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Synthesis planetary-layshaft transmission with single transition shifts

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Abstract. The more speeds in the automatic transmission, the better the fuel economy and the vehicle drivability. More speeds in conventional transmissions consisting only of planetary gear sets or only of gears with fixed axles lead to increase gears and shifting elements, which complicating the transmission design and growing the overall dimensions and weight. One of the ways to obtain simple transmission designs providing a great number of speeds with a relatively small number of gears and shifting elements is to use combinations of planetary gear sets and gears with fixed axles (planetary-layshaft transmissions). The main disadvantage that limits the use of planetary-layshaft transmissions is the presence of double and triple transition shifts in which 2 or 3 pairs of shifting elements are switched on and off at the same time. The problem to find out a set of speed ratios realized by the transmission with single transition shift sequences is solved in the paper on example of a planetary-layshaft transmissions consisting of two differentials and three sets of gears with fixed axles. The gear ratios magnitudes which satisfying obtained set of transmission speed ratios were found.

1. Introduction

One of the main ways to improve the fuel economy of vehicles is to ensure the engine operation in the best efficiency narrow range. The number of speeds in modern automatic transmissions has increased over the past 20 years from 4 to 8 - 10 for this reason.

Obviously, more speeds lead to a transmission design complication, an increase in overall dimensions and weight. The possible planetary-layshaft transmissions structures consisting of gears with fixed axles of gear wheels and two planetary differentials connecting in parallel were synthesized in paper [1]. It was also shown there that such transmissions make it possible to obtain a significantly more speeds at smaller number of gears and shifting elements used. In addition, a reduction in the loads on the internal parts of the transmission by more than 65% is achieved due to the division of power flow by means of differentials into three parallel branches [2]. The main disadvantage of the proposed planetary-layshaft transmissions with three power-flows is the presence of double and triple transition shifts, when not one pair of shifting elements is involved in the speed change, but two or three. Double and triple transition shifts lead to an increase in shifting time and, as a consequence, a decrease in efficiency, comfort and drivability of the vehicle [3].

The purpose of this work is to find a single transition shifts sequence of control elements for a planetary-layshaft transmission with three power-flows and to calculate gear ratios, providing set of transmission speed ratios close to a given.



2. Planetary-layshaft transmission IDDO

We use the planetary-layshaft structure IDDO shown in figure 1 as an example [1], in which I, O are the input and output links; D1, D2 - differentials; GB1, GB2, GB3 - sets of gears with fixed axes of gear wheels and shifting elements switching on these gears; m and n, p and q are the parts of the differentials D1 and D2, respectively; c1 – c6 – brakes and clutches of the differentials D1 and D2.

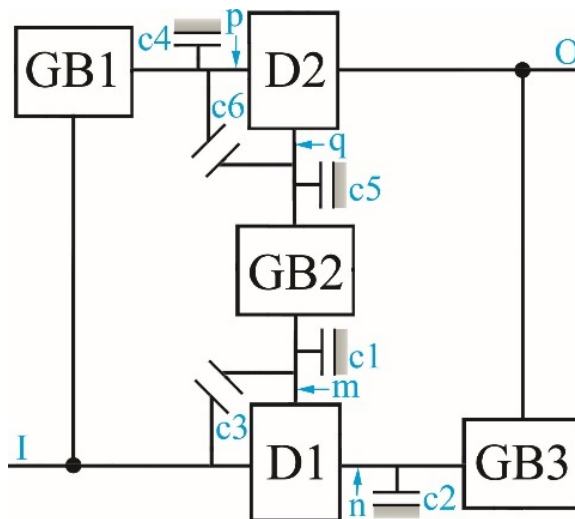


Figure 1. IDDO structure of planetary-layshaft transmission with three power-flows.

If in the IDDO structure (figure 1) the brakes c1, c2, c4 and c6, connecting the differentials parts m, n, p and q with a fixed body, are removed, then the structure will correspond, for example, to the transmission kinematic diagram presented in figure 2. Each GB contains 2 gears and a synchronizer, which switching on these gears. The parasitic gear wheels are additionally installed in GB3 to coordinate the rotation direction of the transmission inner links, since the sun gear n rotates in the opposite direction relative to the input shaft I. The shafts I and O rotate in different directions.

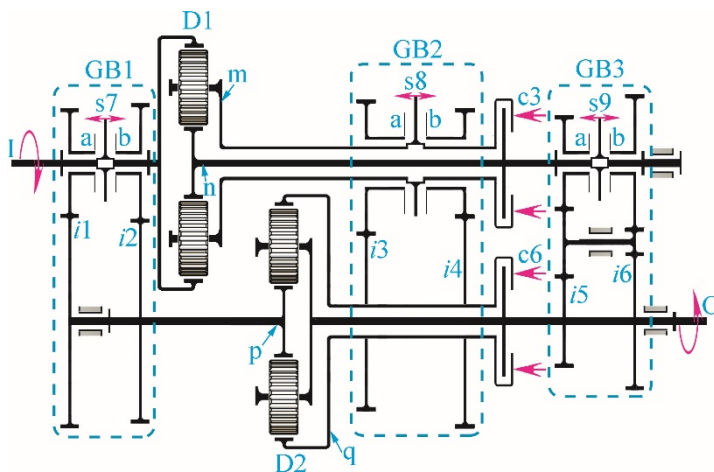


Figure 2. Kinematic diagram of planetary-layshaft transmission according to the IDDO structure.

In figure 2 i_1 , i_2 , i_3 , i_4 are negative gear ratios of the respective gear pairs; i_5 , i_6 – are positive gear ratios of gear pairs including the parasitic gears.

3. Transmission speed ratios

The transmission shown in figure 2 allows up to 20 forward and 2 reverse different speed ratios in accordance with the formula obtained in [1]. We distributed the twenty i_{IO} speed ratios implemented by the transmission by using the methodology from [4] from larger to smaller depending on the shifting elements switched-on. The results were entered into the table 1.

Table 1. Shifting elements sequence and speed ratios of the transmission shown in figure 2.

Speed	s7		s8		s9		c3	c6	i_{IO}	Sequences with single transition shifts		
	a	b	a	b	a	b				1 (x)	2 (Δ)	3 (o)
1			x		x			x	-8.345	1	1	1
2	x		x		x				-6.259	2		
3				o	o			o	-5.480			2
4	x			x	x				-5.129	3		3
5			Δ			Δ		Δ	-4.218		2	
6	Δ		Δ			Δ			-3.665		3	
7		x		x	x				-2.941	4		
8	Δ		(Δ)	(o)				Δ	-2.731		4	4
9	Δ		Δ				Δ		-2.401		5	
10			Δ				Δ	Δ	-2.285		6	
11		x	x		x				-1.977	5		
12	o			o	o				-1.529			5
13				—	—			—	-1.353			
14		x	x			x			-1.158	6		
15		Δ	Δ				Δ		-0.951		7	
16		x		x		x			-0.877	7		
17	o			o			o		-0.740			6
18				o			o	o	-0.565			7
19		x		x			x		-0.503	8	8	8
20		x		(x)				x	-0.400	9	9	9

Double and triple transition shifts occur as can be seen from table 1 between the following steps: 2–3, 4–5, 6–7, 7–8, 10–11, 11–12, 13–14, 15–16, 16–17. An analysis of table 1 shows that there are three possible single transition shift sequences in the resulting 20 speeds in which only one pair of shifting elements is switched on. The shifting elements for each of the three sequences are pointed with the corresponding symbol: “x” for the first sequence; “ Δ ” for the second; and “o” for the third. The numbers of speeds with single transition shifts for each sequence are shown in the three right columns of table 1. Each of the three sequences with single transition shifts has 9 speeds. The thirteenth speed in which the shifting elements are shown with the symbol “—”, is not used in any of the three sequences.

The IDDO structure has 4 degrees of freedom, i.e. 3 shifting elements should be switched on at each speed. While in one-stream modes (in which only GB1 or GB3 is enabled), it is sufficient to enable only two shifting elements. Therefore, a third shifting element can be additionally switched on in these speeds and will not affect to the speed ratio being realized, as in dual-clutch transmissions [5]. The shifting elements additionally switched on in speeds 8 and 20 are shown in table 1 by corresponding symbols in brackets.

Thus, the planetary-layshaft transmission with three power-flows presented in figure 2 implements three dependent single transition shifts sequences with 9 speeds in each. To find out the best sequence from three obtained, requires the development of a criterion and optimization this criterion.

4. Optimization criterion

Let some speed ratios sequence $a_k = a_1, a_2, \dots, a_k$ be preset, where a is the ratio i_{IO} of the corresponding speed; k is the speed number. The a_k sequence is ordered, the magnitudes of its members decrease in absolute value with increasing k , which is due to a decrease the speed ratios of the transmission when shifting from the lowest to the highest speed. It is unlikely that the members of the

a_k sequence will satisfy the speed ratios implemented by the transmission. Therefore, we will look for another sequence b_k , satisfying the speed ratio of the transmission closest to the given sequence a_k . The proximity of the sequences a_k and b_k will be evaluated by the criterion of the minimum of the function (1).

$$F = \sum_{k=1}^9 \left(\frac{a_k - b_k}{a_k} \right)^2 \quad (1)$$

The best sequence will be evaluated by the smallest value of the function F after its minimization for each of the three sequences with single transition shifts.

5. Calculation results

Let the next sequence a_k [6] be preset, see table 2.

Table 2. Preset speed ratio sequence a_k .

Speed	Ratio	Step
1	-4.713	
2	-2.842	1.658
3	-1.909	1.488
4	-1.382	1.382
5	-1.000	1.382
6	-0.808	1.237
7	-0.699	1.156
8	-0.580	1.205
9	-0.480	1.209

5.1. Sequence “x” optimization

As a result of the criterion (1) minimization for the sequence “x” the next sequence b_k was obtained, see table 3.

Table 3. Optimization results of the sequence “x”.

Speed	s7		s8		s9		c3	c6	i_{IO}	Step
	a	b	a	b	a	b				
1			x		x			x	-4.796	
2	x		x		x				-2.629	1.825
3	x			x	x				-2.090	1.258
4		x		x	x				-1.182	1.768
5		x	x		x				-1.108	1.067
6		x	x			x			-0.904	1.226
7		x		x		x			-0.706	1.282
8		x		x			x		-0.535	1.318
9		x		(x)				x	-0.421	1.271
R1					x		x		2.109	
R2						x	x		0.160	

R1 and R2 are the ratios for the reverse speeds. The resulting sequence is implemented with the following magnitudes of gear ratios: $i_1 = -1.342$, $i_2 = -0.421$, $i_3 = -2.034$, $i_4 = -0.654$, $i_5 = 2.109$, $i_6 = 0.16$, $i_{D1} = -1.5$, $i_{D2} = -1.5$. Here and below, the gear ratios of the differentials from the sun gear to the ring gear under the assumption that the carrier has stopped.

The value of the function (1) of the sequence “x” is $F = 0.083$.

5.2. Sequence “ Δ ” optimization

As a result of the criterion (1) minimization for the sequence “ Δ ” the next sequence b_k was obtained, see table 4.

Table 4. Optimization results of the sequence “ Δ ”.

Speed	s7		s8		s9		c3	c6	i_{IO}	Step
	a	b	a	b	a	b				
1			Δ		Δ			Δ	-4.415	
2			Δ			Δ		Δ	-2.646	1.668
3	Δ		Δ			Δ			-2.098	1.261
4	Δ		(Δ)					Δ	-1.326	1.582
5	Δ		Δ				Δ		-0.991	1.338
6			Δ				Δ	Δ	-0.848	1.168
7		Δ	Δ				Δ		-0.659	1.287
8		Δ		Δ			Δ		-0.574	1.148
9		Δ		(Δ)				Δ	-0.494	1.162
R1					Δ		Δ		4.501	
R2						Δ	Δ		1.848	

The resulting sequence is implemented with the following magnitudes of gear ratios: $i_1 = -1.326$, $i_2 = -0.494$, $i_3 = -0.848$, $i_4 = -0.644$, $i_5 = 4.501$, $i_6 = 1.848$, $i_{D1} = -1.5$, $i_{D2} = -1.5$.

The value of the function (1) of the sequence “ Δ ” is $F = 0.027$.

5.3. Sequence “o” optimization

As a result of the criterion (1) minimization for the sequence “o” the next sequence b_k was obtained, see table 5.

Table 5. Optimization results of the sequence “o”.

Speed	s7		s8		s9		c3	c6	i_{IO}	Step
	a	b	a	b	a	b				
1			o		o			o	-4.732	
2				o	o			o	-2.310	2.048
3	o			o	o				-2.164	1.068
4	o		(o)					o	-1.342	1.612
5	o			o		o			-1.074	1.249
6	o			o			o		-0.743	1.446
7				o			o	o	-0.669	1.112
8		o		o			o		-0.617	1.083
9		o		(o)				o	-0.473	1.306
R1					o		o		3.131	
R2						o	o		0.16	

The resulting sequence is implemented with the following magnitudes of gear ratios: $i_1 = -1.342$, $i_2 = -0.473$, $i_3 = -2.36$, $i_4 = -0.669$, $i_5 = 3.131$, $i_6 = 0.16$, $i_{D1} = -2.314$, $i_{D2} = -4.0$.

The value of the function (1) of the sequence “o” is $F = 0.027$.

Thus, the best sequence with single transition shifts for the transmission presented in figure 2 in accordance with the criterion optimized, is the sequence “ Δ ”.

6. Summary and results

One of the main problems of planetary-layshaft transmissions is the presence of double and triple transition shifts in which 2 or 3 pairs of shifting elements are involved. It leads to an increase in shifting time, reduce efficiency, worsen driving comfort and drivability. The problem of finding a sequence with single transition shifts and calculating the gear ratios that provide a range of speed ratios close to the given was solved using the example of a planetary-layshaft transmission consisting of two differentials and three sets of gears with fixed axles. The presented transmission has 5 shifting elements, provides 9 forward speeds with single transition shifts and 2 reverse speeds. The proposed planetary-layshaft transmission if compared with the 9-speed planetary transmission [6] has smaller number of gears and shifting elements and implements a close sequence of speed ratios.

Acknowledgments

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