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Analysis of Cam-Follower of Valve operating mechanism of Diesel engine for the Prediction of Wear-rate

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Review/Article

ABSTRACT The dynamics of cam-follower is greatly influenced by wear which leads to reduction in efficiency, smoke and high stresses. Therefore, it is one of the most vital criterion to be taken into account while considering and designing the engine, valves and inlet port geometry. This paper deals with a methodology to estimate wear rate on the basis of dynamic force assessment and wear calculation done with computer simulation. Dynamic forces, acting during valve operations, subsequently affect the wear and tear, decrement in power and lubrication issues which may further become the cause of the premature failure of the system. Henceforth, the inertial forces become inevitable factors to precisely determine contact stresses. The critical influencers like contacting load and relative motion have been evaluated through dynamic modelling (including kinematic) of valve operating mechanism. Evaluation of wear was predicted having wear model of Archard. The estimation of film thickness (least) carried out with empirical equation given by Dowson-Hamrock. The flat-faced tappet of engine is having Polydyne cam profile. This research work predicts forces on non-conformal geometry taking into account Hertzian stresses and wear rate of cam-tappet.

Keywords: Construction and Demolition Waste (C&D Waste); Recycled Concrete Aggregate (RCA); Second-Generation Concrete; Heat-Processed Recycling; Sustainable Concrete.

INTRODUCTION

Valve operating mechanism is meant to control the outlet and suction valve. Valve operating mechanism greatly influences not only the brake power and required torque but also smoke at exhaust[1,2]. High amplitude accelerations in the cam and tappet system accounts to peak Hertz pressures [6]. This higher side Hertzian pressure eventually causes severe fatigue-subsurface, which menifests tenths of millimeters of cam surface. Surface wear predominant at slow peripheral speeds caused owing to the deteriorated lubricant properties which further leads to extreme frictional forces in interacting cam and follower non-conformal profiles [1, 5, 8, 11, and 12]. Owing to dynamic nature of forces and abrupt variations in relative velocities of tappet and cam, the tribological paraments become very crucial for evaluation. Valve spring and valve seat are having very influential materialistic properties like stiffness and damping which are accountable for operations of valve mechanism. additionally, non-conformal geometry, friction and also affect the severity of wear. These dynamic factors (including vibration) too affect valve lift. Subsequently, the non-linear dynamism of the system must be critically examined. Under detrimental load and relative velocities, the shape and size of lubricant-film thickness plays very

vital role in this anlysis [13, 14 and 15]. Entire modeling of dynamics and wear was done considering flat faced follower. This approach may be applied to other followers as well with certain constraints.

The contacting surfaces have extreme fluid film pressures and undue stresses because of nonconformity of cam-follower. The cam-follower are in continuous line contact during motion and having elastic deformation under Elastohydrodynamic lubrication region. Henceforth, the study must encompass the same to calculate the lubricant film thickness.

Cams may have simple harmonic motion and cycloidal motion. The current study includes polynomial cam profile of 5^{th} or 6^{th}-derivatives/order to achieve optimum results at elevated speed[3].

THE CAM PROFILE DESIGN

To design the polydyne cam profile in current modelling [1, 2, 3 and 4] of the order-6, following equations justify motion profile:

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Lift, $$Y = L\left[64\left(\frac{\theta}{\beta}\right)^3 - 192\left(\frac{\theta}{\beta}\right)^4 + 192\left(\frac{\theta}{\beta}\right)^5 - 64\left(\frac{\theta}{\beta}\right)^6\right]$$

$(0 \le \theta \le \beta)$ (1)

Velocity, $$V = L\omega\left[192\left(\frac{\theta^2}{\beta^3}\right) - 768\left(\frac{\theta^3}{\beta^4}\right) + 960\left(\frac{\theta^4}{\beta^5}\right) - 384\left(\frac{\theta^5}{\beta^6}\right)\right]$$

(2)

Acceleration,

$$A = L\omega^2\left[384\left(\frac{\theta}{\beta^3}\right) - 2304\left(\frac{\theta^2}{\beta^4}\right) + 3840\left(\frac{\theta^3}{\beta^5}\right) - 1920\left(\frac{\theta^4}{\beta^6}\right)\right]$$

(3)

Jerk, $$J = L\omega^3\left[\frac{768}{\beta^3} - 4608\left(\frac{\theta}{\beta^4}\right) + 11520\left(\frac{\theta^2}{\beta^5}\right) - 7680\left(\frac{\theta^3}{\beta^6}\right)\right]$$

(4)

Where $\theta = \omega \times t$

ω =Angular velocity of camshaft $= \frac{2\pi N}{60}$ rad/s

N = Rpm

The tabulated data were used for to plot vital factors graphs using a MATLAB program feeding equations-1,2,3 and 4.

Table 1: Dimensions of valve train Components:

Cam base circle	**20 mm**	**Lubricant type**	**SAE-30**
L	**43.55 mm**	**Spring constant**	**24.46 N/mm**
l_p	**261 mm**	**Cam hardness**	**55 HRc**
R_1	**41 mm**	**Valve lift**	**11.05 mm**
η_o	**0.1033 Pas**	**Pressure viscosity coefficient (α)**	**2.20E-08**
Ramp lift	**0.5 mm**	**Follower lift**	**6.8599 mm**

Below figure-1 dictates the follower displacement has a peak value of 6.86 mm occuring @ 45° of cam angle[1].

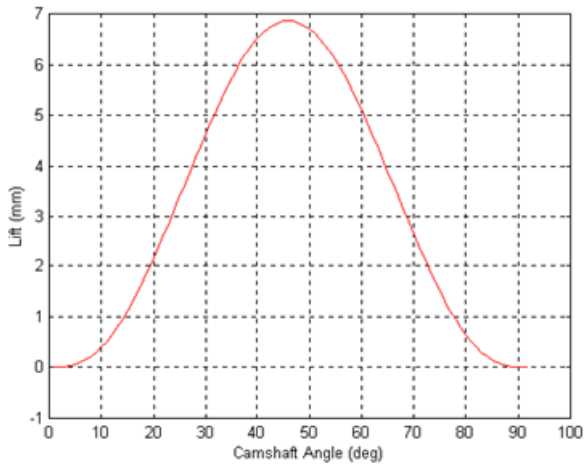


Fig-1. Lift v/s camshaft angle@3200 rpm

While At 10°, of cam angle, the velocity is 1000 mm/s. the peak of velocity is observed at @ 25° and 66° of cam angle as shown in adjacent graph[1].

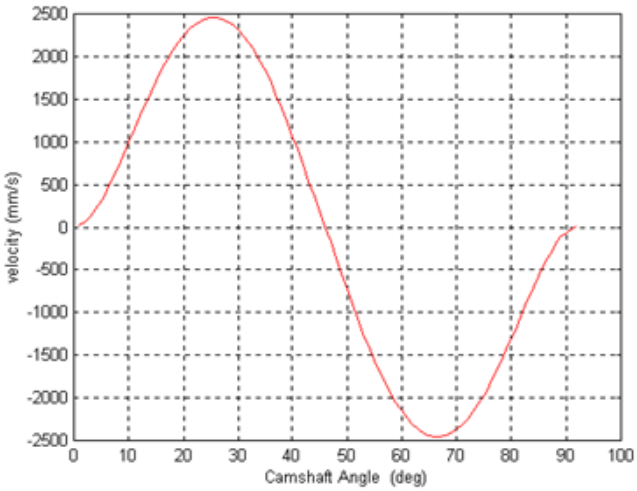


Fig -2. Velocity v/s camshaft angle@3200 rpm

The next graph pictures rate of change of velocity and cam angle[1]. The peak is observed @ 10° and 45°.

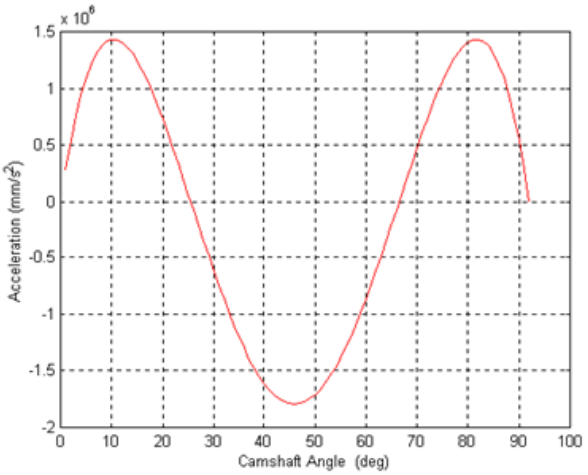


Fig-3. Acceleration v/s camshaft angle@3200 rpm

The next graph pictures jerk corresponding to cam rotation. Jerk accounts for impact force and leads to rough vibrations and concentrated forces[1].

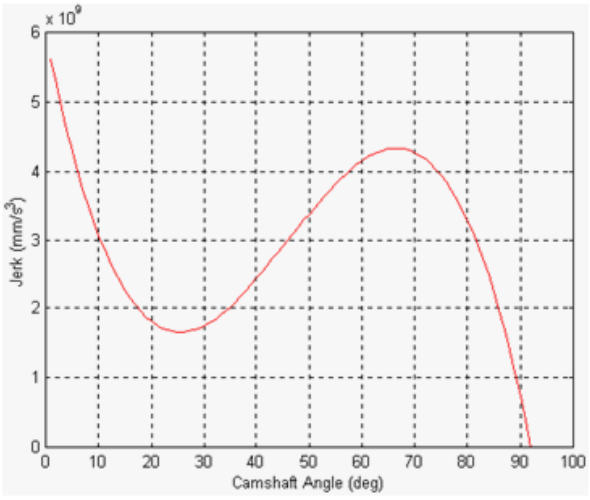


Fig-4. Jerk v/s camshaft angle@3200 rpm

DYNAMIC MODELLING OF VALVE TRAIN SYSTEM

The cam-follower wear intensity is greatly affected by dynamic forces of cam-follower during valve operation, therefore, the analysis of forces acting on different parts of valve train system were made [1,2]. The simulated modelling of the system is formed to consider that Follower and pushrod are following line of symmetry and lift delay is ignored.

The conceived modelling works satisfactorily at moderate and slow speeds excluding the effects of elasticity and damping. A labelled pictorial presentation of cam-follower and pushrod is as follows[1]-

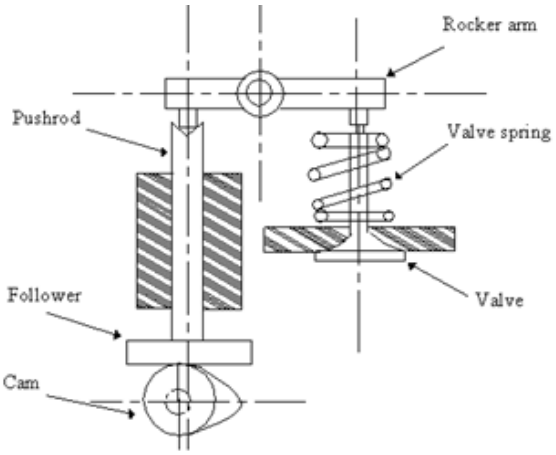


Fig-5. Schematic diagram of pushrod type valve train system with flat-faced follower

Downward force on the valve due to masses of components

$$= -(m_v + m_{kv} + 0.5m_{vz} + m_{vs})(g - a_v) \quad ..(5)$$

Downward force on the valve during suction

$$= -\frac{\pi(D_v{}^2 - d^2)P_c}{4} - \frac{\pi}{4}d^2P_o \quad \ldots (6)$$

Upward force on the valve due to gas pressure

$$= \frac{\pi D_v{}^2 P_g}{4} \quad \ldots(7)$$

Hence contact force between rocker arm and valve

$$F_3 = -(m_v + m_{kv} + 0.5m_{vz} + m_{vs})(g - a_v) + (F_L + K_v) + \frac{\pi D_v{}^2 P_g}{4} - \frac{\pi(D_v{}^2 - d^2)P_c}{4} - \frac{\pi}{4}d^2P_o \quad \ldots(8)$$

Contact force of bearing of rocker arm

$$F_k = \frac{\left[F_3R_1\left(1+\frac{1}{i}\right) + J_k\varepsilon + m_k ge - m_k gR_1\right]}{\left[R_1 - \frac{\mu \times d_k}{2}\right]} \quad (9)$$

Contact force between follower and pushrod

$$F_2 = \left(-F_3 + m_k g + F_k\right) \quad (10)$$

Contact force between cam and follower

$$F_1 = F_2 + \left(m_p + m_s\right)\left(a_p + g\right) \quad (11)$$

HERTZ PRESSURE

In order to calculate the Hertz's pressures, the theory of rolling contact[8] has been used.

$$P_H = 0.418\sqrt{\frac{F_1 E'}{b_n}\left(\frac{1}{R_p} + \frac{1}{RON}\right)} \quad (12)$$

Where $RON = \left(h_p + r_o + \frac{a_p}{\omega^2}\right)$

$$\frac{1}{E'} = 0.5\left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right)$$

MINIMUM FILM THICKNESS

Recalling the equation generated by Hamrock and Dowson to determine and evaluate the lubricant film thickness in EHL region[1,10,13]

$$W = \frac{w}{\left(E'R\right)}$$

The Speed Parameter:

$$U = \frac{\eta_o u}{\left(E'R\right)}$$

The Materials Parameter:

$$G = \alpha \times E'$$

According to Dowson-Hamrock's Curve Fitting formula:
Minimum film thickness is given by

$$H_m = 1.714 R W^{-0.128} U^{0.694} G^{0.568} \quad (13)$$

SPRING FORCE AND INERTIA FORCE

Spring force[2] is given by,
F_s = spring constant × total valve lift
Inertia force is given by,
F_i = total mass of the components × acceleration

MODELING OF WEAR

THE ARCHARD'S WEAR EQUATION

The most influential wear is sliding which accounts for malfunction of valves in long run use. A wear modelling was proposed by Holm and Archard[6] which at higher extent includes the critical variables in the analysis of sliding wear. This modelling can predict the intensity of wear and so the severity by taking into account the wear coefficient k. following is the well-known Archard's wear equation -

$$V = \frac{k \times \Delta L \times W}{3 \times H} \quad \ldots (14)$$

Where k = Wear coefficient. In case of steel-steel mating components, the range is 10^{-13}-10^{-9} for EHL region. (the assumed value is k = 10^{-10} mm^3/Nm)

RESULTS

The results are summarized as follows-

1. When engine was operated at 3200 rpm, the observed Hertz pressure[1] at nose of the intake and outlet cam-follower were tabulated -

Table 2: Hertz pressure at top of the cam (45^0 of cam angle)

Speed(rpm)	Hertz pressure (N/mm^2)	
	Inlet cam	Exhaust cam
3200	441	940

2. The results for wear rate are tabulated and found the maximum at exhaust cam side since there is higher hertz pressure at this side.

Table 3: Wear rate at top of the cam.

Speed (rpm)	Wear rate (mm^3/meter)	
	Inlet cam	Exhaust cam
3200	5.87E-10	2.65E-09

3. Minimum film thicknesses were obtained as tabulated-

Table 4: Minimum film thickness at top of the cam

Speed(rpm)	Minimum film thickness (µm)	
	Inlet cam	Exhaust cam
3200	1.09	0.89

CONCLUSION

Wear estimation of cam-follower have been done having a modelling of valve operating mechanism (valve gear train) and sliding wear. The befitting range of Hertz pressure for most of the materials, used in manufacturing of cam-follower, lies in 850 - 1000 MN/m^2 as per ASTM. Determined data are below the allowable range. Wear rate calculated are close to standard wear data (2.4E-08 mm^3/m) of steel against steel, which is commonly used value for long life of the engine. Exhaust cam requires special attention, since it was found to have higher wear rate than inlet cam. This could be mitigated by reducing hertz pressure at cam contact point. Hertz pressure may be decreased by modifying the cam profile or a higher surface hardness may be used.

Damping and vibration phenomenon have not been taken into account in the current analysis. This may cause deviation in the predicted wear rate (@3200 rpm) through this modelling.

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