Noise and vibration reduction of a vibrating screen

D.S. Yantek, P. Jurovcik and E.R. Bauer
Mechanical Engineer, general engineer and mining engineer, respectively,
NIOSH, Pittsburgh, Pennsylvania

Abstract

Workers in coal-preparation plants, where vibrating screens are significant noise sources, are often exposed to sound levels exceeding 90 dB(A). The National Institute for Occupational Safety and Health, a major coal company and a vibrating-screen manufacturer conducted a cooperative study of vibrating-screen noise. Rubber isolators reduced sound levels by 1 dB. Urethane screen panels did not reduce sound levels in this case. Drive noise was determined to be the dominant noise source on the screen. Damped side plates reduced vibration levels on the screen sides by 1 to 7 dB(A). Detailed results of the study are presented.

Introduction

In 2000, there were 212 preparation plants in operation in the United States and 129 of these plants were located in three states: Kentucky, Pennsylvania and West Virginia (Fiscor and Lyles, 2000). Studies by The National Institute for Occupational Safety and Health (NIOSH) have shown that workers who spend a significant portion of their shift working in a coal preparation plant can experience noise exposures that exceed the Mine Safety and Health Administration (MSHA) Permissible Exposure Level. Noise doses up to 220% have been recorded for preparation-plant workers in jobs with titles such as stationary equipment operator, froth cell operator, plant operator, plant controls man, third floor operator, wet plant attendant, sump floor operator, plant backup and plant mechanic (Bauer, 2004). These job classifications require the worker to spend a considerable portion of a shift in the plant while working around slurry pumps, dryers, centrifuges and vibrating screens. Vibrating screens generate sound levels ranging from 90 to 95 dB(A) during clean bituminous coal processing and 95 to 100 dB(A) during refuse and anthracite processing (Ungar et al., 1974).

Because they are used to size, separate and dewater both coal and refuse (rock) of various sizes, screens may be located on many floors within a preparation plant. The number of screens in a processing plant can range from a single screen to more than a dozen. Consequently, preparation-plant workers can be exposed to high sound levels generated by the screens multiple times during a shift as they move and work throughout the floors within the plant. Vibrating screens are a major noise problem in most coal preparation plants because screens are used extensively in the plants, are usually located in high traffic areas and can generate high noise levels (Rubin et al., 1982).

Due to the revised Health Standard for Occupational Noise Exposure (30 CFR Part 62) passed in 1999, the Mine Safety and Health Administration (MSHA) no longer gives credit for wearing hearing protection in determining a worker's noise dose. This new MSHA regulation has reemphasized the use of engineering noise controls to reduce worker noise exposure. To address the problem of worker noise exposure in coal preparation plants due to vibrating screens, NIOSH, a major coal company and a vibrating screen manufacturer at a coal preparation plant conducted a cooperative study of vibrating screen noise.

The consortium selected a coal preparation plant for the study based on its proximity to all of the involved researchers. In the selected plant, preliminary measurements were made in the area surrounding a group of eight horizontal vibrating screens used to process clean coal. These measurements indicated that the sound levels ranged from 94 to 98 dB(A) with the plant processing coal (see Fig. 1) and from 89 to 97 dB(A) without the plant processing coal and the screens operating with vibration only. The sound levels decrease significantly with increasing distance from the screens, indicating that the screens dominate the overall A-weighted sound level in this area of the preparation plant. To reduce the potential for overexposing preparation-plant workers to noise, the collaborators agreed to test several commonly used engineering noise controls for vibrating screens. In addition, the collaborators initiated a research effort to rank the noise sources on one of the horizontal vibrating screens and to develop engineering noise controls to address the dominant noise sources.
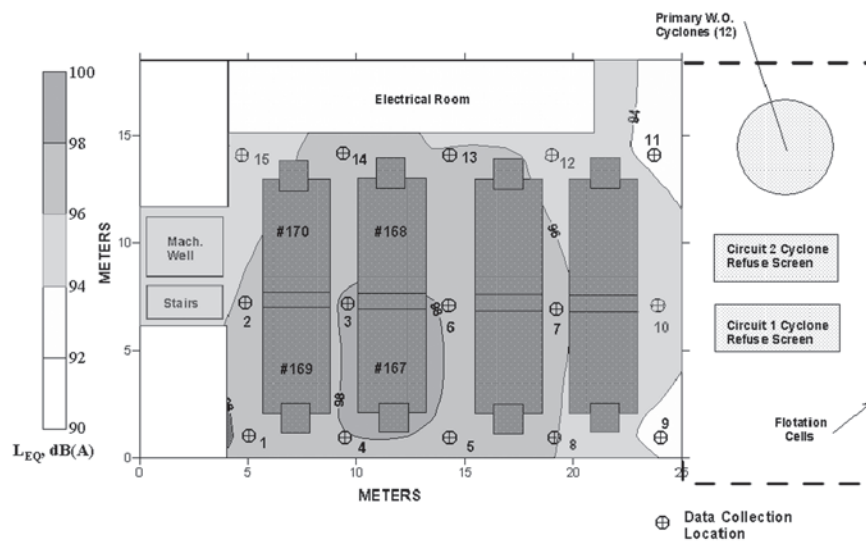
Figure 1 — Contour plot of the A-weighted sound levels around a group of eight clean coal screens.
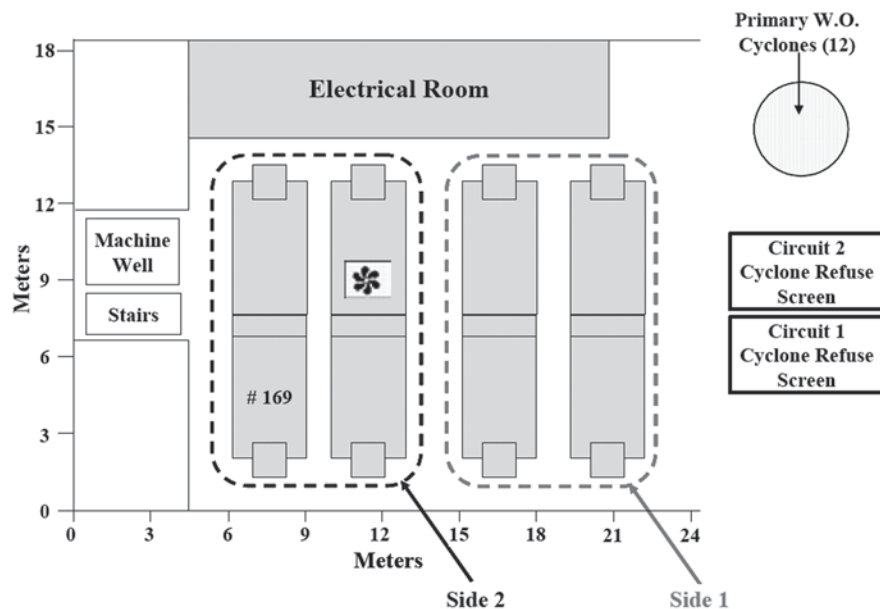
Figure 2 — Overhead view of test area.

Experimental procedures

After the plant operation, test options and available screens were discussed, a single 2.4- x 4.9-m (8- x 16-ft) horizontal vibrating screen used to drain and rinse a 1 x 10-mesh cyclone clean coal product was selected as the test screen. The test screen, Screen 169, operates within a group of eight similar screens as shown in Fig. 2. The plant is designed such that the eight screens are split into two groups of four screens, referred to as Side 1 and Side 2. The sound levels at any location within the test area are influenced by each of the eight screens. Ideally, the sound level around the test screen could be measured with only the test screen processing coal. However, the seven other screens cannot be turned off while

processing coal on only the test screen. All of the screens from either Side 1 or Side 2 can be turned off while processing coal. In addition, one screen from Side 1 or Side 2 can be turned off while processing coal with the remaining screens. Therefore, without some means of reducing the contribution of the other screens to the sound level around the test screen, evaluating the sound levels due to the test screen and the noise reductions due to installing engineering controls would be impossible.

To reduce the background sound levels around the test screen due to the nearby plant machinery, quilted barrier-absorbers consisting of two fiberglass layers with a loaded vinyl septum and covered in a nylon material were hung from the plant support beams around the test screen (see Fig. 3). The barriers were attached to the support beams of the above floor with bolts through grommets. The barriers were sized such that they touched the floor when installed. The seams between individual barrier strips were sealed with Velcro, and gaps around pipe penetrations were kept to a minimum. The above steps were necessary to achieve the maximal reduction in the background noise near the test screen.

Six microphones were positioned at a height of 1.5 m (5 ft) and a distance of 0.6 m (2 ft) from a reference box surrounding the test screen (see Fig. 4). Four microphones were positioned along the left side of the test screen with two microphones behind the feed chute. Due to the proximity of the barrier to the right side of the test screen, it was not possible to position microphones along the right side of the screen. Microphones could not be positioned at the discharge end of the screen between the test screen and Screen 170 due to the screen design. The data-acquisition equipment was positioned in the corner of the building as shown in Fig. 5.

A series of sound-level measurements was performed to assess the impact of the barrier-absorbers on the background noise near the test screen. Sound-level measurements were performed with the plant processing coal with the test screen off. Next, measurements were performed with the screens in Side 1 processing coal and Side 2 off. These conditions represent the background sound levels that would be observed during testing. The barriers were then installed and the measurements were repeated. The measurements indicated that the background sound level was reduced by more than 11 dB(A) with the plant processing coal with the test screen off and by more than 7 dB(A) with the screens in Side 1 processing coal and the screens in Side 2 off. These results show that the barrier was effective in reducing the background noise and would enable the researchers to evaluate noise controls and to study the noise generated by the test screen.

Figure 3 — Barrier-absorber curtain between Screens 169 and 167 showing tight seal around piping.

Figure 5 — Data acquisition equipment location and microphones along left side of Screen 169.
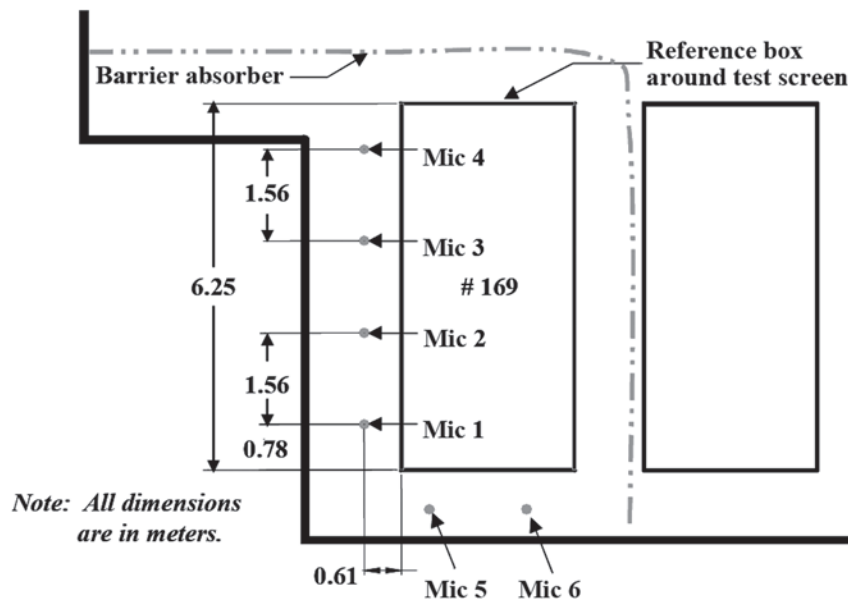
Figure 4 — Overhead view of microphone positions around test Screen 169 showing barrier-absorber curtain location.

Trial testing of commonly used engineering noise controls

In the beginning of the project, the coal company wanted to test several engineering noise controls that are commonly applied to vibrating screens: rubber isolation mounts, urethane screen panels and a urethane sieve. Prior to all testing, a new set of steel coil springs was installed to ensure the baseline sound levels would not be affected by broken or worn springs. Next, rubber isolation mounts replaced the new steel coil springs. After the tests with the rubber mounts were performed, the standard steel screen panels were replaced with urethane screen panels. After examining the test data, it was determined that the coal/water flow was a secondary noise source and the urethane siev would, therefore, not have an impact on sound levels for this screen.

Steel coil springs vs. rubber isolators. After the barrier-absorber curtain was in place, sound-level measurements were performed with new steel coil springs installed. To determine the effect of switching to rubber isolators on building vibration, accelerometers were attached to the screen at the steel coil spring locations on both the screen side and building side of the springs to measure accelerations in both the fore/aft (x) and vertical (z) directions as shown in Fig. 6. Figure 7 shows the accelerometers at one mounting location with the steel coil springs (upper picture) and rubber isolators (lower picture).

First, the sound level due to the test screen processing coal was determined. With the barrier in place and the new steel coil springs installed, the sound levels around the screen and the vibration at the screen mounts were measured with the test
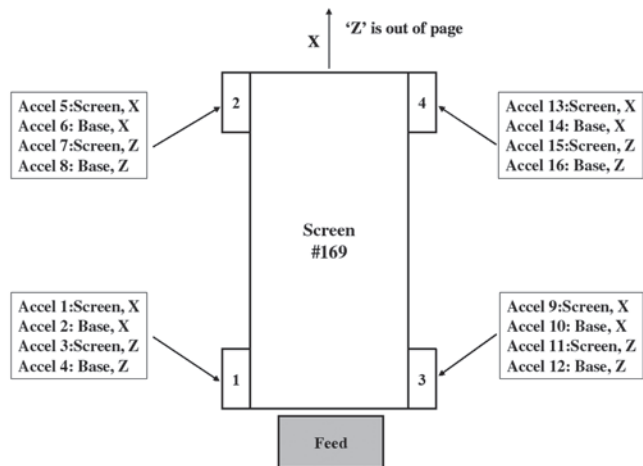
Figure 6 — Accelerometer locations used during steel coil spring and rubber isolator testing.
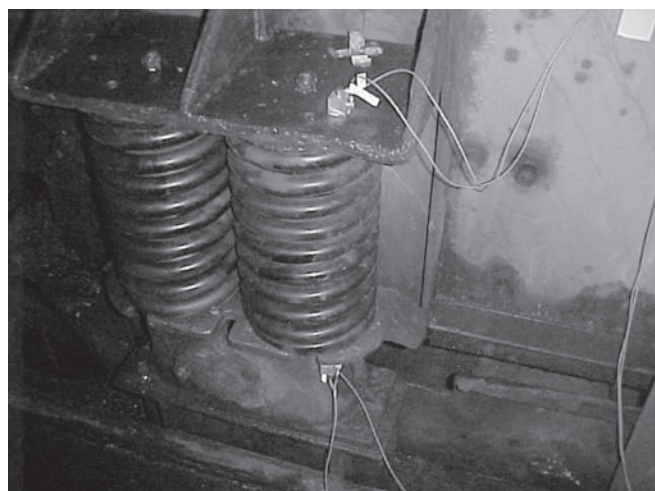
Figure 7 — Accelerometers affixed to screen and base at isolator mount locations.

screen off and all other screens processing coal. The test screen was then turned on and the measurements were repeated with all screens processing coal. The sound level produced due to the test screen processing coal could then be determined by

$$L_{P,w/caol} = 10\text{LOG}\left(10^{0.1L_{P,2}} - 10^{0.1L_{P,1}}\right) \quad (1)$$

where

$L_{P,w/coal}$ is the background-noise-corrected sound level due to the test screen processing coal,

$L_{P,1}$ is the sound level measured with the test screen off and the seven other screens processing coal and

$L_{P,2}$ is the sound level measured with all of the screens processing coal.

It should be noted that all of the equations presented in this report were used to calculate both overall and 1/3-octave band sound levels.

Next, sound-level measurements were performed with Side 1 processing coal and Side 2 off. The test screen, Screen 169, was then operated with vibration only with the Side 1 screens processing coal and the sound level measurements were repeated. The background-noise-corrected sound levels due to the test screen for vibration only operation were then calculated by

$$L_{P,vib} = 10\text{LOG}\left(10^{0.1L_{P,4}} - 10^{0.1L_{P,3}}\right) \quad (2)$$

where

$L_{P,vib}$ is the background noise corrected sound level due to operating the test screen with vibration only;

$L_{P,3}$ is the sound level measured with the Side 1 screens processing coal and the Side 2 screens off; and

$L_{P,4}$ is the sound level measured with the Side 1 screens processing coal, the test screen operating with vibration only and the other screens in Side 2 off.

Similar procedures were followed throughout the study when sound level measurements had to be taken with the non-test screens operating.

Figures 8 and 9 show the background-noise-corrected A-weighted 1/3-octave band sound levels due to the test screen for full operation and vibration only, respectively. The figures show that using rubber isolators reduced the overall sound level by 1 dB(A) for full operation and 2 dB(A) for vibration only operation. The figures show that small reductions occur throughout the 100 to 10 kHz frequency range.

When altering the suspension of a vibrating screen, the effects of the modifications on both the sound levels and the building vibration are important considerations. Vibration measurements at the mounting locations with the steel coil springs and rubber isolators were compared to examine the vibration associated with noise radiation and building vibration. Vibrations at frequencies above 20 Hz are significant in terms of noise radiation. In addition, spikes in the vibration time history are indicative of impacts that would generate significant noise levels. With regard to building vibration, vibrations at the eccentric mechanism rotation frequency, 15 Hz in this case, are most important.

To examine the vibration associated with noise radiation, the time waveform of the accelerometer signal on the building side of the mounting location highlighted in Fig. 10 was compared for the steel coil springs and rubber isolators.

Figure 11 shows the time waveform with the steel coil springs and with the rubber isolators. The figure shows that 15 impacts per second occurred with the steel coil springs. This impact rate corresponds with the rotation frequency of the vibration mechanisms. The impacts cause peak-to-peak high-frequency vibrations on the order of 75 g at this location. Similar results were observed at the other mounting locations.

The impacts could be due to contact between an inner and outer coil spring, a coil spring and its positioning disk, a friction check and the screen, or the screen and a fixed object such as a chute, pipe, or support structure. These impacts cause the screen and its support structure to vibrate and radiate noise. Therefore, these impacts must be eliminated to reduce screen noise in the most efficient manner.

The accelerations in the 16 Hz 1/3-octave band were compared for the steel coil springs and the rubber isolators, because this frequency band contains the mechanism rotation frequency. These low frequency vibrations can be a concern with respect to building vibration.

Figure 12 shows the percentage change in the accelerations in the 16 Hz 1/3-octave band for the fore/aft (x) and vertical (z) directions on both the building (base) and screen side of each mount. The figure shows that the building vibration at the left mount on the discharge end and the right mount on the feed end of the screen increased substantially. Subjectively, an increase in the building vibration could be felt in the corner of the building where the acquisition equipment was located. The low-frequency vibration with the rubber isolators was significantly higher than with the steel coil springs, even though the high-frequency vibrations were lower with the rubber isolators.

The increase in building vibration is probably due to a difference between the system spring rates with the rubber isolators and the steel coil springs. With the steel coil springs, two sets of inner and outer springs are used at each mount at the discharge end whereas one inner and outer spring and a third spring are used at each mount on the feed end. However, with this arrangement, the product of the feed end spring rate with the distance from the feed end mounting location to the screen center of gravity is equal to the product of the discharge end screen rate with the distance from the discharge end mounting location to the screen center of gravity. In this case, the translational and rotational motions of the screen are decoupled so the screen motion is primarily translational in the direction of the force applied by the eccentric mechanisms.

When the rubber isolators were installed, two isolators were used at each mounting location at the discharge end and a single isolator was installed at each mounting location at the feed end. The spring rates of all of the rubber isolators were equal. With this arrangement, the product of the total spring rate at each screen end and the distance from the screen center of gravity to the mounting locations is not equal. This creates coupling between translational and rotational screen motions, causing the screen to translate in the direction of the force applied by the eccentric mechanisms and to rotate with a
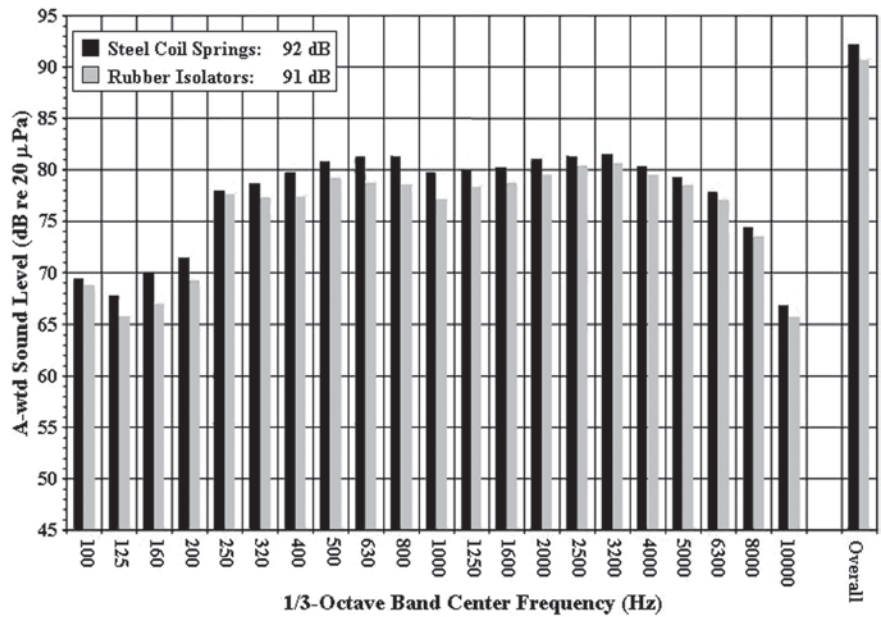

Figure 8 — A-weighted 1/3-octave band sound levels for steel coil springs vs. rubber isolators, full operation.
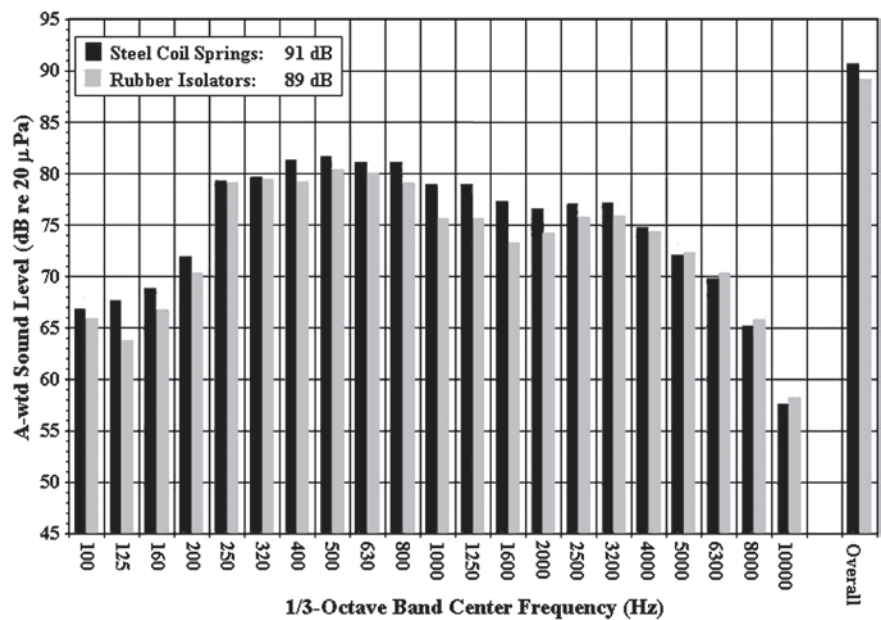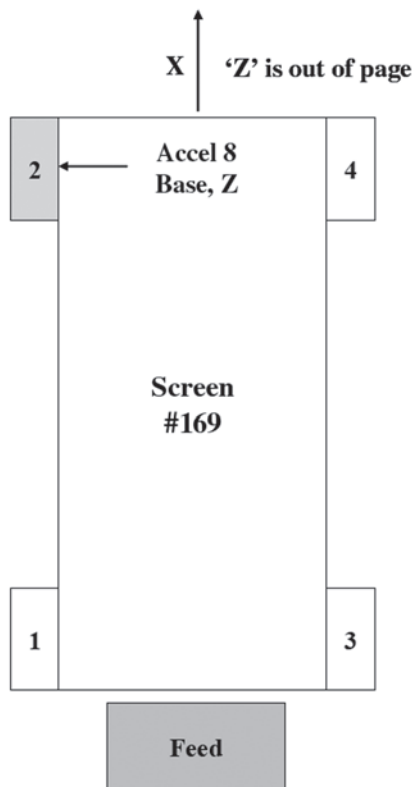

Figure 9 — A-weighted 1/3-octave band sound levels for steel coil springs vs. rubber isolators, vibration only.

pitching motion about the screen center of gravity. This coupling tends to increase the vibration transmitted to the building. If the spring rates at the discharge and feed ends of the screens with rubber isolators could be matched to those with the steel coil springs, the steel-on-steel impacts observed with the steel coil springs could be eliminated without increasing the building vibration.

Standard steel screen panels vs. urethane screen panels. Following the tests with the rubber isolators, the steel screen panels were replaced with urethane screen panels. Sound level measurements were then performed with the screens process-

Figure 10 — Location used to compare the acceleration time history with steel coil springs and rubber isolators.

ing coal. Figure 13 shows the background-noise-corrected A-weighted 1/3-octave band sound levels with the standard and urethane screen panels. A slight increase in sound level was observed. However, the increase was only a few tenths of a decibel, which is within measurement error. The figure indicates that the urethane screen panels did not reduce the sound level around the test screen in this case. These results
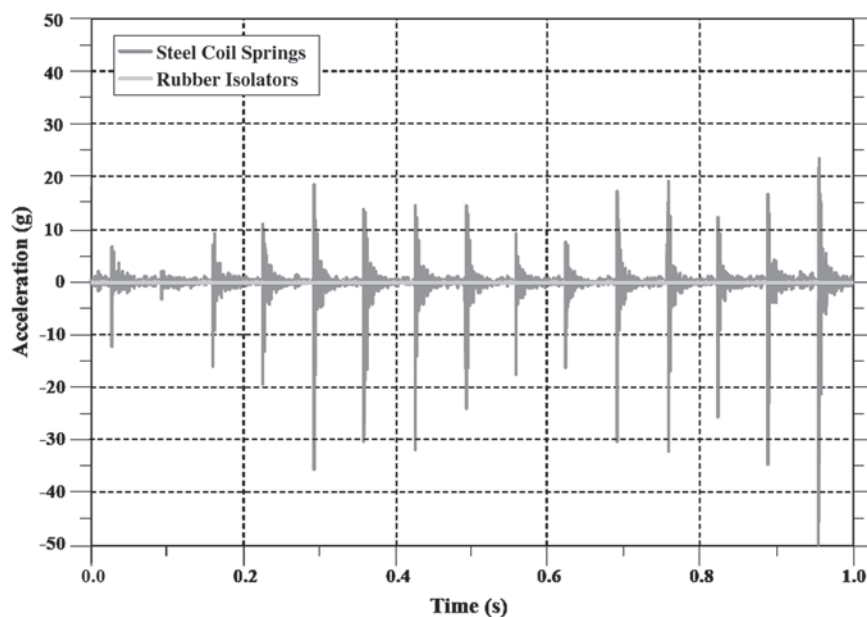
Figure 11 — Acceleration in Z-direction at Mount 2 with steel coil springs and rubber isolators.

are consistent with the findings of a 1977 report to the U.S. Bureau of Mines, which states that particle size must ordinarily be greater than 12-mm diameter before coal impact noise exceeds 90 dB(A) near a screen (Rubin, 1997). Generally, coal flow noise is dominant in screens handling coarse coal, while noise due to the drive mechanism, which may be radiated by the screen body or mechanism housings, is the most significant noise source in screens handling finer coal (Ungar et al., 1976).

Ranking of screen noise sources

The first phase of the study involved a trial-and-error approach to reducing the noise generated by the test screen. In some cases, this approach will result in finding a successful noise control. However, in many cases, the attempted controls will not reduce sound levels because the controls do not directly address the primary noise sources on a piece of equipment. Noise controls that address the dominant noise sources on machinery are the most effective means of reducing noise. Therefore, the next phase of the project consisted of rank ordering the noise sources on the screen.

Contributions of major noise sources. Figure 14 shows the A-weighted 1/3-octave band sound levels for the test screen with the steel coil springs and standard steel return deck while processing coal. The figure shows that the sound level in the 16 Hz 1/3-octave band, which contains the mechanism rotation frequency, is unimportant in terms of the overall A-weighted sound level. The 100 Hz through 10 kHz 1/3-octave bands, however, account for 99% of the overall A-weighted sound level under full operation. Therefore, emphasis was placed on this frequency range in determining the contributions of each noise source on the screen.

A vibrating screen is a complex piece of machinery. However, the noise radiated by a screen is primarily due to two noise sources: screening noise and drive noise (Hennings, 1980). Screening noise consists of the noise generated by the flow of the coal/water mixture down the chute and across the top of the screen deck. As the mixture flows out of the chute and across the screen, impacts between individual pieces of coal and between the mixture and the chutes and screen generate noise. Drive noise refers to the noise radiated due to vibration of the mechanism housings, screen sides and the building, resulting from excitation by the gears, bearings and eccentric weights of the mechanisms. Generally, screening noise is more significant than drive noise for coarse coal screens, while drive mechanism noise is dominant for fine coal screens (Ungar et al., 1976). For drain and rinse screens, such as the test screen in this study, the spray of rinse water onto the processed coal is another potential noise source.

To determine the relative contribution of these sources, several measurements were performed. First, the sound levels around the screen were measured with the test screen processing coal. In this instance, all of the aforementioned noise sources make a contribution to the measured sound levels. Next, the sound levels were measured with the screen vibrating without processing coal. This yields the sound level

due to drive noise. Finally, the sound levels were measured with the rinse water spray on and the screen turned off. The above measurements may be used to calculate the sound level due to screening noise. In these calculations, the assumption was made that the damping and/or mass-loading effects of the flow of the coal/water mixture across the screen has little impact on the noise generated by drive noise.

The total sound level produced by the screen may be calculated by

$$L_{P,tot} = 10\text{LOG}\left(10^{0.1L_{P,scr}} + 10^{0.1L_{Pdr}} + 10^{0.1L_{p,rin}}\right) \quad (3)$$

where
$L_{p,tot}$ is the total sound level,
$L_{p,scr}$ is the sound level due to screening noise,
$L_{p,dr}$ is the sound level due to drive noise and
$L_{p,rin}$ is the sound level due to rinse water spray.

Because the first set of measurements consists of the total screen noise and the second and third measurements consist of the drive noise and the rinse spray noise, the contribution of screening noise to the total sound level can be calculated by rearranging Eq. (3) to yield

$$L_{P,scr} = 10\text{LOG}\left(10^{0.1L_{P,tot}} + 10^{0.1L_{Pdr}} + 10^{0.1L_{p,rin}}\right) \quad (4)$$

Drive noise was found to be the most significant contributor to the total screen noise, yielding a sound level of 91 dB(A). The sound level resulting from the rinse-water spray was found to be only 80 dB(A), which is insignificant in relation to the total sound level around the screen. Using Eq. (4), the sound level due to screening noise was calculated to be 87 dB(A). Table 1 shows a summary of the A-weighted sound levels due to each individual noise source as well as the total sound level.

The A-weighted 1/3-octave band sound levels for the total screen noise, drive noise, rinse-water spray noise and screening noise are shown in Fig. 15. The figure shows that drive noise is dominant in the 100 through 1,600 Hz 1/3-octave bands, whereas screening noise is dominant above the 1,600 Hz 1/3-octave band. Therefore, the most effective approach to reduce the sound level around the screen would be to first address the drive noise in the 100 through 1,600 Hz 1/3-octave bands. Once the sound level due to drive noise is reduced to 87 dB(A), drive noise and screening noise would be equal contributors to the sound level around the screen during full operation. At this time, a decision would have to be made regarding which of these two noise sources would be most easily addressed to further reduce sound levels, if necessary.
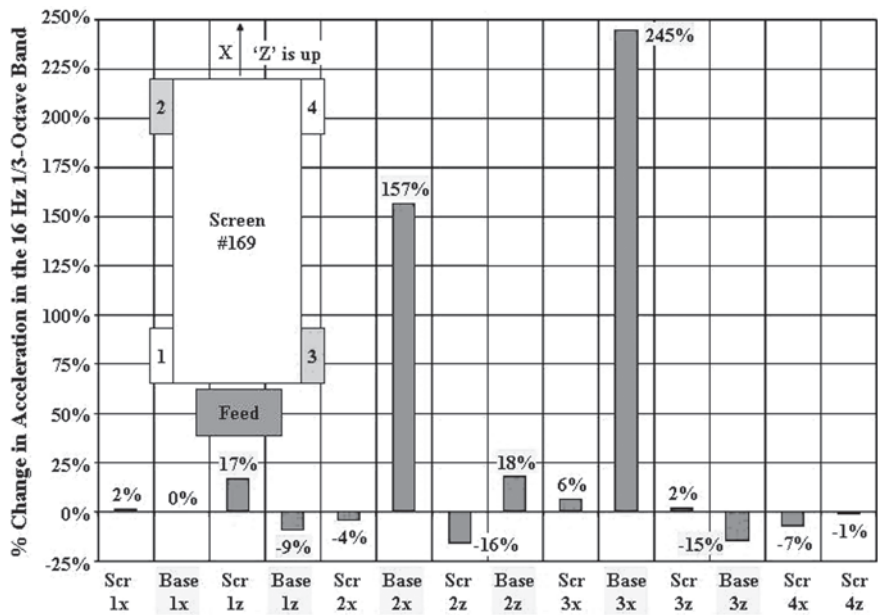

Figure 12 — Percentage change in the acceleration in the 16 Hz 1/3-octave band due to replacing steel coil springs with rubber isolators.
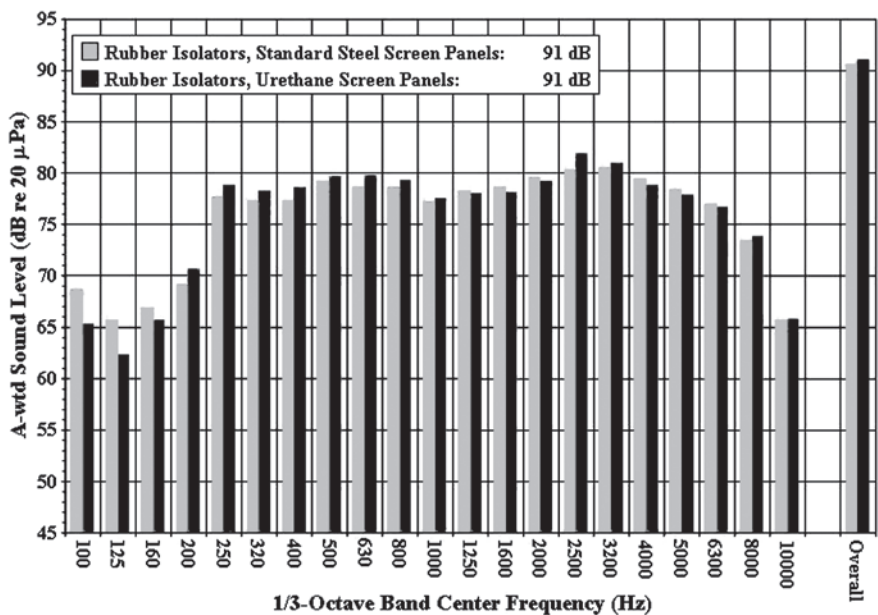

Figure 13 — A-weighted 1/3-octave band sound levels while processing coal with standard steel screen panels or urethane screen panels and rubber isolators coal.

Contributors to drive noise. Because drive noise was determined to be the dominant noise source on the screen, further testing was performed to examine the drive noise. Drive noise can be separated into the noise radiated by the screen sides and the noise radiated from the housings of the eccentric mechanisms. To identify noise controls that will best reduce drive noise, the relative contributions of the screen sides and mechanism housings to drive noise must be determined. The total sound level due to drive noise is

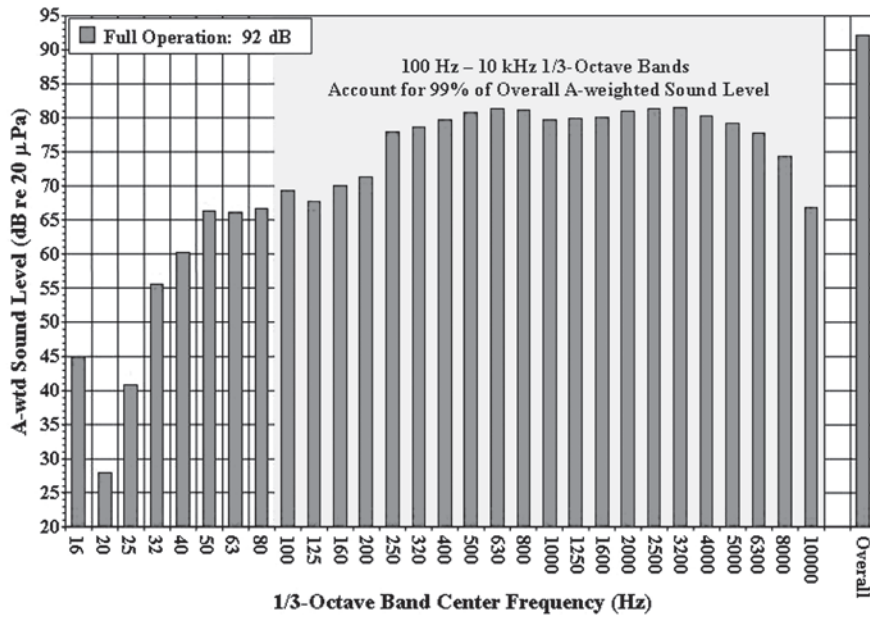$$L_{P,drive} = 10\text{LOG}\left(10^{0.1L_{P,side}} + 10^{0.1L_{p,mech}}\right) \quad (5)$$

Figure 14 — A-weighted 1/3-octave band sound levels for test screen while processing coal.
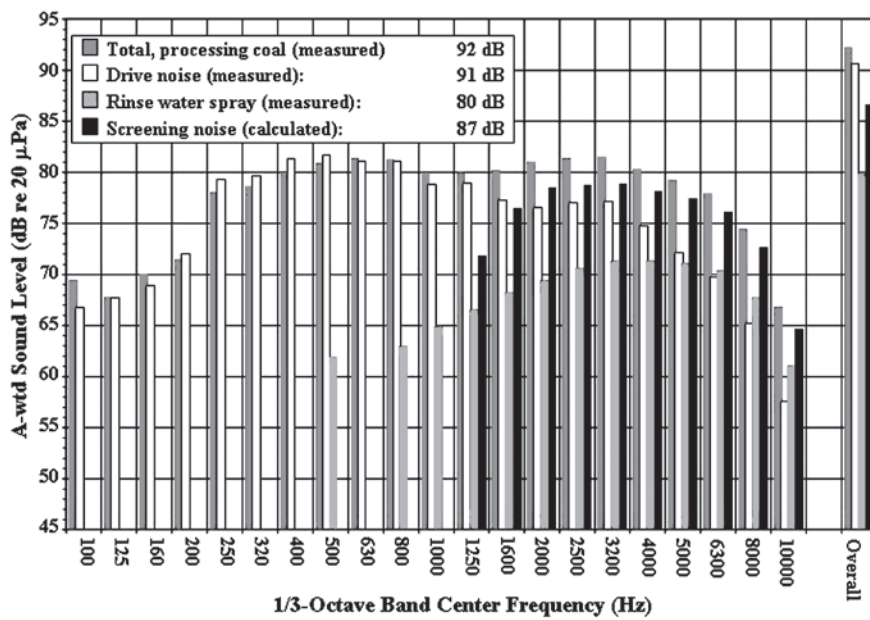

Figure 15 — A-weighted 1/3-octave band sound levels for the total screen noise, drive noise, rinse water spray noise, and screening noise.

Table 1 — Summary of noise sources and their sound levels with steel coil springs and steel deck.

Test condition/source	Overall A-weighted sound level dB re 20 μPa
Total, processing coal (measured)	92
Drive noise (measured)	91
Rinse water spray (measured)	80
Screening noise (calculated)	87

where

$L_{p,side}$ and $L_{p,mech}$ are the vibration-radiated sound levels due to screen side vibration and mechanism housing vibration, respectively.

A series of sound-level measurements was performed to determine the sound levels due to screen side and mechanism housing vibration. Because the goal was to determine contributors to drive noise, these measurements were performed with vibration only on a day when the plant was not operating.

For these tests, the barrier-absorber curtains used to block airborne background noise from the nearby screens at the beginning of the study were taken down. For the first test, the eccentric mechanism housings were wrapped with quilted lead-fiberglass barrier-absorbers. The goal of wrapping the mechanisms was to reduce the contribution of the noise radiated by the mechanism housings by at least 10 dB, thereby enabling the measured levels to be treated as the sound levels due to screen side vibration only. Next, the barrier-absorber wrap was removed to measure the contributions of both screen side and housing vibration to the drive noise. The sound level due to mechanism housing vibration was then calculated by rearranging Eq. (5) to yield

$$L_{P,mech} = 10\mathrm{LOG}\left(10^{0.1L_{P,drive}} + 10^{0.1L_{p,side}}\right) \quad (6)$$

Figure 16 shows the A-weighted 1/3-octave band sound levels due to screen body and mechanism housing vibration. The sound levels due to screen body and mechanism housing vibration were found to be 87 and 84 dB(A), respectively. Because a 3-dB difference in levels indicates a doubling of energy, screen side vibration can be said to have twice the significance of mechanism housing vibration in terms of drive noise. The spectra indicate that the noise radiated by the screen body is dominant in the 100 through 500 Hz 1/3-octave bands. The screen sides account for 83% of the sound energy for this band. In the 630 through 1,600 Hz 1/3-octave bands, the figure shows that the screen sides and mechanism housings generate similar sound levels with each contributing approximately 50% to the drive noise in this frequency range. In the 2 through 10 kHz 1/3-octave bands, the sound levels due to mechanism housing vibration are higher than those associated with screen side vibration. Mechanism housing vibration accounts for 58% of the drive noise sound level in this frequency range.

Reducing vibration-radiated noise

To reduce the noise radiated by the vibrating screens in the most efficient manner, the dominant noise source must be addressed first. The noise source ranking tests showed that

drive noise was the most significant noise source. Therefore, the noise radiated with the screen operating without coal and with vibration on should be reduced prior to attempting to reduce screening noise.

Rebuilt mechanisms. Because drive noise was determined to be the dominant noise source on the screen, several methods of reducing drive noise were considered. The first idea that was discussed was replacing the existing set of mechanisms, which were approximately 5 years old, with rebuilt mechanisms. It was thought that wear of the internal gears and bearings over time could have increased the vibration generated by the mechanisms at audible frequencies (above 20 Hz). These vibrations are then transmitted to the screen sides via the H-beam, causing the screen sides to radiate noise. Therefore, replacing the mechanisms could potentially reduce the drive noise radiated by both the mechanism housings and screen sides.

Baseline sound levels were measured with the existing set of mechanisms, the rubber isolators and the urethane screen panels installed on the test screen with the plant shut down and the test screen operating with vibration only. The mechanisms were then replaced with rebuilt mechanisms and the measurements were repeated. Figure 17 shows the 1/3-octave band sound levels with the original and rebuilt mechanisms. The figure indicates that installing the rebuilt mechanisms reduced the overall A-weighted sound level by 1 dB. According to the screen manufacturer and the coal company, installing rebuilt mechanisms costs approximately $10,000. The results suggest that the cost associated with installing a set of rebuilt mechanisms is high relative to the observed sound level reduction.

Due to the increased building vibration with the rubber isolators, the mounts were switched back to the steel coil springs. In addition, the urethane screen panels were removed and standard steel screen panels were installed, because the urethane screen panels did not reduce the sound levels and were prone to plugging.

Figure 16 — A-weighted 1/3-octave band sound levels for noise radiated by screen body and mechanism housings.

Figure 17 — A-weighted 1/3-octave band sound levels for vibration only operation with original and rebuilt mechanisms.

After approximately one year, a new set of measurements was performed to establish the baseline sound levels for the screen operating with vibration only with the steel coil springs and steel deck. The measurements were performed on a maintenance day with the plant shut down. At this time, an unexpected result was observed. The vibration only sound levels for the test screen with the steel coil springs and rebuilt mechanism decreased to 87 dB(A), a 4-dB reduction compared to the original measurements from Feb. 25, 2003 Figure 18 shows a comparison of the vibration only sound levels with steel coil springs, steel screen panels and the
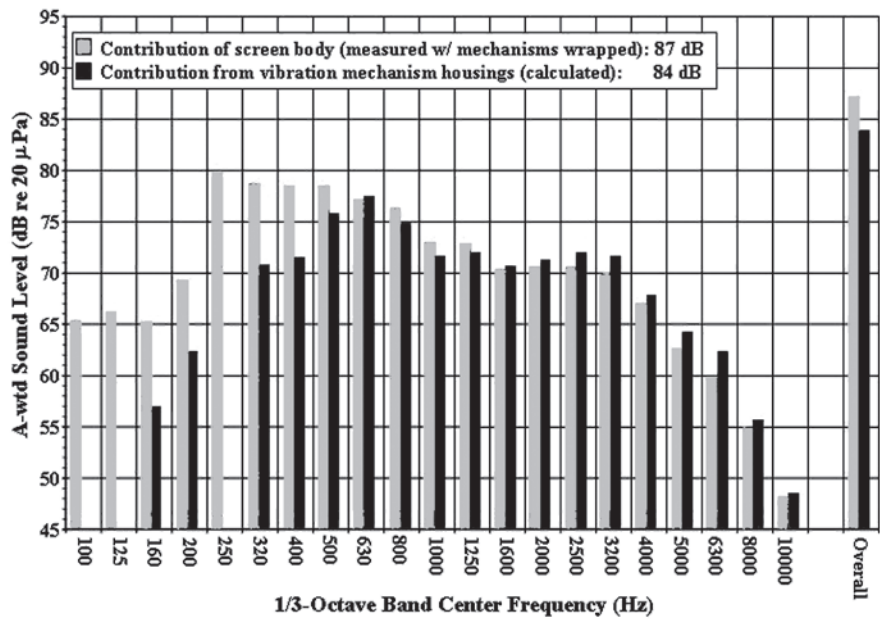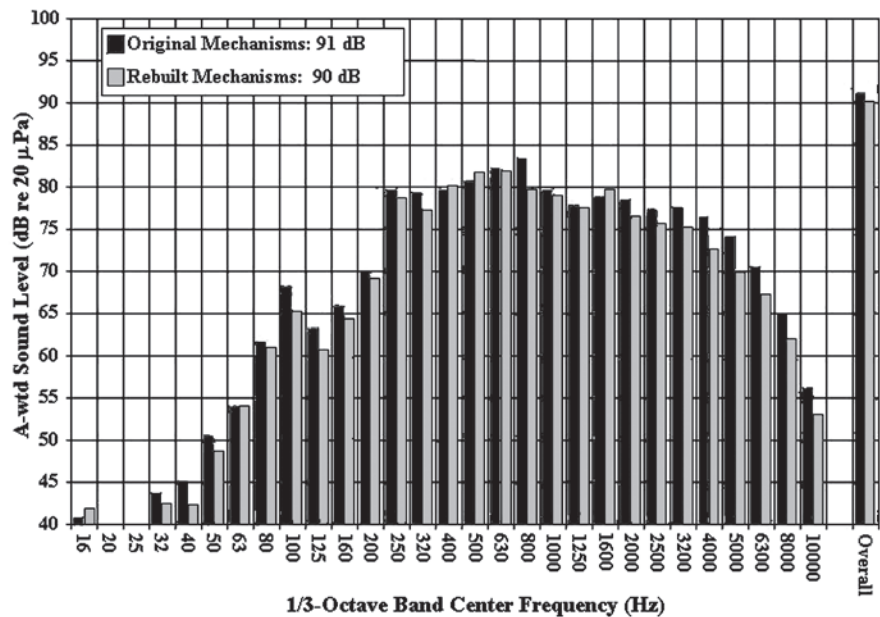
original mechanisms; rubber isolators, urethane screen panels and the original mechanisms; rubber isolators, urethane screen panels and the rebuilt mechanisms; and steel coil springs, steel screen panels and the rebuilt mechanisms. The test dates for these data were Feb. 25, 2003, July 12, 2003, July 18, 2003, and July 23, 2004, respectively.

The figure shows that the average sound level measured on Feb. 25, 2003, with new coil springs, the steel return deck and the original mechanisms was 91 dB(A). When testing was conducted within a one-week period (July 12, 2003, and July 18, 2003) with the original versus rebuilt mechanisms with
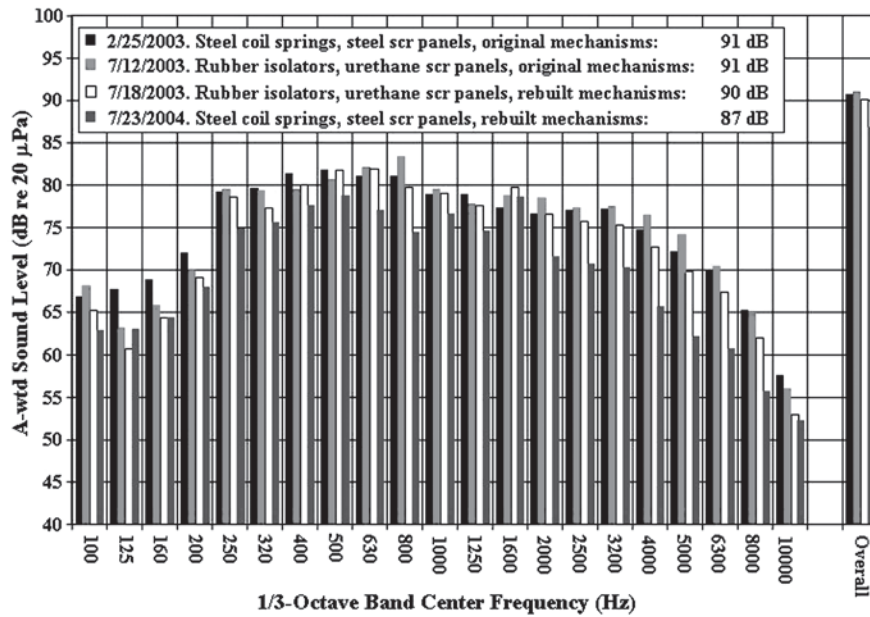
Figure 18 — Comparison of sound levels with original or rebuilt mechanisms, steel coil springs or rubber isolators, and steel or urethane screen panels for several test dates.
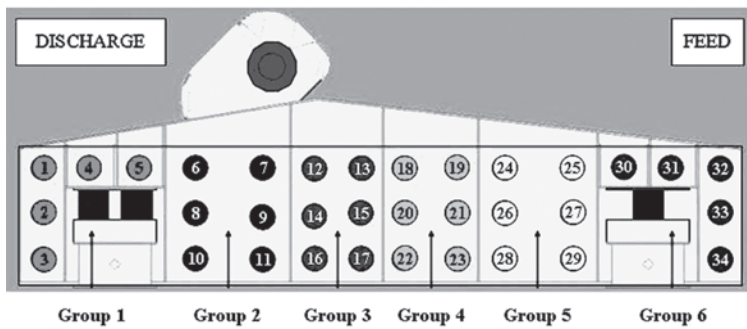

Figure 19 — Approximate accelerometer locations used for vibration measurements on the left side of the screen.
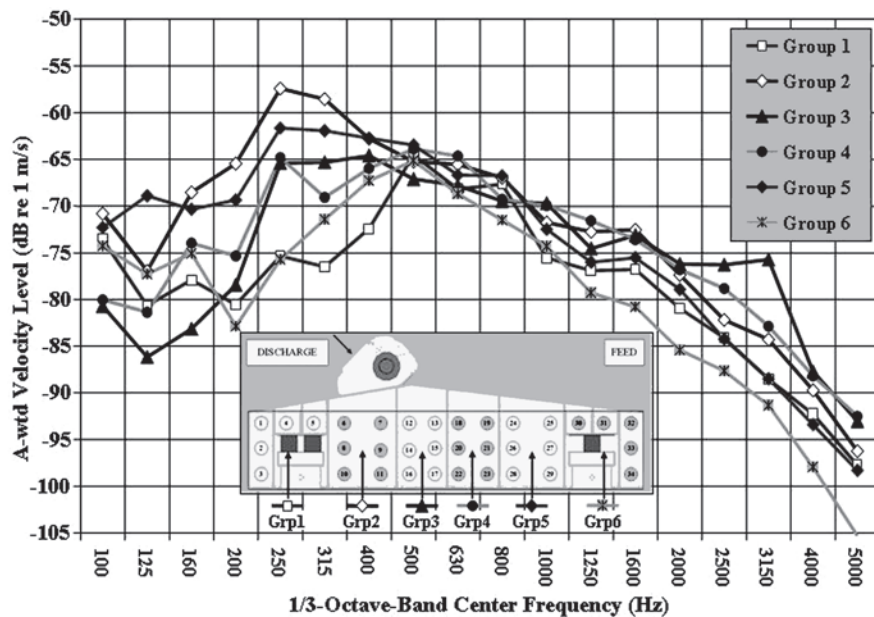

Figure 20 — A-weighted surface-averaged velocity levels.

rubber isolators and urethane screen panels, the sound levels were 91 and 90 dB(A), respectively. However, after the steel coil springs replaced the rubber isolators and the steel screen panels replaced the urethane screen panels and one year had passed since the installation of the rebuilt mechanisms, the sound level for vibration only operation was 87 dB(A).

On the reinstallation of the steel coil springs, vibration measurements were taken at the spring mounting locations to determine the cause of the steel-on-steel impacts observed in the original data (refer to Fig. 11). The steel-on-steel impacts were not observed at this time. It is possible that the original test data were influenced by an installation problem with the springs, which would increase the sound level around the screen. It is also possible that during the year that had passed, any high spots on the gears in the rebuilt mechanisms had worn away resulting in smoother tooth engagement. In addition, with the rubber isolators, more vibration was transmitted to the building, which could have increased the drive noise. Proper installation of the steel coil springs may have eliminated the steel-on-steel impacts and reduced the vibration transmitted to the building, thus decreasing the noise radiated by the screen and the building.

Damped side plates and mechanism cover. Because screen side vibration was determined to be the most significant source of drive noise, constrained layer damping was applied to the screen sides. Constrained layer damping involves bonding a viscoelastic material to the vibrating object using a stiff adhesive and then bonding a stiff constraining layer to the viscoelastic material. When subjected to vibration, the constraining layer places the viscoelastic material in shear. Shearing the viscoelastic material converts the vibration energy into a small amount of heat, thereby reducing vibration and, hence, radiated noise.

Prior to constructing the damped side plates, a series of vibration measurements was performed on the left side of the screen to determine the relative vibration levels along the side of the screen. The goal was to determine if constrained layer damping was necessary on the entire side plate or if some areas, which had insignificant vibration levels, could be left untreated. The left side was divided into six areas and a group of accelerometers was attached to each area as shown in Fig. 19.

The test screen was operated with vibration and without coal. Because noise radiation is directly related to the surface-

averaged velocity, the resulting 1/3-octave band accelerations were integrated to yield vibratory velocities, and the surface-averaged A-weighted velocity levels for the six areas were calculated (see Fig. 20). A-weighting was applied to the vibratory velocities to deemphasize low frequency vibration, which is not a significant contributor to the A-weighted sound level in this case. The figure shows that the velocity levels are highest in the 100 through 400 Hz 1/3-octave bands for the Group 2 and Group 5 accelerometers. However, above the 500 Hz 1/3-octave band, the variation in vibration levels from group to group is generally less than 10 dB. Therefore, each area is a significant contributor to the drive noise radiated by the screen sides and should be treated.

Constrained layer damped side plates were constructed by bonding a layer of viscoelastic damping material to new side plates using epoxy. A 3.2-mm- (1/8-in.-) thick layer of steel was then attached to the viscoelastic material with epoxy to complete the construction. Mechanism covers were fabricated from a barrier-absorber material similar to the material used to exclude background noise from the test area during the initial tests. Prior to installing the damped side plates, baseline sound level measurements were taken during miner's vacation with the plant shut down and the test screen operating with vibration only. In addition, acceleration measurements were taken on the sides of the screen at the locations shown in Fig. 21.

The damped side plates were installed during the first week of September 2004, and sound-level measurements were collected on Sept. 14, 2004, with the mechanism covers and damped screen sides installed with the plant shut down and the test screen operating with vibration only. The mechanism covers were then removed and the measurements were repeated. In addition, acceleration measurements were performed at the locations shown in Fig. 21 for comparison with the baseline vibration measurements collected in July.

Figure 22 shows the A-weighted 1/3-octave band sound levels for baseline conditions and with the engineering noise controls installed. As the figure shows, neither the damped side plates alone nor the damped side plates and mechanism covers reduced the overall A-weighted sound level. It was expected that the sound level would be reduced by 3 dB or more. The figure shows that the sound levels in the 200 through 630 Hz 1/3-octave bands were reduced by 1 to 2 dB with the damped side plates. With the damped sides and mechanism covers, the reductions in this band were again 1 to 2 dB. In the 800 through 10 kHz 1/3-octave bands, the sound levels measured with the engineering controls installed increased from 83 to 84 dB(A). Because it is unlikely that installing these noise controls would increase the sound levels
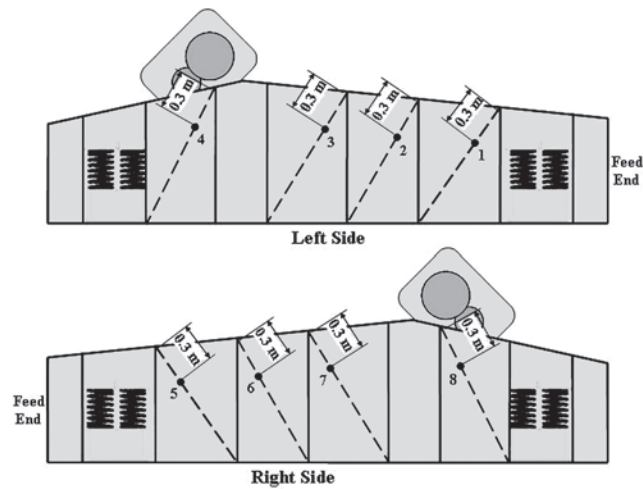
Figure 21 — Accelerometer locations used to evaluate the effects of installing constrained layer damped side plates.
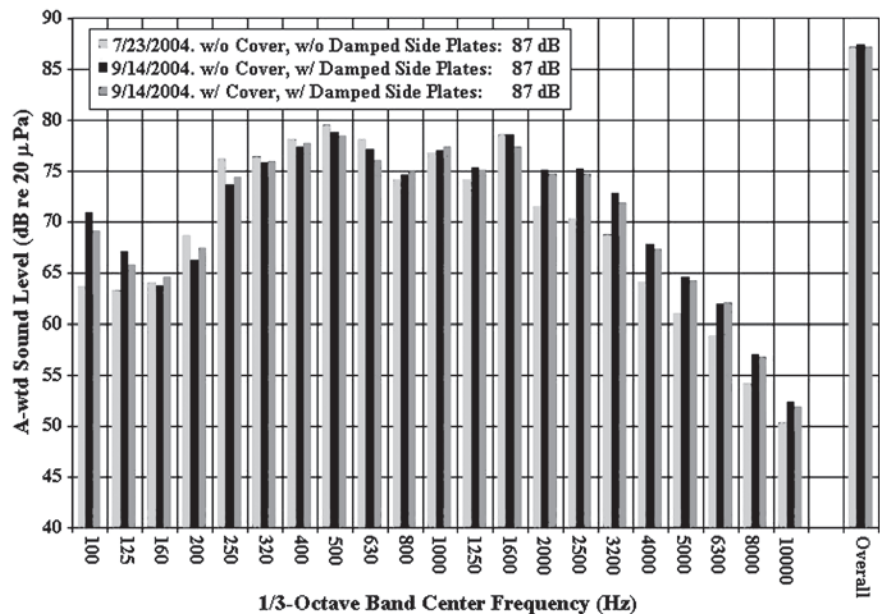

Figure 22 — A-weighted 1/3-octave band sound levels for vibration only operation for mechanism cover and damped side plate tests.

in these bands, something must have changed with the vibrating screen during the period between baseline testing in July 2004 and the testing with the noise controls in September 2004.

Further investigation of the screen revealed a possible cause for the increase in sound levels. One of the inner springs on the discharge end of the screen was broken when the damped side plates were installed. As a result, only three springs were installed on the right side of the discharge end when the damped side plates were installed. With three springs at this location, it is likely that the screen would undergo a pitching/rolling motion that could create steel-on-steel contact between the support springs or with the screen and the base. Either case would increase noise radiation.
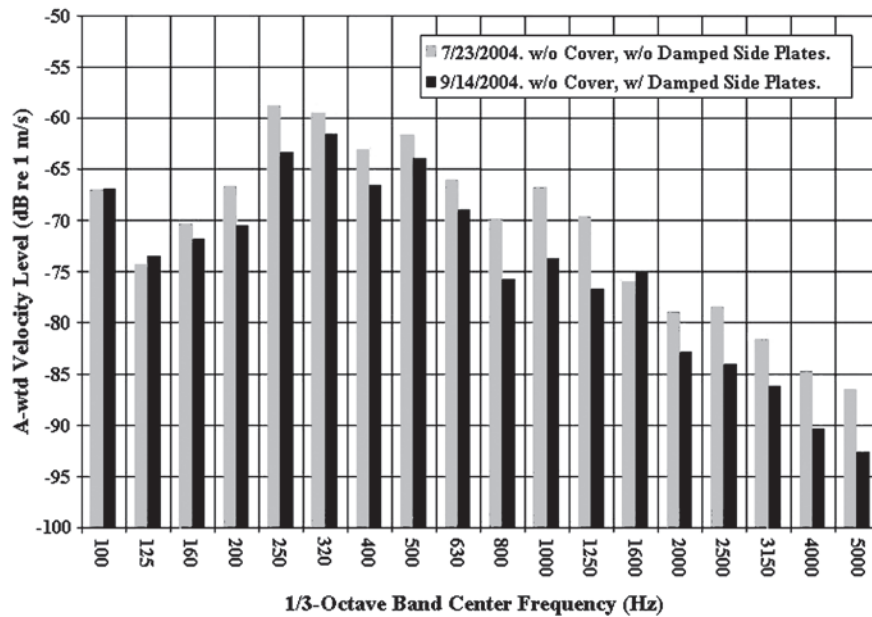
Figure 23 — A-weighted average velocity level on left side of the screen without and with damped sides.
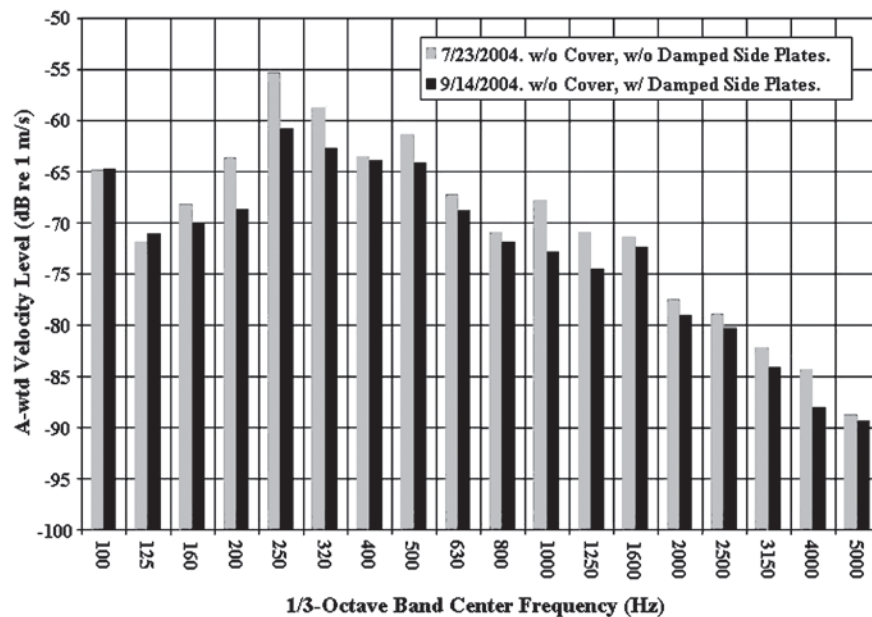
Figure 24 — A-weighted average velocity level on right side of the screen without and with damped sides.

To evaluate the effects of the damping treatment on the screen side vibration, the average A-weighted vibration velocity levels for the left and right sides of the screen were calculated from the measured accelerations at the locations shown in Fig. 21. Although the aforementioned problems with the screen installation would tend to increase the radiated noise, it was thought that the damping treatment would still reduce the screen side vibration. This is because both of the installation problems would tend to cause additional noise to be radiated from the building and from local areas more than from the screen sides. Figures 23 and 24 show the A-weighted 1/3-octave band vibration levels on the left and right sides of

the screen, respectively. Figure 23 shows reductions in the A-weighted velocity levels on the left screen side ranging from 2 to 7 dB in the 200 through 5 kHz 1/3-octave bands with the damped sides. For the right screen side, Figure 24 indicates reductions of 1 to 6 dB in the 200 Hz through 5 kHz 1/3-octave bands. In the 200 through 630 Hz 1/3-octave bands, the A-weighted vibration levels were reduced by 3 dB on the left side and 4 dB on the right side. The A-weighted vibration levels in the 800 Hz through 5 kHz 1/3-octave bands were reduced by 6 dB on the left side and 3 dB on the right side. With these substantial reductions, the A-weighted sound levels should have been reduced by 3 dB or more. More research should be performed to evaluate the effects of mechanism covers and the damped sides on the radiated sound levels.

Future work

Due to problems with unintentional changes to the test screen in the plant, evaluating the barrier-absorber mechanism covers and damped side plates in the plant proved to be difficult. The reductions in vibration levels due to the damping treatments should have produced a corresponding reduction in the drive noise, but instead the overall A-weighted sound level was not reduced. In addition, the sound levels in the higher frequencies increased significantly despite the significant reduction in vibration velocity levels at these frequencies. The variation of test conditions in the preparation plant clearly presents a problem when trying to evaluate engineering noise controls. To properly evaluate these engineering controls, the sound and vibration measurements should be performed on a similar screen at the manufacturer's location in a well-controlled environment.

Summary

Two commonly applied engineering noise controls, rubber isolators and urethane screen panels, were tested on a 2.4- x 4.9-m (8- x 16-ft) horizontal vibrating screen. Replacing the steel coil springs with rubber isolators reduced the sound level when processing coal by 1 dB(A) and the vibration-only sound level by 2 dB(A). However, switching to the rubber isolators resulted in a 157% increase in the building vibration at the left discharge end mounting location and a 245% increase in the building vibration at the right feed end mounting location. These increases are probably due to coupling of translational and rotational vibration due to a spring rate imbalance with the rubber isolators. Steel-on-steel impacts, which can substantially increase noise, were observed at the screen mounting locations with the steel coil springs during the initial testing. After switching from the steel coil springs to the rubber isolators and back to the steel coil springs, the steel-on-steel impacts that were observed during the initial testing

were not observed. It is important to note that without the steel-on-steel impacts, the sound levels with the steel coil springs and rebuilt mechanisms were significantly lower than the sound levels with the rebuilt mechanisms and rubber isolators. Because one year had passed between these tests, it is possible that the mechanisms had gone through a break-in period, reducing drive noise. The differences between the sound levels with correctly installed steel coil springs and a properly designed suspension using rubber isolators would probably be insignificant. However, this must be evaluated in a well-controlled environment. The urethane screen panels were ineffective at reducing the sound level in this instance.

A series of sound level measurements was performed to rank order the screen noise sources. Drive noise was found to be the most significant contributor. Screening noise was found to be the second most important contributor to the noise generated by the test screen. Noise resulting from the rinse-water spray was found to be insignificant. Screen side vibration was found to be the primary contributor to drive noise, while mechanism-housing vibration was determined to be the secondary contributor.

Several engineering noise controls to reduce drive noise were tested. Replacing the original mechanisms with rebuilt mechanisms was initially found to reduce the drive noise by only 1 dB(A). However, subsequent tests showed that installing the rebuilt mechanisms reduced the drive noise by 4 dB(A) compared to the initial measurements. It is possible that some type of installation problem that increased the noise radiated due to steel-on-steel impacts elevated the baseline sound levels. A series of tests should be conducted at the manufacturer's site in a well-controlled environment to compare the sound levels on a similar screen with used and rebuilt mechanisms. Installing a barrier-absorber wrap over the mechanism housings and applying constrained layer damping to the

screen sides did not reduce the drive noise. However, the damped side plates did reduce the vibration velocity levels on the screen sides between 1 and 7 dB(A). The reduced vibration levels on the screen sides should have produced a sound level reduction. It is thought that a missing steel coil spring on the discharge end of the screen influenced the results with the mechanism cover and damped screen sides. The missing spring may have increased vibration transmitted to the building or may have caused steel-on-steel impacts between the other inner and outer coil springs. Either effect could increase radiated noise beyond the baseline levels. Further investigations with the mechanism covers and damped side plates should be performed at the manufacturer's site in a well-controlled environment.

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