IJRAME
ISSN (ONLINE): 2321-3051
INTERNATIONAL JOURNAL OF RESEARCH IN AERONAUTICAL AND MECHANICAL ENGINEERING

Design and Contact Analysis of Crankshaft Using Abaqus

V.Vijayakumar[1], T.Gopalakrishnan[2], Dr.R.Vivekananthan[3]

[1]*P.G Student, Dept. Of Mech. Engg., Govt. College of Engineering, Salem, India,* vsuvijayakumar@gmail.com
[2]*P.G Student, Dept. Of Mech. Engg., Govt. College of Engineering, Salem, India,* vsuvijayakumar@gmail.com
[3]*Asst. professor, Dept. Of Mech. Engg., Govt. College of Engineering, Salem, India,* rvivekapme@gmail.com

Abstract

The design and analysis of a single cylinder crankshaft assembly are analyzed using FEA commercial software using ABAQUS. Three-dimension models of connecting rod and crank shaft were modeled using Pro/ENGINEER software and analyzed using ABAQUS to find the critical stress status and maximum design load for the crank pin. The maximum deformation, maximum stress point is found by creating contact surface between connecting rod and crank pin using contact stress analysis. The validation of contact analysis of crank shaft is compared with hand calculation and the results are found satisfactory.

***Keywords**:* Crankshaft; Connecting rod; Contact analysis.

1. Introduction

Crankshaft of Internal Combustion Engine is a well-known phenomenon. The problem of their premature failure has attracted several investigators for over a century. The extensive studies have been made to identify the cause of failure and several have been listed. Forces acting on the crankpin are complex in nature. The piston and the connecting rod transmit gas pressure from the cylinder to the crankpin. Crankshaft consists of the parts which revolve in the main bearings, the crankpin to which the big ends of the connecting rod is connected,. The crankpin is like a build in beam with a distributed load along its length that varies with crank position. Reasons for Failure of crankshaft assembly and crankpin may be

a. Shaft misalignment
b. Vibration cause by bearings application
c. Incorrect geometry(stress concentration)
d. Improper lubrication
e. High engine temperature
f. Overloading
g. Crankpin material & its chemical composition
h. Pressure acting on piston

Vol.2 Issue.3,
March 2014.
Pgs: 57-63

Results from the FE model are then presented which includes identification of the critically stressed location, variation of stresses over an entire cycle, and a discussion of the effects of engine speed as well as torsion load on stresses.

2. Design Calculation for Crankshaft

The configuration of the diesel engine for this crankshaft is tabulated in Table 1

Table: - 1

Specifications of Honda Engine

Capacity	395 cc
Number of Cylinders	1
Bore Stroke	86 68 mm
Compression Ratio	18:1
Maximum Power	8.1hp @ 3600 rpm
Maximum Torque	16.7 Nm @ 2200 rpm
Maximum gas pressure	25 bar

2.1 Design of Crankshaft When the Crank is at an Angle of Maximum Twisting Moment

Force on the piston

$$\begin{aligned} \mathrm{F} &= \text{Area of the bore } \times \text{Max. combustion pressure} \\ &= \frac{\pi}{4} \times \mathrm{D}^2 \times \\ &= 14.52 \text{ KN} \end{aligned} \tag{1}$$

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

$$\sin\emptyset = \frac{\sin\theta}{\left[\frac{L}{R}\right]} = \sin\frac{35^0}{4} \tag{2}$$

This implies $\emptyset = 8.24^0$

Thrust in the connecting rod

$$F_Q = \frac{F_P}{\cos\emptyset} \tag{3}$$

From this we have,

Thrust on the connecting rod F_Q= 14.67 KN

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

a. Tangential force on the crank shaft,
b. $F_T = F_Q \sin(\theta + \emptyset)$ =10.05 KN
c. Radial force on the crank shaft,
d. $F_R = F\cos(\theta + \emptyset)$ = 10.68 KN

Reactions at bearings due to tangential force is given by

$$\mathrm{HT1} = \mathrm{HT2} = \frac{\mathrm{F_T}}{\mathrm{b}}\mathrm{b}_1 = 5.02\mathrm{KN}(\mathrm{b1} = \mathrm{b2} = \frac{\mathrm{b}}{2}) \tag{4}$$

Similarly, reactions at bearings due to radial force is given by,

$$\mathrm{HR1} = \mathrm{HR2} = \frac{\mathrm{F_R}}{\mathrm{b}}\mathrm{b}_1 = 5.34\mathrm{KN}(\mathrm{b1} = \mathrm{b2} = \frac{\mathrm{b}}{2}) \tag{5}$$

2.2 Design of Crankpin

Let dc = diameter of crankpin in mm

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

$$\begin{aligned} M_C &= H_{R1} \times b_2 \\ &= 5.34 \times 86 \\ &= 459.24\ KN - mm \end{aligned} \tag{6}$$

Twisting moment on the crankpin $= 170.68\ KN - mm$

From this we have the equivalent twisting moment

$$T_e = \sqrt{M_C^2 + T_C^2} = 489.93\ KN - mm$$

We know that equivalent twisting moment

$$T_e = \frac{\pi}{16}(d_c^3) \times \tau \qquad (7)$$

Shear stress value is limited to 35 N/mm^2

so d_c = 47.47 mm

Since this value of crankpin diameter
(d_c = 47.47 mm) is less than the when the crank is at top dead centre already calculated value of crankpin dia. (dc= 48 mm)
therefore, we shall take, dc=48 mm

Result:

Diameter of the crank pin =48mm
Design of crank pin against fatigue loading According to distortion energy theory
The von Mises stress induced in the crank-pin is,

$$M_{ev} = \sqrt{(K_b \times K_C) + \frac{1}{2}(K_t \times T_C)} \qquad (8)$$

$$= 880.3\ \text{KN-mm}$$

Here, Kb = combined shock and fatigue factor for bending (Take Kb=2)
Kt = combined shock and fatigue factor for torsion (Take K t =1.5)

$$M_{ev} = \frac{\pi}{32} \times d_c^3 \times 6v \qquad (9)$$

6v = 80.4 N/mm2 and also calculated shear stress on the shaft = 32 N/mm^2

Results:-

Diameter of the crankpin = 48 mm
Length of the crankpin = 40 mm
Diameter of the shaft = 30 mm
Web thickness (both left and right hand)
t = 23 mm
Web width = 63 mm

2.2 Design calculations

Assumption: Let us assume, Mechanical efficiency = 0.80
BP=Brake power (KW) i.e. Maximum power
IP= Indicated power in KW
IP = 60/0.8 = 75KW

$$IP = P(i) \times L \times A \times n \times K \div 60000 \qquad (10)$$

Where,
Pi=Indicated mean effective pressure
L=Stroke length
D=Bore diameter
K= No. of cylinders
n = N/2 for 4-stroke
N=Speed in rpm

$$Pi = L \times A \times n \times K \div 60000 \times IP \qquad (11)$$

$$Pi = 0.93\ \text{Mpa}$$

At the TDC of the piston, the volume will be reduced by the compression. At this moment, the maximum pressure inside the cylinder will be,

Max. Pressure = B.M.E.P × Compression ratio = 0.93 × 18.5= 17.205 MPa

Now, this value of B.M.E.P acts on the piston head, and the whole force is transmitted to the crankpin through the connecting rod. This force is the most critical in the design of the crankshaft and the design in done on the basis of the above mentioned force.
To find the force exerted on the crankpin by the piston:

Piston force, F (kN) = cylinder bore area (mm^2) × B.M.E.P F = 116.87 kN

Piston force will act at the middle of the crankpin, and it will be balanced by the reactions from the bearings at either side of the crankpin. Let the reactions be R_1 and R_2.
Considering the crankpin as a simply supported beam, we will see that
$R_1 + R_2 = F$ and $R_1 = R_2$
Therefore, $R_1 = R_2 = F \div 2 = 58.435$ kN
Maximum bending moment (M) on the crank pin is given by $M = R1 \times b$ (12)
Where, b is the distance from the centre of the bearing to the centre of connecting rod
Assuming $b = 1.2 \times D = 111.6$ i.e.
$b = 112$mm.
Also, we know that
From the above equation, we get that
Where,
d = diameter of the crankpin = max. bending stress of the material of the crankshaft with suitable factor of safety (350Mpa)
Equating the values of M in the above equations, we can get the value of the crankpin d=58mm
Length of the crankpin (L_c) $= F\times D \div p$
Where, P = maximum permissible stress on the bearing, 50MPa,
$L_c = 116870 \div 58 \times 60$
Crank web thickness is given by 0.25D,
i.e. $= 0.25 \times 93 = 23.25$ mm
Torque, Ts $= 27.822 \times 0.045 = 1252$ Nm.
Now, we know that to design the diameter for a given torque, we use
d = 36.45mm
Where is the maximum shear stress acting, which is taken as 175 MPa.
So, as per the above equation, the diameter of the main journal is obtained as 36.45 mm, which is much lesser than 5 mm, i.e. the design is safe and we can use the main journal diameter as 69.75 mm.

2.3 Dimensions calculated

Table: - 2
Crankshaft dimensions

Requireddesign parameter	Dimension
Crank pin diameter (d)	58mm
Length of crank pin (Lc)	40mm

3. Methodology

3.1 Procedure of contact Analysis
1) First, I have Prepared Assembly in Solid works for crankshaft and Save as this part as IGES for Exporting into HYPER MESH Workbench Environment. Import .IGES Model in ABAQUS Workbench Simulation Module.
2) Apply Material for Crank Shaft (Forged steel).
Material Details
Material Type: - Forged Steel
Designation: - 42CrMo4
Yield strength (MPa):- 680
Ultimate tensile strength (MPa):- 850
Elongation (%):-13
Poisson ratio:-0.3

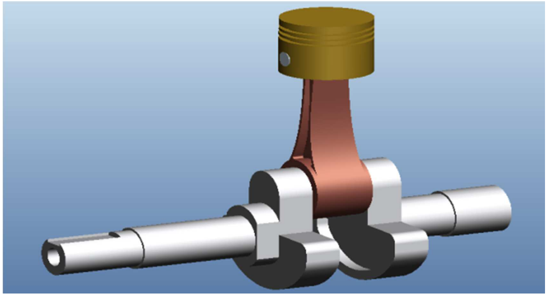

Figure 3.1: Crank Shaft assembly

3) Mesh the Crankshaft.
Mesh Statics:
Type of Element: Tetrahedrons
Number of Nodes : 17119
Number of Elements : 9605

Meshed model of Crank shaft

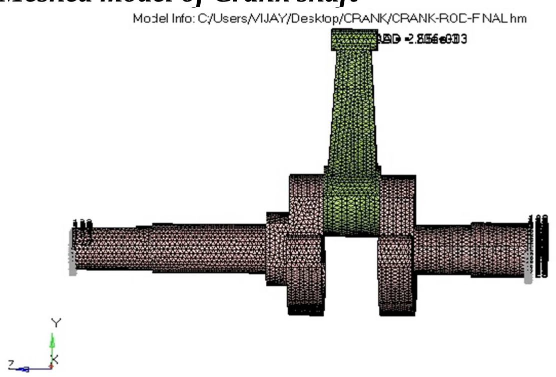

Figure 3.2 Meshed model of Crank shaft

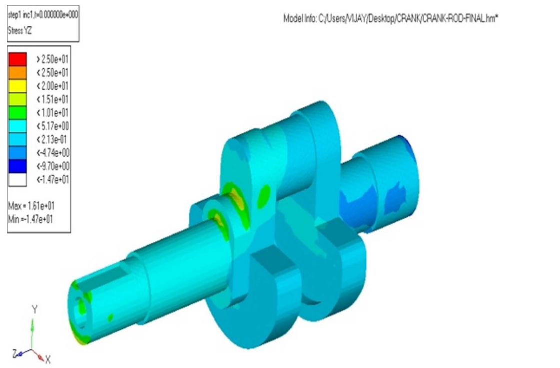

Figure 3.3: Apply Remote Displacement for Bearing Support (Which has only one degree of freedom (Rotational))

4) Define boundary condition for Analysis Boundary conditions play an important role in finite element calculation here; I have taken both remote displacements for bearing supports are fixed.

5) Define type of Analysis
Type of Analysis:-Static Structural

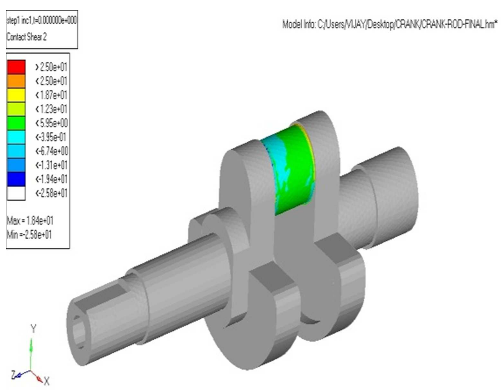

Figure 3.4: contact shear model

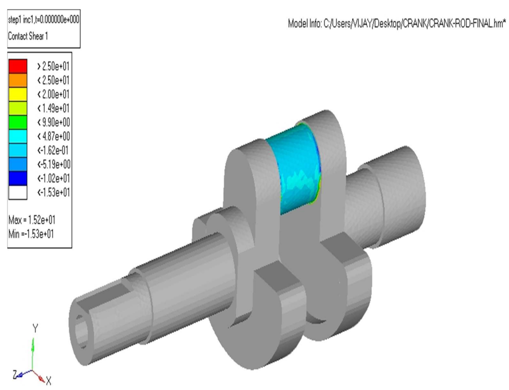

Figure 3.6 Maximum shear stress

6) Run the Analysis
7) Get the Results

Output of the Analysis

contact Analysis of Crankshaft

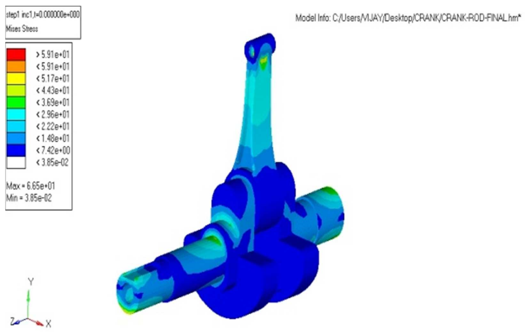

Figure 3.5 Von misses stress

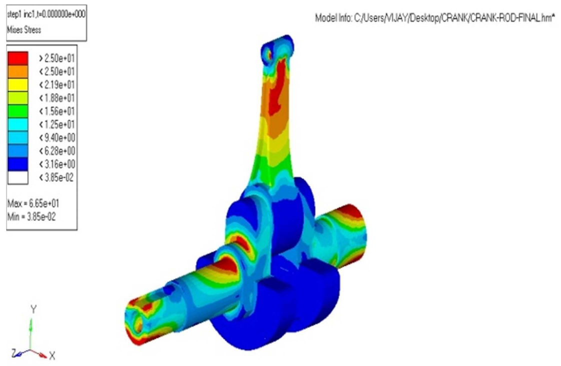

Figure 3.7 Deformation at a phase angle of 355

4. Result

• In this paper, the crankshaft model was created by PRO-E software. Then, the model created by Solid works was imported to ABAQUS software.

Result Table:-3

Sr.no	Types of stress	Theoretical	FEA analysis
1	Von-Misses Stresses (N/mm2)	80.4	66
2	Shear Stresses (N/mm2)	32	18.4

• 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.

Conclusion

Above Results Shows that FEA analysis results matches with the theoretical calculation and hence FEA analysis is used to predicts the stress distribution in the crank shaft. 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 fatigue analysis in order to determine the life cycle of the crank shaft and further optimization to reduce the material and cost in the future.

References

1.C.M.Balamurugan.,R.Krishnaraj.,Dr.M.Sakthive.l,K.Kanthave.l,DeepanMarudachalamM.G.,R.Palani.,2011“ Computer Aided Modeling and Optimization of Crankshaft, ’’International Journal of Scientific & Engineering Research Volume 2, Issue 8 ISSN 2229-5518

2. Yu Ding and Xiaobo Li.,2011, “ Crankshaft Strength Analysis of a Diesel Engine Using Finite Element Method,” Asia-Pacific Power and Energy Engineering Conference

3. Jian Meng., Yongqi Liu., Ruixiang Liu.,2011,"Finite Element Analysis of 4-Cylinder Diesel Crankshaft, " I.J. Image, Graphics and Signal Processing, 5, 22-29
4. MENG Jian., LIU Yong-qi., LIU Rui-xiang., and ZHENG Bin.,2011,"Intension Analysis of 3D Finite ElementAnalysis on 380 diesel crankshaft," International
Conference on Computational and Information Sciences
5. Yu Gongzh.i, Yu Hongliang., Duan Shulin., 2011, "Crankshaft Dynamic Strength Analysis for Marine Diesel Engine," Third International Conference on Measuring Technology and Mechatronics Automation.
6. Gu Yingkui, Zhou Zhibo.,2011,"Strength Analysis of Diesel Engine Crankshaft Based on PRO/E and ANSYS," Third International Conference on Measuring Technology and Mechatronics Automation
7. Xiaorong Zhou., Ganwei Cai., Zhuan Zhang. Zhongqing Cheng., 2009, "Analysis on Dynamic Characteristics of Internal Combustion Engine Crankshaft System," International Conference on Measuring Technology and Mechatronics Automation.