RESEARCH ON PARAMETER MATCHING METHOD OF PURE ELECTRIC HORTICULTURAL MACHINERY DRIVELINE BASED ON WORKING CONDITION CHARACTERISTICS
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考虑作业工况特性的纯电动园艺作业机械动力传动系统参数匹配方法研究

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ABSTRACT
To meet the economy requirement of the horticultural machinery, a two-speed transmission system was proposed based on the characteristics of the driving cycle of the horticultural machinery. Firstly, the test cycle of the horticultural machinery was established based on the data collector that fixed on the machinery. Secondly, the two-speed driveline system was designed. To reduce the energy consumption in the horticultural machinery working cycle, the gear ratio of the two-speed gear box was optimized with the goal of minimum energy consumption by the Genetic Algorithm. The optimized gear ratio were 11.6 and 9.62. The comparison of energy consumption between single gear ratio and two-speed gear ratio was made. The comparison result showed that the energy consumption can reduce 0.25% under one transportation test condition, the energy consumption can reduce 0.41% under one ploughing test condition, and the energy consumption can reduce 0.41% under one rotary test condition.

摘要
针对纯电动园艺作业动力机械使用周期内需具备田间道路行驶和满足不同作业需求的特点，本研究考虑不同作业工况的经济性，提出了一种两挡园艺作业动力机械动力传动系统。根据园艺作业动力机械的工作特点，构建了作业工况模型，以作业过程能耗最小为目标，两挡变速箱的传动比为变量，采用遗传算法对两挡位变速箱的传动比进行优化，得到 1 挡和 2 挡的传动比分别为 11.6 和 9.62。并将优化后的传动比与单挡位的情况进行对比，结果表明：优化后的双挡位变速箱在单个运输测试工况下能耗降低 0.25%，单个犁耕测试工况下，能耗降低 0.41%，单个旋耕测试工况下，能耗降低 0.41%。

INTRODUCTION
Horticultural machinery is the main equipment used to develop family farming and yard economy, it can complete the task of ploughing, rotary tillage and transportation, other operations by equipping with different agricultural machinery. It can also be used in some urban public utilities. Most of the arable land in China is mainly based on family farming, and the types of crop planting and agronomy in different regions are different, therefore, farmers have a greater demand for horticultural machinery. They can not only carry out agricultural production activities during busy agricultural hours, but also carry out some transportation business activities in agricultural leisure time by using these power machinery (*Cao Zhang et al., 2015, Yiqi Huang et al., 2007*).

Pure electric power horticultural operation machinery has the advantages of clean, environmental protection, easy to control, and is the development direction of agricultural machinery in the future (*Ali et al., 2022*). Electric driveline for agricultural tractors has been focused on by researchers (*Rossi, C et al., 2021*). As the core device of pure electric horticultural operation machinery, the driveline system directly affects the performance of the dynamics and the fuel consumption (*Kargar M. et al., 2022, Lee T. et al., 2023, Park J., 2022*).

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Compared with fuel machinery, the motor in the pure electric power mechanical drive system has the characteristics of large starting torque, wide speed regulation range and good NVH (Noise, Vibration, Harshness), which is very suitable for yard, facility agriculture and field operations.

At present, the research on horticultural machinery mainly focuses on the configuration of the drive system and its parameter optimization. Researchers mainly analyzed the performance of single-motor drive and dual-motor drive tractor, and optimizes its key parameters.

Fang et al., (2017), designed a single-motor driven pure electric tractor driveline system based on the characteristics of agricultural tractors in Anhui Province, and he matched the power transmission parameters through simulation. But he did not optimize the key parameters in the transmission system. *Liu et al.,* (*2022*), proposed a two-way coupling device for pure electric tractors in view of the low power utilization rate of dual-motor driven tractors. They analyzed and optimized the topology of the coupling device. *Yu Liu, (2023*), proposed a dual-motor driven pure electric tractor power coupling system in view of the problems existing in the current research on the parameter matching of pure electric tractor drivetrain, such as not considering the power efficiency distribution, excessive weight coefficient of static index optimization and ignoring dynamic load, and using a double-layer collaborative parameter optimization method to optimize the parameters of the power transmission system. *Gaoli Chen, (2023),* proposed an improved undominated ranking genetic algorithm for the optimization parameters and complex constraints of the pure electric tractor power transmission system, and optimized the heat loss and total efficiency of the drive motor based on this method. *Xiaolei Cai et al., (2023),* designed an electric tractor for greenhouse driven by brushless motor, and controlled the brushless DC motor. *Guoxiang Lu et al., (2020*), proposed a method to realize multi-directional power output in view of the multi-directional power output requirements of horticultural tractors, based on which a multi-power take-off device was designed, and the structural parameters and performance of the device were simulated and optimized by three-dimensional modelling method. *Junxiang Zhang et al., (2020),* proposed a driveline using two-speed transmission in view of the actual working characteristics of pure electric tractor, and optimized the gear transmission ratio with the power and working time of the electric tractor, the results show that the two-speed driveline can improve the overall operating performance of the electric tractor, but in this paper, the transmission ratio optimization only considers the optimization of ploughing and rotary tillage speed, and does not analyse the economy of the tractor under driving conditions.

To design a high efficiency and pure electric horticultural machinery driveline that satisfy different working cycle, a two-speed driveline system was proposed and the test working cycle of ploughing, rotary tillage, and transportation were established with the data that collected from the working machineries. The gear ratio of the two-speed transmission was optimized with the goal of minimum energy consumption by the genetic algorithm.

MATERIALS AND METHODS

Construction of test working conditions

The main operation methods of horticultural machinery include: ploughing, rotary tillage and transportation driving. It is necessary to construct test working conditions to analyze its operating conditions and load when matching the power parameters of horticultural operation machinery (*Jingwen Xaio, et al., 1998; Xiaosen Hou et al., 2019)*. Therefore, the force analysis of each test working cycle was done first, and the test working cycle of ploughing, rotary tillage and transportation were established first.

Test working cycle of ploughing

The horticultural machinery will suffer the resistance force from the solid and the equipped agricultural machinery, when it does the ploughing work. Assuming the resistance force of agricultural machinery is F_P, its calculation equation can be expressed by equation (1).

$$F_p = Z \cdot k \cdot b_l \cdot h_k \tag{1}$$

where, k is the specific resistance of the soil, kPa; b_l is the width of a single ploughshare, cm; h_k for depth of the ploughing, cm; Z is the number of ploughshares.

When the horticultural machinery is operated in the field, it is generally a cyclic operating condition, and its main working process includes: acceleration, agricultural tools piercing into the soil, running with constant speed, agricultural tools unearthing and stopping, and in the stage of acceleration and deceleration the agricultural tools piercing into the soil and unearthing the soil. Therefore, it is necessary to construct a suitable test condition, according to the calculated load of ploughing operation and the speed of ploughing operation when matching the dynamic parameters of horticultural machinery.

And considering the reciprocating nature of horticultural machinery during operation, a test working condition (one-way ploughing length of 200 m as the test data) was established, as shown in Figure 1.

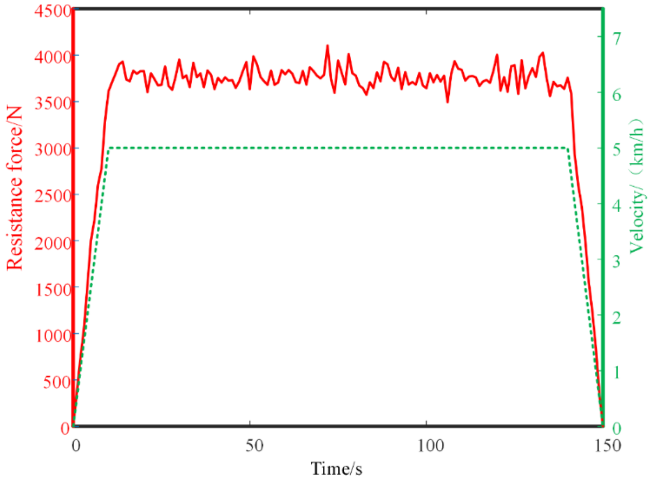

Fig. 1 - Test cycle of ploughing

Test conditions of rotary tillage operation

Force analysis of the rotary tillage knife is complicated, when the horticultural machinery is in the operation of rotary tillage, and many methods to calculate the power consumption of the rotary cultivator were proposed, such as unit analysis method, energy analysis method, specific work method, specific resistance method and other analysis methods. In practice, the consumption power of the rotary tillage is calculated by equation (2) (*Junxiang Zhang et al., 2020*).

$$N = 0.1 \cdot K_{\lambda} \cdot d \cdot v \cdot B \tag{2}$$

where N is the power consumption of the rotary cultivator, kW; v is the forward speed of the machinery, m/s; B is the width of the tillage cultivator, m; d is the depth of the tillage cultivator, cm; K_{λ} is the specific resistance of rotary tillage, N/cm^2.

Referring to the ploughing condition and agricultural machinery manual, the speed of the horticultural machinery was set to 5 km/h, and the test cycle of rotary tillage was established, which is shown in Figure 2.

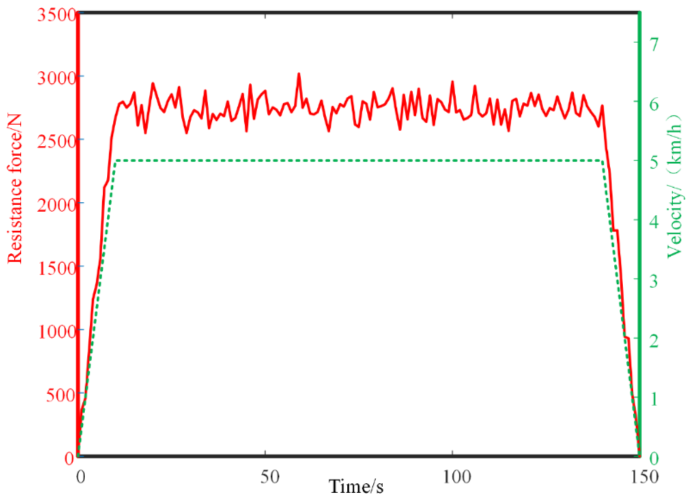

Fig. 2 - Test cycle of rotary tillage

Transportation test conditions

In addition to the operation in the field, the horticultural machinery also carries out transportation operations. During transportation, the machinery suffers the resistance from the air, slope, and soil. Assuming the resistance is F_r, it mainly includes acceleration resistance, F_j, rolling resistance, F_f, air resistance, F_w, and ramp resistance F_i.

$$F_r = F_f + F_w + F_i + F_j \tag{3}$$

where: $F_f = mgf\cos\theta, F_w = 0.5C_D A\rho v^2, F_i = mg\sin\theta, F_j = \delta m\,du/dt$, ρ is the air density, v is the vehicle speed, and δ is the vehicle rotation mass conversion coefficient.

From the above formula, it can be seen that in the case of knowing other structural parameters, the speed of the vehicle can be determined to determine the resistance of the power machinery transportation condition, according to the situation of the horticultural under the transportation condition, referring to the New European Driving Cycle (NEDC) cycle, the transportation test cycle can be established, which is shown in Figure 3.

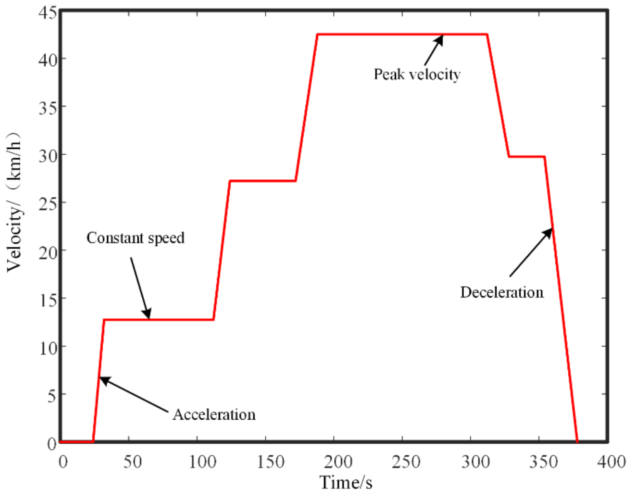

Fig. 3 - Test cycle of transportation

Design of pure electric horticultural machinery driveline system

The main function of horticultural machinery is to pull agricultural machinery in the field to operate at a constant speed, and under pure traction conditions, the working machinery is not only driven by the driving resistance during the driving process, but also by the traction of the agricultural tools, which can be shown by equation (4).

$$F_T = F_r + F_p \tag{4}$$

where F_T is the driving force of the horticultural machinery.

Calculation of motor power

All the power of the horticultural operation machinery comes from the electric motor, so the power of the motor should be higher than the demand power of several working conditions of the horticultural machinery. The power of the ploughing cycle can be calculated by equation (5).

$$P_p = \left(F_r + F_p\right) v_p / \eta_T / 1000 \tag{5}$$

where: P_p is the demand power under ploughing conditions, kW; v_p is the speed, m/s; η_T is the efficiency.

The power of rotary tillage can be calculated by equation (6).

$$P_N = N + F_r v_p / \eta_T / 1000 \tag{6}$$

where: P_N is the required power under rotary tillage conditions, kW.

The power required for transportation conditions can be calculated by equation (7).

$$P_r = F_r v_p / \eta_T / 1000 \tag{7}$$

where: P_r is the demand power under transportation conditions, kW.

Then the required power of the motor P_m is:

$$P_m \geq \max\left\{P_p, P_r, P_N\right\} \tag{8}$$

Design of the battery

As the only power source of electric horticultural machinery, the power battery pack needs to meet the power demand and operation time demand of horticultural machinery.

First, in terms of power demand, it is necessary to satisfy Equation (9).

$$P_{m\_\max} \le n_m P_b \eta_{bat} \tag{9}$$

where: P_{m_max} is the peak power of the motor, and n_m is the number of batteries that is calculated to meet the power demand. η_{bat} is the efficiency of the battery and motor controller, and P_b is the output power of a single battery, kW.

Secondly, in terms of the working time demand, it is necessary to satisfy Equation (10).

$$P_m t_w \le \frac{n_h W_{bat}}{3.6 \times 10^{-6}} \tag{10}$$

where: n_h is the number of batteries calculated to meet the working time requirements, W_{bat} is the energy that can be released by a single battery, and t_w is the working time of the unit, h.

In the design of the power battery pack, the largest of the two n_h and n_m should be selected.

Design of the gearbox

Though the drive motor has the ability of zero speed starting, and can output constant torque below the rated speed, and has the characteristics of constant power above the rated speed compared with the traditional internal combustion engine, it is necessary to make the motor work in the high-efficiency area as much as possible (that is, near the rated speed of the motor). It is stipulated that when the motor is at rated speed, the corresponding vehicle speed is at rated speed, so the energy consumption of horticultural machinery can be reduced and the working time can be extended. The motor can work in the high efficiency area with more gears (*Tao Liu, 2020; Yanni Chen et al., 2018; Kang Huang et al., 2020*).

According to the principle that the driving force curve of adjacent two gears must have at least one intersection point in the design of multi-speed transmission, the number of the gears of the transmission can be calculated, as shown in equation (11).

$$\frac{v_{\max}}{v_{rat}} \le \beta^n \tag{11}$$

where, β is the coefficient of the motor expanding the constant power zone, the general value is 2~4, and n is the number of the gear.

From the above formula, the value range of the number of blocks can be obtained.

$$n \ge f\left(v_e, \beta\right) = \frac{\log\left(\dfrac{v_{\max}}{v_e}\right)}{\log\left(\beta\right)} \tag{12}$$

According to the formula, the relationship between the rated vehicle speed and the transmission gear can be obtained.

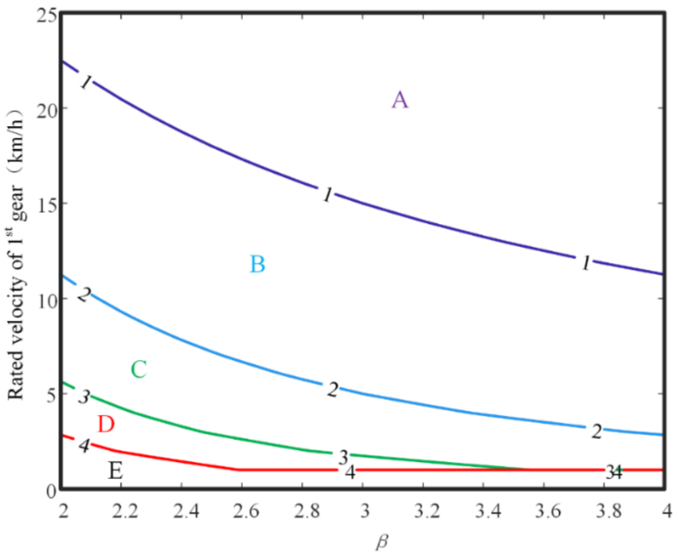

Fig. 4 - The relationship between the number of gears and the rated speed of 1st gear

Figure 4 showed relationship between the transmission gear number and the rated point speed of 1st gear. It can be seen from the figure that the gear number divides it into different areas, wherein area A represents the range of rated speed of 1st gear when selecting single gear, area B represents the range of rated point speed of 1st gear when two gears are set, area C means the range of 1st gear rated point speed when setting three gears, area D represents the range of 1st gear rated point speed when setting four gears, and area E represents the range of 1st gear rated point speed when setting five gears. Considering the wide range of values of the rated point speed of the first gear in the B area and the reduction of the complexity of the transmission system, it is possible to set two gears to improve the power and economy of the horticultural machinery.

Gear ratio parameters

According to the kinematic relationship of the vehicle driveline, the maximum gear ratio of the vehicle gearbox has the following relationship (equation (13)) with the peak speed of the motor and the maximum speed of the vehicle (*Gözen E. et al., 2022*).

$$i_{hg} \le 0.377\frac{n_{\max} r_w}{v_{\max}} \tag{13}$$

Among them, i_{hg} is the gear ratio of the highest gear of the transmission, r_w is the tire radius, and n_{max} is the peak speed of the motor.

Considering the characteristics of the motor, the maximum speed of the corresponding motor at the highest speed is generally 90%~95% of the peak speed, so the range of the 2-speed transmission ratio can be obtained by equation (14).

$$0.377\frac{n_{\max} r_w}{v_{\max}} \times 90\% \le i_2 \le 0.377\frac{n_{\max} r_w}{v_{\max}} \times 95\% \tag{14}$$

According to the principle that the driving force curves of adjacent two gears must have at least one intersection, their constraint relationship can be determined, as shown in equation (15).

$$\begin{cases} 0.377\dfrac{n_e r_w}{i_1} \le 0.377\dfrac{n_e r_w}{i_2} \\ 0.377\dfrac{n_e r_w}{i_2} \le 0.377\dfrac{n_{\max} r_w}{i_1} \end{cases} \tag{15}$$

From the above equation, the relationship between the 1st gear ratio and the 2nd gear ratio can be obtained, which can be shown in equation (16).

$$i_2 < i_1 \le i_2\beta \tag{16}$$

Since the rated speed of the motor has been determined, the relationship between the 1st gear ratio and the rated point speed of 1st gear can be calculated by equation (17).

$$0.377\frac{n_e r_w}{v_{e\_\min}} \ge i_1 \ge 0.377\frac{n_e r_w}{v_{e\_\max}} \tag{17}$$

where: v_{e_min} is the minimum value of the 1st gear rated point speed range; v_{2_max} is the maximum value of the 1st gear rated point speed range.

Combining the above formula, the range of gear ratios of 1 and 2 gears can be obtained, as shown in Table 1.

Table 1

Range of the gear ratio

Gear	Range of gear ratio
1st	9.42~20.54
2nd	9.42~9.94

Optimization of the gear ratio considering the working cycle

Construction of composite working cycle

The main goal of the design of the gearbox is to reduce the energy consumption of the horticultural machinery under different working cycles (*Baohua Zhou, 2010; Xuebing Yin et al., 2022; Li Y et al., 2020; Sun G.B. et al., 2021*).

The gear ratio should be optimized with the goal of minimum energy consumption (*Krüger B. et al., 2022; Peng H. et al.; 2022, Licun F. et al., 2007; Eckert J.J. et al., 2022, Zhu B. et al., 2013; Wang W. et al., 2019*).

In this study the gear ratio should be optimized under different working cycles, therefore, the composite working cycle contains ploughing, rotary tillage and transportation. The composite working cycle should reflect the actual conditions of the horticultural machinery.

Considering the actual operating conditions of horticultural machinery, the composite working cycle (transportation + ploughing + transportation + rotary tillage + transportation) are constructed, which reflect the actual work flow of the horticultural machinery: go to the field, work (repeat process), and go to home. The constructed composite working cycle is illustrated in Figure 5.

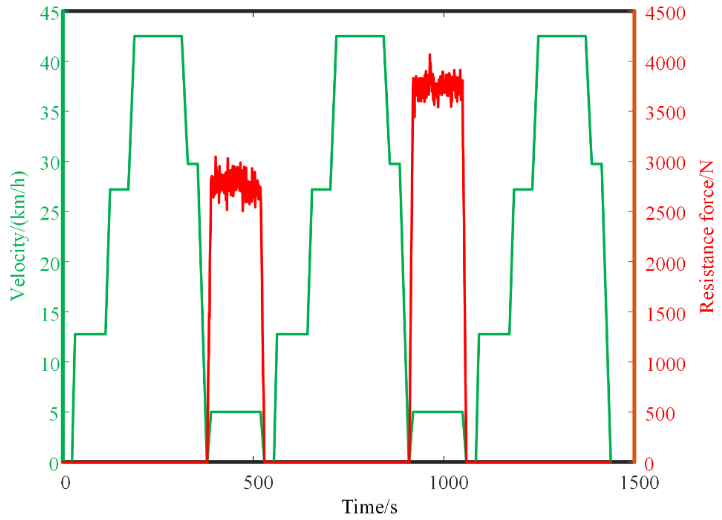

Fig. 5 - The composite working cycles (transportation + ploughing + transportation + rotary tillage + transportation)

RESULTS

Optimization of gear ratio of two-speed transmission based on genetic algorithm

The transmission ratio is optimized by genetic algorithm, the optimization process is shown in Figure 6. The simulation model was built with Matlab/Simulink, the working cycle built above was set in the model and the shift schedule and control strategy were made with Matlab/Stateflow, the energy consumption was chosen as the Fitness function. The maximum number of iterations was set to 50, and the optimization process and results are shown in Figure 7.

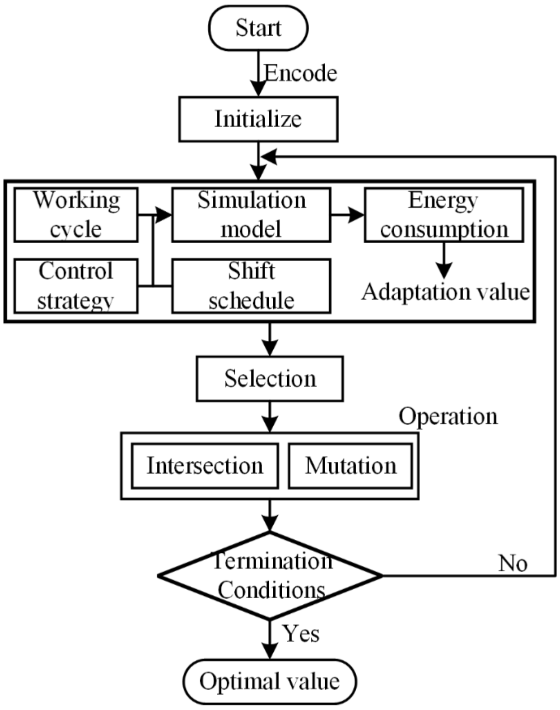

Fig. 6 - Work flow optimization of the gear ratio

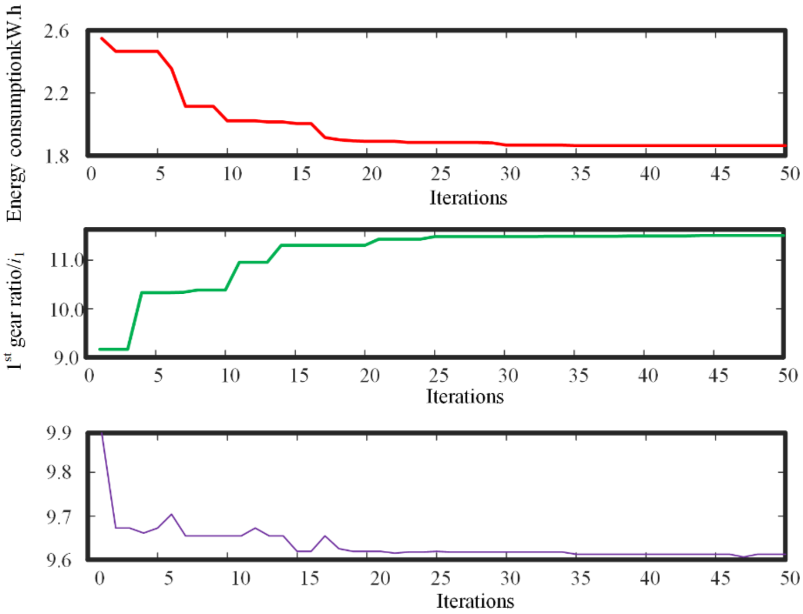

Fig. 7 - Optimization result of the gear ratio

It can be seen from figure 7 that the solution tends to converge when the number of iterations is 25, the first gear ratio finally converges to 11.6, the second gear ratio converges to 9.62, and the optimized gear ratio is shown in the table.

Table 2

Optimization result of the gear ratio

1st	2nd
11.6	9.62

To verify the optimization result, the optimized gear ratio was applied to different working cycle and the comparison was made with single gear ratio. In this study, the single gear ratio was set to be i_{ig} = 9.8. and the comparison of different cycles were made as follows.

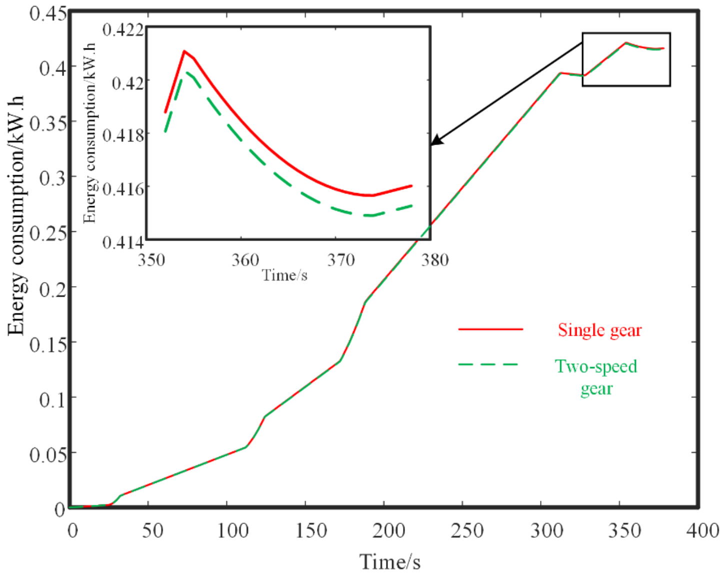

Fig. 8 - Comparison in one transportation test cycle

Figure 8 showed the comparison result under transportation cycle. It can be seen from the figure that the energy consumption of single gear is 4.16×10^{-1} kW.h in one test cycle. The energy consumption of two-speed gear is 4.15×10^{-1} kW.h, the energy consumption can be reduced by 0.25% in one transportation test cycle.

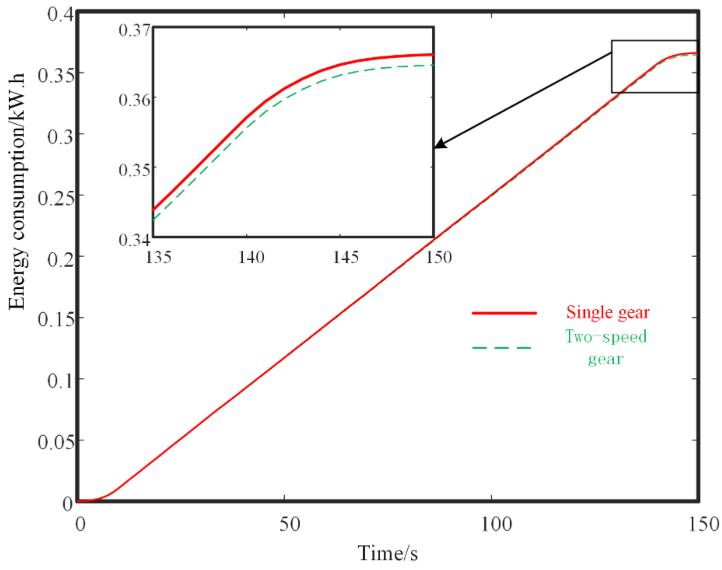

Fig. 9 - Comparison in one ploughing test cycle

Figure 2 showed the comparison result of ploughing test cycle. It can be seen from the figure that the energy consumption of two-speed gear is lower than the single gear. The energy consumption of single gear is 3.661×10^{-1}kW.h in one ploughing test cycle. The energy consumption of two-speed is 3.646×10^{-1}kW.h in one ploughing test cycle. The energy consumption can be reduced by 0.41% in one ploughing test cycle.

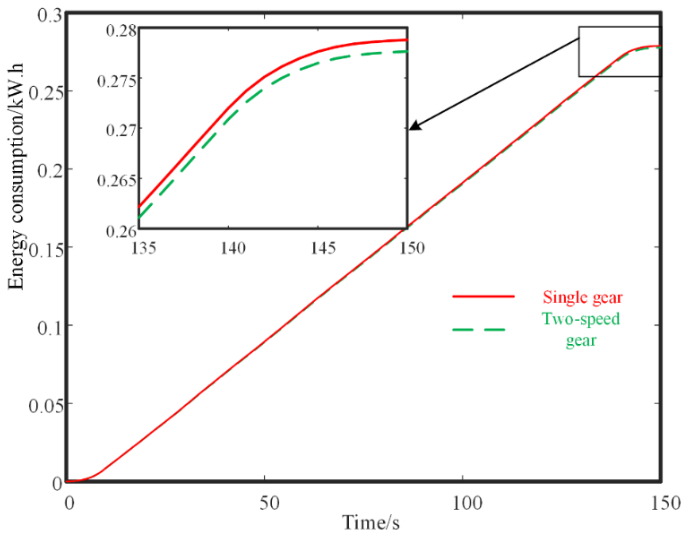

Fig. 10 - Comparison in one rotary tillage test cycle

Figure 10 shows the comparison result of rotary tillage test cycle. It can be seen from the figure that the energy consumption of single gear is 2.788×10^{-1} kW.h in one rotary tillage test cycle. The energy consumption of two-speed gear is 2.776×10^{-1} kW.h in one rotary tillage test cycle. The energy consumption can be reduced by 0.41% in one rotary tillage test cycle.

CONCLUSIONS

In this study, a two-speed transmission system was proposed based on the characteristics of the driving cycle of the horticultural machinery. Firstly, the test cycle of the horticultural machinery was established based on the data collector that fixed on the machinery. Secondly, the two-speed driveline system was designed. To reduce the energy consumption in the horticultural machinery working cycle, the gear ratio of the two-speed gear box was optimized with the goal of minimum energy consumption by genetic algorithm. The optimal gear ratio was obtained, and the comparison was made with the single speed gearbox, the comparison result showed that the energy consumption can be reduced 0.25, 0.41% and 0.41% under the transportation, ploughing and rotary tillage test cycle.

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