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Stress Analysis of Crankshaft Subjected To Dynamic Loading

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Abstract: *In this study a dynamic simulation was conducted on crankshaft, from multi cylinder four stroke engines. Finite element analysis was performed to obtain the variation of stress magnitude. The load diagram was used to calculate the load boundary condition. This load was then applied to the FE model and boundary conditions were applied according to the engine mounting conditions. From this analysis a model of crankshaft is generated in PRO -E, meshes by the HYPER MESH and analyzed in ANSYS using FEM (finite element method) by applying loads and boundary conditions, and then solved for engineering responses and Stress variation over the engine cycle. The scope of the FE Analysis of the crank shaft under static, modal and transient analysis was carried out to predict the stresses and its deformations.*
Keywords: *Dynamic Simulation, Crank Shaft Under Static.*

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.

II. Design Procedure

The following procedure may be adopted for a crankshaft. find the magnitude of the various loads on the crankshaft. Determine the distances between the supports and their position with respect to the loads. For the sake of simplicity and also for safety, the shaft is considered to be supported at the centre of the bearings and all the forces and reactions to be acting at these points. The distances between the supports depend on the length of the bearings, which in turn depend on the diameter of the shaft because of the allowable bearing pressures. The thickness of the cheeks or webs is assumed to be from 0.4ds to 0.6ds, where ds are the diameter of the shaft. It may also be taken as 0.22D to 0.32D where D is the bore diameter of cylinder in mm. Now calculate the distances between the supports. Assuming the allowable bending and shear stresses, determine the main dimensions of the crankshaft.

III. Indentations and Equations

Design of the crankshaft when the crank is at the dead centre
Bore diameter D = 160mm
Stroke L = 120 mm
Mean effective Pressure pm = 0.5 N/mm2
Combustion pressure p = 2N/mm2
Flywheel weight W = 50 KN
Total belt pull (T1+T2) = 17.51 KN Pressure on piston p '= 1 N/mm2
Crank angle θ = 35º We know that the piston gas load,
Fp = (Л/4) x D2 x p (3.1) = (Л/4) x (160)2 x 2 = 40.21 KN
Assume that the distance (b) between the bearings 1& 2 is equal to the piston diameter (D).
b = 2 x (200+200) = 800 mm
And b1 = b2 = (b/2) (3.2) = (800/2) = 400mm
We know that due to the piston gas load, there will be two horizontal reactions H1 & H2 at bearings 1&2respectively,
such that H1 = {(Fp x b1)/b} (3.3) = {(40.21 x 400)/800} = 20.1 KN
And H2 = {(Fp x b2)/b} (3.4) = {(40.21 x 400)/800} = 20.1 KN
Assume that the length of the main bearings to be equal, i.e. c1 = c2 = c/2.
We know that due to the weight of the flywheel acting downwards, there will be two vertical reactions V2 & V3 at bearing 2 & 3 respectively, such that V2 = {(W x c1)/c} = {[W x (c/2)]/c} (3.5) = W/2 = 50/2 = 25 KN
And V3= {(W x c2)/c} = {[W x (c/2)]/c} (3.6) = W/2 = 50/2 = 25 KN

Due to the resultant belt tension (T1 + T2) acting horizontally, there will be horizontal reactions H2' & H3' respectively, such that
H2'= {[(T1 + T2) c1]/c} (3.7) = (T1 + T2)/2 (3.8) = 17.51/2 = 8.755 KN [c1 = (c/2)]
H3' = {[(T1 + T2) c2]/c} (3.9) = (T1 + T2)/2 = 17.51/2 = 8.755 KN [c2 = (c/2)]

Design Of Crankpin
Let dc = Diameter of the crankpin in mm, lc = Length of the crankpin in mm, σb = Allowable bending stress for the crankpin in N/mm2. For Plain carbon steel σb = (480/5) = 96 N/mm2 [From the PSGDB Page.No:1.8] We know that bending moment at the centre of the crankpin, Mc = H1 x b2 (3.10) = 20.1 x 400 = 8040 KN-mm We also know that Mc = (Л/32) x (dc) 3 x σb (3.11) = (Л/32) x (dc) 3 x 96 =9.424 (dc) 3 N-mm Mc = 9.424 x 10-3 (dc) 3 KN-m(dc) 3 = 3946/ 9.424 x 10-3 =94.85 mm dc = 94.85 ≈ 100 mm We know that length of the crankpin IC =Fp/ dc x pb = 40.21 ≈ 40 mm Where [Taking Permissible bearing pressure pb= 10 N/mm2]

Design Of Left Hand Web
We know that thickness of the crank web, t = 0.65 dc + 6.35 mm (3.12) t = (0.65 x 100) + 6.35 = 71.3 ≈ **75 mm** And width of the web, w = 1.125 dc + 12.7 mm (3.13) w = (1.125 x 100) + 12.7 = 125.2 ≈ **130 mm**
We know that maximum bending moment on the crank web, M = H1 b2 (lc/2) (t/2) (3.14)
= 9.815 x 400 (40/2) (75/2)
= 6984.75 KN-mm
Bending stress (σb), σb= 59.60 N/mm2
And direct compressive stress on the crank web, σc = 2.06 N/mm2
Total stress on the crank web = Bending stress + direct compressive stress = σb +σc = 59.6 + 2.06 = 61.66 N/mm2 (or) MPa Since the total stress on the crank web is less than the allowable bending stress of 96 MPa, therefore, the design of the left hand crank web is safe.

Design Of Web Right Hand Crank
From the balancing point of view, the dimensions of the right hand crank web (i.e. thickness and width) are made equal to left hand web.

Design Of Shaft Under The Fly Wheel
Let ds= Diameter of the shaft in mm, Since the lengths of the main bearings are equal, therefore l1 = l2 = l3 = 2[(b/2) - (lc/2) - t] (3.16) =2 [400 - 15- 60] = 650 mm
Allowing space for gearing and clearance, let us take c = 800 mm. c1 = c2 = (c/2) = (800/2) = 400 mm
We know that bending moment due to the weight of flywheel, MW= V3 .c1 (3.17) = 25 x 400 = 10,000 KN-mm = 10 x 106 N-mm
And bending moment due to belt tension,
MT= H3'. c1 = 8.755 x 400 = 3502 KN-mm = 3.5 x 106 N-mm
Therefore, the resultant bending moment on the shaft,
MS = (MW) 2 + (MT) 2 1/2 = (10 x 106) 2 + (3.5 x 106) 2 1/2 MS = 10.56 x 106 N-mm We also know that the bending moment on the shaft,
10.56 x 10 6 = (Л/32) x (ds) 3 x σb = (Л/32) x (ds) 3 x 96 ds =103.96 mm ≈ **110 mm**

IV. Figures and Tables

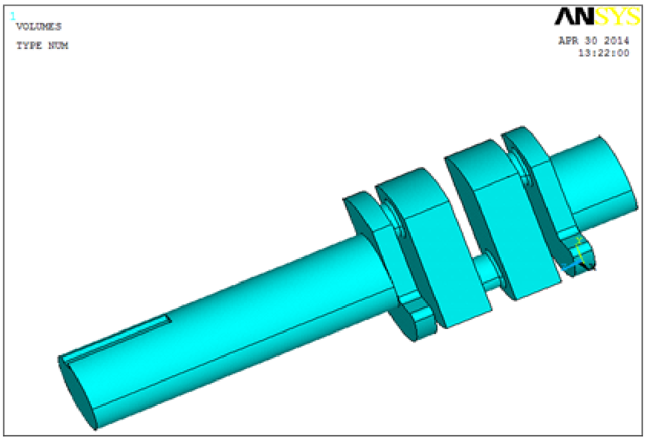

Fig: 1 Ansys Drawing Of Modified Crank Shaft

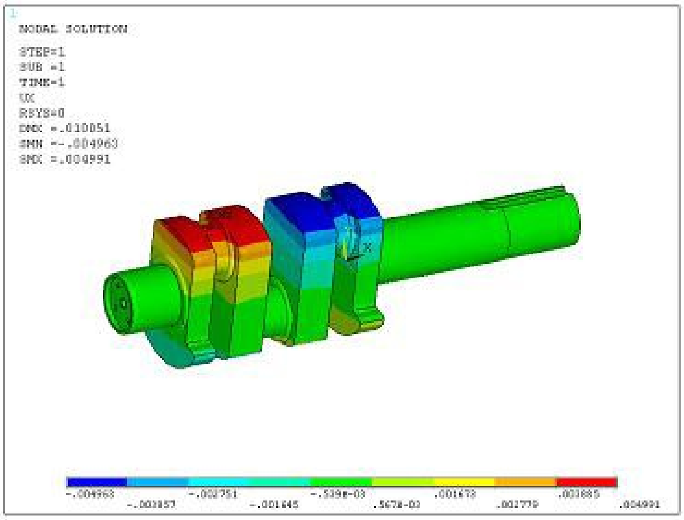

Fig: 2 Displacements In X Direction

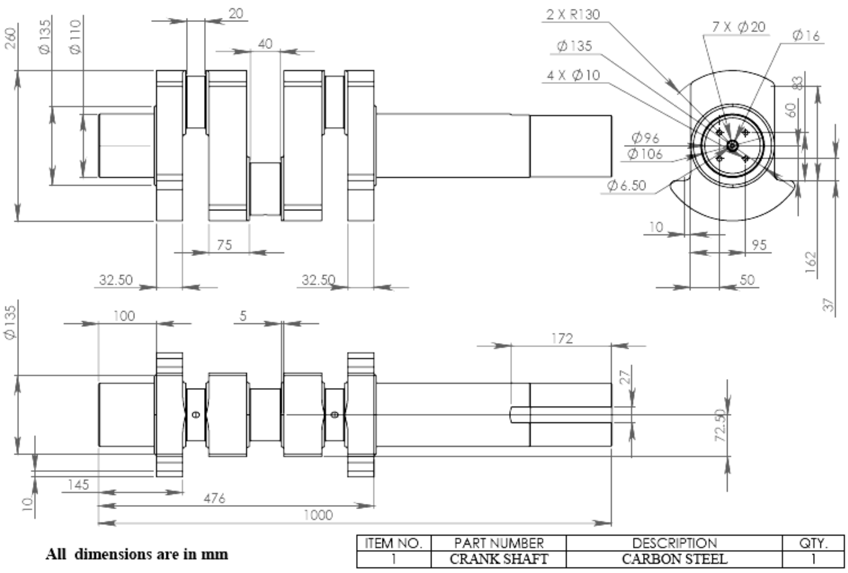

Fig: 3 Detail Drawing Of Crank Shaft

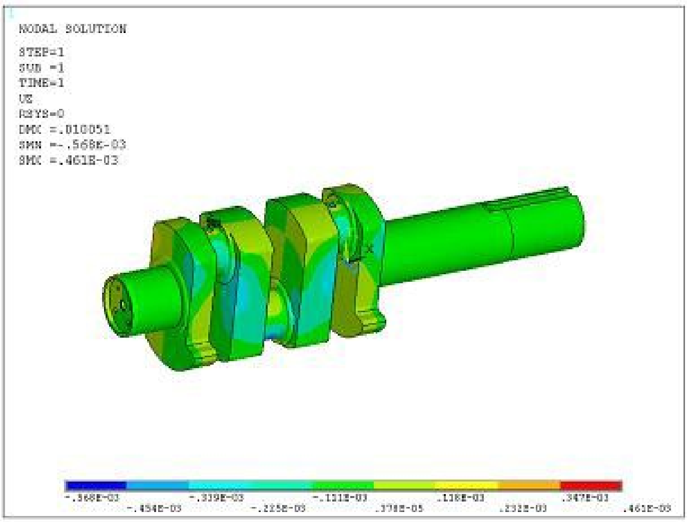

Fig: 4 Displacements In Y Direction

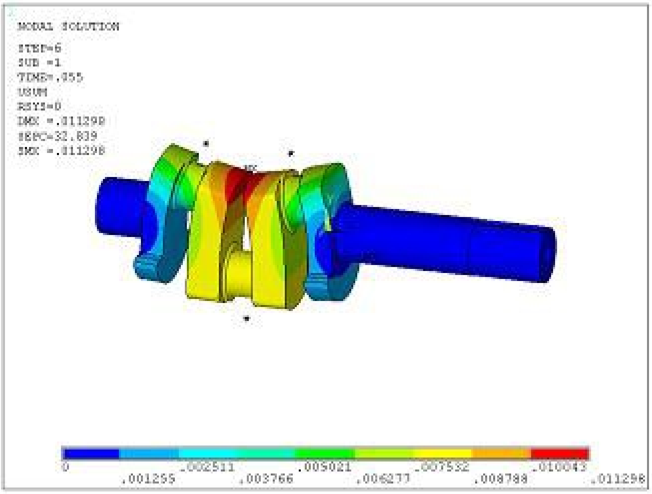

Fig: 5 Equivalent Displacement Diagram

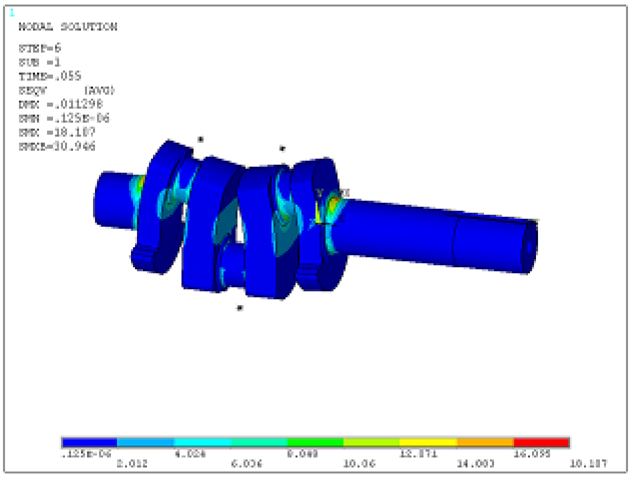

Fig: 6 Equivalent Stress Displacement Diagram

Sl .No	Mechanical Terms	Theoretical Value	Static Analysis	Modal Analysis	Transient Analysis
1	Shaft Diameter In Mm	110	110	110	110
2	Shear Stress In Mpa	43.23	-	-	30.7

Table: 1 Value Of Analysis

V. Conclusion

Thus the Crankshaft is designed based on the shaft whose diameter **110 mm**.

1. 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. Considering torsional load in the overall dynamic loading conditions has no effect on von misses stress at the critically stressed location (in fillet areas). 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.

2. Geometry optimization resulted in 18% weight reduction of the forged steel crankshaft, which was achieved by changing the dimensions and geometry of the crank webs while maintaining dynamic balance of the crankshaft. This stage of optimization did not require any changes in the engine block or connecting rod.

3. Adding fillet rolling was considered in the manufacturing process. Fillet rolling induces compressive residual stress in the fillet areas, which results in 165% increase in fatigue strength of the crankshaft and increases the life of the component significantly.

4. Using micro alloyed steel as an alternative material to the current forged steel results in the elimination of the heat treatment process. In addition, considering better mach inability of the micro alloyed steel along with the reduced material cost due to the 18%.

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