Selection and Analysis of Dual-motor Coupling Device Based on Efficiency Power

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Abstract. In response to the needs of electric tractors under multiple operating conditions, a dual-motor power coupling drive system is matched. According to its power transmission requirements, a dual planetary gear train dynamic coupling structure scheme and a multi-planetary gear train dynamic coupling structure scheme are proposed. Through example design, the specific parameters of the two schemes are obtained. The transmission efficiency formulas of the two schemes in multiple modes are deduced, the transmission efficiency in each mode is calculated, and the final transmission scheme of the dual-motor power coupling device is optimized based on the transmission efficiency. It provides a theoretical basis and method for the design and analysis of the tractor power coupling device.

1. Introduction

In recent years, China has increased its support for agricultural machinery, and "Made in China 2025" lists agricultural machinery and equipment as one of the top ten industries for development[1], and the national agricultural mechanization development "13th Five-Year Plan" points out that the increase in vigorously promote environmentally friendly and energy-saving agricultural power Equipment, accelerate the elimination of high energy consumption, heavy pollution, low performance of old machinery[2]. Tractors, as one of the most widely used agricultural equipment in the agricultural production process, have gradually become a hot spot in recent years for research related to their electric drive systems[3-5].

At present, the research on electric tractors in China is mainly focused on university teams. Chen et al[6] carried out parameter matching and optimized design of a dual-motor electric tractor power-coupled drive system. Li et al[7] proposed a tractor dual-motor coupled drive system and established an electric tractor model for simulation experiments. Xu et al[8] designed an electric tractor dual-motor independent drive system, proposed some important components of the electric drive system parameters, and conducted dynamic performance analysis of the electric tractor. Liu et al[9] designed a crawler type power transmission system scheme for an electric tractor, proposed a method to match the parameters of the core components of the power transmission system, and conducted a field rotary tillage test, a field transportation test and a tillage energy consumption test. Baek et al[10] designed a 120kw class electric tractor four-wheel drive system, established a simulation model, and conducted simulation and experimental verification.

However, the existing research results are mostly the one-way coupling of dual power to the central transmission, which cannot realize the switching of the power distribution between the driving wheel and the PTO, and cannot guarantee the maximum power output of ploughing and rotary tillage
operations, so development Power coupling device capable of realizing bidirectional power flow. This paper takes 132kW tractor as the research object, matches the dual motor drive system, and proposes two basic schemes of power coupling device that can realize multi-mode according to its power transmission demand, and carries out power matching and parameter calculation, and takes transmission efficiency as the main evaluation index to prefer the final transmission scheme.

2. Dual motor electric tractor power coupling device scheme design

According to the above power transfer requirements, two basic schemes of dual-motor power coupling device structure that can realize the coupling and decoupling of dual motors at the same time are proposed based on the creative design method, which are the dual-planetary wheel system power coupling structure scheme and the multi-planetary wheel system power coupling structure scheme, and named as Scheme A and Scheme B, as shown in Figure 1.

As shown in Figure 1(a), the dual planetary wheel system power coupling structure scheme is a combination of multiple pairs of drive gear pairs, two synchronizer shift mechanisms, two planetary gear mechanisms and two brakes. The two inputs are connected to the drive motor and the PTO motor, and the corresponding two outputs are connected to the gearbox and the rotating mechanism. When synchronizer 1 and synchronizer 2 are not engaged in independent drive mode, the two motors are decoupled and the power provided by the two motors is output through the corresponding planetary gear mechanism 4 and planetary gear mechanism 5 respectively; when synchronizer 1 and synchronizer 2 are engaged to the right at the same time in coupled drive mode, the power provided by the PTO motor is coupled with the drive motor through the transmission gears $z_5z_6$ and $z_7z_8$. When synchronizer 1 and synchronizer 2 are engaged to the left at the same time, it is coupled to the rotary mode, and the power provided by the drive motor is coupled to the power provided by the PTO motor through the drive gears $z_1z_2$ and $z_3z_4$ in the planetary gear mechanism 5 and output.

As shown in Figure 1(b), unlike the double planetary wheel system power coupling device, the multi-planetary wheel system power coupling device is composed of multiple pairs of drive gear pairs, a synchronizer shift mechanism, three planetary gear mechanisms and two brakes. The two inputs are connected to the drive motor and the PTO motor, and the corresponding two outputs are connected to the gearbox and the rotating mechanism. When synchronizer 1 and 2 are not engaged in independent drive mode, the two motors are decoupled at this time, and the power provided by the two motors is output through the corresponding planetary gear mechanism 4 and planetary gear mechanism 5 respectively; when synchronizer 1 and 2 are engaged to the right in coupled drive mode, the power provided by the PTO motor is coupled to the drive motor through the drive gear vice $z_5z_6$, planetary gear mechanism 7 and $z_7z_8$. When the synchronizer 1 and 2 are engaged to the left, it is coupled rotary mode, at which time the power provided by the drive motor is coupled to the power provided by the
PTO motor through the drive gear pair $z_1z_2$, planetary gear mechanism 7 and $z_3z_4$ and the power provided by the PTO motor in the planetary gear mechanism 5 and output.

3. Example design

According to the power transmission route of the electric tractor power coupling device and the planetary gear mechanism transmission theory, the speed-torque relationship between the two schemes in the coupled drive mode is analysed.

The rotational speed torque relational expression of A scheme is as follows.

$$
\begin{align*}
    n_b &= \frac{1}{1+k} n_q + \frac{k}{(1+k) i_{56}} n_p \\
    T_b &= (1+k) T_q = \frac{(1+k) i_{56} i_{78}}{k} T_p
\end{align*}
$$

Similarly, the speed-torque relational expression of the B scheme is as follows.

$$
\begin{align*}
    n_b &= \frac{1}{1+k_i} n_q + \frac{k_i (1+k_i)}{(1+k_i) i_{56}} n_p \\
    T_b &= (1+k_i) T_q = \frac{(1+k_i) i_{56} i_{78}}{k_i (1+k_i)} T_p
\end{align*}
$$

Where, $n_q$ is the output speed of the drive motor, r/min; $n_b$ is the output speed of the PTO motor, r/min; $n_b$ is the speed output to the transmission, r/min; $T_q$ is the output torque of the drive motor, Nm; $T_p$ is PTO Motor output torque, Nm; $T_b$ is the torque output to the transmission; $k$ is the characteristic parameter of the planetary gear train; $i_{56}$ is the gear ratio of 5/6, $i_{56} = z_6 / z_5$; $i_{78}$ is the gear ratio of 7/8, $i_{78} = z_8 / z_7$.

According to the power demand of the 132kW tractor, the two schemes were power matched to obtain the main parameters of the drive motor and PTO motor, as shown in Table 1.

| Parameter items | Value |
|-----------------|-------|
| Drive motor rated power/peak power (kW) | 90/150 |
| Drive motor rated torque/peak torque (Nm) | 1140/2100 |
| Drive motor rated speed/maximum speed (rpm) | 760/3000 |
| PTO motor rated power/peak power (kW) | 48/130 |
| PTO motor rated torque/peak torque (Nm) | 230/450 |
| PTO motor rated speed/maximum speed (rpm) | 2000/6000 |

Under the condition of ensuring the same output speed and torque at the end of the power coupling device, the specific parameters of Scheme A can be obtained through parameter calculation, as shown in Table 2.

| Parameter items | Number of teeth $z_1$ | Modulus $m$ | Pressure angle $\alpha$ | Helix angle $\beta$ | Tooth width $b$/mm |
|-----------------|----------------------|-------------|------------------------|-------------------|-------------------|
| Gear vice 1/2   | 31/99                | 3           | 20°                    | 14.12°            | 96/92             |
| Gear vice 3/4   | 93/29                | 3           | 20°                    | 13.79°            | 86/90             |
| Gear vice 5/6   | 29/93                | 3           | 20°                    | 13.79°            | 90/86             |
| Gear vice 7/8   | 31/99                | 3           | 20°                    | 14.12°            | 96/92             |
| Sun Wheel 4     | 25                   | 4           | 20°                    | 14°               | 76                |
The specific parameters of Scheme B can be obtained through parameter calculation, as shown in Table 3.

| Parameter items       | Number of teeth z | Modulus m | Pressure angle α | Helix angle β | Tooth width b/mm |
|-----------------------|-------------------|-----------|------------------|---------------|------------------|
| Gear vice 1/2         | 27/145            | 3         | 20°              | 14°           | 84/80            |
| Gear vice 3/4         | 123/23            | 3         | 20°              | 13.99°        | 67/71            |
| Gear vice 5/6         | 23/123            | 3         | 20°              | 13.99°        | 71/67            |
| Gear vice 7/8         | 27/145            | 3         | 20°              | 14°           | 84/80            |
| Sun Wheel 4           | 25                | 4         | 20°              | 14°           | 76               |
| Planetary Wheel 4     | 19                | 4         | 20°              | 14°           | 76               |
| Gear ring 4           | 62                | 4         | 20°              | 14°           | 76               |
| Sun Wheel 5           | 25                | 4         | 20°              | 14°           | 76               |
| Planetary Wheel 5     | 19                | 4         | 20°              | 14°           | 76               |
| Gear ring 5           | 62                | 4         | 20°              | 14°           | 76               |
| Sun Wheel 7           | 43                | 4         | 20°              | 14°           | 68               |
| Planetary Wheel 7     | 17                | 4         | 20°              | 14°           | 68               |
| Gear ring 7           | 77                | 4         | 20°              | 14°           | 68               |

4. Transmission efficiency analysis and comparison

Mechanical transmission system efficiency $\eta$ is one of the important indicators to evaluate the transmission characteristics of the designed mechanism, for different transmission structure types, its efficiency $\eta$ is different; for the same transmission type, the specific number of teeth and other parameters are different, its efficiency $\eta$ is also different, the efficiency of the transmission system $\eta$ even with the change of the transmission speed and change. The efficiency loss of mechanical transmission system is mainly: gear pair friction loss, bearing friction loss, hydraulic loss. This article needs to consider the transmission efficiency, mainly considering the friction loss of the gear pair, and select the final transmission scheme through comparative analysis.

Firstly, analyse the transmission efficiency of scheme A. When it is in independent drive mode, the calculation formula of transmission efficiency is:
\[
\eta_{eq} = \eta_e \\
\eta_e = 1 - \frac{n_r - n_c}{(1 + k_i) n_e - k_i n_r \phi'_s} \\
n_r = \frac{1}{1 + k_i} n_s + \frac{k_i}{1 + k_i} n_e \\
\phi'_s = \phi'_s + \phi'_c \\
\phi'_c = 2.3 f \left( \frac{1}{z_s} + \frac{1}{z_r} \right) \\
\phi'_m = 2.3 f \left( \frac{1}{z_c} - \frac{1}{z_s} \right)
\]

Where, \(\eta_{eq}\) is always the total efficiency of the transmission system; \(\eta_e\) is the transmission efficiency of planetary gear mechanism 4; \(n_s\) is the input speed of sun wheel 4, rpm; \(n_e\) is the output speed of planetary frame 4, rpm; \(n_r\) is the input speed of tooth ring 4, rpm; \(k_i\) is the characteristic parameter of planetary gear mechanism 4; \(\phi'_s\) is the coefficient of meshing loss of planetary gear mechanism 4; \(\phi'_c\) is the coefficient of meshing loss between sun wheel 4 and planetary wheel 4; \(\phi'_m\) is the coefficient of meshing loss between planet wheel 4 and gear ring 4; \(z_s\) is the number of sun wheel 4 teeth; \(z_c\) is the number of planet wheel 4 teeth; \(z_r\) is the number of gear ring 4 teeth.

When the A scheme is in the coupled drive mode and the PTO rotates synchronously, the transmission efficiency calculation formula is as follows, where \(\phi'_s\), \(\phi'_c\) and \(\phi'_m\) are calculated as shown in formula (3), the same below.

\[
\eta_{eq} = \left( \frac{P_e + P_p \eta_p}{P_p} \right) \eta_e \\
\eta_e = 1 - \frac{n_r - n_c}{(1 + k_i) n_e - k_i n_r \phi'_s} \\
n_r = \frac{1}{1 + k_i} n_s + \frac{k_i}{1 + k_i} n_e \\
n_e = \frac{P_p \eta_p}{k_i P_e} \cdot n_s \\
\eta_p = \eta_{p5/6} \eta_{p7/8} \\
\eta_{p5/6} = \frac{1 - \pi f (\frac{1}{z_s} + \frac{1}{z_c})}{2}
\]

Where, \(P_e\) is the drive motor input power; kW; \(P_p\) is the PTO motor input power; \(\eta_p\) is the transmission efficiency between PTO motor to planetary gear mechanism 4; \(\eta_{p5/6}\) is the transmission efficiency between transmission gear 5/6; \(\eta_{p7/8}\) is the transmission efficiency between transmission gear 7/8; \(\eta_{pab}\) is the transmission efficiency of transmission gear vice; \(\varepsilon\) is the coefficient of overlap; \(f\) is the friction coefficient; \(z_s\) is the number of teeth of the master wheel; \(z_c\) is the number of teeth of the slave wheel.

When the A scheme is in the coupled drive mode and the PTO is at the standard speed, the PTO motor is set to work at the rated power, and the transmission efficiency calculation formula is:
Where, $P_{pe}$ is the rated power of PTO motor, kW; $P_{qe}$ is the rated power of the drive motor, kW; $k_5$ is the characteristic parameter of planetary gear mechanism 5.

Secondly, the transmission efficiency of the B scheme is analysed. When it is in the independent driving mode, its transmission structure and parameters are the same as the dual planetary gear train power coupling structure scheme, so the transmission efficiency calculation formula is also the same.

When the B scheme is in the coupled drive mode and the PTO rotates synchronously, the formula for calculating the transmission efficiency is:

$$
\begin{align*}
\eta_{tot} &= \left( \frac{P_p + P_{pe} \eta_s}{P_p + P_{qe}} \right) \eta_s \\
\eta_s &= 1 - \left[ \frac{n_i - n_e}{(1 + k_e) n_i - k_i n_e \phi_{s'}} \right] \phi_{s'}^e \\
n_i &= \frac{1}{1 + k_e} n_i + \frac{k_e}{1 + k_e} n_e \\
n_e &= \frac{P_{pe} \eta_s}{k_p P_q} \cdot n_i \\
\eta_p &= \eta_{tot} \eta_{p/\eta} \\
\eta_{p/\eta} &= 1 - \left[ \frac{n_i - n_e}{(1 + k_e) n_i - k_i n_e \phi_{s'}} \right] \phi_{s'}^e \\
\eta_{sub} &= 1 - \frac{\pi}{2} e f \left( \frac{1}{z_e} + \frac{1}{z_0} \right)
\end{align*}
$$

Where, $\eta_7$ is the planetary gear mechanism 7 transmission efficiency; $n_i'$ is the sun wheel 7 output speed, rpm; $n_e'$ is the planetary frame 7 input speed, rpm; $\phi_{s'}^e$ is the planetary gear mechanism 7 meshing loss coefficient.

When the B scheme is in the coupled drive mode and the PTO is at the standard speed, the transmission efficiency formula is:
Through the above efficiency formula, and combined with the specific parameters of the transmission system in Tables 1, 2 and 3, the calculated transmission efficiencies of Scheme A and Scheme B under single-motor independent drive, dual-motor coupled drive with proportional power (PTO follower) and dual-motor coupled drive with fixed speed of PTO motor (PTO fixed speed) modes, respectively, are shown in Figure 2.

\[
\eta_{eq} = \frac{P_x + P_{pt} \eta_p}{P_x} \\
\eta_s = 1 - \frac{n_r - n_p}{(1 + k_s)n_r - k_p \phi_p} \\
\eta_c = \frac{1}{1 + k_s} \frac{n_r}{n_p} + \frac{k_s}{1 + k_s} \\
\eta_0 = \frac{540(1 + k_s)}{i_{ax} i_{rs}} \\
\eta_p = \eta_{pt} \eta_{ps} \eta_t \\
\eta_c = 1 - \frac{n_r - n_p}{(1 + k_s)n_r - k_p \phi_p} \\
\eta_{ps} = 1 - \frac{\pi}{2} f \left( \frac{1}{z_a} + \frac{1}{z_c} \right) \\
\eta_{pt} = 1 - \frac{\pi}{\phi_p} f \left( \frac{1}{z_a} + \frac{1}{z_c} \right) \\
\phi_p = \phi_r \phi_{cr} \phi_{cr}
\]

(7)

Analysis in Figure 2 shows that the mechanical system drive efficiency is high and all of them remain around 98%. Among them, scheme A has a constant drive system efficiency of 97.8834% in independent drive mode, and its drive system efficiency is 98.0257% in coupled drive (PTO follower) mode, and in coupled drive (PTO fixed speed) mode, the drive efficiency changes with the change of drive motor speed because the independent variable (drive motor speed) in the formula fails to cancel, and its efficiency changes from 96.8006% to 98.3892%, and then decreases to 98.1829%; Option B has a constant drive train efficiency of 97.8834% in the independent drive mode, 97.9780% in the coupled drive (PTO follower) mode, and increases from 96.6130% to 98.2576%, and then decreases to 98.1829%.

The comparison analysis shows that the transmission efficiency of both solutions is the same when they are in the independent drive mode. In the coupled drive (PTO follower) mode, the transmission efficiency of solution A is higher than that of solution B. In the coupled drive (PTO fixed speed) mode, the transmission efficiency of solution A is also higher than that of solution B. In the coupled drive (PTO fixed speed) mode, the transmission efficiency of solution A is also higher than that of solution B. From the perspective of manufacturing cost, solution B has one more planetary gear mechanism...
than solution A, so the manufacturing cost of solution B is higher. Therefore, from the consideration of both transmission efficiency and manufacturing cost, the A scheme, i.e., the dual planetary wheel system power coupling structure scheme, is finally selected as the dual motor coupling device structure scheme.

5. Conclusion

For electric tractor traction, plough and rotary tillage and other multi-station operation needs, matching the dual-motor electric tractor drive system, according to its power transmission needs, two basic schemes of power coupling device structure that can realize multi-mode are proposed, and the speed-torque relationship is analysed, and the speed-torque relationship equations of the two coupling schemes are obtained.

In this paper, the two proposed schemes are designed by examples, and the specific parameters of the two schemes are obtained. The efficiency equations of the two schemes under various operating modes are derived, and the transmission efficiency range of the dual-planetary wheel system power coupling scheme is from 97.8834% to 98.3892%, which is higher than that of the multi-planetary wheel system power coupling scheme, so the dual-planetary wheel system power coupling scheme is preferred as the final transmission scheme.

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