Synthesis And Analysis of Grinding Wheel of Fly Ash Grinding Machine

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***Abstract* — This paper consists of the redesign, analysis, synthesis and experimental verification of the grinding wheel for fly ash grinding machine. Grinding wheel was designed to take up the forces required to grind fly ash of 250 microns to 50 microns size. Grinding wheel also considered to take centrifugal force generated due to 238 rpm of central shaft and 900 rpm of shaft of wheel. The mathematical calculations of various stresses due to dynamic loading conditions on grinding wheel of fly ash grinding machine is carried out. Contact stress in grinding wheel and race is very prominent parameter of grinding wheel design. Generation of 3D CAD model in CAD software with exact dimensioning and according to manufacturing process of fly ash grinding wheel is considered. The analysis on ANSYS software with exact loading condition is carried out. In this paper, numerical analysis (FEA) of contact stress for grinding wheel is studied by finite element method (FEM). Stress analysis is carried out to find out highly stressed components of grinding wheel which are prone to failure. Verification of Stress on grinding wheel by analytical tool is performed. The parameters for study of comparison are selected as vibration and temperature. On completion of analysis, experimentation and comparison suitable correction is suggested. Verification of improved design is carried out.**

***Keywords* – Fly Ash Grinding Roller, Contact Stress, CAD, Ansys**

I. INTRODUCTION

Floating roller grinding machine is the special sort of grinding machine used to grind fly powder from material after burning (fly ash).[1] In floating roller grinding machine rollers are subjected to high pressure and this high pressure is used to grind the fly ash particles.[2] Power is supplied to the central shaft of floating roller grinding machine with the help of electric motor followed by gearing unit. Central shaft rotate with 238 rpm. Floating roller grinding machine consist of four stages of grinding rollers. Each stage consists of top and bottom separator plates six rollers are secured in between top and bottom turn plates. Six in each stage and total 4 stages i.e. total 24 floating rollers used to grind fly ash. Roller assembly secured in top and bottom turn plate with the help of shaft grooved at both the ends. Two bearings are mounted on the shaft to support outer cylindrical roller. Top and bottom roller plates are bolted to seal the bearing lubrication. Oil seals are also used to avoid leaking of oil from roller assembly. Grinding roller rotate inside stationary outer cylindrical race and fly ash is ground in between roller and outer cylindrical race to give fine fly ash powder. In this paper study of failures

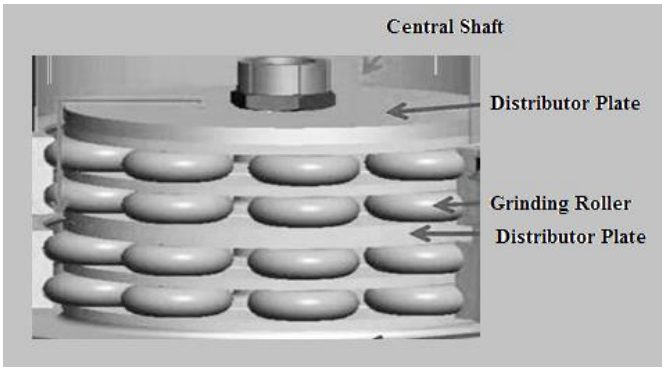

Fig. 1. Schematic diagram of fly ash grinding roller

and faults in the previous design and analysis to avoid failures are carried out. Following shows the schematic diagram of the fly ash grinding machine.

A. Problems in initial design of roller

1. Two spherical roller bearings were used to transfer load and smooth motion of roller. Spherical roller bearings are failing after some time of operation. Technical bearing mounting technique was not used while assembling the roller hence it leads to loose fit with shaft and hence high amplitude vibrations are generated in the machine.

2. Axial force on the shaft is not considered while selecting bearing hence it is one of the reasons of bearing failure.

3. Separate grinding liner was used of same material as of inner cylindrical roller. Grinding linear was fitted with the help of withdrawal sleeve and locknut. After some period time as grinding progress due to some wear grinding liner and inner roller fit become loose and hence it leads to vibration and unnecessary stresses in the roller.

Single intact roller is used in fly ash grinding machine hence roller is redesigned to eliminate grinding liner. Shaft is redesigned as per new bearing selection and axial force to reduce failures vibrations and operating temperature rise.

This work was supported in part by Mechanical (Design) Engineering Department in JSPM's RSCOE, Tathwade, Savitribai Phule Pune University.
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II. Design Calculations

A. Forces and Angular Velocities of Grinding Roller

1. Bond Work Index (for < 3.36 mm particle size) [3]

$$W_{i,B} = \frac{4.9}{{x_{max}}^{0.23} * G^{0.82} * (\frac{1}{\sqrt{x_{80}}} - \frac{1}{\sqrt{x_{80}}})} \; [kWh] \quad (1)$$

$$W_{i,B} = 0.0835 \; [kWh]$$

2. Energy (E) required to grind fly ash is calculated by multiplying work index and feed rate per seconds.[4][5]

$$E = W_{i,B} * feed\ rate\ per\ seconds \quad (2)$$

$$E = 835.305 \; kJ/s$$

3. Angular velocity along central shaft in rad/s[10]

$$w_c = \frac{2 * \pi * N_c}{60} \frac{rad}{s} \quad (3)$$

$$w_c = 24.92 \; rad/s$$

4. Linear velocity calculated as product of radial distance and angular velocity.

$$Liner\ Velocity\ V = r * w_c \quad (4)$$

$$Liner\ Velocity\ V = 11.288 \; m/s$$

5. Force required to grind fly ash is calculated by dividing energy require to linear velocity.

$$Total\ force\ required\ \ F = \frac{E}{V} \; kN \quad (5)$$

$$F = 71.81 kN$$

$$F_s = \frac{Total\ Force}{No.of\ stages}$$

$$F_s = 18 \; kN$$

6. Energy spent at each roller is converted to kinetic energy generated by each roller.

$$Er = E_{KE} \quad (6)$$

$$v = 7.14 \; m/s$$

Linear velocity is the product of radius of roller and angular velocity.

$$w_r = \frac{v}{r} \frac{rad}{s} = 72.92 \frac{rad}{s} \quad (7)$$

Minimum attainable speed of the roller is,

$$w_{r,min} = 72.92 \; rad/s$$

Maximum attainable speed by roller is calculated by considering positive contact between two cylinders.

$$r_c * w_c = r_r * w_r \quad (8)$$

$$w_{r,max} = 115.20 \; rad/s$$

$$w_{r,avg} = 94.06 \; rad/s$$

$$Average\ angular\ velocity = N_{r,avg} = 900 \; rpm$$

B. Grinding Roller Design

1. Minimum mass of roller assembly

Grinding of fly ash is done by centrifugal force acted between two cylinders. Centrifugal force acted between two cylinders is equal to force require to crush fly ash.[4]

$$Centrifugal\ Force\ F_{cent} = Crushing\ Force\ F_c = 18\text{KN}$$

$$Crushing\ Force\ F_c = m_r * r_{rc} * {w_c}^2 \quad (9)$$

$$m_r = 54.62 \; kg$$

Roller assembly can have minimum mass of 54.62 kg.

Standard pipe SA106GrB selected for roller design. From previous design total mass of shaft, roller plates and bearing is 34Kg. From material properties and dimensions of roller mass of roller is 20.72Kg. Total mass of roller assembly become 54.72Kg. Hence grinding force can be achieved with this roller mass.[12]

2. Contact stress induced in the cylinder

If the cylindrical surfaces are in contact, the contact region is approximately along a straight line element before loads are applied. In these cases, the radii ${R_1}'$ and ${R_2}'$, which lie in a plane perpendicular to the paper, are each infinitely large so that $1/{R_1}'$ and $1/{R_2}'$ each vanish identically and the angle = 0. All the stress and deflection calculation require first that values be obtained for B, A and Δ; these are given by following equations [8].

The radii ${R_1}'$ and ${R_2}'$ which lie in the plane perpendicular to the paper, are each infinitely large so that $\frac{1}{{R_1}'} and \frac{1}{{R_2}'}$ each vanish identically and the angle $\alpha = 0$.

From Hertz contact stress equation for cylindrical surfaces in contact [12]

$$B = \frac{1}{2} * \left(\frac{1}{R_1} + \frac{1}{R_2}\right) \quad (10)$$

$$A = 0$$

$$\frac{B}{A} = \infty$$

From values of B and A, value of Δ can be calculated as[9],

$$\Delta = \frac{1}{\left(\frac{1}{2}R_1\right) + \left(\frac{1}{2}R_2\right)} * \left[\left(\frac{1 - {v_1}^2}{E_1}\right) + \left(\frac{1 - {v_2}^2}{E_2}\right)\right]$$

$$= 0.002333 \quad (11)$$

Maximum pressure pmax is given by $= \sqrt{\frac{2 * F}{\pi * b}} = 136.9$

Maximum shear stress is given by $= 0.3 * P_{max} =$
$= 41.1\ Mpa$

Maximum octahedral shear stress is given by $= 0.277 * \frac{b}{\Delta}$
$= 50.131\ Mpa$

C. Bearing Selection

As per analysis from Timken Taper Roller Bearing TRB 30312 selected. Selected Bearing details are as follows:

Position	Bearing Type	Part No	Bore "d" (mm)	O.D. "D" (mm)	Width "T" (mm)	Radial Rating C1 (kN)	Axial Rating, Ca1 (kN)	Static Rating, C0 (kN)	K factor	Mounted End Play (M.E.P) mm
Bottom	TRB	30312	60.000	130.000	33.500	201	119	221	1.69	0.050
Top	TRB	30312	60.000	130.000	33.500	201	119	221	1.69	

Fig.2. TRB 30312 specifications

Timken recommended BARRIERTA L 55/2 (Kluber) lubricant for grinding roller application based on maximum operating temperature and viscosity at 100°C. Analyses with lubricant are as follows:

Condition	Bearing Position	Part no.		Radial load, N	Axial load, N	L10a Life* (Hrs)	Max. Inner Race/Roller contact Stress-MPa	Load zone (Degree)	Lambda ratio, λ	Power loss (Watt)
Condition1	Bottom	30312	Row 1	4,900	1,200	9,867,000	967	100	0.33	14
	Top	30312	Row 2	13,100	6,200	620,000	1,157	360	0.31	74
	Bottom	30312	Row 1	21,600	5,900	160,000	1,979	166	0.30	69
	Top	30312	Row 1	3,600	900	46,059,000	809	121	0.35	10
Weighted	Bottom	30312	Row 1			315,000				
	Top	30312	Row 2			1,224,000				

Fig.3. TRB 30312 Timken analysis report

Standard bearing fitting practices suggested from Timken is used to mount bearing on the roller shaft.

D. Design of roller shaft for selected bearing and loads

$Power\ P = 400kW$
$Speed\ of\ the\ motor\ N_m = 1480\ rpm$

Speed and torque relation,

$$\frac{T_r}{T_m} = \frac{N_m}{N_r} \tag{12}$$

Power is directly proportional to the torque,
$$Torque \propto Power$$

$$\frac{P_r}{P_m} = \frac{1480}{900}$$

$P_r = 657.6\ kw$

Torque generated due to given power can be calculated as:

$T_r = 6977.35\ Nm$

Hence, torque generated at roller shaft is $T_r = 6977.35\ Nm$

$$Maximum\ bending\ moment\ M = F_c * AC \tag{13}$$

$M = 9000 * 0.08825$

$M = 794.25\ Nm$

$$Least\ radius\ of\ gyration\ K = \sqrt{\frac{I}{A}} \tag{14}$$

$K = 0.0125\ m$

Maximum limit of the length of the shaft $L = 236\ mm$
Slenderness ratio is given by,

$Slenderness\ ratio = \frac{L}{K}$
$Slenderness\ ratio = 18.88$

If $\frac{L}{K} < 115$

$\alpha = 1.0905$

$K_m = 1.5\ and\ K_t = 1.0$

Equivalent twisting moment of shaft is,[10]

$$T_e = \sqrt{[K_m * M + \frac{\alpha * F * d}{8}]^2 + [k_t * T]^2} \tag{15}$$

$T_e = 7084.54\ Nm$

Equivalent twisting moment of shaft is,
$T_e = \frac{\pi}{16} * \tau * d^3$

$7084.54 = \frac{\pi}{16} * \tau * 0.06^3$

$\tau = 288.6501\ Mpa$

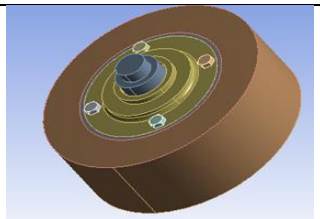
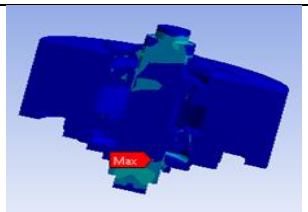
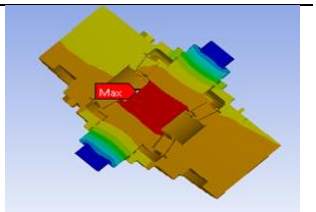
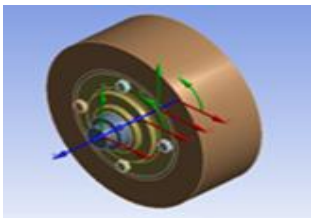
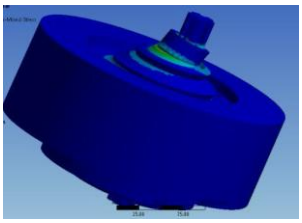
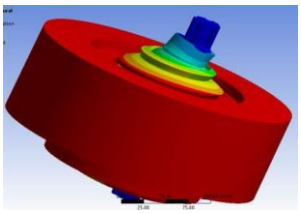
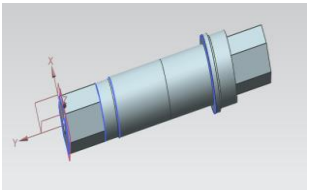
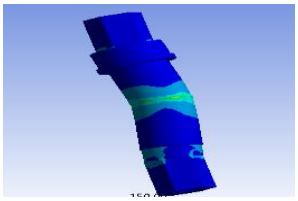
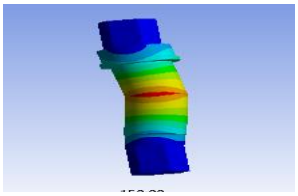
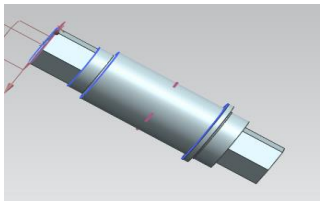
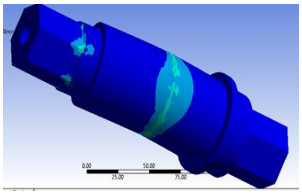
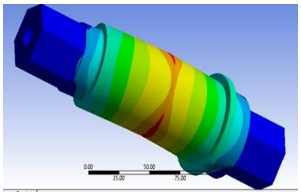
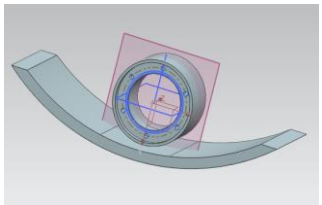
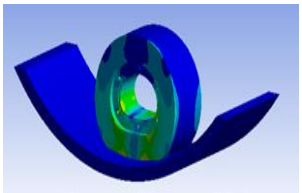
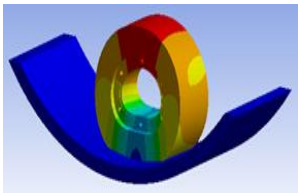
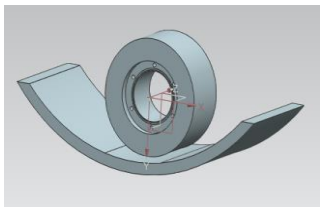
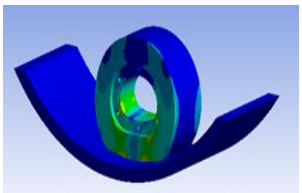
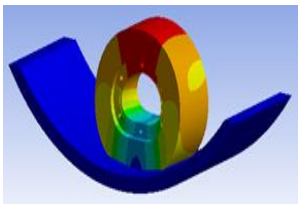
Magnesium steel grade 7 is used to design the shaft which has allowable shear stress 0.58*yield stress = 400.2MPa.

III. Analysis Results of Components in Roller Assembly

Analytical Design is verified with the help of ANSYS software. Exactly similar forces and constraints simulated in ANSYS to get accurate results. Von-mises stresses and Total deformation is calculated and compared for shaft, contact stresses in roller and total assembly.

ANALYSIS RESULTS OF COMPONENTS IN ROLLER ASSEMBLY

TABLE I
ANSYS ANALYSIS RESULTS

	Model	Stress(Mpa)		Total Deformation(mm)	
Analysis of previous roller assembly					
Values		MAX 130.156	MIN 0.000322	MAX 0.07612	MIN 0
Analysis of new roller assembly					
Values	Induced Stress 91.148	MAX 99.318	MIN 0.00094	MAX 0.011428	MIN 0
Analysis of previous shaft					
Values		MAX 305.26	MIN 1.19e-9	MAX 0.01039	MIN 0
Analysis of new shaft					
Values	Induced Stress 288.65	MAX 296.17	MIN 1.25e-9	MAX 0.1364	MIN 0
Contact Stress analysis of previous cylindrical roller					
Values		MAX 44.118	MIN 9.546e-9	MAX 0.0120	MIN 0
Contact Stress analysis of new cylindrical roller					
Values	Induced Stress 41.1	MAX 45.68	MIN 9.146e-9	MAX 0.002247	MIN 0

IV. Experimental Results

A. Temperature Test

Fly ash is sticking to the inner race of roller cylinder due to rise in operating temperature of fly ash grinding. HTC infrared thermometer MT-4 is used to measure temperature. Improved design and reduction in unnecessary vibration leads to operating temperature reduction up to 8℃

TABLE II
Operating Temperature

	Component/Interface	Steady State Temperature(℃)
Previous Design	Grinding Machine	63℃
New Design	Grinding Machine	55℃

B. Vibration Test

Bearing selected in previous design was not considered for operating temperature rise. Bearing expansion leads to misalignment of roller shaft inside bearing and this was the main cause of high amplitude vibrations.

Rion Vibration analyzer VA-12 with FFT analysis function was used to take vibration amplitude readings and FFT plots. Sample frequency 20 KHz used to draw FFT plots. Peak

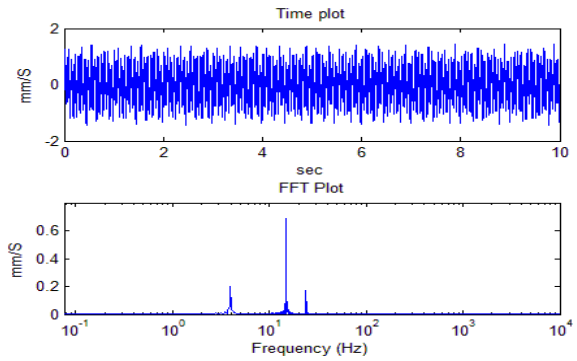


Fig.4. Time and FFT plot of before designing grinding roller.

amplitude of vibrations of fly ash grinding machine was$0.8mm$. After selection of proper bearings and redesigning shaft and roller reduced peak amplitude of machine up to

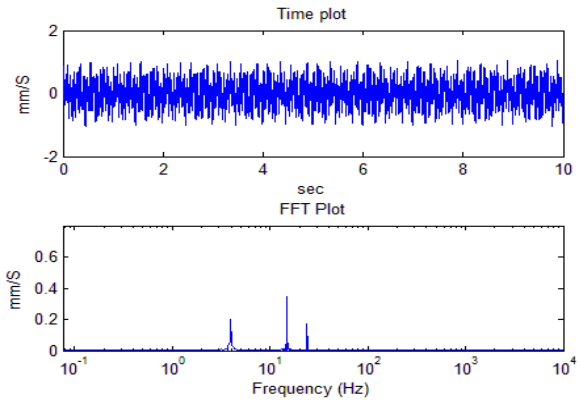


Fig.5. Time and FFT plot of after designing grinding roller.

$0.4mm$.

C. Fly ash size

Sieve BS – 140 Mesh 300 used to measure fly ash size after grinding.[11] Final fly ash size of 50 micron achieved after grinding.

V. Conclusions

1) Bearing failure and high amplitude of vibrations due to lack of contact between inner race and roller shaft is detected.

2) Redesigned the shaft as per bearing selection resulted into stress transfer and stress concentration minimization.

3) Minimization of roller deformation in revised roller design gives added advantage to minimize vibrations.

4) Vibrations and temperature rise are measured on site with the help of standard equipments. Vibrations and temperature rise reduced to acceptable limit.

5) Testing of size of fly ash with standard sieve shows that design of roller is appropriate to grind fly ash up to 50 microns.

Future Scope: Higher capacity grinding machine can be designed and analyzed on the similar guidelines.

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