Study on the Transmission Characteristics of the Multi-gear Multi-degree-of-freedom Hybrid Planetary Gear Automatic Transmission Based on the Line Method

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Abstract. Planetary gear transmission is widely used in vehicle automatic transmission because of its compact structure and good working conditions of gears and bearings. The transmission characteristic analysis of planetary gear automatic transmission has become one of the most important contents of automatic transmission design and maintenance. This paper takes the typical multi-gear multi-degree-of-freedom hybrid planetary gear automatic transmission Ford 10R80 as an example. Firstly, its structure is analysed, and its degree of freedom is calculated to be 5. According to this, further analysis is made on the shift logic of the sequential up and down gears, skipping-level shifting gears and the multi-level down gears such as from ⑩ to ⑦ or from ⑥ to ③. Then, the overall transmission nomogram is established based on the improved line method, and the power transmission characteristics of typical gears are analyzed. Finally, the transmission ratios of all gears are calculated according to the nomogram, and the calculated results are completely consistent with the data provided in the patent documents. The results show that the nomogram can not only clearly describe the speed of all the basic components in each gear but also calculate the transmission ratio of each gear quickly.

1. Introduction
Compared with the fixed shaft gear transmission, when the same power is transmitted, the meshing stress of the planetary gear is smaller, and the transmission shaft basically does not bear radial force, so the working conditions of the gear and the bearing are better. Planetary gear transmission has the advantages of low vibration, low noise and long service life, which is obvious in high-speed and high-power devices. Using the multi-plate clutches and brakes as the shift actuators to reduce the structure size can achieve a larger transmission ratio with a smaller size, so its structure is compact. Because of these advantages, planetary gear automatic transmission is widely used in automobiles [1]. With the requirements of energy saving, emission reduction and improvement of shift smoothness, multi-gear planetary automatic transmissions have been developed one after another successively, such as the 9-gear planetary automatic transmission 9HP48 launched by ZF Company [2], the 9-gear planetary automatic transmission launched by Benz [3], the 10-gear automatic transmission jointly developed by GM and Ford [4], the Direct Shift-10AT developed by Toyota [5], and the 10-gear automatic transmission PMX developed by Honda for the front-wheel drive and engine transverse type [6]. These multi-gear automatic transmissions are all composed of four single-row planetary gear rows which are connected in series and parallel by coaxial rotating elements. They are controlled by several shift actuators to achieve corresponding gears. Their connection and motion characteristics are complex. In order to better design and maintain the multi-gear transmission, it is necessary to analyze...
the power transmission characteristics of the planetary gear automatic transmission. The common analysis methods include the lever graphic method, the numerical analysis method, etc. However, these methods are not enough for the multi-degree-of-freedom hybrid planetary gear automatic transmission. Therefore, this paper takes the automatic transmission 10R80 jointly developed by GM and Ford as an example. On the basis of analyzing its structure and composition, this paper analyzes its shift logic and the power transmission characteristics of typical gears by using the improved line method, which provides an analysis method for the multi-gear multi-degree-of-freedom hybrid planetary gear automatic transmission.

2. Structure analysis of the automatic transmission

Figure 1 is the transmission structure diagram of the automatic transmission Ford 10R80, which is composed of four single-row single-stage planetary gear rows, numbered as the first planetary row, the second planetary row, the third planetary row and the fourth planetary row from left to right. The first planetary sun gear is connected with the second planetary sun gear, the first planetary carrier is connected with the fourth planetary gear ring, the second planetary gear ring is connected with the third planetary sun gear, and the third planetary gear ring is connected with the fourth planetary sun gear. The input shaft is connected with the second planetary carrier, and the output shaft is connected with the fourth planetary carrier. It includes 2 brakes namely B1, B2 and 4 clutches namely C1, C2, C3, C4. The value of Brake \( Z \) is 2, the value of power components (input component and output component) \( D \) is 2, the value of auxiliary component \( F \) is 5, and the number of basic components is \( J \).

\[
J = D + Z + F = 2 + 2 + 5 = 9
\]

Suppose that the number of degree of freedom of the transmission is \( n \), the number of planetary rows of the transmission is \( p = 4 \), and each planetary row has 2 degrees of freedom. Thus, it is necessary to eliminate \((2p-n)\) degrees of freedom, and retain \( J \) basic components through the rigid connection between different components [7].

\[
J = 3p - (2p - n) = p + n
\]

\[
n = J - p = 9 - 4 = 5
\]

3. Shift logic analysis of the automatic transmission

According to the analysis in the previous section, the transmission is a multi-planetary multi-gear hybrid planetary gear mechanism whose degree of freedom is 5. All forward and reverse gears need 4 of the 6 shift actuators to combine into a single degree of freedom mechanism to achieve the corresponding gear. Therefore, it can theoretically achieve up to \( C^4_5 = 15 \) gears: when B1 with B2 is involved in the work, the transmission ratios are the same under the three combination modes of C4 with C1, C4 with C2 and C4 with C3. Because B1 with B2 is braked, the 1st planetary row is fixed, then the 4th planetary gear ring is fixed, and C4 connects the power of input shaft to the 4th planetary sun gear. So the power of the 4th planetary single row is transmitted in these three modes. The combination of B1, C1, C3 and C4 or combination of B1, B2, C1 and C2 cannot transmit power, because C1 with C3 will connect the 2nd and the 3rd planetary rows as a whole, and C1 with C2 will connect the 1st and the 2nd planetary rows as a whole. At this time, the 2nd planetary row will appear as the power input but
be fixed by B1, resulting in the motion interference. Therefore, the actual effective combination is 11 (15 - 2 - 2 = 11).

In gear P/N, combined with the three shift actuators B1, B2 and C3, C4 is increased in the 1st gear and C2 is increased in the reverse gear, both of which can realize a fast start. Figure 2 is the forward gear shift logic diagram of the automatic transmission Ford 10R80, which can not only realize the conventional sequential up and down gears, but also realize the skipping-level shifting gears (such as from ① to ③ to ⑤). Especially for the multi-level downshifts in response to typical road conditions, the fast downshift for general urban roads is directly from ⑥ to ③, and the fast downshift for expressways is directly from ⑩ to ⑦, so as to ensure the speed reduction and driving safety of vehicles as soon as possible.

4. Power transmission analysis of the automatic transmission based on the line method

4.1 Overall nomogram of the automatic transmission Ford 10R80

The characteristic parameters (ratio of the gear ring to the sun gear number) of the transmission planetary row in reference [4] are $\alpha_1 = 2.20$, $\alpha_2 = 1.75$, $\alpha_3 = 1.60$, $\alpha_4 = 3.70$ respectively. The nomogram drawn according to the characteristic parameters of all planetary rows in the transmission and the relative position relationship of the basic components of the planetary row is shown in Figure 3, where S1, P1 and R1 respectively represent the sun gear, the planetary carrier and the gear ring of the 1st planetary row, and so on. The 9 vertical lines correspond to 9 basic components. In particular, the "auxiliary component" shown in the "dotted line" is the shared component among C1, C2 and C3. The input speed and the braking speed are represented by horizontal lines. The braking speed is 0 r/min, and the input speed is the input shaft speed of the transmission. The ratio of the distance from line S1 to P1 to the distance from P1 to R1 is 2.20, the ratio of the distance from line S2 to P2 to the distance from P2 to R2 is 1.75, the ratio of the distance from line S3 to P3 to the distance from P3 to R3 is 1.60, and the ratio of the distance from line S4 to P4 to the distance from P4 to R4 is 3.70. Take the 1st gear, the 5th gear and the reverse gear of the automatic transmission Ford 10R80 as an example to further analyze the power transmission process of typical gears with the line method.
4.2 Nomogram of Typical Gears

In the 1st forward gear, B1, B2, C3 and C4 work while C1 and C2 do not, as shown in Figure 4. For the 1st planetary row, B1 and B2 work, and the sun gear and the gear ring of the 1st planetary row are fixed, then the 1st planetary row is fixed as a whole, and the planetary carrier is fixed accordingly. For the 2nd planetary row, B1 fixes the sun gear, and the power of the input shaft is transmitted to the planetary carrier, then the speed of the gear ring as the output increases. And the speed vector ends of the sun gear, the planetary carrier and the gear ring are collinear. For the 3rd planetary row, C4 transfers the power of the input shaft to the gear ring, and the sun gear of the 3rd planetary row is rigidly connected with the gear ring of the 2nd planetary row (at the same speed). Connect the speed vector ends of the sun gear and the ring gear of the 3rd planetary row, and obtain the speed line of the planetary carrier. For the 4th planetary row, C4 transfers the power of the input shaft to the sun gear, and the gear ring of the 4th planetary row is rigidly connected with the planetary carrier which is fixed and whose speed is 0. Connect the speed vector ends of the sun gear and the ring gear of the 4th planetary row, and obtain the speed line of the planetary carrier of the 4th planetary row, which is the output speed in the 1st gear. Each speed line is indicated in order of ①, ②, ③ and ④ in the drawing, and the speeds of 9 basic components are also clearly drawn in the drawing. According to the proportion relationship between each line and the principle of triangle similarity, the transmission ratio in the 1st gear can be calculated as \( i_1 = 1 + \alpha_4 = 4.70 \).

It can be seen that the transmission ratio of the 1st gear is only related to the 4th planetary row. The two steps ② and ③ in Figure 4 can be omitted, and just the two steps ① and ④ can complete the power transmission analysis of the 1st gear. That is to say, for step ①, if B1 and B2 work, and the sun gear and the gear ring of the 1st planetary row are fixed, then the 1st planetary row is fixed as a whole; and if the 4th planetary gear ring is rigidly connected with the 1st planetary carrier, then the speed of the 4th planetary gear ring is 0. For step ④, the power of the input shaft is transmitted to the sun gear of the 4th planetary row through C4, and the speed vector ends of the sun gear and the gear ring are connected, then the speed line of the 4th planetary carrier is obtained, which is the output speed in the 1st gear. The shift actuator C3 does not play a substantial role in the power transmission of the 1st gear but is used in the 2nd, 3rd and 4th gears.

In the 5th forward gear, B2, C1, C2 and C4 work while B1 and C3 do not. The nomogram of the 5th gear is shown in Figure 5. For step ①, C1 and C2 work, and the four components of P1, R2, S3 and R4 are rigidly connected. For step ②, the sun gear of the 1st planetary rigidly connects with that of the
2nd planetary row, and the planetary carrier of the 1st planetary row connects with the gear ring of the 2nd planetary row. The 1st and the 2nd planetary row become typical Simpson type double planetary row structure, the 2nd planetary carrier is used as the power input component, and the 1st planetary gear ring is fixed by B2. Connect the speed vector ends of the corresponding components, then the speed line of the 2nd planetary gear ring is obtained. For step ③, connect the speed vector ends of the 4th planetary sun gear (linked with the input shaft by C4) and the 4th planetary gear ring (rigidly linked with the 2nd planetary gear ring at the same speed), then the speed of the 4th planetary carrier is gotten, which is the output speed of the 5th forward gear. In the light of the proportion relationship between each line, the transmission ratio of the 5th gear can be calculated by using the triangle similarity principle, that is

$$i_5 = \frac{(1 + \alpha_4)(1 + \alpha_1 + \alpha_2)}{(1 + \alpha_2)(1 + \alpha_4) + \alpha_1} = 1.54.$$  

Thus, the 1st, 2nd and 4th planetary rows participate in the power transmission process of the 5th gear while the 3rd planetary row does not.

![Figure 5. Power transmission nomogram of the 5th gear.](image)

![Figure 6. Power transmission nomogram of the reverse gear.](image)

In the reverse gear, B1, B2, C2 and C3 work while C1 and C4 do not. The nomogram of the reverse gear is shown in Figure 6. For step ①, C2 and C3 work, the planetary carrier of the 3rd planetary row rigidly connects with the gear ring of the 4th planetary row, and 3rd and the 4th planetary row form a double planetary four-component structure. For step ②, the sun gear in the 2nd planetary row is braked by B1, the planetary carrier is linked with the input shaft as the input component, and the gear ring is the output component. The speed vector ends of the three components are shown in Figure 6. For step ③, when B1 and B2 work, the sun gear and the gear ring of the 1st planetary row are fixed. Afterwards, the 1st planetary row is fixed as a whole, so does the planetary carrier. For step ④, the 3rd planetary sun gear (connected with the 2nd planetary gear ring) is used as the input component, and the 3rd planetary carrier and the 4th planetary gear ring are fixed (linked with the 1st planetary carrier). The corresponding speed vector line is shown as ④, and the speed of the 4th planetary carrier is obtained, which is the output speed of the reverse gear. According to the proportion relationship between each line, the transmission ratio of the reverse gear can be calculated by using the triangle similarity principle, that is

$$i_r = -\frac{\alpha_2\alpha_1(1 + \alpha_4)}{1 + \alpha_2} = -4.79.$$  

Therefrom, the 2nd, 3rd and 4th planetary rows participate in the power transmission process of the reverse gear while the 1st planetary row does not.

The dynamic analysis, nomogram drawing and transmission ratio calculation of other gears are similar to those of the above gears and will not be described in detail.
4.3 Transmission ratio calculation
According to the nomogram of all gears, the transmission ratio of each gear is calculated by using the triangle similar principle and the proportion relationship between the lines. The result in Table 1 is completely consistent with the patent document [4] submitted by Ford to the U.S. patent office.

| Gears | Transmission ratio | Gears | Transmission ratio |
|-------|--------------------|-------|--------------------|
| 1     | \(1 + \alpha_4 = 4.70\) | 6     | \(\frac{(1 + \alpha_4)(1 + \alpha_1 + \alpha_2 + \alpha_3 \alpha_4)}{(1 + \alpha_4)(1 + \alpha_2 + \alpha_3 \alpha_4) + \alpha_1} = 1.29\) |
| 2     | \(\frac{\alpha_5(1 + \alpha_4)}{1 + \alpha_2} = 2.99\) | 7     | 1 |
| 3     | \(\frac{(1 + \alpha_4)(1 + \alpha_5)}{1 + \alpha_1 + \alpha_4} = 2.18\) | 8     | \(\frac{\alpha_5(1 + \alpha_4)(1 + \alpha_4)}{\alpha_5(1 + \alpha_4) + \alpha_4(1 + \alpha_2 + \alpha_3 \alpha_4)} = 0.85\) |
| 4     | \(\frac{1 + \alpha_4 + \alpha_5}{1 + \alpha_2} = 1.80\) | 9     | \(\frac{\alpha_5(1 + \alpha_4)}{\alpha_2 + \alpha_4 + \alpha_3 \alpha_4} = 0.69\) |
| 5     | \(\frac{(1 + \alpha_4)(1 + \alpha_4 + \alpha_5)}{(1 + \alpha_2)(1 + \alpha_4) + \alpha_4} = 1.54\) | 10    | \(\frac{\alpha_2}{1 + \alpha_2} = 0.64\) |
| R     | \(-\frac{\alpha_2 \alpha_3 (1 + \alpha_4)}{1 + \alpha_2} = -4.79\) |       |       |

5. Conclusion
Aiming at the transmission characteristic analysis of the multi-gear multi-degree-of-freedom hybrid planetary gear automatic transmission, the paper takes the typical planetary gear automatic transmission Ford 10R80 as an example. The structural connection relationship of the transmission is analyzed firstly, which is a 4-row and 5-degree-of-freedom hybrid structure composed of 2 brakes, 2 power components (1 input component and 1 output component), 5 auxiliary components and 9 basic components.

Secondly, the shift logic diagram is used to analyze that it can realize the conventional sequential up and down gears, as well as the skipping-level shifting gears (such as from ① to ③ to ⑤). In particular, for the multi-level down gears in response to typical road conditions, the fast downshift for general urban roads is directly from ⑥ to ③, and the fast downshift for expressways is directly from ⑩ to ⑦.

Thirdly, the overall structure and the nomograms of typical gears are analyzed with the line method. Furthermore, the transmission ratio of all gears is calculated based on the nomogram, whose result is completely consistent with the patent document submitted by Ford to the U.S. patent office.

By applying the line method, not only the power transmission characteristics of the multi-gear multi-degree-of-freedom hybrid planetary gear automatic transmission can be analyzed accurately and intuitively, but also the transmission ratio can be calculated quickly, which provides a good way for the design and maintenance of other similar automatic transmissions.

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