CONTACT STRESS ANALYSIS OF ENGINE SPEED ON CAM-TAPPET PAIR

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The cam-tappet friction pair of the valve train is one of the three major friction pairs of the engine. The contact stress between the cam and the tappet has an important influence on the fatigue life of the valve train. Taking a certain type diesel engine as the research object, a single mass motion model and computational model of the engine valve train were established. According to the Hertz theory, the kinetic equation of the cam-tappet friction pair was established. Then, the contact stress between the cam and the tappet was simulated by using the multi-body dynamics simulation software ADAMS respectively for the engine running at idling speed, rated speed and over-speed condition. The load history of the cam-tappet contact stress varying with time at different engine speeds was obtained. The research results have important reference value for further optimized design and improving the performance of the valve train.

Keywords: Engine Speed, Cam-Tappet, Contact Stress, ADAMS

1. Introduction

Cam-tappet is a pair of very important and sensitive friction pair in the valve train [1]. The engine usually works chronically in the harsh environments such as high temperature, high speed, pressure change, so the valve will be worn seriously after a period of time. As one of the three major friction pairs of the engine, the wear of the cam-tappet will not only destroy the valve movement laws and affect the ventilation performance of the valve, but also increase the noise during engine operation and reduce engine reliability [2]. The fatigue wear between the cam and the tappet is closely related to the contact stress. When the engine is working, the greater the contact stress between the cam and the tappet is, the greater the frictional heat between the friction pairs will be, excessive friction heat can cause cracking of

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the oil film [3]. When the cyclic stress in the cam-tappet pair exceeds the yield strength of the part material, the contact surface will crack locally and eventually lead to the complete failure of the engine over time. Therefore, it is of great significance to study the change of contact stress of cam-tappet pair [4, 5].

A single mass and multi-body dynamics model for valve train is established in this paper. Based on the cam contact stress theory, the calculation method of the contact stress of the cam tappet is studied. The changing law with time for the contact stress of the cam-tappet with different rotation rate is simulated by ADMAS software. The pressure spectrum of the valve body is obtained and the cam "fly off" condition is identified， which provides the basis for the fatigue wear analysis and wear experiment.

2. Establishment of Valve Train Model

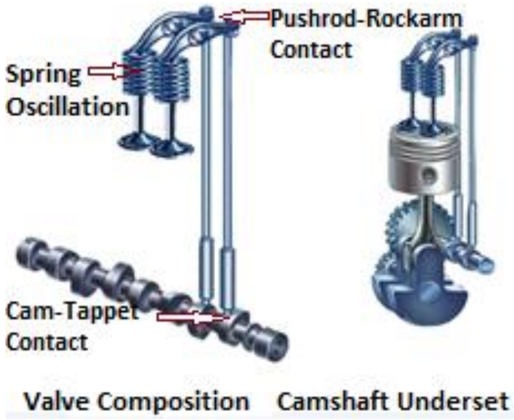


Fig.1. Lower cam type valve train

As shown in Fig.1, the whole drive chain of the engine valve train is composed with various parts of different quality and stiffness [6]. It is very difficult to describe them accurately, so the valve train mechanism can be simplified into a single degree of freedom vibration model shown in Fig.2. The valve motion of the valve train can be represent by the motion of a lumped mass *M*. *M* is the sum of the valve mass and the other transmission parts mass which converted to the valve. One end of *M* is connected to the cylinder head by a valve spring with a stiffness of c' and the other end is connected to a spring of imaginary rigidity c. The spring of with stiffness c actually represents the elasticity of the kinematic chain including the pushrod. The upper end of the model is controlled by the "equivalent cam"[7].

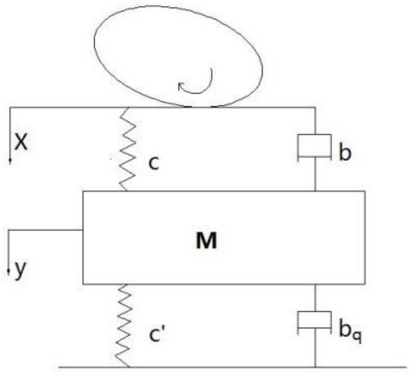


Fig.2. Single mass model of valve train

The rising process function of the valve $x(\alpha)$ can be expressed as

$$x(\alpha) = k \cdot h(\alpha) - x_0. \tag{1}$$

Where k is the rocker arm ratio; $h(\alpha)$ is the tappet lift, x_0 is the valve clearance.

Assuming that the sum of the external forces acting on the mass M is F, F can be given as Eq.2 according to Newton's second law.

$$F = Ma = M \cdot \frac{\mathrm{d}^2 y}{\mathrm{d}t^2} = M \cdot \omega^2 \cdot \frac{\mathrm{d}^2 y}{\mathrm{d}\alpha^2}. \tag{2}$$

Where M is the concentrated mass, kg; ω is the cam rotation angular velocity, rad/s; α is the cam angle, rad; $y(\alpha)$ is the valve displacement, mm.

The sum of the forces acting on the mass M is assumed to be F, which can be expressed as

$$F = c \cdot J + (-c' \cdot y(\alpha)) + (-F_0) + (-b \cdot \omega \cdot J_V) + (-b_P \cdot \omega \cdot \frac{dy}{d\alpha}) + (-R \cdot F_g(\alpha)). \tag{3}$$

In Eq. 3, because the method of determining the damping coefficient is complicated, paper [8] suggests that the external damping coefficient b be taken as $b = 0.017\sqrt{(c + c')M}$ and the external damping coefficient b_P be taken as zero; $c \cdot J$ is the spring restoring force of the valve mechanism ; $-c' \cdot y(\alpha)$ is the spring force of the valve spring； $-F_0$ is the spring preload； $F_g(\alpha)$ is the force that gas in the cylinder acts on the valve; R is the intake and exhaust coefficient, R equal to 0 indicates intake and R equal to 1 indicates exhaust; $-b \cdot \omega \cdot J_V$ is the internal damping force; $-b_P \cdot \omega \cdot (\mathrm{d}y/\mathrm{d}\alpha)$ is the external damping force; among them J and J_V can be expressed as Eq.4 and as Eq.5 respectively.

$$J = \begin{cases} x(\alpha) - y(\alpha) & x(\alpha) - y(\alpha) \ge 0 \\ 0 & x(\alpha) - y(\alpha) \le 0 \end{cases}. \tag{4}$$

$$J_V = \begin{cases} \dfrac{dx}{d\alpha} - \dfrac{dy}{d\alpha} & x(\alpha) - y(\alpha) \geq 0 \\ 0 & x(\alpha) - y(\alpha) \leq 0 \end{cases}. \tag{5}$$

$$F_{g(\alpha)} = A \cdot P(\alpha) - A_1 P_0 \tag{6}$$

P (α) is the gas pressure in the cylinder, A is the valve disc area, P_0 is the pressure of the back airway, and A_1 is the area of the valve affected by P_0. The gas pressure P (α) in the cylinder is measured by one cylinder of the V6 engine used in this experiment. Working conditions of the internal combustion engine are selected as 500r/min, 1050r/min and 1500r/min respectively. The working load is taken as 20%, 60% and 100%, as shown in Fig.3.

Eq. 3, Eq. 4, Eq. 5 and Eq. 6 are substituted into the Eq. 2, and the motion equation of the valve train mechanism is

$$\frac{\mathrm{d}^2 y}{\mathrm{d}\alpha^2} = \frac{c}{M\omega^2} \cdot J + \frac{b}{M\omega} \cdot J_V - \frac{b_p}{M\omega} \cdot \frac{\mathrm{d}y}{\mathrm{d}\alpha} - \frac{c'}{M\omega^2} \cdot y - \frac{F_0 + R \cdot F_g(\alpha)}{M\omega^2}. \tag{7}$$

Eq. 7 is a second-order differential equation about the unknown function whose solution is infinitely many. Two additional initial conditions need to be added before they can get a certain valve lift function, when the initial condition is the valve just opened a moment, that is, when $\alpha = \alpha_0$,

$$y\Big|_{\alpha = \alpha_0} = \frac{\mathrm{d}y}{\mathrm{d}\alpha}\Big|_{\alpha = \alpha_0} = 0 \tag{8}$$

In order to calculate and analyze the convenience of the problem, we can use an unknown function $z(\alpha)$ could be used instead of $y(\alpha)$, and the relationship between them is:

$$z(\alpha) = x(\alpha) - y(\alpha) \tag{9}$$

By Eq.7, it could be found that the differential equation that $z(\alpha)$ should satisfy:

$$\frac{d^2 z}{d\alpha^2} + \frac{b_p}{M\omega} \cdot \frac{dz}{dx} + \frac{b}{M\omega} \cdot J_V + \frac{c'}{M\omega^2} \cdot z + \frac{c}{M\omega^2} \cdot J = \frac{d^2 x}{d\alpha^2} + \frac{b}{M\omega} \cdot \frac{dx}{d\alpha} + \frac{c'}{M\omega^2} \cdot x(\alpha) + \frac{F_0 + R \cdot F_g(\alpha)}{M\omega^2} \tag{10}$$

The right end of Eq.10 is a known function of α, which can be written as $\varphi(\alpha)$. At this point, the initial condition of Eq.11 becomes:

$$z\Big|_{\alpha = \alpha_0} = x(\alpha_0), \qquad \frac{\mathrm{d}z}{\mathrm{d}\alpha}\Big|_{\alpha = \alpha_0} = \frac{\mathrm{d}x}{\mathrm{d}\alpha}\Big|_{\alpha = \alpha_0} \tag{11}$$

If $\frac{dz}{d\alpha} = u(\alpha)$ is remembered and treated it as an unknown function, then the second-order Eq.11 can be rewritten as first-order differential equations about the unknown functions $z(\alpha)$ and $u(\alpha)$:

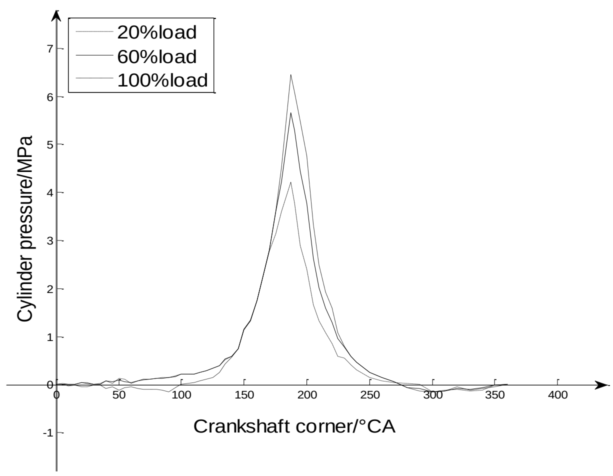
$$\begin{cases} \dfrac{dz}{d\alpha} = u \\ \dfrac{du}{d\alpha} = -\dfrac{b_p}{M\omega} \cdot u - \dfrac{c'}{M\omega^2} \cdot z - \dfrac{b}{M\omega} \cdot J_V - \dfrac{c}{M\omega^2} \cdot J + \varphi(\alpha) \end{cases} \tag{12}$$

The initial conditions become:

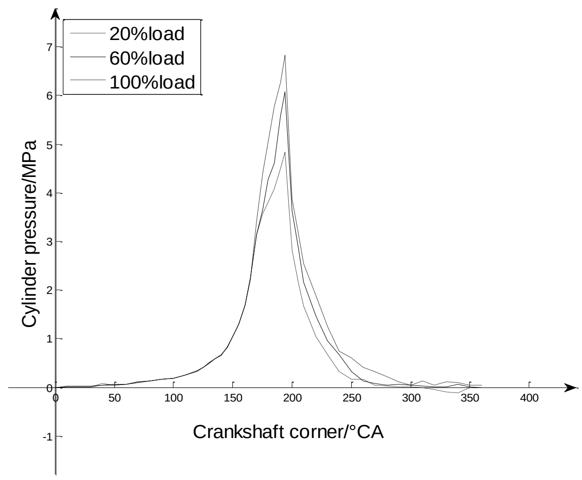
$$z\Big|_{\alpha=\alpha_0} = x(\alpha_0), \qquad u\Big|_{\alpha=\alpha_0} = \frac{dx}{d\alpha}\Big|_{\alpha=\alpha_0} \tag{13}$$

This problem can be attributed to how the first-order differential Eq.12 satisfies the solutions Eq.13 for $z(\alpha)$ and $u(\alpha)$. This problem can be solved by Ronge-Kutta method.

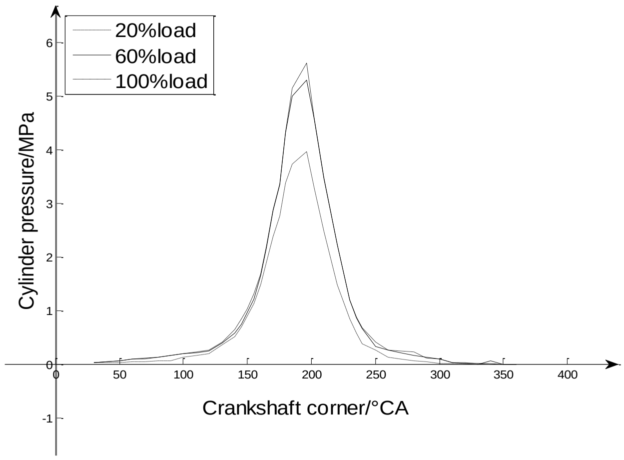
Substituting $z(\alpha)$ and $\frac{dz}{d\alpha}$ into the solution Eq.10 yields $\frac{d^2z}{d\alpha^2}$. $y(\alpha)$, $\frac{dy}{d\alpha}$ and $\frac{d^2y}{d\alpha^2}$ are obtained from Eq.9.



a. n=500r/min



b. n=1050r/min



c. n=1500r/min

Fig.3. Different working conditions at different speeds diesel engine work diagram

3. Calculation Equation of Contact Stress of Valve Train

Hertz theory. Hertz theory is a guiding method proposed by Hertz in 1882 to solve the stress distribution in the contact area by elastic mechanics. The Hertz theory is based on the theory of static elastic contact to deal with the contact problems of general elastic objects [9]. Usually some fundamental assumptions should be made for calculating contact stresses with Hertz theory. The fundamental assumptions are as following [10].

(1) The two contact bodies of contact pairs are all isotropic and uniform linear elastic bodies;

(2) The radius of curvature is much larger than the size of contact surface and only elastic deformation occurs in contact zone before the elastic deformation of contact body occurs;

(3) The contact surface only has vertical pressure and no tangential friction.

The cam and tappet are usually cylinders made of different metal materials. Assuming that there is a line contact between the cam and tappet, the Hertz formula for calculating the contact stress of the cam-tappet can be described by Eq.9[8].

$$\sigma_c = 0.564\sqrt{\frac{F_z(\frac{\rho_1+\rho_2}{\rho_1\rho_2})}{(\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2})W}}. \tag{14}$$

In the Eq.9, F_Z is the normal force between the cam and the tappet; ρ_1 is the radius of curvature of the cam; ρ_2 is the curvature radius of the tappet; E_1 is the cam elastic modulus; E_2 is the Column elastic modulus; μ_1 and μ_2 are Poisson ratios of the cam and tappet respectively; W is the contact line width.

Contact stress equation. When calculating the contact stress, it is unreasonable to take the high flexible valve train as a complete rigid body which working under the high-speed and long-term conditions. The influence of the elastic deformation on the contact stress between the cam and the tappet can not be neglected during the movement of the valve train [11]. In this process, the force acting on the cam consists of three terms.

(1) Inertial force of mass m

$$F_n' = m\cdot\frac{d^2h_T}{dt^2} = m\cdot\omega^2\frac{d^2h_T}{d\alpha^2}. \tag{15}$$

(2) The elastic restoring force of the valve spring is converted to the end of the tappet

$$F_c' = k^2c\cdot\frac{\mathrm{z}(\alpha)}{k} = k\cdot c\cdot \mathrm{z}(\alpha). \tag{16}$$

Where k is the conversion factor; c is the valve spring stiffness.

(3) Damping force

$$F_b' = k\cdot b\cdot\omega\cdot\frac{d\mathrm{z}}{d\alpha}. \tag{17}$$

The normal force between the cam tappets is

$$F_z = F_n' + F_c' + F_b' = m\cdot\omega^2\frac{d^2h_T}{d\alpha^2} + k\cdot c\cdot\mathrm{z}(\alpha) + k\cdot b\cdot\frac{d\mathrm{z}}{d\alpha}. \tag{18}$$

Eq.18 is substituted into the Eq.14, and the contact stress σ_c of cam tappet can be obtained.

4. Dynamic simulation

Simulation model. The cam mechanism of the lower cam type valve train is embedded in the cylinder block of the engine. When modeling, it is necessary to appropriately simplify the solid model of the cam-type air distribution mechanism [12]. The three-dimensional assembly model of the valve train established in Pro/E is shown in Fig.4. The model described in Fig.4 is imported into ADMAS. According to the numerical model of single-degree-of-freedom dynamics established in section 2, the valve mechanism is established by defining the material of each part, applying the constraint relationship according to the mutual topological relationship of each moving part, contacting type and spring damping instead of the valve spring body dynamics model (as shown in Fig.5).

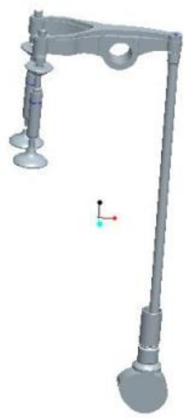


Fig.4. Three-dimensional model of valve train Fig.5. Multi-body dynamics model for valve train

Acceleration and displacement analysis of tappet. The ADAMAS/ Solver module is used to solve the model and kinetic equations, and the data of the simulation results are fed back to the ADAMS/View post-processing module. At the rated speed of the engine (1050 r/min), the simulation curve of the tappet displacement and acceleration of the engine valve train are shown in Fig.6 and Fig.7 respectively.

As can be seen from Fig.6and Fig.7, the displacement and acceleration of the tappet have no obvious changes among the t= 0s to 0.035s periods. Take t=0.035s at the starting point of the cam, the tappet contacts the cam base and the cam starts the push motion.

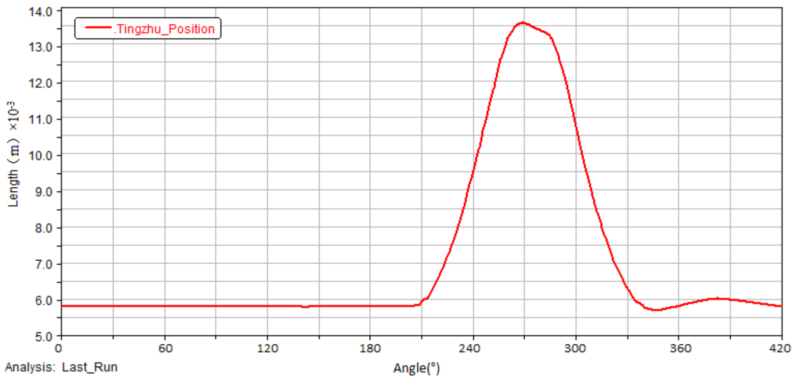


Fig.6. Displacement curve of tappet

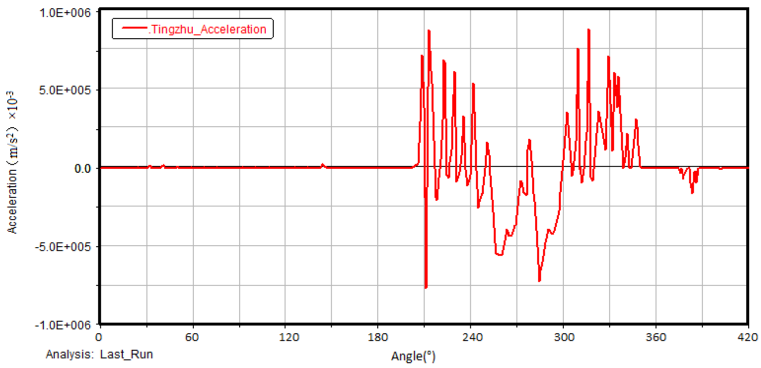


Fig.7. Acceleration curve of tappet

The tappet moves upward as the follower and has an upward acceleration. The cam enters the far stop stage during the period of t=0.044s to 0.046s, the displacement of the tappet does not change significantly and the acceleration direction changes. The cam enters the return motion during the period of t=0.046s to 0.056s, the tappet moves in the opposite direction. The cam enters the near-stoppage stage after t= 0.056s, the tappet displacement and acceleration are almost constant.

Contact stress analysis of cam-tappet. The change of the contact stress of cam-tappet with time are analyzed under the three working conditions of engine idling(500 r/min), the amount of speed(1050 r/min) and speeding work(1500 r/min).

(1) Idle conditions (500 r/min)

Idle conditions refer to the engine in the no-load operation state. In idle conditions, the clutch is in the joint position, the gearbox is in the neutral position, and the engine only maintains its own movement and does not transfer torque outward. The change of cam-tappet contact stress in idle time is shown in Fig.8.

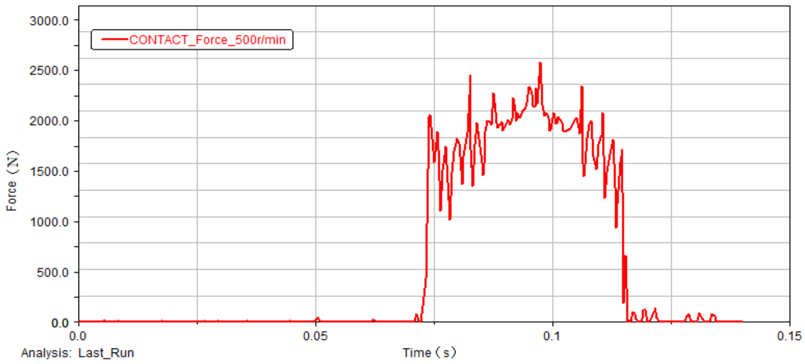


Fig.8. the relationship between contact force and time of cam-tappet at 500r/min

As shown in Fig.8, the contact stresses between the cam and the tappet are positive during the whole simulation. Due to the existence of the valve clearance, the contact force between the cam and tappet is intermittent with 0 values. When t=0.0973s, the contact force is maximum, F_{max}=2459N.

(2) Rated operating conditions (1050 r/min)

The rated operating condition refers to the engine working in the condition of the factory calibration speed and power. At this condition, the engine runs normally and transmits torque outward. The contact force curve of cam tappet column simulated under rated working condition can be obtained, as shown in Fig.9.

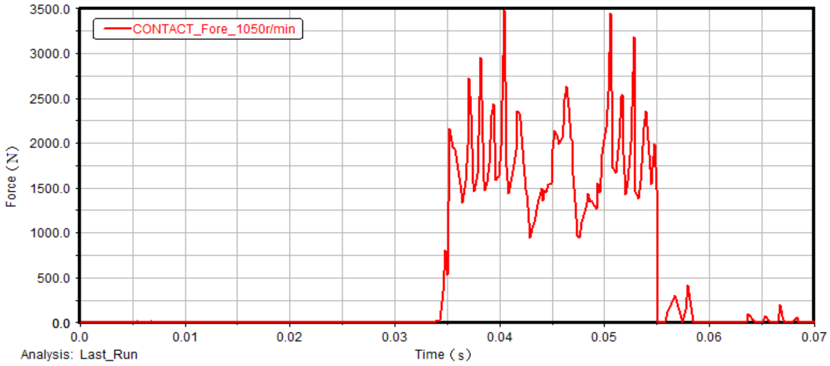


Fig.9. the relationship between contact force and time of cam-tappet at 1050r/min

As can be seen from Fig.9, the contact stress of the cam-tappet is similar to that under the idle condition under the rated operating conditions. Namely, the contact stress values of the cam-tappet are positive and the contact stress is intermittent 0 value of the situation. The difference is that the maximum contact force of the cam tappet pairs increases under the rated speed. When t=0.0404s, the contact force is F_{max}=3480 N.

(3) Over-speed conditions (1500 r / min)

Over-speed conditions refer to the engine speed exceeds the rated speed, the engine works in an abnormal state. The engine running with overspeed conditions will cause violent jitter phenomenon, which will do great harm to the engine.

When the cam speed is 1500 r/min, the contact stress of the cam-tappet changed with time can be obtained by simulation, as shown in Fig.10.

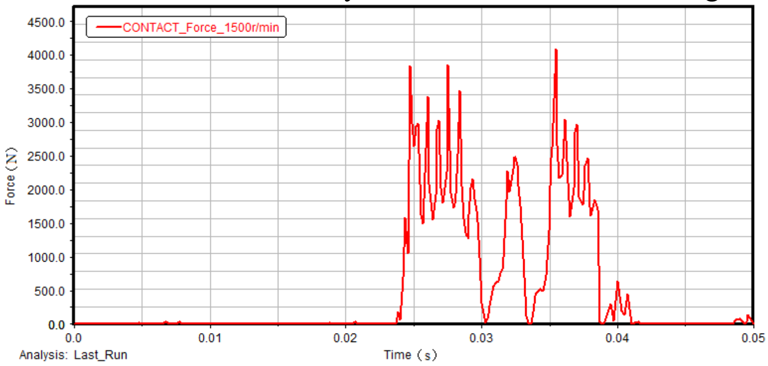


Fig.10. the relationship between contact force and time of cam-tappet at 1500r/min

As can be seen from Fig.10, in the over-speed condition, the maximum contact force between the cam and the tappet is F_{max}=4086 N at t=0.0355s. When the engine is in the over-speed condition, the contact stress value between two peaks suddenly becomes smaller, or even negative. This condition shows that there is no pressure and contact stress between the cam and tappet, namely the phenomenon of "flying off" occurs in the valve train [14]. The intermittent contact and disengagement of the cam-tappet pair causes the cam to collide with the tappet, which might result in severe engine vibration and noise increases, which might affect the engine life seriously.

In order to more intuitively illustrate the effect of different speeds on the contact stress for the cam-tappet pairs of the engine valve train more intuitively, Table 1 gives the simulation statistic results [15].

As can be seen from Table 1, the contact stress between the cam and the tappet is proportional to the engine speed. The higher the engine speed is, the greater the peak value of the cam-tappet pairs' contact stress becomes and the earlier the peak appears. When the engine is operating in the over-speed state (1500r/min), the engine will be violent jitter, the contact stress peak will increases and the valve body will wear severely.

Table 1

Simulation statistics of different speeds on the contact stress

Condition / Engine speed [r/min]	Idle conditions [500 r/min]	rated operating conditions [1050 r/min]	Over speed conditions [1500 r/min]
Maximum contact stress [N]	2459	3480	4086
Appear time [s]	0.0973	0.0404	0.0355

5. Conclusion

The influence of different speeds on the contact stress for the cam-tappet of the engine train is analyzed by using the multi-body dynamics simulation software ADMAS. The load history of the contact stress with time is obtained under the engine running at idle speed, rated speed and over speed conditions. The analysis results show that the engine speed has a great influence on the contact stress of cam-tappet. When the engine is in the over speed state, the peak value of contact stress is the maximum and the peak time appears the earliest, which will greatly affect the stability of engine operation, increase the cam-tappet wear and reduce engine life. The conclusion of the study provides the basis for optimizing the valve train in the future.

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