Evaluation of stiffeners for reducing noise from horizontal vibrating screens

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Sound levels around vibrating screens in coal preparation plants often exceed 90 dB(A). The National Institute for Occupational Safety and Health (NIOSH) is developing noise controls to reduce noise generated by horizontal vibrating screens. NIOSH researchers used finite element analysis (FEA) and a NIOSH-written program to estimate the sound power level reduction resulting from adding rib stiffeners to key locations on the sides of a screen. The researchers evaluated adding rib stiffeners made from steel channel with two different cross-sections (C and T) and the effects of orienting the stiffeners horizontally and vertically. In addition, the researchers investigated the effects of increasing the modal damping from 0.002 to 0.010. The results indicate that for a broadband input, adding stiffeners had little impact on the estimated sound power level, but increasing the modal damping reduced the predicted A-weighted sound power level by 7.8 dB.

1   INTRODUCTION

In 1996, NIOSH published the National Occupational Research Agenda, which identified hearing loss as the most common job-related disease in the United States[1]. Approximately 30 million workers are exposed to hazardous sound levels alone or to hazardous sound levels in conjunction with ototoxic agents[2]. Despite more than 30 years of noise regulation in the mining industry, mine workers develop hearing loss at a significantly higher rate compared to the non-noise exposed population. According to an analysis of audiograms performed by NIOSH, nearly 90% of coal miners have a hearing impairment by the age of 50 whereas only 10% of the population that is not exposed to occupational noise has a hearing loss by the same age[3].

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To reduce the occurrence of noise-induced hearing loss, the Mine Safety and Health Administration (MSHA) modified its rules regarding noise exposure in 1999[4]. MSHA's new rule requires mine operators to use all feasible engineering and/or administrative controls to reduce the noise exposures of overexposed miners'. However, for many machines, such as vibrating screens, noise controls that reduce the operator's noise exposure below the MSHA Permissible Exposure Level (PEL) are not currently available.

In 2000, there were 212 preparation plants in operation in the United States[5]. NIOSH studies show that workers who spend a significant portion of their shift working in a coal preparation plant can experience noise exposures which exceed the MSHA PEL for noise. NIOSH data show that 20 out of 46 coal preparation plant workers had noise exposures that exceeded the MSHA PEL noise dose[6]. MSHA PEL noise doses up to 220% have been recorded for preparation plant workers. Workers in preparation plants spend a significant portion of a shift working around vibrating screens. The number of screens in a processing plant can range from a single screen to more than a dozen. 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[7].

A horizontal vibrating screen (see Fig. 1) is a large machine used to process clean coal that has been separated from refuse materials using a water-magnetite mixture. This magnetite is recovered so it can be reused in the processing plant and because it lowers the heating value of coal. The screen body has two sides made of steel plates with a bottom screening surface made of steel wire welded to a frame with small gaps between the wires. Round cross-tubes are used at either end of the screen to help stiffen the structure. The feed box, also made from steel plates, is where the coal enters the screen. The body of the screen is supported on a steel coil spring suspension. One or more vibration mechanisms are mounted to a steel beam that connects the sides of the screen. These vibration mechanisms, which use rotating eccentric shafts to generate vibration, are belt-driven by an electric motor. The screen is designed such that it vibrates on a 45 degree angle. Coal flows into the feed end of the screen from a delivery chute. As the screen vibrates, the material traverses the deck under a water spray that rinses the magnetite from the coal. The liquid and fine coal particles pass through the gaps in the screening deck as the material flows toward the discharge end of the screen. Finally, the rinsed coal falls off the discharge end of the screen for further processing.

Previous NIOSH studies showed the A-weighted sound level around a group of eight horizontal vibrating screens used to process clean coal ranged from 94 to 98 dB(A)[8]. A series of measurements in an operating preparation plant indicated that noise due to vibration was the dominant noise source whereas noise from material flow was less significant. Further research showed that most of the noise due to vibration is radiated by the screen body and the mechanism housings[9,10]. At frequencies below about 1 kHz, screen body noise is the main noise source whereas mechanism housing noise is the primary source above 1 kHz. The sound energy at frequencies below 1 kHz accounts for about 80% of the overall A-weighted sound power level. Operating deflection shape analysis revealed significant response on the screen sides and feed box[11].

Estimates of the sound power radiated by the screen sides and feedbox using vibration measurements showed that each is significant in terms of the total sound power radiated by the screen. For vibrating screens, it is undesirable to add weight to the screen body because this could adversely affect the performance of the screen. Therefore, it is not possible to make large scale changes to the structure of the screen. This work examines the effects of adding stiffeners or increasing damping on the sound power level radiated by the screen sides and feed box using

a finite element model (FEM) of the screen to estimate vibration response and a NIOSH-written computer program to estimate the sound power radiated by the screen sides and feedbox.

2   SCREEN FINITE ELEMENT MODEL DEVELOPMENT

2.1 Screen Solid Model

Pro/Engineer was used to develop a surface model of the as-built screen. The solid model included nearly all the components of the screen. The drive mechanisms were represented by solid blocks with the density adjusted so the mass of the blocks matched the mass of the mechanisms. The screen deck was not included in the model because it would significantly increase the complexity of the model without significantly changing the results since the screen deck does not contribute substantially to either the stiffness or mass of the structure. In addition, the frame used to support the screen was not included because previous tests did not indicate it as a contributor to noise. The belt guard was also ignored because previous testing showed the belt guard was not a significant noise source unless it rattles against the screen. Belt guard rattling can be eliminated with a slight increase in clearance. In addition, an assumption was made that the belt guard does not have a significant impact on the vibration of the other screen components.

2.2 Screen Finite Element Models

The Pro/Engineer solid model was imported into ANSY finite element analysis (FEA) software (see Fig. 2a). Some connections between screen components, such as the ribs and screen sides, are welded joints. Other components, for example the screen sides and round cross-tubes, are connected via bolted joints. For the purposes of the model, all connections between individual components were modeled as bonded joints within the software. The screen suspension springs were modeled using lumped spring elements at each of the mounting locations. Three spring elements were used at each location to model the spring rates in the x (for/aft), y (vertical), and z (lateral) directions. Each suspension spring in the model was connected to ground with fixed boundary conditions. Steel was used for material properties for the entire screen. An example of the mesh for the finite element model of the screen without added stiffeners is shown in Fig. 2b.

After examining previously collected beamforming, ODS, and modal data, it appeared that several locations along the screen sides, the sides of the feedbox, and the back of the feedbox were dominant contributors to vibration and noise radiation. For ease of reference, the areas between the ribs on the screen are called "panels". Panel 4, panel 7, panel 8, the feedbox side and the feedbox back (refer to Fig. 3a) are the areas that exhibited the most significant vibration response. These areas were considered as potential locations for adding stiffeners. Initially, stiffeners with a T-shaped cross section were considered and one additional model was made using stiffeners with a C-shaped cross section (refer to Fig. 3b). For reference, the screen sides and feedbox are 8 mm thick.

Models were made with stiffeners added as shown in Fig. 4. For the first case, a single T-shaped stiffener was added across the feedbox back with a vertical T-shaped stiffener added to each side of the feedbox. For the second case, horizontal T-shaped stiffeners were added to panels 4, 5, 7, and 8 along with the stiffeners from the first case. Next, for the third case, four vertical T-shaped stiffeners were added across the feedbox back along with vertical stiffeners on the feedbox sides and panels 4, 7, and 8. Finally, for the fourth case, C-shaped stiffeners were used at the same locations with the same orientations as the third case. In every case, the backs of

the stiffeners were modeled as bonded to the screen sides and feedbox. In addition, the ends of the stiffeners considered to be bonded to the existing ribs.

2.3 Screen Vibration Analysis

To estimate the effects of adding stiffeners or damping on the noise radiated by the screen sides and feedbox, a linear dynamic analysis was performed in ANSY. Forces were applied to the screen model at the center of each mechanism perpendicular to the H-beam as shown in Fig. 5 (note: mechanisms hidden for clarity). Previous work showed that during operation, screen natural frequencies are excited due to operation of the mechanisms. It is thought that bearing slap within the mechanisms excites the structure through a series of impacts. These impacts provide broadband excitation to the screen. For the purposes of evaluating the effects of adding stiffeners or damping, each force was assumed to have a flat input spectrum of 4.45 N from 0 to 1 kHz. The actual excitation force may differ from this assumption, but the purpose of this effort is to examine if adding stiffeners or damping will reduce radiated noise. Therefore, it is not necessary to have an exact prediction of noise radiated by the screen; rather the change is what is important.

Within ANSYS, the screen response to these input forces is solved in two steps. First, the natural frequencies and mode shapes are determined in the desired frequency range. Next, the modal superposition technique is used to estimate the screen response to the input forces. The modal superposition technique determines the contribution of each mode of vibration to each input force on a frequency-by-frequency basis. The modal responses are then summed to yield the response of the screen. To perform the analysis, the modal damping must be defined. A constant modal damping ratio of 0.002 was used for all analysis except for the analysis used to examine the effects of increasing damping where a constant modal damping of 0.010 was used. Previous experimental modal test results indicated the modal damping values were much less than 0.010. Table 1 shows the test cases which were simulated.

The displacement responses at 39 locations were computed and stored for each analysis. Figure 6 shows the locations where the responses were computed. Displacement results were stored as follows: five locations on panels 4, 6, 7, and 8; four locations on panel 5; five locations on the side of the feedbox; and ten locations on the back of the feedbox. These locations were used for all analyses. For each analysis, the results were exported as text files that could be read into MATLAB.

3   ESTIMATION OF SOUND POWER RADIATED

The sound power level radiated by a vibrating object can be estimated by

$$L_W = 10\log_{10}\langle v\rangle^2_{s,t} + 10\log_{10}S + 10\log_{10}\sigma + 146 \quad (1)$$

where L_W is the sound power level, $\langle v\rangle^2_{s,t}$ is the surface averaged mean-squared velocity, S is the surface area, and σ is the radiation efficiency[12]. Equation 1 was incorporated into a computer program that carried out all calculations for each screen location on a frequency-by-frequency basis in 1 Hz increments. For each set of displacement results from the FE model (e.g. panel 4, panel 5, panel 6, etc.), the velocity response at each location was calculated by multiplying the frequency data by

$$v = 2\pi f d \quad (2)$$

Where *d* is the displacement magnitude as a function of frequency and *f* is the excitation frequency in Hz. The surface averaged mean-squared velocity response of each panel, the feedbox sides, and the feedbox back were than calculated by squaring and averaging these velocities for each component.

The radiation efficiencies were estimated from the thickness and perimeter of each component using information from *Bies and Hansen*. Within the program, the radiation efficiencies were estimated by curve fitting values from the *Bies and Hansen* data using a 3rd order polynomial. This provided a simple means of estimating the radiation efficiency on a frequency-by-frequency basis.

Estimates for radiated sound power level were performed on each component separately. Since we are interested in reducing worker exposure according to the MSHA PEL, which is based on A-weighted sound levels, the program was used to apply A-weighting to the estimated sound power levels. The sound power levels for each panel on the screen sides were added to yield the estimated sound power level radiated by the screen sides. The estimated sound power level from the feedbox was estimated by adding the sound power levels from the feedbox sides to that of the feedbox back. The estimated A-weighted sound power level was also summed for the 0 to 500 Hz and 500 Hz through 1 kHz frequency bands. In addition, the A-weighted sound power level was computed in 1/3-octave bands.

4   RESULTS AND DISCUSSION

Prior to using the FE model to examine noise radiation, a comparison was made between the frequencies of the first few modes of vibration identified by the FE model as compared to those obtained via experimental modal analysis (EMA). In addition, a visual comparison was made between the animated mode shapes for the first few modes. Figure 7a and shows a comparison between the first structural mode of vibration as identified by FEA and EMA. Both results show indicate this is a torsional mode of vibration. The FE model predicted the resonant frequency for the first mode to be 20.3 Hz where the EMA showed this mode to occur at 18.8 Hz. Figure 7b shows the comparison between the FE and EMA results for a mode that involves only the feedbox back. The frequency for this mode was 44.0 Hz according to the FE model and 42.0 Hz according to the EMA results. For our purposes, these differences were considered to be acceptable.

Figure 8a shows a screen mode at 221 Hz that involves significant contributions from the feedbox back and panels 4, 7, and 8. Many of the modes identified from the FE model showed significant contributions from these areas. The forced response at 221 Hz is shown in Fig. 8b. As expected, the response looks nearly identical to the 221 Hz mode. Significant contributions from these same regions were observed on the other modes predicted by the FE model. In addition, the previous ODS analysis showed these areas to be dominant contributors to screen vibration.

The FE models were used to calculate the forced response of the screen as previously described. In each case, the exported displacement results at the locations shown in Fig. 6 were used to estimate the radiated sound power from the screen sides and feedbox. Table 2 shows the resulting estimates of the radiated sound power level for the screen side and feedbox. All values in the table are with A-weighting applied because all noise exposure regulations in the mining industry are based on A-weighted sound levels. Table 2 shows that none of the cases with added stiffeners made a significant improvement on the predicted sound power levels. In fact, the estimated sound power levels for the vertical T-shaped stiffeners (Case 4) and C-shaped stiffeners (Case 5) were 0.5 and 1.0 dB(A) higher, respectively, than the estimate for the baseline, as built model. The only model that showed a significant improvement was the model

where the modal damping ratio was increased from 0.002 to 0.010. To increase the modal damping, constrained layer damping could be applied at various locations on the screen.

It was expected that adding the stiffeners to the panels would significantly reduce the response of the structure and the resulting radiated sound power; however this was not the case. Figure 9 shows a comparison of the estimated A-weighted sound power level in narrow bands for the as built model (Case 1) compared to the model with vertical T-shaped stiffeners (Case 4) narrowband spectra. It appears that some of the modes of the structure simply shifted to a higher frequency with little change in response and others did not change much at all. Figure 10 shows a comparison of the estimated A-weighted sound power level in narrow bands for the as built model with 0.002 modal damping (Case 1) compared to the as built model with 0.010 modal damping (Case 6). As expected, the significant increase in damping yielded a substantial reduction in the response at each of the peaks which correspond to the modes of the screen. Recall that at resonance, the modal damping has the most influence on the response.

5 CONCLUSIONS

The effects of adding stiffeners to the sides and feedbox of a horizontal vibrating screen and the effects of increasing the damping of a screen were evaluated using the displacement response predicted by an FE model. Surface average mean-square velocities calculated from the FE results were used with estimated radiation efficiencies to estimate the A-weighted sound power level from the screen sides and feedbox. According to the results, adding stiffeners would have little impact on the sound power level radiated by these components. However, increasing the modal damping for the screen from 0.002 to 0.010 resulted in a predicted reduction of 7.8 dB(A).

6 ACKNOWLEDGEMENTS

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Table 1 - Descriptions of the FE models used to examine the effects of adding stiffeners or damping to the screen.

Case	Description	Modal Damping
1	As built	0.002
2	T-shaped stiffeners, feedbox back – horizontal, feedbox sides - vertical	0.002
3	T-shaped stiffeners, feedbox back – horizontal, feedbox sides - vertical screen sides, panels 4, 5, 7, & 8 - horizontal	0.002
4	T-shaped stiffeners, feedbox back and sides – vertical, screen sides, panels 4, 7, & 8 - vertical	0.002
5	C-shaped stiffeners, feedbox back and sides – vertical; screen sides, panels 4, 7, & 8 - vertical	0.002
6	As built	0.010

Table 2 - Estimated A-weighted sound power level as predicted using the FE model and computer program.

Case	Description	Estimated Sound Power Level, dB(A)									
		0 to 500 Hz			500 to 1000 Hz			0 to 1000 Hz			
		Side (each)	Fbox	Total	Side (each)	Fbox	Total	Side (each)	Fbox	Total	Δ
1	As built, ζ = 0.002	60.3	58.0	64.4	56.7	53.2	60.6	61.9	59.3	65.9	-
2	T stiffeners, ζ = 0.002 • fbox back - H • fbox sides – V	59.9	58.4	64.2	57.0	53.0	60.8	61.7	59.5	65.8	-0.1
3	T stiffeners, ζ = 0.002 • fbox back - H • fbox sides - V • panels 4, 5, 7, & 8 – H	57.9	58.8	63.0	57.6	53.7	61.5	60.8	60.0	65.3	-0.6
4	T stiffeners, ζ = 0.002 • fbox back and sides - V • panels 4, 7, & 8 – V	60.7	58.9	65.0	57.2	51.7	60.8	62.3	59.7	66.4	+0.5
5	C stiffeners, ζ = 0.002 • fbox back and sides - V • panels 4, 7, & 8 – V	60.8	59.3	65.2	58.3	54.4	62.1	62.8	60.5	66.9	+1.0
6	As built, ζ = 0.010	52.0	50.1	56.2	50.3	43.1	53.7	54.2	50.9	58.1	-7.8

Note: Fbox denotes feedbox, ζ denotes the modal damping ratio, H denotes horizontal, and V denotes vertical.

(a)   (b)
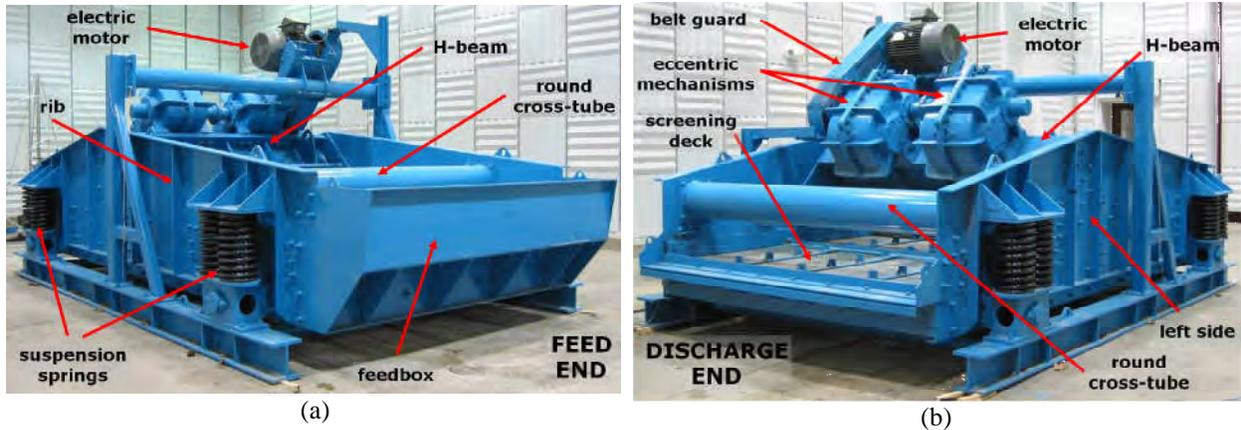
Fig. 1 -   A horizontal vibrating screen used to process coal viewed from (a) feed end and (b) discharge end.
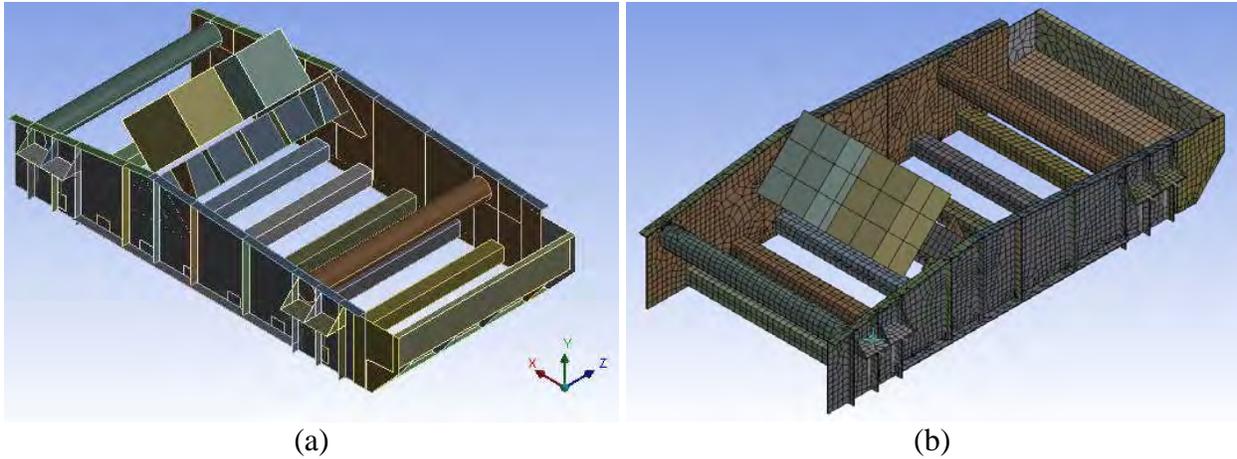
Fig. 2 - *(a) Isometric view of the ANSYS model of screen from the feed and (b)isometric view of the screen model with FE mesh from as observed from the discharge end.*
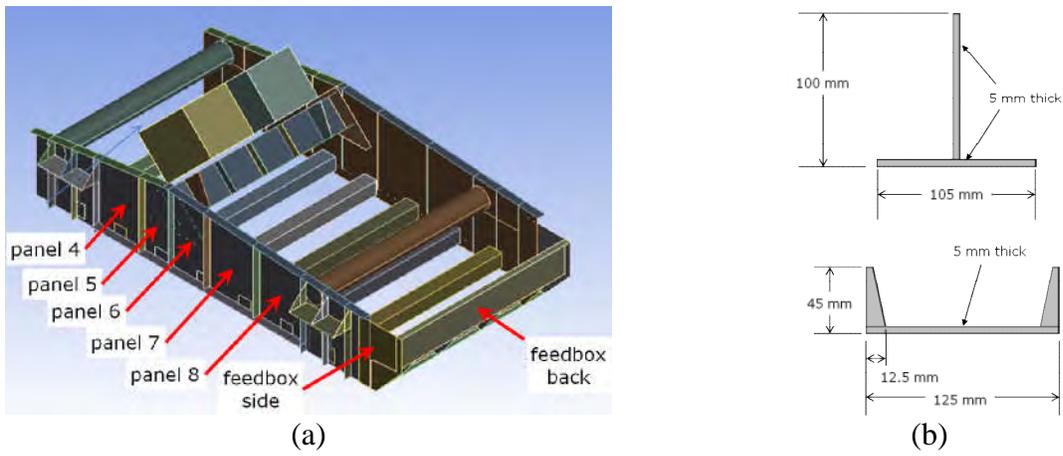

Fig. 3 - *(a) Reference locations on the screen sides and feedbox (note: panels 4,7, and 8 and the feedbox exhibited high vibration response) and (b) stiffener cross sections.*
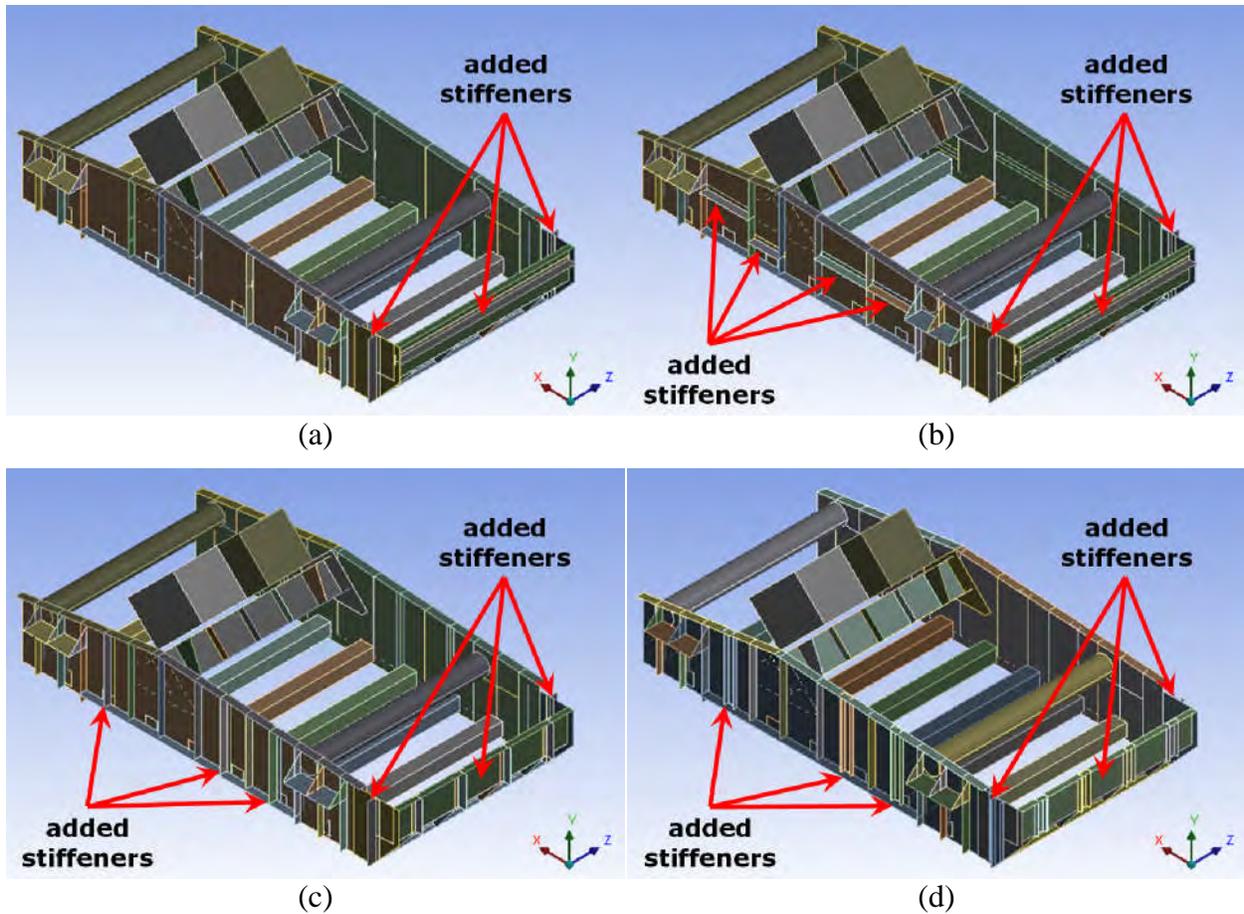
Fig. 4 - Models of screen with: (a) horizontal T-shaped stiffener on feedbox back and vertical T-shaped stiffeners on feedbox sides; (b) horizontal T-shaped stiffener on feedbox back, and panels 4,5,7, and 8 with vertical T-shaped stiffeners on feedbox sides; (c) vertical T-shaped stiffeners on panels 4, 7, and 8, the feedbox sides, and the feedbox back; and (d) vertical C-shaped stiffeners on panels 4, 7, and 8, the feedbox sides, and the feedbox back.
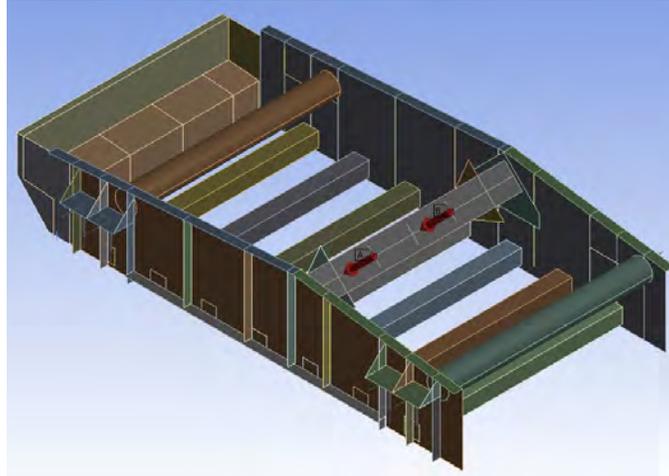
Fig. 5 - Excitation forces applied to the H-beam at the mechanism locations (note: mechanisms hidden for clarity).
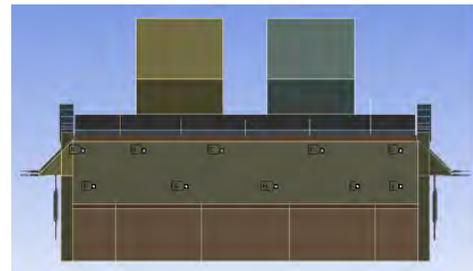
(a)   (b)

Fig. 6 - Locations on (a) the screen side and feedbox side and (b) the feedbox back where displacement results were stored for each analysis.
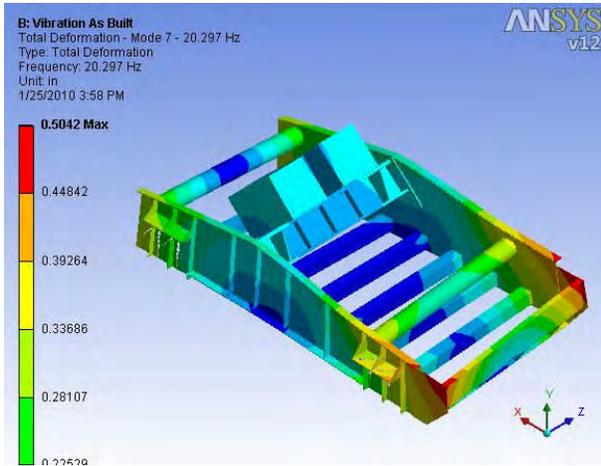
(a)
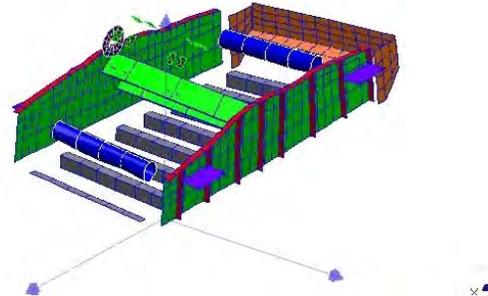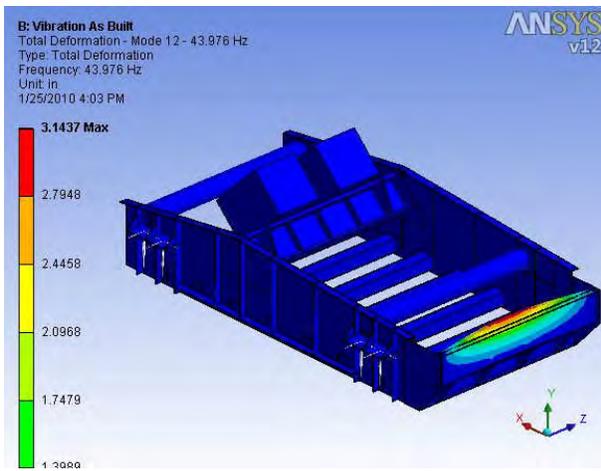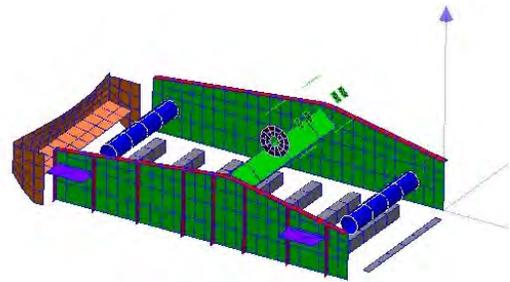
(b)

Fig. 7 - Comparison of the mode shapes for (a) the 20.3 Hz torsional mode predicted by the FE model compared to the 18.8 Hz mode from EMA and (b) the 44.0 Hz feedbox back mode predicted by the FE model compared to the 42.0 Hz mode from EMA.

(a)   (b)
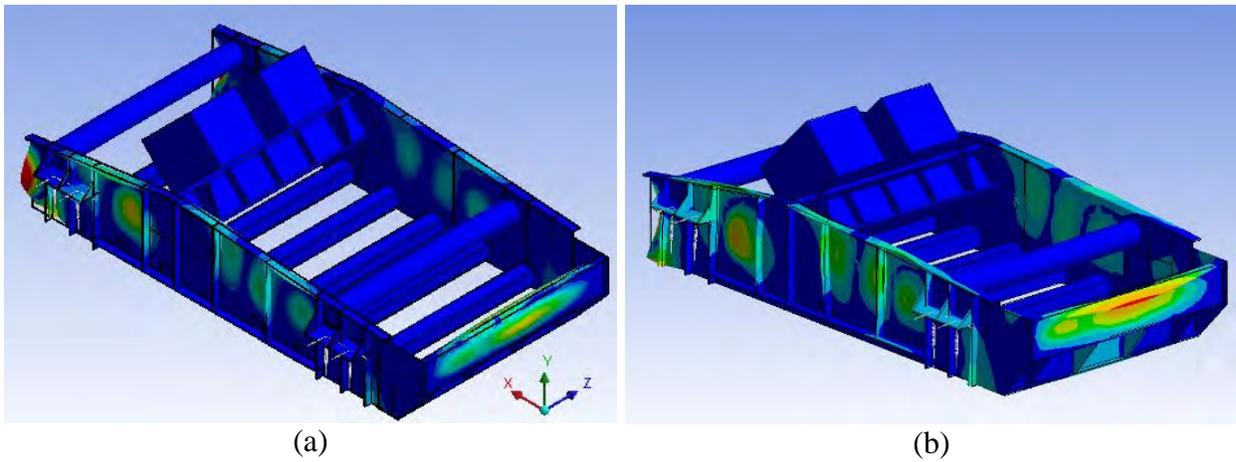
Fig. 8 -   Comparison of (a) screen mode at 221 Hz with (b) the forced response at 221 Hz as predicted by the FE model.
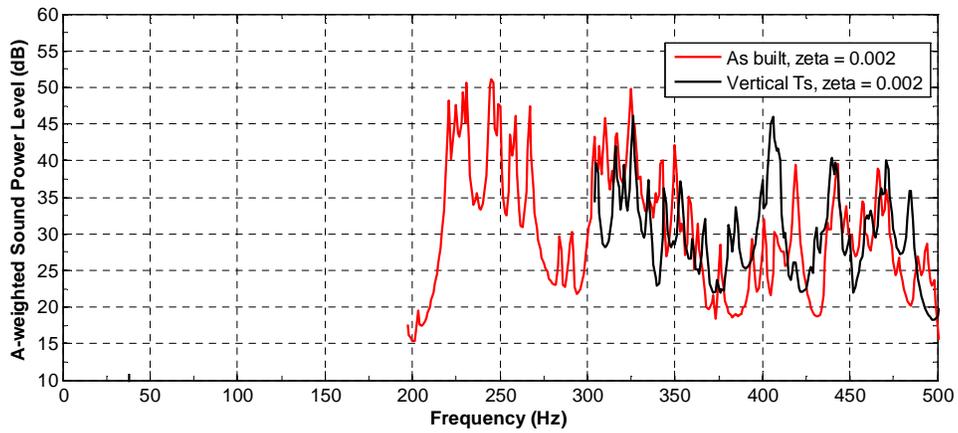

Fig. 9 -   Comparison of the A-weighted narrowband sound power level spectra for the as built model and model with vertical T-shaped stiffeners.
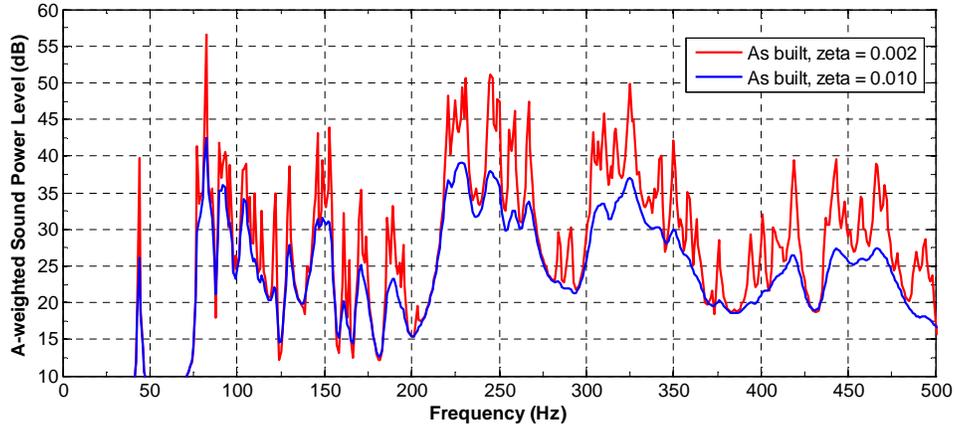
Fig. 10 - Comparison of the A-weighted narrowband sound power level spectra for the as built model with 0.002 and 0.010 modal damping.