Topo1ogy Optimisation Of Crankshaft Using FEA Technique

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ABSTRACT

The main objective of this study is to investigate weight and cost reduction opportunities for a crankshaft. The need 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 critical components for the effective and precise working of the internal combustion engine. In this paper a static simulation is conducted on a crankshaft from a single cylinder 4-stroke petrol engine. A three dimension model of petrol engine crankshaft is created using Pro-E software. Finite element analysis (FEA) is performed to obtain the variation of stress magnitude at critical locations of crankshaft. Simulation inputs are taken from the engine specification chart. The preliminary static analysis will be done on existing design of crankshaft using FEA Software ANSYS. The boundary conditions are applied according to the engine mounting conditions. The analysis is done for finding critical location in crankshaft. The deflection, stress and strain will be obtained from FEA study. The second step consists of Topology optimization of crankshaft for same boundary conditions. The main objective of optimization study is to reduce the weight of a crankshaft. Geometry, material, and manufacturing processes will be optimized considering different constraints, manufacturing feasibility, and cost. The optimization process included geometry changes compatible with the current engine, reduce weight compared to existing design, cost of the crankshaft, without changing connecting rod and/or engine block.

Keywords — Crankshaft, FEM, ANSYS, Topology, Optimization etc.

I. INTRODUCTION

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. The Forces acting on the crankpin are 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. In this project one crank model of TVS Scooty pep will used to calculate the effect of stresses. Crankshaft consists of the parts which revolve in the main bearings, the 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. The crankpin is like a build in beam with a distributed load along its length that varies with
reasons for failure of crankshaft assembly and crankpin may be –
A) Shaft misalignment
B) Vibration causes by bearings application
C) Incorrect geometry (stress concentration)
D) Improper lubrication
E) High engine temperature
F) Overloading
G) Crankpin material & its chemical composition
H) Pressure acting on piston

II. ENGINE SPECIFICATION

<table>
<thead>
<tr>
<th>Engine specifications</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1  Cylinder Bore</td>
<td>51 mm</td>
</tr>
<tr>
<td>2  Stroke</td>
<td>43 mm</td>
</tr>
<tr>
<td>3  Piston Displacement</td>
<td>87.8 cc</td>
</tr>
<tr>
<td>4  Compression ratio</td>
<td>10:1:1</td>
</tr>
<tr>
<td>5  Maximum Power in KW</td>
<td>3.68 @ 6500RPM</td>
</tr>
<tr>
<td>6  Maximum Torque in Nm</td>
<td>5.80 @ 4000RPM</td>
</tr>
<tr>
<td>7  Maximum Speed</td>
<td>60Km/Hr</td>
</tr>
<tr>
<td>8  Bearing Pressure</td>
<td>7-12.5N/mm²</td>
</tr>
</tbody>
</table>

III. OBJECTIVE

Analyze the stresses acting on crank pin due to the gas force. Evaluate maximum deformation, maximum stress point and dangerous areas of failure. Carry out topology optimization on existing design to reduce the weight and cost.

IV. METHODOLOGY

Let P = max. Pressure of gas
D = Diameter of piston
m = mass of reciprocating parts
ω = angular speed of crank
θ = angle of inclination of crank from top dead centre
Ø = angle of inclination of connecting rod with the line of stroke
r = radius of crank
l = length of connecting rod
n = ratio of (l/r)
dc = diameter of crank pin
lc = length of crank pin
FL = Force on piston due to gas pressure i.e. (p*A)
FI = Inertia force of reciprocating part i.e. (mR*ω²*r (cosθ+cos²θ/n))
Fp = net force on crank pin
FL±FI = force on connecting rod
Fc = Fp/cosØ
FL = inertia force on crank pin m*r*ω²
Pc = load on crank pin dc*lc*pbc
FT = tangential force on crank pin FQ sin (θ+Ø)
FR = radial force on crank pin FQ cos (θ+Ø)
HT1 and HT2 = reaction at bearing due to FT FT*b1/b and FT*b2/b
HR1 and HR2 = reaction at bearing due to FR FR*b1/b and FR*b2/b
Mc = Max. Bending moment on crank pin
Te = Max. twist moment on crank pin
Te = equivalent twist moment on crank pin √ [(Mc²) + (Te²)]
τ = max. Shear stress on pin Te/(π/16)*dc³

Let max. Bearing Pressure (Pb) = 12.5 N/mm²
So, load on crank pin (FL) = dc*lc*Pb = (23*40*12.5) = 11500N. Max. Gas pressure on piston (P) = (p/4 * D²)/(FL) = 5.62 N/mm² Net force acting on crank pin (Fp) = (FL), neglecting inertia force. Thrust on crank pin (Fq) = FL/cosØ.

Where θ = 28, Fq = 13024.5 N
V. FEA STUDY METHODOLOGY

When performing the Static Structural analysis, Second-order tetrahedron elements (SOLID 187) are used for whole crankshaft. The sufficient finer mesh has selected for the model to get appropriate results. SOLID187 element is a higher order 3-D, 10-node element. SOLID187 has quadratic displacement behaviour and is well suited to modelling irregular meshes (such as those produced from various CAD/CAM systems). The element is defined by 10 nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions.

FLOWCHART

VI. 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 like x, y and z directions are fix in all directions as shown in the figure below.

VII. MATERIAL SPECIFICATIONS

The 42CrMo4 steel material properties are specified in the analysis. The static structural analysis requires specification elastic modulus, which is 2.1E+5 MPa, and Poisson’s ratio 0.3. The density of the material is also specified, which is 7850 kg/m³. The following table shows material data for crankshaft.

<table>
<thead>
<tr>
<th>Material Details</th>
<th>Material Type:</th>
<th>Forged Steel Designation:</th>
<th>42CrMo4</th>
<th>Yield strength (MPa):</th>
<th>680</th>
<th>Ultimate tensile strength (MPa):</th>
<th>850</th>
<th>Elongation (%):</th>
<th>13</th>
<th>Poisson ratio:</th>
<th>0.3</th>
</tr>
</thead>
</table>

Boundary Conditions: - The forces on crank pin are calculated with the help of analytical calculations. The crankpin subjected to maximum force 13024.5N. 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. Following figure shows load in red color acting on crankpin of the crankshaft.

VIII. POST PROCESSING AND RESULTS

The displacement contour plots are shown in the below figure. The maximum displacement shown by the crankshaft is 0.007 mm. As per distortion energy theory, The maximum equivalent stress observed in the crankshaft model 140 MPa. The yield strength of the material is 680 MPa.
IX. TOPOLOGY OPTIMIZATION

The topology optimisation of crankshaft model is carried out in workbench module of shape Optimisation. The Material data for 42CrMo4 steel remain same as used in the static structural analysis.

![Fig. Optimization Model of Crankshaft](image)

X. CONCLUSIONS

Analytically it is to be concluded that the value of Max. Stress is 140 N/mm² and Max. Deflection is 0.007 mm which is less than the value of allowable stress value of material. The relationship between the crank rotation and force acting on crank pin would provide a valuable theoretical foundation for the material selection and design optimization of crankpin and engine design. The weight of the baseline model is higher as compared to optimized model. The Optimized model is obtained for same boundary conditions. The max. Deflection 0.01 mm for optimized model and max. Stress observed is up to 180 N/mm². The value observed below the yield strength of material. The Weight of baseline model is 970 gm. The weight of the optimized model is 820 gm. The percentage reduction in mass of the crankshaft is 9 %.

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