DESIGN & STATIC STRUCTURAL ANALYSIS OF CRANKSHAFT FOR HIGH PRESSURE PLUNGER PUMP

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Abstract- in this research crankshaft is an important part of pump assembly which transfers energy from prime mover to connecting rod and convert the rotary motion of a crankshaft into reciprocating motion of a plunger rod through the connecting rod. Crankshaft is one of the most critically loaded component in the pump as it experiences cyclic loads in the form of bending and torsion during its working condition. Its failure will cause serious damage to the pump. So, its reliability verification must be performed. In this paper a dynamic simulation is conducted on a crankshaft from a single plunger rod plunger pump. In this work crankshaft of EN24 material is used for high pressure plunger pump. A three-dimension model of crankshaft is created using SOLID EDGE ST6 software. The main objective of this research is to study the static structural analysis of crankshaft. Finite element analysis (FEA) is performed to obtain the variation of stress magnitude at critical locations of crankshaft. The dynamic analysis is done using FEA Software ANSYS14.5 which resulted in the load spectrum applied to crank pin bearing. This load is applied to the FE model in ANSYS, and boundary conditions are applied according to the mounting conditions. The analysis is done for finding critical location in crankshaft. Stress variation over the cycle and the effect of torsion and bending load in the analysis are investigated. Von-mises stress is calculated using theoretically and FEA software ANSYS.

Keywords- Crankshaft, Ansys 14.5, Solid edge ST6, FEA, EN24, Static structural analysis.

I. INTRODUCTION

It is a large component with a complex geometry in the pump, which converts rotary motion to a reciprocating displacement of the plunger rod with a four link mechanism. Crankshaft consists of the shaft parts which revolve in the main bearings, the crankpins to which the big ends of the connecting rod are connected, the crank arms or webs (also called cheeks) which connect the crankpins and the shaft parts. The crankshaft main journals rotate in a set of supporting bearings (main bearings) causing the offset road journals to rotate in circular path around the main journal centres, the diameter of that path is the engine “stroke’.

Crankshaft experiences large forces from fluid pressure. This force is applied to the top of the plunger rod and since the connecting rod connects the plunger rod to the crank shaft, the force will be transmitted to the crankshaft. The magnitude of the forces depends on many factors which consist of crank radius, connecting rod dimensions and weight of the connecting rod, plunger rod, crank slider, and pin. Pressure and inertia forces acting on the crankshaft such as torsional load & bending load. Crankshaft must be strong enough to take the downward force of the compression stroke without excessive bending so the reliability and life of the plunger pump depend on the strength of the crankshaft largely. The crank pin is like a built in beam with a distributed load along its length that varies with crank positions. Each web is like a cantilever beam subjected to bending and twisting.

1. Bending moment which causes tensile and compressive stresses.

2. Twisting moment causes shear stress.
A crankshaft should have sufficient strength to withstand the bending and twisting moments to which it is subjected. In addition, it should have sufficient rigidity to keep the lateral and angular deflections within permissible limits. The crankshaft is subjected to fluctuating stresses and, as such, it should have sufficient endurance limit stress. There are many sources of failure in the engine one of the most common crankshaft failure is fatigue at the fillet areas due to the bending load. The moment of pressure force from plunger rod is transmitted to the crankpin, causing a large bending moment on the entire geometry of the crankshaft. At the root of the fillet areas stress concentrations exist and these high stress range locations are the points where cyclic loads could cause fatigue crank initiation leading to fracture.

In industrial engine, the crankshafts are commonly made from carbon steel such as 40C8, 55C8 and 60C4. In transport engines, manganese steel such as 20Mn2, 27Mn2 and 37Mn2 are generally used for the making of crankshaft. In aero engines, nickel chromium steel such as 35Ni1Cr60 and 40Ni2Cr1Mo28 are extensively used for the crankshaft. The crankshafts are made by drop forging or casting process but the former method is more common. The surface of the crankpin is hardened by case carburizing, nit riding or induction hardening.

It is important for engineers to properly determine the stresses present within the crankshaft to ensure that work even under its worst conditions. By being able to model the crankshaft and determining the forces present on it under all conditions it is possible to use programs such as ANSYS to determine the stresses developed within the crankshaft.

The material used for connecting rod has following properties,

Table 1: Material properties of crankshaft

<table>
<thead>
<tr>
<th>Material selected</th>
<th>EN24</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young's Modulus</td>
<td>210 GPa</td>
</tr>
<tr>
<td>Poisson’s Ratio</td>
<td>0.280</td>
</tr>
<tr>
<td>Tensile Yield Strength</td>
<td>1100 N/mm²</td>
</tr>
<tr>
<td>Density</td>
<td>7800 Kg/m³</td>
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<tr>
<td>Yield stress</td>
<td>785</td>
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</table>

II. LITERATURE REVIEW

Jaimin Brahmbhatt, Abhishek choubey [1] - In this paper a dynamic simulation is conducted on a crankshaft from a single cylinder 4-stroke diesel engine. The dynamic analysis is done using FEA Software ANSYS which resulted in the load spectrum applied to crank pin bearing. The analysis is done for finding critical location in crankshaft. Stress variation over the engine cycle and the effect of torsion and bending load in the analysis are investigated. The relationship between the frequency and the vibration modal is explained by the modal and harmonic analysis of crankshaft using FEA software ANSYS. The maximum deformation appears at the centre of crankpin neck surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks and near the central point Journal. The edge of main journal is high stress area.

K. Thriveni, Dr. B. Jaya Chandraiah [2] – studied the Static analysis on a crankshaft from a single cylinder 4-stroke LC Engine. Finite element analysis (FEA) is performed to obtain the variation of stress at critical locations of the crank shaft using the ANSYS software and applying the boundary conditions. Then the results are drawn Von-misses stress induced in the crankshaft is 15.83MPa and shear stress is induced in the crankshaft is 8.271MPa. The Theoretical von-misses stress is 19.6MPa, shear stress is 9.28MPa. The validation of model is compared with the Theoretical and FEA results of Von-misses stress and shear stress are within the limits.

R. J. Deshbhratar, and Y.R Suple, [3] have been analysed 4-cylinder crankshaft and model of the crankshaft were created by Pro/E Software and then imported to ANSYS software. The maximum deformation appears at the centre of crankshaft surface. The maximum stress appears at the fillets between the crankshaft journal and crank cheeks, and near the central point. The edge of main journal is high stress area. The crankshaft deformation was mainly bending deformation under the lower
frequency. And the maximum deformation was located at the link between main bearing journal and crankpin and crank cheeks. So this area prone to appear the bending fatigue crack.

Rinkle garg and Sunil Baghl, [4] have been analysed crankshaft model and crank throw were created by Pro/E Software and then imported to ANSYS software. The result shows that the improvement in the strength of the crankshaft as the maximum limits of stress, total deformation, and the strain is reduced. The weight of the crankshaft is reduced . There by, reduces the inertia force. As the weight of the crankshaft is decreased this will decrease the cost of the crankshaft and increase the I.C engine performance.

Sri K. Prasad, A.V.S.S.Somasundar, [5] have been analysed a cast Iron crankshaft of a single cylinder 4 –stroke diesel engine. A static analysis was conducted to get variation of stress magnitude at critical locations of the crankshaft. A model was created in CATIA of crank shaft and imported into ANSYS to carryout static analysis. Meshing of crankshaft was done; loads and boundary conditions were applied as per the mounting conditions of the crankshaft on Finite element model of crankshaft. Results obtained from the analysis were then used in optimization of the cast Iron crankshaft. Weight Optimization is achieved by varying the crankpin diameter. That requires the stress range in FE analysis not to exceed the magnitude of the stress range in the original crankshaft. The optimization process involves geometry changing without changing engine block.

Ramanagouda Biradar, Dr. R A Savanur, [6] have been analysed static stress analysis on crankshaft of a single-cylinder 4 stroke diesel engine using finite element method. ANSYS WB14 is used for the analysis using 4 different materials to find the distribution of maximum deformation, maximum shear stress and von-mises stress on the crankshaft. By comparing all the determinants like von mises stress, shear stress, deformation and mass on all type material crankshafts, Gray cast iron crankshaft is best suitable from the study. All the determinants are within safe limit and mass of the Gray cast iron crankshaft is about 9.0277 kg which is lesser amongst different material crankshafts. Gray cast iron crankshaft will impose less inertia effect.

2.1 Objective

The main objective of this research is to study the static structural analysis of plunger pump crankshaft. Theoretical design of Crankshaft is carried out. Modelling of the connecting rod is done using Solid edge ST6 and finite element analysis is done in Ansys 14.5 software to analyse the stresses acting on the crankpin due to fluid force. Analyse the maximum deformation, maximum stress point and dangerous areas of failure.

III. DESIGN OF CRANKSHAFT

The different parameters of the pump and crankshaft are given below.

| Table 2: Plunger pump specification |
| No. of Plunger rod | 1 |
| Speed of Plunger pump | 83 rpm |
| Maximum Pressure | 350 bar |
| Design Pressure | 415 bar |
| L/R ratio of connecting rod | 10 |
| Plunger rod diameter | 20 mm |
| Stroke | 45 mm |
| Factor of safety | 3 |

A crankshaft is subjected to bending and torsional moments due to the following three forces:

I. Force exerted by the connecting rod on the crank pin

II. Weight of counterweight (W) acting downward in the vertical direction

III. Resultant belt tensions acting in the horizontal direction (P₁ + P₂)
In the design of the centre crankshaft, two cases of crank positions are considered. They are as follows:
Case I: The crank is at the top dead centre position and subjected to maximum bending moment and no torsional moment.
Case II: The crank is at an angle with the line of dead centre positions and subjected to maximum torsional moment.

We will consider these cases separately to determine the dimensions of the crankshaft.

Most crankshaft failures are caused by a progressive fracture due to repeated bending or reversed torsional stresses. Thus the crankshaft is under fatigue loading and, therefore, its design should be based upon endurance limit. Neither since the failure of a crankshaft is likely to cause serious pump destruction and neither all the forces nor all the stresses acting on the crankshaft can be determined accurately; therefore a high factor of safety from 3 to 4, based on the endurance limit, is used.

3.1 Design of Centre Crankshaft

Case I: When the crank is at dead centre (crankshaft subjected to pure bending)

At this position of the crank, the maximum fluid pressure on the piston will transmit maximum force on the crankpin in the plane of the crank causing only bending of the shaft the crankpin as well as ends of the crankshaft will be only subjected to bending moment. Thus, when the crank is at the dead centre, the bending moment on the shaft is maximum and the twisting moment is zero.

1) The reactions at the bearings 1 and 2 due to force on the crank pin are denoted by \( R_1 \) and \( R_2 \). The vertical component is given by \( R_v \) and horizontal reaction is denoted by \( R_h \).

2) The reaction at the bearing 2 due to weight of the counterweight and sum of the belt tensions \( (P_1 + P_2) \) are denoted by \( R'_2 \) followed by suffix letters \( v \) and \( h \) such as \( (R'_2)_v \) or \( (R'_2)_h \).

F\(_p\) = Force acting on crank pin

D = diameter of piston

\( P_{\text{max}} \) = maximum fluid pressure inside the cylinder

W = weight of counterweight (N)

\( P_1 + P_2 \) = tension in belt drive

\( b \) = distance between main bearing 1 and 2

\( c \) = distance between 2\(^{nd}\) bearing and counterweight.

\( F_p = \) Pressure × Area = \( P \times (\pi/4) \times (d^2) \)

\( = 41.5 \times 0.7854 \times 20^2 = 13037.63 \text{ N} \)

At the top dead centre position, the thrust in the connecting rod will be equal to the force acting on plunger rod. It is assumed that the portion of the crankshaft between bearing 1 and 2 is simply supported on bearings and subjected to force \( F_p \).

Taking moment of forces, \( F_p \times b_1 = (R_2)_h \times b \) & \( F_p \times b_2 = (R_1)_h \times b \)

\( 13067.63 \times 90 = (R_2)_h \times 180 \) & \( 13067.63 \times 90 = (R_1)_h \times 180 \)

\( \therefore (R_2)_h = (R_1)_h = 6518.815 \text{ N} \)

\( \therefore \) Bending moment at the central plane is given by,

\( (M_b)_c = (R_1)_h \times b_1 \)

But we also know that, \( M_b = (\pi/32) \times (d_c)^3 \times \sigma_b \)

\( = 586693.35 = (\pi/32) \times (d_c)^3 \times (98.1/3) \)

\( \therefore \) Diameter of the crankpin is \( d_c = 56.7485 \text{ mm} \approx 58 \text{ mm} \)

Case II: Centre crankshaft at angle of maximum torque (crankshaft subjected to pure twisting)

The position of the crank when it makes an angle (\( \Theta \)) with the line of dead centres. The torque is maximum when the tangential component of force on the crank pin is maximum. It may be noted that the tangential force will cause twisting of the crankpin and shaft while the radial force will cause bending of the shaft.

The force acting on the connecting rod \( F_Q \) may be resolved into two components, one perpendicular to the crank and the other along the crank. The component of \( F_Q \) perpendicular to the
crank is known as crank-pin effort and it is denoted by $F_T$. The component of $F_Q$ along the crank produces a thrust on the crank shaft bearings and it is denoted by $F_B$.

Resolving $F_Q$ perpendicular to the crank,

$$F_T = F_Q \times \sin (\theta + \phi) = \left(\frac{F_p}{\cos \phi}\right) \times \sin (\theta + \phi) = 13102.63 \text{ N}$$

And resolving $F_Q$ along the crank,

$$F_R = F_Q \times \cos (\theta + \phi) = \left(\frac{F_p}{\cos \phi}\right) \times \cos (\theta + \phi) = 1310.287 \text{ N}$$

It is assumed that the portion of the crankshaft between bearing 1 and 2 is simply supported on bearing and subjectend to tangential force $F_T$ and radial force $F_B$ at the crankpin. Due to the tangential component $F_T$, there are reaction $H_{T1}$ and $H_{T2}$ at bearing 1 and 2 respectively. Similarly, due to the radial component $F_R$, there are reactions $H_{R1}$ and $H_{R2}$ at bearing 1 and 2 respectively.

Taking moment of forces about bearing 1,

$$F_T \times b_1 = H_{T2} \times b\quad \text{&}\quad H_{R2} \times b = F_R \times b_1$$

$$13102.63 \times 90 = H_{T2} \times 180 \quad \text{&} \quad 1310.287 \times 90 = H_{R2} \times 90$$

$$H_{T2} = 6551.315 \text{ N} \quad \text{&} \quad H_{R2} = 655.1435 \text{ N}$$

Taking moment of forces about bearing,

$$F_T \times b_2 = H_{T1} \times b \quad \text{&} \quad H_{R1} \times b = F_R \times b_2$$

$$H_{T1} = 6551.315 \text{ N} \quad \text{&} \quad H_{R1} = 655.1435 \text{ N}$$

The central plane of the crank pin is subjected to bending moment $M_b$ due to $(R_1)b$ and torsional moment $M_t$ due to $(R_1)b$

$$M_b = H_{R1} \times b_1 = 655.1435 \times 90 = 58962.915 \text{ N}$$

$$M_t = H_{T1} \times r = 6551.315 \times 22.5 = 147404.58 \text{ N}$$

The diameter of the crank pin ($d_c$) is calculated by following equation:

$$(d_c)^3 = \left(\frac{16}{\pi \tau}\right) \times \left(\frac{M_b^2 + (M_t)^2}{d^4}\right)^{1/2}$$

$\tau = $ allowable shearing stress & F.O.S = 3

$$= \left(16 \times 3 \times 158759.9935\right) / (\pi \times 6.4 \times 9.81)$$

$$= 38635.2259$$

$$d_c = 33.8060 \text{ mm} \approx 34 \text{ mm}$$

By comparing case I and case II crank pin diameter, we will be choose maximum diameter therefore we will follow when the crank is at dead centre (crankshaft subjected to pure bending) i.e. case I.

### 3.2 Design of shaft under load

To find the diameter of the shaft adjacent to the crank web and which is to be mounted in crankshaft bearing. Whenever shaft is subjected fluctuating load, shaft experiences bending moment and twisting moment. To find diameter of shaft for pure bending, pure twisting and combine effect of bending and twisting moment procedure discussed below.

**Case I: Subjected to pure bending moment**

Consider maximum gas force is applied on shaft; bending moment is produced. This bending moment is given by,

$$M_b = (R_1)b \times b_1 = 6518.815 \times 90 = 586693.35 \text{ Nmm}$$

$K_b =$ combined shock and fatigue factor applied to bending moment = 2 & F.O.S = 3

$$K_b \times M = \left(\frac{\tau}{32}\right) \times (\sigma/fos) \times d^4 \quad \text{Where} \quad \sigma = \text{allowable bending stress} = 98.1 \text{ N/mm}^2 \text{ for E24}$$

$$d = 56.75 \text{ mm}$$

**Case II: Subjected to pure twisting**

When maximum tangential force is applied on the shaft, twisting moment is produced. This twisting moment is given by,

$$T = F_T \times e = 13102.63 \times 22.5 = 294809.175 \text{ Nmm}$$

Where $e =$ eccentricity = half of the stroke length = 22.5 mm

$K_t =$ combined shock and fatigue factor applied to torsional moment = 1.5 & F.O.S = 3
\[K_t \times T = \left(\frac{\pi}{16}\right) \times \left(\frac{\tau}{f_o s}\right) \times d^3 \] …………Where \(\tau\) = allowable shearing stress = 62.784 N/mm\(^2\) for E24\n\[d = 41.55\ mm\]

We will try to find diameter based on combined effect due to bending & torsional moment.

Case III: Diameter based on combined effect

First we will find equivalent twisting moment,
\[\therefore \text{Equivalent twisting moment } T_e = \left[\left(K_b \times M\right)^2 + \left(K_t \times T\right)^2\right]^{1/2}\]
\[= 626974.75\ \text{Nmm}\]

We know that \[T_e = \left(\frac{\pi}{16}\right) \times \left(\frac{\tau}{f_o s}\right) \times d^3\]
\[\therefore d = 53.44\ mm\]

Now we will find out equivalent bending moment,
\[\therefore M_e = \left[\left(K_b \times M\right)^2 + \left(\frac{3}{4}\right) \left(K_t \times T\right)^2\right]^{1/2}\]
\[= 1213668.102\ \text{Nmm}\]

We know that \[M_e = \left(\frac{\pi}{32}\right) \times \left(\frac{\sigma}{f_o s}\right) \times d^3\]
\[d = 73.10\ mm\]

For \(d = 74\ mm\) we will check whether stresses in shaft are within limiting stress for defined material.

Design of crank pin against fatigue loading, According to distortion energy theory,
The Von Mises stress induced in the crankpin is,
\[\sigma_v = \left(\frac{M_e Y}{I}\right)\] …………for \(d = 74\ mm\)

And shear stress;
\[\tau = 8.252296243\ \text{N/mm}^2\] …………for \(d = 74\ mm\)

As a result, stresses induced in crankpin are within the endurance limit of material.

3.3 Design of left hand crank web

First we will find the thickness of the crank web, generally thickness calculated by the empirical relationship as,
\[\therefore t = 0.4\ d_s \text{ to } 0.6\ d_s\]
We are taking \(t = 0.6\ d_s\)
\[= 0.6 \times 58 = 34.8\ mm \approx 35\ mm\]

In our case we are taking crank web thickness as 38 mm

Width of the crank web can be found out by using following empirical relation,
\[w = 1.123\ d_s + 12.7\ mm\]
\[w = 1.125 \times 58 + 12.7 = 77.95\ mm \approx 80\ mm\]

In our case we are taking crank web width as 140 mm

The left hand crank web is subjected to eccentric load \((R_1)\). There are two types of stresses in the central plane of the crank web, viz., direct compressive stress and bending stress due to eccentricity of reaction \((R_1)\).

The direct compressive stress is given by,
\[\sigma_c = \frac{[(R_1)h/(wt)]}{6518.815/(140 \times 38)} = 1.22534\ \text{N/mm}^2\]

The bending moment at the central plane is given by,
\[M_b = (R_1)h \left[\left(b_1-(l/2)-(t/2)\right)\right] \]
\[= 6518.815 \times [90 - (56/2)-(38/2)] = 443279.42\ \text{Nmm}\]
\[\therefore \text{Bending stress is given by, } \sigma_b = \frac{M \times Y}{I} \quad \text{where } y = D/2 \text{ and } I = \frac{\pi D^4}{32}, D = 80\ mm\]
\[\therefore \sigma_b = 8.8187\ \text{N/mm}^2\]

\[\therefore \text{Total stress acting on crank web is equal to, } \sigma = \sigma_c + \sigma_b\]
$\sigma = 10.0440 \text{ N/mm}^2$

Since $\sigma < \sigma_a$, here $\sigma_a = \text{allowable bending stress of material}$, $\sigma_a = 98.1 \text{ N/mm}^2$

### 3.4 Design of right hand crank web

The dimensions of the right hand crank web (i.e. thickness and width) are made equal to left hand crank web from the balancing point of view.

### 3.5 Design of shaft under counterweight

The forces acting on the shaft under the counterweight are shown in fig. The central plane of the shaft is subjected to maximum bending moment.

Let $d_s = \text{Diameter of shaft in mm}$.

We know that bending moment due to the weight of counterweight,

$M_w = W \times c = 40 \times 160 = 6400 \text{ Nmm}$

And bending moment due to belt tension,

$M_t = (P_1+P_2) \times c = 3544 \times 160 = 567040 \text{ Nmm}$

These two bending moments act at right angles to each other. Therefore, the resultant bending moment at the counterweight location,

$M_b = \left[(M_w)^2 + (M_t)^2\right]^{1/2} = \left(\frac{\pi}{32}\right) \times \left(\frac{\sigma}{f_{os}}\right) \times d^3$

$567076.1162 = \left(\frac{\pi}{32}\right) \times \left(\frac{98.1}{3}\right) \times d^3$

$d = 56.1088 \text{ mm}$

This the minimum diameter required for shaft under counterweight.

### IV. MODELING AND STATIC STRUCTURAL ANALYSIS OF CRANKSHAFT

Crankshaft for material modelled by taking the designed parameter and then by using the Solid Edge ST6 software solid modelling has done which is shown in Figure (1) And saved within this program in *.IGES format. The model is imported in Ansys and then the mechanical characteristics of the connecting rod are applied as shown below.

![Figure 1: Isometric 3D View of crankshaft](image1)

![Figure 2: Dimensions of crankshaft](image2)

As shown in the figure 1 the crankshaft is modelled using Solid Edge ST6 Software with the dimensions shown in figure 2 for Steel EN24.

### 4.1 Finite Element Analysis

The basis of FEA relies on the decomposition of the domain into a finite number of sub-domains (elements) for which the systematic approximate solution is constructed by applying the variation or weighted residual methods. In effect, FEA reduces problem to that of a finite number of unknowns by dividing the domain into elements and by expressing the unknown field variable in terms of the assumed approximating functions within each element. These functions (also called interpolation functions) are defined in terms of the values of the field variables at specific points, referred to as nodes. The finite element method is a numerical procedure that can be used to obtain solutions to a large class of engineering problems involving stress analysis, heat transfer, electromagnetism, and fluid flow.
4.2 Meshing of crankshaft

Significant factor that will affect the possibilities to obtain acceptable results from the analysis is how the mesh is defined. A finer mesh will generate more accurate results, at the price of longer calculation time. Hexagonal meshing is done. The numbers of elements are 88143 and the numbers of nodes are 300989. It is necessary to mesh manually in subsequent simulations where the model is more detailed and the geometry is more complex. For this first analysis, element size is found out to be 4 mm for working in convergence zone. Figure 3 shown below is meshed model of crankshaft.

![Meshed model of crankshaft](image)

Figure 3: Meshed model of crankshaft

4.3 Loading and boundary conditions

Boundary conditions play an important role in finite element calculation and also relate to the finite elements mesh generating, some assumptions are set up when considering the boundary conditions:

1) The torque is ignored. Only bending moment, which affects the crankshaft mostly, is included in the calculation and analysis.
2) The mass force is included in the program itself. Therefore the force boundary condition excluded the mass force.

Crankshaft is a constraint with a ball bearing from one side and with a journal on the other side. The ball bearing is press fit to the crankshaft and does not allow the crankshaft to have any motion other than rotation about its main axis. Since only 180 degrees of the bearing surfaces facing the load direction constraint the motion of the crankshaft, this constraint is defined as a fixed semi-circular surface as wide as ball bearing width. Since the crankshaft is in interaction with the connecting rod, the same loading distribution will be transmitted to the crankshaft. In this study a force of 13037.63N is applied at the crankpin at the position of maximum bending moment or is at the dead centre.

![Loading conditions for crankshaft](image)

Figure 4: Loading conditions for crankshaft
4.4 Static structural analysis of crankshaft

![Figure 5: Equivalent von mises stress in crankshaft](image1)

![Figure 6: Shear stresses in crankshaft](image2)

![Figure 7: Total deformation in crankshaft](image3)

Validated result:

<table>
<thead>
<tr>
<th>Sr.No.</th>
<th>Parameter</th>
<th>Theoretical</th>
<th>FEA Analysis</th>
</tr>
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<td>1</td>
<td>Von mises stress (MPa)</td>
<td>17.61128</td>
<td>16.739</td>
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<tr>
<td>2</td>
<td>Shear stress (MPa)</td>
<td>8.2522</td>
<td>6.7492</td>
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V. CONCLUSIONS

By checking and comparing the results of material in above tables and finalizing the results are shown in below.

1) Table 3 results shows that FEA Results conformal matches with the theoretical calculation so we can say that FEA is a good tool to reduce time consuming theoretical work.

2) The maximum deformation appears at the centre of the crankpin neck surface.

3) The maximum stress appears at lubricating hole circular area, transition areas between the crankshaft journal and crank web and the area between crankpin and crank web.

4) The value of von-misses stresses that comes out from the analysis is far less than material yield stress so our design is safe.

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I sincerely appreciate Prof. Sachin S. Mestry for accepting me as his student and for giving me the opportunity to work on this research. I am also grateful for his support and guidance that have helped me expand my horizons of thought and expression.
REFERENCES


