Design and optimization of main reducer based on Baja racing

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Abstract. The main reducer is the main component of the whole vehicle, and its main function is to realize deceleration and torque increase. For Baha racing car, in order to improve the dynamic performance of the whole car, the main reducer is designed from the aspects of layout, transmission ratio distribution, shift mode, overall size and shell structure. Calculate the transmission ratio range of reducer according to the performance parameters of transmission parts, and verify the rationality of transmission ratio; Then determine the parameters of gear according to the transmission ratio and related parameters, and finally design the parameters of gear according to the transmission ratio and related parameters, and finally design the parameters of other parts of reducer. Based on the determined parameters, 3D modeling software UG is used to build 3D models of various parts of the reducer, and finite element analysis software ANSYS is used to simulate and analyze the parts to check whether the comprehensive mechanical properties meet the requirements. In this paper, the design of the main reducer realizes the comprehensive design of small size, light weight, reasonable transmission ratio distribution, high reliability, shifting gears during driving, and the comprehensive mechanical properties also meet the requirements.

Keywords: ANSYS analysis of optimization design of BSC racing main reducer.

1 Introduction

Baja SAE China, a competition for the design, manufacture and testing of off-road vehicles, is a brand-new process of technical education and engineering practice, which is organized by students of automobile and related majors in universities and vocational colleges[1]. Reducer is an important part of Baha off-road racing [2]. In this paper, through data comparison, ANSYS software for transient dynamic analysis of the designed final drive, etc **[2]**, the final drive is optimized from layout, transmission ratio distribution, shift mode, overall size, shell structure and other aspects, so as to design the final drive with better performance.

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2 Analysis of the present situation of Baja's reducer

Analyzes the designed reducer (as shown in fig. 1), and carries out optimization design on this basis.

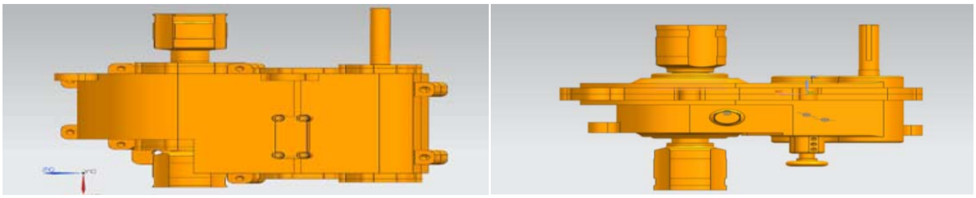

Fig. 1. Three-dimensional model of reducer.

All gears of the reducer are spur gears with small stress area, obvious stress concentration and large vibration, so the strength is weaker than that of helical gears, and large noise will be produced during operation; When shifting gears, it is necessary to stop the car and remove the fork cover, which is extremely inconvenient and wastes a lot of time; The bearing cover and the shell are separated and connected by bolts, which increases the workload of disassembling and assembling the final reducer, reduces the strength of the shell and increases the weight.

3 Calculation of transmission ratio

3.1 Calculate the running resistance of racing car

The basic parameters of the whole vehicle are shown in Table 1:

Table 1. Basic parameters of the whole vehicle

Basic parameters	Symbols	Parameter values
Full load weight/(kg)	Ga	250
Vehicle weight/(kg)	M_1	185
Driver weight/(kg)	M_2	65
Windward area/(m²)	A	1.2
wheel radius(m)	r	0.28

Rolling resistance, air resistance, slope resistance and acceleration resistance are encountered when a car is running, so the total resistance encountered by a racing car is$\Sigma F=F_f+F_w+F_i+F_j$[3].

Bach racing cars mostly run on gravel and muddy roads, according to formula (1):

$$F_\mathrm{f}=G_\mathrm{a}f \tag{1}$$

The air resistance is calculated as shown in formula (2) [3]:

$$F_\mathrm{w}=(C_\mathrm{D}Au_\mathrm{a}^2)/21.15 \tag{2}$$

The calculation method of acceleration resistance is as follows: Formula (3)[3]:

$$Fj=\delta m\frac{du}{dt} \tag{3}$$

Road resistance is the sum of slope resistance and rolling resistance when the racing car goes uphill, and the calculation method is as follows [3]:

$$F_\Psi=F_\mathrm{f}+F_\mathrm{i}= Gf\cos_\mathrm{a}+ G\sin_\mathrm{a} \tag{4}$$

Adhesion of racing cars on 30% ramps (adhesion coefficient of muddy road λ is 0.4), according to formula (5)[3]:

$$\mathrm{F}_\Phi= \mathrm{Ga}\lambda\cos\alpha \tag{5}$$

After calculation, the adhesive force F_{Φ} is 938.67N,$F_{\Psi}<F_{\Phi}$, so meet the startup conditions.

3.2 calculate transmission ratio:

According to the CVT speed check table, it can be concluded that the CVT transmission ratio is 0.59 at this time because the maximum engine speed is 3800 r/min[1]. The transmission ratio of the main reducer is calculated according to the preset maximum speed. According to the formula (6)[4],

$$u_a = 0.377\frac{rn}{i_g i_o} \tag{6}$$

3.3 verify whether the transmission ratio is reasonable:

There are some 30% (slope angle is about 16.7°) uphill sections in the race track. The maximum torque of the engine designated for the race is 21.3 N m [4] when the speed is 2600 r/min, and the CVT transmission ratio is 1.34 at this time. The uphill section is a dirt road, so take the rolling resistance coefficient f as 0.025. According to formula (7)[4], it can be found that the transmission ratio i0 of the final drive at this time should be greater than or equal to 9.48.

$$\frac{T_{tqmax} i_0 i_1 \eta_T}{r} \geq Gf cos\alpha_{max} + G sin\alpha_{max} \tag{7}$$

To prevent the wheels from slipping when the racing car goes uphill, the driving force should be less than or equal to the adhesion force. The uphill section is a dirt road, and the off-road tires used in racing cars have great adhesion, so the adhesion coefficient Φ is 0.4[3]. According to formula (8), the transmission ratio of the final drive should be less than or equal to 12.99.

$$F_{max} \leq F_f = F_z \Phi \tag{8}$$

To sum up, in order to ensure that the racing car has enough driving force to climb the hill without causing wheel slip, the transmission ratio $i_0$of the final reducer should be between 10.49 and 12.99, and the two transmission ratios calculated by comparison are within this range, so the two transmission ratios are available.

4. Design of gear and shaft

4.1 Determine transmission ratio

When the main reducer is in low gear, according to the transmission ratio distribution formula (9)[2,5]

$$1.2X_1^2 = 12.23 \tag{9}$$

It can be found that the first-stage transmission ratio is 3.19 and the second-stage transmission ratio is 3.83. In order to increase the stress area of gears, enhance the strength of gears, improve the transmission quality, and reduce the noise in the running process of the main reducer, all gears adopt helical teeth [6]. Finally, the total transmission ratio is determined to be 12.64 at low speed and 9.87 at high speed.

4.2 Determine the graduation circle diameter of gear and the gear tooth width

The tooth surface contact stress and strength conditions of a pair of steel standard helical gears are shown in Formula (10)[7]:

$$d_1 \geq 2.32\sqrt[3]{\frac{KT_1}{\varphi_d}\frac{u \pm 1}{u}\left(\frac{Z_E Z_\beta}{[\sigma_H]}\right)^2} \tag{10}$$

The calculation of gear bending strength is shown in formula (11)

$$m_n \geq \sqrt[3]{\frac{2KT_1}{\varphi_d Z_1^2}\frac{Y_{Fa}Y_{Sa}}{[\sigma_F]}COS^2\beta} \tag{11}$$

The screw angle of the primary transmission gear is tentatively set at 15°[4], and the data is brought into formula (10) to get the first-class small,according to formula (13), the standard modulus m_n of the first stage pinion is 1.73.

$$\sigma_H = \frac{\sigma_{Hlim}}{S_H} \tag{12}$$

$$d = \frac{m_n}{cos\beta} \tag{13}$$

$$[\sigma_H] = \frac{\sigma_{FE}}{S_F} \tag{14}$$

According to formula (14), m_n should be greater than or equal to 1.65 mm. According to the above calculation and the second series of standard modulus [7], the modulus of the low-speed primary transmission gear is selected as 1.75. According to the above calculation and the first series of standard modulus, the modulus of the secondary transmission gear is selected as 2.In a certain range, with the increase of the spiral angle, the bearing capacity of the gear will be better [8], but the bending strength will suddenly deteriorate when the spiral angle exceeds 30 [9]. Considering comprehensively, the spiral angle of the secondary drive gear is set at 20°.

The calculated graduation circle diameter is shown in Table 2:

Table 2. Parameters of each gear.

Gear	Low-speed primary pinion (mm)	Low-speed primary gear(mm)	High-speed primary pinion(mm)	A high-speed first-stage gear (mm)	Two-stage pinion (mm)	Two-stage big gear (mm)
Standard pitch diameter	27.18	89.96	32.61	81.56	39.34	155.37

According to formula (15):

$$b = d_1\varphi_d \tag{15}$$

The parameters of each reducer obtained from the above calculation data are shown in Figure2:

Large transmission ratio	12.64	Helical angle of primary gear	15°
Small transmission ratio	9.87	Helical angle of secondary gear	33°
First gear module	2	Maximum CVT transmission ratio	3
Secondary gear module	1.75	Gear material	20Cr
Estimated maximum speed	50Km/h	Material of reducer housing	7075aluminum

Fig. 2. Design parameter diagram.

4.3 Shift mechanism and gear setting and Gear and Transmission Shaft Design

Shift fork and combination sleeve are used for switching gear sets inside the shell (the arrangement is shown in Figure3), and shift lever outside the shell is used for shifting gears outside the shell. Firstly, the side shell of the speed reducer is provided with a stop lever limiting boss, and both the boss and the stop lever are provided with three through holes for limiting the stop lever, which respectively correspond to three gears. At the same time, it has the ability to shift gears smoothly at idle speed, and the external gear lever makes shifting easier. In order to reduce the difficulty of cart and adapt to various road conditions better, neutral gear is added on the basis of two forward gears.

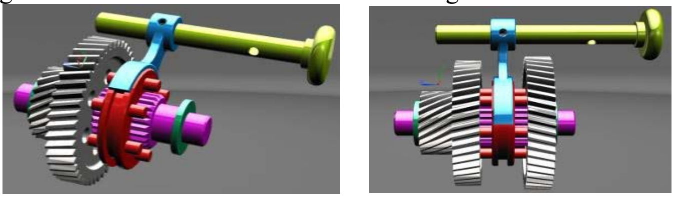

Fig. 3. Gear arrangement.

Since the gear size has been determined, according to formula [5]:

$$d \geq C\sqrt[3]{\frac{p}{n}} \tag{16}$$

The parameters of the shaft are shown in Table 3.

Table 3. Dynamic parameters of each shaft.

	Rotating speed	Torque	Power
One axis	3800	19	7.56
Two axes	1173	59.12	7.26

4.4 Establish 3D model of parts

According to the above parameters, all parts of the main reducer are modeled in 3D by UG, and assembled [10]. Whether there is interference among the parts can be detected through the assembly drawing, so as to know whether the design of the main reducer is compact and reasonable, thus making preliminary improvement according to the 3D drawing. The arrangement is shown in Figure 4. The installation of the main reducer and other parts is shown in Figure 5.

Fig. 4. Layout of speed reducer.

Fig. 5. Installation drawing of main reducer and other parts.

5 ANSYS simulation analysis

5.1 gear analysis

The file format of "stp" of low-speed gear is imported into ANSYS[11], and relevant loads and constraints are applied. Through analysis, the equivalent stress is shown in Figure6. The maximum stress is 147.33MPa, and the maximum stress is also at the gear meshing position. The maximum deformation is 0.00067mm as shown in Figure 7, which meets the strength condition[2].

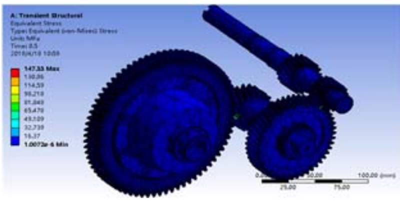

Fig. 6. Low gear stress diagram

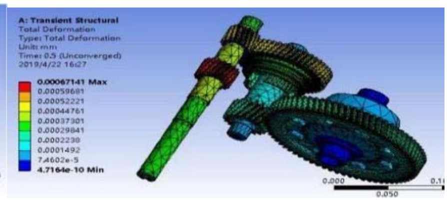

Fig. 7. Low gear strain diagram.

In the same way, the maximum stress is 120.56MPa, and the maximum stress is located at the meshing position of gears as shown in fig. 5.4, and the maximum deformation is 0.00452mm.

5.2 Axis analysis

As the carrier of torque and rotational speed, the strength of the shaft has an important influence on the performance of the reducer. The equivalent effect diagram shown in Figure 9 is obtained by applying the corresponding rotational speed and torque to the shaft. It can be seen from the figure that the maximum stress on the input shaft is 45.50Mpa; The equivalent strain diagram is shown in fig. 10, and the maximum deformation is 0.00914mm.

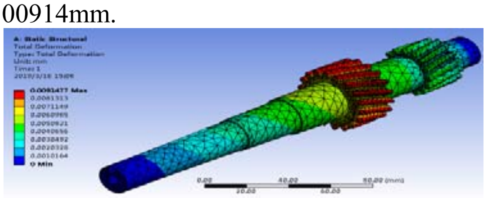

Fig. 8. One-axis equivalent effect diagram.

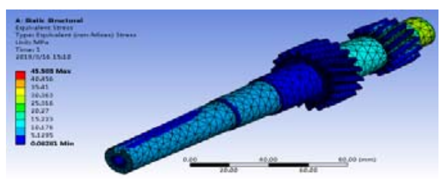

Fig. 9. Axis equivalent strain diagram.

By the same analysis, the maximum stress of two axes is 44.74Mpa, and the maximum deformation is 0.0038 mm.

Through ANSYS static simulation analysis, the stress intensity and deformation of gear and shaft meet the strength requirements, and are safe in use.

6 Design comparison

Aiming at the problems of the designed reducer, the center of gravity of the car is higher and the anti-roll ability of the car is not good. In this paper, optimization is made, and the comparison between the reducer before and after optimization is shown in the figure.

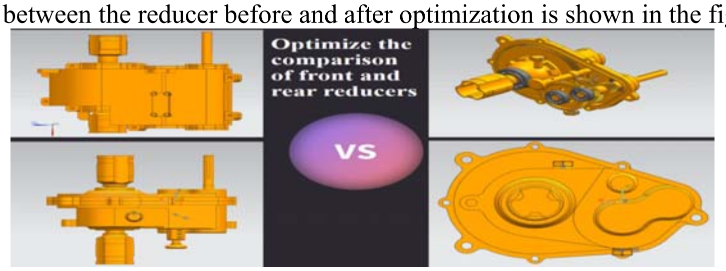

Fig. 10. Comparison of three-dimensional models of reducer before and after optimization.

In the transmission arrangement form, the engine and reducer of the car are arranged obliquely to reduce the center of gravity of the rear of the car; The reducer is designed with three gears: high speed, low speed and neutral, and a differential is added. The high speed transmission ratio is 9.87 and the low speed transmission ratio is 12.64. In terms of volume size, the left and right width is reduced by about 38% before optimization.

6 Conclusion

Based on the designed reducer, this paper designs and optimizes the main reducer from the aspects of layout, transmission ratio distribution, shift mode, overall size and shell structure, and carries out simulation analysis by ANSYS. Finally, the inclined type was designed, which reduced the overall center of gravity of the racing car; The racing reducer is designed with three gears: high speed, low speed and neutral, and a differential is added. The high speed transmission ratio is 9.87 and the low speed transmission ratio is 12.64, which can adapt to various terrains on flat and steep slopes; On the premise of meeting the strength requirements, the overall size and weight of the reducer are lightweight, which makes the performance of the racing car better, and provides a certain reference for the design and optimization of the reducer.

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