Analysis and Optimisation of Crankshaft

Mr. S. M. Nagare, Prof P. N. Narwade

1 sachin.nagare88@gmail.com

1 Department of Mechanical Engg, DPVVP COE, Vilat ghat, Savitribai Phule Pune University

ABSTRACT

Crankshaft is one of the critical components for the effective and precise working of Internal Combustion Engine. The modal analysis of a 6-cylinder crankshaft is discussed using finite element method in this paper. Three-dimension models of diesel engine crankshaft were created using SOLID WORKS software. The finite element analysis (FEM) software ABAQUS was used to analyze the vibration modal of the crankshaft. The maximum stress point and dangerous areas are found by the deformation analysis of crankshaft. We compared result of theoretical, FEA & experimental. The results would provide a valuable theoretical foundation for the optimization and improvement of engine design.

Keywords: Finite Element Analysis, Pro-E, ABAQUS, Crankshaft, 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 six 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.

II. LITERATURE REVIEW

1. Xiaorong Zhou et al 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.

2. Farzin H. Montazersadgh et al. 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.

3. Jian Meng 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.

4. Gu Yingkui et 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.

An extensive literature review on crankshafts was performed by Zoroufi and Fatemi (2005). Their study presents a literature survey focused on fatigue performance evaluation and comparisons of forged steel and ductile cast iron crankshafts. In their study, crankshaft specifications, operation conditions, and various failure sources are discussed. Their survey included a review of the effect of influential parameters such as residual stress on fatigue behavior and methods of inducing compressive residual stress in crankshafts. The common crankshaft material and manufacturing process technologies in use were compared with regards to their durability performance. This was followed by a discussion of durability assessment procedures used for crankshafts, as well as bench testing and experimental techniques. In their literature review, geometry optimization of crankshafts, cost analysis and potential cost saving opportunities are also briefly discussed.

**Specifications of Diesel Engine**

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>No of cylinders</td>
<td>6</td>
</tr>
<tr>
<td>Bore/Stroke</td>
<td>86 mm/ 68 mm</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>18 : 1</td>
</tr>
<tr>
<td>Max. Power</td>
<td>8.1 HP @ 3600rpm</td>
</tr>
<tr>
<td>Max. Torque</td>
<td>16.7 Nm@ 2200rpm</td>
</tr>
<tr>
<td>Maximum Gas pressure</td>
<td>25 Bar</td>
</tr>
</tbody>
</table>

### III. DESIGN OF CRANK SHAFT

**A] When the crank is at dead center**

1. Force on the Piston \( F_p = \text{Area of the bore} \times \text{Max. Combustion pressure} = \frac{\pi}{4} \times D^2 \times P_{\text{max}} \)

\[ = 14.522 \text{ kN} \]

2. Horizontal Reactions at bearings (1&2) due to tangential force is given by,

\[ H_1 = H_2 = \frac{F_p x b1}{b} = 7.26 \text{ kN} \]

3. Similarly, Vertical Reactions at bearings (2& 3) due to piston gas load is given by,

\[ V_2 = V_3 = \frac{(W x C_2) x C}{C_1} = 10 \text{ kN} \]

**Design of Crankpin**

Let \( d_c = \text{Diameter of crankpin in mm} \)

\[ L_c = \text{Length of crankpin in mm} \]

\[ \sigma_b = \text{allowable bending stress for the crankpin} \]

It may be assume as 75 MPa

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

\[ M_c = H_1 \times b = 624.36 \text{ kN.mm} \]

\[ M_c = \frac{\pi}{32} \times d_c^3 \sigma_b \]

\[ d_c = 44 \text{ mm} \]

Length of the crankpin

\[ L_c = \frac{F_p}{(d_c \times b)_1} = 33 \text{ mm} \]

**Design of Left hand crank web:**

Thickness of the crank web,

\[ t = 0.65d_c + 6.35 = 35 \text{ mm} \]

width of the crank web,

\[ w = 1.125d_c + 12.7 = 65 \text{ mm} \]

Max. Bending Moment on the crank web,

\[ M = H_1 (b - l_c - 2/2) = 377.52 \text{ KN.MM} \]

**B] When the Crank Is At An Angle Of Maximum Twisting Moment**

1. Force on the Piston \( F_p = \text{Area of the bore} \times \text{Max. Combustion pressure} \)

\[ = \frac{\pi}{4} \times D^2 \times P_{\text{max}} \]

\[ = 5808.8 \text{ N} \]

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 \( \Omega \)).

We know that \( \sin \Theta = \sin \Theta / (L/R) \)

\[ \Theta = 6.58^\circ \]

2. Thrust in the connecting rod \( F_Q = F_p / \cos \Theta = 5847.3 \text{ N} \)

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

a) Tangential force on the crank shaft, \( F_S = F_Q \sin (\Theta + \Omega) = 3880.45 \text{ N} \)

b) Radial force on the crank shaft, \( F_R = F_Q \cos (\Theta + \Omega) = 4373.7 \text{ N} \)

3. Reactions at bearings (1&2) due to tangential force is given by,

\[ H_T_1 = H_T_2 = \frac{F_T x b_1}{b} = 1940.22 \text{ N} \]

4. Similarly, Reactions at bearings (a & b) due to radial force is given by,

\[ H_R_1 = H_R_2 = \frac{F_R x b_1}{b} = 2186.85 \text{ N} \]

**Design of Crankpin**

Let \( d_c = \text{Diameter of crankpin in mm} \)

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

\[ M_c = H_1 \times b_2 = 188069.1 \text{ N.mm} \]

Twisting moment on the crankpin,

\[ T_e = \sqrt{( M_2^2 + T_c^2)} = 199303.02 \text{ N.mm} \]

Equivalent twisting moment \( (T_e) \)

\[ T_e = \frac{\pi}{16 \times (d_c)^3} \times \sigma_b \]

\[ d_c = 30.72 \text{ mm} \]

Twisting moment on shaft \( T_s = F_T x r = 131935.3 \text{ N.mm} \)
Equivalent Torque \( Te = \sqrt{MS_2 + TS_2} = 864368.49 \text{ N/mm} \)

\[ Te = \pi/16 x (d^3) \times \sigma_b = 20.38 \text{ N/mm}^2 \]

This value is less than allowable shear stress 35 MPa Hence Dia 60 mm is valid

B.M. at junction
\( Ms_1 = R_1(b_2+l_4/2) = 152009.4 \text{ N/mm} \)

\( Ts_1 = F_T x r = 13193.53 \text{ N/mm} \)

\( Te = \sqrt{(Ms_1 + Ts_1)} = 152580.88 \text{ N/mm} \)

\( Ds_1 = 26.44 \text{ mm} \)

Design of right hand crank web:
\( MR = (ob) x Z \)
\( (ob)r = 8.56 \text{ N/mm}^2 \)

\( MT = (ob) x TS \)

\( Direct \ compressive \ stress, \sigma_c = (ob) + (ob)T \)

\( = 12.79 \text{ N/mm}^2 \)

Twisting moment on arm
\( MT = \frac{HT^2}{Z_p} = 132905.07 \text{ N/mm} \)

\( \sigma_b = T/Z_p = 7.51 \text{ N/mm}^2 \)

Max combine stress
\( \sigma_m = \sigma_c/2 + \frac{1}{2} \sqrt{(\sigma_c^2 + 4\sigma_b)} = 16.26 \text{ N/mm}^2 \)

**Design of crank pin against fatigue loading**

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

\( M_{ev} = \sqrt{K_b X M_c} + 3/4(K_t x K_c) \)

Where, \( K_b = \) Combine shock & fatigue factor for bending = 2

\( K_t = \) Combine shock & fatigue factor for torsion = 1.5

Putting the values in above equation we get

\( M_{ev} = 938.33 \text{ KN/mm} \)

Also we know that,

\( M_{ev} = \pi/32 x (d^3) \times 3 x \sigma_v \)

\( \sigma_v = 121.15 \text{ N/mm}^2 \)

**Von Mises Stress = 121.15 N/mm²**

\( T_{ev} = \sqrt{K_b X M_c} + 3/4(K_t x K_c) \)

\( = 953.49 \text{ KN/mm} \)

Also we know that,

\( T_{ev} = \pi/32 x (d^3) x \sigma_b \)

\[ \sigma_b = 57 \text{ N/mm}^2 \]

**IV. RESULTS**

Diameter of crankpin = 44 mm
Length of the Crank pin = 33 mm
Diameter of shaft = 60 mm
Web Thickness (Left & Right Hand) = 35 mm
Web Width (Left & Right Hand) = 65 mm

**V. METHODOLOGY:**

Procedure of static analysis
First I have prepared assembly in solid works for crankshaft & save as IGES for exporting into ABAQUS. Import IGES Model in ABAQUS Workbench for simulation module

1. Apply material for crankshaft

Material detail: C-70 Alloy Steel
Material Type – Forge steel
Yield strength (MPa) – 483
Ultimate Tensile strength (MPa) – 621
Elongation ( % ) – 13
Poisson ratio -0.30

2. Mesh the crankshaft:

Types of Element: Tetrahedron
No of element : 17119
No of Nodes: 9605

3. Defined B. C.

4. Run the analysis
5. Get the result
6. Output of the analysis
Results of Analysis
Maximum Deformation at a Phase Angle 355°

RESULTS AND CONCLUSION
Finite Element analysis of the six cylinder crankshaft has been done using FEA tool ABAQUS

Result Table:-

<table>
<thead>
<tr>
<th>Sr no</th>
<th>Types of stress</th>
<th>Theoretical</th>
<th>FEA Analysis</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Von-Misses Stresses (N/mm²)</td>
<td>121.15</td>
<td>119.3</td>
</tr>
<tr>
<td>2</td>
<td>Shear Stresses (N/mm²)</td>
<td>57</td>
<td>41.35</td>
</tr>
</tbody>
</table>

1. 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.

2. Experimental work will be done at next stage. After experimental work is done we will compare all the result i.e. Theoretical, FEA & Experimental result

3. 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.

4. The weight of the crankshaft reduced from Design 1-Original to Design 1-Modified. Thereby, reduces the inertia force.

5. As the weight of the crankshaft is decreased this will decrease the cost of the crankshaft and increase the engine performance

6. Material C-70 Alloy steel is meeting the maximum no of requirements. & C70 alloy steel given optimum results as compared to the other materials.

ACKNOWLEDGMENT
First and foremost, I would like to express my deep sense of gratitude and indebtedness to my supervisor Mr. P. N. Narwade Professor, Mechanical Engineering Department for his invaluable encouragement, suggestions and support from an early stage of this project stage one report and providing me extraordinary experiences throughout the work. Above all, his priceless and meticulous supervision at each and every phase of work inspired me in innumerable ways. I specially acknowledge him for his advice, supervision, and the vital contribution as and when required during this project stage one. His involvement with originality has triggered and nourished my intellectual maturity that will help me for a long time to come. I am proud to record that I had the opportunity to work with an exceptionally experienced Professor like him. I am highly grateful to Prof R. N. Navtar, PG coordinator P.DV.V.P college of Engineering, Ahmednagar for their kind support and permission to use the facilities available in the Institute.
References


A comparison of cycling SRM crank and strain gauge instrumented pedal measures of peak torque, crank angle at peak torque and power output Rodrigo R. Bini, Patria A. Hume, Andre Cerviri, Proscenia Engineering 13 (2011) 56–61


Model of curvature of crankshaft blank during the heat treatment after forging Andrzej Milenina, Tomasz Reca, Wojciech Walczykb, Maciej Pietrzyka, Procedia Engineering 81 (2014) 498 – 503

