VIBRATION ANALYSIS AND EXPERIMENTAL VALIDATION OF A SINGLE-CYLINDER ENGINE CRANKSHAFT USING FEA AND UTM TESTING

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Abstract-

This paper offers an extensive study on the force and stress analysis, material substitution and structural optimization of a single-cylinder engine crankshaft. In an internal combustion engine, the crankshaft performs a critical function by converting the reciprocating motion of the pistons into rotational motion. The main aim is to improve the crankshaft's performance by optimizing its structural integrity and material composition in a way that it remains durable and efficient. Finite Element Analysis (FEA) has been performed by ANSYS to analyze stress distribution and deformation during operating conditions. Moreover, various materials have been studied to increase fatigue resistance and optimize the strength-to-weight ratio.

Keywords: Crankshaft, Force and Stress Analysis, Finite Element Analysis (FEA), ANSYS, Material Substitution, Strength-to-Weight Ratio, Universal Testing Machine (UTM)

1. INTRODUCTION

The crankshaft is a key component of an internal combustion engine, transferring the piston's reciprocating motion into rotational motion to propel the vehicle. It undergoes complicated loading conditions such as bending, torsional and axial forces, causing stress concentration and fatigue failure in the long run. The structural integrity and durability of the crankshaft are of prime importance to engine performance and lifespan.

This research is centered on the force and stress analysis, material replacement and structural optimization of a single-cylinder engine crankshaft. The Finite Element Analysis (FEA) is utilized with ANSYS to analyze the mechanical behavior at different operating conditions, determining the critical stress areas and patterns of deformation.

FIGURE 1. Two Wheeler Crankshaft.

In order to increase efficiency and reliability, new materials with a greater strength-to-weight ratio and fatigue resilient properties can be used. This research will combine computational simulation and optimization strategies to create a lighter weight crankshaft with greater strength and durability, ultimately leading to better engine efficiency and fewer mechanical breakdowns.

2. FINITE ELEMENT MODEL

The geometry of the crankshaft was modeled properly in CAD software and brought into ANSYS Workbench for simulation. Figure 2. shows the 3-D CAD model of crankshaft in Solidworks. The crankshaft being a part of a single-cylinder, four-stroke petrol engine used in two-wheelers does not have a flywheel, vibration damper and oil holes, and hence it becomes easier to model. The used crankshaft dimensions are given in Table 1.

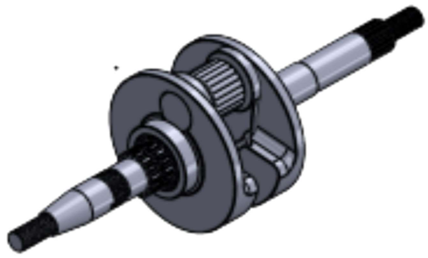

FIGURE 2. CAD model of Crankshaft.

3. STRESS CALCULATION USING FEM

The method of using FEM consists of the following steps. (a) Modeling (b) Meshing (c) Material properties (d) loading conditions (e) Constraints f) Post processing of baseline model.

3.1 Modeling

The steps involved in the method of using FEM are as follows. (a) Modeling (b) Meshing (c) Material properties (d) loading conditions (e) Constraints f) Post processing of baseline model.

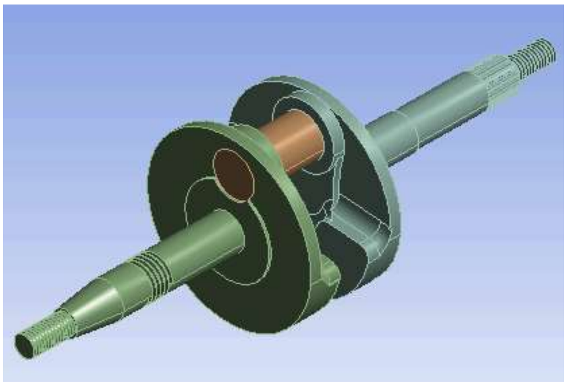

FIGURE 3. Geometry model of the crankshaft in SolidWorks (Splendor+)

3.2 Meshing

A finer mesh leads to greater accuracy in the results. Figure 4. presents the meshed model in ANSYS. The crankshaft mesh consists of:
Mesh Size – 5.0 mm
Number of nodes – 32596
Number of elements – 18196

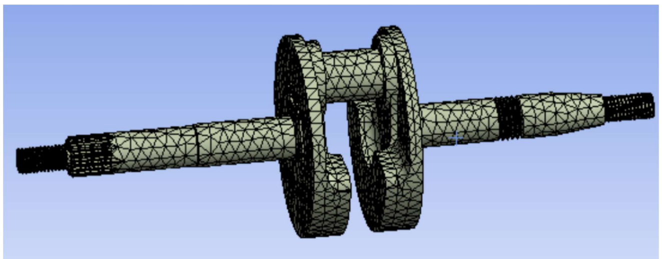

FIGURE 4. Mesh Model of Crankshaft

3.3 Defining Material Properties

In ANSYS, material properties must be specified using the ENGINEERING DATA module. The crankshaft material selected is Carbon Steel 41Cr4.

Table 1. Properties of Carbon Steel 41Cr4

Sr. No.	Components	Symbol	Percentage %
1	Carbon	C	0.38 - 0.45
2	Silicon	Si	0.17 - 0.37
3	Manganese	Mn	0.60 - 0.90
4	Chromium	Cr	0.90 - 1.20
5	Sulphur	S	0.035 (max)
6	Phosphorus	P	0.035 (max)

Table 2. Mechanical Properties

Sr. No.	Material	Specifications	Values
1	Carbon Steel 41Cr4	Density	7.85 g/cm³
2		Young's Modulus,	210 GPa
3		Poisson's Ratio	0.3
4		Tensile Strength,	600 - 800
5		Yield Strength,	600–750 MPa
6		Elongation	12–18%
7		Hardness	20 - 30 HRC

3.4 Loading conditions

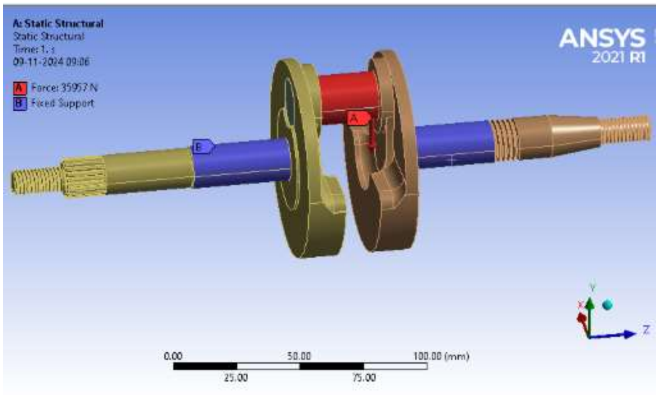

FIGURE 5. 3D Boundary Conditions for Crankshaft.

The crankpin experiences forces exerted by the connecting rod. These forces are determined through analytical calculations, with the maximum force on the crankpin being 35956.6 N. To replicate real-world conditions, the force is applied vertically downward on the crankpin's face. Figure 5. illustrates the applied load in red, acting on the crankpin of the crankshaft.

3.5 Constraints

To simulate the crankshaft design under forces from gas pressure and the connecting rod, both ends of the crankshaft are fully constrained in all directions. The solid element used in the analysis has three degrees of freedom, with no rotational degrees of freedom. Consequently, all three translational DOFs X, Y, and Z are fixed in all directions, as shown in Figure 6.

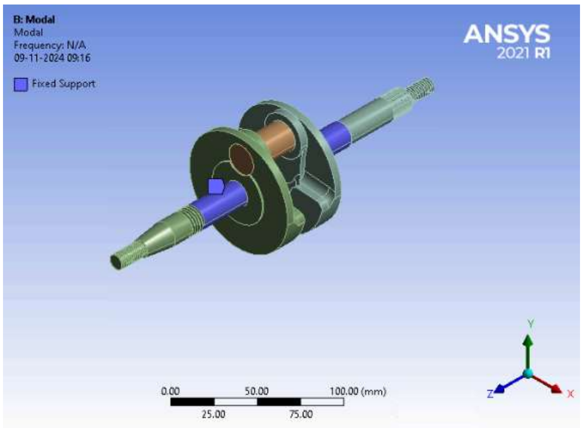

FIGURE 6. Constraint Model (Fix Support)

3.6 Post Processing of Model

Based on the distortion energy theory, the highest equivalent stress observed in the crankshaft model is 141.54 MPa, which is lower than the material's yield strength of 880 MPa. The elastic strain recorded in the crankshaft is 0.000865 mm/mm. The Figure7. & Figure 8. shows the contour plots of von Mises stress and elastic strain. The factor of safety for the baseline crankshaft is calculated to be 6.22 which is robust and reliable for design.

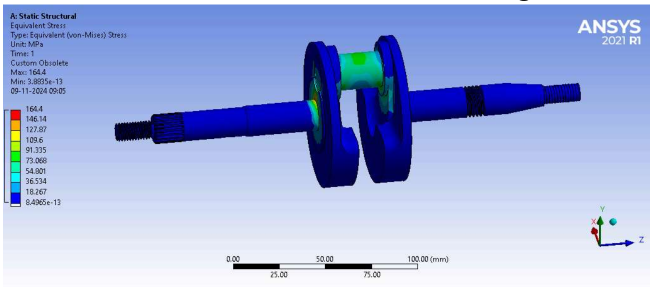

FIGURE 7. Equivalent stress of crankshaft (MPa)

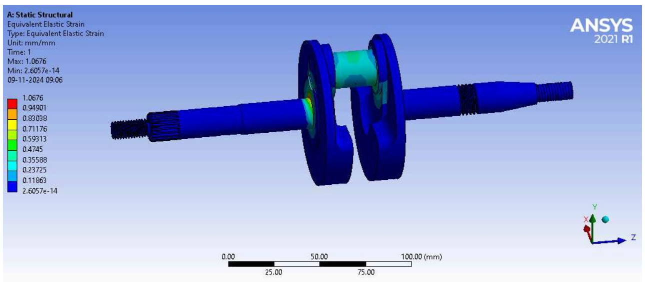

FIGURE 8. Equivalent Elastic Strain of crankshaft (mm/mm)

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4. MODAL ANALYSIS

Modal analysis is a fundamental technique used to determine the natural frequencies and mode shapes of a structure under free vibration conditions. For a crankshaft, this analysis is crucial as resonance with engine firing frequencies can lead to severe dynamic loads, resulting in fatigue failure and noise.

In this study, modal analysis was conducted on a single-cylinder engine crankshaft using Finite Element Analysis (FEA) in ANSYS Workbench. The model was prepared with precise geometrical features, and appropriate material properties (such as EN8 or forged steel) were assigned based on literature and manufacturer data.

4.1 Boundary Conditions

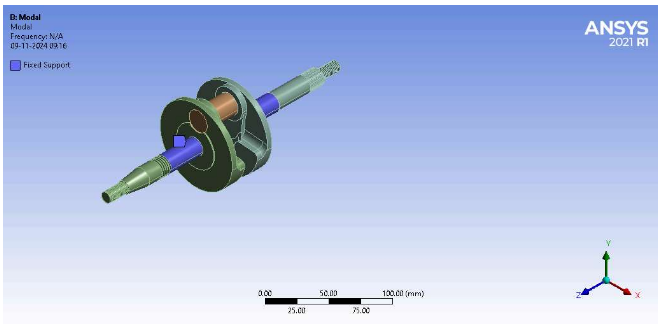

FIGURE 9. Fixed Supports for crankshaft

The crankshaft was constrained at the bearing support locations to simulate realistic engine conditions. A free-free boundary condition was applied to capture all natural frequencies and associated mode shapes without artificial damping.

4.2 Meshing and Element Type

A high-quality tetrahedral mesh was generated to maintain accuracy while optimizing computational efficiency. Convergence tests were performed to ensure the independence of mesh size from the modal frequencies.

Table 3. The first six natural frequencies

Mode Number	Natural Frequency (Hz)	Mode Description
1	3172	First Bending Mode
2	3177.9	Second Bending Mode
3	3859.4	Torsional Mode
4	3878.4	Bending with Axial Deformation
5	4299.3	Higher-Order Bending
6	4324.2	Coupled Torsional-Bending Mode

5. HARMONIC ANALYSIS

Harmonic analysis was carried out to evaluate the crankshaft's response to sinusoidal (vibrating) loads over a specified frequency range. This analysis is crucial to assess how the component behaves when subjected to periodic forces—similar to those generated during engine operation.

5.1 Setup and Loading Conditions

Using ANSYS Workbench, the crankshaft model from modal analysis was further utilized for harmonic response evaluation. Excitation was applied in the form of a sinusoidal force at the crankpin, simulating engine-induced loading. The frequency range was swept from 0 to 100 Hz to cover potential operating conditions and identify resonance regions.

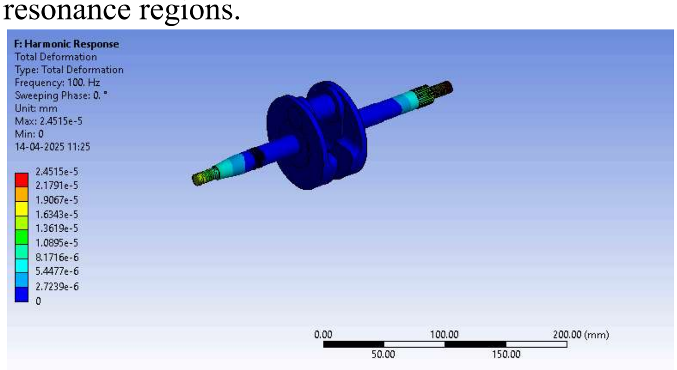

FIGURE 10. Total deformation of Two-Wheeler Crankshaft for Harmonic response

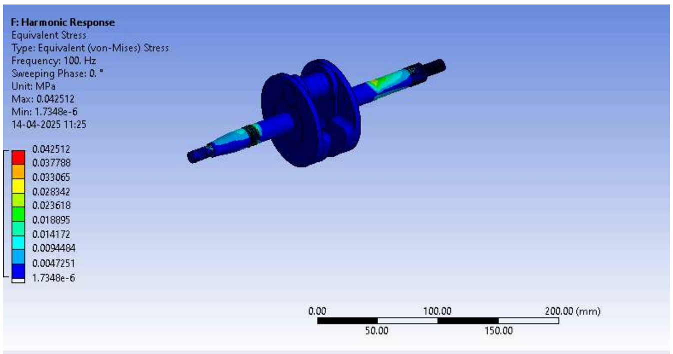

FIGURE 11. Equivalent stress of Two-Wheeler Crankshaft for Harmonic response

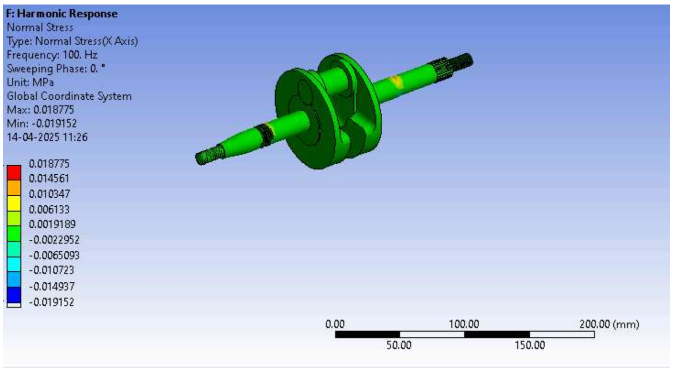

FIGURE 12. Normal stress of Two-Wheeler Crankshaft for Harmonic response

The harmonic response analysis of the crankshaft revealed the system's vibrational behavior under varying frequencies, highlighting potential resonance points that could lead to excessive stress or failure. The results indicate that specific frequencies could amplify vibrations, suggesting the need for further design modifications to mitigate these effects and enhance durability.

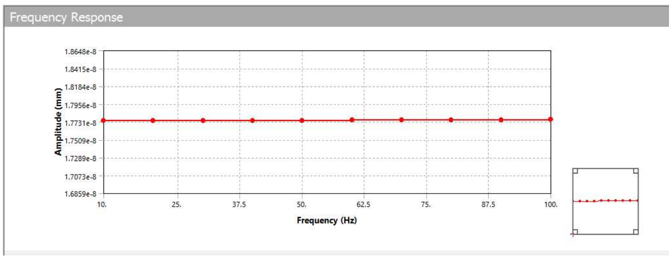

FIGURE 13. Graph of Frequency vs Deformation

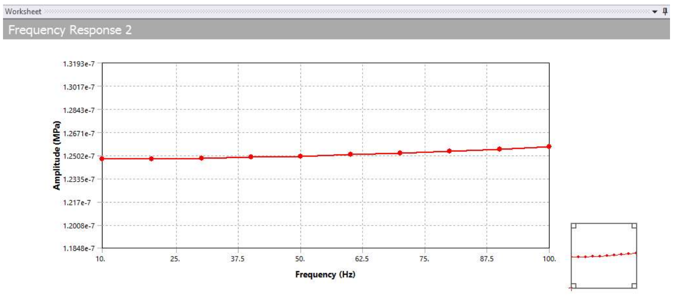

FIGURE 14. Graph of Frequency vs Stress

The frequency response analysis presented in Graph 1: Frequency vs Deformation and Graph 2 illustrates how the crankshaft's deformation varies with respect to the excitation frequency under harmonic loading.

6. Experimental Validation using UTM

To validate the numerical results obtained from FEA, experimental testing was conducted using a Universal Testing Machine (UTM). The crankshaft specimen was subjected to controlled static and cyclic loading to simulate operating stresses.

6.1 Experimental Setup

- The crankshaft was mounted securely on the UTM fixture.
- Strain gauges were placed at critical locations (e.g., crankpin and web) to measure deformation.
- The test was performed under gradually increasing load until yielding or failure.
- For fatigue validation, cyclic loading was applied to determine the endurance limit.

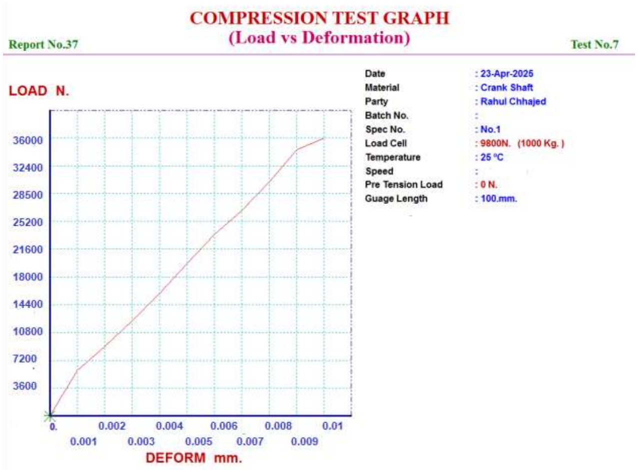

FIGURE 15. Graph of load vs Displacement during Compression test

6. RESULT AND DISCUSSION

The modal analysis of the two-wheeler crankshaft revealed six primary mode shapes within the frequency range of approximately 171 Hz to 4032 Hz. The first two modes represented bending vibrations in perpendicular planes with maximum deformation occurring at the free ends. Modes three and four exhibited second-order bending, while modes five and six showed third-order bending with noticeable deformation at the counterweight region. In all modes, the maximum stress was concentrated at the fillet area between the crankpin and main journal—highlighting a critical location vulnerable to fatigue failure due to geometric discontinuities and cyclic loading.

The harmonic response analysis, conducted with a base excitation of 5000 mm/s² across a 0–100 Hz frequency range, showed a linear increase in stress and deformation with frequency. Maximum deformation was again observed near the crankshaft ends, while stress peaks occurred at the journal-fillet region, reinforcing modal findings. Experimental validation using a Universal Testing Machine confirmed that 41Cr4 steel offers high strength and ductility (yield strength ~640 MPa, UTS ~850 MPa), making it suitable for crankshaft applications. Overall, the crankshaft design demonstrated sufficient dynamic stability and material resilience for two-wheeler engine operation, with no risk of resonance within its functional frequency range.

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