

# Research on the Dynamics of Crankshaft Under the Condition of Operating Mode Based on LS-DYNA

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**Abstract:** A model of crankshaft under working condition is established based on LS-DYNA. The stress distributions of crankshaft and its regularities of the stress along surface of crankshaft and layer depth at the fillet are studied. The results show that the obvious stress concentration exist in the upper end of fillet on main journal and the lower end of fillet on rod journal when the piston arrives at top dead center, and the maximum equivalent stress occurs at the fillet on rod journal, so the fracture of crankshaft is most likely to occur in the lower end of fillet on rod journal. With the increase of depth, the stress on the surface of fillet reduces correspondingly, and the equivalent stress begins to stabilize when the depth reaches up to 2.8 mm. The study of maximum working stress provides the theory basis for the selection of residual stress.

**Keywords:** crankshaft; explicit dynamics; distribution regularities; residual stress

## 1 Introduction

Crankshaft is one of the most important parts of the engine. Due to the impact of cyclic stress, the permanent deformation will occur at the crankshaft<sup>[1]</sup>. How to ensure the crankshaft working life and prevent the occurring of fracture has been the research topic over the years. In order to study the stress distribution of crankshaft under the working condition, this paper builds and analyzes the model of crankshaft based on LS-DYNA, and gets the results of the stress distribution and the regularities of the stress along the fillet surface and layer depth in fillet of connecting rod journal.

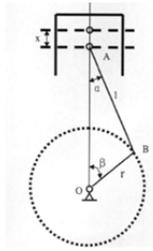
The crankshaft fillet rolling strengthening is an ideal surface strengthening method. Compared with the traditional methods, rolling strengthening has many advantages like low cost, short processing time, obvious strengthening effect<sup>[2]</sup> etc. After rolling strengthening, the crankshaft fatigue strength can increase by 30%-200%<sup>[3]</sup>.

However, how much the residual stress produced in the process of rolling strengthening is uncertain, and it's very hard to confirm the value of residual stress. This paper finds that the maximum working stress can provide the theory basis for the selection of residual stress.

## 2 Research of kinematics of crankshaft connecting rod system and load-bearing performance

### 2.1 Kinematics analysis of crankshaft connecting rod system

Figure 1 shows the structural diagram of crankshaft connecting rod.  $\alpha = 0^{\circ}$ , the piston moves to the top dead center position.  $\alpha = 180^{\circ}$ , the piston moves to the lower dead center position.



**Fig.1** Structural diagram of crankshaft connecting rod

If  $\lambda = l/r$ ,  $\sin \alpha = \lambda \sin \beta$ , the displacement formula is shown below

$$x = r[(1 + 1/\lambda) - \cos \beta - [1 - \lambda^2 \sin^2 \beta]/\lambda]. \quad (1)$$

According to the piston displacement formula, it deduces the velocity and acceleration:

$$v = wr \left[ \sin \beta + \lambda/2 \sin 2\beta [1 - \lambda^2 \sin^2 \beta]^{-1/2} \right]. \quad (2)$$

$$a = w^2 r \left[ \cos \beta + \lambda(1 - \lambda^2 \sin^2 \beta)^{-3/2} \left[ \cos 2\beta(1 - \lambda^2 \sin^2 \beta)^2 + \frac{\lambda^2}{4} \sin^2 2\beta \right] \right].$$

(3)

## 2.2 Gas force and inertia force of connecting rod mechanism

The force of the cylinder is not only related to the size of the cylinder, but also to the pressure between inside and outside of the cylinder. The gas force formula is shown as follows:

$$F_g = (P_1 - P_0)\pi D^2/4 \cos \alpha. \quad (4)$$

$\alpha$ : swing angle of connecting rod;  $D$ : cylinder diameter;  $P_1$ : absolute pressure of gas in cylinder;  $P_0$ : absolute pressure of gas inside the box.

In the theoretical analysis, the whole weight of connecting rod is translated into large mass  $m_1$  and small mass  $m_2$  by the static equivalent principle,  $m_1$  is heavier than  $m_2$ . Reciprocating inertia force is generated by the piston and small mass. It is considered that the weight of piston assembly concentrates on the center of the pin.  $M_p = \sum m_i$ . The reciprocating inertia force is shown as follows:

$$F_1 = (M_p + m_2)rw^2 \left[ \cos \beta + \lambda(1 - \lambda^2 \sin^2 \beta)^{-3/2} \left[ \cos 2\beta(1 - \lambda^2 \sin^2 \beta)^2 + \frac{\lambda^2}{4} \sin^2 2\beta \right] \right].$$

(5)

Centrifugal inertia force is generated by the crankshaft and the large mass of connecting rod. The crankshaft comprises the main crankshaft journal, crank arm, connecting rod journal etc. In order to facilitate the theoretical calculation, the crankshaft's weight is converted to the connecting rod's weight, so the centrifugal inertia force after converting weight is equal

to the centrifugal inertia force before converting. The single crankshaft's weight conversion formula is shown as follows:

$$m_k = m_a + 2m_b p/x. \quad (6)$$

$m_a$ : weight of connecting rod journal;  $m_b$ : weight of crank arm;  $p$ : distance between the center of crank arm and rotating center of crankshaft;  $x$ : piston stroke. The centrifugal inertia force is shown as follows:

$$F_2 = (m_k + m_1)rw^2. \quad (7)$$

## 2.3 Solution of load

There're some parameters of crankshaft showed in table 1.

**Table 1** Parameters of the crankshaft

Parameters	Value	Parameters	Value
crank radius $r$	39mm	Bulk mass quality $m_1$	0.342kg
Connecting rod center distance $l$	130mm	head mass quality $m_2$	0.180kg
main journal diameter $D_1$	50.33mm	single crank quality $m_k$	0.281kg
connecting rod diameter $d$	38.5mm	piston quality $m_p$	0.315kg
cylinder diameter $D$	75mm	angular velocity $w$	255.2rad/s

This paper finds out the similar cylinder size and the size of the cylinder pressure, the maximum pressure in the 465Q cylinder is about 8.12MPa, the inner pressure of the box is about 0.12MPa.

$$F_g = (P_1 - P_n)\pi D^2/4 \cos \alpha = 35325N. \quad (8)$$

$$F_1 = (M_p + m_2)a = 1634N. \quad (9)$$

$$F_2 = (m_k + m_1)rw^2 = 1582N. \quad (10)$$

During the power stroke, the maximum pressure of crankshaft is shown as follows:

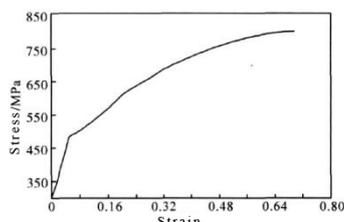
$$F = F_g - F_1 - F_2 = 32109N. \quad (11)$$

## 3 Dynamics analysis of the crankshaft under working condition

### 3.1 Parameter selection

The diameter of the main journal is 50.33mm, the diameter of the connecting rod journal is 38.5mm, the diameter of the cylinder is 75mm. Material of crankshaft is made of QT800 ductile iron, its density is  $7.16 \times 10^3 \text{ kg/m}^3$ , modulus of elasticity is  $7.12 \times 10^5 \text{ Mpa/mm}^3$ , the poisson's ratio is 0.27. The

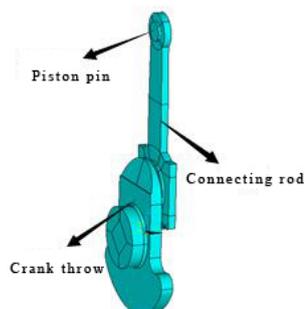
actual stress-strain curve of ductile cast iron is shown in figure 2. Connecting rod is made of 42CrMo, its density is  $7.85 \times 10^3 \text{ kg/m}^3$ , modulus of elasticity is  $2.09 \times 10^5 \text{ MPa/mm}^2$ , the poisson's ratio is 0.28. Piston pin is usually made of T10, its density is  $7.86 \times 10^3 \text{ kg/m}^3$ , modulus of elasticity is  $2.08 \times 10^5 \text{ MPa/mm}^2$ , the poisson's ratio is 0.28.



**Figure.2** Stress-strain curve of ductile cast iron

### 3.2 Simplified simulation model

This paper makes an appropriate simplification of the crankshaft structure and takes a part of the crankshaft as the calculating object. The crankshaft fatigue point occurs in the third crank throw of connecting rod journal, so this paper chooses the third crank throw as the analysis model. The simplified model is composed of half single crank throw, connecting rod and piston pin. The simplified model is shown in figure 3a, and its element type is SOLID164 unit, it is shown in figure 3b.



**Figure.3a** Solid model

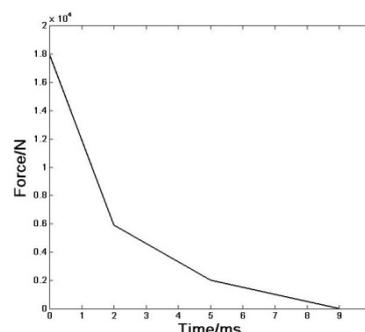


**Figure.3b** SOLID164 unit

### 3.3 Apply Load

During the process of working, the impact load of the crankshaft is maximum when it's in the stage of power stroke which is the key point of leading to the fatigue of the crankshaft. This paper mainly analyzes the crankshaft when it's in the stage of power stroke.

The normal speed of the crankshaft is around 1500~3000r/min, this paper takes the constant speed 2400r/min for analysis speed. The load value is  $F_{max}/2$ . The loading force is not constant, when the gas bursts, the pressure in the cylinder increases rapidly to the maximum. As the piston moves down, the gas volume increases and the pressure begins to fall.

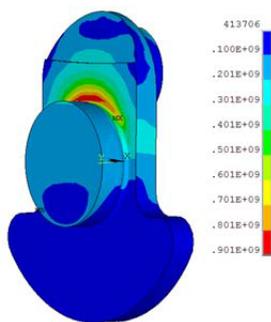


**Figure.4** Force-time curve

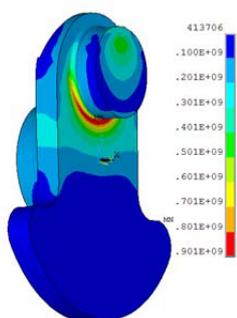
In order to make the load and the actual gas pressure closer, the force-time curve is established in figure 4. With the increase of rotation angle of crankshaft, the load decreases rapidly. When the time is 0.009s, the load decreases to 0. The terminal time in this paper is 0.009s.

### 3.4 Solution and result processing

Figure 5a shows the equivalent stress of the main journal and figure 5b shows the equivalent stress of the connecting rod journal in the maximum load. From the simulation results, it can be seen that the upper end of the main journal and the lower end of the connecting rod journal exist obvious stress concentration phenomenon, and the maximum equivalent stress, 901MPa, occurs at the fillet of the connecting rod journal.

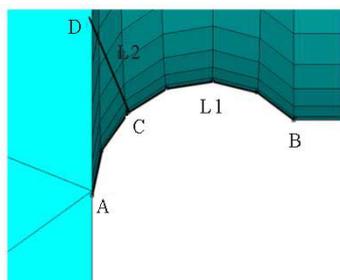


**Figure.5a** Equivalent stress of main journal



**Figure.5b** Equivalent stress of connecting rod journal

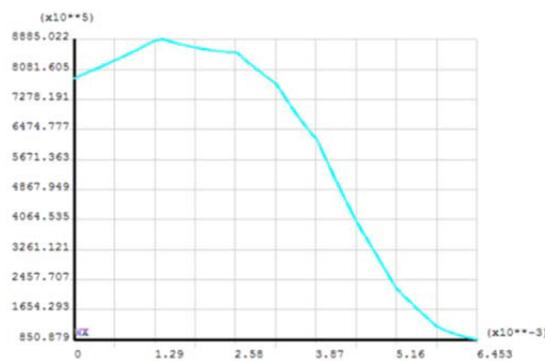
Analysis result shows that the upper end of the main journal withstands the pressure stress and the lower end of the connecting rod journal withstands the tensile stress. Rigid material can withstand the more pressure stress than tensile stress. Therefore, tensile stress in the fillet of the connecting rod journal is the key factor of leading to fatigue failure. This paper analyzes the tensile stress in the fillet of the connecting rod journal. In order to analyze it intuitively, the two paths in the circumferential direction and the rolling direction of the fillet of the connecting rod journal are selected. The paths in two directions are shown in Figure 6.



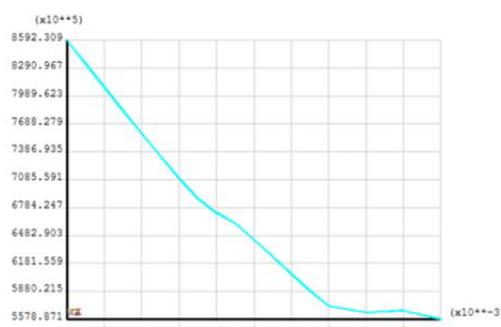
**Figure.6** Paths in two directions

The path L1 is from A to B in the connecting rod journal along the circumferential direction, the path L2 is from C to D in the connecting rod journal along the rolling direction. The equivalent stress values and the stress distributions along path L1 and path L2 are

obtained from the simulation which is shown in Figure 7a and 7b. In Figure 7a, it can be seen a parabolic form of the fillet surface equivalent stress curve, the maximum value of equivalent stress in the fillet of connecting rod journal is 888MPa and the point of maximum equivalent stress locates in between A and C. In Figure 7b, it can be seen that the stress in the fillet of the connecting rod journal gradually reduces with the increase of the depth, when the depth approaches 2.8mm, the equivalent stress is beginning to stabilize, and the maximum equivalent stress on the surface is 860MPa.



**Figure.7a** Simulation result of path L1



**Figure.7b** Simulation result of path L2

## 4 Summary

1. This paper researches the maximum equivalent stress, and gets the stress concentration phenomenon in the lower end of the connecting rod journal and the upper end of the main journal. The maximum equivalent stress occurs at the fillet of the connecting rod. Therefore, the key factor of leading to fatigue failure is the stress in the fillet of the main journal and the connecting rod.

2. The stress distributions along the path L1 and the path L2 are obtained by simulation. They can provide the theory basis for the selection of the depth of the carburizing layer.

3. The maximum working stress, 901MPa, is obtained by simulation. It can provide the theory basis for the selection of the residual stress in the process of crankshaft fillet rolling.

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