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Review

A SIMPLE PLANETARY GEAR IN INTEGRATING MODES

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ABSTRACT

This paper describes constructive methods of implementation of kinematic capabilities of a simple three-element planetary gear (PG) in the integrating (summing) modes. Kinematic capabilities of PG are much wider than used currently in designs. For example, a simple three-element PG can provide 7 gear options in the geared mode, where one of the PG elements is stopped (except of a top gear), as well as three gears in the integrating (summing) modes, where driving torque is applied at different rates on two elements, while is taken-off from the third element.

INTRODUCTION

Planetary gears (PG) in modern land transport vehicles (LTV) are used as two-stage auxiliary gearbox-demultiplicator of the multi-stage gearbox (GB) or in a planetary gearbox (Kudryavtsev and Kyrdyashova, 1977). It is also used in the drive axles as the single-reduction wheel gear of double-spaced drive gears. Planetary gear is often used as symmetrical or asymmetrical differential.

Integrating features of a simple planetary gear

A simple planetary gear, consisting of three elements-a sun gear (a), a ring gear (b), and a pinion carrier (h) with satellites, is characterized by an internal parameter $K=Z_b/Z_{a=}1.5\text{-}5$, which is the ratio of the number of teeth Z_b of ring gear and Z_a of sun gear.

Integrating (summing) capabilities of PG are described by the following dependencies (Kudryavtsev and Kyrdyashova 1977):

1) $n_h=U_{ha}{}^{b}\, n_a+U_{hb}{}^{a}\, n_b=n_a/(K+1)+(n_b\, K)/(K+1);$

2) $n_b=U_{bh}{}^{a}\, n_h+U_{ba}{}^{h}\, n_a=n_h\,(K+1)/K - n_a\,/K;$

3) $n_a=U_{ah}{}^{b}\, n_{h+}U_{ab}{}^{h}\, n_b=(K+1)\, n_h - K\, n_b;$

Where n_a, n_h, n_b - are the rotational speed of a sun gear, pinion carrier and ring gear in rpm.

These dependencies are represented graphically in Fig. 1. Differences of elements rotation frequencies at the PG input in terms of gear ratio U_{in} are built along the horizontal axis in natural and logarithmic form. If we apply driving torque with the rotation speed of the input element $n_{in}=n_{a=}1000$ rpm on one of the PG elements, for example, a sun gear (a), while apply driving torque with lower frequency $n_{b=}500$ rpm on the second element-a ring gear (b), than $U_{in=}1000/500=2.0$; lg $U_{in}=0.3$. This point is to the right of the origin of coordinates. Gear ratios of elements at the PG output relative to those at the input $U_{out=}n_{in}/n_{out}$ are also indicated in natural and logarithmic form at the vertical axis.

For K= 1.62 we get

$n_h=U_{ha}{}^{b}\, n_a+U_{hb}{}^{a}\, n_b=n_a/(K+1)++(n_bK)/(K+ 1)=1000/(1.62+1)+(500*1.62)/(1.62+1)=691.7$ rpm.

$U_{out=}n_{in}/ n_{out}=1000/691.7=1.446$. lg $U_{out}=0.16$ (A)

If we increase U_{in} up 20; lg $U_{in}=1.3$, than $n_{b=}1000/20=50$.

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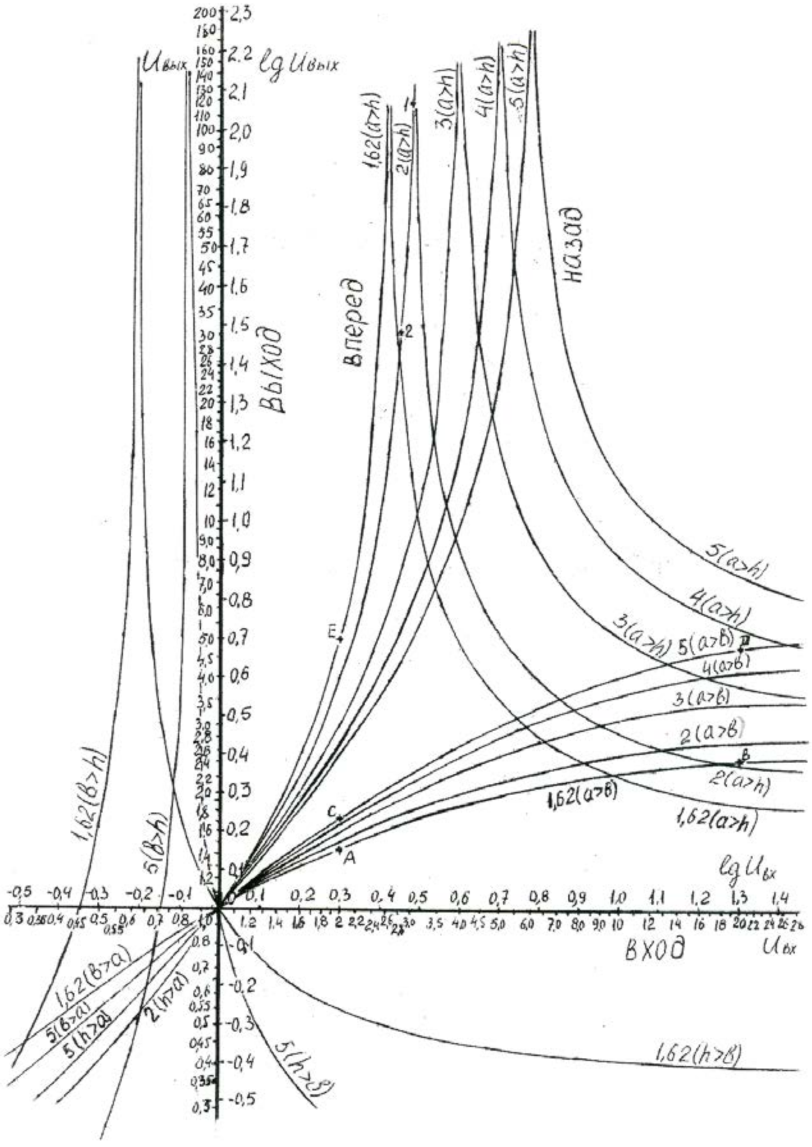


Fig. 1 Change in PG gear ratios in the integrating (summing) modes at various rotation frequencies of PG elements at the input.

n_h=1000/(1.62+1)+(50*1.62)/(1.62+1)=412.7 rpm.

U_{out}=n_{in} / n_{out}=1000/412.7=2.423. lg U_{out}=0.384 (B)

In Fig. 1 this is a bottom sloping line starting from the origin of coordinates to the right.

At K= 5.0 we obtain

n_h=1000/(5+1)+(500*5)/(5+1)=583.17 rpm

U_{out}=n_{in} / n_{out}=1000/583.17=1.715. lg U_{out}=0.234 (C)

n_h=1000/(5+1)+(50*5)/(5+1)=208.32 rpm

U_{out}=n_{in} / n_{out}=1000/208.32=4.8. lg U_{out}=0.68 (D)

In Fig. 1 this is the upper sloping line starting from the origin of coordinates to the right.

The bundle of sloping lines reflects the kinematic capabilities of the PG in integrating modes at its various internal parameters K: 1.62; 2.0; 3.0; 4.0 and 5.0. The mode n_a>n_b is indicated nearby numbers, and only PG elements are distinguished (a>b).

Graphical dependences for the other modes are constructed similarly.

At K=1.62; n_h=500 rpm, for the mode n_b=U_{bh}^a n_h+U_{ba}^h n_a=n_h (K+1)/K-n_a/K we get n_b=500*2.62/1.62-1000/1.62=195 rpm;

U_{out}=n_{in}/n_{out}=1000/195=5.13. lg U_{out}=0.7 (E)

This mode has extrema when changing the rotation direction. The extrema are located above the lg U_{in}=lg (K+1). For example, if K=1.62, we have lg 2.62=0.42.

If lg U_{in}=0.42; than U_{in}=2.62. n_h=1000/2.62=381.7 rpm

$n_b=n_h\ (K+1)/K-n_a\ /K=381.7\ (1.62+1)/1.62-1000/1.62=0$

Change in rotation direction of the PG output element occurs above the point lg U_{in}=0.42. The direction from the origin of coordinates steeply upward to this point corresponds to moving forward, while to the right from this point steeply downward means moving backward-R.

With increasing K the extreme points are shifted upwards to the right.

If lg U_{in}=0.7; than $U_{in=}$5.0. $n_{h=}$1000/5=200 rpm. At that we obtain

$n_{b=}$200*2.62/1.62-1000/1.62=- 293.4 rpm;

Sign (-) indicates the direction of the output element rotation-reverse-R.

$U_{out=}n_{in}/n_{out}$=1000/293.4=3.4. lg U_{out}=0.53.

Considered second dependence is of the greatest interest, as it allows obtaining a significant change in gear ratios at the output of the unit through a slight difference in rotation frequencies of the PG elements.

Non-axial gearbox of type 24R4

Fig. 2 shows a diagram of non-axial MSG (multi-stage gearbox) of type 24R4 according to the patent RU No. 2058234, which allows obtaining 24 gears to move forward and 4 gears for reverse movement, and has a fairly simple design, consisting of a three-element PG, six gears with outer teeth and six strike clutches (Nekrasov, 2001).

Three pairs of gear wheels (1-3), freely mounted on the shafts 2 and 10, provide four transmission options (sub-bands) at the PG input. In each sub-band, PG implements 6 gears in the gear mode. Moreover, the design uses the integrating (summing) capabilities of PG: 1st and 2nd gears are obtained when applying the torque to the sun gear a and pinion carrier h, while the torque is taken-off from the ring gear b; 11th and 14th gears are obtained through the application of the torque on the sun gear a and ring gear b, while the torque is taken-off from the pinion carrier h.

For example, at the 1st gear and K=1.62, the rotation torque is transferred from the input shaft via the clutch A in position I (left position in Fig. 2a) to a pair of gear wheels 1, where the torque is split into two parts. One portion of torque is transferred through the clutch C in a position 3 to the lower shaft and a sun gear a. The other portion of torque is transferred via tubular shaft, the pair of gear wheels 2, and the

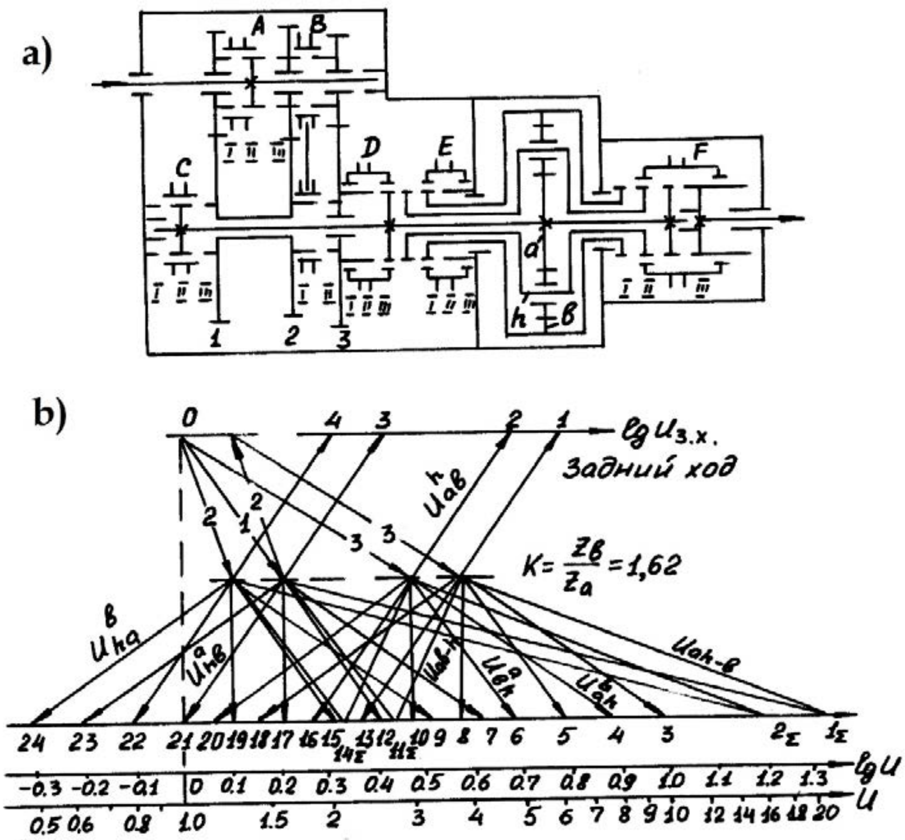


Fig. 2 The diagram and raypath plot of non-axial MSG of type 24R4 according to the patent RU No. 2058234 with the PG in gear and summing modes.

clutch D in position II to the tubular shaft and pinion carrier h. Torques are summarized on the ring gear b and taken-off from it by the clutch F in position I.

In raypath plot this condition is reflected by the half-line 1 directed from the point 0 downwards to the right, the three half-lines 1, 2 and 3, then the two half-lines converging in the point 1 on the bottom horizontal. In the raypath plot the values of gear ratios (U) are reflected by the uneven scale of the lover horizontal. Above this horizontal, a uniform scale of gear ratio values is presented in logarithmic scale (lgU); at that the half-line characterizing the magnitude of the gear ratio, does not change its value.

The rotation of the sun gear a with a gear ratio from the input shaft U_1=1.6; and pinion carrier h with the gear ratio $U_{1,2,3}$=1.6*0.79*2.95=3.73 provides at the MSG output gear ratio U_{out}>20.

The nature of the gear distribution changes when changing the gear parameters and PG. Instead of PG of type 24R4 we can get PG of type 22R6 while increasing the difference in gear ratios between the pairs of gears at K=1.62; at that, the summing gears will shift to the right and then go into reverse mode.

Coaxial gearbox of type 30R5

Multi-stage gearboxes can be both coaxial and non-axial. Fig. 3 shows a diagram and raypath plot of the coaxial MSG of the type 30R5 according to the inventor's certificate N 1379143 with PG at K=2.0 in gear and summing modes.

Lower gears with high gear ratio, provided by the operation of the PG in integrating modes, provide operation of LTV with the "creeping" speed, when maneuvering at cargo operations. Gears with high gear ratio are required first and foremost to ensure the beginning of the movement of a large mass land transport vehicles such as tractors, armored vehicles, etc. (Nekrasov, 2001).

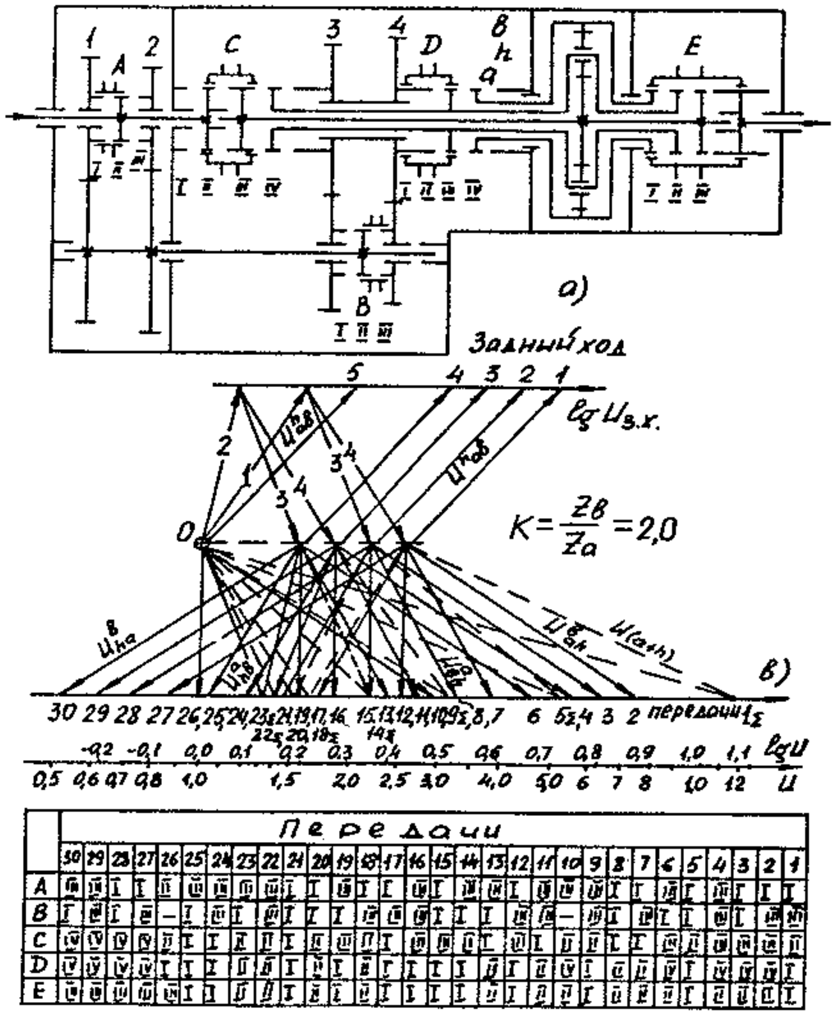


Fig. 3 The diagram and raypath plot of coaxial MSG of the type 30R5 according to the inventor's certificate No. 1379143 in gear and summing modes.

Units for gas turbine power plants

Start of gas turbine power plant as well as dynamic balancing of large mass rotors at the maintenance and repair works of gas turbines is of great importance. An individual barring and launching gears are usually used to start gas turbines. These devices are functionally separated and quite complex. Initially barring gear is activated to withdraw a rotor with a large inertia mass from a rest. Then, the launching gear is activated, which provides a certain rotation frequency of the compressor rotor, sufficient to supply air parameters required to implement the start-up of combustion chamber and turbine. Turbo-expander is used often as launching gear. Its operation requires high pressure gas taken from the main pipeline (Mogil'nitsky and Steshenko, 1971).

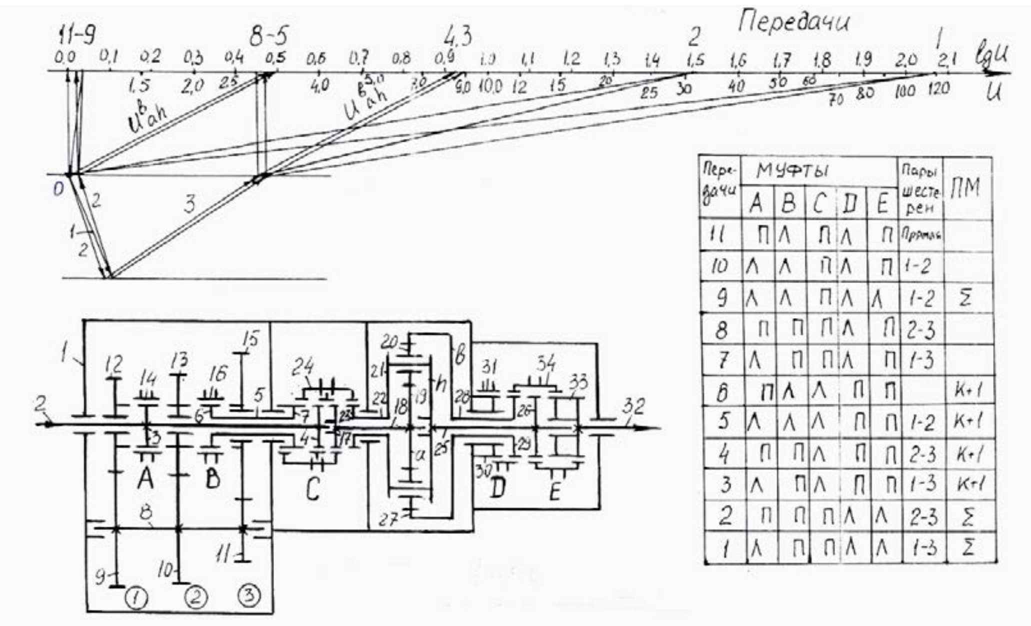
To combine two functions in one technical device we may use a simple three-element PG. Fig. 4 shows a diagram of a coaxial unit for barring gear (BG) according to the patent RU No. 2397344, as well as raypath plot and table indicating positions of strike clutches at different gears.

The unit comprises a three-shaft four-stage GB (gear wheel pairs: 1-3; 2-3; 1-3 and direct drive), as well as a simple three-element PG. The location of the two-position dual strike clutch on the shafts between the GB and PG, as well as the clutch between the gear case wall and toothed rim of the tubular shaft of ring gear provides operation of the PG in three modes: 1) summing (1st, 2nd, and 9th gears); 2) direct gear in the PG (7th, 8th, 10th, and 11th gears); 3) $U^{b}_{ah=}$K+1 mode (3rd-6th gears). Superscript in U^{b}_{ah} denotes the stopped element-ring gear b; subscript denotes elements: input-sun gear a and torque output-pinion carrier h. Theoretically, the unit provides 11 gears at five gear steps (11-9; 8-5; 4-3; 2nd and 1st gears) with almost equal intervals between them that provides reliable start of gas turbine unit.

When unit is operating in start-up mode of the gas turbine plant on the 1st gear, clutches A, D and E are in the left (L) position, while clutches B and C are in the right (R) position. Sun gear receives a direct drive from the input shaft 2 through the inner toothed rim of the clutch 24 (C). The pinion carrier 21 receives the drive through gear pairs 1 and 3 with rotation frequency lower than that of the sun gear 19.

Rotation torque is transferred from the drive motor via the input shaft 2, toothed rim 3 and clutch 14 (A) to gear wheel pairs 1 (12-9), then through the intermediate shaft 8 - to gear wheel pairs 3 (11-15), further through the clutch 16 (B) to the first toothed rim 6 and the tubular output shaft 5, the second toothed rim 7, through outer toothed rims of the clutch 24 (C) to the toothed rim and the tubular shaft 22 of the pinion carrier 21; power flows from the sun gear 19 and the pinion carrier 21 are summarized on satellites 20 and are taken-off from the ring gear 27 through a tubular output shaft 28 to toothed rim 29, then, through the clutch 34 (E)-to the toothed rim 33 and the output shaft 32 of the unit.

In the raypath plot, half-line 1 is directed from the point 0 on the middle horizontal downward to the



Передачи	Муфты A	B	C	D	E	Пары шестерен	ПМ
11	П	Л	П	Л	П	Прямая	
10	Л	Л	П	Л	П	1-2	
9	Л	Л	П	Л	Л	1-2	Σ
8	П	П	П	Л	П	2-3	
7	Л	П	П	Л	П	1-3	
6	П	Л	Л	П	П		K+1
5	Л	Л	Л	П	П	1-2	K+1
4	П	П	Л	П	П	2-3	K+1
3	Л	П	Л	П	П	1-3	K+1
2	П	П	П	Л	Л	2-3	Σ
1	Л	П	П	Л	Л	1-3	Σ

Fig. 4 The diagram and raypath plot of coaxial MSG 11 according to the patent RU No. 2397344 with the PG in gear and summing modes.

right, then the half-line 3 is directed upward to the right to the level lg U_{in}=0.47; at that U=2.95. Two half-lines are directed to a point on the upper horizontal equal to 2.07; one half-line is directed from point 0 of the rotation frequency of sun gear 19, while second one - from rotation frequency of the pinion carrier 21. The rotation frequency of the ring gear 27 and output shaft of the unit at K=2.0 are defined as:

2) $n_b=U_{bh}^{a}\ n_h+U_{ba}^{h}\ n_a=n_h\ (K+1)/K-n_a/K$=(3000/2.95)*3/2-3000/2=25.4 rpm.

$U_{1=}$3000/25.4=118.1; lg 118.1=2.07.

In Fig. 1, this point, denoted as No.1 (lg U_{in}=0.47 and lg U_{out}=2.07), is located just to the left of the extreme point lg (K+1)=lg 3=0.477, at that U_{in}=2.818.

On the 2[nd] gear lg U_{in}=0.45; $n_b=n_h\ (K+1)/K-n_a/K$=(3000/2.818)*3/2-3000/2=96.9 rpm. $U_{1=}$3000/96.9=31; lg 31=1.49.

In Fig. 1 this point, denoted as No. 2 (lg U_{in}=0.45 and lg U_{out}=1.49), is located to the left and below the previous point.

Fig. 5 and 6 show further development of concerned unit design. The lower part presents the kinematic diagram of the unit; gear wheel pair numbers for table are indicated inside the circles.

The upper part presents the raypath plot of gear ratios provided by the unit. Table including data on gears, strike clutches status at indicated gears, and involved unit elements is located under the plot on the right.

Four rows of PG gears, marked with digits inside the circles, provide 8 gear options on a tubular output shaft 5.

Planetary gear mounted at the unit output provides three option modes:

The first mode of PG operation is summing mode (∑); the clutches E and F are in the left position (L), while the clutch D is in the right position (R) and is used in seven gears: 1, 2, 11, 12, 17, 22 and 23.

The second mode of PG operation is $U^{b}_{ah=}$K+1 mode; clutches E and F are in the right position (R), while the clutch D is in the left position (L) and is used in eight gears: 3-8, 13, and 15.

The third mode of PG operation is top gear; clutches D and F are in the right position (R), while the clutch E is in the left position (L) and is used in eight gears: 9, 10, 14, 16, and 18-21.

Gears of the unit are formed by GB and PG options.

1[st] gear-(1[st] option in the GB and the 1[st] option in the PG)

The 2[nd] and 4[th] gear wheel pairs are in operation, at that, clutches A and C are in the right position (R), while the clutch B is in the left position (L).

Rotation torque is transmitted from the drive motor

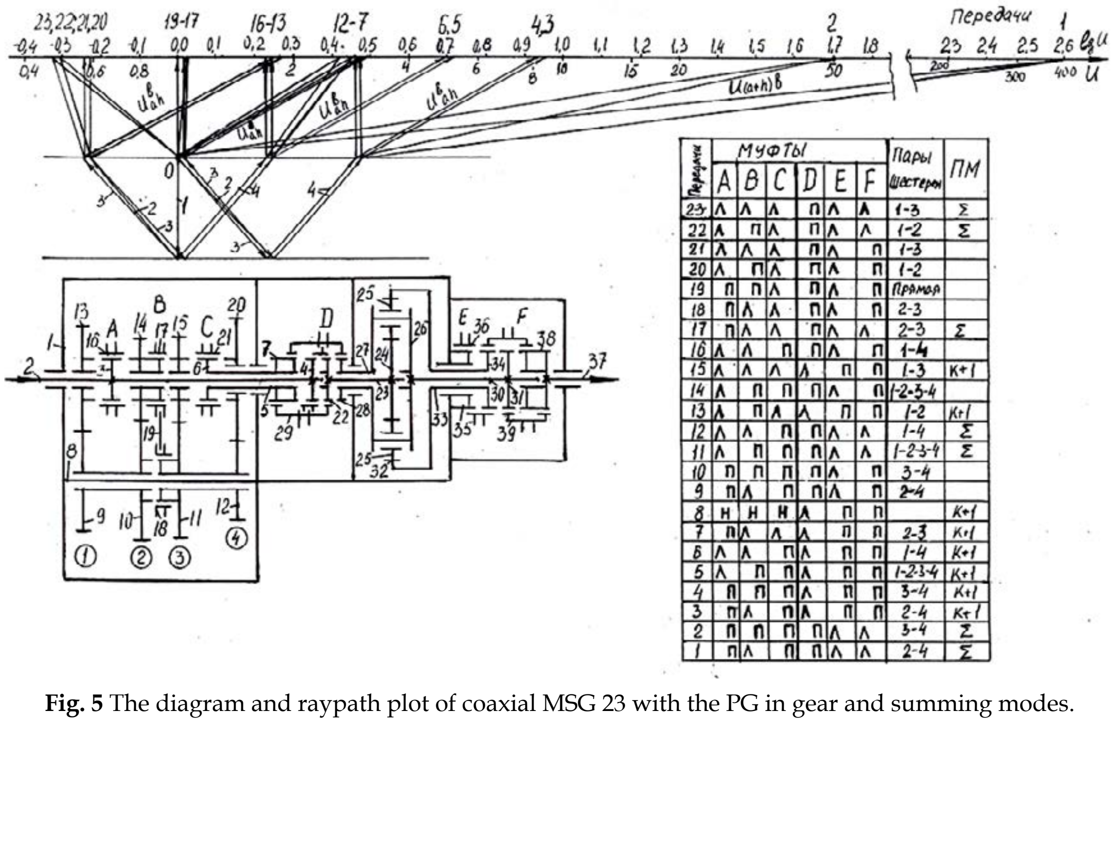


Fig. 5 The diagram and raypath plot of coaxial MSG 23 with the PG in gear and summing modes.

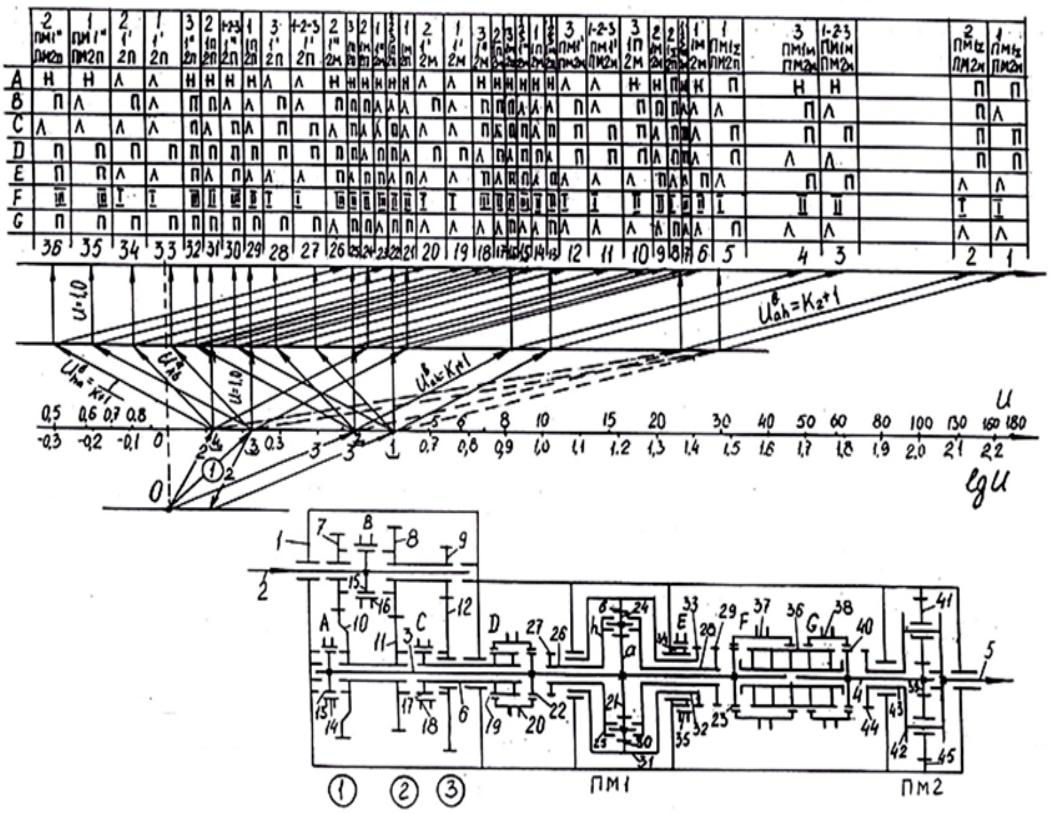


Fig. 6 The diagram and raypath plot of non-axial MSG 36 with two PGs operating in gear and summing modes.

via the toothed rim 3 and the clutch 16 (A) to gear wheels pair 2 (14-10), then through the half-clutch 18- to the second train of gear wheels of axis 8 and a pair of gear wheels 4 (12-20), then through the clutch 21 (C)-to the first toothed rim 6 and the tubular output shaft 5.

In raypath plot, half-line 2 from point 0 on the middle horizontal is directed downward to the right, then the half-line 4 is directed upwards to the right to the level lgU=0.475, at that U_{in}=2.985, then the half-line becomes vertical that corresponds to the 9th gear.

Driving torque from GB to PG is transmitted in two ways: the first portion comes from the input shaft 2 to the second toothed rim 4 through the internal toothed rim of the clutch 29 (D) to the toothed rim 22 of the shaft 23 of the sun gear drive 24; the second portion is transmitted through tubular output shaft 5, its second toothed rim 7, the external toothed rim of the clutch 29 (D) to the toothed rim 28 and a tubular shaft 27 of the pinion carrier 26. Power flows from the sun gear 24 and pinion carrier 26 are summed up at the satellites 25 and taken-off from the ring gear 32 through the tubular output shaft 33 to toothed rim 34, then through the clutch 39 (F) to toothed rim 38 and output shaft 37 of the unit.

In raypath plot, two half-lines are directed to the point on the upper horizontal at the level of lgU_{in}=2.6; one from the point 0 to the rotation frequency of the sun gear 24, and the second-from rotation frequency of the pinion carrier 26.

The rotation frequency of the ring gear 32 and output shaft of the gear 37, as well as the gear ratio of the unit are defined as:

$n_b=u^h_{ba}\ n_a+u^a_{bh}\ n_h=-\ n_a/\kappa+n_h(\kappa+1)/\kappa=-3000/2+3000/2.985*3/2=7.5$ rpm;

$u_{1=}3000/7.5=400$; lg 400=2.6.

A slight change in the gear ratio of the GB leads to a significant change in gear ratio of the unit. For example, U=2.99; $n_b=-\ n_a/\kappa+n_h(\kappa+1)/\kappa=-3000/2+3000/2.99*3/2=5$ rpm; $u_{1=}3000/5=600$; lg 600=2.778. If lgU_{in} › 3.0, the rotation direction of the output shaft 37 is reversed.

Figure 6 shows the non-axial unit, which consists of three pairs of gear wheels and two PG: 1st PG operates in gear and integrating modes, while 2nd PG is dual-mode: a direct drive and $U^b_{ah=}K_2+1$.

Feasibility study of the simple planetary gear in integrating modes

Positive effect: units with the PG in integrating modes replace BG (two complex devices: barring gear and launching gear) and provide a reliable start-up of the gas turbine unit without the use of gas flow energy from trunk pipeline due to gears located in a few steps with the distribution of gear ratios within a wide range.

A comparative assessment of the gearboxes can be carried out using the gear wheel utilization rate K_u, which is the ratio of the number of forward gears to the number of required gear wheels. The higher the K_u the smaller are dimensions and metal consumption of the unit. Even with three degrees of freedom of the planetary GB, the efficiency of the use of gear wheels is small, K_u=8/12=0.67 and K_u=16/16=1.0 (Nekrasov, 2001).

The units presented in Fig. 2-6 have higher gear wheel utilization rates. Assuming that each PG consists of five gear wheels, for the unit in Fig. 6, we will get a very high K_u=36/(2x5+6)=2.25.

RESULTS

A more complete use of the kinematic capabilities of simple PGs in the gear and integrating modes significantly reduces the size and metal content of the units.

CONCLUSION

The proposed design methods allow creating competitive compact units with low metal content.

CONFLICT OF INTEREST

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

REFERENCES

Kudryavtsev, V.N. and Kyrdyashova, Yu.N. 1977. The planetary gears. Leningrad, Mashinostroenie. 536.

Mogil'nitsky, I.P. and Steshenko, V.N. 1971. Gas turbines in the oil and gas industry. Moscow, Nedra. 160.

Nekrasov, V.I. 2001. Multistage transmission. Design, drafting and calculation. Kurgan, Kurgan State University Publishing House. 155.