Orthogonal Test Optimization and Finite Element Analysis of Wheel Reducer

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Abstract: The optimal design method of the orthogonal test is used to optimize the wheel reducer. Orthogonal tests are carried out with the number of teeth, modulus, and tooth thickness factor as independent variables and volume as the dependent variable, and the strength requirements of each test are verified.The range analysis of the orthogonal test results was carried out, a set of optimal solutions for parameter optimization of the wheel reducer were obtained, and the modeling and finite element analysis of the optimal solutions were carried out to verify the results of the orthogonal test.

Keywords: *Wheel reducer,Orthogonal test,Finite Element Analysis*

I. DESIGNANALYSISOFWHEELREDUCER

The heavy-duty mobile belt conveyor generally includes a chassis part and a conveyor part installed on the chassis. The chassis part provides basic support and overall motion functions for the conveyor. The power transmission of the wheel side is the main working part of the chassis[1]. The power transmission of the wheel side is: the output of the hydraulic motor is connected to the wheel through the reducer so that the chassis can walk. The wheel reducer adopts the form of two-stage planetary gear transmission, as shown in Figure 1.The number of teeth of the sun gear, planetary gear, and ring gear of the reducer is z_1, z_2, and z_3 respectively. Although the number of planetary gears np has a great influence on the bearing of the planetary reducer, the value of np is generally limited by the level of production technology[2].

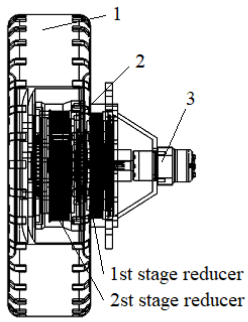


Fig.1 Working Principle Diagram of Wheel Side Reducer

1. Wheel 2. Wheel reducer 3. Hydraulic motor

Due to the limited space structure inside the wheel and the need to install the entire drive device inside the wheel, space is particularly important[3]. If the volume of the reducer is reduced, the material for manufacturing the wheel reducer will be reduced accordingly, and the cost will also be reduced, which not only achieves the purpose of optimizing the reducer but also saves more space for the installation and arrangement of other components[4].

Since the number of teeth satisfies the gear matching conditions, the minimum volume of the sun gear and planetary gear is the performance index for the design[5], and the following formula can calculate each stage of the reducer:

$$f(X)=V_1+3V_2=\pi d^2{}_{m1}b/4+3\pi d^2{}_{m2}b/4$$
$$=\pi\frac{m^2(z_1-0.25)^2}{4}b+3\pi\frac{m^2(z_2-0.25)^2}{4}b$$

In the formula, d_{m1}, d_{m2}——the average diameter of the sun gear and planetary gear;

b——tooth width, $b=mz_1\phi_d$.

II. ORTHOGONAL EXPERIMENT OPTIMIZATION DESIGN

It can be known from f(X) that the parameters that determine the volume of the reducer are the number of teeth z, the modulus m, and tooth thickness factorϕ_d.The chassis weighs 4500kg, the part of the chassis is full of materials and weighs 9500kg. The rolling radius of the wheels is 0.4m. The sun gear and planetary gear are made of 40CrMnMo[6], which has been carburized and surface quenched.According to the existing conventional design experience, the value range of each of the above parameters is selected.

A. Optimal Design of Orthogonal Experiment of First-stage Wheel Reducer

In the first stage of deceleration, since $i_1\geq4$, the sun gear z_1 is a pinion gear, taking z_1 as a factor, and by design experience, then: $20\leq z_1\leq 22$, $3\leq m\leq 5$, $0.2\leq\phi_d\leq 0.3$. The above three parameters have a direct impact on the volume. Since the number of teeth and the modulus are integers, each factor takes three levels[7], as shown in Table 1.

Tab. 1 Optimization design factor level table of orthogonal test

Factor	1	2	3
Number of teeth of sun gear (A)	20(A1)	21(A2)	22(A3)
Gear modules (B)	3(B1)	4(B2)	5(B3)
Tooth thickness factor	0.2(C1)	0.25(C2)	0.3(C3)

It can be seen from Table 1 that the factors A, B, and C are all 3 levels, L9 (3^4) is selected, and the orthogonal test scheme is listed in Table 2.

Tab. 2 Orthogonal test scheme combination table

Test No.	Combination level	Number of Teeth	Modules /mm	Tooth Thickness Factor
1	A1B1C1	20	3	0.2
2	A1B2C2	20	4	0.25
3	A1B3C3	20	5	0.3
4	A2B1C2	21	3	0.25
5	A2B2C3	21	4	0.3
6	A2B3C1	21	5	0.2

7	A3B1C3	22	3	0.3
8	A3B2C1	22	4	0.2
9	A3B3C2	22	5	0.25

From the known conditions of the first-stage wheel reducer, the formula is simplified as follows:

$$f(X)=3.142z_1^3m^3\varphi_d-1.571z_1^2m^3\varphi_d+0.196z_1m^3\varphi_d$$

From the combined level data in the above table, bring in the f(X) calculation results, and then verify the bending fatigue strength (strength 1) and contact fatigue strength (strength 2) of the sun gear, the results are shown in Table 3.

Tab. 3 Index results of each test scheme

Test No.	Combination level	Volume	Strength 1	Strength 2
1	A1B1C1	1.323×10^5	NO	Satisfy
2	A1B2C2	3.921×10^5	NO	Satisfy
3	A1B3C3	9.191×10^5	Satisfy	Satisfy
4	A2B1C2	1.918×10^5	NO	Satisfy
5	A2B2C3	5.455×10^5	Satisfy	Satisfy
6	A2B3C1	7.103×10^5	Satisfy	Satisfy
7	A3B1C3	2.649×10^5	NO	Satisfy
8	A3B2C1	4.186×10^5	Satisfy	Satisfy
9	A3B3C2	1.022×10^6	Satisfy	Satisfy

B. Analysis of range analysis method

From the data in Table 3, calculate the average of the test values of factors A, B, and C at levels 1, 2, and 3, respectively[8]. which is:

$$K_1^A=4.812\times10^5,K_2^A=4.825\times10^5,K_3^A=5.685\times10^5$$

$$K_1^B=1.963\times10^5,K_2^B=4.518\times10^5,K_3^B=8.838\times10^5$$

$$K_1^C=4.204\times10^5,K_2^C=5.353\times10^5,K_3^C=5.765\times10^5$$

Plot the mean of the test values as the ordinate and the factor levels as the abscissa.As shown in Figure 2.

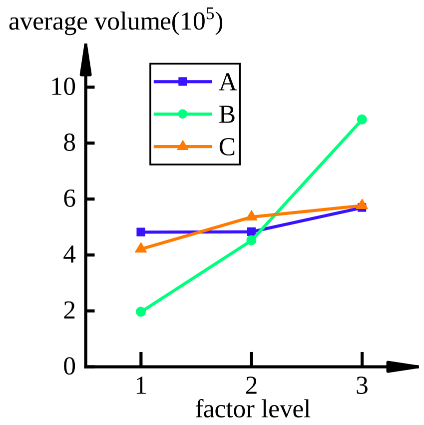


Fig. 2 Plot of Mean Volume Versus Factor Level

From Figure 2, it can be seen that the combination that minimizes the average volume f(X) is A1B1C1, and this combination does not satisfy the detection result intensity 1. It can be seen from the analysis results that the combination A3B2C1 should be selected to minimize the average volume and meet the strength requirements, and the average volume of this combination is 4.186×105. That is, a set of optimal solutions for the first-stage wheel deceleration is as follows:

$$z_1=22,m=4,\phi_d=0.2。$$

C. Orthogonal experimental optimization design of the second-stage wheel reducer

In the second-stage deceleration, because $i_2<4$, the planetary gear z_2 is a pinion, and with z_2 as the factor, according to the design experience, then: $17\le z_2\le19$, $4\le m\le6$, $0.4\le\phi_d\le0.6$. The above three parameters have a direct impact on the volume. Since the number of teeth and the modulus are integers, each factor takes three levels, as shown in Table 4.

Tab. 4 Optimization design factor level table of orthogonal test

Factor	1	2	3
Number of teeth of planet gears (D)	17(D1)	18(D2)	19(D3)
Gear modules(E)	4(E1)	5(E2)	6(E3)
Tooth thickness factor(F)	0.4(F1)	0.5(F2)	0.6(F3)

It can be seen from Table 4 that the factors D, E, and F all have 3 levels, L9 (3^4) is selected, and the orthogonal test scheme is listed in Table 5.

Tab. 5 Orthogonal test scheme combination table

Test No.	Combination level	Number of Teeth	Modules /mm	Tooth Thickness Factor
1	D1E1F1	17	4	0.4
2	D1E2F2	17	5	0.5
3	D1E3F3	17	6	0.6
4	D2E1F2	18	4	0.5
5	D2E2F3	18	5	0.6
6	D2E3F1	18	6	0.4
7	D3E1F3	19	4	0.6
8	D3E2F1	19	5	0.4
9	D3E3F2	19	6	0.5

From the known conditions of the second-stage wheel reducer, the formula is simplified as follows:

$$f(X)=2.909z_2^3m^3\varphi_d-2.269z_2^2m^3\varphi_d+0.262z_2m^3\varphi_d$$

From the combined level data in the above table, bring in the f(X) calculation results, and then verify the bending fatigue strength (strength 1) and contact fatigue strength (strength 2) of the planetary gear. The results are shown in Table 6:

Tab. 6 Index results of each experimental scheme

Test No.	Combination level	Volume	Strength 1	Strength 2
1	D1E1F1	3.492×10^5	NO	Satisfy
2	D1E2F2	8.525×10^5	Satisfy	Satisfy
3	D1E3F3	1.768×10^6	Satisfy	Satisfy
4	D2E1F2	5.195×10^5	NO	Satisfy
5	D2E2F3	1.218×10^6	Satisfy	Satisfy
6	D2E3F1	1.403×10^6	Satisfy	Satisfy
7	D3E1F3	7.349×10^5	NO	Satisfy
8	D3E2F1	9.569×10^5	Satisfy	Satisfy
9	D3E3F2	2.067×10^6	Satisfy	Satisfy

D. 2.4 Analysis of range analysis method

From the data in Table 6, calculate the average of the test values of factors D, E, and F at levels 1, 2, and 3, respectively. which is:

$$K_1^D = 9.899\times10^5, K_2^D = 1.047\times10^6, K_3^D = 1.253\times10^6$$

$$K_1^E = 5.345\times10^5,\ K_2^E = 1.009\times10^6,\ K_3^E = 1.746\times10^6$$

$$K_1^F = 9.030\times10^5,\ K_2^F = 1.146\times10^6,\ K_3^E = 1.240\times10^6$$

Plot the mean of the test values as the ordinate and the factor levels as the abscissa.As shown in Figure 3.

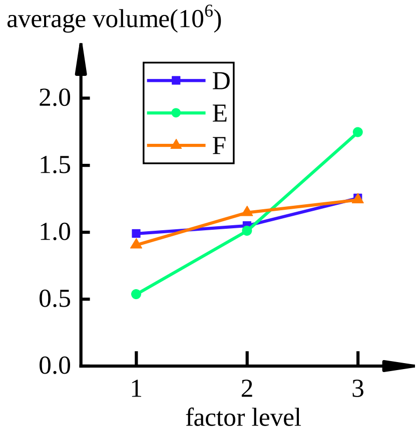


Fig. 3 Plot of Mean Volume Versus Factor Level

From Figure 3, it can be seen that the combination that minimizes the average volume f(X) is D1E1F1, and this combination does not satisfy the detection result intensity 1. It can be seen from the analysis results that the combination D1E2F1 should be selected to minimize the average volume and meet the strength requirements, and the average volume of this combination is 6.8203×105. That is, a set of optimal solutions for the second-stage wheel deceleration is

$$z_2=17, m=5, \phi_d=0.4。$$

III. OPTIMIZE THE RESULTS

Table 7 compares before and after optimization.

Tab. 7 Comparison of optimization results

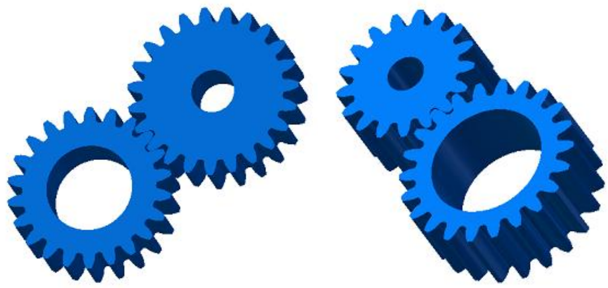
	Total volume(V)	Volume decrease rate (%)
Before	1.4856×10^6	0
After	1.1006×10^6	25.9%

It can be seen from the table that the total volume of the wheel reducer is reduced by 25.9% compared with the original, indicating that the optimization effect is obvious.

IV. OPTIMIZATION RESULT VERIFICATION

A. 3D model establishment

According to the structure of the wheel side reducer, the three-dimensional modeling software Solidworks is used to establish the sun gear and planetary gear models[9], and they are assembled according to the standard center distance to obtain their three-dimensional geometric model, as shown in Figure 4.



(1) High-speed stage (2) Low-speed stage

Fig. 4 3D Model of The Planetary Gear Assembly

B. Material settings

Import the created 3D model into ANSYS, the material is 40CrMnMo, the density is 7.87×10^3kg/m^3, the elastic modulus is 2.07×10^{11}Pa, and the Poisson's ratio is 0.254.

C. Mesh processing

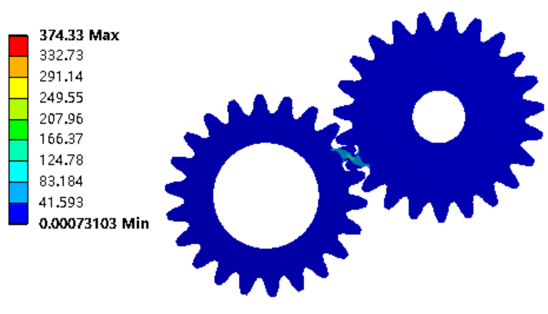
The meshing method is Tetrahedrons[10]. The overall mesh is divided into 4mm, and the gear contact part is refined by 0.5mm. The division results are 123,346 units and 183,258 nodes in the low-speed class, and 103,941 units and 158,064 nodes in the high-speed class.

D. Loads and Boundaries

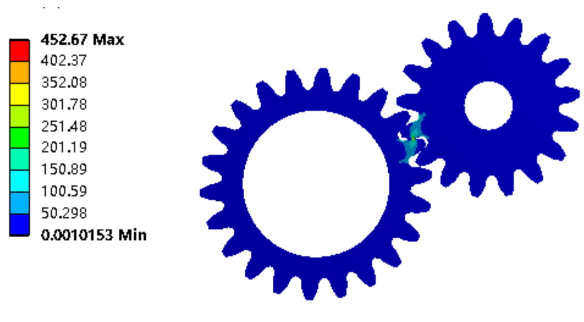
The high-speed and low-speed sun gears are set to rotate around the z-axis, and the other two directions are regarded as rigid bodies, and the displacement can be ignored[11]. The inner surfaces are loaded with torques of 220N·m and 900N·m respectively, and the planetary gear is constrained by full displacement[12].

E. Analysis of finite element results.

It can be seen from Fig. 5 that the overall contact stress cloud diagram of the sun gear and the planet gear shows that the maximum stress value of the high-speed stage is 374.33MPa, and the maximum stress value of the low-speed stage is 452.67MPa.



(1) High-speed stage



(2) Low-speed stage

Fig.5 Contact Stress Cloud Diagram of The Sun Gear and Planet Gear

Both of which appear at the contact position between the sun gear and the planet gear, and the area is small. As shown in Figure 6. Both are less than the allowable stress of the gear material, and the structural dimensions and parameters of the gear after the optimization of the orthogonal test meet the strength requirements.

(1) High-speed stage (2) Low-speed stage

Fig.6 Local Enlarged View of Contact Stress

CONCLUSION

(1) The optimal solution obtained by the orthogonal test optimization design is similar to the results obtained by the general optimization design, which can achieve the purpose of optimizing the planetary gear reducer.

(2) In the ordinary optimization design, the optimization algorithm program needs to be written, the complexity is high, the obtained result is not necessarily an integer, and it often needs to be rounded manually. However, when selecting each factor level in the orthogonal test optimization design, they are artificially selected as integers according to the conventional value range, no rounding processing is required, there is no need to write an optimization algorithm program, and the complexity is low.

(3) The simulation analysis of the sun gear and the planetary gear pair shows that the maximum contact stress of the high-speed and low-speed gears is less than the allowable stress of the material, indicating that the orthogonal experiment optimization is reasonable.

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