Table Saw Noise Control

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This article presents a case study on the radiation, transmission and reduction of noise from a table saw typically used in the construction industry. The National Institute of Occupational Safety and Health (NIOSH) initiated this project through a multi-university student project program. In the construction industry, little attention has been given to control of noise from equipment and power tools. In order to address these issues, the work presented here is focused on reducing noise radiation from a table saw used in the construction industry from an engineering control point of view. The testing methodologies and results presented in this article address both the airborne and structure-borne noise contributors of the table saw. Sound power and sound intensity measurements were used to identify and rank all possible noise sources from the table saw. The use of these testing techniques in conjunction with experimental modal analysis, operational deflection shape analysis, and finite element analysis, provided data from which several acoustic treatments were chosen and investigated. Materials such as free layer paints, blade stabilizers and open-cell foam were used as possible solutions to the table saw's noise emission problem. Several treatments were recommended for either production or aftermarket modifications.

In general, the construction industry has placed little emphasis on the control of damaging noise attributes of power tools. Statistics show that an increasing number of construction workers – as many as 50% in the industry – suffer from hearing impairment due to work-related noise.[1,2] In order to address this concern among construction workers, a study of noise emission from a Craftsman 10 in. table saw was conducted. The National Institute of Occupational Safety and Health (NIOSH) had requested this noise emissions study to reduce the overall sound pressure level (SPL). In the consumer table saw market, quieter models are generally more expensive and designed for use in industrial and professional shops. However, more common consumer grade tools such as the Craftsman table saw used in this experiment, usually suffer from noise problems induced by low cost construction, noisier motors and general lack of emphasis on noise control. These tradeoffs are made for the sake of ruggedness, portability and reduced cost. However, current awareness of hearing damage has prompted this investigation to improve these products while maintaining their functionality and low cost in the marketplace.

Testing Methodology

The primary goals of the study were to identify and rank all possible noise sources, and to develop possible acoustic treatments to attenuate the operating noise of the table saw. This study approached the problem from two directions – airborne noise contributors and structure-borne noise contributors. In order to identify and rank all possible noise sources, two approaches were employed: sound power and sound intensity to assess airborne noise contributions and structural testing techniques such as experimental modal analysis, operational deflection shape (ODS) analysis; and finite element analysis to investigate any structure-borne noise contributors within the table saw.

The most practical and direct way to identify and rank the major airborne noise sources was to perform sound power tests on a component-by-component basis. This procedure allowed major noise sources to be ranked. Once the sources were ranked, the next step was to determine the locations on the assembled table saw where the sound intensity was greatest. This approach allowed targeting of specific areas for noise path treatments. Two methods of determining sound intensity were used – point-by-point sound intensity evaluation and averaged sound intensity scans. These testing techniques provided information, allowing the major noise contributors to be identified. The effectiveness of applied treatments was evaluated using similar techniques.

To investigate any structure-borne noise emitted from the table saw, three structural vibration testing techniques were utilized: ODS analysis, experimental modal analysis, and analytical modal analysis using finite elements. The ODS test was performed to determine at which frequencies the table saw was structurally excited. Once the areas of maximum excitation were found, several experimental and analytical modal analyses were performed to determine if the excitation frequencies coincided with natural frequencies of the structure.

Sound Power Measurements. Sound power is defined as the rate of acoustic energy emitted from a source. This is ideal for ranking noise sources since the acoustic energy emitted by a source is the same regardless of the environment or measurement location. This experiment utilized two methods to calculate sound power – the comparison method and the free field sound intensity method.

The comparison method calculates sound power through the comparison of SPLs (in a reverberant field) of a source of unknown sound power to those of a calibrated source of known sound power.[3,4] Using a source of known sound power, the average sound pressure of the known source is first measured in a given environment. The unknown sound power source is then placed in the same environment and its sound pressure is measured. The sound power for the unknown source is calculated using the relation that the difference between the power levels of the known and unknown sources is equal to the difference in the measured SPLs of the corresponding sources. This process can be performed across all frequency bands in any environment.

The sound power measurements were made, using the comparison method, in a reverberant chamber. A calibrated (81 dB) fan producing white noise was used as a known source. A figure eight pattern was used to acquire the average SPL in the reverberant chamber. The sound power for each component configuration was calculated using the method previously presented. Figure 1 shows the component configurations used in the sound power tests.

The second method used in this experiment was the free field sound intensity method. This method derives sound power from sound intensity. In a free field environment, sound intensity can be approximated as sound pressure. Sound power measurements derived from the free field sound intensity method were performed simultaneously with the sound intensity scan of the fully assembled table saw. The average sound intensity of the fully assembled structure was used to calculate the sound power.

Sound Intensity Measurements. Sound intensity is defined as the average rate of flow of energy through a unit area.[5] The direction of flow is a vector quantity that is normal to the area on which measurements are made. Determination of the magnitude and direction of sound intensity involves approximating the sound pressure gradient between two microphones when using a two-microphone intensity probe. Once the pressure gradient is established, particle velocity can be calculated and used in conjunction with the sound pressure value midway between the two microphones to calculate the sound intensity. The ability to obtain the magnitude and direction of

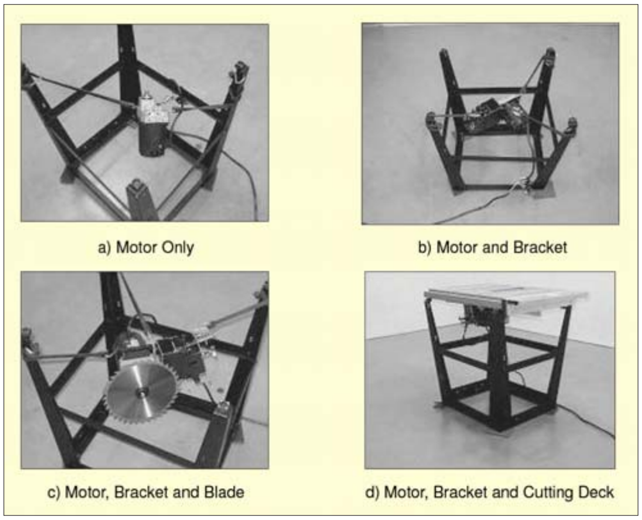

Figure 1. Test setup configurations for component sound power measurements.

noise propagation makes the application of sound intensity measurements ideal for the identification and ranking of the table saw's noise sources. Two types of sound intensity measurements were performed in this experiment: a sweeping scan and point-by-point test. The sweeping scan was performed under loaded and unloaded conditions in accordance with the ANSI standard.[6]

The accuracy and quality of the sound intensity data collected were two major concerns before testing began. To ensure the sound intensity measurements were acquired accurately, a point-by-point grid was constructed. The grid was built to dimensions of $33 \times 30 \times 48$ in. ($83.82 \times 76.2 \times 121.92$ cm) with 48 points on the front and back panels, 40 points on the right and left panels, and 30 points on the top panel. Each grid panel was equidistant from its respective saw surface. The frame was constructed with 3/4-in. PVC pipe. Grid lines were made using a cotton thread to indicate the data acquisition locations while not disrupting the sound field.

To assure that the quality of the data collected was not compromised, any sound reverberation or background noise must be minimized. In order to accomplish this, all of the unloaded intensity tests were performed in an anechoic chamber. The anechoic chamber used was rated down to 250 Hz. This was not a major concern for the intensity testing, since the one-third octave band frequencies of interest were higher than the low frequency limits of the anechoic chamber. In order to reduce the phase mismatch error, a 12 mm spacer was used in the intensity probe, giving a useful range of 250 to 5000 Hz for the intensity scan. Figure 2 shows the grid structure and table saw set up in the anechoic chamber for the sound intensity tests.

To understand how the table saw operated in an unloaded condition, the saw and grid structure were placed in the anechoic chamber. Two intensity sweeps of each of the five sides of the structure were performed. The two scans were averaged together to create the sound power map for each side. Fixed-point measurements were also taken for each of the five sides. From the data collected, a sound intensity contour map of the local intensity field of each side was generated.

The loaded test could not be performed in the anechoic chamber due to space and safety concerns. To approximate the free-field environment, the table saw was tested in an open parking lot. All of the sound power and sound intensity measurements were made with a two-channel Symphony hardware system equipped with dBFA32 software from 01dB-Stell, Inc.

Operational Deflection Shape Analysis. Operating deflection shapes define the dynamic response the table saw exhibits under operating conditions.[7] When the saw is operating under steady state conditions, there exist numerous input forces to the system. These potential inputs include, but are not limited to, multi-axial motor vibration, blade imbalance, and airborne

Figure 2. Setup for point-by-point sound intensity test in the anechoic chamber.

noise created from the moving parts. All of these inputs produce responses across the entire structure. The ODS analysis characterizes these responses at selected points about the structure with respect to a reference point on the structure. Using the data collected at these points, animations of the vibration response can be generated. These animations assist in highlighting areas of large response at frequencies of interest such as the operating frequency and its harmonics. The areas of large response can potentially be correlated to assist in the isolation of particular structure-borne noise components.

For the ODS test, the table saw was set up as it would be under normal operating conditions on a concrete floor. Data acquisition began once the saw had reached its steady state operating speed. The blade's vertical position was maintained at approximately 1.5 in. above the cutting deck. The rip fence was positioned and locked 6 in. to the operator's right of the blade, the same position it was in for loaded sound power and intensity tests. The push-fence and all other accessories except the rip guard were removed. The test was performed indoors under ambient environmental conditions.

Sixty-seven measurement locations were established on the structure to create an acceptable spatial representation of the table saw. Measurement locations were established on the cutting deck, plastic side skirts and motor components. The structural model of the measurement locations can be seen in Figure 3.

Cutting Deck Experimental Modal Analysis. The goal of this test was to determine the frequencies at which the table saw cutting deck would naturally vibrate. Knowing the frequencies and shapes of the modes of the cutting deck, comparisons can be made between natural and forced vibration frequencies. The overlapping of these resonant frequencies can lead to both acoustic and vibration problems. If this overlap does occur, treatments to the current design, or a change of design may be necessary.

Initially, the cutting deck was tested in a free-free condition by placing open cell foam beneath the supports. This allowed for the modes of the cutting deck to be seen in their natural

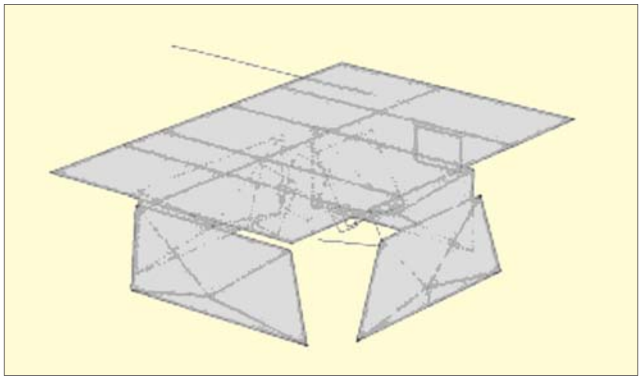

Figure 3. Geometry of the table saw for ODS test.

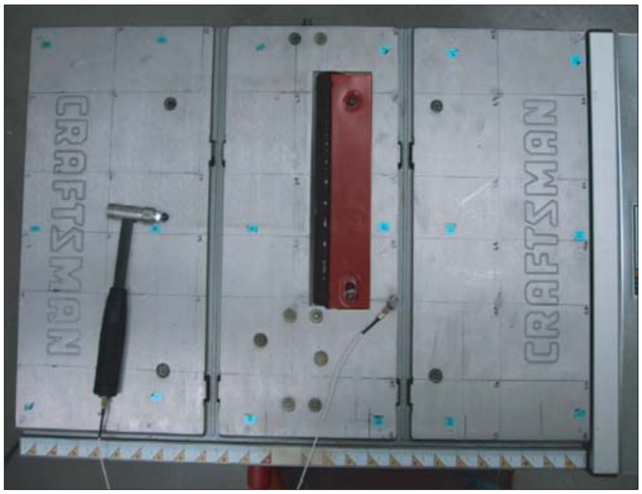

Figure 4. Setup for cutting deck modal analysis.

state. While this condition produces acceptable modes for the cutting deck by itself, changes in boundary conditions produce changes in the structural dynamics of the cutting deck. The free-free boundary condition demonstrates how the cutting deck would react if perfectly isolated.

In order to observe the modes of the cutting deck under operating conditions, the cutting deck was left attached to the rest of the table saw structure. This way, the boundary conditions did not have to be simulated, and the actual modes could be viewed. Choosing a frequency range of interest was an important part of the testing setup. The operating frequencies of the motor under the unloaded and loaded conditions were 83 Hz and 79 Hz respectively. The chosen bandwidth of 0-500 Hz allowed for viewing of the modes near 83 Hz and their harmonic frequencies.

With the test setup complete, the measurement degrees of freedom (DOFs) for the cutting deck needed to be determined. These DOFs were chosen considering a number of factors. Points were chosen where the modal participation was known to be greater than zero. Enough points were chosen to provide an accurate amount of spatial resolution to make an appropriate structure of the cutting deck for animation of the natural modes. A basic grid pattern of 49 points was laid out on the cutting deck, with each point spaced approximately 3×4.5 in. (7.62×11.43 cm). Figure 4 shows the setup for the cutting deck modal test.

A roving impact hammer was used to acquire the frequency response functions (FRFs). The response and input force directions for this test were measured only in the vertical (z-axis) direction normal to the cutting deck surface. Ten averages were taken at each point. The overall coherence for the free-free boundary condition showed a strong correlation between the forced input and response output. The coherence was slightly weaker at some points for the fixed boundary condition. To maintain FRF consistency, minimizing normalized random error, more averages were taken at points of low coherence.

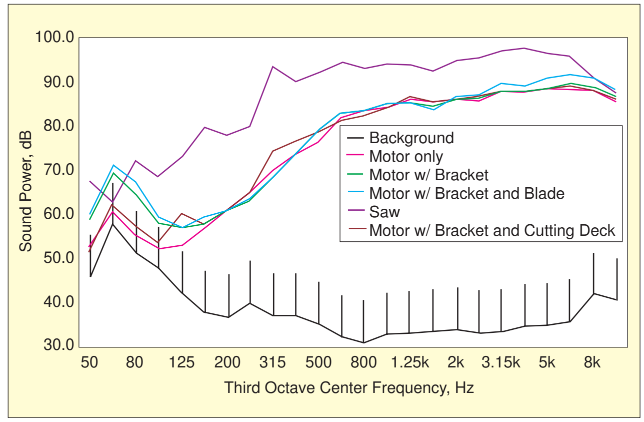

Figure 5. Sound power contribution of various sources within the saw.

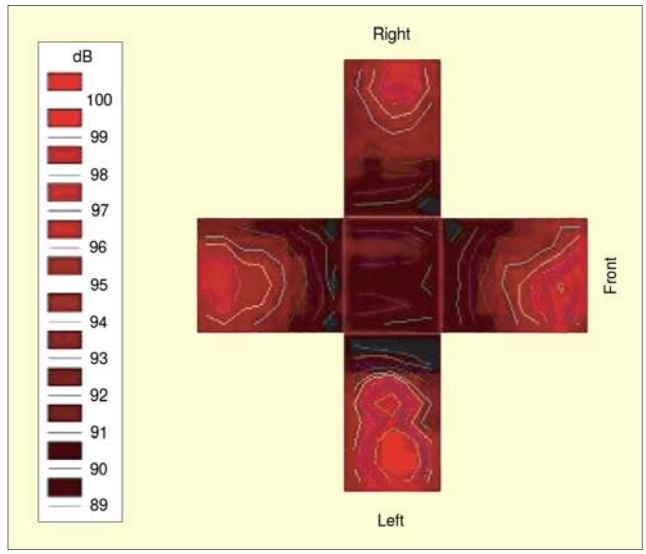

Figure 6. Sound intensity contour map.

After completing all the measurements, the modes were identified. The mode indicator function of the imaginary part located the modal peaks. Single degree of freedom estimation was utilized whenever possible for curve fitting, using the rational fraction polynomial method.[8]

To ensure the validity of the results, three different aspects were observed. First, the structure's modal response was animated to verify that the mode shape vectors truly represented natural modes of vibration of the structure. Second, the modal assurance criterion (MAC) was calculated and monitored on all of the mode shapes. Third, FRF synthesis was used to determine the accuracy of the parameter estimation. The modes lined up well at all but the low frequencies. This misalignment can be attributed to the difficulty of measuring low frequencies with the modal array accelerometer.

Saw Blade Analytical Modal Analysis. In an attempt to correlate the structure-borne vibration to the noise emitted from the table saw, an analytical modal analysis was conducted on the saw blade utilizing SDRC I-DEAS solid modeling software. If the saw blade has a mode at the operating frequency of the motor or one of its harmonics, portions of the blade will be displaced causing a change in sound pressure. This change in sound pressure is a potential source of noise transmitted from the blade.

The Finite Element-based modal analysis in I-DEAS proved the most reliable due to difficulties in adequately exciting the structure experimentally. A quick modal analysis using an impact hammer confirmed the natural frequencies seen in the results of the FEA model.

The material properties assigned to the model were for general isotropic steel, which was assumed to be of similar density, stiffness and yield strength as the blade. The boundary

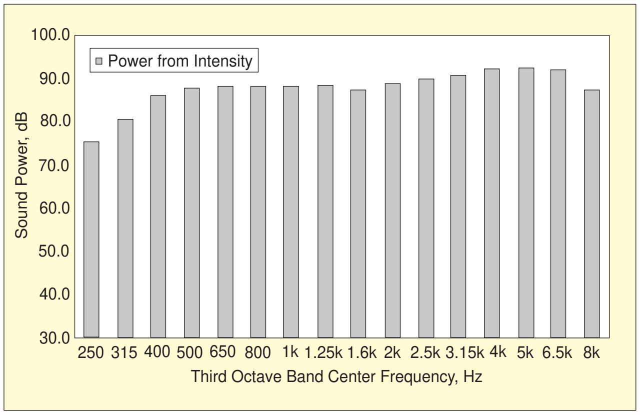

Figure 7. Sound power levels from sound intensity measurements.

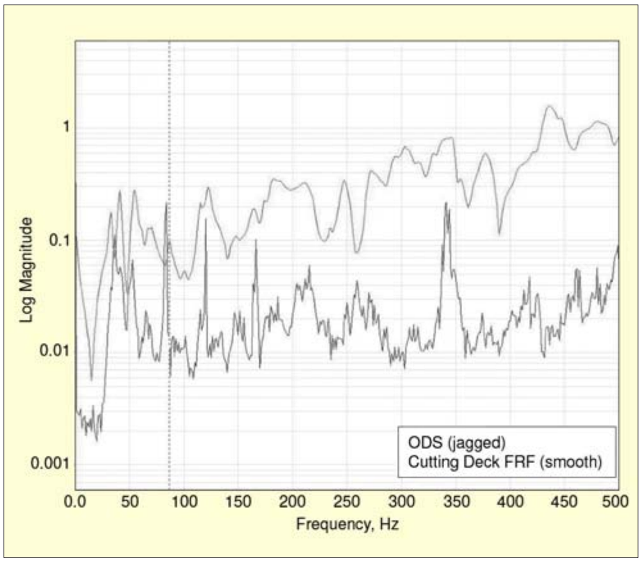

Figure 8. Comparisons of ODS vs. cutting deck modal analysis data.

conditions of the model were set to replicate those found on the table saw assembly: the center of the blade was fixed and clamped around the center hole. The first 10 modes of the system were sought in the analysis, which would include the operating frequency and several of its harmonics.

Results and Discussion

Figure 5 illustrates the sound power contribution of various sources within the table saw. A background noise measurement was taken to ensure the sound power measurements of the components were at least 10 dB above the frequency bands of interest (as indicated by the error bars in Figure 5). This plot indicates that the addition of individual components to the motor does not increase the sound power when compared to the stand alone motor. Therefore, the motor is considered to be the largest sound power source, on a component level, in which the most dominating levels are within the frequency range of 500 to 8000 Hz. The initial recommendation was to investigate ways to reduce SPL by using a quieter motor or treating the existing motor with some path control methods.

The point-by-point sound intensity test produced a detailed one-third octave band sound intensity map (Figure 6) showing where the table saw transmitted the largest amounts of sound energy. Based on the results from this test, the major sources of sound energy were found to be the air paths from the motor and blade. These paths include: air vents on the right and left sides, the adjustment slots on the front, and the open area below the chassis.

Figure 7 shows that the higher frequency octave bands (2500 to 8000 Hz) are the major contributors to the overall sound power. These higher frequency bands were targeted in the treatments applied to the table saw.

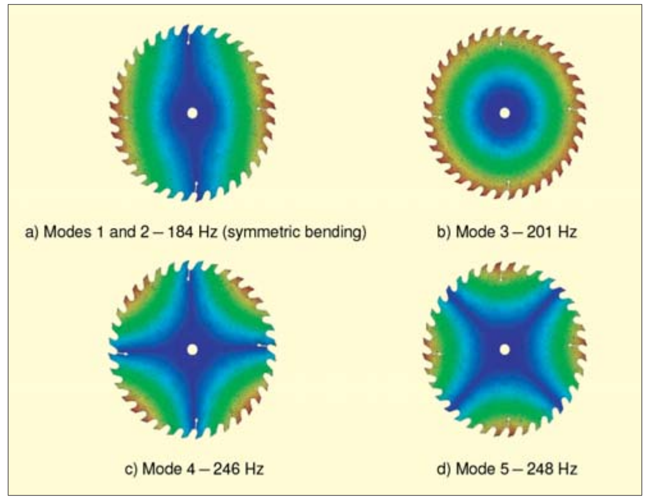

Figure 9. Original saw blade mode shapes for modes 1-5.

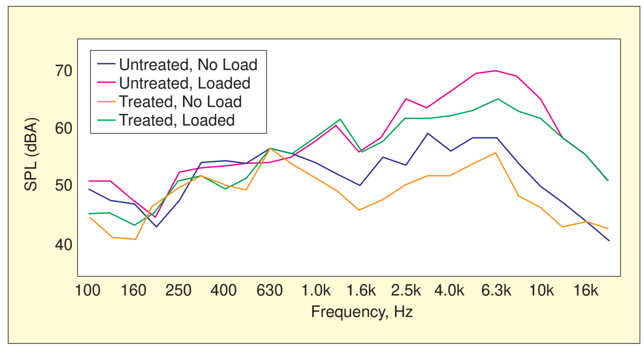

Figure 10. Comparison of the effectiveness of foam and free-layer paint treatments.

The purpose of the ODS testing was to provide a starting point for defining the major sources of structure-borne noise on the table saw structure. The data (not presented here) revealed that the majority of the excitation occurred on the plastic side skirts. This excitation was attributed to airborne noise induced by the blade and motor in conjunction with the low stiffness of the plastic side skirts.

As previously stated, the operating frequency of the table saw under an unloaded condition was around 83 Hz. If the cutting deck were to be the source of any structure borne noise, it would have to be excited by either the operating frequency or one of its harmonics. The modal analysis of the cutting deck was performed to ensure that this did not occur.

The results of the experimental modal analysis revealed that the dominant modes of the cutting deck, under the fixed-fixed boundary condition, occur near the 32, 41, 54, 123 and 250 Hz frequencies. These frequencies show that the cutting deck is not expected to be excited at or near the operating frequency of the table saw under the loaded or unloaded operating conditions. One mode was present near 86 Hz during the unloaded test. This mode, however, was observed to be highly damped in comparison to the other dominant modes. A free-layer paint or other damping treatment can be applied to the cutting deck to further damp this mode, making it less significant. Figure 8 shows a FRF comparison between the ODS and data from experimental modal analysis of the cutting deck. The animated mode shapes exhibited behavior concurrent with predicted mode shapes for a flat plate with the given boundary conditions.

The first five mode shapes from analytical modal analysis of the stock saw blade and the frequencies at which they occur are seen in Figure 9. The fourth and fifth torsional modes lie near the third harmonic of the operating frequency (approximately 249 Hz maximum). Therefore, it could be assumed that these modes could be excited by a harmonic of the saw's operating frequency.

Figure 11. Acoustical foam treatment for chassis interior.

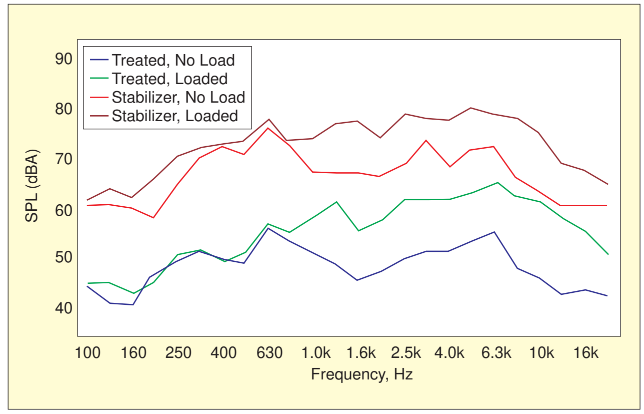

Figure 12. Comparison of effectiveness of blade stabilizer, foam and paint treatments.

Proposed Solution

Determining the best treatments to reduce noise emission from the saw required identification of the major sources and their most dominating frequency ranges. This identification makes it possible to properly match aftermarket treatment options with the primary noise sources. Based on the results of all of the tests described, it is evident that the left side of the structure, where there are several holes/slots, is the source of sound emission with the greatest intensity.

Acoustic foam treatments on the chassis walls can be used to dissipate sound energy. The sound intensity plots indicated that the bulk of the noise emitted from the saw was escaping through ports in and beneath the chassis. Partial sealing of these ports, and the addition of acoustic absorption material inside the chassis are possible. To attack the structure-borne noise elements, damping material can be applied to the chassis walls. Care must be taken in this application, however, as not to introduce hazards such as dust collection, interference with rotating parts, and sufficient cooling/ventilation to the table saw system. Aside from chassis noise, noise emissions from the saw blade were also a concern. Blade stabilizers are a readily available aftermarket treatment designed for noise reduction and cut quality improvement. Also, modifying the saw blade design with cross-drilled hole patterns or gullets can achieve different modal responses, thus tailoring the saw blade to be excited outside of the machine's operating frequencies and harmonics. The following treatment solutions were evaluated based on mock-ups:

1. Free-layer damping paint applied to the bottom side of the cutting deck,
2. Plastic chassis lined with open-cell acoustic foam,
3. After market saw blade stabilizers,
4. Finite element analysis of different saw blade patterns.

Treatments were evaluated using sound intensity in a free field environment for both loaded and unloaded conditions. A comparison of operator SPLs for stock and treated configurations was also performed.

Acoustical Foam and Free-Layer Paint Treatment. The acoustical foam application in conjunction with the free-layer paint treatment proved to be the most effective method of reducing sound pressure and intensity levels. The most noticeable improvement was the reduction of noise propagation from the bottom of the chassis. Figure 10 shows the operator SPL results, comparing the saw with and without treatments in both loaded and unloaded conditions. A picture of the acoustical foam treatment is shown in Figure 11.

The combination of the acoustical foam and free-layer paint treatments significantly decreased high frequency noise, specifically between 1 and 10 kHz. Noticeable improvements, up to 6 dBA, were seen in SPL at the operator's ear location.

Saw Blade Stabilizer. In order to validate the manu-facturer's claim that blade stabilizers quiet the operation of table saws, a blade stabilizer was purchased and tested. The stabilizer used consisted of a pair of steel washers that clamp the saw blade when tightened by the arbor nut. The larger clamping surface provided by the stabilizer is designed to stabilize blade vibration, reducing noise and creating a cleaner, more accurate cut. However in this test, the metal-on-metal clamping configuration seemed to amplify operator SPL results, as seen in Figure 12. Increases of up to 10 dB in critical bands of 1 to 10 kHz indicate the tested stabilizer is a detrimental addition to the saw blade from an acoustic standpoint.

Modification of the Saw Blade Design. Based on ideas from available blade designs on the market and design material recommendations, several iterations of the 10 in. saw blade were designed and tested using the finite element modal analysis (FEMA). Several different approaches to treatment were evaluated: addition of cross-drilled holes in various patterns, changes in gullet size, and changes in gullet configuration (gullets are the symmetrical notches near the outer circumference of the blade between the teeth). The modified blades were designed as shown in Figure 13.

The goal is to shift the natural modal frequencies of the blade further from the operating harmonic frequencies of the table saw. Since modes 4 and 5 were of most concern (246 Hz and 248 Hz respectively), these were the modes of focus for the design changes. The changes were designed to move the natural blade frequencies above the third harmonic for both the loaded and unloaded operating conditions.

Design 2 with the widened gullets shifted the natural frequency for modes 4 and 5 the greatest, out of range of the third harmonic of the saw. Figure 14 shows the deepened gullet design (Design 1) and response at modes 4 and 5, while Figure 15 shows the widened gullet design (Design 2) response at modes 4 and 5. Notice these modes have an increased frequency of excitation at 252 Hz and 253 Hz for designs 2 and 3 respectively. This is safely outside the reach of the third harmonic (approximately 250 Hz maximum at unloaded condition), and should be seriously considered for structural vibration improvement. These recommendations must be validated with experimental testing using prototype blades.

Conclusions

Sound power and sound intensity measurements were used to identify and rank the major airborne noise sources from the table saw. ODS, FEM and experimental modal analysis were used to pinpoint structure-borne vibrations. Several noise control treatments were examined for both feasibility and effectiveness. The treatments selected were validated using average sound intensity and operator SPL measurements.

The main sources of noise emission were identified and ranked as first the motor, then the blade. Structure-borne paths, in order of decreasing severity were the chassis, cutting deck and motor bracket. The airborne paths, also listed in order of decreasing severity, were below and inside the chassis, the

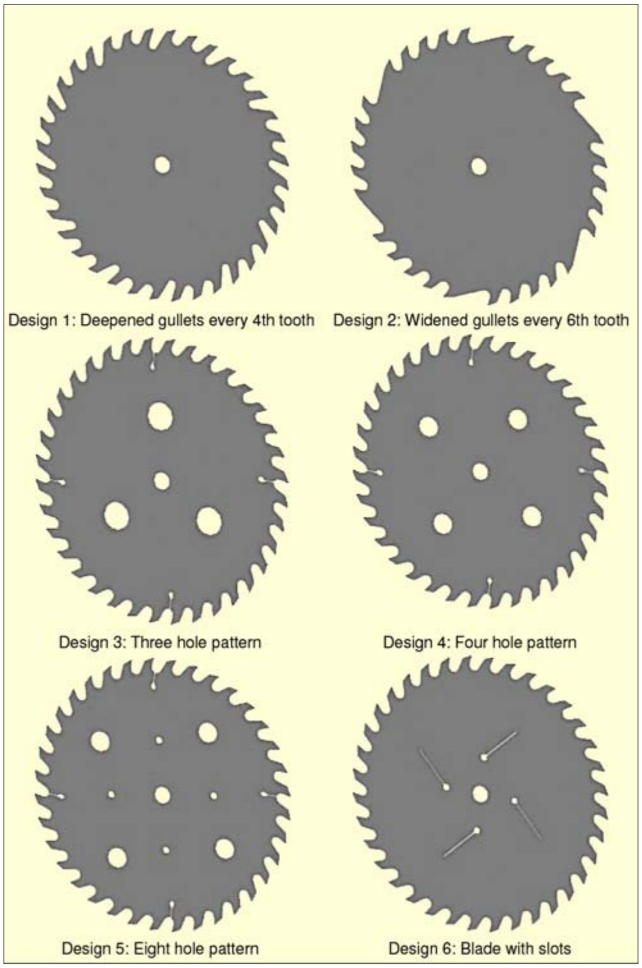

Figure 13. Modified saw blade designs.

exposed blade, and the component adjustment slots/ports.

Sound power reduction of the motor would be the most direct approach to improving operator SPL. However, this is difficult to do without impeding function or significantly increasing the cost of the table saw. Therefore, the more viable solutions lie in path treatments.

Acoustical open-cell foam was applied and found to be effective in attenuating high frequency noise radiating from the chassis openings. Improvements up to 9 dBA in the unloaded condition, and 6 dBA in the loaded state were witnessed using the acoustical foam treatment in conjunction with the free-layer paint treatment in a mock-up study. The saw blade stabilizer was evaluated, and found to have no effect in reducing the operator SPL. It is likely this stabilizer's worth lies more in its cut quality improvement than in its noise reduction attributes. Operator SPL increases of more than 10 dBA in certain third-octave bands make this a poor choice for noise reduction.

Recommendations

The most low cost and effective way to attenuate the harmful noise contributors of the table saw was to apply open-cell acoustic foam to the inside of the plastic chassis. It would be wise to consider open-cell foam with an acoustically transparent yet particle-impervious membrane to avoid collection of sawdust. This membrane could be made of materials such as aluminum, mylar, perforated steel or polyurethane. Furthermore, the design of a more rigid, acoustically damped chassis is recommended as an original manufacturer solution. The saw blade stabilizer is not recommended based on its performance in the operator SPL evaluation.

Further investigation into different saw blade designs, as well as other types of blade stabilizers are also recommended. The modal analysis of different saw blade designs should indicate that blades with widened gullets would stiffen the blade, thus shifting the fourth and fifth modes above the third harmonic of the saw's operating frequency. Another recommendation is to simultaneously correlate SPLs with forced and natural structural vibrations. Analytical software packages, using finite element methods, could also be used to correlate and estimate the acoustic relationships.

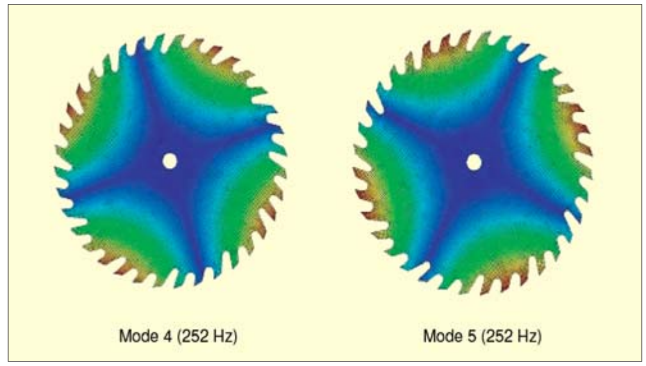

Figure 14. Deep gullet (every 4th tooth) blade – mode shapes 4 and 5.

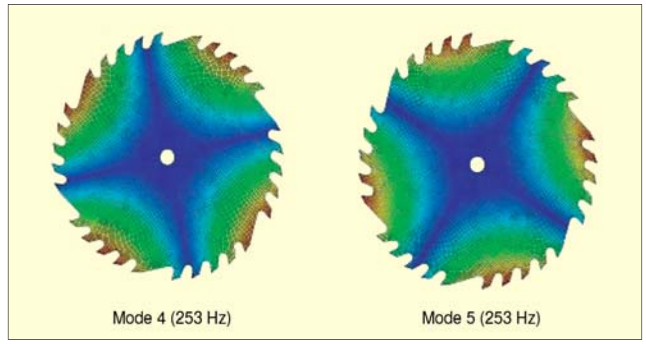

Figure 15. Wide gullet (every 6th tooth) blade – mode shapes 4 and 5.

Just a few examples of treatment solutions are presented here. Many more are possible and should be investigated. However, it has been shown that simple efforts can successfully reduce the amount of harmful noise emission from this consumer-grade table saw.

Acknowledgments

This project was sponsored by the division of Applied Research and Technology of the National Institute of Occupational Safety and Health (NIOSH) with Chuck Hayden serving as technical monitor.

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