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AN EXPERIMENTAL ANALYSIS OF VIBRATION OF A BALL MILL WITH VARIABLE LOAD

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ABSTRACT
This paper deals with the experimental investigation of vibrations of a ball mill under variable load. The common causes of vibrations of the ball mill are described in short. The exciting frequencies at which the possible causes of vibration can be recognised are estimated for all principle parts of the ball mill. After that the measurement of accelerations of vibration at housings of bearings has been performed. Due to the production requirements the capacity of the ball mill is variable and therefore the torque on the ball mill drive is also variable. Measurement has been done for two different capacities. Vibration signatures have shown different vibration levels at different loads. It has been noticed that vibration amplitudes at the same frequencies decrease with increase of load of the mill.
Keywords: ball mill, exciting frequencies, vibration signatures, time and frequency domain.

1. INTRODUCTION
A ball mill investigated in this study is used for grinding of limestone in a cement plant. It consists of a large cylindrical shell which ends with inlet- and outlet tubes that are supported in journal bearings. The drive of the ball mill consists of an induction electric motor, two-stage gearbox and the pinion gear. This pinion gear is mated with the girth gear mounted on circumference of the ball mill shell and in that way it transfers the motion from the drive to the shell, see Figure 1, taken from [1].

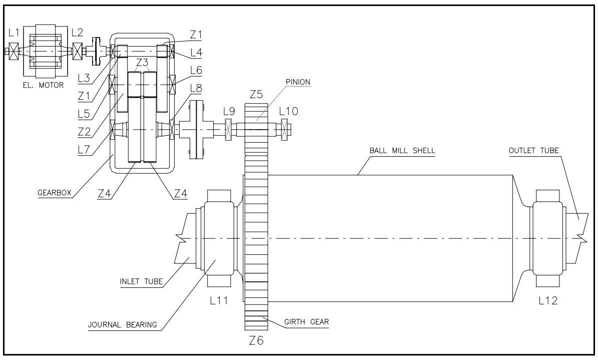

Figure 1. Mechanical system of a ball mill with drive

The ball mill shell is filled up with a certain amount of the steel balls of various sizes. Together with material which is being ground they are called grinding media. By the rotation of a shell a kidney-shaped cross section of grinding media is formed at the rising side of the shell. Particles of grinding media that reach the top of a "kidney" fall down creating cataracts or cascades depending on the speed of rotation. At lower speeds cascading motion prevails. This motion produces mechanical vibration of lesser intensity. With the increase of the ball mill speed the balls begin to cataract. This increases the number of impacts of balls and through it the intensity of vibration of the ball mill.

2. LOAD OF THE BALL MILL

Grinding bodies, new material which is fed into the shell and material which goes back from separator into the shell (circulating charge) represent the load of a ball mill. In Fig. 2a is shown a longitudinal section of the grinding media inside of the ball mill shell. It is obvious that the cross section area of this load varies with the distance L, from the ball mill inlet tube. Besides this the mass of the newly fed and circulating charge can vary due to technological demands and operating factors. Thus, the load is a function of the location inside of the shell and of the time. For practical investigation one can assume that the cross section area of the charge is constant for a given capacity of the ball mill.

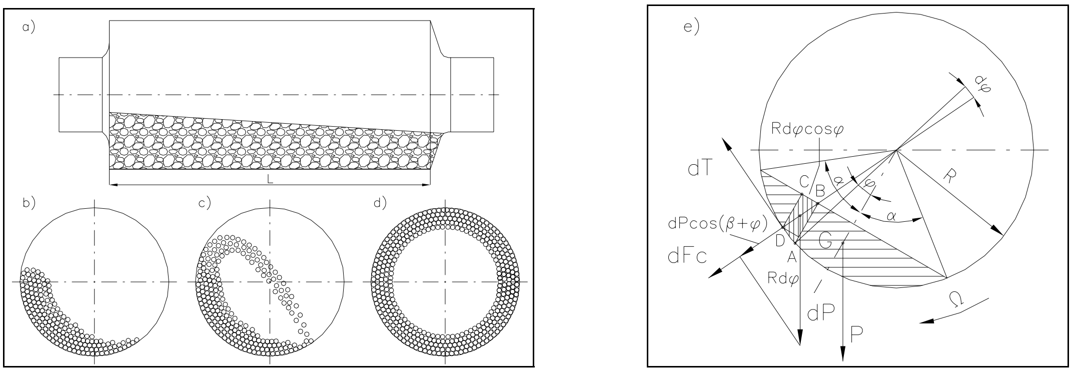

Figure 2: a) A shape of the charge inside the ball mill, b) cascading motion, c) cataracting motion, d)motion of charge at critical speed, e) forces acting on the ball mill shell

With this assumption the cross section area becomes only a function of a central angle 2α, as given in Eq. (3). Figure 2e shows the forces acting on the charge and its elementary sector ABCD, with surface dA. The following elementary forces act on that element: friction force dT, gravity force, dP=ρgLdA and the centrifugal force $dF_C=R\Omega^2dP/g$. From the balance of forces the force dT is determined, [2] as:

$$dT = f \cdot dP\left[\frac{R\cdot\Omega^2}{g} + \cos(\beta+\varphi)\right] = f\cdot\rho\cdot L\cdot g\cdot dA\left[\frac{R\cdot\Omega^2}{g} + \cos(\beta+\varphi)\right] \tag{1}$$

From the Fig. 2e the elementary surface dA is: $dA = R^2\cos\varphi(\cos\varphi - \cos\alpha)d\varphi$, and the moment of friction force acting between the charge and the shell wall, is determined by the integral

$$M_T = \int_{-\alpha}^{+\alpha} RdT = R^3\cdot f\cdot\rho\cdot L\cdot g\cdot\int_{-\alpha}^{+\alpha}\left[\frac{R\cdot\Omega^2}{g} + \cos(\beta+\varphi)\right]\cdot\left(\cos^2\varphi - \cos\alpha\cos\varphi\right)d\varphi \tag{2}$$

After integration we have the formula for the moment in the next form:

$$M_T = R^3\cdot f\cdot\rho\cdot L\cdot g\cdot\left[\frac{R\cdot\Omega^2}{g} + (\alpha - \frac{\sin 2\alpha}{2}) + \cos\beta\left(\frac{4}{3}\sin\alpha - \frac{1}{6}\sin 2\alpha\cos\alpha - \alpha\cos\alpha\right)\right] \tag{3}$$

The obtained formula is satisfactory for determination of moment for the case of prevaling cascading motion of the charge. For cataracting motion a certain corrections should be needed, as done in [3].

3. CAUSES OF VIBRATION OF A BALL MILL AND EXPECTED APPEARANCE OF VIBRATION SPECTRA

The most common faults that appear in a ball mill are imbalance, misalignment, eccentricity of rotating parts, increased clearances in bearings, gears, keys and the keyways in shafts, damages and faults in bearings and gears such as pitch error, meshing errors, wear and cracks, looseness of machine elements, deformations such as bent shafts in the gearbox or a bent shell of a ball mill. Mechanical defect in electric motor such as stator and rotor eccentricity, or broken rotor bars are the next source of vibrations, etc. All these faults cause vibrations at specific forcing frequencies that can be found in the vibration spectra as well as their harmonics and different sidebands. Specific forcing frequencies of all rolling element bearings installed in the ball mill assembly are given in Table 1, [1] .They can be calculated according to formulae given in many literature, for example in [4].

Table 1: Specific forcing frequencies of rolling element bearings of the ball mill drive

Location:	L3, L4	L5, L6	L7, L8	L9, L10
Rolling element bearing:	NU2338 EC	NU2356 MA	22264 CC/W	23168 CCK/W
RPS-rev/sec of inner race	16.53 Hz	3.50 Hz	1.30 Hz	1.30 Hz
FTF-fundamental train freq.	6.61 Hz	1.47 Hz	0.55 Hz	0.57 Hz
BSF-ball spin frequency	39.68 Hz	10.80 Hz	4.13 Hz	5.02 Hz
BPFI-ball pass freq. inner race	128.96 Hz	32.45 Hz	13.43 Hz	15.31 Hz
BPFO-ball pass freq. outer race	85.97 Hz	23.59 Hz	9.91 Hz	11.92 Hz

The most important forcing frequencies of gears are running speeds of pinion and gear, F_{PN} and F_{GE}, and the gear mesh frequency F_{GM}. Other forcing frequencies of gears can be found in literature, [5].

Table 2: Ball mill drive gearbox specific forcing frequencies

Gear pair:	$T_P=Z_1=25$, $T_G=Z_2=118$	$Z_3=47$, $Z_4=127$	$Z_5=39$, $Z_6=180$
F_{PN}	16.53 Hz	3.5 Hz	1.29 Hz
F_{GE}	3.5 Hz	1.29 Hz	0.28 Hz
$F_{GM}=F_{PN}\cdot T_P=F_{GE}\cdot T_G$	413.25 Hz	164.5 Hz	50.31 Hz

The specific forcing frequencies of vibrations in an electric motor are: line frequency F_L=50Hz, and rotor bar passing frequency $F_{RB}=N_{RB}\cdot RPM$=2083.2 Hz. Here N_{RB}=126 is number of rotor bars.

4. A BALL MILL VIBRATION MEASUREMENT AND INTERPRETATION OF RESULTS

Vibration measurement has been done by using a piezo electric accelerometer SA620 with sensitivity 100mV/g (Metrix, USA), an amplifier type Spider 8 with software for signal acquisition Catman 5.0 Professional, HBM, Germany. Measuring points were at locations marked with L_i (i=1,2,.,12), Fig. 1. Measurement has been done for two capacities of ball mill, Q_1=72t/h, Q_2=30t/h, with constant speed. A part of the results of vibration acceleration measurement is given in the following text.

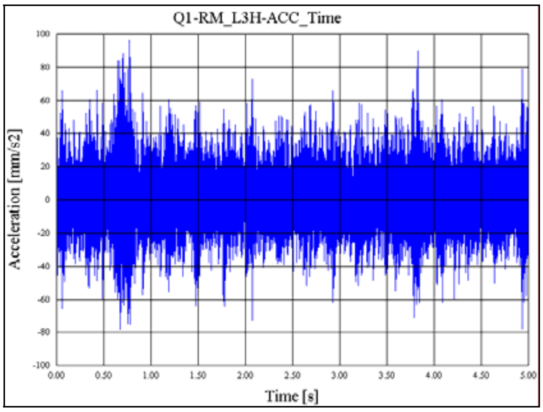

Figure 3. Q1-Waveform of acc. of vibration at L3H

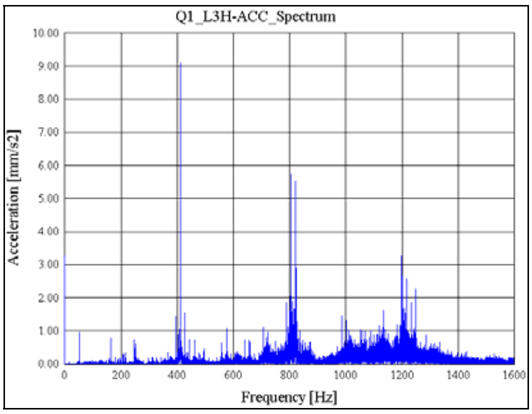

Figure 4. Q1-Spectrum of acc. of vibration at L3H

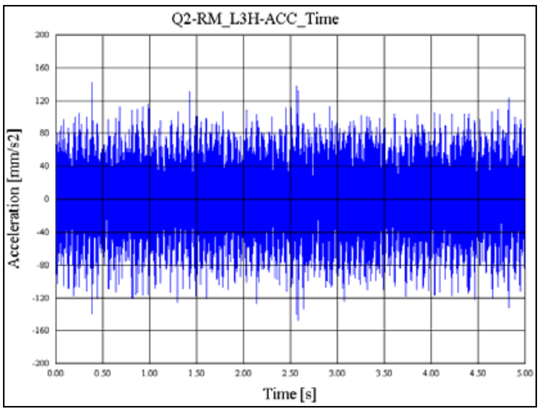

Figure 5. Q2-Waveform of acc. of vibration at L3H

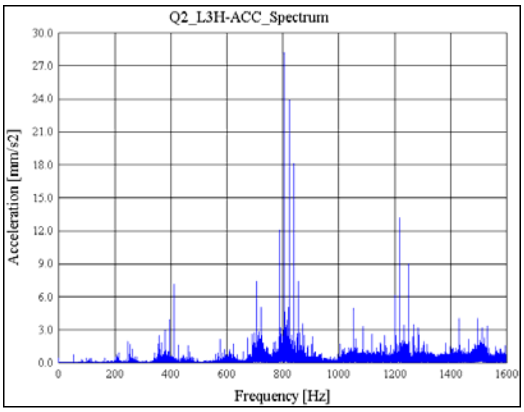

Figure 6. Q2-Spectrum of acc. of vibration at L3H

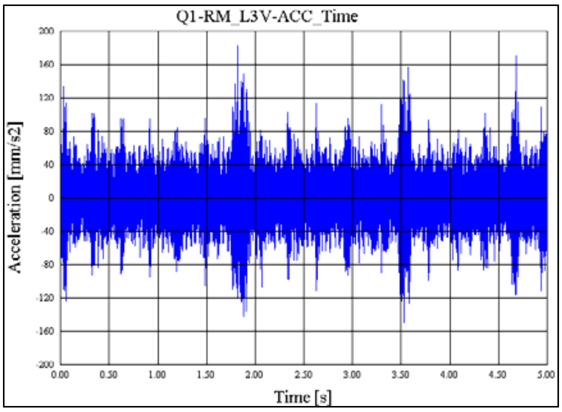

Figure 7. Q1-Waveform of acc. of vibration at L3V

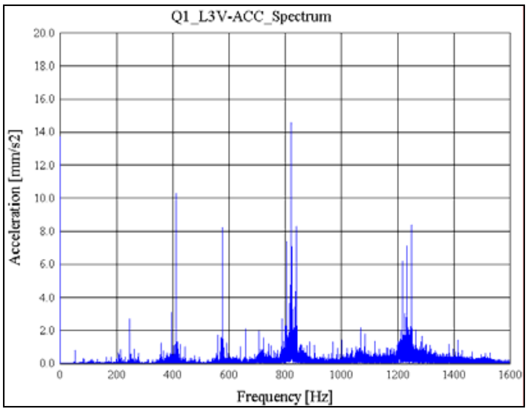

Figure 8. Q1-Spectrum of acc. of vibration at L3V

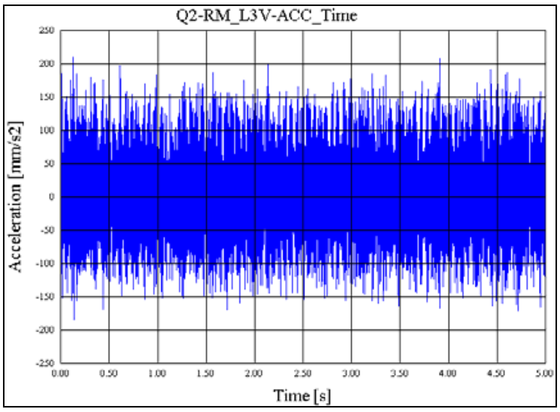

Figure 9. Q2-Waveform of acc. of vibration at L3V

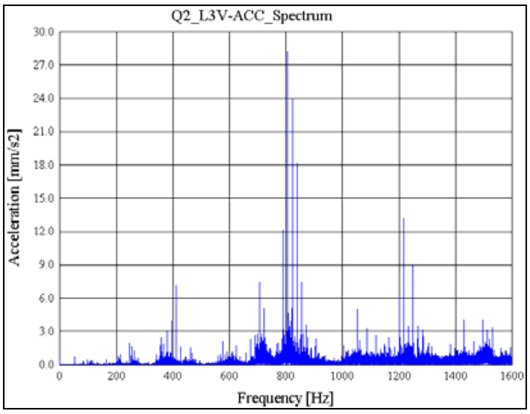

Figure 10. Q2-Spectrum of acc. of vibration at L3V

Time waveforms of vibration acceleration measured at ball mill capacity Q_2=30 t/h show higher overall readings and their peaks indicate the problems with gears which are better seen in the given frequency spectra. The prominent peak at frequency F_{GM}=410.3 Hz which is the gear mesh frequency of the gear pair Z_1/Z_2 (see Table 2), and its harmonics are present in all obtained spectra. The 2nd harmonic of this frequency at 820.6 Hz is dominant in spectra in Fig. 6,8 and 10 and it indicates the misalignment of gear pair Z_1/Z_2. The sidebands spaced around this peak at running speed frequency of gear Z_1, F_{PN}=16.53 Hz, indicate that the main problem is with gear Z_1 which most probably is wear.

5. CONCLUSION

An important conclusion of this study is that the amplitudes of accelerations of vibration at the same frequencies decrease with increase of load showing that there exist correlation between the load and the intensity of vibration of the ball mill components. The causes of high vibration at gear mesh frequency of gear pair Z_1/Z_2 are wear, backlash and misalignment of that gear pair. This fault has been recognized much easier in the spectra of vibration measured at the lower load of the ball mil.

6. REFERENCES

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