# GEAR RATIOS CALCULATION FOR PLANETARY-LAYSHAFT TRANSMISSIONS WITH THREE POWER-FLOWS

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**Abstract.** One of the main ways to improve fuel economy of vehicles is to provide operation of the engine in a narrow range with the best efficiency. The number of speeds in modern automatic transmissions has significantly increased for this reason. Obviously, an increase in the number of speeds leads to an increase in the number of gears and shifting elements, transmission complication and an increase in overall dimensions and weight. The article discusses the planetary-layshaft transmission, which has parallel connection of two differentials and three sets of gears with fixed axles of wheels. Transmissions of such arrangement allow to transmit the power in parallel by three flows from the input shaft to the output one and provide a significantly larger number of speeds with fewer gears and shifting elements, due to combinations of different operating modes of the differentials. Applying of planetary-layshaft transmissions limits the presence of double and triple transition shifts between adjacent speeds, when two or three pairs of control elements (clutches or brakes) are simultaneously switching off and on during gearshift. The purpose of the article is to find a sequence of single transition shifts for a planetary-layshaft transmission and to calculate the gear ratios that provide a set of speed ratios close to the given.

**Keywords:** transmission, gearbox, planetary-layshaft, gear shift, single transition shift, gear ratio.

## Introduction

Parallel connection several gears with fixed axles of gear wheels (layshaft gears) through differential epicycle gears significantly expands the functionality of the mechanisms, the range and the number of speeds. Changing the differentials' operation modes and switching gears with fixed axes allow the latter to work both separately and in parallel with each other. The using of differentials and layshaft gears connection has an additional advantage in obtaining compact transmissions that are composed of simple mechanisms.

Transmission structures with one and two differentials having two and three power flows correspondingly and only one layshaft gear set were studied in [1;2]. The work [3] is devoted to the analysis and synthesis of planetary-layshaft transmissions with one differential and two layshaft gear sets installed in each power flow and variable operating modes of the differential. Transmissions with two power flows allow to realize a greater number of speeds as compared with transmissions consisting of the mechanisms of the same type. A double reduction in the loads acting on the internal parts of the transmission is achieved with parallel power pass along two branches [4]. Planetary-layshaft transmissions contain two differentials and three sets of gears with fixed axles, it allows to get 2.5-4 times more speeds compared to the transmissions with two power flows [5] and the loads reduction acting on the internal links is achieved by more than 65 % on three stream mode [6].

The main lack of the planetary-layshaft transmissions is double and triple transition shifts [7], when two or three pairs of shifting elements (clutches or brakes) are simultaneously switching off and on during gearshift between neighbour speeds. It relates with changing the differential operation modes as it does on heavy-duty vehicles, which have epicycle range change unit for changing the operating range of the main 3 or 4-speed gearbox [8]. The double or triple transition shifts occur during the range change in such transmissions. Such gear shifts lead to an increase in the speed shifting time and, consequently, a decrease in efficiency, comfort and controllability of the vehicle. One of the main challenges during transmission synthesis is to obtain the shift sequence, which has only single transition shifts between neighbour speeds.

This article discusses the synthesis of a kinematic diagram of a planetary-layshaft transmission with three power flows, drawing up a number of speed ratios implemented by a synthesized transmission, finding a sequence of single transition shifts of control elements and calculating gear ratios of gear pairs.

## Materials and methods

As shown in [5], only four planetary-layshaft transmission structures with two differentials are possible in principle. The IDD structure synthesized in [5] is shown in Fig. 1, the corresponding transmission kinematic diagram is shown in Fig. 2.

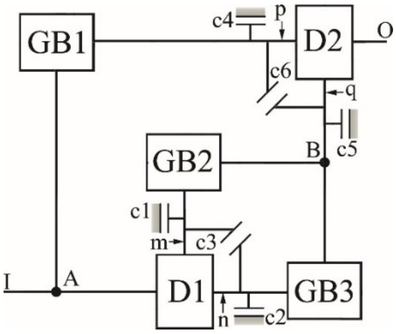


Fig. 1. **IDD structure of planetary-layshaft transmission with three power flows**

In the IDD structure (Fig. 1), D1, D2 are epicycle three-link mechanisms (differentials); GB1 – GB3 gear sets with fixed axles of gears, each GB has shifting elements, which engage the corresponding layshaft gear pair; the kinematic links between the units are shown by the straight lines in Fig. 1. When the control elements c1 - c6 are turned off and the GB1, GB2, GB3 are involved, the transmission implements a three-stream operation mode: the power flow is divided at the node A of the input shaft I between the differential D1 and GB1. The differential D1 additionally divides the power flow into two streams, which pass through GB2 and GB3 and are summed at the node B. Next, the power flows from the node B and GB1 are summed by the differential D2.

The brakes c1, c2, c4, c5 and clutches c3, c6 determine the operation of the differentials D1, D2: stopping one of the links or locking the differential. If one of the links of both differentials D1, D2 is stopped by the brakes or two links of each differential are coupled to each other by clutches, the transmission will operate in a one-stream mode. In the case of stopping one differential link or locking it, a two-stream mode is implemented.

Table 1

**Combinations of the elements involved, operating modes and formulas for calculating speed ratios from input I to output O for the IDD transmission structure (Fig. 1)**

| Operating mode | Elements switched on | | | Speed ratio calculation formula $i_{IO}$ |
|---|---|---|---|---|
| – | GB1 | – | c5 | $i_{GB1} / i^{q}_{Op}$ |
| | GB1 | – | c6 | $i_{GB1}$ |
| | GB2 | c2 | c4 | $i_{GB2} i^{n}_{Im} / i^{p}_{Oq}$ |
| | GB2 | c2 | c6 | $i_{GB2} i^{n}_{Im}$ |
| | GB2 | c3 | c4 | $i_{GB2} / i^{p}_{Oq}$ |
| | GB2 | c3 | c6 | $i_{GB2}$ |
| | GB3 | c1 | c4 | $i_{GB3} i^{m} / i^{p}_{Oq}$ |
| | GB3 | c1 | c6 | $i_{GB3} i^{m}_{In}$ |
| | GB3 | c3 | c4 | $i_{GB3} / i^{p}_{Oq}$ |
| | GB3 | c3 | c6 | $i_{GB3}$ |

Table 1 (continued)

| Operating mode | Elements switched on | | | Speed ratio calculation formula $i_{IO}$ |
|---|---|---|---|---|
| = | GB1 | GB2 | c2 | $1/\left(i_{Op}^{q}/i_{GB1}+i_{Oq}^{p}/\left(i_{GB2}i_{Im}^{n}\right)\right)$ |
| | GB1 | GB2 | c3 | $1/\left(i_{Op}^{q}/i_{GB1}+i_{Oq}^{p}/i_{GB2}\right)$ |
| | GB1 | GB3 | c1 | $1/\left(i_{Op}^{q}/i_{GB1}+i_{Oq}^{p}/\left(i_{GB3}i_{In}^{m}\right)\right)$ |
| | GB1 | GB3 | c3 | $1/\left(i_{Op}^{q}/i_{GB1}+i_{Oq}^{p}/i_{GB3}\right)$ |
| | GB2 | GB3 | c4 | $\left(i_{GB2}i_{Im}^{n}+i_{GB3}i_{n}^{m}\right)/i_{Oq}^{p}$ |
| | GB2 | GB3 | c6 | $i_{GB2}i_{Im}^{n}+i_{GB3}i_{In}^{m}$ |
| ≡ | GB1 | GB2 | GB3 | $1/\left(\frac{i_{Op}^{q}}{i_{GB1}}+\frac{i_{Oq}^{p}}{i_{GB2}i_{Im}^{n}+i_{In}^{m}i_{GB3}}\right)$ |

Possible combinations of the elements c1-c6 of the transmission structure (Fig. 1), operating modes ("–" – one-stream; " = " – two-stream; "≡" – three-stream) and formulas for calculating speed ratios $i_{IO}$ from input I to output O are presented in the Table 1. In the formulas of gear ratios: $i_{GB}$ is the gear ratio of corresponding GB; $i_{Im}^{n}$ – the gear ratio of the differential from the link I to link m under the assumption that the link n is stopped.

As it can be seen from Table 1, for the implementation of any speed, except for one-stream mode through GB1, three control elements must be switched on in the transmission. To get one-stream mode through GB1, it is enough to turn on two control elements (one in GB1 and brake c5 or clutch c6). At the same time, in order to reduce idle rotation losses, one of the unused control elements in GB2 or GB3, or one of the elements c1, c2, c3, may additionally be switched on.

If we denote that $l_1$, $l_2$, $l_3$ are the numbers of GB1, GB2, GB3 one-stream operation modes respectively; $l_4$, $l_5$, $l_6$ are the numbers of two-stream operation modes GB1 and GB2, GB1 and GB3, GB2 and GB3, respectively, than the number of speeds $N$ implemented by the transmission according to the IDD structure will be defined by the expression:

$$\begin{aligned} N = N_{GB1}\ l_1 + N_{GB2}\ l_2 + N_{GB3}\ l_3 + N_{GB1}\ N_{GB2}\ l_4 + \\ + N_{GB1}\ N_{GB3}\ l_5 + N_{GB2}\ N_{GB3}\ l_6 + N_{GB1}\ N_{GB2}\ N_{GB3} \end{aligned} \quad (1)$$

where $N_{GB1}$, $N_{GB2}$, $N_{GB3}$ – number of gear ratios in GB1, GB2, GB3 correspondingly.

Table 1 shows that $l_2 = l_3 = 4$; $l_1 = l_4 = l_5 = l_6 = 2$ in the IDD structure. Thus, if $N_{GB1} = N_{GB2} = N_{GB3} = 2$, for example, the transmission will have 52 speeds.

The variety of epicycle mechanisms, layshaft gears and ways to connecting their parts determine plenty kinematic diagrams of transmissions corresponding to the IDD structure under consideration (Fig. 1). To truncate this plenty, we introduce the following restrictions:

1. Let each GB have two gear pairs ($N_{GB1} = N_{GB2} = N_{GB3} = 2$) and one three positions synchronizer. The synchronizer connects the GB input link with the corresponding gear pair in the two extreme positions (a, b).
2. The simple epicycle mechanisms are used as D1 and D2, which contain three main parts: the sun gear, ring gear and carrier with satellites.
3. The D1 ring gear is connected to the input shaft I. The differential D1 links m and n are the carrier and the sun gear respectively.
4. The D2 carrier is connected to the output shaft O. The links p and q are connected to the D2 sun gear and the ring gear respectively.
5. The parasitic gears in GB3 introduced for unidirectional rotation of all transmission links due to the D1 sun gear rotates in the opposite direction relative to the ring when the carrier is stopped.

6. The controls c1, c2, c4 and c5 that are mainly used in one-stream modes will not be included in the synthesized kinematic diagram in order to increase the number of multi-stream modes among the transmission speeds.

These restrictions allow to build a kinematic diagram in Fig. 2 of a transmission, corresponding to the IDD structure (Fig. 1). The input shaft I and output shaft O rotate in the opposite directions.

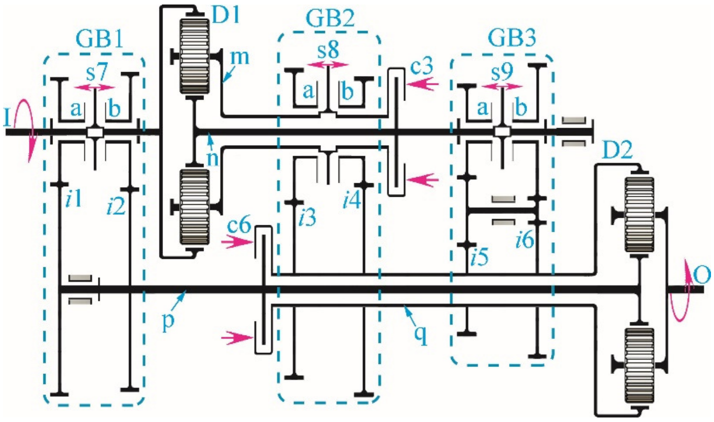


Fig. 2. **Transmission kinematic diagram corresponding to IDD structure**

The parameters $l_1 = l_2 = l_3 = l_4 = l_6 = 1$ and the parameter $l_5 = 0$, since the two-stream mode of the transmission via GB1 and GB3 with the c3 clutch engaged will result in unidirectional rotation of the links m and n of the differential D1 and, as a result, to the internal recirculation of power [1;2;4;9]. Substituting the parameters $l$ into expression (1), we obtain in the synthesized transmission 22 speeds, two of which are reverse speeds (one-stream operation mode via GB3 at switched on clutches c3, c6).

The restriction formulation is the basis of the proposed synthesis method of diagrams of planetary-layshaft transmissions. The more restrictions, the less possible options for the arrangement of mechanisms and the simpler the synthesis. Also, it must be noted that the set of accepted restrictions is largely determined by the technical specification, experience and intuition of the designer and it is aimed to distinguish the potentially acceptable diagrams with the required functions from the whole variety of diagrams for further consideration and analysis.

The parameter synthesis problem arises after obtaining the kinematic diagram. The problem consists in determining the gear ratios of the gear pairs, at which the transmission provides a specified or close to the specified set of speed ratios $i_{IO}$ from the input shaft to the output one.

The transmission in Fig. 2 has 8 independent parameters – internal gear ratios: $i1$, $i2$, $i3$, $i4$, $i5$, $i6$, $i_{nI}^{m}$ , $i_{pq}^{O}$ . The locations of one-, two- and three-stream operation modes in a set of speed ratios are unknown in general case. The independent parameters will be limited by the recommended [10] range for layshaft gears: $-3.0 \leq i1, i2, i3, i4 \leq -0.4$; $0.16 \leq i5, i6 \leq 9.0$ (since there are parasitic gears); for differentials: $-4 \leq i_{nI}^{m}, i_{pq}^{O} \leq -1.5$ .

A set of speed ratios without taking into account the reverse speeds is presented in Table 2. The set obtained with the following values of internal gear ratios: $i1$ = -1.3; $i2$ = -0.4; $i3$ = -0.5; $i4$ = -1.1; $i5$ = 2.5; $i6$ = 0.16; $i_{nI}^{m}$ = -1.5; $i_{pq}^{O}$ = -4.0.

The control elements switched on each speed are shown with symbols "□", "Δ", "o", "•", "x" in Table 2.

As it can be seen from Table 2, with sequential shifting between the speeds 2-3, 6-7, 7-8, 8-9, 10-11, 11-12, 15-16, 16-17 two pairs of controls and between the speeds 4-5, 9- 10, 12-13, 14-15 three pairs of controls are involved (double and triple transition shifts).

It was stated above that the one-stream mode via GB1 occurs by means of two controls in the IDD planetary-layshaft structure: the synchronizer in GB1 and the clutch c6, which blocks the differential D2. At the same time, the inclusion of one control element from the remaining three will not affect the speed ratio. This property is used for single transition shift to one-stream GB1 speeds (8th and 20th speeds in Table 2), the additionally controls switched on are shown in Table 2 in brackets.

Table 2

**Set of forward speed ratios for the gearbox in Fig. 2**

| Speed, operation mode | s7 | | s8 | | s9 | | c3 | c6 | $i_{IO}$ | Sequences with single transition shifts | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | a | b | a | b | a | b | | | | 1 (□) | 2 (Δ) | 3 (o) | 4(•) |
| 1 = | | | | □ | □ | | | □ | -3.40 | 1 | 1 | 1 | 1 |
| 2 ≡ | □ | | | □ | □ | | | | -2.58 | 2 | | 2 | 2 |
| 3 = | | | x | | x | | | x | -2.43 | | | | |
| 4 ≡ | □ | | □ | | □ | | | | -2.08 | 3 | | 3 | |
| 5 = | | | | Δ | | Δ | | Δ | -1.86 | | 2 | | |
| 6 ≡ | Δ | | | Δ | | Δ | | | -1.71 | | 3 | | |
| 7 ≡ | | • | | • | • | | | | -1.36 | | | | 3 |
| 8 – | □ | | (□) | (Δ) | | | | □ | -1.31 | 4 | 4 | 4 | |
| 9 ≡ | | • | • | | • | | | | -1.21 | | | | 4 |
| 10 = | Δ | | | Δ | | | Δ | | -1.09 | | 5 | | |
| 11 ≡ | | x | | x | | x | | | -1.07 | | | | |
| 12 – | | | | Δ | | | Δ | Δ | -1.05 | | 6 | | |
| 13 ≡ | □ | | □ | | | □ | | | -0.95 | 5 | | 5 | |
| 14 = | | | □ | | | □ | | □ | -0.89 | 6 | | | |
| 15 = | | Δ | | Δ | | | Δ | | -0.79 | | 7 | | |
| 16 ≡ | | □ | □ | | | □ | | | -0.72 | 7 | | | 5 |
| 17 = | o | | o | | | | o | | -0.54 | | | 6 | |
| 18 – | | | o | | | | o | o | -0.47 | | | 7 | |
| 19 = | | □ | □ | | | | □ | | -0.46 | 8 | 8 | 8 | 6 |
| 20 – | | □ | (□) | | | | | □ | -0.40 | 9 | 9 | 9 | 7 |

There are four dependent sequences identified as a result of the Table 2 analysis. In each sequence only one pair of controls is involved during gear shift. Three sequences contain 9 speeds and one sequence contains 7 speeds. These sequences are presented in the right column in Table 2 and denoted by the corresponding symbol. The speeds with the "x" symbol are not used in any of the four sequences with single transition shifts obtained. The sequence 4 (•) containing 7 speeds is not considered further in view of the smaller number of speeds.

Each of the three 9-speed sequences with single transition shifts contains a different number of single and multi-stream modes. The number of modes for each sequence is presented in Table 3.

Table 3

**Number of speeds in dependence of operation modes**

| Sequence | Gearbox operation mode | | |
|---|---|---|---|
| | – | = | ≡ |
| **1 (□)** | 2 | 3 | 4 |
| **2 (Δ)** | 3 | 5 | 1 |
| **3 (o)** | 3 | 3 | 3 |

As mentioned above, the loads transmitted by the transmission links on multi-stream modes are significantly lower than on one-stream modes. Therefore, preference should be given to the sequence with the smallest number of one-stream speeds. The best sequence for this criterion is sequence 1 (□).

Let $a_k$ speed ratios $i_{IO}$ set be given:

$$a_k = a_1, a_2, \ldots, a_9 , \tag{2}$$

where $k$ – speed number.

Moreover, the members of the $a_k$ set have negative values, since the rotation direction of the output shaft O is opposite to the direction of the shaft $I$ ($a_k < a_{k+1}$) in the transmission in Fig. 2.

The task of parameter synthesis is to determine the values of the parameters $i1$, $i2$, $i3$, $i4$, $i5$, $i6$, $i_{nI}^{m}$, $i_{pq}^{O}$, at which a set of speed ratios $a_k$ is achieved, satisfying the formulas in Table 1. If the members of a given set $a_k$ do not satisfy the formulas in Table 1, the transmission will not ideally satisfy the set (2). But one can find some close to (2) set $b_k$ by searching for the values $i1$, $i2$, $i3$, $i4$, $i5$, $i6$, $i_{nI}^{m}$, $i_{pq}^{O}$. The proximity of the sets $a_k$ and $b_k$ will be evaluated by the functions:

$$F_1 = \min \sum_{k=1}^{9} \left( \frac{a_k - b_k}{a_k} \right)^2 \tag{3.1}$$

$$F_2 = \min \sum_{k=1}^{9} \left( \frac{a_k / a_{k+1} - b_k / b_{k+1}}{a_k / a_{k+1}} \right)^2 = \min \sum_{k=1}^{9} \left( 1 - \frac{b_k a_{k+1}}{a_k b_{k+1}} \right)^2 \tag{3.2}$$

where $F_1$, $F_2$ – functions dependent on gear ratios $i1$, $i2$, $i3$, $i4$, $i5$, $i6$, $i_{nI}^{m}$, $i_{pq}^{O}$ only;
$b_k$ – set of gear ratios satisfying formulas in Table 1.

The function $F_1$ determines the sum of the quadratic relative deviation of the members of a given set from the set implemented by the transmission. The function $F_2$ is the sum of the quadratic relative deviation of a given step between the speed ratios of neighbour speeds and the step implemented by the transmission. The speed ratio and step are the most important indicators of the sequences. For the purposes of this work, we assume that one of the functions $F_1$ or $F_2$, which will have the minimum value, determines the set $b_k$ closest to the given.

There are various methods to find the extremum of the nonlinear functions $F_1$ and $F_2$ of several variables [11;12]. Many of them are implemented in modern mathematical computer software packages, such as Mathcad and MATLAB, which allow to simply and quickly solve optimization problems.

Using the described method, we will carry out parameter synthesis for each of the three 9-speed sequences with single transition shifts.

Let the next set $a_k$ (see Table 4) be given, which kinematic range is 10.9.

Table 4

**Given set of speed ratios**

| Speed | Ratio | Step |
|---|---|---|
| 1 | -5.45 | |
| 2 | -3.30 | 1.65 |
| 3 | -2.20 | 1.50 |
| 4 | -1.57 | 1.40 |
| 5 | -1.12 | 1.40 |
| 6 | -0.86 | 1.30 |
| 7 | -0.72 | 1.20 |
| 8 | -0.60 | 1.20 |
| 9 | -0.50 | 1.20 |

The values of the functions $F_1 = 0.059$, $F_2 = 0.023$ after minimizing (3.1) and (3.2) for the sequence 1 (□) were obtained. Function $F_2 = 0.023$ corresponds to the set given in Table 5 with the following gear ratios: $i1$ = -1.18; $i2$ = -0.4; $i3$ = -0.52; $i4$ = -2.12; $i5$ = 3.82; $i6$ = 0.16; $i_{nI}^{m}$ = -3.22; $i_{pq}^{O}$ = -2.58.

Table 5

**Set of speed ratios after minimization $F_2$ according to the sequence 1 (□)**

| Speed, operation mode | s7 | | s8 | | s9 | | c3 | c6 | $i_{IO}$ | Step |
|---|---|---|---|---|---|---|---|---|---|---|
| | a | b | a | b | a | b | | | | |
| 1 = | | | | □ | □ | | | □ | -3.97 | |
| 2 ≡ | □ | | | □ | □ | | | | -2.39 | 1.66 |
| 3 ≡ | □ | | □ | | □ | | | | -1.61 | 1.49 |
| 4 – | □ | | (□) | | | | | □ | -1.18 | 1.36 |
| 5 ≡ | □ | | □ | | | □ | | | -0.82 | 1.44 |
| 6 = | | | □ | | | □ | | □ | -0.73 | 1.12 |
| 7 ≡ | | □ | □ | | | □ | | | -0.60 | 1.23 |
| 8 = | | □ | □ | | | | □ | | -0.48 | 1.24 |
| 9 – | | □ | (□) | | | | | □ | -0.40 | 1.20 |
| R1 | | | | | □ | | □ | □ | 3.82 | |
| R2 | | | | | | □ | □ | □ | 0.16 | |

R1 and R2 in Table 5 and further are reverse speeds.

The values of the functions $F_1 = 0.011$, $F_2 = 0.023$ after minimizing (3.1) and (3.2) for the sequence 2 (Δ) were obtained. Function $F_1 = 0.011$ corresponds to the set given in Table 6 with the following gear ratios: $i1$ = -1.55; $i2$ = -0.52; $i3$ = -0.70; $i4$ = -0.88; $i5$ = 5.59; $i6$ = 2.68; $i_{nI}^{m}$ = -1.5; $i_{pq}^{O}$ = -1.5.

Table 6

**Set of speed ratios after minimization $F_1$ according to the sequence 2 (Δ)**

| Speed, operation mode | s7 | | s8 | | s9 | | c3 | c6 | $i_{IO}$ | Step |
|---|---|---|---|---|---|---|---|---|---|---|
| | a | b | a | b | a | b | | | | |
| 1 = | | | | Δ | Δ | | | Δ | -5.19 | |
| 2 = | | | | Δ | | Δ | | Δ | -3.25 | 1.60 |
| 3 ≡ | Δ | | | Δ | | Δ | | | -2.26 | 1.44 |
| 4 – | Δ | | | (Δ) | | | | Δ | -1.55 | 1.46 |
| 5 = | Δ | | | Δ | | | Δ | | -1.06 | 1.46 |
| 6 – | | | | Δ | | | Δ | Δ | -0.88 | 1.21 |
| 7 = | | Δ | | Δ | | | Δ | | -0.69 | 1.27 |
| 8 = | | Δ | Δ | | | | Δ | | -0.61 | 1.12 |
| 9 – | | Δ | (Δ) | | | | | Δ | -0.52 | 1.18 |
| R1 | | | | | Δ | | Δ | Δ | 5.59 | |
| R2 | | | | | | Δ | Δ | Δ | 2.68 | |

The values of the functions $F_1 = 0.047$, $F_2 = 0.009$ after minimizing (3.1) and (3.2) for the sequence 3 (o) were obtained. Function $F_2 = 0.009$ corresponds to the set given in Table 7 with the following gear ratios: $i1$ = -1.2; $i2$ = -0.4; $i3$ = -0.56; $i4$ = -2.19; $i5$ = 4.09; $i6$ = 0.16; $i_{nI}^{m}$ = -3.38; $i_{pq}^{O}$ = -2.52.

As a result, all three speed ratio sequences with single transition shifts have a close range: 9.93 for the row 1 (□); 9.98 for the row 2 (Δ); 9.85 for the row 3 (o).

Table 7

**Set of speed ratios after minimization $F_2$ according to the sequence 3 (o)**

| Speed, operation mode | s7 | | s8 | | S9 | | c3 | c6 | $i_{IO}$ | Step |
|---|---|---|---|---|---|---|---|---|---|---|
| | a | b | a | b | a | b | | | | |
| 1 = | | | | o | o | | | o | -3.94 | |
| 2 ≡ | o | | | o | o | | | | -2.36 | 1.66 |
| 3 ≡ | o | | o | | o | | | | -1.60 | 1.47 |
| 4 – | o | | (o) | | | | | o | -1.18 | 1.36 |
| 5 ≡ | o | | o | | | o | | | -0.85 | 1.37 |
| 6 = | o | | o | | | | o | | -0.65 | 1.32 |
| 7 – | | | o | | | | o | o | -0.55 | 1.18 |
| 8 = | | o | o | | | | o | | -0.50 | 1.11 |
| 9 – | | o | (o) | | | | | o | -0.40 | 1.25 |
| R1 | | | | | o | | o | o | 4.09 | |
| R2 | | | | | | o | o | o | 0.16 | |

## Results and discussion

Three-stream planetary-layshaft transmissions containing a parallel connection of two differentials and three sets of gears with fixed axles, compared with transmissions made up of mechanisms of the same type, realize a significantly larger number of speeds with a smaller number of controls, load reduction acting on the internal links is achieved by more than 65 % on the three stream mode.

The kinematic diagram of the transmission was synthesized by using one of the four possible structures of three-stream planetary-layshaft transmissions. The synthesized transmission implements 20 forward speeds and 2 reverse ones, contains only 5 controls, three of which are switched on at each speed, which allows reducing losses in control elements switched off.

The sequence of speed shift for synthesized 20-speed transmission was analysed in the article. The obtained sequence contains double and triple transition shift between some neighbor speeds. Such speed shifts are hard to control, lead to increase of the shifting time, lower the efficiency and controllability of the vehicle. Three sequences were identified, each of which contains nine speeds with single transition shifts between neighbour speeds. The obtained sequences were analysed by the number of one-, two- and three-stream speeds. The transmission gear ratios synthesis method was proposed. The gear ratios of the transmission mechanisms are obtained for each of the three sequences with single transition shifts.

The synthesized planetary-layshaft transmission has the same kinematic range and fewer controls compared with the 9-speed planetary transmission ZF 9HP [13].

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