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Modelling & Analysis of a (1000cc, Single Cylinder Engine) Crankshaft using ANSYS Software

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Abstract

Crankshaft is large volume manufacture constituent with a multifaceted geometry in the I.C Engine. This transforms the reciprocating or the to and fro movement of the piston into a rotary motion of the crank. An effort is made in this research paper to comprehend and study the Static analysis on a crankshaft from a single cylinder 1000cc 4-stroke I.C Engine. The designing of the crankshaft is done using CATIA-V5 Software. Finite element analysis (FEA) is carried out to gain the variation of stress at critical and serious locations and positions of the crank shaft using the ANSYS software and applying the suitable boundary conditions as per the need. Franz Langer held the record for the biggest and the largest displacement single cylinder motorbike engine, with a 1000cc thumper built in 1998; and this invention really made a dent in the universe. Further the enhancements were made in this design and presently the beast holds a capacity of 2000 cc, commissioning a single cylinder only.
Keywords: Crankshaft, Finite Element Analysis (FEA), ANSYS Software, Static Analysis

I. INTRODUCTION

Crank shaft is a big component with a complex geometry in the I.C engine, which converts the reciprocating drive of the piston to a rotary motion with a four-bar link mechanism.

Crankshaft involves shaft parts, two journal bearings and one crankpin bearing. In addition to this, the linear displacement of an engine is not smooth; as the displacement is caused by the combustion chamber therefore the displacement has sudden shocks. The idea of employing crankshaft is to modify these sudden displacements to a very even rotary output, which is obviously the input to various devices such as generators, pumps and compressors. It should also be specified that the use of a flywheel helps in levelling the shocks.

Combustion and inertia forces acting on the crankshaft.
1) Torsional load
2) Bending load.

Crankshaft must be robust enough to take the downward force of the power stroke without excessive bending so the reliability and life of the internal combustion engine depend on the strength of the crankshaft largely.

The crank pin is like a built-in beam with a distributed load along its length that varies with crank positions. Each web is like a cantilever beam subjected to bending and twisting.
1) Bending moment which causes tensile and compressive stresses.

2) Twisting moment causes shear stress.

There are many sources of failure in the engine one of the most common crankshaft failure is fatigue at the fillet areas due to the bending load causes by the combustion. The moment of combustion the load from the piston is transmitted to the crankpin, causing a large bending moment on the entire geometry of the crankshaft. At the root of the fillet areas stress concentrations exist and these high stress range locations are the points where cyclic loads could cause fatigue crank initiation leading to fracture.

II. RECOGNITION OF NEED

Franz Langer held the record for the largest displacement single cylinder motorcycle engine, with a 1000cc thumper built in 1998.Further the improvements were made in this design and currently the beast holds a capacity of 2000 cc, employing a single cylinder only.

Now, for such a beast there must be a finely and carefully designed crankshaft to ensure the proper working of the I.C engine; thus, a need for designing this crankshaft was felt.

However, in our project we are trying to reproduce a crankshaft that would have been used in that original engine by following the fundamental steps of machine design.

III. PROBLEM STATEMENT (SPECIFYING THE PROBLEM)

To design a crankshaft for a NSU 1000 cc single cylinder thumper engine.

IV. MATHEMATICAL MODEL FOR CRANKSHAFT

Cylinder bore (in mm)	*125*
(L/R) ratio	*4.5*
Maximum gas pressure (in Mpa)	*2.5*
Length of stroke (in mm)	*150*
Weight of flywheel (in kN)	*1*
Total pull on sprocket (in kN)	*2*

The crank is at angle with the line of dead centre positions and subjected to maximum torsional moment.

1) Step 1: The crank angle (θ) for maximum torsional moment condition is given as 25° .

Since, p_1 is gas pressure on the piston top for maximum torque condition.

$P_P = (\pi D^2/4)\ p_1$

= 24543.69 N

$\sin \emptyset = sin\theta/(\frac{L}{r})$

= 0.09392

Thus, $\emptyset = 5.39°$

Now, the thrust on connecting rod (P_Q) is given by,

$P_Q = P_P/ \cos\emptyset$

= 24652.69 N

Therefore, $P_t = P_Q \sin(\theta+\emptyset)$

= 12471.38 N

$P_R = P_Q \cos(\theta+\emptyset)$

= 21265.46 N

2) Step 2: Bearing reactions

The forces acting on the centre crankshaft are at an angle of maximum torque.

The crankshaft is supported on three bearings.

Let us assume,

b = 250 mm and c = 300 mm

$b_1 = b_2 = b/2 = (250/2) = 125$mm

$c_1 = c_2 = c/2 = 150$ mm

By symmetry,

$(R_1)_v = (R_2)_v = (P_R/2) = 10632.73$ N

$(R_1)_h = (R_2)_h = (P_t/2) = 6235.69$ N

$(R_2')_v = (R_3')_v = 500$ N

$(R_2')_h = (R_3')_h = 1000$ N

3) Step 3: Design of crank pin the central plane of the crank pin is subjected to the bending moment M_b due to $(R_1)_v$ and torsional moment M_T due to $(R_1)_h$.

Let us assume, allowable shear stress = 40 Mpa.

The diameter of the crank pin (dc) is,

$d_c^3 = (16*\sqrt{(M_b^2 + M_T^2)})/2$
$= 179.4 * 10^3$
$d_c = 56.4$ mm
But, considering the bending stress also into account let us assume the $d_c = 65$ mm $= l_c$

4) Step 4: Design of shaft under flywheel

Suppose, d_S = diameter of shaft under flywheel (mm)
The central plane of the shaft is subjected to maximum bending moment due to the reaction R_3.
$M_b = (R_3)* c_2$
It is also subjected to torsional moment M_t due to tangential component P_t.
$M_T = P_T * r$
$R_3 = \sqrt{(500)^2 + (1000)^2}$
$= 1118.03$ mm
$d_s^3 = (16*\sqrt{(M_b^2 + M_T^2)})/\tau\pi)$
$= 120.99 * 10^3$
$d_s = 50$ mm

5) Step V: Design of shaft at the juncture of right-hand crank web

The cross-section of the shaft under flywheel at the juncture of the right-hand crank web is subjected to the following moments:
Bending moment in vertical plane, $(M_b)_v$
Bending moment in horizontal plane, $(M_b)_h$
Torsional moment M_T
$(M_b)_v = (R_1)_v (b_1 + l_c/2 + t/2) – P_r (l_c /2 + t/2)$
$= 738.97 *10^3$ N-mm
$(M_b)_h = (R_1)_h(b_1 + l_c/2 + t/2) – P_t (l_c /2 + t/2)$
$= 433.38 * 10^3$ N-mm
The resultant bending moment M_b is:
$856.68* 10^3$ N-mm
The diameter of the shaft d_{s1} is:
$d_{s1}^3 = (16*\sqrt{(M_b^2 + M_T^2)})/\tau\pi)$
$= 161.5 * 10^3$
$d_{s1} = 55$mm

6) Step 6: Design of right-hand crank web

The right-hand crank web is subjected to the following stresses:
Bending stresses in vertical and horizontal planes due to radial component Pr and tangential component Pt respectively.
Direct compressive stress due to radial component Pr.
Torsional shear stresses.
The bending moment due to the radial component is given by,
$(M_b)_r = (R_2)_v [b_2 – l_c/2 – t/2]$
$= 738.97 * 10^3$ N - mm
Also, $(M_b)_r = (\sigma_b)_r *(wt^2/6)$
Thus, $(\sigma_b)_r = 27.94$ Mpa
Also,
The tangential stress due to tangential component at the juncture of the crank web and shaft is:
$(\sigma_b)_t = 13.74$ Mpa
The direct compressive stress due to radial component is given by,
$(\sigma_c)_d = P_r/2wt = 3.08$ Mpa
The maximum compressive stress (sc) is given by:
$(\sigma_c) = (\sigma_c)_d + (\sigma_b)_t + (\sigma_b)_r$
$= 44.76$ Mpa
The torsional moment on the arm is
$M_t = (R_2)_h * (b_2 – l_c/2)$
$= 576.80 *10^3$ N-mm
The maximum compressive stress is given by,
$(\sigma_c)_{maX} = 0.5 * [\sigma_{c} + \sqrt{\{ (\sigma_c)^2 + 4\tau^2 \}}]$
$= 50.10$ Mpa

7) Step 7: Design of left-hand crank web

The left-hand crank web is not severely stressed to the extent of the right-hand crank web. Therefore, it is not necessary to check the stresses in the left-hand crank web. The thickness and width of the left-hand crank web are made equal to that of the right-hand crank web from balancing consideration.

8) Step 8: Crankshaft model on Catia

Date March 22, 2018

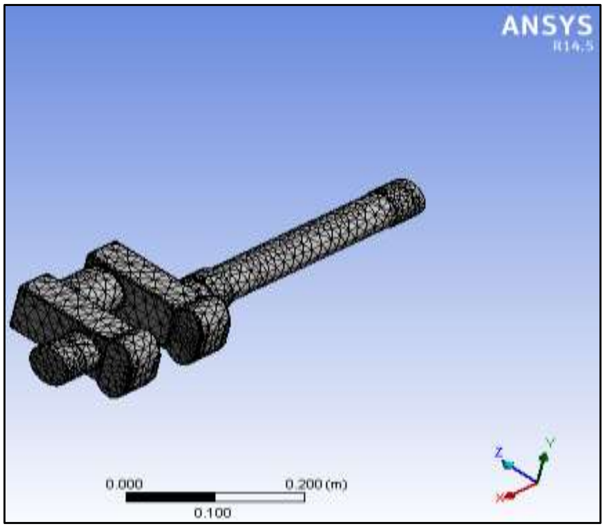

Fig. 1:

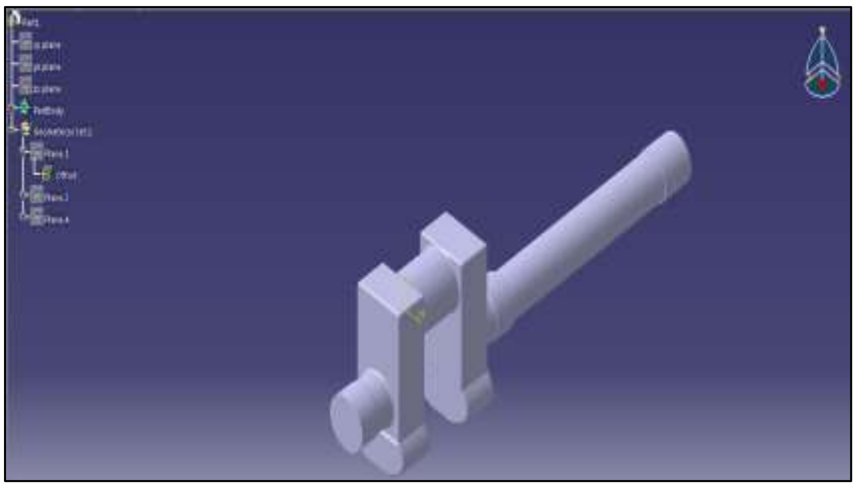

Fig. 2:

Mesh type= tetrahedron
30884=no of nodes
17824=no of elements

9) Step 9: Equivalent Stress analysis on ANSYS

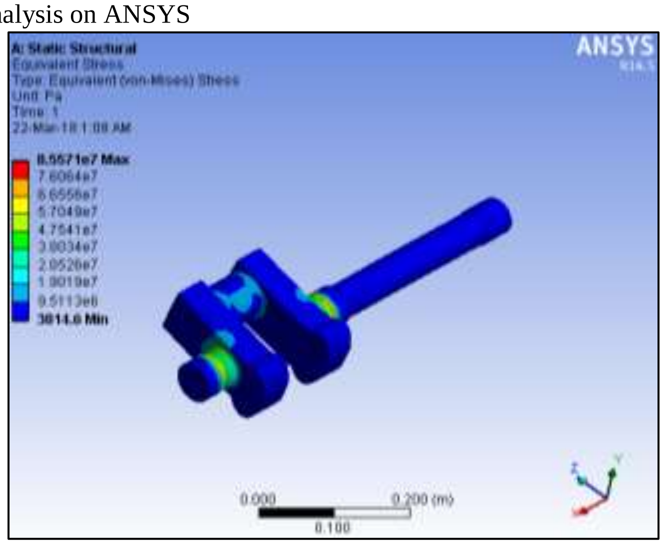

Fig. 3:

Maximum Shear Stress analysis on ANSYS

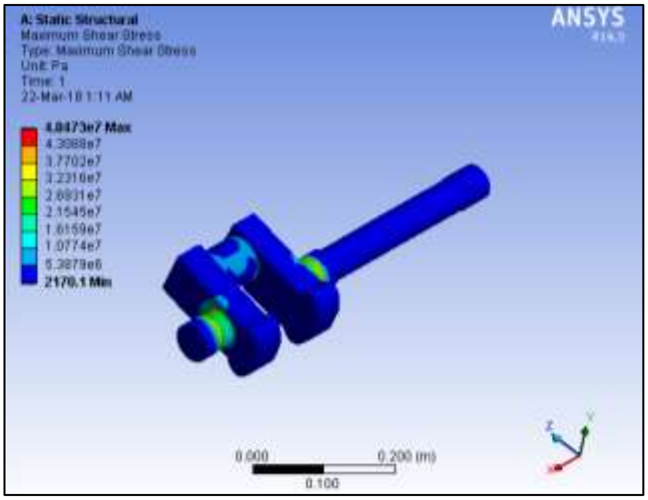

Fig. 4:

Safety Factor analysis on ANSYS

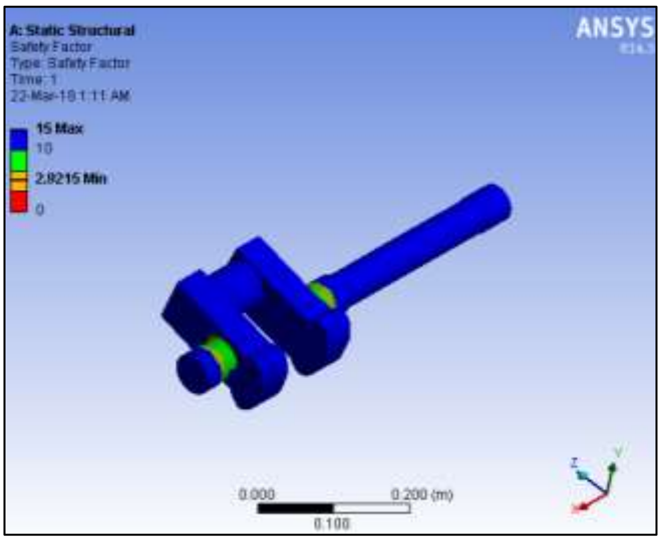

Fig. 5:

Maximum Shear Elastic Strain analysis on ANSYS

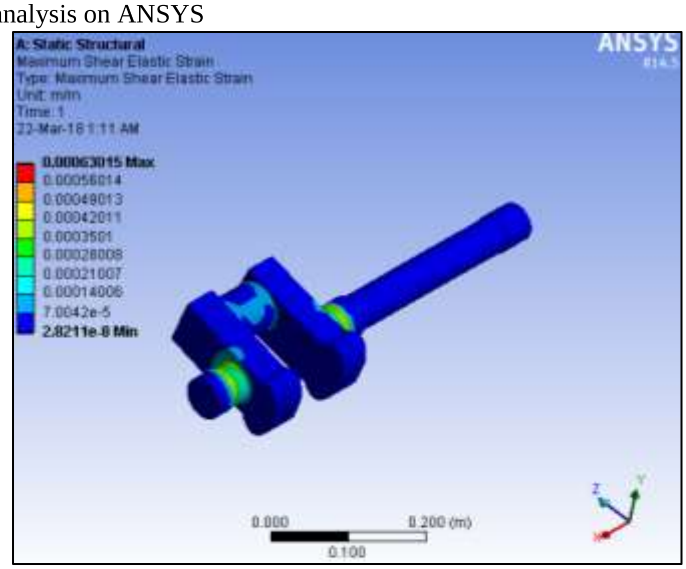

Fig. 6:

Total Deformation analysis on ANSYS

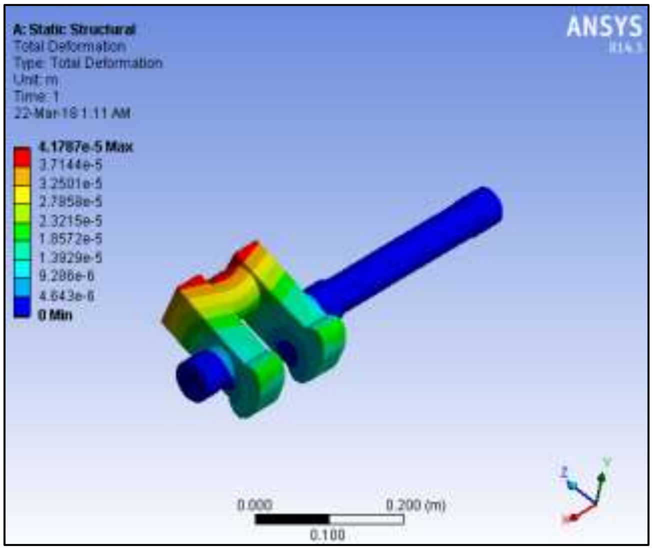

Fig. 7:

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