Failure Analysis of Power Press Crankshaft

Satish A. Patel[1], H. G. Patil[2]

[1]Research Scholar, [2]Associate Professor
[1,2]Department of Mechanical Engineering
[1,2]D.N. Patel College of Engineering Shahada, Maharashtra, India

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Abstract - *Crank shaft is a critical component in n power press industry. Because of impact load continuously acting on it, crank shaft of a press machine always experiences continuous shear stress and over loading. Press machine continuously deals with stress and because of that crank fails in machine under uneven conditions, which leads to complete shutdown of machine. So, as crank is a critical part of power press, it must be designed after carefully considering all the factors responsible for its failure. This paper is focused on finding the failure of crankshaft in industry. It consists of creating a 3-D model of crankshaft in Solidworks software and performing Static and Fatigue Analysis in Ansys Workbench 15.0 applying the boundary conditions and forces. The result obtained from the Ansys analysis is validated using the analytical method. It is based on the obtained results redesign the crankshaft.*

Key Words: **Power press machine, Crankshaft, Solidworks, Static analysis, FEM.**

1. INTRODUCTION

Crankshaft is a large component with a complex geometry in the Power Press, which converts the rotary motion of motor into the reciprocating motion of the Ram with a connecting rod link mechanism. Al-Jazari was the first engineer to invent the crankshaft, which is considered the single most important invention after the wheel. This system is used to transform linear motion into rotating motion, and vice versa, and is central to the modern machinery such as internal combustion engines and Power press.

Since the crankshaft experiences a large number of load cycles during its service life, fatigue performance and durability of the components have 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.

The crankshaft must be strong enough to take the downward force during downward stroke without excessive bending. Crankshafts have altered very little in their basic design since the very first steam reciprocating engines were put into ships during the nineteenth century. What has been changed is the material and level of design and engineering to ensure that crankshaft can cope with the high powers and speeds required by modern machinery. Crankshaft is an important part of Power press machine. They need to be rigid, with high torsional strength, be able to withstand forces and, without compromise, need to be compact.

Fig -1: Crankshaft

As shown in figure.1 above, a crankshaft has three main components;

1. A crank pin journal or "big end" (the running surface of the shaft) which receives the energy

2. A main journal which carries the crankshaft within the main bearings.

3. The crank webs, which connect the two journals together

2. TYPES OF FAILURES IN CRANKSHAFT

Crankshaft journals (being running surfaces) are hardened. The hardness layer is thin and is essential to prevent wear to the crankshaft itself; the hardness level has manufacturers' recommended limitations for polishing and machining. A journal with defects beyond the hardness layer limitations will require replacement or expensive repairs. The hardness layer can also be affected by overheating, resulting in the surface becoming excessively hard. Taking all the above into consideration, it hardly seems surprising that these components, which are complex by design and receive so many varied forces are subject to failure. The most common types of failure are:

1. Damage to journal surface hardening beyond recommended limits

Fig -2: Failure in surface hardening [12].

2. Bending of thecrankshaft

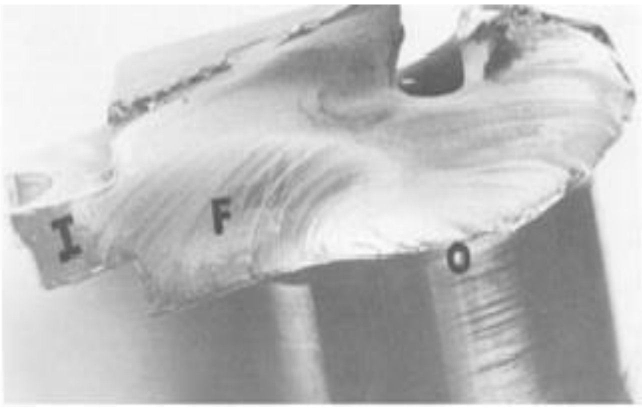


Fig -3: Failure in Bending.

3. Cracking of the crankshaft

Fig -4

Fig -5

Fig. 4 &5. Cracking of crankshaft between the crank pin and the crank web.

The figures 2, 3, 4 & 5 show the different failures most commonly seen in crankshafts.

3. MATERIAL SPECIFICATION

Table -1: Specifications of EN24 Material

EN-24 specification (Composition)	
Carbon	0.36 – 0.44 %
Silicon	0.10 – 0.35 %
Manganese	0.45 – 0.70 %
Sulphur	0.040 Max
Phosphorus	0.035 Max
Chromium	1.00 – 1.40 %
Molybdenum	0.20 – 0.35 %
Nickel	1.30 – 1.70 %
EN-24 Mechanical properties	
Max Stress	850 -1000 N/mm^2
Yield Stress	680 N/mm^2 Min.
Poison's Ratio	0.3
Elongation	0.13

4. DESIGN AND 2D DRAWINGS OF POWERPRESS MACHINE COMPONENTS

The figures given below show the 2D & 3D designs of all the components of a crank shaft.

1. Crankshaft

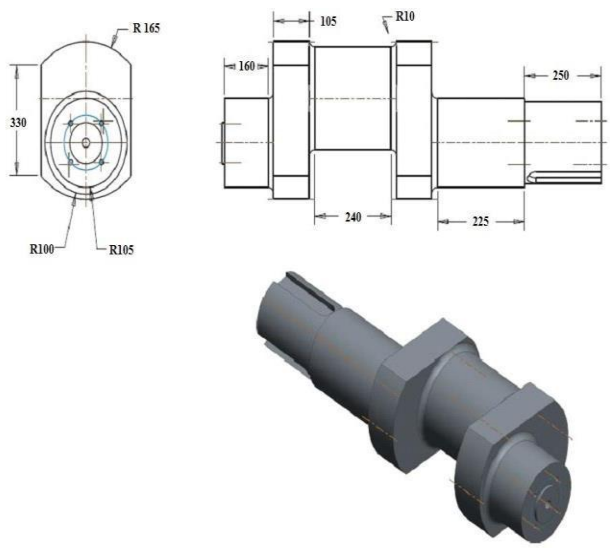


Fig -6: 2D & 3D design of Crankshaft.

Figure.6 shows the schematic diagram of assembly of a crankshaft

2. Ball screw.

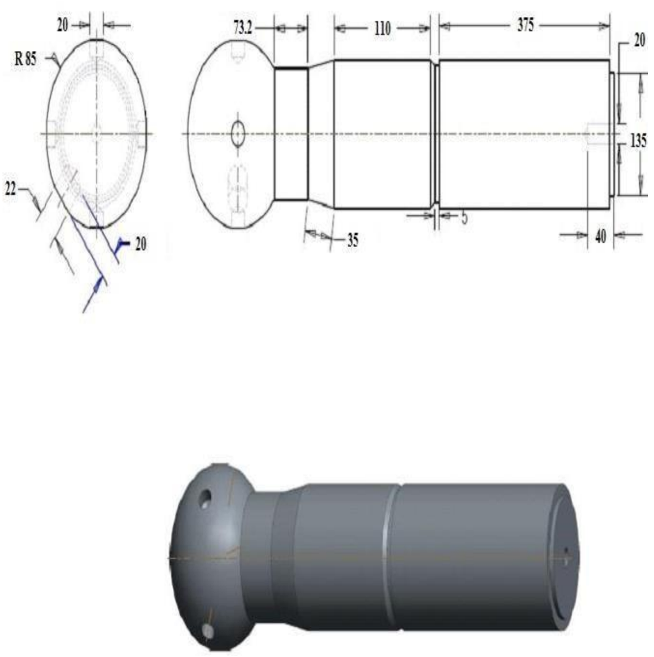


Fig -7: 2D & 3D design of ball screw.

The above figure shows the design of a ball screw.

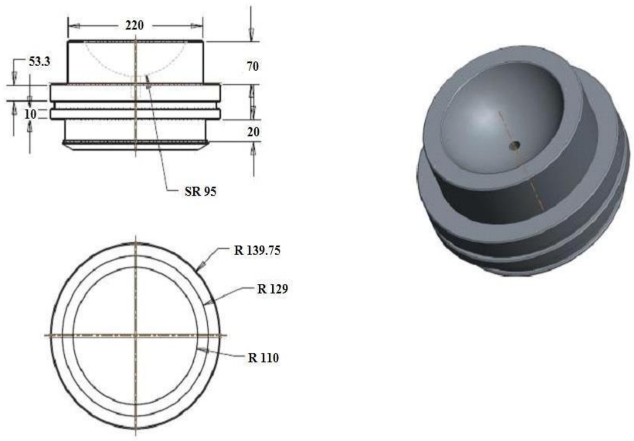


Fig -8: 2D & 3D design of ball seat.

Figure.8 shows the design of a ball seat which supports the ball screw.

3. Cylinder

Fig -9: 2D & 3D design of cylinder.

Figure.9 depicts the schematic design of a cylinder which supports the ball seat.

4. Connecting rod

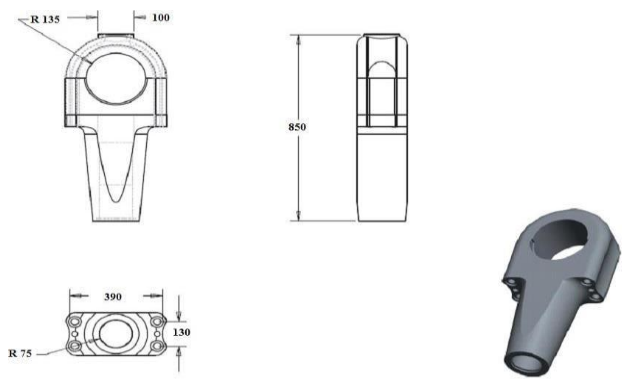


Fig -10: 2D & 3D design of connecting rod

The above figure shows the design of connecting rod.

Table -2: Technical Specifications of Power Press

Description	Dimension
Model	SNX-200
Tonnage	200
Fixed stroke (mm)	220
Rating point (mm)	6
Die height (mm)	450
Slide area (mm^2)	860 x 650
Bolster area (mm^2)	1400 x 840
Major motor (hp x P)	20 x 4

5. CALCULATION OF POWER TRANSMISSION IN POWERPRESS

- Motor power P_1= 14.91399744 KW (20HP)
- Motor RPM N_1 = 1440rpm
- So, Angular Speed of motor W1 = 2πN1/60

=150.7964 rad/s

- Motor Shaft torque T1 = P1/W1 = 98.90152Nm
- Velocity Ratio = R_1/ R_2; R_1 = flywheel outer radius

= 525 mm, R $_2$= Pulley radius = 94.5 mm

Vr= 0.18

- Vr= W_2'/W_1; Where, W_2'= Angular speed of

Flywheel

W_2' = V_2 * W_1 = 27.14336 rad/s

Considering Slip in belt drive, Slip loss in belt drive = 0.78 %

Actual flywheel Angular speed W_2 = W_2'* (100-0.78)/100 =

26.93164 rpm

- Power available at flywheel P_2 = P_1 – Power loss in belt drive

Efficiency of B section V groove belt drive = 86 to 94 %

P_2 = 0.94 * 14.91399744 =

14.01916 KW

Flywheel:

- Torque available at flywheel T2=P2/W

=. 520546 KNm

= 520.546 Nm

Pinion:

- Angular speed of Pinion shaft W_3 =W_2

= 26.93164 rpm

- Torque available at pinion T_3= T_3 = 520.546Nm
- Power available at pinion P_3= P_2 = 14.01916KW

Crank Gear:

Gear ratio = Tp/Tg; Tp= Number of teeth on Pinion = 12

Tg= Number of teeth on Crank gear = 75

- Angular speed of crank gear W_4 = G_r *W_3

=4.309063 rad/sec

- P_4 = Power available at crank gear

= Efficiency *P_3

= 0.99 *14.01916

= 13.87897 KW

- Torque available at gear shaft = P_4 /W_4

= 3220.87 Nm

6. CALCULATION OF C.G. FOR CRANKSHAFT

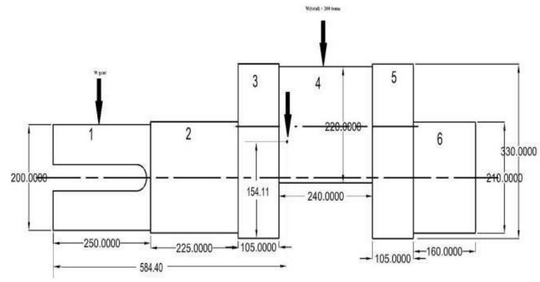


Fig -13: C.G. calculation.

Figure.13 shows the calculation of the centre of gravity on the crankshaft.

Part 1:

Volume $V_1 = 785 \times 10^4\ mm^3$ $X_1 = 125$ mm

$Y_1 = 115$ mm

Part 2:

Volume $V_2 = 779 \times 10^4 mm^3$

$X_2 = 362.5$ mm

$Y_2 = 115$ mm

Part 3:

Volume $V3 = 751 \times 10^4 mm^3$

$X3 = 527.5$ mm

$Y3 = 165$ mm

Part 4:

Volume $V4 = 912 \times 10^4\ mm^3$

$X4 = 700.0$ mm

$Y4 = 225$ mm

Part 5:

Volume $V5 = 751 \times 10^4\ mm^3$

$X5 = 872.5$ mm

$Y5 = 165$ mm

Part 6:

Volume $V6 = 554 \times 10^4\ mm^3$

$X6 = 1005$ mm

$Y6 = 115$ mm

$R_A + R_B = W_S + W_{total} + W_g + 200$ tone---------------- (1)

Part 7:

Volume $V7 = 47 \times 10^4 mm^3$

$X7 = 120$ mm

$Y7 = 115$ mm

7. STATIC STRESS ANALYSIS BY ANALYTICAL METHOD

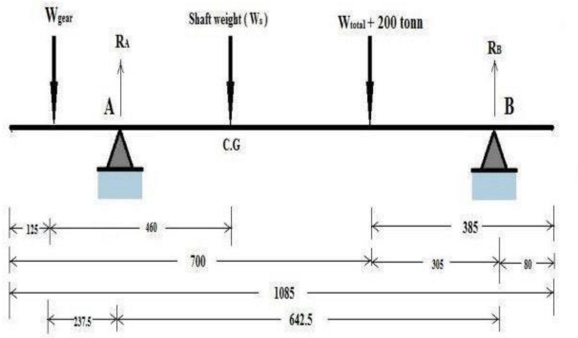


Fig -14: Line and force diagram

Figure.14 shows the forces acting on the crankshaft considering it to be a simply supported beam.

Shaft weight Ws = 340 Kg

= 3335.4 N

W_{total} = 81.47 (Ball screw) + 110(ball cylinder) + 49(Ball seat) + 57(CR cap) + 122(CR) + 45(Retainer) + 1500(Ram)

= 1965 Kg = 19276.7 N

Gear weight Wg = 950 Kg

= 9320 N

Here, at point A and B simply supported beam is provided.

Resolving vertical force

Where, R_A – Reaction Force at A

R_B – Reaction Force at B

W_S – Shaft Weight

W_{total} – Total Weight including Ram assembly

W_g – Gear Weight

Moment at point A, $M_A = 0$.

$0 = (W_g \times 237.5) - [(W_t + 1962000) \times 337.5] + (R_B \times 642.5) - (W_S \times 222.5)$------------------(2)

Put the value of W_S and W_t and W_g in equation-(2)

$642.5\ R_B = 667209469$

Moment at point B,

$0 = (W_S \times 420) + [(W_t + 1962000) \times 305] - (R_A \times 642.5) +$

$(W_g \times 880)$

$642.5\ R_A = 613891832.975$

So, Maximum Bending moment @ mid- point of crank pin:

$= R_B \times 305$

$= 316729.79$ Nmm

According to Maximum shear stress theory:

$\sigma b = 32\ M / \pi d^3$

$= 403.27 \times 10^6\ N/m^2$

$\sigma b = 403.27$MPa

Shear stress due to twisting moment

$\tau = 16T/ \pi\ d^3$

$= 2.05$MPa

$\tau_{max} = 1/2 * (\sigma^2 + 4\ \tau^2)^{1/2}$

$= 201.6454$ MPa

$\sigma b_{max} = (½)\ \sigma b + \tau_{max}$

$= 403.28$MPa

8. STATIC STRESS ANALYSIS BY USING SOFTWARE

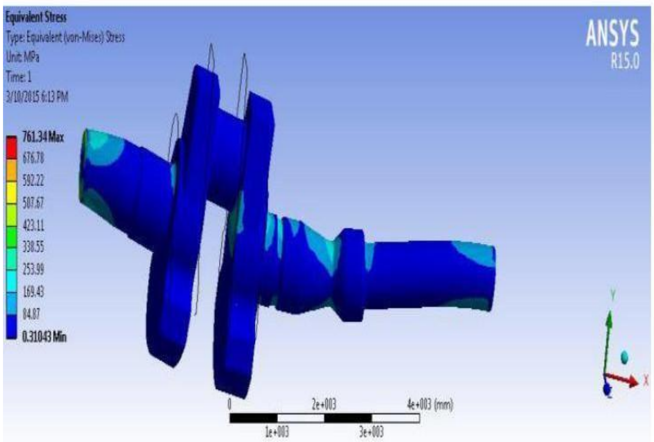
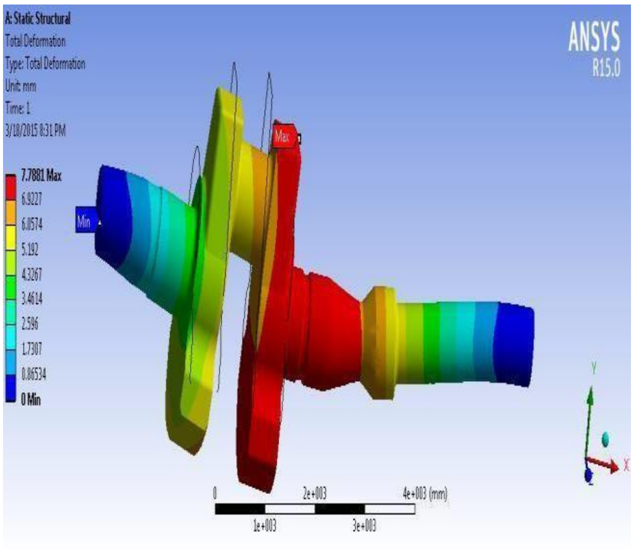


Fig -15: Von-Misses stress induced in crankshaft.

After completing pre-processing steps like material properties defined, boundary conditions and applying forces on crankshaft, the analysis in the static structural analysis was carried out. It was found that the von misses stress induced in the crankshaft is nearly 761.34 MPa as shown in figure.15. Hence the design is safe according to von-misses stress and static structural analysis.



b

Fig -16: Total deformation in crankshaft.

Fig.16 shows the total deformation of the crankshaft after performing the static structural analysis. The deformation is nearly about accurately and approximately answer equal to 7.338 mm.

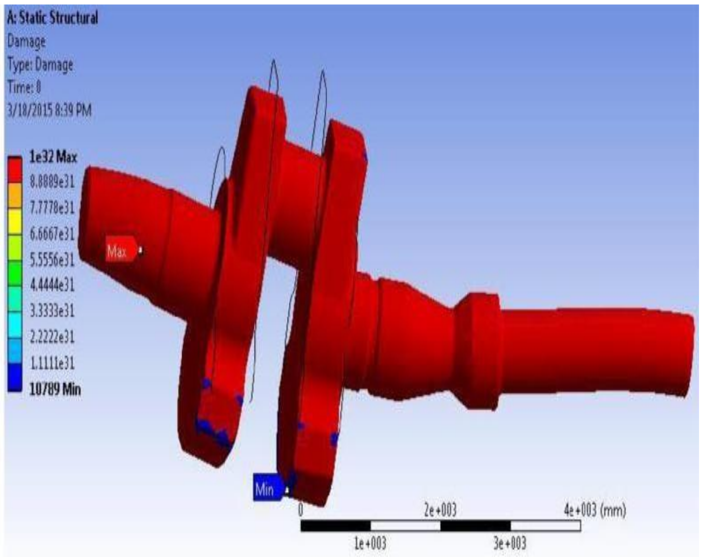


Fig -17: Fatigue damage

These two figures 16 & 17 show the fatigue life and fatigue failures. In this analysis, the red portion is very large and these portions show the almost zero life and maximum damage under fatigue during critical loading conditions.

9. CONCLUSIONS

After performing the analysis and conformable matches with the theoretical and software results, the design is safe in static structural analysis and the maximum deformation is produced between crankpin and main journal area.

The suitability of alternative compatible materials like SG iron 500/7, SG Iron 700/2 grades are considered for different analysis.

On the basis of interpretation and results, we found that the SG Iron 500/7 grade material is best suited for this crankshaft instead of EN24. FEA and Fatigue is good tool to reduce theoretical and costly experimental work.

The life and strength of this crankshaft can be improved to some extent by changing the chemical composition of different materials.

Another possible solution is design modification and optimization like increasing radius of crankpin to reduce stress concentration at critical locations to reduce the tendency of fatigue failure.

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