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Design, Modeling, FEM & Experimental Analysis of Crankshaft and Camshaft of a Passenger Car

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Abstract: **-- A crankshaft can be called as the heart of any I.C. engine since it is the first recipient of the power generated by the engine. Its main function is to convert the oscillating motion of the connecting rod into rotary motion of the flywheel. The main function of a camshaft is to convert rotary motion of the crankshaft into vertically reciprocating motion of the valves required to open and close the intake and exhaust valves of engine cylinders, with the assistance of cams located on it and an intermediate mechanism. The crankshaft is subjected to bending stress and torsional shear stress, whereas the camshaft is mainly subjected to compressive stress due to contact pressure, galling and wear and tear. This project aims at designing of I.C. engine multi-crankshaft and camshaft using standard design procedures. Further, Creo software is used to create 3-D models of crankshaft and camshaft. After creating the models, static structural analysis is performed for both of these using different materials and boundary conditions using ANSYS software. A static load testing is performed on the crankshaft and camshaft of a TATA Vista Quadrajet car using UTM and experimental stresses are compared with analytical stresses for validation purpose. Finally, the results of total deformation and equivalent (von-Mises) stresses obtained for different crankshaft materials like ASTM 100-70-03, GS-70, AISI 1045 and Inconel X-750, different camshaft materials like ASTM A532 and ASTM A536 are evaluated and compared with each other to select the best suitable material for manufacturing of crankshaft and camshaft.**

***Index Terms*— ANSYS workbench, crankshaft, camshaft, Creo, design, experimentation, finite element method, materials, modeling, static loading, stress analysis.**

1. INTRODUCTION

A crankshaft as shown in Fig. 1 can be considered as the heart of an I.C. engine, without which it cannot work. It allows the pistons to continuously reciprocate inside the cylinder by means of the unbalanced masses called crank webs. It has a intricate solid geometry. The crankshaft consists of three main parts namely the crank pin, crank web and shaft. The big end of the connecting rod is connected to the crank pin; the crank web connects the crank pin to the shaft portion which is rotated by the main bearings and transmits power to the outside source through the belt drive, gear drive or chain drive. Load of the gas forces inside the combustion chamber is transferred and distributed over each crank pin through the connecting rod. Every crank web is subjected to bending moment and twisting moment. There is a flywheel attached at the end of the crankshaft to bring uniformity in torque of a four-stroke I.C. engine by storing energy during the power stroke and releasing the same during the other three strokes. The crankshaft should have sufficiently high strength so as to sustain the gas force acting vertically downwards during the expansion stroke. Strength of the crankshaft should be high enough to avoid bending failure. Hence, the crankshaft greatly affects the life and reliability of an I.C. engine. The mechanical arrangement of a crankshaft in an I.C. engine is as shown in Fig. 2.

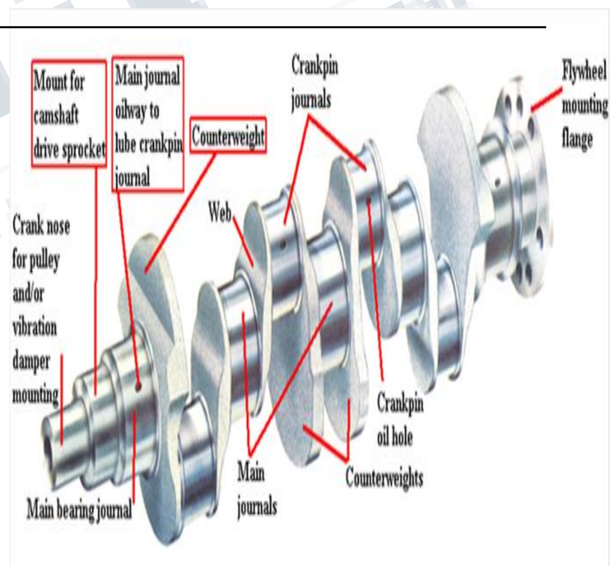

Fig. 1 Nomenclature of Crankshaft

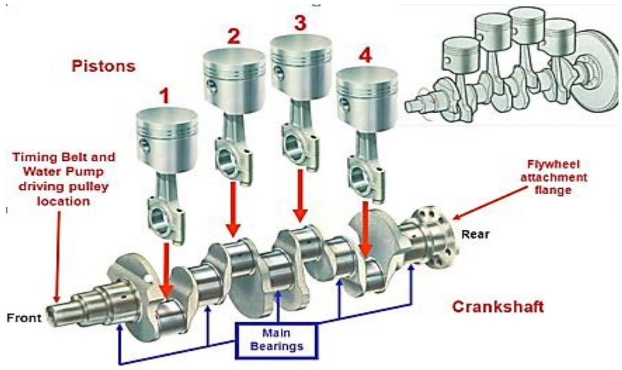

Fig. 2 Mechanical Arrangement of Crankshaft of a 4-Cylinder I.C. Engine

Cam is a mechanical member having a lobe shape used to transfer a desired motion to a follower by means of direct contact. In the pair of a cam and follower, cam is the driving element and follower is the driven element. A camshaft, as shown in Fig. 3 is a shaft which essentially consists of number of cam lobes protruding from it. One cam lobe is provided for one valve. Other supplementary components of a camshaft are bearing journals, push rods, rocker arms, valve springs and tappets. In addition, a camshaft can include a gear to drive the distributor and an eccentric to drive a fuel pump. Valve train operation in an internal combustion (I.C) engine is done by the camshaft. The camshaft, alongwith its supplementary components together called as camshaft mechanism is as shown in Fig. 4. There is a follower lift or valve opening side and an analogous follower fall or valve closing side on the cam profile. These life and fall phases can be divided into three phases namely the cam ramp, cam flank and cam nose as shown in Fig. 5.

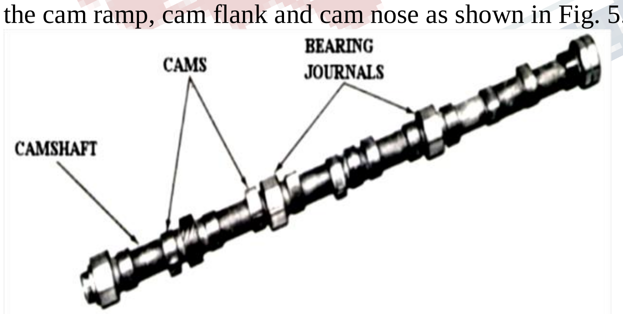

Fig. 3 Camshaft of an I.C. Engine

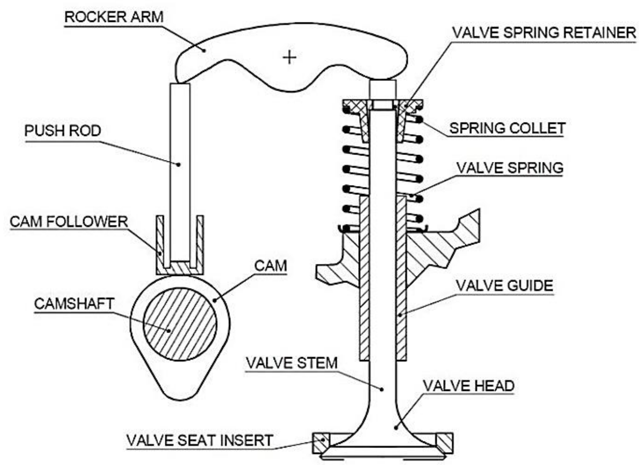

Fig. 4 Overhead Camshaft Mechanism

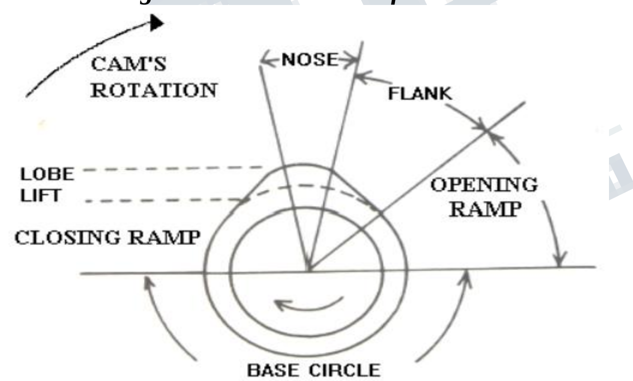

Fig. 5 Camshaft terminology

Crankshaft of an I.C. engine drives the camshaft using chain, belt or gears. The camshaft rotates at half of the speed of crankshaft. Its main function is to open the valve of an I.C. engine cylinder during the outstroke or life, keep it open during the dwell period and close it during the return stroke. When the intake valve opens, the air fuel mixture is sucked inside the engine cylinder, whereas the exhaust gases go out from the engine cylinder when the exhaust valve opens. Both of these valves are controlled by the crankshaft. If a four cylinder engine is considered, there will be one cam lobe for controlling the intake valve and one for the exhaust valve, per cylinder.

Hence, for an engine having two valves per cylinder the total number of cam lobes required would be eight. In such cases, Single Over-head Camshaft (SOHC) will be used. If an engine has four valves per cylinder, the number of cam lobes required would be sixteen. In such cases, Dual Over-head Camshaft (DOHC) having eight lobes per shaft would be used. The followers used alongwith the cams, according to their shapes are namely knife edged, roller, flat-faced and spherical-faced follower. These are as shown in Fig-6.

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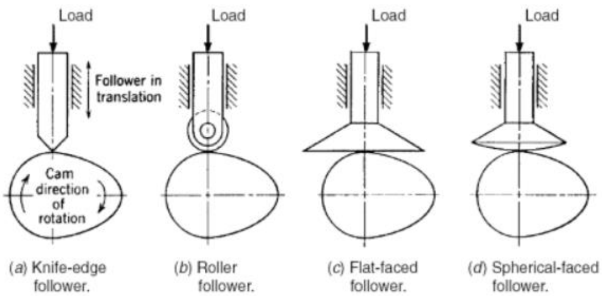


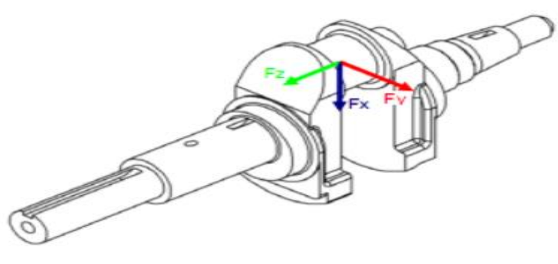
Fig. 6 Types of followers

Most of the I.C. engines in the market use roller cam and follower mechanisms which have a line contact between the cam and roller follower. [1]

II. STRESSES IN CRANKSHAFT AND CAMSHAFT

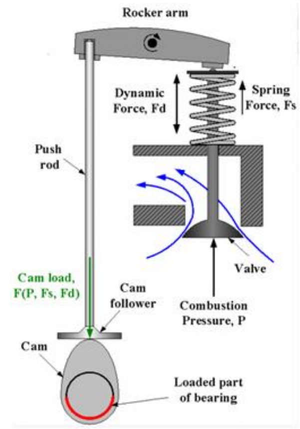
Following stresses directly affect the crankshaft of an I.C. engine:-

1. Bending Stress: - The gas force generated by the burning of air-fuel mixture in the combustion chamber, above the piston head, forces the piston downwards. This force is transmitted to the crankpin bearing. It is a bending load, causing corresponding bending stress.
2. Torsional Shear Stress: - Crankshaft is a rotating component, and it runs at high speeds. Any rotating mass generates centrifugal force. Greater the engine speed, greater is the centrifugal force. Connecting rod also generates centrifugal force. Due to this, both, bending and torsional shear stress are developed in the crankshaft. These are as shown in Fig. 7.



Crankshaft geometry and bending (F_x), torsional (F_y), and longitudinal (F_z) force directions

A camshaft not only controls the opening and closing of the valves of an I.C. engine, but also takes the enormous force of the gas pressure inside the I.C. engine cylinder. Since it is a rotating element, it is also affected by vibrations and deformations especially when the engine is not maintained properly. Forces acting on the camshaft can be analysed dependent on the configuration of the camshaft mechanism. There are three configurations of camshaft mechanism as follows as shown in Fig. 8, Fig. 9 and Fig. 10. [2]



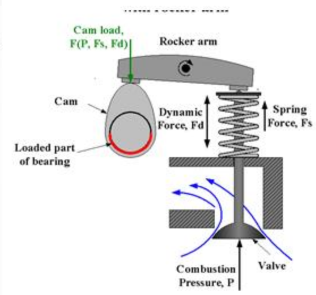


Fig. 8 OHV *Fig. 9 OHC with rocker arms*

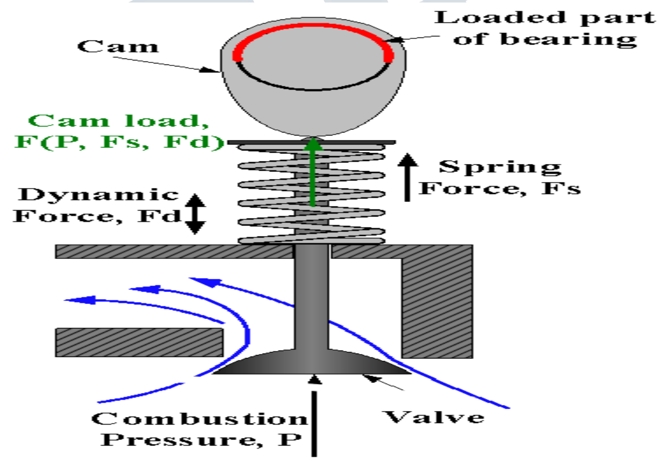


Fig. 10 OHC with direct cam operation

III. MATERIALS AND MANUFACTURING METHODS

The most commonly used materials for the manufacturing of automobile crankshafts are Nodular Cast Iron, Cast Steel and Forged Alloy Steel, depending upon the end application, whether it is used for two wheelers, passenger cars or heavy commercial vehicles. The volume of production also matters. Some special materials like Aluminium-Silicon composite, Inconel X-750 have also been used for exotic applications like racing cars, where performance outdoes cost reduction. While manufacturing camshafts, materials like chilled cast iron, billet steel and aluminium alloys are used.

Crankshafts are manufactured using processes like casting, forging or machining, while camshafts are produced using processes like casting, forging or assembling.

IV. LITERATURE REVIEW

Farzin H. Montazersadgh and Ali Fatemi (2007) [3] carried out a study on crankshaft applying dynamic loading. The analysis was conducted four-stroke engine crankshaft of a single cylinder engine. The values of stress and deformation at critical locations in the crankshaft were obtained using ANSYS software. The analytical results from ANSYS software were compared with the FEA results of simulation performed in ADAMS software. This analysis was conducted considering different engine speeds. The values of stress obtained for engine cycle and effect of fatigue were analyzed. An experimental analysis was also performed by attaching strain gauges at various critical locations in an actual crankshaft. These results were employed to calculate fatigue life of the crankshaft and to further optimize it. Materials considered for analysis were cast iron and forged steel. Results obtained from ANSYS software, ADAMS software and experimentation were compared with each other. In this study, they concluded that dynamic analysis gave realistic results as compared to static structural analysis. There was no effect of twisting load on the values of von-Mises stresses, even at the highly stressed areas. Torsion had a very small effect on the stresses. This made the authors to reach an important conclusion that the crankshaft analysis could be simplified by considering only bending load, without considering torsional load. Areas of the crankshaft geometry which were susceptible to failure were the ones having uneven or quick change in the gradient, such as fillets. These were found to have a high concentration of stresses.

Amit Solanki and Jaydeepsinh Dodiya (2014) [4] performed a static simulation on the crankshaft of a 4-stroke diesel engine having a single cylinder. In their work, they designed a crankshaft using standard design formulae. After this, a 3-D model of the crankshaft was prepared using Pro/Engineer software. This model was imported in ANSYS software as and meshed using tetrahedron 10 elements. Boundary conditions were applied, fixing the bearing supports at both ends. Load was applied and the analysis was run after applying material as cast iron. The values of von-Mises stress and shear stress were found out and compared with the theoretical values of the same for validation purpose. It was found that centre of the neck surface of crank pin had the highest deformation. The transitional surface between the journal of the crankshaft and cheek of the crank, called as fillet, showed the highest intensity of stresses. In addition to this, main journal edge was also a critical area. The analytical value of von-Mises stress was very less as compared to the material yield stress, which ensured a safe design.

P. Vivekandan and M. Kumar (2013) [5] in their work highlighted dynamic force analysis of the forces acting on the follower of the camshaft mechanism. They found that three forces like spring force, inertia force and frictional force act against the motion of the follower. They assumed the frictional force to be negligible because it was very less as compared to the other two forces. They calculated the normal force between the cam and the follower at 1200 RPM. They concluded that the maximum force acts at the nose, transferring quickly to the ramp with its intensity getting reduced and finally maintaining a constant value at the base circle. According to the authors, contact stresses are the compressive stresses at the point of contact of cam and follower due to tangential loads and normal loads. A model of camshaft was prepared using design software, which was later imported in ANSYS software and the analysis applying carbon steel S55C as the material for the camshaft. The obtained equivalent (von-Mises) stress was compared with the yield strength of the material to obtain the factor of safety.

D Jagan and V Ganesh (2016) [6] conducted the design and finite element analysis of I.C. engine camshaft using a new composite metal matrix Al-SiC, manufactured by powder metallurgy, to determine whether it can be a possible alternative to any of the existing materials. They stressed that metal matrix composites have been recently used for the packing of electronics since they have suitable physical characteristics, good manufacturing flexibility and reasonable costs. Their thermal properties could be controlled by modifying the matrix. The main drawback of these matrix composites was found to be their complex manufacturing process. A 3-D model of the camshaft was created using design software. The same was meshed in ANSYS. The analysis was run after applying the boundary conditions which included static loading and results were obtained for determining total deformation, directional deformation, von-Mises stress, shear stress and elastic strain.

V. PROBLEM DEFINITION

In the present automotive market, the industries which manufacture automotive components always aim at manufacturing the components with the highest quality, excellent reliability and minimum possible cost. It is highlighted in many studies that engine related components are maximum prone to failure, followed by the drivetrain components. Owing to the intricate geometry and sudden changes in area in a crankshaft, it has high chances of accumulation of stresses, leading to failure. In addition, it is acted upon by bending and torsional loads since it is a rotating element. Similar is the case with a camshaft. Due to this, it is very complicated to determine the exact values of

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loads acting on the crankshaft and camshaft. The life of any component is mainly dependent on its design, material and manufacturing method. If the design is faulty and the selected material is incorrect, the crankshaft and camshaft can fail before its lifespan, decreasing its reliability and safety.

VI. OBJECTIVES

1. To design the crankshaft and camshaft of a passenger car using standard mathematical design formulae to obtain the dimensions.
2. To create 3-D models of multi-crankshaft and camshaft using Creo 2.0 software.
3. To perform finite element method based static structural analysis on 3-D models using ANSYS Workbench 17.0 software for different cases of loading and different materials.
4. To perform experimentation on crankshaft and camshaft of TATA Indica Vista Quadrajet car using static load testing on universal testing machine for validation of results.
5. To determine the best material for manufacturing crankshaft and camshaft based on the results.

VII. DESIGN CALCULATIONS

For calculating the dimensions of crankshaft and camshaft, general data of a four cylinder four stroke gasoline engine of a passenger car as shown in Table 1 is considered.

Table 1 Specifications of passenger car engine

Sr. No.	Parameter	Value
1.	Cylinder bore diameter (D)	69.6 mm
2.	Stroke length (L)	82 mm
3.	Engine speed (N)	5500 rpm
4.	Maximum combustion pressure (p)	5MPa
5.	Weight of flywheel	150 N
6.	Total belt pull	2500 N

Design of Crankshaft

Case I: When crank is at TDC position:-

1) Bearing Reactions:

Piston gas load

$$F_p = \frac{\pi}{4} D^2 \times p$$

$$F_p = \frac{\pi}{4} (69.6)^2 \times 5$$

$$F_p = 19023 \text{ N}$$

Due to piston gas load, there will be two vertical reactions $(R_1)_v$ & $(R_2)_v$

b = 2D = 2 × 69.6 = 139.2 = 140 mm

Due to piston gas load, there will be two vertical reactions $(R_1)_v$ & $(R_2)_v$

b = 2D = 2 × 69.6 = 139.2 = 140 mm

$b_1 = b_2 = \frac{b}{2} = 70$ mm

By symmetry

$(R_1)_v = (R_2)_v = \frac{Fp}{2} = \frac{19029}{2} = 9511.5$

$(R_1)_v = (R_2)_v = 9511.5$ N

There will be two vertical reactions due to flywheel weight (w) , by symmetry,

$(R_5')_v = (R_6')_v = \frac{W}{2} = = \frac{150}{2} = 75$ N

Due to the resultant belt tension (T_1+T_2) acting horizontally; there will be two horizontal reactions

$(R_5')_H = (R_6')_H = (\frac{T1+T2}{2}) = \frac{2500}{2} = 1250$ N

2) Design of crankpin

Assumptions:

a. Allowable bending stress $(\sigma_b) = 75$ N/mm²
b. Allowable bearing pressure for crankpin bushing $(P_b) = 10$ N/mm²

$L_c/d_c = 1.3$

$(M_b)_c = (R_1)_v \times b_1$
$= 9511.5 \times 70$
$= 665805$ N.mm

$(M_b)_c = (\frac{\pi dc^3}{32}) \times \sigma_b$

$665805 = \frac{\pi dc^3}{32} \times 75$

dc = 44.88

$d_c = 45$ mm

$l_c = 1.3 d_c = 1.3 \times 45 = 58.5$

$l_c = 59$ mm

3) Design of left hand crank web

$w = 1.14\, d_c = 1.14 \times 45 = 51.3$

w= 52 mm

$t = 0.7\, d_c = 0.7 \times 45 = 31.5$ mm

t = 32 mm

Web is subjected to direct compressive stress and bending stress due to reaction $(R_1)_v$

$\sigma_c = \frac{(R1)v}{Wt} = \frac{9511.5}{52 X 32}$

$\sigma_c = 5.72$ N/ mm²

$\sigma_b = \frac{6(R1)v\,[\,b1 - \frac{lc}{2} - \frac{t}{2}]}{wt^2}$

$\sigma_b = \frac{6(9511.5)\,[\,70 - \frac{59}{2} - \frac{32}{2}]}{52 X 32^2}$

$\sigma_b = 26.26$ N/ mm²

Considering yield strength of the material = 75 N/mm²

31.98 N/ mm² < 75N/mm²

Hence, the design of crank web is safe.

4) Design of right hand crank web

The right hand and left hand web is made identical to the right hand crank web for perfect balancing.

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5) Design of shaft under flywheel

C = 20 mm (considering the width of the flywheel and length of bearings)

$C_1 = C_2 = \frac{C}{2} = \frac{20}{2} = 10\text{mm}$

$(M_b)_v = (R_6)_v \times C_1 = 75 \times 10 = 750$ N.mm

$(M_b)_h = (R_6)_H\, C_2 = 1250 \times 10 = 12500$ N.mm

Resultant bending moment,

$M_b = \sqrt{(Mb)v^2 + (Mb)h^2}$

$M_b = \sqrt{(3712.5)^2 + (61875)^2}$

$M_b = 12.522 \times 10^3$ mm

$M_b = (\frac{\pi ds^3}{32}) \times \sigma_b$

$12.522 \times 10^3 = (\frac{\pi ds^3}{32}) \times 75$

$d_s = 11.93$ mm

Diameter of shaft $d_s \approx 12$ mm

Case II: When the crank is at an angle with the line of dead centre position and subjected to maximum torsional moment.

1) Components of force on crank pin

$\Theta = 30°$ (Crank angle for maximum torsional moment)

P' = 4 MPa (pressure for maximum torque)

Piston gas load,

$F_p = \frac{\pi}{4} D^2 \times p' = \frac{\pi}{4} (69.6)^2 \times 4$

$F_p = 15218$ N

Connecting rod length (l) = 300 mm

Crank radius $r = \frac{l}{2} = \frac{82}{2} = 41$ mm

$\sin\phi = \frac{\sin\Theta}{(l/r)} = \frac{\sin 30}{(300/41)}$

$\phi = \sin^{-1}(0.06833)$

$\phi = 3.92°$

Thrust on the connecting rod (F_g)

$F_g = \frac{Fp}{\cos\phi} = \frac{15218}{\cos(3.92)} = 15253.69$ N

$F_t = F_g \sin(\Theta + \phi) = 15253.69 \sin(30 + 3.92)$

$F_t = 8512.09$ N

$F_r = F_g \cos(\Theta + \phi) = 15253.69 \cos(30+3.92)$

$F_r = 12657.78$ N

2) Bearing reactions

The crankshaft is supported on six bearings 1, 2, 3, 4, 5 and 6

b = 140mm

$b_1 = b_2 = b_3 = b_4 = b_5 = b_6 = \frac{140}{2} = 70$ mm

$(R_1)_v = (R_2)_v = (\frac{Fr}{2}) = \frac{12657.78}{2}$

= 6328.89 N

.... upto $(R_4)_v$

$(R_1)_H = (R_2)_H = (\frac{Ft}{2}) = \frac{8512.09}{2}$

= 4256.05 N

$(R_1)_H = (R_2)_H = (R_3)_H = (R_4)_H =$

$(R_5)_v = (R_6)_v = \frac{W}{2} = \frac{150}{2} = 75$ N

$(R_5)_H = (R_6)_H = \frac{T1+T2}{2} = \frac{2500}{2} = 1250$N

3) Design of crank pin (Assume z= 40 N/ mm^2)

$d_c^3 = \frac{16}{\pi z}\sqrt{(Mb)^2 + (Mt)^2}$

$= \frac{16}{\pi z}\sqrt{[(R1)v \, X \, b1)]^2 + [(R1)h \, X \, r)]^2}$

$= \frac{16}{\pi X40}\sqrt{(6328.89 \, X \, 70)^2 + (4256.05 \, X \, 41)^2}$

$d_c = 39.28$

$d_c = 40$mm

In previous case dc is greater than this value.

So taking

$d_c = 45$mm & $l_c = 59$mm

4) Design of shaft under flywheel

Mb = 12522.

Mt = Ft × r

$ds^3 = \frac{16}{\pi z}\sqrt{(Mb)^2 + (Mt)^2}$

$= \frac{16}{\pi X40}\sqrt{[(R6) \times c2)]^2 + (Ft \times r)^2}$

$= \frac{16}{\pi X40}\sqrt{(75 \times 150)^2 + (8512.09 \times 41)^2}$

ds = 35.42

ds = 36 mm

In previous case, ds=12 mm is less than the ds of this case

So, ds= 36mm

5) Design of shaft at the juncture of right hand crank web

$(M_b)_v = (R_4)_v [\, b1 + \frac{lc}{2} + \frac{t}{2}] - Fr [\frac{lc}{2} + \frac{t}{2}]$

$= 6328.89 [\, 70 + \frac{59}{2} + \frac{32}{2}] - 12657.78 [\frac{59}{2} + \frac{32}{2}]$

= 730986.80 – 575928.99

$(M_b)_v = 155.06 \times 10^3$ N.mm

$(M_b)_H = (R_4)_H [\, b1 + \frac{lc}{2} + \frac{t}{2}] - R_1 [\frac{lc}{2} + \frac{t}{2}]$

$= 4256.05 [70 + \frac{59}{2} + \frac{32}{2}] - 8512.09 [\frac{59}{2} + \frac{32}{2}]$

= 491573.78 – 387300.095

$(M_b)_H = 104.27 \times 10^3$ N.mm

Resultant bending moment

$M_b = \sqrt{(Mb)v^2 + (Mb)H^2}$

$M_b = 186.86 \times 10^3$ N.mm , $M_t = f_t \, X \, r$

$= 348.99 \times 10^3$ N mm

Diameter of shaft (ds_1)

$d_{s1}^3 = \frac{16}{\pi z}\sqrt{(Mb)^2 + (Mt)^2}$

$= \frac{16}{\pi X 40}\sqrt{(186.86 \times 10^3)^2 + (348.99 \times 10^3)^2}$

= 36.93

$d_{s1} = 37$mm

6) Design of right hand crank web

$(M_b)_r = (R_2)_v [\, b1 - \frac{lc}{2} - \frac{t}{2}]$

$= 6328.89\ [70 - \frac{59}{2} - \frac{32}{2}]$
$= 155.06 \times 10^3$ N.mm

$(M_b)_r = (\sigma_b)_r\ [\frac{wt^2}{6}]$

$155.06 \times 10^3 = (\sigma_b)_r\ [\frac{52 \times 32^2}{6}]$

$(\sigma_b)_r = 17.47$ N/mm²

$(M_b)_t = F_t\ [r - \frac{ds1}{2}]$
$= 8512.09\ [41 - \frac{37}{2}]$
$= 191.52 \times 10^3$ N.mm

$(M_b)_t = (\sigma_b)_t\ [\frac{w^2t}{6}]$

$191.52 \times 10^3 = (\sigma_b)_t\ [\frac{52^2 \times 32}{6}]$

$(\sigma_b)_t = 13.28$ N/mm²

$(\sigma_c)_d = [\frac{Fr}{2wt}] = [\frac{12657.78}{2\times52\times32}] = 3.80$ N/mm²

$(\sigma_c) = (\sigma_b)_r + (\sigma_b)_t + (\sigma_c)_d$

$(\sigma_c) = 34.55$ N/mm2

$M_t = (R_2)_h\ [\ b_2 - \frac{lc}{2}\ ] = 4256.05\ [70 - \frac{59}{2}]$

$M_t = 172.37 \times 10^3$ N.mm

$\tau = \frac{Mt}{Zp} = \frac{4.5\ Mt}{wt^2} = \frac{4.5 \times 172.37 \times 10^3}{52 \times (32)^2}$

$\tau = 14.56$ N/mm²

$(\sigma_c)_{max} = 0.5 \times [(\sigma_c) + \sqrt{(\sigma c)^2 + 4z^2}]$
$= 0.5 \times [34.55 + \sqrt{(34.55)^2 + 4z^2}]$
$= 39.87$ N/mm²

It is less than 75N/mm², so design is safe.

7) Design of left hand crank web
No need to find stresses for left hand crank web. W & t will be same.

8) Design of crankshaft bearing
Reaction at bearing

$R_5 = \frac{Fp}{2} + \frac{W}{2} + \frac{T1+T2}{2}$
$= \frac{19.023}{2} + \frac{0.150}{2} + \frac{2.5}{2}$
$= 10.8365$ KN
$= 10836.5$ N

Total bearing pressure

$d_5 = 1.3d_{s1} = 48.1 = 49$mm

$P_b = \frac{R5}{ds1 \times d5} = \frac{10836.5}{37 \times 49} = 5.98$ N/mm²

Bearing pressure for the material = 10 N/mm²

$P_b < 10$ N/mm²

Hence the design is safe. [7]

Design of Camshaft

1) Design of Valve

1) Diameter of valve port

$$a = \frac{\pi D^2}{4} = \frac{\pi(69.6)^2}{4} = 3804.59 \approx 3805\ mm^2$$

$$v = 2L\left(\frac{N}{60}\right) = 2(0.082)\left(\frac{5500}{60}\right) = 15.08\ {}^{m}/_{s}$$

Assume

$$v_p = 50\ {}^{m}/_{s}$$

$$a_p = \left(\frac{\pi d_p^2}{4}\right) mm^2$$

$$a \times v = a_p \times v_p$$

$$3805 \times 1503 = \left(\frac{\pi d_p^2}{4}\right) \times 50$$

$$d_p = 38.16 \approx 39\ mm$$

2) Diameter of Valve Head
For a seat angle 45°

$$w = 0.06\ d_p = 0.06 \times 39 = 2.34\ mm$$

Diameter of Valve Head (d_v)

$$d_v = (d_p + 2w) = 39 + 2(2.34) = 43.68\ \approx 44\ mm$$

3) Thickness of the valve head
For a steel valve,

$$k = 0.42$$

Assume

$$\sigma_b = 70\ {}^{N}/_{mm^2}$$

$$t = kd_p\sqrt{\frac{P_{max}}{\sigma_b}} = 0.42 \times 39\sqrt{\frac{5}{70}} = 4.38 \approx 5\ mm$$

4) Diameter of Valve Stem

$$d_s = \left[\frac{d_p}{8} + 6.35\right] to \left[\frac{d_p}{8} + 11\right]$$
$$= \left[\frac{39}{8} + 6.35\right] to \left[\frac{39}{8} + 11\right]$$
$$= 11.225\ to\ 15.875$$
$$d_s = 14\ mm$$

5) Maximum lift of Valve

$$h_{max} = \frac{d_p}{4\cos\alpha} = \frac{39}{4\cos(45)} = 13.79 \approx 14\ mm$$

2) Forces acting on Rocker arm
The gas load (P_g)

$$P_g = \left(\frac{\pi d_v^2}{4}\right) P_c = \left(\frac{\pi(44^2)}{4}\right)(0.4) = 608.21\ N$$

Initial Spring force (P_i)

$$P_i = \left(\frac{\pi d_v^2}{4}\right) P_s = \left(\frac{\pi(44^2)}{4}\right)(0.02) = 30.41\ N$$

Acceleration of Valve

$$Speed\ of\ Camshaft = \frac{1}{2}(speed\ of\ crankshaft)$$
$$= \frac{1}{2}(5500) = 2750\ rpm$$

Angle turned by camshaft/s

$$1\ rev. = 360°$$

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$$angle\ turned\ by\ camshaft/s = \left(\frac{2750}{60}\right)(360)$$
$$= 16500°\ per\ sec$$

Total crankshaft angle when the valve is open $= 33 + 180 + 1 = 241$

Total angle of cam action$= \frac{1}{2}(angle\ of\ camshaft) = \frac{1}{2}(214) = 107°$

Cam angle during constant acceleration $= 26.75°$

Valve lift during constant acceleration $(S) = 13.5mm = 13.5 \times 10^{-3}m$

Time taken by valve during constant acceleration period $(t) = \frac{angle\ of\ cam\ action}{angle\ turned\ by\ camshaft/s} = \frac{26.75}{16500} = 0.00162\ s$

$$s = ut + \frac{1}{2}\alpha t^2$$
$$s = 13.5 \times 10^{-3}, \quad u = 0, \quad t = 0.001625\ sec$$
$$\alpha = 5334.6\ m/s^2$$
$$P_a = m\alpha = 0.5 \times 5534.6 = 2667N$$
$$P_e = P_g + P_a + P_i = 608.21 + 2667 + 30.41$$
$$= 3305.62 \approx 3306\ N$$

3) Design of fulcrum pin

$$R_f = \sqrt{(P_e)^2 + (P_c)^2 - 2P_eP_c\cos 0} = 6555.43\ N$$
$$P_b = \frac{R_f}{d_1 l_1} \qquad \left(P_b = \frac{5N}{mm^2}\right)\left({}^{l_1}/{}_{d_1} = 1.25\right)$$
$$s = \frac{6555.43}{d_1(1.25d_1)}$$
$$d_1 = 32.38 \approx 34\ mm$$
$$l_1 = 1.25d_1 = 43.55 \approx 44\ mm$$
$$z = \frac{R_f}{2\left(\frac{\pi d_1}{4}\right)^2} = \frac{6555.43}{2\left(\frac{\pi(34)}{4}\right)^2} = 3.61N/mm^2$$
$$z < 42\ N/mm^2$$
$$Outside\ dia\ of\ rocker\ arm\ boss\ D_i = 2d_i = 2(34)$$
$$= 68\ mm$$
$$Inside\ dia = d_i + (3 \times 2) = 40\ mm$$
$$M_b = P_e \times a = 3306 \times 150 = 495.9 \times 10^3 Nmm$$
$$I = \left(\frac{(44) \times (68)^3}{12}\right) - \left(\frac{(44) \times (40)^3}{12}\right)$$
$$= 918.25 \times 10^3\ mm^4$$
$$y = {}^{68}/{}_{2} = 34\ mm$$
$$\sigma_b = \frac{M_b y}{I} = \frac{495.9 \times 10^3 \times 34}{918.25 \times 10^3} = 18.36\ N/mm^2$$

$\sigma_b < 70\ N/mm^2$ 4 : Design of forked end

$$P_c = P_e = 3306$$
$$P_o = \frac{P_c}{d_2 l_2} \qquad \left(P_0 = 5\frac{N}{mm^2}\right), \left({}^{l_2}/{}_{d_2} = 1.25\right)$$
$$5 = \frac{3306}{d_2(1.25d_2)}$$
$$d_2 = 23.47 \approx 24\ mm$$
$$l_2 = 1.25d_2 = 30\ mm$$
$$z = \frac{P_c}{2\left(\frac{\pi d_2}{4}\right)^2} = \frac{3306}{2\left(\frac{\pi(24)}{4}\right)^2} = 3.65\ N/mm^2$$
$$z < 42\ N/mm^2$$
$$Thickness\ of\ eye = l_2/2 = 15\ mm$$
$$Outer\ dia\ of\ eye = D_2 = 2d_2 = 48\ mm$$
$$The\ diameter\ of\ roller = D_2 + 5 = 48 + 5$$
$$= 53\ mm \approx 55\ mm$$
$$M_b = \left(\frac{5}{24}\right)P_c l_2 = \left(\frac{5}{24}\right) \times 3306 \times 30$$
$$= 20.66 \times 10^3\ Nmm$$
$$z = \frac{\pi d_2^3}{32} = \left(\frac{\pi(24)^3}{32}\right) = 1357.17\ mm^3$$
$$\sigma_b = \frac{M_b}{2} = \frac{20.66 \times 10^3}{1357.17} = 15.22\ N/mm^2$$
$$\sigma_b < 70\ N/mm^2$$

5) Design of cross section of rocker arm

$$M_b = P_c(a - d_1) = 3306(150 - 34)$$
$$= 383.496 \times 10^3 Nmm$$
$$I = 37t^4, y = 3t$$
$$\sigma_b = \frac{M_b y}{I}$$
$$70 = \frac{383.496 \times 10^3 \times (3t)}{37t^4}$$
$$t = 7.63 \approx 8\ mm\ thickness$$
$$Width = 2.5t = 2.5(8) = 20\ mm$$
$$Depth = 6t = 6(8) = 48\ mm$$

6)Design of tappet

$$d_c = core\ dia\ of\ stud$$
$$\sigma_c = \frac{P_e}{\left(\frac{\pi d_1^2}{4}\right)}\ or\ 50 = \frac{3306}{\frac{\pi d_c^2}{4}}$$
$$d_c = 9.17\ mm$$
$$nominal\ dia\ of\ stud\ d = \frac{d_c}{0.8} = \frac{9.17}{0.8} = 11.46$$
$$\approx 12\ mm$$

Diameter of circular end of rocker arm (D_3) and Depth (t_3)

$$D_3 = (2d) = 2(12) = 24\ mm$$
$$t_3 = (2d) = 2(12) = 24\ mm$$

7: Design of Valve Spring

Assumptions

Spring index =8

Stiffness of spring is 10 N/mm^2

Permissible torsional shear stress : 300 N/mm^2

Modulus of rigidity = $84 \times 10^3\ N/mm^2$

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Total gas between consecutive coils is 15% of max compression.

$$P_i = 30.41\ N$$

$$P_{max} = P_i + k8 = 30.41 + 10(14) = 170.41\ N$$

Wire diameter

$$k = \frac{4c-1}{4c-4} + \frac{0.615}{c} = \frac{4(8)-1}{4(8)-4} + \frac{0.615}{8} = 1.184$$

$$z = k\left(\frac{8P_{max}c}{\pi d^2}\right)$$

$$300 = 1.184\left(\frac{8(170.41)(8)}{\pi d^2}\right)$$

$$d = 3.7\ \approx 4\ mm$$

Mean coil diameter $D = cd = 8(4) = 32\ mm$

Number of active turns $N = \frac{Gd_2^4}{8D^3K} = \frac{(84\times10^3)(4^4)}{8(32^3)(4^4)} = 8.20\ \approx 9$

Total no. of turns $N_t = N + 2 = 9 + 2 = 11$

Max compression of spring

$$\delta_{max} = \frac{8P_{max}.D^3N}{Gd^4} = \frac{8(170.41)(32^3)\times 9}{(84\times10^3)(4^4)} = 18.65\ mm$$

Solid length of spring

$$Spring\ length = N_t d = (11\times 4) = 44\ mm$$

$$free\ length\ of\ spring = solid\ length + \delta_{maz} + 0.15\delta_{max}$$

$$= 44 + 18.65\ (0.15\times 18.65) = 66\ mm$$

$$Pitch\ of\ coils = \frac{Free\ length}{(N_t - 1)} = \frac{66}{(11-1)} = 6.66\ mm$$

8) Design of Cam

Diameter of camshaft $(D') = 0.16D + 12.5 = 0.16(69.6) + 12.5 = 23.64$

D' ≈ 23 mm

The base circle diameter of cam $= 25 + 5 = 30\ mm$

Roller Diameter $= 55\ mm$

Roller width $= 30\ mm$

Cam width = Roller width $= 30\ mm$

Lift of Valve $= 14\ mm$ [7]

VIII. MODELLING AND ANALYSIS OF CRANKSHAFT

A. Modeling of Crankshaft

A 3-D model of Crankshaft is created in Creo 2.0 software using the dimensions obtained from the design calculations above. It is as shown in Fig. 11.

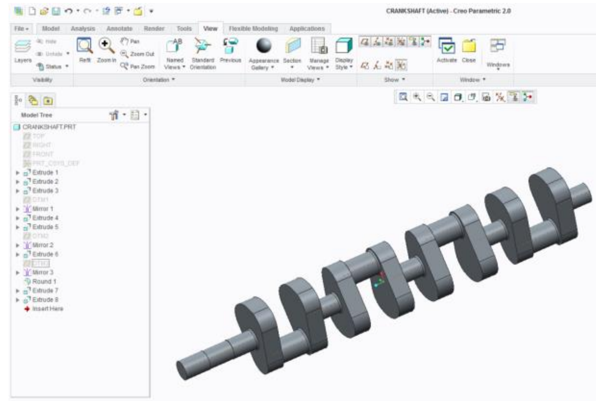

Fig. 11 3-D model of Crankshaft in Creo 2.0

B. Static Structural Analysis of Crankshaft (for Case-II)

The 3-D model of crankshaft created using Creo 2.0 software is imported in ANSYS 17.0 software. It was meshed using tetrahedron elements. The mesh statistics are as shown in Fig. 12. The meshed model is as shown in Fig. 13

Statistics	
Nodes	480524
Elements	315653
Mesh Metric	None

Fig. 12 Mesh statistics for Crankshaft

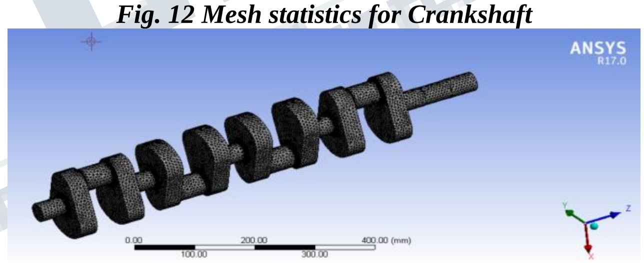

Fig. 13 Meshed model of crankshaft

Boundary conditions are applied to the meshed crankshaft which includes force boundary conditions and restriction boundary conditions. It is as shown in Fig. 14.

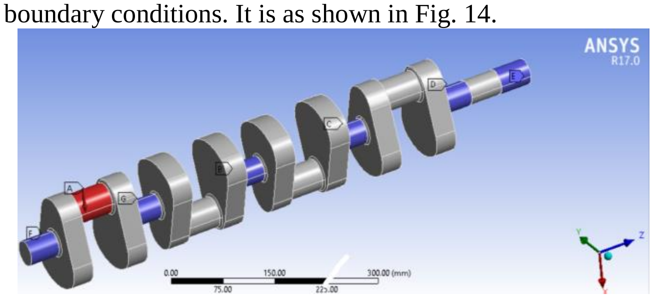

Fig. 14 Boundary conditions for Crankshaft

The static structural analysis was run by applying the materials namely nodular cast iron, cast steel, forged steel and inconel X-750. Results were obtained for total

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deformation and equivalent (von-Mises) stress for the materials as shown in Fig. 15 to Fig. 22 respectively for Case-II.

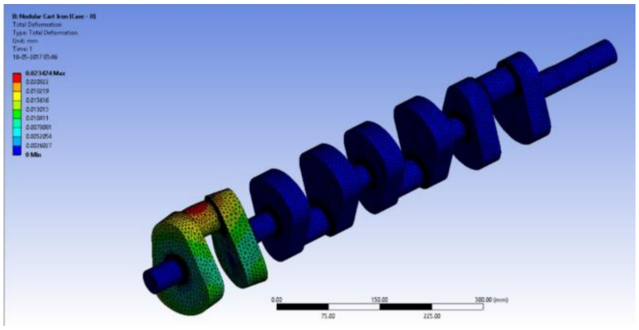

Fig. 15 Total deformation for nodular cast iron

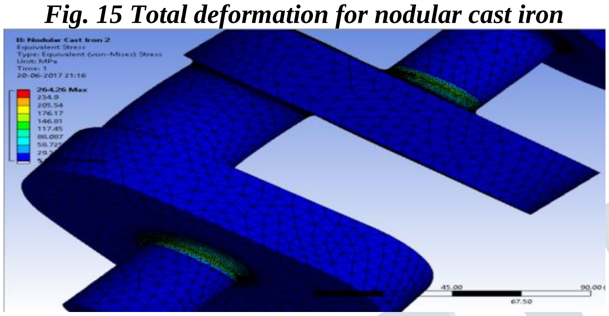

Fig. 16 Von-Mises stress for nodular cast iron

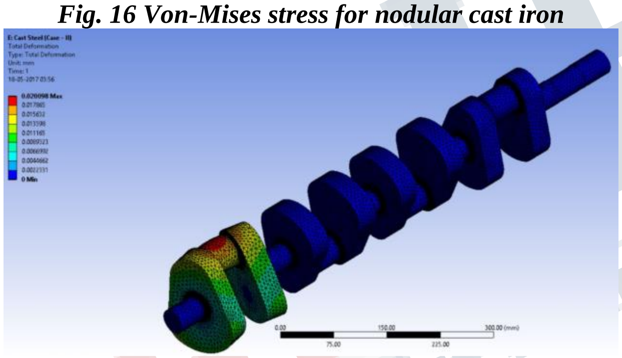

Fig. 17 Total deformation for cast steel

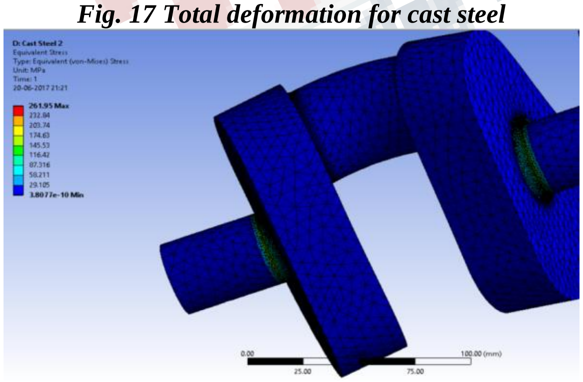

Fig. 18 von-Mises stress for cast steel

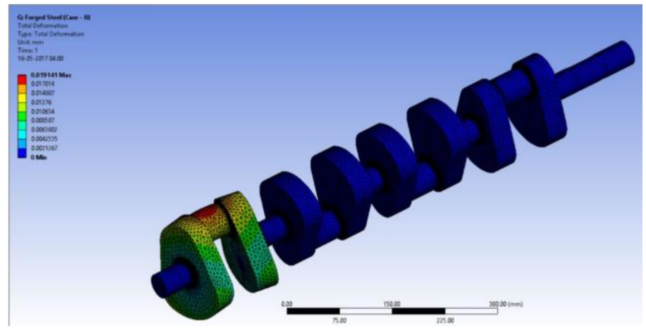

Fig. 19 Total deformation for forged steel

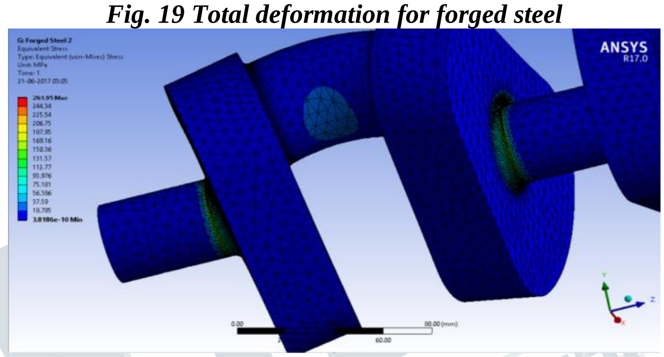

Fig. 20 von-Mises stress for forged steel

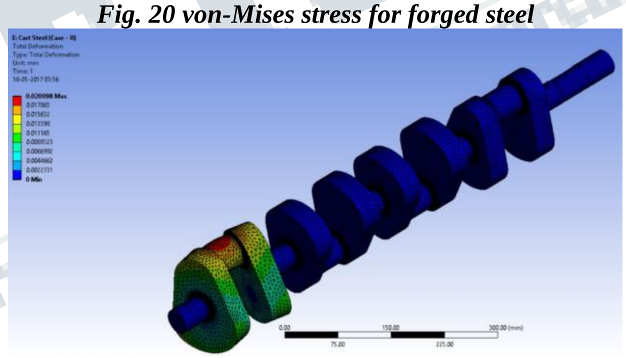

Fig. 21 Total deformation for Inconel-X750

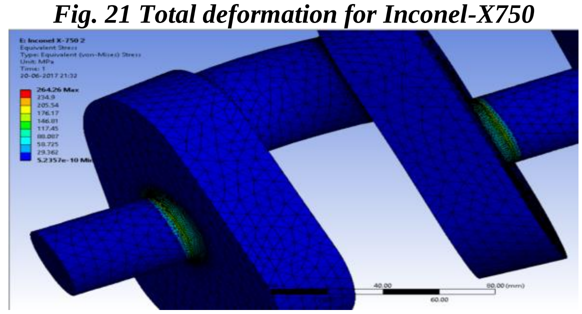

Fig. 22 von-Mises stress for Inconel X-750

IX. MODELLING AND ANALYSIS OF CAMSHAFT

A. Modeling of Camshaft

A 3-D model of camshaft is created in Creo 2.0 software using the dimensions obtained from the design calculations above. It is as shown in Fig. 23.

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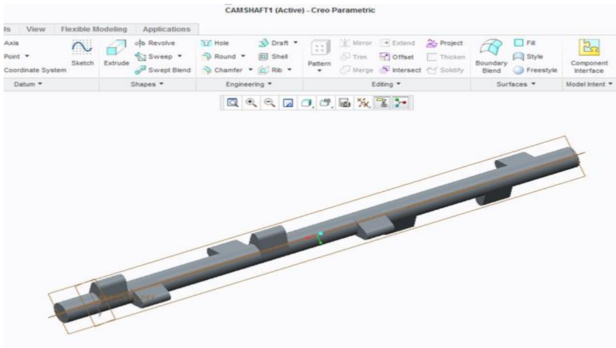

Fig. 23 3-D model of camshaft

B. Static Structural Analysis of Camshaft

The 3-D model of crankshaft created using Creo 2.0 software is imported in ANSYS 17.0 software. It is meshed using tetrahedron elements. The mesh statistics are as shown in Fig. 24. The meshed model is as shown in Fig. 25.

Statistics	
Nodes	60851
Elements	38022
Mesh Metric	None

Fig. 24 Mesh statistics for camshaft

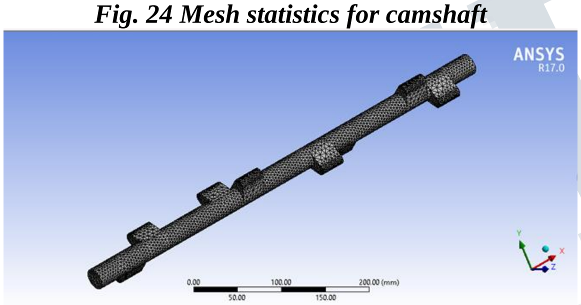

Fig. 25 Meshed model of camshaft

Boundary conditions are applied to the meshed crankshaft which includes force boundary conditions and restriction boundary conditions. It is as shown in Fig. 26.

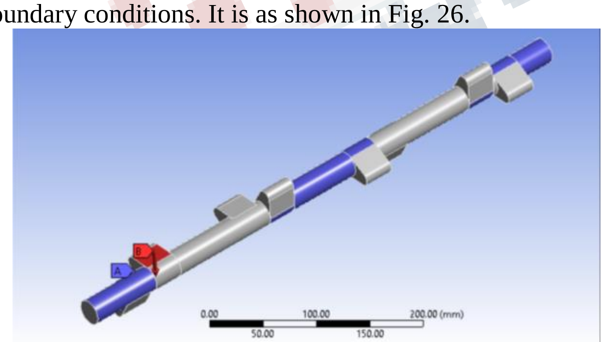

Fig. 26 Boundary conditions for camshaft

The static structural analysis was run by applying the materials namely white cast iron and nodular cast iron. Results were obtained for total deformation and equivalent (von-Mises) stress for the materials as shown Fig. 27 to Fig. 30 respectively

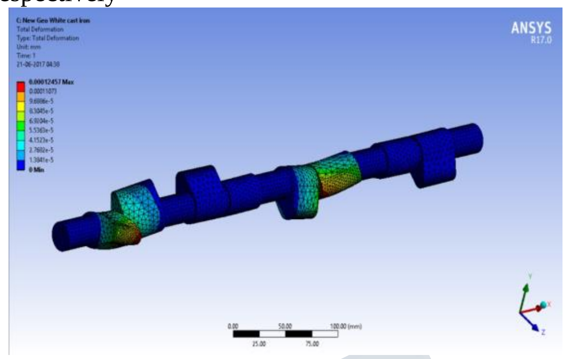

Fig. 27 Total deformation for white cast iron

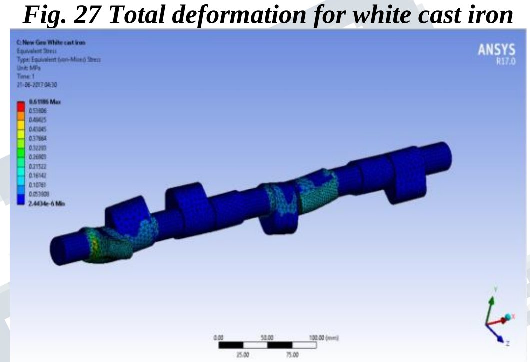

Fig. 28 von-Mises stress for white cast iron

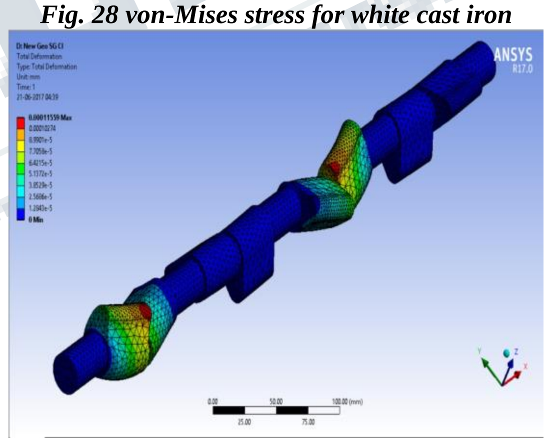

Fig. 29 Total deformation in nodular cast iron

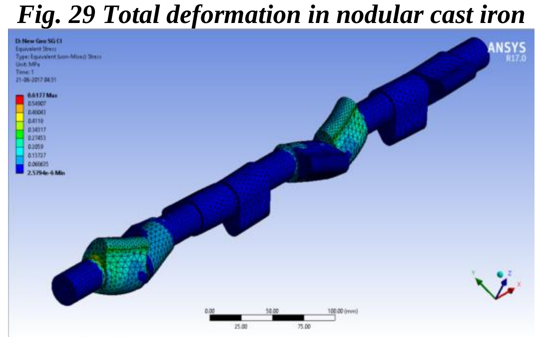

Fig. 30 von-Mises stress in nodular cast iron

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X. EXPERIMENTAL ANALYSIS

After carrying out FEM based stress analysis using ANSYS software, a basic level experimentation is also carried out on the crankshaft and camshaft on a universal testing machine for static loading conditions. The crankshaft and camshaft used for this experimentation was obtained from a TATA Indica Vista Quadrajet (Diesel version). The material specifications of these are as shown in Table 2. According to the chemical composition, the crankshaft material was equivalent to EN8 (forged steel), while that of camshaft was equivalent to nodular cast iron (SG iron).

Table 2 Material properties of TATA Indica Vista Quadrajet car crankshaft and camshaft obtained from testing

PRAJ Metallurgical Laboratory
B-38, Indira Shankar Nagari, Paud Road, Near Kothrud PMT Depo, Kothrud, Pune - 411 038.
Tel.: +91 - 20 - 2528 1584, Cell : 96650 11314, 9158658634
E-mail : prajlab.ashok@gmail.com

- Testing of Metal, Rubber, Plastic, Gasket, Foam, Dental.
- Calibration of Durometer, Temp. Sensors, Vernier & Load cells.
- Failure analysis

TEST REPORT

PRAJ/2017-05/054B — Date: 26/05/2017

Customer Name : Ketan V. Karandikar
Address : JSPM's Rajarshi Shahu College of Engineering, Tathawade, Pune.
Reference : Request Dated 22/05/2017
Subject : Testing of samples

1.0 Chemical Composition
(Optical Emission Spectroscopy)

Sr. No.	Sample Identification	Chemical Composition, (%)							
		C	Mn	Si	S	P	Cr	Ni	Cu
1	Cam Shaft	3.388	0.642	1.662	0.072	0.085	0.350	0.041	0.151
2	Crank Shaft	0.460	0.783	0.200	0.026	0.019	0.191	0.055	0.021

Remark: Crankshaft material is equivalent to En 8.

TEST CONDUCTED BY

SACHIN
TEST ENGINEER

A.M.BHAGAT
PROPRIETOR

PRAJ METALLURGICAL LABORATORY PUNE

The crankshaft was tested using a big universal testing machine having a capacity of 10 tons. It was mounted on the engine block of TATA Indica Vista Quadrajet car on the machine bed. In case-I (crank is at TDC position) of loading, a maximum static load of 19023 N was applied at the centre of one crankpin, whereas in case-II (crank is at 30º from TDC), a load of 15218 N was applied at the centre of one crankpin. This was done one after the other. The arrangement is as shown in Fig. 31 and Fig. 32. The values of deformation and stress obtained from the testing are as shown in Table 3.

Fig. 32 Testing of crankshaft on UTM (mounting)

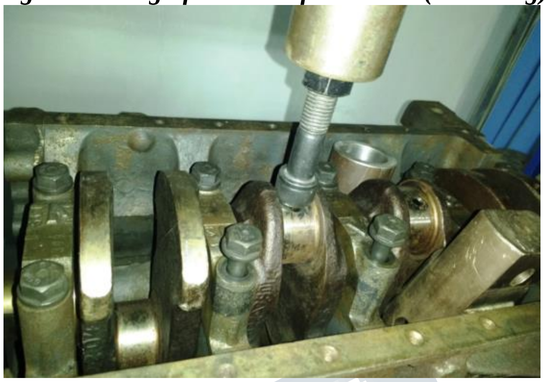

Fig. 33 Testing of crankshaft on UTM (loading)

Table 3. Test results of crankshaft

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Tel.: +91 - 20 - 2528 1584, Cell : 96650 11314, 9158658634
E-mail : prajlab.ashok@gmail.com

- Testing of Metal, Rubber, Plastic, Gasket, Foam, Dental.
- Calibration of Durometer, Temp. Sensors, Vernier & Load cells.
- Failure analysis

TEST REPORT

PRAJ/2017-05/050A — Date: 13/05/2017

Customer Name : Ketan V. Karandikar
Address : JSPM's Rajarshi Shahu College of Engineering, Tathawade,Pune.
Reference : Request Dated 12/05/2017
Subject : Load testing of Crankshaft sample.

Machine specifications : Universal Testing Machine (computerized, software based)
Company: Star Testing Systems, India.
Model No. STS 248, Speed: 3 mm/min.
Accuracy of the machine: ±1%

Crankshaft_TDC			Crankshaft_30Degree		
Displacement (mm)	Load (N)	Stress (MPa)	Displacement (mm)	Load (N)	Stress (MPa)
0.2	1068.20	6.18	0.3	646.80	10.17
0.4	4547.20	26.31	0.6	1969.80	31.00
0.5	6928.60	40.08	0.9	3998.40	62.92
0.6	9515.80	55.06	1.2	6497.40	102.25
0.7	12181.40	70.48	1.4	8839.60	139.11
0.8	15111.60	87.43	1.6	11524.80	181.37
0.9	18251.60	105.60	1.8	14317.80	225.33
1.0	21119.00	122.19	2.0	15866.20	249.70
1.1	22206.80	128.48	2.2	17620.40	277.31

The camshaft was tested on a small universal testing machine having a capacity of 1 ton. A static loading of 171 N was applied at the tip of a cam, after the camshaft was firmly held at its two ends using bench vices. This is as shown in Fig. 34. The test results are as shown in Table 4.

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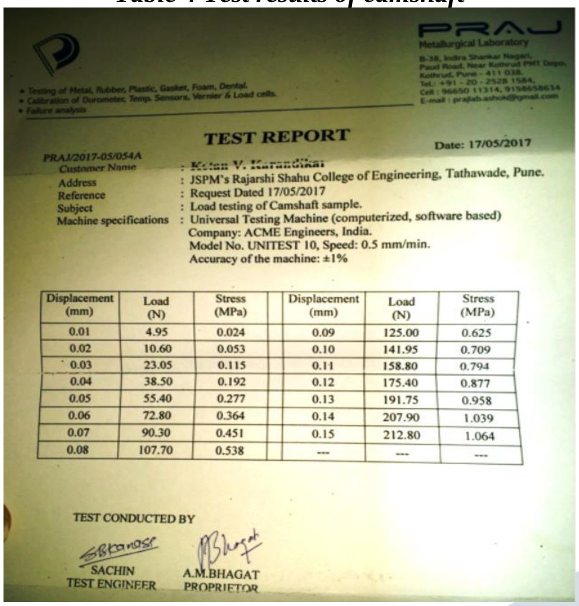
Fig. 34 Testing of camshaft on UTM (loading)

Table 4 Test results of camshaft



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Tel.: +91 - 20 - 2528 1584, Cell : 96650 11314, 9158658634
E-mail : prajlab.ashok@gmail.com

- Testing of Metal, Rubber, Plastic, Gasket, Foam, Dental.
- Calibration of Durometer, Temp. Sensors, Vernier & Load cells.
- Failure analysis

TEST REPORT

PRAJ/2017-05/054A   Date: 17/05/2017

Customer Name : Ketan V. Karandikar
Address : JSPM's Rajarshi Shahu College of Engineering, Tathawade, Pune.
Reference : Request Dated 17/05/2017
Subject : Load testing of Camshaft sample.
Machine specifications : Universal Testing Machine (computerized, software based)
Company: ACME Engineers, India.
Model No. UNITEST 10, Speed: 0.5 mm/min.
Accuracy of the machine: ±1%

Displacement (mm)	Load (N)	Stress (MPa)		Displacement (mm)	Load (N)	Stress (MPa)
0.01	4.95	0.024		0.09	125.00	0.625
0.02	10.60	0.053		0.10	141.95	0.709
0.03	23.05	0.115		0.11	158.80	0.794
0.04	38.50	0.192		0.12	175.40	0.877
0.05	55.40	0.277		0.13	191.75	0.958
0.06	72.80	0.364		0.14	207.90	1.039
0.07	90.30	0.451		0.15	212.80	1.064
0.08	107.70	0.538		---	---	---

TEST CONDUCTED BY

SACHIN
TEST ENGINEER

A.M.BHAGAT
PROPRIETOR

XI. RESULTS

The comparison of analytical (ANSYS software) results for different materials of crankshaft is as shown in Table 5

Table 5. Comparison of analytical results for crankshaft (for Case-II)

Sr. No.	Parameter	Nodular Cast Iron (ASTM 100-70-03)	Cast Steel (GS 70)	Forged Steel (AISI 1045)	Inconel X-750
1.	Total deformation (mm)	0.023424	0.020098	0.019384	0.018734
2.	Von-Mises stress (N/mm^2)	264.26	261.95	261.95	262.94

The comparison of analytical (ANSYS software) results for different materials of camshaft is as shown in Table 6.

Table 6. Comparison of analytical results for camshaft (for Case-II)

Sr. No.	Parameter	White Cast Iron	Nodular Cast Iron (SG Iron)
1.	Total deformation (mm)	0.00012457	0.00011559
2.	Von-Mises stress (N/mm^2)	0.61186	0.6177

XII. VALIDATION OF RESULTS

The analytical results were compared with the experimental results for validation purpose as shown in Table 7.

Table 7. Validation of Results

Crankshaft					
Case - I					
Sr. No.	Analytical Value AISI 1045		Experimental Value EN 8		Discrepancy
	Load (N)	Stress (N/mm^2)	Load (N)	Stress (N/mm^2)	
1.	19023	110.41 N	21119 N	122.19	10.66 %
Case - II					
1.	15218	261.95	15886	249.40	4.79 %
Camshaft					
Sr. No.	Analytical Value ASTM A536		Experimental Value ASTM A536		Discrepancy
	Load (N)	Stress (N/mm^2)	Load (N)	Stress (N/mm^2)	
1.	171	0.6177	175.40	0.877	29.55%

Some variation or discrepancy was found in the analytical and experimental results because of variation in material properties, unavailability of a highly costly and dedicated test rig and related fixtures. These variations were reasonably small.

XIII. CONCLUSIONS

The crankshaft and camshaft are designed to calculate their dimensions using theoretical design equations. The respective stresses calculated are less than the yield strength of the material. This ensures safety of the design. These dimensions are used to design the 3-D models in Creo 2.0 software.Concluding from the analytical results of Case-II for the crankshaft, none of the materials break. Inconel X-750 shows the minimum deformation and a high factor of safety at 3.23 based on its yield strength. Also, it is lighter in weight than other materials. This undoubtedly makes it the best alternative for existing materials from which the crankshaft is manufactured. It though is about three times costlier than steel grades, making it expensive and requires a special manufacturing process. Hence, it can be ideal for high performance cars like racing cars, supercars and high end luxury cars. Forged steel (AISI 1045) shows minimum value of stress and total deformation as compared to nodular cast iron and cast steel. Thus, it becomes the best suited material for manufacturing crankshafts of passenger cars and SUV's,

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where a combination of strength and refinement is required. The analytical results of camshaft show that nodular cast iron shows less deformation than white cast iron, both materials showing almost same stress. Thus, it is a better material for manufacturing camshafts as compared to white cast iron since it can inhibit the development of cracks.

It was observed from the stress analysis using ANSYS software, maximum deformation or displacement in crankshaft happens at the top central portion of the crankpin. The maximum stresses befall in the regions where there is quite a sudden change in the geometry of the crankshaft. This area is the junction where the crank web connects the shaft. This is a natural because the stress lines suddenly change themselves. This leads to the accumulation of high stresses over there, making the material weak. These fillets should be designed such that the crankshaft is least affected by these high stresses. Also, high surface finish should be maintained to minimize these stresses. This may be the reason that the manufacturers nowadays prefer forged materials over cast materials. Since the crankshaft is a rotating element and is imperilled by cyclic loading, there are chances of fatigue cracks being developed in these locations over time. This finally leads to fatigue failure.

REFERENCES

[1] Akkamahdevi G. Chanagond and L.B. Raut, "Finite Element Analysis of Roller Cam by Optimization of Surface Contact Area", International Journal of Advanced Engineering Research and Studies (IJAERS), E-ISSN 2249-8974, Vol. IV, Issue II, Jan-March 2015, pp. 05-06.

[2] Dr. Dmitri Kopeliovich, "Camshaft Bearings", Engine Professional, Jan-March 2015, pp. 82-86.

[3] Farzin H. Montazersadgh and Ali Fatemi, "DynamicLoad and Stress Analysis of a Crankshaft," SAE International, 2007-01-0258.

[4] Amit Solanki and Jaydeepsinh Dodiya, "Design and Stress Analysis of Crankshaft for Single Cylinder 4-Stroke Diesel Engine", International Journal for Research in Applied Science and Engineering Technology (IJRASET), ISSN 2321-9653, Vol. 2, Issue V, May 2014, pp. 320-324.

[5] Vivekanandan.P and Kumar. M, "Modelling, Design and Finite Element Analysis of Cam Shaft", International Journal of Current Engineering and Technology (IJCET), ISSN 2277-4106, Vol. 3, Issue 1, March 2013, pp. 220-223.

[6] D Jagan and V Ganesh, "Design and Analysis of a Camshaft using Al-SiC Composite with a Study of Latest Trends in Diesel Technologies", International Journal of Innovations in Engineering and Technology (IJIET), ISSN: 2319-1058, Vol. 7, Issue 1, June 2016, pp. 566-574.

[7] Machine Design by R.S. Khurmi and J.K. Gupta, Eurasia Publishing House (P) Ltd., New Delhi

[8] Prajakta P. Pawar, Dr. Santosh D.Dalvi, Santosh Rane and Dr. Chandra Babu Divakaran, "Evaluation of Crankshaft Manufacturing Methods -An Overview of Material Removal and Additive Processes", International Research Journal of Engineering and Technology (IRJET), Vol. 2, Issue 4, July 2015.

[9] Sujata Satish Shenkar and Nagraj Biradar, "Design and Static Structural Analysis of Crank Shaft", International Journal of Scientific Engineering and Technology Research (IJSETR), ISSN 2319-8885, Vol. 4, Issue 7, March 2015, pp. 1393-1398.

[10]P. Preetham and S. Srinivasa Prasad, "Design and Analysis of Six Cylinder Four Stroke Engine Crank Shaft", Journal of Technological Advances and Scientific Research (JTASR), eISSN- 2454-1788, ISSN- 2395-5600, Vol. 2, Issue 1, January-March 2016.

[11] Jian Meng, Yongqi Liu and Ruixiang Liu,"Finite Element Analysis of 4-Cylinder Diesel Crankshaft, " International Journal of Image,Graphics and Signal Processing, 5, 22-29, August 2007.

[12] B. Mounika and Madhuri .R .P, "Structural Static Analysis of Crankshaft", International Journal of Science and Research (IJSR), ISSN (Online)-2319-7064, Vol. 6, Issue 5, May 2017, pp. 987-987.

