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Finite Element Analysis of Two Wheeler Honda Bike Crank Shaft

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Abstract— **The most important objective of this study is to study weight and cost reduction opportunities for a crankshaft. The necessity of load history in the FEM analysis necessitates performing a detailed load analysis. Therefore, this study consists of two major sections: (1) FEM and stress analysis of existing crankshaft, (2) optimization for weight and cost reduction. Crankshaft is one of the vital components for the effective and precise working of the internal combustion engine. In this study, a static simulation is conducted on a crankshaft from a single cylinder 4-stroke petrol engine. A three dimension model of crankshaft is created using CATIA software. Finite element analysis (FEA) is performed to get the variation of stress magnitude at critical locations of crankshaft. The preliminary static analysis will be done on existing design of crankshaft with FEA Software ANSYS. The boundary conditions are applied according to the engine mounting conditions. The deflection, stress and strain will be obtained from FEA study. The second step consists of Topology optimization of crankshaft for similar boundary conditions. The main purpose of optimization study is to reduce the weight of a crankshaft. The optimization will be carried out in Altair Optistruct software. The existing design will be divided in design and non-design space. Geometry, material, and manufacturing processes will be optimized considering special constraints, developing feasibility, and cost. The optimization process include geometry changes well matched with the current engine, reduce weight compared to existing design, cost of the crankshaft, without changing other components.**

Index Terms— **FEM, FEA, CATIA, ANSYS.**

I. Introduction

In automotive engine Crankshaft is one of the most significant moving component which converts the reciprocating displacement of the piston into a rotary motion of crankshaft with a four link mechanism. Since the crankshaft experiences a large number of load cycles throughout its Service life, fatigue performance and durability of this component have to be considered in the design process. Design development is always an important matter in the Crankshaft production industry, in order to produce a less expensive component with the minimum weight possible and proper fatigue strength and other functional needs. These improvements result in lighter and smaller engines with improved fuel efficiency
and high power output. The load acting on the crankpin are complex in nature. The piston and the connecting rod convey gas pressure from the cylinder to the crankpin. It also exerts forces on the crankpin, which is time varying. In this study one crankshaft model of Honda PCX 150cc is used to compute the effect of stresses. Crankshaft consists of the parts which rotate in the main bearings, the crankpin to which the big ends of the connecting rod are connected, the crank arm or web which is connected to the crankpin and the shaft parts. The crankpin is manufacture in beam with a distributed load along its length that varies with crank position. Reasons for Failure of crankshaft assembly and crankpin due to

A) Shaft misalignment
B) Vibration root out by bearings application
C) Incorrect geometry (stress concentration)
D) Inappropriate lubrication
E) High temperature
F) Overload
G) Crankpin material & its chemical composition
H) Force acting on piston.

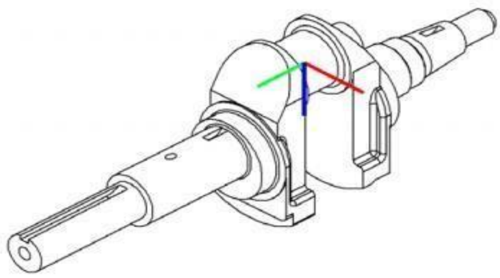

Fig.1 Typical four stroke engine Crankshaft

The crankshaft is manufactured by different modern processes; there are sand cast, wrought forged, and powder metallurgy. The materials used for crankshafts are medium carbon steels (having 0.35 to 0.45 percent carbon) and alloy steels (Chromium-nickel or chromium-molybdenum steels).These alloys are used for different applications depending upon the ultimate tensile strength required for the particular application. Till now, huge research is going on in the field of metallurgy. And resulted in large number of newly developed materials are available to select materials and its particular applications. Focusing on this concern, in this study the Crankshaft is modulate and simulated by using CATIA software and ANSYS is used for analysis.

II. FINITE ELEMENT MODEL

Fig.2 shows the 3-Dimensional model in CATIA. As the crankshaft is of a single cylinder four stroke petrol engines are used in two wheelers, it does not have a flywheel attached to it, a vibration damper and oil holes building the modeling even simpler. The dimensions of crankshaft are listed in Table 1.

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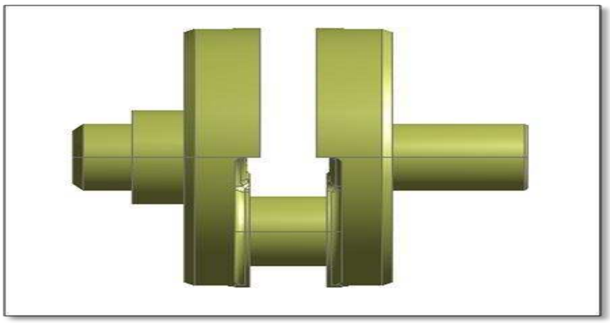

Fig 2. CAD model of Crankshaft - Pulsar 150

III. STRESS CALCULATION USING FEM

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

A. Modeling

The CAD model of crankshaft is generated in the CATIA software. CATIA is the high end solid modeling software of Dassault's system incorporation. CATIA Geometry export in the form of '.step' format and same has been imported in Ansys design modeler software.

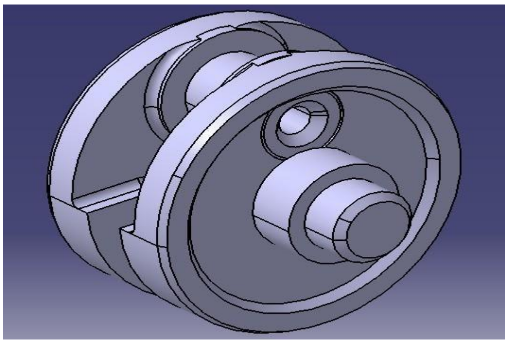

Fig.3 CATIA geometry model of the crankshaft (Pulsar-150 DTS-i)

B. Meshing

Greater the fineness of the mesh better the accuracy of the results. The Fig. 4 shows the meshed model in ANSYS.
Crankshaft mesh contains: Mesh
Size – 2.5 mm
Number of nodes – 313760
Number of elements – 217407

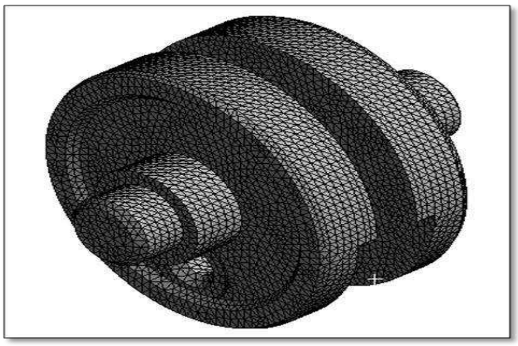

Fig. 4 Mesh Model of Crankshaft

C. Defining Material Properties

The ANSYS demands for material properties which are defined using module ENGINERING DATA. The material used for crankshaft is **Carbon Steel – 41Cr4.**

Table – 1 Properties of Carbon Steel

S.No.	**Components**	**Symbol**	**Percentage %**
1	Iron	Fe	95.3 – 98.1
2	Chromium	Cr	0.5 – 0.9
3	Carbon	C	0.34 – 0.43
4	Silicon	Si	0.18 – 0.53
5	Sulphur	S	0 – 0.04
6	Phosphorous	P	0 – 0.035

Table – 2 Mechanical Properties

Sr. No.	**Material**	**Specifications**	**Values**
1	Carbon Steel 41Cr4	Density , Kg/mm3	8000
2		Young's Modulus, MPa	200000
3		Poisson's Ratio	0.3
4		Tensile strength, MPa	650-880
5		Yield Strength, MPa	350-550
6		Elongation	8%
7		Fatigue Strength, MPa	275

D. Loading conditions

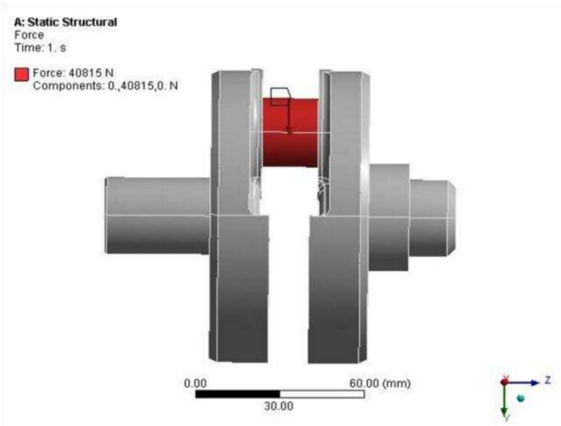

Fig. 5 Load application on the Crankshaft

The forces from the connecting rod are acting on the crankpin. The forces on crank pin are calculated with the help of analytical calculations. The crankpin subjected to maximum force **40815 N**. The force is applied on the face of the crankpin in vertical downward directions so as to simulate the actual scenario of the crankshaft and crankpin. Above fig. 5 shows Load in red colour acting on crankpin of the crankshaft.

E. Constraints

To simulate the Crankshaft design to forces acting from gas pressure and connecting rod, the two side of the crankshaft are constraint in all directions. The solid element has 3 degrees of freedom. The rotational degrees of freedom are not present in solid elements Therefore, three translational DOF's like x, y and z directions are fix in all directions as shown in the fig. 6.
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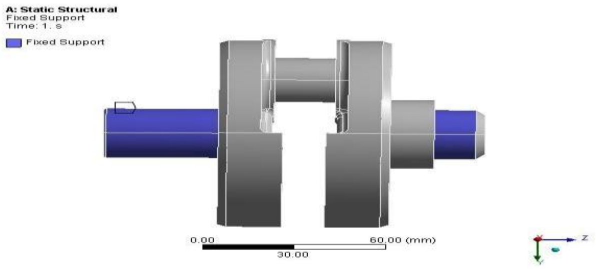

Fig. 6 Post Processing of Baseline Model

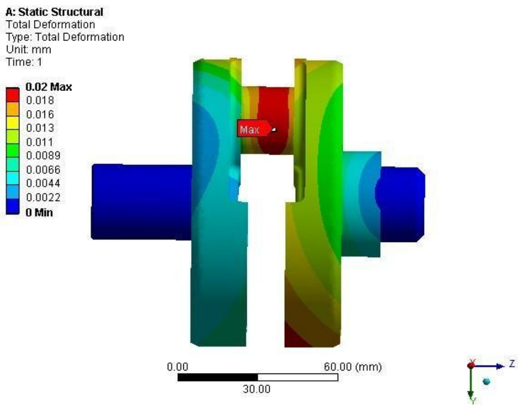

Fig. 7 Total Deformation contour plots (Baseline design)

The displacement contour plots are shown in the below figure. The maximum displacement shown by the crankshaft is 0.018 mm. As per distortion energy theory, the maximum equivalent stress observed in the crankshaft model 204 MPa. The yield strength of the material is 350 MPa. According to results, the von-Mises stress 204 MPa is less than yield strength of the material. The factor of safety of the baseline crankshaft is 1.71.
The elastic strain observed in the crankshaft is 0.0015452. The following pictures show contour plots of the von-Mises stress and elastic strain.

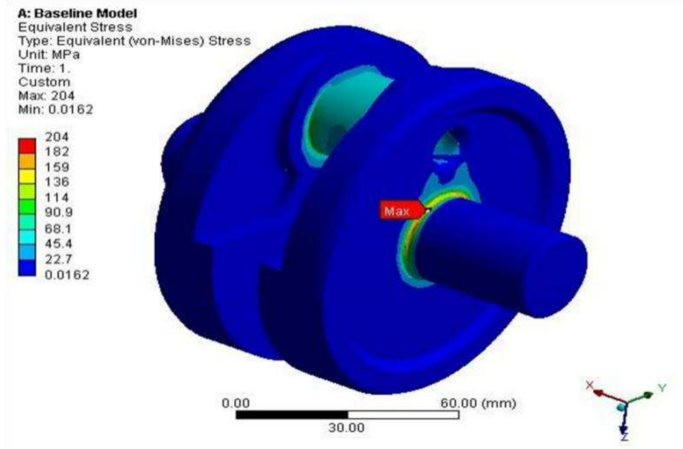

Fig.8 Equivalent stress contour plots (Baseline design)

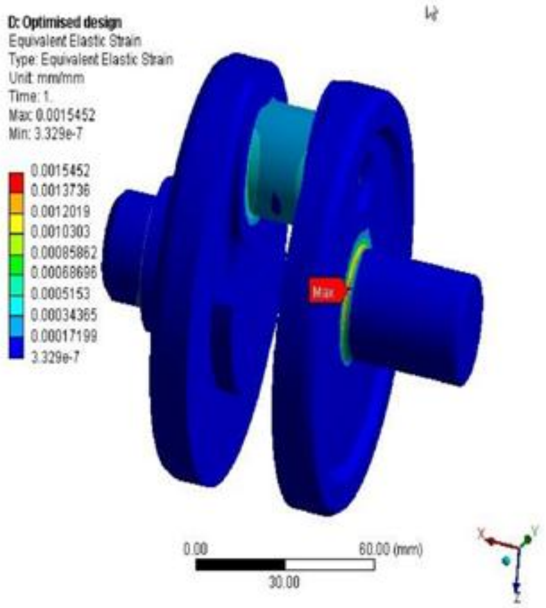

Fig 9 Equivalent elastic strain contour plots (Baseline design)

IV. TOPOLOGY OPTIMIZATION

Topology optimization is perform on a model to generate a new topology for the structure, removing any unnecessary material. The resulting structure is lighter and satisfy all design constraints. The topology optimization of crankshaft model is carried out in Altair's Optistruct software. The Material data for 41Cr4 carbon steel remain same as used in the static structural analysis. The shape optimization includes same boundary conditions as used in the static analysis of baseline model.
Topology optimization carried out for the following objective.

A. Problem definition

Optimization Objective: To minimize volume (reduce weight)
Constraint: von-Mises stress < 250 MPa (Yield strength of

material)

Design variables: The density of each element in the design space.

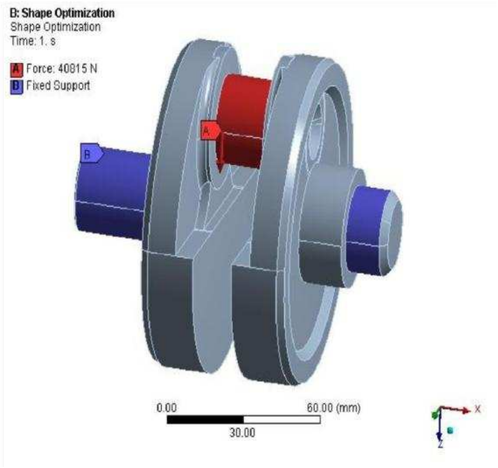

Fig.10 Boundary Conditions for Topology Optimization

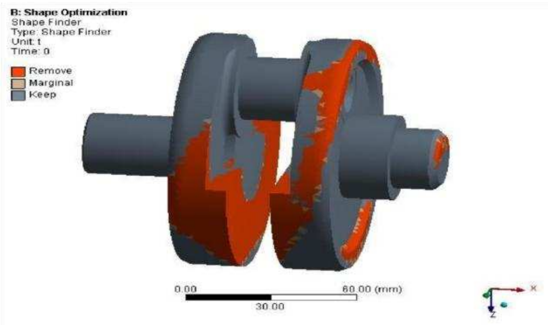

Fig.11 Optimized Geometry

The red region in crankshaft indicates the unnecessary material to be removed from the crankshaft. The optimized design is extracted from the raw design obtained through analysis. The optimized design is prepared in the CATIA software.

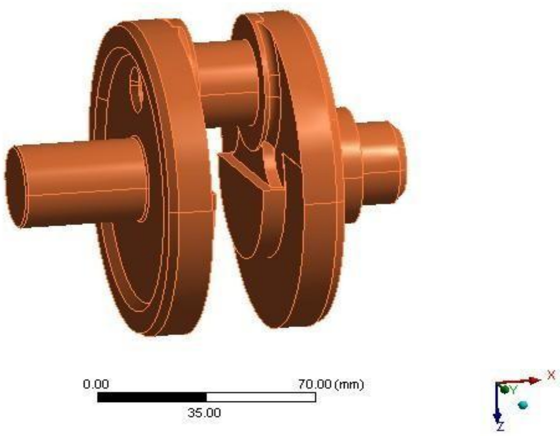

Fig.12 Optimized geometry

V. STATIC ANALYSIS OF OPTIMIZED DESIGN OF CRANKSHAFT

A. *Geometry*

As per optimization criteria, the material is removed from baseline CAD model in CATIA. The optimized model is shown in the figure below. CATIA model of optimized design is exported in the form of '.step' format and same has imported in Ansys.

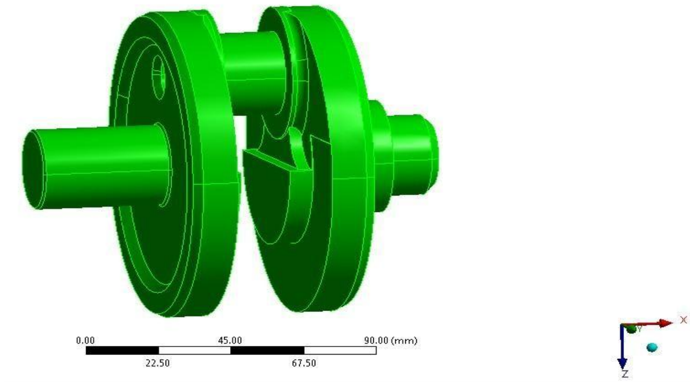

Fig.13 Optimized CATIA geometry

After performing the topology optimization analysis, Second-order tetrahedron elements (SOLID 187) are used for whole crankshaft. The sufficient finer mesh has selected for the optimized model to get appropriate results. SOLID187 element is a higher order 3-D, 10node element. SOLID187 has quadratic displacement behaviour and is well suitable to modeling asymmetrical meshes (such as those produced from various CAD/CAM systems).

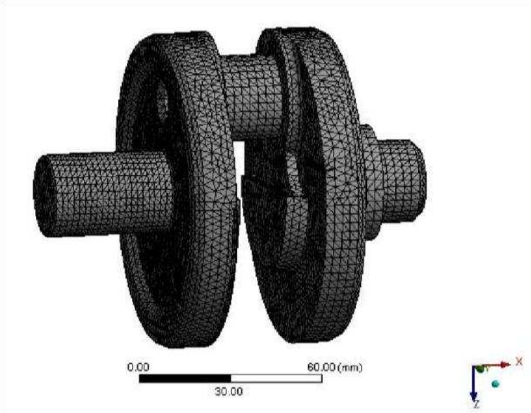

Fig.14 Mesh Model Optimized design

The element is defined by 10 nodes having three degrees of freedom at each node: translation in the nodal x, y, and z directions. The element has plasticity, hyper elasticity, creep, stress stiffening, large deflection, and large strain capabilities. Crankshaft is meshed with 10 node Tetrahedron second order element shown in the image below. Crankshaft mesh contains

Number of nodes - 185419

Number of elements - 124685

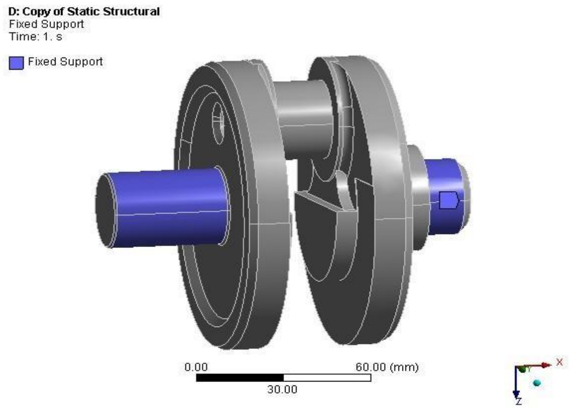

Fig.15 Constraints of optimized geometry

The constraints are remaining same as per baseline design model. The above picture shows nodes in the blue colour region are fixed in all DOF's. and the forces are acting on the crankpin along vertical downward direction.

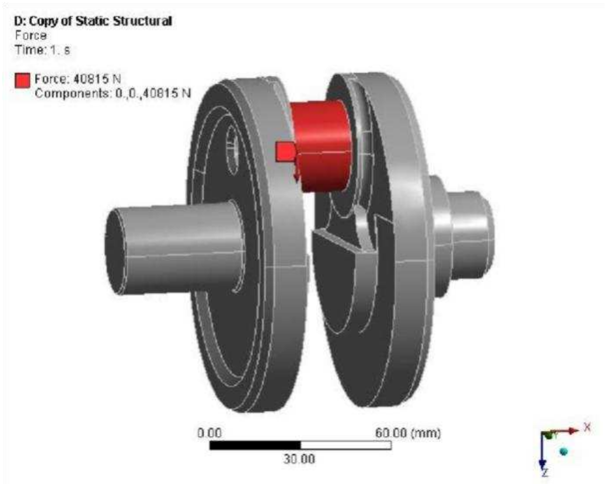

Fig.16 Loading Scenario of Optimized Geometry

B. Post Processing of Optimized model

The effect of structural loading on the optimized design of Crankshaft was investigated by comparing the stress distributions within the model under the loading scenarios. The deflection and von-Mises stress contour plots for optimized crankshaft are shown in the figure below.

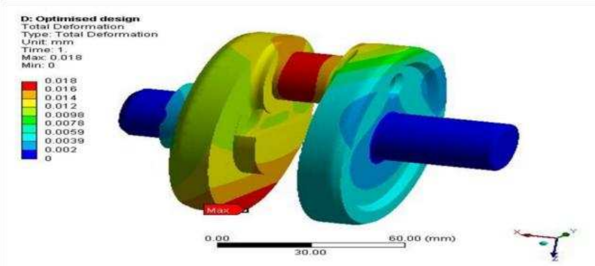

Fig.17 Deflection Contour Plot of Optimized Geometry

The maximum deflection of the optimized crankshaft is 0.018 mm. Optimized crankshaft shows maximum von-Mises stress up to 241 MPa. The observed value occurs within the yield strength of the material. The tensile yield strength of the material is 350 MPa. The factor of safety of the optimized design is calculated by following formula-

Factor of safety = (Yield strength of the material)/(Observed value of maximum stress) value of maximum stress) Factor of safety = 350/241

Factor of Safety = 1.45

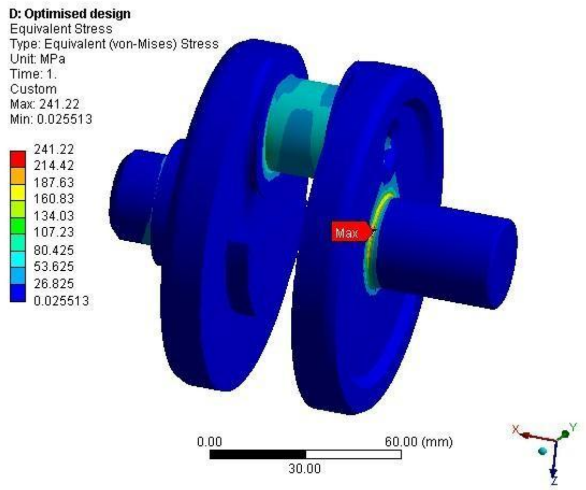

Fig.18 Von-Misses Stress Contour Plot of Optimized Geometry .

As per distortion energy theory, the stresses are observed within yield criteria of the material. Therefore, the crankshaft is safe for applied structural loading conditions. The stresses at crankpin corner are termed as singular stresses. These high stresses exist in close proximity of the crank and crankpin.

VI. RESULT AND DISCUSSION

The comparison between baseline and optimized design of the crankshaft for vonMises stress shows better improvements. The stress increased from 204MPa to 241MPa but, the stress values always remain within yield criteria of material. The deflection in crankshaft for baseline design and optimized design are 0.0182 mm and 0.018 mm respectively. The total mass of the baseline model was 3.5 Kg which is decreased up to 2.95 Kg. The percentage reduction in mass of the crankshaft observed up to 19**%.** The baseline analysis is compared with the testing of the baseline specimen. The maximum variation observed between simulation and testing is up to 18.64% which is acceptable.

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