DESIGN AND FAILURE ANALYSIS OF FOUR STROKE SIX CYLINDER DIESEL ENGINE CRANKSHAFT

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Abstract

Crankshaft is one of the critical components for the effective and precise working of the internal combustion engine. Study is done on the crankshaft and found that there are tremendous stress induced in crankshaft. The stress are calculated first analytically and then also by Finite element analysis (FEA). A static simulation is conducted on a four stroke six cylinder diesel engine crankshaft. A three-dimension model of engine crankshaft is created using CATIA-V5 software. FEA is performed in Hypermesh for preprocessing and FEMAP is used for processing. The load is applied to the FE model in FEMAP, and boundary conditions are applied according to the engine mounting conditions. In an arbitrary position of the crank, due to tangential force, the crank arm will be subjected to transverse shear, bending and twisting, while due to radial component it is subjected to direct stress and bending. It will be laborious to consider all these straining actions in several positions of the crank. Generally, the crank is designed for two positions; those are maximum twisting moment and maximum bending moment. 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-misses stress is calculated theoretically and FEA, and thus validated.

Index Terms: Crankshaft, Finite Element Analysis (FEA), Diesel engine, Software.

I. INTRODUCTION

The function of crankshaft is to convert the reciprocating displacement of piston into rotary motion. An automotive crankshaft is (fig. 1) consist of main journals, crankpins, oil holes, counter weights and a thrust bearing journals. Flywheel is attached at one side of the crankshaft to reduce the pulsation characteristics of four stroke cycle, and sometime vibrational or torsional damper at the opposite end, to reduce the torsional vibrations often caused along the length of the crankshaft. Crankshaft is the important component of the engine. Any damage occurring to the crankshaft may put the mechanical system out of order. Nowadays, manufacturers are compelled to ensure the downsizing concept to permit the increase of engines power and torque without increasing cylinder capacity in order to reduce fuel consumption [3]. Crankshaft 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.

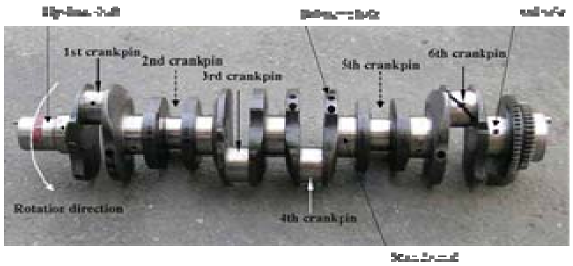
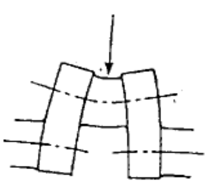


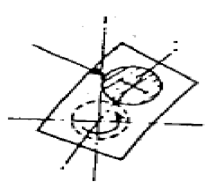
Fig.1: Automotive Crankshaft

II. STRESSES IN CRANKSHAFT

Crankshaft of an I. C. engine is a well-known phenomenon. Forces acting on the crankpin of crankshaft is complex in nature. The piston and the connecting rod transmit gas pressure from the cylinder to the crankpin. It also exerts forces on the crankpin, which is time varying. The crankpin is like a built in beam with a distributed load along its length that varies with crank position. Each web like a cantilever beam subjected to bending & twisting. Journals would be principally subjected to twisting. Fig.2 represent the stresses acting on the crankshaft [1].

1. Twisting causes shear stress.
2. Bending causes tensile and compressive stresses.
3. Due to shrinkage of the web onto the journals, compressive stresses are set up in the journals and tensile hoop stresses in the webs.





Bending of Pin Bending of Web and
Torsion of Journal

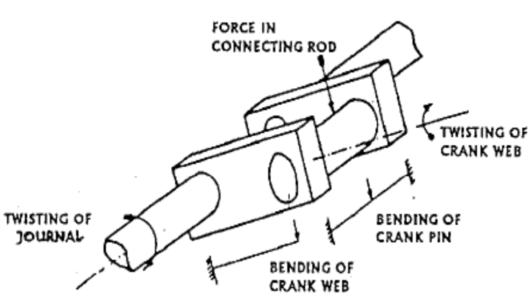


Fig. 2: Stresses in Crankshaft and their Effect on Crankshaft

Crankshaft consists of parts which revolves in the main bearings, crankpin to which the big ends of the connecting rod is connected, the crank arms or webs (also called cheeks) which connect the crankpins and the shaft parts [2]. Reasons for Failure of crankshaft assembly and crankpin may be –

1. Shaft misalignment
2. Incorrect geometry (stress concentration)
3. Vibration cause by bearings application
4. High engine temperature
5. Improper lubrication
6. Overloading
7. Pressure acting on piston
8. Crankpin material & its chemical composition

III. DESIGN CALCULATION FOR CRANKSHAFT

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

Table I: Specification of Six Cylinder Diesel Engine Crankshaft

Parameter	Value
No of cylinder	6 inline
Bore/Stroke	97mm × 128 mm
Capacity	5675 cc
Max Engine Output	96 kW @ 2400 rpm for
Max Torque	410 N m @ 1400-1700 rpm
Compression Ratio	17.5:1
Firing order	1-5-3-6-2-4
Maximum Gas Pressure	55.70

A. Crank is at dead Centre:

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.

Bore diameter (D) = 97 mm

F_Q = Area of the bore × Max. Combustion pressure

$= \frac{\pi}{4} \times D^2 \times P_{max} = 14.161$ kN

$H_1 = \frac{Fp \times b1}{b} = \frac{41.161 \times 97}{194} = 20.58 \text{ kN} = H_2$

[1] Design of Crankpin

Let d_c = Diameter of the crankpin in mm = 70 mm

$b = 2D = 2 \times 97 = 194$

$\therefore b_1 = b_2 = b/2 = 194/2 = 97$ mm

Bending moment at the centre of the crankpin

$M_c = H_1 \times b_2 = 20.58 \times 97 = 1996.26$ kN mm

We also know that

$$M_c = \frac{\pi}{32} \times (d_c)^3 \times \sigma_v$$

$$\sigma_v = 59.28 \text{ N/mm}^2 \text{ or MPa}$$

B. Crank is at an angle of maximum twisting moment Force on the piston:

The twisting moment on the crankshaft will be maximum when the tangential force on the crank (F_T) is maximum. In order to find the thrust in the connecting rod (F_Q), we should first find out the angle of inclination of the connecting rod with the line of stroke (i.e. angle $Ø = 35^0$).

We know that,

$$\sin \varnothing = \frac{\sin\theta}{l/r} = \frac{\sin 35}{5} = 0.1147$$

$$\varnothing = \sin^{-1}(0.1147) = 6.58°$$

We know that thrust in the connecting rod,

$$F_Q = \frac{F_P}{\cos\varnothing} = \frac{41.64}{\cos(6.58)} = 41.43 \text{ kN}$$

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

1) Tangential force on the crank shaft,

$F_T = F_Q \sin(\varnothing + \theta) = 41.43 \sin(6.58 + 35) = 27.49$ kN And

2) Radial force acting on the crankshaft,

$F_R = F_Q \cos(\varnothing + \theta) = 41.16 \cos(6.58 + 35) = 30.99$ kN

Reactions at bearings (1&2) due to tangential force is given by HT1=HT2 = FT/2= 13.745 kN

Similarly, reactions at bearings (1&2) due to radial force is given by HR1 = HR2= FR/2= 15.495kN

[1] Design of Crankpin.

We know that,

Bending moment at the centre of the crankshaft

$MC = HR1 \times b_2 = 15.495 \text{ X } 97 = 1503.015$ kN- mm

Twisting moment on the crankpin

$T_c = HT_1 \times r = 13.745 \times 64 = 879.68$ kN m

From this we have equivalent twisting moment

$$T_e = \sqrt{(M_c)^2 + (T_c)^2} = 1741.63 \text{ kN mm}$$

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

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

$M_{ev} = 3214.06$ kN mm

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)

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

$$3214.06 \times 10^3 = \frac{\pi}{32} \times (70)^3 \times \sigma_v$$

$$\sigma_v = 95.44 \text{ N/mm}^2 \text{ or MPa}$$

IV. PROCEDURE FOR STATIC ANALYSIS

A. Modeling of Crankshaft

First, prepare assembly in Catia for crankshaft and save as this part as .iges for exporting into Hypermesh environment. Import .iges model in hypermesh module.

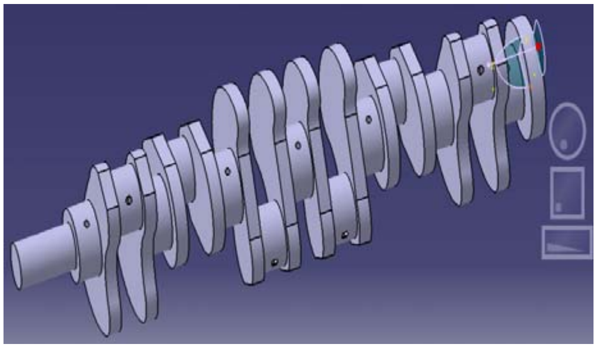


Fig. 3: 3D Modelling of Crankshaft Using Catia.

B. Meshing Of Crankshaft

Meshing of crankshaft is done using tetra. Select the meshing size as 10 mm for main bearing and 8 mm for crankpin and web of the crankshaft because the maximum stress concentration is occur at the crankpin and web of the crankshaft. Meshing details of crankshaft is as shown in below.

Type of Element: Tetrahedrons

Number of Nodes: 25112

Number of Elements: 108050

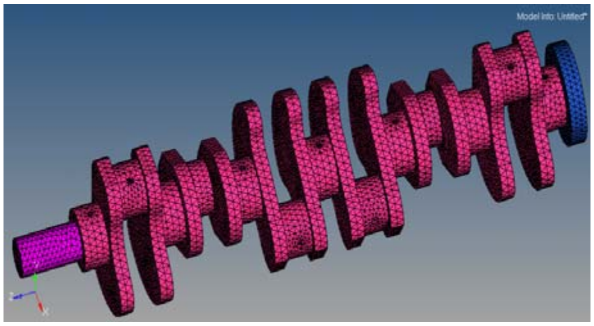


Fig. 4: Meshing of Crankshaft

C. Material Details for static structural analysis

Material Type: EN 16 Grade
Density: 7.85 g/cm^3
Tensile Strength: 570 MPa
Yield Strength: 295 MPa
Poissions Ratio: 0.3
Elastic Modulus: 190-210 GPa

D. Apply Load and boundary condition.

Boundary conditions play an important role in FEA.The Tangential load of 27.49 KN is applied by using rigid RBE_2 elements in y-direction. Static analysis is performed with application of load on crankpin .The radial force is applied 30.99 kN in z-direction and tangential force is applied in y-direction. Boundary conditions applied are left end of crankshaft is constrained in all degrees of freedom in all directions except z-rotation, The right end is constrained in all degrees of freedom in all directions except z translation as shown in fig.5. It is carried out to find displacement, von - misses stress, strain. Under static loading condition neglecting inertia and damping effect.

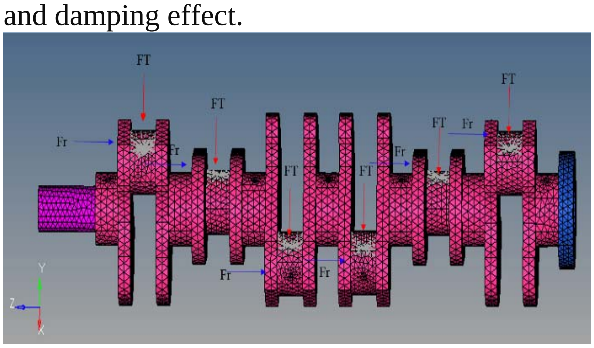


Fig. 5: Boundary Condition Diagram

E. ANALYSIS RESULTS

The maximum stress induced in the crankshaft is 93.922 N/mm^2 at the crankpin fillet region. Yield Strength of Grade En 16 material is 295 N/mm^2.Crankshaft Von-Misses stress Simulation result is shown in Fig.7. The maximum deformation value is 0.022 mm. A crankshaft total deformation is shown in Fig.6.

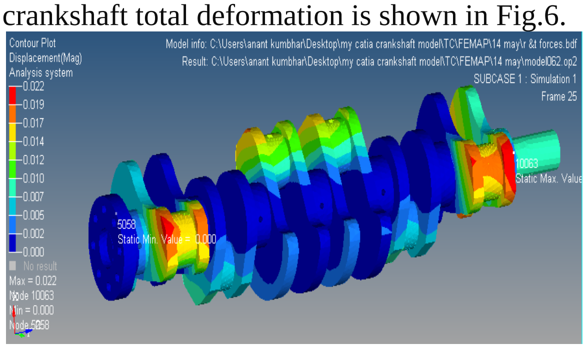


Fig. 6: Displacement Diagram

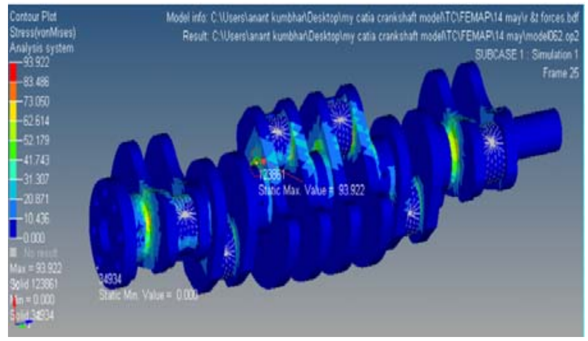


Fig. 7: Von – Misses Stress Diagram

V. RESULTS AND CONCLUSION

In this paper, the crankshaft model was created by Catia V5 software. The maximum stress appears at the fillet areas between the crankshaft journal and crank cheeks and near the central point journal. The value of von-misses stresses that comes out from the analysis is far less than material yield stress so our design is safe.

Table II: Validated Results

Sr. No.	Type of Stress	Theoretical (MPa)	FEA Result (MPa)
1	Von – Misses stress	95.44	93.922

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VI. ACKNOWLEDGEMENT

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