ISSN 2395-695X (Print)
ISSN 2395-695X (Online)

International Journal of Advanced Research in Biology Engineering Science and Technology (IJARBEST)
Vol. 2, Issue 4, April 2016

Design and Analysis of Solid Crank Shaft Using Catia and ANSYS Workbench

Ankit Basnet[1], Saurav Rajgadia[2], Anush Karki[3], Debayan Das[4], Pawan Jaiswal[5], Rakesh Jaiswal[6], Anupam Raj Jha[7], Rabindra Nath Barman[8]

[8]Assistant Professor, Mechanical Engineering Department, National Institute of Technology Durgapur. [1,2,3,4,5,6,7] B.Tech Students, Mechanical Engineering, National Institute of Technology Durgapur, Durgapur-713209, West Bengal, India,

***Abstract*—The crankshaft is an important component of Internal Combustion (IC) engine that converts the reciprocating motion into rotary motion through the connecting rod. 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. This paper represent the design and analysis of crank shaft with a motive to decrease weight. Here, the shaft is modelled using Catia V-5 and analysed in ANSYS 16.0**.

***Index Terms*— Crank shaft, Catia V-5, ANSYS 16.0, Finite Element Analysis (FEA)**

I. INTRODUCTION

The crankshaft is an important machine element of IC engine that converts the reciprocating motion of the piston into rotary motion through the connecting rod. The crankshaft consists of three portions i.e. crank pin, crank web and shaft. The big end of the connecting rod is attached to the crank pin. The crank web connects the crank pin to the shaft portion. The shaft portion rotates in the main bearings and transmits power to the outside source through the belt drive, gear drive or chain drive.

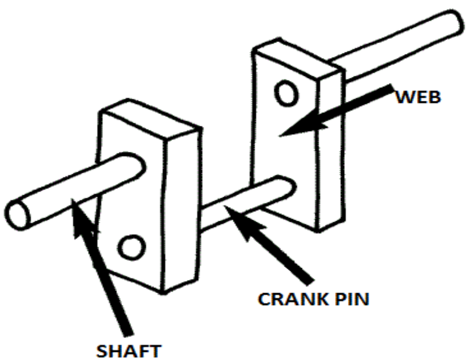

Fig. 1. Crank shaft

There are two types of crankshafts- side crankshaft and centre crankshaft. The side crankshaft is also called the 'overhung' crankshaft. It has only one crank web and requires only two bearings for support. It is used in medium-size engines and large-size horizontal engines. The centre crankshaft (as shown in Fig.1.) has two webs and three bearings for support. It is used in radial aircraft engines, stationary and marine engines. It is more popular in automotive engines [4]. Crankshafts are also classified as single-throw and multi-throw crankshafts depending upon the number of crank pins used in the assembly. The crankshafts illustrated in Fig.1 have one crank pin and are called single-throw crankshafts. Crankshafts used in multi-cylinder engines have more than one crank pin. They are called multi-throw crankshafts.

A crankshaft should have sufficient strength to withstand the bending and twisting moments to which it is subjected. [5] 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.[13]

A. Material and its properties

[1]The crankshaft is usually manufactured by drop forging, casting or powdered metallurgy. The commonly used materials for the crankshaft are carbon steels (40C8, 45C8 and 50C4), alloy steel (16Ni3Cr2, 35Ni5Cr2), cast iron, aluminium alloy (T62024, T651-707) and titanium alloy. Crankshaft made from aluminium is light and can absorb high impact at expense of durability while the Crankshaft made using titanium is used for combination of light and strength at expense of affordability. The material used in this paper i.e. Aged Grade 250 Maraging steel has ultra-high strength, good toughness and superb transverse properties and is widely used in motorsport industry. It derives its strength not from carbon, but from precipitation of intermetallic compound. Secondary alloying elements such as Cobalt (Co), Manganese (Mn) and Titanium (Ti) are added to Nickel to produce intermetallic precipitates. The addition of chromium produces stainless grades resistant to corrosion.[3] Due to low carbon content maraging steels have good machinability prior to aging. It may also be cold rolled up to 90% without cracking. Ti offers good weldability. High alloy content maraging steels have a high hardenability.

II. DESIGN OF CRANKSHAFT

A crankshaft is subjected to bending and torsional moments due to the following three forces:-[2]

i. Force exerted by the connecting rod on the crank pin.
ii. Weight of flywheel (W) acting downward in the vertical direction.
iii. Resultant belt tensions acting in the horizontal direction$(P_1 + P_2)$.

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.[14]

ISSN 2395-695X (Print)
ISSN 2395-695X (Online)

International Journal of Advanced Research in Biology Engineering Science and Technology (IJARBEST)
Vol. 2, Issue 4, April 2016

CENTER CRANKSHAFT AT TOP DEAD CENTER POSITION:
The forces acting on the centre crankshaft at the top dead centre position are shown in Fig.2. The crankshaft is supported on three bearings 1, 2 and 3.

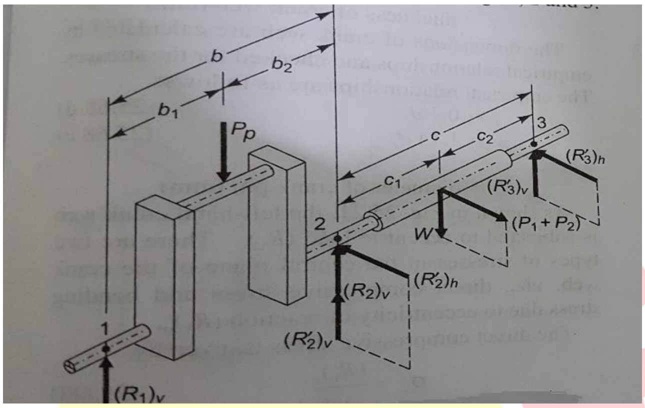

Fig 2. Different Forces acting on Crank shaft

(i) Bearing Reactions

(a) The reactions at the bearings 1 and 2 due to force on the crank pin (P_p) are denoted by R_1 and R_2 followed by suffix letters v and$h$. The vertical component of reaction is denoted by the suffix letter v such as$(R_1)_v$. The

(b) Horizontal component of reaction is denoted by the suffix letter h such as$(R_1)_h$.

(c) The reactions at the bearings 2 and 3 due to weight of the flywheel (W) and sum of the belt tensions $(P_1 + P_2)$ are denoted by R'_2 and R'_3 followed by suffix letters v and h such as $(R'_2)_v$ or $(R'_3)_h$.

Nomenclature:

$P_p =$ Force acting on crank pin (N)
$D =$ Diameter of piston (mm)
p_{max} = Maximum gas pressure inside the cylinder (MPa or $\mathrm{N/mm^2}$)
$W =$ Weight of flywheel (N)
$P_1 =$ Tension in tight side of belt (N)
$P_2 =$ Tension in slack side of belt (N)
$b =$ Distance between main bearings 1 and 2
$c =$ Distance between bearings 2 and 3

Given data,

Cylinder bore (D) = 125 mm
(L/r) = 4.5
Maximum gas pressure (P_{max}) = 2.5 MPa
Stroke length (l) = 150 mm
Weight of fly wheel cum belt pulley (W) = 1 KN = 1000 N
Total belt pull ($P_1 + P_2$) = 2 KN = 2000 N
Width of hub for flywheel cum belt pulley = 200 mm
Considering crank angle from top dead centre to be Ɵ = 25° and
The gas pressure in cylinder at this instant be (P') = 2 MPa.

The crank radius r is given by, R = $\left(\frac{L}{2}\right) = \frac{150}{2}$ = 75 mm

Step I: Bearing reactions

At the top dead centre position, the thrust in the connecting rod will be equal to the force acting on piston.

$P_p = \left(\frac{\pi D^2}{4}\right) p_{max.} = \left(\frac{\pi(125)^{\wedge}2}{4}\right)(2.5)$ = 30679062 N

Assumption 1: The centre to centre distance between the main bearing 1 and 2 is twice of the piston diameter. Therefore,

b = 2* Piston diameter = 2*125 = 250 mm

It is further assumed that $b_1 = b_2$, i.e.$b_1 = b_2 = \frac{b}{2} = \frac{250}{2}$ = 125 mm

It is assumed that the portion of the crankshaft between bearings 1 and 2 is simply supported on bearings and subjected to force P_p.

Taking moment of forces,

$P_p \times b_1 = (R_2)_v \times b$ Or $(R_2)_v = \frac{P_p \times b_1}{b}$ (a)

Similarly, $P_p \times b_2 = (R_1)_v \times b$ or $(R_1)_v = \frac{P_p \times b_2}{b}$ (b)

By symmetry,

$$(R_1)_v = (R_2)_v = \frac{P_p \times b_1}{b} = \frac{30679.62}{2} = 15339.81\ N$$

It is also assumed that the portion of the crankshaft between bearings 2 and 3 is simply supported on bearings and subjected to a vertical force W and horizontal force $(P_1 + P_2)$. Taking moment of forces,

$$W \times c_1 = (R'_3)_v \times c \text{ or } (R'_3)_v = \frac{W \times c_1}{c} \quad \text{(c)}$$

$$W \times c_2 = (R'_2)_v \times c \text{ or } (R'_2)_v = \frac{W \times c_2}{c} \quad \text{(d)}$$

$$(P_1 + P_2) \times c_1 = (R'_3)_h \times c \text{ Or } (R'_3)_h = \frac{(P_1+P_2) \times c_1}{c} \quad \text{(e)}$$

$$(P_1 + P_2) \times c_2 = (R'_2)_h \times c \text{ Or } (R'_2)_h = \frac{(P_1+P_2) \times c_2}{c} \quad \text{(f)}$$

Similarly It is assumed that, $c_1 = c_2$

Therefore, by symmetry,

$$(R'_2)_v = (R'_3)_v = \frac{W \times c_1}{c} = \frac{w}{2} = \frac{1000}{2} = 500N$$

$$(R'_2)_h = (R'_3)_h = \frac{(P_1 + P_2) \times c_1}{c} = \frac{(P_1 + P_2)}{2} = \frac{2000}{2} = 1000\ N$$

The resultant reactions at the bearings are as follows:

$\boldsymbol{R_1} = (R_1)_v$ = 15339.81 N

$\boldsymbol{R_2} = \sqrt{[(R_2)_v + (R'_2)_v]^2 + [(R'_2)_h]^2} = \sqrt{[15339.81 + 500]^2 + [1000]^2}$ = 15871.3346 N

$\boldsymbol{R_3} = \sqrt{[(R'_3)_v]^2 + [(R'_3)_h]^2} = \sqrt{[500]^2 + [1000]^2}$=1118.03398 N

ISSN 2395-695X (Print)
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Step II: Design of Crank Pin

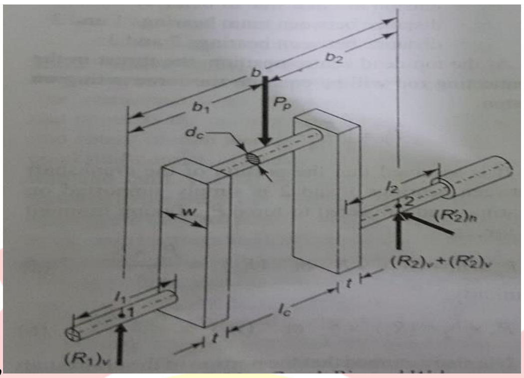

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Fig. 3. Crank pin

Assumption 2: The allowable bending stress for the crank pin is 75 N/mm^2
Assumption 3: The allowable bearing pressure for the crank pin bush is 10 N/mm^2
Suppose,
$d_c =$ Diameter of crank pin (mm)
$l_c =$ Length of crank pin (mm)
$\sigma_b =$ Allowable bending stress for crank pin (N/mm^2) **= 75 N/mm^2**

The bending moment at the central plane is given by,
$(M_b)_c = (R_1)_v b_1$ = 15339.81*125 =1917480 N-mm
$(M_b)_c = \left(\frac{\pi d_c^3}{32}\right)\sigma_c$
Substituting,
1917480 = $\left(\frac{\pi d_c^3}{32}\right)$ 75
i.e. d_c^3= 260420 Therefore, d_c= 63.86 or 65 mm
Again,
$I = \left(\frac{\pi d_c^4}{64}\right), y = \left(\frac{d_c}{2}\right)$ and $\sigma_b = \frac{(M_b)_c y}{I}$
Substituting,
$(M_b)_c = \left(\frac{\pi d_c^3}{32}\right)\sigma_c$
Assumption 4: the (L/r) ratio for crank pin bearing is 1.
$\left(\frac{Lc}{dc}\right)$=1
$L_c = d_c$ = 65 mm
$P_b = \frac{Pp}{dclc} = \frac{30679062}{65*65}$=7.26 N/mm^2 i.e. P_b < 10N/mmm^2

Step III: Design of left hand crank web

Suppose,
W= width of crank web (mm)
t = thickness of crank web (mm)

ISSN 2395-695X (Print)
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Crank web dimension are calculated by empirical relationship and checked for stress. The empirical relation are:

t = 0.7*d_c = 0.7*65 = 45.5 mm i.e. t = 46 mm

w = 1.14*d_c = 1.14*65 =74.1 mm i.e. w = 75 mm

The left-hand crank web is subjected to eccentric load $(R_1)_v$.

There are two types of stresses in central plane of crank web i.e. direct compressive stress and bending stress due to reaction $(R_1)_v$.

The direct compressive stress is given by,

$\sigma_c = \frac{(R_1)_v}{w*t} = \frac{15339.81}{75*46}$ = 4.45 N/mm^2

$\sigma_b = \frac{6((R_1)_v)\left[b1-\frac{Lc}{2}-\frac{t}{2}\right]}{w*t^2} = \frac{6*15339.81\left[125-\frac{65}{2}-\frac{46}{2}\right]}{75*46*46}$ = 40.31 N/mm^2

The total compressive stress is given by,

$(\sigma c)_t = \sigma_c + \sigma_b$ = 4.45 + 40.31 = 44.76 N/mm^2

The total compressive stress is less than the allowable bending stress of 75 N/mm^2 and the design of crank web is safe.

Step IV: Design of right hand crank-web

Right web is made identical from balancing consideration. Therefore the thickness of right and left are made equal.

Step V: Design of shaft under flywheel

The width of the hub for flywheel cum belt pulley is given as 200 mm. it is observed from figure that the centre to centre distance between bearing 2 and 3 should be more than 200 mm to accommodate the bearings. We will assume

C = 200 + margin for length of two bearings 2and 3

C = 200 + 100 = 300 mm

$C_1 = C_2 = \frac{C}{2} = \frac{300}{2}$= 150 mm

Suppose,

d_s = Diameter of shaft under flywheel (mm)

The central plane of the shaft is subjected to maximum bending moment. The bending moment in the vertical plane due to weight of the flywheel is given by,

$(M_b)_v = (R'_3)_v c_2 = 500(150) = 75 \times 10^3$ N-mm

The bending moment in the horizontal plane due to resultant belt tension is given by,

$(M_b)_h = (R'_3)_h c_2 = 1000(150) = 150 \times 10^3$ N-mm

The resultant bending moment is given by,

$M_b = \sqrt{(M_b)_v^2 + (M_b)_h^2} = \sqrt{(75 \times 10^3)^2 + (150 \times 10^3)^2} = 167.71 \times 10^3$ N-mm

The allowable bending stress is 75 N/mm^2

From Equations:-

$M_b = \left[\frac{\pi d_s^3}{32}\right] \sigma_b$ Or $167.71 \times 10^3 = \left[\frac{\pi d_s^3}{32}\right](75)$

$d_s^3 = 22.78 \times 10^3$

ISSN 2395-695X (Print)
ISSN 2395-695X (Online)

International Journal of Advanced Research in Biology Engineering Science and Technology (IJARBEST)
Vol. 2, Issue 4, April 2016

$d_s = 28.35\ or\ 30$ mm

THE CRANK IS AT AN ANGLE WITH THE LINE OF THE DEAD CENTRE POSITION:

Step I: Components of force on crank pin

The crank angle (θ) for maximum torsional moment is given as 25°.
Since, p' is the gas pressure on the piston to for maximum torque condition.

$$P_p = \left[\frac{\pi D^2}{4}\right] p' = \left[\frac{\pi(125)^2}{4}\right](2) = 24543.69 \text{ N}$$

$$\sin \emptyset = \frac{\sin \theta}{(L/r)} = \frac{\sin 25}{4.5} = 0.0939 \text{ i.e. } \emptyset = \sin^{-1} 0.09392 = 5.39°$$

The thrust on the connecting rod (P_q) is given by,

$$P_q = \frac{P_p}{\cos \emptyset} = \frac{24543.69}{\cos 5.39} = 24652.69 \text{ N}$$

P_t and P_r are the tangential and radial components of P_q at the crank pin. Therefore,

$$P_t = P_q \sin(\theta + \emptyset) = 24652.69 \sin(25 + 5.39) = 12471.38 \text{ N}$$

$$P_r = P_q \cos(\theta + \emptyset) = 24652.69 \cos(25 + 5.39) = 21265.46 \text{ N}$$

Step II: Bearing reactions

The forces acting on the centre crankshaft at an angle of maximum torque. The crankshaft is supported on three bearings 1, 2 and 3. As decided in the previous cage,

$b = 250$ mm And $c = 300$ mm

$$b_1 = b_2 = \frac{b}{2} = \frac{250}{2} = 125 \text{ mm}$$

$$c_1 = c_2 = \frac{c}{2} = \frac{300}{2} = 150 \text{ mm}$$

By symmetry,

$$(R_1)_v = (R_2)_v = \left[\frac{P_r}{2}\right] = \left[\frac{21265.46}{2}\right] = 10632.73 \text{ N}$$

$$(R_1)_h = (R_2)_h = \left[\frac{P_t}{2}\right] = \left[\frac{12471.38}{2}\right] = 6235.69 \text{ N}$$

$$(R'_2)_v = (R'_3)_v = \left[\frac{W}{2}\right] = \left[\frac{1000}{2}\right] = 500 \text{ N}$$

$$(R'_2)_h = (R'_3)_h = \left[\frac{P_1 + P_2}{2}\right] = \left[\frac{2000}{2}\right] = 1000 \text{ N}$$

Step III: Design of crank pin

The central plane of the crank pin is subjected to the bending moment M_b due to $(R_1)_v$ and torsional moment M_t due to$(R_1)_h$. In absence of data, the allowable shear stress is taken as 40 N/mm^2.
The diameter of crank pin $(dc)$is calculated by

$$d_c^3 = \frac{16}{\pi\tau}\sqrt{(Mb)^2 + (Mt)^2} = \frac{16}{\pi\tau}\sqrt{[(R_1)_v *\ b_1]^2 + [(R_1)_h * r]^2}$$

$= \frac{16}{\pi(40)}\sqrt{[(10632.73 * 125)]^2 + [(6235.69 * 75)]^2} = 179.4*10^3$

d_c= 56.4 mm

In calculation of previous case, the diameter (dc) is 65 mm. and length of crank pin (Lc) is 65 mm. since the diameter is more, the first case is criterion of deciding the diameter of the crank pin. Therefore,

$d_c=l_c$=65 mm

Step IV: The forces acting on the shaft under the flywheel.

Suppose,

d_s = diameter of the shaft under flywheel (mm).

The central plane of shaft is subjected to maximum bending moment due to the Reaction R_3.

$M_b=(R3)*C_2$

It is also subjected to torsional moment M_t due to tangential component P_r

$M_t= P_t* r$

$R_3= \sqrt{[(R'_3)_v]^2 + [(R'_3)_h]^2} = \sqrt{(500)^2 + (1000)^2} = 1118.03$ N

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

$d_s^3 = \frac{16}{\pi\tau}\sqrt{(\mathrm{Mb})^2 + (Mt)^2} = \frac{16}{\pi\tau}\sqrt{[R3 * \mathrm{C2}] + [Pt * r]}$

$= \frac{16}{\pi(40)}\sqrt{(1118.03 * 150)^2 + (12471.38 * 75)^2} = 120.99*10^3$

d_s =49.46 mm *or* 50 mm

In calculation of previous case, the diameter (ds) is 30 mm. since the diameter is less, the second case is the criterion of deciding the diameter of the shaft under flywheel. Therefore,

d_s = 50 mm.

Step V: Design of the shaft at the juncture of the right hand crank web

Suppose,

d_{s1}= diameter of the shaft at the juncture of the right-hand crank web (mm)

The cross-section of the shaft under flywheel at the juncture of the right-hand crank web is subjected to the following moments:

(i) Bending moment in vertical plane $(Mb)_v$ due to forces in vertical plane, viz., $(R_1)_v$ And P_r.

(ii) The bending moment in horizontal plane, viz., $(R_1)_h$ *and* P_t

(iii)Torsional moment M_t due to tangential component P_t

$$(M_b)_v = (R_1)_v\left[b_1 + \frac{l_c}{2} + \frac{t}{2}\right] - P_r\left[\frac{l_c}{2} + \frac{t}{2}\right]$$

$$= (10632.73)\left[125 + \frac{65}{2} + \frac{46}{2}\right] - (21265.46)\left[\frac{65}{2} + \frac{46}{2}\right] = 738.97 \times 10^3 \text{ N} - \text{mm}$$

$$(M_b)_h = (R_1)_h\left[b_1 + \frac{l_c}{2} + \frac{t}{2}\right] - P_t\left[\frac{l_c}{2} + \frac{t}{2}\right]$$

ISSN 2395-695X (Print)
ISSN 2395-695X (Online)

International Journal of Advanced Research in Biology Engineering Science and Technology (IJARBEST)
Vol. 2, Issue 4, April 2016

$$= (6235.69)\left[125 + \frac{65}{2} + \frac{46}{2}\right] - (12471.38)\left[\frac{65}{2} + \frac{46}{2}\right] = 433.38 \times 10^3 \text{ N} - \text{mm}$$

The resultant bending moment M_b is given by,

$$M_b = \sqrt{[(M_b)_v]^2 + [(M_b)_h]^2} = \sqrt{[738.97 \times 10^3]^2 + [433.38 \times 10^3]^2}$$
$$= 856.68 \times 10^3 \text{ N} - \text{mm}$$

The diameter of the shaft d_{s1} is calculated by the following expression:

$$d_{s1}^3 = \frac{16}{\pi\tau}\sqrt{(M_b)^2 + (M_t)^2} = \frac{16}{\pi(40)}\sqrt{(856.68 \times 10^3)^2 + (935.36 \times 10^3)^2} = 161.5 \times 10^3$$

$$d_{s1} = 54.46 \text{ or } 55 \text{ mm}$$

Step VI: Design of right-hand crank web

The right-hand crank web is subjected to the following stresses:

(i) Bending stresses in vertical and horizontal planes due to radial component P_r and tangential component P_t respectively.
(ii) Direct compressive stress due to radial component P_r.
(iii) Torsional shear stresses.

The bending moment due to the radial component is given by,

$$(M_b)_r = (R_2)_v\left[b_2 - \frac{l_c}{2} - \frac{t}{2}\right] = (10632.73)\left[125 - \frac{65}{2} - \frac{46}{2}\right] = 738.97 \times 10^3 \text{ N} - \text{mm}$$

Also,

$$(M_b)_r = (\sigma_b)_r\left[\frac{1}{6}wt^2\right] 738.97 \times 10^3 = (\sigma_b)_r\left[\frac{1}{6}(75)(46)^2\right]$$

$$(\sigma_b)_r = 27.94 \text{ N/mm}^2$$

The bending moment due to tangential component at the juncture of the crank web and shaft is given by,

$$(M_b)_t = P_t\left[r - \frac{d_{s1}}{2}\right] = (12471.38)\left[75 - \frac{55}{2}\right] = 592.39 \times 10^3 \text{ N} - \text{mm}^2$$

Also,

$$(M_b)_t = (\sigma_b)_t\left[\frac{1}{6}tw^2\right]$$

Or $529.39 \times 10^3 = (\sigma_b)_t\left[\frac{1}{6}(46)(75)^2\right]$

$$(\sigma_b)_t = 13.74 \text{ N/mm}^2$$

The direct compressive stress due to radial component is given by,

$$(\sigma_c)_d = \frac{P_r}{2wt} = \frac{21265.46}{2(75)(46)} = 3.08\frac{\text{N}}{\text{mm}^2}$$

The maximum compressive stress (σ_c) is given by,

$$\sigma_c = (\sigma_b)_r + (\sigma_b)_t + (\sigma_c)_d = 27.94 + 13.74 + 3.08 = 44.76 \text{ N/mm}^2$$

The torsional moment on the arm is given by,

$$M_t = (R_2)_h\left[b_2 - \frac{l_c}{2}\right] = (6235.69)\left[125 - \frac{65}{2}\right] = 576 \times 10^3 \text{ N} - \text{mm}$$

$$\tau = \frac{M_t}{Z_p} = \frac{4.5M_t}{wt^2} = \frac{4.5(576.80 \times 10^3)}{(75)(46)^2} = 16.36 \text{ N/mm}^2$$

The maximum compressive stress is given by,

$$(\sigma_c)_{max} = \frac{1}{2}\left[\sigma_c + \sqrt{(\sigma_c)^2 + 4\tau^2}\right] = \frac{1}{2}\left[44.76 + \sqrt{(44.76)^2 + 4(16.36)^2}\right] = 50.10\ N/mm^2$$

The above value of $(\sigma_c)_{max}$ is less than the allowable compressive stress $(75\ \mathrm{N/mm^2})$ and the design is safe.

Step VII: Design of left-hand crank web

The left-hand crank web is not severely stressed to the extent of the right-hand crank web. Therefore, it is not necessary to check the stresses in the left-hand crank web. The thickness and width of the left-hand crank web are made equal to that of the right-hand crank web from balancing consideration.

Step VIII: Design of crankshaft bearing

Bearing 2 is subjected to maximum stress. The reaction at this bearing is given by

$$R_2 = \sqrt{[(R_2)_v + (R'_2)_v]^2 + [(R_2)_h + (R'_2)_h]^2}$$

$$= \sqrt{[10632.73 + 500]^2 + [6235.69 + 1000]^2} = 13277.53\ \mathrm{N}$$

The diameter of the journal at the bearing 2 is (d_{s1}). i.e.$d_{s1} = 55$ mm

The (l/d) ratio for bearing is assumed as 1.

$l_2 = d_{s1} = 55$ mm

The bearing pressure is given by, $p_b = \frac{R_2}{d_{s1} l_2} = \frac{13277.53}{(55)(55)} = 4.39 \frac{\mathrm{N}}{\mathrm{mm}^2}$

$$\therefore \quad p_b < 10 \frac{\mathrm{N}}{\mathrm{mm}^2}$$

The bearing pressure is within the limits and design is safe.

TABLE 1. SPECIFICATION OF CRANKSHAFT

S.no.	Parameters (mm)
1	Crank pin diameter (d_c) = 65
2	Crank pin length (l_c) = 65
3	Width of Crank web (W) = 75
4	Thickness of Crank web (t) = 46
5	Diameter of shaft under flywheel (d_s) = 50
6	Diameter of shaft at the juncture of the right hand crank web (d_{s1}) = 55
7	The diameter of the journal at the bearing 2 (d_{s1}) = 55

III. METHODOLOGY

For the analysis of the crankshaft we have designed our model in catia V5 and then save it as IGES format for exporting the part into ANSYS 16.0 Workbench environment.

ISSN 2395-695X (Print)
ISSN 2395-695X (Online)

International Journal of Advanced Research in Biology Engineering Science and Technology (IJARBEST)
Vol. 2, Issue 4, April 2016

A. Meshing

Catia and ANSYS workbench software are used for the Finite Element Analysis of the crankshaft. At first the crankshaft is designed in the Catia software and then the file is saved as IGES format and imported in the ANSYS workbench software. The next step was to mesh the model as shown in the fig 7, the 10 node tetrahedral element are used as shown in the fig 4.The finite element was generated using the tetrahedral element of size 1mm.We have divided the part into 50833 element. The reason for choosing this huge number of element was to make our part very complex which enable us to gain more authentic results based on the high technique of fatigue life calculation.[8]

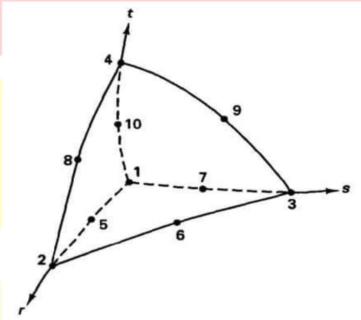
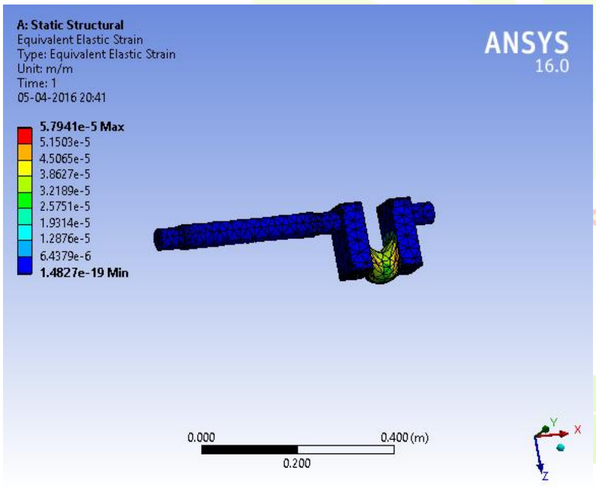


Fig 4: Meshing type: tetrahedral

IV. ANALYSIS OF CRANKSHAFT



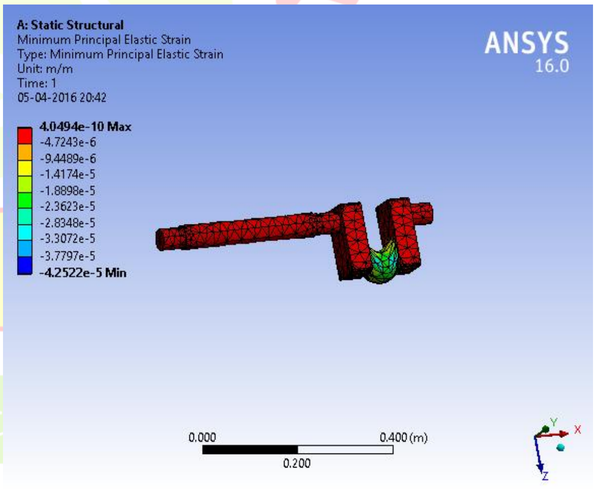


Fig. 5. Equivalent Elastic Strain

Fig. 6. Minimum Principal Elastic Strain

ISSN 2395-695X (Print)
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International Journal of Advanced Research in Biology Engineering Science and Technology (IJARBEST)
Vol. 2, Issue 4, April 2016

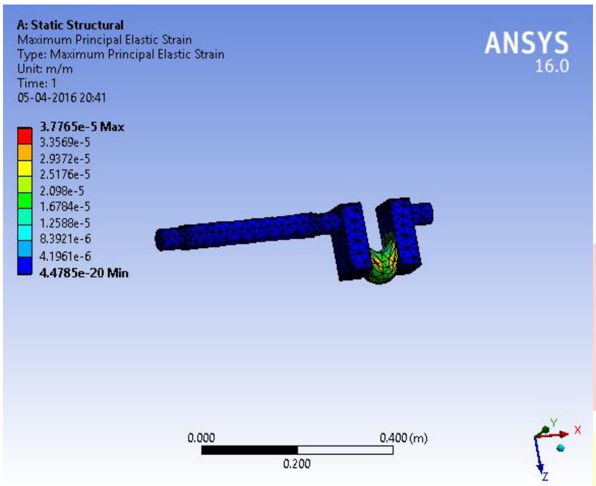

Fig. 7. Maximum Principal Elastic Strain

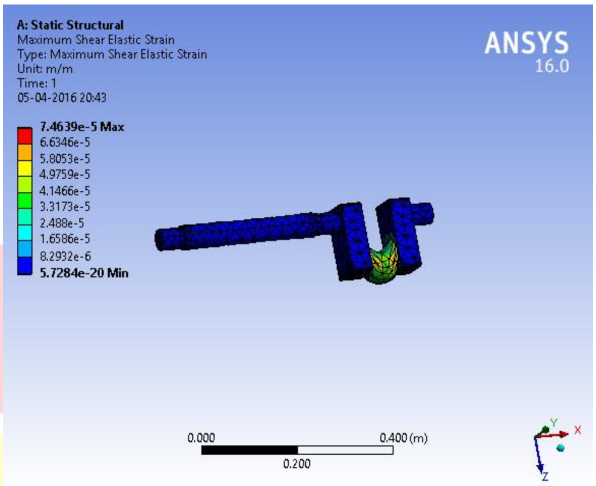

Fig. 8. Maximum Shear Elastic Strain

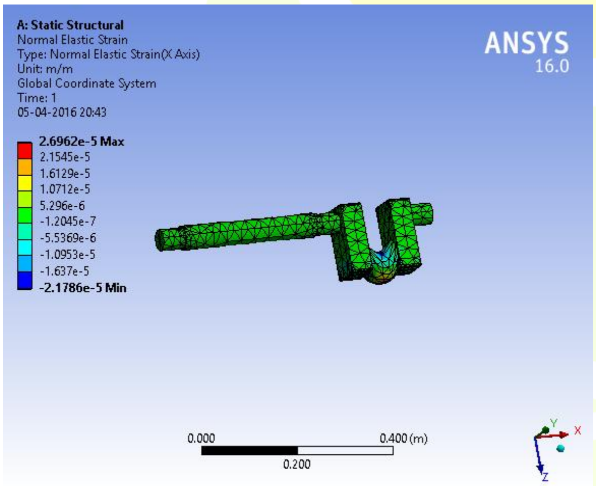

Fig. 9. Normal Elastic Strain (X-Axis)

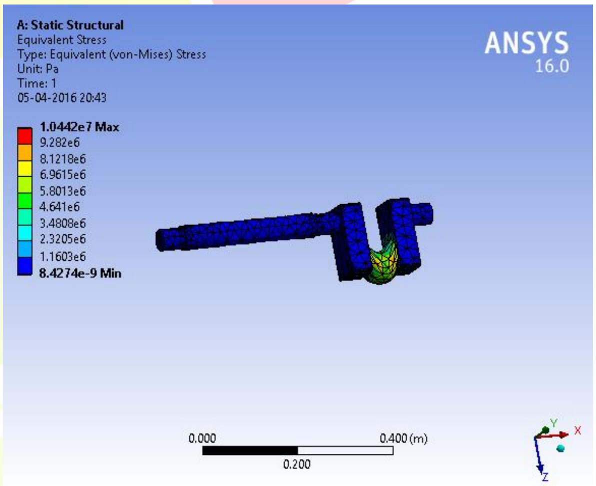

Fig. 10. Equivalent (Von-Mises) Stress

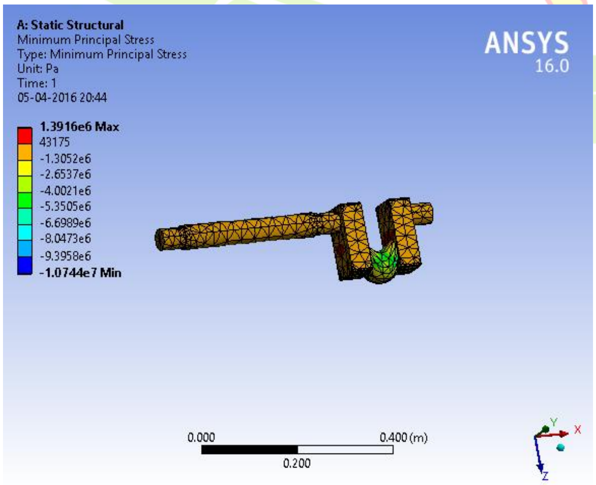

Fig. 11. Minimum Principal Stress

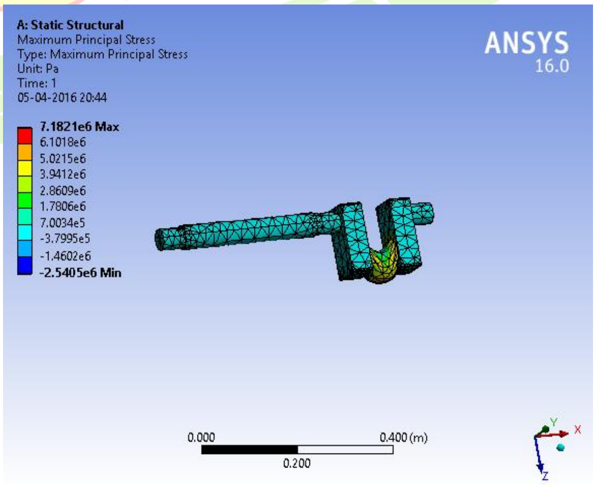

Fig. 12. Maximum Principal Stress

ISSN 2395-695X (Print)
ISSN 2395-695X (Online)

International Journal of Advanced Research in Biology Engineering Science and Technology (IJARBEST)
Vol. 2, Issue 4, April 2016

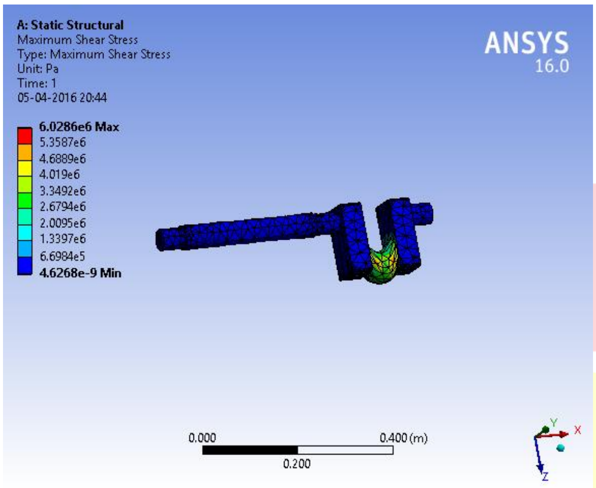

Fig. 13. Maximum Shear Stress

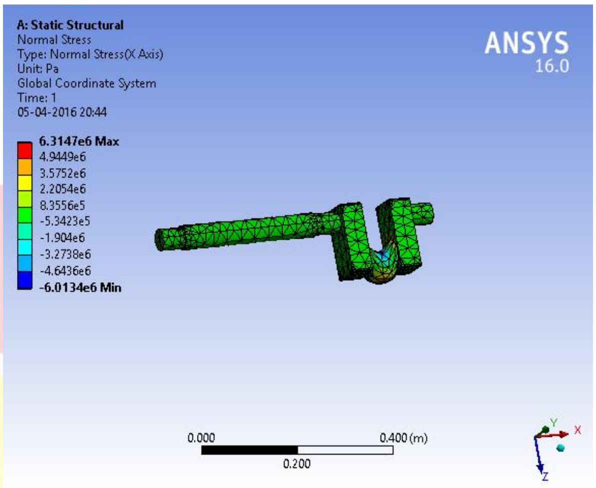

Fig. 14. Normal Stress (X-Axis)

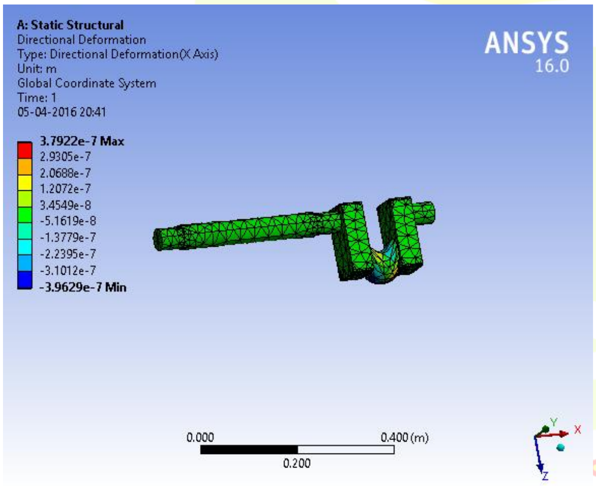

Fig. 15. Directional Deformation (X-Axis)

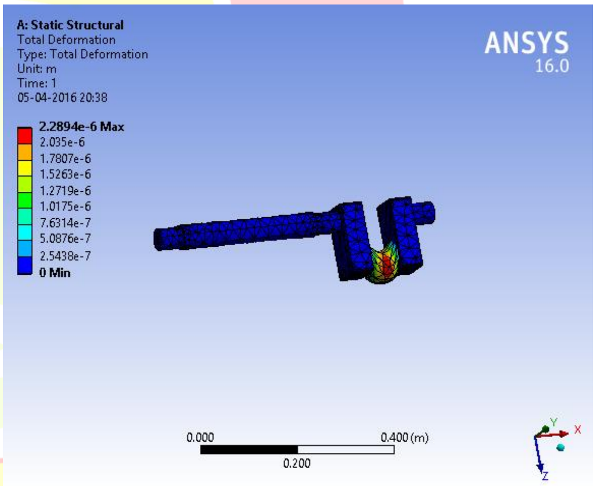

Fig. 16. Total Deformation

TABLE 2: RESULT

Parameter	Maximum	Minimum
Equivalent Elastic Strain (m)	5.7941e-5	1.4827e-19
Minimum Principal Elastic Strain (m)	4.0494e-10	-4.2522e-5
Maximum Principal Elastic Strain (m)	3.7765e-5	4.4785e-20
Maximum Shear Elastic Strain (m)	7.4639e-5	5.7284e-20
Normal Elastic Strain (X-Axis) (m)	2.6962e-5	-2.1786e-5
Equivalent (Von-Mises) Stress (Pa)	1.0442e7	8.4274e-9
Minimum Principal Stress (Pa)	1.3916e6	-1.0744e7
Maximum Principal Stress (Pa)	7.1821e6	-2.540e6

Maximum Shear Stress (Pa)	6.0286e6	4.6268e-9
Normal Stress (X-Axis) (Pa)	6.3146e7	-6.0134e6
Directional Deformation (X-Axis) (m)	3.7922e-7	-3.9629e-7
Total Deformation (m)	2.2894e-6	0

V. CONCLUSION

The Von-Misses stress values for Crankshaft made of Maraging V250 is 1.0442e7 Pa. (max) and 8.4274e-9 Pa. (min). The total maximum deformation is 2.2894e-6 mm and minimum is 0 mm.

From the above results, it can be observed that a light weight and considerably high strength maraging V250 Steel can be used as Crankshaft material.

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