International Journal of Scientific Research in Engineering and Management (IJSREM)
Volume: 09 Issue: 03 | March - 2025 SJIF Rating: 8.586 ISSN: 2582-3930

Design Optimization and Structural Analysis of an Internal Combustion Engine Crank Shaft

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ABSTRACT

The crankshaft is a critical mechanical component in internal combustion engines, responsible for converting the reciprocating motion of the pistons into rotational motion to drive the vehicle or machinery. This project focuses on the design, modeling, and analysis of a high-performance crankshaft using advanced engineering tools. A 3D model of the crankshaft was developed using solid works software to achieve optimized geometry that balances strength, weight, and manufacturability. Finite Element Analysis (FEA) was conducted by using Ansys to evaluate the stress distribution, deformation, and fatigue life under dynamic loading conditions. The study also explores material selection and surface treatments to enhance durability and efficiency. The results provide valuable insights into improving crankshaft reliability and performance in modern engines, contributing to reduced mechanical failures and enhanced powertrain longevity.

Keywords: Crank shaft, internal combustion engine, Solid works, finite element analysis, Ansys.

I. INTRODUCTION

The crankshaft is a fundamental component in an internal combustion engine, responsible for converting the reciprocating motion of the pistons into rotational motion. This transformation is essential for transferring power to the vehicle's wheels or other machinery driven by the engine. The crankshaft's role is critical because it directly influences the engine's performance, efficiency, and durability. To achieve this, the crankshaft must endure intense mechanical stresses, including torsional forces, bending loads, and heat exposure. As a result, its design and construction are meticulously engineered to ensure optimal functionality.

A crankshaft's operation is closely linked to the movement of the pistons. During the combustion process, the expanding gases force the pistons downward in a linear motion. This force is transferred to the crankshaft through the connecting rods, which are attached to offset crankpins. As the connecting rods push and pull the crankpins, the crankshaft rotates, converting the pistons' linear movement into rotary motion. This rotation is ultimately transferred to the flywheel, ensuring smooth power delivery to the drivetrain.

The crankshaft is composed of several essential parts, each serving a distinct purpose. Main journals are the primary bearing surfaces that allow the crankshaft to rotate within the engine block. These journals are precision-machined to ensure minimal friction and proper alignment. Crankpins, located on offset segments of the crankshaft, connect to the pistons via connecting rods and are critical in defining the engine's stroke and displacement. To maintain balance and minimize vibrations, counterweights are strategically integrated into the crankshaft. These counterweights reduce the dynamic forces created by the pistons' motion, improving the engine's smoothness and efficiency. Additionally, the crankshaft is equipped with internal oil passages that provide lubrication to critical bearing surfaces, reducing wear and heat buildup.

Material selection plays a significant role in the crankshaft's strength and durability. Crankshafts are commonly made from forged steel, cast iron, or billet steel, depending on the engine's performance demands.

Forged steel crankshafts offer exceptional strength and are commonly used in performance and heavy-duty applications. Cast iron crankshafts, while more cost-effective, are ideal for standard passenger vehicles. High-performance engines may utilize billet steel crankshafts, which are CNC-machined from solid steel blocks for superior precision and strength.

The manufacturing process of a crankshaft is complex and involves multiple stages. Initially, the crankshaft is forged or cast into its rough shape. Precision machining processes then refine its dimensions to ensure proper fitment and alignment. Heat treatment is applied to harden the crankshaft's surface, enhancing its resistance to wear and fatigue. Finally, grinding and polishing techniques are used to achieve smooth journal surfaces, reducing friction and improving the crankshaft's efficiency.

Crankshafts are designed to suit different engine configurations. Flat plane crankshafts, commonly found in high-performance engines, provide quicker acceleration due to reduced rotational inertia but tend to produce more vibrations. In contrast, cross plane crankshafts, typical in V8 engines, offer superior balance and smoother operation, making them ideal for passenger vehicles. In some cases, large industrial engines may use built-up crankshafts, which allow individual components to be assembled for easier maintenance and repair.

In conclusion, the crankshaft is a vital component that dictates the engine's performance, reliability, and efficiency. Its intricate design, durable material composition, and precision engineering ensure smooth power delivery under extreme conditions. By understanding the crankshaft's working principles, components, and potential failure points, vehicle owners and mechanics can enhance engine longevity and maintain peak performance.

II. 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].

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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 | ISO 9001:2008 Certified Journal | Page 1312 International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056 Volume: 08 Issue: 09 | Sep 2021 www.irjet.net 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].

III. **OBJECTIVE**

- To design the crankshaft by using standard mathematical design procedure.
- To create a three dimensional model of crankshaft by using Solid works parametric software as per calculations.
- To run FEA analysis on designed crankshaft by considering engine gas combustion load and software to evaluate the total deformation, von mises stress.
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IV.EXPERIMENTAL PROCEDURE:

The primary force acting on a crankshaft is generated by the combustion pressure exerted on the top of the piston. This intense force is transferred to the crankshaft rod journal, resulting in significant torsional and bending moments. Consequently, the crankshaft experiences various stresses, including shear, compressive, and tensile forces. To ensure the crankshaft's durability and performance, these factors must be carefully evaluated and incorporated into its design.

V.SPECIFICATION OF ENGINE

Parameter	Specification
Number of Cylinders	1
Rated Speed	5100 RPM
Bore x Stroke	70 x 90 mm
Maximum Power	20.3 PS @ 5250 RPM
Maximum Torque	58 Nm @ 4000 RPM
Maximum Combustion Pressure	75 Bar
Length of Connecting Rod	124

VI. MATERIAL SELECTION

Material selection for a crankshaft is a crucial step that directly affects its performance, durability, and resistance to mechanical stresses. Crankshafts are typically made from materials that offer high strength, fatigue resistance, and the ability to endure extreme loads. Common materials used for crankshaft manufacturing include forged steel, cast iron, and billet steel.

VI.I APPENDIX

CHEMICAL COMPOSITION OF ALUMINIUM D16

Element	**Composition (%)**
Aluminium (Al)	**90.0 - 93.0**
Copper (Cu)	**3.8 - 4.9**
Magnesium (Mg)	**1.2 - 1.8**
Manganese (Mn)	**0.3 - 0.9**
Iron (Fe)	**$\leq$ 0.5**
Silicon (Si)	**$\leq$ 0.5**
Zinc (Zn)	**$\leq$ 0.3**

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Nickel (Ni)	**≤ 0.1**
Titanium (Ti)	**≤ 0.15**

CHEMICAL COMPOSITION OF ALUMINIUM ALLOY

Element	Composition (%)
Al	85 - 95
Si	4 - 13
Cu	0.5 - 5
Mg	0.3 - 5
Zn	0.5 - 6
Mn	0.1 - 1.5
Fe	0.2 - 1
Ti	0.01 - 0.2
Cr	0.05 - 0.3
Ni	0.1 - 1

CHEMICAL COMPOSITION OF TITANIUM ALLOY

lement	Composition (%)
Ti	88 - 99
Al	2 - 8
V	1 - 5
Fe	0.1 - 0.5
O	0.03 - 0.2
C	0.01 - 0.08
N	0.01 - 0.05
H	0.001 - 0.015

CHEMICAL COMPOSITION OF GREY CAST IRON

Element	Composition (%)
Fe	92 - 95
C	2.5 - 4
Si	1 - 3
Mn	0.1 - 1
P	0.05 - 1
S	0.02 - 0.2

CHEMICAL COMPOSITION OF MEDIUM CARBON STEEL

Element	Composition (%)
Fe	Balance
C	0.3 - 0.6
Mn	0.6 - 1.65
Si	0.15 - 0.35
S	≤ 0.05
P	≤ 0.04

VII DESIGN CALCULATIONS

The highest gas combustion pressure acting on the piston generates the greatest force on the crankpin, resulting in bending of the crankshaft. In this condition, the crankpin and the ends of the crankshaft experience only bending moments. Consequently, when the crank is positioned at the dead center, the bending moment on the shaft reaches its peak, while the twisting moment becomes zero.

Input data:

- Cylinder diameter = 90mm
- Stroke (l) = 90 mm
- Crank throw (r) = 90/2 = 45 mm
- Length of connecting rod (L) = 124 mm
- L/r ratio = 124/45 = 2.75
- Angle of Inclination of crank with line of dead center (θ) = 25
- Angle of inclination of connecting rod with the line of dead center (φ): sin φ = sin θ / (L/r) φ = 8.84°

vii.i. 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

$PP = P' \times \pi/4\, d^2$

PP = 28863.382 N

Evaluate Thrust Force on connecting rod (Pq):

$Pq = Pp / \cos \varphi$

Pq = 29210.36 N

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

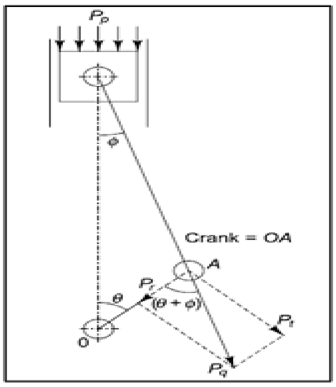
Thrust on the crank shaft can be split into Tangential and radial component.

Tangential component of force on crank pin (Pt):

Pt = Pq sin (θ+φ) = 16266.537 N

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

Pr = Pq cos (θ+φ) = 24262 N



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

vii.vi. 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.

b1 = Distance of crankpin from bearing 1 near to flywheel side = 70mm.

b2 = Distances of crankpin from bearing 2 near to Power Take Off side = 70mm.

b = Total distance between bearings 1 & 2 = 140 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= 8133.26 N

- Horizontal forces at bearing 2:

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

(R1)h= 8133.26 N

- Vertical forces at bearing 1:

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

(R2)v = 12131 N

- Vertical forces at bearing 2:

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

(R1)v = 12131 N

vii.v. 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=849170 N.mm

• Torsional moment (Mt) due to (R1)h

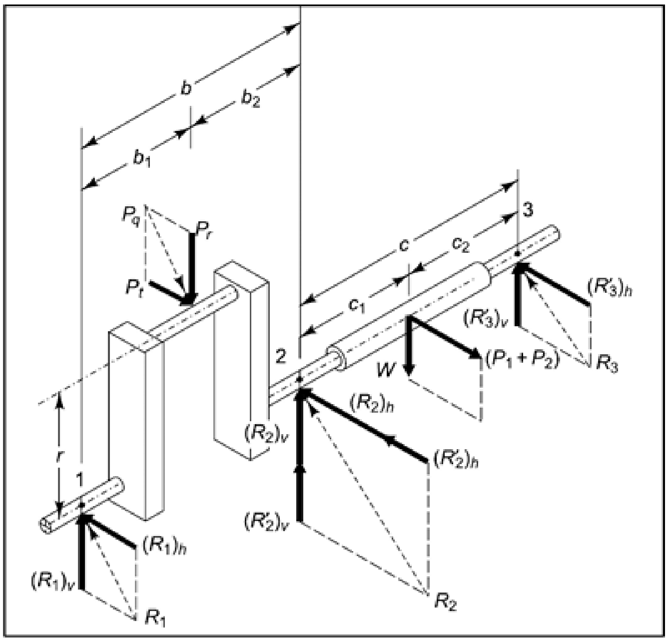
Mt = (R1)h X r

Mt=284664.1 N.mm

Now, we can get equivalent twisting moment

Te = Te $=\sqrt{(MB2)\ +(MT^2)}$

Te =895613.38 kN.mm



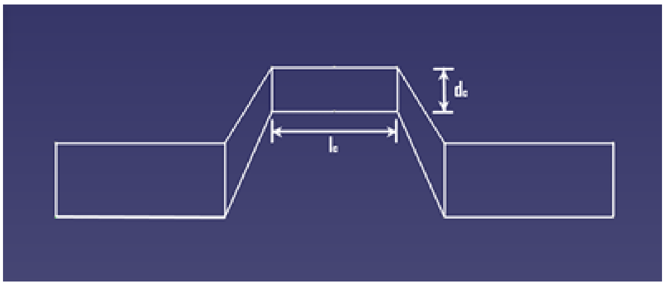
Support reactions in crankshaft(Ref. V.B. Bhandari book)

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

$$dc3 = \frac{16}{\pi\tau}\sqrt{(MB2)\ + (MT^2)}$$

Where τ = Allowable shear stress = 40 N/mm2 (Assuming as per V. B. Bhandari book) dc = 48.5 mm

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.



Schematic diagram of crankpin

Design of crank pin length (lc):

Lc/dc = 1.2 …. (Assuming as per V. B. Bhandari book). Hence,

lc = 1.2 x dc = 58.19 mm

$$Pb = \frac{Pp}{dc*lc}$$

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

So we get, Pb = 10 N/mm2

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

vii.vi. 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

The width of crank web (w) is taken as,

w = 1.14 x dc w= 55.29 mm

The thickness (t) of the crank web is given by

t = 0.7 x dc t= 33.39 mm

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

(Mb)r = (R2)v * [b2 – lc /2 – t/2]

= 253901.83 N.mm

Section Module:

$$Z= \frac{w*t2}{6}$$

Z= 11256.14 mm

Bending stress induced in the crank web:

σb =$(MB)r\ Z$

σb= 22.55 N/mm2

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

vii.vii. 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

Where, c1 = c2 = 125 mm (assumed)

(R´3)v = (w 2)
(w is flywheel weight load in N)

(R´3)v = (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

(R´3)h = $\frac{(P1+P2)}{2}$

(R´3)v = (1200/2)

(R´3)v = 600 N) …(P1 + P2 is total belt pull load in N)

Hence, (Mb)h = 75000 N.mm

Now, the resultant bending moment is given by,

Mb = $\sqrt{(\mathrm{MB})\mathrm{V2} + (\mathrm{MB})\mathrm{h2}}$

= 97628.12 N.mm

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

Mb = (∏ ∗ ds3 32) x σb

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 R3

Mb = (R3) x c2 where, c1 = c2 = 125 mm

R3 = $\sqrt{[(R13)V2 + (R13)H2}$

= 781.02494 N

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

= 1164859.2 N.mm

ds^3 = $\frac{16}{\pi\tau}\sqrt{(MB)2 + (MT)2}$

ds = 45.47 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 = 45.47 mm ≈ 45.5 mm

Results: Diameter of the crankpin (dc) = 48.5 MM

Length of the crankpin (lc) = 58.2MM

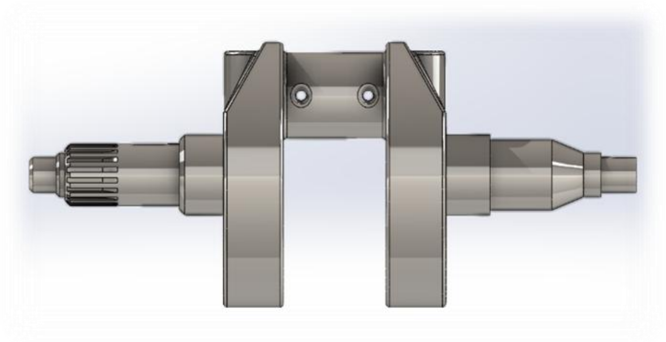
Diameter of the shaft (ds) = 45.5MM

Web thickness (both left and right hand) (t) = 34MM

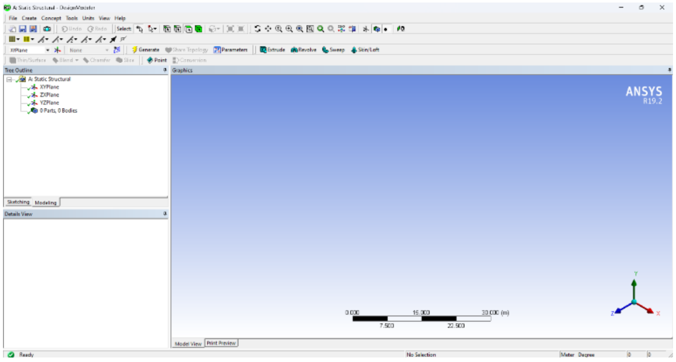
Web width (both left and right hand) (w)= 55.3MM.

VIII. 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



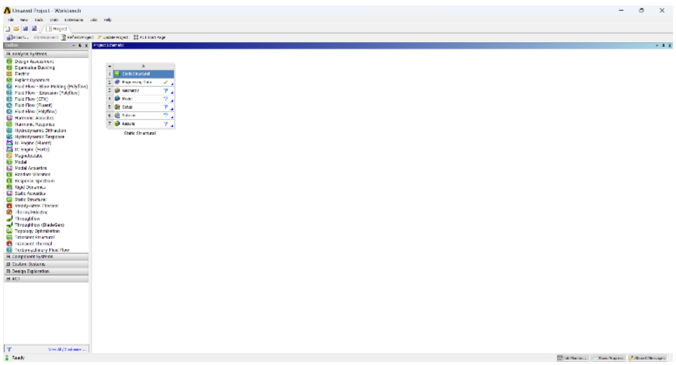
3D Model of designed crankshaft



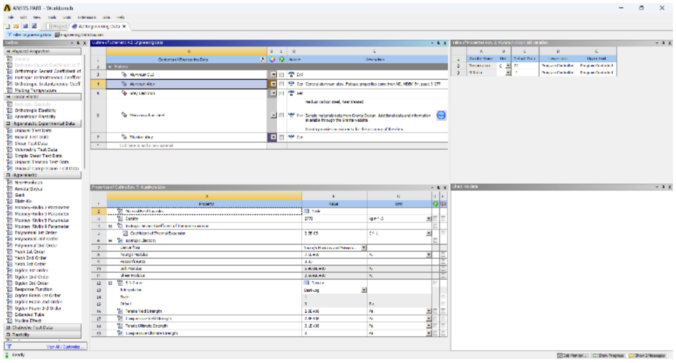
Ansys workpace

IX. FEA ANALYSIS OF CRANKSHAFT:

Finite Element Analysis (FEA) of a crankshaft is a crucial engineering process used to assess its structural integrity, performance, and durability. The crankshaft experiences complex loading conditions such as torsional, bending, and axial forces during engine operation. FEA allows engineers to create a detailed 3D model of the crankshaft, apply material properties, and simulate real-world conditions to predict stress distribution, deformation, and potential failure points. By analyzing these results, engineers can identify weak zones, optimize geometry, and improve material selection to enhance overall performance and reliability.



Ansys workbench



Engineering Materials

IX.I APPENDIX II

MECHANICAL COMPOSITION OF ALUMINIUM D16

Property	Value
Material Type	Aluminium Alloy (D16)
Density	2.77 g/cm³
Tensile Strength	470 MPa
Yield Strength	280 MPa
Elastic Modulus	70 GPa
Hardness (Brinell)	120 HB
Elongation at Break	10-12%
Thermal Conductivity	120-140 W/m·K
Poisson’s Ratio	0.33

MECHANICAL COMPOSITION OF ALUMINIUM ALLOY

Property	Typical Range
Density	2.6 - 2.8 g/cm³
Melting Point	~660°C
Yield Strength	35 - 500 MPa
Tensile Strength	90 - 600 MPa
Elongation (%)	5 - 30%
Hardness (Brinell HB)	15 - 150 HB
Modulus of Elasticity	~69 GPa
Poisson's Ratio	~0.33
Thermal Conductivity	~205 W/m·K

MECHANICAL COMPOSITION OF TITANIUM ALLOY

Density	4.4 - 4.5 g/cm³
Melting Point	~1,668°C
Yield Strength	275 - 1,200 MPa
Tensile Strength	345 - 1,400 MPa
Elongation (%)	10 - 30%
Hardness (Brinell HB)	200 - 400 HB
Modulus of Elasticity	~110 GPa
Poisson's Ratio	~0.32
Thermal Conductivity	~7 - 22 W/m·K

MECHANICAL COMPOSITION OF GREY CAST IRON

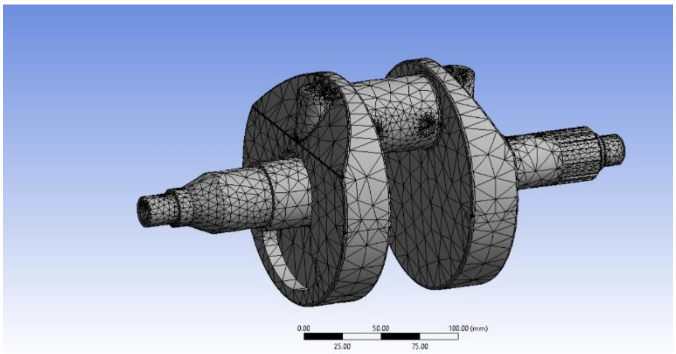
Property	Typical Range
Density	6.8 - 7.5 g/cm³
Melting Point	~1,150°C - 1,200°C
Yield Strength	100 - 350 MPa
Tensile Strength	150 - 400 MPa
Compressive Strength	800 - 1,600 MPa
Elongation (%)	<1% - 3%
Hardness (Brinell HB)	150 - 300 HB
Modulus of Elasticity	~100 - 150 GPa
Poisson's Ratio	~0.26 - 0.30
Thermal Conductivity	~35 - 50 W/m·K
Damping Capacity	Excellent
Wear Resistance	High

MECHANICAL COMPOSITION OF MEDIUM CARBON STEEL

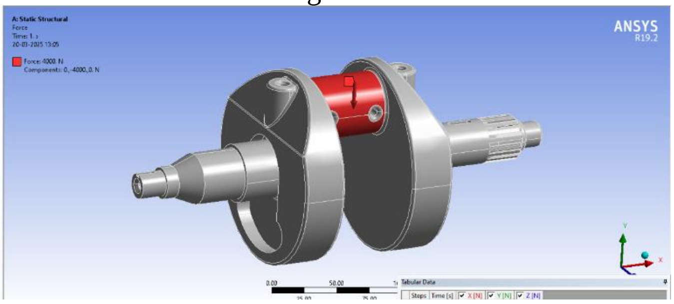
Property	Typical Range
Density	~7.85 g/cm³
Melting Point	~1,370°C - 1,540°C
Yield Strength	290 - 600 MPa
Tensile Strength	500 - 800 MPa
Elongation (%)	10 - 25%
Hardness (Brinell HB)	140 - 300 HB
Modulus of Elasticity	~200 GPa
Poisson's Ratio	~0.29
Thermal Conductivity	~50 W/m·K
Impact Toughness	Moderate
Weldability	Moderate
Machinability	Good

IX.II MESHING AND BOUNDARY CONDITIONS OF CRANKSHAFT

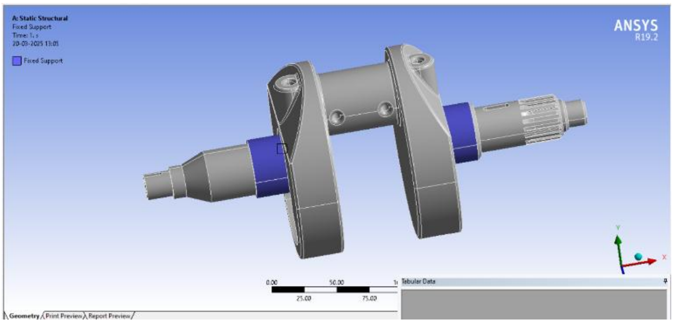
Meshing in ANSYS is the process of dividing a model into smaller elements to perform accurate simulations. These elements can be 1D, 2D, or 3D depending on the geometry and analysis type. A good mesh ensures accurate results by capturing important details like stress concentration and deformation. Smaller elements improve precision but increase computation time, so balancing mesh density is crucial. Key factors like element size, shape quality, and refinement in critical areas help improve performance. Proper meshing ensures that the simulation results are reliable, efficient, and meaningful for engineering analysis.



Meshing of crankshaft



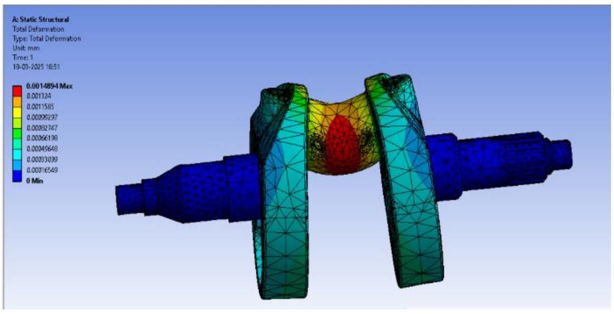
Force acting on crankpin



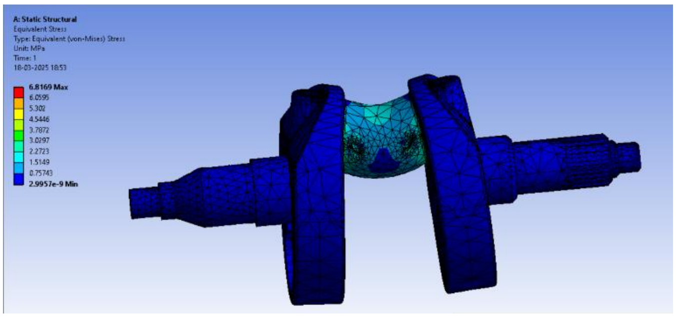
Fixed support on both ends

X. RESULTS AND DISCUSSION

In ANSYS software analysis, the process is conducted using predefined boundary conditions. The resulting output parameters typically include total deformation, Von Mises stress.



(a) Total deformation analysis



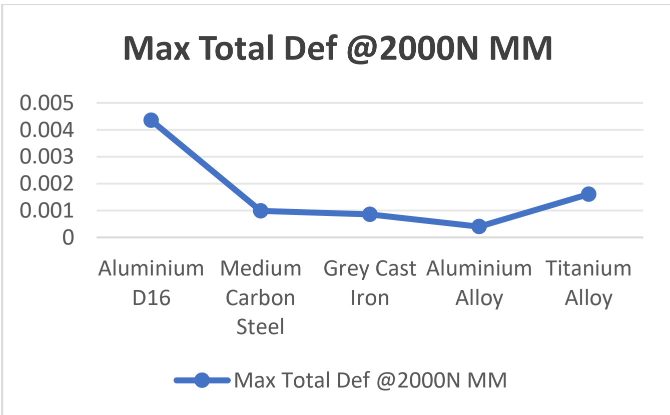
(b) Von Mises Stress analysis

The results obtained can be seen in above figures. It can be noticed from figure (a), that total deformation is 0.0014894 mm and occurs at crank pin location. Similarly, in figure (b) highest von mises stress is observed 6.8169 MPa 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.

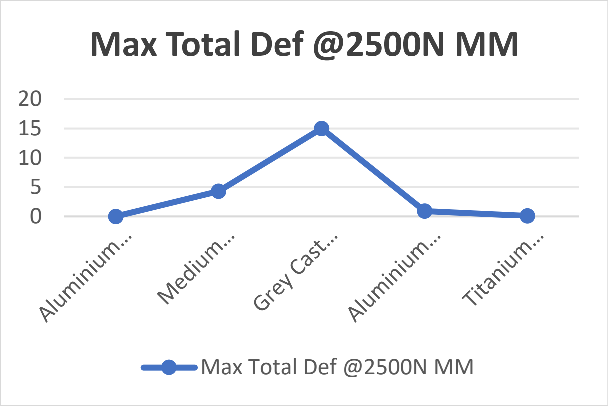
TOTAL DEGFGORMATION:

Material	Max Total Def @2000N MM	Max Total Def @2500N MM	Max Total Def @3000N MM	Max Total Def @3500N MM	Max Total Def @4000N MM
Aluminium D16	0.00435	0.00181	0.0021	0.0025	0.0062
Medium Carbon Steel	0.00099	4.27	6.1784	4.97	9.831
Grey Cast Iron	0.00085	15	4.41	4.97	8.854
Aluminium Alloy	0.0004	0.894	0.342	0.065	0.789
Titanium Alloy	0.0016	0.078	0.016	0.386	0.155

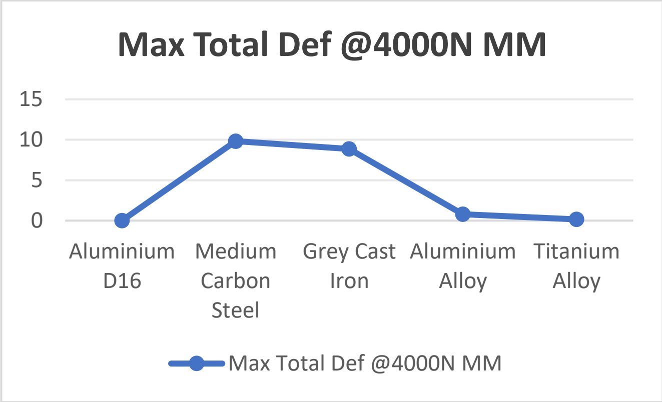
The total deformation obtained from diferrent materials



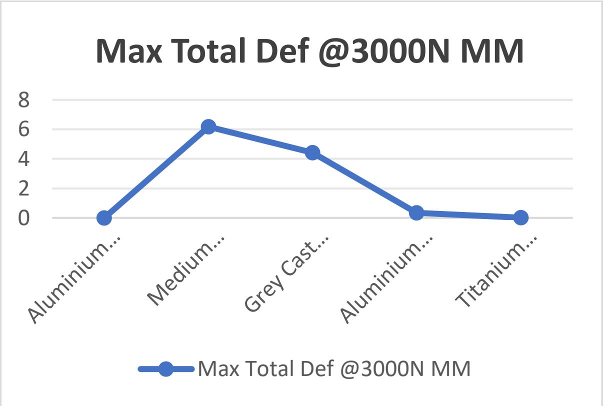
Max Total Def @2000N in MM



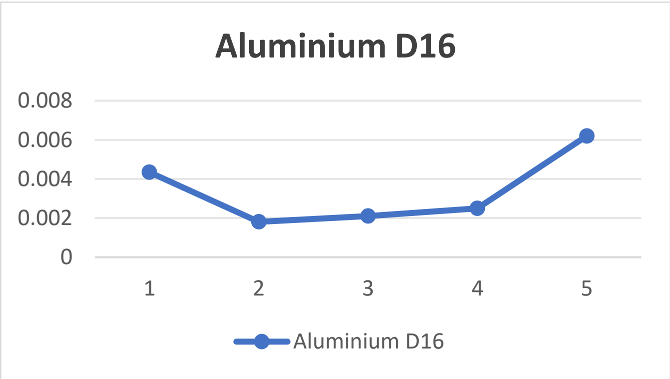
Max Total Def @2500N MM



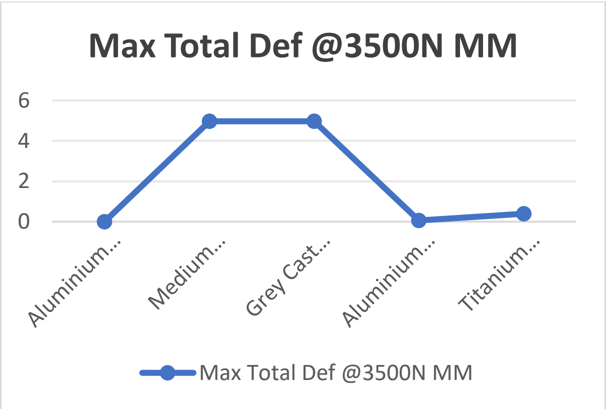
Max Total Def @4000N MM



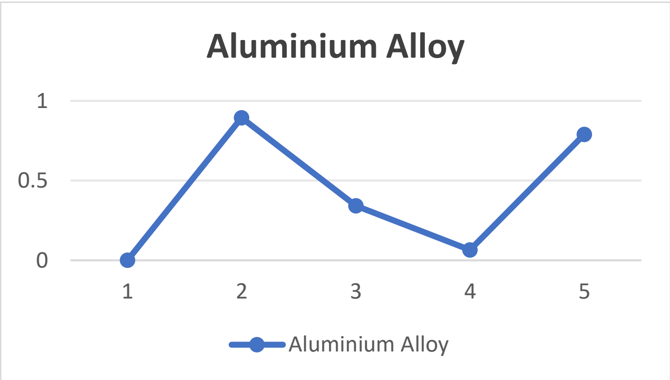
Max Total Def @3000N MM



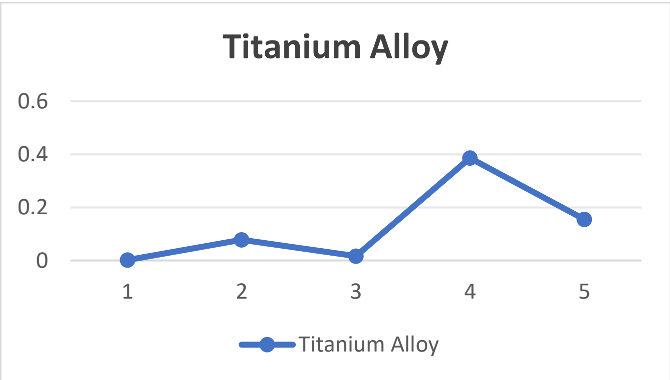
Max Total Def of Aluminium D16 at diff loads



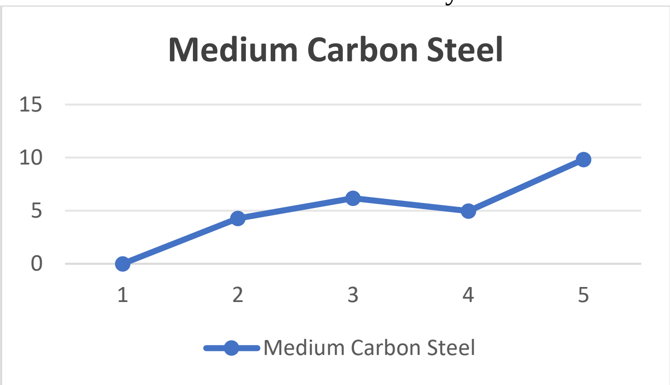
Max Total Def @3500N MM



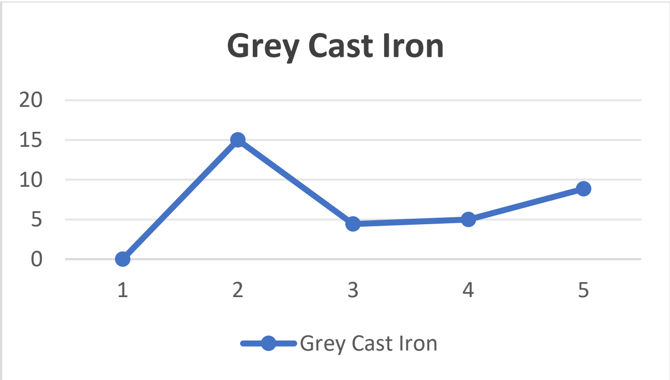
Max Total Def of Aluminium Alloy at diff loads



Max Total Def of Titanium Alloy at diff loads



Max Total Def of Medium Carbon Steel at diff loads

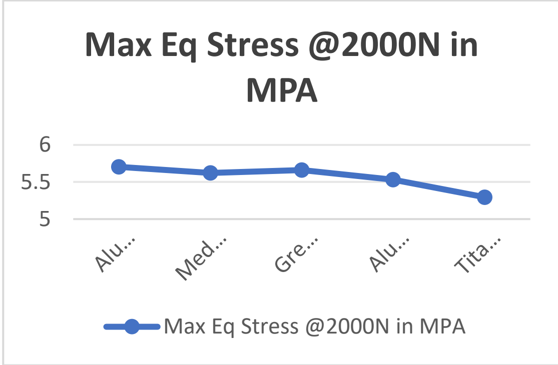


Max Total Def of Grey Cast Iron at diff loads

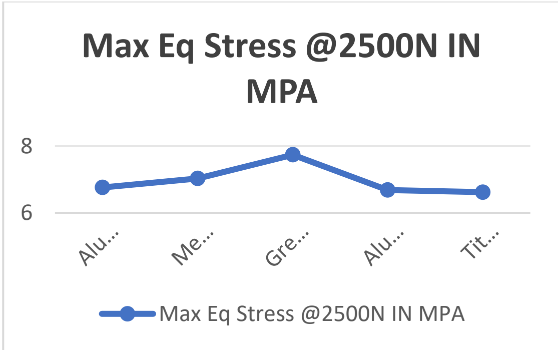
EQUIVALENT STRESS (VON MISES):

Material	Max Eq Stress @2000N in MPA	Max Eq Stress @2500N IN MPA	Max Eq Stress @3000N MPA	Max Eq Stress @3500N MPA	Max Eq Stress @4000N MPA
Aluminium D16	5.7041	6.7556	8.1067	9.4578	10.809
Medium Carbon Steel	5.6226	7.0282	8.4338	9.8375	11.245
Grey Cast Iron	5.6591	7.739	8.9886	9.903	11.318
Aluminium Alloy	5.5335	6.6819	8.1803	9.5437	10.907
Titanium Alloy	5.2961	6.6202	7.9442	9.2683	10.592

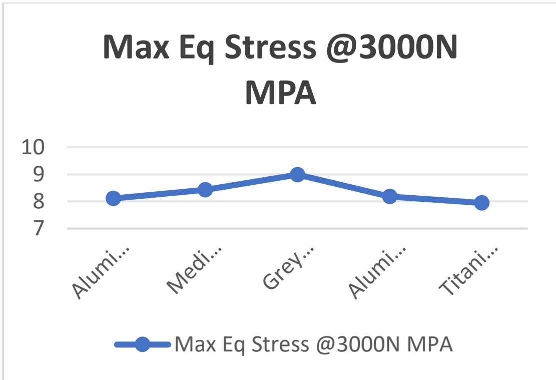
The Equivalent von mises stresses obtained from diferrent materials



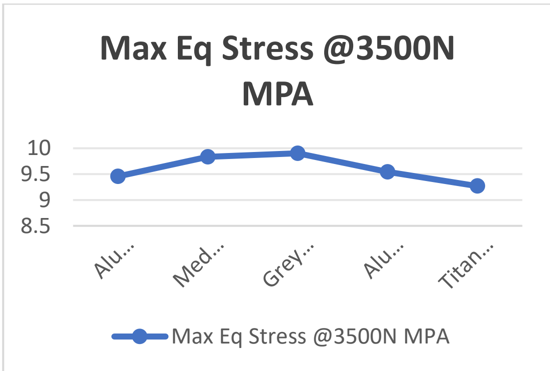
Max Eq Stress @2000N in MPA



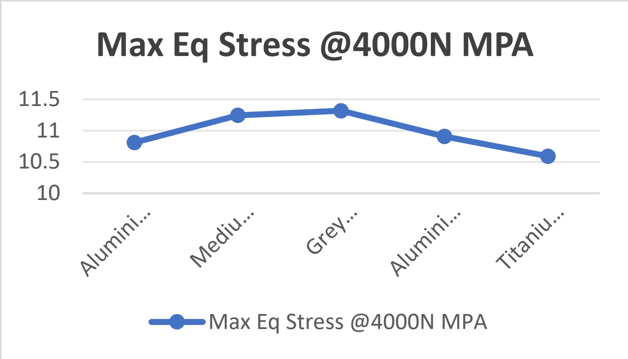
Max Eq Stress @2500N IN MPA



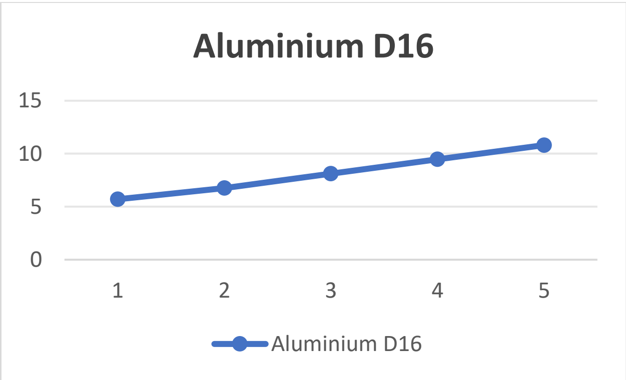
Max Eq Stress @3000N MPA



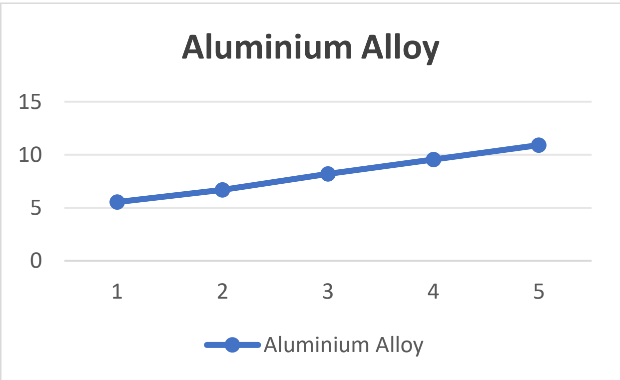
Max Eq Stress @3500N MPA



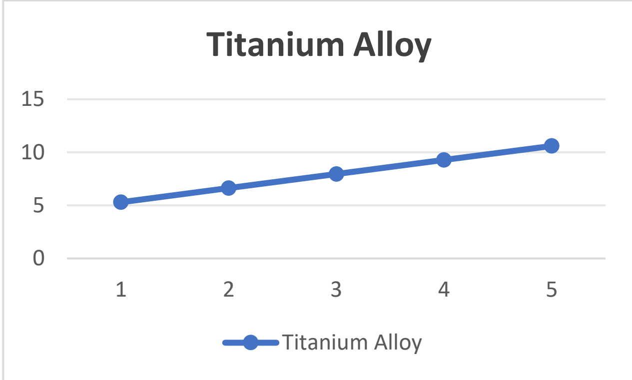
Max Eq Stress @4000N MPA



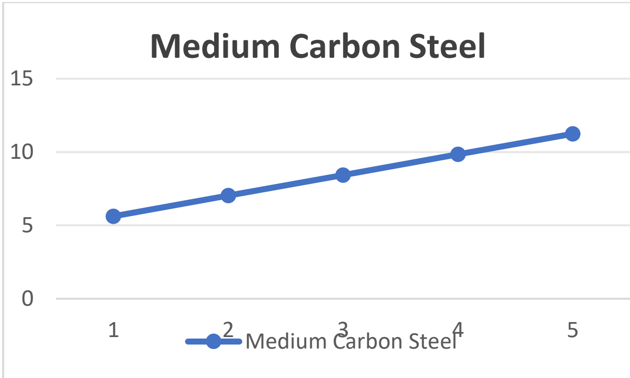
Max Equivalent von stress of Aluminium D16 at Diff loads



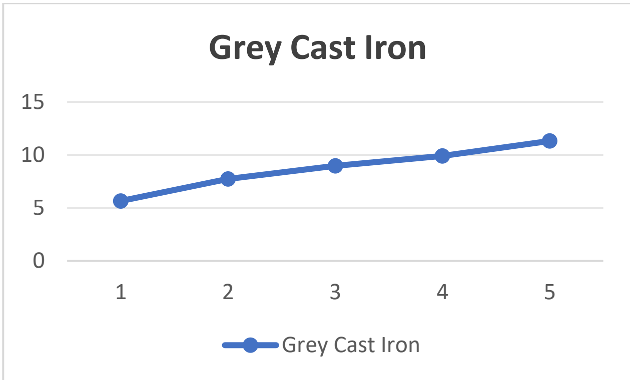
Max Equivalent von stress of Aluminium Alloy at Diff loads



Max Equivalent von stress of Titanium Alloy at Diff loads



Max Equivalent von stress Medium carbon steel at Diff loads



Max Equivalent von stress Grey Cast Iron at Diff loads

XI. CONCLUSION:

This research outlines a numerical method for determining the optimal crankshaft design. The initial stage involves performing manual calculations to establish the design parameters. Developing a wellstructured crankshaft design is crucial to minimize stress concentration and prevent potential failures.

In this study, the crankshaft model was created using Creo software and subsequently imported into ANSYS software in STEP format. The crankshaft was analyzed under the influence of inertia load and torque, necessitating a thorough evaluation of stress distribution by incorporating appropriate constraints and loads in the analysis.

The results obtained from the Finite Element Analysis (FEA) provide essential insights for determining whether the crankshaft design is structurally sound. These observations play a key role in ensuring the design's safety and reliability.

- Maximum total deformation occurs at center of the crank pin.
- Von Mises stress under the combustion gas load are within the yield strength limit and ultimate strength limit of the material respectively.
- Crankshaft with above mentioned materias is safe for 75 bar peak cylinder pressure operation without any failure.

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