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Finite Element Analysis of the Fatigue Life for the Connecting Rod Remanufacturing

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Abstract: One important technical issue is whether the residual fatigue life of products meeting the needs of its next life cycle.This study analyzes the failure mechanism of the connecting rod, establishes its three dimensional model, uses dynamic simulation software ADAMS to calculate its time-load spectrum of the connecting rod; uses finite element analysis software ANSYS to get local stress-strain distribution; uses the traditional anti-fatigue methods to calculate the condition limited fatigue strength and then based on Miner fatigue damage theory and the stress of the connecting rod to make analysis, finally, uses Goodman fatigue theory to get fatigue strength and to estimate its total fatigue life, combined with its historical service time to predict its residual fatigue life. Provide reliable data to support how to calculate the residual fatigue life of these parts.

Key words: Connecting rod, fatigue life, remanufacturing, finite element analysis

INTRODUCTION

The connecting rod is an important driving part of diesel engine. It is loaded in compression and in tension. Its main motions are shown in Fig. 1.

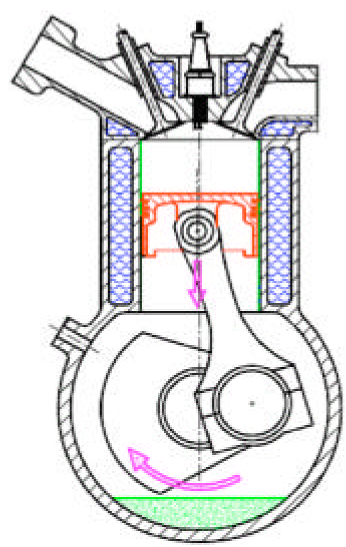

Fig. 1: Main motions of the piston-connecting rod system (Vertical arrow: Oscillating, Circular motion: Rotating)

The finite element analysis is an effective analysis method, used widely in the design of connecting rod. In 2000s, some studies tried to carry out the shape optimization using the special strategies, such as topology and genetic optimization, bounded constraints of the stress and fatigue life of connecting rod (Yuan and Zhang, 2012; Sun *et al*., 2012; Haiba *et al*., 2005). Xu and Yu (2007) studied the failure investigation of a diesel engine connecting rod. The fatigue cracks initiated from the axial grooves by the alternative load so that the multiple-origin fatigue fracture took place on the connecting rod. Griza *et al*. (2009) evaluated the relation between tightening force and fatigue crack propagation in connecting rod bolts with an analytical fracture mechanics approach. Baldizzone *et al*. (2012) studied the spalling phenomena of the crankpin surface. The non-metallic inclusions underneath the surface was considered the

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cause of the damage and the varieties and contents of non-metallic inclusions were identified. Bo *et al.* (2010) studied the method of fatigue life prediction based on the material S-N curve is able to predict the critical position and fatigue life in connecting rod fatigue test.

There is great significance to predict remanufacturing remaining fatigue life of connecting rod. Remanufacturing blanks has undergone many service cycle. Whether the parts have residual service life or its remaining useful life is able to sustain the next life cycle which is a important technology of the remanufacturing and is the primary problem before remanufacturing process.

Based on the traditional theory and methods of the fatigue life analysis, the engine connecting rod fatigue life is predicted by finite element analysis method and dynamic simulation method (Chen *et al*., 2006). The time-load spectrum of engine connecting rod is obtained using Dynamics simulation by ADAMS software. The rod local stress-strain distribution is received by ANSYS finite element analysis. The limit fatigue strength of rod is calculated (Milan *et al*., 2004). Based on the rod stress and Miner fatigue damage theory, the Goodman fatigue method is used to calculate the fatigue strength of the rod and to predict its fatigue life.

To predict the fatigue life of the connecting rod, the time-load spectrum analysis of the rod of a diesel engine is performed, the finite element analysis software ADAMS is used to simulate the real force of the actual situation. The stress distribution of connecting rod was calculated by ANSYS. Goodman fatigue theory was used to get fatigue strength and to estimate total fatigue life of the rod, then its residual fatigue life was predicted combing with its historical service time.

DYNAMICS SIMULATION METHODOLOGY

Setting parameters of dynamics simulation:
Materials:

Crankshaft:	42CrMoA
Piston:	ZL109 (ZAlSil2CulMglNil)
Connecting rod:	45Cr

Methodology: The three dimensional model of the connecting rod is established by PRO-E software. The time-load spectrum of the connecting rod is calculated by ADAMS software using dynamic simulation method.

PRO-E model of the rod is imported into ADAMS software. The material properties of piston, connecting rod and crankshaft are defined and the mass of them are calculated, respectively. The result of PRO-E model of the rod imported into ADAMS software is shown in Fig. 2.

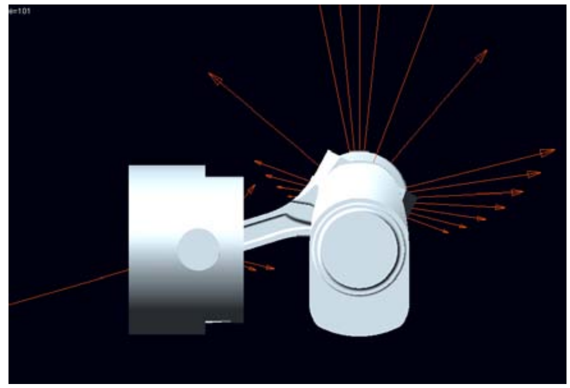

Fig. 2: Result of PRO-E model of the rod imported into ADAMS software

Rated speed of crankshaft is 2400 r min^{-1}, so the crank every turn around time is 0.025 sec. For six-cylinder four-stroke engine, the time of load cycle analysis of connecting rod should be two-circle, i.e., 0.05 sec. The corresponding rotation angle is 720°. Pressure is 13.5 Mpa when the fuel exploded. The piston diameter is 124 mm. The pressure F is 16, 3000 N. The explosion force working angle is 120° in a six-cylinder engine. The explosive power at this time is assumed to be constant.

Analysis results of dynamics simulation: When the crankshaft turns 0.003 sec, i.e., turns 43.2 degrees, cylinder explodes after avoiding dead center. The impact force is 163,000 N. The instantaneous compression load of connecting rod is 99, 071 N; The crankshaft turned 0.003 sec, i.e., turned 43.2 degree, it escapes the dead point. At that time the cylinder explosion produce the impact force of 163,000 N and the connecting rod is instantaneous subjected to a compressive load of 99,071 N.

At 0.004 sec, i.e., 57.6 degree, the compression load reduced to 16,001 N. Due to increasing angular acceleration in 0.007 sec, i.e., 100.8 degree, the maximum compression load rating is 24,800 N. At 0.0125 sec, i.e., 180 degree, the compression load reduced to 16,001 N. At 0.019 sec, i.e., 273.6 degree, the maximum tension load is 12771 N. At 0.025 sec, i.e., 360 degree, the tension load is 7542 N. The load time history from 0.029-0.05 sec and from 0.004-0.025 sec is same. The cycle load simulation results of the connecting rod big end are shown in Fig. 3.

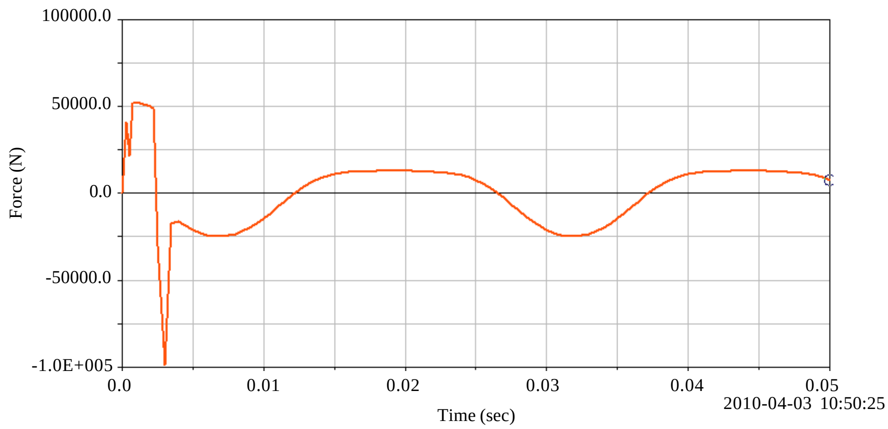

Fig. 3: Time-load spectrum of the connecting rod in a load cycl

CONNECTING ROD FINITE ELEMENT ANALYSIS OF CONTACT FATIGUE

Finite element model:

- **Material property:** 45Cr, E = 2.06×10^{11}, = 0.3
- **Unit type:** Solid brick 8 nodes 185
- **Boundary conditions:** According to the actual working condition of the connecting rod, the big and small ends are respectively fixed to analysis its stress and strain
- **Meshing:** There is more fillet on the connecting rod. So, it is difficult to divide mesh of the connecting rod, automatic grid method is used to mesh model. Figure 4 shows the mesh of the connecting rod. It contained about 8956 elements

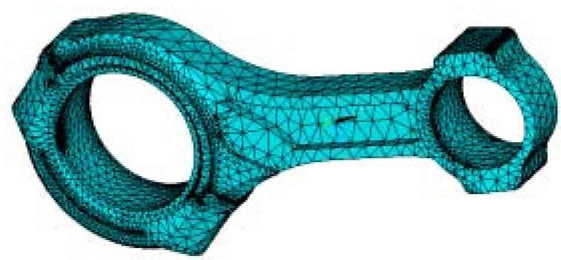

Fig. 4: Finite element model of connecting rod

RESULTS ANALYSIS

Using ANSYS, stress distribution of connecting rod is obtained. As shown in Fig. 5-8. Figure 5 shows that the maximum stress occurs in the bar of the connecting rod near the small end when the big end loaded under compressive load. The maximum stress is about 26.6 Mpa in 0.004 sec and the maximum stress is about 41.2 Mpa in 0.007 sec. Figure 6 shows that the maximum stress occurs in the small end when the big end loaded under compressive load. The maximum stress is about 22.5 Mpa in 0.004 sec and the maximum stress is about 25.3 Mpa in 0.007 sec. Figure 7 show that when the big end loaded under tension load, the maximum stress is about 8.5 Mpa in 0.0125 sec, 49.2 Mpa in 0.019 sec and 24.5 Mpa in 0.025 sec, respectively. Figure 8 show that when the small end loaded under tension load, the maximum stress is about 8.4 Mpa in 0.0125 sec, 58.5 Mpa in 0.019 sec and 24 Mpa in 0.025 sec, respectively.

Received in the connecting rod the maximum tension 12771 N in 0.019 sec rod small end of a point receives the maximum tensile stress is 57.547 Mpa while the big end received the maximum tensile stress 49.338 Mpa, so statistics stress results way in accordance with the large end fixed to the small end of the stress value statistics. In 0.019 sec, the maximum tension load of the connecting rod is 12771 N, rod small end of a point receives the maximum tensile stress of the small end is 57.547 Mpa, the maximum tensile stress of the big end is 49.338 Mpa. So, the stress results statistics method. Stress results of statistical way is according to the big end being fixed, the small end being forced.

Abscissa 1-10 represent the ten force point on the crankshaft from 0.004-0.05 sec, the ordinate represents the stress value of the point A (i.e., the maximum stress points on the small end at the maximum tensile force effecting). Statistical results are shown in Fig. 9.

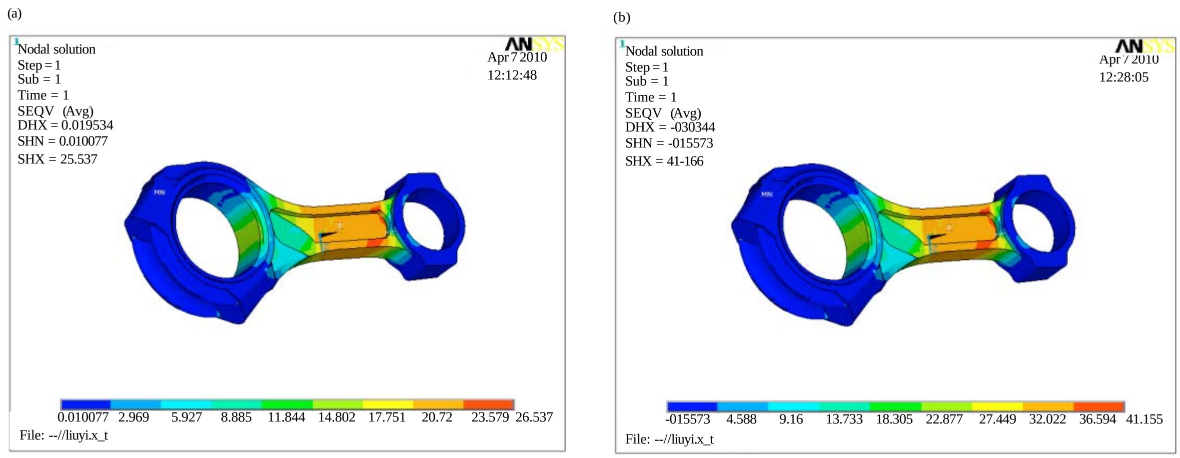

Fig. 5(a-b): Stress distribution of big end under compression load, (a) Stress distribution at 0.004 sec and (b) Stress distribution at 0.007 sec

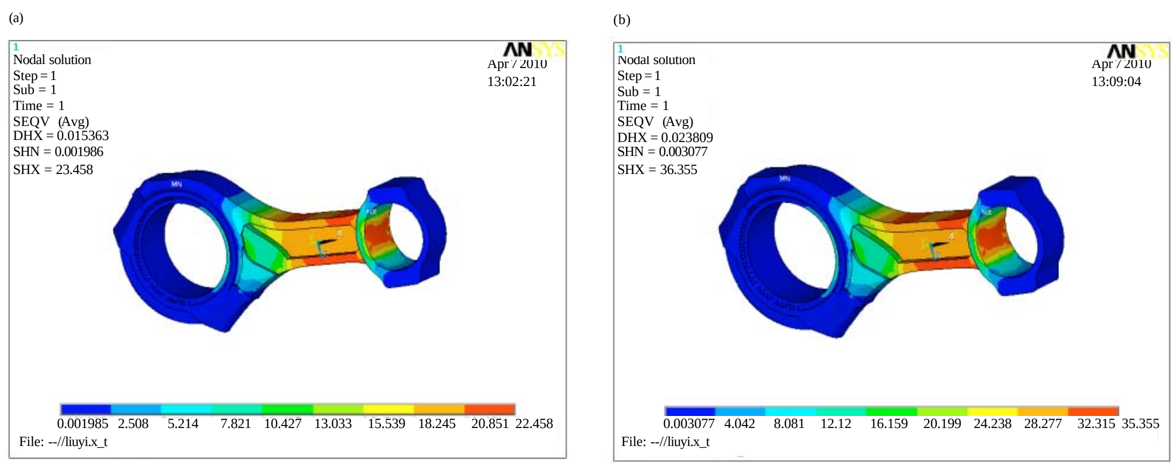

Fig. 6(a-b): Stress distribution of small end under compression load, (a) Stress distribution at 0.004 sec and (b) Stress distribution at 0.007 sec

CONNECTING ROD LIFE CALCULATION

Connecting rod fatigue limitation calculation: The connecting rod materials is 45Cr. Fatigue limit is σ_{-1} = 241.8 Mpa, the conditional fatigue limit of The connecting rod is σ_{-1D} = 91.59 Mpa, after taking into account the factor of safety, the conditional fatigue limit is:

$$\sigma_{-1D}' = \frac{\sigma_{-1D}}{n_1} = \frac{91.59}{1.8} = 50.88\ \text{MPa}$$

Prediction service life of connecting rod based on the miner fatigue damage theory: Based on the finite element analysis results it can be concluded that the maximum stress is 57.55 Mpa. At the rated load, the maximum stress is less than fatigue limit, so the life of rob can be considered infinite.

In rated loading cyclic, the curve of the stress changes in a four-stroke cycle is drawn with the simulation, as shown in the Fig. 10. The connecting rod is endured a fatigue damage while crankshaft runs each lap. As the red line section shown in the figure, during this

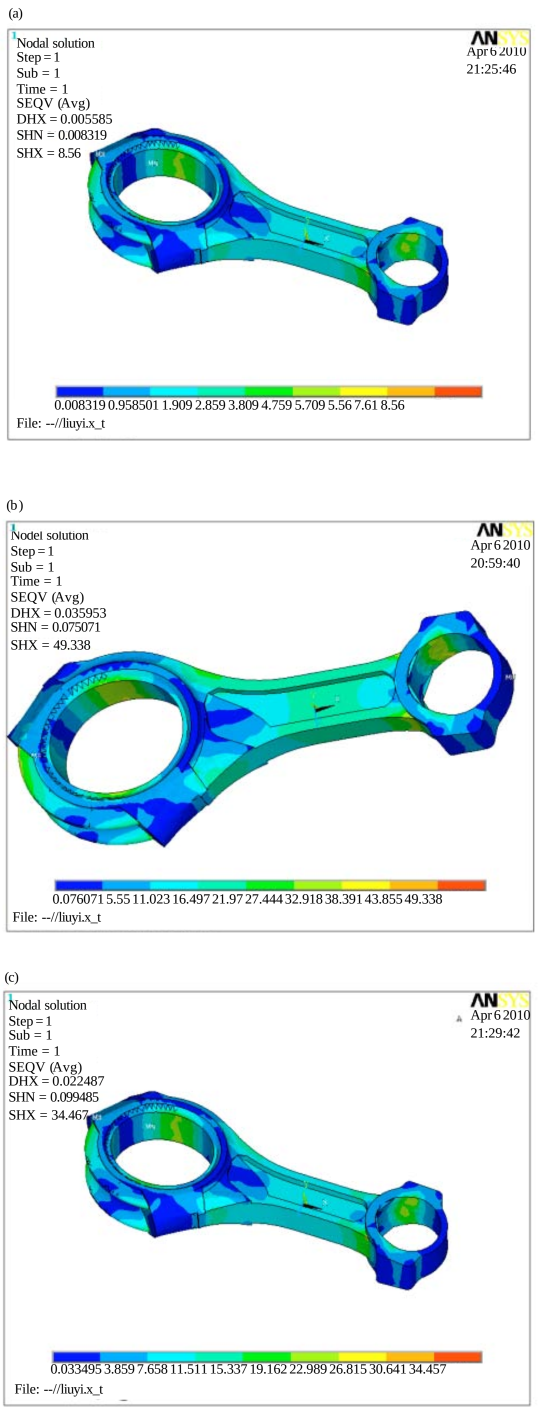

Fig. 7(a-c): Stress distribution of big end under tension load, (a) Stress distribution at 0.0125 sec (b) Stress distribution at 0.019 sec and (c) Stress distribution at 0.025 sec

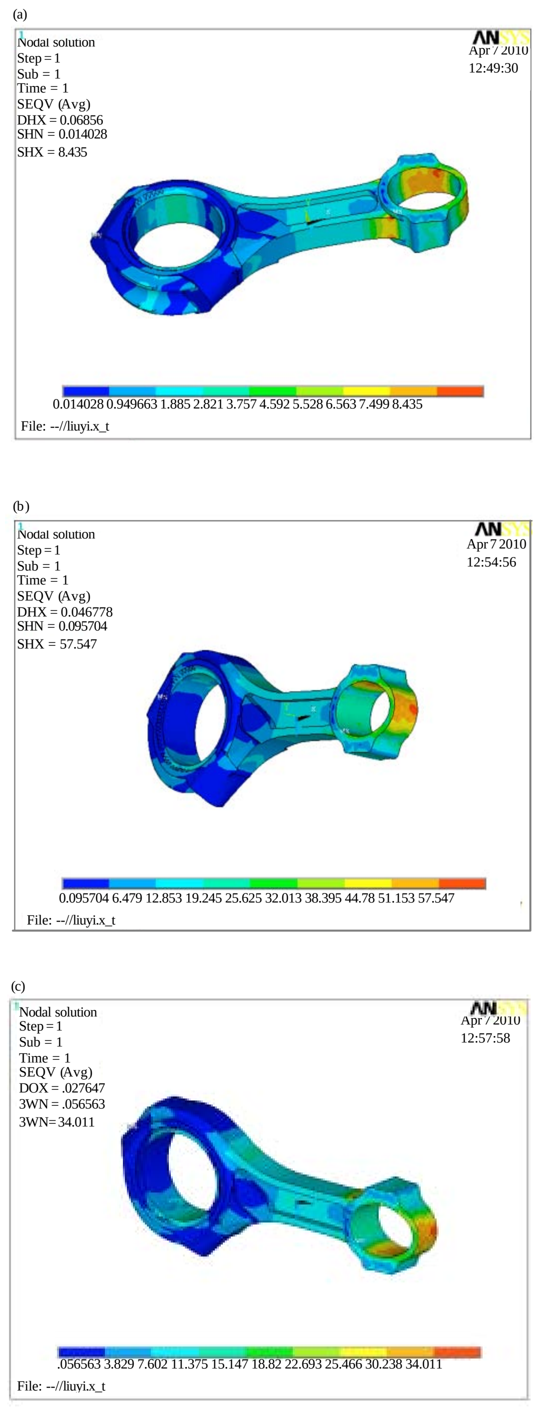

Fig. 8(a-c): Stress distribution of small end under tension load, (a) Stress distribution at 0.0125 sec (b) Stress distribution at 0.019 sec and (c) Stress distribution at 0.025 sec

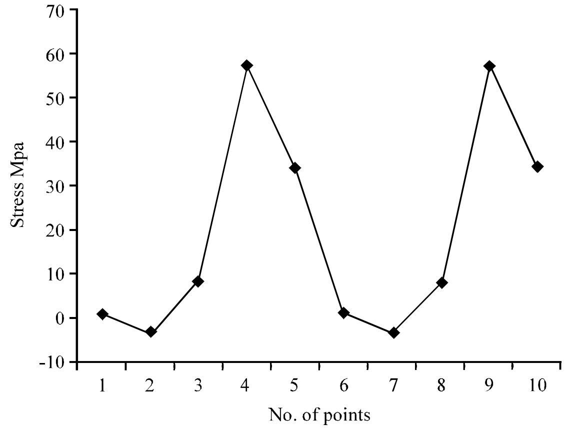

Fig. 9: Curve of the maximum stress (A point) on the connecting rod changing with time

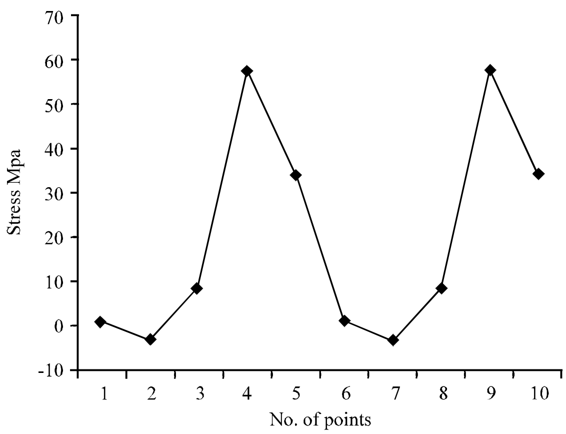

Fig. 10: Variety of stress in a four-stroke cycle

Table 1: Calculation parameters of Equivalent symmetric cyclic stresses σ_i

I	σ_{min}	σ_{max}	σ_m	σ_a
1	8.44	57.55	32.99	49.11
2	57.55	34.01	45.78	23.54

I: Level i, σ_{min}: Minimum stress, σ_{max}: Maximum stress, σ_m: Average stress, σ_a: Stress amplitude

time the connecting rod bearing the maximum tension force. Other times the stress of the rod is less than the conditional fatigue limit 50.88 Mpa.

To predict Fatigue life, Goodman fatigue calculation method takes into account the fatigue stress amplitude, mean stress and material mechanical performance parameters. σ_i is calculated according to Goodman fatigue limit diagram. The calculation results of Equivalent symmetric cyclic stresses σ_i are shown in Table 1.

The calculated σ_1 is 50.9 Mpa, σ_2 is 24.75 Mpa.

The equivalent symmetric cyclic stress σ_1 and σ_2 corresponding to material fatigue failure limit cycles were, respectively:

$$N_1' = 10^7 \times (\frac{241.8}{50.90})^{4.17} = 6.64 \times 10^9$$

$${}'_2 = {}^7 \times \frac{241.8}{24.75}{}^{4.17} = \times {}^9$$

The cycle number of the connecting rod under rated load is obtained:

$$N = \frac{1}{\frac{1}{N_1'} + \frac{1}{N_2'}} = 6.325 \times 10^9$$

The vehicles speed is Assumed to be 80 km h^{-1}, Crankshaft rated speed is 2400 r min^{-1}, so crankshaft rotation 1800 rpm km^{-1} is calculated. The vehicle can safely travel 3,514,000 km. Assuming the engine to work 12 h a day and uninterrupted work 300 days a year, it can be calculated motor vehicle safe and reliable working time is about 12 years and 2 months.

Remaining fatigue life calculation of the connecting rod: The vehicle mileage specified in "Motor vehicle standard specifies mandatory retirement" is generally not more than 600,000 km. while the rated speed of the crankshaft is 2400 r min^{-1}, the traveling speed is 80 km h^{-1}, so the equivalent life N_0 is equal to 1.08×10^9. Rated load rod work cycles 6.325×10^9 minus the rod remaining work cycles 5.245×10^9 and then the rod remaining safe use mileage of 2.8 million km is calculated. It can meet the requirements of the next life cycle of the connecting rod.

CONCLUSION

The breaking down of an engine connecting rod is analyzed with the remaining fatigue life analysis process. Firstly, the three-dimensional solid model of the connecting rod is established by Pro/E software. Then, the time-load spectrum of the rod is calculated by ADAMS software. Import three-dimensional model into ANSYS, to analysis contact fatigue of the connecting rod, the stress-strain distribution is calculated. The conditions limit fatigue strength of the connecting rod is investigated by traditional fatigue analysis methods. And then based on the rod stress the Miner fatigue theory is used in analysis. The rod fatigue strength and life prediction are researched with the Goodman fatigue Law. The results showed that the remaining fatigue life of the rod is able to meet the next life cycle.

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