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THEORETICAL SUBSTANTIATION OF IMPROVING THE TRACTION EFFICIENCY OF ELECTRIC TRACTORS THROUGH A PTO-BASED POWER ASSIST SYSTEM

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Abstract

This article develops the theoretical basis of a power assist system connected to the power take-off (PTO) shaft to increase the traction capability of electric tractors under high-load operating conditions. The study theoretically compares three powertrain layouts: a baseline dual-motor coupled powertrain (DMCP), a speed-coupling configuration with a power assist system, and a mixed-coupling configuration with a power assist system. The longitudinal motion of the tractor, wheel traction force, kinematic relationships of the planetary gear set, motor power demand, and energy consumption are described using mathematical equations. For 46 kW and 55 kW ploughing workload conditions, the power envelope, maximum traction force, and relative energy consumption were evaluated. The calculation results show that the integration of a PTO-based assist motor increases the traction force by 24.92-30.83% compared with the baseline DMCP. The speed-coupling configuration provides the highest traction force, whereas the mixed-coupling configuration is distinguished as a structurally simpler solution because it delivers torque directly to the output shaft. The proposed article can serve as a theoretical basis for selecting the power transmission architecture of electric tractors intended for agricultural working units.

Keywords: electric tractor, PTO, power assist system, DMCP, planetary gear, traction force, dynamic model, energy efficiency.

1. Introduction

The electrification of agricultural machinery is one of the key directions for reducing fuel consumption, lowering maintenance costs, and improving environmental safety during field operations. The advantages of an electric tractor are not limited to the replacement of diesel fuel. The high starting torque of the electric motor, fast torque control, and accurate power distribution under different operating modes make it a promising technical solution for agricultural mechanization [2, 6, 8, 13].

A major factor limiting the wider use of electric tractors under field conditions is the shortage of traction force during heavy ploughing, deep loosening, and operations on wet soils. In such processes, a tractor is evaluated not only by its nominal power but also by its ability to provide high torque for short periods. If the power transmission system cannot cover the peak load points, the tractor speed decreases, wheel slip increases, and energy consumption rises [1, 4, 12, 14].

In conventional electric tractor layouts, the main focus is usually placed on battery capacity, traction motor power, and inverter efficiency. In practical field operations, however, mechanisms driven through the PTO shaft also play an important role. Therefore, the design of an electric tractor requires a joint consideration of traction motion, PTO power, and power exchange between auxiliary systems [2, 3, 7, 13].

In this study, a PTO-based power assist system is considered as a method for increasing the traction capability of an electric tractor without a complete redesign of the existing powertrain layout. The research concept is based on connecting an additional electric motor to the PTO shaft so that, under heavy operating conditions, it supplies additional torque to the main power transmission path. As a result, the maximum traction force of the tractor increases, motor loading is redistributed, and the useful operating point of the system is improved [1, 3, 9, 10].

2. Research Aim and Scientific Objectives

The aim of the study is to provide a theoretical substantiation of a PTO-based power assist system for increasing the traction force of electric tractors under high-load conditions and to evaluate its operating efficiency using a mathematical model [1, 3, 9].

To achieve this aim, the following scientific objectives were addressed: the main elements of the electric tractor powertrain architecture were systematized; a layout for connecting an additional motor through the PTO path was proposed; a longitudinal dynamic model of the tractor was developed; torque and speed relationships based on a planetary gear set were formulated; baseline and power-assisted configurations were compared through the F-V power envelope; and energy consumption was evaluated using the principle of dynamic programming [3, 7, 9, 10].

The object of the research is the dual-motor coupled electric tractor powertrain system. The subject of the research is the change in traction force, motor operating mode, and energy consumption resulting from the connection of an additional motor through the PTO shaft [3, 9].

Table 1. Configurations considered in the study

Configuration	Composition	Main feature
Baseline DMCP	Main motor + auxiliary motor + planetary gear set	The torque of two motors is combined at the carrier
Speed-coupling + assist	DMCP + assist motor through the PTO path	The assist motor adds supplementary torque to the ring gear
Mixed-coupling + assist	DMCP + assist motor near the output shaft	The assist torque is added directly to the carrier

3. Theoretical Basis of the PTO-Based Power Assist System

The main function of the PTO shaft is to transmit tractor power to external working units. In the proposed approach, this shaft is considered not only as a power output point but also as a mechanical connection that reintroduces the torque of an additional electric motor into the power transmission system. This solution allows a short-term increase in traction power under heavy operating conditions and enables motors to operate at more economical points under light-load conditions [1, 11, 15].

The physical concept of the system is shown in Figure 1. The battery and inverter transmit electrical energy to the main, auxiliary, and assist motors. The main and auxiliary motors combine torque through the planetary mechanism. The assist motor provides additional torque

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through the mechanical PTO path. The electronic control unit manages power distribution among the motors depending on load, speed, and the selected operating mode [1, 3].

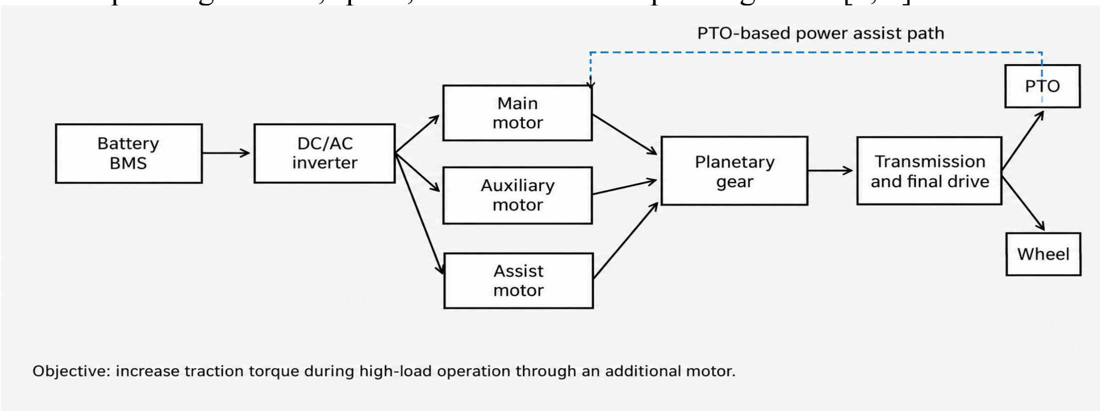

Figure 1. Powertrain layout of an electric tractor equipped with a PTO-based power assist system.

The total traction power of the tractor is determined by the travel speed and traction force [6, 13]:

$$P_t = F_t \cdot v \quad (1)$$

where P_t is the traction power, kW; F_t is the traction force, kN; and v is the tractor travel speed, m/s. The traction force based on wheel torque is determined as follows:

$$F_t = \sum \left(\frac{T_w}{r_w} \right) \quad (2)$$

where T_w is the wheel torque, $N{\cdot}m$; and r_w is the wheel radius, m. During ploughing, the total resistance that the tractor must overcome consists of the following components:

$$R_\Sigma = R_i + R_f + R_a + R_s \quad (3)$$

where R_i is the soil resistance of the working unit; R_f is rolling resistance; R_a is aerodynamic resistance; and R_s is slope resistance. Under low-speed ploughing conditions, R_i and R_f dominate. Therefore, the power assist system provides the greatest benefit in low-speed, high-torque operating modes [12, 14, 15].

4. Planetary Gear Model and Torque Distribution

The central element of the DMCP system is the planetary gear set. In this mechanism, the ring gear, sun gear, and carrier are kinematically coupled. The main motor is connected to the ring gear, the auxiliary motor is connected to the sun gear, and the output is connected to the carrier. The connection point of the assist motor differs depending on the selected configuration [3, 5, 9].

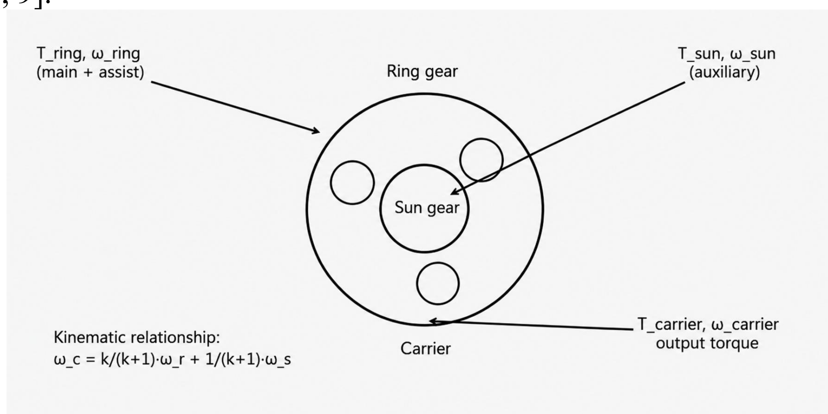

Figure 2. Theoretical relationship between torque and angular velocities in the planetary gear set.

For the planetary gear set, the torque relationship is expressed as [3, 5]:

$$T_c = \min\left(\frac{k+1}{k}\, T_r,\ (k+1)\, T_s\right) \quad (4)$$

where T_c is the torque on the carrier; T_r is the torque on the ring gear; T_s is the torque on the sun gear; and k is the ratio between the number of ring gear teeth and the number of sun gear teeth. The kinematic relationship is expressed by the following equation:

$$\omega_c = \frac{k}{k+1}\,\omega_r + \frac{1}{k+1}\,\omega_s \quad (5)$$

In the speed-coupling power-assisted configuration, the assist motor is connected to the ring-gear path. Therefore, the equivalent torque on the ring-gear side is written as follows [1, 3]:

$$T_r = (T_{\text{main}} - I_{\text{main}}\dot{\omega}_{\text{main}})\, i_{\text{main}} + (T_{\text{assist}} - I_{\text{assist}}\dot{\omega}_{\text{assist}})\, i_{\text{assist}} \quad (6)$$

The auxiliary motor provides torque to the sun gear:

$$T_s = (T_{\text{aux}} - I_{\text{aux}}\dot{\omega}_{\text{aux}})\, i_{\text{aux}} \quad (7)$$

In the mixed-coupling configuration, the assist motor adds torque to a point close to the output shaft independently of the internal kinematics of the planetary gear set. In this case, the transmission input torque is determined as follows:

$$T_{\text{out}} = T_c + (T_{\text{assist}} - I_{\text{assist}}\dot{\omega}_{\text{assist}})\, i_{\text{assist}} \quad (8)$$

The traction torque transmitted to the wheel is determined by the final drive and the selected transmission gear ratio:

$$T_t = T_{\text{out}} \cdot i_0 \cdot i_j \cdot \eta_m \quad (9)$$

where i_0 is the final drive ratio; i_j is the transmission ratio of the j-th gear; and η_m is mechanical transmission efficiency. The wheel traction force is then:

$$F_w = \frac{T_t}{r_w} \quad (10)$$

Theoretically, the speed-coupling configuration provides a higher torque multiplication effect because the assist motor torque enters the planetary mechanism through the ring gear. In the mixed-coupling configuration, the torque multiplication effect is lower, but the transmission path is simpler and the control algorithm is more stable [1, 3, 9].

5. Energy Consumption and Optimal Control Model

Maximum traction force alone is not a sufficient indicator for evaluating electric tractor efficiency. The operating speed and torque point of the motors, inverter efficiency, transmission losses, and mode switching must also be taken into account. Therefore, energy consumption is evaluated using the following expression [6, 7, 10]:

$$P_{\text{req}}(k) = \frac{T_{\text{main}}\omega_{\text{main}}}{\eta_{\text{main}}} + \frac{T_{\text{aux}}\omega_{\text{aux}}}{\eta_{\text{aux}}} + \frac{T_{\text{assist}}\omega_{\text{assist}}}{\eta_{\text{assist}}} \quad (11)$$

The total energy consumption over a discrete time interval is:

$$E = \frac{\sum \left[P_{\text{req}}(k) \cdot \Delta t\right]}{3600} \quad (12)$$

The optimal control problem is aimed at minimizing total energy consumption by selecting motor torque, motor speed, and operating mode at each time step. In dynamic programming, this is expressed by the state variable *x(k)*, control variable *u(k)*, transition function *f(x,u)*, and cost function *L(x,u)* [7, 10]:

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$$x(k+1)=f(x(k),u(k)) \quad (13)$$

$$J^*(x(k))=\min_u\left[L(x(k),u(k))+J^*(x(k+1))\right] \quad (14)$$

Abrupt changes in motor speed and excessive switching between operating modes may cause thermal loading, clutch wear, and additional energy losses in a real system. Therefore, penalty terms are introduced into the cost function [7, 10]:

$$L(x,u)=P_{req}(k)+\alpha\cdot|\Delta\omega|+\gamma\cdot S_{mode} \quad (15)$$

where α is the penalty coefficient for motor speed variation; γ is the penalty coefficient for mode switching; and S_{mode} is an indicator of operating mode switching. Figure 3 shows the general logic of the dynamic programming process.

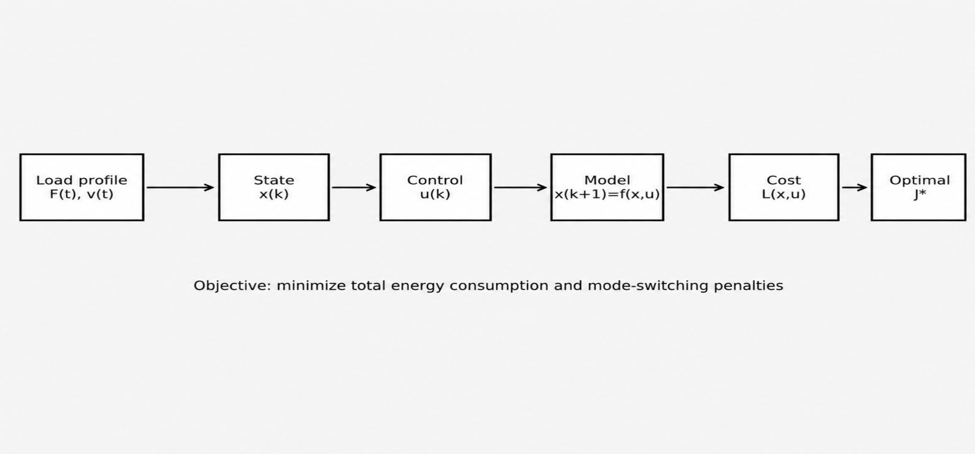

Figure 3. General logic of selecting optimal power distribution based on dynamic programming.

6. Calculation Results and Analysis

The theoretical calculations were performed for three configurations using identical motor parameters. The nominal power of the main motor was assumed to be 30 kW, while the nominal power of the auxiliary and assist motors was assumed to be 10 kW. For each motor, the nominal rotational speed was taken as 4000 rpm, the maximum rotational speed as 6000 rpm, the maximum torque of the main motor as 72 N·m, and the maximum torque of the auxiliary and assist motors as 24 N·m [1, 3, 9].

Table 2. Motor parameters used in the calculations

Component	Nominal power, kW	Nominal speed, rpm	Maximum speed, rpm	Maximum torque, N·m
Main motor	30	4000	6000	72
Auxiliary motor	10	4000	6000	24
Assist motor	10	4000	6000	24

The capability of the configurations to cover ploughing load demand was evaluated through the F-V power envelope. The baseline DMCP reached a maximum traction force of 24.36 kN. When the power assist system was added, the speed-coupling configuration provided a maximum traction force of 31.87 kN, while the mixed-coupling configuration provided 30.43 kN. These values correspond to increases of 30.83% and 24.92%, respectively, compared with the baseline case [1, 9]:

$$K_F=\frac{F_{\text{assist}}-F_{\text{base}}}{F_{\text{base}}}\cdot 100\% \quad (16)$$

$$K_F(\text{speed})=\frac{31.87-24.36}{24.36}\cdot 100=30.83\% \quad (17)$$

$$K_F(\text{mixed})=\frac{30.43-24.36}{24.36}\cdot 100=24.92\% \quad (18)$$

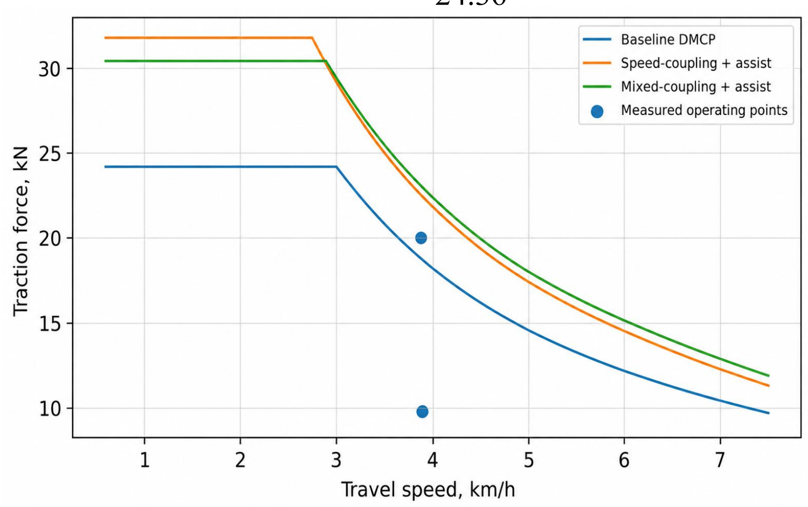

Figure 4. F-V power envelope for the baseline and PTO power-assisted configurations.

Under the 46 kW ploughing load, all configurations were able to cover the required operating points. Under the 55 kW load condition, the baseline DMCP moved outside the force envelope, whereas both power-assisted configurations remained within the required operating range. This result indicates that the PTO-based additional motor extends the practical operating boundary of the tractor under high-load conditions [1, 9].

Table 3. Traction indicators by configuration

Configuration	**Maximum traction force, kN**	**Increase relative to baseline, %**	**Ability to perform 55 kW load**
Baseline DMCP	24.36	0	No
Speed-coupling + assist	31.87	30.83	Yes
Mixed-coupling + assist	30.43	24.92	Yes

For energy efficiency analysis at the 46 kW workload, the baseline DMCP was conditionally taken as 100%. The speed-coupling power-assisted configuration reduced energy consumption by 2.40%, while the mixed-coupling configuration reduced it by 1.78%. Relative energy saving was determined as follows [7, 10]:

$$K_E=\frac{E_{\text{base}}-E_i}{E_{\text{base}}}\cdot 100\% \quad (19)$$

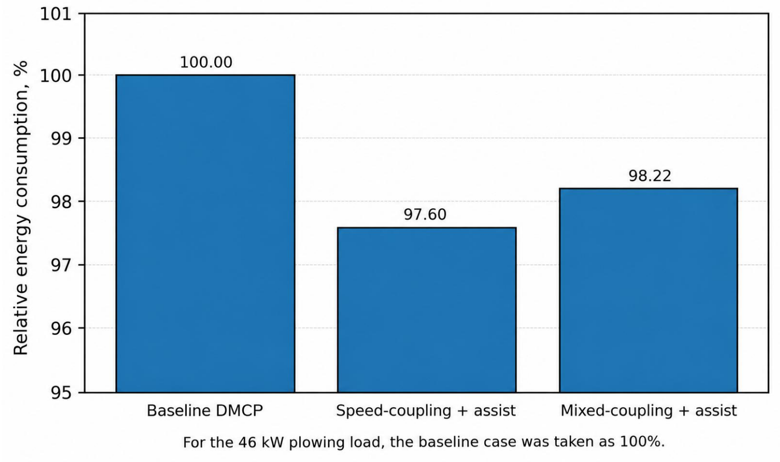

Figure 5. Relative energy consumption under the 46 kW load condition.

The speed-coupling configuration provided higher energy saving because the three-motor layout distributed the load more appropriately among the motors and moved the main motor away from an unfavorable high-torque operating point. In the mixed-coupling configuration, energy saving was slightly lower, but the simplicity of the mechanical connection and the direct torque transfer to the output shaft are important advantages for practical design [10].

Table 4. Theoretical assessment of the power assist system

Criterion	Speed-coupling + assist	Mixed-coupling + assist
Traction force	Highest, 31.87 kN	High, 30.43 kN
Maximum speed	Decreases due to kinematic limitation	Relatively stable
Structural complexity	Higher	Low to moderate
Energy consumption	Decreases by up to 2.40%	Decreases by up to 1.78%
Practical application	Suitable for heavy ploughing	Convenient for multi-mode field operations

7. Discussion

The obtained results show that the PTO-based power assist system increases the traction capability of the electric tractor without requiring major structural changes to the power transmission system. This approach is particularly useful for upgrading an existing platform. Increasing the power of the main motor or installing a larger battery increases tractor mass, cost, and thermal load. Connecting an assist motor through the PTO path, by contrast, provides additional torque when it is needed [1, 2, 13].

The advantage of the speed-coupling configuration lies in the torque multiplication effect generated through the planetary gear set. Since the assist motor supplies torque to the ring-gear path, the output traction force increases substantially. At the same time, the torque required from the sun-gear side also increases, which limits the maximum travel speed. Therefore, this configuration is suitable for ploughing, deep tillage, and field operations with high draft resistance [1, 3].

In the mixed-coupling configuration, the assist motor torque is added directly to the carrier or to the transmission input shaft. In this case, the torque multiplication effect is lower than in the speed-coupling layout, but the mechanical requirements of the system are simpler. Mode-switching control is also more stable. Therefore, this layout can be considered a convenient solution for transport, cultivation, and mixed work cycles requiring PTO operation in addition to ploughing [1, 3, 9].

The control strategy for electric tractors is as important as the mechanical layout. The variation in energy consumption among different configurations with the same motors confirms the importance of power distribution. Dynamic programming brings the motors closer to the optimal efficiency zone, limits abrupt speed changes, and reduces excessive operating mode switching. For a real tractor, this strategy can later be implemented as model predictive control or a rule-based adaptive algorithm [7, 10, 11].

The practical value of the study is that the proposed theoretical model can serve as a preliminary calculation platform for designing an electric tractor prototype. The model makes it possible to jointly evaluate motor power, gear ratio, PTO assist path, battery power, and control limits. This reduces technical risk before developing an experimental prototype [9, 14, 15].

8. Conclusion

- The PTO-based power assist system is theoretically substantiated as a solution for increasing the traction capability of an electric tractor under high-load operating conditions.
- The kinematics of the planetary gear set shows that the connection point of the assist motor directly affects both maximum traction force and maximum travel speed.
- The speed-coupling power-assisted configuration achieved a maximum traction force of 31.87 kN, which corresponds to a 30.83% increase compared with the baseline DMCP.
- The mixed-coupling power-assisted configuration achieved a traction force of 30.43 kN and can serve as a convenient solution for multi-mode field operations due to its structural simplicity.
- Under the 46 kW load condition, all configurations can operate, whereas under the 55 kW load condition, only configurations equipped with the power assist system remain within the required power range.
- Energy evaluation based on dynamic programming showed an energy saving of up to 2.40% in the speed-coupling configuration and up to 1.78% in the mixed-coupling configuration.
- Future research should verify the proposed theoretical model using a hardware-in-the-loop test bench, field tests under real soil conditions, and thermal regime analysis.

9. Practical Recommendations for Prototype Testing

When moving from the theoretical model to a practical prototype, the test program should be organized in stages. First, electrical safety, inverter protection, battery temperature, and motor current limits should be checked under laboratory conditions. Then, wheel torque, travel speed, and PTO shaft rotational speed should be measured on a load test bench. Field testing should begin only after the braking, clutch, and emergency shutdown systems operate reliably.

Three operating modes should be evaluated separately during testing. In the first mode, only the baseline DMCP should be used, and the traction force should be taken as the reference value. In the second mode, the speed-coupling power assist system should be activated. In the third mode, the mixed-coupling power assist system should be evaluated. In each mode, soil moisture, tillage depth, travel speed, and wheel slip should be maintained within the same range. This will ensure a correct comparison of results.

The measurement system should continuously record at least four parameters: motor torque, motor speed, battery current, and wheel rotational speed. Based on these data, traction power, electrical energy consumption, and the actual effectiveness of the power assist system should be determined for each work cycle. The test results should then be compared with the theoretical F-V envelope. If the practical traction force is significantly lower than the model prediction, the main causes may be wheel slip, transmission losses, or motor thermal limitations.

Table 5. Recommended measurement indicators for prototype testing

Indicator	Measurement device	Evaluation purpose
Wheel torque	Torque sensor	Determination of traction force
Motor speed and torque	Inverter data	Evaluation of the efficiency operating point
Battery current and voltage	BMS and power analyzer	Calculation of energy consumption
PTO rotational speed	RPM sensor	Verification of assist-path stability
Soil moisture and tillage depth	Moisture meter and depth sensor	Unification of test conditions

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