

# Noise controls for vibrating screen mechanisms

**M.J. Lowe, D.S. Yantek, H.E. Camargo and L.A. Alcorn**

Mechanical engineer, mechanical engineer, research engineer, engineering technician,  
National Institute for Occupational Safety and Health (NIOSH), Pittsburgh, PA

**M. Shields**

Vice president of engineering, Conn-Weld Industries, Inc., Princeton, WV

## Abstract

*U.S. National Institute for Occupational Safety and Health (NIOSH) studies show that 43.5% of surveyed coal preparation plant workers had noise exposures exceeding the U.S. Mine Safety and Health Administration permissible exposure level of 90 dB(A). 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 Office of Mine Safety and Health Research laboratory in Pittsburgh. 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 in this range from 1-10 kHz. 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.*

Key words: Noise control, coal, vibrating screens

## Introduction

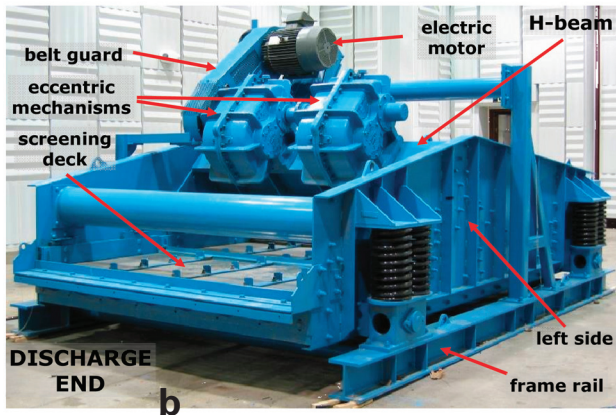
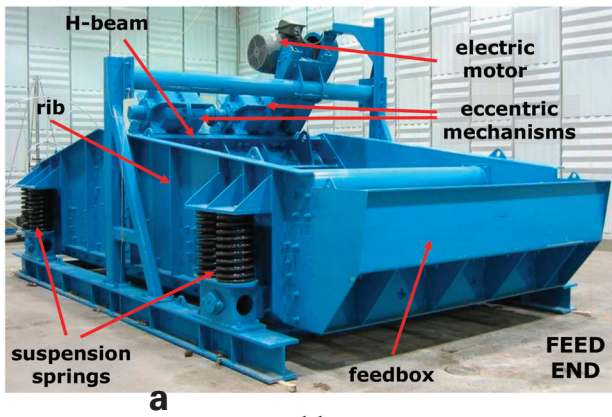
Hearing loss is one of the most common occupational illnesses in the United States (Franks et al., 1996). However, in the mining industry, hearing loss is 2.5-3 times greater than what is expected for the average of the population that is not exposed to occupational noise. Additionally, the same National 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 that is 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 conventional frequency domain beamforming techniques (Mueller, 2002;

Christensen and Hald, 2004) to locate noise sources on a Conn-Weld G-Master 1000 dewatering 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 (Fig. 1) is a large machine used to process coal. Vibrating screens can be used for a variety of applications, such as sizing, separating and dewatering both coal and refuse (rock) of various sizes. 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.

This screen is designed in such a way that it vibrates on roughly a 45° 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



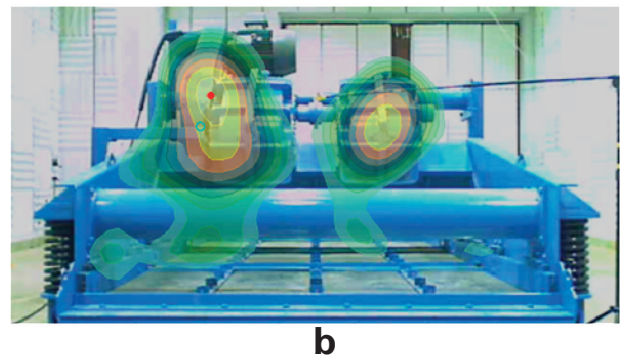
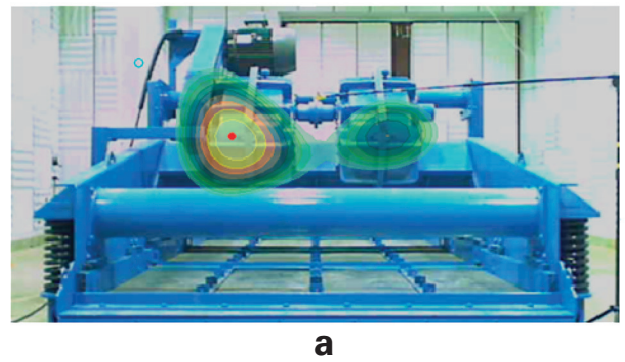
**Figure 1** — A horizontal vibrating screen used to process coal viewed from (a) feed end and (b) discharge end.

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 sources were identified using the beamforming technique. The screen was positioned in the NIOSH Office of Mine Safety and Health Research (OMSHR) hemi-anechoic chamber in Pittsburgh 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 m long, 10.1 m wide and 7.0 m high. To collect the acoustic data, a Bruel & Kjaer Pulse data acquisition system simultaneously recorded the sound pressures from a 42-microphone, 1.9-m-diameter wheel array. Based on previous testing, it was known that 99% of the overall A-weighted sound level of the vibrating screen came from the 100 Hz to 10 kHz one-third-octave bands. Additionally, NIOSH researchers discovered that the screen body was the main low-frequency noise source and the mechanism housings were the main high-frequency noise source (Yantek et al., 2005; Yantek and Camargo, 2009). This gave a starting point to determine the frequency ranges that were influenced by the mechanism housings and screen body.

The beamforming 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 since the light colors line up with the mechanism housings. Figure 2b indicates the belt guard might also be a source of



**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.

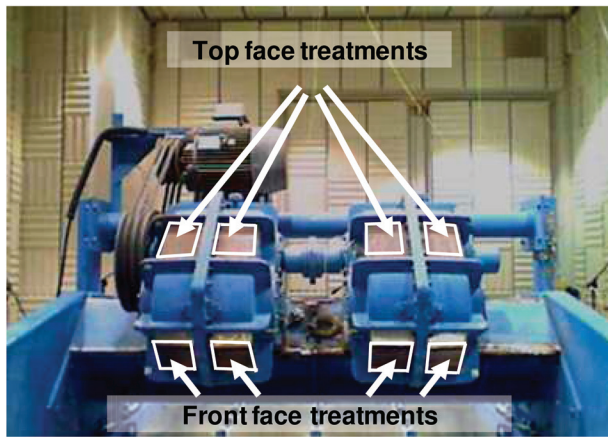
noise. Close inspection revealed that during operation the belt guard was rattling against the screen structure due to a lack of clearance. Since it is easy to eliminate the rattling and it would interfere with evaluating other screen noise sources, the belt guard was removed for all other tests.

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-m-diameter array. The vibrating screen was positioned in the center of the NIOSH Office of Mine Safety and Health Research (OMSHR) hemi-anechoic chamber in Pittsburgh, PA. 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, which implements a conventional frequency domain beamforming algorithm. To this end, scanning grids in the planes of interest are generated and potential noise sources are assumed at each grid point. Then, the relative phase measured between microphones is compared to the relative phase that would be induced between the microphones by the assumed sources (Ravetta, 2005). The results were obtained in the form of one-third-octave band acoustic maps. 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, noise controls need to be developed for both noise from the vibration mechanism housings and noise from the screen body. The work presented here focuses on the noise radiated by the vibration mechanism housings. As mentioned in the previous section, the vibration mechanism housings are the dominant





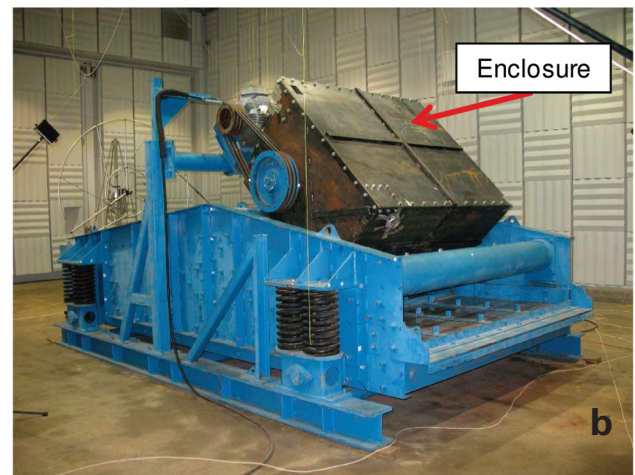
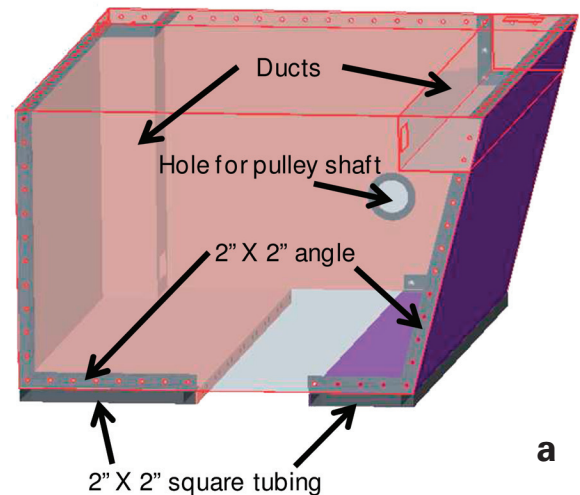
**Figure 3** — Constrained layer damping treatments applied to the top and front faces of the mechanism housings.

noise source in the one-third-octave bands from 1 to 10 kHz. NIOSH researchers developed two separate noise controls to reduce the mechanism housing noise in this frequency range. 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.** Constrained layer damping applies a flexible damping material— which is typically a rubber-like material— to a surface that is vibrating and generating noise. The firmness of this rubbery material is measured by a property called durometer. The higher the durometer, the higher the shear modulus of the material is. In other words, a higher durometer value means the material is less pliable. The other side of the damping material is constrained by a more rigid material such as steel. As the original base layer vibrates, it transmits motion to the damping layer. Since the damping layer is constrained on both sides, the vibration energy from the base layer is transformed into a shear deformation within the damping layer, which reduces the noise that is radiated by the system.

For this application, a thin sheet of 80 durometer, 0.64-mm (0.025-in.) thick elastomeric damping material was bonded on one side to the flat faces on the front, top and back of each housing by using an adhesive. These layers of damping material were then constrained on the opposite side by bonding them to 6.4-mm-(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.

**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. 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 (a rectangular prism modified to have an angled face) to maximize air space around the mechanism housings (see Fig. 4). The enclosure walls were comprised of three pieces of 3.2-mm-(1/8-in.)-thick steel joined together by bolts through



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

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 installed to move the pulley away from the mechanism housing, providing additional clearance between the pulley and the right side of the enclosure. The entire enclosure was lined with 50.8-mm-(2-in.)-thick Polydamp acoustic foam to prevent buildup of reverberant noise inside. Ducts were incorporated into the design as shown in Fig. 4a 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 a solid concrete floor and wooden wedges were driven under the frame rails in order to prevent the screen from rocking during operation. In a preparation plant, screens will often be mounted directly to a steel structure underneath, where the concrete plant floor has been cut away. However, this type of modification was not practical for the test chamber and plant configurations vary. The noise reductions from this research in the laboratory will vary in the field since each screen and mounting condition will be different.

The screen was running without load (i.e., no coal or other material was flowing over the screen deck.) This should not significantly affect the results, since earlier field testing of similar vibrating screens showed that mechanism and screen body noise is a far greater contributor to overall screen noise than sound generated by material flow (Yantek et al., 2005). The belt guard was removed to avoid rattling that was discovered in previous testing. The configurations tested were: baseline screen with no noise controls, constrained layer damping on the mechanism housings, and constrained layer damping treatments combined with the enclosure.

The tests were performed in the NIOSH OMSHR hemi-anechoic chamber in Pittsburgh, PA. 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). Three 30-second sound samples were measured and averaged for each configuration. These data were then converted to sound power levels following the reference source method listed in the ISO 3744 standard.

## Experimental results and discussion

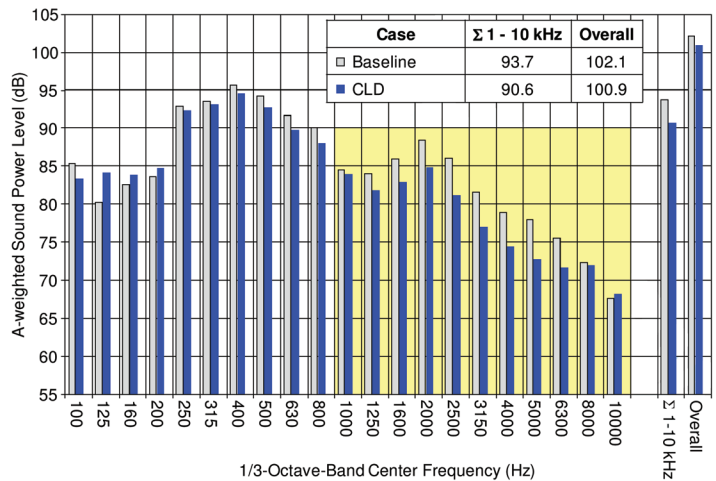
### Constrained layer damping (CLD) treatments.

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.

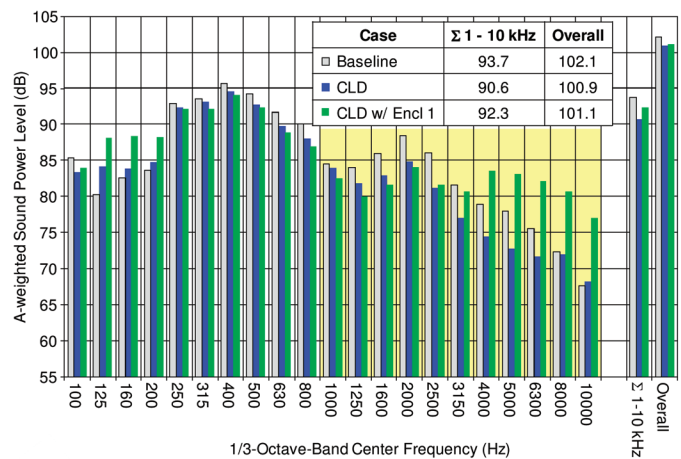
**First enclosure.** The first enclosure prototype was tested in conjunction with the CLD treatments. Tests could not be performed with only the enclosure, due to the failure of several enclosure welds during this first test. As can be seen in Fig. 6, the combination of the enclosure and the CLD treatments 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.

The lower frequency degradation is most likely attributable to panel vibration 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 on the enclosure which allowed pieces of it to slap together as the screen vibrated. This created a noticeable jackhammer-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 corrected. Besides the necessity of better welds to correct the high frequency performance degradation, NIOSH and their collaborators wanted a stiffer design in order to address the low frequency noise and durability issues. Furthermore, they desired a design that would easily accommodate preventive



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



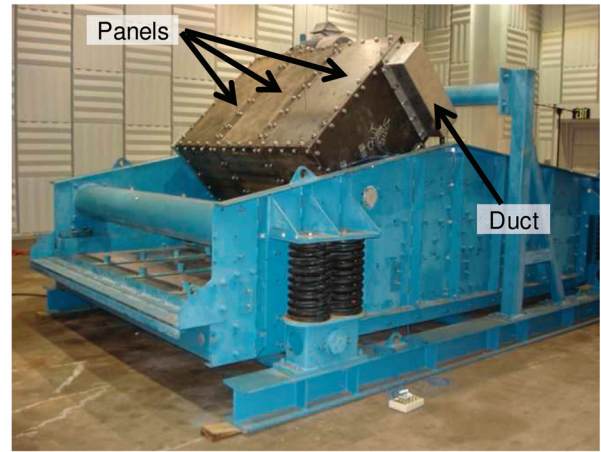
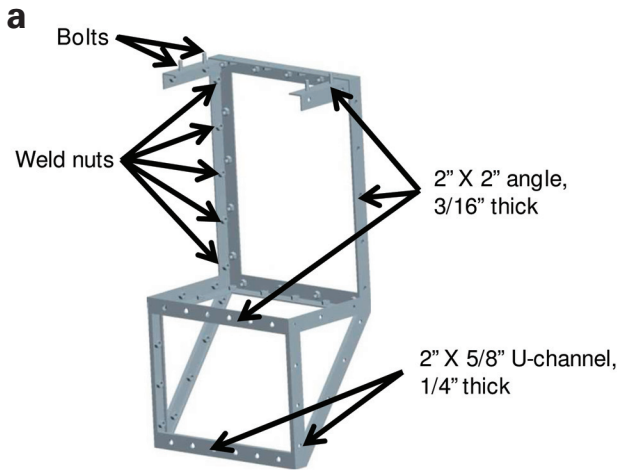
**Figure 6** — A-weighted sound power level in one-third-octave bands for baseline, with CLD treatments, and with CLD plus enclosure.

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 whole enclosure. Finally, the design team wanted each component of the enclosure to weigh no more than 222 N (50 lb) to allow the parts to be handled and installed more easily.

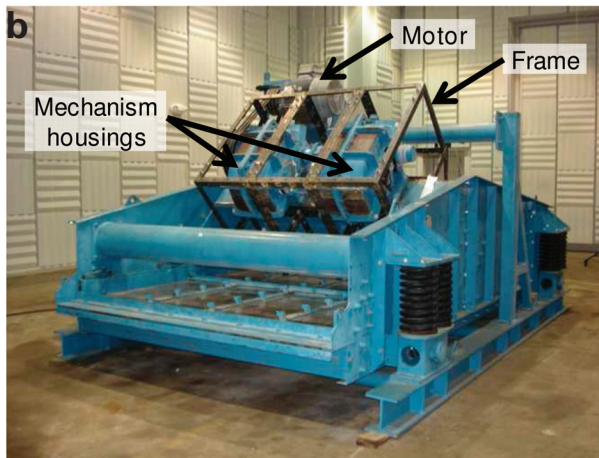
Based upon the above considerations, NIOSH researchers created a modular panel-on-frame design. The steel frame provides a relatively stiff structure for the individual panels. Using smaller panels further increases panel stiffness, thereby reducing the effect of panel modes on low frequency performance. The new enclosure accommodates different numbers and spacing of vibration mechanism housings and has no pieces that exceed the 222-N (50-lb) 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 Fig. 7).

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 the use of interchangeable panels made of various materials with different types of sound absorption and/or damping treatments for easier design optimization. Cooling ducts bolt onto panels separately and

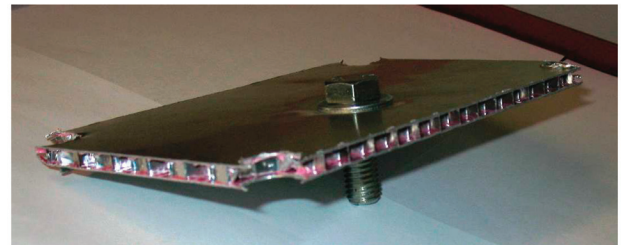




**Figure 8** — Second enclosure installed on vibrating screen.



**Figure 7** — Steel frame for the second enclosure showing (a) one frame section and (b) the entire frame assembly installed on a vibrating screen.



**Figure 9** — Example of aluminum honeycomb material.

can be reconfigured as necessary (Fig. 8).

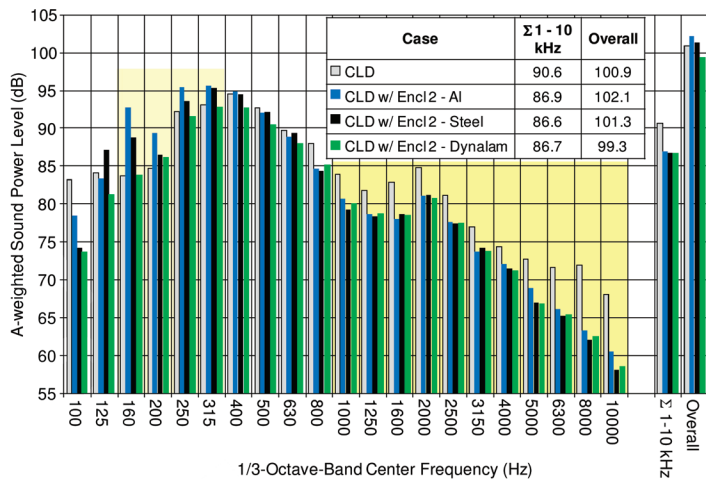
The frame was composed of steel angle stock and U-channel. It was isolated from the H-beam using strips of 57 durometer natural rubber. Four sets of panels were made for the second enclosure: aluminum, steel, Dynalam damped steel (a CLD steel) and Paneltec aluminum honeycomb. All panels were 3.2 mm (1/8 in.) thick, except the honeycomb panels, which were 6.4 mm (1/4 in.) thick (Fig. 9). Panels were lined with 25.4-mm (1-in.) thick Polydamp acoustic foam. The right panel was isolated from the bearing cover plate by a boss made of 57 durometer natural rubber. The bolts on the bearing cover plate were countersunk to increase clearance between the right panel and the bearing cover.

The sound power level was measured with the second enclosure using each of the aforementioned panel materials in addition to the CLD treatments on the mechanism housings. 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 on the mechanism housings. Each type of panel material reduced the A-weighted sound power level in the 1 to 10 kHz one-third-octave bands. The aluminum, steel and Dynalam™ panels reduced the A-weighted sound power level in the 1 to 10 kHz one-third-octave bands

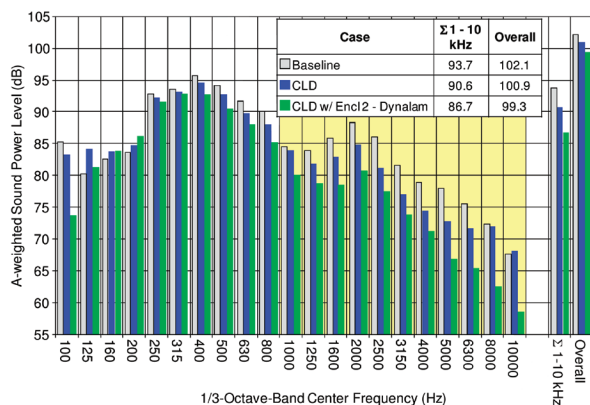
by 3.7, 4.0 and 3.9 dB, respectively. For all practical purposes, the results are the same because changes on the order of a few tenths of a decibel are insignificant and can be a result of test-to-test variation.

While the honeycomb panels reduced noise from 1-10 kHz, the overall A-weighted sound power level increased by 2 dB. Due to the construction of the honeycomb 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 honeycomb 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. 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 Fig. 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 only the CLD treatments on the mechanism housings. For the aluminum and steel panels, the low frequency degradation 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



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



**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.

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.

## 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 in the lab to translate into a reduction of operator noise exposure in the field for an estimated production cost of less than 15% of the screen price. 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. Field testing of this package is planned for multiple vibrating screens to evaluate durability as well as effectiveness at reducing noise.

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## Disclaimer

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. Reference to specific brand names does not imply endorsement by the National Institute for Occupational Safety and Health.

## References

- Christensen, J.J., and Hald, J., 2004, "Beamforming," *Bruel & Kjaer Technical Review*, No. 1-2004, Naerum, Denmark: Bruel & Kjaer Sound & Vibration Measurement A/S.
- Franks, J.R., Stephenson, M.R., and Merry, C.J., 1996, *Preventing Occupational Hearing Loss—A Practical Guide*, Technical Report No. 96-110, National Institute for Occupational Safety and Health, June.
- Franks, J.R., 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.
- 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.
- Mueller, T. (ed.), 2002, *Aeroacoustic Measurements*, Springer.
- Ravetta, P.A., *LORE Approach for Phased Array Measurements and Noise Control of Landing Gears*, Ph.D. dissertation, Virginia Polytechnic Institute and State University, 2005.
- Seiler, J.P., Valoski, M.P., and Crivaro, M.A., 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.
- Vipperman, J.S., Bauer, E.R., and Babich, D.R., 2007, "Survey of noise in coal preparation plants," *Journal of the Acoustical Society of America*, Vol. 121 pp. 197–205.
- Yantek, D.S., Jurovcik, P., and Bauer, E.R., 2005, "Noise and vibration reduction of a vibrating screen," *Transactions of the Society for Mining, Metallurgy and Exploration*, Vol. 318, pp. 201–213.
- Yantek, D.S., Camargo, H.E., and Matetic, R.J., 2008, "Application of a microphone phased array to identify noise sources on a horizontal vibrating screen," *NOISE-CON 2008: Proceedings of the 2008 National Conference on Noise Control Engineering*, Vol. 2008:1-15, C. Burroughs, T. Lim, J. Kim, and G. Maling, eds., Dearborn, MI, July 28–31, 2008, Indianapolis, IN, Institute of Noise Control Engineering of the USA.
- Yantek, D.S., and Camargo, H.E., 2009, "Structural vibration as a noise source on vibrating screens," *Proceedings of the American Society of Mechanical Engineers International Mechanical Engineering Congress and Exposition, IMECE2009-12272*, November 13–19, 2009, Lake Buena Vista, FL, ASME.