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MATHEMATICAL MODELING AND STRESS ANALYSIS OF VALVE GEAR TRAIN OF DIESEL ENGINE AT VARIABLE VALVE LIFT

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Abstract: Dynamic forces are the results of varying turning moment, vibration and changes in loads on the engine. This leads to massive contact stresses in cam and follower of the engine which is responsible for changes in the timing of valve opening and closing. This paper defines a methodology to estimate the Hertzian contact stresses through the dynamic analysis with the aid of computer simulation (MATLAB). A dynamic force, which leads to wear and loss of power, may be the cause of the premature failure of the system. The inertia forces provide more accurate analysis of stresses of mating parts in line contact, which is crucial in estimation of wear also. The crucial variables, contact loads and relative motion, have been deduced by mathematical modeling of kinematics and dynamics of valve train system. The flat-faced tappet has been used with Polydyne cam profile. This modeling provides estimates of dynamic forces, Hertzian stresses in cam and follower which may prolong the life of engine.

Keywords: Six phase transmission line; fault analysis.

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INTRODUCTION

Befitting valve gear train system of any four stroke internal combustion engine is the key to control the operation of inlet and exhaust valves. Due to the strict norms of BIS, it is desirable to minimize the exhaust emission in accordance with engine's power output and torque [1]. Sudden changes in acceleration result in greater Hertz pressures in the cam and follower system [6,15]. These higher Hertz stresses follow the acute subsurface fatigue, which initiates below tenths of millimeters of cam surface. Also considerable sliding wear occurs at lower velocities since the elastohydrodynamic lubrication properties are degenerated leading to higher friction between cam and follower [4, 5, 7, 10, and 11]. Extreme tribological conditions are observed in valve gear train system owing to fluctuations in cycling loading and abrupt change of relative sliding speed. Also the operation of valves is influenced by stiffness and damping characteristics of elements such as valve spring and valve seat, inertia and geometry of components and frictional behavior of mating components. Valve opening and closing is dependent upon the mechanical and dynamic factors, hence to have smooth operation, it relies upon geometry of elements as well as on vibration. Both shape and thickness of fluid-film between cam and follower depend on the applied load and relative speed [12, 13, and 14]. In the current analysis a flat faced follower is used. However the analysis can also be easily exposed to other geometries of followers.

Earlier cams have SHM and Cycloidal motion but nowadays they have polynomial motion of 5th or 6th-order, which gives better performance at high speeds [2, 16].

THE KINEMATIC MODEL OF CAM AND FOLLOWER

In this analysis, a Polydyne [2 and 3] of 6-order is taken. The equation of motion for this cam is:

Lift, $$Y = L\left[64\left(\frac{\theta}{\beta}\right)^3 - 192\left(\frac{\theta}{\beta}\right)^4 + 192\left(\frac{\theta}{\beta}\right)^5 - 64\left(\frac{\theta}{\beta}\right)^6\right] \quad (0 \leq \theta \leq \beta) \qquad (1)$$

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

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

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

Where $\theta = \omega \times t$

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

N = Rpm

β = Constant (max. angle of rise or fall)

The dimensions and other parameters are given in table-1 below. The pertaining graphs have been obtained through a MATLAB program for the above equations showing displacement, velocity, acceleration and jerk at 3200 rpm.

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**

Figure-1 shows the displacement of follower with a maximum value of 6.8599 mm that occurs at 45° of cam rotation.
Figure-2 shows the velocity (in mm/s) v/s camshaft angle (degree). At 10°, the value of velocity is 1000 mm/s. maximum velocity occurs at 25° and 66°.
Figure-3 shows acceleration (in mm/s^2) v/s camshaft angle. Maximum acceleration occurs at 10° and 45°.
Figure-4 shows jerk (in mm/s^3) v/s camshaft angle. Jerk produces shock loads and causes undue vibrations and stresses. Jerk has been a main source of trouble in earlier cam profiles but in this case jerk is finite and results into smaller shock loads.

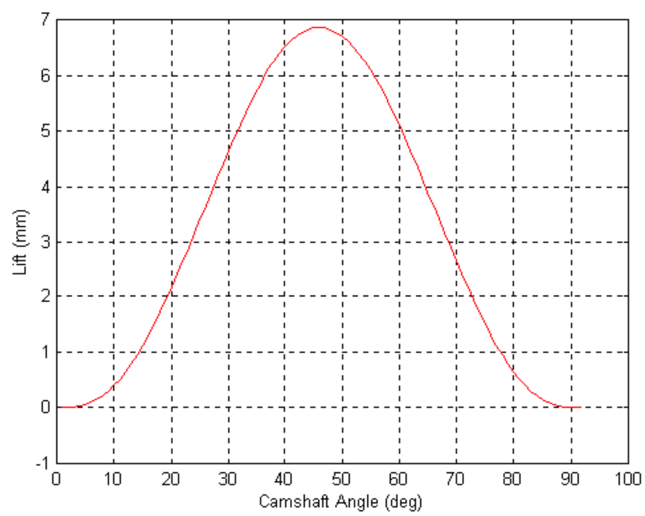

Fig-1. Lift v/s camshaft angle

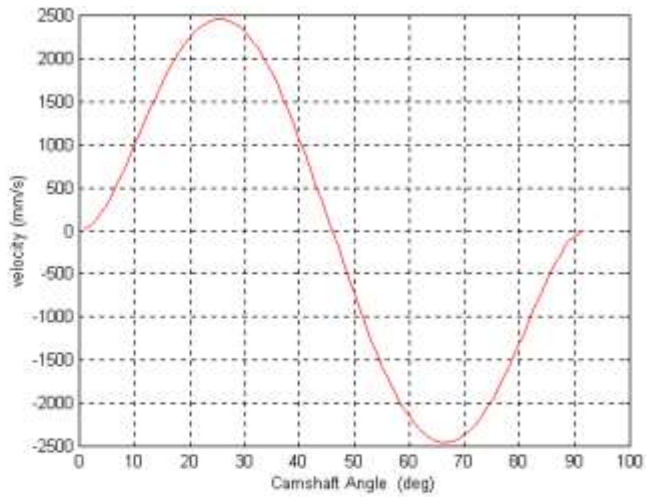

Fig -2. Velocity v/s camshaft angle

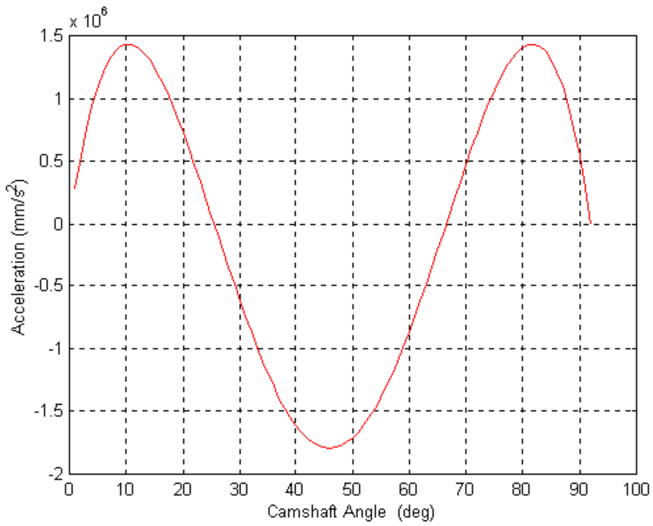

Fig-3. Acceleration v/s camshaft angle

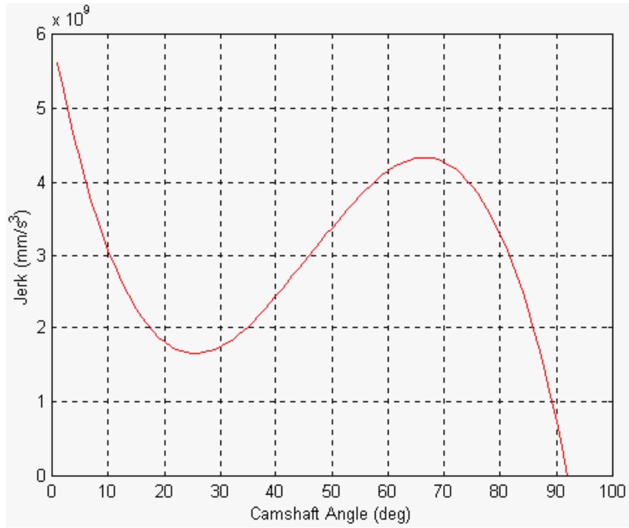

Fig-4. Jerk v/s camshaft angle

DYNAMIC MODEL OF VALVE GEAR TRAIN

Following assumptions are made in development of the model for the dynamic analysis of the valve gear system[1]:

1. All the components are rigid.
2. Tappet and pushrod are coplanar and in the axis of symmetry.
3. Lift delay due to the cam follower inclination is ignored.
4. Valve inclination in the any other plane is ignored.

The current model gives satisfactory results at medium and low speeds since at higher speed vibration affect the performance of engine. Hence in this case, the elasticity and damping does not influence much. The contact forces between the valve gear elements have been determined with the following analysis. A schematic arrangement of cam-follower and pushrod is shown in fig- 5.

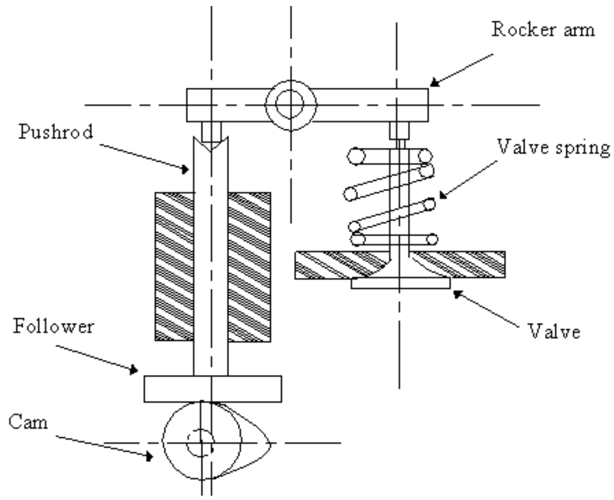

Fig-5. Schematic diagram of pushrod type valve train system with flat-faced follower

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

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

Upward force on the valve due to gas pressure

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

Hence contact force between rocker arm and valve

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

Contact force of bearing of rocker arm

$$F_k = \frac{\left[F_3 R_1\left(1 + \frac{1}{i}\right) + J_k \varepsilon + m_k g e - m_k g R_1\right]}{\left[R_1 - \frac{\mu \times d_k}{2}\right]} \quad (9)$$

Contact force between follower and pushrod

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

Contact force between cam and follower

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

HERTZ PRESSURE

For calculating the Hertz's pressures, the theory of rolling contact [6] has been used.

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

Where

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

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

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

RESULTS

Hertz pressures 441 and 940 MPa have been obtained at 3200 rpm of engine at top of the Inlet and Exhaust cam (45^0 of cam angle) respectively. Maximum shear stress was 282 N/ mm^2 in the exhaust cam.

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

Table 3: Spring force and Inertia force (at the top of the cam)

Speed (rpm)	Spring force (N)	Inertia force (N)	
		Inlet	Exhaust
3200	340	341	553

CONCLUSION

The design of diesel engine valve gear train is so intricate, leading to many problems while analyzing the contact forces. For cam materials, the usual permissible Hertzian pressure is between 850 to 1000 MPa. In this paper, obtained Hertz pressures are below the permissible limit. Exhaust cam had higher stress than inlet cam. Hence at the exhaust cam side, Hertz pressure shall be minimized by modifying the cam profile or a higher surface hardness may be used.

This dynamic model has limitation of damping and vibration effects hence the prediction of wear at higher speeds (above 3200 rpm) may deviate considerably from the actual results, because this has not been included in the analysis.

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