Optimized design of the power consumption test of mountain orchard transporters

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Abstract: In order to study the influence rule of various factors on the operating power consumption of the traction orchard transporter and realize the optimal design of the operation power consumption of the transporter, according to the traditional experience and the existing research foundation, the monorail transporter test bench was designed and built on the basis of the whole structure and operation characteristics of the transporter. Taking the motor frequency, track gradient and load as the investigation factors, and the driving shaft power, shaft power transmission and mechanical efficiency as the evaluation indices, the orthogonal test was conducted, and the range analysis of the influence effect was carried out according to the test results. The primary and secondary orders of the influence of various factors were obtained that motor frequency was greater than track gradient and track gradient was greater than load. According to the orthogonal test results, the second-order response surface method was used to establish the optimization model of the power consumption of the transporter, and the model was verified on the test bench. The results showed that the relative error between the model optimization value and the test value based on the response surface power optimization model was less than 10%, which indicated that the power optimization model had satisfactory performance. The research can provide a reference for the orchard conveyor to choose the parameter combination which can save power consumption and the motor that matches power consumption.

Keywords: mountain orchard transporter, power consumption, orthogonal test, optimization model, analysis
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1 Introduction

The orchard transporter is a kind of agricultural transportation machine that is used in mountain orchards, which rides on the track and runs along with the track under the traction of a driving device. Orchard transporter has become one of the main types of machinery is used in orchard production due to its high efficiency, stable operation and flexibility[1,2]. In recent years, a series of transport machines suitable for mountain orchard work has been developed with the strong support of the national agricultural mechanization policy[3,4]. So far, a number of transport machines have been developed, such as 7YGD-35 self-propelled single-track orchard transporter, 7YGS45 self-propelled high-slope double-track orchard transporter, 7YGDQ-50 remote-controlled traction single-track transporter, 7YGWQ-50 remote-controlled traction track-less transporter, and mountain orchard unpowered transporter[5-9], the orange orchard chain-type circulating cargo ropeway and the dismantling single-direction traction double-track transporter[10,11]. Various forms of agricultural transporters are of great significance to alleviate the orchard labor shortage, improve operation efficiency and bring more benefits to orchard growers.

In addition to the design of various types of machines, scholars and practicing engineers have also carried out a lot of research on the key components of the transporter. Regarding the transmission system of battery-driven monorail transporter, Liu et al.[12] designed a transmission system of the two-way transmission chain to improve the transmission efficiency of the transporter. Li et al.[13] concluded the rack of the chain wheel tooth form is more suitable for rail transport by the optimization of rack tooth profile of the self-propelled single track mountain orchard transporter. Ouyang et al.[14] designed and studied the test platform for the damage of the steel wire rope of the transporter, and the test platform can accurately detect the number and the accurate position of the broken wire. However, there is little research on the power consumption and efficiency of the transporter. In contrast, experts and scholars have done a lot of research on the power consumption of ditchers. Wang et al.[15] conducted the parameter comparison

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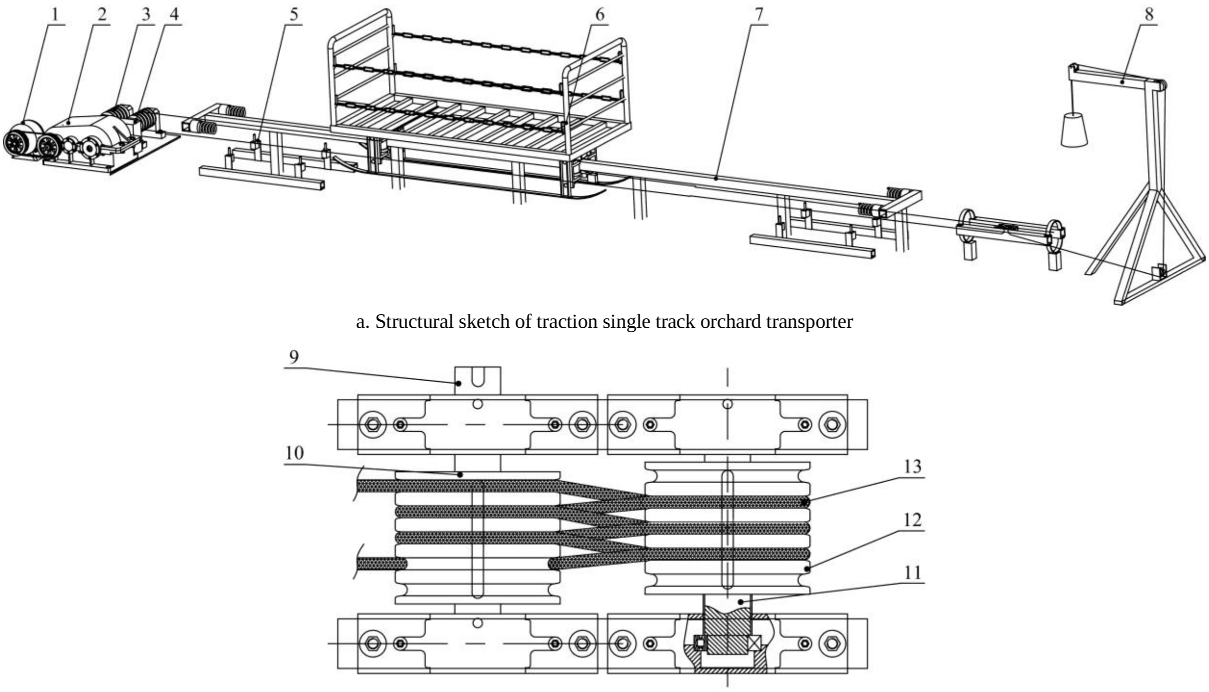
and verification test to analyze the influence of ditching operation parameters on the power consumption of ditching, and the power consumption of ditching after optimization was reduced by 12.80%. Liu et al.[16] analyzed and tested the influencing factors on the power consumption of ditching parts of the ditcher, and verified that the theoretical model of power consumption resulted in higher accuracy. Yao et al.[17] carried out orthogonal test and optimization analysis on the power consumption of the ditcher. The research on the power consumption of ditchers provides ideas and methods for power consumption analysis in this study and offers a feasible scheme for the test principle, design and optimization analysis. To sum up, the above research provides a theoretical reference for power consumption analysis and optimization of the transporters.

In order to study the influences of various factors on the operating power consumption of traction orchard transporters and realize the optimal design of the operation power consumption of the transporter, it is desirable to analyze and determine the critical degree of various factors affecting the operation power consumption of the transporter and the optimal level of each factor according to the traditional experience and existing research[16]. The test bench of the monorail transporter was designed and built. The motor frequency, rail slope and load were taken as the factors to investigate, and the driving shaft power, shaft power transmission efficiency and mechanical efficiency were taken as the evaluation indexes to design and carry out the orthogonal test. A traction orchard transporter was analyzed and studied by the response surface modeling optimization method. To improve the utilization rate of the power consumption of the transporter and determine the importance of various factors that affect the power, and realize the optimal design of the power consumption of the transporter.

2 Structure and working principles of orchard conveyor

The traction single-track orchard transporter is composed of a motor, a transmission system, a braking device, a driving device, a trailer, a track, a weighing device and a travel control device. The driving device is the core part of the whole machine, and the structure is shown in Figure 1[5]. When the conveyor is running, the output power of the motor transmits the power to the driving device through the belt and the deceleration transmission mechanism. The driving device is mainly composed of the driving wheel and the driven wheel, they are winded and connected by the traction rope into a shape of “8”, which is used to prevent the slippage of the traction rope and the wheel pair during the driving process. In addition, this provides the driving force, and the two ends of the traction rope are fixed at both ends of the track. The guide wheels and lower pressure wheels installed at the front and rear ends of the transporter frame are used to import and export the wire rope in the rope groove of the driving wheels, realize the driving of the transport vehicle through the friction between the steel wire rope and the driving wheels.



a. Structural sketch of traction single track orchard transporter

b. Drive system structure sketch

1. Electric machinery 2. Drive system 3. Driver 4. Brake rigging 5. Travel control device 6. Trailer 7. Track 8. Weighing device 9. Drive shaft 10. Drive wheel 11. Driven shaft 12. Driven wheel 13. Traction rope

Figure 1 Schematic diagram of traction single track orchard transporter and drive system

3 Orthogonal test analysis of power consumption

3.1 Test principle and method of power consumption

According to the engineering test theory[17], the relationship between shaft power, torque and rotating speed during the operation of the hoisting system is as follows:

$$P = \frac{T \cdot n}{9550} \tag{1}$$

where, P is the shaft power, W; T is the torque, N·m; N is the rotational speed, r/min.

During the operation of the driving device, the power consumption of the device can be calculated only by measuring the torque and speed of the driving axle on the device. Three groups of data should be obtained from the test: the driving shaft power, the transfer efficiency, and the mechanical efficiency of the shaft power[18]. The ratio of the driven shaft power to driving shaft power is shaft power transfer efficiency, the ratio of wire rope tension power to driving shaft power is mechanical efficiency, driving shaft power is optimization index, and shaft power transfer efficiency and mechanical efficiency are constraints index. The test is carried out on the test bench of a rail transport vehicle. The selected test system is based on a wireless network sensor. The TQ201 wireless network sensor (Beijing Bichuang Technology Co., Ltd, Beijing, China) is an embedded device that integrates a sensor, controller, computing power and wireless communication. The strain gauge is pasted on the axis to be tested, the node is installed on the corresponding rollers after connecting the wireless torque node. The tension sensor is installed between the wire ropes under the transport vehicle, and the node is installed on the transport vehicle after connecting with the wireless tension node[19]. During the operation of the driving device, the torque node collects the data of torque and speed, transmits them to the computer through the wireless gateway to obtain the shaft power, the tension node collects the tension data, transmits them to the computer through the wireless gateway, and obtains the wire rope tension power by combining with the stable speed of the transporter measured at the same time[20].

In this study, the orthogonal design method is used to experiment. After obtaining the test data, the range analysis can help get a more comprehensive comparison between the test factors and results. The mathematical modeling and parameter optimization of operation power consumption are carried out according to the method shown in Figure 2[17].

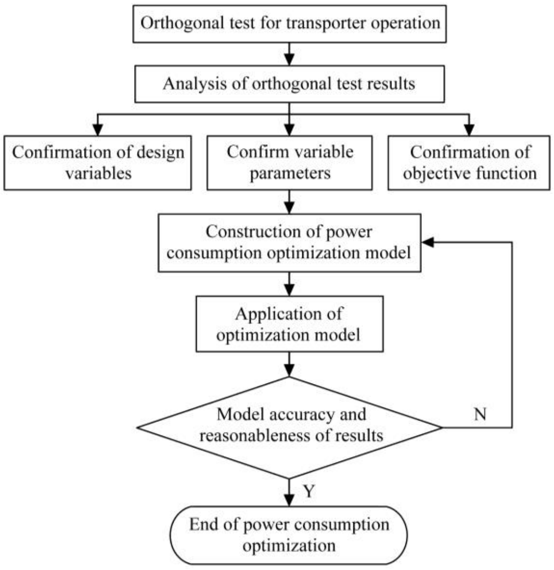
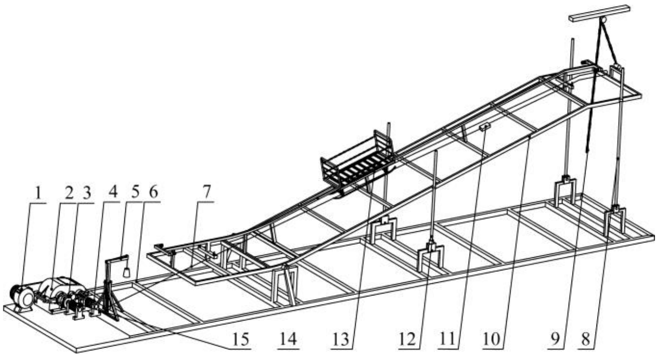


Figure 2 Power consumption test and optimization method flow of transporter

3.2 Test equipment

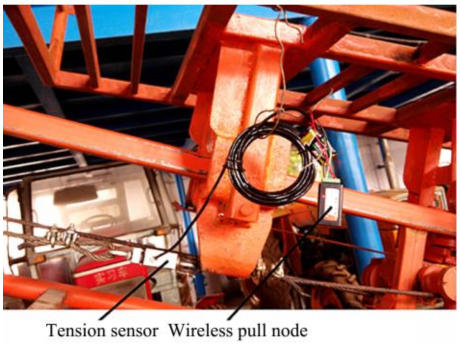
The test bench is built on the basis of a monorail transport vehicle, and the diagram of the test bench is shown in Figure 3. The selected variable frequency motor is a Y2-132S-4 three-phase asynchronous motor with a rated power of 5500 W and a rated speed of 1440 r/min. The diameter of the traction rope is 0.01 m, and the internal structure is 6×19 FC (fiber core) and the reduction ratio of the reducer is 15.75[21]. The lifting platform is hinged with the base, the long and short lifting rods are used to support the lifting platform, and one end of the hand-pulled hoist device is welded with the lifting platform. Pulling the chain drives the lifting platform to rotate around the base to change the climbing angle of the trailer[22]. The tension sensor is connected with the traction rope and is used to measure the tension value and convert the physical signal into a measurable electrical signal. The torque sensor strain gauge is attached to the drive shaft and the transmission shaft to measure the torque. The part specifications of the test system are listed in Table 1, and the installation of the test bench and the tension and the torque sensors of the test system are shown in Figure 4.



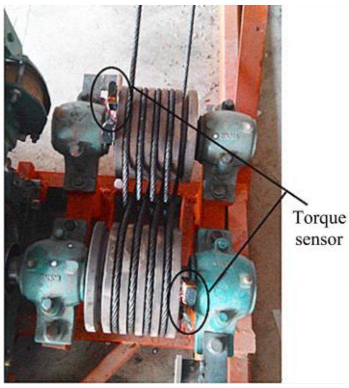
1. Electric machinery 2. Reducer 3. Brake rigging 4. Driver 5. Weighing device 6. Base 7. Traction rope 8. Long lifting rod 9. Hand-pulled hoist device 10. Lifting platform 11. Tension sensor 12. Short lift bar 13. Trailer 14. Track 15. The torque sensor

Figure 3 Test bench sketch of traction single track orchard transporter

a. Practical test bench



b. Installation position of tension sensor



c. Installation position of torque sensor

Figure 4 Installation drawing of test bench and measuring device

Table 1 Test system components of transporter test bench

Parts name		Type	Specification
Torque sensor	Torque strain gauge	BF350-3HE-A(11)N2-P200	Sensitivity coefficient: (2.09±1.00)% Resistance: (350.0±1.0) Ω
	Wireless transmission nodes	TQ201HD	Strain range: ±3000/±1500/±750 $\mu\varepsilon$ Measurement accuracy: 0.1% red±0.02 $\mu\varepsilon$ Resolving power: ±0.1 $\mu\varepsilon$ Speed range: 3-1200 r/min Stability: 0.05%±2 $\mu\varepsilon$ Synchronization accuracy: 1 ms Distance: 200 m
Tension sensor	Tension gauge	H3-C3	Maximum tension: 1.0 t Sensitivity coefficient: (2.0004±0.0200)%
	Wireless transmission nodes	TQ201	Range: ±15 000 $\mu\varepsilon$ Resolving power: ±0.5 $\mu\varepsilon$ Measurement accuracy: 0.1% red±2 $\mu\varepsilon$ Sampling rate:80 Hz Synchronization accuracy: ≤20 ms Distance: 200 m
Wireless data acquisition software		BEEDATA	

3.3 Orthogonal experimental design

A single-factor experimental study on the spindle torque was conducted in the previous research[18]. Through the analysis of variance, influence trend analysis and optimal combination design of the motor frequency, load, slope and pre-tightening force, it is determined that motor frequency, load and slope have significant influence on spindle torque. However, the pre-tightening force has insignificant effect on the torque of the spindle. Thus it is concluded that the influence degree of various factors on the spindle is as slope, load, pre-tightening force, and frequency. Therefore, the frequency conversion frequency, load and slope of the motor are the three main factors that affect the power consumption of the release power transmission device. The orthogonal test of three factors at four levels was designed. The setting of different levels of the load was achieved by accumulating paint buckets. The specifications of the four paint buckets were consistent and each bucket was filled with clods. The quality of each paint bucket was accurately weighed and numbered with a calibrated scale. The weight of each paint bucket was added to the transporter in turn from small to large according to the number, and different levels of load were obtained. The setting of different levels of motor frequency conversion was realized by the frequency conversion control box. The frequency conversion control box controlled one motor, which adopted triangular start, and the control voltage was 380 V AC, in which the frequency conversion controller adopted Siemens MM440, the motor adopted VFG132M-1500-7.5 frequency conversion motor (Realland Electrical Technology Co., Ltd, China). According to the application of frequency conversion motor in transport machinery and hoist, the frequency was generally in the range of 10-50 Hz, and four levels were taken in this range[23]. $L_{16}(4^5)$ orthogonal table was selected, which contained two empty columns that can be used as test error[24], the factor level is listed in Table 2.

Table 2 Factors and levels of orthogonal test

Level	Factor		
	Slope/(°)	Frequency/Hz	Load/kg
1	15	20	47.9
2	25	30	97.1
3	35	40	145.9
4	45	50	195.4

3.4 Analysis of test results

According to the three factors, three sets of data were obtained by orthogonal test, as shown in Table 3, Table 4 and Table 5.

As shown in Table 3, the range of the frequency conversion factor is the largest among the three factors, which indicates that the frequency of the motor has the greatest influence on the power of the driving shaft. The second factor is the slope of the mountain orchard, and the load had the least influence on the power of the drive shaft. The range of the two blank columns in Table 3 is 102 and 147, which are much smaller than the range 358 of the load column, indicating that no interaction was ignored among the three factors of the test, and other factors that have an important impact on the test results were not omitted.

Table 3 Orthogonal test of drive shaft power

Test number	Experimental factor					Driving shaft power/W
	Slope/(°)	Frequency/Hz	Load/kg	Blank	Blank	
1	1(15)	1(20)	1(47.9)	1	1	780
2	1	2(30)	2(97.1)	2	2	1363
3	1	3(40)	3(145.9)	3	3	1932
4	1	4(50)	4(195.4)	4	4	2596
5	2(25)	1	2	3	4	981
6	2	2	1	4	3	1281
7	2	3	4	1	2	2336
8	2	4	3	2	1	2739
9	3(35)	1	3	4	2	1191
10	3	2	4	3	1	1913
11	3	3	1	2	4	1920
12	3	4	2	1	3	2691
13	4(45)	1	4	2	3	1416
14	4	2	3	1	4	2039
15	4	3	2	4	1	2474
16	4	4	1	3	2	2847
k_1	1668	1092	1707	1962	1977	
k_2	1834	1649	1877	1860	1934	
k_3	1929	2166	1975	1918	1830	
k_4	2194	2718	2065	1886	1884	
R	526	1626	358	102	147	

Note: ( ): the internal value represents the horizontal value of each experimental factor; Blank columns are test error; k_1, k_2, k_3, and k_4 are the mean values of the sum of driving shaft powers at each level; R indicates range, the same as below.

Let k_1, k_2, k_3, and k_4 represent the mean values of the sum of driving shaft powers at each level. To determine the better level combination of each factor in this test, k_1, k_2, k_3, and k_4 were compared. In the slope factor column: $k_4>k_3>k_2>k_1$; in the frequency conversion factor column: $k_4>k_3>k_2>k_1$; in the load factor column: $k_4>k_3>k_2>k_1$. According to the results of the orthogonal test, the optimal scheme is that the slope is 15°, the load is 47.9 kg, the motor frequency is 20 Hz, and the power of the drive shaft is the minimum.

According to Table 4, the range of the load factor is 5.03, which is the largest among the three factors. This indicates that the load has the greatest impact on the shaft power transfer efficiency. The secondary factor is the slope, and the frequency conversion has the least impact on the shaft power transfer efficiency.

Table 4 Orthogonal test of shaft power transfer efficiency

Test number	Experimental factors					Shaft power transfer efficiency/%
	Slope/(°)	Frequency/Hz	Load/kg	Blank	Blank	
1	1(15)	1(20)	1(47.9)	1	1	90.24
2	1	2(30)	2(97.1)	2	2	94.12
3	1	3(40)	3(145.9)	3	3	95.97
4	1	4(50)	4(195.4)	4	4	97.06
5	2(25)	1	2	3	4	95.34
6	2	2	1	4	3	92.78
7	2	3	4	1	2	98.77
8	2	4	3	2	1	95.28
9	3(35)	1	3	4	2	96.52
10	3	2	4	3	1	97.99
11	3	3	1	2	4	93.63
12	3	4	2	1	3	95.28
13	4(45)	1	4	2	3	97.32
14	4	2	3	1	4	97.45
15	4	3	2	4	1	97.33
16	4	4	1	3	2	94.38
R	2.27	1.57	5.03	0.74	0.84	

It can be seen from Table 5 that the range of slope factor is 19.37, which is the largest among the three factors. Therefore, the factor that has the greatest influence on mechanical efficiency is the slope, the second is load, followed by the influence of frequency conversion on mechanical efficiency.

Table 5 Orthogonal test of mechanical efficiency

Test number	Experimental factors					Mechanical efficiency/%
	Slope/(°)	Frequency/Hz	Load/kg	Blank	Blank	
1	1(15)	1(20)	1(47.9)	1	1	45.93
2	1	2(30)	2(97.1)	2	2	47.21
3	1	3(40)	3(145.9)	3	3	50.76
4	1	4(50)	4(195.4)	4	4	56.13
5	2(25)	1	2	3	4	57.01
6	2	2	1	4	3	58.73
7	2	3	4	1	2	70.58
8	2	4	3	2	1	57.51
9	3(35)	1	3	4	2	66.67
10	3	2	4	3	1	79.44
11	3	3	1	2	4	64.46
12	3	4	2	1	3	63.82
13	4(45)	1	4	2	3	80.39
14	4	2	3	1	4	69.65
15	4	3	2	4	1	67.59
16	4	4	1	3	2	59.89
R	19.37	4.42	14.38	0.72	2.34	

In this study, the drive shaft power, the transmission efficiency and the mechanical efficiency of the shaft power are taken as the test indexes, the drive shaft power is the main objective, and the transmission efficiency and mechanical efficiency of the shaft power are the secondary objectives. It is only necessary to draw the trend chart of factors and results according to the data obtained from the drive shaft power[25], and combined with range R to analyze the importance of various factors on the drive shaft power. The greater the range is, the more important the factor is[26]. The drive shaft power trend is shown in Figure 5. From the range R and trend chart, it can be seen that the frequency conversion has the greatest impact on the power of the drive shaft and the load has a minimal impact on the power of the drive shaft.

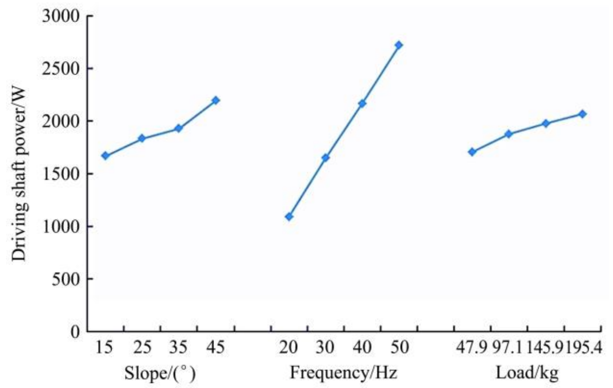


Figure 5 Drive shaft power trend

4 Optimal modeling of power consumption

4.1 Response surface modeling

Response surface modeling is a non-linear approximate modeling method, the response surface model was established based on the data obtained from orthogonal test or uniform test, and then the values of other test points were predicted according to the model[27,28]. The response surface model had various orders, and the second-order response surface model was easy to solve and has high accuracy, which was widely used in the optimization model[29]. Its general expression is as follows:

$$Y = F(X) = a_0 + \sum_{i=1}^{n} a_i x_i + \sum_{i=1}^{n} a_{ii} x_i^2 + \sum_{i<j}^{n} a_{ij} x_i x_j \tag{2}$$

where, the coefficients a_0, a_i, a_{ii} and a_{ij} in the above equation are coefficients to be calculated, the independent variable x is the data obtained from the test; i is the design point for the orthogonal test (i=1, 2, 3, …); j is the original model explanatory variable (j=1, 2, 3, …). The test data $\boldsymbol{X}$ is changed into the form of a matrix as follows:

$$X = \begin{bmatrix} 1 & x_{11} & x_{12} & \cdots & x_{1j} \\ 1 & x_{21} & x_{22} & \cdots & x_{2j} \\ \cdots & \cdots & \cdots & \cdots & \cdots \\ 1 & x_{i1} & x_{i2} & \cdots & x_{ij} \end{bmatrix} \tag{3}$$

Then the matrix equation for solving unknown coefficient matrix $\boldsymbol{A}$ is as follows:

$$A = (X^T X)^{-1} X^T Y \tag{4}$$

The coefficient matrix A is obtained and the response surface optimization model can be obtained by substituting the coefficient into Equation (2).

4.2 Model establishment

Among the three factors of slope, load and frequency conversion, the slope is related to the topography of each area, which is highly random and uncontrollable. The slope can be determined as the model variable parameter t, and the load and frequency as variables x_1 and x_2 respectively to establish the second-order model. P is the driving shaft power, then, the second-order response surface model equation is as follows:

$$P_{\min} = F(x_1, x_2, t) = a_0 + a_1x_1 + a_2x_2 + a_3t + a_{11}x_1^2 + a_{22}x_2^2 + a_{33}t^2 + a_{12}x_1x_2 + a_{23}x_2t + a_{31}tx_1 \quad (5)$$

The data obtained from the orthogonal test are imported into the Matlab software, and the data are processed according to Equations (3) and (4), and the power consumption optimization model is established with the driving shaft power P as the objective equation:

$$\begin{aligned}F(x_1, x_2, t) = {} & 262.1701 + 2.1079x_1 + 19.6443x_2 - 22.1653t \\ & - 0.0084x_1^2 - 0.0096x_2^2 + 0.2457t^2 + 0.0988x_1x_2 \\ & + 0.7801x_2t + 0.01x_1t\end{aligned} \quad (6)$$

The constraint model was established with the transfer efficiency and mechanical efficiency of the shaft power as conditional equations:

$$\begin{cases}M(x_1, x_2, t) = 80.8171 + 0.1028x_1 + 0.1773x_2 + 0.2457t \\ \quad - 0.0001x_1^2 - 0.0041x_2^2 - 0.0011t^2 + 0.0002x_1x_2 \\ \quad + 0.0019x_2t - 0.0014x_1t \\ N(x_1, x_2, t) = 13.745 - 0.1777x_1 + 0.9343x_2 + 2.01t \\ \quad + 0.0009x_1^2 - 0.0132x_2^2 - 0.0254t^2 - 0.0001x_1x_2 \\ \quad - 0.0009x_2t + 0.0017x_1t\end{cases} \quad (7)$$

According to the variable range set in the test and the performance parameters of the monorail transporter in Reference [18], the constraints are set as $50 \leq x_1 \leq 200$, $20 \leq x_2 \leq 50$.

4.3 Model validation and application

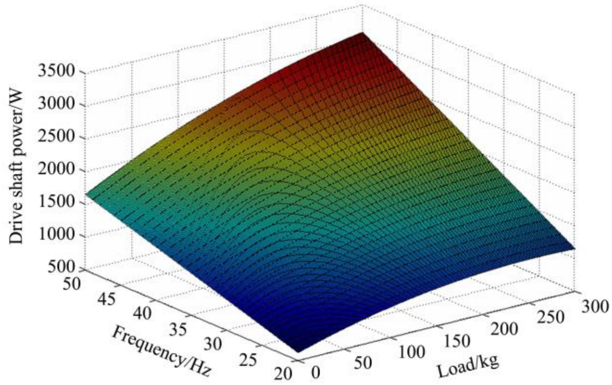
Three different gradient levels are set on the test bench, which are t_1=20°, t_2=30° and t_3=40°, which are substituted into Equations (6) and (7) respectively, and the response surface optimization model and constraint model under three different slope levels were obtained as follows:

$$\begin{cases}F_{t_1}(x_1, x_2) = -82.8559 + 2.3079x_1 + 35.2463x_2 - 0.0084x_1^2 \\ \quad - 0.0096x_2^2 + 0.0988x_1x_2 \\ M_{t_1}(x_1, x_2) = 85.2911 + 0.0748x_1 + 0.2153x_2 - 0.0001x_1^2 \\ \quad - 0.0041x_2^2 + 0.0002x_1x_2 \\ N_{t_1}(x_1, x_2) = 43.785 - 0.1437x_1 + 0.9163x_2 + 0.0009x_1^2 \\ \quad - 0.0132x_2^2 - 0.0001x_1x_2\end{cases} \quad (8)$$

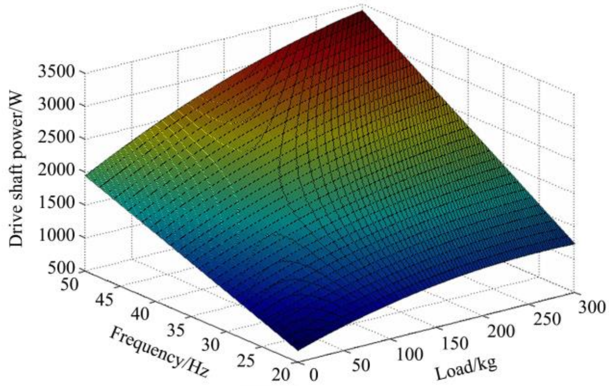
$$\begin{cases}F_{t_2}(x_1, x_2) = -181.6589 + 2.4079x_1 + 43.0473x_2 - 0.0084x_1^2 \\ \quad - 0.0096x_2^2 + 0.0988x_1x_2 \\ M_{t_2}(x_1, x_2) = 87.1981 + 0.0608x_1 + 0.2343x_2 - 0.0001x_1^2 \\ \quad - 0.0041x_2^2 + 0.0002x_1x_2 \\ N_{t_2}(x_1, x_2) = 51.185 - 0.1267x_1 + 0.9073x_2 + 0.0009x_1^2 \\ \quad - 0.0132x_2^2 - 0.0001x_1x_2\end{cases} \quad (9)$$

$$\begin{cases}F_{t_3}(x_1, x_2) = -231.3219 + 2.5079x_1 + 50.8483x_2 - 0.0084x_1^2 \\ \quad - 0.0096x_2^2 + 0.0988x_1x_2 \\ M_{t_3}(x_1, x_2) = 88.8851 + 0.0468x_1 + 0.2533x_2 - 0.0001x_1^2 \\ \quad - 0.0041x_2^2 + 0.0002x_1x_2 \\ N_{t_3}(x_1, x_2) = 53.505 - 0.1097x_1 + 0.8983x_2 + 0.0009x_1^2 \\ \quad - 0.0132x_2^2 - 0.0001x_1x_2\end{cases} \quad (10)$$

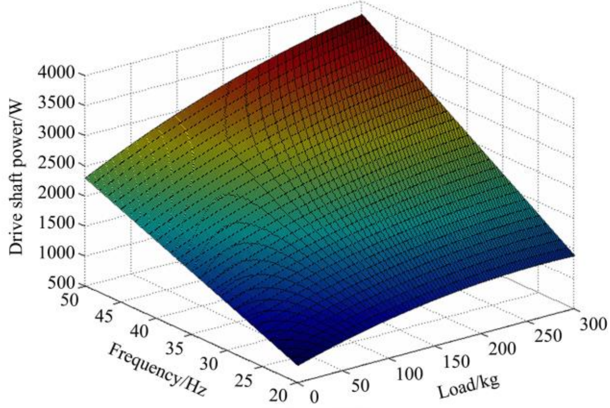
The constraint object is that the shaft power transfer efficiency is greater than 98% and the mechanical efficiency is greater than 50%. The corresponding response surface models can be obtained from Equations (8), (9), and (10), as shown in Figure 6.



a. Equation (8)



b. Equation (9)



c. Equation (10)

Figure 6 Response surface of models

As shown in Figure 6, when the slope is 20°, 30° and 40° respectively, the power of the driving shaft basically conforms to the law that increases with the increase of frequency conversion and load. The optimal area for minimizing the power of the drive shaft is about the lower limit of the frequency conversion factor. From the response surface diagram, it can be concluded that the influence of frequency conversion on the power of the drive shaft is greater than that of the load on the power of the drive shaft, which is consistent with the conclusion of the orthogonal test. Therefore, in the actual work process, under the premise of reducing power consumption, in order to ensure transport efficiency, reducing motor frequency is the optimal choice.

Then, input constraints ($50 \leq x_1 \leq 200$, $20 \leq x_2 \leq 50$) and Equations (8), (9), and (10) into Matlab software, the optimal solution is obtained by genetic algorithm[29]. Using the basic genetic algorithm to set the running parameters, the coding length is 10, the coding accuracy is 0.0029, the population size is 50, the crossover probability is 0.7, the mutation probability is 0.07, and the number of terminating evolutionary iterations is 500. The optimal power consumption, frequency and load variables of the response model under three slopes are obtained. In order to verify and analyze the established optimization model, the load and frequency conversion under the three slopes are tested respectively. To ensure the accuracy of the test, the actual slope test is completed, as shown in Figure 7.

In the field test of the lifting system of the orchard conveyor, the slope of the track is changed by changing the height difference of the track, so as to achieve the effect of simulating different slopes. Compare the optimum power consumption obtained by the model with the results measured by the test, as shown in Table 6.

Figure 7 Field testing of hoisting system for hilly orchard transporter

Table 6 Result of the optimization model verification

Slope/(°)	Variable optimization value		Driving shaft power/W		Relative error/%	Power transfer efficiency/%	Mechanical efficiency/%
	Load/kg	Frequency/Hz	Optimum value	Test value		Test value	Test value
20	192	20.12	1131	1210	6.53	98.14	62.25
30	147	24.90	1418	1509	6.03	98.42	67.31
40	133	28.80	1788	1938	7.73	97.88	64.11

Note: With the slope in Table 6 as the parameter, the constraints ($50\le x_1\le 200$, $20\le x_2\le 50$) and Equations (8), (9) and (10) are input into Matlab software to obtain the optimum value using the genetic algorithm.

As can be seen from Table 6, the optimum values of the driving shaft power are relatively smaller than the test values of the driving shaft power, this is because the optimized frequency conversion value appeared with two decimal digits, and the minimum resolution of the frequency converter used in the test was 1 Hz. The frequency conversion value used in the test is obtained by taking the optimized frequency conversion value which can be rounded up so that the actual test value is higher than the theoretical optimization power. In addition, there is a certain deviation in the actual track slope and actual load of the test, as well as a certain turning radius of the track and the loss of power consumption caused by the inevitable friction and wear in the operation process. But the error between the theoretical optimization value and the test value is less than 10%, and the shaft power transfer efficiency is more than 97%, and the mechanical efficiency is more than 50%. It indicates that the response surface optimization model has good engineering practicality and has theoretical guiding significance for obtaining the combination of parameters with the lowest power consumption in the operation of transporter[30].

5 Conclusions

1) A set of test platforms for power consumption analysis of mountain orchard transporter was built, which effectively realized the real-time monitoring of parameters such as trailer climbing angle, trailer load, traction rope tension, driving wheel torque and rotational speed. The test system based on a wireless network sensor was easy to operate, eliminated the noise interference caused by long cable transmission, and improved the measurement accuracy and anti-interference ability.

2) The primary and secondary relationship between the influencing factors through the orthogonal test can be obtained. The motor frequency conversion had the greatest influence on power consumption, followed by the slope and load. In the operation of the lifting system during the start-up stage of the uphill of the orchard conveyor, it was more practical to reduce the frequency conversion properly than to reduce the weight of the cargo.

3) The power response surface optimization model, the power transfer efficiency constraint model and the mechanical efficiency constraint model of the drive shaft established with the orthogonal test data showed that the relative error of the model was within 10%, which had high engineering practicability. It can provide a theoretical basis for the selection of power-saving parameter combinations and power-matching motors for orchard conveyor, the operation power consumption can also be predicted according to the working conditions of the transporter, which provided a reference for operators to select motors with matching power, which was conducive to the environmental effect of energy saving and emission reduction in agricultural mechanization production.

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