A New Fatigue Analysis Method for Rotating Parts Based on Fatigue Cumulative Damage Criterion

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Abstract. The camshaft is one of the most important rotating parts in diesel engine. As a continuous rotating part, the high-cycle fatigue problem of camshafts has always been a hotspot in engineering. Generally speaking, nominal stress method is used to analyze high-cycle fatigue problems, but the continuous operation of camshafts determines that nominal stress method can not be used directly. Based on this, a load-discretization method based on the fatigue cumulative damage criterion and nominal stress method is proposed to solve the fatigue calculation problem of continuous rotating parts.

1. Introduction
The fatigue problem of shafts has always been a hotspot in engineering. Scholars at home and abroad have adopted various theories to study the fatigue problem of shafts. Early research \cite{1}, \cite{2} has limitations because they only focus on point of maximum stress rather than consider the effect of stress state on fatigue as a whole. Based on the fatigue cumulative damage criterion and considering the effect of stress-time history, a fatigue life analysis method of crankshafts considering global stress distribution is established by Wu Xiugen \cite{3}. Taylor \cite{4} proposed a crack model method to study the fatigue of camshafts by using the crack theory, and obtained fatigue results of camshafts under different loading conditions. Lou Xiaojing \cite{5} combined the results of Adams dynamic analysis, used rain flow counting method to process the stress spectrum of camshafts, and obtained fatigue life contour of camshafts. This method is still the most widely used one now.

Current research has fully considered the effect of load-time history, but because the stress spectrum is still simplified by the rain flow counting method, it can’t fully reflect the actual stress state of parts in a whole cycle. Based on this, a load-discretization method based on fatigue cumulative damage criterion is proposed in this paper. The load-time history is considered, and the actual state of the stress is fully considered.

2. Load-discretization method
The rotation of camshafts is a continuous loading process. The position and value of the loads on the cams vary with the rotation of the cams. During a cycle of camshaft rotation, the position and value of loads on the cams are different. According to finite element thought, a cycle of camshafts in divided into several discrete cases. In theory, when the number of the case is large enough, the solution result
of this method will be infinitely close to the actual operation of the camshafts. As shown in Figure 1, one cycle of camshaft is dispersed into 24 static load cases.

![Figure 1. Schematic diagram load-discretization method.](image1)

The loading condition of each load case is different, so the fatigue effect of each load case need to be superposed. In this paper, Palmgren-Miner cumulative damage criterion is adopted to superimpose the cumulative fatigue effects under various loading conditions and calculate their combined effects. The expression of the Palmgren-Miner criterion is shown in formula 1.

\[
\frac{n_1}{N_1} + \frac{n_2}{N_2} + \ldots + \frac{n_{i-1}}{N_{i-1}} + \frac{n_i}{N_i} < 1
\]

Where,
\(n_1, n_2, \ldots, n_i\) — Using Frequencies of each load cases;
\(N_1, N_2, \ldots, N_i\) — Permissible using frequencies of each load cases.

3. Static Stress Analysis

3.1. Geometric model
The research object of this paper is a six-cylinder diesel engine, in which different cylinders correspond to different shaft segments on camshaft. The complete model of the camshaft is shown in Figure 2.

![Figure 2. Complete camshaft model.](image2)
3.2. Static stress results

Load and boundary conditions of different load cases are input ABAQUS as different load steps. The maximum von Mises stress of each load case is shown in Table 1.

| Load Case | Mises Stress | Load Case | Mises Stress |
|-----------|--------------|-----------|--------------|
| Step-1    | 239.39       | Step-13   | 273.32       |
| Step-2    | 249.26       | Step-14   | 275.94       |
| Step-3    | 256.86       | Step-15   | 279.06       |
| Step-4    | 252.53       | Step-16   | 327.51       |
| Step-5    | 253.69       | Step-17   | 227.15       |
| Step-6    | 238.39       | Step-18   | 165.93       |
| Step-7    | 240.84       | Step-19   | 165.41       |
| Step-8    | 258.38       | Step-20   | 165.47       |
| Step-9    | 268.00       | Step-21   | 174.66       |
| Step-10   | 255.39       | Step-22   | 227.00       |
| Step-11   | 265.14       | Step-23   | 264.81       |
| Step-12   | 267.43       | Step-24   | 249.93       |

From the stress results in Table 1, it can be seen that the maximum stress value of camshaft varies with different load cases. Among them, the von Mises stress value corresponding to load case 16 is the largest, and the maximum stress value is 327.51 MPa. Von Mises stress contour of load case 16 is shown in Figure 4.

Figure 3. Model and main components.

Figure 4. Static solution results of Load Case 16.
From the stress results above, it can be seen that the maximum stress in one cycle of camshaft is 327.51 MPa. But the yield limit is 1178 MPa, which is far more than the maximum stress. This also confirms the rationality of the nominal stress method adopted in this paper.

4. Fatigue life
In this paper, MSC. Fatigue in MSC. Patran 2014 software is adopted to analyse the fatigue life of the camshaft. The stress results are converted into ASII coded files, and then imported into MSC. Fatigue software. The steps of fatigue analysis are shown in Figure 5.

![Figure 5. Steps of nominal stress method.](image)

After the calculation, the fatigue life contour is presented. MSC. Fatigue provides two ways of expressing fatigue life contour, namely fatigue life contour (Figure 6) and fatigue life logarithmic contour (Figure 7). Logarithmic contour can express the change of fatigue life gradient more clearly, but the value of minimum fatigue life is to round the power directly, which causes large result error. Although the gradient of fatigue life change is not as clear as that of logarithmic contour, the value of fatigue life is rounded directly in fatigue life calculation, which avoids the large error in the minimum fatigue life value of logarithmic contour. Therefore, the fatigue life contours in this paper use logarithmic life contours, but the minimum fatigue life values of camshafts are obtained in the fatigue life contour.

![Figure 6. Fatigue life contour of camshaft.](image)

The result of fatigue life calculation is $9.32 \times 10^{10}$, and the weak point of fatigue life lies in the shoulder of supporting shaft and cam 1-2. Comparing with the results in table 1, it can be seen that the maximum stress of the camshaft is far below the yield limit of the material, but after certain cyclic load, the camshaft will also suffer from fatigue damage. At the same time, the fatigue life contour on cams is discontinuous, which will not cause significant error because the weak point of the camshaft lies on the shaft necks rather than the cams. According to refs [6], the position of failure in this paper
are basically the same as that in engineering cases. Based on the result, we can conclude that load-discretization method proposed in this paper is almost accurate and has a certain value in engineering.

Figure 7. Fatigue life logarithmic contour of camshaft.

5. Conclusion

1. Compared with the traditional fatigue analysis method, the load-discretization method proposed in this paper considers every load case in a cycle, and ensures the accuracy and conservativeness of the results.
2. The load-discretization method based on fatigue cumulative damage criterion can be used to analyse the fatigue life of rotating parts, and the calculation results are basically reliable.
3. Load-discretization method proposed in this paper will cause certain error on the cams of fatigue life contour, but the result of weak point is basically accurate.

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