Modal and Harmonic analysis of a small-scale ball mill based on ANSYS

Justin Byiringiro, James K. Kimotho and Hiram M. Ndiritu

Abstract—Ball mill is a grinding machine used in mineral processing, comprising of a rotating cylindrical shell filled with steel balls and the material to be ground. The ball mill is inevitably accompanied by vibration during operation. This vibration may lead to resonance in the structural components if the natural frequency matches the operating frequency. Therefore, it is important to study the dynamic characteristics of the ball mill. Modal analysis is an essential technique which helps in predicting the possibility of resonance by determining the natural frequencies as well as mode shapes. In this study, modal analysis of a small-scale ball mill developed in Jkuat is carried out in ANSYS Workbench 16.0. The first ten modes of vibration are extracted for two scenarios, one, for the ball mill and frame support (drum, shaft and frame support) and two, for the drum and shaft. Harmonic analysis is done to visualize the response of the ball mill structure under dynamic loading. The results show that some frequencies fall within the expected operating frequencies.

Keywords—Ball mill, Finite Element Method, Modal Analysis, Resonance, Harmonic response.

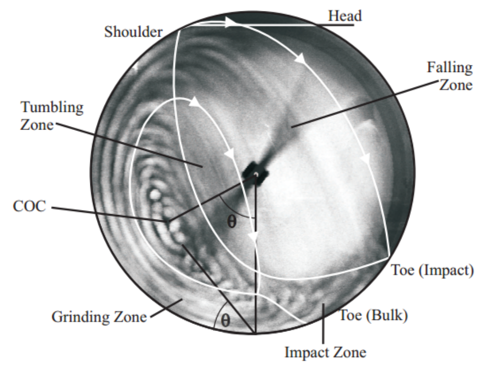

Fig. 1. A typical ball mill motion [2].

I. INTRODUCTION

BALL mill is a machine used in grinding minerals. It is a rotating cylindrical shell filled with steel balls and material to be reduced. Due to the rotation of the drum , the charge is lifted toward the shoulder, the point at which the charge material separates from the mill shell. From the shoulder, the charge falls towards the toe, the zone of intersection where the tumbling charge impacts the material below as demonstrated in Fig. 1. As the charge transits from the toe back to the shoulder, it is subjected to grinding forces. Throughout the cycle, the charge is subjected to both impact and grinding forces. The tumbling of steel balls and the material to be ground are the main source of vibration in a ball mill [1]. This vibration may cause operational problems in ball mill structure even leading to resonance if not well modeled.

Resonance is the tendency of a mechanical structure to absorb more energy when the frequency of its oscillations matches the system's natural frequency of vibration than it does at other frequencies, resonance may cause severe vibration even leading to failure.

Modal analysis is a technique to study the dynamic characteristics of a structure under vibrational excitation [3]. Natural frequencies, modes shapes and mode vectors of a machine can be determined using modal analysis. Modal analysis allows the design to avoid resonance and gives an idea of how the design will respond to different types of dynamic loads. Ball mill is an example of a machine whose dynamic characteristics can be better studied by modal analysis.

Modal analysis since 1960s, has been widely used as an important method in fault diagnosis [4]. A lot of researches on the modal analysis of the ball mill have been conducted. Chen et al. [4] have used ANSYS workbench, to carry out modal analysis of an oversize ball mill tube where first ten natural frequencies were calculated, it was found that resonance could not happen since the lowest natural frequency of ball mill tube was much higher than the operating frequency, also from vibration modes it was found that crack could happen in the tube much easier than in other area since all vibrations modes was in radial direction and the maximum amplitude occurred in the middle of the tube. However this study considered only the ball mill tube. In [5] Porto B.Thiago, Mendonca Q.Beatriz and Carvalho S.G.Lucas used Finite Element Method to establish the basic design requirements for structures subjected to dynamic action, Ball mill and dryer were discussed as case studies. The analysis of the natural frequency of the base of equipments was performed and it was found that there were no risk of resonance since these frequencies was outside the limits of the operating frequency. However the finite element method in this study considered only the base of equipments. Quan [7] explored the tumbling mill resonance using six different kinds of mill FE models. Modal analysis was performed to investigate modes shapes and natural frequencies at different mill rotation speeds in order to investigate resonance behavior. However in this study the simulation model used, only includes the mill tube while the other parts of the mill system was not

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taken into consideration.
Radziszewski [8] investigated the resonance of ball mill by studying the natural frequency. The author study the ball mill as a rotating machinery. Unlike stationary machines, in rotating machineries resonance is related to the rotation speed, because their natural frequencies vary along the operating speed. Fig. 2, shows that the resonant behavior related to mill rotation speed.

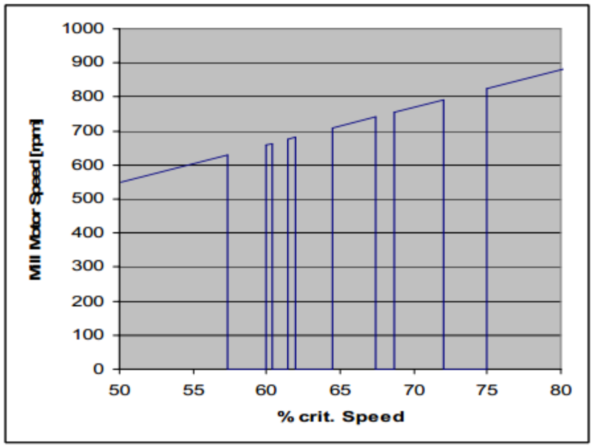

Fig. 2. Mill resonance behavior related with mill rotation speed [8].

The objective of this study is to determine natural frequencies, study the mode shapes of the ball mill and subject to a harmonic loading in order to predict the dynamic characteristics of a ball mill and to avoid possible resonance. Modal analysis and harmonic response are carried out in Finite Element Analysis ANSYS workbench 16.0. Natural frequencies are extracted for two conditions; first for ball mill drum with the frame support and for drum with shaft. The geometric modeling of ball mill was done in SOLID WORKS.

II. Finite Element Modeling

A. Theory of Modal and Harmonic response analysis

Finite element calculation mode is structural dynamics of the eigenvalue. Eigenvalues and eigenvectors is natural frequencies and mode shapes modal analysis. The dynamic equation of motion [9], is shown in Equation 1.

$$[m]\{\ddot{u}\} + [c]\{\dot{u}\} + [K]\{u\} = \{F(t)\} \tag{1}$$

where; $[m]$ is mass Matrix, $[c]$ is damping matrix, $[K]$ is stiffness matrix, $\{\dot{u}\}$ is nodal velocity, $\{\ddot{u}\}$ is nodal acceleration, $\{u\}$ is nodal displacement, $\{F(t)\}$ is excitation Force.

The eigenvalue problem is solved with undamped modal analysis as shown in Equation 2.

$$[m]\{\ddot{u}\} + [K]\{u\} = \{0\}. \tag{2}$$

The free vibration mode of the structure is harmonic vibration, so the displacement is a sine function.

$$X = A\sin(\omega t). \tag{3}$$

Combining Equation 3 and 2 result in:

$$([K] - \omega^2[M]) = 0. \tag{4}$$

Equation 4 represents an eigenvalue problem , where ω_i^2 is eigenvalues and ω_i is natural circular frequency.

Modal analysis is actually solving eigenvalue and eigenvector.

III. Modal and Harmonic analysis

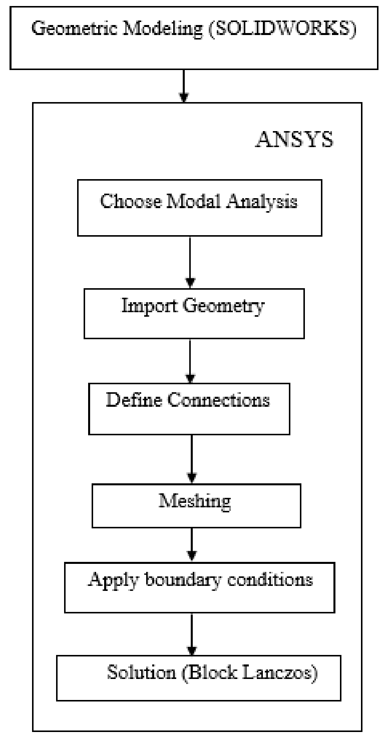

Fig. 3. Flow chart of procedure used in modal analysis

A simplified 3D model of ball mill developed in jkuat is modeled in SOLIDWORKS then imported to ANSYS workbench 16.0 using Initial Graphic Exchange Specification (IGES) file extension. Procedure used in modal analysis is illustrated in Fig. 3.
Modal analysis is done for two conditions;

1) Ball mill and frame support
2) Drum and shaft, considering the rotation velocity.

For condition 1 modal analysis of ball mill and frame support as shown in Fig. 4, is done for predicting the dynamic characteristic of the ball mill tube together with the frame support. For condition 2 the rotor dynamic analysis (modal analysis) of only the drum and the shaft as shown in Fig. 6, is performed in order to check whether the drum and the shaft are safe from resonance caused by the whirling of the shaft.
In operation, the ball mill is subjected to two major excitation frequencies, that is, the excitation frequency from the comminution process and the rotation frequency of the drum. For now, only the rotation frequency has been considered in

analysis as the modeling of the internal process of the ball mill to get the excitation frequency from the comminution process is still continuing.

A. Condition 1: Modal analysis of ball mill with frame support

Modal analysis of ball mill structure is done to obtain the first ten natural frequency and vibration mode. There are 182725 nodes and 140385 elements generated in total in the model of the structure. A simplified model used in modal analysis is shown in Fig. 4.

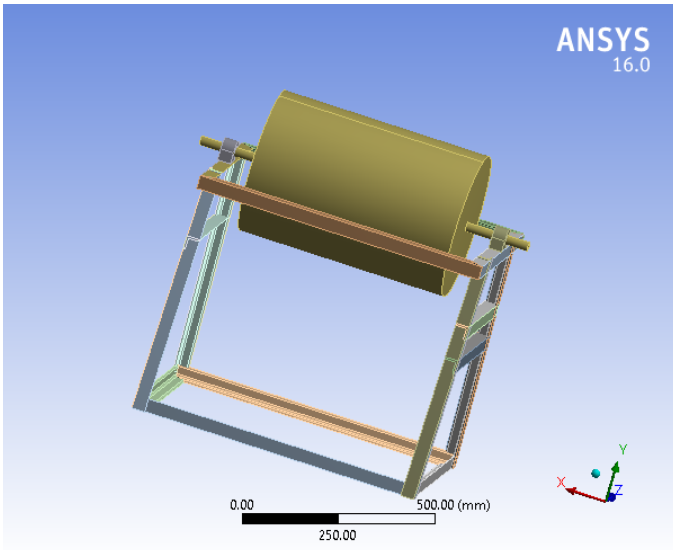

Fig. 4. A simplified model of Ball mill from jkuat

1) Mesh Design: The model is divided into 182725 nodes and 140385 Elements. The mesh used and quality evaluation parameters are shown in Table I.

TABLE I
MESH QUALITY EVALUATION PARAMETERS

No	Parameters	Averages Value Used	Evaluation Standard and Value Range
1	Aspect Ratio	1.6522	Optimum value is 1 Warning value is 20
2	Element Quality	0.85032	ranges from 0 to 1 Optimum value is 1
3	Jacobian Ratio	1.0532	Optimum value is 1 Warning value is 40
4	Wraping Factor	2.57E-03	Optimum value is 0 limit is 7
5	Parallel Deviation	5.7603	Optimum value is 0 Warning value is 70
6	Maximum Corner Angle	96.835	Optimum value is 90 Warning value is 155
7	skewness	0.19596	ranges from 0 to 1 Optimum value is 0
8	Orthogonal Quality	0.89292	ranges from 0 to 1 Optimum value is 1

The element quality and skewness are the most important parameters to evaluate the mesh quality. From Table I the element quality value is 0.85032 and skewness is 0.19596. The mesh quality is excellent when the skewness value is less than 0.25 [10]. Therefore, the mesh quality are good for the model calculation.

2) Boundary conditions: While carrying out modal analysis of the ball mill, it is only needed to set constraints without considering the force. For the boundary condition in the modal analysis, the base of the frame support is fixed. Fig. 5, shows the boundary conditions applied.

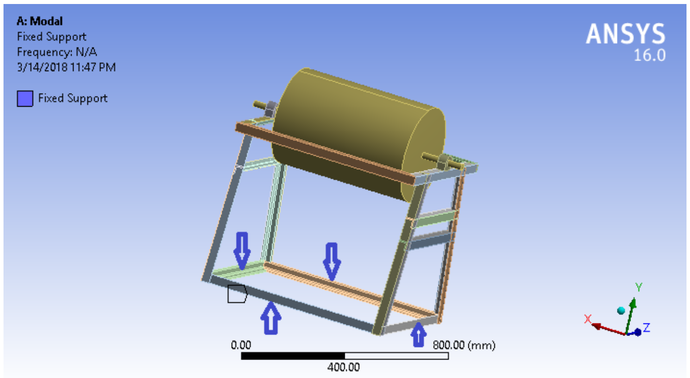

Fig. 5. Boundary conditions applied

3) Solution scheme: According to the vibration theory, the lower order modes have a huge impact on vibration [10]. Therefore the first 10 natural frequencies and vibration modes of the ball mill are solved by Block Lanczos method. The advantage of this method is that the mesh quality is low and the running speed is fast [11]. Settings used in ANSYS are shown in Table II and Table III.

TABLE II
ANSYS SETTINGS USED

object name	**Modal (A5)**
State	Solved
Physics Type	Structure
Analysis Type	Modal
Solver Target	Mechanical APDL
Environment Temperature	22 oC
Generate Input Only	No

TABLE III

Object Name	Pre-stress (None)
State	Fully Defined
Pre-Stress Environment	None

B. Condition2: Ball mill drum and shaft

For this analysis, only the drum and shaft are analyzed as shown in Fig. 6. The drum is simplified to simply supported constrained beam and restricted in the ad-axial surfaces which lie in the shaft at the ends of the roller. It was assumed that the total mass of the drum is a point mass acting at the center of the drum which is 200kg (mass of the drum with the grinding medium).

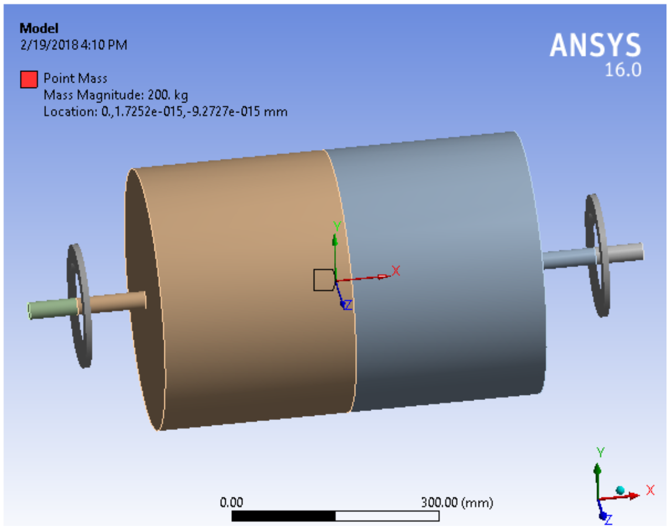

Fig. 6. Model used for rotor Dynamic analysis

Harmonic response analysis is used to predict the sustained dynamic behavior of the structure under predetermined excitation, verifying whether or not structure will successfully overcome harmful effects of forced vibration. In harmonic response analysis the constraint is as same as modal analysis since they share some informations as shown in Fig. 7. The frequency range is set to 0-200Hz. In a harmonic analysis, the peak response will correspond with the natural frequencies of the structure. The analysis settings used for harmonic response are shown in Table IV.

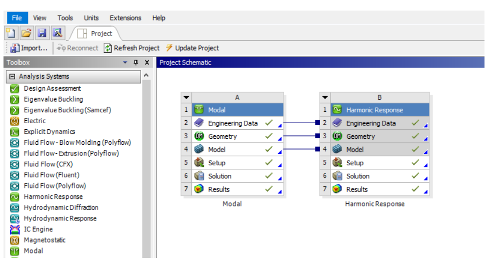

Fig. 7. Setting harmonic response

TABLE IV
ANALYSIS SETTING FOR HARMONIC RESPONSE

State	Fully Defined
Range Nininum	0.Hz
Range Maximum	200.Hz
Solution Intervals	80
Solution Method	Mode Superposition
Cluster Results	No
Modal Frequency Range	Program Controlled

IV. RESULTS AND DISCUSSIONS

Modal analysis determines the vibration characteristics of the structure, by determining the mode shapes and natural frequencies. First ten natural frequency results are obtained between 0-150 Hz range. Six of the ten mode shapes calculated are illustrated in Fig. 9, Fig. 10, Fig. 12, Fig. 13 and Fig. 14. Mode shapes frequencies and characteristic are tabulated in Table V.

A. *Modal analysis results for condition 1*

TABLE V
FREQUENCIES AND CORRESPONDING VIBRATION MODES

Mode	Frequency (Hz)	Type of mode
1	3.6	Bending along X axis
2	3.6562	Bending along X axis
3	9.152	Bending along Z axis
4	19.655	Bending along Y axis
5	19.788	Twisting along X axis
6	39.576	Bending along Y axis
7	70.911	Bending along X axis
8	121.95	Bending along X axis
9	128.5	Bending along Z axis
10	146.45	Bending along Y axis

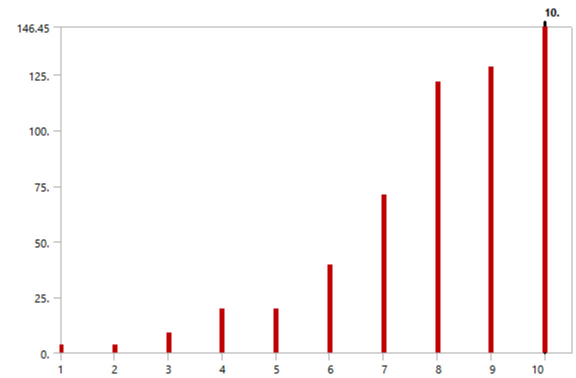

Fig. 8. Variation of number of modes vs frequency. X-axis contains number of modes and Y-axis contains frequency

B. *Mode shapes*

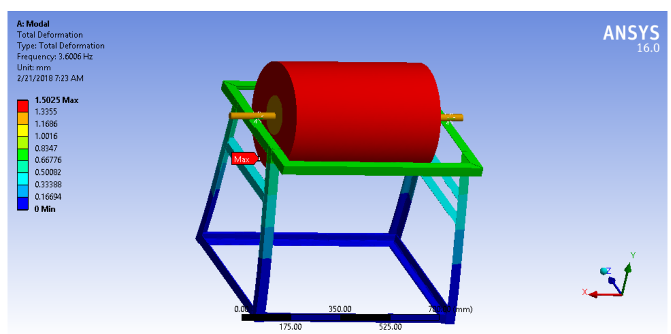

Fig. 9. The 1st mode

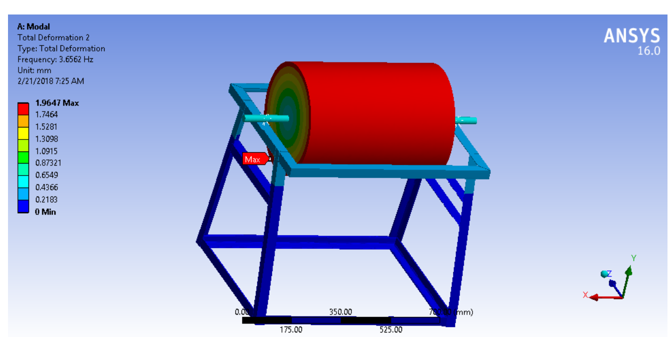

Fig. 10. The 3rd mode

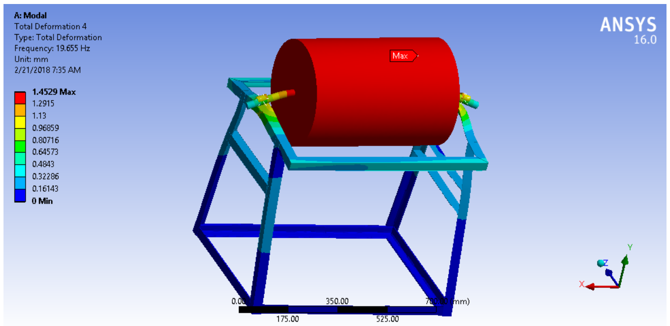

Fig. 11. The 4th mode

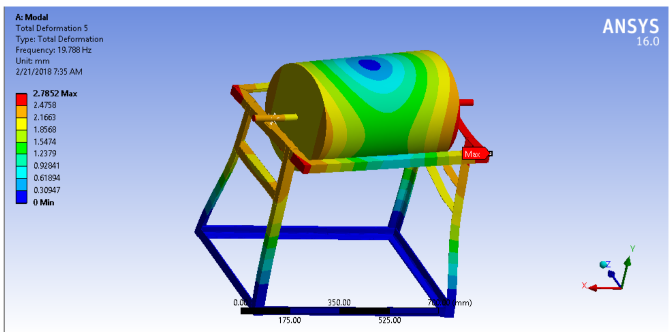

Fig. 12. The 5th mode

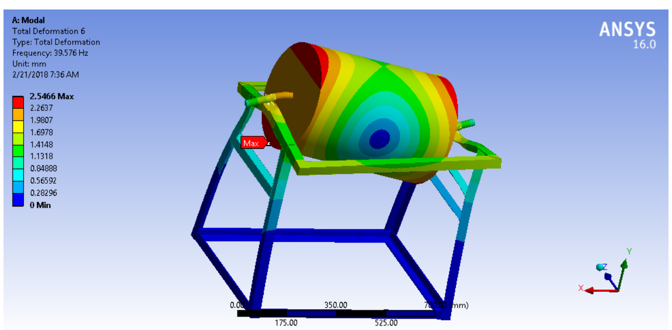

Fig. 13. The 6th mode

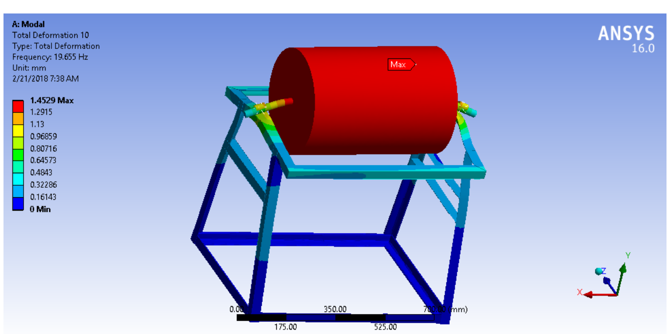

Fig. 14. The 10th

As shown in Table V, the first and second order natural frequencies are close and their vibration mode is bending along X axis, with maximum deformation located on the drum. The vibration mode of the third order natural frequency is bending along Z axis with maximum deformation on part of frame support and part of drum. The fourth and fifth order natural frequencies are also close but their modes of vibration are different, the vibration of the fourth is bending along Y axis with the maximum deformation on the drum, vibration mode of the fifth is twisting along X axis with the maximum vibration on the upper part of the frame support. The mode of vibration of the sixth order natural frequency is bending along Y axis with maximum deformation on the drum. The seventh and eighth order natural frequencies are the same as the first and second. The ninth vibration mode is the same as the third. The tenth vibration mode is the same as the fourth.

From modal analysis shows that the first natural frequency is higher than the operating frequency of the ball mill, as said above only the rotation frequency is considered. The first natural frequency which is responsible for the resonance vibrations is equal to 3.6006Hz and is not close to the rotation frequency of the drum. The operating speed of ball mill is 50rpm, that is 0.83Hz. However for the fourth, sixth and tenth order vibration modes as shown in Figs (11 , 13, 14), it obvious that the whirling of shaft due to shaft deflection caused by loading of the drum, introduces some vibration which could have a high amplitude.

C. Modal analysis results for condition 2

Damped frequency (Hz), stability (HZ), modal damping ratio and logarithmic decrement are illustrated in Fig. 15.

Set	Solve Point	Mode	Damped Frequency [Hz]	Stability [Hz]	Modal Damping Ratio	Logarithmic Decrement
1.	1.	1.	9.834e-004	3.2577e-011	-3.3127e-008	2.0814e-007
2.		2.	6.2216e-003	-2.773e-009	4.4571e-007	-2.8005e-006
3.		3.	20.247	-0.37175	1.8358e-002	-0.11537
4.		4.	20.34	-0.37701	1.8532e-002	-0.11646
5.		5.	86.108	-6.9027	7.9907e-002	-0.50368
6.		6.	86.451	-6.6066	7.6197e-002	-0.48016
7.		7.	0.	-200.14	1.	N/A
8.		8.		-204.07		
9.		9.	302.63	-29.084	9.5663e-002	-0.60384
10.		10.	306.49	-29.822	9.6844e-002	-0.61136
11.	2.	1.	9.834e-004	3.1092e-010	-3.1617e-007	1.9865e-006
12.		2.	6.2216e-003	-2.7317e-009	4.3906e-007	-2.7587e-006
13.		3.	20.246	-0.37169	1.8356e-002	-0.11535
14.		4.	20.341	-0.37707	1.8534e-002	-0.11648
15.		5.	74.044	-5.7744	7.775e-002	-0.49
16.		6.	100.54	-7.704	7.6405e-002	-0.48147
17.		7.	0.	-200.21	1.	N/A
18.		8.		-204.06		
19.		9.	302.63	-29.084	9.5663e-002	-0.60384
20.		10.	306.49	-29.822	9.6844e-002	-0.61137

Fig. 15. Damped frequency (Hz), stability (Hz), modal damping ratio and logarithmic decrement

Fig. 16, shows the frequency at each calculated mode

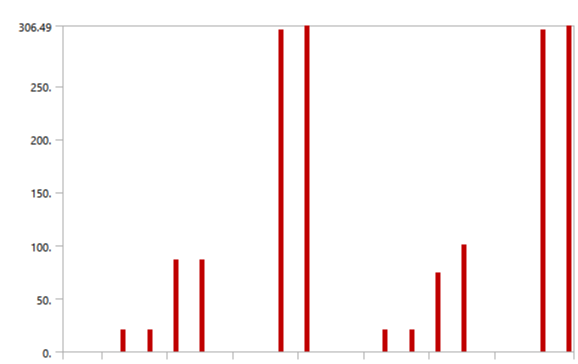

Fig. 16. Frequency at each calculated mode

D. Modal shapes

The first ten order modes extracted using Block Lanczos method, of the ten modes calculated five modes are illustrated in Fig. 17, Fig. 18, Fig. 19, Fig. 20 and Fig. 21.

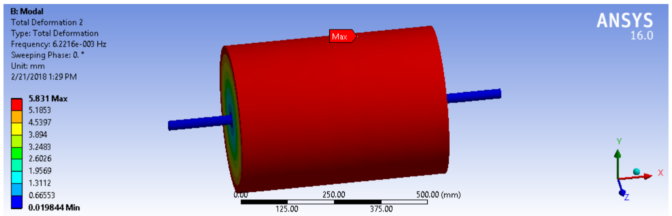

Fig. 17. The 2nd mode

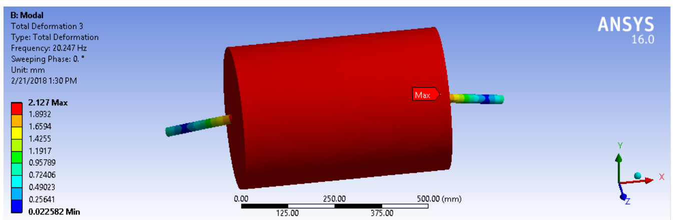

Fig. 18. The 3rd mode

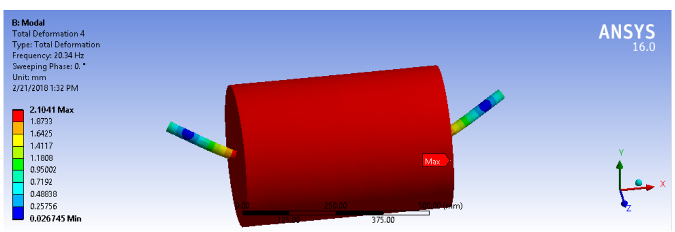

Fig. 19. The 4th mode

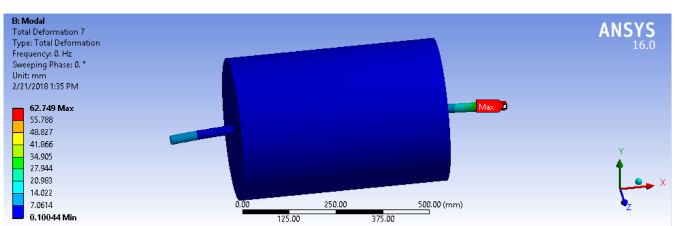

Fig. 20. The 7th mode

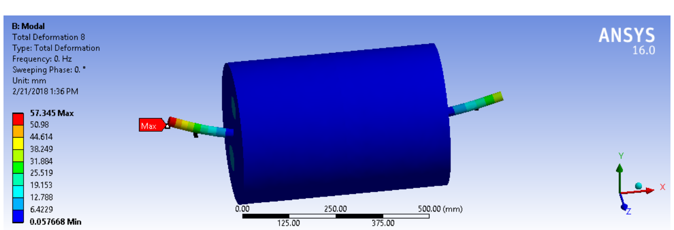

Fig. 21. The 8th mode

E. Campbell diagram

The Campbell diagram is one of the most important tools for understanding the dynamic behavior of the rotating machines. The Campbell diagram used in this research consists of a plot of the natural frequencies as a functions of the rotation velocity.

Fig. 22, shows the Campbell diagram of natural frequencies (Hz) as a function of rotation velocity (rad/s).

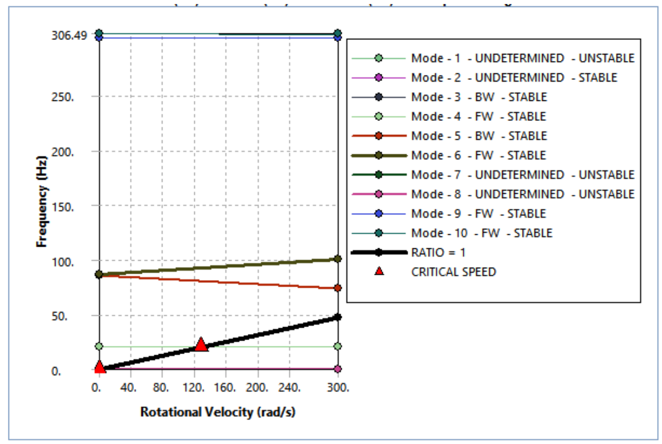

Fig. 22. Campbell diagram (a)

Mode	Whirl Direction	Mode Stability	Critical Speed	0. rad/s	300. rad/s
1.	UNDETERMINED	UNSTABLE	NONE	9.834e-004 Hz	9.834e-004 Hz
2.	UNDETERMINED	STABLE	3.9092e-002 rad/s	6.2216e-003 Hz	6.2216e-003 Hz
3.	BW	STABLE	127.21 rad/s	20.247 Hz	20.246 Hz
4.	FW	STABLE	127.8 rad/s	20.34 Hz	20.341 Hz
5.	BW	STABLE	NONE	86.108 Hz	74.044 Hz
6.	FW	STABLE	NONE	86.451 Hz	100.54 Hz
7.	UNDETERMINED	UNSTABLE	NONE	0. Hz	0. Hz
8.	UNDETERMINED	UNSTABLE	NONE	0. Hz	0. Hz
9.	FW	STABLE	NONE	302.63 Hz	302.63 Hz
10.	FW	STABLE	NONE	306.49 Hz	306.49 Hz

Fig. 23. Campbell diagram (b)

The highest deflections are 62.749mm and 57.345mm on seventh and eight modes respectively as shown in Fig. 20, and Fig. 21.

Critical speeds are 0.039092 rad/s , 127.21 rad/s and 127.8 rad/s as shown in Fig. 23. The first critical speed is within the operating range of the ball mill.

F. Harmonic response results

Harmonic analysis result is used to verify the steady-state response of a structure, which enable researches and ball mill designers to determine whether the ball mill can withstand resonance or other structure problems related to vibration during its operating life. Harmonic analysis calculates the response of a structure to cyclic loads over a frequency range and plot the response of a structure on amplitude versus frequency graph. Post-processing option in ANSYS is used to plot the amplitude versus frequency graph as shown in Fig. 24.

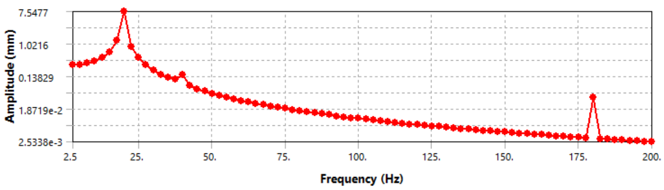

Fig. 24. Variation of displacement amplitude with different exciting frequencies

Fig. 24, shows that the maximum amplitude is generated in 20Hz and the corresponding value of amplitude is 7.5477mm. Resonance occurred at this frequency since it generate the highest amplitude. This is close to the 4th and 5th order predicted natural frequencies. Therefore the 4th and 5th modes shapes are the resonance frequency for ball mill. The drum and the shaft will be the most affected since the highest amplitude of the 4th and 5th natural frequencies are located there. The structure of the ball mill needs to be optimized in order to enhance the stiffness.

V. Conclusion

This study presents modal and harmonic analysis of small-scale ball mill developed in Jkuat. The first ten natural frequency and corresponding vibration mode of two conditions are determined. Campbell diagram has been plotted graphically and critical speeds were obtained. The vibration of number of modes versus frequency has been plotted graphically. The description of mode with the corresponding frequency has been tabulated. Harmonic response of the ball mill for the excitation in the range of 0-200Hz has been studied. Variation of amplitude with respect to frequency has been graphically plotted. From modal the above analysis, the following Conclusions are drawn:

1) The structure of the ball mill need to be optimized especially the ball mill drum and the shaft, in order to avoid not only resonance but also vibrations which can cause higher deflection.
2) In working condition it is advised to avoid the above natural frequencies and critical speeds of the ball mill in order to prevent resonance.
3) The modal analysis can provide reference for the design of ball mill and the selection of reasonable parameters.

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