=K/(K+1)$. For K=5, the gear ratio is $U_{hb}^{a}=5/6=0.833$; lg 0.833=- 0.08; steep half-line is directed upwards to the left towards the point 4. For K=1.5 gear ratio $U_{hb}^{a}=1.5/2.5=0.6$; lg 0.6=-0.22. Between points 0.08 and -0.08 at the bottom and points 0.22 and -0.22 at the top we have obtained two symmetrical divergent lines, characterizing the operation of the PG at the stopped sun gear. The vertical half-line in the point 3 indicates the gear ratio U=1.0.

Reverse gears are formed at stopped pinion carrier (h) (1). For K=5 gear ratio $U_{ab}^{h}=-K=-5$; lg 5=0.7. The sign "-" indicates the change in rotation direction of the link at the output of THE PG. Slow reverse gear is represented by the U_{ab}^{h} half-line directed downwards to the right to the point 1R. For K=1.5, gear ratio $U_{ab}^{h}=-1.5$; lg 1.5=0.176. The half-line is directed upwards to the right to the point 1R.

Replacing links of the driving torque input and output we receive the following status of the PG: $U_{ba}^{h}=-1/K$. For K=5, gear ratio $U_{ba}^{h}=-1/5=0.2$; lg 0.2=-0.7. The half-line is directed downwards to the left to the point 2R. For K=1.5 gear ratio $U_{ba}^{h}=-1/1.5=0.667$;

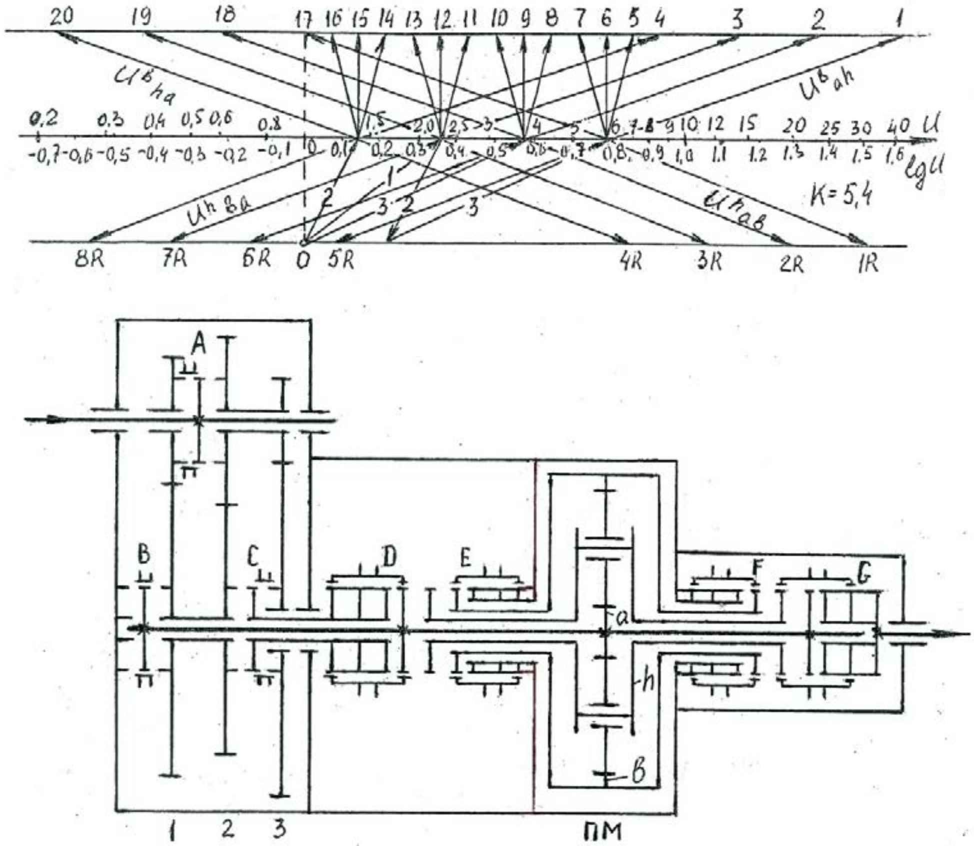

Fig. 4 Raypath plot and diagram of type 20R8 MSG based on the patent RU No. 2047506.

lg 0.667=-0.176. The half-line is directed upwards to the left to the point 2R.

Analysis of half-lines characterizing different conditions of the PG indicates to the fact that reverse gear ratios are equal to the difference between points 1-2 and 4-5.

$U_{ab}^{h}=U_{ah}^{b}-U_{bh}^{a}$. For example, for K=5: $U_{ab}^{h}=0.78-0.08=0.7$.

The upper part of Fig. 2 shows the raypath plot of the PG with the internal parameter K=55/34=1.62. A distinctive feature of this raypath plot lies in the fact that the segments intercepted by half-lines on the horizontals, are equal lg (K+1)=lg 2.62=0.42-the point 1. This segment is divided in half by the point 2: lg (K+1)/K=lg 2.62/1.62=lg 1.62=0.21, etc. These segments 0.21 are projected by the dotted lines to a general dome-shaped dependence of the segment values on K. The dotted line below the line K=1.5 shows the line for K=1.62. It is clearly seen that the segments for the different conditions of the PG are equal.

Raypath plots of the planetary gears with the gearbox input

Fig. 3 shows raypath plots with different values of the internal parameter K: 1.62; 2.0; and 4.3. The top three plots for the four-stage gearboxes at the input to the PG are denoted by points 01, 02, 03 and 04. Eventually we obtain the MSG of type 20R8: at the upper horizontals, there are 4 × 5=20 forward gears, while at the lower horizontals 4 × 2=8-reverse gears: 1R-8R. The bottom two plots for three-stage input gearboxes correspond to the MSG of type 15R6.

The distribution of gear ratios within the range of the multi-stage gearbox (MSG) is not a simple task. On the top plot for K=1.62 we obtained equal segments of 0.21/2=0.105. The interval between neighboring gears $q=U_1/U_2=8.0/6.3=1.27$ that will provide for quite comfortable gear shift. Though, intervals in the middle part of the plot between the 4th and 17th gears are small. A range of the multi-stage gearbox, equal to the relative gear ratios of lower and higher gears will be $D=U_1/U_{20}=8/0.38=21$ that is quite sufficient for modern trucks.

The transmission of four-wheel drive vehicles often includes the PG with K=2.0 used as an interaxle differential. This kind of the PG can be used as a reducer at the MSG output. Some of gear ratio distributions over the range are shown in three diagrams in the middle part of Fig. 3 for K=2.0.

Closely located gears of four-stage gearbox (points 0_1, 0_2, 0_3 and 0_4 on the middle horizontal) significantly reduce the range of the MSG of type 20R8, which is $D=U_1/U_{20=}5.2/0.33=15.76$.

The middle diagram shows a case with almost equal intervals for most gears that is achieved by pairwise input to the PG from the GB (points 0_3 and 0_4 are located at a considerable distance from points 0_1 and 0_2). Here a very large range of gear ratios is obtained, though a significant interval (more than 5) between the 2nd and 3rd gears of the GB complicates its design and leads to an increase in the overdriving gears (U ‹ 1) that is undesirable.

The third option with K=2.0 for the MSG of type 15R6 as compared to the first option (MSG of type 20R8) when increasing the interval to q=1.26 slightly reduces the range from D=15.76 down to D=4.7/0.33=14.24. This is not enough for modern heavy trucks.

It is desirable to have the MSG with small equal intervals between the gears providing comfortable gear shift.

Equal intervals can be provided subject to the equality

lg (K+1)=z lg (K+1)/K

Where z=3 n-1; n-is the number of gears of the GB at the PG input.

For the lower plot in Fig. 3, n=3; z=3 × 3-1=8 (see the bottom part of generalizing diagram in Fig. 2.

Equality is respected under the condition K=4.3.

lg 5.3=8 lg (5.3/4.3); 0.724=8 × 0.091=0.726 ≈ 0.724.

We have obtained equal intervals between gears 3 and 13 with q=1.23 that is convenient to shift gears. Two starting gears can be used for slow motion when maneuvering in limited spaces at cargo operations.

Raypath plot and diagram of the multistage gearbox of type 20R8

Raypath plot with equal intervals between 4th and 17th gears and the diagram of the MSG of 20R8 type, based on the patent RU N° 2047506, is presented in Fig. 4. The scale of gear ratios is compressed as compared with the previous diagrams.

For the diagram in Fig. 4, n=4; z=4 × 3-1=11.

Equality is respected under the condition K=5.4.

lg 6.4=11 lg (6.4/5.4); 0.806=11 × 0.0738=0.812 ≈ 0.806.

Three pairs of gear wheels, freely mounted on the shafts, provide four gears (Nekrasov, 2001). Numbers of gear wheel pairs are indicated at the bottom of the diagram: 1, 2 and 3.

The first gear is formed by subsequent connection of all three pairs of gear wheels. In raypath plot this gear is represented by three half-lines: half-line 1 is directed from the point 0 on the lower horizontal upwards to the right toward middle horizontal, then the half-line 2 is directed in the overdriving mode downward to the left to the bottom horizontal, and then the sloping half-line 3 is directed upward to the right to the middle horizontal.

The gear ratio of this gear is $U_{1g}=U_1\ U_2\ U_3=2.3 \times (1/1.4) \times 3.7=6.08$;

lg 6.08=0.784.

Half-line 3 represents the 2nd gear. $U_{2g}=U_3=3.7$; lg 3.7=0.57.

Half-line 1 corresponds to the 3rd gear. $U_{3g}=U_1=2.3$; lg 2.3=0.36.

Half-line 2 corresponds to 4th gear. $U_{4g}=U_2=1.4$; lg 1.4=0.146.

In the 1st gear, clutch A is in the left position, clutch B-in neutral position, and clutch C-in right position. For the 2nd gear, clutch A is switched to the right position. For the 3rd gear, clutch A is returned to the left position, and clutch C is switched to the left position. For the 4th gear, clutch A is switched to the right position, and the clutch C remains in the left position.

Planetary gear provides five forward gears and two reverse gears.

The first gear corresponds to the condition $U_{ah}^{b}=K+1$. The ring gear (b) is stopped by the clutch E in the right position and is doubled by the clutch F in the left position. Drive torque is supplied to the sun gear (a) from the clutch D in the left position. The output of driving torque is taken off from the pinion carrier (h) by clutch G in the middle position.

Second gear corresponds to the condition $U_{bh}^{a}=(K+1)/K$. The sun gear (a) is stopped by clutch B in the left position, the input to the ring gear (b) from the clutch D in the right position, the output from the pinion carrier (h) by clutch G in the middle position. Clutches E and F are in neutral switched-off position nearby the casing wall.

The third gear with U=1 is provided by the transmission of driving torque through one of the PG links, for example, through the pinion carrier (h). The input to the pinion carrier is implemented through clutch D in the middle position, the output from the pinion carrier (h) is taken off through the clutch G in the middle position. Clutches B, E and F are in neutral position.

The fourth gear corresponds to the condition $U_{hb}^{a}=K/(K+1)$. Sun gear (a) is stopped by clutch B in the left position, the input to the pinion carrier (h) is supplied by clutch D in the middle position, the output from the ring gear (b) is taken off through clutch G in the left position.

The fifth gear corresponds to the condition $U_{ha}^{b}=1/(K+1)$. Ring gear (b) is stopped by the clutch E in the right position and is doubled by the clutch F in the left position. Driving torque is supplied to the pinion carrier (h) through the clutch D to the middle position. Driving torque output is taken off from sun gear (a) through the clutch G in the right position. Clutches E and F are in neutral position.

Slow reverse motion corresponds to the condition $U_{ab}^{h}=-K$. Pinion carrier (h) is stopped by the clutch E in the middle position. Driving torque is supplied to the sun gear (a) from the clutch D in the left position. Driving torque output is taken off from the ring gear (b) by clutch G in the left position. Clutch F is in a neutral position.

Fast reverse motion corresponds to the condition $U_{ba}^{h}=-1/K$. Pinion carrier (h) is stopped by the clutch E in the middle position. Driving torque is supplied to the ring gear (b) from the clutch D in the right position. Driving torque output is taken off from sun gear (a) by clutch G in the right position. The clutch F is in the neutral or middle position.

In the raypath plot, the first gear of the MSG is shown by the following half-lines: half-line 1 is directed from the point 0 upward to the right, then half-line 2 is directed downward to the left, then half-line 3 is directed upward to the right, and then the half-line U_{ah}^{b} is directed to the point 1 on the upper horizontal.

The diagram in Fig. 4 illustrates the first gear of multi-stage gearbox. The ring gear (b) is stopped by the clutch E. Driving torque is transmitted from the input shaft through the clutch A in the left position to the 1st couple of gear wheels, then by a tubular shaft of the lower gear train it is transmitted to the 2nd pair of gear wheels, then through the tubular shaft of the upper gear train-to the 3rd pair of gear wheels, through the clutch C it is transmitted then to the tubular shaft with three-position toothed rim, wherefrom it is transmitted through clutch D to the toothed rim of the drive shaft of sun gear (a). Sun gear rotates satellites, which when rolled over stopped ring gear (b), rotate pinion carrier (h) and the output shaft with reduced rotation frequency and increased driving torque.

Feasibility study of the multi-stage gearbox with a wide use of the free installation method of gear wheels on the shafts and kinematic features of the planetary gear

Traditional assembly of the multi-stage gearbox for 20 forward gears will require 10-stage shaft gearbox and two-stage planetary gear. For example, 10-stage gearbox of KAMAZ vehicles has 12 gear wheels of forward drive: two gear wheels of the front divider and 10 gear wheels of five-stage gearbox. Additional reverse PG is assimilated to five gear wheels (sun gear, three satellites, and ring gear). Multi-stage gearbox requires 12+5=17 gear wheels.

Gear wheel utilization rate, equal to the ratio of the number of forward gears to the number of required gear wheels, $K_u=20/17=1.176$. The higher the K_u, the smaller are dimensions and specific quantity of metal of the unit. In Fig. 4, the MSG is formed from six gear wheels of the shaft gearbox and five gear wheels of the PG. $K_u=20/(6+5)=1.818$-is a very high factor, indicating relatively small dimensions and specific quantity of metal of multi-stage gearbox.

RESULTS

A more complete use of kinematic capabilities of a simple PG significantly reduces the overall dimensions and specific quantity of metal of the transmission units. The effect is enhanced by the additional application of the method consisting in free installation of gear wheels on the shafts.

CONCLUSION

The proposed design methods of the MSG allow creating competitive compact gear units with low specific quantity of metal, which are protected by several patents.

CONFLICT OF INTEREST

The authors confirm that the submitted data do not contain conflict of interests.

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