

Simulation of Cam Wear in Valve Trains of Diesel Engines

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ABSTRACT

In order to predict cam wear in valve trains of engines quantitatively, multi-body system dynamic analysis, finite element quasi-statics contact analysis and numerical calculation of elastohydrodynamic lubrication are applied to obtain the dynamic loads, the time histories of contact pressure and the oil film thicknesses in a cam-tappet pair. Using this methodology, the wear depth of the cam in the valve train of a heavy load diesel engine is calculated, which is in good agreement with the measured value in the practical tests. The results show that the cam-tappet is operated in a mixed lubricant or a boundary lubricant regime and the wear depths on both the sides of the cam have been found to be the greatest.

CITATION: Qin, W. and Duan, L., "Simulation of Cam Wear in Valve Trains of Diesel Engines," *SAE Int. J. Fuels Lubr.* 7(3):2014, doi:10.4271/2014-01-2879.

INTRODUCTION

Cam-follower kinematic pairs in valve trains of engines always work under complicated conditions of mechanical loads and wear during operation, which account for about 20-30% of automotive engine friction under the conditions of ordinary use at engine speeds below 2000 r/min [1, 2]. A cam and follower contact is a form of concentrated contact, which is traditionally reflected as operating in the mixed or boundary lubrication regime [2, 3, 4, 5]. Therefore wear in cams and followers is impossible to be avoided. As the characteristics of valve lift motion that influence the engine performance greatly are mainly determined by cam profiles, it is necessary to study the wear behaviors of cams and design measures for low wear.

The wear modes of cams and followers are pitting, scuffing and polishing [2, 6], which can be characterized as surface fatigue or adhesive wear. Surface fatigue is not considered in this study as it can be prevented by controlling of local stresses. Adhesive wear is usually predicted by the modified Archard formula [6, 7], which is proportional to the contact load and the sliding distance and inverse proportional to the hardness of surface under no lubricant conditions. Under mixed lubricant conditions, wear is closely related to the thickness of lubricant oil film and the composite root mean square (RMS) roughness of the contact surfaces [6, 8]. Therefore simulation of cam wear is a multidisciplinary problem including kinematic considerations, dynamic load and stress calculations and elastohydrodynamic film thickness evaluations.

In this study, as the cam wear in the valve train of a heavy load diesel engine is concerned, the kinematic parameters and dynamic loads are obtained using multi-body system dynamic analysis of the valve train, the oil film thickness is calculated by the numerical analysis of elastohydrodynamic lubrication (EHL), and the time history of the contact pressure between the cam and the tappet is simulated by finite element quasi-statics analysis.

MULTI-BODY SYSTEM DYNAMICS MODELING AND ANALYSIS OF THE VALVE TRAIN

In this paper, the multi-body system dynamics model of the pushrod-type valve train is established, and its dynamics characteristics are simulated in ADAMS software [9]. The three-dimension solid models of the parts in the valve train such as the cam, tappet, pushrod, rocker arm and valve are built firstly. Thus their masses, inertia and locations of the mass centers can be calculated and their rigid body models can be built. The cam profile is modeled as a smooth curve through a series points. The parts are constrained by kinematic joints as listed in Table 1. The damping of springs and friction of contact pairs are also included in the model. In addition, the constant angular velocity generator is added to the cams, and gas pressures on the valve varied with the crankshaft rotation angle and impact forces between the valves and seats are applied as well. The finished valve train model is shown in figure 1. Using this model, the dynamic contact force between the cam and the tappet at various engine speeds are simulated and shown in figure 2.

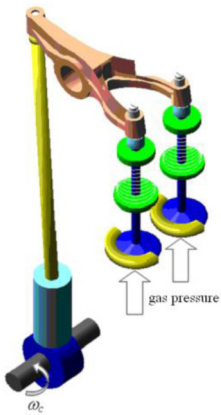


Figure 1. Multi-body system dynamics model of the valve train.

Table 1. Constraints between parts.

Part I	Constraint	Part J
Cam	Curve plane contact	Tappet
Tappet	Translational joint	Cylinder head
Tappet	Hooke joint	Pushrod
Pushrod	Spherical joint	Rocker arm
Rocker arm	Revolute joint	Cylinder head
Rocker arm	Sphere flat contact	Valve
Valve	Translational joint	Cylinder head
Top spring seat	Fixed joint	Valve
Top spring seat	Hooke joint	Valve spring
Valve spring	Fixed joint	Bottom spring seat

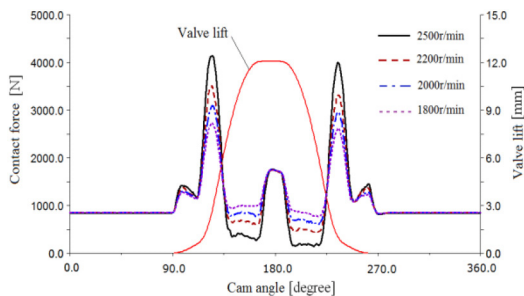


Figure 2. Contact forces between the cam and the tappet.

NUMERICAL ELASTOHYDRODYNAMIC LUBRICATION ANALYSIS

The contact between the cam and the tappet is a line contact. The basic equations of line-contact elastohydrodynamic lubrication (EHL) are as follows [10]:

$$\frac{\partial}{\partial x} \left(\frac{\rho h^3}{\eta} \frac{\partial p}{\partial x} \right) = 12u \frac{\partial}{\partial x} (\rho h) + 12 \frac{\partial}{\partial t} (\rho h) \quad (1)$$

where ρ is the density of lubricant (kg/m^3), η is the viscosity of lubricant ($\text{Pa}\cdot\text{s}$), u is the mean surface velocity (m/s), p is the pressure (N/m^2), and h is the film thickness (m) which is calculated by the geometric equation given below.

$$h = h_0(t) + \frac{x^2}{2R} + \delta(x, t) \quad (2)$$

where h_0 is the central film thickness of the rigid bodies (m), R is the equivalent radius of curvature (m), and $\delta(x, t)$ is the surface deformation (m) which is given by

$$\delta(x, t) = -\frac{2}{\pi E'} \int_{x_i}^{x_o} p(s, t) \ln(x-s)^2 ds + c \quad (3)$$

where E' is the equivalent modulus of elasticity (N/m^2), x_i and x_o are the inlet and outlet locations of the lubricant (m), and c is the integral constant.

The Roelands viscosity pressure relation is used and the density of lubricant equation is

$$\rho = \rho_0 \left(1 + \frac{0.6 \times 10^{-9} p}{1 + 1.7 \times 10^{-9} p} \right) \quad (4)$$

where ρ_0 is the density under atmosphere (kg/m^3).

The load equilibrium equation is

$$P(t) = \int_{x_i}^{x_o} p(x, t) dx \quad (5)$$

The calculated minimal oil film thickness with respect to the cam angle at various engine speeds is shown in figure 3, in which h_c is the lower film thickness bound for full film lubrication and h_b is the upper film thickness bound for boundary lubrication. It can be seen that the minimal oil film thickness of every point on the cam is increasing with the improvement of the speed. Most points on the cam are in the mixed lubrication regime, whereas some local locations at the two sides of the cam nose (at the cam angle of 180°) are in the boundary lubrication regime. The thin films are resulted from the small radii of curvature or the small entrainment velocities at these locations.