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Empirical Kinematic Design of Cam and Follower System in Valve Gear Train

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ABSTRACT

In this paper, the valve's operating system of push-rod type has been studied and profile of cam and follower has been designed. The follower model in this work is of flat shaped and kinematic and dynamic interaction between cam and follower is valid to various curvatures of followers. The mathematical analysis has been done using MATLAB.

I. INTRODUCTION

The contact of cam with follower [3, 15, 16] involves the intense tribological behaviors due to variation of loads and relative speed. The non-conformal geometry of valve train system [7] gives independent prediction to experimental data. Kinematics and dynamic behavioral pattern of valve train system [11, 13] is generally affected by contact mechanism of cam and follower.

The force required in opening inlet and exhaust valves [11, 15, 16] are much influenced by stiffness, masses and geometries of components and friction existing between mating components. Hence it becomes very vital that design of cam and follower system must felicitate the optimum wear rate, safe contacting stresses and sustenance of required film thickness of lubricant.

Tappet faces are super finished to about 6 micro-in. In a some applications, shot-peened tappet faces seem to retain lubricant slightly better [4, 7, 6]. Oxide coating of super finished tappet faces on chilled iron or phosphate coating of hardened steel or gray iron, improves frictional qualities.

II. CAM PROFILE

A cam acts as a device [7, 11] for converting one motion into another. The rotary motion of the cam is transformed into follower's oscillatory, translatory motion or both.

Cam Kinematics:

The intake and exhaust valves in automobile engines are opened by cams. Unlike linkages, cams can be designed easily [7, 11] but much more difficult and expensive to manufacture. It is four-bar linkage mechanism of which the coupler link is changed by a half joint as shown in figure [1].

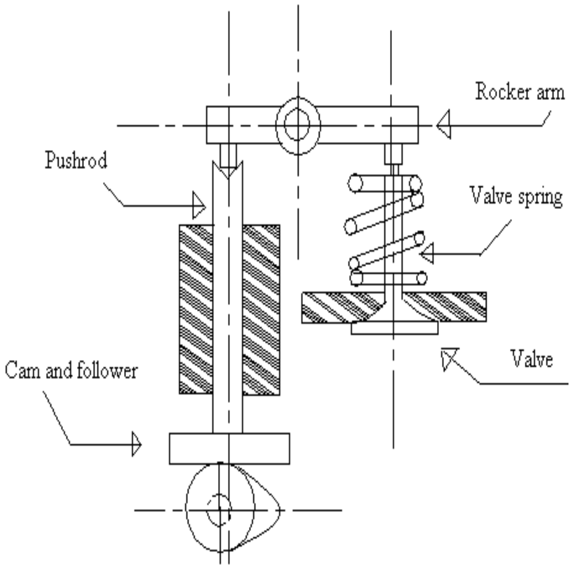

Figure 1: Valve Train System with Flat-Faced Follower

In I C engines, during suction and exhaust strokes, peak accelerations take place on the valves and tappets leading to severe contacts between cams and followers. This subsequently causes wear and tear of cam surface.

Cam Dynamics

While analyzing the dynamics of the cam-follower system, the contacting forces at cam surface, spring forces and tangential forces [13] are to be determined. Usually the speeds are high and members are so elastic that an elastic-body analysis is preferred.

In flat-faced tappet, the pressure angle is zero for all states of cam and follower. As shown in the figure [2], as the contact point changes its location, the point of application of the force between cam and tappet also shifts with it.

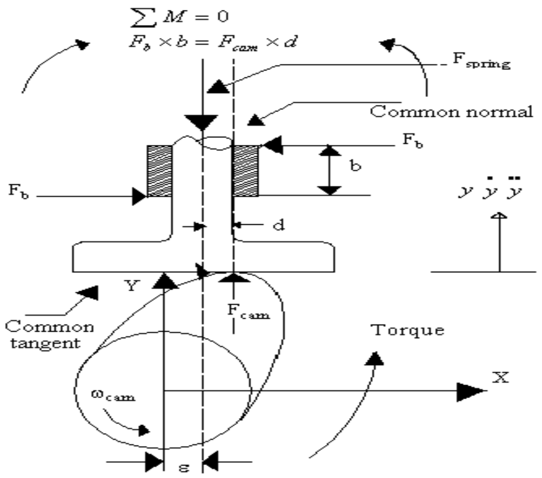

Figure 2: Overturning moment on a flat-faced follower

The overturning moment on the tappet [11] subjected to eccentric force tends to jam it in its guides, similar to large pressure angle in the roller follower. Hence to optimize the moment arm of the force, the cam is kept as small as possible. The radius of curvature of the cam should be opted large enough to have no undercutting.

III. KINEMATICS AND DYAMIC MODELING OF CAM-FOLLOWER SYSTEM

Geometrical Modeling

Since the flat tappet does not follow a concave cam, hence in a flat-faced tappet, negative radius of curvature on the cam [7, 11] should not be provided.

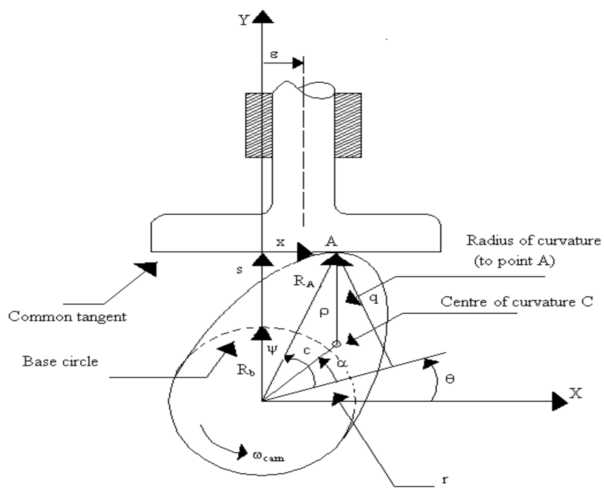

Figure 3: Geometry of Curvature and Cam Profile

Figure [3] shows a cam and flat-faced follower in an arbitrary position. We can define the location of contact point A from two vector loops (in complex notation).

$$R_A = x + j(R_b + s)$$

And

$$R_A = ce^{j(\theta+\alpha)} + j\rho$$

which deduces

$$x = v \qquad \text{.. (1)}$$

$$Facewidth > v_{\max} - v_{\min} \qquad \text{...................... (2)}$$

$$\rho = R_b + s + a \qquad \text{................................ (3)}$$

And the minimum base circle radius can be defined as

$$R_b = \rho_{\min} - s@a_{\min} - a_{\min} \qquad \text{.................. (4)}$$

To apply this equation, a minimum radius of curvature $\rho_{\min}$ must be opted for the cam surface as a design parameter. Because the Hertzian contact stresses at the point of contact are function of local radius of curvature, that criterion can be used to select $\rho_{\min}$.

Cam Contour

The coordinates of the physical cam surface should be provided for a flat-faced follower. Hence from figure-3

Real (x component):

r = (R_b+ s) sinθ + v cosθ .... (5)

Imaginary (y component):

q = (R_b+ s) cosθ - v sinθ .... (6)

Above identities can be used to manufacture and machine the cam in contact with flat-faced follower. None of the above equations involve the eccentricity ε. Geometry of a flat follower cam is not influenced by it.

III. Valve Gear Kinematics Modeling

The general form of polynomial function [15, 16] is:

$$s = C_o + C_1x + C_2x^2 + C_3x^3 + C_4x^4 + ... + C_nx^n \qquad \text{....... (7)}$$

Where

s =follower displacement, and

x is the independent variable, and in this case it will be replaced by $\dfrac{\theta}{\beta}$.

Now the equation of displacement is:

$$s = C_o + C_1\left(\frac{\theta}{\beta}\right) + C_2\left(\frac{\theta}{\beta}\right)^2 + C_3\left(\frac{\theta}{\beta}\right)^3 + C_4\left(\frac{\theta}{\beta}\right)^4 + C_5\left(\frac{\theta}{\beta}\right)^5 \qquad \text{..... (8)}$$

Velocity

$$v = \frac{1}{\beta}\left[C_1\left(\frac{\theta}{\beta}\right) + 2C_2\left(\frac{\theta}{\beta}\right) + 3C_3\left(\frac{\theta}{\beta}\right)^2 + 4C_4\left(\frac{\theta}{\beta}\right)^3 + 5C_5\left(\frac{\theta}{\beta}\right)^4\right] \qquad \text{...... (9)}$$

and

Acceleration

$$a = \frac{1}{\beta^2}\left[2C_2 + 6C_3\left(\frac{\theta}{\beta}\right) + 12C_4\left(\frac{\theta}{\beta}\right)^2 + 20C_5\left(\frac{\theta}{\beta}\right)^3\right] \qquad \text{.. (10)}$$

Assuming suitable boundary conditions and solving above equations (8, 9, 10) into a matrix form by putting values of C_0, C_1, C_2

We have

$$C_3 = 10L;$$

$$C_4 = -15L;$$

$$C_5 = 6L$$

Hence, the equation for the five-order Polydyne cam design is:

Lift,

$$Y = L\left[10\left(\frac{\theta}{\beta}\right)^3 - 15\left(\frac{\theta}{\beta}\right)^4 + 6\left(\frac{\theta}{\beta}\right)^5\right](0 \le \theta \le \beta) \quad \text{......(11)}$$

Velocity,

$$V = L\omega\left[30\left(\frac{\theta^2}{\beta^3}\right) - 60\left(\frac{\theta^3}{\beta^4}\right) + 30\left(\frac{\theta^4}{\beta^5}\right)\right] \quad \text{...... (12)}$$

Acceleration,

$$A = L\omega^2\left[60\left(\frac{\theta}{\beta^3}\right) - 180\left(\frac{\theta^2}{\beta^4}\right) + 120\left(\frac{\theta^3}{\beta^5}\right)\right] \quad \text{.......(13)}$$

Jerk,

$$J = L\omega^3\left[\frac{60}{\beta^3} - 360\left(\frac{\theta}{\beta^4}\right) + 360\left(\frac{\theta^2}{\beta^5}\right)\right] \quad \text{...(14)}$$

Where N = Rpm

β = Constant (total angle of rise or fall interval)

L = Total lift or rise

Jerk causes shock loads and undue vibrations and stresses.

IV. RESULT

The kinematic design of cam and follower has grave concern in view of tribological aspects such as wear and lubrication point of view since these components are always under immense contact stresses. Empirical relations for polynomial cam and flat faced follower have been deduced from suitable geometric consideration.

V. CONCLUSION

This paper could be very useful for the analysis of forces acting on cam and follower which leads to wear and tear of mating components and lubrication problems as well. The analysis can be extended to other shapes of cam and follower combinations. Further, vibration effects of cam-follower may also be considered for kinematics of system.

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