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Design and Stress Analysis of Crankshaft for Single Cylinder 4-Stroke Diesel Engine

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Abstract*— Crankshaft is one of the critical components for the effective and precise working of the internal combustion engine. In this paper a static simulation is conducted on a crankshaft from a single cylinder 4- stroke diesel engine. A three-dimension model of diesel engine crankshaft is created using Pro-E 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 is calculated using theoretically and FEA software ANSYS. The relationship between the frequency and the vibration modal is explained by the modal and harmonic analysis of crankshaft using FEA software ANSYS.

***Keywords*—- Diesel engine, Crankshaft, Finite Element Analysis, Stress analysis**

I. INTRODUCTION

Crankshaft is one of the most important moving parts in internal combustion engine. Crankshaft is a large component with a complex geometry in the engine, which converts the reciprocating displacement of the piston into a rotary motion. This study was conducted on a single cylinder 4- stroke diesel engine. It must be strong enough to take the downward force during power stroke without excessive bending. So the reliability and life of internal combustion engine depend on the strength of the crankshaft largely. And as the engine runs, the power impulses hit the crankshaft in one place and then another. The torsional vibration appears when a power impulse hits a crankpin toward the front of the engine and the power stroke ends. If not controlled, it can break the crankshaft. Jian Meng et al. [3] analyzed crankshaft model and crank throw were created by Pro/ENGINEER software and then imported to ANSYS software. The crankshaft deformation was mainly bending deformation under the lower frequency. And the maximum deformation was located at the link between main bearing journal, crankpin and crank cheeks. Gu Yingkui Et Al. [6] researched a three-dimensional model of a diesel engine crankshaft was established by using PRO/E software. Using ANSYS analysis tool, it shows that the high stress region mainly concentrates in the knuckles of the crank arm & the main journal and the crank arm & connecting rod journal ,which is the area most easily broken.

Xiaorong Zhou et al. [7] described the stress concentration in static analysis of the crankshaft model. The stress concentration is mainly occurred in the fillet of spindle neck and the stress of the crankpin fillet is also relatively large. Based on the stress analysis, calculating the fatigue strength of the crankshaft will be able to achieve the design requirements. From the natural frequencies values, it is known that the chance of crankshaft resonant is unlike. This paper deals with the dynamic analysis of the whole crankshaft. Farzin H. Montazersadgh et al. [8] investigated first dynamic load analysis of the crankshaft. 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.

II. DESIGN CALCULATION FOR CRANKSHAFT

The specification of diesel engine for crankshaft is TABULATED below:

Table 1: Specification of engine

Type	**Single Cylinder Diesel engine**
No of cylinders	1
Bore/Stroke	86 mm/ 68 mm
Capacity	395 cc
Compression Ratio	18 : 1
Max. Power	8.1 HP @ 3600rpm

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Max. Torque	16.7 Nm@ 2200rpm
Max. Gas Pressure	25 bar

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

At this position of the crank, the maximum gas 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.

Let, *D* = Piston diameter or cylinder bore in mm, *p* = Maximum intensity of pressure on the piston in N/mm2

The thrust in the connecting rod will be equal to the gas load on the piston (F_p). We know that gas load on the piston,

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

$$F_p = 14.52 \text{ kN}$$

Distance between two bearings is given by,

$$b = 2D = 2 \times 86 = 172 \ mm$$

$$\therefore b_1 = b_2 = \frac{b}{2} = 86 \ mm$$

Due to this piston gas load (*FP*) acting horizontally, there will be two horizontal reactions *H*1 and *H*2 at bearings 1 and 2 respectively, such that

$$H_1 = H_2 = \frac{F_p}{2} = 7.26 \ kN$$

B. Design of Crankpin

Let, d_c =Diameter of crankpin

l_c = Length of crankpin

σ_b = Allowable bending stress for the crankpin $= 75 \ N/mm^2$

Bending moment at the centre of the crankpin,

$$M_b = H_1 \times b_2 = 624.36 \ kN \cdot mm$$

$$M_b = \frac{\pi}{32} \times d_c{}^3 \times \sigma_b$$

$$d_c = 44 \ mm$$

The length of the crankpin is given by

$$l_c = \frac{F_p}{d_c \times p_b}$$

Where, p_b = Permissible bearing pressure

$= 10 \ N/mm^2$ (Assuming)

$$l_c = 33 \ mm$$

C. Design of left hand Crank web

The crank web is designed for eccentric loading. There will be two stresses acting on the crank web, one is direct compressive stress and the other is bending stress due to piston gas load (F_p).

Let, w = Width of crank web

t = Thickness of crank web

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

$$w = 1.125 \ d_c + 12.7 \ mm$$

$$w = 63 \ mm$$

The thickness (*t*) of the crank web is given empirically as,

$$t = 0.28 \times D$$

$$t = 24 \ mm$$

We know that maximum bending moment on the crank web,

$$M = H_1 \left(b_2 - \frac{l_c}{2} - \frac{t}{2} \right) = 417450 \ N \cdot mm$$

And Section Modulus is,

$$Z = \frac{w \times t^2}{6} = 6048 \ mm$$

Bending stress bending stress induced in the crank web is,

$$\sigma_b = \frac{M}{Z} = 69 \ N/mm^2$$

Here, induced bending stress is less than the allowable bending stress which is ($\sigma_b = 75 \ N/mm^2$). Hence the design is safe.

D.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.

E.Design of shaft

Let, d_s = Diameter of shaft in mm.

We know that bending moment on shaft is,

$$BM = F_p \times c$$

Where, *c* =clearance = 30 *mm* (assuming)

$$BM = 435.6 \ kN \cdot mm$$

And twisting moment on shaft is,

$$TM = F_p \times r$$

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Where, $r =$ Offset of Crankpin

$$= \frac{stroke}{2} = 34\ mm$$

$$TM = 493.7\ kN \cdot mm$$

Equivalent moment on shaft is given by,

$$M_s = \sqrt{BM^2 + TM^2} = 658.4\ kN \cdot mm$$

Now, we know that

$$M_s = \frac{\pi}{32} \times d_s{}^3 \times \sigma_b$$

Hence, shaft diameter

$$d_s = 45\ mm$$

F. Design of crank pin against fatigue loading

According to distortion energy theory, the Von-Misses stress induced in the crank-pin is,

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

Where,

$K_b =$ combined shock and fatigue factor for bendin $= 2$ (Assume)
$K_t =$ combined shock and fatigue factor for torsion $= 1.5$ (Assume)

Putting the values in above equation we get

$$M_{ev} = 938.33\ kN \cdot mm$$

Also we know that,

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

$$938.33 \times 10^3 = \frac{\pi}{32} \times \sigma_v \times 44^3$$

$$\textbf{Von - Mises stress } \sigma_v = 121.15\ N/mm^2$$

Now,

$$T_{ev} = \sqrt{(K_b \times M_c)^2 + \ (K_t \times T_c)^2}$$

$$T_{ev} = 953.49\ kN \cdot mm$$

Also we know that,

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

$$953.49 \times 10^3 = \frac{\pi}{16} \times \tau \times 44^3$$

$$\textbf{shear stress } \tau = 57\ N/mm^2$$

RESULTS:

Diameter of crank pin – 44 mm
Length of crankpin – 33 mm
Width of crank web – 63 mm
Thickness of crank web – 24 mm
Diameter of shaft – 42 mm

III. DESIGN METHODOLOGY

A. Procedure of static Analysis

First, we prepare a model of crankshaft in Pro- E software and save as .IGES file format for Analysis of crankshaft in ANSYS WORKBENCH 14.5. Import .IGES model in ANSYS Workbench simulation module.

B.Applying material for crankshaft Material details:

Material type:- Cast steel

Poisson ratio:- 0.268

Yield strength:-620 Mpa

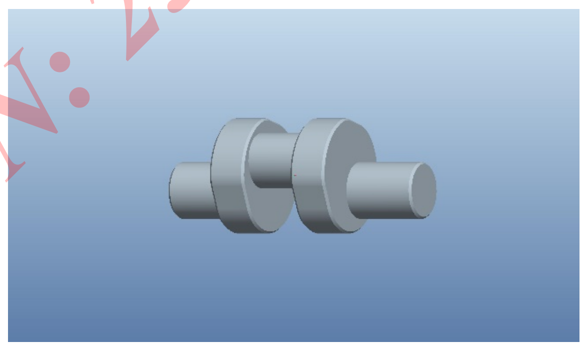

Fig. 3.1 Crankshaft in Ansys

C.Meshing of Crankshaft :

Mesh statics:
Type of element:- Tetrahedron10
Number of nodes:- 50934
Number of elements:- 29685

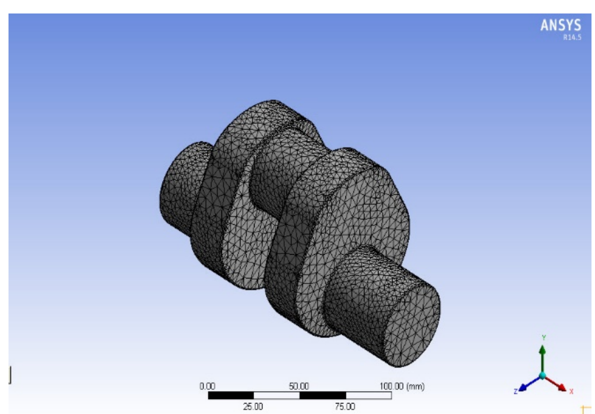

Fig. 3.2 Meshed Model of Crankshaft

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D. Define boundary condition for analysis:

Boundary conditions play an important role in Finite Element Analysis. Here we have taken both remote displacements for bearing supports are fixed.

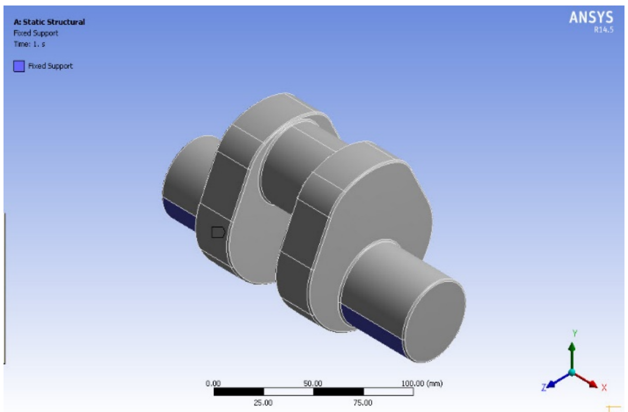

Fig. 3.3 Apply boundary condition

E.Define type of analysis:

Type of Analysis: Static structural

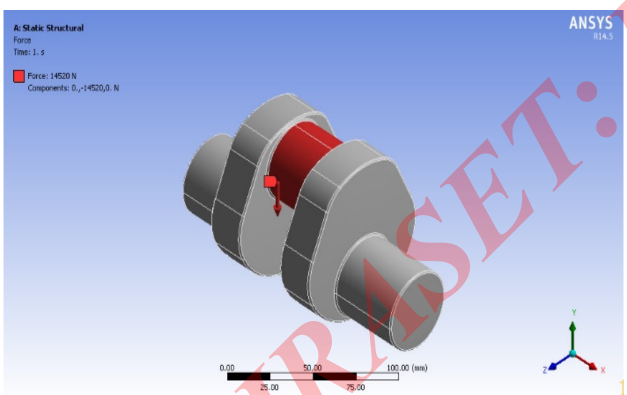

Fig. 3.4 Apply tangential force

F. Run the analysis and Get Results:

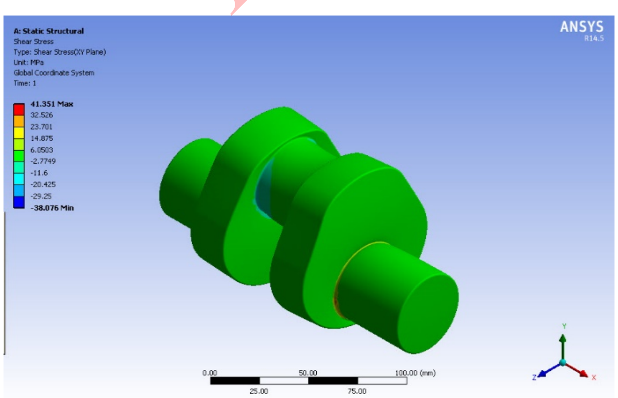

Fig. 3.5 Shear stress analysis

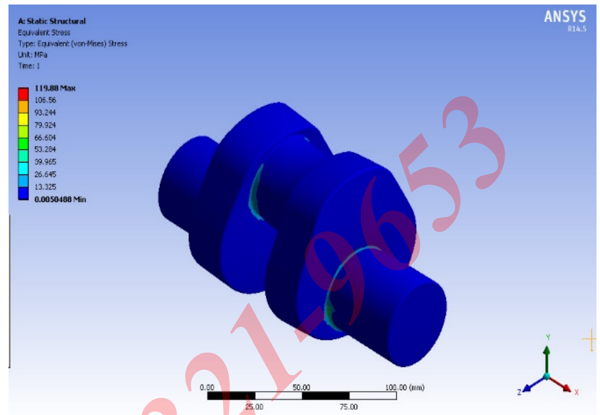

Fig. 3.6 Von-Misses Stress analysis

IV. RESULTS AND CONCLUSION

- In this paper, the crankshaft model was created by Pro-E software. Then, the model created by Pro-E was imported to ANSYS software.

Result table:-

Sr. No.	Types of stress	Theoretical	FEA Analysis
1	Von-Misses Stresses, N/mm^2	121.15	119.88
2	Shear Stresses, N/mm^2	57	41.35

- 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.
- After Performing Static Analysis I Performed 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.

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- Analysis Results. So we can Say that Dynamic FEA is a good tool to reduce Costly experimental work.

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REFERENCES

1. Jaimin Brahmbhatt Design And Analysis Of Crankshaft For Single Cylinder 4-Stroke Deisel Engine, International Journal of Advanced Engineering Research and Studies- E-ISSN2249–8974

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. 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.
5. Gu Yingkui, Zhou Zhibo.,2011,"Strength Analysis of Diesel Engine Crankshaft Based on PRO/E and ANSYS," Third International Conference on Measuring Technolog and Mechatronics Automation

6. Shenoy, P. S. and Fatemi, A., 2006, "Dynamic analysis of loads and stresses in connecting rods,"IMechE, Journal of Mechanical Engineering Science, Vol. 220, No. 5, pp. 615-624