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# Numerical Prediction of Fatigue Life of Crankshaft

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**Abstract**

In Automobile manufacturing plant, it was noticed that 7 out of 1000 crankshafts are prone to failure. This work is concentrated to identify the cause of failure in the crankshaft and to improve the fatigue life of the crankshaft by performing parametric study. In parametric study, the crankshaft is designed and analyzed in ANSYS WORKBENCH by varying the fillet radius of journal bearing. The maximum equivalent alternating stress and cycles to failure were predicted in numerical analysis. The cycles to failure are increased by varying the parameter and hence the fatigue life of crankshaft is improved.
**Keyword- Numerical Model, Dynamic Analysis, Torsion Test Analysis, Bending Test Analysis, Fatigue life**

## I. INTRODUCTION

Crankshaft is a large component with a complex geometry in the engine, which converts the reciprocating displacement of the piston into a rotary motion. This study was conducted on a four cylinder petrol engine. It must be strong enough to take the downward force during power stroke without excessive bending. [3] Due to the repeated bending and twisting load the crankshaft fails as cracks form in fillet area and prediction of fatigue life is important to ensure safety of components. [5] The journal pin fillet area is identified as the critical location. The stress concentration and cycles to failure is improved using ANSYS WORKBENCH 14.5.

## II. MODELLING OF THE CRANKSHAFT

Figure 1 shows the crankshaft developed in creo 2.0 software and figure 2 shows it’s dimensions.

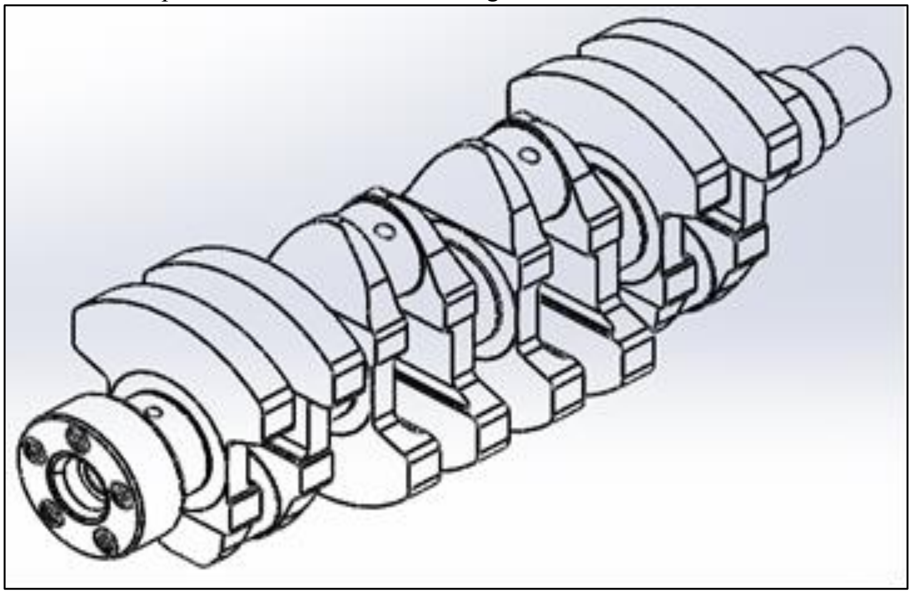

Fig. 1: CAD Model of Crankshaft

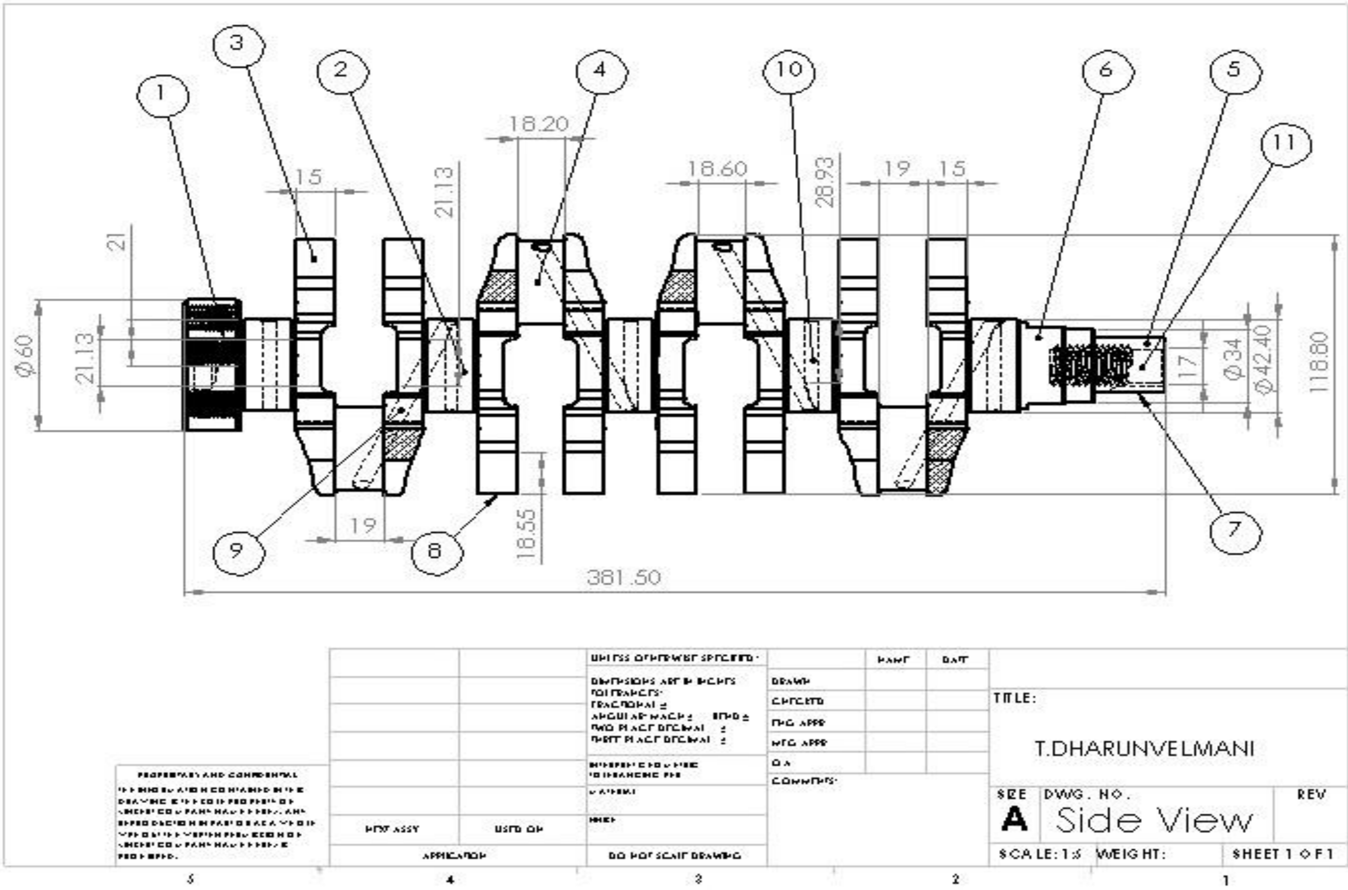


Fig. 2: Dimensions of the Crankshaft

| *Part No* | *Description* |
|---|---|
| *1* | *Crankshaft Flange* |
| *2* | *Main Journal* |
| *3* | *Counter Weight Or Balancing Weight* |
| *4* | *Crank Pin* |
| *5* | *Pulley* |
| *6* | *Pulley Key Seat* |
| *7* | *Key Way* |
| *8* | *Balancing Completion* |
| *9* | *Inclined Oil Hole Through Condition* |
| *10* | *Vertical Oil Hole Through Condition* |
| *11* | *Pulley Bolt Hole* |

*Table 1: Parts of the Crankshaft*

| *Properties* | *Symbols* | *Value* |
|---|---|---|
| *Modulus of elasticity,* | *E, GPa* | *178* |
| *Yield Strength ,* | *YS, MPa* | *412* |
| *Ultimate strength,* | $S_u$*, MPa* | *658* |
| *Percent elongation,* | *%EL* | *10%* |
| *Strength coefficient,* | *K, MPa* | *1199* |
| *Strain hardening exponent,* | *n* | *0.183* |
| *Fatigue strength coefficient,* | $\sigma_f$*, MPa* | *927* |
| *Cyclic yield strength,* | *YS, MPa* | *519* |
| *Fatigue strength* | $S_f$ *, MPa* | *263* |

*Table 2: Properties of ductile cast iron*

## III. DETERMINATION OF FORCES

### A. Determination of Gaseous Forces

The gas forces acting on the crankshaft are determined using Get Data software. It converts the graphical data into digital format. Hence, it is possible to obtain the values at any point on the curve. Minimum and maximum values of the volume of engine are obtained using the P-V diagram shown in figure 3.

| Volume | Unit in $cm^3$ |
|---|---|
| $V_{max}$ | 275 |
| $V_{min}$ | 3.75 |

*Table 3: Maximum and minimum volume of engine during combustion*

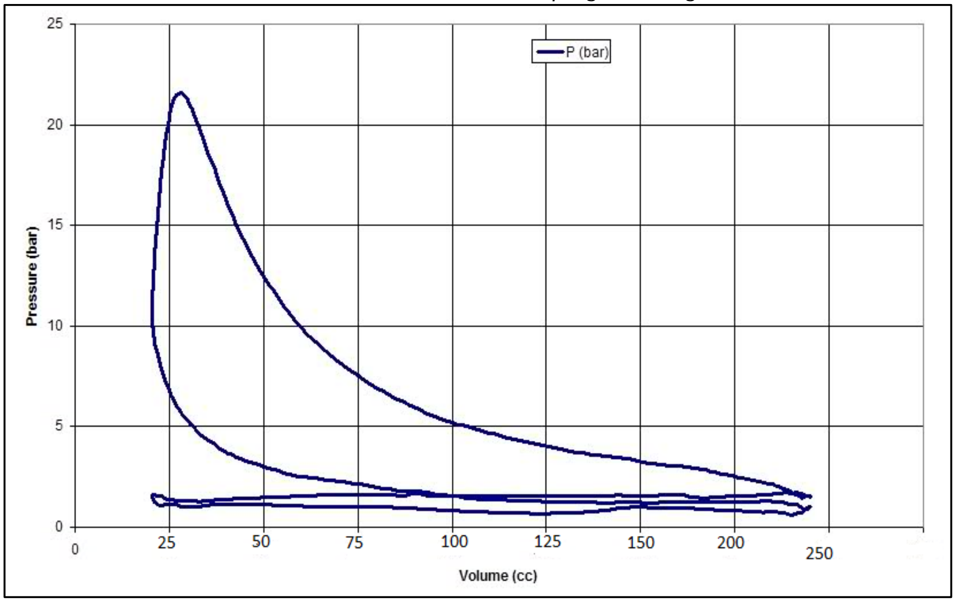


Fig. 3: P-V diagram of Hyundai petrol engine

Table 3 shows the values of $V_{min}$ and $V_{max}$ and the formulas used to obtain the swept volume of the engine is shown below.

$$V_{swept} = V_{max} - V_{min}$$

Swept volume can be written as

$$V_{swept} = \pi R^2 S$$

Where,
R is radius of piston and
S is stroke length of the engine.

Therefore the volume of the engine is changing with respect to ϴ. From the P-V diagram shown in fig 4 values of pressure at volumes are obtained. Using these relations values of pressures are obtained for different crank angles. (Crank angles in ϴ)

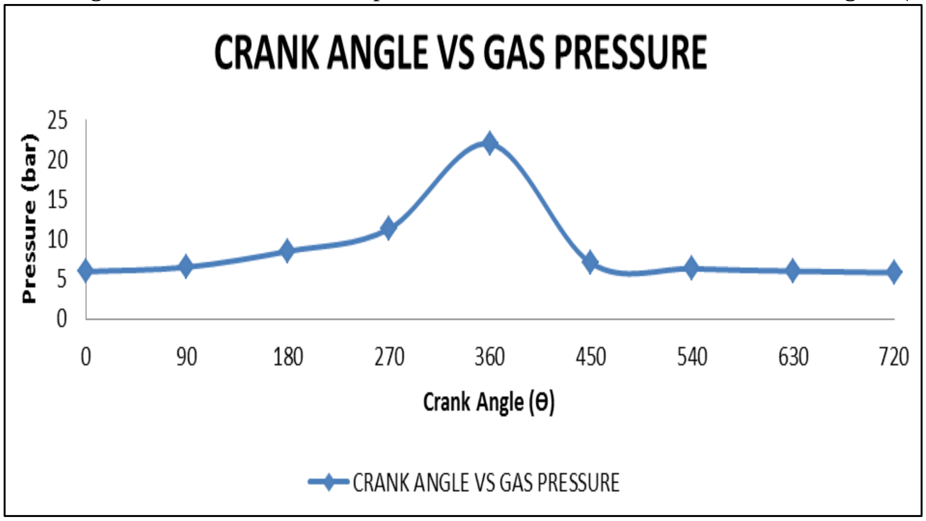


Fig. 4: Plot for crank angle vs engine pressure

– Forces acting on the crankshaft due to gas force in X and Y direction:
$(F_{cx})_G = -P$
$(F_{cx})_G = P \tan\beta$

### *B. Determination of Inertial Forces*

The parametric representation of the crankshaft, connecting rod and piston assembly is represented in figure 5.

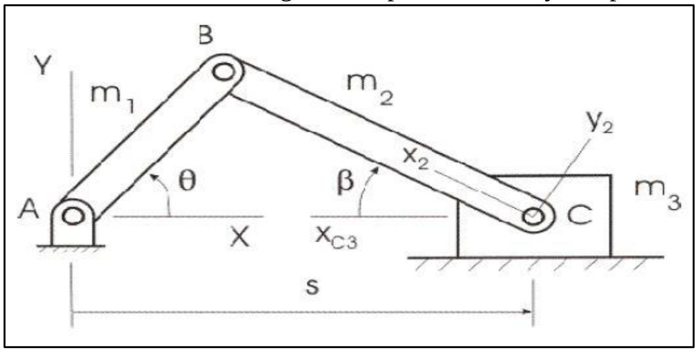


Fig. 5: Parametric representation of the crankshaft, connecting rod, and piston assembly

– The linear velocity and acceleration of piston are given by:

$$v_p = \omega\, r\, (\sin\theta + (\sin 2\theta/n))$$
$$a_p = \omega^2\, r\, (\cos\theta + (\cos 2\theta/n))$$

Where
$\omega$ is the angular velocity of crank shaft (rad/s)
r is the radius of crank angle (m)
$\theta$ is the crank angle with the horizontal
$n = l/r$
l is the length of the connecting rod (m)

– Forces acting on the crankshaft in X and Y direction considering effect of inertia force due to piston:

$$(F_{cx})_p = -m_p\, a_p$$
$$(F_{cy})_p = m_p\, a_p \tan\beta$$

Where
$m_p$ is the mass of the piston (10 Kg)
$\beta$ is the angle connecting rod makes with the horizontal

– Forces acting on the crankshaft in X and Y direction considering effect of inertia force due to lumped mass of connecting rod at wrist pin C:

$$(F_{cx})_c = -\, m_w\, a_p$$
$$(F_{cy})_c = m_w\, a_p \tan\beta$$

Where
$m_w$ is the lumped mass of connecting rod at wrist pin C (0.1Kg)

– Forces acting on the crankshaft in X and Y direction considering effect of inertia force due to lumped mass of connecting rod at crank pin B:

$$(F_{cx})_B = -\, m_s\, r\, \omega^2 \cos\theta$$
$$(F_{cy})_B = m_s\, r\, \omega^2 \sin\theta$$

Where - $m_s$ is the lumped mass of the connecting rod at crank pin B (0.19Kg)
$m_r$ is the mass of the connecting rod ( 300g)
The total bearing load is

$$F_{cx} = -m_p\, a_p - m_w\, a_p - m_s\, r\, \omega^2 \cos\theta - P$$
$$F_{cy} = m_p\, a_p \tan\beta + m_s\, a_p \tan\beta + m_s\, r\, \omega^2 \sin\theta + P \tan\beta$$

$F_{cx}$ and $F_{cy}$ are expressed in the global coordinate system, which is not rotating with the crankshaft. Forces expressed in a coordinate system attached to the crankshaft are given by:

$$F_X = F_{cx} \cos\theta + F_{cy} \sin\theta$$
$$F_Y = F_{cy} \cos\theta - F_{cx} \sin\theta$$

## IV. PARAMETRIC TECHNIQUES

The fatigue life of the crankshaft for the existing crankshaft is 22014 number of cycles to failure at 4500 rpm. The critical region for the fatigue crack initiation is identified at the groove region on the main journal bearing of the crankshaft [10]. So in order to reduce the number of cycles to failure the fillet radius has to be modified. The journal groove near the fillet region of main bearing is prone to fatigue failure because most of the degrees of freedom are constrained there [7]. To reduce the number of cycle to failure the profile of the existing fillet in the journal bearing of the crankshaft is to be changed with semi elliptical groove as shown in figure 6.

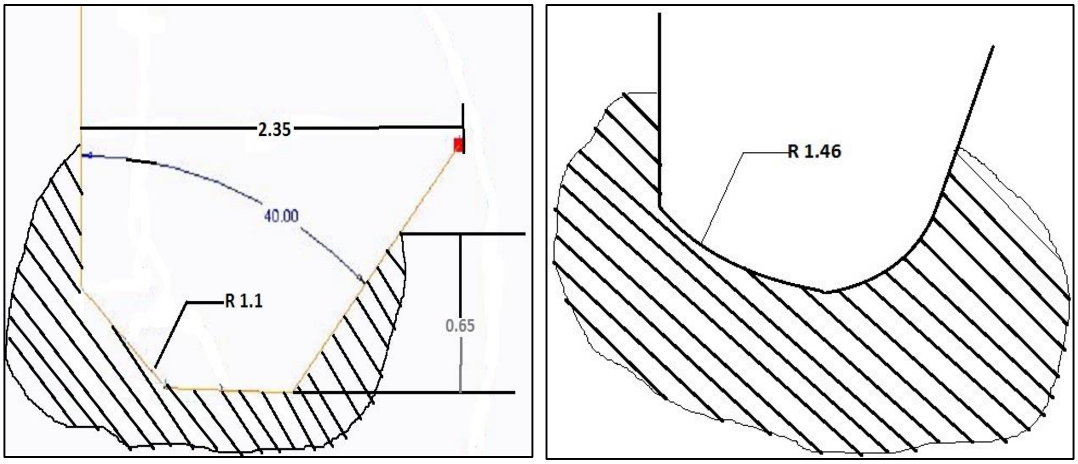


Fig. 6: Existing groove design near main journal fillet (left) semi elliptical groove design near main journal ( right)

## V. Numerical analysis

Figure 7 shows the numerical model of the crankshaft and its parameters are shown in Table 2.

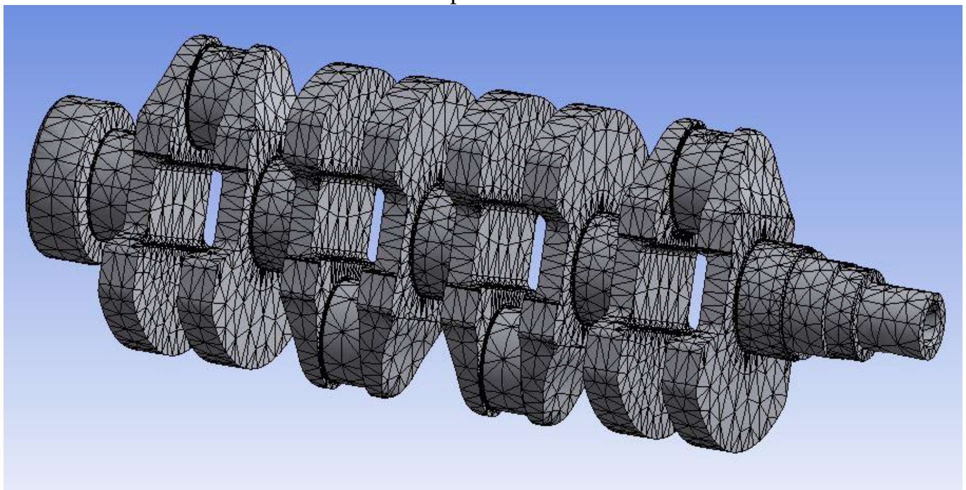

Fig. 7: Numerical Model of the Crankshaft

| *PARAMETERS* | *VALUE* |
|---|---|
| *Mesh Type* | *Quadratic tetrahedral* |
| *Elements* | *67495* |
| *Nodes* | *117114* |
| *Element Size* | *1.9664e-002 mm* |
| *Relevance Centre* | *Fine* |

*Table 4: Numerical parameters of the crankshaft*

### *A. Dynamic Analysis*

The crankshaft is subjected to two different load sources

- Inertia of rotating components (e.g. Connecting rod) applies forces to the crankshaft and this force increases with the increase of engine speed.
- The second load source is the force applied to the crankshaft due to gas combustion in the cylinder. The slider-crank mechanism transports the pressure applied to the upper part of the slider to the joint between crankshaft and connecting rod [2].

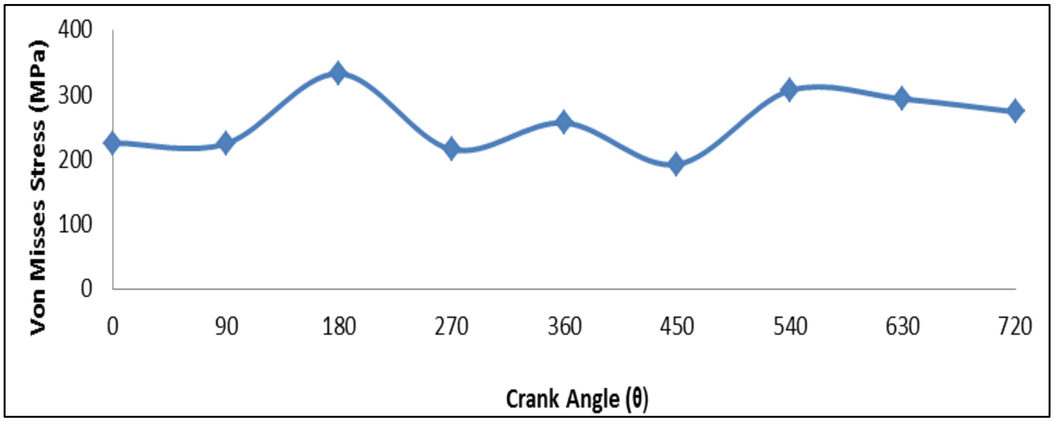


Fig. 8: Plot for Crank Angle vs Von Misses Stress

Figure 8 shows the plot for crank angle vs von misses stress for optimized crankshaft at 4500 rpm.

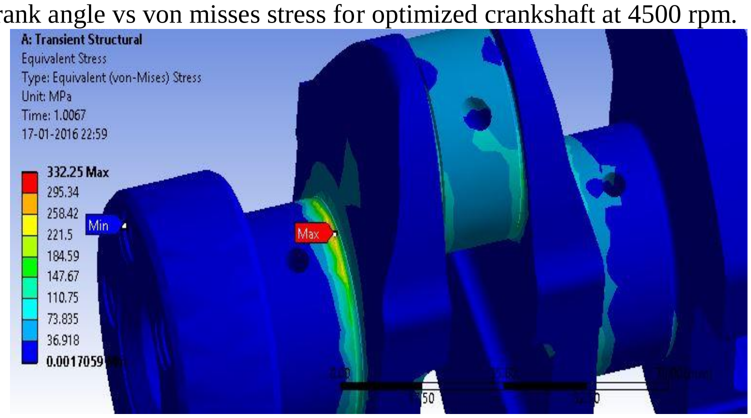


Fig. 9: Reduction in the stress concentration level near the journal groove region

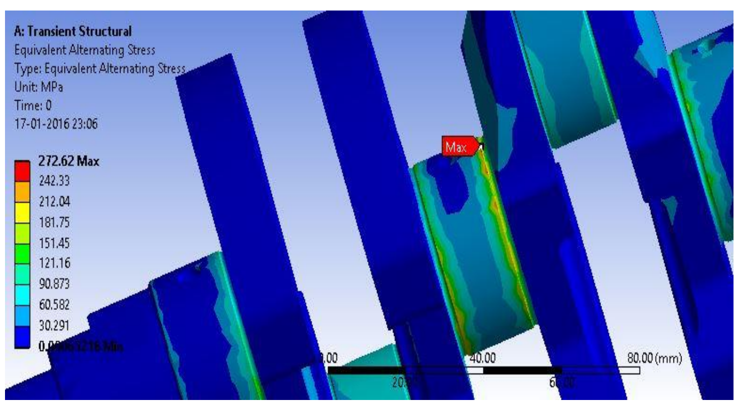


Fig. 10: Equivalent alternating stress for the crankshaft

Figure 9 shows the reduction in stress concentration level near the journal groove region at 4500 rpm

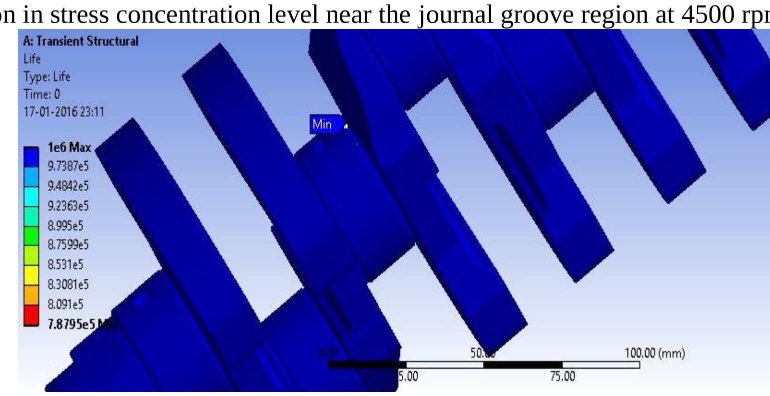


Fig. 11: Fatigue life for the crankshaft at 4500 rpm

 

Figure 10 shows the reduction in equivalent alternating stress to 272 MPa and figure 11 shows the increase in cycles to failure to $7.879 \times 10^5$ at 4500 rpm by dynamic analysis.

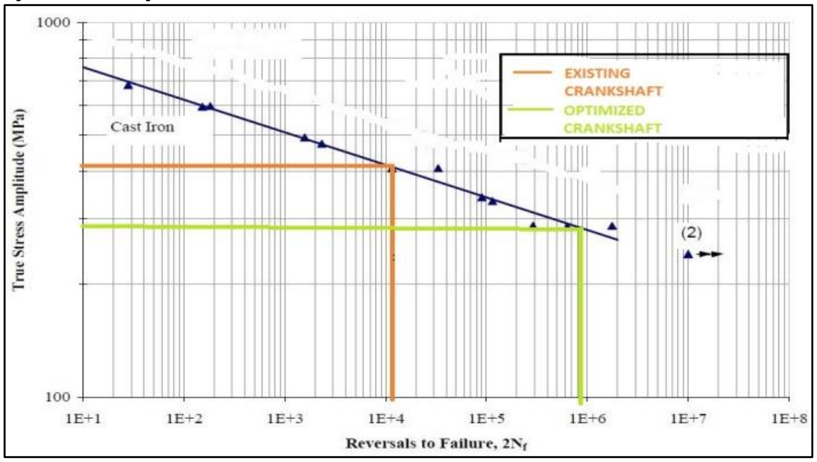

Fig. 12: Comparison of the fatigue life crankshaft

Figure 12 shows the comparison of fatigue life of the parameter studied crankshaft with the existing crankshaft

*B. Bending Test Analysis*

The maximum bending load acting on the crankpin bearing is at 355 degrees. The critical loads due to the slider crank mechanism induces bending and shear forces on the crankshaft.[5] The bending moment acting on the centre of the crankshaft is calculated based on the following equation

$$M / I = \sigma_b / y$$

Where

M is the maximum bending moment acting on crankpin (N.m)

I is the area moment of inertia = $(\pi \times d_p)/64$ ($m^4$)

Y is the distance of C.G from neutral fibre = $d_p/2$ (m)

$\sigma_b$ is the maximum allowable bending stress for nodular cast iron (210 MPa)

$d_p$ is the diameter of the crankpin (0.0384m)

M = 1166.78 N.m

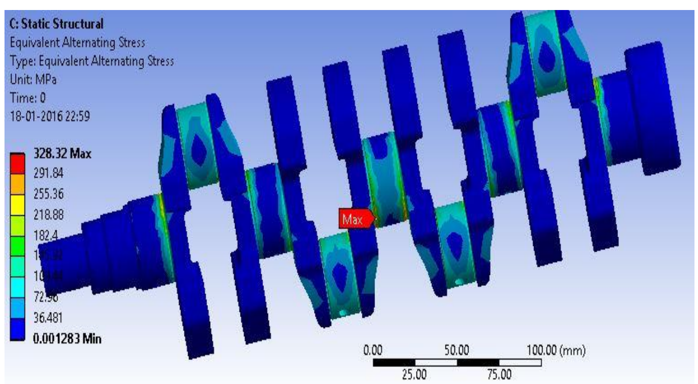

Fig. 13: Equivalent alternating stress for crankshaft

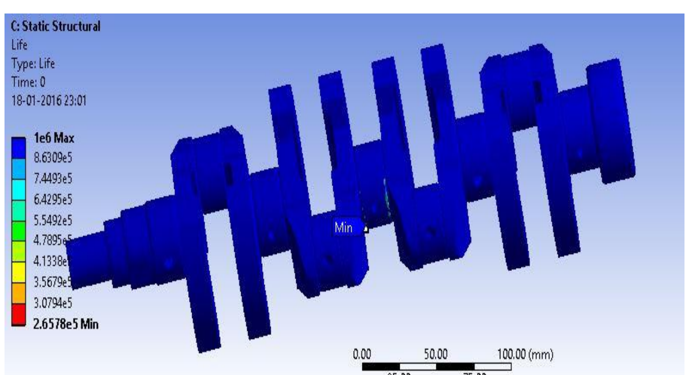

Fig. 14: Fatigue life contour for the crankshaft

Figure 13 shows the reduction in equivalent alternating stress to 328.23 MPa and figure 14 shows the increase in cycles to failure to $2.657 \times 10^5$ at 4500 rpm after conducting bending test analysis.

*C. Torsional Test Analysis*

The twisting load on the crankshaft is a negligible force when compared with the bending load acting on the crankpin.The maximum twistinng moment on the crankshaft occurs at an crank angle of 25 to 35 degrees. The torque or twisting moment on the crankshaft is calculated as follows [4]

Piston effort ($F_p$) = (Net force acting on piston at 35 deg) - $m\,\omega^2\, r\,(\cos\theta + (\cos 2\theta/n)) + mg$

= 1410 – 3989 + 9.81

= -2569.19 N

$T = F_T\ r = F_p\ r\ (\sin\theta + (\sin 2\theta / 2\,(n^2 - \sin^2\theta)^{1/2}))$

= - 36.4 N.m

where

m is the mass of the reciprocating parts = 1 kg

ω is the angular velocity of crankshaft = 471 rad/s

r is the crank radius = 0.0192 m

θ is the crankangle = 35 deg

$F_p$ is the piston effort, N

$F_T$ is the tangential force acting on the crankshaft, N

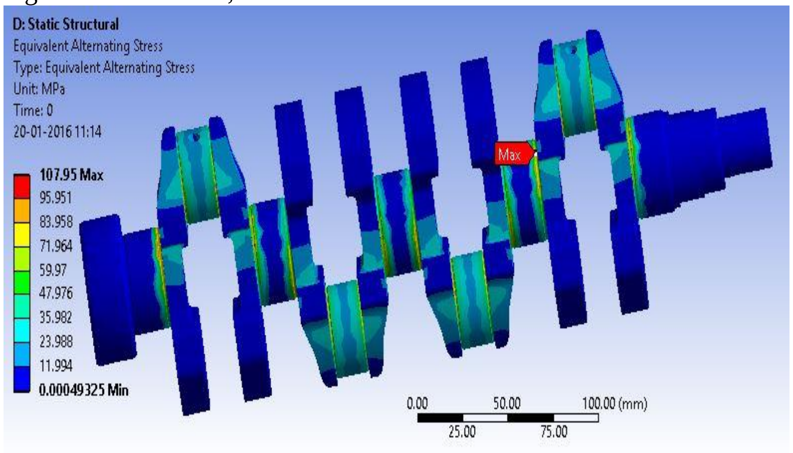


Fig. 15: Equivalent alternating stress contour for the crankshaft

Figure 15 shows the reduction in equivalent alternating stress to 108 MPa and table 5 shows the tabulated values of equivalent alternating stress and cycles to failure is increased to $10^6$ cycles.

## VI. RESULTS AND DISCUSSION

*A. Dynamic Analysis*

- The critical failure speed for crankshaft is maximum speed of 4500 rpm as the von misses stress at 3600 rpm is well under the yield stress and fatigue strength of the material.
- The maximum von misses stress for the crankshaft after parametric study was reduced from 453 MPa to 332 MPa which is a 19 % reduction in the maximum von misses stress under dynamic condition at a maximum operating speed of 4500 rpm.
- The fatigue life of the crankshaft after design optimization was improved from 22104 to $7.895 \times 10^5$ number of cycles to failure under dynamic condition at a maximum speed of 4500 rpm.
- The stress concentration near the journal groove at the fillet region was reduced by changing the profile of existing groove of radius 1.1 mm to semi elliptical shape of radius 1.46 mm as shown in figure 6 so the stress gets equally distributed at the groove region.
- The notch sensitivity was reduced by increasing the r/d ratio to unity so that plasticity of the fillet surface is controlled.

*B. Bending Test Analysis*

- The maximum von misses stress for the crankshaft after parametric study was reduced from 359 MPa to 328 MPa- a 8.6 % reduction in the stress level.
- The fatigue life of the crankshaft under bending condition after design optimization was improved from $8.347 \times 10^4$ to $2.657 \times 10^5$ number of cycles to failure which is due to minimization of stress near the groove region of the crankshaft.

*C. Torsion Test Analysis*

- In torsion test analysis the maximum von misses stress for the crankshaft was reduced from 114 to 108 MPa - a 5 % reduction in the stress level, by modifying the fillet radius in the journal bearing as fillet radius is inversely proportional to the stress concentration.

## VII. CONCLUSION

- In dynamic analysis there was a 19% reduction in the maximum von misses stress and also the cycles to failure was increased to 34%
- The maximum von misses stress for the crankshaft after design optimization was reduced from 359 to 328 Mpa which is a 8.6 % reduction in the stress level.
- The cycles to failure of the crankshaft under bending condition was increased by 21%.
- In torsion test analysis there was a 5% reduction in the maximum von misses stress.
- The fatigue life of the crankshaft in terms of cycles to failure has improved by modifying the existing crankshaft design.

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