# Design & Analysis of Crankshaft for Single Cylinder Diesel Engine

**Priyanka M. Chavan[1], Prof. R. H. Shinde[2]**

[1]*Post Graduate Student, Department of Mechanical Engineering, Deogiri Institute of Engineering and Management Studies, Aurangabad, Maharashtra, India*
[2]*Assistant Professor, Department of Mechanical Engineering, Deogiri Institute of Engineering and Management Studies, Aurangabad, Maharashtra, India*

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**ABSTRACT:** *Internal combustion engine has a wide range of application in different engineering field and automobile sector. Crank train is one of the essential driving part of internal combustion engine and its purpose is to translate the linear motion of piston into rotary motion for extracting useful work. The present study focuses on the design, performance analysis of a four stroke single cylinder diesel engine as crankshafts is one of the most critical component of internal combustion engines which subjected to very high dynamic loads during engine use. The crankshaft is essentially the backbone of the internal combustion engine; its failure will cause serious damage to the engine so it is important to verify its design. Design calculations are carried out theoretically and CREO 3.0 software used for 3D modeling of crankshaft. Finite element analysis is run to validate the design by using ANSYS software. It will determine the values of total deformation, Von mises stress and maximum principle stress.*

***KEY WORDS:*** *Crankshaft, CREO, Ansys, Finite element analysis*

## 1.1. INTRODUCTION:

The diesel engine is known as internal combustion engine or a compression ignition engine, in which fuel is injected into the combustion chamber and air-fuel mixture compressed at TDC which leads to increase in pressure & temperature in the combustion chamber. This increases the air temperature inside the cylinder to high degree that atomized injected diesel fuel into the combustion chamber ignites spontaneously. The Crank train is the heart of the reciprocating piston engine and its purpose is to translate the linear motion of the pistons into rotary motion or vice versa [(1).] Durability of this component and fatigue performance has to be considered in the design process, since the crankshaft subjected to a large number of load cycles during its service life Design and development of crankshaft have always been an important issue in the crankshaft industry, for manufacturing a less expensive part with the minimum weight and proper fatigue strength along with other functional requirements [(2).] Crankshaft vibration is one of the most crucial factors affecting operation of engine. Lateral vibrations and torsional vibrations affecting the crankshaft [(6)]. During service life, crankshaft operates under high forces resulting from gas combustion and inertia forces. These forces acting on the crankshaft cause two types of fluctuating loading i.e. torsional load and bending load. To survive safely during its service life, crankshaft requires high bending and fatigue strength and better balancing characteristics [(7).] The crankshaft experiences a cyclic load, due to this fatigue failure occur over a period. The fatigue analysis need to considered in the initial design stage itself. These improvements result in smaller and lighter engines with higher power output and better fuel efficiency [(9).] There are more challenges occurred while designing of crankshaft due to higher efficiency, increasing lower weight requirement, vehicle payloads and longer durability life. Crankshaft has to demonstrate durability throughout the entire engine life cycle hence the critical nature of the crankshaft requires consideration of optimization plans and calculations. Most of the design of crankshaft or any components has been utilized with help of finite element method (FEM) for the stress calculations [(10).] Two types of loading such as torsional and bending loading caused by inertia and combustion forces which are acting on the crankshaft structure [(11).]

Finite element analysis is widely used to obtain the variation of stress magnitude at critical locations [(12)].

As a result of a large number of moving components, inertial forces occur in the engine itself. In terms of overall energy efficiency and performance, industries have not received any suitable alternative for the IC engine. Hence for many machine's drive, IC engine will be the base for a long time [(13).]

## 1.2. MANUFACTURING PROCESS:

The manufacturing of crankshafts is done by forging or by casting in ductile steel. But forging have certain advantages such as more compact dimensions, lighter weight and better inherent dampening [(3)]. Nowadays, the more common crankshafts manufactured as:

a. assembled crankshafts, where the main journal, crankpin journal and crank webs are manufactured separately and then fitted together using a shrink fitting method;

b. Semi-built crankshaft where main journals and/or crank webs and crankpin journals are forged as one piece and shrink fitted together [(4)]. There are casting and closed die forging for an automobile engine. Die forging is mainly used for engine that needs the high rigidity and strength. However, in recent years, the forces to act on crankshaft become higher, because the high performance required by engines (7). Crankshafts can be cast in ductile steel or forged from a steel bar usually through roll forging. Due to lighter weight, better inherent damping and compact dimension, most of the manufacturers tend to use forged crankshafts (8).

## 1.3. HEAT TREATMENT:

Induction heating process has been considered as a high productivity, repeatable quality and green heating technology compared to fuel-fired furnaces in hot forging industry. Hence induction heating is considered as a best available heating technology and mostly preferred in forging manufacturing. Induction hardening requires a huge amount of electrical energy for heating a steel work piece with a huge volume from the ambient temperature to 1150 - 1250 C approximate. Therefore, the increase in the thermal efficiency of the heating system significantly saves the consumed energy (5). In the process of being forged All hot forged parts receive a certain amount of heat treatment and after that may be used without additional heat treatment. Many forgings are heat treated one or more times before being put into service for maximum usefulness. In order to improve wear and fatigue properties of the material, bearing sections and fillet areas of crankshafts are heat treated at these certain locations. Usually forgings are being heat treated before and after their machining. The objective of the initial treatment is to secure uniform structure of the metal and contribute to ease of machining of the forged part. The final treatment is very important to any component to makes it possible to use it for the service intended.

## 1.4. APPLICATION OF CRANKSHAFT:

Crankshafts has wide applications in various branches of engineering. They are used whenever there is the requirement to translate reciprocating linear motion into rotation or vice-versa. Crankshaft has more variety of configurations which generally used in internal combustion engines. The internal combustion engines cover different fields of uses, from small scale model planes to large maritime engines. So crankshafts mainly used in engines for rail, road and maritime transport, electrical generators, portable machinery, industrial and agricultural machinery. Crankshafts are also used in driven machinery such as air reciprocating pumps and compressors (8). Crankshafts is mainly used in internal combustion engines and consist of crankpin to which connecting rod attached. It rotates within the engine block through use of main bearings and the crankpin rotate within the connecting rod big end bearing.

## 2.1. LITERATURE REVIEW:

S. S. Shenkar and N. Biradar, studied the stress analysis of a crankshaft of single cylinder engine by using finite element method. 3D model of crankshaft was created in Proe software. ANSYS software was used to analyse the distortion and stress of the crank throw. The maximum deformation, dangerous areas and maximum stress point are found by the stress analysis. The results would provide a valuable theoretical foundation for the improvement and optimization of engine design [1].

Anbu T. conducted analysis in three modules: modal, static, and transit. The research was carried out on crankshaft of a multi-cylinder engine in order to determine the stresses and deformation. They took values from the load characteristics of the engine as boundary values. In the end, they presented a proposal to reduce the cost of production of materials that could be used [2].

Naik, described the reliability of the crankshaft of four-cylinder engine. It has been concluded that the crankshaft fracture occurs mostly on the first flying sleeve which is closest to the engine's flywheel. By using Stress analysis of the crankshaft, it also represents that maximum amplitudes of the stresses at critical cross sections [3].

M. Fonte, studied the fatigue life of crankshafts of marine engine and its maintenance. Estimation of fatigue life is very important to ensure reliability and safety of components and by taking into the account a design improvement. Crankshafts are subjected to rotating bending combined with torsion on main journals and bending on crankpins, which are responsible for fatigue failure mostly. An example of a semi-built crankshaft failure is also presented along with probable root cause of failure and at the end some final remarks are presented [4].

R. Metkar, represents finite element analysis is the most favourite method to solve fatigue analysis and stress and it is widely used for analysing problems of engineering. He also studied strain life, stress life and LEFM methods to solve fatigue analysis. There are lot of software available for use in finite element analysis applications, such as Abacus, Nastran, ANSYS and MSC [5].

Amit Chaudhari, modified crankshaft geometry by adding some material at inclination face on crankshaft and found out there is improvement in results as compared to original geometry of crankshaft for torsional vibration [6].

Vijaykumar Khansis, studied fatigue life, balancing and stress of crankshaft by modifying crankshaft geometry by adding very small amount of material at bevel section. It is observed that maximum stresses generated at bevel section were reduced in modified geometry as compared to original crankshaft [7].

K. Durga Prasad,K. V. J. P. Narayana,N. Kiranmayee, concluded that FEA is a good tool to reduce time consuming theoretical work. It is observed that Maximum stress appears near the central point Journal and at the fillets between the crankshaft journal and crank cheeks and from analysis. Maximum deformation appears at the center of crankpin neck surface Von-Misses Stresses that comes out from the analysis is far less than material yield stress so our design is safe. Deformation and Stresses and are critical input to fatigue analysis and optimization of the crankshaft [8].

K.Sandya, M Keerthi, K.Srinivas, analyze the crankshaft in several positions of the crank. The static analysis is conducted on the crankshaft with three different materials such as Structural steel, Al 6061, Inconel x750 in different orientations. The results are validated with theoretical calculations for two crank positions for all materials and it is observed that Al6061 is subjected to high von-mises stresses compared to remaining two materials. Inconel x750 is subjected to little deformation when compared to remaining two materials [9].

R. J. Deshbhratar, and Y.R Suple carried out analysis on crankshaft of four cylinder and 3D model were created by Pro/E Software and then imported to ANSYS software. From the analysis, it is observed that the maximum deformation occurs at the center of crankshaft surface. The maximum stress observed at the fillets between the crankshaft journal and crank cheeks, and near the central point. The high stress area is edge of main journal. The maximum deformation observed at the link between crankpin and crank cheeks and main bearing journal [10].

Ashwin kumar Devaraj determined nature, frequencies and critical stress of the crankshaft and the mode shapes. For analysis purpose, ansys software has been used and it is found that at 355 degrees of crank angle, the maximum load appears for this specific engine. It is also observed that there is no effect of torsional load on von Mises stress in the overall dynamic loading conditions at the critically stressed location. It is necessary to keep the excitation frequencies far away from the natural frequencies which will help to reduce resonance problems [11].

Farzin H. Montazersadgh, determined the critically stressed location and stress variation in the crankshaft over an entire cycle from the FE model & dynamic load analysis. Also stated the effects of torsion load on engine speed and stresses [12].

Xiaorong Zhou, determined that the stress concentration of stress is largely observed in the spindle neck fillet and the stress at crankpin fillet is also higher in static analysis of the crankshaft. Fatigue strength calculated on the basis of of stress analysis [13].

## 2.2. OBJECTIVE:

- To design the crankshaft by using standard mathematical design procedure.
- To create a three dimensional model of crankshaft by using CREO 3.0. parametric software as per calculations.
- To run FEA analysis on designed crankshaft by considering engine gas combustion load and torque parameters by using ANSYS Workbench software to evaluate the total deformation, von-mises stress, and maximum principal stress.

## 3.1. EXPERIMENTAL PROCEDURE:

The main source of forces applied to a crankshaft is the product of combustion pressure which is acting on the piston top. This kind of huge force exerted onto a crankshaft rod journal which produces substantial torsional and bending moments and the resulting shear, compressive and tensile stresses. Hence we need to considered all these parameters and design the crankshaft accordingly.

## 3.2. SPECIFICATION OF ENGINE:

To design crankshaft, we need to understand engine specification first as tabulated in Table 3.2.

Table 3.2. Engine Specification

| | |
|---|---|
| No. of cylinder | Single |
| Rated Speed | 3600 RPM |
| Bore x Stroke | 90 x 90 mm |
| Max. Power | 8 kW @ 3600RPM |
| Max. Torque | 24 N.m @ 1800-2200 RPM |
| Max. Combustion Pressure | 75 bar |

## 3.3. MATERIAL SELECTION:

Crankshaft is working under heavy load and continuous changeable bending moment and torsion moment, common failure modes are the bending fatigue fracture and journal wear. Therefore, requires the crankshaft material with high rigidity, fatigue strength and good wear resistance. Crankshaft materials should be readily sharped, machined and heat treated and has adequate strength, toughness, hardness and high fatigue strength. The crankshafts are generally manufactured from steel material either by casting or forging. Forged crankshafts are stronger in nature than casted crankshafts. EN19 is a high quality alloy steel with good tensile strength. With a combination of shock resistance and good ductility, EN19 is suitable for very high loading applications.

### 3.3.1. CHEMICAL COMPOSITION OF EN19:

EN19 material is considered for the crankshaft of selected diesel engine. The detailed composition of selected material is as mentioned in below Table 3.3.1.

Table 3.3.1. Chemical composition of EN19

| C | Si | Mn | Cr | Mo | S | P |
|---|---|---|---|---|---|---|
| 0.45 | 0.30 | 0.80 | 1.50 | 0.40 | 0.05 | 0.035 |

| | | | | | | |
|---|---|---|---|---|---|---|
| % | % | % | % | % | % | % |

Crankshafts require the following characteristics:

- High stiffness and strength to withstand the high loads in engines and to offer opportunities for weight reduction and downsizing.
- Resistance to fatigue in bending and torsion.
- Resistance to wear in the Low vibration and bearing areas.

Thus the forged steel crankshafts offer higher stiffness and strength along with other material characteristics than the cast iron.

### 3.4. BLOCK DIAGRAM OF EXPERIMENTAL PROCEDURE:

Flow of Procedure of design & analysis of crankshaft for selected engine is as illustrated in below figure 3.4.

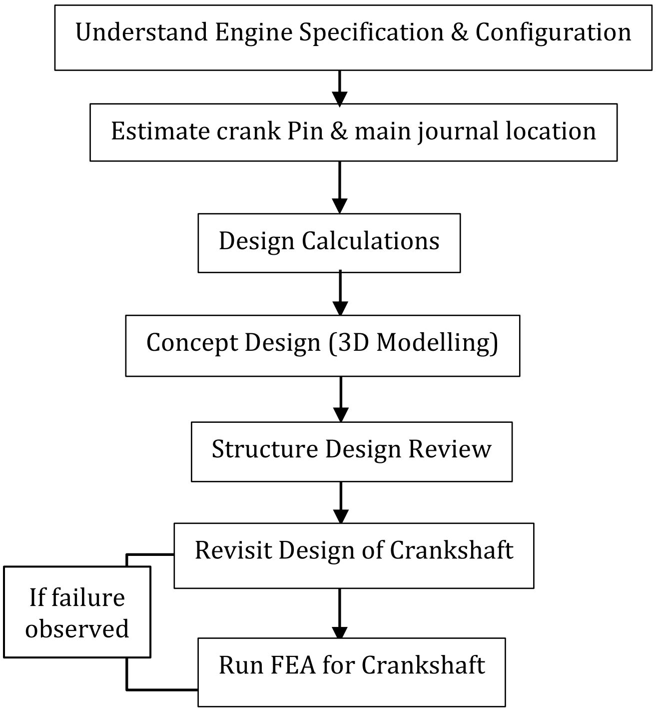


Figure 3.4. Block Diagram of Experimental Procedure

### 3.5. DESIGN CALCULATIONS:

The maximum gas combustion pressure on the piston will transmit maximum force on the crankpin causing only bending of the shaft. The crankpin and ends of the crankshaft will be subjected to bending moment only. Thus, the bending moment on the shaft is maximum and the twisting moment is zero when the crank is at the dead center.

Input data:

- Cylinder Bore Diameter (D) = 90mm
- Stroke (l) = 90 mm
- Crank throw (r) = 90/2 = 45 mm
- Length of connecting rod (L) = 145 mm
- L/r ratio = 3.222
- Maximum gas combustion pressure on the piston top for maximum torque condition (P') = 75 bar
- Angle of Inclination of crank with line of dead center (θ) = $25^0$
- Angle of inclination of connecting rod with the line of dead center (φ):
  sin φ = sin θ / (L/r)
  φ = $7.5370\ ^0$

3.5.1. Evaluate Gas Force acting on piston due to gas pressure (Pp):

The crank is at the top dead center position and experience no torsional moment and maximum bending moment. The thrust in the connecting rod will be equal to the forces acting on the piston at the top dead center position.

Force on the piston = Combustion pressure x Area of the bore

$P_P = P' \times \frac{\pi}{4} d^2$

$P_P = 47712.938$ N ... (3.1)

3.5.2. Evaluate Thrust Force on connecting rod (Pq):

Pq = Pp / cos φ

Pq = 48128.74481 N ... (3.2)

3.5.3. Evaluate component of force on crank pin (P):

Thrust on the crank shaft can be split into Tangential and radial component as shown in figure 3.5.3.

3.5.3.1. Tangential component of force on crank pin (Pt):

Pt = Pq sin (θ+φ)

= 25885.76 N ... (3.3)

3.5.3.2. Radial component of force on crank pin (Pt):

Pr = Pq cos (θ+φ)

= 40574.66 N ... (3.4)

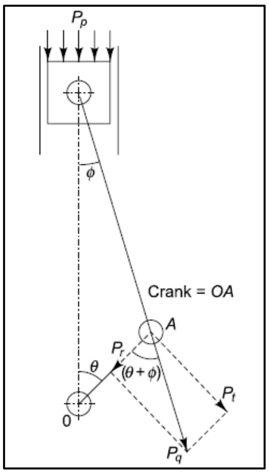


Figure 3.5.3. Forces acting on crank (Ref. V.B. Bhandari book)

3.5.4. Evaluate reactions at bearings:

Crankshaft is simply supported at bearings 1 & 2 and subjected to tangential force Pt & radial force Pr at the crank pin as shown in figure 3.5.4.

b1 = Distance of crankpin from bearing 1 near to flywheel side = 83.65 mm

b2 = Distances of crankpin from bearing 2 near to Power Take Off side = 107.65 mm

b = Total distance between bearings 1 & 2 = 191.3 mm

Due to tangential component Pt, there are reactions (R1)h an(Ref. V.B. Bhandari)d (R2)h at bearings 1 & 2 respectively. Similarly, there are reactions (R1)v and (R2)v at bearings 1 & 2 respectively due to the radial component Pr.

Horizontal forces at bearing 1:

(R2)h = (Pt x b1) / (b1+b2)

(R2)h= 11319.099 N .... (3.5)

- Horizontal forces at bearing 2:

(R1)h = (Pt x b2) / (b1+b2)

(R1)h= 14566.66 N ... (3.6)

- Vertical forces at bearing 1:

(R2)v = (Pr x b1) / (b1+b2)

(R2)v = 17742.13 N ... (3.7)

- Vertical forces at bearing 2:

(R1)v = (Pr x b2) / (b1+b2) ... (3.8)

(R1)v = 22832.525 N

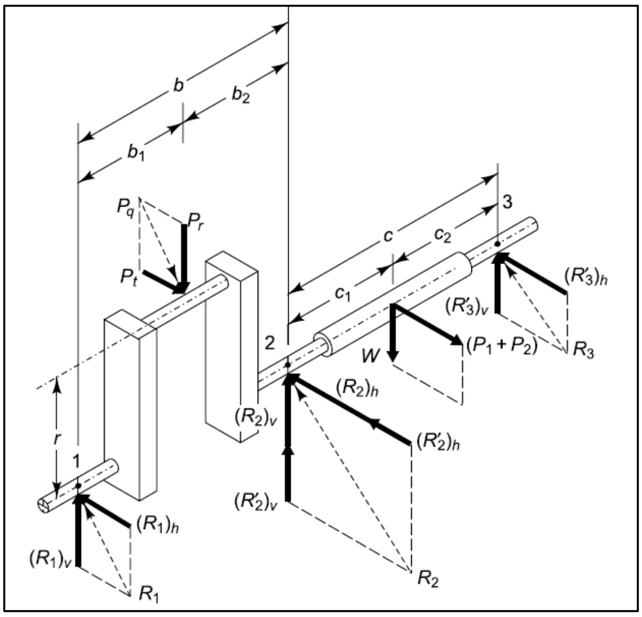


Figure 3.5.4. Support reactions in crankshaft(Ref. V.B. Bhandari book)

3.5.5. Design of crank pin (dc):

Let, crankpin diameter in mm = dc

The crankpin central plane is subjected to:

- Bending moment (Mb) due to (R1)v

  Mb = (R1)v X b1

  Mb=1909940.71N.mm ... (3.9)

- Torsional moment (Mt) due to (R1)h

  Mt = (R1)h X r

  Mt=655499.7N.mm ... (3.10)

Now, we can get equivalent twisting moment,

Te = $\sqrt{(Mb)2 + (Mt)2}$

Te =2019.2952 kN.mm ... (3.11)

The diameter of the crank pin (dc) is calculated by below equation:

dc$^3$ = $\frac{16}{\pi\tau}\sqrt{(\mathrm{Mb})2 + (\mathrm{Mt})2}$

Where, $\tau$= Allowable shear stress

= 40 N/mm2 (Assuming as per V. B. Bhandari book) dc = 63.58 mm .... (3.12)

Crankpin is the crucial part as full gaseous load, (load due to combustion) will be acting on the crankpin directly. The force generated when the peak firing pressure is achieved in the engine is the maximum force, would have to withstand. The schematic diagram for crankpin is shown in below figure 3.5.5.

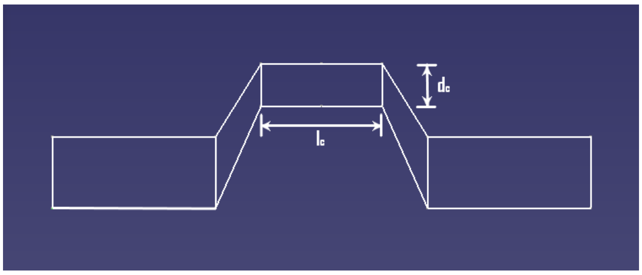


Figure 3.5.5. Schematic diagram of crankpin

3.5.6. Design of crank pin length (lc):

$\frac{lc}{dc}$ = 1.2 .... (Assuming as per V. B. Bhandari book).

Hence, lc = 1.2 x dc = 76.29 mm

Pb = $\frac{Pp}{dc \times Pb}$ .... (3.13)

Where, Pb =Allowable bearing pressure at the crank pin bush

So we get, Pb = 9.8366 N/mm2

Hence, Pb < 10 N/mm2, means the calculated bearing pressure is within the limits and design is safe.

3.5.7. Design of left hand crank web:

The left-hand web and right-hand webs are made identical from balancing considerations. Therefore, the width and thickness of the left-hand crank web are made equal to that of the right-hand crank web. Generally, the crank web is designed for eccentric loading. There will be two stresses acting on the crank web, one is bending stresses in vertical and horizontal planes due to radial component and tangential component and another is direct compressive stress due to radial component.

Let,

t = Thickness of crank web

w = Width of crank web

3.5.7.1. The width of crank web (w) is taken as,
w = 1.14 x dc = 72.4812 mm ... (3.14)

3.5.7.2. The thickness (t) of the crank web is given by empirically relation as,
t = 0.7 x dc = 44.506 mm ... (3.15)

3.5.8. We know that maximum bending moment on the crank web is given by,

(Mb)r = (R2)v * [ b2 – $\frac{lc}{2}$ - $\frac{t}{2}$ ] ... (3.16)

= 838297.900 N.mm (From eqn. no. 3.7, 3.13 & 3.15)

3.5.9. Section Module is,

Z = $\frac{w \times t^2}{6}$ ... (3.17)

Z = 23928.26 mm

3.5.10. Bending stress induced in the crank web:

σb= $\frac{M}{Z}$ ... (3.18)

σb= 35.03 N/mm2
Here, induced bending stress is less than the allowable bending stress which is (75 N/mm2). Hence the design is safe.

3.5.11. Design of shaft under flywheel:
Design of shaft carried out by considering two cases:

- First case: The central plane of the shaft is subjected to maximum bending moment due to weight of flywheel and resultant belt tension.
- Second case: The central plane of the shaft is subjected to maximum bending moment due to reaction R3 & tangential component Pt.

Determine the shaft diameter by considering first case:

a. The bending moment due to weight of the flywheel in vertical plane is given by,
(Mb)v = (R′3)v x c2 ... (3.19)

Where, c1 = c2 = 125 mm &

(R′3)v = ($\frac{W}{2}$) (w is flywheel weight load in N)... (3.20)

(R′3)v = ($\frac{1000}{2}$)

(R′3)v = 500 N

Hence, (Mb)v = 62500 N.mm

b. The bending moment due to resultant belt tension in horizontal plane is given by,
(Mb)h = (R′3)h x c2 ... (3.21)

$(R'3)h = (\frac{P1+P2}{2})$ ...(P1 + P2 is total belt pull load in N)

$(R'3)v = (\frac{1200}{2})$ ... (3.22)

$(R'3)v = 600$ N

Hence, (Mb)h = 75000 N.mm

Now, the resultant bending moment is given by,

$$Mb = \sqrt{(Mb)v2 + (Mt)h2} \quad .... (3.23)$$

= 97628.12 N.mm

the diameter of the shaft (ds) is calculated by the following equation,

$$Mb = (\frac{\pi * ds3}{32}) \text{ x } \sigma b \quad ... (3.24)$$

Assume, the allowable bending stress is 75 N/mm2

(as per V. B. Bhandari book)

ds = 23.66 mm

Determine the shaft diameter by considering second case:

a. The central plane of the shaft is subjected to maximum bending moment due to reaction $R_3$:

Mb = (R3) x c2 .... (3.25)

where, c1 = c2 = 125 mm

$$R_3 = \sqrt{[(R'3)v]2 + [(R'3)h]2} \quad .... (3.26)$$

= 781.02494 N (From eqn. 3.20 & 3.22)

Hence, Mb = 97628.1175 N.mm

b. The central plane of the shaft is subjected to maximum bending moment due to tangential component Pt:

Mt = Pt x r ... (3.27)

= 1164859.2 N.mm (From eqn. 3.3)

The shaft diameter (ds) is calculated by the following equation:

$$ds^3 = \frac{16}{\pi\tau}\sqrt{(Mb)2 + (Mt)2} \quad \text{(From eqn. 3.25 \& 3.27)}$$

Assume, allowable shear stress τ is 40 N/mm$^2$

ds = 52.99 mm

In calculations of the first case, the diameter (ds) is 23.66 mm. Since the diameter is less, the second case is the criterion of deciding the diameter of the shaft under flywheel. Therefore,

Ds = 52.99 mm ≈ 53 mm

Results:

Diameter of the crankpin (dc) = 63.58 mm ≈ 64mm

Length of the crankpin (lc) = 76.29 ≈ 76.3 mm

Diameter of the shaft (ds) = 52.99 mm ≈ 53 mm

Web thickness (both left and right hand) (t) = 44.5 mm

Web width (both left and right hand) (w)= 72.48 ≈ 72.5 mm

### 4.1. MODELLING OF CRANKSHAFT:

CAD software assist designers and engineers in a wide variety of industries to design physical products [15]. Creo supports multiple stages of 3D product design whether started from scratch or from 2D sketches. Creo is able to read and produce STEP format files for various purpose including analysis. Modelling of crankshaft done in Creo software on the basis of calculated design parameters as shown in figure 4.1.

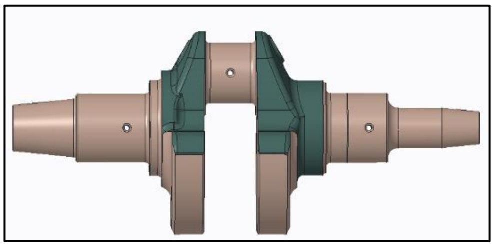

Figure 4.1. 3D Model of designed crankshaft

### 4.2. FEA ANALYSIS OF CRANKSHAFT:

The finite element method (FEA) is numerical analysis technique for obtaining approximate solutions to a wide variety of engineering problems. To check the performance of designed crankshaft, analysis carried out by using ANSYS tool. In this software, boundary condition in the model is based on the engine configuration.

- Input for FEA:

a. Model: 3D Solid model of Crankshaft

b. Pre-processor: Meshing of the Crankshaft

Material Properties of crankshaft are tabulated in below table 4.2.

Table 4.2. Material Properties of crankshaft

| Description | Young's Modulus | Density | Poisson Ratio | Yield Strength | Tensile Strength |
|---|---|---|---|---|---|
| Unit | GPa | Kg/mm³ | -- | MPa | MPa |
| | 210 | 780 | 0.3 | 660 | 870 |

### 4.3. MESHING AND CONSTRAINT OF CRANKSHAFT:

Meshing is very important components for obtaining accurate results from an FEA model. By meshing, you break up the domain into pieces, each piece representing an element as shown in figure 4.3. hence it plays a vital role in the accuracy and reliability of the analysis performed. Due to the combustion in the engine, gas forces are caused during the power stroke which are directly related to the peak firing pressure. The gas force is applied as a load of the crankshaft. The cylinder pressure was recorded for each degree of the crankshaft during the pressure measurement in the engine cylinder. Since the operating cycle of the four-stroke IC engine takes place in two full turns of the crankshaft, or 720 degrees, cylinder's pressure values are recorded, including 0 degrees. Therefore, the maximum gas force recorded is 70 bar at 11 degrees after TDC at engine rated speed of 3600 rpm.

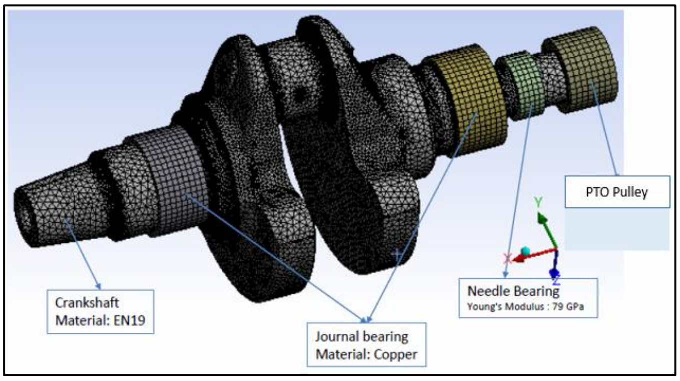


Figure 4.3. Meshing of crankshaft

## 5. RESULT AND DISCUSSION:

ANSYS software analysis is performed based on the mentioned boundary conditions, we get output parameters such as total deformation, Von Mises stress and maximum principle stress.

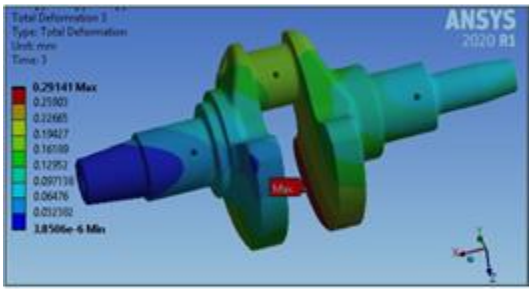


Figure 5.1. Total deformation analysis

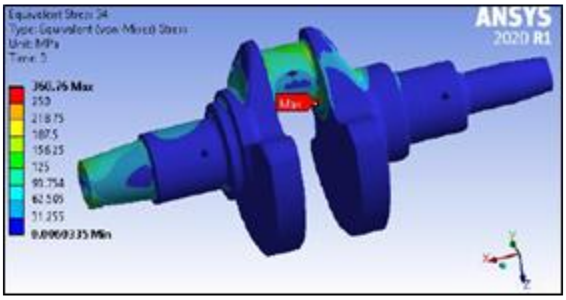


Figure 5.2. Von Mises Stress analysis

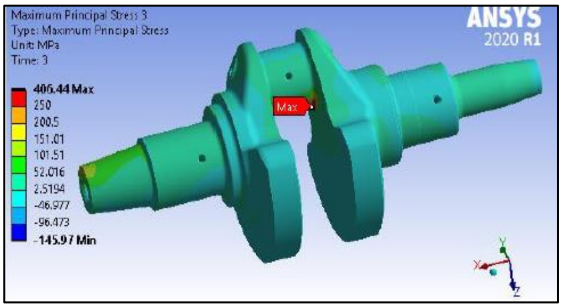


Figure 5.3. Maximum Principle Stress analysis

The results obtained can be seen in figures 5.1., figure 5.2. and figure 5.3. It can be noticed from figure 5.1. that total deformation is 0.29141 mm and occurs at crank web location. Similarly, in figure 5.2 & figure 5.3, highest von mises & maximum principle stress is observed 360.76 MPa and 406.44 MPa respectively at crank pin fillet location due to gas pressure. By considering the obtained values and material of crankshaft, it can be concluded that the reliability of the crankshaft is not get distorted at such deformation and stresses.

## 6. CONCLUSION:

This research shows a numerical method to determine the design of crankshaft. Hand calculation is the first step to finalized the design parameters. The design of the crankshaft is necessary for the view that there is less stress concentration and avoid the failure. In this paper, the crankshaft model was created by Creo software and then imported to ANSYS software in the step format. The crankshaft is subjected to inertia load & torque; hence it was necessitude to evaluate the stresses considering constraint and load in analysis. FEA Analysis results are important information to conclude that whether design is safe or not with following observations:

- Maximum total deformation occurs at crank web location.
- Von Mises stress & Maximum principle stress under the combustion gas load are within the yield strength limit and ultimate strength limit of the material respectively.
- Maximum stress concentration observed at crank pin fillet location.
- Deformation and stresses are important input to fatigue analysis and optimization of the crankshaft,
- Crankshaft with EN19 material is safe for 75 bar peak cylinder pressure operation without any failure.

**ACKNOWLEDGEMENT:**
The support extended by the DIEMS (Deogiri Institute of Engineering and Management Studies) and college authorities is highly appreciated and acknowledged with due respect.

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