

Noise Control of Hydraulically Power Vehicles

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ABSTRACT

For many industrial vehicles, hydraulic or hydrostatic drive systems offer advantages over standard drive systems. This includes high torque at low speeds, elimination of complicated gears and belts, and increased efficiency. Many times these vehicles have hydraulically powered implements, so using a hydraulic drive system simplifies the vehicle design. Acoustically, hydrostatic and hydraulic drives have unique noise control considerations. The sound produced by the pumps and motors can be tonal leading to operator and/or spectator annoyance, even at relatively low sound pressure levels. Sound levels produced by the pumps and motors can also be high, and vibration isolation of the pumps and lines can be difficult as the pressures may exceed 35000 kPa. This in turn can lead to high in-cab and/or high exterior sound pressure levels. This paper presents an overview of hydraulic power systems and several case studies for the noise control treatment development of hydraulically powered machines.

Keywords: Hydraulic, Noise Control, Non-Highway Mobile Equipment
I-INCE Classification of Subjects Number(s): – 11.3, 13.7, 30

1. INTRODUCTION

A hydraulic drive or hydrostatic system uses pressure and flow from a hydraulic fluid to power a machine. Hydraulic systems typically utilize a constant pressure pump system and vary the fluid flow rate to do work. Hydrostatic systems, on the other hand, vary the pressure. Although these systems may have fundamental differences, we will treat them the same from an acoustics point of view and refer to all of the systems in this paper simply as hydraulic systems.

Hydraulic systems have three main noise and vibration sources: A hydraulic pump typically driven by the engine, valves and piping to guide and control the energy in the fluid, and hydraulic motors or other actuators to move the machine or operate the various implements. Having one system to drive the machine and operate the various implements simplifies the design and has the potential to increase the machine's overall efficiency. Since vehicle speed or implement power is controlled via fluid pressure and flow, an infinite number of settings are possible. While these systems have been around for many years, improvements in controls and other system aspects have recently increased their popularity.¹

Depending on the size and design of the machine, the hydraulic system noise sources may be located close together or spread out. Often, the pump is located near the engine and the motors near the wheels or drive mechanism. High pressure piping is used to connect the pumps and motors and can be rigidly attached to the body structure, providing numerous paths for airborne or structure-borne sound to be transmitted to the operator or external receivers. This presents challenges to enclosing the noise sources or designing vibration isolation systems as the source may be distributed throughout the vehicle, or soft mounts may not be possible due to durability concerns. A typical hydraulic pump is shown in Figure 1 along with a prototype enclosure. Effective production enclosures can be difficult due to the number of openings, manufacturing tolerances, and space limitations. In addition, the sound from the pumps and motors can be very tonal, leading to operator and community annoyance. Figure 2 presents a typical in-cab sound level spectrum. The spectrum external to the machine would be similar. Note the high levels and the various tones and harmonics.

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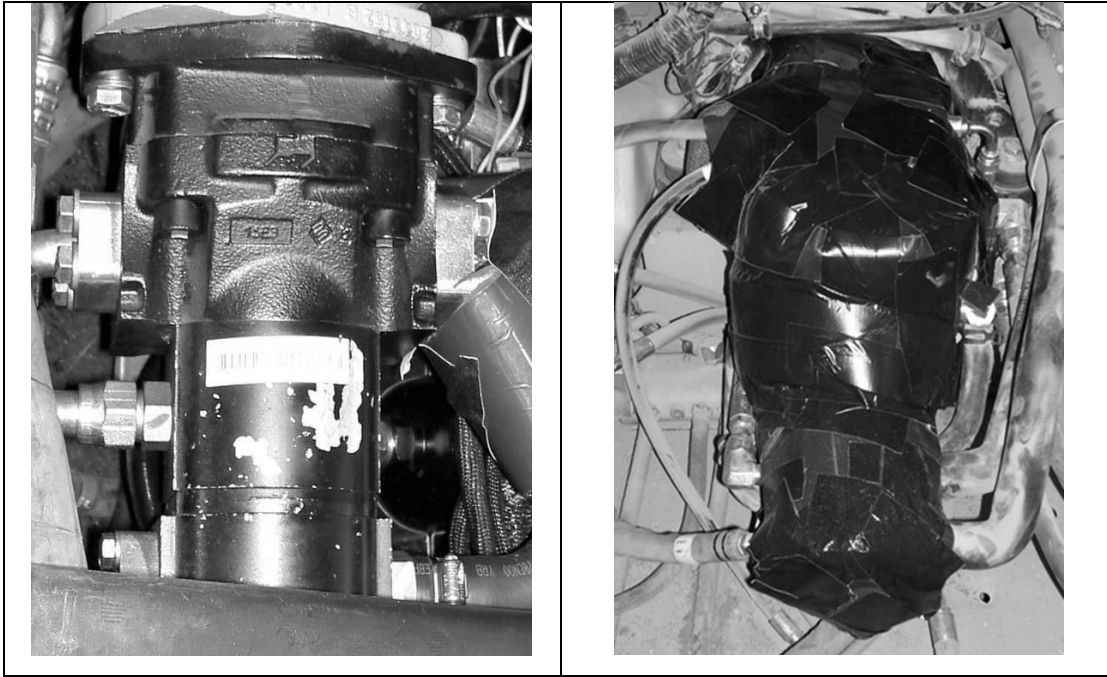


Figure 1 - Typical hydraulic pump with and without an acoustical enclosure

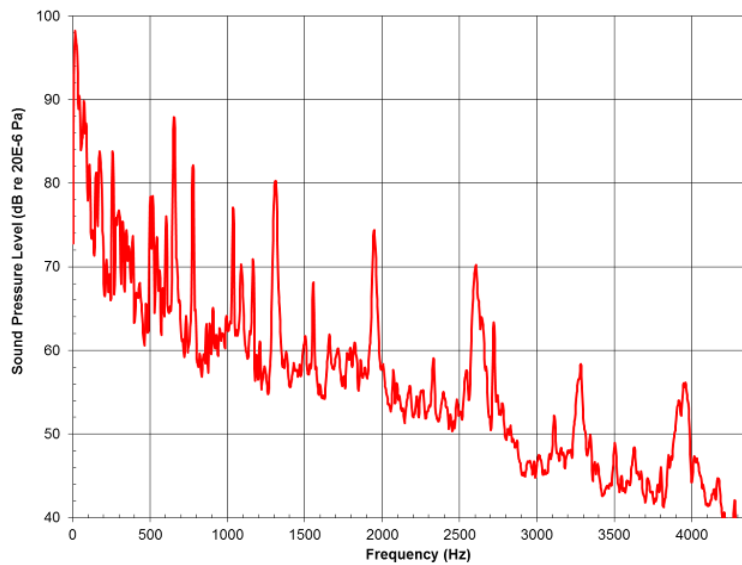


Figure 2 – Typical in-cab spectra

2. CASE STUDIES

In this paper, three case studies are presented: an on-highway machine where the objective is to reduce the sound level in the cab, an off-highway machine where the objective is to reduce the sound power level, and an off-highway machine where the objective is to improve the sound quality inside the cab.

2.1 Case Study 1: On-Highway Machine Cab Interior

The objective in this study was to reduce the sound level at the operator. Depending on the operating condition, the engine, cooling fan, hydraulic implements or drive system, or a combination of these dominated the in cab sound level. For example, at a low-speed, low-load condition, the engine was the dominant noise source and the sound level at the operator's position was about 83 dBA.

As additional load was added to the hydraulic system, additional sound emanated from the pumps and motors so that at high-speed high-load conditions, the dominant noise source became the hydraulics. For this operating condition, the sound level at the operator increased to about 91 dBA.

A next generation unit was in the design process and lessons learned from this project were to be applied to the new machine. Since changing components or the design of the current machine was not possible, enclosures were designed for the hydraulic pumps and motors. The enclosures were made of steel with absorption added to the interior. Photos of the enclosures are shown in Figures 3 and 4.



Figure 3. Implement motor enclosures



Figure 4. Drive motor with and without enclosure

Installation of the enclosures provided a 3 dBA reduction at the operator. A one-third octave band spectrum in the cab with and without enclosures during the high speed high load condition is provided in Figure 5. In the next generation machine, a redesigned cab, lower noise hydraulic pumps and motors, and the addition of sound absorption in the engine compartment provided a noise reduction of about 10 dBA at the operator. For the new machine, enclosures were not utilized.

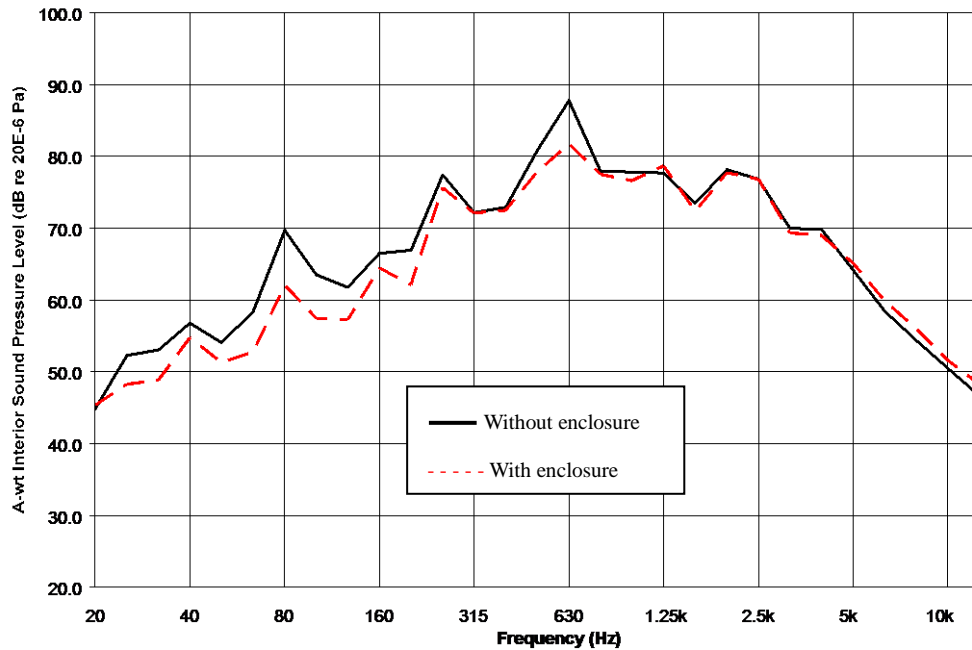


Figure 5. High speed high load condition with and without enclosures

2.2 Case Study 2 – Off-Highway Machine Sound Power Level

The objective in this study was to reduce the sound power level of an off-highway machine. Acoustical holography measurements indicated high sound levels from the hydraulic reservoir tank, shown in Figure 6. It was thought the tank was being excited through structure-borne and fluid-borne connections from the pump. An acoustical cover was developed for the tank and installed along with additional sound absorption in the engine compartment around the tank. A separate add-on treatment was desired as only some of the machines produced required a reduction in the sound power level.

The resulting treatments provided a sound power level reduction of only 0.3 dBA, much less than anticipated. Additional measurements indicated that a bypass valve located near the tank may be the cause. This valve was used to control the speed of the cooling fan which operated on demand instead of at fixed speeds to minimize the sound from the machine. In addition to high sound levels near the valve, high sound levels were also measured near various body panels of the machine. This indicated that the valve was exciting the external body panels, which caused them to be a significant contributor to the overall machine sound power level. Isolation of the valve provided a 1 to 3 dBA reduction in the sound power level depending on the operating condition.



Figure 6. Acoustical holography image of engine compartment

2.3 Case Study 3 – Off-Highway Machine Cab Sound Quality

The objective of this study was to improve the sound quality in the cab of an off-highway machine

during transit, while driving the unit from one job site to the next. The manufacturer had received complaints about the in-cab sound levels during this operating condition. While the overall sound level was acceptable, a high frequency tone was very prominent. The tone was due to the hydraulic system, which was installed in a compartment under the cab. No engineering changes to the system were viable at the time, so an acoustical treatment package was deemed the best solution.

Baseline tests determined that the main tonal component was located in the 2500 Hz one-third octave band, as shown in Figure 7. Narrowband analysis showed the main tone in the 2500 Hz band was approximately 20 dB above the noise floor. It had been established that the noise was primarily airborne, so the dominant paths into the cab were determined using a panel contribution analysis (PCA) technique.

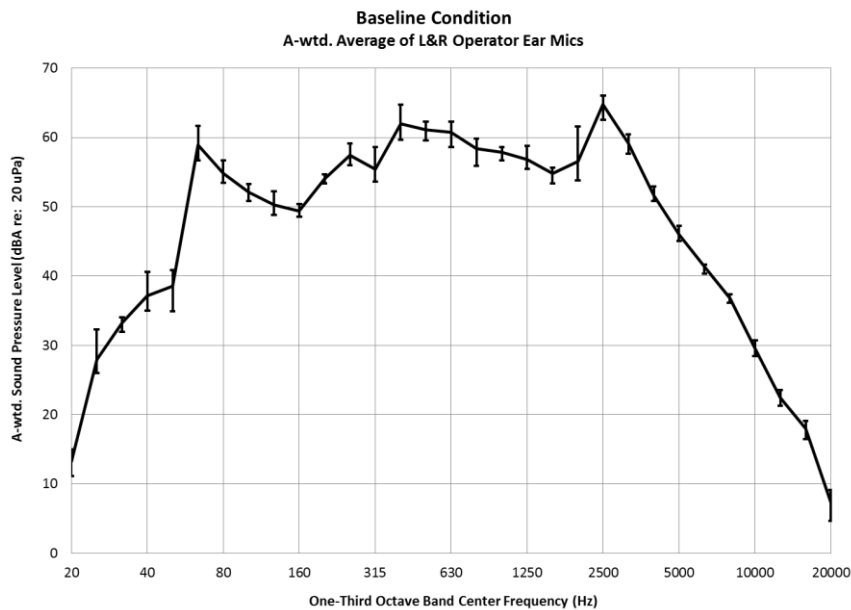


Figure 7. Baseline noise levels for case study 3.

The PCA technique uses sound intensity measurements to determine the sound paths into a large vehicle cab, and has been in use for several years.²⁻³ With this procedure, the cab interior is typically divided into finite surfaces, and each of the dominant sound paths (floor, engine cover, dash, etc.) are then treated with thick, heavy, noise control treatments. Foam and barrier composites with a 25 mm foam decoupler layer and 9.8 kg/m² acoustical barrier are common for this application. With the treatments in place, the sound intensity level of each of the untreated surfaces is measured with the vehicle operating at a steady state condition. Then, one by one, each of the treatments is removed, the sound intensity is measured, and the treatment replaced. The sound power level radiated from each of the interior surfaces is then calculated according to equation 1.

$$\text{Sound Power (L}_w\text{)} = \text{Positive Sound Intensity (L}_I\text{)} + 10 \cdot \log_{10}(\text{Surface Area}) \quad (1)$$

Modeling the cab interior as a small room, the sound power level from each surface is then propagated to each receiver location using equation 2.

$$Lp_i = Lw_i + 10 \log_{10} \left[\frac{Q}{4r^2\pi} + \frac{4}{A} \right] + K \quad (2)$$

- where: Lw_i = the sound power level from each surface
- Lp_i = the sound pressure level at the receiver
- r = the distance from the measurement area to the receiver
- A = the total sound absorption inside the cab
- Q = the directivity factor (in this case $Q = 2$)
- K = a correction factor

While the negative sound intensity is useful for understanding the sound field inside the cab, only the sound that propagates from the measurement surfaces to the receiver is of interest. Therefore, only the positive sound intensity is accounted for in equations 1 and 2.

To determine the sound absorption inside the cab, the reverberation time (T_{60}) is measured and the total absorption is calculated according to equation 3.

$$A = \frac{0.161 * V}{T_{60}} \quad (3)$$

where: V is the interior cab volume

The correction factor K is measured using an omnidirectional loudspeaker of known sound power (Lw_{speaker}). The source is placed near each of the measurement surfaces, the sound pressure level at the receiver (Lp_{receiver}) is measured, and the correction factor calculated from equation 4.

$$K = Lp_{\text{receiver}} - Lw_{\text{speaker}} - 10 \log_{10} \left[\frac{Q}{4r^2\pi} + \frac{4}{A} \right] \quad (4)$$

The sound pressure level from each surface is then logarithmically summed to determine the level at the receiver. PCA results from the test vehicle are shown in Figure 8. The solid black line indicates the predicted sound pressure level, and the solid red line indicates the average measured level. Error bars are included to show variation in the measured in-cab level.

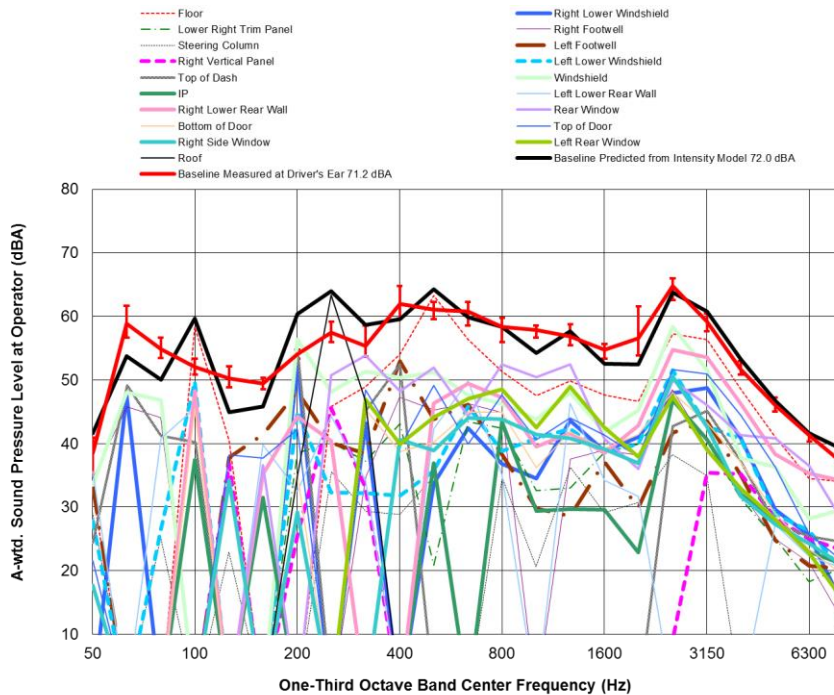


Figure 8. Panel contribution analysis of each scanned surface.

Each of the surfaces can then be ranked according to their contribution to the sound level at the operator's ear, as shown in Figure 9a. Also shown is the path ranking for only the 2500 Hz one-third octave band, as seen in Figure 9b. This shows that the main sound paths in the 2500 Hz band were through the windshield and floor. In addition, the total path contribution due to all of the cab glass is 57%, a significant portion of the noise path into the cab. The cab floor contributes another 22%, bringing the total contribution of the cab glass and floor to 79%. Therefore, the focus of the acoustical treatments needed to address these two main paths.

Sound Path Ranking of As-Received Vehicle (250-8k Hz)			
Location	Sound Level (dBA)	%	Rank
Floor	66.4	32	1
Roof	63.5	17	2
Windshield	61.8	11	3
Rear Window	61.0	9	4
Right Lower Rear Wall	59.1	6	5
Top of Door	57.7	4	6
Left Rear Window	56.0	3	7
Left Footwell	55.3	2	8
Left Lower Windshield	54.4	2	9
Right Footwell	54.2	2	10
Top of Dash	54.2	2	11
Bottom of Door	53.9	2	12
Right Side Window	53.8	2	13
Right Lower Windshield	53.8	2	14
Left Lower Rear Wall	52.9	1	15
Lower Right Trim Panel	51.9	1	16
IP	49.6	1	17
Right Vertical Panel	46.7	0	18
Steering Column	44.4	0	19
A-Weighted Level (250-8k Hz)	71.3	100	
Articulation Index (%)	37.0		
SIL	63.9		

(a)

Sound Path Ranking of Vehicle (2500)			
Location	Sound Level (dBA)	%	Rank
Windshield	58.3	28	1
Floor	57.2	22	2
Right Lower Rear Wall	54.7	12	3
Top of Door	51.6	6	4
Left Lower Windshield	51.6	6	5
Right Side Window	50.5	5	6
Rear Window	49.9	4	7
Right Footwell	48.5	3	8
Bottom of Door	48.1	3	9
Right Lower Windshield	47.9	3	10
Left Rear Window	47.5	2	11
IP	46.8	2	12
Lower Right Trim Panel	46.7	2	13
Top of Dash	42.6	1	14
Left Footwell	41.7	1	15
Steering Column	38.3	0	16
Right Vertical Panel	9.1	0	17
Roof	0.0	0	18
Left Lower Rear Wall	0.0	0	19
2500 Hz A-wtd. Level	63.8	100	

(b)

Figure 9. Path ranking for measured frequency range (a) and in only the 2500 Hz band (b).

Because of the difficulty in treating all of the components of the hydraulic system, an enclosure around the under-cab compartment was created. This not only reduced the sound under the floor, but also the sound emanating out of the under-cab compartment, and into the cab through the glass. Noise control treatments would also be needed to increase the sound transmission loss (STL) of various cab panels such as the floor and glass.

Developed treatments included extensive sound absorption in the under-cab compartment, adding absorption to the rubber skirts along the machine sides and front, utilizing decoupled barrier treatments under the floor and replacing the cab glass with acoustical laminated glass. Prototype acoustical treatments were developed for the machine. The under-cab compartment was treated extensively with a 50 mm polyester fiber absorber. A 9.8 kg/m² acoustical barrier material was adhered to the cab glass to simulate acoustical laminated glass. To increase the floor STL, the interior cab floor was treated with a 7.8 kg/m² barrier and a 25 mm foam decoupler, while the underside of the cab floor was also treated with a 25 mm foam absorber. Finally, the non-glass interior cab walls were treated with a 6 mm foam-7.8 kg/m² barrier-6 mm foam material to increase the STL in other parts of the cab. Figure 10 shows the amount of noise reduction achieved with these treatments. In the 2500 Hz one-third octave band, the sound level was reduced by over 10 dBA. Similar treatments were installed in the production machine and customer response has been positive.

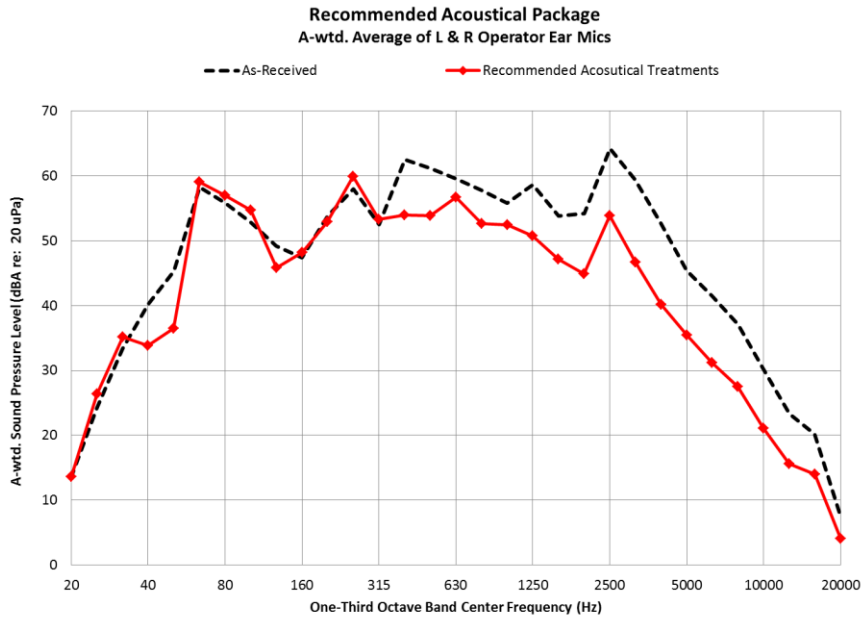


Figure 10. Noise reduction achieved in cab with proposed acoustical treatments.

3. CONCLUSIONS

Three case studies for reducing the sound levels from hydraulically powered machines are presented. While it is possible to provide some modest noise reduction for a current production machine, significant reduction may only be possible through redesign. Improving the sound quality can be equally challenging as the tones from the system are much higher than adjacent levels. Even with substantial noise reduction, the tones can still be audible.

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