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DEVELOPMENT AND EVALUATION OF A URETHANE JACKETED TAIL ROLLER FOR CONTINUOUS MINING MACHINES

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ABSTRACT

Occupational noise-induced hearing loss continues to be one of the most pervasive health problems in the mining industry, despite over 25 years of regulation. One of the loudest pieces of equipment used in underground mining is the continuous mining machine. Noise sample data collected by the Mine Safety and Health Administration indicate that 42% of noise overexposures between 2000 and 2005 involved continuous mining machine operators. Previously conducted field and laboratory tests have determined that the continuous mining machine conveyor system is a dominant noise source. Loud impacts occur as the conveyor chain flight bars, used to move the mined aggregate, traverse their path from the top to the underside of the conveyor deck. Various noise control treatments have been applied to abate noise caused by the conveyor system. A durable polyurethane coating has been developed for the conveyor flight bars, resulting in a time-weighted average reduction of 3 dB(A) for an eight-hour work shift. In an attempt to further reduce continuous mining machine operator overexposures, a similar urethane coating has been applied to the tail roller component of the conveyor system. Laboratory results showed a 2 dB(A) reduction in sound power levels, but the component failed during underground durability testing. An outer steel sleeve has been added to the urethane coating of the tail roller to enhance wear resistance during mining. The urethane jacketed tail roller is the latest effort, combined with previous noise treatments, to bring the continuous mining machine into compliance with federal noise regulations.

NOMENCLATURE

L_w - sound power level
L_p - sound pressure level
a - acceleration
L_a - acceleration level
ω - frequency
$H(\omega)$ - complex frequency response function
$X(\omega)$ - frequency force input response
$Y(\omega)$ - frequency acceleration output response
$|H(\omega)|$ - FRF magnitude response
$\Phi(\omega)$ - FRF phase response

Subscripts

r - referring to the reference sound source
n - number of measurement locations
i - measurement number
rms - root mean square average
$*$ - complex conjugate
T - combined response

INTRODUCTION

Hearing loss caused by exposure to occupational noise results in a disability that is particularly severe in the mining industry. Previous studies conducted by the National Institute for Occupational Safety and Health (NIOSH)* have shown that approximately 90% of coal miners and 49% of metal/non-metal miners develop a hearing impairment by age 50 (based on an average hearing threshold level of 25 dB or greater for the 1000, 2000, 3000, and 4000 Hz frequencies), while only 10% of those who are not exposed to occupational noise experience a hearing loss by the same age [1]. In addition, NIOSH studies of 17,260 audiograms for 2871 coal miners show that 80% of coal miners have a moderate to profound high-frequency hearing loss by age 64.

The Mine Safety and Health Administration (MSHA) evaluated the effects of past legislation and issued an occupational noise exposure rule to help reduce hearing impairment in the mining industry [2,3]. This legislation, among other things, defined a Permissible Exposure Level (PEL), gave no credit for the use of personal hearing protection, and gave engineering and administrative controls supremacy to reduce worker exposure. Since the inauguration of this legislation the overall annual median noise dose for all underground coal mine workers has decreased by 24% [3]. However, the annual median noise dose for continuous mining machine (CMM) operators has remained unchanged.

CMMs are one of the principal pieces of equipment used in underground mining, and they are also one of the loudest. Sound levels around these machines can range from 78 to 109 dB(A), and operator noise dose can range from 44 to 347% [4]. Sixty-five percent of workers who were overexposed to noise operated one of seven types of equipment, according to MSHA Coal Noise Data collected from 2000 to 2002 [5]. Accounting for 35% of noise overexposures shown in Figure 1, CMMs produced the most noise overexposures of all surveyed mining equipment. A recent examination of this data shows that 42 % of all noise overexposures from 2000 to 2005 involved continuous mining machine operators [6].

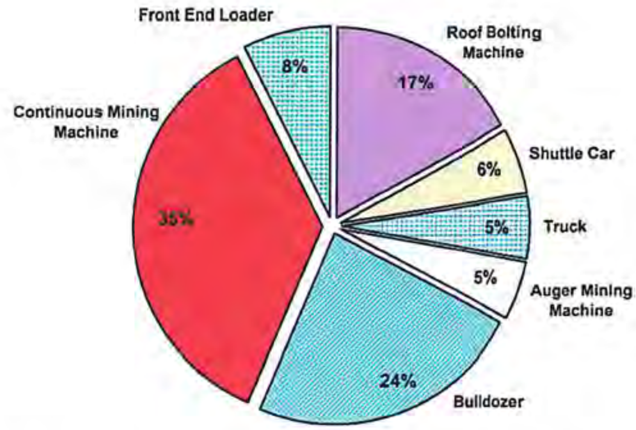

Figure 1: MSHA coal noise Sample Data - Percentage of Equipment Operators that Exceed 100% Dose

BACKGROUND

CMMs are used to cut, gather, and remove material from underground mining facilities. These machines have a series of rotating drums that are fitted with carbide cutting bits, which gouge the mine seam (known as the face). The cutting drum advances into the face and material falls to the floor. Particulate matter created from the cutting process that is not suppressed by a water spray is collected by a vane axial fan system, called a dust scrubber. The fractured pieces of coal are scooped up by the gathering arms, which are located underneath the cutting head. Steel bars perpendicular to the conveyor chain, called flight bars, span the width of the conveyor and move the aggregate to the rear of the machine. As the mined product is dumped off the end of the conveyor, the chain and flight bars are sent to the underside of the machine via the tail roller. The operator typically maneuvers the machine, by remote control, close to the end of the conveyor. Finally, the mined material is either picked up by a loader or dumped directly onto a shuttle car where it is hauled out of the mining facility for processing.

CMM noise is generated by the combined contribution of several "independent" noise sources. These sources can be categorized by machine operation and are a result of the cutting, dust collection, and conveying systems. Past studies conducted under the Bureau of Mines (BOM) examined the relative contributions of cutting and conveying noise to the overall sound level, concluding that cutting was the dominant noise source [7]. However, in these studies the average sound level was examined in the reverberant field (to simulate in-mine conditions) and did not include the sound level at the operator's position. There have also been advances in designing more efficient cutting heads since the time of these studies. The dust collection system is a relatively new CMM

* The finding 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

component that generates high sound levels. Recent studies have examined treatments applied to the vane-axial fan of the dust collection system that reduced the overall A-weighted sound level by 8 dB in a laboratory environment [8].

Of all machine operation noise sources, the conveyor system is the largest contributor to CMM noise. A CMM with an unloaded conveyor produces sound levels between 104 dB and 108 dB under dry conditions, depending on chain tension [9]. This noise source is also of particular interest because of its close proximity to where the operator is typically positioned.

Numerous research efforts conducted in the past have concentrated on reducing conveyor system noise. This noise is predominantly caused by the interaction that occurs between the chain flight bars and conveyor deck at the tail roller and gear sprocket regions. The low damping of the conveyor flight bars causes them to 'ring' when impacted. In response, a urethane coating has been developed that adds damping to the flight bars and reduces the energy transmitted to the conveyor deck. This noise control reduced the overall sound level by 5 dB(A) and exposure by a time-weighted average (TWA) of 3 dB(A) over an 8-hour work shift [9,10]. Constrained layer damping treatments have also been applied to the top, bottom, and sides of the conveyor deck to reduce impact and scraping noise caused by the flight bars [11]. Initially the constrained layer damping treatments showed a promising reduction of 2 dB(A), but after one year of operation no significant reduction was observed.

Treatments have also been applied to the tail roller of the CMM. Impact forces exerted on the tail shaft by the conveyor chain are transmitted as vibrations through the rest of the structure. Resilient materials have been placed between the chain and roller in an attempt to reduce these forces. Studies conducted under the Bureau of Mines investigated an isolated tail roller design, where an elastomer was bonded to the tail shaft and protected with a steel tube [11,12]. This treatment achieved a 2 dB(A) reduction in sound level at the operator's position, but was never tested underground. Recently, a urethane coating, similar to the coating used for the coated flight bar design, was applied to the outer diameter of the tail roller [13]. This design also achieved a 2 dB(A) reduction, but failed underground due to high point contact loading from material under the chain.

EXPERIMENTAL DESIGN AND ANALYSIS

The CMM used for this study was a 14CM-9 manufactured by Joy Mining Machinery, shown in Figure 2. By design, the tail section of CMMs can swing 90° to the left or right off center in order to deposit mined material strategically during the mining of crosscuts. This machine arrangement is usually louder due to chain tension and the flight bars contacting the conveyor side plates. Thus, only the straight conveyor configuration is examined for each tail roller for comparison purposes. The chain tension was made constant for all cases by measuring the slack in the chain. The gear sprocket that drives the conveyor chain is invariable, making the chain speed constant. Also, a water spray of approximately 1 gal/min was applied to the conveyor to simulate the wet environment encountered during mining.

Figure 2: 14CM-9 Continuous Mining Machine in Reverberation Chamber

The urethane jacketed tail roller used in this study consisted of a standard tail roller machined down to accommodate for a 3/8-inch-thick layer of urethane that was bonded to a 1/4-inch-thick steel outer tube. The overall diameter of the urethane jacketed tail roller is one inch larger than the standard tail roller, which required the slide plate to be slightly cut. The properties of the urethane material can be seen in Table 1.

Table 1: Material Properties of Urethane

Polyurethane Material (PO 650)	
Durometer	84A
Max Elongation	550%
Compression	45% Max
Ultimate Tensile Strength	6000 psi

Sound power levels, vibration levels, and Frequency Response Functions (FRFs) were examined in this study to quantify the performance of the standard and urethane jacketed tail rollers. Sound power measurements were conducted in a large reverberation chamber. At the same time, vibration measurements were taken using 54 accelerometers mounted at various locations on the CMM tail section. The FRF data were obtained separately on only the tail roller component using a roving impact hammer and accelerometers.

Sound pressure levels that CMM operators are exposed to result from the sound power radiated by the machine and the acoustic properties of the environment. Sound power, which is independent of the acoustic environment, can be used to compare noise treatments directly. Sound power testing for this study was conducted in the reverberation chamber at the Pittsburgh Research Laboratory [14,15]. All sound power measurements were in accordance with ISO 3743-2 using the comparison method with the equation:

$$L_w = L_{w_r} + \left(\overline{L}_p - \overline{L}_{p_r}\right), \tag{1}$$

where L_{w_r} is the sound power level of the reference sound source, and $\overline{L}_p$ is the space-averaged sound pressure level for the source under test. The sound pressure data were collected with a Bruel & Kjaer Pulse acquisition system using 15 measurement channels. The results were A-weighted and calculated in one-third octave bands.

Vibration levels were examined at 54 measurement locations on the CMM tail section for both tail rollers. The tail section was divided into six measurement areas: left flex plate, right flex plate, left side, right side, top deck, and return deck. Nine accelerometers were mounted at each measurement area using cyanocrylate gel. For the top deck measurement area, accelerometers were embedded into the deck by drilling 6-mm-diameter holes approximately 6 mm deep. In order to run the cables to the outside of the machine, grooves were made in the underside of the top deck. The acceleration levels were digitized, A-weighted, and filtered in one-third octave bands using a LMS Pimento™ analyzer. The RMS acceleration for each measurement area was calculated from the acceleration levels using the equation:

$$\overline{a}_{rms} = \sqrt{\frac{1}{n}\sum_{i=1}^{n} a_i^{\,2}}\,, \tag{2}$$

where a_i is the acceleration at the individual measurement locations. The RMS acceleration levels were converted logarithmically using the equation:

$$L_a = 20 \cdot \log_{10}\left(\frac{\overline{a}_{rms}}{1g}\right), \tag{3}$$

where L_a is the average acceleration in one-third octave bands.

Impact testing was performed on the both the standard and urethane jacketed tail rollers to gain insight about the structural modes of vibration. The tail roller component was disconnected from the CMM tail section and Styrofoam was used to simulate free boundary conditions during testing, as shown in Figure 3. Two accelerometers were installed on each end of the tail roller shaft using cyanocrylate gel. A PCB™ model 086C03 impulse hammer was used to excite the system and data were collected using the same LMS Pimento™ analyzer as mentioned above. The frequency response between the force inputs and accelerometer locations were found using the relationship [16]:

$$H(\omega) = \frac{X^*(\omega)Y(\omega)}{X^*(\omega)X(\omega)}, \tag{4}$$

where $X^*(\omega)$ is the complex conjugate frequency response of the impact hammer, and $Y(\omega)$ is the frequency response of the accelerometers. The magnitude and phase responses were translated from the complex FRF data using the equations [16,17]:

$$|H(\omega)| = \sqrt{\mathrm{Re}[H(\omega)]^2 + \mathrm{Im}[H(\omega)]^2}$$

and (5)

$$\Phi(\omega) = \tan^{-1}\left(\frac{\mathrm{Im}[H(\omega)]}{\mathrm{Re}[H(\omega)]}\right),$$

Three impact locations were utilized on the tail roller at: the midpoint of the outer sleeve, four inches to the left of the midpoint, four inches to the right of the midpoint. The impact locations were chosen based on where the conveyor chain is in contact with the tail roller. The FRF magnitude data were combined logarithmically using the equation:

$$|H_T(\omega)| = 20 \cdot \log_{10}\left(|H_i(\omega)|\right), \tag{6}$$

where $|H_T(\omega)|$ is the average response for the tail rollers. Also, coherence between force inputs and acceleration outputs were also examined to ensure linearity between signals.

Figure 3: Impact Test Setup for Tail Roller Component

RESULTS AND DISCUSSION

The FRF data were examined to determine the modal shapes and frequencies of the treated and untreated tail rollers. The magnitude response from the impact testing is shown in Figure 4. The resonant peak of the urethane jacketed tail roller, just below 6 kHz, is 6 dB lower than that of the standard tail roller. While the magnitude frequency response at 7 kHz has been eliminated, the energy appears to have shifted to the next higher mode for the treated tail roller. A similar phenomenon is observed at the lower frequency mode below 2 kHz.

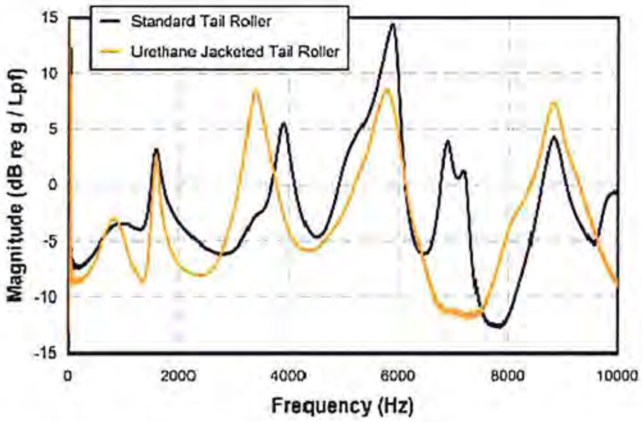

Figure 4: Magnitude Frequency Response of Tail Roller Components

The results of the sound power tests conducted in the reverberation chamber are displayed in Figure 5. The results show levels above 100 dB(A) at the one-third octave band frequency between 400 Hz and 3,150 Hz. This accounts for 85-90% of the sound energy emitted by the CMM. The urethane jacketed tail roller sound power test shows modest reductions of around 1 dB(A) for all one-third octave band frequencies except at 160 Hz, which exhibited a 2 dB(A) increase. The overall sound power level of the CMM with standard tail roller is 116 dB(A), and 115 dB(A) with the urethane jacketed tail roller installed. This translates to an overall reduction of just 1 dB(A) in sound power level when the urethane jacketed tail roller was installed, which will be explained below.

Vibration measurements were taken during the sound power testing in the reverberation chamber. The measurements were divided into 6 component areas on the CMM and are displayed in Figures 6-11. The results show a similar pattern to the sound power level spectra, with the highest acceleration levels occurring between 400 Hz and 5 kHz. A possible resonant peak is observed at the 800 Hz band for the left and right side measurement locations, shown in Figures 6 and 7.

The top deck, shown in Figure 8, demonstrates the largest overall reduction in acceleration level of 3 dB(A) with the urethane jacketed tail roller installed. The return deck measurement area shown in Figure 9 displays highest overall acceleration level of 19 dB(A) with the standard tail roller installed.

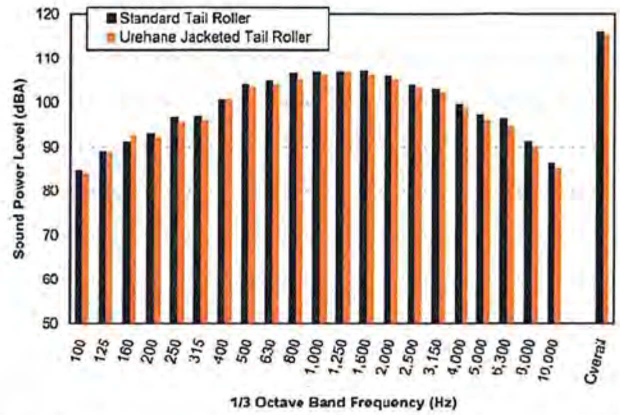

Figure 5: A-weighted Sound Power Level Comparison of CMM Standard and Urethane Jacketed Tail Rollers

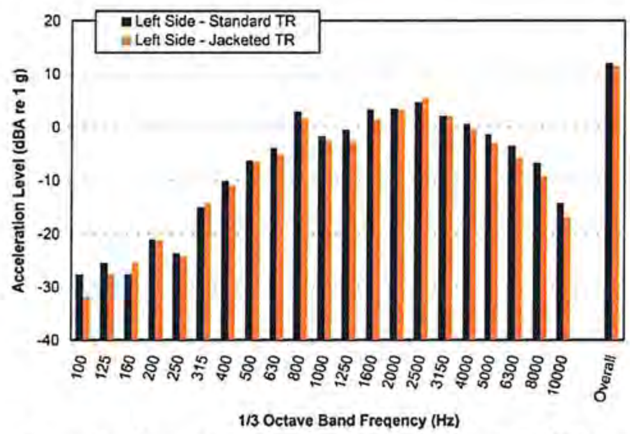

Figure 6: A-weighted Acceleration Level at CMM Left Side

All vibration measurement areas on the CMM show an overall reduction in A-weighted acceleration levels of at least 1 dB using the treated tail roller, except the left flex plate. Figure 10 shows how the acceleration levels in each one-third-octave frequency band at this measurement area increased after the jacketed tail roller was installed. At the left flex plate measurement segment an overall increase of 5 dB acceleration is observed. This is not seen at the right flex plate measurements displayed in Figure 11, where an average reduction of 4 dB is observed in one-third octave band frequencies greater than 2 kHz.

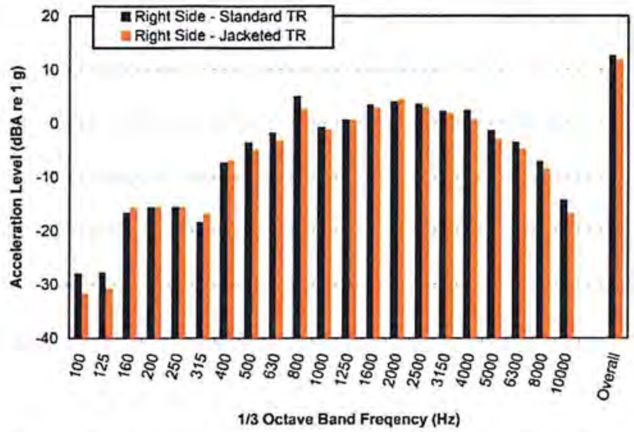

Figure 7: A-weighted Acceleration Level at CMM Right Side

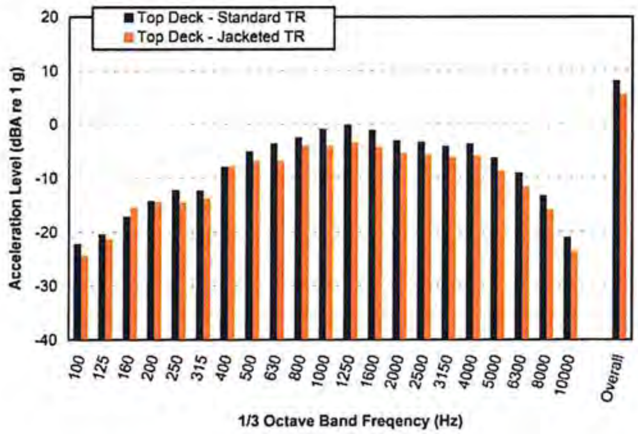

Figure 8: A-weighted Acceleration Level at CMM Top Deck

To better quantify the performance of the urethane jacketed tail roller, the FRF, sound power, and vibration measurements can be analyzed together. The higher frequency reductions shown in the magnitude response of Figure 4 are annulled by modal shifts and increases at lower frequency. This somewhat explains the slight increases in the sound power levels at the frequencies below 500 Hz. However, this study only examines A-weighted levels, which should limit low frequency contributions to the overall level. As stated above, the left flex plate measurement area showed an increase in acceleration levels. As the chain travels down the conveyor the flight bars make contact with the flex plates. These ¼-inch-thick steel plates guide the chain along the conveyor deck on each side. Although chain tension was consistent for all tests, there is no mechanism to ensure that the chain is traveling straight down the conveyor deck. It appears from the vibration data shown in Figure 10 and sound power data shown in Figure 5 that the chain flight bar contacting the left plate is radiating noise. This would explain why reductions in sound power levels are not achieved at the higher frequencies with the urethane jacketed tail roller installed.

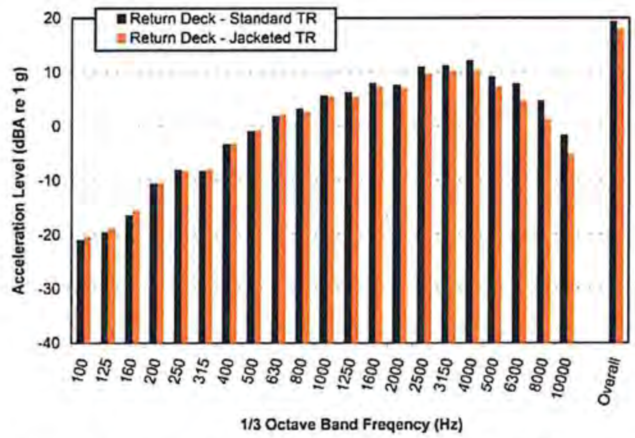

Figure 9: A-weighted Acceleration Level at CMM Return Deck

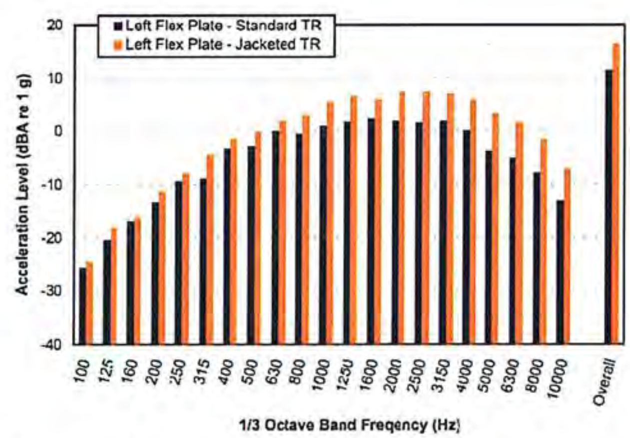

Figure 10: A-weighted Acceleration Level at CMM Left Flex Plate

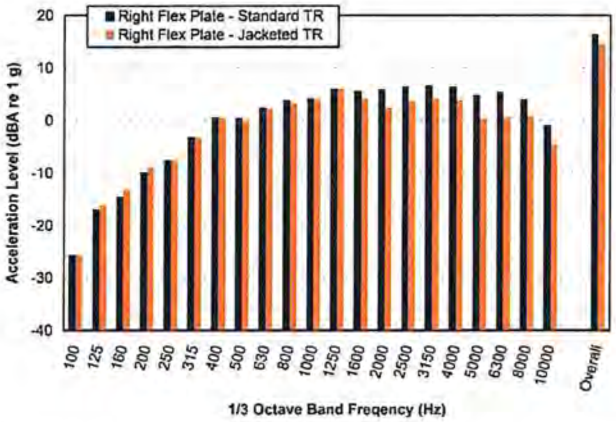

Figure 11: A-weighted Acceleration Level at CMM Right Flex Plate

CONCLUDING REMARKS

A treated tail roller has been developed in order to reduce sound emission from a CMM. The urethane jacketed tail roller was developed based on previous noise control treatments and consists of a standard tail roller with a layer of urethane bonded to a steel outer shell. This noise control is the latest attempt to reduce noise exposure of CMM operators. Sound power levels, FRF, and vibration data were examined to quantify the performance of the treated tail roller. A modest overall reduction of 1 dB(A) in sound power level was achieved with the urethane jacketed tail roller. However, the urethane jacketed tail roller could be combined with proven noise controls to further reduce noise exposure caused by CMMs. Examining FRFs and acceleration levels gives insight as to why higher reductions were not achieved. The magnitude response from impact hammer testing indicated insufficient reductions and shifting of modes toward low frequencies. The urethane layer could be made with a lower durometer that may improve low frequency performance, but could also reduce durability. The accelerometers positioned on the CMM left flex plate reveal that the chain flight bars may be making contact with the plate, causing this component to radiate noise. Further investigation is needed to determine how this possible noise source is contributing to the overall conveyor system noise. Future studies will examine flex plate noise and using different materials for the resilient layer between the tail roller and steel outer shell.

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