IOTA-Vol-1,
Issue -2, 2015

Modelling for Maximization of Useful Life of a Disc Cam under Multiple Combinations of Pressure Angle and Speed

Koushik Choudhury[1], Prithwiraj Jana[2], Pooja Mazumder[3]

[1]Indian Institute of Science, Bengaluru, India
[2,3]Industrial Engineering and Management, West Bengal University of Technology, West Bengal, India

dextar12@gmail.com[1]
prithwiraj.janahit@gmail.com[2]
mazumder.pooja@gmail.com[3]

ABSTRACT: This research project ventures one of the major concerns in the field of internal combustion engine, which is to predict the service life of cam (here in this research simple harmonic motion (SHM) cam is taken), used in automobile. This project addresses the phenomenon of characterizing the stress analysis of the ground cam and to calculate the service life of 3 cams (camA, camB, camC) where each of them has different pressure angle and hence, the cam with maximum fatigue life is selected.

Keywords: SHM cam, stress analysis, fatigue life, life cycles

INTRODUCTION

Among the different factors that have been considered, while designing the cam, the base circle and the material of the cam have been kept fixed and the pressure angle has been varied. With three different values of the pressure angle, three different cams have been designed and stress analysis have been done in Autodesk AutoCAD 2013 and in Autodesk Inventor 2013 and plotted in the graph. The mean stress and the amplitude of the stress are found and with the help of these values the S-N (stress amplitude versus number of cycles) diagram is drawn and ultimately the fatigue life cycle of the cams is calculated and ultimately the cam with maximum service life is found to be best suited for engines.

METHODOLOGY

Cam-Follower is very important machine to actualize accurate positive-motion drive of fuel-injection pump. The wear of Cam-Follower would worsen rolling contact and would cause increased probability of surface fatigue. This could emerge more serious devastating effects. It is necessary to carry out high pressure-dynamic contact stress study to improve contact capability and extend the service life. Cam is mounted or fixed on a camshaft, which gets its motion from the rotation of flywheel by coupling it with a toothed wheel mounted on crankshaft. As flywheel rotates at high speed therefore a smaller wheel is coupled with a big wheel so that its motion can be reduced.

DESIGN OF CAM

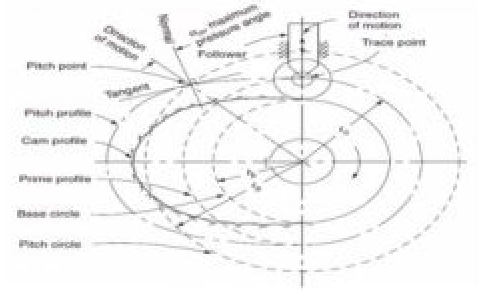


Figure 1: CAM

Base Circle: It is the smallest circle, drawn tangential to cam profile. The base circle decides the overall size of the cam and thus is fundamental feature.

Trace Point: It is a point on the follower, and its motion describes the movement of the follower.

Pitch Curve: If we hold the cam fixed and rotate the follower in a direction opposite to that of the cam, then the curve generated by the locus of the trace point is called pitch curve.

Pressure Angle: The angle between the direction of the follower movement and the normal to the pitch curve at any point is called pressure angle. Pressure angle varies from maximum to minimum during complete rotation. Higher the pressure angle higher is side thrust and higher the chances of jamming the translating follower in its guide ways.

The pressure angle should be as small as possible within the limits of design. The pressure angle should be less than 450 for low speed cam mechanisms with oscillating followers, whereas it should not exceed 300 in case of cams with translating followers. The pressure angle can be reduced by increasing the cam size or by adjusting the offset.

Pitch Point: Pitch point corresponds to the point of maximum pressure angle, and a circle drawn with its centre at the cam centre, to pass through the pitch point, is known as the pitch circle.

Prime Circle: The prime circle is the smallest circle that can be drawn so as to be tangential to the pitch curve, with its centre at the cam centre.
picture given in Fig 1.

Working curve: The working surface of a cam is in contact with the follower. For the knife-edge follower of the plate cam, the pitch curve and the working curves coincide. In a close or grooved cam there is an inner profile and an outer working curve.

Pitch circle: A circle from the cam centre through the pitch point. The pitch circle radius is used to calculate a cam of minimum size for a given pressure angle.
Stroke or throw: The greatest distance or angle through which the follower moves or rotates.
Follower displacement: The position of the follower from a specific zero or rest position (usually it's the position when the follower contacts with the base circle of the cam) in relation to time or the rotary angle of the cam.

Kinematic Coefficients OF Cam

The displacement diagram is a plot of the cam displacement vs the cam angle e.g. $y = f(\theta)$ It is possible to plot additional graphs as follows

The First order Kinematic Relationship
$f'(\theta) = dy/d\theta$

This is a plot of the slope of the displacement graph and thus the rate of movement of the follower. High values of $f'(\theta)$ result in very steep cam slopes with a risk the follower will jam.

The Second order Kinematic Relationship
$f''(\theta) = d^2y/d\theta^2$

This is related to the curvature of the cam. If $f''(\theta)$ becomes very large the curvature of the cam approaches zero (a point). This is highly unsatisfactory as it results in very high contact stresses and consequent wear.

Cam is mounted or fixed on a camshaft, which gets its motion from the rotation of flywheel by coupling it with a toothed wheel mounted on crankshaft. As flywheel rotates at high speed therefore a smaller wheel is coupled with a big wheel so that its motion can be reduced.

The base circle of a cam can be divided into four parts that is angle of ascent, angle of dwell, angle of descent and angle of dwell again. During the first sector. i.e. angle of ascent, cam lifts the follower which with the help of arm and guide opens the valve then period of dwell comes into picture in which there is negligible displacement of follower takes place and the valve remains open for some time. During third sector. i.e. angle of descent, cam descents down and also the follower which closes the valve and then again dwell period comes into picture as a result valve remains closed for some time .These events occur in a cycle and in this way cam functions.

For SHM cam

Let s= follower displacement (instantaneous)
h= maximum follower displacement
v= velocity of the follower
f= acceleration of the follower
θ= cam rotation angle (instantaneous)
φ= cam rotation angle for the maximum follower displacement
β= angle on the harmonic circle

Construction: the follower rises through a distance h while the cam turns through an angle φ. Construction of the follower displacement curve as follows:

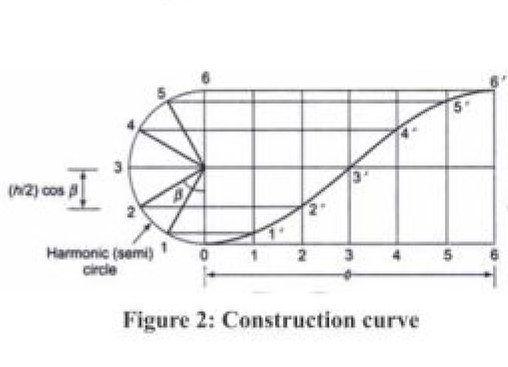


Figure 2: Construction curve

1. Draw a semicircle with cam rise (or fall) as the diameter. This is, usually known as the harmonic (semi) circle. Divide this semicircle into n equal arcs (n even).

2. Divide the cam displacement interval into n equal divisions.

3. Project the intercepts of the harmonic semicircle to the corresponding divisions of the cam displacement interval.

4. Join the points with a smooth curve to obtain the required harmonic curve.
   Displacement: at any instant, displacement of the follower is given by,

$$S = h/2 - (h/2) \cos \beta \tag{1}$$

For the rise (or fall) h of the follower displacement, the cam is rotated through an angle φ whereas a point on the harmonic semicircle traverses an angle ∏. Thus, the cam rotation is proportional to the angle turned by the point on the harmonic semicircle, i.e.

$$\beta = \pi \theta/\varphi \tag{2}$$

Thus   can be replaced by θ and φ in equation 1 above,

$$S = h/2 * (1 - \cos\pi \theta/\varphi) \tag{3}$$

The expression is also valid for β more than 90°. In that case,   or   becomes negative so that s is again positive and more than h/2

Let ω = angular velocity of the cam

$$\theta = \omega t \tag{4}$$

$$\text{and } s = h/2 * (1 - \cos\pi \theta/\varphi) \tag{5}$$

$$v = ds/dt = h/2 * (\pi\omega/\varphi) * \sin(\pi\omega t/\varphi) \tag{6}$$

$$v = h\pi\omega/2\varphi \text{ at } \theta = \phi/2 \tag{7}$$

$$f = dv/dt = h/2 * ((\pi\omega/\varphi))^2 * \cos(\pi\omega t/\varphi) \tag{8}$$

$$f = h/2(\pi\omega/\varphi)^2 \text{ at } \theta = 0 \tag{9}$$

It can be seen from the plots of the figure that there is an abrupt change is acceleration from zero to maximum at the beginning of the follower motion and also from maximum (negative) to zero at the end of the follower motion when the follower rises. Similar abruption would also be there at the start and end of the return motion. As these abrupt changes result in infinite jerk, vibration and noise, the programme should be adopted only for low or moderate cam speeds.

Designing of the cam profile

Initially the cam profile along with its specification was drawn in AutoCAD so that proper shape could be obtained. With the help of the displacement diagram the profile is drawn as shown below.

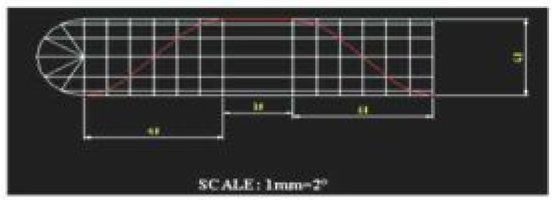


Figure 3: Displacement diagram of SHM cam

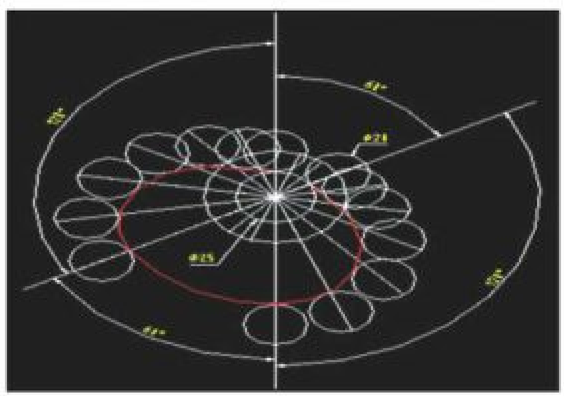


Figure 4: cam profile of SHM cam

Only the cam profile with simple harmonic motion is done still now. In the above sketch drawn in AutoCAD shows the cam profile of a SHM motion with a roller follower. The cam has D- R- D- F as 60°- 120°- 60° - 120°. The base circle of the cam is 25mm and the radius of the roller is 20mm.

Cam –The base circle diameter of the cam is 25mm. The lift is 10mm. Angle of rise is 120°, angle of dwell is 60°, angle of fall is 120° and again angle of dwell is 60°, i.e. R- D- F- D is 120°- 60°- 120°- 60°.

Roller – The roller thickness is 5mm. the radius of the roller is 20mm.

Designing of the structure

After the completion of the cam profile the structure of the model was designed in AutoCAD Mechanical to get an idea of the model. The figure drawn is shown below:

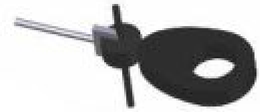
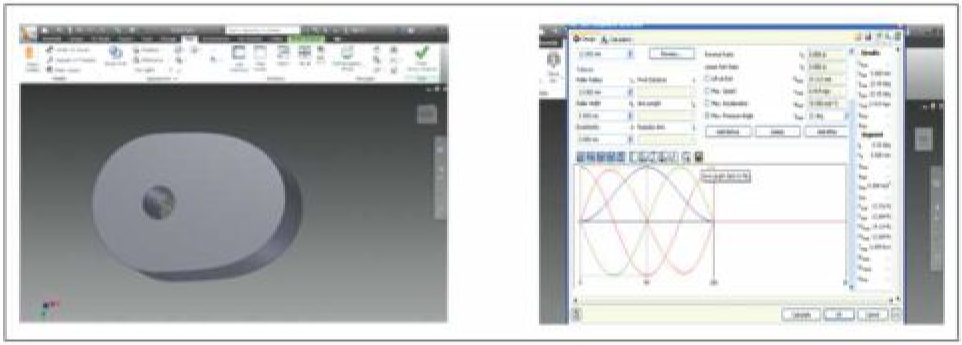


Figure 5: different pressure angles different designs and displacement diagram of cams

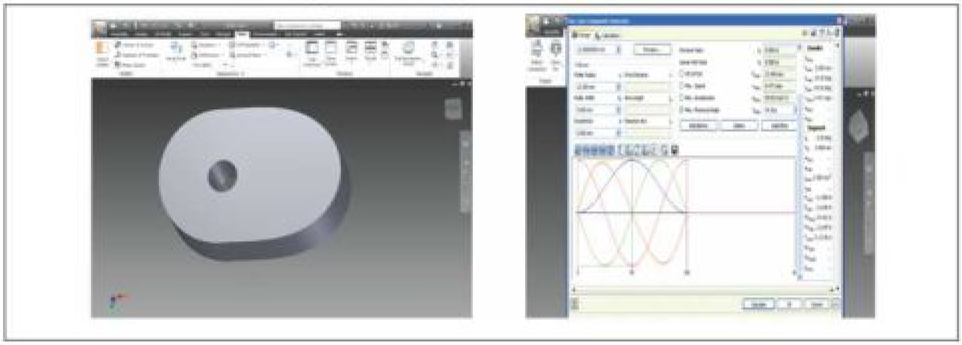
For cam A when the pressure angle is 22°

Design and the displacement diagram of the cam



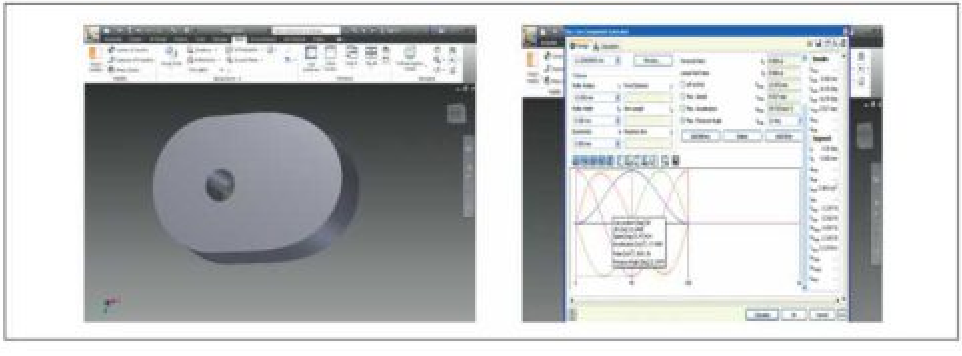
For cam B when the pressure angle is 24°

Design and the displacement diagram of the cam



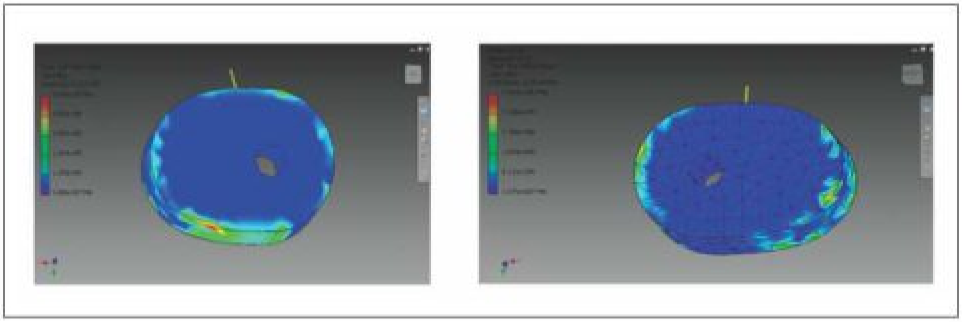
For cam C when the pressure angle is 26°

Design and the displacement diagram of the cam

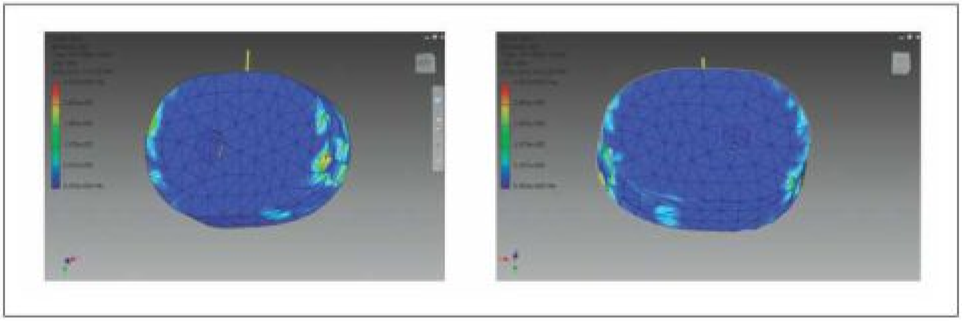


For each of them stress analysis will be as follows

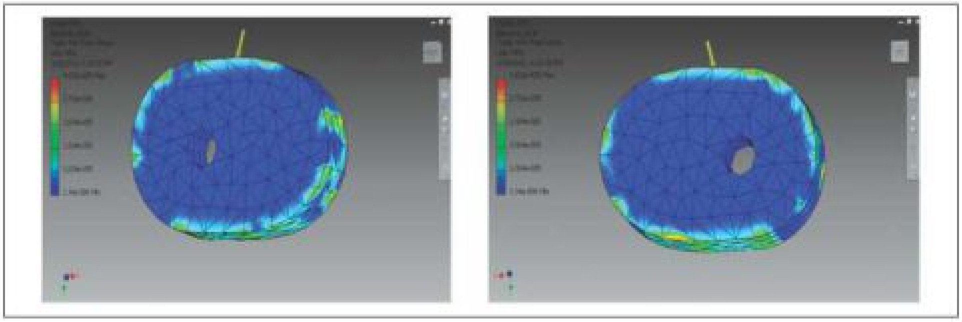
For cam A when the pressure angle is 22°



For cam B when the pressure angle is 24°



For cam B when the pressure angle is 26°



Fatigue life calculations from stress analysis

The various modes of contact-fatigue failure between a cam and a follower can be classified according to their appearance and the factor which promote their initiation and propagation.

The main failure modes of the cam-follower configuration are scuffing and pitting. The probability of one of these occurring depends on several parameters such as material properties, lubricants, loads, engine speed, and temperature.

Here due to change in pressure angles the contact stress also varies so by using Morrow's model which uses the Tresca effective stress which represents the multi axial state of stress at a point. Because the stresses at the interface of the cam and follower are all compressive, the effective stress is assumed to have a negative value. Knowing the principal stresses, the Tresca effective stress can be computed:

$\varepsilon_a = \{(\sigma'_f - \sigma_m)/E\} + (2.N_f)^b$

N_f = fatigue life of the component
σ'_f = fatigue structure coefficient
= 3.1DPH + 500 = 2205
DPH stands for Diamond Pyramid Hardness, for that particular steel
(SAE 52100) the value of DPH is=550
σ_m=mean stress= $1/2(\sigma_{maximum} + \sigma_{minimum})$
b = Basquin's exponent or fatigue strength exponent
= -1/6 log [{(2xDPH) +1380}/DPH}
= -0.109

So the value of
$N_f = \frac{1}{2}\ [\{\sigma_a/(\sigma'_f - \sigma_m)\}^{1/b}]$

From the stress analysis diagram we have the $\sigma_{maximum}$ and $\sigma_{minimum}$ values and from that values we can calculate the value of stress amplitude and mean stress ,
In the following table the different values are shown,

Table 1: Sensitivity

Pressure Angle	$\sigma_{maximum}$	$\sigma_{minimum}$	Mean Stress	Stress Amplitude
22°	3.981E-005	1.875E-007	1.99875E-005	1.98113E-005
24°	4.937E-005	8.003E-008	2.4725E-005	2.4644E-005
26°	4.633E-005	1.34E-006	2.385E-005	2.2495E-005

From the above table we can calculate the value of fatigue life Nf from that value we calculated the no of cycles for each of the cams

For cam A, pressure angle 220 no. of life cycles = 5847.9

For cam B, pressure angle 240 no. of life cycles = 4886.5

For cam C, pressure angle 260 no. of life cycles = 5328.4

CONCLUSION

From the stress analysis of three different cams having different pressure angles of 220 , 240 and 260 , cam A has the maximum fatigue life and the maximum life cycle and cam B has the minimum life cycles .

Future scope of study

The project research can be taken to the next level by designing the 3 three different cams in Catia and finding the stress analysis in Ansys and implementation of Finite Element Analysis (FEA) and henceforth comparing the life cycles. Application of software like Delcam would convert this theoretical approach to the final product, which in turn, would be of great help in automobile industries and the ultimate aim will be fulfilled.

REFERENCES

An Optimum Design of Cam Mechanisms with Roller Follower for Combined Effect of Impact and High Contact Loads, Fathi Al-Shamma, Department of Mechanical Engineering/ University Of Baghdad.

Analysis on Cam-roller Contact Stress and Optimization on Roller Figuration of Fuel-injection System Tan Di Naval University of Engineering.

Fatigue Life Calculation, SEÇIL ARIDURU, Middle East Technical University.

Stress and Fatigue Analysis of SVI-Tested camshaft lobes, Jose Alejandro Escobar, Virginia Polytechnic University.

The Wear Analysis for Cam Mechanism of Diesel Fuel Injection Pump Based on Multibody

Predicting Fatigue Properties through Hardness Measurements, James McMohan, UIUC