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Non-Coaxial 24-Speed Shaft-Planetary Gearbox

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Abstract*: *The article presents the results of the analysis of existing designs of non-coaxial planetary gearboxes, their advantages and disadvantages. The invention is related to transport machinery area, transmission of transport vehicles, land transport systems, including motor vehicles, tractor-trailers, construction and road machines, tractors and ground-based equipment for special purposes. The design and kinematic scheme of a non-coaxial multiple-speed shaft-planetary gearbox, type 24R10, are presented. The work describes the operation of a non-coaxial multiple-speed shaft-planetary gearbox, type 24R10 (24 direct gears and 10 reverse gears), as well as gear ratios for each gear. A technical and economic assessment of the non-coaxial multiple-speed shaft-planetary gearbox, such as 24R10, is given. It is shown that the task is solved by compacting the layout of the multiple-speed gearbox and the planetary mechanism both in gear and summing modes, as the torque enters two links and is removed from the third one. The research proves that the design of a non-coaxial multiple-speed shaft-planetary gearbox, type 24R10, provides an improvement in the layout and operational characteristics of the gearbox and, accordingly, the operational capabilities of the vehicle.

Keywords* : *non-coaxial gearbox, multiple-speed gearbox, shaft-planetary gearbox, planetary gear, transmission unit, vehicle.

I. INTRODUCTION

The gearboxes used in automobiles with staging gear ratios, which include wheel gears, shafts, supports and gear shifting devices, are structurally complex and metal-consuming units [1, 2]. Gearboxes with coaxial drive axle shafts are widely used as they allow the same main gear to be used (interchangeably) for the front and rear drive axles. However, in this case, the drive gear of the main transmission of the front axle, having the left-hand direction of the helix teeth, will work on "screwing in". Therefore, when loosening its bearings, the main transmission of the front-drive axle may become wedged [3].

Gearboxes with non-coaxial driven shafts do not have an intermediate shaft. They are more compact, less metal-consuming, more silent during operation and have a higher efficiency [4].

Transmission units – gearboxes with staging gear ratios, containing gears, shafts, supports, gear shifting devices – are used in Russian and foreign cars [5]. These units are structurally complex and metal-consuming.

Analysis of various designs of gearboxes allows one to conclude that attention is paid to this issue. Researchers propose various design solutions to simplify the design, reduce metal consumption or improve the performance of the device.

The gearbox (patent RU 2025304 C1) [6] has a simpler configuration as compared to typical designs of transmission units and provides a sufficient range of gear ratios (the ratio of the gear ratios from the lower gear to the highest). However, it has limited operational capabilities due to the non-optimal (uneven) distribution of gear ratios over the range and requires the use of a gear pair with a large gear ratio, which increases the size and intensity of the unit. The presence of several sleeve coaxial shafts on each side of the planetary mechanism (PM) complicates the design of the unit and requires the use of five clutches and shifters (five-way gearbox) [7].

The design of the non-coaxial shaft-planetary gearbox (patent RU 2295456 C1) [8] allows optimizing the distribution of the gear ratios over the range at equal intervals between gears. It reduces the size and intensity of the structure and improves the mechanism for shifting the planetary gear. This technical result is achieved because the non-coaxial planetary gearbox includes an aggregate body, in which a two-shaft non-coaxial gearbox with pairwise intermeshed external gears is located. Two gears on the mainshaft are combined into a unit. Opposite to this unit, between the gears of the primary shaft, there is a gear shift clutch. Gears on the shafts of the gearbox are installed freely. At the output of the two-shaft gearbox, there is a simple three-stage planetary gear with a carrier, satellite, sun gear and epicyclic wheel with the mainshaft of the gearbox with the sun gear fixed on it. The output shaft of the carrier and the output shaft of the unit are coaxial. The carrier and the casing of the epicyclic wheel have adjacent ring gears and sleeve shaft of the epicyclic wheel casing. The output shaft of the carrier and the output shaft of the unit have adjacently located ring gears. A striker clutch is installed on the ring gear of the output shaft. A gear is freely mounted on the mainshaft. A striker clutch is located on the hub of the mainshaft and between this gear and the gear train. The same way two gears on the primary shaft are combined into a train and are located opposite to the striker clutch of the mainshaft. The ring gear of the output shaft of a carrier is two-position; the ring gear of the output shaft is three-position. The striker clutch on this crown has three internal gears of the crown. On the slide of the striker clutch, there is a rigidly fixed ring gear with internal and external gearing. The external gearing is connected with the three-position ring gear of the unit casing.

In non-coaxial multistage shaft-planetary gearbox with electric starter (based on patent RU 2652485) [9], the vehicle's layout and operational capabilities are expanded by sealing the multistage gearbox

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layout and the PM, not only in gearboxes, but also in summing modes, when the torque enters two links and is removed from the third link. In the gearbox between the flywheel and the crankshaft of the engine, there is a two-position clutch. At the entrance of a multi-stage gearbox on two shafts, four rows of direct front gears are freely installed. A number of gears of an electric inertial starter with an electric motor at the output of the primary shaft of the mechanical gearbox (MG), a simple three-link PM consisting of a carrier with satellites that are meshed to the sun gear and epicyclic wheel, as well as six gearshifts, are also installed.

Closest to the proposed design of a multi-stage gearbox is the gearbox design based on patent RU No. 2177882 C2 [10]. The shaft-planetary gearbox 16R4 consists of a non-coaxial two-shaft four-stage unit and a simple three-link PM operating in five modes: slow gear Ub ah=K+1; reducing gear Ua bh=(K+1)/K; direct gear U=1.0; increasing gear Ua hb=K/(K+1) and reverse gear Uh ba=-K. Six wheel gears, five shift clutches (one of them is double) and three PM linkages (sun gear, epicyclic wheel, and planetary) provide 16 forward gears and 4 reverse gears. The disadvantage of the described transmission unit is its significant longitudinal dimension.

The task, at which the described device is aimed, is to expand the layout and operational capabilities of the vehicle. The task is solved by compaction of the layout of the MG and the work of the PM not only in gear but also in summing modes when the torque enters the two links and is removed from the third link.

II. PROPOSED METHOD

A. General description

Theoretical (study, systematization, analysis, synthesis of literature, patent search on the problem under consideration; analysis of the subject matter of the study; comparison and grouping of theoretical material on the problem of the research; modeling and designing; generalization of the research results); experimental (methods of mathematical results processing).

B. Algorithm

Design and kinematic scheme of the non-coaxial shaft-planetary multiple-speed gearbox, type 24R10

A kinematic diagram of a non-coaxial shaft-planetary multiple-speed gearbox, type 24R10 (24 front gears and 10 reverse gears) is shown in Figure 1. The MG consists of a simple three-link planetary gear, six front-wheel gears freely mounted on two parallelly arranged shafts and two rows of reverse gears mounted in front of the forward gears.

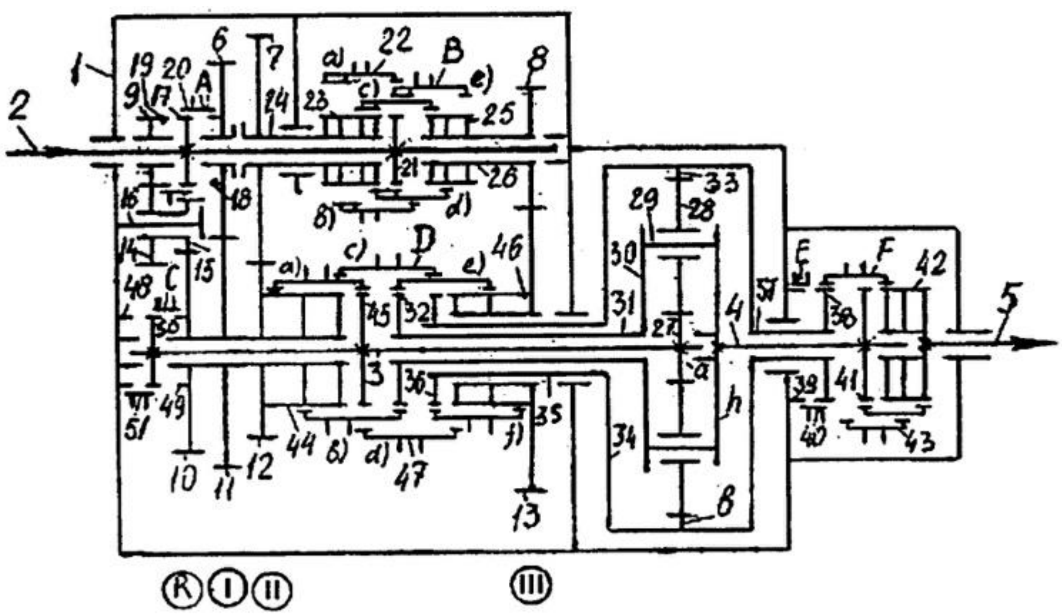

Fig. 1. Kinematic scheme of the non-coaxial shaft-planetary multiple-speed gearbox, type 24R10 [11].

The input shaft 2 is located in the base of MG 1. The mainshaft 3, which is the drive shaft of the PM sun gear, is installed in parallel to it. The planetary gear 4 and the output shaft 5 are coaxially mounted. Separate front gears 6, 7 are freely mounted on the input shaft 2, 8. The reverse gear 9, a block of three gears 10-12 and a separate gear 13 are freely mounted on the mainshaft 3. The driven reverse gear 10, through a train of reverse gears 14-15, mounted on the axis 16, is engaged with the main reverse gear 9. Wheel gears of the first and second rows 11 and 12 are meshed respectively with the gears 6 and 7 of the input shaft 2. The separate driven gear of the third row 13 is meshed with the gear 8 of the input shaft 2. The first gear 17, mounted on the input shaft 2 between the separate gears 9 and 6, with a three-position clutch 20 (A) of the input shaft 2, is mounted with the ring gear 18 and 19. On the input shaft 2 between the gears 7 of the second and 8 third rows, there is a second ring gear 21 with a five-position striker clutch 22 (B) having a wide left ring gear for random compound with four-position ring gear 23 of the sleeve shaft 24, gear 7 and the second row with three-position crown 25 of the sleeve shaft 26 and the driving gear 8 of the third row. The sun gear 27 (a) is fixed on the mainshaft, which is engaged with the satellites 28 mounted on the axes 29 of the carrier 30 (h). A sleeve shaft 31 with a ring gear 32 is fixed at the entrance of the carrier 30 and a shaft 4 – at the output of the carrier. Satellites 28 are also engaged with an epicyclic wheel 33 (b). An axial shaft 35 with a ring gear 36 is fixed at the inlet of the casing 34 of the epicyclic wheel 33 (b). A sleeve shaft 37 with a ring gear 38 is located next to the gear ring 39 of the casing 1 and the clutch 40 of the epicyclic wheel stop 33 (b). A ring gear 41 is fixed on the shaft 4 of the carrier 30 and a three-position ring gear 42 is mounted on the output shaft 5. A three-position clutch 43 (F) of the output shaft 5 is mounted on it. On the gears train 10-12 of the mainshaft 3, a three-position ring gear 44 is located from the side of the ring gear 12. Next to it, there are ring gears, which are fixed as follows: 45 – on the mainshaft 3, 32 – on the sleeve shaft 31 of the carrier 30, 36 – on the axial shaft 35 of the casing 34 of the epicyclic wheel 33 (b). Next, there is a three-position ring gear 46 fixed on the driven gear 13 of the third row. A multi-position striker clutch 47 (D) is mounted between these ring gears and the ring gear 50. A three-position striker clutch 51 (C) is also fixed on the mainshaft 3 between the ring gears 48 of the gearbox casing 1 and 49 and driven reverse gear 10.

Operation of the non-coaxial multiple-speed shaft-planetary gearbox, type 24R10

Clutch A (20) in the right position connects the ring gear 17 of the input shaft 2 with the ring gear 18 of the gear pinion 6 of the first row. In a neutral position, it is located on the ring gear 17. In the left position, it connects the ring gear 17 of the input shaft 2 with the ring gear 19 of the reverse pinion gear 9.

Clutch B (22) in the first position "a" connects the ring gear 23 of the sleeve shaft 24 of the top gear 7 of the second row (top left view) with the ring gear 21 of the input shaft 2. In the second position "b", the neutral state is ensured when the right clutch crown is between the ring gears 21 and 25 and the left one – in the left part of the ring gear 23. In the third position "c", it connects the ring gear 23 of the sleeve shaft 24 of the top gear 7 of the second row with the gear crown 25 of the sleeve shaft 26 of the drive gear 8 of the third row. In the fourth position "d", the clutch connects the ring gears 23 of the sleeve shaft 24 of the gear 7 and 25 of the sleeve shaft 26 of the gear 8 with the gear crown 21 of the input shaft 2. In the fifth position "e", the clutch connects the ring gears 21 of the input shaft 2 and 25 of the sleeve shaft 26 of the gear 8.

Clutch C (51) in the left position connects the ring gear 50 of the mainshaft 3 to the ring gear 48 of the MG casing 1. In a neutral position, it is located on the ring gear 50. In the right position, it connects the ring gear 50 of the mainshaft 3 with the ring gear 49 of the lower driven reverse gear 10 (with the gear train 10-12).

Clutch D (47) in the first position "a" connects the ring gears 44 of the lower gear 12 of the second row (gear train 10-12) and 45 of the mainshaft 3. In the second and fourth positions, the neutral state is ensured when the right ring gear of the clutch is between the ring gears 45 and 32, as well as 32 and 36. In the third position "b", it connects the ring gears 44 of the lower gear 12 of the second row and 32 of the sleeve shaft 31 of the carrier 30 (h). In the fifth position "c", the clutch connects the ring gears 44 of the lower gear 12 of the second row and 36 of the axial shaft 35 of the casing 34 of the epicyclic wheel 33. In the sixth position "d", it connects the ring gear 46 of the driven gear 13 of the third row with the ring gear 45 of the mainshaft 3 and to the sun pinion 27 (a). In the seventh position "e", it connects the ring gear 46 of the driven gear 13 of the third row with the ring gear 32 of the sleeve shaft 31 of the carrier 30. In the eighth position "f", it connects the ring gear 46 of the driven gear 13 of the third row to the ring gear 36 of the axial shaft 35 of the casing 34 of the epicyclic wheel 33.

Clutch E (40) in the right position connects the ring gear 39 of the MG casing 1 with the ring gear 38 of the sleeve shaft 37 of the casing 34 of the epicyclic wheel 33. In the off, neutral position it is located on the ring gear 39.

Clutch F (43) in the left position connects the ring gear 38 of the sleeve shaft 37 of the casing 34 of the epicyclic wheel 33 with the ring gear 42 of the output shaft 5 (upper view). In the neutral position, the left crown of the casing is located between the ring gears 38 and 41. In the right position, it connects the ring gear 41 of the shaft 4 of the carrier 30 (h) with the ring gear 42 of the output shaft 5.

First gear: clutches A, E, and F are in the right position, clutch C is in a neutral position. The pairs of gears I, II, III, and PM in the $U^{b}{}_{ah}$=K+1 mode are put into service. From the drive motor through the input shaft 2, the ring gear 17 and the clutch 20 the torque reaches the pair of gears I (6-11), through the sleeve shaft of the train – the pair of gears II (12-7), through the sleeve shaft 24, the ring gear 23, the casing 22 (B), the ring gear 25 (upper-middle view), the sleeve shaft 26 – the pair of gears III (8-13), through the ring gear 46, the casing 47 it reaches the ring gear 45 and mainshaft 3, then it reaches sun gear 27 (a) of PM. The epicyclic wheel 33 (b) with the casing 34, the sleeve shaft 37, the ring gear 38, the clutch 40 is fixed relative to the ring gear 39 of the casing 1 of MG. The sun gear 27 (a) rotates the satellites 28, which roll around the stopped epicyclic wheel 33 (b) and rotate the carrier 30 (h), shaft 4, ring gear 41, clutch F (43), ring gear 42 and output shaft 5 of the MCP with low speed and increased torque. The gear ratio of the first MG gear:

$U_{1g}=U_1U_2U_3U^{b}{}_{ah}=1.32\times1.32\times2.29\times6.0=24.0$; lq 24.0=1.38.

where U is the gear ratio. For example, the gear $U^{b}{}_{ah}$: the index at the top indicates a stopped PM link, "b" is an epicyclic wheel, the indices at the bottom "ah" are the links of the input (sun gear) and output (carrier) of the flow power.

Second gear: PM in $U^{b}{}_{ah}$=K+1 mode. From the input shaft 2, through the ring gear 21 and the clutch 22 (B), the torque reaches the ring gear 25, the sleeve shaft 26, through the pair of gears III (8-13) and then the same way as in the first gear. The gear ratio of the second MG gear:

$U_{2g}=U_3U^{b}{}_{ah}=2.29\times6.0=13.74$; lq 13.74=1.14.

Third gear: the pair of gears I and PM are operating in $U^{b}{}_{ah}$=K+1 mode. Drive to the mainshaft 3 is also possible by the clutch D; clutch C can be switched to the neutral position. From the input shaft 2, through the ring gear 17 and the clutch 20 (A), the torque flows to the pair of gears I (6-11), through the sleeve shaft of the gear train to gear 10, ring gear 49, through the clutch C (51) to the ring gear 50 (top view), to the mainshaft 3, on the PM sun gear 27 (a), and then correspondingly as in the first gear. The gear ratio of the third MG gear:

$U_{3g}=U_1U^b{}_{ah}=1.32\times6.0=7.9$; lq 7.9=0.9.

Fourth gear: PM works in summing mode (a+h). From the input shaft 2 through the ring gear 17 and the clutch 20 (A), the torque enters the pair of gears I (6-11), through the clutch C (51) to the output shaft 3 and the sun gear 27 (a). Then, to the pair of gears III, correspondingly as in the first gear from gear 13 through ring gear 46, clutch D (47) to ring gear 32 (upper right view), sleeve shaft 31 and carrier 30 (h). From the sun gear 27 (a) and the carrier 30 (h) by the satellites 28, the forces are transmitted to the planetary wheel 33 (b), casing 34, sleeve shaft 37, the ring gear 38, clutch F (43) to the ring gear 42 and output shaft 5. The summing capabilities of the PM, in this case, are described by the dependence $n_b=U_{bh}{}^a n_h+U_{ba}{}^h n_a=n_h(K+1)K-n_a/K$, where n_a, n_h, n_b are the rotational speeds of the sun gear, carrier and epicyclic wheel in rpm. Conditionally assume $n_{in}=n_2=1{,}000$ rpm. In this case, the frequency of rotation of the sun gear $n_a=n_2/U_1=1{,}000/1.32=757.6$ rpm. The frequency of rotation of the carrier $n_h=n_2/U_1U_2U_3=1{,}000/(1.32\times1.32\times2.29)=250.6$ rpm.

The rotational speed of the planetary wheel 34 (b) and the output shaft 5 will be: $n_b=n_h(K+1)/K-n_a/K=250.6(6/5)-757.6/5=300.7-151.5=149.2$ rpm. The gear ratio of the fourth MG gear: $u_{4g}=n_2/n_b=1{,}000/149.2=6.7$; lq 6.7=0.826.

Fifth gear: the sun gear 27(a) is stopped by the mainshaft 3, the ring gear 50, the clutch C (51) relative to the ring gear 48 of the casing 1. From the input shaft 2 to the driven gear 13 of the third row and the ring gear 46, the torque enters as in the first gear, clutch D to the ring gear 36 (lower right view), on the axial shaft 35, the casing 34 and the epicyclic wheel 33(b), satellites 28 on the axis 29 and carrier 30(h), shaft 4 and then correspondingly as in the first, second and third gears. The gear ratio of the fifth MG gear:

$u_{5g}=U_1U_2U_3U^a{}_{bh}=1.32\times1.32\times2.29\times(6.0/5.0)=4.8$; lq 4.8=0.68.

Sixth gear: from the input shaft 2 through the ring gear 21 and the clutch 22 (B), the torque enters the pair of gears II 7-12, through the ring gear 44, clutch C (51) (upper view) to the ring gear 50, then as in the third gear. The gear ratio of the sixth MG gear: $U_{6g}=U_2U^b{}_{ah}=(1/1.32)\times6.0=4.55$; lq 4.55=0.66.

Seventh gear: in comparison with the fifth gear, switch the clutch C to the neutral position. It is possible to transmit torque along the casing of the planetary wheel or the carrier. From the input shaft 2 to the driven gear 13 and the ring gear 46, the torque arrives as in the first and the fifth gears, through the clutch D (upper right view) on the ring gear 32, on the sleeve shaft 32, carrier 30, shaft 4, ring gear 41, clutch 43, ring gear 42 and the output shaft 5. The gear ratio of the seventh gear:

$U_{7g}=U_1U_2U_3U_{bh}=1.32\times1.32\times2.29=4.0$; lq 4.0=0.6.

Eighth gear: PM works in summing mode (a+h). Compared with the fourth gear, switch clutch A to the neutral position. From the input shaft 2, through the gear ring 21 and the clutch 22 (B) the torque from the left ring reaches the pair of gears II (7-12), through the clutch C (51) to the output shaft 3 and the sun gear 27 (a). The second power flow from the ring gear 21 along the right clutch ring 22 (B) enters the ring gear 25, the sleeve shaft 26 and the pair of gears III (8-13), from the gear 13 through the gear crown 46, clutch D (47) to the ring gear 45 (lower average view), then correspondingly to the fourth gear. The summing capabilities of the PM, in this case, are described by the relationship: $n_b=U_{bh}{}^a n_h+U_{ba}{}^h n_a=n_h(K+1)/K-n_a/K$, where n_a, n_h, n_b – the frequency of rotation of the sun gear, carrier and epicyclic wheel in rpm.

Conditionally assume $n_{in}=n_2=1{,}000$ rpm. In this case, the frequency of rotation of the sun gear $n_a=n_2/U_2=1{,}000/(1/1.32)=1{,}320$ rpm. The frequency of rotation of the carrier $n_h=n_2/U_3=1{,}000/2.29=436.7$ rpm.

The rotational speed of the planetary wheel 33 (b) and the output shaft 5 will be: $n_b=n_h(K+1)/K-n_a/K=436.7(6/5)-1{,}320/5=524-264=260$ rpm. The gear ratio of the eighth gear MCP: $u_{8g}=n_2/n_b=1{,}000/260=3.85$; lq 3.85=0.585.

Ninth gear: the sun gear 27 (a) is stopped by the secondary shaft 3, the ring gear 50, the clutch C (51) relative to the ring gear 48 of the casing 1. From the input shaft 2 to the driven gear 13 and the ring gear 46, the torque arrives correspondingly as at the first and the fifth gear, through clutch D on the ring gear 32, on the sleeve shaft 31, carrier 30 (h), axis 29, satellites 28 on the planetary wheel 33 (b), casing 34, sleeve shaft 37, the ring gear 38, clutch F (43) on the ring gear 42 and the output shaft 5. The gear ratio of the ninth gear:

$U_{9g}=U_1U_2U_3U^a{}_{hb}=1.32\times1.32\times2.29\times(5.0/6.0)=3.32$; lq 3.32=0.52.

Tenth gear: PM operates in summing mode (a+b). From the input shaft 2 through the ring gear 17 and the clutch 20 (A), the torque enters the reverse gears (9-16 and 15-10), through the clutch C (51) to the output shaft 3 and the sun gear 27 (a). The second power flow through the clutch 22 (B) enters the pair of gears III (8-13), from gear 13 through the ring gear 46, clutch D (47) to ring gear 36 (lower right view), axial shaft 35, casing 34 and the epicyclic wheel 33 (b), from the sun gear 27 (a) and the planetary wheel 33 (b) by satellites 28, the forces are transmitted to the axis 29, the carrier 30 (h), the shaft 4, the ring gear 41, the clutch F (43) on the ring gear 42 and the output shaft 5. The summation capabilities of the PM, in this case, are described by the relation: $n_h=U_{ha}{}^b n_a+U_{hb}{}^a n_b=n_a/(K+1)+(n_b K)/(K+1)$, where n_a, n_h, n_b – the frequency of rotation of the sun gear, carrier and epicyclic wheel in rmp. Assume $n_{in}=n_2=1{,}000$ rpm. In this case, the frequency of rotation of the sun gear $n_a=n_2/U_R=1{,}000/3=-333.3$ rpm. The minus sign indicates a change in the direction of rotation. The rotational speed of the epicyclic wheel $n_b=n_2/U_3=1{,}000/2.29=436.7$ rpm.

The rotational speed of the carrier n_h and the output shaft 5 will be:

$n_h=n_a/(K+1)+(n_bK)/(K+1)=-333.3/6.0+436.7(5/6)=-55.56+363.8=308.2$ rpm.

The gear ratio of the tenth gear: $u_{10g}=n_2/n_h=1{,}000/308.2=3.24$; lq 3.24=0.511.

Eleventh gear: the PM operates in summing mode (a+b).

From the input shaft 2 through the ring gear 17 and the clutch 20 (A), the torque enters the pair of gears I (6-11), through the clutch C

(51) to the output shaft 3 and the sun gear 27 (a). The second power flow through the sleeve shaft of the train enters the pair of gears II (12-7), the sleeve shaft 24 onto the ring gear 23, and through the clutch 22 (B) (upper-middle view) enters the gear pair III (8-13), from gear 13 on ring gear 46, clutch D (47) to gear ring 36 (lower right view), axial shaft 35, casing 34 and to planetary wheel 33 (b), from sun gear 27 (a) and planetary wheel 33 (b) by the satellites 28, the forces are transmitted to the axis 29, the carrier 30 (h), the shaft 4, the ring gear 41, the clutch F (43) to the ring gear 42 and the output shaft 5.

Summing capabilities of the PM are described by the dependence:

$n_h=U_{ha}{}^{b}n_a+U_{hb}{}^{a}n_b=n_a/(K+1)+(n_b\ K)/(K+1)$, where n_a, n_h, n_b – the frequency of rotation of the sun gear, carrier and planetary wheel in rpm.

Assume $n_{in}=n_2=1{,}000$ rpm. In this case, the frequency of rotation of the sun gear $n_a=n_2/U_1=1{,}000/1.32=757.6$ rpm. The rotational speed of the epicyclic wheel: $n_b=n_2/U_1U_2U_3=1{,}000/(1.32\times1.32\times2.29)=250.6$ rpm.

The speed of the carrier n_h and the output shaft 5 will be:

$n_h=n_a/(K+1)+(n_bK)/(K+1)=757.6/6.0+250.6(5/6)=126.3+208.7=335$ rpm.

The gear ratio of the eleventh gear: $u_{11g}=n_2/n_h=1{,}000/335=3$; lq 3=0.475.

Twelfth gear: the sun gear 27 (a) is stopped by the mainshaft 3, the ring gear 50, the clutch C (51) relative to the ring gear 48 of the casing 1. From the input shaft 2 to the driven gear 13 and the ring gear 46, the torque arrives as on the second gear, then as on the fifth gear. The gear ratio of the twelfth MG gear: $u_{12g}=U_3U^{a}{}_{bh}=2.29\times(6.0/5.0)=2.75$; lq 2.75=0.44

Thirteenth gear: PM works in summing mode (a+h). From the input shaft 2 through the ring gear 17 and the clutch 20 (A), the torque enters the pair of gears I (6-11) via the clutch C (51) to the output shaft 3 and the sun gear 27 (a), as in fourth and tenth gears. The second power flow from the ring gear 21 through the clutch 22 (B) enters the ring gear 25 of the sleeve shaft 26 (top-right view), the pair of gears III (8-13), from gear 13 through the ring gear 46, clutch D (47) on the ring gear 32, the sleeve shaft 31 and the carrier 30 (h). From the sun gear 27 (a) and the carrier 30 (h) by the satellites 28, the forces are transmitted to the epicyclic wheel 33 (b), casing 34, sleeve shaft 37, the ring gear 38, clutch F (43) to the ring gear 42 and output shaft 5.

Summing capabilities of the PM are described by the dependency:

$n_b=U_{bh}{}^{a}n_h+U_{ba}{}^{h}n_a=n_h(K+1)/K-n_a/K$, where n_a, n_h, n_b are the rotational speeds of the sun gear, carrier and epicyclic wheel in rpm.

Conditionally accept $n_{in}=n_2=1{,}000$ rpm. In this case, the frequency of rotation of the sun gear $n_a=n_2\backslash U_1=1{,}000/1.32=757.6$ rpm. The frequency of rotation of the carrier $n_h=n_2/U_3=1{,}000/2.29=436.7$ rpm.

The rotational speed of the epicyclic wheel 33 (b) and the output shaft 5 will be: $n_b=n_h(K+1)/K-n_a/K=436.7(6/5)-757.6/5=524-151.5=372.5$ rpm;

The gear ratio of the thirteenth gear: $u_{13g}=n_2/n_b=1{,}000/372.5=2.68$; lq 2.68=0.43.

Fourteenth gear: from the input shaft 2 to the epicyclic wheel 33, the torque arrives as in the twelfth gear, then along the casing 34, along the sleeve shaft 37, then as in the ninth gear. The gear ratio of the fourteenth gear:

$u_{14g}=U_3\times U_{(b\text{-}b)}=2.29\times1.0=2.29$; lq 2.29=0.36.

Fifteenth gear: PM operates in summing mode (a+b). From the input shaft 2 through the ring gear 17 and the clutch 20 (A), the torque enters the pair of gears I (6-11), the clutch C (51) to the mainshaft 3 and the sun gear 27 (a). The second power flow from the ring gear 21 of the input shaft 2 through the clutch 22 (B) enters the pair of gears III (8-13), from the gear 13 through the ring gear 46, the clutch D (47) to the ring gear 36, axial shaft 35, casing 34 and the epicyclic wheel 33 (b), from the sun gear 27 (a) and the planetary wheel 33 (b) by satellites 28 the forces are transmitted to the axis 29, the carrier 30 (h), shaft 4 and clutch F (43) to the gear ring 42 and the output shaft 5.

The summing capabilities of the PM un this case are described by the relationship: $n_h=U_{ha}{}^{b}/n_a+U_{hb}{}^{a}n_b=n_a/(K+1)+(n_bK)/(K+1)$, where n_a, n_h, n_b are the rotation frequencies of the sun gear, carrier and planetary wheel in rpm.

Accept $n_2=1{,}000$ rpm. In this case, the frequency of rotation of the sun gear $n_a=n_2/U_1=1{,}000/1.32=757.6$ rpm. The rotation speed of the epicyclic wheel $n_b=n_2/U_3=1{,}000/2.29=436.7$ rpm.

Rotating frequency of the carrier n_h and the output shaft 5 is:

$n_h=n_a/(K+1)+(n_bK)/(K+1)=757.6/6.0+436.7(5/6)=126.3+363.8=490$ rpm

The gear ratio of the fifteenth gear: $u_{15g}=n_2/n_h=1{,}000/490=2.04$; lq 2.04=0.31.

Sixteenth gear: the sun gear 27 (a) is stopped by the mainshaft 3, the ring gear 50, clutch C (51) as relating to ring gear 48 of casing 1. From the input shaft 2, torque arrives at the gear pair III as in the ninth gear, then as in the previous gear. The gear ratio of the sixteenth gear: $u_{16g}=U_3\times U^{a}{}_{hb}=2.29\times0.833=1.9$; lq 1.9=0.28.

Seventeenth gear: PM works in summing mode (a+b). From the input shaft 2 through the ring gear 21 and the clutch 22 (B), the torque enters the pairs of gears II (7-12) and III (8-13), through the clutch C (51) on the output shaft 3 and the sun gear 27 (a). Then the second and total power flows are transmitted the same way as in the fifteenth gear. The summing capabilities of the PM, in this case, are described by the relationship: $n_h=U_{ha}{}^{b}n_a+U_{hb}{}^{a}n_b=n_a/(K+1)+(n_b\ K)/(K+1)$.

Assume $n_2=1{,}000$ rpm. In this case, the frequency of rotation of the sun gear $n_a=n_2/U_2=1{,}000/0.76=1{,}320$ rpm; rotating frequency of the planetary wheel $n_b=n_2/U_3=1{,}000/2.29=436.7$ rpm. The rotating frequency of the carrier n_h and the output shaft 5 $n_h=n_a/(K+1)+(n_bK)/(K+1)=1{,}320/6.0+436.7(5/6)=220+363.8=583.8$ rpm. The gear ratio of seventeenth gear: $u_{17g}=n_2/n_h=1{,}000/583.8=1.71$; lq 1.71=0.234.

Eighteenth gear: PM works in summing mode (a+h). Compared with the thirteenth gear, switch clutch A to the left (L) position. From the input shaft 2 through the ring gear 17 and the clutch 20 (A), the torque enters the pair of reverse gears (9-16 and 15-10), the clutch C (51) to the output shaft 3 and the sun gear 27 (a), then the second and total power flows are received as in the thirteenth gear. The summing possibilities of PM, in this case, are described by the dependence: $n_b=U_{bh}{}^a n_h+U_{ba}{}^h n_a=n_h(K+1)/K-n_a/K$.

Accept $n_{in}=n_2=1{,}000$ rpm. In this case, the frequency of rotation of the sun gear $n_a=n_2/U_R=1{,}000/3.0=333.3$ rpm; the frequency of rotation of the carrier $n_h=n_2/U_3=1{,}000/2.29=436.7$ rpm.

The rotational frequency of the epicyclic wheel 33 (b) and the output shaft 5 will be: $n_b=n_h(K+1)/K-n_a/K=436.7(6/5)-333.3/5=524-66.67=590.7$ rpm. The gear ratio of the eighteenth gear: $u_{18g}=n_2/n_b=1{,}000/590.7=1.69$; lq 1.69=0.229.

Nineteenth gear: The sun gear 27 (a) is stopped by the mainshaft 3, the ring gear 50, the clutch C (51) relative to the ring gear 48 of the casing 1. From the input shaft 2 through the ring gear 17 and the clutch 20 (A) the torque enters the pair of gears I (6-11), then the same as in the fifth and twelfth gears. The gear ratio of nineteenth gear: $u_{19g}=U_1\times U^a{}_{bh}=1.32\times1.2=1.584$; lq 1.584=0.2.

Twentieth gear: from the input shaft 2 to the carrier 30, the torque enters from the input shaft 2 through the clutch A to the pair of gears I, then through the ring gear 44, clutch D to the ring gear 32, sleeve shaft 31 and to the carrier 30, then through the shaft 4 the same way as in the seventh gear. The gear ratio of the twentieth gear:

$U_{20g}=U_1\times U_{(h-h)}=1.32\times1.0=1.32$; lq 1.32=0.12.

Twenty-first gear: the pair of gears I and PM are switched on in the $U^a{}_{hb}$ mode. The sun gear 27 (a) is stopped by the mainshaft 3, the ring gear 50, clutch C (51) relative to the ring gear 48 of casing 1. From the input shaft 2, torque arrives at the pair of gears I as in the thirteenth gear, then as in the sixteenth gear. The gear ratio of the twenty-first gear:

$U_{21g}=U_1\times U^a{}_{hb}=1.32\times0.833=1.1$; lq 1.1=0.041.

Twenty-second gear: the sun gear 27 (a) is stopped by the mainshaft 3, the ring gear 50, the clutch C (51) relative to the ring gear 48 of the casing 1. From the input shaft 2 through the ring gear 21 and the clutch 22 (B) the torque enters the pair of gears II 7-12, then as in nineteenth gear. The gear ratio of the twenty-second gear:

$U_{22g}=U_2\times U^a{}_{bh}=(1/1.32)\times1.2=0.91$; lq 0.91=-0.04.

Twenty-third gear: from the input shaft 2 to the carrier 30, the torque enters from the input shaft 2 through the clutch B to the pair of gears II, then the same as in the twentieth gear. The gear ratio of the twenty-third gear:

$U_{23g}=U_1\times U_{(h-h)}=(1/1.32)\times1.0=0.758$; lq 0.758=-0.12.

Twenty-fourth gear: the sun gear 27 (a) is stopped by the mainshaft 3, the ring gear 50, clutch C (51) relative to ring gear 48 of casing 1. From the input shaft 2, the torque flows to the pair of gears II as in previous gears, then the same way as in the twenty-first gear. The gear ratio of the twenty-fourth gear: $U_{24g}=U_2\times U^a{}_{hb}=(1/1.32)\times0.833=0.63$; lq 0.63=-0.2.

1R – first reverse gear: PM is in $U^b{}_{ah}=K+1$ mode. From the drive motor, through the input shaft 2 (Fig. 1), the gear ring 17 and the clutch 20 (A) the torque enters the pair of reverse gears 9-16 and 15-10, through the sleeve shaft of the train to the pair of gears II (12-7), through the sleeve shaft 24, thorough the ring gear 23, through the clutch 21 (B) to the ring gear 25, sleeve shaft 26, to the pair of gears III (8-13), through the ring gear 46, clutch 47 (D) to the ring gear 45 and the mainshaft 3, then to the PM sun gear 27 (a). The epicyclic wheel 33 (b) of the casing 34, the sleeve shaft 37, the ring gear 38, the clutch 40 (E) is fixed relative to the ring gear 39 of the casing 1. The sun gear 27 (a) rotates the satellites 28, which, rolling around the stopped planetary wheel 33 (b), rotate the carrier 30 (h), shaft 4, ring gear 41, clutch F (43) on the output shaft 5 with a low rotation frequency and an increased torque in the opposite direction to the input shaft. The gear ratio of the first reverse gear of MG:

$u_{1R}=U_RU_2U_3U^b{}_{ah}=-(3.0\times1.32\times2.29\times6.0)=-54.4$; lq 54.4=1.74.

2R – second reverse gear: compared with first reverse gear, switch clutches B and D to the neutral position, clutch C to the right position disconnect a pair of forward gears. A pair of reverse gears and a PM in $U^b{}_{ah}=K+1$ mode are switched on.

From the drive motor, through the input shaft 2, the ring gear 17 and the clutch 20 (A), the torque enters a pair of reverse gears 9-16 and 15-10, through the sleeve shaft of the train, through the gear rim 49, the clutch 51 (C) to the ring gear 50 and the output shaft 3, then as in the previous gear. The gear ratio of the second MG reverse gear 2R:

$U_{2R}=U_RU^b{}_{ah}=-3.0\times6.0=-18.0$; lq 18.0=1.255.

3R – third reverse gear: the PM works in summing mode (a+ h). From the input shaft 2 through the ring gear 17 and the clutch 20 (A), the torque flows to a pair of reverse gears (9-16 and 15-10), through the clutch C (51) to the mainshaft 3 and the sun gear 27 (a). The second power flow through the sleeve shaft of the train to the pair of gears II (12-7), through the sleeve shaft 24, through the ring gear 23, through the clutch 22 (B) to the ring gear 25 (upper-middle view), sleeve shaft 26, to the pair of gears III (8-13), through the ring gear crown 46, clutch 47 (D) to the ring gear 32 (top-right view), sleeve shaft 31, carrier 30 and axle 29. The satellites 28 transmit the force from sun gear 27 and carrier 30 to the epicyclic wheel 33, the casing 34, sleeve shaft 37 and the ring gear 38, through the clutch F to the ring gear 42 and the output shaft 5.

The summing capabilities of the PM, in this case, are described by the relationship: $n_b=U_{bh}{}^a n_h+U_{ba}{}^h n^a=n_h(K+1)/K-n_a/K$;

Conditionally accept $n_{in}=n_2=1{,}000$ rpm. In this case, the rotating frequency of the sun gear $n_a=n_2/U_R=1{,}000/(-3)=-333.3$ rpm. The rotating frequency of the carrier $n_h=n_2/U_RU_2U_3=1{,}000/(-3\times1.32\times2.29)=-110.25$ rpm.

The rotating frequency of the epicyclic wheel 33 (b) and the output shaft 5 is: $n_b=n_h(K+1)/K-n_a/K=-110.25(6/5)-(-333.3/5)=-132.3+66.67=-65.63$ rpm;

The gear ration of the third MG reverse gear 3R:

$u_{3R}=n_2/n_b=1{,}000/-65.63=-15.237$; lq 15.237=1.183.

4R – fourth reverse gear: pairs of reverse gears, pairs of forward gears II and III, and PM are put into operation in $U^a{}_{bh}=(K+1)/K$ mode. The sun gear 27 (a) by the mainshaft 3 and the ring gear 50 is fixed by a clutch (51) relative to the ring gear 48 of the MG casing 1. From the drive motor on the input shaft 2 through the ring gear 46 of the gear 13, torque is transmitted as in the first reverse gear, then through the clutch 47 (D) to the ring gear 36 and the axial shaft 37, casing 34, epicyclic wheel 33 (b) which rotates the satellites 28, run around the stopped sun gear 27 and rotate the carrier 30 (h), then as in the first reverse gear. The gear ratio of the fourth reverse gear 1R of MG: $u_{4R}=U_R U_2 U_3 U^a{}_{bh}=-3.0\times1.32\times2.29\times1.2=-10.9$; lq 10.9=1.04

5R – fifth reverse gear: from the drive motor to the ring gear 46 of the gear 13, the torque arrives as in the previous gear, then through the ring gear 46, clutch 47 (D) to the ring gear 32, sleeve shaft 31, carrier 30 (h), shaft 4, then by a ring gear 41, clutch F (43) to the ring gear 42 and an output shaft 5 of MG. The gear ratio of the fifth reverse gear of MG: $u_{5R}=U_R U_2 U_3 U_{hh}=-3.0\times1.32\times2.29\times1.0=-9.07$; lq 9.07=0.96.

6R – sixth reverse gear: from the drive motor to the ring gear 46 of the gear 13, the torque arrives as in previous gears, through the gear ring 46, the clutch 47 (D) to the ring gear 32, the sleeve shaft 31, the carrier 30 by axles 29 rotates the satellites 28 relatively to the stopped sun gear 27 (a), the satellites 28 rotate the epicyclic wheel 33 (b), the casing 34, the sleeve shaft 37, the ring gear 38 by the clutch F (43) to the ring gear 42 and the MG output shaft 5. The gear ratio of 6R gear of MG:

$u_{6R}=U_R U_2 U_3 U^a{}_{hb}=-3{,}0\times1.32\times2.29\times0.833=-7.55$; lq 7.55=0.88.

7R – seventh reverse gear: PM operates in summing mode (a+b). From the input shaft 2 to the sun gear 27 (a) and to the pair of gears III (8-13), gear 46 the torque is transmitted as in third reverse gear, then through the clutch 47 (D) to the ring gear 36 (upper-lower view), axial shaft 35, casing 34, to the epicyclic wheel 33. By the satellites 28, the force from the sun gear 27 and the epicyclic wheel 33 are transmitted to the axis 29, carrier 30, shaft 4, the ring gear 41, by the clutch F to the ring gear 42 and output shaft 5. Summing capabilities of the PM, in this case, are described by the dependence: $n_h=n_a/(K+1)+(n_b K)/(K+1)$.

Conditionally accept $n_{in}=n_2=1{,}000$ rpm. In this case, the rotating frequency of the sun gear $n_a=n_2/U_R=1{,}000/(-3)=-333.3$ rpm; the rotational speed of epicyclic wheel 33 (b):

$n_b=n_2/U_R U_2 U_3=1{,}000/(-3\times1.32\times2.29)=-110.25$ rpm.

The rotating frequency of the carrier n_h and the output shaft 5 will be:

$n_h=n_a/(K+1)+(n_b K)/(K+1)=-333.3/6.0+(-110.25)(5/6)=-55.56-91.84=-147$ rpm.

The gear ratio of the seventh MG reverse gear 7R:

$u_{7R}=n_2/n_b=1{,}000/-147.4=-6.78$; lq 6.78=0.83.

8R – eighth reverse gear: a pair of reverse gears and PM gears in $U^a{}_{bh}=(K+1)/K$ mode are put into service. The sun gear 27 (a) by the mainshaft 3, the ring gear 50 is fixed by the clutch (51) relative to the ring gear 48 of the casing 1 of MG. From the drive motor through the input shaft 2, through the gear ring 17, the clutch 20 (A), by the reverse gears 9-16 and 15-10 to the ring gear 44 of the gear train, the torque is transmitted as in second reverse gear, then along the clutch 47 (D) to the ring gear 36 and axial shaft 37, casing 34, epicyclic wheel 33 (b), which rotates the satellites 28,. They run around the stopped sun gear 27 and rotate the carrier 30 (h), then as in the fourth reverse gear. The gear ratio of the eighth MG reverse gear 8R:

$u_{8R}=U_R U^a{}_{bh}=-3.0\times1.2=-3.6$; lq 3.6=0.556.

9R – ninth reverse gear: from the drive motor to the ring gear 44 of the gear unit, torque arrives as in the previous gear, through the ring gear 44, clutch 47 (D) to the ring gear 32, sleeve shaft 31, clutch 30 (h), shaft 4, to the ring gear 41, then through the clutch F (43) to the ring gear 42 and an MG output shaft 5. The gear ratio of the MG ninth reverse gear 9R: $u_{5R}=U_R U_{hh}=3.0\times1.0=3.0$; lq 3.0=0.477.

10R – tenth reverse gear: from the drive motor to the ring gear 44 of the gear train, the torque arrives as in previous gears, through the ring gear 44, clutch 47 (D) to the ring gear 32, sleeve shaft 31, carrier 30, axles 29 rotates the satellites 28 relatively to the stopped sun gear 27 (a), the satellites 28 rotate the epicyclic wheel 33 (b), the casing 34, the sleeve shaft 37, the ring gear 38, the clutch F (43) to the ring gear 42 and the MG output shaft 5. The gear ratio of the tenth MG reverse gear 10R: $u_{10R}=U_R U^a{}_{hb}=3.0\times0.833=2.5$; lq 2.5=0.4.

When K – the internal parameter of the PM – is changed and the gear ratios of the pairs of gears are changed, the distribution of gear ratios over the range will be changed as well. By increasing the gear ratios of pairs of gears, one can reduce the number of override gears for which the gear ratio is less than unity. Redistribution of summing gears is possible under changes in the output shaft rotation.

A compact transmission unit with a large range of gear ratios $D=u_1/u_{22}=24.0/0.63=38.1$ creates opportunities for the operation of the vehicle under a variety of conditions. Besides, there are crawler gears with high gear ratios that allow low speeds, for example, for maneuvering during loading and unloading.

Small intervals of gear ratios between adjacent gears $q=u_i/u_{i+1}=1.2$ provide the convenience of shifting gears.

III. RESULT ANALYSIS

Technical and economic assessment of non-coaxial shaft-planetary multiple-speed gearbox, type 24R10

The task is solved by compaction of the layout of the MG and the operation of the PM both in gear and summing modes when the torque enters the two links and is removed from the third link.

The achievement of the technical result is ensured due to the fact that the non-coaxial shaft-planetary gearbox contains a simple three-link PM. Three pairs of direct gears with ring gears are freely mounted on the input of the PM on two parallel shafts. The sun gear unit is mounted on the mainshaft and two gears of the direct gears are freely installed. A ring gear with a three-position striker clutch is fixed on the input shaft above the gears train. A reverse gear, a three-position ring gear, is located on the gear train. Next to it, ring gears are fixed on the mainshaft, on the sleeve shaft of the carrier, on the axial shaft of the epicyclic wheel. There is a three-position ring gear mounted on the third-row driven gear; a multi-position striker clutch is installed between these ring gears. Between the third and fourth gear rows of the mainshaft, there is a five-position striker clutch fixed for random connection with ring gears: four-position sleeve shaft of the second-row gear and three-position drive gear of the third row. On the carrier shaft mounted in the output ring gear base, a ring gear with a three-position clutch is fixed next to it on the sleeve shaft of the epicyclic wheel and on the gearbox casing with the striker clutch. The reverse drive gear is freely mounted next to the ring gear of the input shaft and meshes with the gear train of the unit mounted on the reverse axis. The gear meshed to the gearbox is also attached to the driven gear train of three output shaft's gears.

IV. CONCLUSION

The design of a non-coaxial multiple-speed shaft-planetary gearbox, such as 24R10, relates to transport engineering, transmission of transport vehicles, land transport systems, including tractor trains, construction and road vehicles, tractors, ground-based special-purpose vehicles, etc.

The non-coaxial shaft-planetary gearbox contains two shafts, on which two rows of reverse gears and three pairs of direct gears are freely installed, and a simple three-link PM consisting of a carrier with satellites that are engaged with the sun gear and epicyclic wheel. Six striker clutches provide the planetary gear with four modes: slow, delaying, direct and accelerating. The proposed design of the non-coaxial shaft-planetary multiple-speed gearbox of 24R10 type provides an improvement in the layout and performance characteristics of the gearbox and, accordingly, the operational capabilities of the vehicle.

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