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Analysis and Modification of Drive Shaft for 5-Ton Capacity Ball Mill

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Abstract — **Ball mill drive shaft frequently fails due to alternative load acting. In this project deals with analysis of exiting drive shaft and design modification of drive shaft. Shaft model prepare with help of Creo 2.0. Equivalent stress, maximum shear stress, deformation and fatigue life analysis of existing drive shaft with help of ANSYS 15.0. Modification of existing drive shaft design and analysis modified drive shaft. In modified drive shaft to minimize stresses and improve the working life of drive shaft. In existing drive shaft made of AISI 1018 material also compare with EN8 material.**

Keywords-Ball Mill, Flanged shaft, FE Analysis, Fatigue life, least square method

I. INTRODUCTION

A ball mill is types of grinder used to grind material into extremely fine powder for use in paints, ceramic, pharmaceuticals and cement industries. Ball mill also known as centrifugal or planetary mill, are device used to rapidly grind material to colloidal fineness (Approximately 2 micron and below) by developing high grinding energy via centrifugal and planetary action. An internal cascade and cataract effect reduces the material to a fine powder [1].Particle of material between 5 to 250 mm are reducing to 20 to 300 µm.

Drive shaft of the ball mill is one of the importance parts of mill. Shaft is rotating member and supported the ball mill. In ball mill two shaft used, in drive side shaft known as the drive shaft and another side shaft known as none drive shaft. The drive shaft generally made of two stepped shaft. One side step is connected with ball mill and another side step with plumber block show in the figure 1. Generally shaft are not uniform diameter but are stepped to provide on it change of cross section area produce high stress and fatigue failure [2]. Failure of an elevator shaft due to torsion-bending fatigue was given in [3] the fatigue cracks originated mainly at top edge of key slot [4]. Find the accurate stress concentration factor using least square method [5]. Turbine shaft crack was originated by a process of high cycles and high stress at change area section [6] the stress concentration factor for bending and torsion accurate find use of least square method [7]. Due to fatigue crack produce at high stress region and crack extended and finally facture of shaft [8].

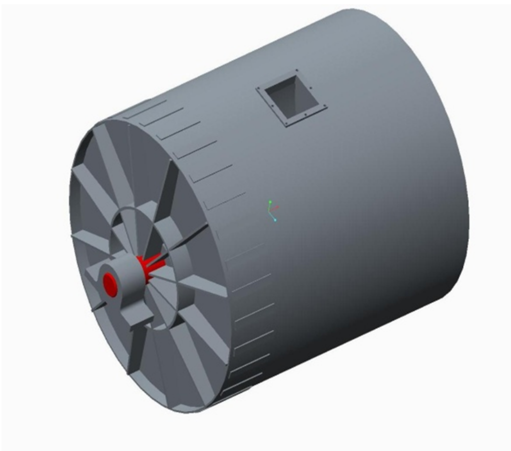

Figure 1, Ball mill

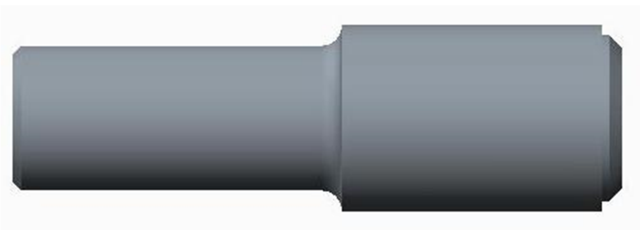

Figure 2, Drive shaft

II. LOAD CALCULATION

Here ball used the belt drive system. The belt tension force acting on the ball mill, also ball mill self weight, grinding material weight, grinding media weight and centrifugal force also acting on the drive shaft.

2.1. Basic specification of mill

- Types of mill : Ball mill
- Types of grinding : Wet grinding
- Capacity : 5 Ton
- Ball mill filling : 45%
- Electric motor : 75 HP
- Mill rotation speed : 15 RPM
- Types of drive system : Belt drive
- Diameter of ball mill =3064 mm
- Diameter of drive pulley = 510 mm
- Distance between drive pulley and driven pulley = 2988 mm

2.2. Belt tension force

T1 = Tension in tight side, N
T2 = Tension in slack side, N
θ = Angle of contact=3.99634 radian

$$\frac{T_1}{T_2} = e^{\mu\theta} = 2.7156$$

$$P = (T_1 - T_2)V$$

$$T_2 = 13.552 \times 10^3 \text{ N}$$

$$T_1 = 36.802 \times 10^3 \text{ N}$$

Here the ball mill and motor pulley inclined with 45 degree, there for here vertical and horizontal components.

$$F_V = (T_1 + T_2)\cos\theta$$

$$F_V = 35.6057 \times 10^3 \text{ N}$$

$$F_H = 35.6057 \times 10^3 \text{ N}$$

Torque on the shaft of ball mill:

$$T = (T_1 - T_2) \times \frac{D}{2}$$

$$T = 35.6190 \times 10^6 \text{N} - \text{mm}$$

2.3. Weight calculation

Here the mass calculate using cad software Creo 2.0

- Mass of ball mill shell = 5177.3 kg
- Mass of two end disc parts = 2 × 2294.9 =4589.8 kg
- Mass material weight = 5000 kg
- Grinding media Mass = 5000 kg
- Water Mass = 5000 kg
- Total mass = 24.7671 × 10^3 kg

Ball mill supported by two shafts, drive shaft load acting,

$$W = \frac{W_{total}}{2}$$

W = 121.4826 × 10^3 N

2.4. Centrifugal force

Ball mill fill level 45%. Show in the figure center of gravity ball mill. Internal radius of ball mill, R = 1500 mm
Center of gravity of mass is at distance of,

$$r = \frac{3R}{8} = 0.5625 \text{ m}$$

$$F_c = m\omega^2 r$$

$$=15000 \times 1.57079^2 \times 0.5625$$

$$=20.8185 \times 10^3 \text{ N}$$

Maximum effect of centrifugal force is downward direction

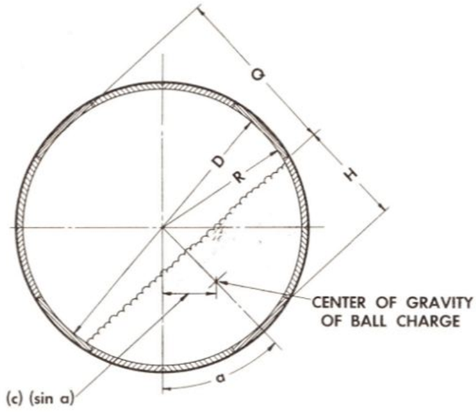

Figure 3, C.G of ball mill

Total force in vertical direction

$F_{v\ total}$ = Tension in vertical + weight of ball mill and material + centrifugal force = 177.9565×10^3 N

Total force in horizontal direction

$F_{H\ total}$ = Tension in horizontal = 35.6057×10^3 N

III. ANALYSIS OF EXISTING DRIVE SHAFT

Use Ansys workbench to analysis of existing drive shaft of ball bill. Finite element analysis involve three phase.

1. Pre processor phase
2. Solution phase
3. Post processor phase

Using of creo parametric to create the cad model of drive shaft and import into Ansys workbench. Show in the figure of meshing of drive shaft.

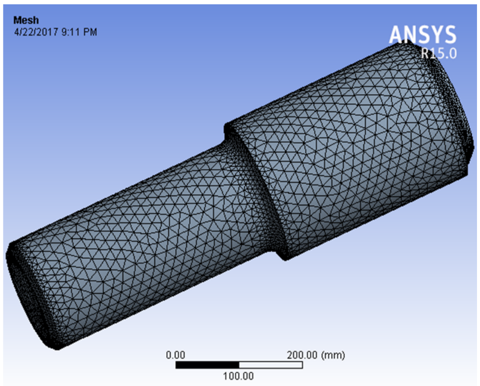

Figure 4, Meshing of shaft

After meshing the boundary condition apply on the drive shaft.

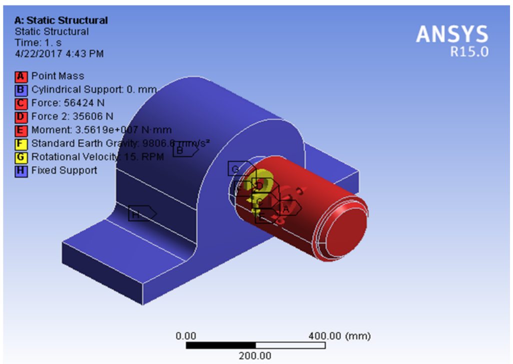

Figure 5, Boundary condition

Apply boundary condition after solution. Here equivalent stress, maximum shear stress, maximum principal stress, deformation, fatigue life and safety factor analysis use to Ansys workbench. Analysis results show the table 1.

Table 1, analysis result of existing drive shaft

Analysis type	Result	Figure no.
Equivalent stress(Mpa)	236.58	6
Maximum shear(Mpa)	131.43	-
Maximum principal(Mpa)	201.38	-
Deformation(mm)	0.13214	-
Life(cycle)	14185	7
Safety factor	0.36437	8

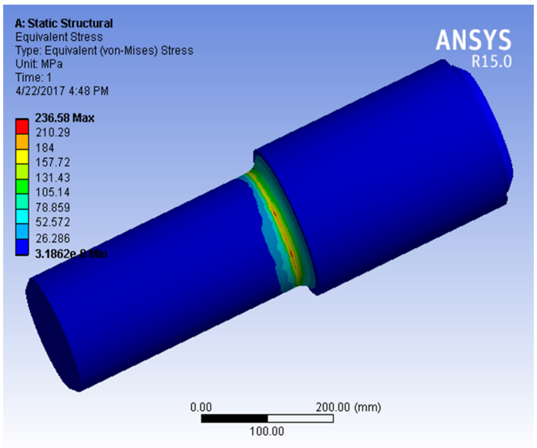

Figure 6, Equivalent stress

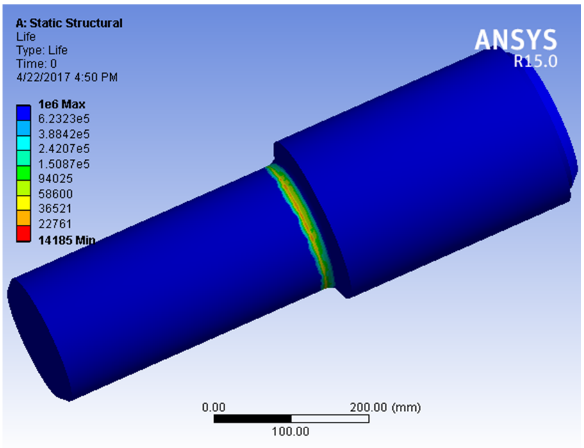

Figure 7, life

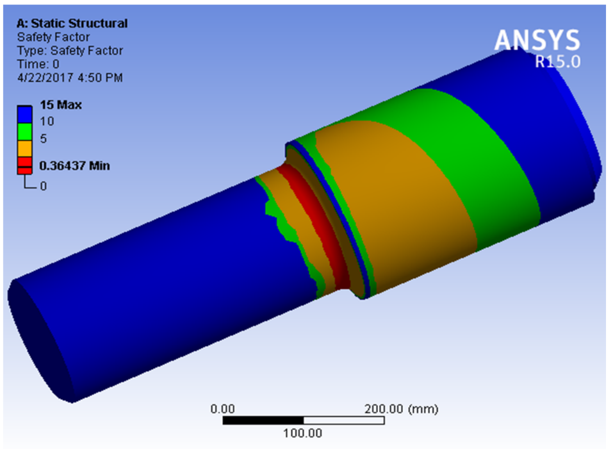

Figure 8, safety factor

As per above analysis result existing drive shaft failure due to high stress at region of change the cross section area. Drive shaft fatigue failure accrue 14185 working cycle.

IV. MODIFICATION OF DRIVE SHAFT

The modification design of drive shaft to minimize the stress and shear stress also improve the working life. Replacement existing drive shaft very difficulties because drive shaft and ball mill end disc both are joint with help of welding. In modification of drive shaft like as a flanged type drive shaft to easy replacement with less time and effort. Also existing drive shaft material used AISI 1018 with compare with the EN8 material, material properties listed below table 2.

Table 2, Material properties

Name of materials	Density	Shear modulus	Poisson ratio	Tensile strength	Ultimate Tensile strength
1018	7870	80	0.28	370	440
EN8	7850	80	0.30	530	625

4.1. Design of flanged type drive shaft.

As per above load calculation the horizontal load, vertical load and torque acting on the drive shaft of the ball mill. Shaft design consideration of torsion and bending or combine stress acting on shaft. Here horizontal and vertical load acting on shaft to find the resultant moment.

$M_{resultant} = \sqrt{(M_v)^2 + (M_H)^2} = 67.13077 \times 10^6$ N-mm
$T = 35.6190 \times 10^6$ N-mm
A shaft subjected to combine bending and torsion, the equivalent twisting moment,
Km = combined shock and fatigue factor for bending = 2.5,
Kt = combined shock and fatigue factor for torsion = 2.5
$T_{equvalient} = \sqrt{(K_m \times M)^2 + (K_t T)^2} = 189.987 \times 10^6 \text{ N} - \text{mm}$
Equivalent bending moment,
$M_{equvalient} = \frac{1}{2}\left[(K_m \times M) + \sqrt{(K_m \times M)^2 + (K_t T)^2}\right] = 178.9073 \times 10^6 \text{ N} - \text{mm}$
Diameter of the shaft is calculate on the basically two types of failure theory,
1)According to the maximum shear stress theory,

$$\tau = \frac{16 \times T_e}{\pi d^3}$$

d = 250.41mm = 255 mm
2)According to the principle stress theory

$$\sigma = \frac{32 \times M_e}{\pi d^3}$$

d = 245.38 mm = 250 mm
d = 250 mm
Consider the d= 255 mm of drive shaft diameter.
Flange design as per standards empirical relation [9,10],

- Outer diameter of flange = 2.70 d = 690 mm
- Diameter of bolt circle = 2.37 d = 605 mm
- Thickness of flange = 0.15 d = 38 mm
- Length of flange = 0.4 d = 102mm
- Number of bolts = 16

4.2. CAD model of flanged type drive shaft.

Figure 9, Modified drive shaft model

4.2. Analysis of flanged type drive shaft.

Analysis of flanged type drive shaft same boundary and loading condition apply on it. Analysis carried out of two different material properties apply the two materials used AISI 1018 and EN8. The analysis results show in table 3.

Table 3, analysis result of modified drive shaft

Analysis type	**Flanged type drive shaft (AISI 1018)**	**Flanged type drive shaft (EN8)**	**Figure no.**
Equivalent stress(Mpa)	86.001	59.056	10,11
Maximum shear(Mpa)	47.75	34.035	-
Maximum principal(Mpa)	78.551	44.03	-
Deformation(mm)	0.041572	0.04102	-
Life(cycle)	9.5641E5	10E6	-
Safety factor	0.99229	1.4596	12,13

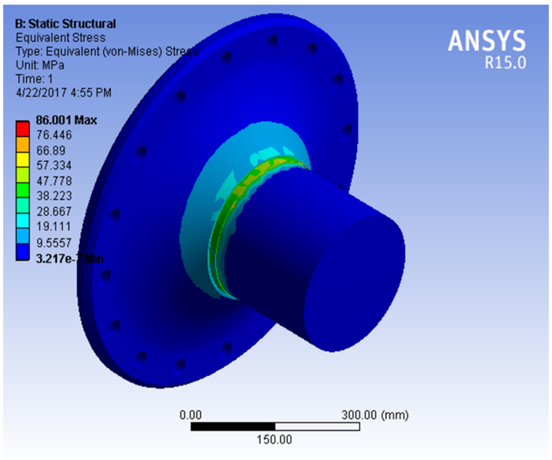

Figure 10, Equivalent stress (AISI 1018)

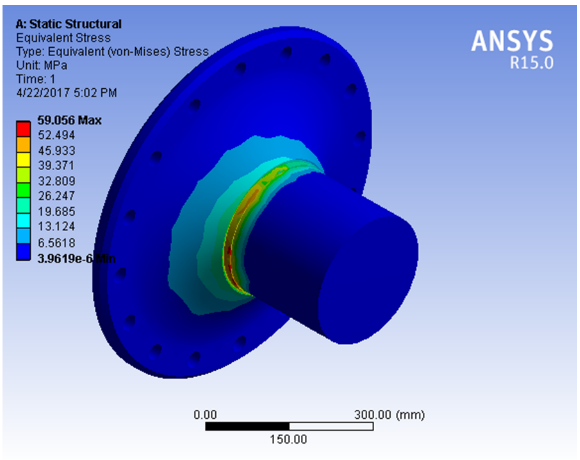

Figure 11, Equivalent stress (EN8)

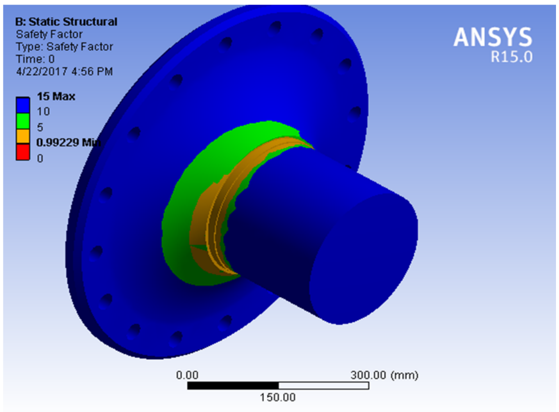

Figure 12, Safety factor (AISI 1018)

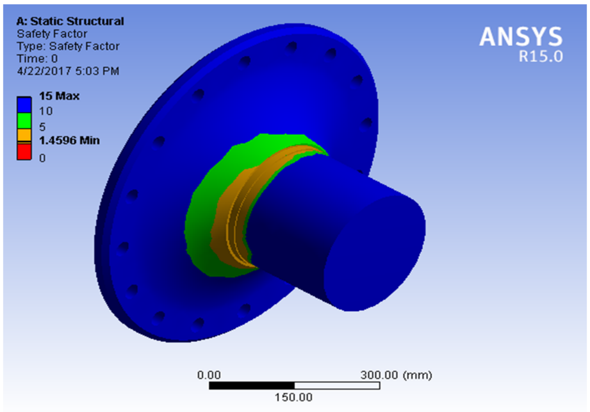

Figure 13, Safety factor (EN8)

V.CONCLUSION

As per above analysis of existing drive shaft and modified design analysis, results compression below table 4.

Table 4, Results comparission

	Exiting shaft (1018)	Modified shaft (1018)	Modified shaft (en8)
Equivalent stress(Mpa)	236.58	86.001	59.056
Maximum shear(Mpa)	131.43	47.75	34.035
Maximum principal(Mpa)	201.38	78.551	44.03
Deformation(mm)	0.13214	0.041572	0.04102
Life(cycle)	14185	9.5641E5	10E6(infinite)
Safety factor	0.36437	0.99229	1.4596

- Modified drive shaft made of EN8 material equivalent stress, maximum shear stress, maximum principal stress and deformation reduce.
- Modified drive shaft improves the working life.
- Safety factor improve therefore less chance to damage of drive shaft.
- Easy to replacement of modified drive shaft, less time and effort required replacement it.

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