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Design of a typical Autogenous Mill: Part-I

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ABSTRACT

An Autogenous Milling defined as used in this study, the term Autogenous milling means a process in which the size of the constituent pieces of a supply of rock is reduced in a tumbling mill purely by the interaction of the pieces, or by the interaction of the pieces with the mill shell, no other grinding medium being employed. The definition thus covers both 'run-of-mine' and 'pebble' milling, the only difference from the mathematical modeling viewpoint being that the feed to the first has a continuous, and the second a non-continuous, size distribution. This paper describes the detail design of a typical Autogenous mill.

Introduction

Over the past twenty or thirty years, mathematical modeling of ball and rod mills has been widely investigated, and reasonably satisfactory models are now available. The Autogenous mill has, however, received very little attention in this respect, and in view of the increasing importance of this type of mill, the Julius Kruttschnitt Mineral Research Centre in the Department of Mining and Metallurgical Engineering, University of Queensland, in 1970 commenced a programme of research into mathematical modeling of the Autogenous mill.

Special Characteristics of the Autogenous Mill Autogenous milling differs fundamentally from non-Autogenous milling in two respects. (1) Size reduction occurs by two main modes, namely the detachment of material from the surface of larger particles (referred to as 'abrasion') on the one hand, and disintegration of smaller particles due to the propagation of crack networks through them (called 'crushing') on the other. Abrasion and crushing breakage overlap on the size scale. This contrasts with non-Autogenous milling, in which only crushing breakage, however caused, is regarded as significant. (2) The grinding parameters of the Autogenous mill load are not independent of the mill feed; the load is continually generated from the feed, and its parameters therefore depend directly on those of the feed. These two characteristics must be specifically included in the model of the Autogenous mill (George, 1947).

A special development is the Autogenous or semi Autogenous mill. Autogenous mills operate without grinding bodies; instead, the coarser part of the ore simply grinds itself and the smaller fractions. To semi Autogenous mills (which have become widespread), 5 to 10 percent grinding bodies (usually metal spheres) are added. Autogenous and semi-Autogenous mills are designed for grinding or primary crushed ore, and are the most widely used in concentrators globally. Autogenous mills are so-called due to the self-grinding of the ore: a rotating drum throws larger rocks of ore in a cascading motion which causes impact breakage of larger rocks and compressive grinding of finer particles. It is similar in operation to a SAG mill as described below but does not use steel balls in the mill. Also known as ROM or "Run of Mine" grinding. Autogenous Mills operate, mechanically, similar to the ball mill. They differ in the media they use to break or grind the ore. Autogenous Mills use large particles of ore instead of steel or other balls for grinding media. Autogenous mills use large pieces of ore as grinding media. The grinding is facilitated in Autogenous mills by attrition with limited grinding by impact. For an ore to successfully grind autogenously, the ore must be competent, and it must break along grain boundaries at the desired product size. Another requirement is that the finer sizes should break easily and should be removed from the mill, otherwise, there will be a critical size buildup. Autogenous grinding has two advantages, (1) it reduces metal wear and (2) eliminates secondary and tertiary crushing stages. Thus it offers a savings in capital and operating costs. Autogenous mills are available for both wet and dry grinding. The diameter of Autogenous mills is normally two to three times the length. The ore charge is usually 25 to 35% of the mill volume. Autogenous mills have grate discharges to retain the coarse grinding media in the mill (Andrew et al., 2000).

Background

The Autogenous mill is designed for dry grinding of raw bath (cryolite) mixed with other materials as aluminium modules or sheets, iron scraps, papers, etc. The Raw bath, which is coming from the mill feed-belt conveyor, is provided with the mill spout feeder. The mill body is the driven unit of the driving pinion and ring gear. The raw bath is lifted during the rotation of the mill by the internal mill liners and it is then broken during free falling. As the size of the particle is reduced, the gas stream through the discharge trunnion removes the ground product. It is then allowed to enter in process bag filter. The reduced size of bath is obtained by impact work and attrition work. A fan exhausts the air and it is fitted with motorised dampers. An air flow meter is located at stack inlet and it controls the position of this damper. It is required to maintain the air sweeping flow at a constant value corresponding to the size of bath particles to be exhausted from the mill.

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The air drawn up by the fan then ensures 3 major functions such as 1) Reclaiming finished product from Autogenous mill, 2) Conveying ground bath up to the process bag filter 3) Making Autogenous Mill grinding process dust free, working under negative pressure. The Autogenous Mill is equipped with a lubrication unit consisting of a) One tank, b) One low-pressure motor pump unit and c) A set of accessories. The oil is pumped from the tank by a low-pressure motor pump unit and it feeds the mill bearings. The oil is filtered before reaching the bearing and cooled by an oil/air or oil/water exchanger. It is then distributed by flow regulating valves. The oil returns from the bearings to the lubrication unit by gravitational pressure. Based on the stated background and historical citation a typical Autogenous Mill has been designed and presented in this paper. Thus the objective of this research work was to design a Autogenous Mill for small and medium scale mill based on the available input parameters.

Materials and Methods:

The specifications for the design and developed Autogenous Mill are given below:

Table 1: Autogenous Mill specifications

Equipment	Autogenous Grinding Mill (Size: 3900x1450x50mm)
Type	Air Swept Discharge
Capacity	Rated- 20 TPH, Design- 30 TPH
Feed Material	Anode Bath for Aluminium Smelter
Bulk Density	1.8 – 2 T/Cu m
Desired Product Size	100% < 5 mm 95% < 3 mm 30% < 74 microns
Gearing Ratio	9.16
Pinion speed	150 rpm
Material Characteristics	Very Abrasive
Auxiliary Systems	• Tramp Metal Butts Extractor with Feed Launder Movement Device and Dust collection Outlets • Access Doors • Main Driving Mechanism and Motor (150 kW-1500 rpm) • Inching Drive with Motor (3.7 kW- 1500 rpm) • Lubrication System
Mill Shell	• 3900 mm dia inside liners • 1250 mm length inside liners
Mill Operating Speed	16.37 rpm
Design Mill Charge	45 % by volume maximum

Fig.1 shows the complete Autogenous mill. The total Autogenous mill is consists of some of the critical major assemblies like Mill body assembly, Driving gear and pinion assembly, Spout Feeder Assembly, Outlet Duct Assembly, Babbit Bearing Assemblies, Metallic Butt Extractor Assembly, Drive Base Assembly.

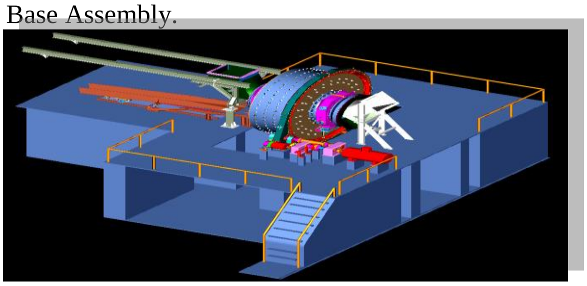

Fig.1: Complete designed and developed Autogenous mill

Design Based Calculation of the typical Autogenous Mill (Malhotra, 1969)

Power calculation of AG mill

Mill inside liner diameter (3.9 m - 2 x 0.185 m), D =3.53m
Mill inside liner length (1.35 m - 2 x 0.13 m), L =1.09m
Bulk density, BD = 2MT/m^3
Average % fraction of critical speed, C_f = 77%
Volumetric % fraction of mill charge V_F = 35%
Product size 80 % passing, P = 740 mm
Feed size 80 % passing (F) = 350000 mm
Short ton to metric tone conversion factor, k_1 =1.102
Kilo Watt to HP conversion factor, K_2 = 1.341
Work index, WI =13.50 STPH
Mill charge with 15% over loading, W = 7.47 MT

Power required based on material parameter and degree of size reduction

Optimum feed size, F_O = 3925mm
Size reduction ratio, R_r = 473:1
Specific energy by Bond Equation E = 7.00 HP/MT
Efficiency Factors: Grinding factor, f_1 = 1.3 for dry
Open circuit grinding factor, f_2 =1.2 for 80% passing
Diameter factor, f_3 = 0.93
Over size factor, f_4 = 2.21
Fineness of grinding factor, f_5 = 0.89
Specific Energy, P_{MS} = 19.85 HP/MT
Total Mill Power with margin of 15% and 90% efficiency, HP_{AG} =189 HP or 141kW

Power required based on physical parameter like size, speed, loading etc.

Critical speed, N_{cr} = 22.52 rpm
Mill speed, N = 16.37 rpm
Peripheral speed, N_{ph} = 3.20m/sec.
Specific energy, PPS = 11.31 kWh/MT
Mill Power with margin of 15% and 90% efficiency,
HPAG = 108HP
Power Model based on speed & CG of charge = 77 kW
CG distance from mill center, R_g = 0.975m
Energy constant factor, c = 1/1200 = 0.000833333
Mill Power with margin of 15% and 90% efficiency,
HPAG =134HP or 100 kW

Calculation of Load Capacity of Spur And Helical Gears(As per IS: 4460-95)

Input Data

Number Of teeth (Pinion), Z_1 = 31
Number Of teeth (Gear), Z_2 = 284
Normal Module, m_n = 16 mm
Normal Pressure Angle, α_n = 20 degree
Helix Angle, b = 0 degree
Pinion Speed, n_1 = 150 rpm
Transmited Power, P = 150000 W
Face Width, b = 150 mm
Life, L_H = 72000 h
Probality Of Failure, P_f = 0.01
Accuracy Grade = 6
Material Hardness, HB_1 = 250 BHN
Material Hardness, HB_2 = 210 BHN
Modulus Of Elasticity, E_1 = 206000 MPa
Poisson's Ratio, m = 0.3
Viscosity Grade Of Lubricant = ISO VG 32
Lubricant Viscosity,VG32, v40 = 32 mm^2/s
Lubricant Viscosity,VG32, v50 = 20 mm^2/s

Endurance Limit, σ_{Hlim1}	=	600Mpa
Endurance Limit, σ_{Hlim2}	=	460 Mpa
Nominal Endurance Limit, σ_{FE1}	=	420 Mpa
Nominal Endurance Limit, σ_{FE1}	=	300 Mpa
Profile Correction Factor, x_1	=	0
Profile Correction Factor, x_2	=	0
Mean Roughness, R_{Z1}	=	6.3 μm
Mean Roughness, R_{Z2}	=	6.3 μm
Protuberance Angle α_{pro1}	=	20^o
Protuberance Angle α_{pro2}	=	20^o
Buckling Height, h_{k2} (α_n=α_{npro})	=	0 mm

Supplementary Data

Ratio Of Gearing, u	=	9.16
Gear Speed, N_2	=	16.37 rpm
Pinion PCD, d_1	=	496 mm
Gear PCD, d_2	=	4544 mm
Outside diameter (Pinion), d_{a1}	=	528 mm
Outside diameter (Gear), d_{a2}	=	528 mm
Linear Speed, v	=	3.9 m/s
Tangential Load, F_t	=	38505.23 N
Transverse Module, m_t	=	16 mm
Base Circle Dia.(Pinion), db_1	=	466.09 mm
Transverse Pressure Angle α_t	=	20^o
Trans. Working Pr. Angle, α_{wt}	=	20^o
Base Helix Angle, $ß_b$	=	0^o
Tip Pressure Angle, α_{a1}	=	28.03^o
Tip Pressure Angle, α_{a2}	=	21.07^o
Transverse Contact Ratio, ε_{a1}	=	0.83
Transverse Contact Ratio, ε_{a2}	=	0.97
Transverse Contact Ratio, ε_a	=	1.8
Normal Contact Ratio, ε_{an}	=	1.8
Overlap Ratio, $\varepsilon_ß$	=	0
Base Circle Dia. (Gear), d_{b2}	=	4269.96 mm

Factor of safety for contact stresses: For Pinion SH1=1.50 and For Gear SH2=1.17

Factor of safety for bending stresses: For Pinion SB1=3.56 and For Gear SB2=1.44

Load factor for surface durability:

Application factor (U/M) K_A = 1.25

Load distribution factors (For contact stress)

Longitudinal $K_{H\beta}$ = 1.19

Transverse $K_{H\sigma}$ = 1.0

Dynamic load factor K_V (For contact stress)

Auxiliary value K_V =

$$\left(Z_1.\frac{v}{100}\right)\left(\sqrt{\frac{u^2}{1+u^2}}\right) = 1.20$$

Factor $K_{V\sigma}$ = 1

Factor $K_{V\beta}$ = 1

Factor K_V = KVσ - Є$_\beta$ ($K_{V\sigma}$ - $K_{V\beta}$) = 1

Zone factor Z_H:

$$Z_H = \sqrt{\frac{(2\cos\beta_b.\cos\sigma_{wt})}{\cos\sigma_1.\sin\sigma_{wt}}} = 2.49$$

Elasticity factor Z_E (For contact stress)

$$Z_E = \sqrt{\left(\frac{1-u^2}{E_1}+\frac{1-u^2}{E_2}\right)} = 189.81$$

Factor of safety for contact stresses: For Pinion SH1=1.50 and For Gear SH2=1.17

Factor of safety for bending stresses: For Pinion SB1=3.56 and For Gear SB2=1.44

Load factor for surface durability:

Application factor (U/M) K_A = 1.25

Load distribution factors:

For contact stress

Longitudinal $K_{H\beta}$ =1.19

Transverse $K_{H\sigma}$ =1.0

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Auxiliary value K_V = $\left(Z_1.\frac{v}{100}\right)\left(\sqrt{\frac{u^2}{1+u^2}}\right)$ = 1.20

Factor $K_{V\sigma}$ = 1

Factor $K_{V\beta}$ = 1

Factor K_V = KVσ - Є$_\beta$ ($K_{V\sigma}$ - $K_{V\beta}$) = 1

Zone factor Z_H:

$$Z_H = \sqrt{\frac{(2\cos\beta_b.\cos\sigma_{wt})}{\cos\sigma_1.\sin\sigma_{wt}}} = 2.49$$

Elasticity factor Z_E (For contact stress

$$Z_E = \sqrt{\left(\frac{1-u^2}{E_1}+\frac{1-u^2}{E_2}\right)} = 189.81$$

Lubrication factor Z_L

Factor of Safety for contact stress (For contact stress)

Factor C_{ZL1} = ((σ_{HLM} -850)/350)*0.8 +0.81 = 0.83

Factor C_{ZL2} = ((σ_{HLM} -850)/350)*0.8 +0.81 = 0.83

Z_L = C_{ZL} + 4 (1-C_{ZL})/(1.2+v^2) = 0.86

(For both pinion and gear)

Work hardening factor Z_W (For contact stress)

Z_{W1} = 1.2 – ((HB_1 -130)/1700) = 1.13

Z_{W2} = 1.2 – ((HB_2 -130)/1700) = 1.15

Roughness factor Z_R (For contact stress)

Z_{R1} = 1, Z_{R2} = 1

Velocity factor Z_V (For contact stress)

C_V= ((H_{LIM}- 850)/350)*0.8 +0.85

$$Z_V = C_V + 2.\frac{1.2-C_V}{\sqrt{0.8+\frac{32}{V}}} = 0.97$$

(For Pinion and Gear)

Size factor Z_S (For contact stress)

Z_{S1} = 1 Z_{S2} = 2

Reliability factor K_R (For contact stress)

K_R = 0.79 – 0.105 $\log_{10}$ (p_f) = 1

Calculated contact stress σ_{HO}:

$$\sigma_{HO} = Z_H.Z_E.Z_\varepsilon.Z_\beta\sqrt{\frac{F_t(1+u)}{bdu}}$$

Permissible contact stress σ_{HP}

$$\sigma_{HP} = \frac{\sigma_{H\lim}}{\sqrt{K_R}} Z_R.Z_V.Z_W.Z_L.Z_1$$

σ_{HP1} = 562.34 MPA σ_{HP2} = 497.41 MPA

Factor of safety S_H = $\frac{\sigma_{HP}}{\sigma_{HO}}$

For pinion S_{H1} = 1.79, For gear SH2 = 1.58

Load Factor for Bending Strength

Table 2 describes the supplementary data for the gear while Fig.2 shows the shear force and bending moment diagram.

Table 2 Supplementary Data

Sl.No.	Description	Values
1	Basic Rack Addendum, h_{ao1}	31.25 mm
2	Basic Rack Addendum, h_{ao2}	31.25mm
3	Tip Radius of Basic Rack, θ_{ao1}	5.00 mm
4	inv a_t	0.01
5	inv a_{a2}	0.08
6	Virtual Number Of Teeth, z_{v1}	182.00
7	Auxiliary Angle, θ_1	0.25^0
8	Tip Tr. Pressure Angle, θ_{ta1}	32.68^0
9	Tip Tr. Pressure Angle, θ_{ta2}	22.04^0
10	Tip Helix Angle, θ_{a1}	0.00^0
11	Tip Helix Angle, θ_{a2}	0.00^0
12	Tip Nor. Pr. Angle, θ_{an1}	32.68^0
13	Tip Nor. Pr. Angle, θ_{an2}	22.04^0

Application factor (U/M) K_A = 1.25

Load distribution factors (For contact stress)

Longitudinal $K_{H\beta}$ =1.21

Transverse $K_{H\sigma}$ =1

Dynamic load factor K_V (For contact stress)

Auxiliary value $K_V = \left(Z_1 \cdot \frac{v}{100}\right)\left(\sqrt{\frac{u^2}{1+u^2}}\right) = 0.86$

Factor $K_{V\sigma}$ = 1

Factor K_V = $K_{V\sigma}$ - $\epsilon_\beta((K_{V\sigma} - K_{V\beta})$ = 1

Life factor, Y_N

For bending strength

Life cycle = 6E+8 cycles

Life cycle L_{n2} = 7E + 8 cycles

Y_{N1} = 1, Y_{N2}= 1

Stress concentration factor

Notch parameter q_{n1} = 3.04

Notch parameter qn_2 = 3.34

Auxiliary value L_{a1} = 0.78

Auxiliary value L_{a2} =0.95

Y_{K1} = (1.2+. 13La1)*q_{n1}*(1/1.21+2.3/L_{a1}) = 1.81

Y_{K2} = (1.2+. 13La2)*q_{n2}*(1/1.21+2.3/L_{a2}) = 1.75

Contact ratio factor

Y_ϵ = 2.5 + .75/ $\epsilon\sigma$ = 0.72

Helix angle factor

Y_β = 1-ϵ_β*β/120= 1

Notch sensitivity factor

Y_{n1} =1, Y_{n2}= 1

Roughness factor

Y_{R1}=1.03, Y_{R2}= 1.03

Size factor

Y_{s1}= 1.05-1.03*m_n =0.88

Y_{s2}= 1.05-1.03*m_n =0.88

Reliability factors

Reliability factor K_R|

For bending strength

K_R = 0.79 – 0.105 $\log_{10}(p_f)$ = 1

Maximum nominal stress

$\sigma_{FO\,1}$= (F_t/bm_n)*Y_{FA}*Y_{K1}*Y_ϵ*Y_β =29.71 MPA

$\sigma_{FO\,2}$= (F_t/bm_n)*Y_{FA}*Y_{K2}*Y_ϵ*Y_β =40.68 MPA

Maximum permissible stress

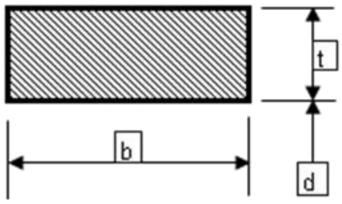
σ_{FP1} = (σ_{FE}/K_R)*Y_N*Y_n*Y_S*Y_r =381.14 MPA

σ_{FP2} = (σ_{FE}/K_R)*Y_N*Y_{n2}*Y_{S2}*Y_r =344 MPA

Factor of safety

For pinion =8.50, For gear = 5.60

Weight of the gear

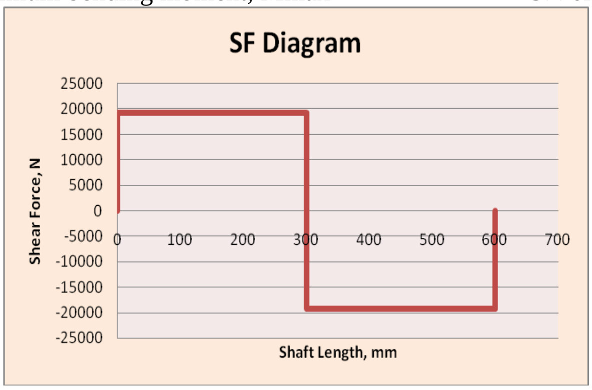


Face width b = 200 mm, Ring thickness t = 92.3 mm, Inner diameter, d = 4329 mm

Weight of the gear W = 2.03

Calculation of Pinion Shaft and Bearing:

Number Of teeth (Pinion), z1	=	31
Normal Module	=	16 mm
Pinion PCD, d1	=	496 mm
Pinion Speed, n1	=	150 rpm
Linear Speed, v	=	3.9m/s
Transmitted Power, P	=	150000W
Tangential Load, Ft	=	38505 N
Bearing span, l	=	600 mm
Pinion distance, l1 (= l/2)	=	300 mm
Bearing reaction, R1	=	19253 N
Bearing reaction, R2	=	19253 N
Maximum bending moment, Mmax	=	5776N m



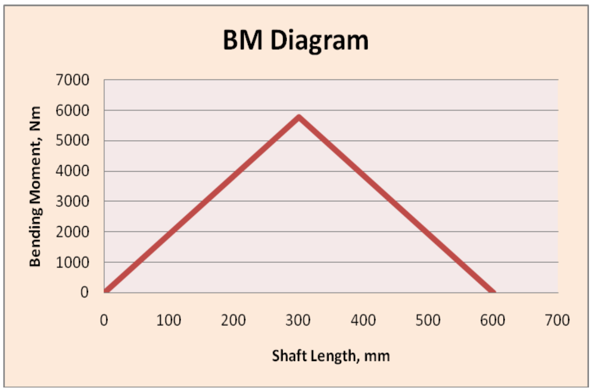


Fig.2 Shear force and bending moment diagram

Babbit Bearing Length Calculations

$$p_\psi = p_{max} . \cos^{1.5} \psi$$

$$F = p_\psi . rd\psi . \cos\psi$$

$$F_{max} = \int_{+60}^{-60} r.p_{max} \cos^{2.5} \psi . d\psi$$

Applying Simpson 1/3 rule and considering P_{max} = Allowable bearing pressure, S_{max}

hc yo	**yc**	**fn = cos2.5 y**	
0.175	60	1.050	0.17464 f0

0.175	50	0.875	0.32896	f1
0.175	40	0.700	0.51160	f2
0.175	30	0.525	0.69654	f3
0.175	20	0.350	0.85525	f4
0.175	10	0.175	0.96225	f5
0.175	0	0.000	1.00000	f6
0.175	-10	-0.175	0.96225	f7
0.175	-20	-0.350	0.85525	f8
0.175	-30	-0.525	0.69654	f9
0.175	-40	-0.700	0.51160	f10
0.175	-50	-0.875	0.32896	f11
0.175	-60	-1.050	0.17464	f12

G = 1/3 h[f0 + 4f1 + 2f2 +4f3 + ….. + 2f10 + 4f11 + f12] = 1.166

For Lade Base Bearing Material, Bearing Load. F_{max} = r. S_{max}.G = 8743 N/mm

Bearing radius, r = 1000 mm

Allowable bearing pressure, smax = 7.5 MPa or N/mm^2, For Lead Bas

Bearing length, L = 170 mm

Bearing reaction load, Rw ~ 42/2 = 21 MT

Bearing angle, y1, y2 = + 60^o to - 60^o

Allowable reaction load, Ra = (Fmax l)/2 =70 MT

Factor of safety (for l = 160 mm) = Ra/Rw = 3.3

Conclusions

An Autogenous Milling has been designed based on the customer need and presented in this paper. During the design process the two main character of an Autogenous Mill e.g. size reduction and the grinding parameters has been specifically included in design of the Autogenous mill. Design process was carried out following Simpson's 1/3rd rule and satisfactorily results were obtained.

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