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Research Article

Modelling, Design and Finite Element Analysis of Cam Shaft

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Abstract

The goal of the Research is to modelling design and analysis of a camshaft. In FEM, behaviour of cam shaft is obtained by analysing the collective behaviour of the elements to make the cam shaft robust at all possible load cases. This analysis is an important step for fixing an optimum size of a camshaft and knowing the dynamic behaviours of the camshaft. Initially the model is created by the basic needs of an engine with the available background data such as power to be transmitted, forces acting over the camshaft by means of valve train while running at maximum speed. Here the approach becomes fully CAE based. CAE based approach enriches the Research and limits the time duration. Camshafts are rotating components with critical loads. Hence the determination of exact load values becomes the challenging one compared with other rotating members. This Research provides the guidelines to solve such situation. The objective is to determine the stress distribution on the cam shaft for both static and dynamic case and finding out the factor of safety.

Keywords: *Cam shaft, Finite Element Method, Force analysis*

1. Introduction

Cam is a mechanical member for transmitting a desired motion to a follower by direct contact. The driver is called cam and driven is called follower. Cam mechanism is a case of a higher pair with line contact. Camshaft is the Brain of the engine must include cam lobes, bearing journals, and a thrust face to prevent fore and after motion of the camshaft. In addition camshaft can include a gear to drive the distributor and an eccentric to drive a fuel pump. Camshaft is controlling the valve train operation. Camshaft is along with the crankshaft it determines firing order. Camshaft is along with the suction and exhaust systems it determines the useful rpm range of the engine.

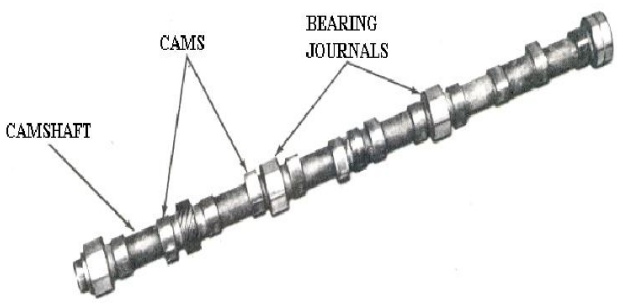
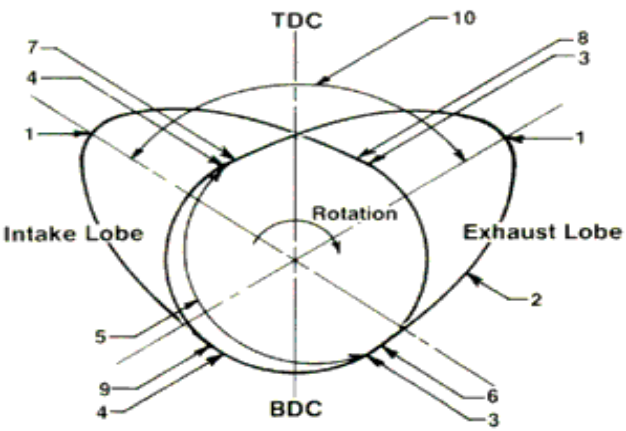


Figure 1. Cam and Camshaft



1. Max lift or nose 2. Flank Opening clearance ramp 3. Closing clearance ramp 4. Base circle 5. Exhaust opening timing figure 6. Exhaust closing timing figure 7. Intake opening timing figure 8. Intake closing timing figure 9. Intake to exhaust lobe separation

Figure 2. CAM Specifications

2. Cam Measurements

2.1. Lift

The valve should be lifted fully to its maximum as soon as the valve starts opening and should be closed in the same fashion. From the figure the distance from base circle to nose edge of 8.95 mm is the lift and Lobe lift is the

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difference in measurement between the nose of the lobe and the base circle of the lobe. Increasing the lift opens the valve further. This reduces the restriction to airflow at the valve and allows air to flow more freely into the cylinder.

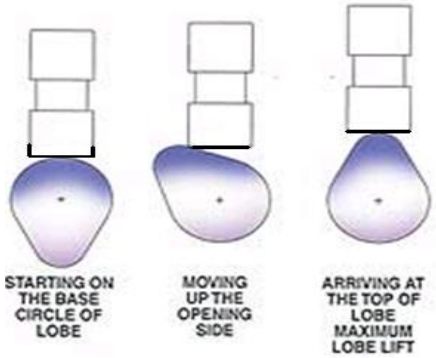

Figure 3. Lift

2.2. Duration

Duration is the length of time (measured in degrees of crankshaft rotation) that the valve remains open. At higher engine speeds the valve opens and shuts in a shorter amount of time. This limits how completely the cylinder can be filled.

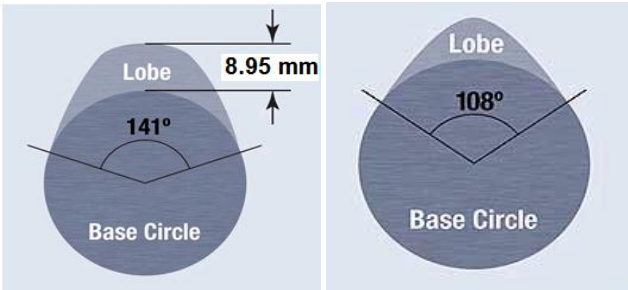

Figure 4. Lobe

2.3. Lobe Separation Angle

Lobe separation angle (LSA) is the number of degrees separating the point of peak exhaust lift and peak intake lift. Lobe separation angle directly impacts the amount of valve overlap. Because of this, production vehicles usually employ a wide LSA to reduce valve overlap and increase idle quality. Because of this, production vehicles usually employ a wide LSA to reduce valve overlap and increase idle quality.

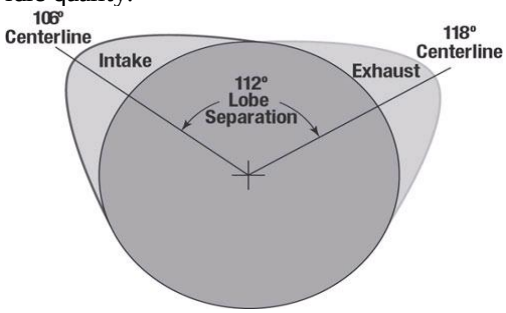

Figure 5a. Lobe separation angle

2.4. Valve Overlap

Valve overlap is the time in which both the intake and exhaust valves are open. Valve overlap is affected by Lobe separation angle and duration. Valve overlap is used because of the principle of exhaust scavenging (the exiting exhaust gases help pull in the fresh intake charge, especially at higher rpm when fill time is limited). At low RPM when intake port speed is low, a long valve overlap period will cause reversion into the intake port (the cylinder pressure exceeds the force of the air in the intake port and exhaust gasses are forced into the intake port). This causes the lumpy idle associated with big camshafts.

2.5. Intake Valve Closing

It is most critical valve opening/closing point. It is too early of an intake valve closing and the cylinder may not have time to fill completely. Intake Valve Closing is too late of an intake valve closing and the cylinder pressure will overcome the inertia of incoming airflow and revert flow back into the intake port. This causes a serious disruption to flow and destroys any pressure waving tuning.

2.6. Exhaust Valve Opening

Exhaust Valve Opening is second most critical valve opening/closing event. It determines the balance between power event efficiency and exhaust pumping losses. It is too early of an exhaust valve opening will reduce the amount of energy converted from cylinder pressure to mechanical force on the piston. It is too late of an Exhaust valve opening will cause an increase in the amount of power needed to expel the burned exhaust gases from the cylinder

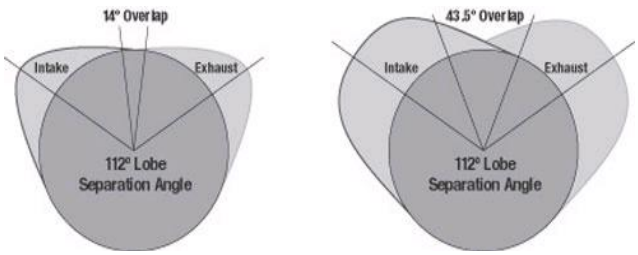

Figure 5b. Valve overlap

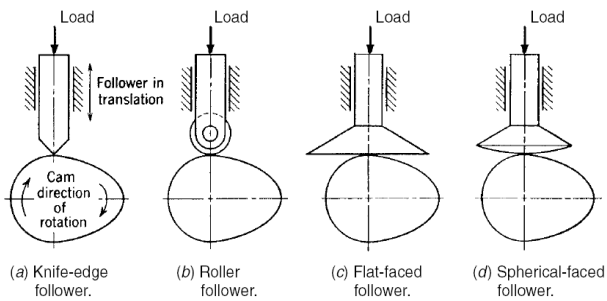

Figure 5c. Exhaust Valve Opening

2.7. Types of Followers

Followers may be classified into three ways. The Construction of the surface is in contact for example knife – edge, roller, flat-faced, mushroom or special shape mushroom. The type of motion is translator and oscillatory.

3. Force Analysis

The forces acting on the follower can now be analysed. As a first approximation, the forces acting against the direction of motion of the follower can be broken down into three components, i) Spring forces ii) Inertial forces and iii) Frictional forces
The spring force, F_s, on the follower follows Hooke's law of the form,

$F_s = k(y-y0)$

Where,

k = spring constant (N/mm)
y = spring compression (mm) and
y_0 = spring compression at start of lift (mm)

The inertial force can be calculated as follows,

$F_{inertial} = ma$

Where, m = mass of the valve train (kg)
a = acceleration of camshaft
d^2r/dt^2 = angular speed (1200 rpm) and
d^2r/d^2 = second derivative of radius (mm/rad^2)

Finally the normal force between the cam and the follower, F_n, can be calculated from the following equation,

$Fn = (Finertial + Fspring)/\cos\alpha$

In this analysis, the frictional force is assumed to be negligible compared to inertial and spring forces. In order to understand the effect of angular velocity on F_n, force calculation were made at 1200 rpm. The results are given in the figure shown below. The 0° position is taken at the nose of the cam.

Note that the greatest normal force occurs at the nose, which decays rapidly throughout the ramps, and maintains a constant value of 148.5 N at the base circle.

4. Contact Stress Analysis

The contact stresses studied herein pertain to compressive stresses developed at the contact surface between the cam and follower due to tangential and normal loads of the valve train. Graph below shows the typical force distribution along the surface of the cam lobe by the valve train spring.

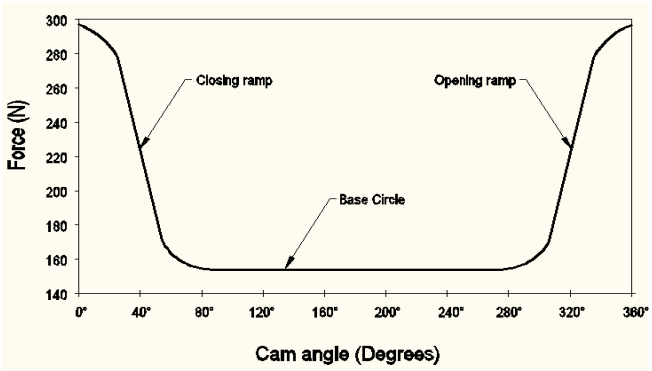

Figure 6. Contact Stress Analysis

4.1.1. Linear Analysis
Assumptions normally made when conducting a linear analysis. The material properties of the component remain linear after the yield limit. The deflections of components are small compared to the overall component size. The components are rigid and ductile; for example, metal components (not rubber). The components deform equally in all three directions; for example, material properties are isotropic.

4.1.2. Mechanical Properties of S55C

S55C is a 0.55 carbon steel supplied at a brinnell hardness of 220. They valued ease of machinability and the corresponding time savings as well as ease of reparability over hardness. Adhering to strict preventive maintenance allowed the molders to successfully complete their production runs.

Table 1. Mechanical Properties of S55C

Material	Alloy	Processing	Shape	Post treatment
Fe	S55C	Unknown	No Form	Annealing

Table 2. Percentage Weight for Elements

Element	Weight %
C	0.55
Si	0.25
Mn	0.75
Fe	98.45

Table 3. Constant values

Quantity	Value	Unit
Young's modulus	210000 - 210000	MPa
Tensile strength	600 - 800	MPa
Elongation	16 - 16	%
Yield strength	340 - 400	MPa

Table 4. Mechanical Properties

Section	Property name	Value
Tensile property	Ultimate tensile strength (MPa)	720
Tensile property	Tensile yield strength (MPa)	431
Elastic property	Tension elastic modulus (GPa)	206
Hardness	Vickers hardness (HV)	230

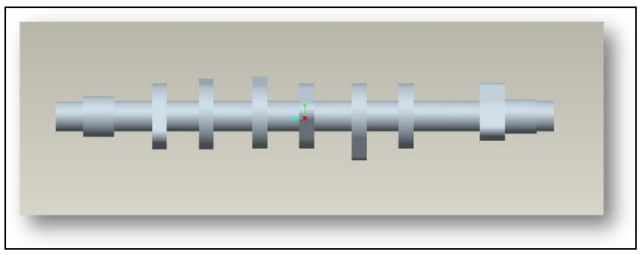

Figure 7. Component Model

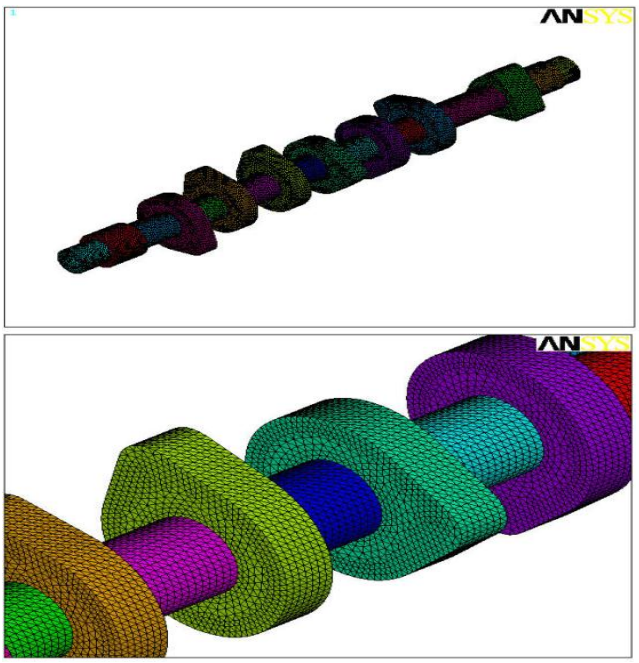

Figure 8. Mesh model

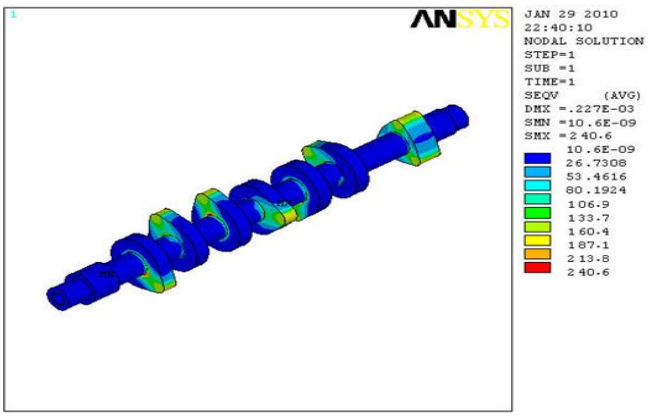

Figure 9. FEA Model

Conclusions

We found that the factor of safety is the ratio of yield tensile strength to the design strength for ductile materials and ultimate tensile strength to the design strength. From the FEA output the maximum design strength is 240.6 N/mm^2 from the material property the ultimate tensile strength of the material is 720 N/mm^2, then the factor of safety becomes within that safety limit.

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