

COMPARATIVE ANALYSIS OF CRANKSHAFT OF PULSAR 180 DTS-I

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ABSTRACT

The crankshaft is also referred as crank. It is responsible for conversion between reciprocating motion and rotational motion. Here the failure of crankshafts for two wheelers mostly occurs in the crankpin. Thus the crankpin is an important component that mostly decides the life of the crankshaft. The crankshaft considered here is of Pulsar 180 DTSi. It is a petrol engine crankshaft made from Alloy steel 41Cr4. abnormal sound was heard in crankshaft while it is in operation. It was identified as failure of crankshaft. Severe wear has been observed at crankpin bearing location where the oil hole is provided. Here the analysis of the two wheeler crankshaft is done. Its results are then compared and verified numerically, by the use of ANSYS software. The results compared here are Von Mises Stresses, the strain occurring on the crankshaft and number of life cycles of the crankshaft.

Keywords: *Crankshaft, Crankpin, Strain, Stress, Force, Moment, Fatigue, Cycles.*

I. INTRODUCTION

1.1 Crankshaft

The crankshaft, also referred as crank, is responsible for conversion between reciprocating motion and rotational motion. In a reciprocating engine, it translates reciprocating linear piston motion into rotational motion, whereas in a reciprocating compressor, it converts the rotational motion into reciprocating motion. In order to do the conversion between two motions, the crankshaft has "crank throws" or "crankpins", additional bearing surfaces whose axis is offset from that of the crank, to which the "big ends" of the connecting rods from each cylinder attach.

The crankshaft main journals rotate in a set of supporting bearings also called as main bearings. They cause the offset rod journals to rotate in a circular path around the main journal centres, the diameter of which is twice the offset of the rod journals. The diameter of that path is the engine "stroke": the distance the piston moves up and down in its cylinder. The big ends of the connecting rods also called as connecting rods, contain bearings (rod bearings) which ride on the (offset) rod journals.



Fig1: A Crankshaft

1.2 Fatigue and Fatigue Life

Fatigue is the condition where a material cracks or fails as a result of repeated (cyclic) stresses applied below the ultimate strength of the material. Fatigue life is the number of stress cycles of a specified character that a specimen sustains before failure of a specified nature occurs.

II. FORCES ACTING ON THE CRANKSHAFT

A major source of forces imposed on a crankshaft, namely Piston Acceleration. The combined weight of the piston, ring package, wristpin, retainers, the connecting rod small end and a small amount of oil are being continuously accelerated from rest to very high velocity and back to rest twice each crankshaft revolution. Since the force it takes to accelerate an object is proportional to the weight of the object times the acceleration (as long as the mass of the object is constant), many of the significant forces exerted on those reciprocating components, as well as on the conrod beam and big-end, crankshaft, crankshaft, bearings, and engine block are directly related to piston acceleration. Combustion forces and piston acceleration are also the main source of external vibration produced by an engine. Here in this case Piston Force is considered.

III. CALCULATIONS FOR PROJECT

3.1 Engine Specifications

- i) Displacement: 178.6cc
- ii) Type: 4 stroke, DTS-i, air cooled, single cylinder
- iii) Bore x Stroke: 63.5 x 56.4 (in mm)
- iv) Maximum Power:
17.02 @ 8500 (ps @ RPM)
12.518 @ 8500 (kW @ RPM)
- v) Maximum Torque: 14.22 @ 6500 (Nm @ RPM)

3.2 Pressure Calculations

Density of petrol (C_8H_{18}):

$$\rho = 750 \text{ kg/m}^3 = 750 \times 10^{-9} \text{ kg/mm}^3$$

Operating Temperature:

$$T = 20^\circ\text{C} = 273.15 + 20 = 293.15 \text{ K}$$

$$\begin{aligned} \text{As Mass} &= \text{Density} \times \text{Volume} \\ &= 0.13395 \text{ kg} \end{aligned}$$

3.3 Molecular Weight of Petrol

$$M = 114.228 \times 10^{-3} \text{ kg/mole}$$

3.4 Gas Constant for Petrol

$$R = 8314.3 \div (114.228 \times 10^{-3}) = 72.7868 \times 10^3 \text{ J/kg/mol K}$$

$$\text{As } pV = mRT$$

$$p = 16.003 \text{ MPa} = 16.003 \text{ N/mm}^2$$

3.5 Design Calculations

1) Gas Force (F_p):

$$F_p = \text{Pressure (P)} \times \text{Cross Section Area Of Piston (A)}$$

2) Moment on the Crankpin:

$$M_{max} = \frac{F_p}{2} \times \frac{L_c}{2}$$

By the given dimensions of the crankpin,

$$\text{Diameter of the crankpin} = (d_c) = 30 \text{ mm}$$

$$\text{Length of the crankpin} = (l_c) = 53.7 \text{ mm}$$

3) Section Modulus of Crankpin:

$$Z = \frac{\pi}{32} \times d_c^3$$

4) Torque Obtained At Maximum Power Given Engine:

$$P = \frac{2\pi NT}{60}$$

5) Von Misses Stresses Induced:

$$T = \text{Torque}$$

$$M_{max} = \text{Bending Moment}$$

$$K_b = \text{Combine shock, fatigue factor for bending}$$

$$K_t = \text{Combine shock, fatigue factor for torsion}$$

Equivalent Bending Moment:

$$M_{ev} = \sqrt{(K_b \times M_{max})^2 + \left(\frac{3}{4}\right) \times (K_t \times T)^2}$$

$$\text{Thus } \sigma_{von} = \frac{M_{ev}}{Z}$$

$$\sigma = \sigma_{von}$$

6) Strain:

$$\epsilon = \frac{\sigma}{E}$$

3.6 Fatigue Calculations

σ_{max} = Maximum Stress

σ_{min} = Minimum Stress

Amplitude Stress

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

Mean Stress:

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

For Alloy Steel 41Cr4:

Yield Strength (S_y) = 660 MPa

Ultimate Tensile Strength (S_{ut}) = 1000 MPa

Fatigue Endurance Strength (S_e'):

$$S_e' = 0.5 \times S_{ut}$$

Endurance Limit Considering Factors:

$$S_e = K_a \times K_b \times K_c \times K_d \times S_e'$$

i) K_a = Surface Finish Factor (Machined Surface)

$$a = 4.51; b = -0.265$$

$$K_a = a(S_{ut})^b$$

ii) K_b = Size Factor ($7.5 < d \leq 50$)

iii) K_c = Reliability Factor (95 % Reliability)

iv) K_d = Modifying Factor taking into account Stress Concentration

For K_d , Notch Sensitivity = q Theoretical

Stress Factor = K_t

Here K_f = Fatigue Stress Concentration Factor Thus $K_d = 1/K_f$

3.7 Finite Life Calculation

$$N = \left(\frac{\sigma_a}{a} \right)^{1/b}$$

Where N= Number of life cycles

$\sigma_a = \sigma_{von}$ = Von Misses Stress

Taking f = 0.82

$$i. \quad a = \frac{(f \times S_{ut})^2}{S_e}$$

$$ii. \quad b = - \frac{\log_{10} \left(\frac{f \times S_{ut}}{S_e} \right)}{3}$$

3.8 S-N Curve

1. Goodman Line

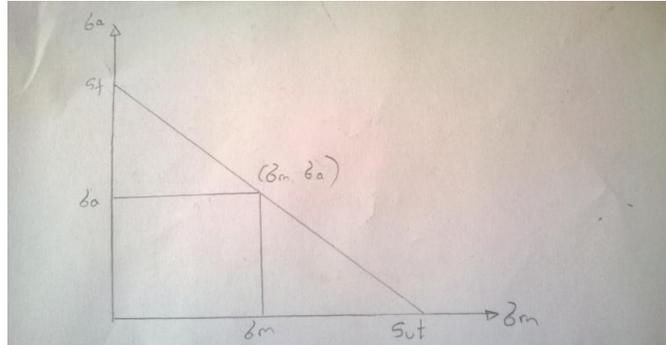


Fig2: Goodman line diagram

The conditions taken into consideration are:

- The factor of safety i.e. FOS is taken as 1.
- **The Mean Stress σ_m** is calculated as zero from previous calculation

The equation of Goodman line:

$$\frac{\sigma_m}{S_{ut}} + \frac{\sigma_a}{S_e} = \frac{1}{\text{FOS}}$$

As FOS=1, the equation obtained is:

$$\frac{\sigma_m}{S_{ut}} + \frac{\sigma_a}{S_e} = 1$$

After Further Modifications,

$$S_f = S_{ut} \times \frac{\sigma_a}{(S_{ut} - \sigma_m)}$$

Put $\sigma_m = 0$,

$$S_f = \sigma_a$$

2. Use of S-N Curve

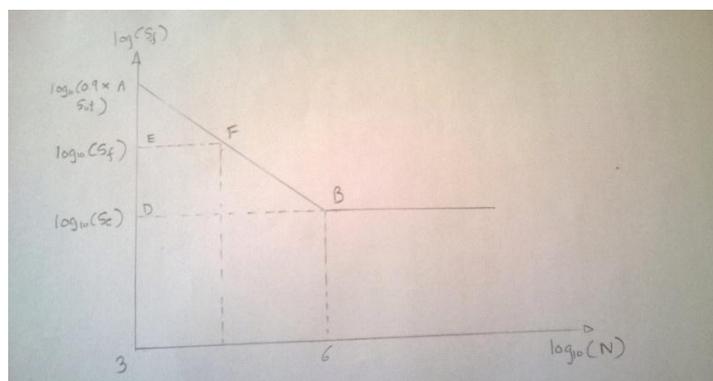


Fig3: S-N Curve

$$EF = DB \times \frac{AE}{AD}$$

Thus number of cycles is finally calculated from: $\log_{10}N$

IV. ANSYS RESULT

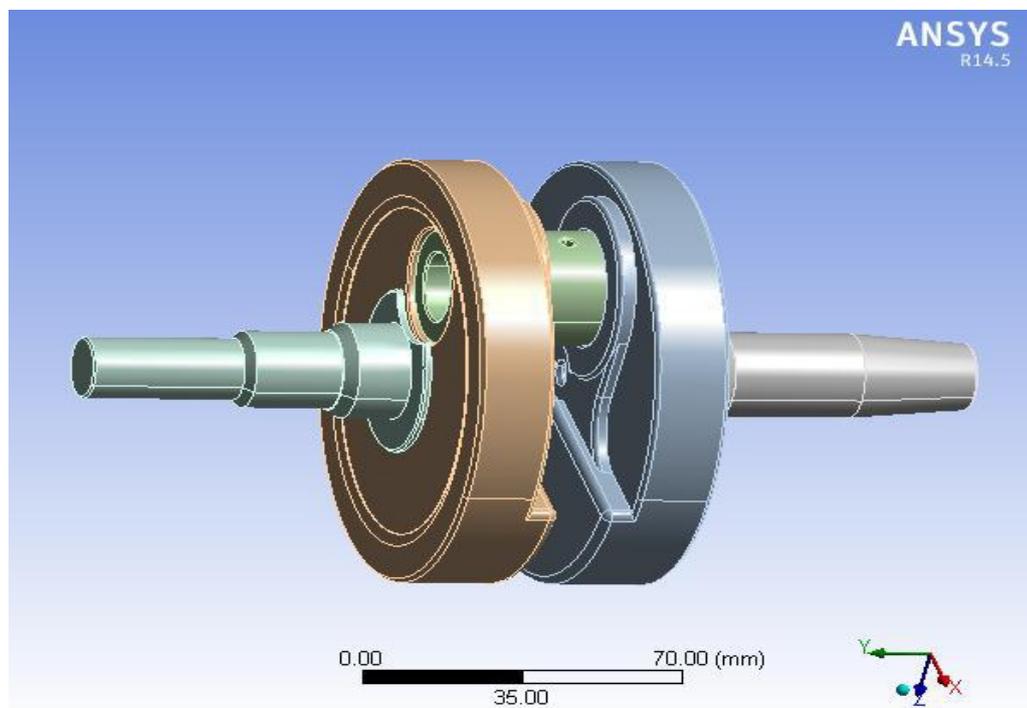
Here the ANSYS simulation is done. In this case, Von Misses stresses, Strain and Number of Cycles is obtained. For this scenario, Gas force of 50.6802×10^3 N. From this case, the following results are obtained.

4.1 Assumptions

Parameters	Values
Combine shock, fatigue factor for bending = K_b	1
Combine shock, fatigue factor for torsion = K_t	1
Density = ρ (kg/m^3)	7700
Ultimate Tensile Strength = S_{ut} (MPa)	1000
Yeild Strength = S_{yt} (MPa)	660

The assumption taken here is the crank web and the shafts are considered as fixed. The only critical component considered here is the crankpin.

4.2 Model and Meshing



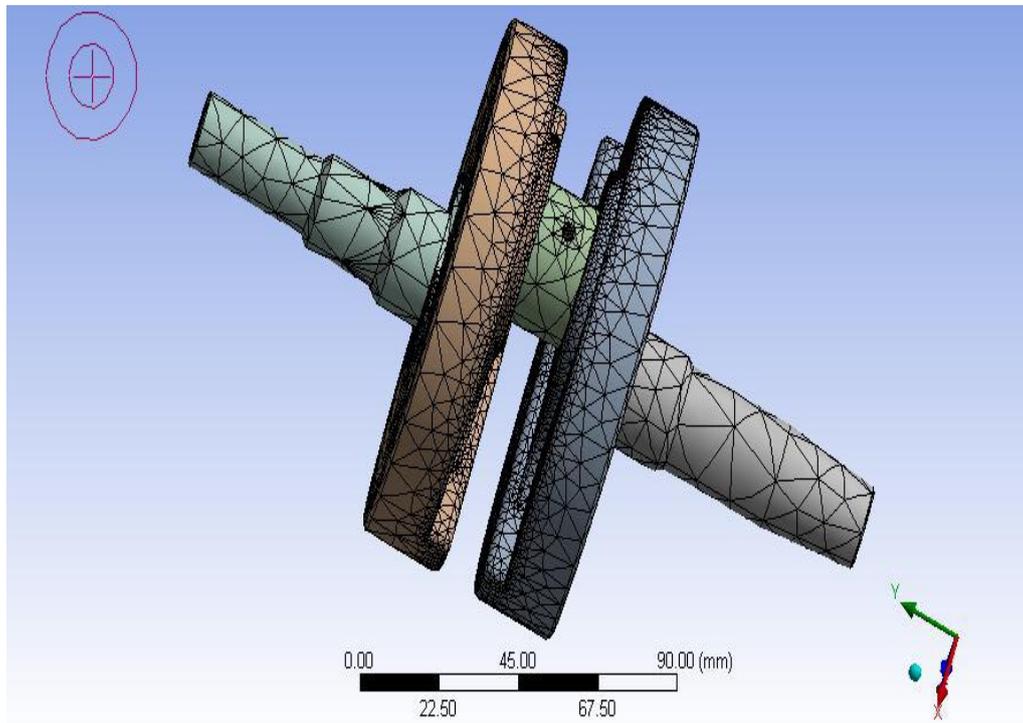


Fig4: Crankshaft Model and Meshing

4.3 Initial Conditions

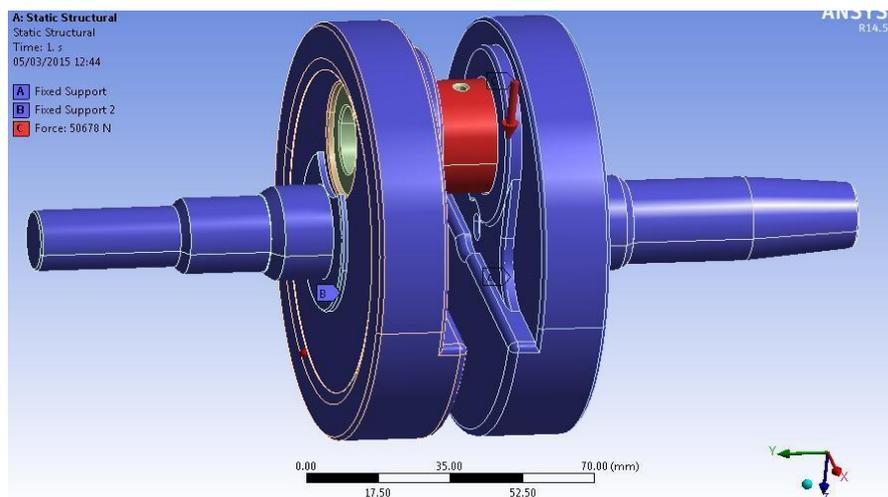


Fig5: Initial Conditions

4.4 Result

4.4.1 Von Misses Stresses and Deformation

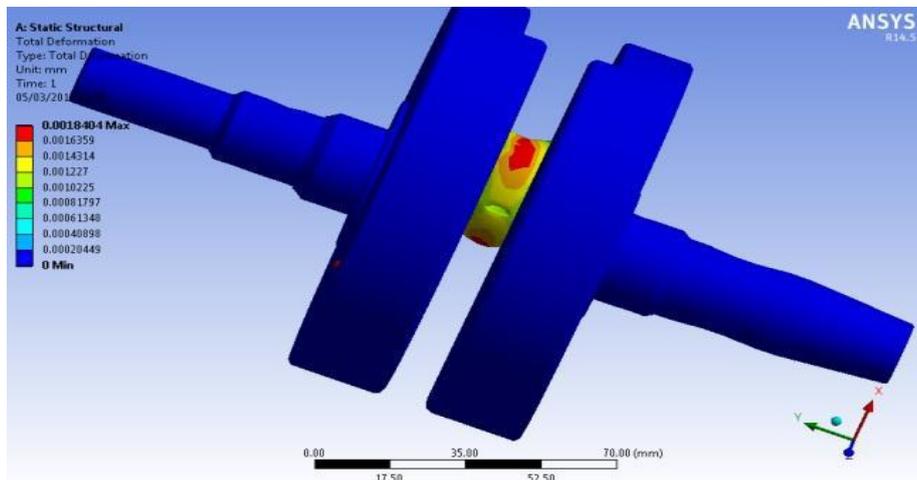
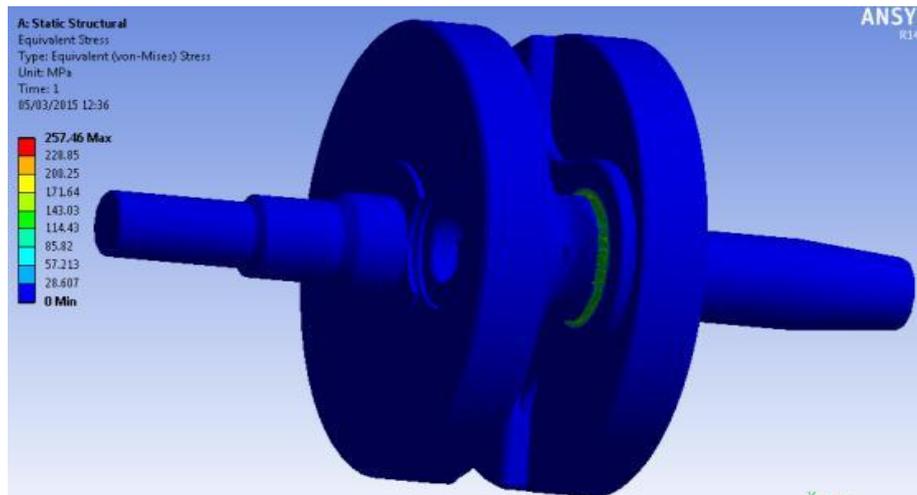
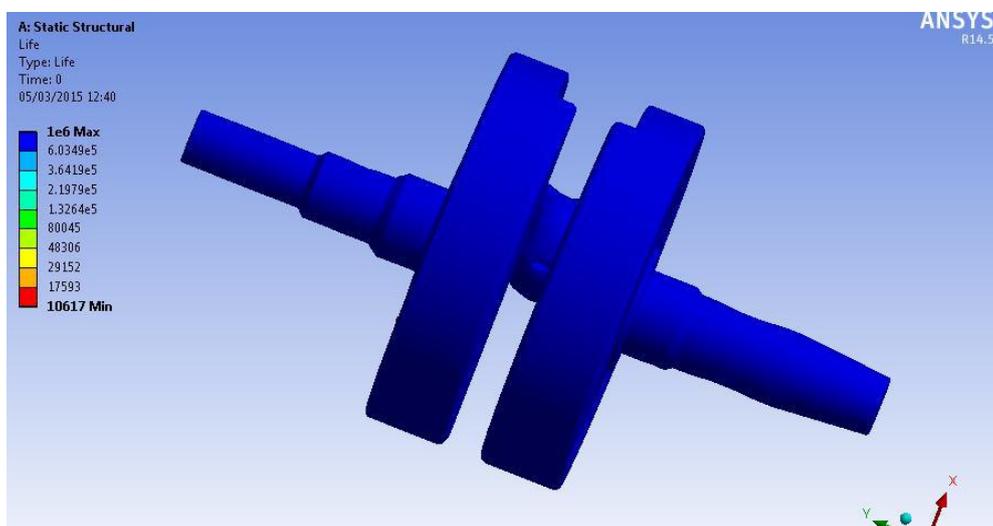


Fig6: Von Misses Stresses and Deformation

4.4.2 Number of cycles and Goodman Diagram



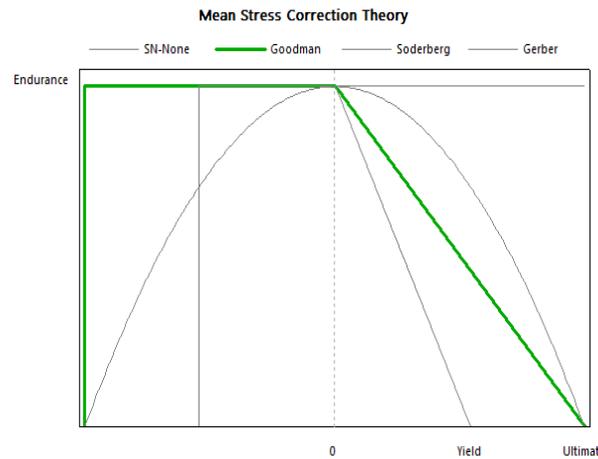


Fig7: Number of Cycles and Goodman Diagram

V. CONCLUSIONS

5.1 Static Analysis

Parameters	Numerical	Analytical	Error (%)
Von Misses Stresses (MPa)	256.805	256.47	0.13
Total Deformation (mm)	0.06895	0.0018404	36.4646

5.2 Fatigue Analysis:

Parameter	Finite Life Calculation	S-N Curve	Analytical
Number of Cycles	1.4222×10^5	0.8596×10^5	0.80045×10^5

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