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Structural Design of Single Crank Shaft Using Analytical and FEM

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ABSTRACT:

The overall objective of this study was to evaluate and compare the Structural performance of two competing manufacturing technologies for automotive crankshafts, namely forged steel and composite material. Crankshaft is one of the vital elements of the internal combustion engine. The main purpose of crankshaft in automobile is to transform reciprocating linear motion to rotary motion i.e. it is used to translate from piston to crank In this study a dynamic simulation was conducted on crankshaft, Cast Iron and forged steel, from similar single cylinder four stroke engines. Finite element analysis was performed to obtain the variation of stress magnitude at critical locations. The pressure-volume diagram was used to calculate the load boundary condition in dynamic simulation model, and other simulation inputs were taken from the engine specification chart.

The Structural analysis was done analytically and was verified by simulations in ANSYS. Results achieved from aforementioned analysis were used in optimization of the forged steel crankshaft. Geometry, material, and manufacturing processes were optimized considering different constraints, manufacturing feasibility, and cost. Comparisons for the properties such as Structural results of crankshafts made up of forged steel & Cast Iron were determined and the results were compared. Model theoretical calculations are performed for clear analysis. The output of result would provide a possible recommendation for optimization of crank shaft and development of engine design.

1. INTRODUCTION:

1.1. Background:

Crankshaft is a large component with a complex geometry in the engine, which converts the reciprocating displacement of the piston to a rotary motion with a four link mechanism. Since the crankshaft experiences a large number of load cycles during its service life, fatigue performance and durability of this component has to be considered in the design process. Design developments have always been an important issue in the crankshaft production industry, in order to manufacture a less expensive component with the minimum weight possible and proper fatigue strength and other functional requirements. These improvements result in lighter and smaller engines with better fuel efficiency and higher power output.

This study was conducted on a single cylinder four stroke cycle engine. Two different crankshafts from similar engines were studied in this research. The finite element analysis was performed in four static steps for each crankshaft. Stresses from these analyses were used for superposition with regards to dynamic load applied to the crankshaft. Further analysis was performed on the forged steel crankshaft in order to optimize the weight and manufacturing cost. Figure 1.1 shows a typical picture of a crankshaft and the nomenclature used to define its different parts.

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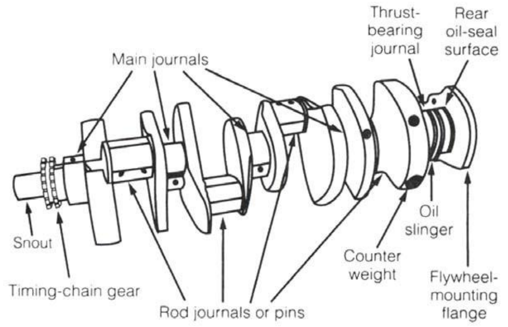

Figure 1.1.Typical crankshaft with main journals that support the crankshaft in the engine block. Rod journals are offset from the crankshaft centerline

4 STRESS ANALYSIS AND FEA:

In order to evaluate the FEA results, a component test was conducted with strain gages. FEA boundary conditions were changed according to the test setup. Strain gages were mounted on the forged steel crankshaft and results from FE analysis and experimental data were compared in order to show the accuracy of the FE model. Finally, results from dynamic FE analysis, which consist of stress history at different locations, were used as the input to the optimization process.

4.1 FINITE ELEMENT MODELING:

Finite element modeling of any solid component consists of geometry generation, applying material properties, meshing the component, defining the boundary constraints, and applying the proper load type. These steps will lead to the stresses and displacements in the component. In this study, similar analysis procedures were performed for both forged steel and cast iron crankshafts.

4.1.2 MESH GENERATION:

FEA analysis was performed on both crankshafts for the dynamic load analysis, as well as for the test setup. Since boundary conditions of dynamic FEA and test setup FEA are different, separate FE models were needed. In this section, meshing of both dynamic FEA and test setup FEA are presented for the forged steel and cast iron crankshafts.

4.1.2.1 DYNAMIC FEA:

Quadratic tetrahedral elements were used to mesh the crankshaft finite element geometry. Tetrahedral elements are the only option for meshing the imported complex geometries to the ANSYS software. Using linear tetrahedral elements will result in a rigid model with less accuracy, whereas using quadratic tetrahedral elements will increase the accuracy and lessen the rigidity of the geometry. In order to mesh the geometry with this element type, the free meshing feature of ANSYS software was used. In this feature, the global mesh size could be defined, while for critical locations free local meshing could be used to increase the number of elements for accurate stresses at locations with high stress gradients.

4.1.3. LOADING AND BOUNDARY CONDITIONS :

4.1.3.1 DYNAMIC FEA:

The engine manual of the forged steel crankshaft was used to determine the proper boundary conditions at bearing locations was taken from the engine manual of the forged steel crankshaft and shows different components at a cut view of the engine. It can be seen that the crankshaft is constraint with a ball bearing from one side and with a journal bearing on the other side. The ball bearing is press fit to the crankshaft and does not allow the crankshaft to have any motion other than rotation about its main central axis. Since only 180 degrees of the bearing surfaces facing the load direction constraint the motion of the crankshaft, this constraint was defined as a fixed semicircular surface as wide as the ball bearing width on the crankshaft. The other side of the crankshaft is on a journal bearing. Therefore, this side was modeled as a semicircular edge facing the load at the bottom of the fillet radius fixed in a plane perpendicular to the central axis and free to move along central axis direction.

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Boundary conditions rotate with the direction of the load applied such that the inner face of the fixed semicircular surface and sliding ring face the direction of the load. For example, if the load is applied downward the inner curve of both boundary conditions will face upward. Definition of a fixed edge is based on the degrees of freedom in a journal bearing, which allows the crankshaft to have displacement along its central axis. Also, the section that is located on the journal bearing can have limited rotation in the direction perpendicular to the plane which central axis and load vector. The distribution of load over the connecting rod bearing is uniform pressure on 120° of contact area. This load distribution is based on experimental results from Webster. The explanation of load distribution in the Webster et al. study is for connecting rods, but since the crankshaft is in interaction with the connecting rod, the same loading distribution will be transmitted to the crankshaft. For pressure P_O on the contact surface, the total resultant load is given by:

$$F = \int_{\frac{\pi}{3}}^{\frac{\pi}{3}} P_o Cos(\varphi) r\, t\, d\,\varphi = P_o\, r\, t\, \sqrt{3} \qquad (4.1)$$

where r is the crankpin radius and t is the crankpin length. As a result, the pressure constant is given by:

$$P_o = \frac{F}{r\, t\, \sqrt{3}} \qquad (4.2)$$

Force F, which is the magnitude of the total force applied to the crankshaft, can be obtained from dynamics analysis at different angles. According to the geometry of the forged steel crankshaft a unit load of 1 kN will result in the pressure of 1.142 MPa, as follows:

$$P_o = \frac{1000}{18.48 \times 27.37 \times \sqrt{3}} = 1.142 MPa$$

The same boundary conditions and loading were used for the cast iron crankshaft. Since some of the dimensions are different in the two crankshafts, the applied pressure resulting from a unit load of 1kN is calculated to be 1.018 MPa,

$$p_o = \frac{1000}{16.51 \times 34.35 \times \sqrt{3}} = 1.018 MPa$$

4.2.FINITE ELEMENT ANALYSIS RESULTS AND DISCUSSION:

It was pointed out that the analysis conducted was based on superposition of four basic loadings in the FE analysis. The unit load applied on the connecting rod bearing was a pressure of magnitude 1.142 MPa and 1.018 MPa for forged steel and cast iron crankshafts, respectively. Note that the resultant load F was 1 kN and because of differences in dimensions of the two crankshafts, the pressure is somewhat different.Section changes in the crankshaft geometry result in stress concentrations at intersections where different sections connect together. Although edges of these sections are filleted in order to decrease the stress level, these fillet areas are highly stresses locations over the geometry of crankshaft. Therefore, stresses were traced over these locations.

At a crank angle for which stress components are aimed to be calculated, load components at that crank angle defined in the local rotating coordinate system are taken from the dynamic analysis with consideration of their sign. Since the analysis is based on linear elastic behavior of the material, stress magnitude has linear relation with load. Identical stress components are then added together resulting in stress components of the aimed loading situation. Replacing these stress components in the following equation gives the von Mises stress.

$$\sigma_{von\ Mises} = \frac{1}{\sqrt{2}} \sqrt{\left(\sigma_{xx} - \sigma_{yy}\right)^2 + (\sigma_{xx} - \sigma_{zz})^2 + \left(\sigma_{yy} - \sigma_{zz}\right)^2 + \sigma\left(\sigma_{xy}^2 + \sigma_{xz}^2 + \sigma_{yz}^2\right)} \qquad (4.3)$$

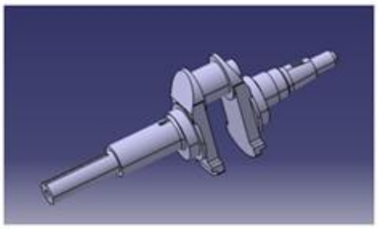
Following the above mentioned procedure, von Mises stress results for both FE models of forged steel and cast iron crankshafts were obtained.

PROBLEM STATEMENT:

As per load Condition or design Pressure developed on Piston is Calculated and load transfer from piston to Crank shaft is Calculated in MATLAB. Experimental results taken from base paper and Compare FEM results as per loadCalculated using MATLAB. For Modeling using CATIA V5, Calculation MATLAB & Finite Element Results ANSYS 15.0 WORKBENCH . Find out best suit model for Crank Shaft and suit for Static and Dynamic condition.

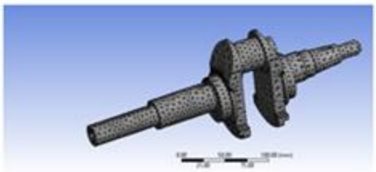
RESULT:

Crank shaft model is done in CATIA V5, two model of Forged Steel and Cast Iron as per CAD drawing Shown below.

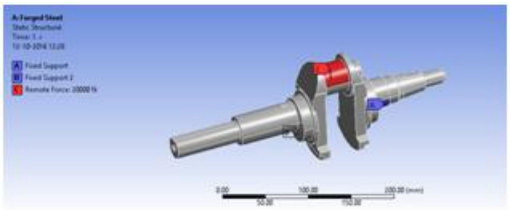


Forged Steel Model done in CATIAV5

Case1: Forged Steel is analyzed as per Static load Condition as per load design 20K N on Crank pin side by MATLAB. find the results of Stress, Strains & Deformation.

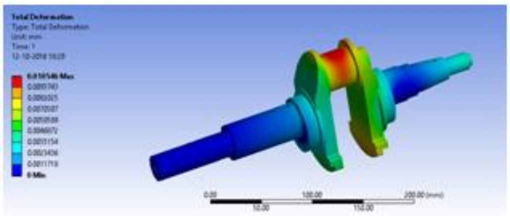


Mesh model of Forged Steel with fine Mesh with Element size of Minimum edge length 7.69E-3mm, Nodes:22058 and Elements:12398 Load & Boundary Conditions for Crankshaft is given as explain about above and shown below image :

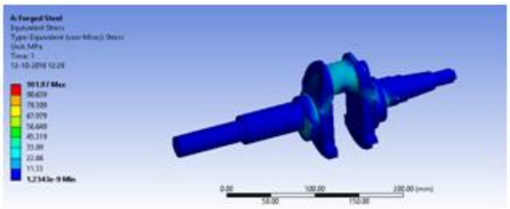


As in image Boundary Conditions A & B are position of fixed with All DOF as there bearing supports and Maximum load acts Vertically 20KN as Remote Force.
After Running for Solving the problem and find out

Results are:

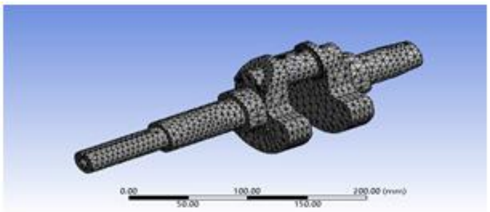


Total Deformation as per load condition is max : 0.0105mm as per Static load Conditions

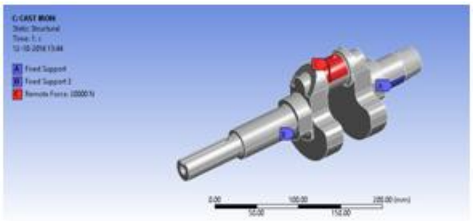


Von Mises Stress as per load Condition is Max :101.97 MPA as per Static Load Conditions

Case2: Cast Iron is analyzed as per Static load Condition as per load design 20K N on Crank pin side by MATLAB. find the results of Stress, Strains & Deformation.



Mesh model of Cast Iron with fine Mesh with Element size of minimum edge length 1.013E-2 mm, Nodes:19850 and Elements:11109
Load & Boundary Conditions for Crankshaft is given as explain about above and shown below image :



As in image Boundary Conditions A & B are position of fixed with All DOF as there bearing supports and Maximum load acts Vertically 20KN as Remote Force.

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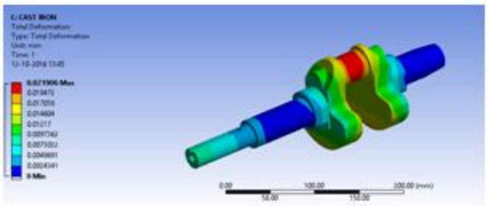
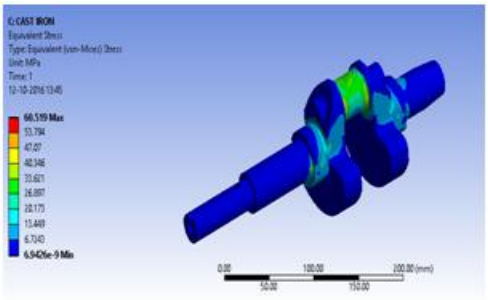
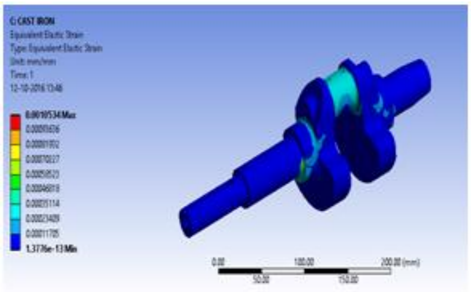
After Running for Solving the problem and find out Results are:



Total Deformation as per load condition is max : 0.0219 mm as per Static load Conditions



Von Mises Stress as per load Condition is Max :60.51 MPA as per Static Load Conditions

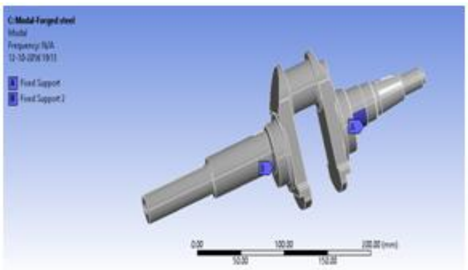


Von Mises Strain as per Load Condition is Max: 0.001053 mm/mm as per Static Load Conditions

DYNAMIC ANALYSIS:

Case 3: Model Analysis for Forged Steel and Model shapes are Find out and know the Resonance frequency and in the range of 0 to 5000 Hz.

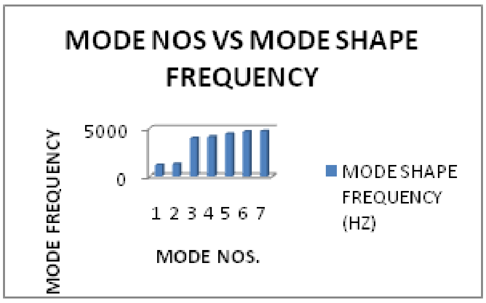
Boundary Condition applied to body of Forged Steel:



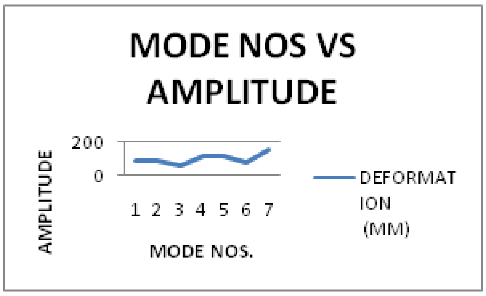
Load applied is zero as it is free Vibrations and fixed Condition are show in above figure

MODE NOS.	MODE SHAPE FREQUENCY (HZ)	DEFORMATION (MM)
1	1178.4	85.74
2	1277.2	87.63
3	3988.1	54.95
4	4115.2	114.02
5	4408.1	108.47
6	4662.2	75.73
7	4714.2	150.97

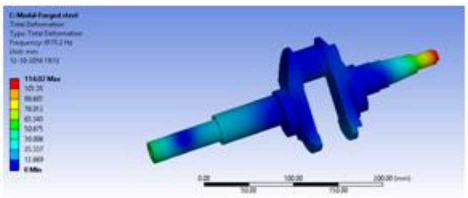
From the Model analysis Mode shape frequency are got and given below:



From result table mode nos against Mode Shape Freq is plot in range of frequency of 0 to 5000 Hz



From result table Mode nos vs Amplitude is plot and at model nos.4, 5 & 7 are at highest Amplitudes Deformation at Mode nos.4 & 5 are shown below:



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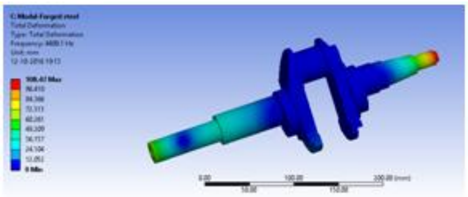
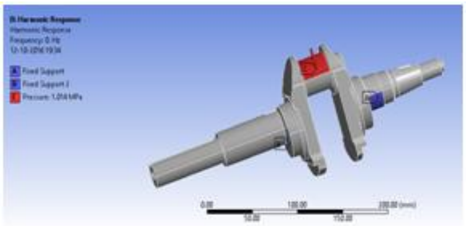
At frequency 4115.2 Hz and Mode no.04, Max deformation is 114.02 mm for Forged Steel Material

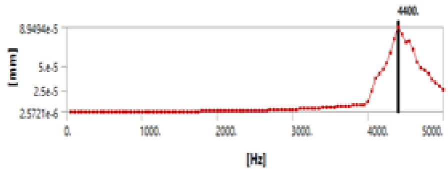


At frequency 4408.1 Hz and Mode no.05, Max deformation is 108.47 mm for Forged Steel Material

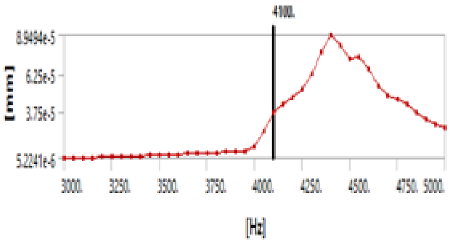
Case 4: Harmonic Analysis is done in frequency range of 0 to 5000 Hz and Stress and deformation are found as per load condition Calculated pressure (1.014 MPA) is applied on Crank Pin top.



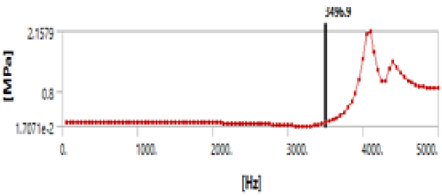
Load and Boundary Condition are applied as shown in fig in frequency Range of 0 to 5000 Hz.



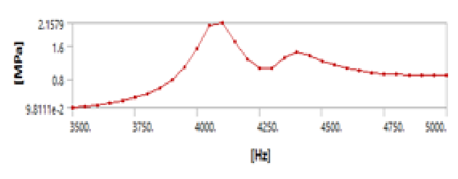
Frequency vs Deformation Range from 0 to 5000 Hz and at 4400Hz Maximum deformation is shown.



Zoom into frequency range of 3000 to 5000 Hz from 4000 to 4400 Hz Deformation is increases from minimum value to Maximum Value

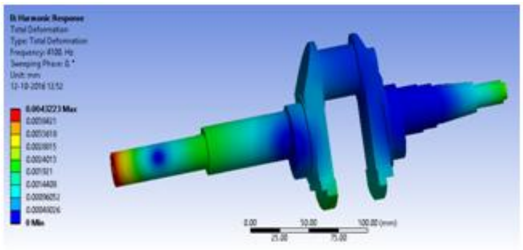


Frequency vs Stress Range from 0 to 5000 Hz and at 4400Hz Maximum Stress is shown.

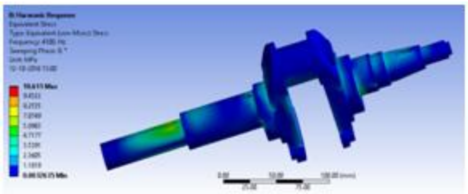


Zoom into Frequency range of 3000 to 5000 Hz from 3500 to 4400 Hz Stress is increases from minimum value to Maximum Value.

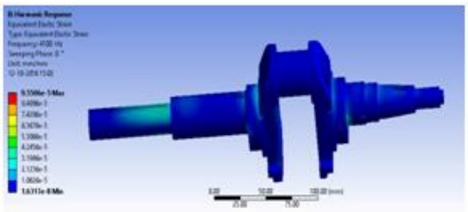
Result Plot at 4100 Hz frequency :



Deformation at Frequency Value 4100 Hz and Maximum deformation .0043 mm

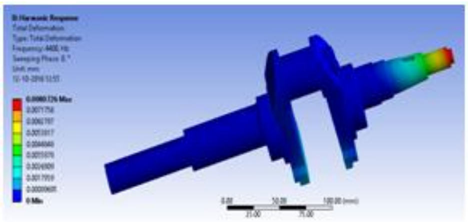


Stress at Frequency Value 4100 Hz and Maximum Stress .10.611 MPA



Strain at Frequency Value 4100 Hz and Maximum Strain .9.55E-5 mm/mm

Result Plot at 4400 Hz frequency :



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CONCLUSION:

A forged steel and a ductile cast iron crankshaft were chosen for this study, both of which belong to similar single cylinder four stroke air cooled gasoline engines. First, both crankshafts were digitized using a CMM machine. Load analysis was performed based on dynamic analysis of the slider crank mechanism consisting of the crankshaft, connecting rod, and piston assembly, using analytical approach and verification of results by MATLAB. FEA model of each crankshaft was created and superposition of stresses from unit load analysis in the FEA, according to dynamic loading, resulted in stress history at different locations on the crankshaft geometry during an entire engine cycle.

The following conclusions can be drawn from the analysis conducted in this study:

1. Dynamic loading analysis of the crankshaft results in more realistic stresses whereas static analysis provides overestimated results. Accurate stresses are find out.

2. There are two different load sources in an engine; inertia and combustion. These two load source cause both bending and torsional load on the crankshaft. The maximum load occurs at the crank angle of 355 degrees for this specific engine. At this angle only bending load is applied to the crankshaft.

3. Considering torsional load in the overall dynamic loading conditions has no effect on von Mises stress at the critically stressed location. The effect of torsion on the stress range is also relatively small at other locations undergoing torsional load. Therefore, the crankshaft analysis could be simplified to applying only bending load.

4. Superposition of FEM analysis results from perpendicular loads is an efficient and simple method of achieving stresses for different loading conditions according to forces applied to the crankshaft from the dynamic analysis.

5. Experimental stress and FEA results showed close agreement, within 7% difference. These results indicate non-symmetric bending stresses on the crankpin bearing, whereas using analytical method predicts bending stresses to be symmetric at this location. The lack of symmetry is a geometry deformation effect, indicating the need for FEA modelling due to the relatively complex geometry of the crankshaft.

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