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DESIGN OF POLYDYNE CAM FOR HIGH SPEED I.C. ENGINE

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ABSTRACT

Cams play very important role in the I.C. engine valve gear mechanisms. Dynamic behaviour of cams for high speed engine causes cross over shock, jump, follower spring surge etc. A proper selection of suitable polydyne cam will reduce these adverse effects, to a large extent. In this paper, an attempt has been made to suggest a best suited cam profile for the valve gear mechanism of the existing high speed IC engine. A mathematical model of the cam is generated. A source code is developed in C language, to compute the displacement, velocity and acceleration of the cam follower. The methodology adopted tests the suitability of the polynomial cam profile for a given engine specifications and suggests modifications. The results are shown graphically and are also compared with the existing cam profile.

Key words: Polydyne cam, displacement, velocity, acceleration of cam, profile, dynamic behaviour, under cutting, jump of follower.

1. INTRODUCTION AND REVIEW

Present day requirements usually dictate that the machine components having relative motion should have a predetermined exact path, in order to achieve more and more accuracy. A cam mechanism is one answer for such requirements and plays an important role in the design of typical application like IC engine valve gear. Proper design of the cam and its profile adds a lot to the performance of engine.

In early days cam design was done mainly by kinematic analysis. This was carried out by choosing trigonometric equations. These curves were suitable and used because of their simplicity of construction and ease of analysis. However, for many machine requirements which are operated at high speeds, their performance was inadequate. For example, the dynamic behaviour of high speed IC engine is severely affected by the elasticity of links, dynamic forces and excessive amplitudes of vibration. Thus, valve gear dynamics became an important research field. Use of polydyne cam is one of the solutions evolved.

For many years, cam design has been improved in various aspects. Duadley[1] first suggested and used the polydyne method to design automobile valve cams. He developed the cam contour, which can be computed to produce definite predictable valve motion in high-speed engines. Theorn *et al*, showed that the cams designed by polydyne approach produce actual valve motion at design speed which is very close to the desired valve motion. It was followed by Barkan[3] who derived theory and experimentally proved the valve motion prediction from cam design.

Kanazaki and Itao[4], made polydyne cam design for type head position in high speed teleprinters during 1972. They found that polydyne cams were devised to reduce residual vibrations of a single degree of freedom follower system. F.Y.Chen[5] developed a digital computer program for analysing the non-linear cam driven mechanisms. He also presented design strategies and a rational procedure of closed loop computer-aided design by preparing a flow chart for the same.

The polydyne cam combines the polynomial equation with the dynamics of a follower system. The result is an excellent approach to a high speed, highly flexible system. The polydyne method was originally presented by Daudley and elaborated by Stoddart. This approach recognizes that much faulty operations of high speed, highly flexible systems can be attributed to the difference between cam command and follower mass response.

Basic advantages of polydyne cam are:

a) By direct means it can eliminate jump.
b) By direct calculation it provides the only means of controlling the exact position of the follower end
c) It limits vibrations to a minimum amplitude if run at the design speed.

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2. NOTATIONS

Y = follower displacement at cam rotation θ
Y' = velocity of the follower at cam rotation θ
Y'' = acceleration of the follower at cam rotation θ
Yc = cam displacement at cam rotation θ
Yc' = velocity of cam at cam rotation θ
Yc'' = acceleration of cam at cam rotation θ
Bcr = base circle radius, mm
e = eccentricity, mm
h = lift of the follower, mm
β = cam rotation during rise/fall of the follower in deg
r_a = ramp height, mm
r_s = initial static deflection of linkage, mm
r_k = clearance or backlash in the linkage, mm
L = external load acting on the follower, N
S1 = initial compression spring force, N
k_f = spring rate of follower linkage, N/mm
k_r = equivalent spring ratio of the follower linkage
k_s = spring rate of compression spring, N/mm
C = dynamic constant
m = w/g, equivalent mass at the follower end, kg
w = equivalent weight at the follower end, N
N = cam speed, rpm
lever arm ratio = (d1/d2)
C_p, C_q, C_r, C_s .. = constants
p, q, r, s .. = polynomial terms

3. MATHEMATICAL MODELLING

The class of polynomial functions is one of the more versatile types that can be used for cam design. The general form of a polynomial function is,

$$Y = C_0 + C_p^* \theta^p + C_q^* \theta^q + C_r^* \theta^r + C_s^* \theta^s + \ldots \ldots\ldots\ldots \quad (1)$$

The derivatives of the above polynomial equation are;

$$Y' = dy/d\theta = pC_p^* \theta^{p-1} + qC_q^* \theta^{q-1} + rC_r^*\theta^{r-1} + sC_s^* \theta^{s-1} + \ldots$$

$$Y'' = d^2y/ d\theta^2 = p(p-1)C_p^* \theta^{p-2} + q(q-1)C_q^* \theta^{q-2} + r(r-1)C_r^* \theta^{r-2} + s(s-1)C_s^* \theta^{s-2} + \ldots$$

$$Y''' = d^3y/ d\theta^3 = p(p-1)(p-2)C_p^* \theta^{p-3} + q(q-1)(q-2)C_q^* \theta^{q-3} + r(r-1)(r-2)C_r^* \theta^{r-3} + s(s-1)(s-2)C_s^* \theta^{s-3} + \ldots$$

$$Y'''' = d^4y/ d\theta^4 = p(p-1)(p-2)(p-3)C_p^* \theta^{p-4} + q(q-1)(q-2)(q-3)C_q^* \theta^{q-4} + r(r-1)(r-2)(r-2)C_r^* \theta^{r-4} + s(s-1)(s-2)(s-2)C_s^* \theta^{s-4} + \ldots$$

Considering the boundary conditions,
at $\theta=1$, $Y=Y'=Y''=Y'''=0$;
at $\theta=0$, $Y=1$ and $Y'=Y''=Y'''=0$; $C_0=1$; the coefficients become,

$$Cp = C_0*q*r*s / ((q-p)(r-p)(s-p))$$
$$C_q = C_0*p*r*s / ((p-q)(r-q)(s-q))$$
$$C_r = C_0*p*q*s / ((p-r)(q-r)(s-r))$$
$$C_s = C_0*p*q*r / ((p-s)(q-s)(r-s))$$

The dynamic equation for the rise of cam in mm, is

$$Yc = r_a + k_r Y + C Y' \quad \ldots 2$$

Where, $r_a = r_s + r_k$
$r_s = (L+S1)/k_f$
$k_r = (k_s + k_f)/k_f$
$C = 36mN^2/k_f$

The derivates of the above dynamic equation are,

$$Yc' = k_r Y' + C Y''$$
$$Yc'' = k_r Y'' + C Y'''$$
$$Yc''' = k_r Y''' + C Y''''$$

In general sense, a model is a conceptualisation of real system. If the quantitative behaviour of the model accurately represents the behaviour of the real system, the designer then has a concise method of describing the system. Fig.1 shows the physical model of pitter type IC engine valve gear system. This physical model can be reduced to one degree of freedom as shown in Fig.2.

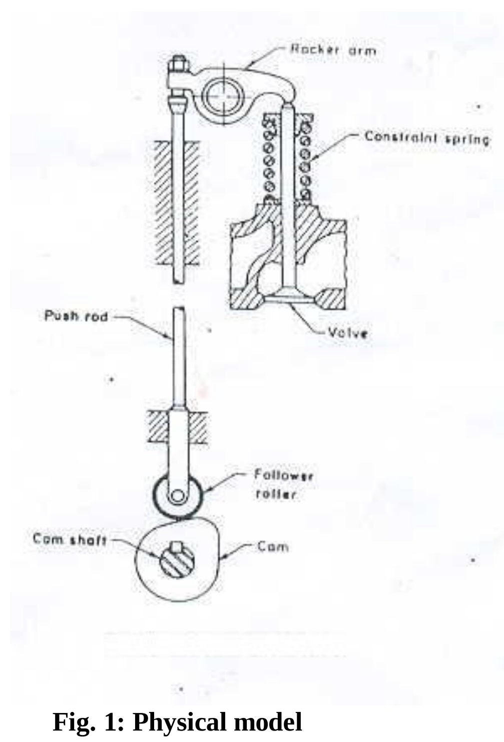

Fig. 1: Physical model

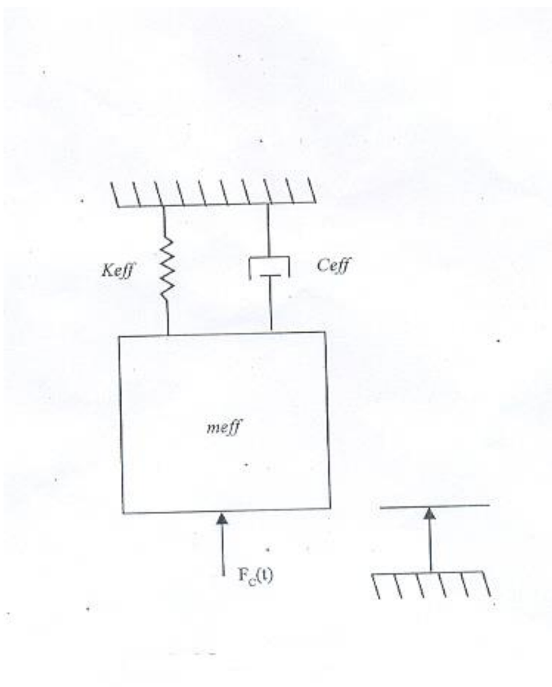

Fig.2: One degree of freedom model

In the mathematical modelling of cam follower system it has been a common practice to reduce it to a simple spring mass system. The values of the effective mass and the spring stiffness can be calculated based on the principle of kinetically equivalent systems.

Usually as a first step towards elastic analysis, single degree of freedom is used. If there is a significant difference between the actual output parameter and the theoretical output parameter as determined by single degree of freedom analysis, one has to investigate the problem by using two degrees of freedom model.

4. ANALYSIS

In the present analysis, a suitable follower motion Y versus cam angle θ is assumed and the cam profile Y_c is developed.

Choosing a proper polynomial and establishing the follower system flexibility, using Eq.2 the displacement, velocity and acceleration are determined.

Then, sizing of the cam has been done, where the base circle radius and the follower offset are found, after satisfying the following conditions:

a) Controlling the pressure angle between the limits to prevent jamming of the follower stem and its guide and to minimise thrust against the follower.
b) Minimizing the radius of curvature of the cam profile to avoid under cutting, which result in incorrect follower movement and produce large contact stresses between the cam and the follower.

As the smallest radius of curvature occurs at the point of maximum negative acceleration, the undercutting may result at higher negative acceleration. Hence to prevent under cutting the following equation is to be tallied.

$$[(r_a + Y)^2 + (Y')^2]^{3/2} / [(r_a + Y)^2 + 2(Y')^2 - (r_a + Y)\, Y''] \geq 1$$

c) Eliminating jump phenomenon, which occurs during the negative acceleration. With the jump, the cam and the follower separate owing to excessively unbalanced forces exceeding the spring force. Hence, it is suggested to use larger initial spring force and larger ramp.

Significant features of the polydyne program developed in this work are,

a) This is a generalised program for designing cam profiles following any degree polynomial.
b) This program calculates maximum pressure angle on the cam profile and ensures that it is lying between 30 to 40 degrees. If the maximum pressure angle is below 30 deg then the base circle radius is reduced slightly or else if the maximum pressure angle is above 40 deg then the base circle radius is increased slightly. This modification of base circle radius is repeated until the maximum pressure angle is falling in between 30 to 40 degrees.
c) This program checks and indicates warning message about the phenomenon of under cutting on the cam. It calculates the radius of curvature and verifies whether it is more than 1.
d) This program also ensures that the jump of the follower does not take place. If jump occurs then it increases the minimum required spring force to eliminate jump phenomenon.

The result obtained from this polydyne program is converted to curves for displacement, velocity and acceleration with the help of MS-excel and corresponding cam shapes are plotted using Sigmaplot software.

5. RESULTS AND DISCUSSION

The program is tested for different data and a specimen data is illustrated below.
Input:

- Cam base circle radius Bcr = 15mm and eccentricity e = 0mm
- Speed N = 1000 rpm
- Lift h = 7.8 mm
- Spring rate of follower linkage k_f = 23950 N/mm
- Spring rate of compression spring k_s = 0.56784 N/mm
- External load L = 49.05 N
- Initial compression spring force S1 = 196.2 N
- Lever arm ratios d1 = 42mm, d2 = 52mm
- Effective weight of follower on cam & follower end w1 = 3.122N, w2 = 2.043N
- Cam angle rotation β = 50°
- Dwell period in deg = 5
- Polynomial terms = 2 [1-2]

When the input data is tried for (1-2) polynomial, the output of the program is as follows.

- Coefficients of polyterms = 2, 1
- k_r =1.000024 w=0.449204 c=0.675212 r_a= 0.001024
- Pressure angle = 38.098
- Radius of curvature for cam profile = 35.1482mm
- Undercutting does not occur.
- Follower will not jump.

Figs.3, 4 and 5 shows displacement, velocity and acceleration curves during the rise respectively and Fig.6 shows the final cam profile for the given data.

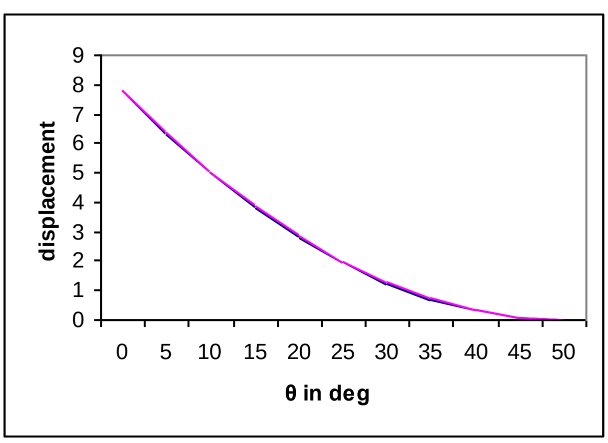

Fig. 3: Displacement curve

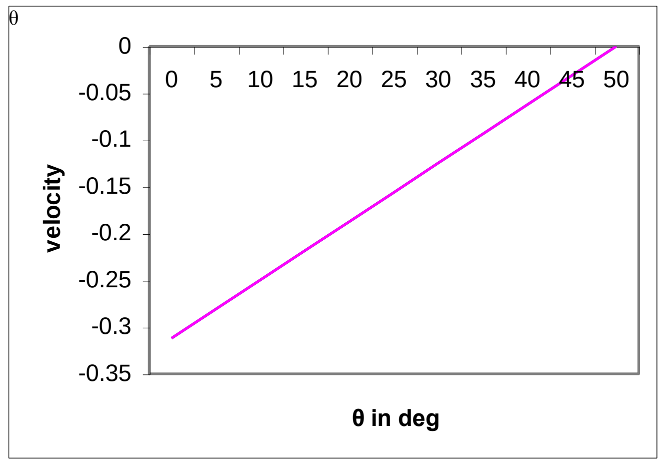

Fig. 4: Velocity curve

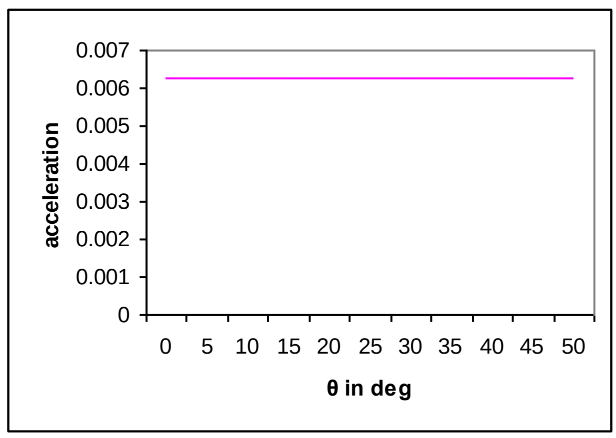

Fig. 5: Acceleration curve

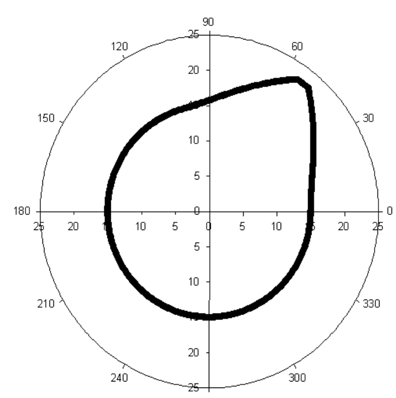

Fig. 6: Cam profile

The same input data is also tested with different polynomial terms, viz., (2-3), (3-4-5) and (4-5-6-7). It is observed that the cam profile for (1-2) polynomial is simple and the curve is steep at total rise period. However, it is observed that the cam displacement curves are becoming smoother with the increase of polynomial order. This results into slower initial and final displacement of the follower during rise and fall, with the increase of order of polynomials.

6. CONCLUSION

The software developed is tested to design the polydyne cam profile suitable for a high-speed diesel engine. Besides this, the software is tested for sensitivity analysis of valve spring force at different speeds. The software developed can be successfully used to design the polydyne cam profile for any high speed engine. It is observed that the computer generated cam profile is closely in agreement with the profile of existing cam used for high speed diesel engine. It is noted that curvature of rise and fall part of the cam profile becomes more and more concave as the degree of polynomial increase, leading to increase of area under displacement curve. From the sensitivity analysis, it is seen that (3-4-5) in addition to (1-2) polydyne cam is also suitable for the specific engine considered here.

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