Cam Design using Polydyne Approach

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Abstract – *The valve train often faces a challenge with noise and vibration due to the variation which exists in the actual cam profile and the calculated cam profile. In conventional approaches, the design of cam includes only valve lift curve and excludes stiffness as well as the elasticity in the linkages which causes discrepancy in the cam command and follower response. Therefore, Polydyne Cam approach can be used which accounts for the elasticity and stiffness in the design process and in turn reduces this discrepancy.*

*Key Words***: Valvetrain, Cam, Follower, Polydyne**

1. INTRODUCTION

The Valve train is an essential mechanical system that controls the operation of intake and exhaust valves in an internal combustion engine. The stiffness of valve train has significant impact on the operation of the engine. The analysis of the valve trains mainly relies on valve acceleration data, because it provides information on the dynamic characteristics. Cam and follower are an integral mechanism in the valve train. Cams are used to convert the rotary motion to linear motion. As cam rotates, the follower rises and falls in a process known as reciprocating motion. Despite its advantages, cams have severe limitations namely that they cannot transmit large forces and significant additional acceleration forces are generated when the cam is operated at high speed. The inherent flexibility of a cam device may induce unwanted vibration of the follower, which will reduce positional accuracy and cause increased forces, noise and operating cost. Because the property of cam profile directly affects the performance of the cam mechanism, numerous cam profiles have been proposed to reduce vibration of cam follower system. However, in many cases, cams are subjected to high operating speed, which makes the matter of vibration reduction tougher. In such situations, the factors of elasticity and backlash must be taken into consideration if vibration and impact loads are to be avoided and minimized. This can be achieved by using Polydyne cam design method.

1.1 Literature Review

Preben W. Jensen [1] in his book proposes clear introduction to those problem-solving methods used in the design, application, and manufacture of cams based for the most part on exhaustive studies of the English and German literature on the subject, the book unifies this scattered information in the single practical treatment, concentrating on both the graphical and analytical methods needed to design and produce cams and cam systems. It also focuses on the synthesis and analysis of polynomial equations for follower motion in the foregoing literature. A comparative study between 3-4-5, 4-5-6-7 and 5-6-7-8-9 polynomial curves has been depicted. Harold A. Rothbart [2] discusses the basics of cam profile the theoretical and practical design considerations for high-speed cam-follower performance. He has also said that the maximum acceleration values of the cam should be as small as possible to give small inertia loads. Stoddart David A. [3] based on polynomial equations offers a versatile and comprehensive approach encompassing the dynamic aspects of machine operations while designing cams using Polydyne Cam method. The effects of dynamic factors on cam design and operation are evaluated and specific example to demonstrate practical application of polydyne approach is demonstrated. Tushar Kiran [4] has presented analyses of 2-3 polynomial cam profile, 3-4-5 polynomial cam profile and 4-5-6-7 polynomial cam profile are presented. Kinematic and dynamic analyses are carried out using motion equations. The kinematic analysis presents follower characteristics of displacement, velocity and acceleration. Dynamic analysis presents pressure angle, spring force, inertial force and resultant force. Combined plots enlisting the follower characteristics of displacement, velocity and acceleration are presented for above mentioned polynomial cam profiles. A. S. More [5] has discussed valve train analysis procedures that are carried out in two stages kinematic and dynamic analysis. Kinematic analysis is used for design of a valve lift profile and find out static forces and Oil film characteristics, etc. Dynamic analysis is used to determine the dynamic movement of valve train component considering the effect of inertia and stiffness. Dong-Joon Chun [6] discussed the mass property data associated with the tuning of the valve train that can be

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measured, but the stiffness data associated with the tuning cannot be measured; it can instead be calculated using the natural frequency and deformation of the rocker arm, cam shaft, rocker arm shaft, mounting brackets, etc. R. L. Norton [7] has taken the subject from an introductory level through advanced topics needed to properly design, model, analyze, specify, and manufacture cam-follower systems. Beginning with a description of "how not to design a cam" in order to point out pitfalls that may not be obvious to the beginner, the proper way to design a cam for multiple and single-dwell situations is developed intimately. All the suitable (and some unacceptable) classical cam functions are described and their mathematics defined for the common double-dwell application. Polynomial functions are introduced and used for both double- and single-dwell examples.

1.2 Problem Statement

In high speed valve train systems, a discrepancy is observed between cam command and follower mass response due to which vibrations are produced. This phenomenon can be minimized or avoided by taking into account the elasticity in the linkages which is possible by using Polydyne method.

2. POLYDYNE APPROACH

The Polydyne Cam Approach combines the polynomial equation with the dynamics of a follower system; the result is an excellent approach to a high speed, highly flexible system. This method consists of two parts 1. The use of polynomials to obtain a wide variety of curves having certain characteristics, and 2. Modification of the polynomials to take into account any elasticity of the system which causes a discrepancy between the cam command and its follower mass response, The polynomial profile created by this method extends the control feature producing zero jerks at the end. Basic advantages of this approach are:

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

By using Polydyne cams residual vibrations are extinguished at plural arbitrarily chosen rise times and reduced over a comparatively wide range near these rise times. This can be obtained by determining polynomial equations of a higher order than the number of boundary conditions of the output displacement curve so as to increase the degree for the design.

3. DESIGN AND INPUT PARAMETERS

Engine Specifications:

*   Capacity: Single Cylinder, 435 cc
*   Max Power: 6 kW, 8.5 HP @3600 rpm
*   Max Torque: 18-19 Nm @3600 rpm
    Valve Timing Diagram:
*   IVO= 13° before TDC
*   IVC= 27° after BDC
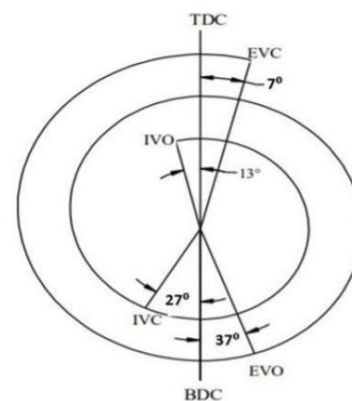*   EVO= 37° before BDC
*   EVC= 7° after TDC

Fig.1 Valve Timing Diagram

Valve Train Data:

• Cam Design speed: 2106 rpm
• Valve Train Stiffness: 5250 N/mm
• Tappet end effective mass: 19.92 gm
• Spring Preload: 226 N
• Spring Stiffness: 22.54 N/mm
• Valve Lash: 0.15 mm
• Rocker Ratio: 1.54
• Base circle Radius: 14 mm
• Contact width 8.5 mm
• Modulus of Elasticity of Cam: 210000 N/mm²
• Modulus of Elasticity of Follower: 210000 N/mm²

3.1 Selection of Polynomial Equations

Polynomial function is given by y = f(x)

Here f(x) = $C_0 + C_p x^p + C_q x^q + C_r x^r + C_s x^s + \ldots + C_n x^n$ …. (i)

The values of coefficients are given by

$$C_p = -C_0 \frac{qrst \ldots}{(q-p)(r-p)(s-p)(t-p)..} \text{ and so on}$$

Therefore, by selecting 5-6-7-8-9 polynomial, the general equation is given by:

$$y = 1 - 126x^5 + 420x^6 - 540x^7 + 315x^8 - 70x^9$$

Similarly, the equation for velocity, acceleration and jerk are as follows:

$$y' = -630x^4 + 2520x^5 - 3780x^6 + 2520x^7 - 630x^8$$
$$y'' = -2520x^3 + 12600x^4 - 22680x^5 + 17640x^6 - 5040x^7$$
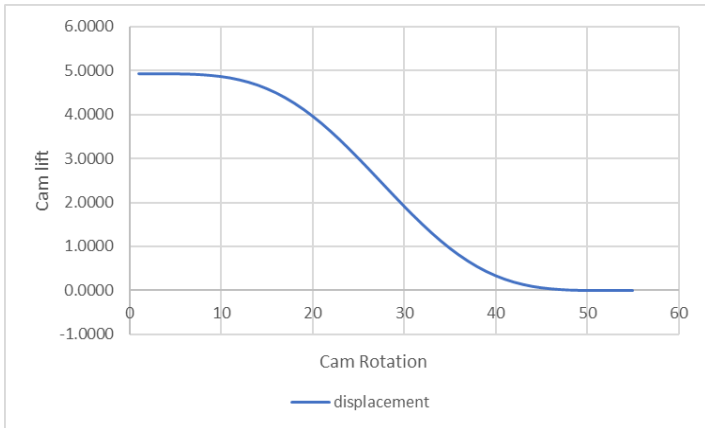$$y''' = -7560x^2 + 50400x^3 - 113400x^4 + 105840x^5 - 35280x^6$$
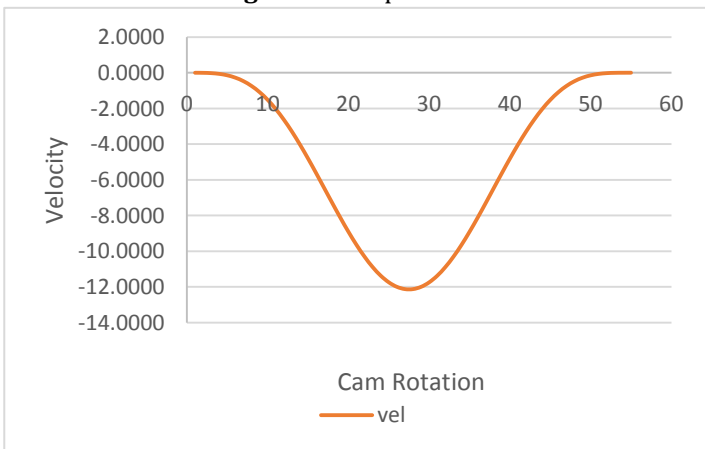

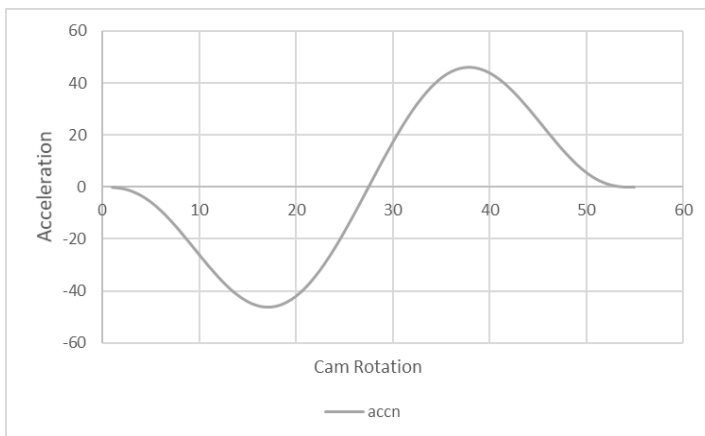**Fig 2.** Cam Displacement

Fig 3. Cam Velocity

Fig 4. Cam Acceleration
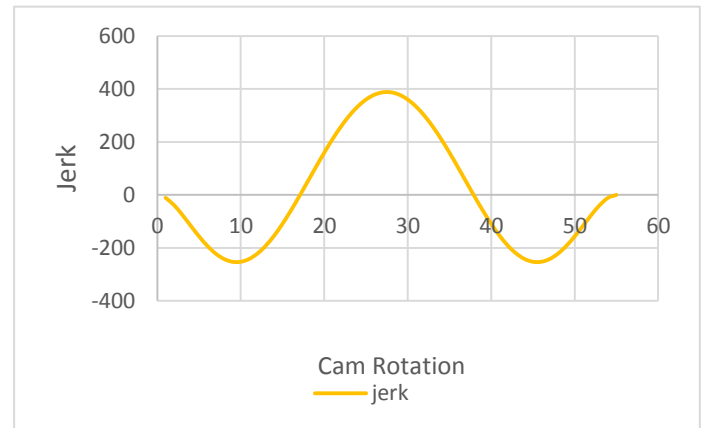
Fig 5. Cam Jerk
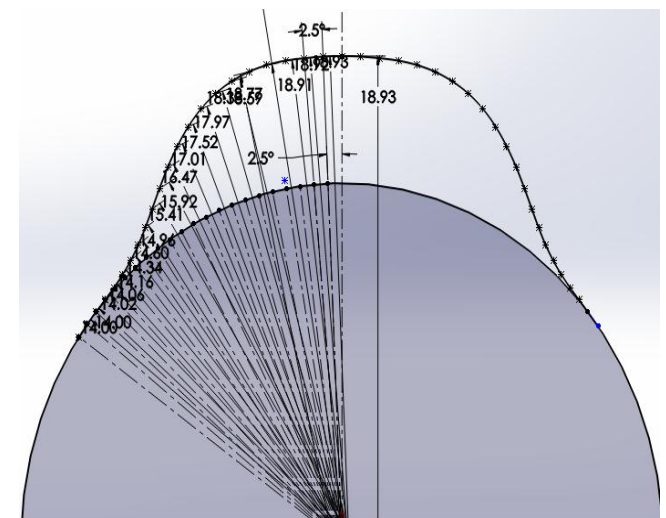
Fig 6. Cam Contour created by using 5-6-7-8-9 Polynomial

3.2 Design
3.2.1 Pushrod
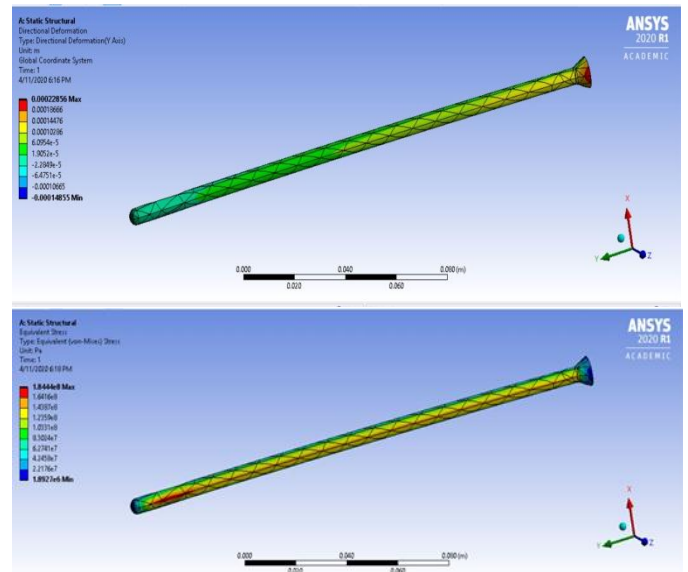
Fig 7. Deformation and stress distribution of Pushrod

The theoretical deformation of the pushrod is found to be 2.8 x 10⁻¹ mm while the maximum stress induced in the pushrod is found to be 184 MPa.
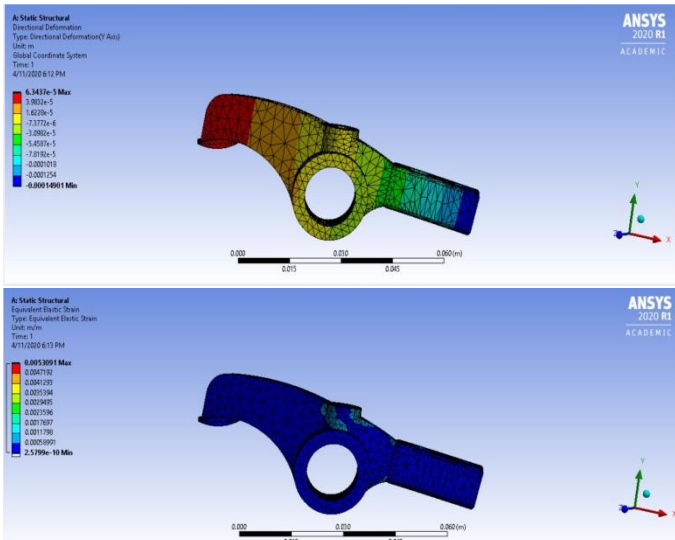
3.2.2 Rocker Arm

Fig 8. Deformation and strain distribution of Rocker Arm

The analysis shows that the maximum deformation of the rocker arm is found to be 14.9 x 10⁻² mm while the maximum strain induced is 0.000530.
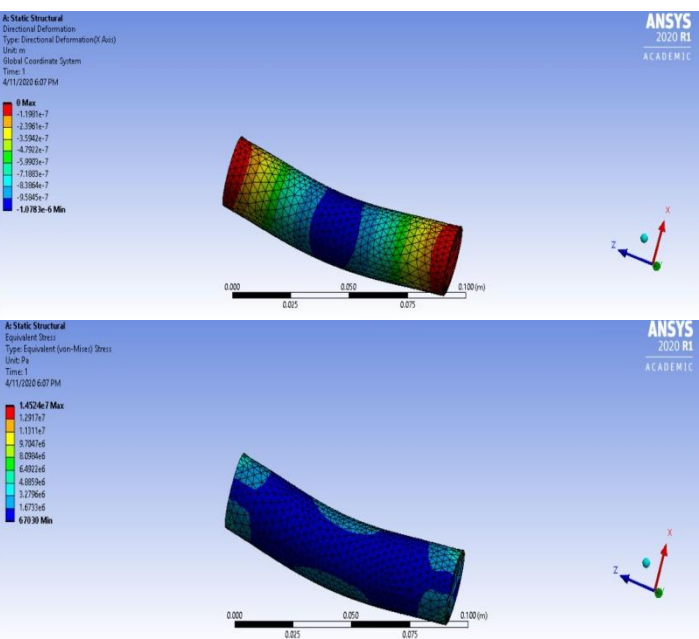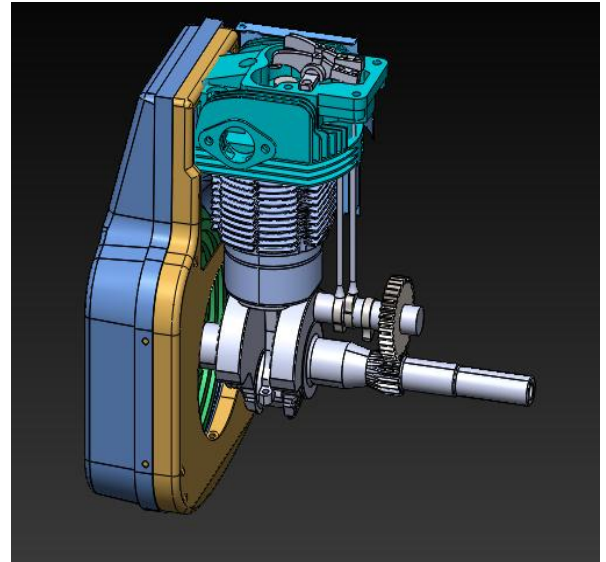
3.2.3 Camshaft

Fig 9. Deformation and stress distribution of camshaft

The theoretical maximum deflection of the camshaft is found to be 1.07 x 10⁻³ mm and the maximum stress induced is 14.5 MPa by using Ansys Workbench.

3.3 CAD Model

Fig 10. CAD Model of Greeves GL435 Engine

4. RESULTS

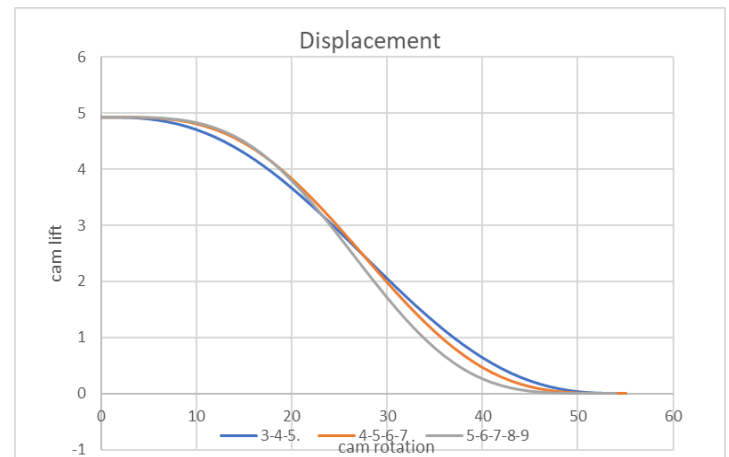Comparison between 3-4-5, 4-5-6-7 and 5-6-7-8-9 polynomial curves:
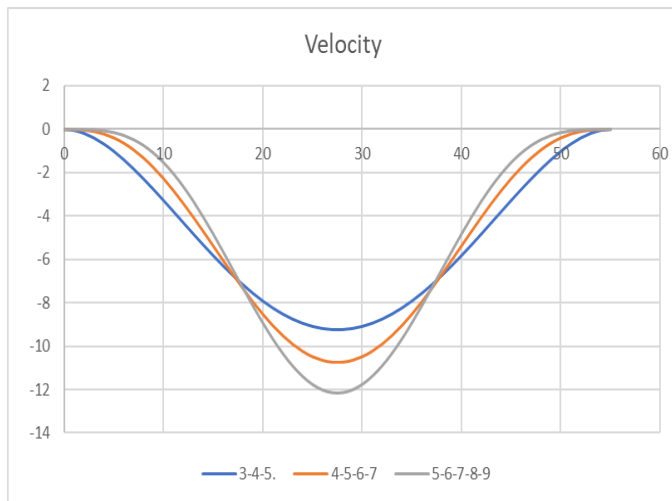

Fig 11. Cam Displacement Comparison

Fig 12. Cam Velocity Comparison
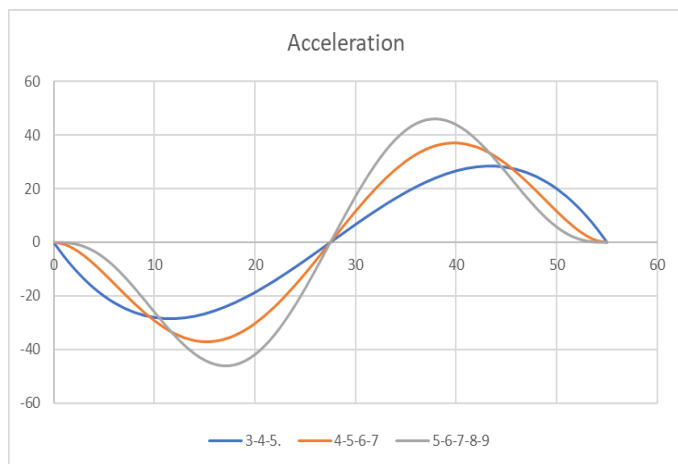

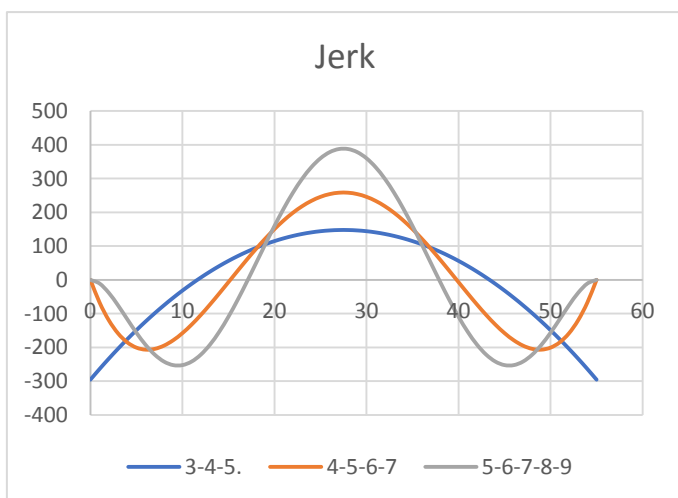**Fig 13.** Cam Acceleration Comparison

Fig 14. Cam Jerk Comparison

5. CONCLUSIONS

1. The 5-6-7-8-9 polynomial showed better kinematic characteristics and yielded smoother curves during the opening and closing of valves than the lower order polynomials like 4-5-6-7 and 3-4-5 polynomials.
2. The Design is safe as per the design validation done through mathematical modelling and theoretical calculations.
3. The deformation of the components of valve train such as pushrod, camshaft, rocker-arm were found to be safe using numerical computation as well as Finite Element Analysis.

Table 1. Deformation of the valve train components (all dimensions in mm)

Part	Numerical	FEA	Design
1. Camshaft	$8.6578 * 10^{-3}$	$1.07 * 10^{-3}$	Safe
2. Rocker Arm	$7.5 * 10^{-2}$	$14.9 * 10^{-2}$	Safe
3. Pushrod	$9.17 * 10^{-2}$	$2.8 * 10^{-1}$	Safe

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