inter.noise 2012
august 19-22
new york city, usa
Quieting the World's Cities

Development of a validated finite element model of a longwall cutting drum

Hugo E. Camargo[a]
David S. Yantek[b]
Adam K. Smith[c]
National Institute for Occupational Safety and Health
626 Cochrans Mill Road, Pittsburgh, PA 15129

The findings and conclusions in this report are those of the author(s) and do not necessarily represent the views of the National Institute for Occupational Safety and Health (NIOSH). Mention of any company name, product, or software does not constitute endorsement by NIOSH.

One of the most pervasive diseases in the mining industry is noise-induced hearing loss (NIHL). Exposure of miners to noise levels above the permissible exposure limit (PEL) results in hearing loss of approximately 80% of coal miners by retirement age. In this context, the Office of Mine Safety and Health Research (OMSHR) of the National Institute for Occupational Safety and Health (NIOSH) is conducting research to develop engineering noise controls for longwall mining systems, which account for half of the national underground coal production. From field measurements, it was determined that the dominant sound-radiating components at the shearer of a longwall system are the two cutting drums used to remove coal from the face of the mine. Due to the dimensions and complexity of longwall systems, the approach used for this project consists of developing finite element (FE) models of the dominant sound-radiating components. These FE models will be the basis for predicting the radiated sound and the development of engineering noise controls. This paper presents the development of a finite element model of the cutting drum which, in order to be an accurate representation of the actual drum, was validated using data from an experimental modal analysis test.

1 INTRODUCTION

As we enter the 21st century, noise-induced hearing loss continues to be one of the most pervasive diseases in the mining industry. According to a study on the hearing difficulty attributable to employment and occupation conducted by the National Institute for Occupational

[a] email: HCamargo@cdc.gov
[b] email: dcy6@cdc.gov
[c] email: ASmith9@cdc.gov

Safety and Health (NIOSH), out of 40 industry categories, workers in the mining industry displayed the second largest prevalence of hearing difficulty in the United States from 1997 to 2003[1]. This result corroborates with an analysis made of over 20,000 audiograms from 3,449 coal miners, showing that by age 64, approximately 80% of coal miners have from moderate to profound high-frequency hearing loss[2].

Several factors contribute to the noise overexposure of miners, including confined workplace conditions, the utilization of heavy duty equipment, and the proximity of the operators to the mining machines. Furthermore, production and efficiency demands have geared the industry towards the development of more powerful and larger mining equipment and the utilization of continuous mining methods. This is especially true in coal mines, which provide one of the most inexpensive and abundant sources of energy. According to the Energy Information Administration (EIA), coal-generated electricity accounted for 45% of the total nation's demand in 2010[3]. Therefore, coal continues to be the primary fuel firing power plants across the Unites States. The EIA also cites that approximately 50% of the underground coal production in 2010 was extracted using longwall mining systems, while the other 50% was extracted using the room and pillar method that employs continuous mining machines.

In this context, the Office of Mine Safety and Health Research (OMSHR) of NIOSH is conducting a project to develop engineering noise controls for longwall mining systems. Considering the dimensions and complexity of longwall mining systems, the approach used to develop noise controls for these machines consists of developing numerical models to predict the vibratory and acoustic response of the dominant sound-radiating components and to use these numerical models to explore various noise control options. From noise assessments conducted by NIOSH, it was determined that the dominant sound-radiating components on a longwall shearer are the two cutting drums used to remove coal from the face of the mine. Furthermore, from acoustic recordings conducted by the Mine Safety and Health Administration (MSHA) on an operating cutting drum, it was determined that the sound radiated by the drum has a dominant frequency content of up to 1,000 Hz. Therefore, the present study focuses on developing noise controls for these cutting drums.

Previous research was also conducted by the United States Bureau of Mines (USBM) to measure the excitation forces that arise from the interaction of the cutting bits with the coal and are transmitted to the drum through the bit holders. These forces were measured in a laboratory setup using a linear cutting rig and resulted in a flat spectrum up to 150 Hz, which simply represents the frequency of bit impacts. Above this "cut off" frequency, the force magnitude rolls down at a rate proportional to the inverse of the frequency square[4].

This paper presents the tasks performed during the development of a validated finite element model of a cutting drum for acoustic prediction in the 0-1,000 Hz frequency range. These tasks include the generation of the FE model in ANSYS, the conduction of an experimental modal analysis test on an actual drum, and the correlation of the experimental modal data and the modal FEA data.

2 DESCRIPTION OF A LONGWALL SYSTEM

Longwall systems are sets of machines that work in full synchrony to extract ore from underground mines. Although there are two basic types of longwall systems—shearers and ploughs—in the United States approximately 98% of longwall mines use shearers. For this reason, the work presented in this paper focuses on longwall shearer systems. These systems are mainly used in coal and in a few trona mines throughout the United States; therefore, coal and ore will be used interchangeably throughout the paper. As shown in Figure 1, a longwall system is comprised of the following components: a shearer that traverses back and forth along the face ripping coal; an armored face conveyor (AFC) that runs along the face and transports the ripped coal to the stageloader; powered self-advancing longwall shields that provide temporary roof support for the shearer and the AFC; and the stageloader which, after crushing the coal, loads it onto a belt conveyor to be taken out of the mine. The shearer measures from 8 to 12 meters in length, and by virtue of its ranging arms can perform cuts of 2 to 6 meters in height. Each shield measures from 1.5 to 2 meters in width, and therefore on a typical 400-meter-long face there are over 200 shields providing the temporary roof support. Since the AFC runs along the face, typical AFCs can measure 400 meters in length.

The longwall shearer is the main component of a longwall system. It is usually controlled by three operators that move along with it as it traverses the face. Its function is to rip the coal and push it into the AFC. In order to effectively accomplish these two tasks, the shearer is provided with two cutting drums which due to their location with respect to the headgate and tailgate entries are called the headgate drum and the tailgate drum. When the shearer travels from headgate to tailgate, the tailgate drum is the leading drum that performs most of the coal cutting while the headgate drum is mostly in charge of a cleaning operation by pushing coal left by the leading drum into the AFC. When the shearer travels from tailgate to headgate, the drums switch functions with the headgate drum being the leading drum and the tailgate drum performing the cleaning job.

2.1 The Longwall Shearer Cutting Drum

The longwall shearer cutting drum examined in this study consists of a cylindrical body with a 0.987-meter outside diameter, a 1.067-meter height, and a 0.05-meter-thick wall. Inside this cylindrical body, there is a circular mounting plate 0.10-meter-thick having a square opening at the center of the cylinder (refer to Figure 2). The drum is fully made of steel and weighs 4,707 Kg. Around the cylindrical body, four helical vanes are welded, starting in the face ring and winding around the cylindrical body towards the discharge side of the drum. The function of the helical vanes is to push the cut coal into the AFC as the drum rotates. The vanes have a 1.91-meter outside diameter. On the outermost edge of the vanes, there are 28 bit holders that hold the cutting bits at various angles of attack. There are also 12 bit holders on the outermost edge of the face ring and 4 bit holders in the flange of the face ring, making a total of 44 bit holders. Water is carried through conduits inside the vanes to the bit holders, where the water is sprayed through nozzles to reduce the risk of ignition of mine gases and for dust control purposes.

As mentioned earlier, previous noise assessments indicated that the cutting drums are the dominant sound-radiating components on a longwall shearer. These drums are set into vibration by the excitation forces that arise from the interaction of the cutting bits with the coal, and transmitted to the drum through the bit holders. Due to the adverse conditions at the face while the drum is in operation—i.e. as the drum is sumped into the coal—vibration measurements are extremely difficult to conduct on an operating drum. In addition, the presence of explosive gases at the face of coal mines, as well as the lack of instrumentation approved by MSHA for underground use, further restricts the ability to perform any type of vibration and/or force measurements. Therefore, in order to reduce the sound radiated by the cutting drums, a finite element model will be used to explore the effect of structural modifications on the response, i.e. surface vibration, of the drum.

3 FINITE ELEMENT MODEL

A FE model of the cutting drum was developed in ANSYS based on a solid model drawing with a high degree of geometrical accuracy. A small change was made on the cylindrical part of the solid model in order to achieve a more regular mesh. The original solid model had a 0.5-mm step increase in its inner diameter used for alignment purposes of the support plate. This step increase was removed in order to avoid having a much finer mesh around this region. Some details on the surface of the bit holders were also simplified to reduce the element count. The resulting mesh had 238,980 elements. Brick HEX20 and tetrahedral TET10 element types with 20 and 10 nodes respectively were used to mesh the structure, resulting in a 1,349,116 degree of freedom (DOF) model.

The welds between the various components of the drum were modeled using contact type elements. This type of element is defined by 8 nodes with three translational degrees of freedom per node, and has the same geometric characteristics as the solid element face with which it is connected. A minimum of two elements along the thickness of each drum component were used to allow variability along this dimension.

3.1 Modal FEA Results

The FE model was used to conduct a modal analysis using the PCG Lanczos iterative eigensolver. From this analysis the structural modes in the 0-1,000 Hz frequency range were computed. Figure 3 shows the first 12 structural modes. Note that as seen from the discharge side, the drum presents a 2-fold symmetry; that is, the first and the third vanes are symmetric, whereas the second and the fourth vanes—which are slightly different than the first and the third vanes—are also symmetric. As a result, there are various symmetric modes with close natural frequencies. For example, the first and the second modes, the third and the fourth modes, and the seventh and eighth modes are symmetric. These results also show that the discharge side of the drum undergoes larger deformations as compared to the face side of the drum due to the added rigidity provided by the face ring.

4 EXPERIMENTAL MODAL ANALYSIS

An experimental modal analysis (EMA) test was conducted on an actual drum. The objective of this test was to obtain experimental data to validate the FE model. Before the actual EMA test, a pretest analysis was conducted in FEMtools in order to determine the optimal locations for the measurement points. The sensor elimination method was used for the pretest analysis, iteratively eliminating sensors from a set of candidates in order to optimally maintain linear independence between mode shapes. The pretest analysis was based on the modal FEA results and yielded a total of 106 points on the drum surface, shown in Figure 4, which constituted the test model.

4.1 Experimental Setup

The test was conducted in the machine shop of a collaborating mine. For the test, the drum was placed with the face ring down on nine inflatable rubber supports known as AirRides inflated to a pressure of 95 psi as shown in Figure 5. At this pressure, each AirRide has a loading capacity of 770 Kg, a natural frequency of 2.68 Hz, and a spring rate of 2.27×10^5N/m. Two different types of excitations were used: a modal hammer with a maximum input force of 4.448

N, and two electromechanical shakers with a maximum input force of 489 N. Twelve triaxial accelerometers were roved around the structure to measure the response in all three directions at each data point. In order to avoid mass loading effects while roving the accelerometers, a dummy mass with the same mass as the accelerometers was placed on each measurement point. Not all the points on the cylindrical body and the face ring were readily accessible, mainly due to interference of the vanes, and a few points had to be moved by 0.1 to 0.15 meters. Each test was conducted and completed in the same day to avoid temperature variability effects.

4.2 EMA Results

The Frequency Response Function (FRF) for each degree of freedom of each measuring point was obtained. From these data, the mode shape vectors were computed using the AF Polynomial curvefitter in the frequency range of interest, i.e. 0-1,000 Hz. The FRF data were also curvefitted to extract the modal parameters. Although two different datasets were collected—one with the modal hammer and one with the electromechanical shakers—the data obtained with the modal hammer were used for the correlation analysis because the FRFs had better quality than the shaker data. Figure 6 shows typical results of the magnitude plot of the FRFs of points 8, 27, and 63, measured on the cylindrical body, the face ring, and the vanes, respectively, in the direction of maximum displacement. The location of these three points can be found in Figure 4. The FRF shown in Figure 6a is clearly the drive point FRF, since all the resonances are separated by antiresonances.

5 MODEL CORRELATION

The correlation analysis was also conducted using the FEMtools software. The first step in this process consisted of performing the spatial correlation. To this end, both the FE model and the test model were put in the same reference coordinates. Then, the points on the test model were paired with the closest nodes on the FE model using a 0.05-m distance criterion.

For the mode shape correlation, the MAC value was computed for each mode shape pair. The MAC value measures the degree of correlation between two vectors (mode shapes) according to the following equation[5].

$$MAC_{ij} = \frac{\left(\{V_i\}^T\{V_j\}\right)^2}{\left(\{V_i\}^T\{V_i\}\right)\left(\{V_j\}^T\{V_j\}\right)} \tag{1}$$

As seen from Eqn. (1), when two vectors are perfectly correlated (same magnitude and direction) their MAC value will equal unity; however, if the vectors are not perfectly correlated their MAC value will be less than 1.

The results from the computation of the MAC values across the FEA modes and the EMA modes are listed in Table 1. From this table, it can be seen that a high degree of correlation, i.e. above 90%, was achieved for the first 6 structural modes in the frequency range of 0-200 Hz. More generally, a good degree of correlation, i.e. above 70%, was achieved for the first 23 modes in the frequency range of 0-450 Hz, with the exception of modes pairs 10, 11, 14, 15, and 22, whose MAC values were less than 70%. Above 450 Hz, the MAC values ranged between 20% and 60% indicating poor correlation, with the exception of mode pairs 26 and 29, with MAC values of 75.9% and 77.10%, respectively.

To improve the correlation between the FEA modes and the EMA modes above 450 Hz, an updating of the FE model parameters using the FEMtools software is to be conducted. However, issues of contact-type element compatibility between ANSYS and FEMtools delayed the completion of this task. Once the element compatibility issue is resolved and the model updating analysis is performed, the correlation between the FEA modes and the EMA modes is expected to improve.

6 CONCLUSIONS

In an effort to reduce NIHL among miners, NIOSH is conducting a project to develop engineering noise controls for longwall mining systems. Due to the dimensions and complexity of longwall systems, the approach used for noise control development consist of using numerical models of the most significant sound-radiating components to predict their vibratory and acoustic response and to explore various noise control options. The most significant sound-radiating components at the shearer are the two cutting drums; therefore, an FE model of a longwall cutting drum in the frequency range of interest of 0-1,000 Hz was developed in ANSYS. An EMA test was also conducted on an actual cutting drum, resulting in the measurement of the mode shapes in the frequency range of interest. Then the FE model was correlated using the experimental modal analysis test data. Upon conducting the correlation analysis, it was determined that the FE model is an accurate representation of the actual drum in the 0-200 Hz frequency range, and in general a good representation of the actual drum in the 0-450 Hz frequency range. Above 450 Hz, poor correlation, i.e. MAC values less than 50%, between the FEA modes and the EMA modes was obtained. There are various reasons why the average MAC values were not higher. First, the limited number of measuring points, i.e. test model mesh density, which includes only a very small fraction of the degrees of freedom of the FE model. There are also differences between the FE model and the actual drum; more specifically, the FE model does not account for differences in the material properties of a hard wear coat present on the actual drum at the beginning of the vanes. Another difference is related to the modal analysis conditions where the AirRides used for the experimental modal analysis test were modeled using a simple spring-to-ground connection in the modal FEA. In terms of excitation forces, inspection of the mode shape pairs and the FRFs shows that the use of the 4.448 N modal hammer provided insufficient excitation for some of the modes, e.g., mode 10 at 258.83 Hz where the drive point was close to a nodal line. Finally, the machine shop where the experimental modal test was conducted constituted a noisy environment with various heavy equipment moved in and out for repair purposes.

In order to improve the correlation of the FEA modes and the EMA modes, a model updating analysis should be conducted. For this study, this analysis was delayed by element compatibility issues between the software used for FEA and that used for the model updating. Once the compatibility issues are resolved and the model updating conducted, it is expected that the optimization of the FE model parameters based on the EMA test data will improve the correlation in the frequency range of interest, i.e. 0-1,000 Hz, resulting in a validated model useful for noise control development purposes.

7 ACKNOWLEDGEMENTS

The authors would like to acknowledge Nigel Goff, Michael Fidel and Jordan Taylor from BHP Billiton for their help and support during the experimental modal analysis test of the cutting drum. Thanks are also due to Jim Steedman from Navcon Engineering for his collaboration during the data analysis; and to Jessie Mechling from NIOSH for his help during the experimental modal analysis test.

8 REFERENCES

1. Tak, S.W., and Calvert, G.M., "Hearing Difficulty Attributable to Employment by Industry and Occupation: An Analysis of the National Health Interview Survey – United States, 1997 to 2003", *J. of Occ. and Env. Med.*, **50**(1), January 2008.

2. Franks, J.R., "Analysis of Audiograms for a Large Cohort of Noise-Exposed Miners".

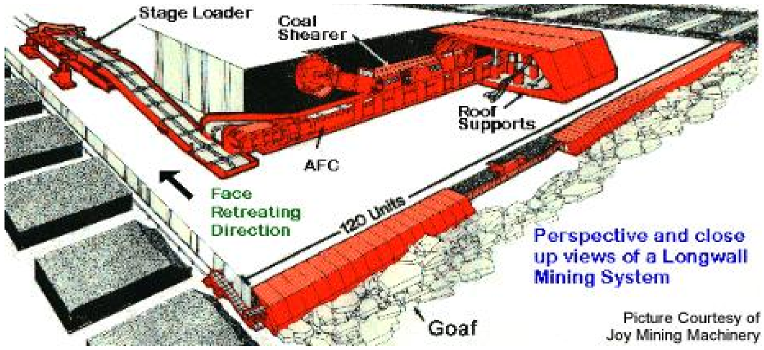
3. Energy Information Administration (EIA) website: www.eia.gov

4. Aljoe, W.W., Bartholomae, R.C., and Stein R.R., "Coal Cutting Noise Control", Proceedings of NoiseCon 87, State College PA, June 8-10, 1987.

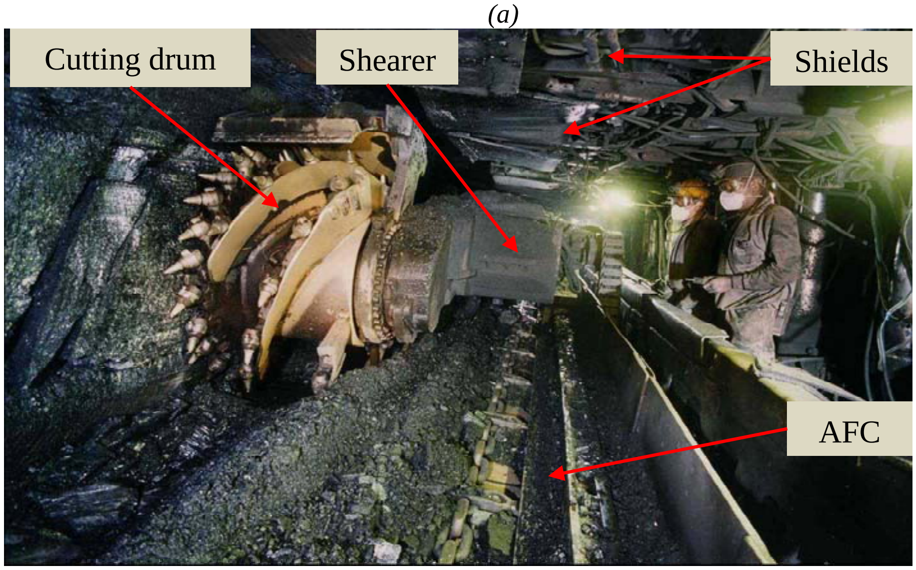
5. Ewings, D.J., Modal Testing: Theory, Practice and Application, Second Edition, Research Studies Press, LTD, Baldock, Hertfordshire, England, 2000.

Table 1 - FEA and EMA mode shape pairs, natural frequencies, and MAC value.

Pair #	FEA Mode	Frequency (Hz)	EMA Mode	Frequency (Hz)	Diff. (%)	MAC (%)
1	7	121.25	1	120.08	0.98	93.30
2	8	148.89	2	145.68	2.20	92.50
3	9	150.74	3	151.84	-0.72	95.70
4	10	157.42	4	160.48	-1.91	91.50
5	11	165.56	5	167.39	-1.09	97.70
6	12	196.56	6	206.78	-4.94	91.20
7	13	203.20	7	212.95	-4.58	88.70
8	14	213.28	8	220.16	-3.12	87.90
9	15	218.36	9	225.60	-3.21	83.90
10	16	242.35	11	258.83	-6.37	35.80
11	17	253.24	10	249.69	1.42	64.10
12	18	270.09	12	283.45	-4.71	89.20
13	19	274.91	13	289.71	-5.11	94.30
14	20	306.90	15	331.93	-7.54	68.00
15	21	306.91	16	338.01	-9.20	62.40
16	23	322.77	17	345.71	-6.64	92.20
17	24	331.56	18	354.03	-6.35	91.00
18	25	357.13	19	378.83	-5.73	86.70
19	26	374.87	21	395.07	-5.11	87.70
20	27	377.45	20	391.20	-3.51	79.70
21	28	384.25	22	409.86	-6.25	89.80
22	29	437.42	25	456.59	-4.20	51.90
23	30	439.16	23	449.14	-2.22	77.60
24	31	440.64	24	453.12	-2.75	50.90
25	32	452.18	26	458.25	-1.32	35.30
26	33	473.04	27	488.23	-3.11	75.90
27	34	483.93	29	515.39	-6.10	63.10
28	35	487.38	28	508.27	-4.11	55.00
29	36	497.54	30	526.33	-5.47	77.10
30	37	528.72	31	534.29	-1.04	23.80
31	38	537.20	32	574.24	-6.45	41.70
32	39	556.18	33	596.08	-6.69	48.20
33	40	564.87	34	602.76	-6.29	43.50
34	41	582.25	35	639.08	-8.89	60.30
35	42	614.20	37	663.57	-7.44	30.20
36	43	617.22	43	708.60	-12.90	22.10
37	44	617.67	49	774.24	-20.22	20.40
38	45	623.94	41	687.64	-9.26	37.30
39	47	653.08	53	800.96	-18.46	22.20
40	50	686.23	50	786.20	-12.72	22.90
41	52	693.87	54	804.01	-13.70	25.10
42	66	765.52	45	729.07	5.00	20.60
43	70	795.65	36	644.93	23.37	37.60
44	72	799.21	40	681.51	17.27	23.40



(a)



(b)

Fig. 1 - (a) Schematic of a longwall system, and (b) picture of a shearer in operation.

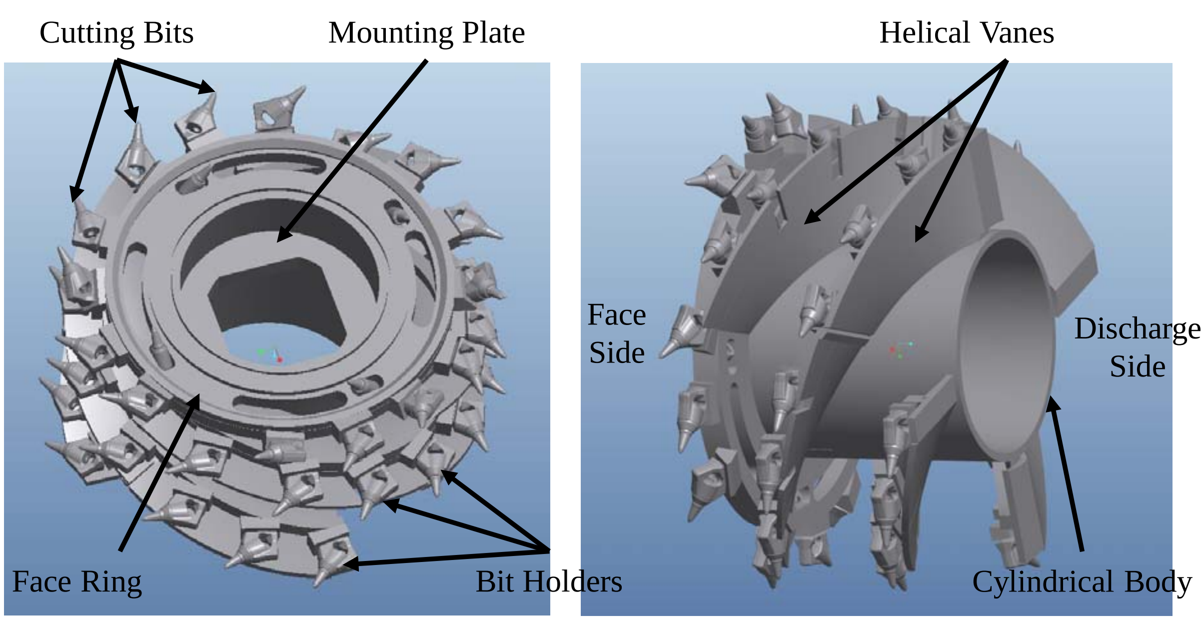
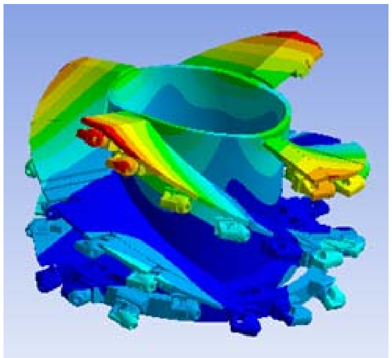
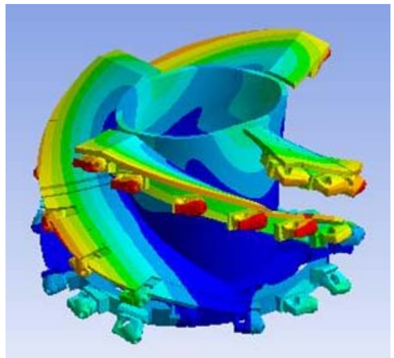
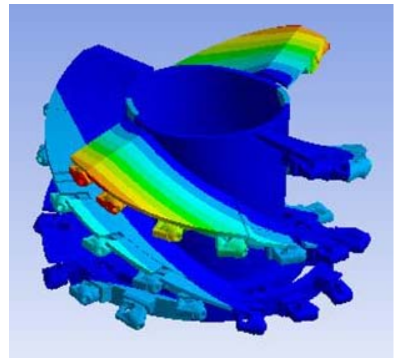
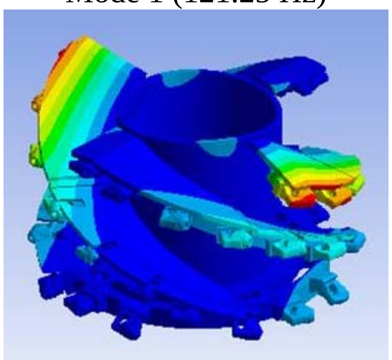


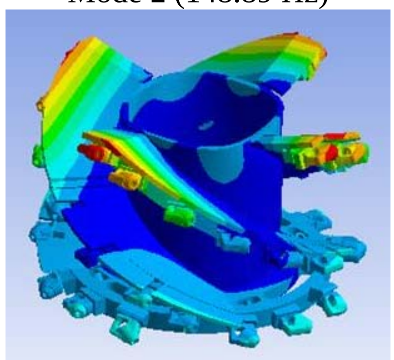
Fig. 2 - Drawing of a longwall cutting drum showing its various components.

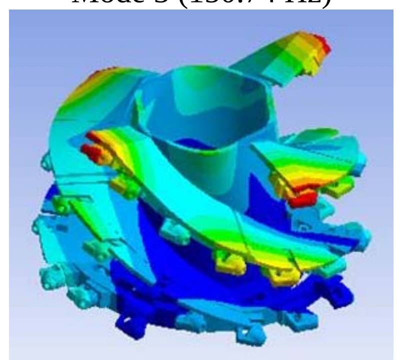

Mode 1 (121.25 Hz)

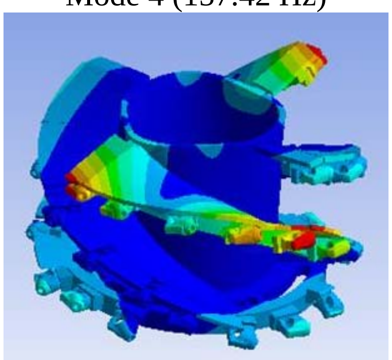

Mode 2 (148.89 Hz)

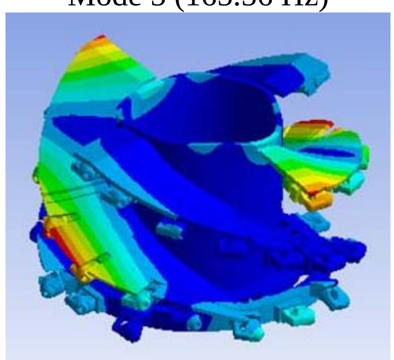

Mode 3 (150.74 Hz)

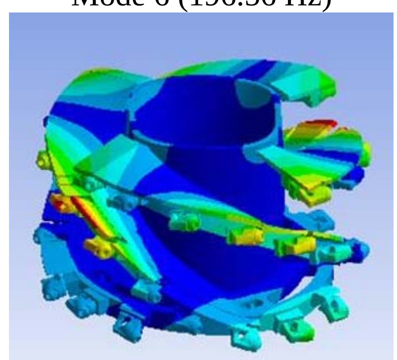

Mode 4 (157.42 Hz)

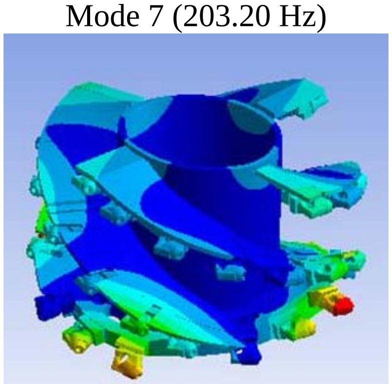

Mode 5 (165.56 Hz)

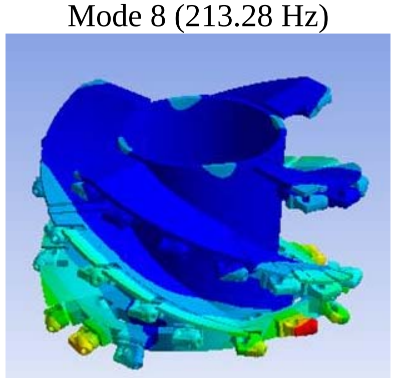

Mode 6 (196.56 Hz)


Mode 7 (203.20 Hz)


Mode 8 (213.28 Hz)


Mode 9 (218.36 Hz)


Mode 10 (242.35 Hz)


Mode 11 (253.24 Hz)

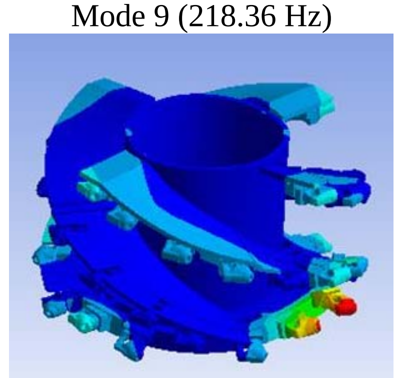

Mode 12 (270.09 Hz)

Fig. 3 - Modal FEA results for the first 12 structural modes.

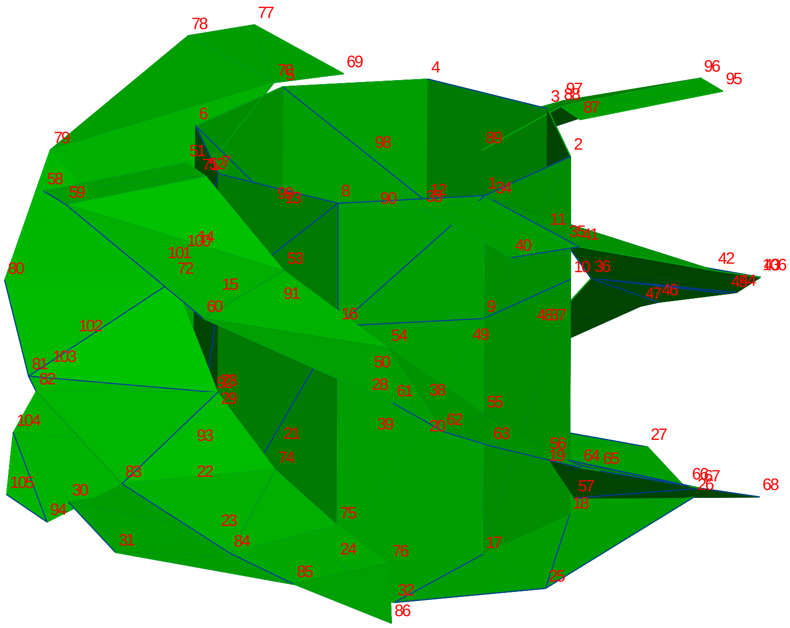


Fig. 4 – Test model obtained from the pretest analysis.

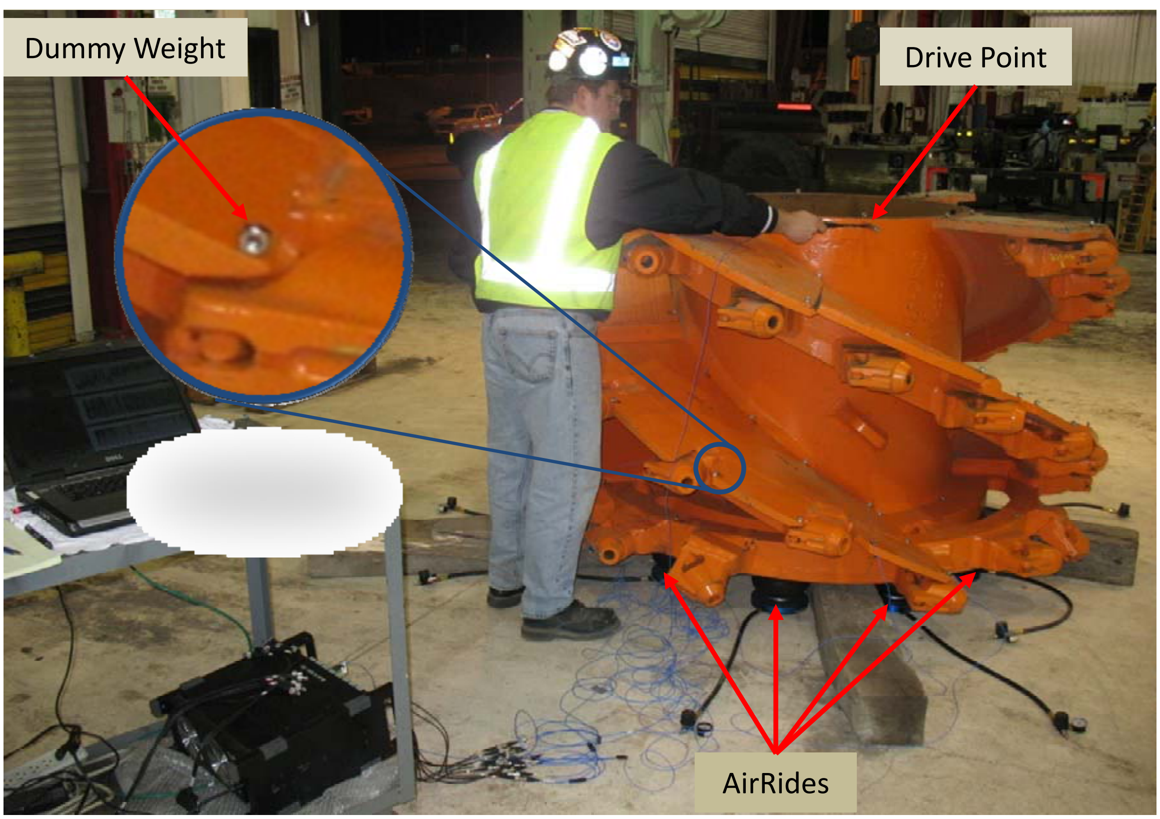
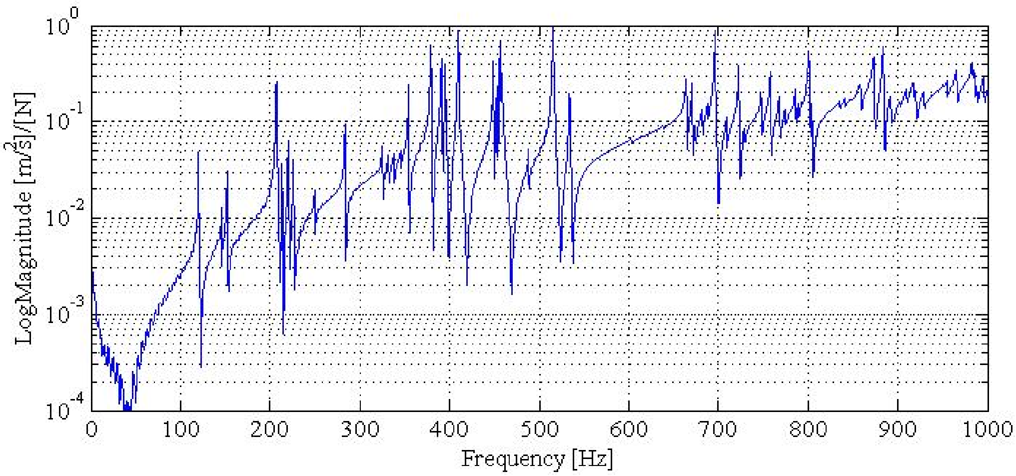
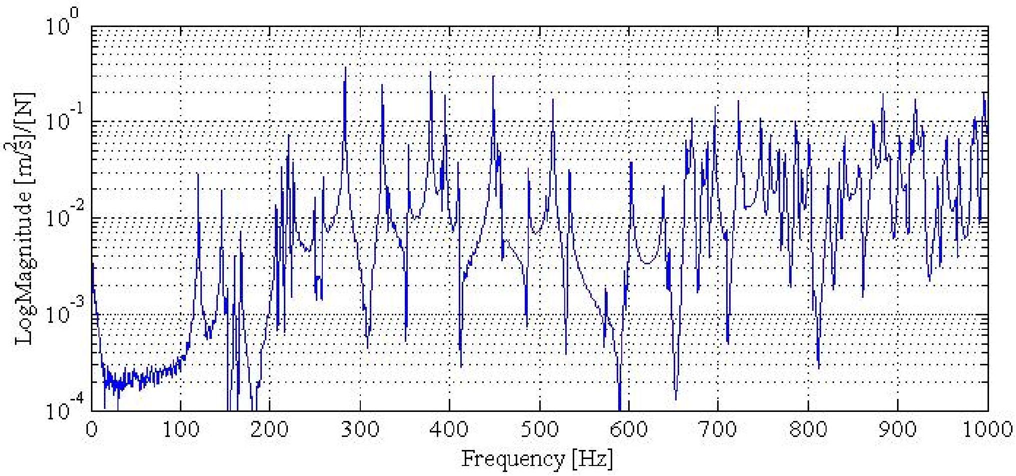


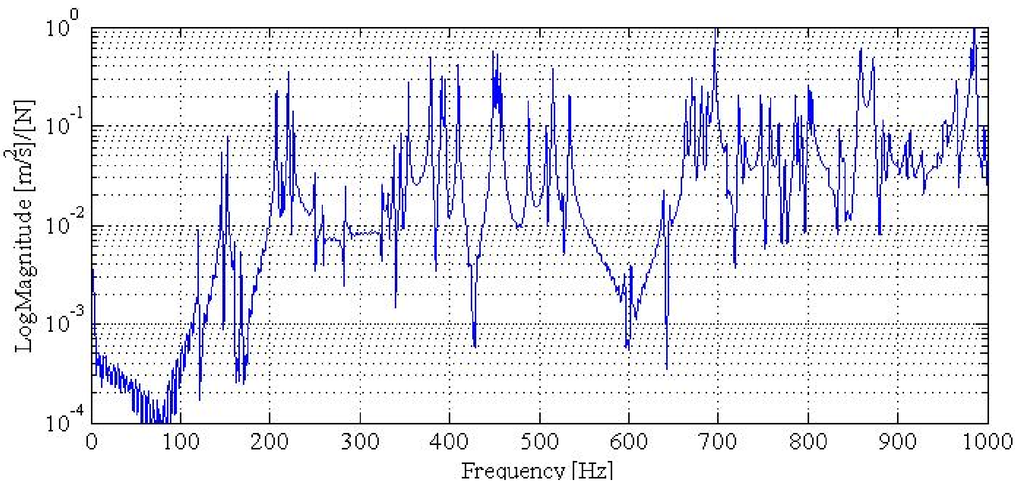
Fig. 5 – Setup for the experimental modal analysis test.



(a) Point 8—Cylindrical body (Drive point).



(b) Point 27—Face ring.



(c) Point 63—Vane.

Fig. 6 – FRF measured during the EMA test.