

POLITECNICO DI TORINO

Department of Mechanical and Aerospace Engineering

Master Degree

Major in Automotive Engineering

**The control of product shape in the
Product Lifecycle Management**



Supervisor:

Prof. Paolo Chiabert

Candidate:

Song Yu

Abstract

In the process of assembling mechanical parts, parts with errors generally need to be offset or squeezed by external forces to fully fit. The error of the bearing hole of the automobile gearbox results in uneven contact between the bearing hole and the shaft. The uneven contact will cause the wear of the shaft bearing mating to increase, and then affect the transmission efficiency and service life of the gearbox. Based on orthogonal tests and finite element analysis methods, this paper studies the contact stress caused by the fit between the bearing and the shaft due to errors in the bearing holes of the gearbox.

Firstly, the development status and research methods of tolerance analysis are studied. An analysis method of tolerance assembly based on orthogonal test and finite element method is proposed. Based on the orthogonal test and finite element analysis method, the contact stress caused by the fit between the bearing and the shaft due to the error in the bearing hole of the gearbox was studied.

The finite element analysis of the gearbox is performed to obtain the stress distribution of the gearbox. Then four tolerance factors of the bearing holes are selected: perpendicularity, position degree, circular runout and parallelism. The values of each tolerance factor are selected based on orthogonal experiments the method is used for experimental grouping to establish tolerance mathematical model. The influence of the above factors on the maximum stress value of the gear shaft and bearing mating is obtained through the analysis of the orthogonal test results, and the tolerance factor that has the greatest influence on the maximum stress value of each mating point is obtained through the range analysis. It is pointed out that different positions of the gearbox need to be adjusted. Tolerance factors are controlled to varying degrees.

Key words: Gearbox, Orthogonal tests, Finite element analysis

Catalogue

Abstract	2
Catalogue	3
List of Figures	5
List of Table	6
Chapter 1 GD&T.....	7
1.1 GD&T in automobile industry.....	7
1.2 Study of GD&T	8
1.3 Geometric characteristic symbols	11
1.4 GD&T applied in gearbox	12
Chapter 2 Overview of orthogonal test.....	14
2.1 Orthogonal test introduction.....	14
2.2 Orthogonal test features.....	15
2.3 Orthogonal symbol	17
2.4 Tolerance selection based on orthogonal tests.....	17
Chapter 3 Assembly analysis method	19
3.1 Finite element analysis	19
3.2 Contact setup features and ANSYS Workbench.....	21
3.3 Contact Analysis of Tolerance Assembly based on ANSYS Workbench...	24
Chapter 4 Gearbox traditional method analysis.....	27
4.1 Gearbox introduction.....	27
4.1.1 Basic parameters of gearbox	28
4.2 Gearbox condition analysis	28
4.3 Gearbox mechanical analysis	29
4.3.1 Gear meshing force calculation	30
4.3.2 Gearbox dynamics model	32
4.4 Gearbox Finite Element Analysis.....	35

4.4.1	Gearbox model simplified.....	35
4.4.2	Finite element analysis of gearbox transmission statics	36
4.5	Summery	39
Chapter 5	Influence of bearing hole error on gearbox	41
5.1	Box tolerance constraint model.....	41
5.2	Orthogonal test grouping for tolerance values	44
5.3	Analysis of results based on orthogonal experiments	47
5.3.1	Influence of tolerance on bearing maximum stress	47
5.3.2	Analysis of the maximum stress range at the bearing.....	51
5.4	Summary	54
Chapter 6	Conclusion.....	56
Reference	58

List of Figures

Figure 3-1 Finite element method.....	20
Figure 3-2 Uneven contact between bearing and bore.....	21
Figure 3-3 Adjust to Touch	22
Figure 3-4 Add offset	23
Figure 3-5 Interference part when error	24
Figure 3-6 Gearbox stress diagram without error.....	25
Figure 3-7 Axial stress diagram under error	25
Figure 3-8 Axis displacement diagram under error	26
Figure 4-1 Gearbox	27
Figure 4-2 Gearbox layout	28
Figure 5-1 Bearing hole tolerances	41
Figure 5-2 Coordinate system established in bearing bore	42
Figure 5-3 Four bearing stress	48
Figure 5-4 Input shaft bearing stress.....	48
Figure 5-5 Output shaft bearing stress	48
Figure 5-6 Intermediate shaft bearing stress (L).....	49
Figure 5-7 Intermediate shaft bearing stress (R).....	49

List of Table

Table 2-1 9 diff. combinations	16
Table 4-1 Gear ratio	28
Table 4-2 Gear parameter.....	31
Table 4-3 Force component of gear	32
Table 5-1 L9 (3 ⁴) orthogonal table.....	45
Table 5-2 Tolerance value	45
Table 5-3 Orthogonal test design	46
Table 5-4 Bearing stress orthogonal test results	50
Table 5-5 First shaft bearing hole average & range	52
Table 5-6 Intermediate shaft bearing hole (left) average & range	53
Table 5-7 Intermediate shaft bearing hole (right) average & range	53
Table 5-8 Output shaft bearing hole average & range	54

Chapter 1 GD&T

1.1 GD&T in automobile industry

In the automobile manufacturing industry, GD & T (Geometric Dimensioning and Tolerancing) is a precise and professional language used to explain the main characteristics of components such as the dimensions, geometric tolerances and positioning methods of mechanical parts. It is a method to express the design requirements and process requirements of each component and assembly with specific size and shape tolerance symbols.

With the development of the automobile industry and the increase in the per capita possession of automobiles, people have increasingly higher requirements for various performances of automobiles. As a complex mechanical product, automobiles are made up of thousands of parts. During the production and processing of parts, affected by tooling, fixtures and technological processes, each geometric element will produce certain position errors and shape errors. If the geometric and positional errors of these geometric elements are not properly controlled, they will directly affect the performance of the various components of the automobile, such as performance, wear resistance, assimilability, and service life, and even affect the overall quality and performance of the automobile. In order to control the accuracy of parts and ensure the interchangeability and economy of parts, GD & T information should be given to the relevant geometric elements of the parts according to the geometric accuracy requirements when designing the parts.

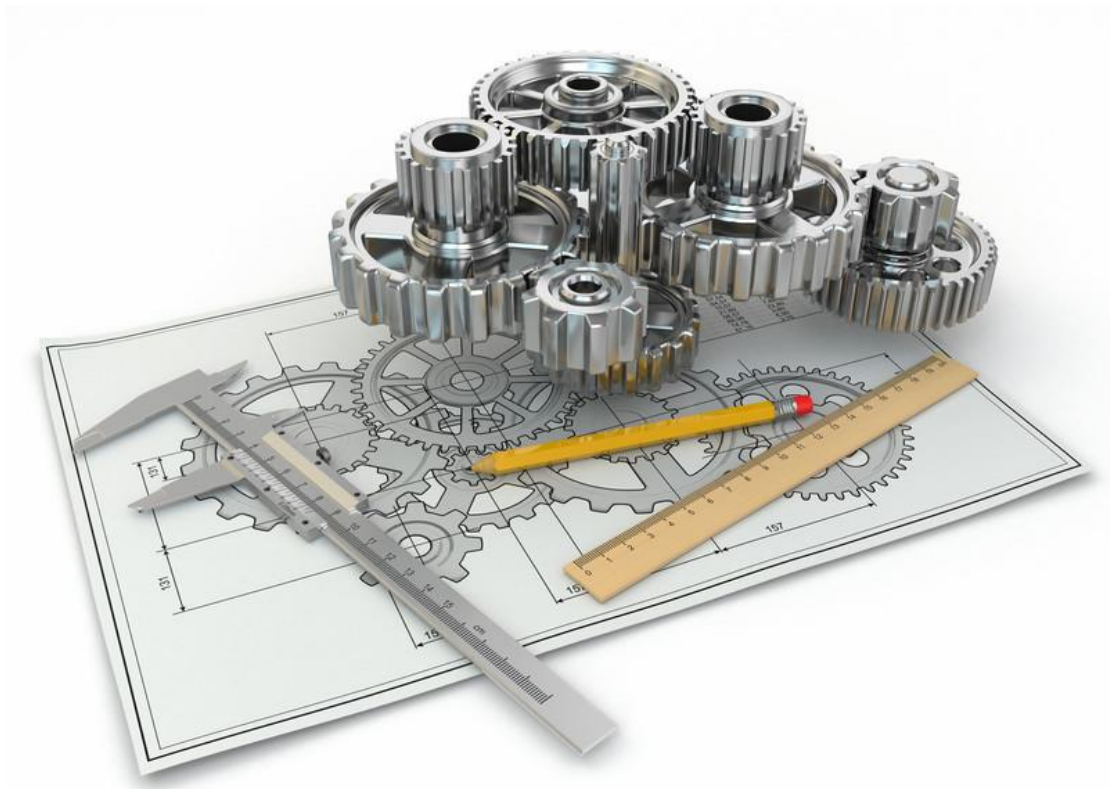
The main features of automotive parts GD & T can be divided into the following points: first, there is a large amount of information, which can accurately describe various deviations and size information; second, it has good sharing and transmission, which is suitable for each step of product development. Because its meaning and the content it refers to are unique and highly accurate, it is not easy to cause confusion and misunderstanding during the transmission, and it can be used by technical personnel

and responsible persons in relevant departments. Although different companies have different standards, a unified standard is generally used in the development of a complete vehicle. Proper use of GD & T can effectively control various deviations and errors in the development process of automobiles. The design and use of GD & T system run through the entire process of automobile development, and involves multiple departments, including design, process, manufacturing, quality inspection and procurement. It is the fundamental basis for controlling and testing the quality of parts and vehicles.

1.2 Study of GD&T

The research on tolerance is mainly for the computer-aided tolerance design CAT. Since 1978, Professor Bjorke started the computer-aided tolerance analysis research, and the computer-assisted thinking has been developed vigorously in the field of tolerance research ^[1]. Later, with the in-depth study of scholars from various countries, many different research theories appeared. Professor Requicha published his mathematical model of drift point set in 1983. The tolerance zone is expressed by the drift zone. Different size, shape and position tolerances can be displayed in the drift zone, which lays a theoretical foundation for computer-aided tolerance design ^[2]. In 1988, Israeli scholar R. Well introduced mathematical ideas to establish the mathematical functions of tolerance models, which was recognized by scholars in the field of tolerance research worldwide ^[3]. Braid proposes different mathematical models. The geometric entities are regarded as physical frames based on the mathematical vectors of geometric constraint changes, and the tolerance value is the allowable variation range of the part size. From this mathematical model gradually brought into the study of tolerance design analysis ^[4]. In the 1980s, tolerance design was given full attention, and many tolerance design research teams appeared and published certain results. Among them, the French representative team, led by Clement and Bourdet, has been devoted to the research of tolerance analysis and the development of tolerance design software, and has published many internationally recognized articles. The team led by Dr. Salomon focused on the

development of tolerance analysis systems. The development results of the FROOM system can realize two-dimensional tolerance analysis with computer assistance, which has greatly promoted tolerance research [5]. In recent years, with the maturity of computer technology, some commercial tolerance design analysis software has appeared, such as CE / TOL, Sigmund3D, etc.



Tolerance analysis is to analyze the tolerance range of a known part through the accumulation of transmission, which generates interference or gaps between components during assembly, which has a certain impact on the performance of the manufactured product. By analyzing the actual manufactured products, it is checked whether the tolerance is reasonable in design and correction is given. At present, the main methods of tolerance analysis are divided into the following three categories: extreme value method, statistical method and Monte Carlo method.

(1) The extreme tolerance method is a method of performing algebraic calculations using known tolerance values of parts. This method takes the extreme size of each assembly when calculating the closed tolerance value, and then performs the superposition calculation of the size ring. The calculation of the extreme tolerance

method is mainly used for calculation of dimensional chains with fewer tolerance composition rings, higher tolerance levels or more tolerance composition rings. This method requires that the marked tolerances must ensure interchangeability or tolerance requirements. It is more lenient and does not require very accurate dimensional chain calculations of tolerance values, such as all primary processing machinery. This method is relatively simple and requires a small amount of calculation, but the disadvantage is that the tolerance value obtained according to this calculation method is usually small, and the possibility of tolerances of individual parts at the same time is extremely small. Factors, the superposition of each part is not a linear relationship, so the value ratio obtained will have a smaller range, which increases the manufacturing cost.

(2) The statistical method assumes that the tolerances of each part are subject to a certain statistical distribution during the assembly calculation process. The typical assumption is a normal distribution. It is still considered that the calculation of the assembly tolerance is based on the linear calculation of the part tolerance. The mathematical optimization calculation method or the non-linear relationship is transformed into a linear calculation using a mathematical formula, such as Taylor expansion. The statistical tolerance analysis method considers the actual distribution of the tolerances of the part, which is closer to the actual tolerances of the part than the extreme value method, and reduces the possibility of tolerance extremes coexisting. It allows the part to have a loose tolerance band, has a better application in machinery manufacturing.





(3) The Monte Carlo method uses statistical test methods to perform actual statistics on the size distribution of each assembly part, and then uses a random number generator to generate corresponding design values that can be used to calculate the fit. By generating samples from these values, the corresponding functions are obtained. Calculate its characteristic distribution. Monte Carlo tolerance analysis method can better handle the assembly relationship and the nonlinear relationship caused by elastic deformation. It has been widely used in mechanical manufacturing. However, the Monte Carlo method also has certain shortcomings: this method requires a lot of actual statistics to effectively guarantee the correctness of the calculation. If the sample

changes during the calculation, it needs to be re-calculated, which takes a lot of time and effort.




1.3 Geometric characteristic symbols

There are thirteen geometric characteristic symbols used in the language of G.D.&T. They are divided into five separate categories: form, orientation, location, runout and profile [6].




(1) Form

Geometric Characteristic	Symbol
Straightness	
Flatness	
Circularity	
Cylindricity	



(2) Orientation

Geometric Characteristic	Symbol
Angularity	
Perpendicularity	
Parallelism	



(3) Location

Geometric Characteristic	Symbol
Position	
Concentricity	
Symmetry	

(4) Runout

Geometric Characteristic	Symbol
Circular Runout	
Total Runout	

(5) Profile

Geometric Characteristic	Symbol
Profile of a Line	
Profile of a Surface	

1.4 GD&T applied in gearbox

The main tolerances of the gearbox are restricted by several tolerance factors such as perpendicularity, parallelism, circular runout and position.

The perpendicularity is the condition that the surface or center plane or axis is 90 ° from the datum. The perpendicularity tolerance is the amount that allows a surface, axis, or center plane to vary from an angle perpendicular to the datum.

Parallelism refers to the condition that all points on the surface, center plane or axis are equal to the distance from the reference plane or axis. Parallelism tolerance is the amount that allows a curved surface, center plane, or axis to change from a parallel state.

The parallelism control establishes the tolerance zone of two parallel planes or cylinders, and all points of the controlled surface, center plane or axis must be located in it.

Round runout is a composite control that affects the shape and position of round elements of a part feature. It is called a composite control because it affects both shape and position. Circular runout is often used to control the position of a circular element with a diameter. When applied to diameter, it controls the circular runout and concentricity of the diameter with respect to the reference axis. When applied to a surface at 90° to the reference axis, it only controls the attitude of the circular element. Circular runout applies to each circular element of a surface that is independent of each other.

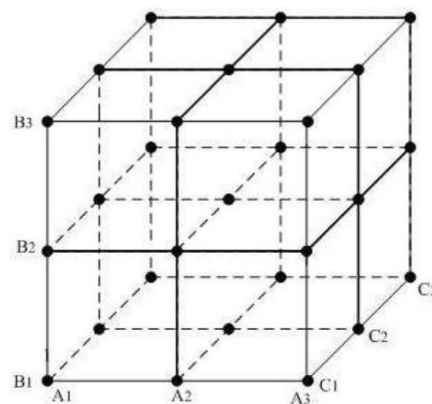
Position is the range within which a body's axis or center plane allows its own position to change, that is, the actual range of a body's axis or center plane that is allowed to fluctuate. The position tolerance is used to evaluate the accuracy of the actual element position based on the ideal position given on the drawing.

Chapter 2 Overview of orthogonal test

Orthogonal test method is used to calculate the tolerance value, and then a geometric tolerance model is established to expand the finite element analysis.

2.1 Orthogonal test introduction

Orthogonal experiment is an experimental design method used to design and arrange multiple experimental factors and examine the influence of each factor on the results. Since being proposed by mathematics scholars in 1942, it has been widely used and developed rapidly in various fields. At present, such scientific design experiments have developed into a set of standardized test tables-orthogonal tables, which can be used to arrange the test work. It is mainly used for multi-factor test arrangements, and at the same time, each level of multiple factors can be compared to the test results. The effect of the index is to reduce the number of tests and select the best combination of test conditions, that is, to select the optimal level of each factor to form a more ideal condition. This is the best selection condition for the expected result. On the other hand, the analysis of orthogonal experiments can also help the experiment designer to grasp the main factors among a variety of condition factors, and judge the degree of influence of each factor on the test, and whether the factor acts alone or synthesizes with other factors. Effect, and can also give error estimates from mathematical statistics [7].

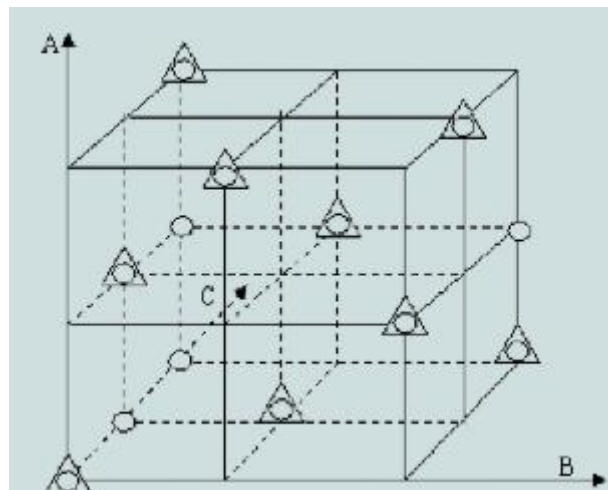


When factorial design requires too many experiments, a very natural idea is to select a representative level combination from factorial design level combinations for testing.

Therefore, fractional factorial designs have emerged, but for practical workers who have little knowledge of experimental design, it is still difficult to choose appropriate fractional factorial designs. For example, for a three-factor three-level experiment, according to the requirements of a comprehensive experiment, an experiment of $3^3 = 27$ combinations must be performed, and the number of repetitions of each combination has not been considered. If the experiment is arranged according to the L9 (3^4) orthogonal table, it only needs to be done 9 times, and the experiment is performed 15 times according to the L15 (3^7) orthogonal table, which obviously reduces the workload significantly. Therefore, orthogonal design has been widely used in many fields.

2.2 Orthogonal test features

Orthogonal test method is a kind of designing method to study many factors and levels. It conducts tests by selecting a suitable number of representative test cases from many test data, which have evenly dispersed, neat comparable characteristics.^[8]



(1) Equilibrium dispersion

The coordination between the various factors designed according to the orthogonal test is uniform, or the orthogonal test design can disperse the factors of the experimental conditions in a complete horizontal combination, making it more representative and more It can reduce the number of tests, and it is easy to obtain the optimal test conditions with fewer tests. The feature that the factors in the design are evenly matched is called

the balanced dispersion of the orthogonal design. This property of orthogonal design can average the test result values to eliminate small errors caused by non-equilibrium and improve the accuracy and reliability of the test results.

For example, in Fig 2-2, each plane on the cube contains three points, each line segment on the cube contains a point. Therefore, these points are highly representative and can better reflect the situation of the comprehensive test.

(2) Neat comparable

In the orthogonal design experimental group, not only the levels of each factor are evenly matched, but also there is a certain regularity in the change of each factor, that is, the level of other factors is in the experimental group when examining the influence of a single factor on the test result Uniform distribution, that is, the extent to which other factors affect it can be considered consistent.

In this way, by comparing the effect of each level of each factor on the result, the interference caused by changes in other factors can be largely excluded, the role of the factor is more prominent, and the effect of each factor can be more clearly distinguished and the size estimated.

For example, among the three factors of A, B, and C, there are three levels of factor $A_1A_2A_3$, and there are three different levels of B and C under the conditions.

	B_1C_1		B_1C_2		B_1C_3
A_1	B_2C_2	A_2	B_2C_3	A_3	B_2C_1
	B_3C_3		B_3C_1		B_3C_2

Table 2-1 9 diff. combinations

In these 9 different combinations, three levels of factor B and C are included under each level of factor A. Although the matching methods are different, B and C are in the same position. When comparing different levels of factor A, the effects of different levels of factor B cancel each other, and the effects of different levels of factor C also cancel each other. Therefore, the three levels of factor A are neat comparable. Similarly, the three levels of factors B and C are neat comparable.

2.3 Orthogonal symbol

For the convenience of description, use L to represent the conventional table, commonly used are L8 (2^7), L9 (3^4), L16 (45), L8 ($4 \times$), L12 (2^{11}), etc. The meaning of each number of this symbol is as follows:

$$Ln(r^m)$$

L: Orthogonal table code

n: Number of rows in orthogonal table (number of test)

r: Number of factor levels

m: Number of columns in orthogonal table (Maximum number of factors that can be scheduled)

2.4 Tolerance selection based on orthogonal tests

In the traditional mechanical design process, the value of the tolerance is to query related manuals, and different levels of tolerance are selected according to the required accuracy, and each level will only give the limit of the tolerance within that level range. After the geometric features of the part are detected within the tolerance limit, the geometric features of the part are considered to meet the tolerance requirements, and the geometric features are constrained within a certain range. The tolerance given is not a specific value but a precision range, called the tolerance zone. Generally, the selection of tolerance is the selection of grade accuracy, and the establishment of a geometric model requires specific values, such as the specific size or angle of a geometric feature. Therefore, for the analysis of error characteristics, a specific value must be selected from the tolerance band constrained by the tolerance class. To perform geometric modeling, it is required that the selected tolerance value should be representative. By using the orthogonal dispersion of the orthogonal test, a representative tolerance value can be selected from a wide range of tolerance levels.

Each feature of a geometric model does not just have a tolerance constraint. Generally, geometric features are defined by the joint constraints of geometric tolerances and

geometric tolerances. For example, the center position of a hole can be constrained by a positional tolerance or a coaxiality tolerance with another center, and the direction of the hole axis can also be constrained by perpendicularity or parallelism. The marking of tolerances is a complex issue. And more tolerance constraints require a lot of choice of tolerance values, making it difficult to carry out later tolerance analysis.

This paper first determines the tolerance factors that have a large impact on each geometric feature, and selects representative tolerance values for them. However, the combination of all tolerance values will bring more test solutions. It is found that even if there are only three factors, there are 27 test combinations that only take three level values for each factor. More, the test combination will also be more, so that it will bring more test analysis times and a larger workload. The tests arranged using the characteristics of neat comparability of orthogonal test methods can not only effectively reduce the number of test analyses, but also ensure the comparability between the test schemes and improve the analysis efficiency. Therefore, for the complex and complicated geometrical values of geometric features, the orthogonal test design method is introduced in this paper.

Chapter 3 Assembly analysis method

3.1 Finite element analysis

With the breakthrough of some mathematical methods and the expansion of computer applications, the finite element principle has been expanded and the calculation efficiency of the finite element has been improved. The development of the finite element method has gradually matured, so it is also increasingly applied in the error analysis of mechanical products.

The finite element method is a new type of mathematical analysis method proposed in the last century. This method uses the concept of discretization to decompose the complex whole into mutually continuous individuals. The whole continuous physical is transformed into piecewise continuous, thereby transforming the overall complex analysis into simple analysis of individuals, forming an efficient and simple mathematic method. Continuous individuals are generally called units, and the units are connected by nodes to form a whole. The internal force between the units is transmitted through a common node. When the whole is subjected to external force, each unit deforms, so the nodes produce different displacement changes, which is called node displacement. In the finite element calculation process, mechanical knowledge or variational principle is used to establish the mechanical relationship between the node force and the node displacement, and the equation with the node displacement as an unknown quantity is obtained. Then, the relationship between nodes and external elements is established for the overall analysis, and the function interpolation is used to solve the problem ^{[9][10]}.

Finite element is applied to the structural error analysis of mechanical products. Its principle is based on a mathematical solution method, which is simple and easy to understand. Moreover, the method is generally applicable to many fields and has a high practicality in the application of tolerance. The simulation analysis in the computer can provide guidance for the actual experiment, while also avoiding some unnecessary operations in the actual experiment. With the development of technical information

means, especially the advancement of computer-aided functions, it is more practical to use finite element methods to analyze mechanical products containing errors ^[11].

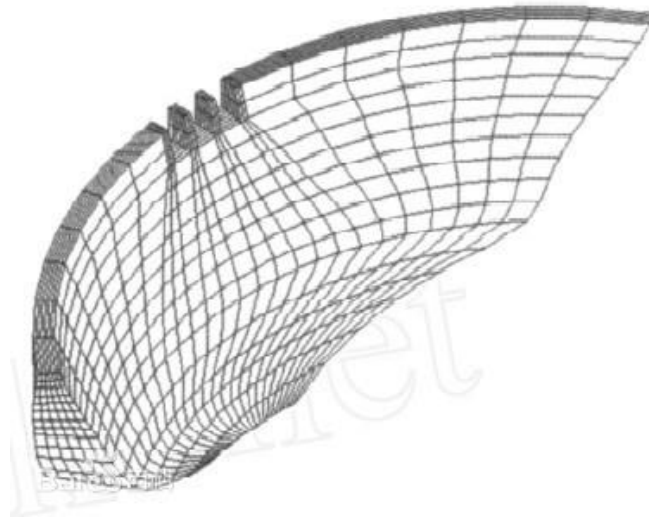


Figure 3-1 Finite element method

The establishment of the finite element method equations is based on the application of mathematical integration, which can solve the complicated or complicated model system. The finite element analysis of the tolerance model is mainly for the analysis of the impact of the mating parts at the contact position. Using the finite element method to solve the contact problem will effectively simulate the uneven contact between mating parts due to tolerances. The treatment of contact in the finite element is an indefinite boundary problem, which has complicated elastic mechanics calculation and nonlinear solution. The contact surface is usually non-linear due to changes in the contact area or contact pressure distribution, and also causes non-linear problems due to the friction between the two sides of the contact. Therefore, iterative calculations must be repeated to solve the contact problem. When solving the contact problem, the finite element method must not only ensure that the interiors of the objects in contact with each other are coordinated with each other in deformation, and cannot penetrate into the other contact surface. Also ensure that the contact surfaces have certain connection conditions. Before meshing the contact surface before solving, make sure that the node coordinates of the contact positions of the two objects in contact on the initial contact surface are the same and form a contact pair with each other.

3.2 Contact setup features and ANSYS Workbench

When there is no error, there will be no interference or gap between the mating parts under the ideal size and constraint conditions. When the finite element analysis is performed, the two contact surfaces, that is, the contact surface and the target surface just contact at the initial position without adjustment. When the error exists, the distance between the two contact surfaces in the geometric model will increase or decrease, as shown in the following non-uniformity, that is, there will be some interference and some gaps between the contact surface and the target surface.

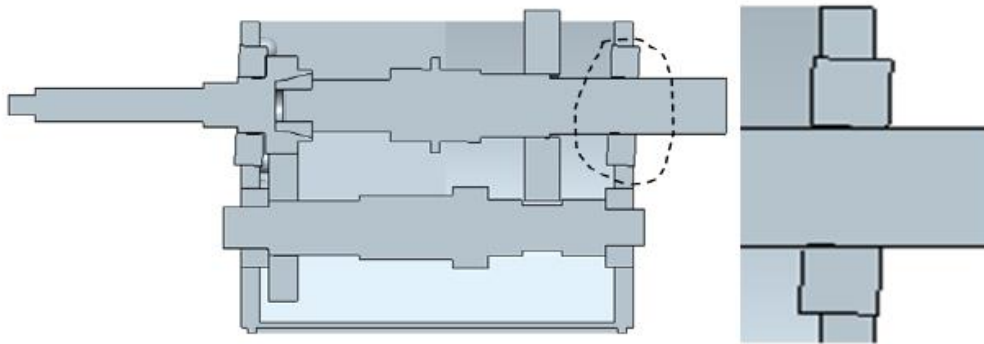


Figure 3-2 Uneven contact between bearing and bore

Actually, the mating parts may have gaps between the contact surfaces in the assembly, but when the two parts interfere to each other, they do not penetrate each other like the established geometric model, but the two parts squeeze each other, and local contact stress occurs. Usually, the finite element analysis is performed according to the ideal geometric model. The initial positions of the two contact surfaces in the geometric model are not considered. Although this calculation result has high accuracy, there is a certain deviation from the actual situation. In order to simulate the local stress of extrusion in actual assembly, the initial adjustment function of the contact of ANSYS software is needed to adjust the position of the contact surface of the geometric model with errors.

During the actual assembly process, due to the existence of tolerances, some mating components cannot be fully mated, but they are mated together after some small deformation caused by artificial or mechanical external forces. There must be internal

stress between these mating components, which will affect their respective performance. Using ANSYS Workbench software to adjust the position of the contact surface and analyze the tolerance model, the internal stress of the geometric assembly can be simulated. In this way, the internal stress of the contact between the parts that are forced to fit together after slight twisting or bending during actual assembly can be obtained through analysis, so as to provide a reference guide for further analysis of the rationality of the tolerance selection.

Due to the existence of errors, the feature position on the part always deviates from the ideal position, which will lead to uneven contact between the two contact surfaces, which will often cause contact nonlinearity in finite element analysis. For the non-linear contact between the target surface and the contact surface, there are three methods in ANSYS Workbench: Adjust to Touch, Add Offset, Ramped Effects, Add Offset, No Ramping [12].

(1) Adjust to Touch: The contact surface is set to this method. The software ignores the gap or interference between the target surface and the contact surface during calculation, and considers that the two contact surfaces just reach a stress-free contact state. In this contact method, even if there is a gap between two contact surfaces in the model, the initial state is considered to be contact. There may still be gaps or infiltrations in the iteration process.

After adjusting to touch adjustment, the gap and penetration of the initial state are adjusted to just touch

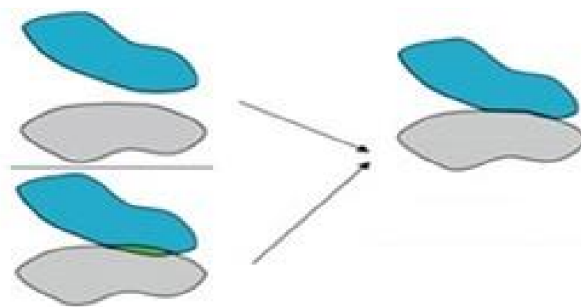


Figure 3-3 Adjust to Touch

(2) Add Offset, Ramped Effects: By entering the value of Offset, the adjusted distance

of the two contact surfaces is equal to the original distance of the two surfaces plus the given input value. Enter a positive value in Offset, and the two contact surfaces move closer to each other. If you enter a negative value, the contact distance of the contact surfaces increases. The contact stress generated during the adjustment process is equivalent to the application of a certain load. The contact surface is set so that the load is divided into several sub-steps and superimposed.

After adding offset, the initial position is adjusted according to the corresponding offset value

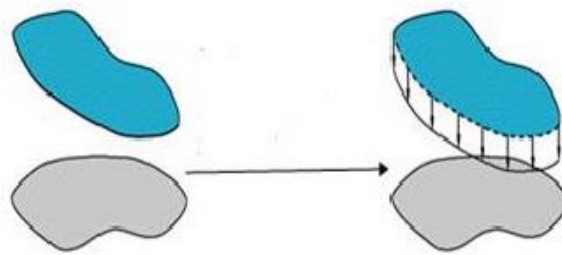


Figure 3-4 Add offset

(3) Add Offset, No Ramping: The setting of this contact surface is basically the same as Add Offset, Ramped Effects. The difference is that when the load is applied, this processing method makes the load all applied in the first load step without stepping.

During the actual assembly process, due to the existence of tolerances, some mating parts cannot be fully mated, but they are only mated together after some slight deformation caused by artificial or mechanical external forces. There must be internal stress between these mating parts, which will affect them. Respective performance.

Using ANSYS Workbench software to adjust the position of the contact surface and analyze the tolerance model, the internal stress of the geometric assembly can be simulated. In this way, the internal stress of the contact between the parts that are forced to fit together after undergoing slight distortion or bending during actual assembly can be obtained through analysis, so as to provide a reference guide for further analysis of the rationality of the tolerance selection.

3.3 Contact Analysis of Tolerance Assembly based on ANSYS Workbench

General 3D modeling software such as UG, Pro / E, etc. are used to establish the 3D model of the part for assembly in accordance with the ideal size and position. If there is an error in the component, it is assembled according to the assembly rules of these 3D software during assembly. There must be gaps or interferences in the model, and the interference part between them is mutual penetration.



Figure 3-5 Interference part when error

In ANSYS Workbench, for an error-free model, there will not be too much stress during the analysis, so there is no interference, and the contact parts are not squeezed; however, there will be stress due to contact when the error model is analyzed.

For the above simple shaft-to-hole contact model, analyzed in ANSYS Workbench, the bottom surface of the box is fixed, no other external force is applied, and only the internal stress caused by the contact between the mating parts is viewed. The contact setting uses non-linear friction contact, and the target surface is the inner surface of the box bearing hole, and the contact surface is the outer surface of the shaft. The initial adjustment is set to zero, so that the distance between the two contact surfaces maintains the distance of the error geometric model, so that the stress due to interference can be simulated during analysis. As shown in Figure 3-5 and Figure 3-6, the analysis results of the error-free model show that the overall maximum stress is 5.7691×10^{-4} MPa,

which is relatively small and has almost no contact. The maximum stress of the model with error reaches 82.635 MPa, which is much larger than that without error. From the perspective of the stress distribution of the shaft in the figure, the maximum value appears at both ends of the shaft, which is the position of contact with the hole, indicating that the existence of errors will have a certain effect on the local stress of the contact between the parts.

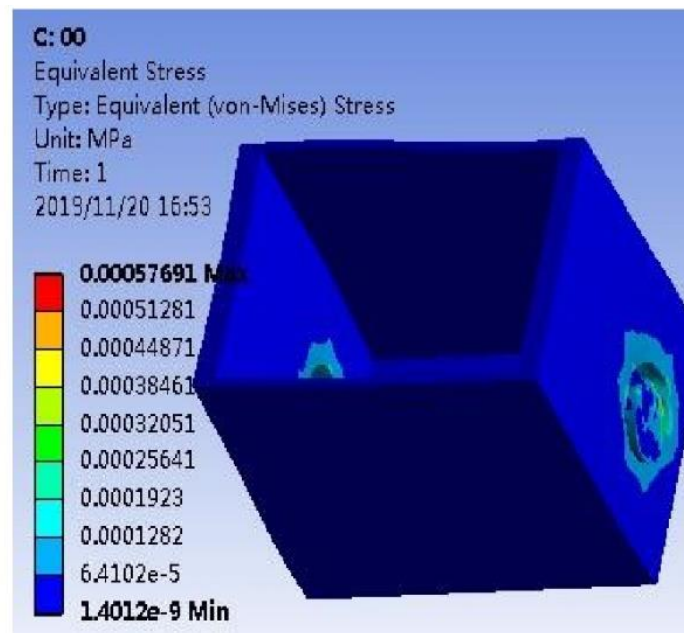


Figure 3-6 Gearbox stress diagram without error

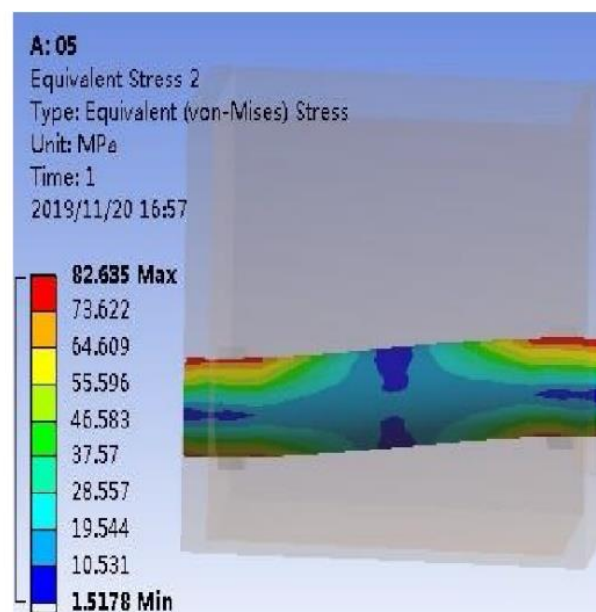


Figure 3-7 Axial stress diagram under error

The following Figure 3-7 is the overall displacement distribution in the error model

results. The displacement of the end of the shaft with respect to the other end is 0.5116mm, which is close to the position tolerance of 0.5mm. The target surfaces fit together, and the shaft and hole are also assembled with each other after deformation, simulating the assembly process when the error exists in practice. At the same time, it is proved that it is correct to analyze the error model by using the finite element initial adjustment method.

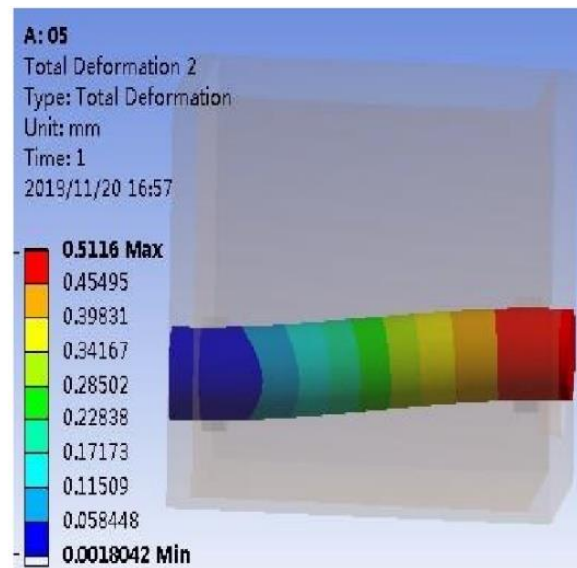


Figure 3-8 Axis displacement diagram under error

Chapter 4 Gearbox traditional method analysis

4.1 Gearbox introduction

Gearboxes usually have an input shaft and an output shaft. Compared to output shafts, input shafts typically run at higher speeds and relatively low torque. This configuration is called a gear reducer. The input shaft is usually coupled to the motor directly or through some type of coupling. The output shaft is used for the drive mechanism.

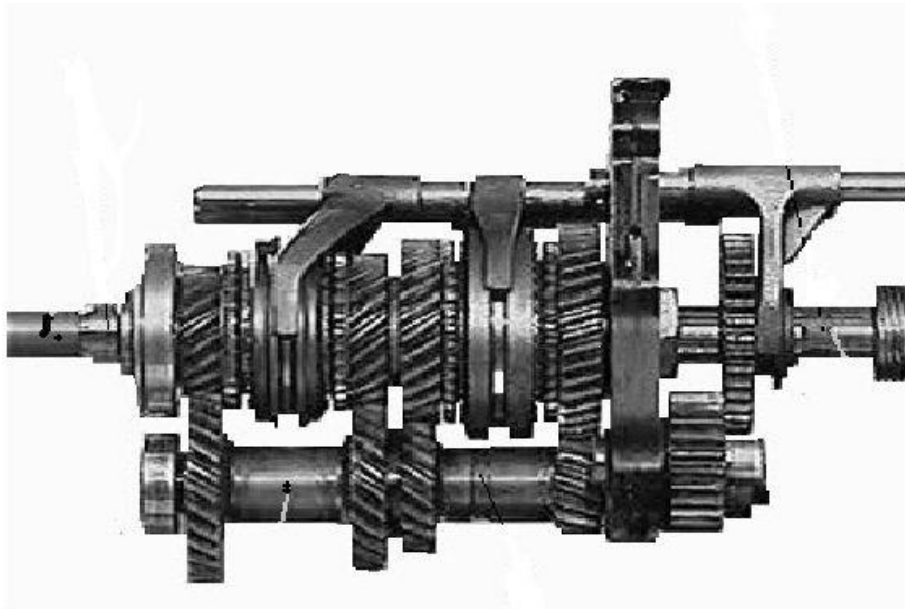


Figure 4-1 Gearbox

There are two main types of commercially available transmissions: inline and right angle. In-line gearboxes have parallel offset or collinear input and output shafts. The input and output shafts of a right-angle gearbox are perpendicular to each other, or on the same plane, or in the offset direction. In-line gearboxes usually use spur gears, planetary gears, cycloidal mechanisms or harmonic generators. Right-angle gearboxes usually have worm or bevel gears.

The gearbox studied in this paper is a three-axle transmission box, which uses a fixed-shaft transmission design, and the box structure also adopts a fixed-shaft design. The

interior uses a single intermediate shaft design, using a full synchronizer structure, with six forward gears and one reverse gear. Helical gears are often meshed. The main features of such a transmission are light weight, small size, compact structure, and convenient vehicle layout.

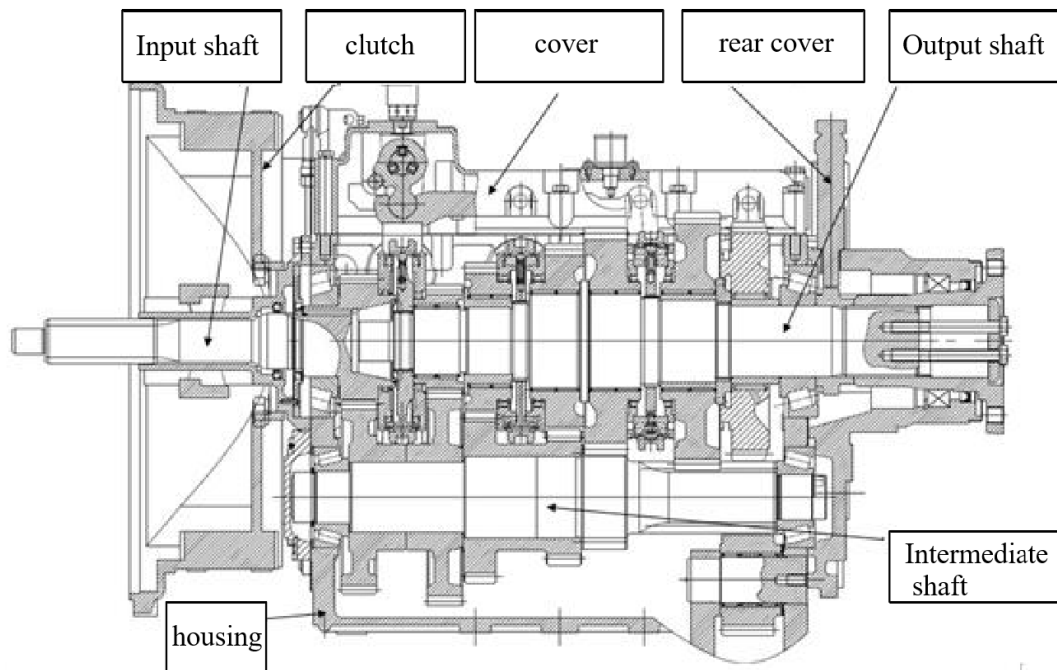


Figure 4-2 Gearbox layout

4.1.1 Basic parameters of gearbox

The gear ratio of this gearbox is below:

<i>Gear</i>	#1	#2	#3	#4	#5	#6	Reverse
<i>Ratio</i>	6.25	3.58	2.22	1.36	1	0.73	5.75

Table 4-1 Gear ratio

Maximum input torque: 1600Nm

4.2 Gearbox condition analysis

Usually the gearbox has multiple gears, and the force and torque received by the gearbox is the largest in the first gear, so the general analysis is performed in the first gear. The first gear power transmission route is shown in the figure. The power

transmitted by the engine or motor is input through the input shaft, and is transmitted to the intermediate shaft through the normally meshing gear pair. The first gear of the intermediate shaft and the first gear of the output shaft mesh. The output of the gearbox output shaft is transmitted to the outside of the gearbox, and the car is driven forward by a drive system such as a drive axle, a universal joint, and a reducer.

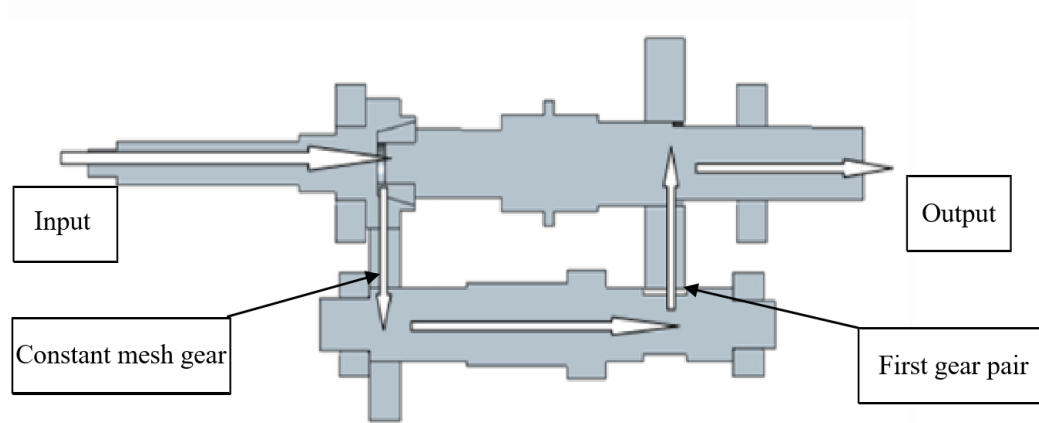


Figure 4-3 First gear power transmission

The first gear ratio i is determined by the product of the constant mesh gear ratio and the first gear pair ratio:

$$i = i_c * i_{01} = 6.25$$

in which i is the first gear ratio, i_c is constant mesh gear ratio, i_{01} is first gear pair ratio.

The gear ratio of the first gear in this paper is 6.25, and the gear ratio of the first gear of the gearbox is the maximum gear ratio that this gearbox can provide. The input torque is increased by the first gear of the gearbox, which can provide the vehicle with greater traction through the obstacle or sloped road. At the same time, the speed of the output shaft decreases, that is, the function of decelerating and increasing the distance is realized. Therefore, under normal circumstances, the load received by the gearbox in the first gear transmission is the largest in each gear position. Therefore, the analysis of the box in this article is performed under the first gear load condition.

4.3 Gearbox mechanical analysis

When the gearbox works in the first gear, two sets of gears mesh with each other. The

gear tooth surface receives the gear meshing force, and the transmission shaft receives the torque and the reaction force from the gear. According to the classic mechanical calculation of gears in mechanical design, the component forces of the gear meshing force in all directions can be obtained, and the knowledge of material mechanics can be used to obtain the forces and moments on the parts of the transmission system when the gear is working. Obtain the bearing force of the bearing at the position of each bearing hole in the box.

4.3.1 Gear meshing force calculation

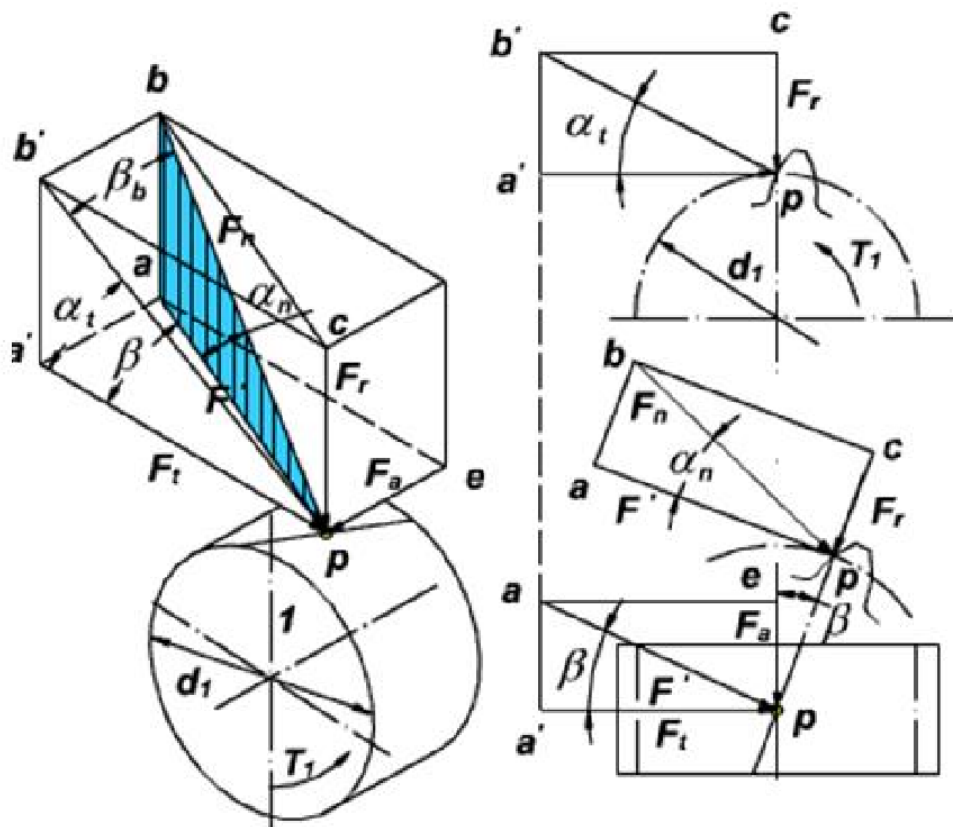


Figure 4-4 Helical gear force

Generally, the friction between the gear meshing surfaces is not taken into account in the calculation. The force distributed on the tooth surface generated by the gear meshing is used as the concentrated force F_n at the midpoint of the tooth width. At the same time, F_n is decomposed into F_t , F_r , F_a , which point to the circumferential direction, the center of the circle and the axial direction of the gear, respectively ^[13].

$$F_n = \frac{F_t}{\cos\alpha_t \cos\beta_b} = \frac{F_t}{\cos\beta \cos\alpha_n}$$

$$F_t = \frac{2T_1}{d_1}$$

$$F_r = \frac{F_t \tan\alpha_n}{\cos\beta}$$

$$F_a = F_t \tan\beta$$

In which T_1 is torque of gear on shaft [N*mm]; α_n is normal pressure angle of helical gear tooth surface; α_t is pressure of helical gear surface; β is helical gear indexing circle helix angle.

	Always meshing drive gear	Always meshing passive gear	First gear drive gear	First gear passive gear
Normal Module [Mn]	4	4	4.75	4.75
Number of Teeth	28	42	12	50
Normal Pressure Angle	23	23	24	24
Helix Angle	24.5	24.5	7.5	7.5
Addendum Coefficient	0.9497	0.9002	0.964	0.8658
Dendendum Coefficient	1.2128	1.2873	1.457	1.4605
Modification Coefficient	0.14	-1.031	0.5	0.75
Tooth Width [mm]	40	35	45	

Table 4-2 Gear parameter

From the above gear parameters, the direction of the force of the gear can be obtained. The directions of the circumferential force F_t and the radial force F_r can be obtained according to the rotation direction of the gear meshing, and the direction of the axial force F_a is directed to the meshing surface of the corresponding gear along the gear axis direction. It is only necessary to find the force on the driving gear. The magnitude of the force on the driven gear is equal to that in the opposite direction. According to the basic parameters of the gears of the constant mesh gear pair and the first gear pair, the component forces in all directions when the gear meshes can be obtained by the above formula.

	Always meshing drive gear	Always meshing passive gear	First gear drive gear	First gear passive gear
Tangential force F_t[N]	25999.35	25999.35	83492.78	83492.78
Radial force F_r[N]	12128.06	12128.06	37494.15	37494.15
Axial force F_a[N]	11848.59	11848.59	10992.03	10992.03
Torque T[N]	1600	2400	2400	10800

Table 4-3 Force component of gear

4.3.2 Gearbox dynamics model

Based on the above calculations, the magnitude of the forces in the gears of the constant meshing gear set and the gear of the first gear set in the first gear transmission can be obtained. The schematic diagram of the force when the gearbox is working is shown below ^{[14][15]}:

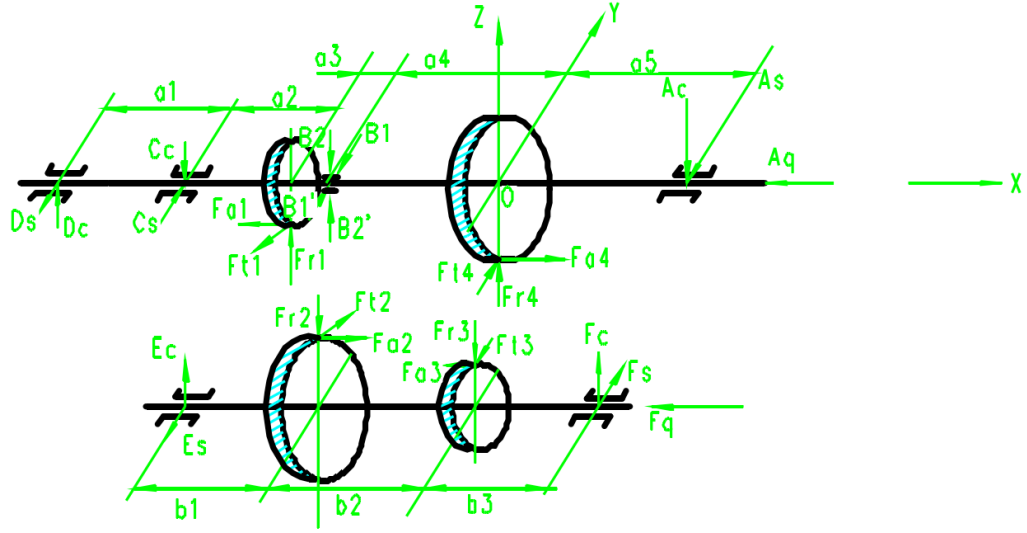


Figure 4-5 Force analysis for gearbox

in which $a_1 = 323mm$, $a_2 = b_1 = 35mm$, $a_3 = 15.5mm$, $a_4 = 317mm$, $b_2 = 332.5mm$, $a_5 = b_3 = 98.5mm$.

For the intermediate axis, on the vertical plane (ZOX plane, the same below):

$$F_{a3} * \frac{d_{048}}{2} + F_c * (b_1 + b_2 + b_3) - F_{a2} * \frac{d_{056}}{2} - F_{r2} * b_1 - F_{r3} * (b_1 + b_2) = 0$$

in which d_{048} is indexing circle diameter of first gear drive gear, d_{056} is indexing circle diameter of constant mesh passive gear.

After the calculation, the component force of the bearing force at the shaft hole of the intermediate shaft on the vertical plane can be obtained.

$$F_c = 32148.84 N$$

$$E_c = F_{r2} + F_{r3} - F_c = 17473.37 N$$

For the intermediate axis, on the horizontal plane (XOY plane, the same below):

$$F_{t2} * b_1 - F_{t3} * (b_1 + b_2) + F_s * (b_1 + b_2 + b_3) = 0$$

After the calculation, the component force of the bearing force at the shaft hole of the intermediate shaft on the horizontal plane can be obtained.

$$F_s = 63891.89 N$$

$$E_s = F_{t2} - F_{t3} - F_s = -6398.46 N$$

The axial force at the shaft hole of the intermediate shaft due to gear transmission is:

$$F_q = F_{a2} - F_{a3} = 865.56 N$$

in which the direction is opposite to the picture, and it acts on the right end of the gearbox.

For the output shaft, on the vertical plane:

$$B_2 * (a_4 + a_5) - F_{r_4} * a_5 + F_{a_4} * \frac{d_{111}}{2} = 0$$

in which d_{111} is first gear passive gear indexing circle diameter.

Enter the data to calculate, the component force received by the bearing at the connection between the input shaft and the output shaft in the vertical direction.

$$B_2 = 5719.88 \text{ N}$$

Force on the output shaft bearing hole in the vertical plane is:

$$A_c = F_{r_4} - B_2 = 31774.27 \text{ N}$$

For the output shaft, on the horizontal plane:

$$B_1 * (a_4 + a_5) - F_{r_4} * a_5 = 0$$

Enter the data to calculate, the component force received by the bearing at the connection between the input shaft and the output shaft in the horizontal direction.

$$B_1 = 1989.62 \text{ N}$$

Force on the output shaft bearing hole in the horizontal plane is:

$$A_s = F_{t_4} - B_1 = 81503.16 \text{ N}$$

For input shaft, on the vertical plane:

$$-D_c * a_1 - F_{a_1} * \frac{d_{031}}{2} + F_{r_1} * a_2 + B'_2 * (a_2 + a_3) = 0$$

in which d_{031} is indexing circle diameter of constant meshing driving gear, $B'_2 = B_2 = 5719.88 \text{ N}$

Enter the data to calculate, the component force of the input shaft's outer end in the vertical plane can be obtained.

$$D_c = 48.997 \text{ N}$$

Force on the input shaft bearing hole in the vertical plane is:

$$C_c = F_{r_1} + D_c + B'_2 = 17896.94 \text{ N}$$

For the input shaft, on the horizontal plane:

$$D_s * a_1 - F_{t_1} * a_2 + B'_1 * (a_2 + a_3) = 0$$

Enter the data to calculate, the component force of the input shaft's external end in the

horizontal plane can be obtained.

$$D_s = 2506.196 \text{ N}$$

Force on the input shaft bearing hole in the horizontal plane is:

$$C_s = D_s + F_{t_1} - B'_1 = 26515.926 \text{ N}$$

Axial force on the input and output shaft's shaft holes due to gear transmission is:

$$A_q = F_{a_1} - F_{a_4} = 856.56 \text{ N}$$

in which the direction is shown as figure, and it acts on the input shaft's shaft hole.

4.4 Gearbox Finite Element Analysis

Since we have done mechanical calculations of the transmission in the first gear of the gearbox, the following will analyze the gearbox assembly in the ideal gear state without tolerances by using finite element software to observe the stress distribution of each component. The gearbox assembly has a complex structure, and some part features can be simplified during analysis to improve the efficiency of analysis and calculation.

4.4.1 Gearbox model simplified

In order to improve the meshing quality during the finite element analysis of the transmission case, some structural features of the transmission case can be simplified. When simplifying the gearbox model, it is necessary to ensure the efficiency of the analysis and calculation, the short calculation time, and the accuracy of the analysis results.

- (1) The boss structure and the bolt holes, pin holes, oil grooves, and some chamfers and fillets on the boss are omitted.
- (2) In order to improve the meshing quality, the spline grooves and snap ring grooves on the transmission shaft have been removed.
- (3) When considering the transmission in a certain gear, only the gears in that gear are retained, and all gears in other gears are omitted.
- (4) The bearing will be replaced by a rigid unit, so the bearing is removed from the model.

(5) In order to improve the calculation efficiency and avoid the contact between the meshing gears, the gears are replaced by cylinders with equal division circles. The gear meshing force can be calculated and applied to the indexing cylinder.

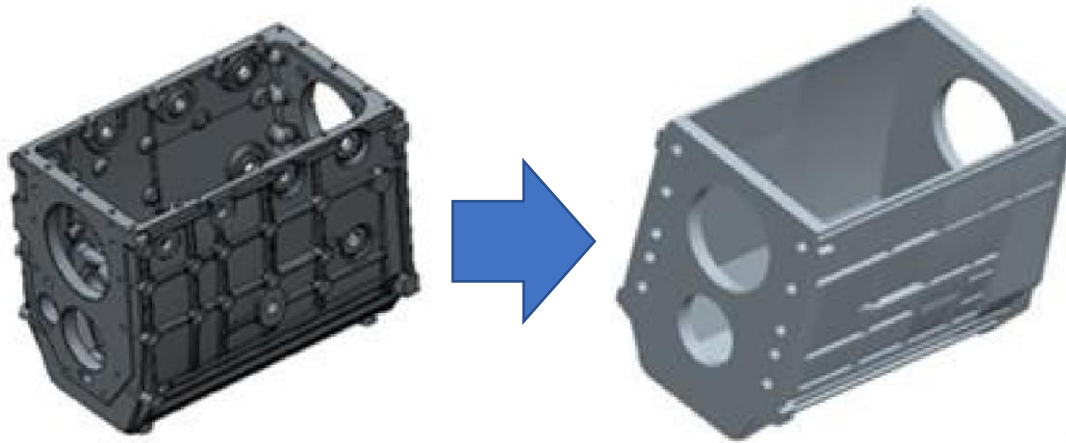


Figure 4-6 Gearbox model Simplified

4.4.2 Finite element analysis of gearbox transmission statics

In the previous article, the classic mechanical calculation of the transmission has been completed. In this section, the finite element analysis of the transmission statics will be performed with reference to the calculation results. The finite element analysis in this article is performed in ANSYS Workbench software. The specific steps are as follows:

(1) Material settings. The gearbox is made of cast iron with an elastic modulus of 1.1×10^5 MPa and a Poisson's ratio of 0.28. The gear and transmission shaft use structural steel with an elastic modulus of 2.1×10^5 MPa and a Poisson's ratio of 0.3.

(2) Rigid units are installed instead of bearings. REMOTE POINT is established on the surface of the box bearing hole and the surface of the corresponding drive shaft. There are three ways to connect the surface and REMOTE POINT: rigid connection, variable connection, and beam unit connection. Here a rigid connection is chosen.

(3) Contact settings. ANSYS Workbench can automatically identify contact parts and establish automatic contact. Then set the required contact algorithm and contact symmetry as needed. The bearing geometry model is not considered here, and there is no need to set the contact at the bearing. Gear meshing contact is also ignored. Just set the binding connection between the gear and the drive shaft. The contact algorithm

settings and other contact setting features are selected by software by default.

(4) Mesh division. Due to the complicated structure of the box, a tetrahedral mesh is used for free division. The gear cylinder can be divided by sweep, and the three transmission shafts are divided by multiple regions.

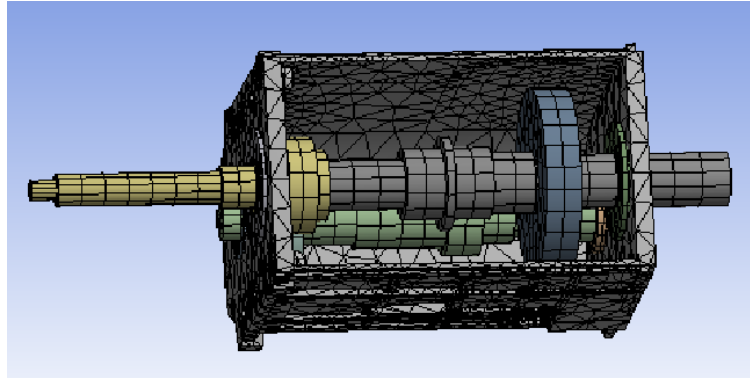


Figure 4-7 Gearbox Finite Element Mesh

(5) Load constraints imposed. The gearbox is tested on the test bench using a fixed front end. When it is actually installed on a vehicle, the front end is connected to the clutch, and the rear end is connected to the frame and fixed on the frame. In this paper, the restraint is loaded in a fixed manner during the test, that is, the bolt hole at the front end of the box is fixed and restrained. The load is applied as described in the previous model simplification. The meshing force of the gear during first gear transmission is calculated and directly applied to the gear cylinder of the equal division circle.

The calculation results are shown in the figure below:

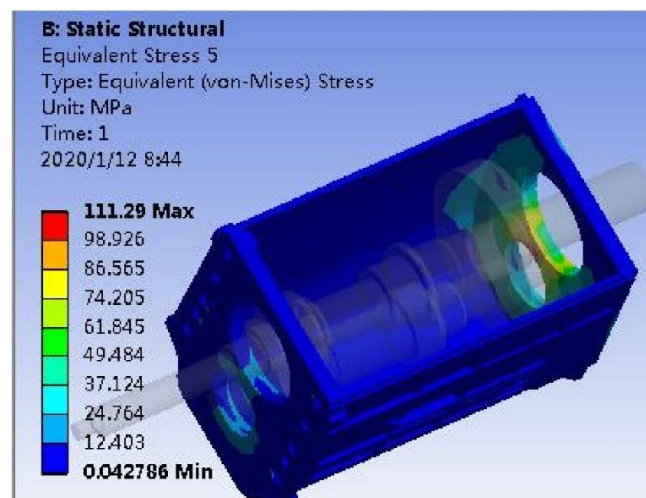


Figure 4-8 Gearbox stress distribution

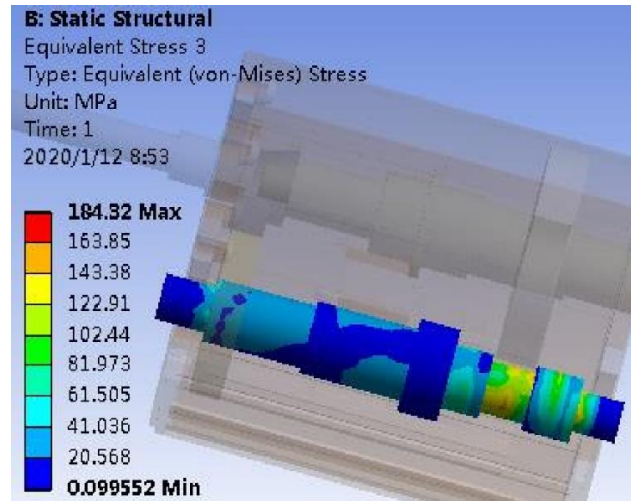


Figure 4-9 Intermediate axis stress distribution

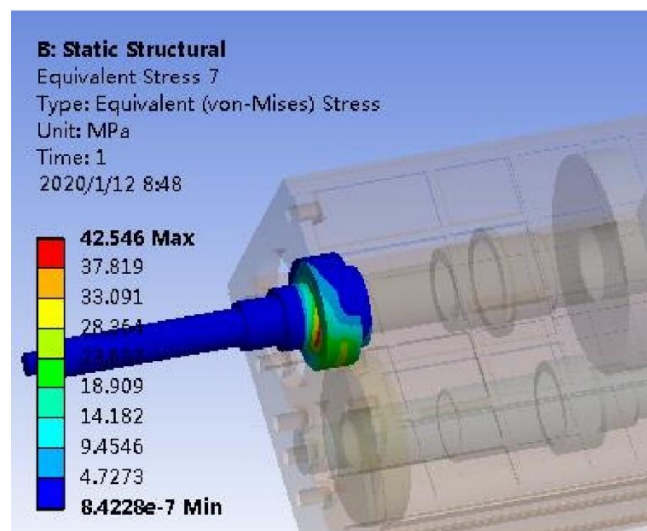


Figure 4-10 Input axis stress distribution



Figure 4-11 Output axis stress distribution

It can be seen from the stress distribution diagram that when the gearbox operates in

the first gear, the maximum stress occurs at the fixed position of the transmission shaft and gear. The maximum stress on the box is 111.29MPa, which occurs between the two bearing holes at the rear end of the box. Because the gear of the first gear is close to the rear end, the meshing force of the gear passes through the bearing to generate greater stress at the bearing holes of the box. A certain stress distribution also occurs at the front bearing hole, which is because the meshing gear is often close to the front bearing hole. When the gearbox is working, the meshing force of the meshing gear often acts on the front bearing hole of the box through the bearing, resulting in stress distribution. Certain stresses at the four bearing holes of the box are caused by normal work, and all are within the acceptable range, indicating that the strength of the gearbox box meets the requirements.

It can be seen from the stress distribution diagram of the three shafts that the strength of the three transmission shafts meets the transmission requirements. The maximum stress value of the intermediate shaft and the output shaft is relatively close to each other, and they occur near the meshing position of the first gear. The maximum stress value of the input shaft is close to the normal. The meshing gear position indicates that the stress distribution is correct. The stress distribution of the three shafts is larger the closer to the meshing gear, and the stress at the shaft shoulder is generally larger, which is in line with the normal stress distribution of the transmission shaft when it works.

Therefore, designers generally use small chamfers or rounded corners on the shoulder of the drive shaft to avoid stress concentration or change the stress distribution on the shaft shoulder, which prolongs the service life of the drive shaft. Not only does the transmission shaft have better strength, it also saves material to a certain extent and reduces the weight of the shaft.

4.5 Summery

This chapter firstly introduces the basic characteristics of the gearbox studied in this paper, calculates the gear meshing force according to the formula of gear meshing, and then establishes a force analysis model of the gearbox in the first gear transmission

condition. Combined with the finite element analysis software to analyze the model of the gearbox in the ideal state of error-free cooperation, the stress distribution diagram of each part of the gearbox was obtained, and the maximum stress occurrence position and the maximum stress value during the operation of the gearbox were observed and understood.

Chapter 5 Influence of bearing hole error on gearbox

5.1 Box tolerance constraint model

This article mainly analyzes the tolerances of the bearing holes of the gearbox. For the other parts of the case, such as the tolerances of the bolt holes and the roughness of the plane, etc. will not be considered for the time being.

The main tolerances of gearboxes are constraints of perpendicularity, circular runout, parallelism, and position. The tolerances limit the range of change in the centerline of gearbox bearing holes. The bearing hole centers for these tolerance constraints A mathematical model is established for the range of the line. Each center line is represented by two points. The range of movement of a center line is determined by the change of the position of the two points, and the tolerance of the bearing hole radius is not considered, so that the bearing hole can be determined under different errors. And its location.

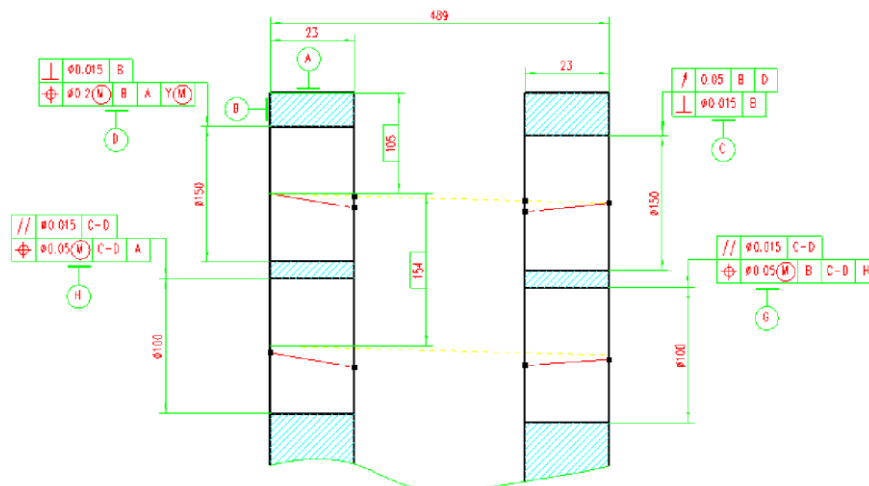


Figure 5-1 Bearing hole tolerances

The input shaft's shaft hole is marked with the perpendicularity based on the end face of the box B and the position degree with the top surface A as the reference, and the output shaft's shaft hole (the shaft hole at the upper right end of the box) is marked with the left end axis circular runout as the reference and perpendicularity with the B end

surface as the reference. After the two shaft holes are processed, the intermediate shaft bearing holes (two holes in the lower layer of the box) are machined based on the line CD of the center of the two shaft holes, where the left end of the intermediate shaft has parallelism and position based on CD. Degree tolerance, the right end bearing hole not only has parallelism and position degree tolerance constraints based on CD, but also the left end bearing hole has positional constraints.

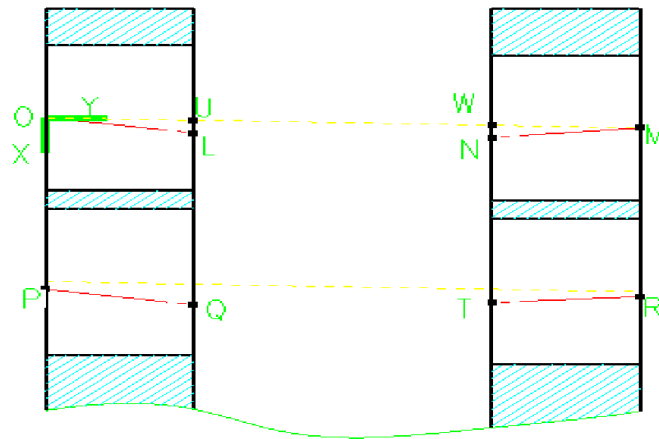


Figure 5-2 Coordinate system established in bearing bore

The two points O and L determine the center line of the first shaft hole. When there is an error, only the positions of these two points need to be calculated to determine the error position of the bearing hole when the error exists. Similarly, MN, PQ, and TR determine the positions of the four bearing hole centerlines in the box.

From the tolerances indicated, the processing order of the four bearing holes of the box is to first process a shaft bearing hole based on the top surface of the box, and then process the output shaft bearing hole based on the one shaft bearing hole and the top surface of the box. Then, use the two bearing holes processed above as a reference to make dimensional constraint on the intermediate shaft bearing hole, and use its center line (dashed line in the figure) as a reference to make parallelism constraint on the intermediate shaft bearing hole to process the bearings on the left and right ends of the intermediate shaft hole.

This article first considers the effect of the change of the error in the vertical plane on the overall gearbox. The position of the bearing hole in the gearbox box finally determines and affects the transmission of the gear installed in the box. From the

processing sequence, we can see that The dimensional tolerance constraint on the top surface of the box has no effect on the transmission of the gear, so the dimensional error of the center line of the shaft bearing hole from the top surface of the box is ignored, and a plane coordinate system is established at the center line of the one axis. Take the O point as the origin, the left end of the box as the X axis, and the ideal horizontal line as the Y axis to establish a plane rectangular coordinate system. In this way, all the points that determine the center line of the bearing hole can be represented in this coordinate system, and the tolerance constraints can be converted into constraints of mathematical expressions. Let α represent the degree of perpendicularity, β represent the degree of position, γ represents the degree of circular runout, p represents the degree of parallelism, and s represents the ideal center distance dimension.

(1) The position of the shaft hole of the one shaft and the top surface A is only to ensure the positioning during processing. After processing a shaft hole, the holes at other positions are processed based on this hole, so this position degree meshes with the gear on the upper and lower shaft There is no direct impact, and the position error at the shaft hole is not considered, but the perpendicularity has a certain effect on the gear shaft installed in the hole, so the tolerance of the perpendicularity tolerance of the shaft should be considered. The perpendicularity and parallelism in the vertical plane constrain the same. The one-axis perpendicularity tolerance constraint is expressed in the coordinate system as:

$$-\alpha \leq L_x \leq \alpha$$

(2) The output shaft bearing hole has a circular runout constraint relative to the center of the left shaft hole, that is, the centerline of this shaft hole should be within the range of circular runout error value, expressed in the coordinate system as:

$$\begin{aligned} -\frac{\gamma}{2} &\leq M_x \leq \frac{\gamma}{2} \\ -\frac{\gamma}{2} &\leq N_x \leq \frac{\gamma}{2} \end{aligned}$$

The bearing hole centerline also has certain perpendicularity tolerance constraints:

$$-\alpha \leq M_x - N_x \leq \alpha$$

(3) The left end of the middle axis has C-D as a reference, and there is a position constraint:

$$\begin{aligned} -\frac{\beta}{2} &\leq P_x - s \leq \frac{\beta}{2} \\ -\frac{\beta}{2} &\leq Q_x - (s + U_x) \leq \frac{\beta}{2} \end{aligned}$$

Here, the U point is the intersection point between the center line C-D on the left and right ends of the upper layer and the inner wall of the cabinet, and the intersection point W also exists on the right end. The parallelism constraint is approximately in:

$$-p \leq P_x - Q_x \leq p$$

(4) The constraint datum at the right end of the middle axis is a bit more. There are two datums for the position degree constraint. The position degree constraint based on the C-D line is expressed as:

$$\begin{aligned} -\frac{\beta}{2} &\leq R_x - (s + M_x) \leq \frac{\beta}{2} \\ -\frac{\beta}{2} &\leq T_x - (s + W_x) \leq \frac{\beta}{2} \end{aligned}$$

The position constraint based on the center of the left hole of the middle axis is expressed as:

$$\begin{aligned} -\frac{\beta}{2} &\leq R_x - P_x \leq \frac{\beta}{2} \\ -\frac{\beta}{2} &\leq T_x - P_x \leq \frac{\beta}{2} \end{aligned}$$

There is still a constraint on the parallelism of the bearing hole centerline:

$$-p \leq R_x - T_x \leq p$$

The tolerance constraints of the bearing hole of the box are expressed by the inequality in the mathematical model. For the figure that determines the center line of the bearing hole, the constraint expressions must be satisfied. The equal sign in the inequality is the case when the tolerance takes the limit.

5.2 Orthogonal test grouping for tolerance values

According to the previous introduction, the orthogonal test method is used to determine the tolerance of bearing hole characteristics and design test groups. The main tolerances

of the gearbox case are constrained by several tolerance factors such as perpendicularity, circular runout, parallelism, and positional degree. This article designs orthogonal tests using these four tolerance factors as test factors. Generally, the processing tolerance accuracy of automotive gearboxes is based on the use of 6 to 8 precision tolerances. Each tolerance accuracy has a limit value at that accuracy, and the limit value at each accuracy is a horizontal value. Each factor takes three levels of 6-8 precision, which constitutes the four-factor three-level orthogonal test used in this paper. Experimental design using L9 (3^4) orthogonal table ^[16].

Experimental group	Element 1	Element 2	Element 3	Element 4
1	1	1	1	1
2	1	2	2	2
3	1	3	3	3
4	2	1	2	3
5	2	2	3	1
6	2	3	1	2
7	3	1	3	2
8	3	2	1	3
9	3	3	2	1

Table 5-1 L9 (3^4) orthogonal table

According to the data, the limit values of the accuracy of the above four tolerance factors in the 6 ~ 8 level can be obtained ^[17]. The values of the four factors with different accuracy are shown in Table 5-2.

Tolerance Grade	Perpendicular ity [mm]	Positio n [mm]	Circular Runout [mm]	Parallelis m [mm]
6	15	20	25	15
7	25	31.5	40	25
8	40	50	60	40

Table 5-2 Tolerance value

The four factors and their respective levels are taken into the above-mentioned orthogonal test table to obtain nine groups of test plans. See table below

Experimental group	#1Perpendicularity [mm]	#2Position [mm]	#3Circular Runout [mm]	#4Parallelism [mm]
1	15	20	25	15
2	15	31.5	40	25
3	15	50	60	40
4	25	20	40	40
5	25	31.5	60	15
6	25	50	25	25
7	40	20	60	25
8	40	31.5	25	40
9	40	50	40	15

Table 5-3 Orthogonal test design

According to the above tolerance expression, it is known that the tolerance of the bearing hole is represented by each point that determines the center line of the bearing hole, and the position of each point in the coordinate system can be determined by the tolerance accuracy. In this paper, the limit values under each tolerance accuracy are analyzed during the analysis, which can effectively verify the rationality of the tolerance values. So, each point in the coordinate system is expressed by a mathematical formula.

$$L_x = \alpha$$

$$M_x = \frac{\gamma}{2}$$

$$N_x = M_x - \alpha = \frac{\gamma}{2} - \alpha$$

$$Q_x = U_x + s + \frac{\beta}{2}$$

$$U_x = \frac{23}{489} M_x$$

$$P_x = Q_x - p$$

$$R_x = M_x + s - \frac{\beta}{2}$$

$$T_x = R_x + p$$

Where U is the intersection of the inner wall of the left end of the box with the connection line of an axis and the output shaft, as shown in Figure 4-2. s represents the ideal dimension value between the upper and lower transmission shafts, that is, the center distance without tolerance. When each tolerance factor obtains different values according to the orthogonal test design table, the position of each point in the coordinate system can be obtained by substituting in the above formula, and then the position parameter of the point can be changed in Pro / E to obtain the test Set of error geometric models.

5.3 Analysis of results based on orthogonal experiments

5.3.1 Influence of tolerance on bearing maximum stress

After calculation and analysis, the experimental analysis results can be obtained after the solution is completed. Tolerance test groups obtained in accordance with the orthogonal test method in Section 4.2 above, take the corresponding tolerance values of each group and analyze them according to the finite element analysis step to obtain the change in contact stress at the bearing under different tolerance values. Taking the test analysis results of the first group of the test group as an example, the vertical tolerance in this group is 15um, the position is 20um, the circular runout is 25um, and the parallelism is 15um.

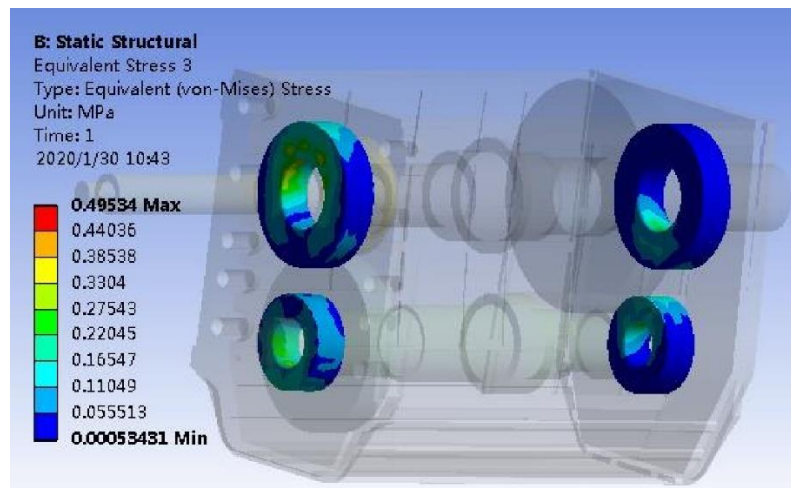


Figure 5-3 Four bearing stress

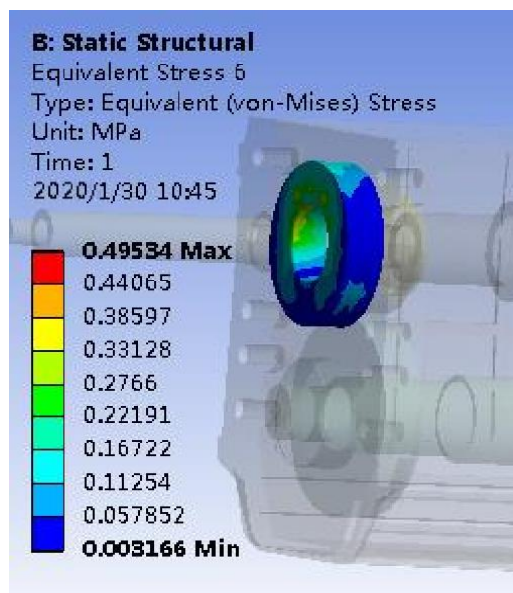


Figure 5-4 Input shaft bearing stress

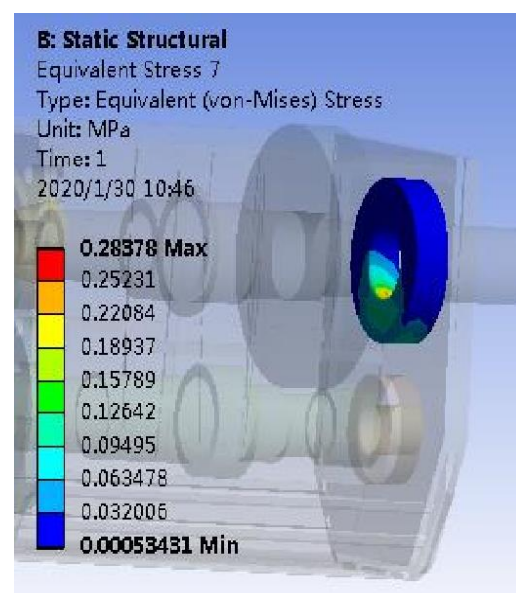


Figure 5-5 Output shaft bearing stress



Figure 5-6 Intermediate shaft bearing stress (L)

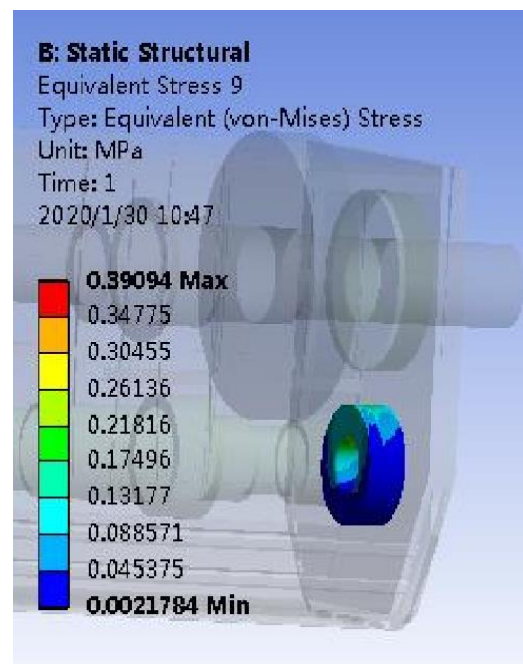


Figure 5-7 Intermediate shaft bearing stress (R)

In the stress distribution diagram, it can be seen that when the gearbox bearing hole adopts the tolerance of the first group, the maximum stress after assembly is 0.495 MPa, which occurs at a shaft bearing hole. The maximum bearing stress at both ends of the intermediate shaft is relatively close, and the stress distribution is also symmetrical. Because the intermediate shaft is a complete shaft, but the input shaft and the output shaft are not one, they are connected by bearings, so the bearing stress at the input shaft's shaft hole and the maximum bearing stress at the output shaft's shaft hole are different larger.

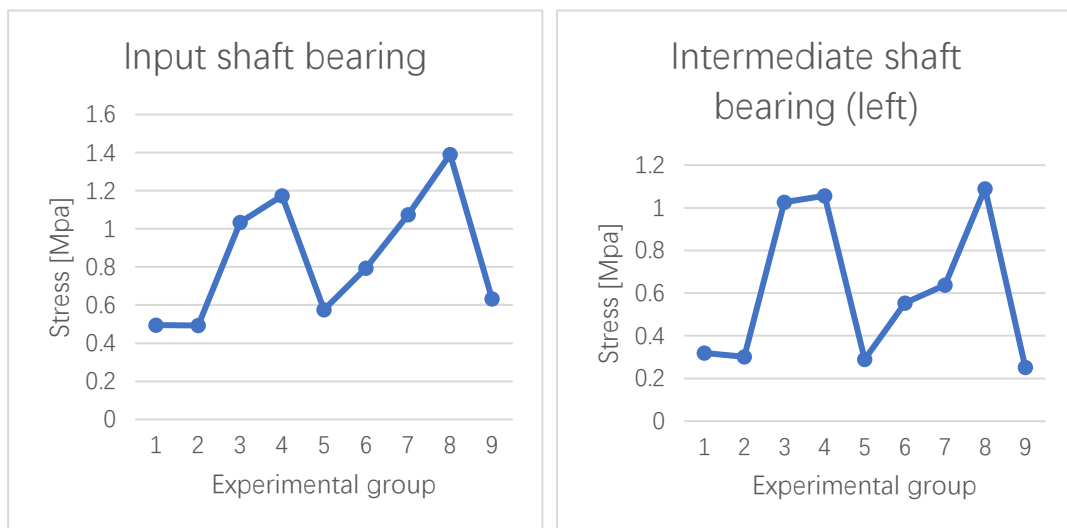
After the tolerance values of the other eight groups are brought into the parameterized geometric model, the finite element calculation analysis is performed to obtain the maximum stress values at the four bearings.

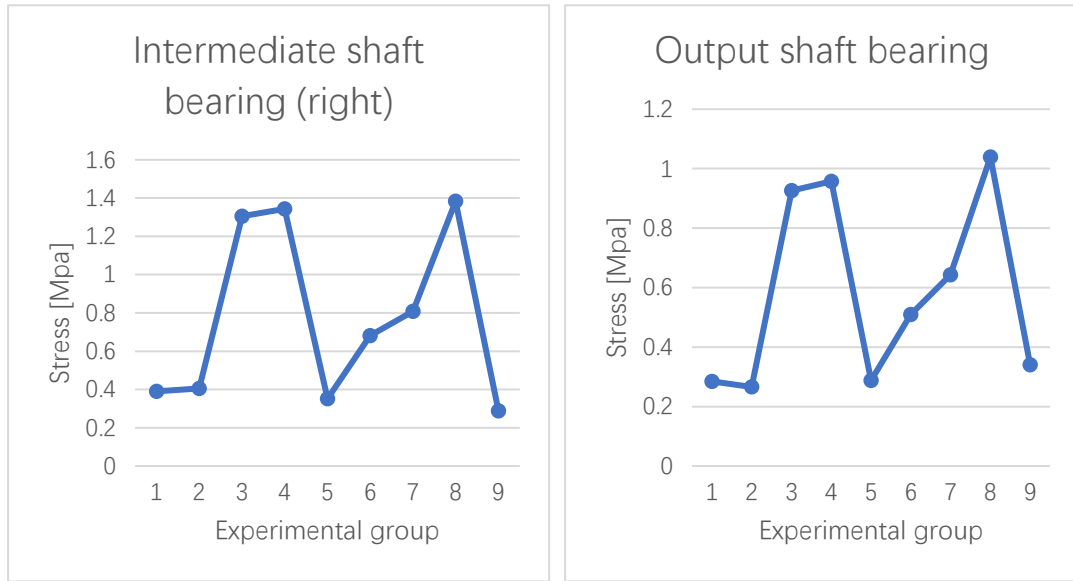
Experimental group	Input shaft bearing	Intermediate shaft bearing (left)	Intermediate shaft bearing (right)	Output shaft bearing
1	0.495	0.319	0.391	0.284

2	0.493	0.302	0.405	0.266
3	1.034	1.027	1.306	0.926
4	1.174	1.057	1.343	0.957
5	0.576	0.289	0.351	0.288
6	0.794	0.554	0.681	0.51
7	1.075	0.637	0.809	0.643
8	1.391	1.09	1.383	1.039
9	0.632	0.252	0.288	0.34

Table 5-4 Bearing stress orthogonal test results

In order to clearly show the changes of the stress of each bearing in the nine groups of tests, the stress value of each bearing is converted into a line chart to reflect the fluctuation of the maximum stress value with the tolerance value.





From the above line graph, it can be seen that the maximum stress of the four bearings fluctuates with the difference of the error value, but all fluctuates within 2MPa, that is, the maximum stress of the gearbox bearing does not change with the tolerance value. There are a wide range of changes.

5.3.2 Analysis of the maximum stress range at the bearing

The results of the orthogonal test method can be analyzed using multiple result analysis methods, because there are more than one method for measuring results, and there are often multiple indicators. Different results analysis methods should be used for different actual test situations. The results of orthogonal test methods mainly adopt analysis of variance method and range analysis method. In this paper, a range analysis is performed for the above-mentioned nine sets of test results. The specific analysis of the change in bearing stress with error factors at each position shall adopt the result analysis method of orthogonal test, that is, by taking a certain tolerance value for a certain tolerance. The average value of the maximum value of bearing contact stress in several sets of tests is compared to estimate the effect of this tolerance factor on the maximum stress at the bearing contact when taking this tolerance value, and the same tolerance factor is used to take the bearing contact when different tolerance values. The range of the average value of the maximum stress value is used to estimate the degree of influence of the tolerance factor on the test result.

Take the effect of the maximum stress value of the bearing at the first shaft bearing hole as an example to analyze the results. If you observe the effect of the vertical value of 15um on the maximum stress of the bearing at the first shaft bearing hole, it is found that a test of 15um vertical In the 1-3 test groups, the average of the results of the three groups of tests is:

$$K_1 = \frac{y_1 + y_2 + y_3}{3} = 0.674 \text{ MPa}$$

where y represents the maximum bearing stress at the first shaft bearing hole in each group of test results;

The test with the same perpendicularity of 25um appeared in the 4 ~ 6 test groups. The average of the results of the three groups of tests is

$$K_2 = \frac{y_4 + y_5 + y_6}{3} = 0.848 \text{ MPa}$$

The test with a perpendicularity of 40um appeared in the 7 ~ 9 test groups. The average of the results of the three groups of tests is

$$K_3 = \frac{y_7 + y_8 + y_9}{3} = 1.033 \text{ MPa}$$

From these three groups of average values, it can be observed that the test result is best when the perpendicularity tolerance is 15um, because the average value of the maximum bearing stress at the first shaft is the smallest. It is also possible to calculate and observe the effects of the values of other tolerance factors on the maximum bearing stress at a shaft hole. The four tolerance factors of perpendicularity, position, circular runout and parallelism are calculated according to the above calculation method to obtain the table

	Perpendicularity [mm] α	Position [mm] β	Circular Runout [mm] γ	Parallelism [mm] P
K1	0.674	0.915	0.893	0.568
K2	0.848	0.82	0.766	0.787
K3	1.033	0.82	0.895	1.2
Range	0.359	0.095	0.126	0.632

Table 5-5 First shaft bearing hole average & range

It can be seen from the table that the best combination of the four tolerance factors should be $\alpha_1\beta_2\gamma_2p_1$, that is, the perpendicularity α is 15um, the position β is 31.5um, the circular runout γ is 40um, and the parallelism p is 15um. In this way, the best combination of experimental factors is obtained. The calculation method of the range R is the difference between the maximum average value and the minimum average value of the same tolerance factor, which reflects the degree to which the tolerance factor affects the results when different levels are used, such as the extreme tolerance of perpendicularity at the bearing hole of a shaft $R = 0.359\text{MPa}$.

The same method can be used to observe that the bearings at other bearing holes are affected by tolerance factors and the optimal combination of tolerance factor values when the maximum stress value of each bearing is minimized.

	Perpendicularity [mm] α	Position [mm] β	Circular Runout [mm] γ	Parallelism [mm] P
K1	0.549	0.671	0.654	0.287
K2	0.633	0.56	0.537	0.498
K3	0.66	0.611	0.651	1.058
Range	0.111	0.11	0.117	0.771

Table 5-6 Intermediate shaft bearing hole (left) average & range

	Perpendicularity [mm] α	Position [mm] β	Circular Runout [mm] γ	Parallelism [mm] P
K1	0.7	0.848	0.818	0.343
K2	0.792	0.713	0.679	0.632
K3	0.827	0.758	0.822	1.344
Range	0.127	0.135	0.143	1.001

Table 5-7 Intermediate shaft bearing hole (right) average & range

	Perpendicularity [mm] α	Position [mm] β	Circular Runout [mm] γ	Parallelism [mm] P
K1	0.492	0.628	0.611	0.304
K2	0.585	0.531	0.521	0.473

K3	0.674	0.592	0.619	0.974
Range	0.182	0.097	0.098	0.67

Table 5-8 Output shaft bearing hole average & range

The optimal combination schemes for minimizing the maximum bearing stress in four places are consistent, and they are all $\alpha_1\beta_2\gamma_2p_1$. Although the bearings in different places are observed separately, but the results are obtained in the same analysis process, the best combination of the overall test results is $\alpha_1\beta_2\gamma_2p_1$, indicating that the test result analysis method is correct.

From the analysis of the calculation results of the extreme difference, it is seen that the four error factors have the greatest impact on the overall structure is parallelism. In addition to the parallelism that affects the overall test results, each bearing is affected differently by different error factors.

The most influential error factor at the input shaft bearing hole is perpendicularity, and the factor that has the biggest effect on the stress change of the intermediate shaft is the circular runout at the output hole. This is related to the method of modeling, because the position of the intermediate shaft bearing hole is the position of the center line of the output bearing hole is determined based on the circular runout at the output bearing hole. The factor that has the greatest effect on bearing stress at the output bearing hole is the parallelism at the input shaft. This is also because the position of the center line of the output hole is It is established based on the determination of the centerline position of the input shaft bearing hole, and it also has a perpendicularity error.

Therefore, the overall parallelism tolerance should be strictly controlled, and since the degree of influence of each error factor is different, different tolerance control strategies are required for different positions.

5.4 Summary

This chapter studies the main tolerance factors of box-type parts and the influence of each factor on the gearbox after assembly, and then establishes a mathematical model of tolerance constraints based on the gearbox's tolerance labeling, determines the main

analyzed tolerance factors, combines orthogonal test methods and After consulting the relevant manuals, the nine sets of analysis tests and the test tolerance values of each group were determined. After that, a geometric model was established and the bearing hole position was parameterized and connected with the parameterization module of the finite element analysis software ANSYS Workbench to form a complete parameterized analysis Process. Nine sets of test analysis results were obtained by changing the test values of each group without changing other analysis conditions, making the test results highly comparable. Finally, the results of the finite element analysis were analyzed in accordance with the orthogonal test method. The degree of influence of the four tolerance factors considered on the stress value distribution at the four bearing holes was compared. It was found that the tolerance factor that had the greatest influence on the overall test results was the parallelism tolerance of the two axes, and the degree of influence of the tolerance factors on each bearing hole They are also not the same, that is, the degree of control of the tolerance factor is different at each bearing hole, and corresponding tightening control strategies must be adopted.

Chapter 6 Conclusion

This paper focuses on the development of tolerance analysis, and comparatively analyzes the main tolerance research methods, including statistical method, range method and Monte Carlo method. Most of the methods did not carry out simulation analysis, and did not consider factors such as material characteristics and installation deformation. Based on this, I put forward my own analysis method of geometric assembly with tolerance combined with orthogonal test and finite element method.

The method of dealing with contact problems in the finite element method is studied. Taking simple shaft hole matching with a certain position deviation as an example, the initial contact position is adjusted using the finite element method, the process of error assembly is simulated, and the change of the contact stress due to the existence of the error is obtained. Explain the correctness and feasibility of this method.

In-depth study of the processing and measurement process of the gearbox, and the effects of various tolerances on the gearbox on the gearbox are explained in detail. Combined with the tolerance marking method, the main tolerance factors at the bearing holes of the box are selected: perpendicularity, circular runout, parallelism and position as the main research objects. The contact stress of the gear shaft and the bearing is taken as the research target. Impact of transmission mechanical properties.

The following conclusions are obtained:

(1) The tolerance value is relatively small compared to the overall size of the box. Although it will not cause a wide range of contact stress between the bearing and the shaft, the maximum value of the contact stress on the bearing will still be certain when the four tolerance factors are loosely controlled and stricter. The fluctuation of the amplitude, from the perspective of long-term use, the tightness of the tolerance factor control affects the wear of the bearing to a certain extent, thereby affecting the service life and transmission efficiency of the gearbox.

(2) Of the above four tolerance factors, the greatest degree of influence on the maximum contact stress at the four bearing holes of the gearbox is the parallelism

tolerance. At the same time, in addition to parallelism, other tolerance factors have an effect on the maximum contact stress of the four bearing holes. It varies with different bearing holes. Therefore, different tightening control strategies must be adopted for different tolerance factors at different bearing holes.

Reference

- [1] Øyvind Bjørke. Computer-aided tolerancing[M]. S.l. Tapir, 1978
- [2] A.G. Requicha. Toward a Theory of Geometric Tolerancing. The International Journal of Robotics Research, 1983
- [3] Weill R, Clément A, et al. Tolerancing for function[J]. CIRP Annals-Manufacturing Technology, 1988
- [4] Hillyard R, Braid I. Characterizing non-ideal shapes in terms of dimensions and tolerances[C]. ACM SIGGRAPH Computer Graphics, 1978
- [5] Salomons O W. Computer support in the design of mechanical products: constraint specification and satisfaction in feature based design for manufacturing[M]. 1995
- [6] Krulikowski, Alex. Self-Study Workbook. Westland.1997
- [7] Qi Yi. DPS data processing system for practical statistics[M]. Science Press, 2002
- [8] B.H. Wang, Y. Jin, Y.G. Luo. Parametric optimization of EQ6110HEV hybrid electric bus based on orthogonal experiment design. Int J Auto Tech-Kor, 11 (2010),
- [9] G.R. Liu, S.S. Quek. The Finite Element Method: A Practical Course[M]. Elsevier Ltd, 2004
- [10] Xucheng Wang. Finite element method. Tsinghua University Press, 2003
- [11] Guozhi Zhang. Finite Element Analysis and Research in Tolerance Design of Miniature Clockwork[J]. Mechanical Design and Research. 2007
- [12] ANSYS Help Document. The ANSYS. Inc, 2019
- [13] Lianggui Pu. Mechanical Design[M]. Sinopec Press, 2012
- [14] Xiaofeng Wang. Automotive chassis design [M]. Tsinghua University Press, 2010
- [15] Zhilin Wei. Check calculation of automobile transmission bearing life [J]. Journal of Guangxi University of Technology, 2000
- [16] Yun Yan. Experiment design and data processing [M]. Chemical Industry Press, 2008
- [17] Daxian Cheng. History of Mechanical Technology (3) Proceedings of the Third China-Japan International Conference on History of Mechanical Technology. 2002