Preprint 10-084

NOISE CONTROLS FOR VIBRATING SCREEN MECHANISMS

M. J. Lowe, NIOSH, Pittsburgh, PA
D. S. Yantek, NIOSH, Pittsburgh, PA
H. E. Camargo, NIOSH, Pittsburgh, PA
L. A. Alcorn, NIOSH, Pittsburgh, PA
M. Shields, Conn-Weld Industries, Inc., Princeton, WV

ABSTRACT

National Institute for Occupational Safety and Health (NIOSH) studies show that 43.5% of surveyed coal preparation plant workers had noise exposures exceeding the Mine Safety and Health Administration Permissible Exposure Level. Sound levels around vibrating screens in these plants often exceed 90 dB(A). NIOSH is currently developing noise controls for horizontal vibrating screens. To characterize noise sources, NIOSH researchers performed sound pressure level (SPL) measurements on a vibrating screen at their Pittsburgh Research Laboratory. The results show that the entire screen contributes to noise below 1 kHz and the vibration mechanism housings are most significant above 1 kHz. Constrained layer damping (CLD) treatments and an enclosure were used to reduce mechanism housing noise. These were evaluated using sound power level measurements according to ISO 3744. The CLD treatments reduced the A-weighted sound power level by 3.1 dB in the 1 to 10 kHz one-third-octave bands. A panel-on-frame vibration mechanism enclosure using various types of panels further reduced the A-weighted sound power level from the CLD configuration in the 1 to 10 kHz one-third-octave bands by 3.7, 4.0, and 3.9 dB for aluminum, steel, and Dynalam panels, respectively. The combination yielded a 7 dB reduction from baseline in A-weighted sound power for the same frequency range.

INTRODUCTION

Hearing loss is one of the most common occupational illnesses in the United States (Franks et al. 1996). However, in the mining industry, hearing loss is 2.5-3 times greater than what is expected for the average of the population that is not exposed to occupational noise. Additionally, the same National Institute for Occupational Safety and Health (NIOSH) studies have shown that by the age of 50, 90% of coal miners have a hearing impairment versus only 10% of the population not exposed to occupational noise (Franks 1996). Noise-induced hearing loss is not just a problem in underground mining. In fact, a Mine Safety and Health Administration (MSHA) study of 60,000 full shift noise surveys showed that based upon federal noise regulations, 26.5% of workers from surface mining operations were overexposed to noise, compared to 21.6% of workers in underground mines (Seiler et al. 1994).

Above ground at coal preparation plants, a NIOSH study shows that 43.5% of employees are overexposed to noise. Furthermore, the study found that not only were vibrating screens one of the loudest pieces of equipment at the preparation plants, they were also the most numerous, thus making vibrating screens a key noise source to address (Vipperman et al. 2007).

In that light, a team of NIOSH researchers in partnership with Conn-Weld Industries, Inc., have used acoustic beamforming techniques to locate noise sources on a Conn-Weld G-Master 1000 de-watering vibrating screen. From the noise sources identified, they developed noise controls to mitigate the sound radiated by the mechanism housings. Additional noise controls for the screen body will be the subject of future investigations.

A horizontal vibrating screen (Figure 1) is a large machine used to process coal. The screen body has four sides made of steel plates with a bottom screening surface— also known as a screen deck— made of steel wire welded to a frame with small gaps between the wires. The body of the screen is supported on a steel coil spring suspension. For the Conn-Weld screen tested at NIOSH, two vibration mechanisms are mounted to a steel beam that spans the width of the screen. These vibration mechanisms, which use rotating eccentric shafts to generate vibration, are belt-driven using an electric motor.

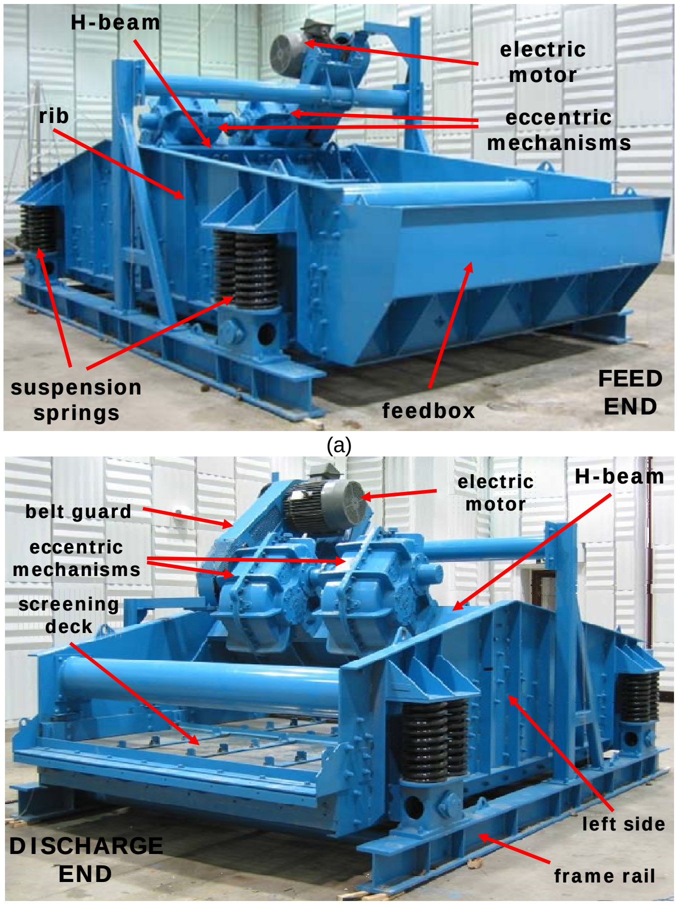


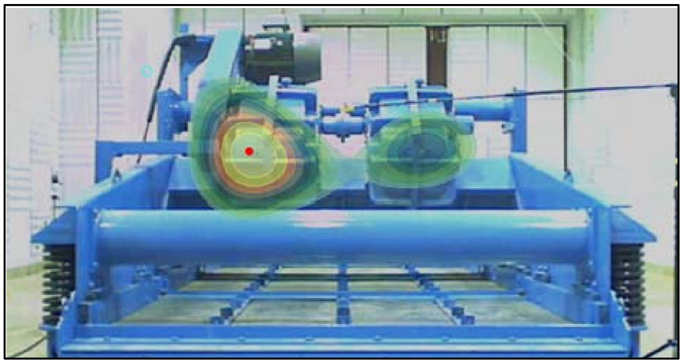
Figure 1. A horizontal vibrating screen used to process coal viewed from (a) feed end and (b) discharge end.

The screen is designed in such a way that it vibrates on roughly a 45 degree angle. In operation, coal flows into the feed end of the screen from a delivery chute. As the screen vibrates, the material moves along the deck and under a water spray that rinses the coal. The liquid and fine coal particles pass through the gaps in the screening deck as the material flows toward the discharge end of the screen. Finally, the rinsed coal falls off the discharge end of the screen to continue with further processing.

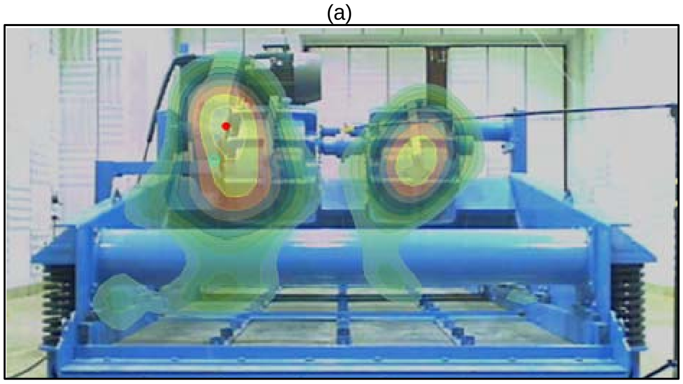
Noise Source Identification

Noise source identification was performed using the beamforming technique (Christensen and Hald 2004). The screen was positioned in the NIOSH Pittsburgh Research Laboratory (PRL) hemi-anechoic chamber with the screen directly on the chamber floor with wooden wedges driven under the frame rails to prevent rocking. The chamber dimensions are approximately 16.7 meters long, 10.1 meters wide, and 7.0 meters high. To collect the acoustic data, a Bruel & Kjaer Pulse data acquisition system simultaneously recorded the sound pressures from a 42-microphone, 1.9-meter-diamter wheel array.

The results showed that above 1 kHz, the vibration mechanisms are the most significant noise source (Yantek et al. 2008). Figure 2 shows examples of the beamforming results for the 1.6 and 2 kHz one-third-octave bands. In the figure, the light colors indicate the locations of high noise radiation. The figure clearly shows the vibration mechanism housings to be the dominant noise sources at these frequencies. Figure 2b indicates the belt guard might also be a source of noise. Close inspection revealed that during operation the belt guard was rattling against the screen structure due to a lack of clearance. Since it is easy to eliminate the rattling and it would interfere with evaluating other screen noise sources, the belt guard was removed for all other tests.



(a)



(b)

Figure 2. Beamforming results viewed from the discharge end of the screen for the (a) 1.6 kHz and (b) 2 kHz one-third-octave bands. Light colors indicate areas of high noise radiation.

To examine noise sources below 1 kHz, NIOSH contracted Acoustical and Vibration Engineering Consultants (AVEC) to perform beamforming measurements using their 121-microphone, 3.5-meter-diameter array. The vibrating screen was positioned in the center of the NIOSH PRL hemi-anechoic chamber. AVEC's phased array was mounted to a movable truss to position the array for measurements from each screen surface. This data was post-processed using AVEC beamforming analysis software. The results were examined in one-third-octave bands. The results indicated that below the 1 kHz one-third-octave band, the screen body is the main source of noise radiation (Yantek and Camargo 2009).

NOISE CONTROLS

Noise Controls Tested

To reduce noise from the screen, both noise from the vibration mechanism housings and noise from the screen body must be addressed. The work presented here focuses on the noise radiated by the vibration mechanism housings. NIOSH researchers developed two separate noise controls to address the mechanism housing noise. The first was a set of constrained layer damping treatments that were bonded directly to the outside of the mechanism housings. The second was an acoustic enclosure which surrounded both mechanism housings to block noise from reaching plant workers.

Constrained Layer Damping Treatments

The vibration mechanism housings were treated using constrained layer damping. To this end an 80 durometer, 0.025-in. thick elastomeric damping material was bonded to the flat face on the front, top, and back of each housing. These layers of damping material were then constrained using 1/4-in. thick steel plates. Figure 3 shows constrained layer damping treatments on the front face and top face. To ensure a good bond between the housings, damping material, and constraining plates, the paint from the housings was removed using a grinder and the constraining plates were sandblasted prior to applying the treatments.

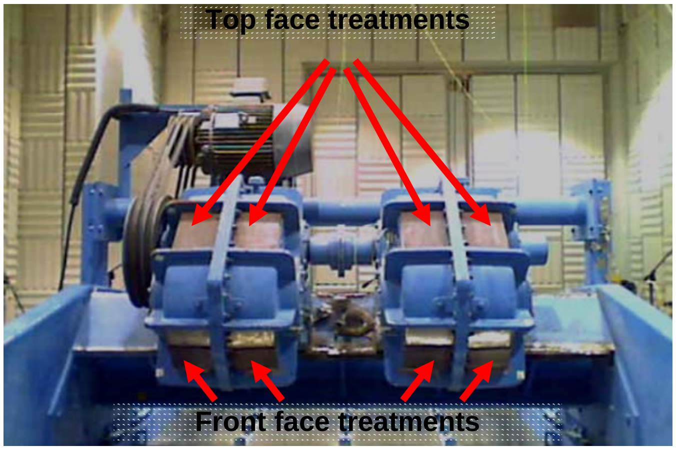


Figure 3. Constrained layer damping treatments applied to the top and front faces of the mechanism housings.

Acoustic Enclosure

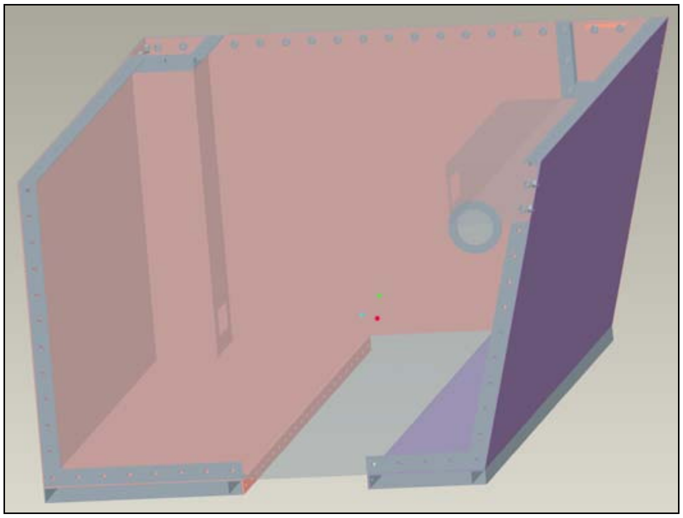
An acoustic enclosure was designed to enclose both mechanism housings and attach to the same H-beam that the mechanism housings attach to (see Figure 4). The motor and drive belts were not enclosed due to space constraints in coal preparation plants, plus they were not found to be significant contributors to noise. This first prototype was saltbox-shaped to maximize air space around the mechanism housings. The enclosure walls were comprised of three pieces of 1/8-in. thick steel joined together by bolts through 1/8-in. thick angle brackets with weld nuts on the back. A hole in the right-hand side of the enclosure was cut for the tapered pulley shaft to pass through.

A shim was used to move the pulley away from the mechanism housing to provide additional clearance between the pulley and the right side of the enclosure. The entire enclosure was lined with 2-in. thick Polydamp® acoustic foam to prevent build up of reverberant noise inside. Ducts were incorporated into the design in such a way as to provide both convective cooling and structural support for the enclosure.

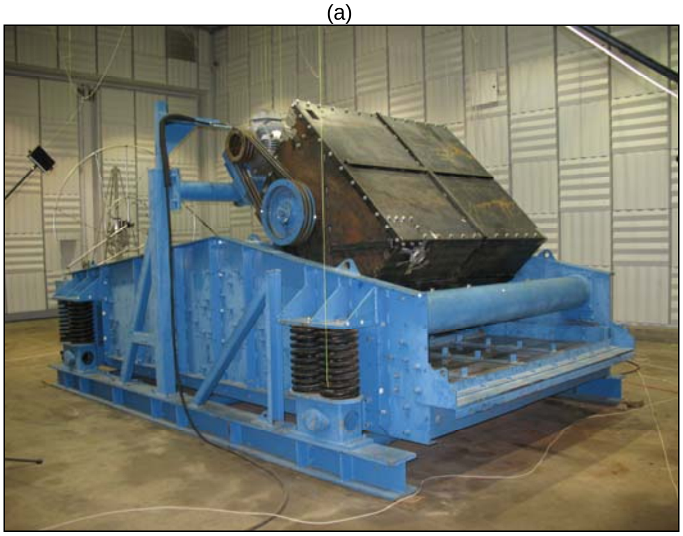
Test Setup

All measurements were taken on a Conn-Weld G-Master 1000 horizontal vibrating screen with dual vibration mechanisms and a screening deck that was 2.44 m × 4.88 m in size. The screen rested on the floor and wooden wedges were driven under the frame rails in order to prevent the screen from rocking during operation. The belt guard was removed to avoid rattling discovered in previous testing.

The tests were performed in the NIOSH PRL hemi-anechoic chamber. Sound pressure levels were measured using 21 B&K Type 4188 and 4189 microphones set up in a parallelepiped configuration surrounding the screen per ISO 3744 (ISO 1994). These data were converted to sound power levels following the reference source method listed in the ISO 3744 standard.



(a)



(b)

Figure 4. First enclosure constructed using a three-piece design. (a) Model showing internal duct structure and (b) enclosure installed on the vibrating screen.

EXPERIMENTAL RESULTS AND DISCUSSION

Constrained Layer Damping (CLD) Treatments

Figure 5 shows the A-weighted sound power level in one-third-octave bands for the baseline and with the constrained layer damping (CLD) treatments on the mechanism housings. The figure shows the CLD treatments reduced the sound power level in the 250 Hz through 8 kHz one-third-octave bands. In the frequency range that is dominated by mechanism housing noise, 1 to 10 kHz, the CLD treatments reduced the A-weighted sound power level by 3.1 dB. In addition, the overall A-weighted sound power level was reduced by 1.2 dB.

First Enclosure

The first enclosure prototype was tested in conjunction with the CLD treatments. Tests could not be performed with only the enclosure due to the failures of several welds during this first test. As can be seen in Figure 6, the combination of the enclosure and the CLD treatments reduced the sound power level in the 250 Hz to 2 kHz one-third-octave bands. However, the combination was worse for the 100 to 200 Hz one-third-octave bands and the 2.5 to 10 kHz one-third-octave bands.

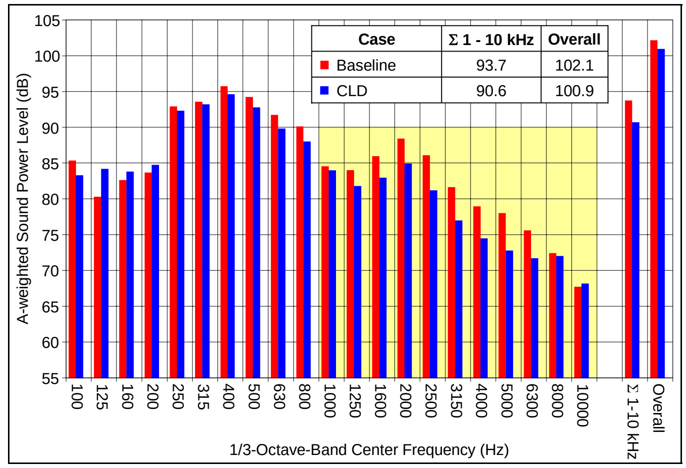


Figure 5. A-weighted sound power level in one-third-octave bands for baseline and CLD treatment configurations.

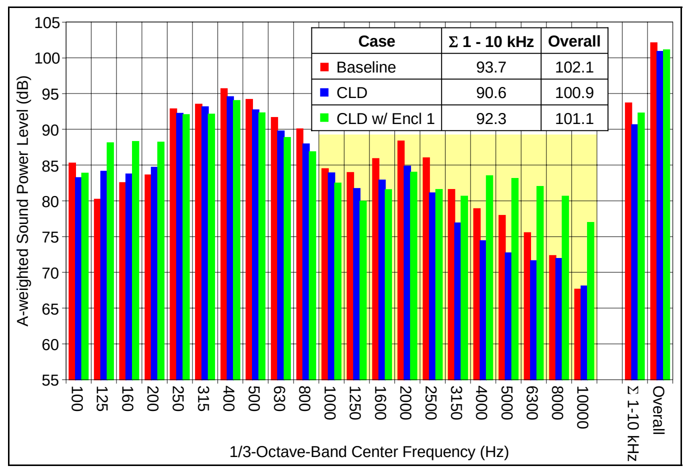


Figure 6. A-weighted sound power level in one-third-octave bands for baseline, with CLD treatments, and with CLD plus enclosure.

The lower frequency degradation is most likely attributable to panel modes from the large sides of the enclosure. An attempt was made to stiffen the panels by welding on reinforcement ribs; however, the additional low frequency noise was still present. The high frequency noise increase is likely caused by the metal-on-metal contact resulting from cracked welds which allowed pieces of the enclosure to slap together as the screen vibrated. This created a noticeable jack-hammer-like sound.

Second Enclosure

While the first enclosure showed some promise in the mid-frequency range, there were obvious manufacturing flaws and design issues that needed to be addressed. Besides the necessity of better welds to correct the high frequency performance degradation, the team wanted a stiffer design in order to address the low frequency noise and durability issues. Furthermore, we desired a design that would easily accommodate preventive maintenance and repairs in the field. Removable panels would allow easy access to fill ports, drain plugs, or entire mechanism assemblies without the need to remove the entire enclosure. Finally, we wanted each component of the enclosure to weigh no more than 50 pounds to allow the parts to be handled and installed more easily.

Based upon the above considerations, we created a modular panel-on-frame design. The steel frame provides a relatively stiff structure for the individual panels. Using smaller panels further increases panel stiffness thereby reducing the effect of panel modes on low frequency performance. The new enclosure accommodates different numbers and spacing of vibration mechanism housings and has no pieces that exceed our 50 pound weight target. This design consists of a series of steel frames that can be bolted together to make a larger or smaller enclosure as needed (see Figure 7).

Figure 7. Steel frame for the second enclosure installed on vibrating screen.

Bolt-on panels block the noise and can be easily removed to reach a fill port, drain plug, or bearing cover. An entire frame section with the panels attached can be removed to change a mechanism. Further, this design allows us to make interchangeable panels of various materials with different types of sound absorption and/or damping treatments for easier design optimization. Cooling ducts bolt on to panels separately and can be reconfigured as necessary (Figure 8).

Figure 8. Second enclosure installed on vibrating screen.

The steel frame was composed of 3/16-in. thick, 2-in. wide angle stock and 1/4-in. thick, 2-in. wide by 5/8-in. deep U-channel. It was isolated from the H-beam using strips of 1/2-in. thick, 57 durometer natural rubber. Four sets of panels were made for the second enclosure: aluminum, steel, Dynalam damped steel, and Paneltec aluminum honeycomb. All panels were 1/8-in. thick, except the honeycomb panels which were 1/4-in. thick (Figure 9). Panels were lined with 1-in. thick Polydamp® acoustic foam. The right panel was isolated from the bearing cover plate by a boss made of 1/4-in. thick, 57 durometer natural rubber. The bolts on the bearing cover plate were countersunk to increase clearance between the right panel and the bearing cover.

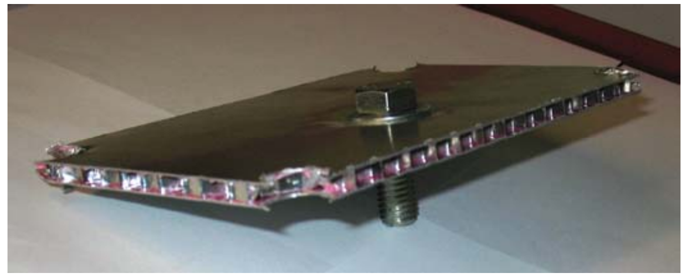


Figure 9. Example of aluminum honeycomb material.

The sound power level was measured with the second enclosure using each of the aforementioned panel materials in addition to the CLD treatments. Each type of panel material reduced the A-weighted sound power level in the 1 to 10 kHz one-third-octave bands. However, with the honeycomb panels, the overall A-weighted sound power level increased by 2 dB. Due to the construction of the panels, the bolts that attach the panels to the frame could not be sufficiently tightened without crushing the panels. The lack of sufficient clamping force allowed the panels to rattle against the frame thereby increasing noise. This problem might be resolved with press-fit sleeve inserts into the panels for each bolt, but this would be cost-prohibitive to manufacture.

Figure 10 shows the A-weighted sound power level in one-third-octave bands for the enclosure with aluminum, steel, and Dynalam panels with the CLD treatments compared to the data with only the CLD treatments. The aluminum, steel, and Dynalam panels reduced the A-weighted sound power level in the 1 to 10 kHz one-third-octave bands by 3.7, 4.0, and 3.9 dB, respectively. For all practical purposes, the results are the same because changes on the order of a few tenths of a decibel are insignificant and can be a result of test-to-test variation.

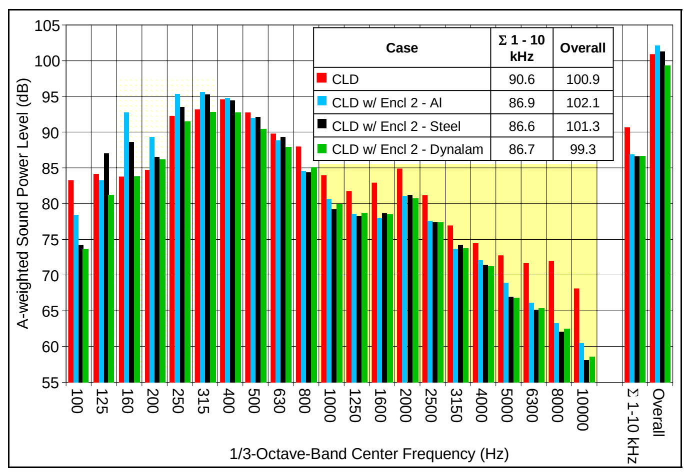


Figure 10. A-weighted sound power level in one-third-octave bands for CLD treatments and CLD plus second enclosure.

With the aluminum and steel panels, the overall A-weighted sound power level was increased by 1.2 and 0.4 dB, respectively. Close inspection of Figure 10 shows the enclosure with either aluminum or steel panels increased the sound power level in the 160 through 315 Hz one-third-octave bands by several decibels. This increase is probably due to the excitation of panel modes and the resulting noise radiation. Using the Dynalam panels reduced overall A-weighted sound power level by 1.6 dB. With the Dynalam panels, levels in the 160 through 315 Hz one-third-octave bands were approximately the same as those with the CLD treatments. For the aluminum and steel panels, lower frequency degradations from baseline may be avoided by adding a rib pattern to the panels for stiffening, or perhaps using thicker panels.

Figure 11 shows a comparison of the A-weighted sound power level spectra for the baseline, CLD treatments, and CLD treatments with the second enclosure using Dynalam panels. The figure shows the A-weighted sound power level in the 1 to 10 kHz frequency range was reduced by 7 dB with the CLD treatments and the Dynalam enclosure. Together, the combination reduced the overall A-weighted sound power level by 2.8 dB, which is nearly a 50% reduction in terms of sound energy.

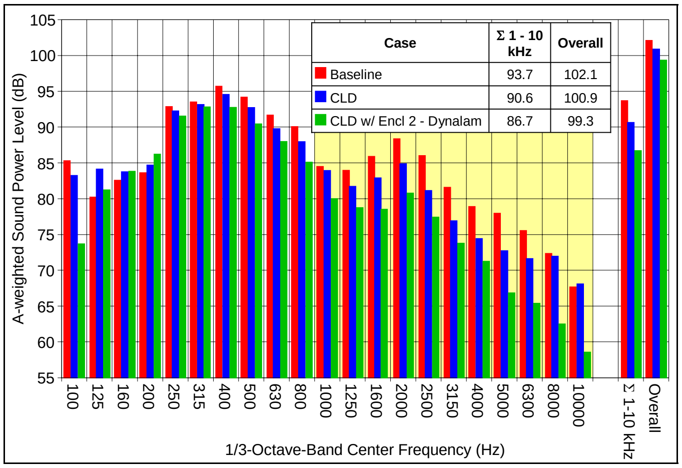


Figure 11. A-weighted sound power level in one-third-octave bands for baseline, with CLD treatments, and with CLD plus second enclosure using Dynalam panels.

CONCLUSIONS

Noise source identification data show that the main sources of noise on the Conn-Weld G-Master 1000 vibrating screen are the screen body and the vibration mechanism housings. Below 1 kHz, the screen body is the dominant noise source whereas noise radiated from the vibration mechanism housings is the primary source above 1 kHz. Constrained layer damping (CLD) plates and an acoustic enclosure were designed to reduce the noise radiated by the vibration mechanism housings.

CLD treatments on the mechanism housings reduced the A-weighted sound power level in the 1 to 10 kHz one-third-octave bands by 3.1 dB. In addition, the CLD treatments reduced the overall A-weighted sound power level by 1.2 dB. Adding an enclosure with Dynalam steel panels in conjunction with the CLD treatments reduced the A-weighted sound power level in the 1 to 10 kHz frequency range by 7 dB versus the baseline values. In addition, this combination reduced the overall A-weighted sound power level by 2.8 dB compared to the baseline. We expect these reductions to translate into a reduction of operator noise exposure in the field. To further reduce noise exposure, controls must be developed to reduce noise radiated by the screen body. A complete package of noise controls for vibrating screens will be the subject of our future work.

ACKNOWLEDGEMENTS

The authors would like to thank Pat McElhinney, Pete Kovalchik, and Arc Weld, Inc. for their valuable assistance in the second enclosure design process; Bob Michael and Corry Rubber Corporation for their donation of engineering samples and expertise for isolation of the enclosure; Conn-Weld Industries, Inc. for providing the vibrating screen; John Pack for assistance with technical questions on the vibrating screen; and Jessie Mechling, Rob Nahay, Tim Matthews, Ben Lewis, Shawn Peterson, Alexander Salas, Art Hudson, Adam Smith, and Kurt Pawlak for additional help with enclosure assembly and testing.

REFERENCES

1. John R. Franks, Mark R. Stephenson, and Carol J. Merry (1996), “Preventing occupational hearing loss—A practical guide”, Technical Report No. 96-110, National Institute for Occupational Safety and Health, June.

2. J. R. Franks (1996), “Analysis of audiograms for a large cohort of noise-exposed miners”, National Institute of Occupational Safety and Health, Internal Report, Cincinnati, OH, 7 pp.

3. J. P. Seiler, M. P. Valoski, and M.A. Crivaro (1994), Noise Exposure in U.S. Coal Mines, U.S. Department of Labor, Mine Safety, and Health Administration, Informational Report No. IR 1214, 46 pp.

4. Jeffrey S. Vipperman, Eric R. Bauer, and Daniel R. Babich (2007), “Survey of noise in coal preparation plants”, *Journal of the Acoustical Society of America*, 121:197–205.

5. JJ Christensen and J. Hald (2004), “Beamforming”, *Bruel & Kjaer Technical Review* No. 1-2004, Naerum, Denmark: Bruel & Kjaer Sound & Vibration Measurement A/S.

6. David S. Yantek, Hugo E. Camargo, and Rudy J. Matetic (2008), “Application of a Microphone Phased Array to Identify Noise Sources on a Horizontal Vibrating Screen”, in *NOISE-CON 2008: Proceedings of the 2008 National Conference on Noise Control Engineering* 2008:1-15, Dearborn, MI, July 28–31. C. Burroughs, T. Lim, J. Kim, and G Maling, eds., Indianapolis, IN: Institute of Noise Control Engineering of the USA.

7. David S. Yantek and Hugo E. Camargo (2009), “Structural Vibration as a Noise Source on Vibrating Screens”, presented at the American Society of Mechanical Engineers International Mechanical Engineering Congress and Exposition, Lake Buena Vista, FL, November 13–19.

8. ISO 3744 (1994), *Acoustics— Determination of Sound Power Levels of Noise Sources Using Sound Pressure— Engineering Method in an Essentially Free Field Over a Reflecting Plane*, Geneva: ISO.