Design and Analysis of a novel Mechanic-Electronic-Hydraulic Powertrain System for Agriculture Tractors

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This work was supported by the National Natural Science Foundation of China (Grant No. 51805222) and the Open Fund of State Key Laboratory of Automotive Simulation and Control (Grant No. 20201201).

ABSTRACT This paper introduces a mechanic-electronic-hydraulic powertrain system (MEH-PS) composed of an electro-mechanical hybrid system and hydro-mechanical composite transmission according to current mainstream drive and transmission technologies. Firstly, the structural design concept of the system is introduced, and the power and transmission components are selected according to the actual working requirements of the tractor. Then, the principles of various drive modes and transmission modes of the powertrain system are explained, and the speed regulation characteristic curves of the hydro-mechanical transmission (HMT) are given; the speed characteristics, torque characteristics, power split characteristics, and efficiency characteristics of the powertrain system are analyzed. Finally, a tractor simulation and test model was developed to verify its performance under certain operating conditions. Simulation results show that: the tractor acceleration performance is improved, the speed range is wider, the power components and hydraulic components can also meet the requirements, the HMT in a wide range of speed to maintain the average efficiency of above 86%. The bench test results show that: the step-less speed regulation characteristics and efficiency characteristics of the powertrain system are basically consistent with the simulation results.

INDEX TERMS Agriculture tractors, Performance study, Hybrid system, Hydro-mechanical transmission (HMT), Powertrain design.

I. INTRODUCTION

There are two important research trends for vehicle powertrain nowadays, one is to improve the performance of the transmission to meet the driving needs for vehicles based on traditional power sources, such as the continuously variable transmission (CVT) which has been widely used in recent years to make the engine have the optimal power output under different working conditions[1, 2]. Another trend is to study the coordinated work of power sources, such as overcoming the shortcomings of single power source operation by a dual power source consisting of an engine and an electric motor[3], with motor assistance to fill the load not covered by the engine, while improving the economic performance of the system[4].

Tractors work in a variety of operations, When performing farm operations, they need to provide greater traction[5]; when performing road transportation, they have high requirements of tractor speed[6]. To meet these requirements, tractors equipped with stepped transmissions provide speed ratios for different speeds and drive force needs by adding extra gears, but this also makes the structure and operation of the transmission complex[7, 8]. HMT has the CVT feature which is suitable for agricultural and construction machinery[9, 10]. Due to its large speed ratio range, compared with traditional tractors, the number of gears is significantly reduced, the space structure is optimized, and the weight of the whole machine is reduced[11]. Furthermore, the driving comfort of the tractor is improved as a result of fewer gears and avoidance of frequent gear shifts[12].

The hydro-mechanical transmission developed by Fendt splits power through planetary gear, eliminating the necessity of complex multi-gear transmission and allowing the tractor to be continuously shifted over a wide speed range by simply controlling the swing angle of the pump and motor (0–45°)[13], but the dual-variable hydraulic
components are technically challenging and expensive to promote[14]. The 4-planetary gears hydro-mechanical transmission developed by Steyr can output variable speed at constant engine speed and achieve step-less speed regulation of the forward and reverse 0-50 km/h speed range without power interruption, but the multi-planetary gears structure convergence and shift control strategy are complex [15]. Liu proposed a multi-speed HMT with a two-stage input coupling layout, in which the maximum speed of the vehicle can be significantly increased by changing the two-phase (positive and negative) power split of the hydrostatic circuit, but the device suffers from low efficiency in multiple speed ranges[16]. Wang designed a single planetary HMT, which has a simplified structure compared to a multi-planetary transmission but has a limited speed range to meet the increasing speed requirements of tractors[17].

Toyota Hybrid System (THS) achieves a multi-coupling output of the engine and motor power through planetary gears and fixed shaft gears, enabling the vehicle to meet both comfort and power requirements in different operating modes, and the multi-coupling approach gives more degrees of freedom to the powertrain, more flexible torque and speed adjustment, and a significant vehicle performance improvement[18]. Joshi investigated the planetary gear and shaft fixed-gear mechanism to develop a hybrid powertrain with torque coupling and speed coupling. This powertrain can decouple the engine speed and torque from the vehicle drive wheels without the need for a complex transmission, and the engine can operate within its efficient speed and torque range to obtain greater drive torque and drive speed with the assistance of an electric motor, while the fuel consumption of the vehicle is lower[19, 20]. Francesco studied the application of an electro-mechanical hybrid system in tractors, which can lower the engine specifications with the assistance of electric motors while satisfying the whole tractor operation[21]. Moreda's research points out that a hybrid power system allows more freedom of torque and speed control of the tractor while reducing noise[22].

There is no research on combining electromechanical hybrids with HMT in the field of tractors or other vehicles till now. The application of this technology in tractors seems to be promising given the application of road vehicles, as the dual power source has additional degrees of freedom, which can simplify the structure of transmission to some extent while lowering the requirements for transmission component specifications. Combining the design ideas of the above scheme and proposes a MEH-PS for tractors. This paper firstly analyzes the design concept of the powertrain, then analyzes its speed characteristics, torque characteristics, power split, and efficiency characteristics, finally, verify the theoretical analysis results through simulation and test bench.

II. DESIGN CONCEPT

A. DESIGN CONCEPT OF THE SCHEME

The design concept of the MEH-PS proposed in this paper is shown in Fig 1, the structure has two power sources: internal combustion engine (ICE) and motor-generator (MG), both of which can output power individually or simultaneously.

When the tractor is operating in the unloaded transit condition, the required power is low, the pure electric drive mode can meet the power requirements. When the tractor is operating in plowing condition, the pure engine drive mode is the main drive mode. When the required torque is high, such as the power take-off (PTO) output power or acceleration process can be driven through the torque coupling (TC) mode. When the required speed is high, through the K1 (1st planetary gear set) to achieve speed coupling (SC) mode. The motor can regulate the torque and speed of the drivetrain independently of the engine through TC mode and SC mode.

The hydraulic system input power is split through the G1 (fixed shaft gear set) and passes through G2 to merge with the power from the mechanical path in the K2. Considering that the hybrid system provides more degree of freedom, the structure uses only a single planetary gear set for power merging. The power can be outputted through hydrostatic transmission mode, hydro-mechanical transmission mode, or mechanical transmission mode as needed, and finally adjusted and outputted to the drive axle through the sub-transmission to drive the tractor forward or backward.

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![MEH-PS structure principle](image)

B. SCHEME REALIZATION

A MEH-PS with independent intellectual property rights for tractors was designed based on the above design concepts, and the details of the powertrain are shown in Fig 2.
FIGURE 2. Configuration Scheme. 1. ICE 2. Input shaft 3. Power split mechanism 4. Hydraulic system 5. PTO 6. TC mechanism 7. MG 8. SC mechanism 9. Power merging mechanism 10. Sub-transmission 11. Output shaft. C is the clutch, B is the brake, e is the displacement ratio, $i_1$, $i_2$ is the gear ratio.

C. MAIN COMPONENTS MATCHING

To facilitate the following analysis of the performance of the MEH-PS, the relevant parameters of the power source and transmission components need to be determined.

The longitudinal dynamics of tractors can be expressed by the following equation:

$$F_t = F_i + F_r + F_a + F_j + F_T$$  \hspace{1cm} (1)

Where, $F_i$ is the wheel driving force, $F_r$ is the rolling resistance, $F_a$ is the air resistance, $F_j$ is the acceleration resistance, and $F_T$ is the hook tension.

Engine selection: the power required for tractor plowing operation is the highest under various operating conditions, and the hook tension during the plowing operation can be expressed by the following formula\[23\]:

$$T = Zx Bx Hx Kx$$  \hspace{1cm} (2)

Where $Z_x$ is the number of the ploughshare, $B_x$ is the monomer ploughshare, $H_x$ is the ploughing depth, and $K_x$ is soil specific resistance. The parameters of the tractor-matched 1LF-550 reversible plow are as follows: $Z_x=5$, $B_x=0.45m$, $H_x=0.25m$, $K_x=7N/cm^2$; considering that the tractor needs a power reservation, according to the calculation: $F_{T_{max}}=1.2F_t=47.25kN$.

Tractor plowing operation speed generally is 5~10km/h, this paper selected for 7km/h, the traction efficiency of wheeled tractor generally is $\eta_t=0.7$. According to the above conditions, the engine rated power is calculated as $P_e = F_{T_{max}}v/3.6\eta_t = 131.25kW$, after comparison, the WP6.180E40 diesel engine was selected to meet the requirements.

$$\text{DOH} = \frac{P_m}{P_e + P_m}$$  \hspace{1cm} (3)

According to the data related to the tractors in operation, when the tractor is in transit or sowing and other low load operations, the required power of the whole tractor can be up to 50%\[25\], given the overload capacity of the motor $\beta=P_{m_{max}}/P_m \geq 2$, the motor power is accounted for 25% of the whole tractor, it is calculated that the rated power of the motor is $P_m = 45kW$. Considering the size of the motor and the need to expand the speed range of the tractor, the hybrid system tends to choose a high-speed motor, and the TZ205XS85K01 electric motor is selected after comparison. The specific parameters of power sources are shown in Tab 1.

TABLE.1 Power source parameters

| Model       | Engine | Rated power | Rated speed | Maximum torque |
|-------------|--------|-------------|-------------|----------------|
| WP6.180E40  | Rated   | 132kW       | 2200rpm     | 750N·m         |

| Model       | Motor  | Rated (Maximum) power | Maximum torque | Maximum speed |
|-------------|--------|-----------------------|----------------|--------------|
| TZ205XS85K01| Rated   | 45(95)kW              | 250 N·m        | 11000 rpm    |

The selection of hydraulic components should be able to match the selected power source. The engine is the main power source of the tractor, therefore the parameters of the engine are mainly considered for matching.\[26\]. The pump is connected to the engine and the relevant parameters of the hydraulic components should satisfy the following equation.

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TABLE.2 Parameters of hydraulic components

| Model      | Rated displacement (cm³/r) | Rated speed (r/min) | Maximum pressure difference (bar) | Torque (N·m/bar) | Flow rate (L/min) |
|------------|----------------------------|---------------------|-----------------------------------|------------------|------------------|
| 055-VDP    | 55                         | 3900                | 480                               | 0.88             | 215              |
| 055-FDM    | 55                         | 3900                | 480                               | 0.88             | 215              |

\[ P_{p_{\text{max}}} = \frac{Q_{p_{\text{max}}} \Delta P_{\text{max}}}{600 \eta_p} \geq P_{e_{\text{max}}} \] (4)

Where \( P_{p_{\text{max}}} \) is the maximum pump power [kW]; \( Q_{p_{\text{max}}} \) is the maximum pump flow rate [L/min]; \( \Delta P_{\text{max}} \) is the maximum pump oil pressure [bar]; if the pump is in the case of maximum flow, the pump efficiency \( \eta_p = 0.95 \), according to the parameter of SAUER-DANFOSS 90-series hydraulic components \( \Delta P_{\text{max}} = 480 \text{bar} \), therefore \( Q_{p_{\text{max}}} \geq 156.7 \text{L/min} \). Considering the cost-effectiveness and size for this study, 0-55 series variable displacement pump (VDP) and fixed-displacement motor (FDM) are suitable to selected[27], and the specific parameters of the hydraulic components are shown in Tab. 2.

### III. OPERATING PRINCIPLES OF THE MEH-PS

#### A. DRIVE MODES ANALYSIS

The switching element engagement states of the four drive modes of the MEH-PS are shown in Tab. 3.

### TABLE.3 Element engagement status of each drive mode

| Drive modes           | C1 | C2 | C3 | B1 |
|-----------------------|----|----|----|----|
| Pure electric drive (1)| ▲ |    |    | ▲ |
| Pure engine drive (2)  | ▲ | ▲ |    | ▲ |
| Torque coupling drive (3)| ▲ | ▲ | ▲ | ▲ |
| Speed coupling drive (4)| ▲ | ▲ | ▲ | ▲ |

The torque coupling drive mode can meet the operating conditions with high requirement of torque, and its power flow is shown in Fig 3(a), the speed and torque relationship between the input and output shaft is expressed as,

\[
\begin{align*}
n_3 &= n_{in1}/i_{g1} \\
n_{in2} &= n_{in1}i_{g2}/i_{g1} \\
T_3 &= T_{in1}i_{g1} + T_{in2}i_{g2}
\end{align*}
\] (5)

Where \( n_{in1} \) and \( n_{in2} \) and \( n_3 \) are the speed of input shaft 1, input shaft 2 and shaft 3 respectively, \( T_{in1} \), \( T_{in2} \) and \( T_3 \) are the torque of input shaft 1, input shaft 2 and shaft 3 respectively, \( i_{g1} \) and \( i_{g2} \) are the gear ratio.

According to the structure shown in Fig 2, if shaft 1 is connected to the engine and shaft 2 is connected to the motor, and \( i_{g1} = i_{g2} = 1 \) then the speed and torque of shaft 3 are \( n_3 = n_c = n_m \) and \( T_3 = T_c + T_m \); the above speed and torque relationship for torque coupling mode can also satisfied with pure electric drive mode and pure engine drive mode.

### B. TRANSMISSION MODES AND SPEED CHARACTERISTICS ANALYSIS

The MEH-PS has three transmission modes, which can be realized by controlling C4, B1, B2, and B3 for hydrostatic transmission mode, hydro-mechanical transmission mode, and mechanical transmission mode.

According to Fig 2, the speed relations can be described as,

\[
\begin{align*}
n_{in1} &= n_{in} \\
n_{in2} &= -i_t n_{in} \\
T_{in1} &= n_{in}n_{in2} = e n_M = -i_z n_M
\end{align*}
\] (8)

Where \( n_{in} \) is the carrier speed of the 1st planetary gear set, the subsequent speed and torque of the other planetary gears are named in the same way. \( n_{in} \) is the speed of input shaft, \( n_{p}, n_{M} \) is the speed of VDP and FDM respectively.

The relationship of the speed between each element of the planetary gear is as follows:

\[ n_1 + kn_t - (1+k)n_c = 0 \] (9)
If C4 and B2 are engaged, \( n_{c2} = 0 \). According to Eq (8) and Eq (9), the speed relationship of hydrostatic transmission mode can be described as,

\[
n_{c2} = \frac{en_{in}}{i_i i_k (1 + k_2)} \quad (10)
\]

If C4 and B1 are engaged, \( n_{c1} = 0 \). According to Eq (8) and Eq (9), the speed relationship of hydro-mechanical transmission mode can be described as,

\[
n_{c1} = \frac{ek_{k_1} n_{in} + k_2 i_i i_k (1 + k_1) n_{in}}{i_i i_k (1 + k_2)} \quad (11)
\]

If B1 and B3 are engaged, \( n_{s1} = n_{s2} = 0 \). According to Eq (8) and Eq (9), the speed relationship of mechanical transmission mode can be described as,

\[
n_{c3} = \frac{k_2 (1 + k_1) n_{in}}{k_k (1 + k_2)} \quad (12)
\]

The sub-transmission consists of planetary gears K3 and K4 as well as clutch C5 and brakes B4 and B5. Engaging any of the shift elements can form a gear set with a fixed transmission ratio. The function of sub-transmission is to expand the transmission ratio, which can realize the power demand of low speed and high speed at the same time; another function is to achieve synchronous shift between hydrostatic transmission mode and hydro-mechanical transmission mode, thus to realize the expansion of transmission ratio.

The ratio of sub-transmission is defined as,

\[
i_{st} = \frac{n_{c2}}{n_{out}} \quad (13)
\]

According to Fig 2, the speed relations of the sub-transmission is obtained as,

\[
\begin{align*}
    c3 & \quad c4 \\
    r4 & \quad s3 \quad s4 \quad c2
\end{align*}
\]

According to Eqs (10), (14), and (15), if C5 is engaged then \( n_{c3} = n_{c4} \), \( i_a = 1 \); if B4 is engaged, then \( n_{c3} = 0 \), \( i_a = \frac{k_4 (1 + k_3)}{k_k k_4 - 1} \); if B5 is engaged, then \( n_{c4} = 0 \), \( i_a = -k_k \).

Each of the above transmission modes theoretically has three gears, which are matched to obtain the transmission ratio between the input and output shafts of each gear is shown in Tab 4.

### TABLE 4. Element engagement status and transmission ratio of each gear

| Transmission modes | Gear | C4 | C5 | B1 | B2 | B3 | B4 | B5 | \( i_g \) |
|-------------------|-----|----|----|----|----|----|----|----|--------|
| Hydrostatic transmission | F(H) | ▲ | ▲ | ▲ | ▲ |
|                      | R(H1) | ▲ | ▲ | ▲ | | |
|                      | R(H2) | ▲ | ▲ | ▲ | | |
| Hydro-mechanical transmission | F(HM1) | ▲ | ▲ | ▲ | | |
|                      | F(HM2) | ▲ | ▲ | ▲ | | |
|                      | R(HM) | ▲ | ▲ | ▲ | | |
| Mechanical transmission | F(M1) | ▲ | ▲ | ▲ | | |
|                      | F(M2) | ▲ | ▲ | ▲ | | |
|                      | R(M) | ▲ | ▲ | ▲ | | |

Where F is forward gear, R is reverse gear, H is hydrostatic transmission, M is mechanical transmission, HM is hydro-mechanical transmission; \( i_g \) is the transmission ratio of HMT.

### TABLE 5. Transmission element parameters

| parameters | \( k_1 \) | \( k_2 \) | \( k_3 \) | \( k_4 \) | \( i_1 \) | \( i_2 \) |
|------------|---------|---------|---------|---------|---------|---------|
| Value      | 1.80    | 1.60    | 1.65    | 1.65    | 0.62    | 1.00    |
the motor speed increases from 0 to -5000rpm, there is an error between the theoretical and experimental values, but it's within the 5%, therefore the theoretical and experimental values of the transmission output speed are in good consistency. Speed coupling drive mode expands the transmission's range of speed regulation for higher tractor speed when the transmission ratio is limited.

C. TORQUE CHARACTERISTICS ANALYSIS

According to Fig 2, the torque relations can be described as,

\[
\begin{align*}
T_{e} &= \frac{T_{m}}{k_1} \frac{i_{1}i_{2}}{i_{3}}(1+k_{2}) \\
T_{c1} &= \frac{T_{e}}{k_2} \frac{i_{1}i_{2}}{i_{3}}(1+k_{2}) \\
T_{c2} &= \frac{T_{e}}{k_2} \frac{i_{1}i_{2}}{i_{3}}(1+k_{2})
\end{align*}
\]

(16)

Where \( T_{out} \) is the torque of the transmission output shaft and \( T_{P} / T_{M} \) is the pump/motor torque.

The relationship of the torque between each element of the planetary gear is as follows:

\[
T_{s} = \frac{T_{e}}{k} = \frac{T_{e}}{1+k}
\]

(17)

According to Eqs (17) and (18) the input shaft torque of the hydrostatic transmission mode is:

\[
T_{in} = \frac{T_{P}}{i_{1}} = \frac{\epsilon T_{out}}{i_{1}i_{2}i_{3}(1+k_{2})}
\]

(18)

The input shaft torque of the hydro-mechanical transmission mode is:

\[
T_{in} = T_{c1} + \frac{T_{P}}{i_{1}} = \frac{k_{2}(1+k_{1})}{k_{1}} \frac{\epsilon T_{out}}{i_{1}i_{2}i_{3}(1+k_{2})}
\]

(19)

The input shaft torque of the mechanical transmission mode is:

\[
T_{in} = T_{c1} = \frac{k_{1}(1+k_{1})T_{out}}{k_{1}(1+k_{2})i_{3}}
\]

(20)
coupling drive mode is 1000N·m. From the figure, it shows that under the same torque, the input torque required in low-gear is smaller than that required in high-gear, and the higher the gear, the smaller the maximum torque the transmission can withstand, and vice versa. Gear F (HM1) is suitable for high-load operation, and gear F (HM2) is suitable for low-load operation. Torque coupling drive mode has the ability of a low-power engine to complete high-power operation tasks to some extent.

D. POWER SPLIT AND EFFICIENCY CHARACTERISTICS ANALYSIS

The power split characteristic can be expressed as $\rho = \frac{P_H}{P_I}$, where $P_H$ is the output power transferred by the hydraulic path (the input power of the sun gear of K2), $P_I$ is the power of input shaft [28]. Fig 7 shows a simplified transmission chain.

![FIGURE.7 HMT closed drive circuit](image)

According to Eqs (9), (14), (17), and (21), it can be calculated that:

$$\rho = \frac{n_H T_H}{n_I T_I} = \frac{e}{i_{sl} k_i (1 + k_i)/k_i + e}$$

(21)

The relationship between the inverse of the forward transmission ratio and the power split ratio $|\rho|$ shown in Fig 8 is obtained from the Eq (21), in which the transmission power split ratio is kept at a lower level, which is good for the efficiency of transmission.

![FIGURE. 8 Hydraulic power split ratio curve](image)

For the 0-55 series VDP and FDM shown in Tab 2, the efficiency expression was obtained by fitting the data obtained from a self-designed test bench as follows,

$$\eta = 0.87\left[\left(\frac{n}{i_{sl}}\right)^{\frac{1}{1.035}} + 0.035\left(\frac{4n}{i_{sl}}\right)\right]$$

*$$exp\left(\frac{-33T_d}{D_{max}}\right) - \exp\left(\frac{-50T_d}{D_{max}}\right) + \exp\left(\frac{0.57k_i}{D_{max}}\right)$$*

(22)

$$\eta_\omega = 0.87\left[\left(\frac{en}{i_{sl}}\right)^{1.035} + 0.035\left(\frac{4en}{i_{sl}}\right)\right]$$

*$$exp\left(\frac{-33T_d}{D_{max}}\right) - \exp\left(\frac{-50T_d}{D_{max}}\right) + \exp\left(\frac{0.57k_i}{D_{max}}\right)$$*

(23)

Where $n_{P_{max}}/n_{M_{max}}$ is the rated speed of the VDP/FDM; $D_{P_{max}}/D_{M_{max}}$ is the rated displacement of the VDP/FDM.

For the closed transmission chain consisting of the fixed shaft gear set and the planetary gear set, the transmission efficiency can be calculated by the gear meshing efficiency method[29, 30], and according to the structural relationship in Fig 7, the relevant ratio is calculated as,

$$i_{sl} = \frac{1 + k_1}{k_1}$$

$$i_{sl} = \frac{e}{i_{sl} k_i (1 + k_i)/k_i + e}$$

$$i_{sl} = \frac{e}{i_{sl} k_i (1 + k_i)/k_i + e}$$

$$i_{sl} = \frac{e}{i_{sl} k_i (1 + k_i)/k_i + e}$$

(24)

Where $i_{C1}^a$, $i_{C1}^b$ are the ratios of C to I when a and b are fixed, respectively.

![FIGURE.9 Normal power flow and circulating power flow](image)
If $e > 0$, $\dot{i}_{C1}^b > 0$, as is shown in Fig 9(a), there is no circulating power in the closed transmission chain and the transmission efficiency is:

$$\eta_t = \left[1 + \left| e \right| \left| \frac{i_c^b - i_c^b}{i_c^b} \right| \varphi + \left| i_c^b \right| \left(1 - \eta_0\right)\right]^{-1} \eta_a \quad (25)$$

If $e < 0$, $\dot{i}_{C1}^b < 0$, as is shown in Fig 9(b), there is circulating power in the closed transmission chain and the transmission efficiency is:

$$\eta_t = \left[1 + \left| e \right| \left| \frac{i_c^b - i_c^b}{i_c^b} \right| \varphi + \left| i_c^b \right| \left(1 - \eta_0\right)\right]^{-1} \eta_a \quad (26)$$

Where, $\varphi$ is the planetary gear power loss coefficient, this paper takes 0.019, $\eta_0$ is the efficiency of the b-I transmission chain, $\eta_0 = \eta_1 \eta_2 \eta_3 \eta_4 \eta_5$, $\eta_a$ is the efficiency of the sub-transmission. The meshing efficiency of the gear set is often regarded as a constant[31], the efficiency of the external-external gear set is 99%, and the efficiency of the internal-external gear set is 99.5%.

Substituting Eq (25) into Eqs (26) and (27) to obtain the transmission efficiency shown in Fig 10.

![FIGURE.10 Transmission efficiency map](image)

**FIGURE.10 Transmission efficiency map**

Fig 10 shows that if $|e|$ is equal, the efficiency corresponding to the normal power flow is significantly higher than the efficiency corresponding to the circulating power flow, and the smaller the $|e|$, the higher the efficiency, and the highest transmission efficiency is achieved when $e=0$, the reason is that no power is transmitted through the hydraulic system, so the transmission energy loss is minimal. In addition, the larger the load torque of the hydraulic motor, the higher the efficiency of the transmission, to ensure higher efficiency of the transmission, the hydraulic system should avoid working in low load conditions.

**IV. SIMULATION AND TEST**

A. SIMULATION MODELING

According to the structure of MEH-PS shown in Fig 2 and the gear switching logic shown in Tab 3 and Tab 4, the simulation model of the whole tractor shown in Fig 11 was established in SimulationX. Because the acceleration process of full load transport requires high tractor traction and speed, it provides a valuable test condition for the performance of the tractor equipped with the MEH-PS[16, 21]. The tractor operation mass is 8260kg, the mass of cargo and trailer is 16440kg, the total mass of the tractor is 26350kg, the axle ratio is 23.68, the tire rolling radius is 0.75m.

![FIGURE.11 SimulationX model of the proposed MEH-PS](image)

**FIGURE.11 SimulationX model of the proposed MEH-PS**

B. SIMULATION RESULTS IN ACCELERATION CONDITION

The simulation results of the acceleration process of the tractor equipped with MEH-PS for full load transport are shown in Fig 12. Speed 1 and speed 2 in Fig 12(a) are the speed comparison of the tractor with motor-assisted (tractor 1) and tractor (tractor 2) without motor-assisted. Under the same shift conditions and engine throttle opening, the tractor speed increased from 0 to 40km/h, tractor 1 reached the speed in 58.6s and tractor 2 reached the speed in 51.5s. The acceleration results of 0~40km/h indicate that the tractor has better acceleration performance and dynamics in the torque coupling mode.

It is worth noting that in Fig 12(b), when switching from gear F (HM1) to gear F (HM2) at the moment of 20s, the displacement ratio changed instantaneously in steps and the transmission ratio decreased slightly, but it had no significant effect on the tractor speed because of the large vehicle inertia when the tractor was fully loaded. In addition, the gear or mode switching was also undertaken at the moments of 10s, 40s, and 65s, and the vehicle drive force changed abruptly at the time of switching, but the relevant parameters of the engine, motor, and hydraulic system changed within an accepted range.

The tractor switched to speed coupling drive mode in 65s, the speed increased from 40km/h to 60km/h in 65s~120s, the motor speed reached -5200rpm then, the speed increase was achieved through the motor speed
regulation, but the acceleration of the tractor is smaller compared to the acceleration of 0–40 km/h, which is the difference between speed coupling and torque coupling drive, expanding the speed range but lose some of the acceleration ability. In Fig 12(b), although the displacement ratio of the hydraulic system remains basically unchanged during the acceleration of 40–60 km/h, the power split ratio decreases, which has a deviation from the theoretical analysis results in Fig 8. The reason is that the proportion of the output power of the motor gradually increases during the acceleration process, so that the proportion of the output power of the hydraulic system decreases, and the power split ratio remains constant at the end of the acceleration, which is consistent with the real case. The efficiency of the transmission increases and this is related to the decrease of the hydraulic power split ratio, and the efficiency of the transmission in Figure 12(c) remains mostly above 86%.

Fig 12(g) shows that the accelerating process of the tractor only experienced four shifts to increase the speed from 0 to 60 km/h. Compared to conventional tractors equipped with stepped transmissions[32], the number of gear shifts is less, which is beneficial to the service life and driving comfort of the tractor. In Fig 12(d), 12(e), and 12(f), according to the parameters shown in Tab 1 and Tab 2, the speed and torque of the power source and oil pressure of the hydraulic system during the operation of the tractor are within its permissible parameters, and the designed MEH-PS can meet the requirements of tractor transport.
C. BENCH TEST

The test bench shown in Fig 13 is built according to the structural principle of Fig 14 of the MEH-PS, and the power of the traction motor (simulated engine) and the drive motor are combined to drive the transmission through the coupling mechanism, and the performance of the powertrain system can be tested by loading on the dynamometer with the control of the load torque. The parameters of the test bench were shown in Tab 5.

**FIGURE 12** Performance simulation results

**FIGURE 13** MEH-PS test bench. 1. Oil cooling system 2. Motor cooling system 3. Traction motor 4. Drive motor and coupling mechanism 5. HMT 6. Reduction gear 7. Power cabinet of dynamometer 8. Electrical dynamometer 9. Controller cabinet

**FIGURE 14** Test bench operating principle

**TABLE 5. Parameters of the test bench.**

| Components                  | Specifications                  |
|-----------------------------|---------------------------------|
| Traction motor              | Rated power: 500kW              |
|                             | Rated torque: 1250 N·m          |
| Electrical dynamometer      | Rated power: 1000kW             |
|                             | Rated torque: 2900 N·m          |
| Torque and speed sensor     | Range: 0–5000Nm; 0–8000r/min   |
|                             | Accuracy: ±0.05%                |

Fig 15 shows the comparison of the results about the simulation and experiment transmission ratio, the figure shows the ratio in the range of 0.41–12. When the tractor speed is low, the ratio slope changes greatly, and the tractor can get more acceleration at a low speed. When the speed is less than 5.6km/h, the difference between simulation and experiment data is larger due to gear F (H) is hydrostatic transmission, and the leakage of the hydraulic system has a greater impact on the transmission ratio.
FIGURE 15 Simulation and experiment transmission ratio

Fig 16 shows the comparison between simulation and experiment of transmission efficiency. There are two local highest efficiency critical points on the curve, both of which are the efficiency of mechanical transmission gears, and the experiment efficiency is lower than the simulated efficiency, which is attributed to the simplification of the simulation modeling. In addition, the efficiency of the experiment is significantly smaller than that of the simulation during mode switching because the simulation modeling considers only the transient effect and does not consider the energy loss of the shift element during the switching process. The results of the experiment and simulation are in good consistency overall.

FIGURE 16 Simulation and experiment transmission efficiency

V. CONCLUSION

In this paper, according to the characteristics of current mainstream drive and transmission technologies, a MEH-PS integrating electromechanical hybrid system and hydro-mechanical composite transmission is proposed to meet the operational requirements of the tractor. The scheme conception and structure design are completed, and the power sources and transmission components are selected according to the relevant work content of the tractor. The drive and transmission principles of the powertrain are discussed. The speed characteristics, torque characteristics, power split, and efficiency characteristics of the proposed system are theoretically analyzed. The simulation model of the whole tractor is established in SimulationX, and the acceleration process of full load transport is taken as the object of the study. The results show that the tractor equipped with MEH-PS has better dynamics and a wider speed range compared with the conventional tractor while maintaining a high transmission efficiency, and the selected power sources and transmission components are within the permitted parameters, which verifies the reliability of the scheme. Compare the simulation and experiment results found that: Although there are differences between the simulated and experimental data on speed regulation characteristics and efficiency characteristics in the tractor speed range, the overall is in good agreement, which confirms the advantages of the scheme in terms of speed regulation and efficiency performance. The powertrain can also apply to a loader, grader, and other vehicles, which has a relatively wide application prospect.

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This article has been accepted for publication in a future issue of this journal, but has not been fully edited. Content may change prior to final publication. Citation information: DOI 10.1109/ACCESS.2021.3126667, IEEE Access

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**Nomenclature**

| Symbol | Description |
|--------|-------------|
| \( F_t \) | wheel driving force [kN] |
| \( F_i \) | rolling resistance [kN] |
| \( F_i \) | ramp resistance [kN] |
| \( F_s \) | air resistance [kN] |
| \( F_j \) | acceleration resistance [kN] |
| \( F_t \) | hook tension [kN] |
| \( Z_x \) | number of the ploughshare |
| \( B_x \) | monomer ploughshare width [m] |
| \( H_x \) | ploughing depth [m] |
| \( K_s \) | soil specific resistance [N/cm²] |
| \( P_e \) | rated power of diesel engine [kW] |
| \( P_m \) | rated power of electric motor [kW] |
| \( \Delta P_{\text{max}} \) | maximum pressure of pump [bar] |
| \( Q_{\text{pmax}} \) | maximum flow rate of pump [L/min] |
| \( v \) | tractor speed [km/h] |
| \( n_e \) | rated speed of engine [rpm] |
| \( T_e \) | output torque of engine [N·m] |
| \( n_m \) | rated speed of motor [rpm] |
| \( T_m \) | output torque of motor [N·m] |
| \( K \) | planetary gear set |
| \( G \) | fixed shaft gear set |
| \( e \) | displacement ratio |
| \( i_g \) | gear ratio of transmission |
| \( k_1 \sim k_4 \) | characteristic parameter of the planetary gear set |
| \( K_1 \sim K_4 \) | \( i_{st} \) transmission ratio the sub-transmission |
| \( i_t \) | transmission ratio of engine and pump |
| \( i_2 \) | fixed-displacement motor gear ratio |
| MEH-PS | mechanic-electronic-hydraulic powertrain system |
| HMT | hydro-mechanical transmission |
| CVT | continuously variable transmission |
| ICE | internal combustion engine |
| MG | motor-generator |
| PTO | power take-off |
| TC | torque coupling |
| SC | speed coupling |
| DOH | degree of hybridization |
| VDP | variable displacement pump |
| FDM | fixed-displacement motor |

**GREEK LETTERS**

| Symbol | Description |
|--------|-------------|
| \( \eta_T \) | traction efficiency |
| \( \beta \) | overload capacity coefficient |
| \( \eta_p \) | efficiency of pump |
| \( \eta_M \) | efficiency of motor |
| \( \rho \) | power split ratio |
| \( \phi \) | planetary gear power loss coefficient |
| \( \eta_t \) | transmission efficiency |

**SUBSCRIPT AND SUPERSCRIPTS**

| Symbol | Description |
|--------|-------------|
| in | input shaft |
| out | output shaft |
| s | sun gear of the planetary gear set |
| c | carrier of the planetary gear set |
| r | ring gear of the planetary gear set |