Design and Finite Element Analysis of Crank Shaft by using Catia and Anysys

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Abstract

Crankshaft is one of the critical components for the effective and precise working of the internal combustion engine. This converts the reciprocating displacement of the piston in to a rotary motion of the crank. An attempt is made in this Project is to study the Static analysis on a crankshaft from a 4 cylinder I.C Engine. A three-dimension model of IC engine crankshaft is created using CATIA V5 software. Finite element analysis (FEA) is performed to obtain the variation of stress magnitude at critical locations of crankshaft. Simulation inputs are taken from the engine specification chart. The Static analysis is done using FEA Software ANSYS 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 engine mounting conditions. 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. Von-mises stress Shear Stresses are calculated using theoretically and FEA software ANSYS.

Keywords — *CATIA V5, FEA, ANSYS, CRANK SHAFT*

I. INTRODUCTION

A. Crankshafts

Crankshafts are common machine elements which transfer rotational movement into linear. Crankshaft design in modern internal combustion engines is driven by the desire for more power at higher efficiency rates and reduced weight. The demands on crankshaft material, therefore, are increasing, while the crankshafts themselves become smaller. The many different designs of crankshaft vary considerably, and even during mass production there can be subtle differences from one to another.

To machine such a variety of cranks, Sandvik Coromant has developed tool systems which are based on well designed and production tested components and inserts.

Fig 1: Four cylinder crankshaft - Flange, Stub end and Main bearings are machined

A crankshaft is used to convert reciprocating motion of the piston into rotary motion or vice versa. The crankshaft consists of the shaft parts, which revolve in the main bearings, the crank pins to which the big ends of the connecting rod are connected, the crank arms or webs, which connect the crankpins, and the shaft parts. The crankshaft, depending upon the position of crank, may be divided into the following two types.

The crankshaft is the principal member of the crank train or crank assembly, which latter converts the reciprocating motion of the pistons into rotary motion. It is subjected to both torsional and bending stresses, and in modern high-speed, multi-cylinder engines these stresses may be greatly increased by resonance, which not only renders the engine noisy, but also may fracture the shaft. In addition, the crankshaft has both supporting bearings (or main bearings) and crankpin bearings, and all of its bearing surfaces must be sufficiently large so that the unit bearing load cannot become excessive even under the most unfavorable conditions. At high speeds the bearing loads are due in large part to dynamic forces-inertia and centrifugal. Fortunately, loads on main bearings due to centrifugal force can be reduced, and even completely eliminated, by the provision of suitable counterweights. All dynamic forces increase as the square of the speed of rotation. (i.e. $F_{Dynamic} \uparrow \Rightarrow Speed^2 \uparrow$)

II. DESIGN CALCULATIONS:

In the design of the crankshafts, it is assumed that the crankshaft is a beam with two or more supports. Every crankshaft must be designed or checked at least for two crank positions, one when the bending moment is maximum, and the other when the twisting moment is a maximum. In addition, the additional moments due to the flywheel weight, belt tension and other forces must be considered.

To make the calculations simpler, without loosing accuracy, it is assumed that the effect of the bending forces does not extend two bearings between which a force is applied.
There are two considerations, which determine the necessary dimensions of the crankpin. One is that its projected bearing area (diameter times length) must be large enough so it will safely sustain the bearing loads imposed upon it by gas pressure, inertia and centrifugal force; the second, that the crankshaft as a whole must be sufficiently rigid so that it will not vibrate perceptibly under the periodic forces to which it is subjected in service. When the crankshaft of a given engine is made more rigid, the so-called critical speeds-that is, speeds at which there is synchronous vibration-are raised, and in this way at least the most important critical speeds can be moved outside the normal operating range.

III. INTRODUCTION TO CAD

Computer-aided design (CAD), also known as computer-aided design and drafting (CADD), is the use of computer technology for the process of design and design-documentation. Computer Aided Drafting describes the process of drafting with a computer. CADD software, or environments, provide the user with input-tools for the purpose of streamlining design processes; drafting, documentation, and manufacturing processes. CADD output is often in the form of electronic files for print or machining operations. The development of CADD-based software is in direct correlation with the processes it seeks to economize; industry-based software (construction, manufacturing, etc.) typically uses vector-based (linear) environments whereas graphic-based software utilizes raster-based (pixelated) environments.

CADD environments often involve more than just shapes. As in the manual drafting of technical and engineering drawings, the output of CAD must convey information, such as materials, processes, dimensions, and tolerances, according to application-specific conventions.

CAD may be used to design curves and figures in two-dimensional (2D) space; or curves, surfaces, and solids in three-dimensional (3D) objects.

The design of geometric models for object shapes, in particular, is often called *computer-aided geometric design* (*CAGD*).

Current computer-aided design software packages range from 2D vector-based drafting systems to 3D solid and surface modelers. Modern CAD packages can also frequently allow rotations in three dimensions, allowing viewing of a designed object from any desired angle, even from the inside looking out. Some CAD software is capable of dynamic mathematic modeling, in which case it may be marketed as CADD — *computer-aided design and drafting*.

IV. INTRODUCTION TO CATIA

CATIA name is an abbreviation for Computer Aided Three-dimensional **I**nteractive **A**pplication graphics on papers, for that, we prefer CATIA because it provides us with all the tools that we need. Before we come to learning any 3d modeling software's, You must know their classification as a drawing program, Where CATIA classified under the following software packages: Sheet Metal, Mold Design, Welding, Aerospace Sheet Metal

A. Crank shaft Design in CATIA

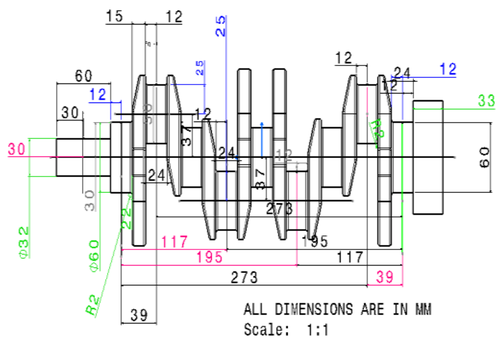
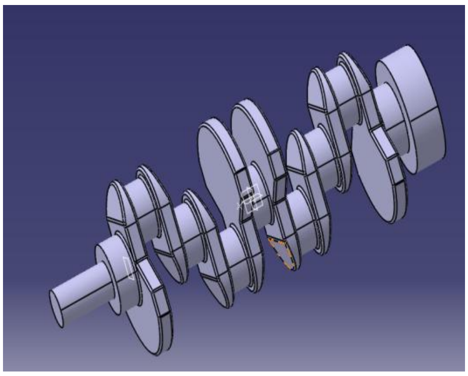


Fig :2D Drawing of Crank shaft Model

Fig :3D Model of Crank shaft



V. MATHEMATICAL MODEL FOR CRANKSHAFT

The configuration of the 4 cylinder Diesel engine crankshaft is tabulated in Table I

Crank pin Dia, d_c	50 *mm*
Shaft Diameter , d_s	32 *mm*
Thickness of the Crank web, *t*	15 *mm*
Bore diameter, *D*	55 *mm*
Length of the crank pin , *l*	24 *mm*
Maximum pressure , P_{max}	3.5 N/mm^2

A. Design of crankshaft when the crank is at an angle of maximum twisting Moment

The Thrust load in the connecting rod will be equal to Gas load on the Piston(F_p) ,We know that Gas load on the piston is equal to,

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

Where D is the diameter of the piston in mm and p_{max} is the maximum gas pressure

$$F_p = \frac{\pi}{4} 55^2 \times 3.5$$

$$F_p = 8.315\ KN$$

In order to find the thrust in the connecting rod (FQ), we should first find out the angle of inclination of the connecting rod with the line of stroke (i.e. angle Ø).

We know that

$$sin\emptyset = \frac{sin\theta}{\frac{l}{r}}$$

$$sin\emptyset = \frac{sin35^0}{4}$$

where Ø = Angle of inclination of the connecting rod with the line of stroke.

Which implies , Ø = 8.24°

We know that thrust in the connecting rod

$$F_Q = \frac{F_P}{\cos\phi}$$

$$F_Q = \frac{8.315}{cos\ 8.24^0}$$

Thrust in the connecting rod , $F_Q = 8.40\ KN$

The thrust in the connecting rod (F_Q) may be divided into two components, one perpendicular to the crank and the other along the crank. The component of F_Q Perpendicular to the crank is tangential force (F_T) and the component of F_Q along the crank is the radial force (F_R) which prodces thrust on the crankshaft bearings.

1) Tangential force on the crank shaft, $F_T = F_Q\, Sin(\theta + \emptyset)$

$$= 8.4\ Sin(35 + 8.24)$$

$$F_T = 5.75\ KN$$

2) Radial force on the crank shaft, $F_R = F_Q \cos(\theta + \phi)$

$$= 8.4\ Cos\ (35 + 8.24)$$

$$F_R = 6.12\ KN$$

Case 1:When the load is applied at Crank Pin -1

B	**b1**	**b2**
312mm	***39 mm***	***273 mm***

Where $\boldsymbol{b_1, b_2, b}$ are distance between bearings

Reactions at bearings (1 & 2) due to Tangential force (F_T) is given by,

$$H_{T_1} = \frac{F_T \times b_1}{b}$$

$$H_{T_1} = \frac{5.75 \times 39}{312}$$

$$H_{T_1} = 0.71875\ KN$$

$$H_{T_2} = \frac{F_T \times b_2}{b}$$

$$H_{T_2} = \frac{5.75 \times 273}{312}$$

$$H_{T_2} = 5.01325\ KN$$

Reactions at bearings (1 & 2) due to Radial force (F_R) is given by,

$$H_{R_1} = \frac{F_R \times b_1}{b}$$

$$H_{R_1} = \frac{6.12 \times 39}{312}$$

$$H_{R_1} = 0.765\ KN$$

$$H_{R_2} = \frac{F_R \times b_2}{b}$$

$$H_{R_2} = \frac{6.12 \times 273}{312}$$

$$H_{R_2} = 5.355\ KN$$

B. Design of crankpin -1:

Let d_c = Diameter of crankpin in mm.

We know that the bending moment at the centre of the crankpin,

$M_c = H_{R1} \times b_2$ = 0.765 x 273 = 208.845 KN-mm

Twisting moment on the crankpin,

$T_c = F_T \times r$

where ,r = distance between center of the crankpin to shaft in mm

$$T_c = 5.75 \times 37 = 212.75\ KN - mm$$

From this we have the equivalent twisting moment,

$$T_e = \sqrt{M_c^2 + T_c^2}$$

$$T_e = \sqrt{208.845^2 + 212.75^2}$$

T_e= 298.1255 mm

According to distortion energy theory The von Mises stress induced in the crank-pin -1 is,

$$M_{ev} = \sqrt{(k_b \times M_c)^2 + \frac{3}{4}(k_t \times T_c)^2}$$

= 500.844 KN-mm

Here, K_b = combined shock and fatigue factor for bending (Take K_b=2)

K_t = combined shock and fatigue factor for torsion (Take K_t=1.5)
we know that ,

$$M_{ev} = \frac{\pi}{32} \times d_c^3 \times \sigma_v$$

$$500.844 = \frac{\pi}{32} \times 50^3 \times \sigma_v$$

von Mises stress induced in the crank-pin -1 , σ_v = 40.812 MPa

Shear stress ,

$$T_e = \frac{\pi}{16} \times d_c^3 \times \tau$$

$$298.1255 = \frac{\pi}{16} \times 50^3 \times \tau$$

Shear stress , $\tau = 12.146\ MPa$

Case 2:When the load is applied at Crank Pin -2

B	**b_1**	**b_2**
312 mm	***117 mm***	***195 mm***

Where ***b_1,b_2,b*** are distance between bearings
Reactions at bearings (1 & 2) due to Tangential force (F_T) is given by,

$$H_{T_1} = \frac{F_T \times b_1}{b}$$

$$H_{T_1} = \frac{5.75 \times 117}{312}$$

$$H_{T_1} = 2.156\ KN$$

$$H_{T_2} = \frac{F_T \times b_2}{b}$$

$$H_{T_2} = \frac{5.75 \times 195}{312}$$

$$H_{T_2} = 3.593\ KN$$

Reactions at bearings (1 & 2) due to Radial force (F_R) is given by,

$$H_{R_1} = \frac{F_R \times b_1}{b}$$

$$H_{R_1} = \frac{6.12 \times 117}{312}$$

$$H_{R_1} = 2.295\ KN$$

$$H_{R_2} = \frac{F_R \times b_2}{b}$$

$$H_{R_2} = \frac{6.12 \times 195}{312}$$

$$H_{R_2} = 3.825\ KN$$

C. Design of crankpin -2:

Let d_c = Diameter of crankpin in mm.
We know that the bending moment at the centre of the crankpin,
$M_c = H_{R1} \times b_2$ = 2.295 x 195 = 447.525 KN-mm
Twisting moment on the crankpin,
$T_c = F_T \times r$
where ,r = distance between center of the crankpin to shaft in mm

$$T_c = 5.75 \times 37 = 212.75\ KN - mm$$

From this we have the equivalent twisting moment,

$$T_e = \sqrt{M_c^2 + T_c^2}$$

$$T_e = \sqrt{447.525^2 + 212.75^2}$$

T_e= *495.5211 mm*

According to distortion energy theory The von Mises stress induced in the crank-pin -1 is,

$$M_{ev} = \sqrt{(k_b \times M_c)^2 + \frac{3}{4}(k_t \times T_c)^2}$$

= 936.747 KN-mm

Here, K_b = combined shock and fatigue factor for bending (Take K_b=2)
K_t = combined shock and fatigue factor for torsion (Take K_t=1.5)
we know that ,

$$M_{ev} = \frac{\pi}{32} \times d_c^3 \times \sigma_v$$

$$936.747 = \frac{\pi}{32} \times 50^3 \times \sigma_v$$

Von Mises stress induced in the crank-pin -2 , σ_v = 76.33 MPa

Shear stress ,

$$T_e = \frac{\pi}{16} \times d_c^3 \times \tau$$

$$495.5211 = \frac{\pi}{16} \times 50^3 \times \tau$$

Shear stress , $\tau = 20.18\ MPa$

Case 3:When the load is applied at Crank Pin -3

B	**b_1**	**b_2**
312 mm	***195 mm***	***117 mm***

Where ***b_1,b_2,b*** are distance between bearings
Reactions at bearings (1 & 2) due to Tangential force (F_T) is given by,

$$H_{T_1} = \frac{F_T \times b_1}{b}$$

$$H_{T_1} = \frac{5.75 \times 195}{312}$$

$$H_{T_1} = 3.593\ KN$$

$$H_{T_2} = \frac{F_T \times b_2}{b}$$

$$H_{T_2} = \frac{5.75 \times 117}{312}$$

$$H_{T_2} = 2.156\ KN$$

Reactions at bearings (1 & 2) due to Radial force (F_R) is given by,

$$H_{R_1} = \frac{F_R \times b_1}{b}$$

$$H_{R_1} = \frac{6.12 \times 195}{312}$$

$$H_{R_1} = 3.825\ KN$$

$$H_{R_2} = \frac{F_R \times b_2}{b}$$

$$H_{R2} = \frac{6.12 \times 117}{312}$$
$$H_{R2} = 2.295\ KN$$

D. Design of crankpin -3:

Let d_c = Diameter of crankpin in mm.
We know that the bending moment at the centre of the crankpin,
$M_c = H_{R1} \times b_2 = 3.825 \times 117 = 447.525$ KN-mm
Twisting moment on the crankpin,
$T_c = F_T \times r$
where ,r = distance between center of the crankpin to shaft in mm

$$T_c = 5.75 \times 37 = 212.75\ KN - mm$$

From this we have the equivalent twisting moment,

$$T_e = \sqrt{M_c^2 + T_c^2}$$
$$T_e = \sqrt{447.525^2 + 212.75^2}$$
$$T_e = 495.5211\ mm$$

According to distortion energy theory The von Mises stress induced in the crank-pin -1 is,

$$M_{ev} = \sqrt{(k_b \times M_c)^2 + \frac{3}{4}(k_t \times T_c)^2}$$
$= 936.747$ KN-mm

Here, K_b = combined shock and fatigue factor for bending (Take K_b=2)
K_t = combined shock and fatigue factor for torsion (Take K_t=1.5)

we know that ,

$$M_{ev} = \frac{\pi}{32} \times d_c^3 \times \sigma_v$$
$$936.747 = \frac{\pi}{32} \times 50^3 \times \sigma_v$$

Von Mises stress induced in the crank-pin -3 , $\sigma_v = 76.33$ MPa

Shear stress , $T_e = \frac{\pi}{16} \times d_c^3 \times \tau$

$$495.5211 = \frac{\pi}{16} \times 50^3 \times \tau$$

Shear stress , $\tau = 20.18\ MPa$

Case 4: When the load is applied at Crank Pin -4

B	b_1	b_2
312 mm	273 mm	39 mm

Where b_1,b_2,b are distance between bearings
Reactions at bearings (1 & 2) due to Tangential force (F_T) is given by,

$$H_{T1} = \frac{F_T \times b_1}{b}$$
$$H_{T1} = \frac{5.75 \times 273}{312}$$
$$H_{T1} = 5.0132\ KN$$
$$H_{T2} = \frac{F_T \times b_2}{b}$$
$$H_{T2} = \frac{5.75 \times 39}{312}$$
$$H_{T2} = 0.71875\ KN$$

Reactions at bearings (1 & 2) due to Radial force (F_R) is given by,

$$H_{R1} = \frac{F_R \times b_1}{b}$$
$$H_{R1} = \frac{6.12 \times 273}{312}$$
$$H_{R1} = 5.355\ KN$$
$$H_{R2} = \frac{F_R \times b_2}{b}$$
$$H_{R2} = \frac{6.12 \times 39}{312}$$
$$H_{R2} = 0.765\ KN$$

E. Design of crankpin -4:

Let d_c = Diameter of crankpin in mm.
We know that the bending moment at the centre of the crankpin,
$M_c = H_{R1} \times b_2 = 5.355 \times 39 = 208.845$ KN-mm
Twisting moment on the crankpin,
$T_c = F_T \times r$
where ,r = distance between center of the crankpin to shaft in mm

$$T_c = 5.75 \times 37 = 212.75\ KN - mm$$

From this we have the equivalent twisting moment,

$$T_e = \sqrt{M_c^2 + T_c^2}$$
$$T_e = \sqrt{208.845^2 + 212.75^2}$$
$$T_e = 298.1255\ mm$$

According to distortion energy theory The von Mises stress induced in the crank-pin -1 is,

$$M_{ev} = \sqrt{(k_b \times M_c)^2 + \frac{3}{4}(k_t \times T_c)^2}$$
$= 500.844$ KN-mm

Here, K_b = combined shock and fatigue factor for bending (Take K_b=2)
K_t = combined shock and fatigue factor for torsion (Take K_t=1.5)
we know that , $M_{ev} = \frac{\pi}{32} \times d_c^3 \times \sigma_v$

$$500.844 = \frac{\pi}{32} \times 50^3 \times \sigma_v$$

von Mises stress induced in the crank-pin -4 , $\sigma_v = 40.812$ MPa

Shear stress , $T_e = \frac{\pi}{16} \times d_c^3 \times \tau$

$$298.1255 = \frac{\pi}{16} \times 50^3 \times \tau$$

Shear stress , $\tau = 12.146\ MPa$

VI. INTRODUCTION TO ANSYS

ANSYS is general-purpose finite element analysis (FEA) software package. Finite Element Analysis is a numerical method of deconstructing a complex system into very small pieces (of user-designated size) called elements. The software implements equations that govern the behaviour of these elements and solves them all; creating a comprehensive explanation of how the system acts as a whole. These results then can be presented in tabulated, or graphical forms. This type of analysis is typically used for the design and optimization of a system far too complex to analyze by hand. Systems that may fit into this category are too complex due to their geometry, scale, or governing equations.

ANSYS is the standard FEA teaching tool within the Mechanical Engineering Department at many colleges. ANSYS is also used in Civil and Electrical Engineering, as well as the Physics and Chemistry departments.

ANSYS provides a cost-effective way to explore the performance of products or processes in a virtual environment. This type of product development is termed virtual prototyping.

With virtual prototyping techniques, users can iterate various scenarios to optimize the product long before the manufacturing is started. This enables a reduction in the level of risk, and in the cost of ineffective designs. The multifaceted nature of ANSYS also provides a means to ensure that users are able to see the effect of a design on the whole behavior of the product, be it electromagnetic, thermal, mechanical etc.

VII. INTRODUCTION TO FEA

Finite Element Analysis (FEA) was first developed in 1943 by R. Courant, who utilized the Ritz method of numerical analysis and minimization of variational calculus to obtain approximate solutions to vibration systems. Shortly thereafter, a paper published in 1956 by M. J. Turner, R. W. Clough, H. C. Martin, and L. J. Topp established a broader definition of numerical analysis. The paper centered on the "stiffness and deflection of complex structures".

Cylinder Crank shaft Structural Analysis in ANSYS

1. First, Prepared Assembly in CATIA V5 for crankshaft and Save as this part as IGES for Exporting into Ansys Workbench Environment. Import .IGES Model in ANSYS Workbench Simulation Module.

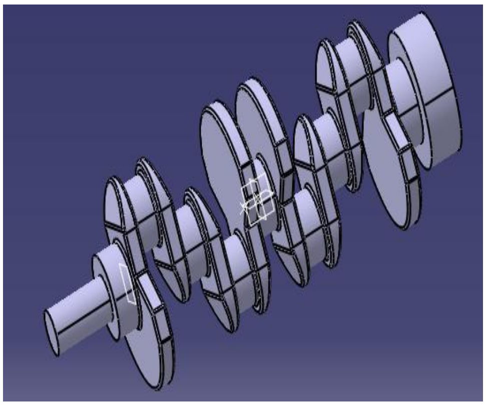


Fig 1: Crank shaft 3D Model

2. Apply Material for Crank Shaft (Forged steel).

Material Details: **Cast Iron and chromium nickel steel**

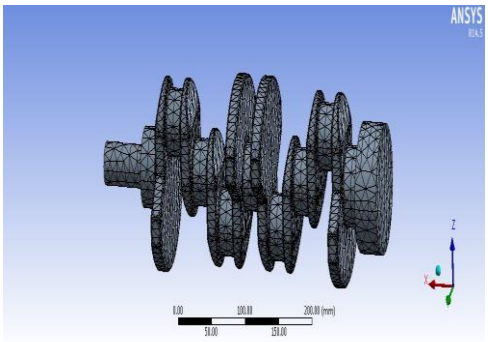
Properties of castiron

Structural	
Young's Modulus	1.78e+005 M Pa
Poisson's Ratio	0.3
Density	7.197e-006 kg/mm³

Properties of chromium nickel

Young's modulus (GPa)	210
Poisson's ratio	0.3
Density (Kg/m^3)	8030

3. Mesh the Crankshaft.



4. Define boundary condition for Analysis Boundary conditions play an important role in finite element calculation here; take both remote displacements for bearing supports are fixed.
5. Define type of Analysis
   Type of Analysis:-Static Structural
6. Apply Radial Force on Crank Pin
7. Apply Tangential force on Crank Pin

8. Run the Analysis
9. Get the Results

VIII. ANSYS RESULTS FOR CAST IRON

Case-1: Crank Pin -1

A. Von Misses Stresses:

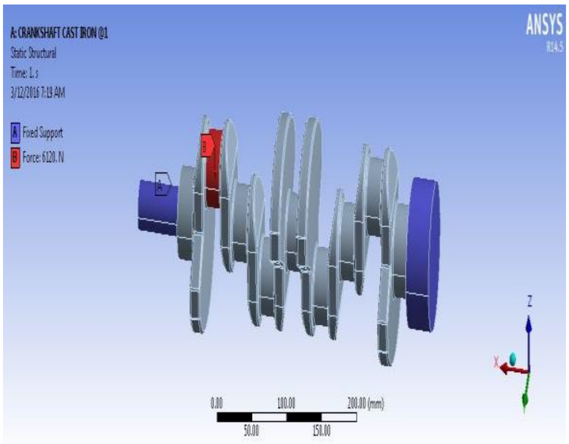


Fig2: Radial Force Applied on Crank Pin-1

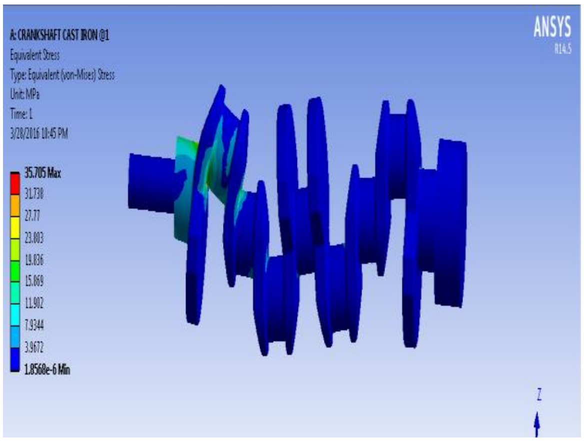


Fig 3: Von Mises stress - Crank Pin-1

B. Shear Stresses:

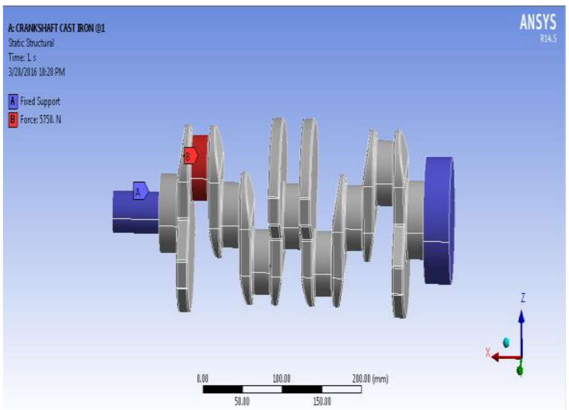


Fig4: Tangential Load applied on Crank Pin -1

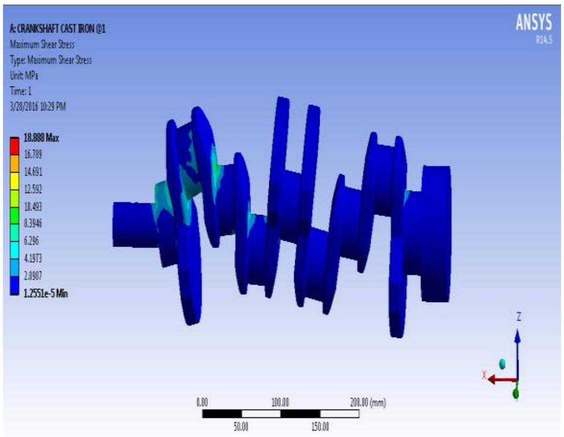


Fig5: Shear Stresses n Crank Pin -1

Case-2: Crank Pin -2

C. Von Misses Stresses:

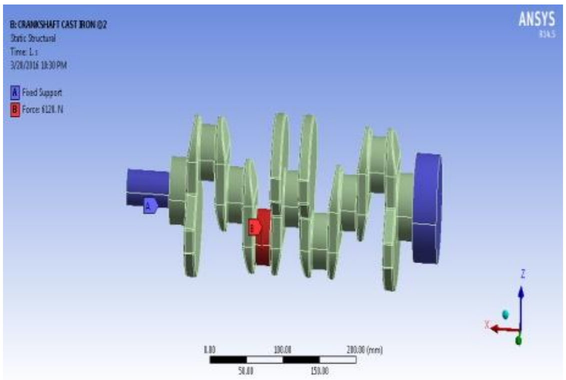


Fig6: Radial Force Applied on Crank Pin-2

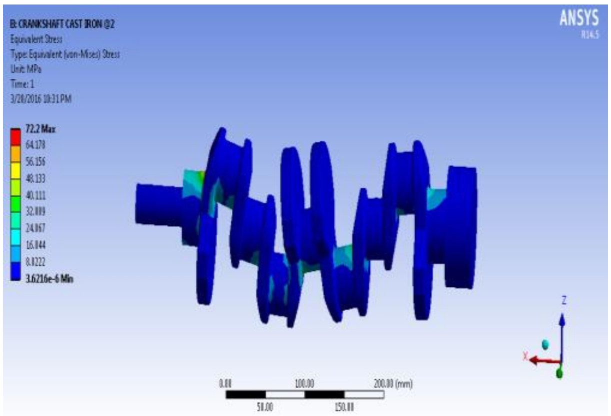


Fig 7: Von Mises stress - Crank Pin-2

D. Shear Stresses:

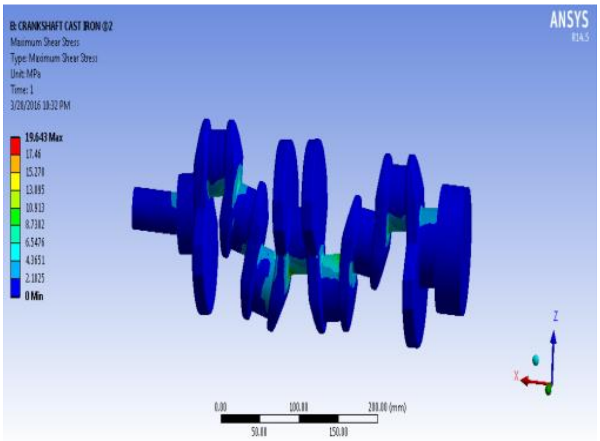


Fig8: Shear Stresses on Crank Pin-2

Case-3: Crank Pin -3

E. Von Misses Stresses:

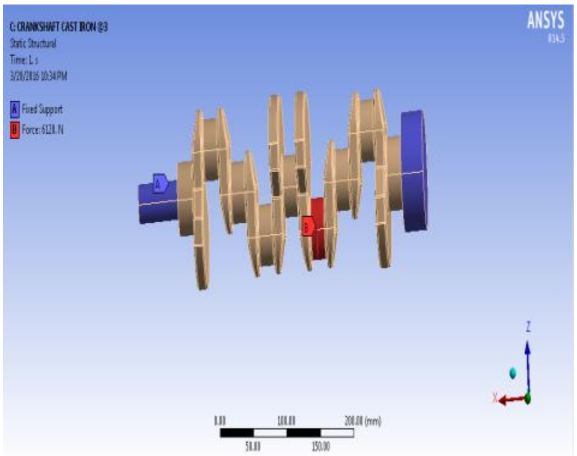


Fig9: Radial Force Applied on Crank Pin-3

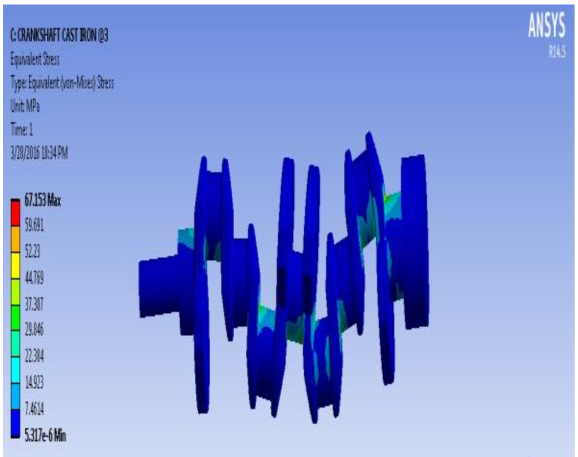


Fig10 : Von Mises stress - Crank Pin-3

F. Shear Stresses:

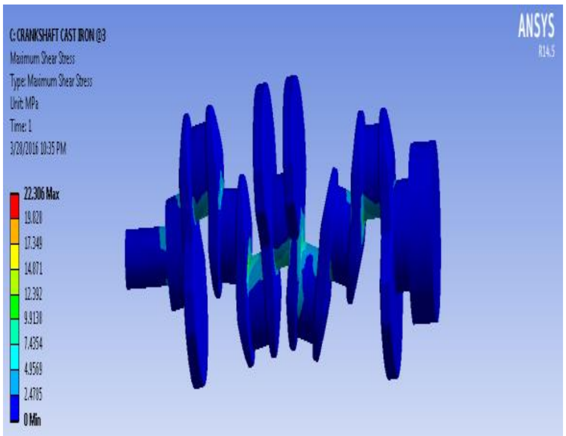


Fig11: Shear Stresses on Crank Pin-3

Case-4: Crank Pin -4

G. Von Misses Stresses:

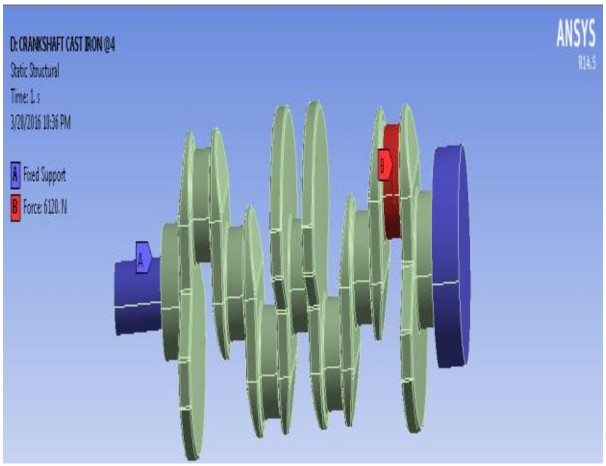


Fig12: Radial Force Applied on Crank Pin-4

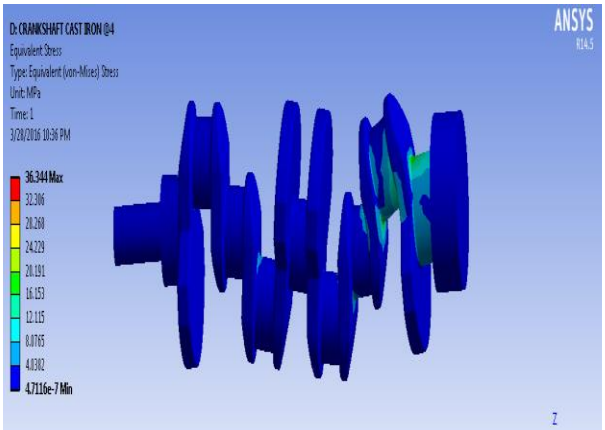


Fig13 : Von Mises stress - Crank Pin-4

IX. ANSYS RESULTS FOR AISI 1040 CHROMIUM NICKEL ALLOY

Case-1: Crank Pin -1

A. Von Misses Stresses:

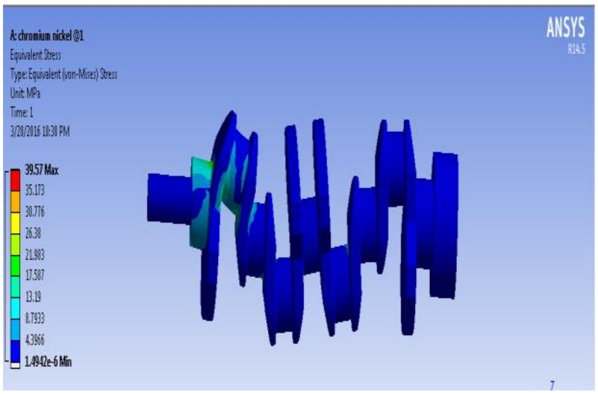


Fig 15: Von Mises stress - Crank Pin-1

B. Shear Stresses:

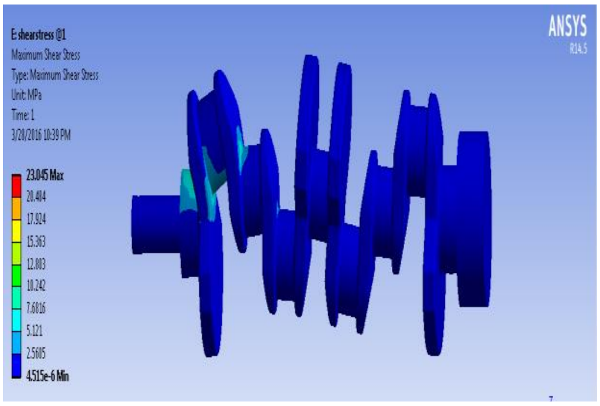


Fig16: Shear Stresses on Crank Pin-1

Case-2: Crank Pin -2

C. Von Misses Stresses:

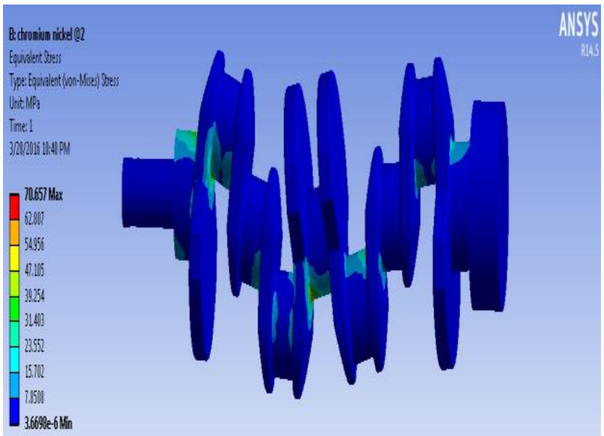


Fig 17: Von Mises stress - Crank Pin-2

D. Shear Stresses:

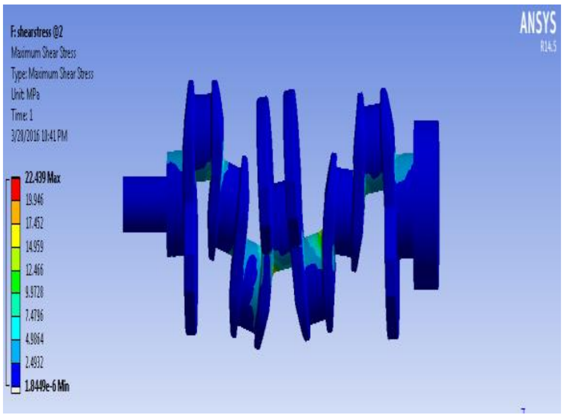


Fig18: Shear Stresses on Crank Pin-2

Case-3: Crank Pin -3

E. Von Misses Stresses:

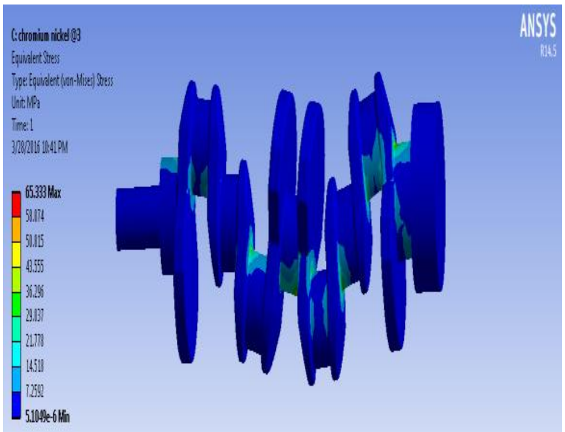


Fig19 : Von Mises stress - Crank Pin-3

F. Shear Stresses:

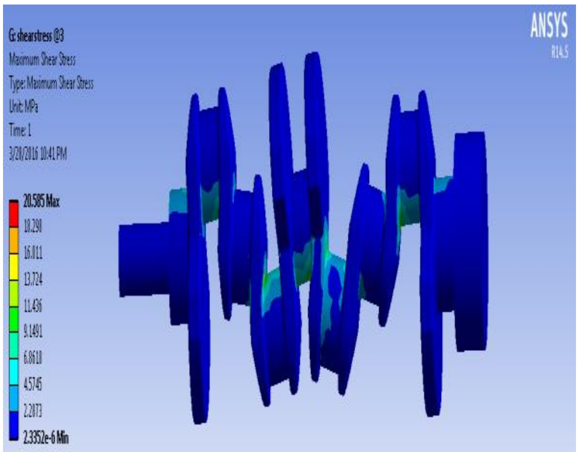


Fig 20: Shear Stresses on Crank Pin-3

Case-4: Crank Pin -4

G. Von Misses Stresses:

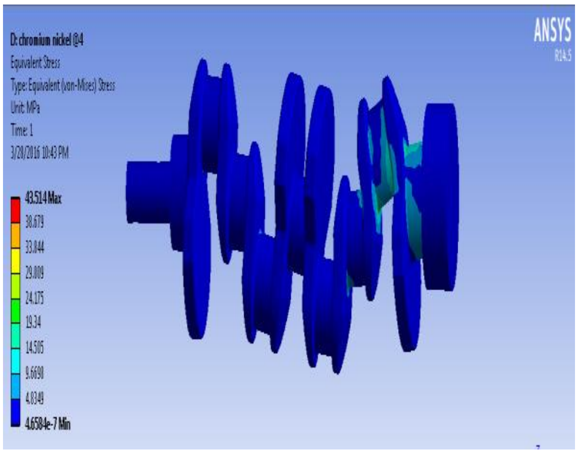


Fig21 : Von Mises stress - Crank Pin-4

H. Shear Stresses:

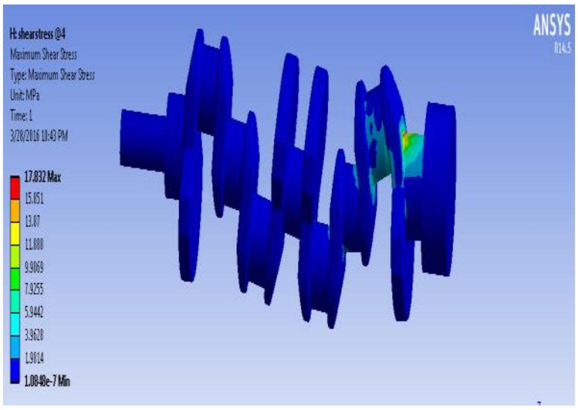


Fig 22: Shear Stresses on Crank Pin-4

X. COMPARISION BETWEEN ANALYTICAL AND FEA RESULTS

The FEA results are in close agreement with the theoretical results. The variation in the Von-Mises stress is 15.88%. The variation in the Shear Stress is 12.2%. So the FEA results are good. Comparison of results is shown in the following Table.

S.NO	Von Mises Stress in Mpa		Shear Stress Mpa	
	Analytical	FEA	Analytical	FEA
1	40.812	35.70	12.146	18.88
2	76.333	72.2	20.189	19.643
3	76.333	67.15	20.18	22.30
4	40.81	36.33	12.146	14.99

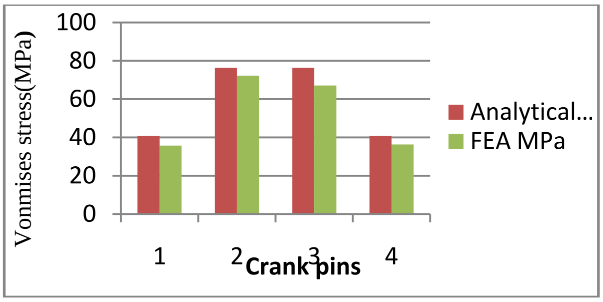


Fig 23: Comparision of VonMises Stresses

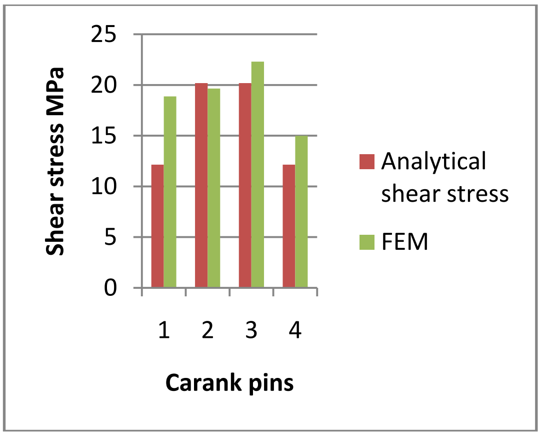


Fig 24: Comparision of shear Stresses

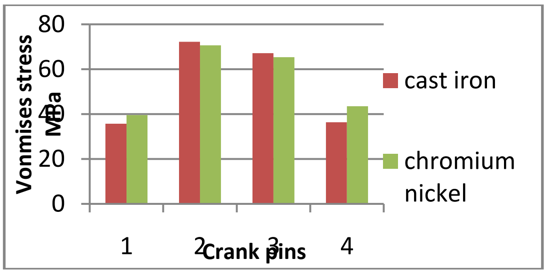


Fig 25: Comparision of Vonmises stresses in cast iron and chromium nickel alloy

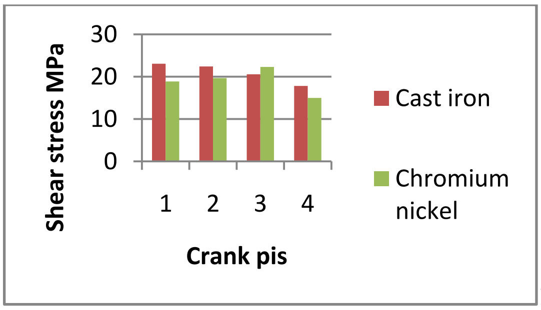


Fig 26: Comparision of shear stresses in cast iron and chromium nickel alloy

XI. RESULTS

In this Project, the crankshaft model was created by CATIA V5 R20 software. Then, the model created by CATIA was imported to ANSYS software.

S.NO	Von Mises Stress in Mpa		Shear Stress Mpa	
	Analytical	FEA	Analytical	FEA
Crank Pin 1	40.81	35.70	12.14	18.88
Crank Pin 2	76.3	72.2	20.18	19.6
Crank Pin 3	76.3	67.15	20.18	22.3
Crank Pin 4	40.8	36.33	12.14	14.99

Result Table

XII. CONCLUSION

Above 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. The maximum deformation appears at the center 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. The Value of Von-Misses Stresses that comes out from the analysis is far less than material yield stress so our design is safe and we should go for optimization to reduce the material and cost. From table 2 , we observed that less von mises and shear stress was developed in chromium nickel alloy compared to cast iron as shown in fig 25 and 26 . After Performing Static Analysis ,Dynamic analysis of the crankshaft which results shows more realistic whereas static analysis provides an overestimate results. Accurate stresses and deformation are critical input to fatigue analysis and optimization of the crankshaft. Analysis Results. So we can Say that Dynamic FEA is a good tool to reduce Costly experimental work.

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