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Dynamic Analysis and Fatigue Life Evaluation of Single Herringbone Planetary Gear System with Crack Fault

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ABSTRACT

The herringbone planetary gear transmission is widely used due to its exquisite compact structure, excellent load-bearing capacity, and stable transmission characteristics. However, root crack is one of the most common failure modes of gears. A cracked herringbone planetary gear system is taken as research object. The sun gear with crack fault is studied from the aspect of modal and transient dynamic analysis, and the related results are compared with the healthy gear. Based on the transient dynamic analysis of the herringbone teeth with cracks, a fatigue life evaluation is conducted. Firstly, the results show that no significant difference between the natural frequencies and main vibration modes of healthy gears and faulty gears could be observed. Secondly, through transient dynamic analysis, it can be found that, during the rotation process of gears and with the increase of crack depth, the stress at the crack area increases gradually. As the crack penetration length increases, the stress at the crack endpoint of the gear increases almost linearly. Additionally, by comparing cracks with different depths and penetration lengths, the depth of the crack has a greater impact on the stress of the gear compared to the crack penetration length. Finally, based on the fatigue life analysis, the depth of the crack has a significant impact on the gear life; even if the depth of the crack is not very large, it will still have a significant impact. The related research is valuable and could guide the design and optimization of herringbone planetary gear system from the point of practical application.

Keywords: *Fatigue, FEA, herringbone tooth, planetary gear system, tooth crack*

I. Introduction

The herringbone planetary gear transmission system uses a symmetrical configuration of a pair of helical gears with opposite rotation directions, which has the characteristics of greater overlap and smoother transmission. Therefore, it is widely used in high-speed and heavy-duty situations where noise characteristics need to be considered. However, in harsh working environments such as excessively complex structures, long-term fatigue loads, and high-power densities, gears often encounter problems and failures, leading to the failure of the entire transmission system, with root cracks being the most common form. The dynamic analysis of a cracked herringbone planetary gear system is an innovative and difficult point in current research. The research results can be applied to the design process of planetary transmission systems, clarify their fault mechanisms, and optimize transmission performance, which has strong theoretical significance and engineering application value.

A computational framework was established to model linear root crack propagation in pinions, utilizing an enhanced potential energy approach to determine time-dependent

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engagement stiffness in mating gear pairs [1]. The vibration characteristics of perforated gear systems were analyzed, with particular emphasis on how rim-traversing crack paths influence dynamic stiffness variations and oscillatory behaviors [2]. A theoretical formulation for spur gear time-varying stiffness was created, incorporating dual-tooth contact mechanics, updated fillet-foundation stiffness, nonlinear interfacial compliance, and spalling defect interactions [3]. A generalized tooth profile equation reflecting manufacturing constraints was introduced to refine meshing stiffness predictions for fractured gear teeth [4]. An integrated analytical methodology was developed to assess mesh stiffness through combined contributions from flexural, shear, axial compression, Hertzian contact, and foundation deformation mechanisms [5]. A three-degree-of-freedom spur gear model incorporating backlash nonlinearities and bearing clearance effects was constructed to evaluate stiffness alterations across varying crack dimensions [6]. Finite element validation confirmed the analytical algorithm's accuracy in modeling elastic deformations (contact, bending, compression), while parametric studies revealed helical gear stiffness dependencies on crack geometry (length/depth/angle) [7]. A coupled locomotive-track dynamic system incorporating power transmission dynamics was formulated, integrating both gear root crack-induced stiffness fluctuations and rail irregularity inputs for vibration prediction [8]. Governing equations for parallel-axis gear transmissions were derived, enabling systematic investigation of operational parameters (rotational speed, damping coefficients, material modulus) on system dynamics [9]. Transition curve functionality was incorporated into a potential energy-based stiffness model, enabling quantitative assessment of ten distinct crack length scenarios through full-profile analysis [10]. Planetary gear dynamics were simulated using lumped-parameter modeling with adaptive meshing stiffness parameters derived from a modified energy formulation applicable to both intact and damaged teeth [11]. System eigenproperties (modal classifications, eigenvalue multiplicities) were determined through lumped-mass dynamic equation formulations [12]. Simulation-based quantification methods demonstrated consistent stiffness degradation across multiple tooth failure modes, establishing transmission fault-stiffness correlations [13]. An energy decomposition principle was proposed, expressing cracked gear potential energy as the superposition of pristine component energy and crack-induced surface energy liberation [14]. A novel dynamics formulation accounting for non-uniform crack distribution along tooth widths improved bearing excitation predictions through asymmetric mesh force modeling [15]. Acoustic signature analysis combined with information entropy metrics and machine learning techniques was implemented for obstruction detection [16]. Coordinated deformation analysis elucidated cracked tooth load distributions, while elastohydrodynamic lubrication modeling revealed crack-induced tribological alterations [17]. A coupled tribo-dynamic spur gear model synthesized lubricant film stiffness/damping characteristics with crack-induced mechanical compliance changes [18]. Finite element investigations of tapered cantilever beam deflections under tip loading provided foundational insights relevant to gear root stress analysis methodologies [19]-[21].

Although many studies related to gears with cracks were conducted, the research on dynamic and fatigue analysis about herringbone planetary gear transmission systems with cracks has not been achieved till now. To study the dynamic characteristics and fatigue life evaluation of single herringbone planetary gear transmission with crack fault, this paper constructed a three-dimensional model of a parameterized herringbone planetary gear transmission system and deeply explores the impact of crack faults at the root of the herringbone sun gear on the performance of the gear system. A static analysis was conducted on a cracked herringbone planetary gear system, and the effects of different penetration depths of tooth root cracks in the tooth width direction and tooth profile direction on the equivalent stress and total deformation of the gear were studied. Modal and transient dynamic analysis was conducted on cracked herringbone planetary gear systems, and the stress situation at the crack tip was studied. Based on the dynamic analysis of cracked herringbone gears, fatigue life analysis was conducted on faulty herringbone gears. The above-mentioned research could guide the optimization design of herringbone planetary gear system from the point of practical application.

II. Methods

2.1 Physical Methods

The crack fault is normally located on the root of tooth and has different parameters including crack depth and penetration length. These parameters not only directly affect the elastic deformation characteristics of the gear teeth under load, but also significantly affect the distribution of bending stress at the tooth root, which may lead to the formation of root cracks. Therefore, from the perspectives of academic research and engineering practice, precise control measures need to be taken to ensure the accuracy of these parameters. In order to ensure the overall performance and service life of the gear teeth and accurately describe the geometric characteristics of the gear end face profile, a method that combines the gear meshing principle with the tooth profile normal method is adopted. By analyzing the geometric relationships of different conjugate point positions in depth, a comprehensive tooth profile equation covering involute and root transition curves was derived. This method not only ensures the accuracy and completeness of the tooth profile equation but also reflects the importance of precise geometric shape control in gear design, providing powerful mathematical tools for precise manufacturing and performance analysis of gears. The parametric model of herringbone tooth planetary gear system is established based on a group of parameters based on a kind of practical planetary gear system, and the 3D model (by ANSYS software) is shown in Figure 1 and Figure 2.

In order to study the influence of cracks on the dynamics and fatigue life of herringbone teeth, it is necessary to model the cracks. According to existing research, cracks can propagate in two directions, including crack depth and crack length, as shown in Figure 3(a) and Figure 3(b). Figure 3(a) shows the propagation of cracks in the depth direction. It can be seen from the figure that the propagation in the depth direction includes two parameters, namely depth and propagation angle. According to research, the propagation angle β is generally set at 45 degrees. After the crack is generated, it will follow a predetermined path, that is, from point A to point B, and then further to point C. This propagation process will continue until complete fracture. Figure 3(b) shows the propagation of cracks in the length direction. This article will study the parameters of crack depth and length in two directions.

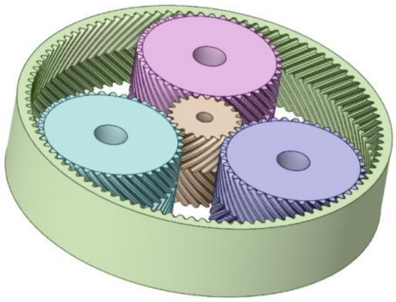


Fig.1. Herringbone tooth planetary gear system model

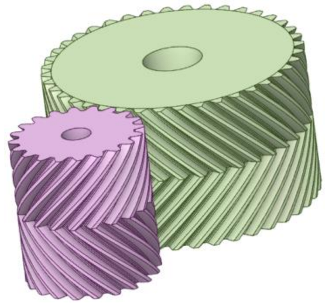
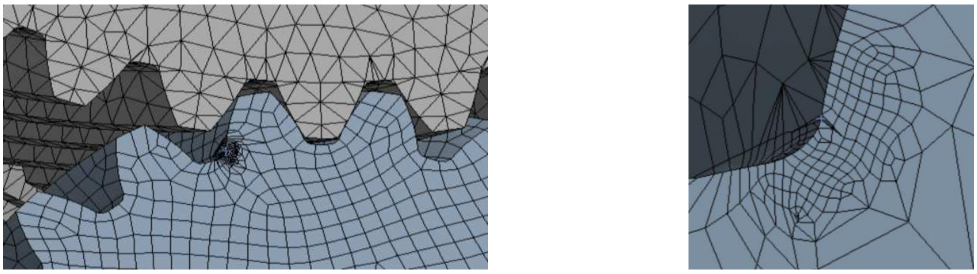


Fig.2. Sun gear-planetary gear model

(a) (b)

Fig.3. Theoretical model of crack propagation. (a) Crack propagation in depth direction; (b) Crack propagation in length direction

The meshing model of planetary gear system with cracks is shown in Figure 4, and Figure 4(a) shows the local view of sun gear and planetary gears, and Figure 4(b) shows the meshing area of cracks, and the meshing grids at crack area is refined significantly.



(a) (b)

Fig.4. Local crack view with meshing. (a) Local model view; (b) Local crack view

2.2 Evaluation Flow Chart

Figure 5 shows the dynamic and fatigue life evaluation flow chart and algorithm for herringbone planetary gear system with crack fault.

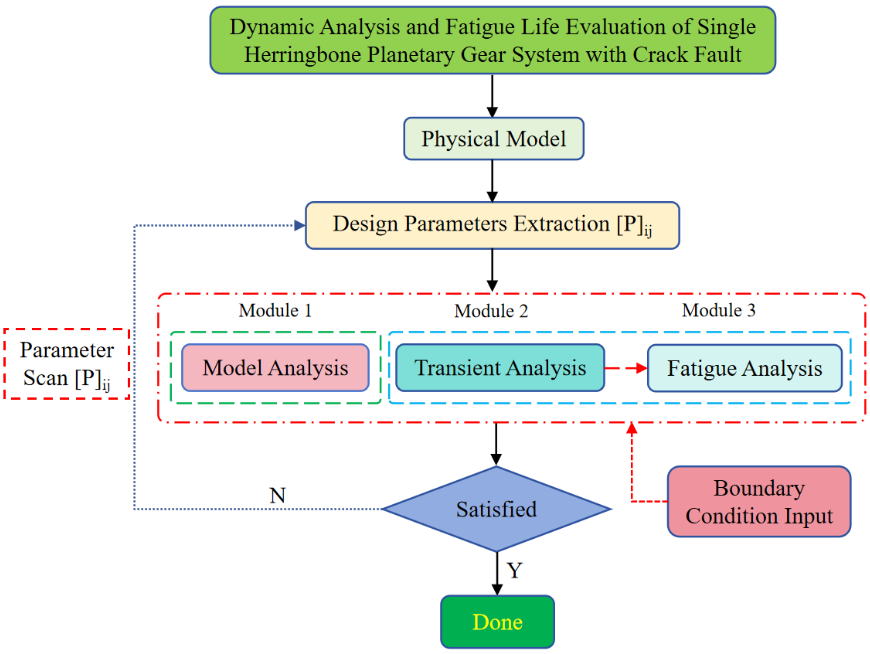


Fig.5. Dynamic and fatigue life evaluation flow chart and algorithm for herringbone planetary gear system

As can be seen from Figure 5, the design parameters of the herringbone planetary gear system with crack fault need to be extracted and set as the variables to conduct the optimization analysis. Besides, the boundary conditions for each analysis should be inputted correctly. The optimized parameter group could be obtained after iterative calculations.

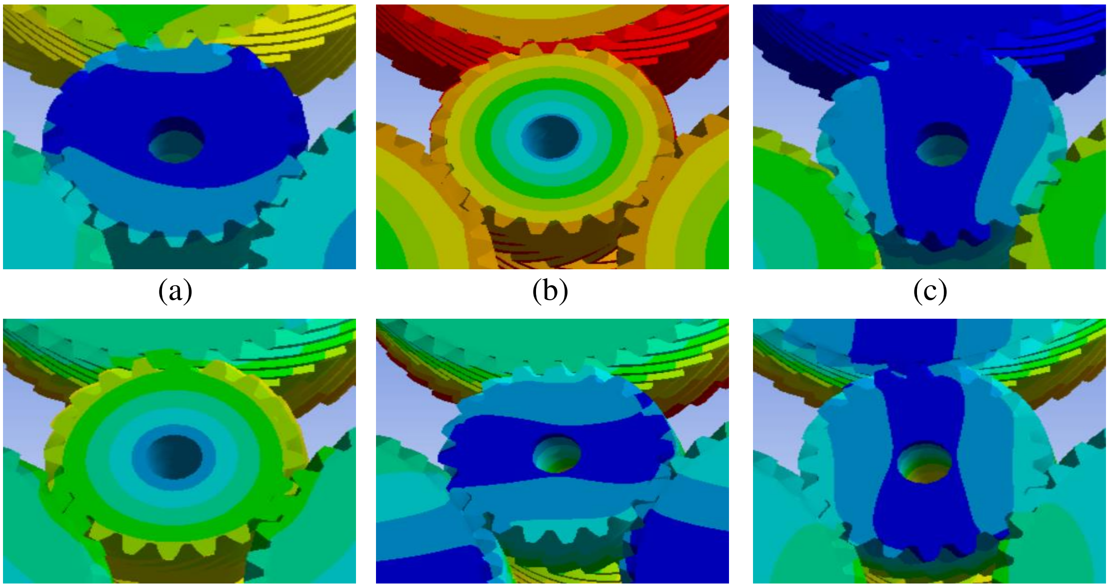
III. Results and Discussions

1. Modal Analysis

In order to get the natural frequency and vibration shape of herringbone tooth planetary gear system, the boundary condition is set according to the practical application, including the fixed ring gear and rotated degree of sun/planet gear. By calculation, the first six-order natural frequency and vibration shape of healthy gear and the gear with crack fault are obtained.

Besides, it should be noted that, as the complex structure shape, the tetrahedron element is adopted. At the same time, based on the consideration of time cost and result accuracy, several rough calculations under the different element sizes are conducted, and 2 mm element dimension is chosen finally for this model, which could lead to a converged result and no better result could be obtained when smaller element is set. The models in section II are meshed with the same size. Furthermore, related boundary condition is set accordingly and the contacts between gears are set to be bounded, and the linear calculation is conducted.

Figure 6 shows the vibration shape of the healthy gear, and Figure 7 shows the curve of the first six-order natural frequency. As can be seen from Figure 6, the curve behaves linear generally from the first to fifth order but increases dramatically in the sixth order. By the same method, the first six natural frequencies of herringbone tooth planetary gear systems with different crack depths are obtained and shown in Table 1.



(a) (b) (c)

(d) (e) (f)

Fig. 6. The first six modal shapes of healthy gear. (a) First order; (b) Second order; (c) Third order; (d) Fourth order; (e) Fifth order; (f) Sixth order.

Additionally, the natural frequency of healthy gear and gear with different crack penetration lengths are compared as shown in Figure 8. It could be found that the difference in natural frequency values between healthy gears and faulty gears is not significant, which has similar trends to the analysis in reference [12].

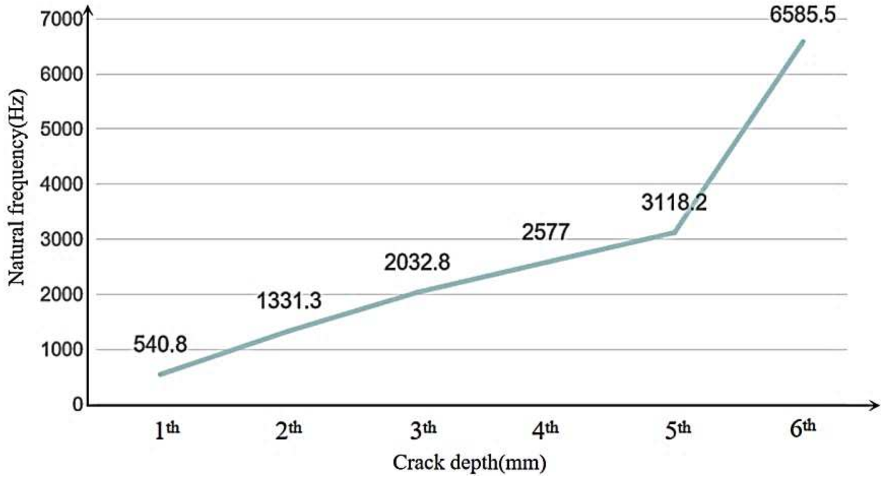


Fig.7. Natural frequency of healthy gear

Table 1. Natural frequency (Hz) of gears with different crack depth (mm)

Order	q=10%	q=30%	q=50%	q=70%	q=90%
1th	502.5	504.2	510.4	496.5	502.1
2th	1300.7	1301.5	1299.6	1303.3	1305.9
3th	2038.4	2037.5	2036.5	2036.8	2038.4
4th	2534.9	2536.4	2549.3	2561.4	2568.9
5th	3140.1	3145.8	3150.8	3149.1	3155.2
6th	6250.2	6259.1	6262.5	6258.4	6240.6

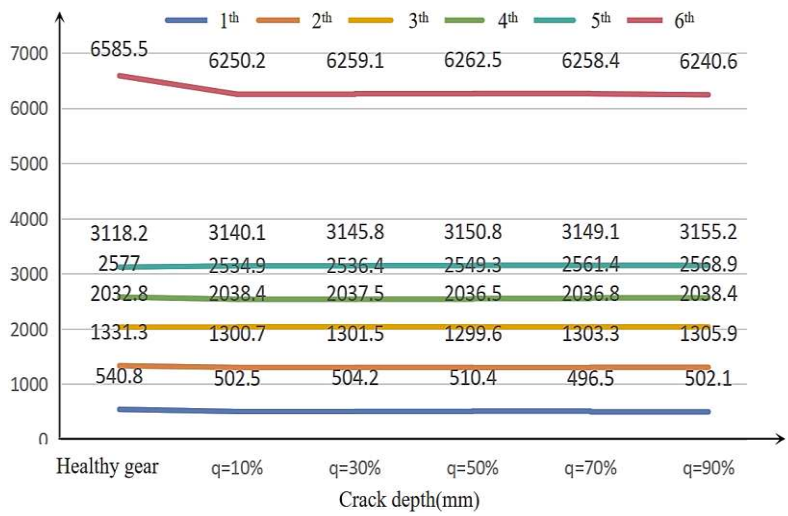
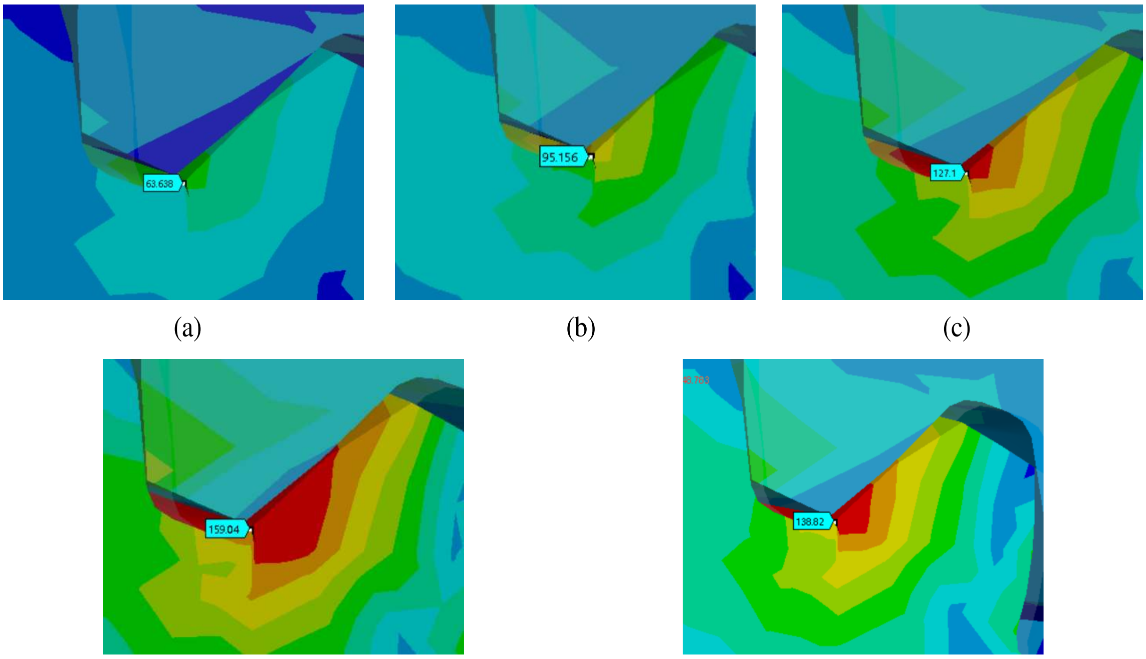


Fig.8. Natural frequency (Hz) curves of the first 6 orders for healthy gears and different crack depths

2. *Transient Dynamics Analysis Under Different Crack Parameters*

A. Transient Dynamics Under Different Crack Depth

The transient dynamics of gear system could simulate the practical condition more precisely. Different from the modal analysis in section III(1), the contacts between gears are set to be frictional under normal lubrication conditions and the friction coefficient is set to be 0.1. This setting makes the FEA (Finite Element Analysis) calculation become nonlinear and causes the high time consuming of calculation. Additionally, the sun gear and planet gear are extracted separately to reduce the scale of the computing and this setting will not affect the results compared to the calculation of a fully loaded model. The stress value at the tip of the crack and different times, when crack depth equals 10% q, are obtained in Figure 9. By the way, the unit of the value shown in Figure 9 is MPa.



(a) (b) (c)

(d) (e)

Fig.9. Stress distribution (MPa) at crack tip and crack depth of 10% q. (a) 0.016s; (b) 0.0165s; (c) 0.017s; (d) 0.0175s; (e) 0.018s

Besides, to find out the rules of stress level under the parameter of different crack depth, a group of cases are conducted and calculated, the detail results of different crack depth and different time are shown in Table 2, and the data is curved and shown in Figure 10. Based on Figure 10, it could be found that the stress level is increased with the crack depth, and reaches the max value at the time of 0.0175s, after that the stress decreases gradually.

Table 2. Stress level at different crack depth

Crack depth (mm)	Time (s)				
	0.016	0.0165	0.017	0.0175	0.018
	Stress (MPa)				
10% q	63.6	95.1	127.1	159.0	138.8
30% q	78.5	118.4	158.6	198.4	173.8
50% q	78.6	117.9	157.1	197.6	174.2
70% q	84.4	127.2	169.0	211.1	186.7
90% q	90.4	136.2	180.8	222.3	199.6

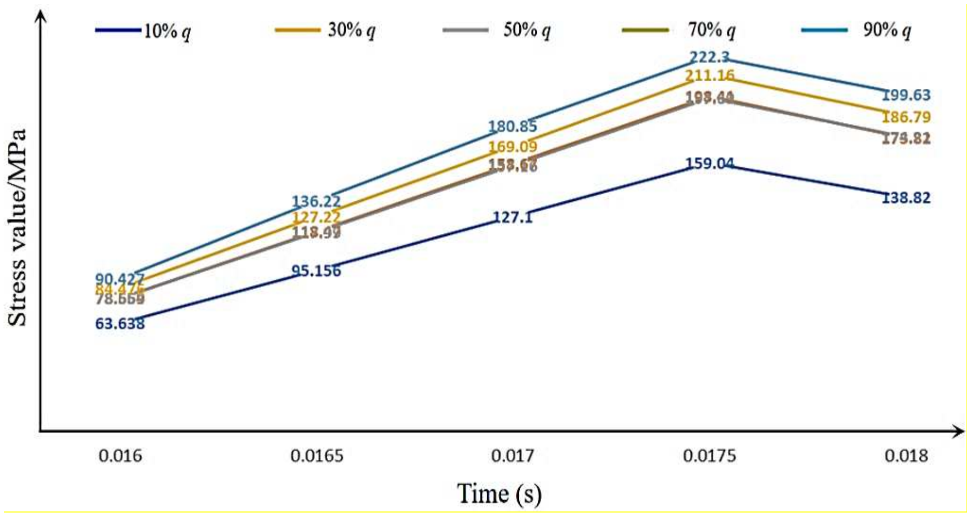
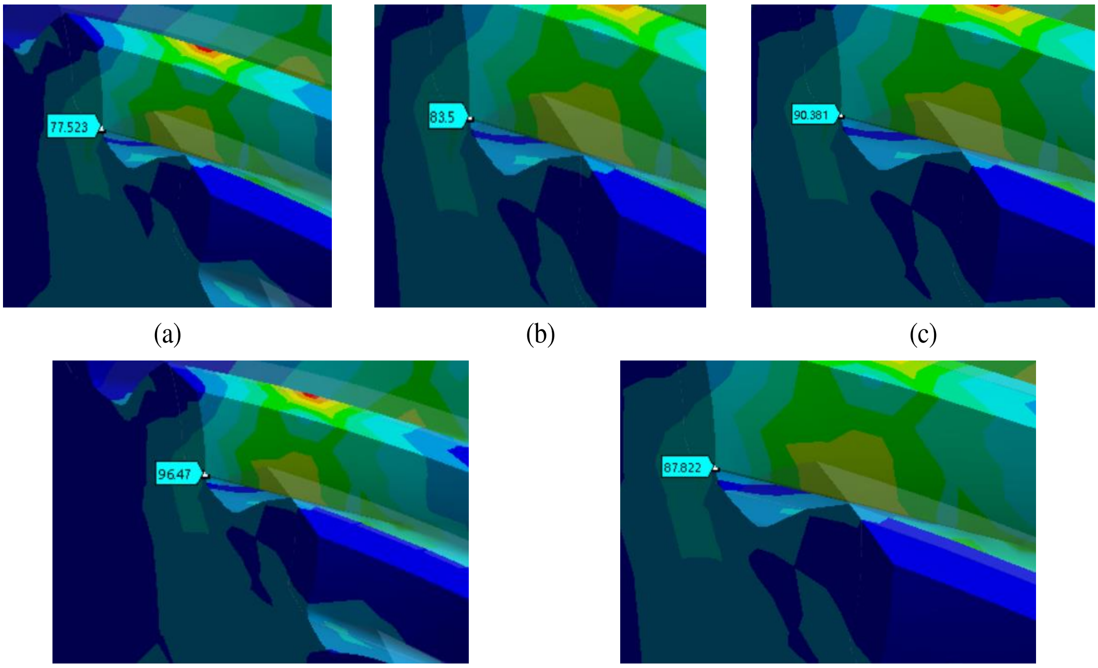


Fig.10. Stress level (MPa) vs. time at different crack depth

B. Transient Dynamics Under Different Penetration Length

By the same method, the stress value at the tip of the crack and different time, when penetration length equals 10mm, are obtained in Figure 11. The stress level under the parameter of different penetration length, a group of cases are conducted and calculated as well, the detail results of different penetration length and different time are shown in Table 3, and the data is curved and shown in Figure 12. Based on Figure 12, it could be found that the stress level is increased with the crack penetration depth, and reaches the max value at the time of 0.0175s, after that the stress decreases gradually, which is quite similar to that of different crack depth vs. time.



(a) (b) (c)

(d) (e)

Fig.11. Stress distribution (MPa) at crack tip and penetration length equals 10mm. (a) 0.016s; (b) 0.0165s; (c) 0.017s; (d) 0.0175s; (e) 0.018s

Table 3. Stress value (MPa) of crack penetration length vs. time

Penetration length(mm)	Time (s)				
	0.016	0.0165	0.017	0.0175	0.018
	Stress (MPa)				
10	77.5	83.5	90.3	96.4	87.8
20	115.1	127.7	139.3	150.2	135.8
30	168.7	185.6	202.6	219.9	197.6
40	138.7	150.3	165.1	182.7	165.6
50	70.3	77.3	85.3	92.8	83.7

According to Figure 10 and Figure 12, it can be seen that as the depth of the crack increases, the stress on the crack at the end point of the gear end face does not follow a non-linear increase pattern. Due to the structural characteristics of herringbone gears, compared to straight teeth, when two gears mesh together, multiple teeth mesh simultaneously, sharing the contact stress borne by a single tooth with the bending stress formed by the gear itself. Based on the stress cloud map, it was found that there is a penetrating crack, and when the crack endpoint is located at the larger meshing position of the herringbone gear, the stress on the gear crack is greater. Moreover, compared to the increase in crack depth, the increase in crack penetration leads to a larger crack volume and a smaller increase in stress at its endpoints at the same time. It can be seen that the depth of cracks has a greater impact on the stress of gears compared to the degree of crack penetration. The result could be compared qualitatively with reference [15], although the type of tooth is different.

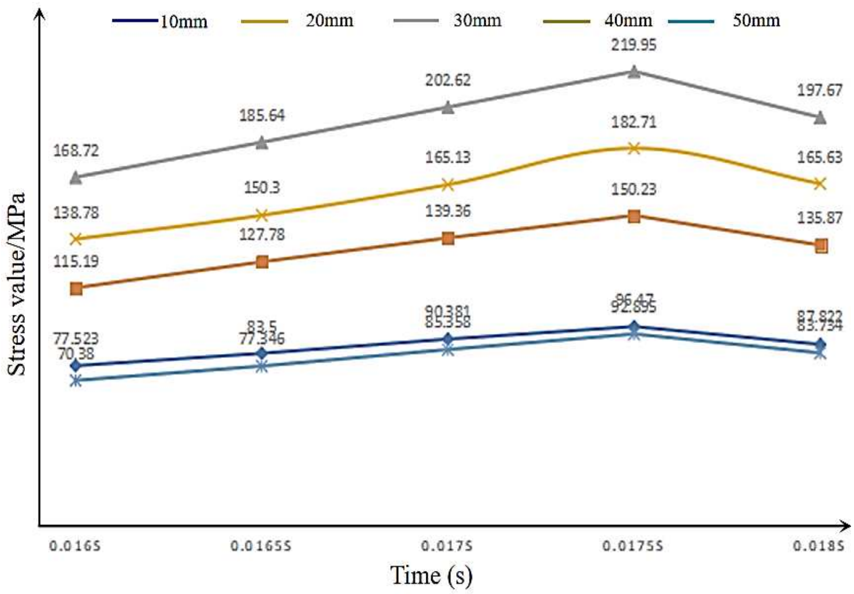


Fig.12. Stress level vs. time at different crack penetration length

3. Fatigue Life Evaluation Under Different Crack Depth

With the rapid progress of computer technology and the continuous deepening of research on gear damage mechanisms and fatigue failure mechanisms, the application of CAD/CAE/CAM collaborative simulation technology in the field of gear fatigue life prediction has gradually become widespread and widespread. The application of this technology not only improves the accuracy of prediction but also significantly accelerates the iterative process of gear design and optimization. The core of this technology lies in constructing a digital prototype model of the gear pair through 3D modeling software and using dynamic simulation software to simulate the gear motion under actual working conditions, in order to obtain its mechanical dynamic characteristics and accurately predict the fatigue life of the gear.

This paper analyzes the fatigue life assessment of gears based on dynamic analysis. Table 4 shows the fatigue life of gears at different crack depths, and the fatigue life of different crack penetration length can be obtained through similar methods. The data in Table 4 is curved in Figure 13. It can be seen that as the depth of the crack increases, the overall life of the gear decreases. When the depth of the crack first increases, the fatigue life of the gear sharply decreases. When the depth of the crack increases to a certain extent, the decreasing trend of the gear life gradually decreases. Therefore, it can be seen that the depth of the crack has a significant impact on the gear life, even if the depth of the crack is not very large, it will still have a significant impact.

Table 4. Fatigue life at different crack depth

Crack depth (mm)	10% q	30% q	50% q	70% q	90% q
Fatigue life (min. stresscycle)	58947	28655	26062	20646	16294

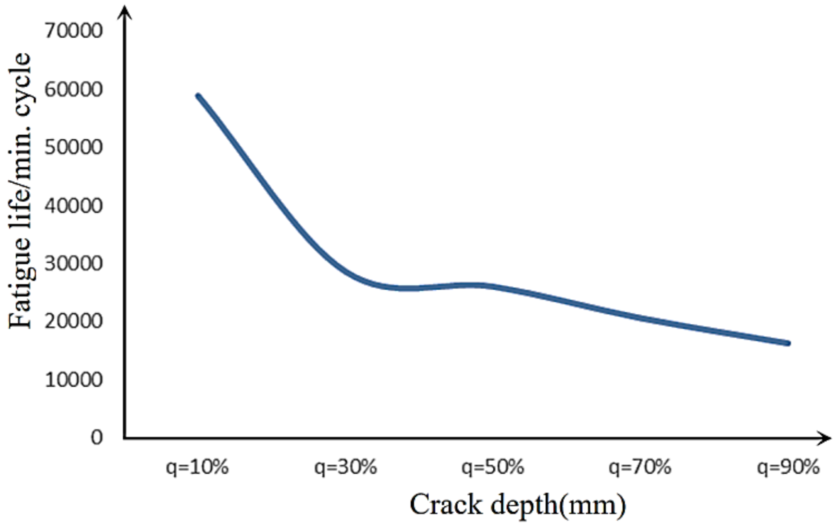


Fig.13. Fatigue life curve at different crack depth

IV. Conclusions

The herringbone tooth planetary gear system is chosen as the main research object in this paper, and modal analysis, static analysis, and transient dynamic analysis are conducted. Based on the above analysis, the natural frequency and corresponding vibration mode of the system, as well as the dynamic behavior characteristics, were obtained. Finally, based on transient dynamic analysis, the fatigue life of the gears was evaluated. The related analysis and optimization method could be used to guide the design of herringbone tooth planetary gear system. The main conclusions could be drawn based on the above analysis, including:

(1) The natural frequency of healthy gear and gear with different crack penetration length are compared, and could be found that the difference in natural frequency values between healthy gears and faulty gears is not significant.

(2) The stress level is increased with the crack penetration length, and increases and reaches the max value at a special time, after that the stress decreases gradually, which is quite similar to that of different crack depth vs. time.

(3) When the depth of the crack first increases, the fatigue life of the gear sharply decreases. When the depth of the crack increases to a certain extent, the decreasing trend of the gear life gradually decreases. Therefore, it can be seen that the depth of the crack has a significant impact on the gear life, even if the depth of the crack is not very large, it will still have a significant impact.

(4) Although the modal, transient dynamics and fatigue life evaluation are done in this paper, the influences caused by several other parameters including dynamic heat generation and different lubrication condition, and so on, are not considered in this paper, which will be studied further in future works as well as the experimental research;

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