Linnéuniversitetet
Institutionen för teknik

Estimate of cabin noise in an articulated hauler with new exhaust piping

Växjö, June
15 points
Tutor: Joacim Linder, Volvo CE
Tutor: Börje Nilsson, Linnaeus University, School of Mathematics
Examiner: Izudin Dugić, Linnaeus University, School of Engineering

Jarosław Jędrzejak
Hamza Ben Saleem

Organisation/ Organization
Linnéuniversitetet
Institutionen för teknik

Linnaeus University
School of Engineering

Författare/Author(s)
Hamza Ben Saleem
Jarosław Jędrzejak

Dokumenttyp/Type of Document
Examensarbete/Diploma Work

Handledare/tutor
Börje Nilsson

Examinator/examiner
Izudin Dugić

Titel och undertitel/Title and subtitle

Estimate of cabin noise in an articulated hauler with new exhaust piping

Abstract (in English)

The method employed in our study estimates the sound pressure level in the cabin when the exhaust pipe is moved to a new location. Using this method we quantified the sound power level of the exhaust pipe and estimated the sound pressure level in the cabin due to the sound source in the cavity of the articulated hauler. This method takes airborne sound into account but not structure borne sound.

In our investigation we have examined a model of the A30F articulated hauler at Volvo Construction Equipment, Braås. All measurements were done at the Volvo CE sound testing facility in Braås.

We investigated two states of the engine, idle and high idle. Idle is when the engine is turned on and standing still on neutral gear when the speed of the engine is 700 rpm, while the high idle is when the engine is standing still and the operator of the articulated hauler pushing in the acceleration pedal, and the speed of the engine is 1900 rpm.

For the idle case the sound level in the cabin is estimated to 16.6 dBA and for the high idle is estimated to 26.7 dBA. The exhaust pipe has a stiff structure resulting in low vibrations and is not a good radiator of sound.

The result we obtained from the measurements and calculations suggests that moving the exhaust pipe to the new position doesn't affect the sound pressure level in the cabin.

Key Words Volvo CE, sound pressure level, sound power level, idle, high idle, exhaust pipe

Utgivningsår/Year of issue	Språk/Language	Antal sidor/Number of pages
2011	English	52

Internet/WWW http://www.lnu.se

Acknowledgement

We wish to thank our academic advisor at Linneaus University, Professor Börje Nilsson for the support and direction that he has given us, along with the assistance of Joacim Linder, Niclas Eliasson and Patrick Ek, Manager at Volvo CE, without their professional help and encouragement, this thesis would not have been completed.

.... ……………………

Jarosław Jędrzejak Hamza Ben Saalem

Abstract

The method employed in our study estimates the sound pressure level in the cabin when the exhaust pipe is moved to a new location. Using this method we quantified the sound power level of the exhaust pipe and estimated the sound pressure level in the cabin due to the sound source in the cavity of the articulated hauler. This method takes airborne sound into account but not structure borne sound.

In our investigation we have examined a model of the A30F articulated hauler at Volvo Construction Equipment, Braås. All measurements were done at the Volvo CE sound testing facility in Braås.

We investigated two states of the engine, idle and high idle. Idle is when the engine is turned on and standing still on neutral gear when the speed of the engine is 700 rpm, while the high idle is when the engine is standing still and the operator of the articulated hauler pushing in the acceleration pedal, and the speed of the engine is 1900 rpm.

For the idle case the sound level in the cabin is estimated to 16.6 dBA and for the high idle is estimated to 26.7 dBA. The exhaust pipe has a stiff structure resulting in low vibrations and is not a good radiator of sound.

The result we obtained from the measurements and calculations suggests that moving the exhaust pipe to the new position doesn't affect the sound pressure level in the cabin.

Table of content

1 Introduction

It is quite commonplace these days to explore natural sources of our planet on a big scale. Big corporations have long engaged in mega construction projects such as building roads, bridges, etc. which normal vehicles cannot reach. In all such big projects around the world, the use of articulated haulers is the best solution to transport tons of material in inaccessible regions. But besides their main function, these huge vehicles should be environmental friendly and safe for people that utilize these vehicles in their work every day. Many engineers are involved in developing technologies; government institutions set standards and regulate emission of noise from machines. Due to aforementioned reasons, engineers are constantly devising new methods involved in reducing noise generated by the machines.

1.1 Background

With the new emission standards taking in 2014, concept studies are being conducted to evaluate different concepts for the new engine installation. In one of the many such concepts, the exhaust pipe will have a different position, which might be worse for internal noise level in the cabin.

To get an indication of how the new piping will affect the internal noise level in the cabin a study is conducted. In this study measurements are done on today's machines. This can provide an early input for the project in the concept studies.

In addition, the study will provide a method to early investigate and predict effects on the internal noise level in the cab from constructional changes in the surrounding.

1.2 History of the company

Figure 1 Bolinder and the Munktell brothers[1]

Volvo CE is the oldest construction equipment company in the world, and remains at the forefront of construction machinery production. The company was established in 19th century when Johan Thoefron Munktell (1805-1887) started, in 1832, a mechanical workshop on behalf of the city of Eskilstuna with the aim of developing the local mechanical industry. Munktell constructed the first Swedish harvester and the first Swedish steam locomotive.

In 1883, two brothers, Jean Bolinder (1813-1899) and Carl Gerhard Bolinder (1818-1892) became famous thanks to the design of the first armed submarine. In the following years they invented the first Swedish combustion engine and the steam roller.

In 1927 in Stockholm, Assar Gabrielsson and Gustaf Larsson established an automobile company aiming at the public. It was the early keystone of Volvo CE.

In 1932, Sweden faced mounting economical problems, as a result of the Great Depression in USA. Then the Handelsbanken bank decided to save assets within the engineering industry. Consequently, companies of Munkel and Bolinder brothers were merged into AB Bolinder-Munktell company.

After the Second World war, the future of AB Bolinder-Munktell was uncertain, as the Handelsbanken bank wanted to break free the dependence of the industry. AB Bolinder-Munktell recieved two offers, one was International Harvester from USA, while the second one was Volvo, which successfully acquired Bolinder-Munktell in May of 1950.

In the late 1950s, in Braås, Livab was experimenting with machines to connect driven hauler trailers and tractors. Development of the machine picked up speed in 1965, when Bolinder-Mounktell signed agreement with Livab. One year later Volvo BM introduced world's first series manufactured articulated hauler DR 631.

Twenty years later Volvo merged with the American companies Clark and Euclid and formed the VME-group, which established Volvo Construction Equipment in 1995.

Nowadays Volvo CE has about 35 per cent of the worldwide market. With the new F-series of articulated hauler the company wants to win in the Asian market. [1]

1.2.1 Articulated hauler Volvo A30F

The machine on which we carried out measurements is one out of four available models; A30 F has 6-cylinder engine, which has 11 liters of capacity and 357 horsepower. Maximum load is 28 000 kg, and then articulated hauler weight is 51 200 kg.

Figure 2 Articulated Hauler Volvo A30F

1.3 Problem formulation

Moving the exhaust pipe to the shoulder cavity of the articulated hauler will affect the sound pressure level in the cabin. We have to quantify the sound power level of the exhaust pipe in the present machine and then place a sound source in the shoulder cavity to determine the sound pressure level in the cabin. This will lead to the following problem:

Determine the sound pressure level L_{p2} in the cabin from the airborne sound from vibration pipe at the new position 2, which is the shoulder cavity of the articulated hauler.

1.4 Purpose

The purpose of this project is to quantify the sound source strength of the pipe and use a reference sound source (RSS) with the same strength in the shoulder cavity of the articulated hauler to determine the effect on the internal noise in the cabin.

1.5 Limitations/delimitations

We do not have access to a prototype. Therefore the measurements are done on the present machine Volvo A30F which would give us an estimate.

The method we are using is not exact but it is better than calculation not based on measurement result.

The assumptions and their limitations that are used in the method we use are the following.

These assumptions and their limitations are:

1. The sound power level of the reference sound source (L_{wRSS}) is not constant. It has negligible effect and we can neglect it.
2. The reference sound source (RSS) and real sound source has not the same directional radiation properties.
3. The real sound source has dimensions that are not small in comparison with the distance to the cabin.
4. RSS will only give airborne sound where as the real sound source will also give structure borne sound.

The errors associated with the first assumption can easily be neglected. Errors associated with the 2nd and 3rd assumptions are important errors which will gives us error of several dB.

Errors associated with the 4th assumption are inherit in the method since the method only takes airborne sound into account and not structure borne sound.

2 Research methodology

To estimate the sound level in the cabin by moving the exhaust pipe to a new location, two methods can be employed.

The first method is to use sound intensity which is theoretically the best one but practically it is complicated since the contribution from each component of the machine has to be isolated. Therefore, we did not use this method.

The second method is to determine the sound power level of the exhaust pipe, calibrate the sound source, and then place a sound source in the cavity to determine the sound pressure level in the cabin. This is the method we have used in our study. Below there is a schematic diagram of this method along with the equations.

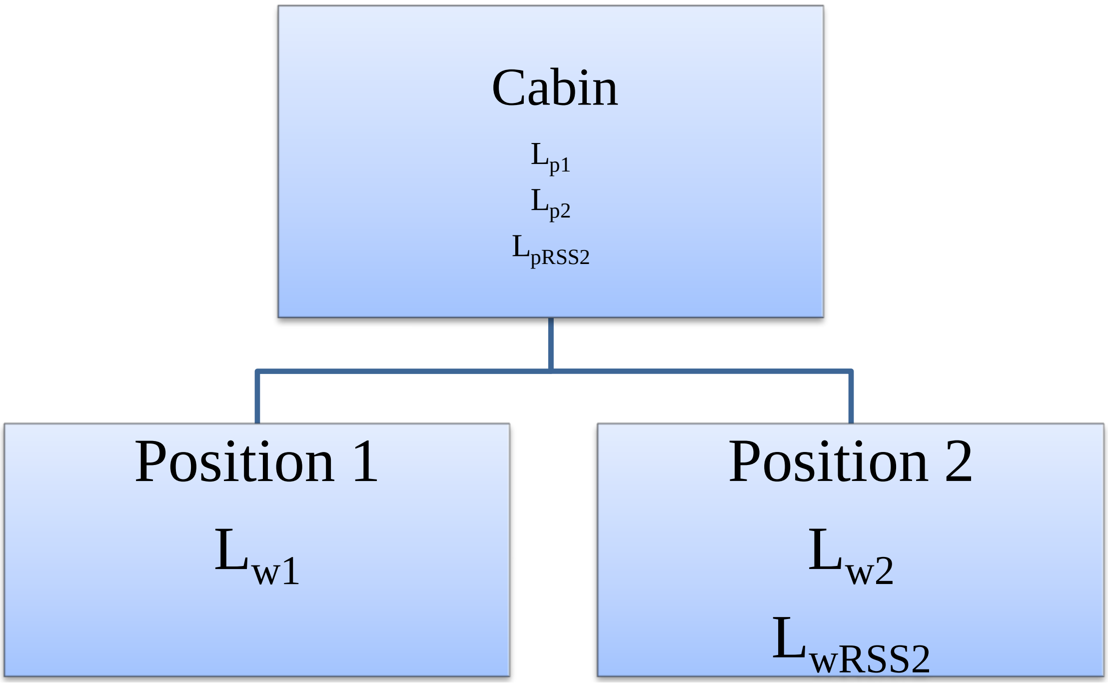


Figure 3 Schematic drawing

Position 1: The location of the exhaust pipe.
Position 2: The location where the exhaust pipe is going to be in the future.

Assumptions:

$$L_{w1} - L_{p2} = L_{wRSS2} - L_{pRSS2} \qquad \text{Eq. 2.1}$$

$$L_{w1} = L_{w2} \qquad \text{Eq. 2.2}$$

leads to:

$$L_{p2} = L_{w1} - L_{wRSS2} + L_{pRSS2} \qquad \text{Eq. 2.3}$$

which is valid for each frequency band. The radiation is same in all directions.

Here:
- L_{w1} - sound power level of the exhaust pipe
- L_{p2} - sound pressure level in the cabin (this is what we have to find)
- L_{wRSS} - sound power level of the reference sound source
- L_{pRSS2} - sound pressure level in the cabin due to the sound source in the cavity

Below we explain how L_{w1}, L_{wRSS2}, L_{pRSS2} are determined.

The sound power level L_{w1} of the exhaust pipe is determined by mounting accelerometers on the exhaust pipe and using the equation below. The equation is valid for each frequency band:

$$L_{w1} = L_v + 10 \cdot \log S + C + R_e \quad \text{[dB]} \qquad \text{Eq. 2.4}$$

Here:
- L_v – vibration velocity level (dB)
- S – surface of the pipe
- C – correction factor
- R_e - radiation efficiency of the pipe
- logarithm base is 10

The correction factor C and the radiation efficiency R_e of the pipe are given in appendices 3-5.

The accelerometers measured the velocity of the vibrating exhaust pipe in the normal direction to the surface only since the exhaust pipe doesn't radiate sound in other directions. We measured it using the 1/3 octave band. The velocity data we obtained from the measurements were used in the equation for each frequency band, and the logarithmic mean value of the sound power level was calculated.

L_{wRSS2} is determined by calibrating the sound source outdoors at the Volvo CE sound testing facility in Braås and using the equation below, which is valid for each frequency band:

$$L_w = L_p + 10 \cdot \log S \quad \text{[dB]} \qquad \text{Eq. 2.5}$$

Here:
- L_p – sound pressure level (dB)
- S – surface of the measurement half sphere (radius is 2 meters)
- L_w – sound power level of the sound source (speaker)

The calibrated sound power level of the sound source L_{wRSS2} as function of feeding voltage, valid for each frequency band, is:

$$L_{wRSS2} = L_{pRSS2} + 10 \cdot \log S - 20 \cdot \log\left(\frac{U_1}{U_2}\right) \quad \text{[dB]} \qquad \text{Eq. 2.6}$$

Here:
- U_1 – the voltage at which the sound pressure level is calibrated.
- U_2 – the voltage at which the sound pressure level in the cabin is measured.

- L_{wRSS2} – calibrated sound power level of the sound source.
- L_{pRSS2} – sound pressure level in the cabin

Both U_1 and U_2 are 1/3 octave band values that are functions of frequency.

L_{pRSS2} is determined by placing two microphones inside the cabin which measure the sound pressure level due to the sound power level of the sound source in the shoulder cavity of the articulated hauler when the engine is turned off.

The equation we use to calculate the logarithmic mean value for each frequency band is:

$$L_{pRSS2} = 10 \cdot \log\left(\frac{10^{0,1 \cdot L_{p1}} + 10^{0,1 \cdot L_{p2}}}{2}\right) \quad [\text{dB}] \qquad \text{Eq. 2.7}$$

Here:
- L_{p1} - sound pressure level of the left microphone
- L_{p2} - sound pressure level of the right microphone

3 Theory

In this chapter basic definition are explained. Description of measurement components you can find in appendix 2.

3.1 Sound, frequency

Sound can be defined as vibrations transmitted by gas, fluid, solid or elastic material, that can be detected by human ear[2]. These vibrations are periodic pressure variations, where the number during one second is called frequency. The frequency is measured in Hertz (Hz). The range of hearing for human ear is from 20 Hz up to 20 kHz [3]. Sound wave travel through different media with different speed. For example, for air it is 340 m/s, while for longitudinal waves in steel it is around 5000 m/s. In acoustics exist two basic types of waves: longitudinal and transverse waves. Particles of longitudinal waves move parallel to the direction of the disturbance transmission, while particles in a transverse wave move perpendicular to the direction of the propagation [4].

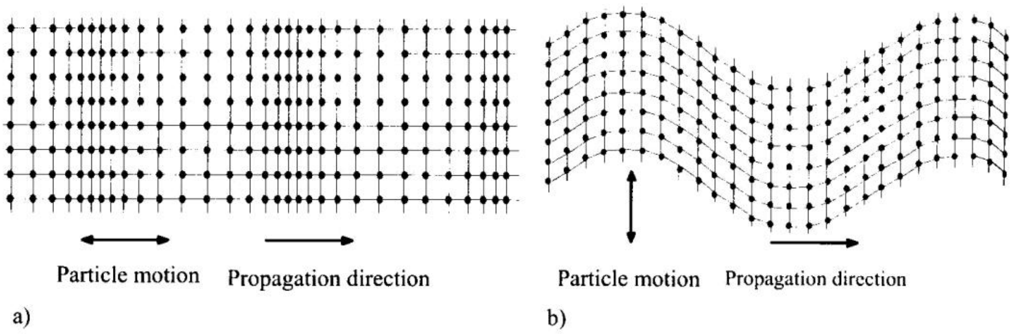


Figure 4 Wave types in solid media: a) longitudinal, b) transverse [4]

3.2 Sound pressure level

The Sound Pressure Level (SPL) is the logarithmic measure of the instantaneous pressure at a certain point related to the sound reference value [5]. The unit is decibel (dB). That is:

$$L_p = 10 \cdot \log \frac{p^2}{p_{REF}^2} = 20 \cdot \log \frac{p}{p_{REF}} \quad [\text{dB}] \qquad \text{Eq. 3.2.1}$$

Here:
- p is the RMS value (unless otherwise stated) of the sound pressure in Pascal,
- p_{REF} is 20 µPa, which is commonly considered as the threshold of human hearing.

3.3 Sound power level

The Sound Power Level is defined as the logarithmic measure of the absolute value of the sound power generated by a source related to the specified sound power reference level [6]. The unit is watt (W). That is:

$$L_w = 20 \cdot \log\left(\frac{|P|}{P_{REF}}\right) \quad [\text{dB}] \qquad \text{Eq. 3.3.1}$$

Here:
- P is the value of sound power of the source,
- P_{REF} is 10^{-12}W.

3.4 Weighting filters

Weighting filters are commonly used in acoustics to amplify differently the signal of the microphone in different frequency ranges. Human hearing adjusts to both frequency and strength of the sound [4]. The human ear is most sensitive in the medium range of audible frequencies, while in very low and very high ones it is insensitive. Weighting filters emphasize or attenuate frequencies and in case of human perception of sound, A-filter is the most common one.

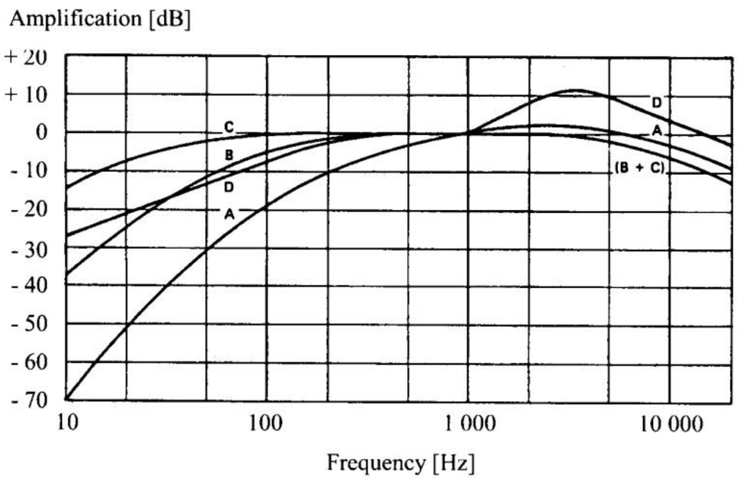


Figure 5 A, B, C, D-weighting curves [4]

4 Measurement setup

Schematic diagrams of each measurement setup are given in appendix 1.

4.1 Setup for measuring the sound power of the exhaust pipe

Figure 6 Three accelerometers mounted on the exhaust pipe

Three plates were welded on the exhaust pipe and the accelerometers were mounted on these welded plates. The exhaust pipe is 1 meter long and has a diameter of 0.127 meters. The accelerometers can stand a maximum temperature of 230˚C. But the temperature of the exhaust pipe exceeds 230˚C. Therefore we have to cool the accelerometers so they can function properly.

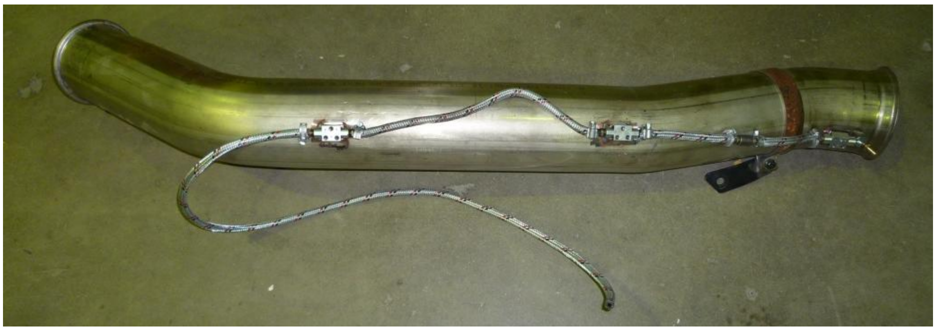


Figure 7 Metal pipe for cooling system mounted on top of the accelerometer

Figure 8 Accelerometer with cooling system mounted on the exhaust pipe

Figure 9 Cooling system

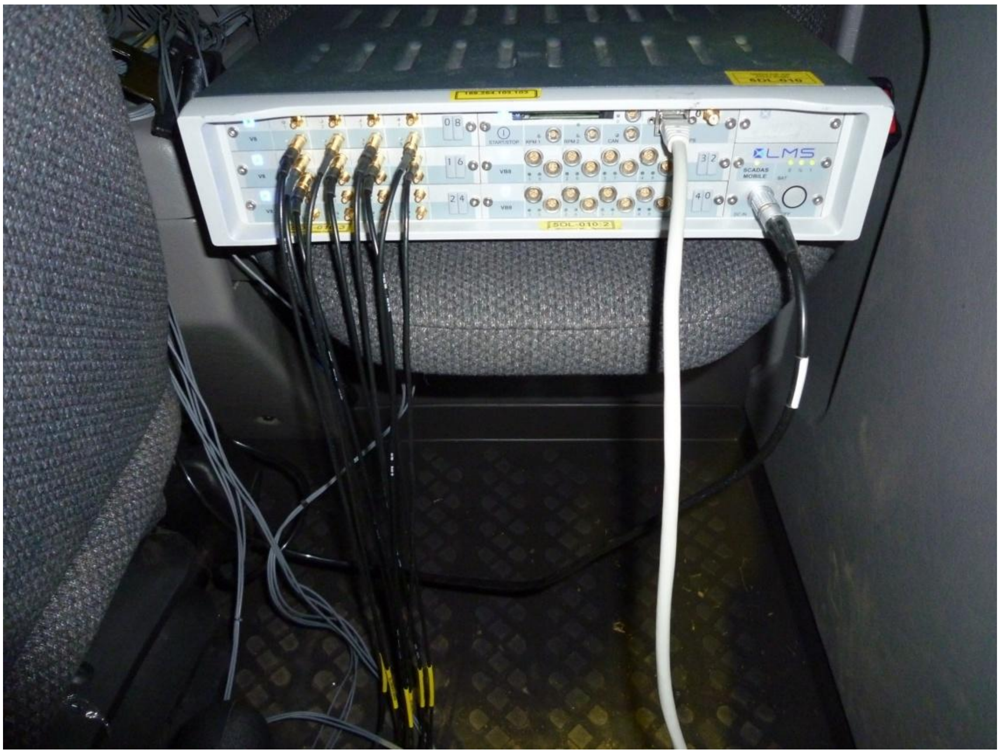


Figure 10 LMS SCADAS Mobile Front-end during measurements

The accelerometer has 3 directions x,y,z and are attached to the channels in the front-end measuring system. The front end is attached to the PC and using LMS software the data obtained from the measurements.

4.2 Setup for calibrating the sound source

Figure 11 Setup for calibrating sound source - on the picture sound source and two microphones

This is the sound testing facility at VOLVO CE. Two microphones placed in the half of a sphere to measure the sound pressure level. The ground is hard, reflecting and there are no reflections from buildings etc.

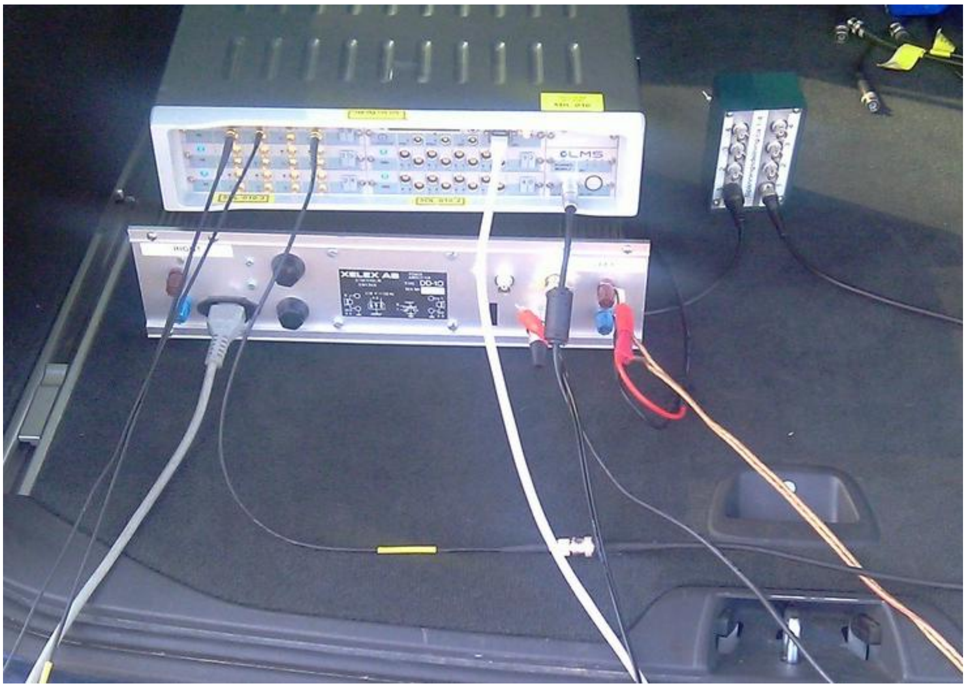


Figure 12 Setup for calibrating sound source – on the picture amplifier, front-end and splitter

LMS Front-end and Amplifier. The sound source is connected to the amplifier and the LMS front End is connected to the PC. The splitter divides the voltage of the sound source since the front-end can only take 12 V.

4.3 Setup for measuring the sound pressure level inside the cabin

Figure 13 The Cavity where we put the sound source

The cavity where we put the sound source(speaker). A wave file was generated from the accelerometers measurement. We used it as a signal generator to simulate the sound from the exhaust pipe. The microphones measured the sound pressure level in the cabin while the engine was turned off and the only source of sound was through the speaker.

Figure 14 The sound source is mounted inside the cavity

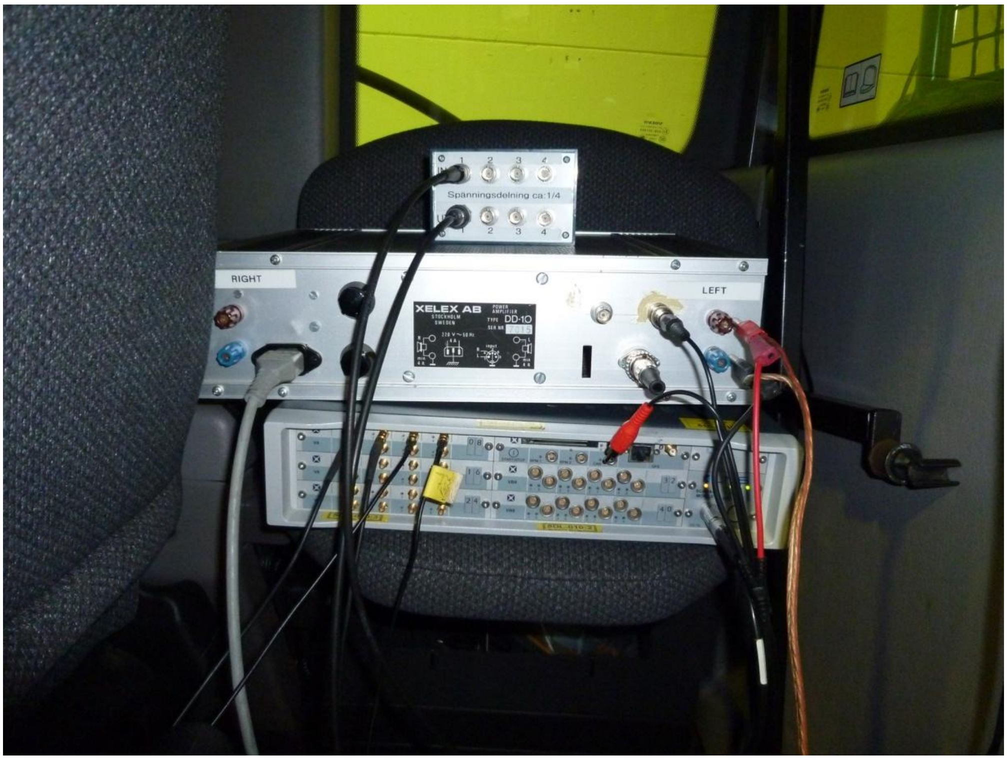


Figure 15 Setup for measuring the sound pressure level inside the cabin

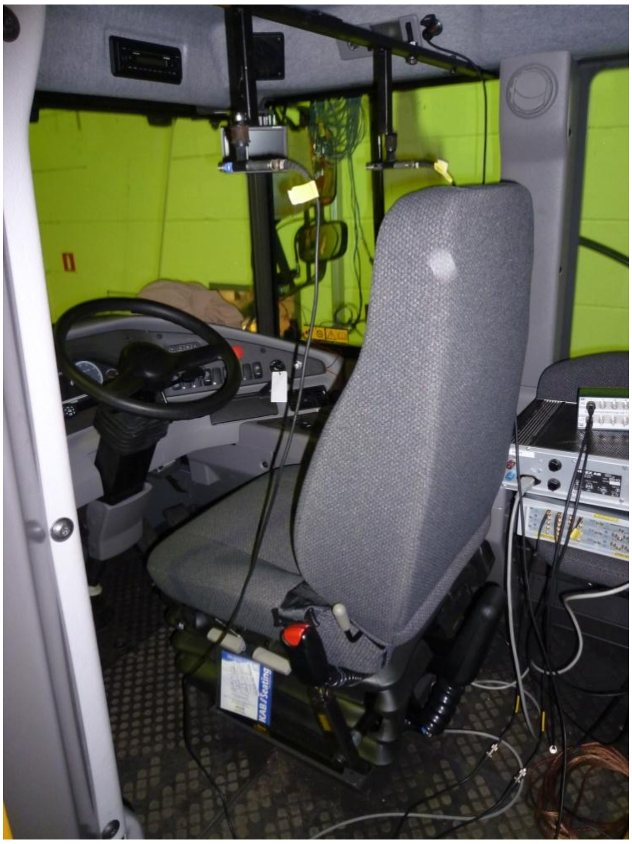


Figure 16 Two Microphones mounted on a stand to measure the sound pressure level inside the cabin

Two Microphones mounted on a stand to measure the sound pressure level inside the cabin when we played the exhaust pipe sound in the speaker.

5 Results

Here, we present the final result of our calculations in total levels in dBA. We did all the measurements and calculations using 1/3 octave bands. Results in 1/3 octave bands are shown in appendix 8. However, these results don't take frequencies below 80 Hz, including the first harmonic and the fundamental frequency, into account.

The sound power level L_{w1} of the exhaust pipe is determined by Equation 2.4 and the sound pressure level L_{p2} in the cabin is determined by Equation 2.3, that is rewritten as:

$$L_{p2} = L_{w1} - (L_{wRSS2} - L_{pRSS2})$$

We use the reference sound source to determine $L_{wRSS2} - L_{pRSS2}$, and the result is presented in the figure 3 in appendix 8.

It is worth to mention here that the sound source was wrapped in plastic as shown in figure 14. We assume that it has negligible effect on the reference sound pressure level L_{pRSS2}.

The results presented below are for 3 cases. in all cases we have $L_{w1}=L_v+10\log S+C+Re$, where C=146,1 dB. For the three cases we have the following:
Case 1: Re=0
Case 2: Re for n=0 (monopole)
Case 3: Re for n=1 (dipole)

For details on *Re*, see appendices 4 and 5.

We investigated two states of the engine, idle and high idle. Idle is when the engine is turned on and standing still on neutral gear when the speed of the engine is 700 rpm, while the high idle is when the engine is standing still and the operator of the articulated hauler pushing in the acceleration pedal, and the speed of the engine is 1900 rpm.

The final results are given below in total dBA-levels:

The total A-weighted sound power level L_{w1A} of the exhaust pipe (High idle) for 3 cases:
- Case 1 L_{w1A}=38.6 [dBA],
- Case 2 L_{w1A}=51.8 [dBA],
- Case 3 L_{w1A}=50.3 [dBA].

The total A-weighted sound power level L_{w1A} of the exhaust pipe (Idle) for 3 cases:
- Case 1 L_{w1A}=27.8 [dBA],
- Case 2 L_{w1A}=28.3 [dBA],
- Case 3 L_{w1A}=35.3 [dBA].

.

The total A-weighted sound pressure level in the cabin (High idle) for 3 cases:
- Case 1 L_{p2A}=26.7 [dBA],
- Case 2 L_{p2A}=19.9 [dBA],
- Case 3 L_{p2A}=18.4 [dBA].

The total A-weighted sound pressure level in the cabin (Idle) for 3 cases:
- Case 1 L_{p2A}=16.6 [dBA],
- Case 2 L_{p2A}=2.4 [dBA],
- Case 3 L_{p2A}=1.1 [dBA].

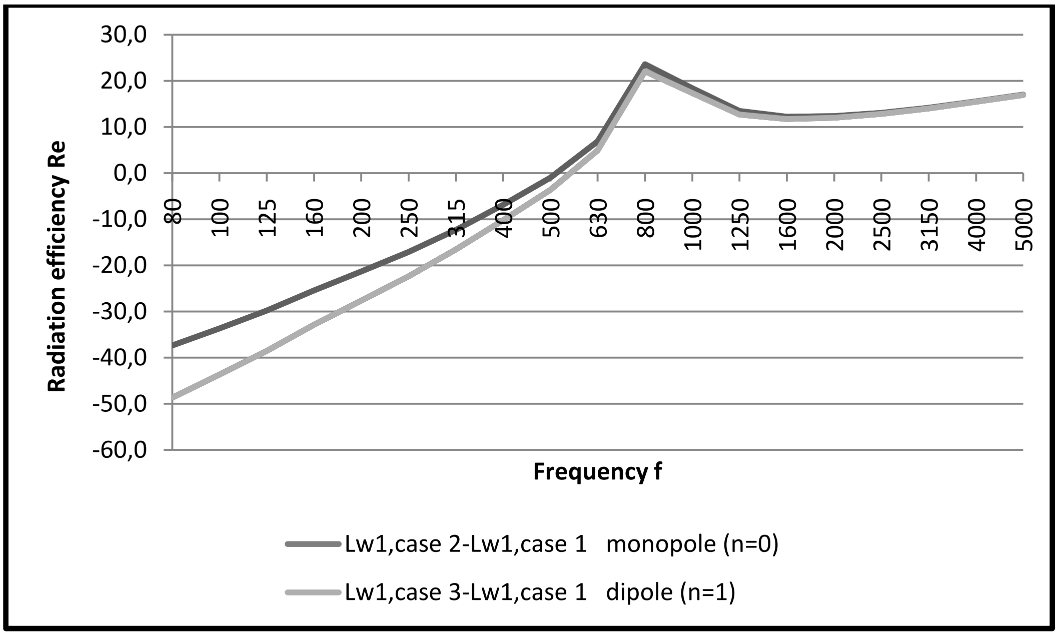


Figure 17 Difference between cases

Figure 17 shows the radiation *Re* of the exhaust pipe as function of frequency. At any given frequency we can find the corresponding radiation efficiency of the exhaust pipe. Note that $L_{w1,case2}$-$L_{w1,case1}$=*Re* for *n*=0 and $L_{w1,case3}$-$L_{w1,case1}$=*Re* for *n*=1.

On the next page we present the final results in 1/3 octave band.

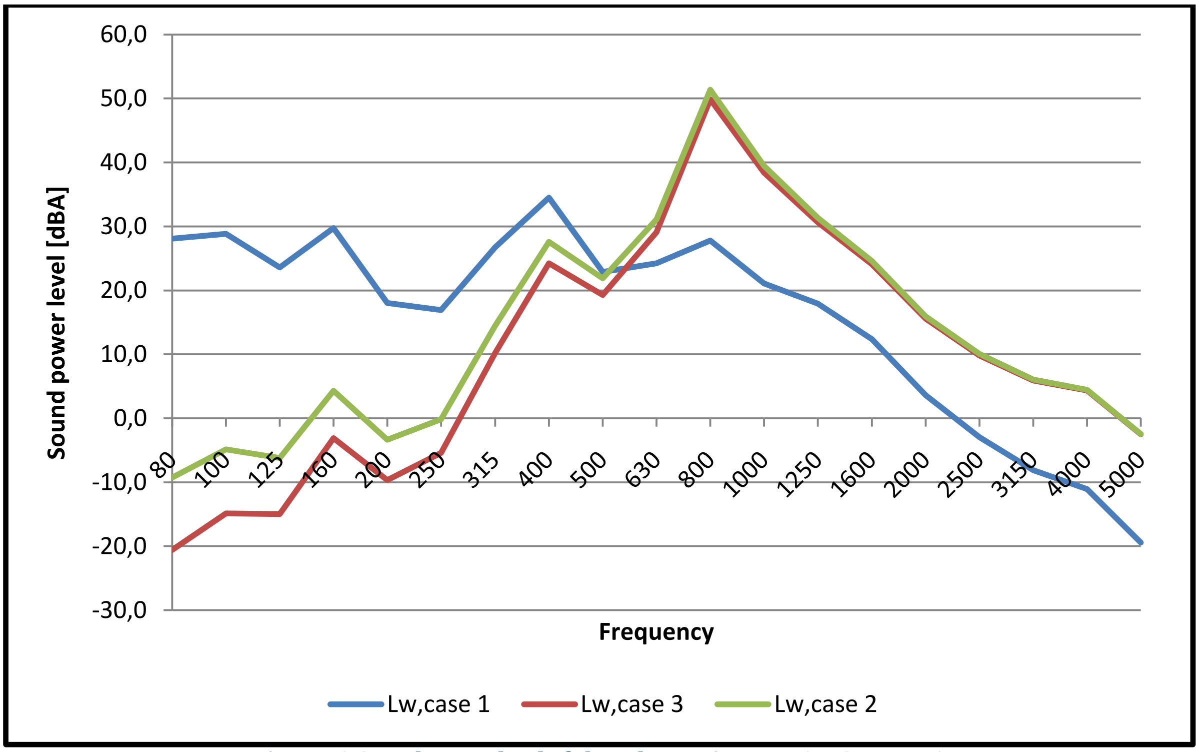


Figure 18 Sound power level of the exhaust pipe Lw1 (HIGH IDLE)

Figure 18 shows the sound power level L_{w1} of the exhaust pipe as a function of frequency for the high idle state. At any frequency a corresponding value of sound power level L_{w1} can be obtained from the graph.

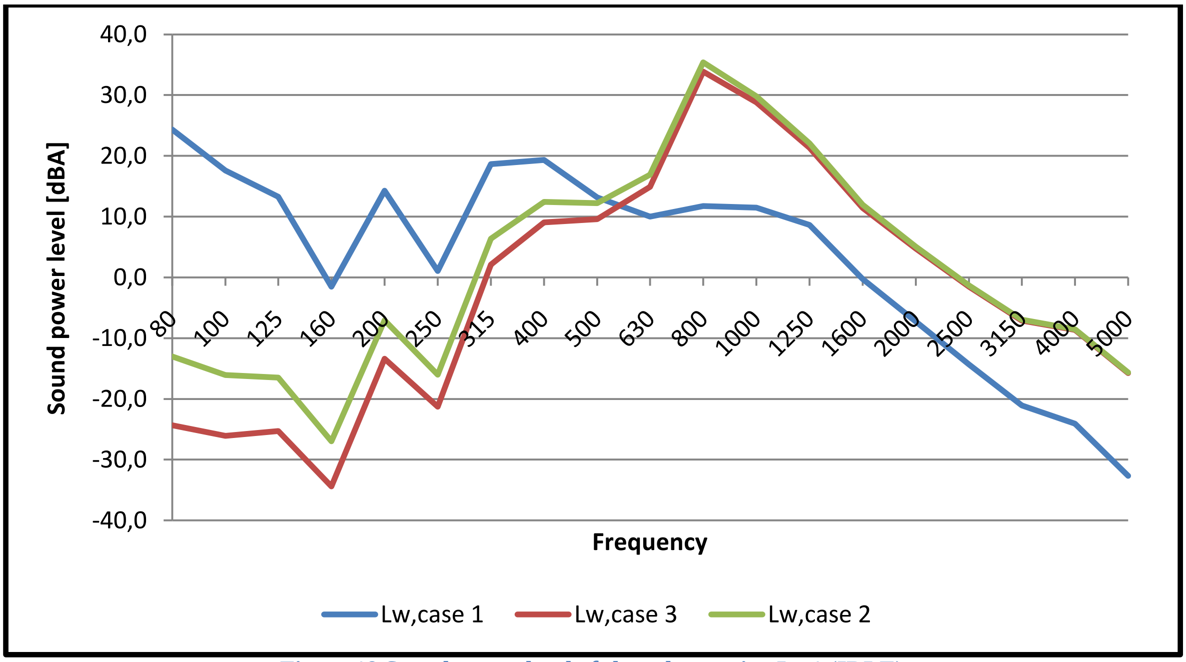


Figure 19 Sound power level of the exhaust pipe Lw1 (IDLE)

Figure 19 shows the sound power level L_{w1} of the exhaust pipe as a function of frequency for the idle state. At any frequency a corresponding value of sound power level L_{w1} can be obtained from the graph.

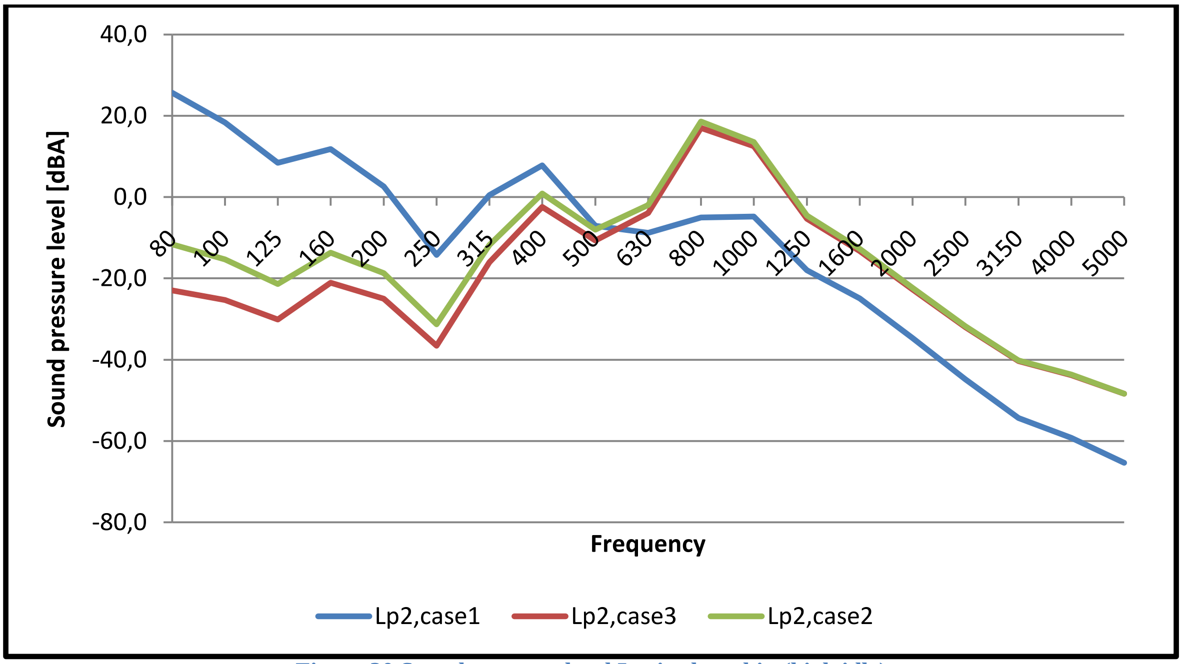


Figure 20 Sound pressure level L_{p2} in the cabin (high idle)

Figure 20 shows the sound pressure level L_{p2} as a function of frequency for the high idle state. At any frequency a corresponding value of sound pressure level L_{p2} can be obtained from the graph.

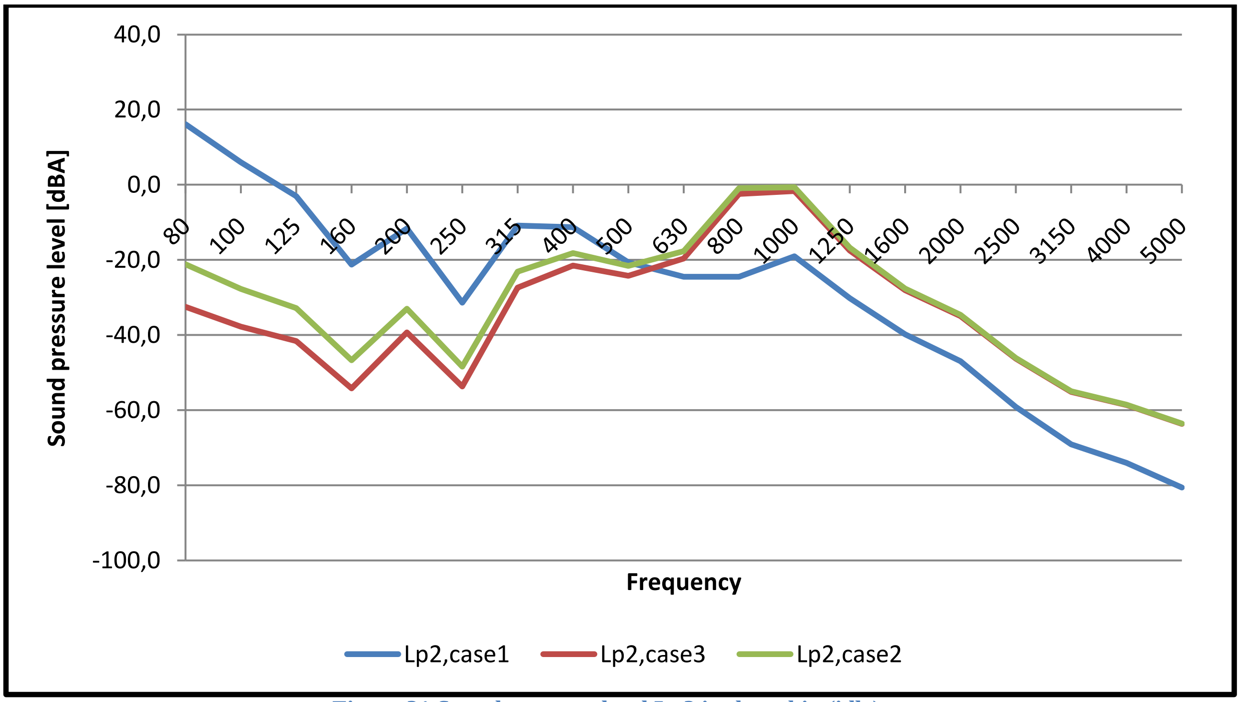


Figure 21 Sound pressure level Lp2 in the cabin (idle)

Figure 21 shows the sound pressure level L_{p2} as a function of frequency for the idle state. At any frequency a corresponding value of sound pressure level L_{p2} can be obtained from the graph.

6 Analysis and discussion

Here, we analyze the results presented in the previous section. First we analyze the radiation efficiency *Re* of the pipe for different cases (Fig.17), then the sound power level L_{w1} (Fig. 18, 19) of the exhaust pipe, finally we discuss result for the main purpose of interest, namely the sound pressure level L_{p2} in the cabin (Fig. 20, 21) are given below.

Figure 17 shows that at low frequencies there is a large difference between the radiation efficiency for the 2 cases whereas at high frequencies above 800 Hz, the radiation efficiency for both cases are identical . At 800 Hz the radiation efficiency for both cases is peaking there. Figure 18 shows that at low frequencies the sound power level of the exhaust pipe for each case is not dominating. For case 2 and 3, the sound power level is at its peak around 800 Hz. The reason for this is that the radiation efficiency is peaking there.

For frequencies above 2000 Hz the sound power level of the exhaust pipe is of minor importance for all three cases due to the spectrum of the sound source. For frequencies below 300 Hz the sound power level is of minor importance for case 2 and 3 due to reduced radiation efficiency.

Figure 19 shows that at low frequencies the sound power level of the exhaust pipe for each case is not dominating. For case 2 and 3, the sound power level is at its peak around 800 Hz the reason for this is that the radiation efficiency is peaking there.

For frequencies above 2000 Hz the sound power level of the exhaust pipe is of minor importance for all three cases due to the spectrum of the sound source. For frequencies below 300 Hz the sound power level is of minor importance for case 2 and 3 due to reduced radiation efficiency

Finally we discuss the sound pressure level in the cabin.

Figure 20 shows that at 80 Hz the sound pressure level for case 1 has its maximum value within the presented frequency range, but for other 2 cases it's insignificant at this frequency. Around 800 Hz the sound pressure level for case 2 and 3 is peaking there.

For frequencies above 1250 Hz the sound pressure level for all three cases is of minor importance.

Figure 21 shows that at 80 Hz the sound pressure level for case 1 has its maximum value within the presented frequency range, but for other 2 cases it is insignificant at this frequency. Around 800 Hz the sound pressure level for case 2 and 3 is peaking there. For frequencies above 125 Hz the sound pressure level for case 1 is of minor importance.

For frequencies above 80 Hz and 1000 Hz the sound pressure level for case 2 and 3 is insignificant.

The highest sound level in the cabin is 26,7 dBA for high idle assuming that radiation efficiency *Re* = 0 (case 1). This is much lower than the sound level during the operation, which is 74 dBA. It is worth commenting that the total highest sound power level is not always producing the highest sound level in the cabin. The main reason is that the frequency dependence of the radiation efficiency is different for the three cases.

7 Conclusion

Based on the assumption that the accelerometers were properly calibrated, the movement of the exhaust pipe to the new location does not affect the sound level in the cabin. The sound pressure level for the both states, idle and high idle, is below 30 dBA while the reference sound pressure level measured by Volvo CE in the cabin is 74 dBA. The difference between these levels is over 40 dBA. Such a big difference does not increase the sound pressure level in the cabin.

The results we obtained from the calculations do not take the fundamental, the first harmonic of the firing frequency that are in 1/3 octave frequency bands below 80 Hz.

8 Recommendations

It is recommended to check the calibration of the accelerometers after dismounting them. The difference in sensitivity of the accelerometers between measured and factory data can be used to correct the results in this report.

In the current case the calculated sound level in the cabin from the pipe at its in new location was much lower than the sound level during operation. So despite that the accuracy of the method is rather poor it is possible to draw the conclusion that sound level in the cabin will not be affected by the movement of the pipe. if, however the calculated had been closer to the operating sound level a more accurate method had to be used. The most critical part is the radiation efficiency *Re* that on this case require the more precise determination.

It is also worth mentioning that correlation measurements could sometimes be of value.[8]

9 References

Books:

4. Wallin, H.P. and Carlsson, U. and Åbom, M. and Bodén, H. and Glav, R. (2010). Sound and Vibration. Kungl Tekniska Högskolan

Internet:

1. Volvo CE. (2007). *175 years Volvo Construction Equipment 1832 - 2007*. Http://www.volvo.com/NR/rdonlyres/39862192-0CD5-4770-9D50-1ECAEA20377B/0/175years21A10039610711.pdf (accessed 20 may, 2011, time 11.50)

2. Sound, http://dictionary.reference.com/browse/sound (accessed 20 may, 2011, time 12.55)

3. Brüel & Kjær, (1984). Measuring the sound. http://www.bksv.com/Library/Primers.aspx (accessed 6 may, 2011, time 19.30)

5. Brüel & Kjær, (1996) Microphone handbook vol. 1. http://www.bksv.com/Library/Primers.aspx (accessed 6 may, 2011, time 19.32)

6. LMS Theory Tutorial. http://www.lmsintl.com/downloads/tutorials (accessed 2 may, 2011, time 11.00)

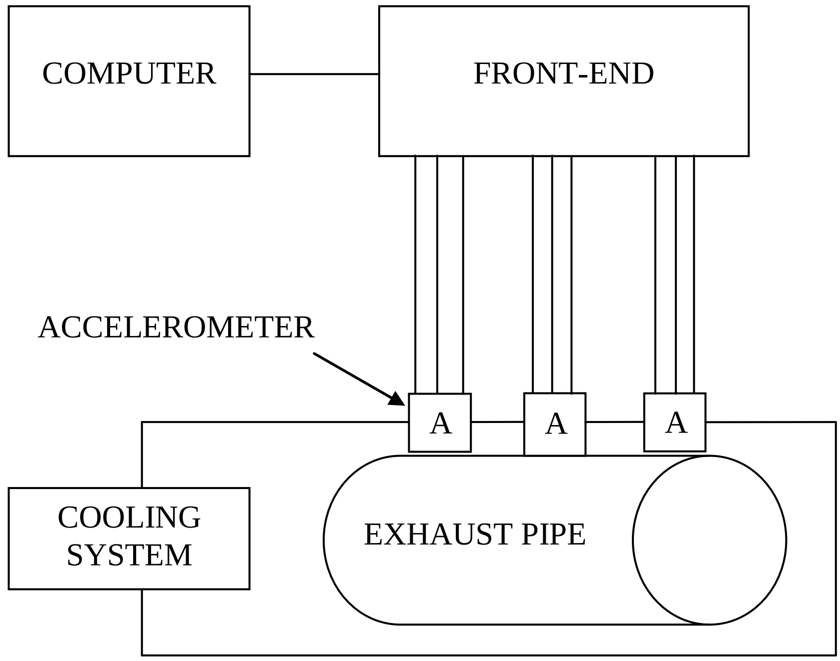
7. Brüel & Kjær, (1982) Measuring Vibration. http://www.bksv.com/Library/Primers.aspx (accessed 6 may, 2011, time 19.34)

8. Ovcina, A. and Petersson, M. (2005) Ljud- och vibrationsnivåer i en dumper mätningar och analys. http://w3.msi.vxu.se/~bni/akustik-volvo2.pdf (accessed 9 may, 2011, time 16.47)

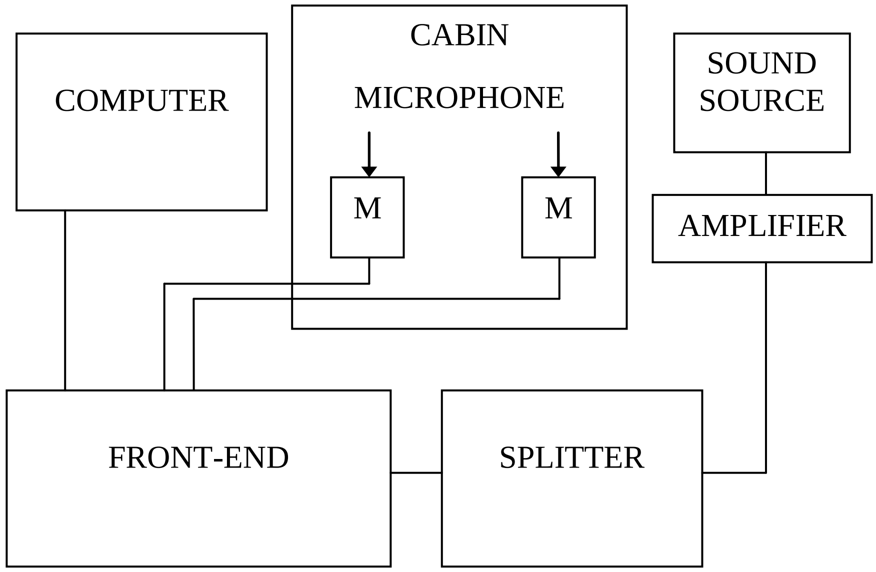
10 Appendix

Appendix 1 Schematic drawings of measuring position

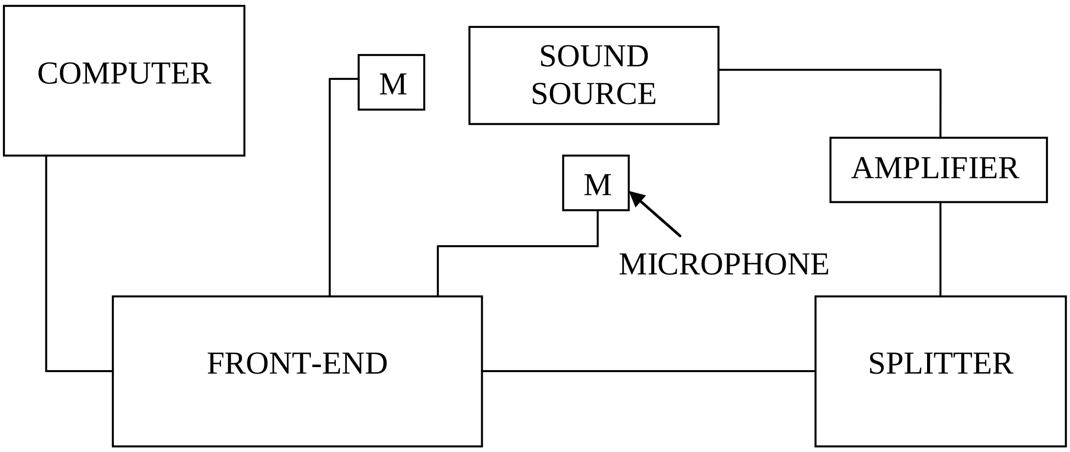
Sound Power Level measuring position schematic diagram.



Sound Pressure Level measuring position schematic diagram



Calibration of the sound source schematic diagram.



Appendix 2

Measurement system components

Accelerometer

Accelerometers can measure the acceleration in different directions. The triaxial piezoelectric charge accelerometer Brüel & Kjær type 4326-A-001 can stand high temperatures in comparison to other available ones. A mechanical stress generates proportional electrical charge to the force, which was applied to accelerometer [7].

Figure 1 Accelerometer type 4326-A-001

table 1 the triaxial piezoelectric charge accelerometer Brüel & Kjær type 4326-A-001

THE TRIAXIAL PIEZOELECTRIC CHARGE ACCELEROMETER BRÜEL & KJÆR TYPE 4326-A-001	
Frequency	1 - 8000 Hz
Temperature	-55 - 230 ºC (-67.0 - 446.0 °F)
Weight	17 gram
Sensitivity	3 pC/g
Residual Noise Level in Spec Freq Range (rms) ±	0.30 mg
Maximum Operational Level (peak)	2000 g
Electrical Connector	10-32 UNF
Mounting	Adhesive
Accessory Included	None
Clip/Stud/Screw included	None
Output	Charge-PE
Unigain	No
Triaxial	Yes
TEDS	No
Resonance Frequency	30 kHz
Maximum Shock Level (± peak)	5000 g

Front-end

LMS SCADAS Mobile SCM05, produced by LMS Engineering Innovation, is the high-end measurement system, which perfectly suits to advanced engineering noise and vibration testing. The system consists of 40 input channels, CAN bus system, Bluetooth as well as GPS.

Figure 2 LMS SCADAS Mobile SCM05

Table 2 LMS SCADAS Mobile SCM05

LMS SCADAS Mobile SCM05	
Number of slots	6 (1 for system controller)
Max number of channels per frame	40
Tacho inputs	2 (standard on board)
Generator outputs	2 (standard on board)
Dimensions (WxHxD)	340 x 78 x 295 mm 13.38 x 3.07 x 11.69 inch
Weight	6.2kg max / 13.67 lbs max
AC power input	110/220V
DC power input	9-36V
Max power consumption	40W
Battery operation (minimum)	1 hour (4 hours with additional slot battery)
Host interface	Ethernet
Operating temperature	-10°C to +55°C / 14° to 131°F
Sensor type	V, ICP, MIC, charge, strain, digital audio

Cooling system

The cooling system have a task to cool, in this case, accelerometers and the whole system consisting of pump, oil tank, heat exchangers and oil pipes, which allow circular flow though the whole system.

Figure 3 Metal pipe with radiators

Figure 4 Oil tank with pump

Microphones

Microphones in other words are electro-acoustical transducers, which convert acoustic waves to electrical signal. There are three types of microphones: diffuse-field, free-field and pressure-field. The free field microphones measure sound waves, which can propagate without any disturbances or reflections in continuous media [4]. Brüel & Kjær type 4189 is high precision, free-field 1/2-inch microphone. On the photo below the microphone is presented with the Brüel & Kjær microphone preamplifier Type 2671.

Figure 5 Microphone B&K type 4189

Table 3 BRÜEL & KJÆR TYPE 4189

BRÜEL & KJÆR TYPE 4189	
Capacitance	13 pF
Diameter	1/2 inch
Dyn. Range	15.2 - 146 dB
Freq. Range	6.3 - 20000 Hz
Inherent Noise	14.6 dB A
Lower Limiting Frequency -3dB	Hz
Optimised	Free field
Polarization	Prepolarized
Polarisation Voltage	0 V
Preamplifier Included	No
Pressure Coefficient	–0.01 dB/kPa
Sensitivity	50 mV/Pa
Standards	IEC 61094-4 WS2F IEC 61672 Class 1ANSI 1.4 Type 2
TEDS UTID	
Temperature Coefficient	- 0.006 dB/ºC
Temperature Range	-30 - 150 ºC (-22.0 - 302.0 °F)
Venting	Rear

LMS Test.Lab Standard 11B

The LMS Test.Lab is a powerful tool made for advanced noise and vibration measurements. A step-by-step configuration process intuitive interface causes the software are easy to use. After measurements, there are possibilities to analyze data which was collected the during measurements.

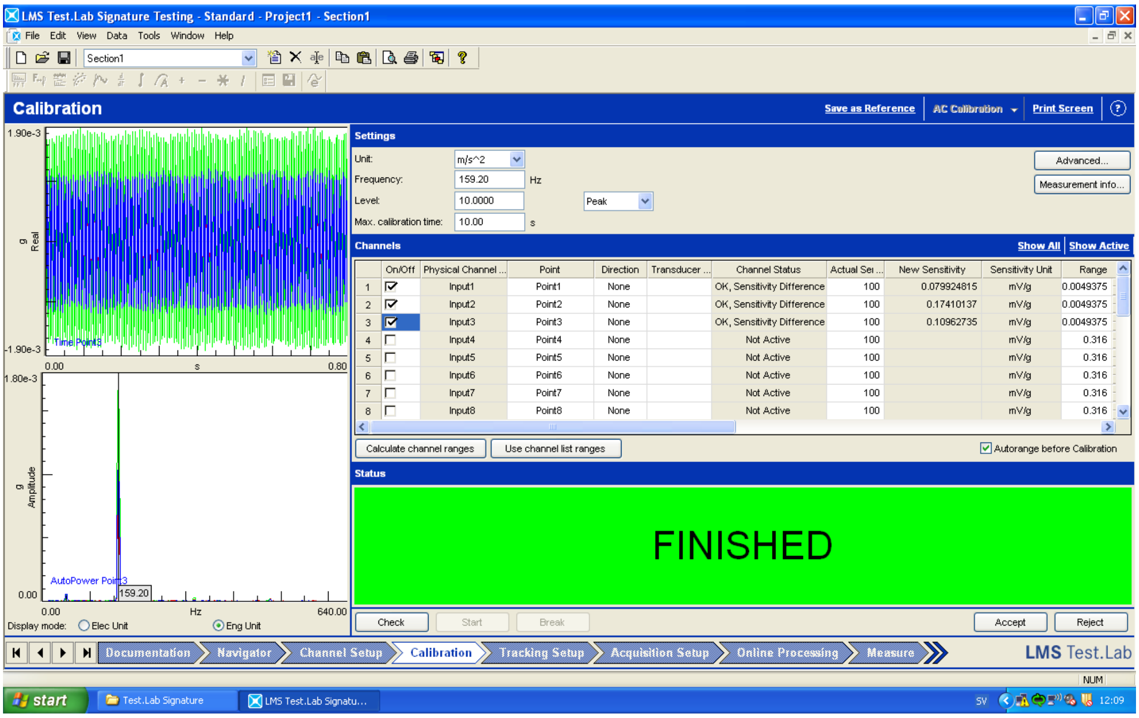


Figure 6 LMS Test.Lab interface

Sound source

As the sound source served OmniSource Loudspeaker type 4295 produced by Brüel & Kjær. The sound source radiates same in all directions.


Figure 7 OmniSource Loudspeaker type 4295

Splitter

With the help of a splitter, it is possible to adjust the voltage of the amplifier to the front-end. The splitter divides the voltage in the ratio of 1:4. The splitter has 4 input and 4 output sockets.

Figure 8 Splitter

Appendix 3 Derivation of the correction factor

We assume first a plane wave and therefore we can use $v = \frac{p}{\rho_0 c}$

$$\bar{W} = pv \cdot S \qquad \text{eq.12.1}$$

$$\left[v = \frac{p}{\rho_0 c}\right] \rightarrow p = v\rho_0 c \qquad \text{eq.12.2}$$

We need to find a relationship between the velocity and the sound power. Therefore we isolate p (eq.1.2) and inserting this expression into eq.1.1 .We get the formula below (eq.1.3).

$$\bar{W} = v\rho_0 cS \qquad \text{eq.12.3}$$

$$L_w = 10log\frac{\bar{W}}{W_{REF}} \qquad \text{eq.12.4}$$

Inserting eq.1.3 into $\bar{W}$ above we get equation 1.5 and we introduce v_{REF}^2 so we can get an expression for Lv

$$L_w = 10log\frac{\bar{v}^2 \cdot \rho_0 \cdot c \cdot S \cdot v_{REF}^2}{W_{REF} \cdot v_{REF}^2} \qquad \text{eq.12.5}$$

Following the logarithmic rules for multiplication we get eq 1.6

$$L_w = 10\log\frac{\bar{v}^2}{v_{REF}^2} + 10\log S + 10\log\frac{\rho_0 c}{W_{REF}} \qquad \text{eq.12.6}$$

Where the first term in the eq.1.6 is a velocity level and the last term in the eq.1.6 is a correction factor.

$$L_w = L_v + 10\log S + C \qquad \text{eq.12.7}$$

Nomenclature:
W – Power,
v – velocity,
c – speed of sound in the air, app. 340m/s
ρ_0 - density of air, app. 1,2 kg/m^3
L_w - Sound power level
S – Surface of the area
p – pressure
v_{REF}^2 – reference value of the velocity
L_v - vibration velocity level
C – correction factor
W_{REF} - power reference value
Log - base 10

Next we assume a more complicated radiation pattern, please see appendix 5 for a derivation.

Appendix 4 Calculation of the sound power level of the exhaust pipe

Calculations of Sound Power Level are identical for idle and high idle.

Surface area of the pipe S:

$$S = 2\pi a h = 2 \cdot \pi \cdot 0{,}0635 \cdot 1 = 0{,}1330\ m^2 \qquad \text{Eq. 13.1}$$

Here:
- a - radius of the pipe (r)
- h - height/length of the pipe (m)

Calculation of correction factor C:

$$C = 10 \cdot \log\left(\frac{\rho_0 \cdot c}{W_{REF}}\right) = 10 \cdot \log\left(\frac{1{,}2 \cdot 340}{10^{-12}}\right) = 146{,}11\ [\text{dB}] \qquad \text{Eq. 13.2}$$

Here:
- ρ_0 - density of air 1,2 kg/m^3
- c – velocity of sound propagation in the air 340 m/s
- W_{REF} – reference value for sound power 10^{-12} W
- logarithm base is 10

Calculation of angular frequency ω:

$$\omega = 2 \cdot \pi \cdot f \quad [\text{rad/s}] \qquad \text{Eq. 13.3}$$

Here:
- f – frequency (Hz)

Calculation of k coefficient. k is variable and depends on ω.

$$k = \frac{2 \cdot \pi \cdot f}{340} \qquad \text{Eq. 13.4}$$

Calculation of radiation efficiency R_e in appendix 5.

At the end L_w is calculated according to the formula:

$$L_{w1} = L_v + 10 \cdot \log S + C + R_e \quad [\text{dB}] \qquad \text{Eq. 13.5}$$

Here:
- L_v – vibration velocity level (dB)
- S – surface of the pipe
- C – correction factor
- R_e - radiation efficiency (Appendix 5)
- logarithm base is 10

Appendix 5 Acoustic radiation form acoustic cylindrical multipoles

Acoustic radiation from acoustic cylindrical multipoles

by

Börje Nilsson

The radiation efficiency from a vibrating surface depends on the vibration pattern. We study here acoustic radiation from a circular cylinder with radius a and length L. To simplify it is assumed that the sound field is not varying in the direction of the cylindrical axis and that only the sound power from a finite part of length L of an infinite cylinder is of interest. Thus the problem is two-dimensional depending only on the polar co-ordinates ρ and φ where $\rho = a$ corresponds to the surface of the cylinder.

To simplify further the calculations, the -iω-method (corresponding to the jω-method in alternating current theory) is used writing

$$\begin{cases} p_n^r(\rho,\varphi,t) = \mathrm{Re}\left[p_n(\rho)\mathrm{e}^{\mathrm{i}(n\varphi-\omega t)}\right] \\ v_{\rho,n}^r(\rho,\varphi,t) = \mathrm{Re}\left[v_{\rho,n}(\rho)\mathrm{e}^{\mathrm{i}(n\varphi-\omega t)}\right] \end{cases} . \tag{1}$$

Here, $p_n^r(\rho,\varphi,t)$ is the measurable (real) acoustic pressure and $p_n(\rho)$ is the corresponding complex-valued quantity. A similar notations $v_{\rho,n}^r(\rho,\varphi,t)$ and $v_{\rho,n}(\rho)$ are used for the radial velocity. Further, ω is the angular frequency and n is the order of the multipole; $n = 0$ is a monopole, $n = 1$ is a dipole, $n = 2$ is a quadropole etc. It is expected that the dominating contribution in the current case comes from transverse vibrations from a stiff cylinder. This corresponds to dipole radiation with $n = 1$.

The vibrations of the cylinder is specified through $v_n^0 = v_{\rho,n}(a)$ having the amplitude $|v_n^0|$ that is given by measurements. From [2] follows that

$$\begin{cases} p_n(\rho) = -\mathrm{i}\rho_0 c v_n^0 \dfrac{\mathrm{H}_n^{(1)}(k\rho)}{\mathrm{H}_n^{(1)\prime}(ka)} \\ v_{\rho,n}(\rho) = v_n^0 \dfrac{\mathrm{H}_n^{(1)\prime}(k\rho)}{\mathrm{H}_n^{(1)\prime}(ka)} \end{cases} . \tag{2}$$

Here, ρ_0 is the density of air, c is the sound velocity, $k = \omega/c$ is the wave number, $\mathrm{H}_n^{(1)}$ is Hankel's function of order n and type 1 giving a radiating wave and $\mathrm{H}_n^{(1)\prime}$ is the derivative of $\mathrm{H}_n^{(1)}$. Thus (2) describes an radiating wave with the normal velocity at the cylinder $v_{\rho,n}(a)$ equals the prescribed velocity v_n^0.

According to [2], the time mean value of the radial sound intensity is

$$\overline{I_{\rho,n}(\rho)} = \mathrm{Re}\left[p_n(\rho)^* v_{\rho,n}(\rho)\right], \tag{3}$$

where * denote complex conjugation. Then we have

$$\overline{I_{\rho,n}(\rho)} = \rho_0 c \left|\frac{v_n^0}{\mathrm{H}_n^{(1)\prime}(ka)}\right|^2 f_n(k\rho), \tag{4}$$

where

$$f_n(k\rho) = \mathrm{Re}\left[\mathrm{i}\mathrm{H}_n^{(1)}(k\rho)\mathrm{H}_n^{(1)\prime}(k\rho)^*\right]. \tag{5}$$

It is easier to calculate the sound power for large $k\rho$ giving the same value as for $\rho = a$ due to the absence of losses. From asymptotic expressions for Hankel functions [1] follows that

$$f_n(k\rho) \sim \frac{2}{\pi k\rho}, \text{ when } k\rho \to \infty \tag{6}$$

providing the time mean value of the acoustic radial intensity

$$\overline{I_{\rho,n}(\rho)} \sim \frac{2\rho_0 c}{\pi k\rho} \left| \frac{v_n^0}{\mathrm{H}_n^{(1)\prime}(ka)} \right|^2, \text{ when } k\rho \to \infty \tag{7}$$

and the time mean value of the acoustic power

$$\overline{W_{\rho,n}} = \frac{4\rho_0 c L}{k} \left| \frac{v_n^0}{\mathrm{H}_n^{(1)\prime}(ka)} \right|^2 . \tag{8}$$

The next step consists of converting (8) to levels in dB. Introducing the reference value $W_{ref} = 1\text{pW}$, the sound power level from the multipole of order n is

$$L_{Wn} \underset{def}{=} 10 * {}^{10}\log \frac{\overline{W_{\rho,n}}}{W_{ref}} = L_{v,n} + 10 * {}^{10}\log S + C + R_e, \tag{9}$$

where

$$\begin{cases} S = 2\pi a L \\ C = 10 * {}^{10}\log \dfrac{\rho_0 c}{W_{ref}} \\ R_e = 10 * {}^{10}\log \dfrac{2}{\pi ka} \dfrac{1}{\left|\mathrm{H}_n^{(1)\prime}(ka)\right|^2} \end{cases} . \tag{10}$$

R_e is called the radiation efficiency. It is understood that it is the variables with their SI-units that should be used in the arguments of the logarithm in (9-10). To calculate the derivative of the Hankel function the following relation [1] can be used:

$$\mathrm{H}_n^{(1)\prime}(ka) = \mathrm{H}_{n-1}^{(1)}(ka) - \frac{n}{ka}\mathrm{H}_n^{(1)}(ka). \tag{11}$$

References

[1] M. Abramovich and I. A. Stegun. *Handbook of Mathematical Functions.* Dover, third edition, 1965.

[2] P. Morse and U. Ingard. *Theoretical acoustics.* McGraw-Hill Book Company, New York, 1968.

Appendix 6 Calculation of the cabin reference sound pressure level

During measurements of the sound pressure level in the cabin there are 2 microphones to increase accuracy and quality of data, which is taken into consideration. Therefore there are 2 different sound pressure levels in one measurement. Using the formula below, it is possible to get the logarithmic mean value of the sound pressure level of two microphones:

$$L_{prss} = 10 \cdot \log\left(\frac{10^{0,1 \cdot L_{p1}} + 10^{0,1 \cdot L_{p2}}}{2}\right) \quad [\mathrm{dB}] \qquad \text{Eq. 15.1}$$

Here:
- logarithm base is 10
- L_{p1} - sound pressure level of first microphone
- L_{p2} - sound pressure level of second microphone

Appendix 7 Calculation of calibration of the sound source

Calculation of the sphere area S:

$$S = 2 \cdot \pi \cdot r^2 \quad [\mathrm{m}^2] \qquad \text{Eq. 16.1}$$

Here:
- r - radius of the sphere

Calculation of logarithmic mean value of sound pressure levels of microphones $L_{p,av}$:

$$L_{p,av} = 10 \cdot \log\left(\frac{10^{0,1 \cdot L_{p1}} + 10^{0,1 \cdot L_{p2}}}{2}\right) \quad [\mathrm{dB}] \qquad \text{Eq. 16.2}$$

Here:
- logarithm base is 10
- L_{p1} - sound pressure level of first microphone
- L_{p2} - sound pressure level of second microphone

Calculation of Sound Power Level of sound source L_{w1}:

$$L_{w1} = L_{p,av} + 10 \cdot \log S \quad [\mathrm{dB}] \qquad \text{Eq. 16.3}$$

Here:
- S – surface area of half sphere
- $L_{p,av}$ – logarithmic mean value of sound pressure levels of microphones, which were used during calibration.
- logarithm base is 10

Calculation of the calibrated Sound Power Level L_{wrss}

$$L_{wrss} = L_{w1} - 20 \cdot \log\left(\frac{U_1}{U_2}\right) \quad [\mathrm{dB}] \qquad \text{Eq. 16.4}$$

Here:
- logarithm base is 10
- L_{w1} - Sound Power Level of the sound source
- U_1 – voltage of Sound Pressure Level of calibration
- U_2 - voltage of Sound Pressure Level in the cabin
- L_{wrss} – calibrated Sound Power Level of sound source

Appendix 8 Measurements and results.

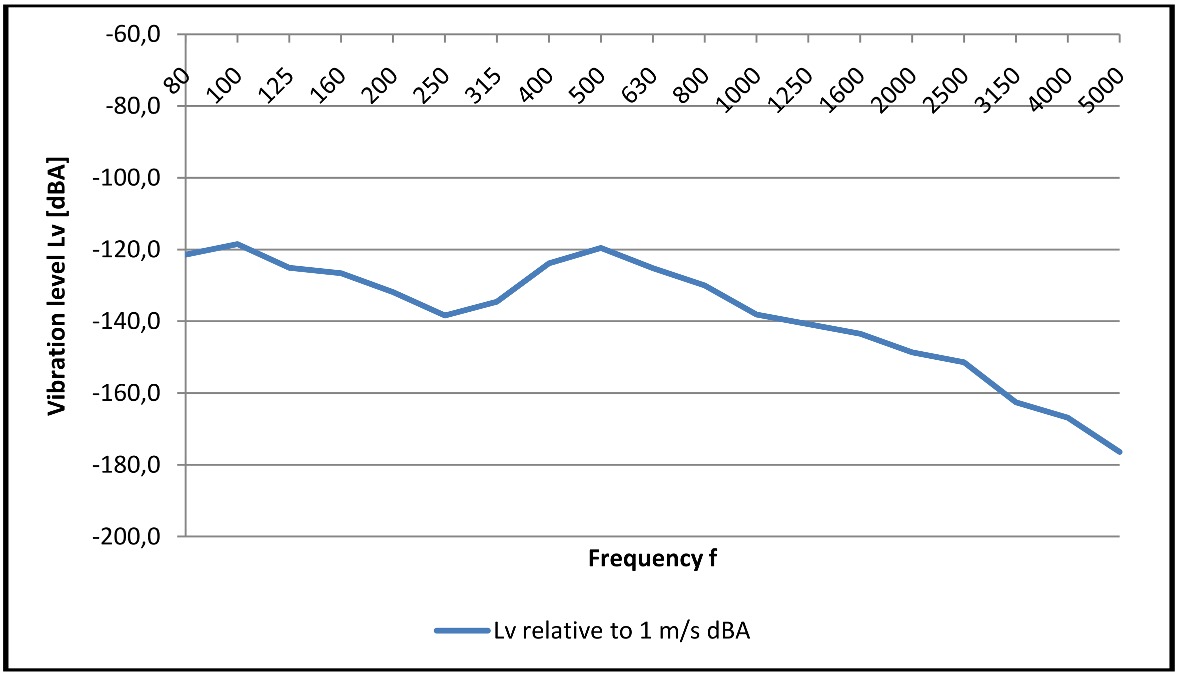


Figure 1 Vibration level of the exhaust pipe (high idle)

Figure 1 shows the vibration levels of the exhaust pipe as a function of frequency. The vibration level was obtained from the measurement system using 1/3 octave frequency band. At any given frequency a corresponding value of the vibration level can be obtained from the graph. Note that the vibration levels are negative because the reference value is 1m/s.

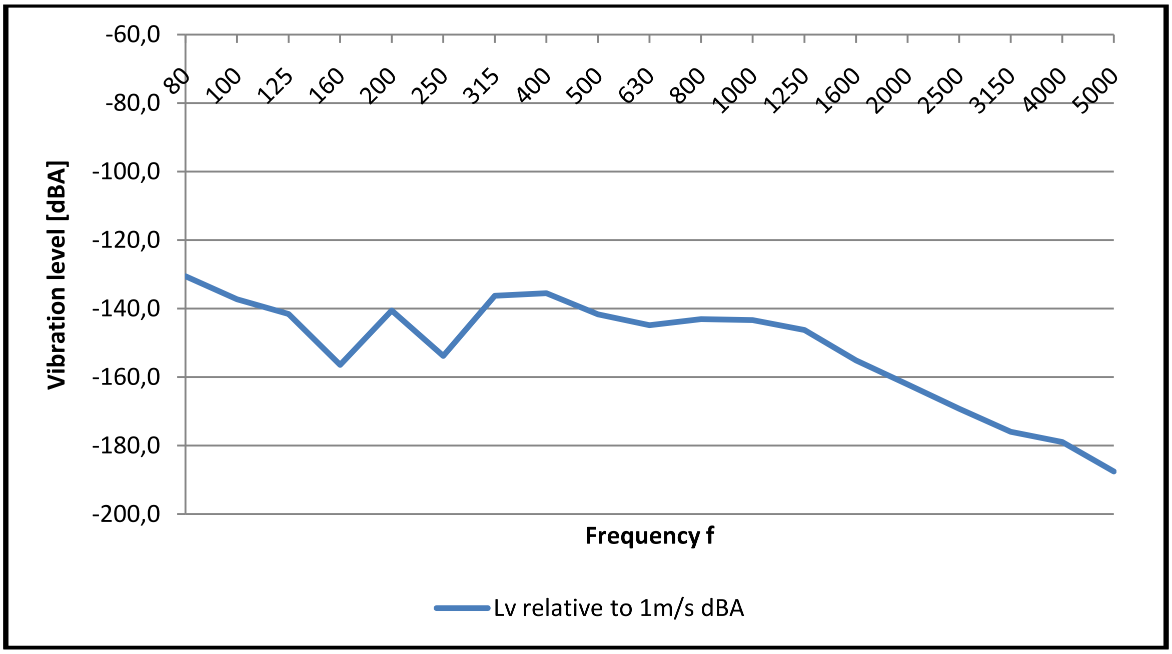


Figure 2 Vibration level of the exhaust pipe (idle)

Figure 2 shows the vibration levels of the exhaust pipe as a function of the frequency. The vibration level was obtained from the measurement system using 1/3 octave frequency band. At any given frequency a corresponding value of the vibration level can be obtained from the graph. Note that the vibration levels are negative because the reference value is 1m/s.

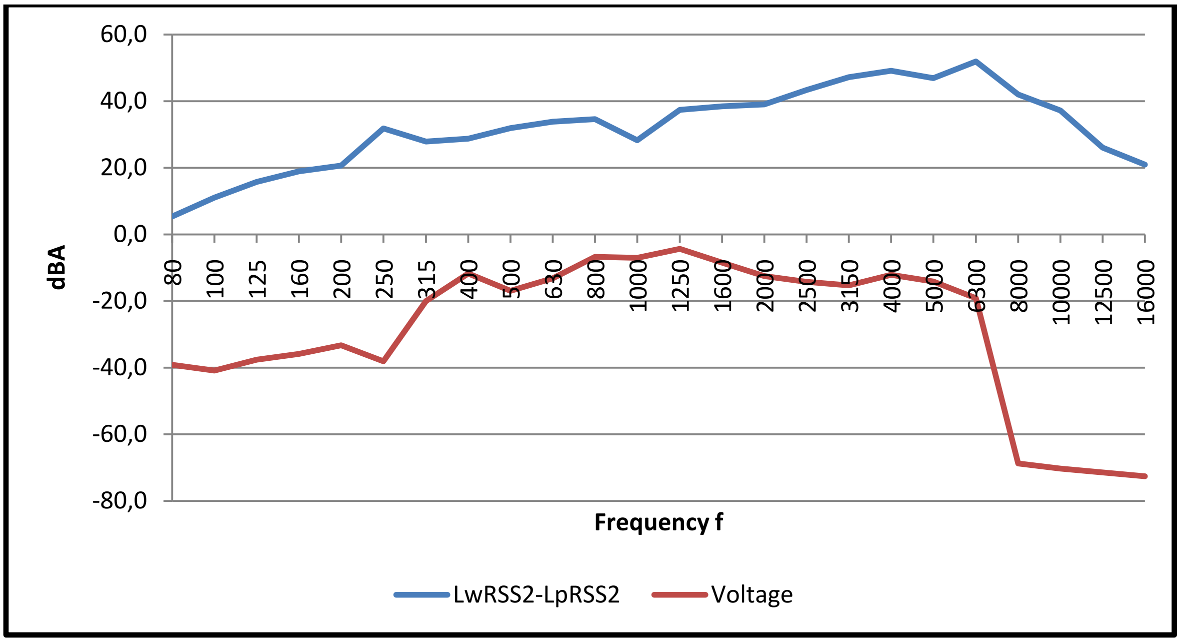


Figure 3 LwRSS2-LpRSS2

Figure 3 shows the difference between L_{wRSS2}-L_{pRSS2} as a function of frequency and the voltage as a function of frequency. At any frequency we can determine the corresponding voltage and the difference between L_{wRSS2}-L_{pRSS2}. At 6300 Hz the difference between L_{wRSS2}-L_{pRSS2} is maximum within this frequency range. Note that LwRss2-LprSS is average value of the two states because the L_{wRSS2}-L_{pRSS2} we obtained for two states were different.

Appendix 9 Example of calculations

We are going to show how we obtained the sound pressure level L_{p2} in the cabin from the equation 2.3.

$$L_{p2} = L_{w1} - L_{wRSS2} + L_{pRSS2}$$

All calculations were done for high idle state, case 1 and for frequency of 800 Hz. Case 1 take only correction factor into consideration. All logarithms base is 10.

To calculate L_{w1}we used equation 2.4, that is given below:

$$L_{w1} = L_v + 10 \cdot \log S + C + R_e$$

From the equation above we have given L_v from measurement and it is equal to -129.2 dB. Now we have to calculate second component, which is $10 \cdot \log S$. We assumed that length of the pipe is cylindrical, straight, 1m long and diameter is 5 in, which is 0.127m. Hence:

$$10 \cdot \log S = 10 \cdot \log(2\pi \cdot (0.127)^2 \cdot 1) = -8.76$$

Because result is negative, we put result in absolute value.

To calculate correction factor C we used equation, where ρ_0 is density of air 1,2 kg/m^3, c is velocity of sound propagation in the air 340 m/s, and W_{REF} is reference value for sound power 10^{-12} W. Having all necessary data we can do following calculation:

$$C = 10 \cdot \log\left(\frac{\rho_0 \cdot c}{W_{REF}}\right) = 10 \cdot \log\left(\frac{1{,}2 \cdot 340}{10^{-12}}\right) = 146{,}11 \text{ [dB]}$$

Due to the fact that our calculation were made for case 1, radiation efficiency R_e is equal to 0. Then equation 2.4 looks:

$$L_{w1} = L_v + 10 \cdot \log S + C = -126 + 10 \cdot \log 8.76 + 146.11 = 28{,}6\ dB$$

So now we calculated first component of equation 2.3. Now we can calculate second component, which is calibrated sound power level of the reference sound source L_{wRSS2} using equation 2.6, which you can see below:

$$L_{wRSS2} = L_{pRSS2} + 10 \cdot \log S - 20 \cdot \log\left(\frac{U_1}{U_2}\right) \quad \text{[dB]}$$

$L_{pRSS2}$is the mean value (Eq.2.7) of the microphones in the cabin, which is equal to 50,4 dB. In the equation 2.6 we have to calculate surface S of the halfsphere (radius is 2m), which is equal to 25,14 m^2. U_1 is the voltage at which the sound pressure level is calibrated and is equal to 0,67V. U_2 is the voltage at which the sound pressure level in the cabin is measured and is equal to 0,94. Having all components from equation 2.6 we can calculate value of L_{wRSS2}:

$$L_{wRSS2} = L_{pRSS2} + 10 \cdot \log S - 20 \cdot \log\left(\frac{U_1}{U_2}\right)$$
$$= 50{,}4 + 10 \cdot \log(25.14) - 20 \cdot \log\left(\frac{0{,}67}{0{,}94}\right) = 83{,}2\ dB$$

Now we calculated value of second component of equation 2.3. The last component, which is sound pressure level L_{pRSS2} of the reference sound source. L_{pRSS2} was used to calculate L_{wRSS2} in equation above. To get more detailed way of calculating L_{pRSS2} you can find in appendix 6.

Having all necessary components of equation 2.3 we can calculate sound pressure level L_{p2} in cabin in position 2, hence:

$$L_{p2} = L_{w1} - L_{wRSS2} + L_{pRSS2} = 25{,}63 - 83{,}2 + 50{,}4 = -4{,}3$$

When we calculate values of L_{p2} for every frequency in our measurement, we can calculate total sound pressure level in the cabin in position 2, using the formula for mean value of logarithm.

Appendix 10 Summary and analysis of measurements on a VOLVO Articulated Hauler

This document summarizes and analyses differences between three different evaluation models. More details are found in the full report.

Measurement method

The main goal of the investigation was to determine the sound level in the cabin from air borne sound from a vibrating exhaust piping. The investigation was made in two parts.

A. Determination of the difference Lw-Lp between the sound power level Lw and sound pressure level Lp for a reference sound source located at the same place as the exhaust piping. Here, Lw was determined under three field conditions. The analysis was done in 1/3 octave bands.
B. Determination of Lw for the exhaust piping using measurements of the velocity level Lv on the surface of the exhaust piping.

The model was used to compute Lw-Lp in part A:

1) $Lw\text{-}Lp = Lw_{RSS2} + Lp_{RSS2}$

Three models were used to determine Lw-Lv in part B:

1) The far field assumption of a plane wave
   $Lw\text{-}Lv = 10 \cdot \log S + C$
2) Cylindrical monopole
   $Lw\text{-}Lv = 10 \cdot \log S + C + Re|n=0$
3) Cylindrical dipole
   $Lw\text{-}Lv = 10 \cdot \log S + C + Re|n=1$

Here, C is a frequency independent correction factor that describes the relation between velocity level Lv and pressure level Lp in a plane wave. Re is the radiation efficiency.

Case 1, which has the advantage of being frequency independent, describes the physical conditions poorly, can give an estimate only. Cases 2 and 3 should be able to describe the situation better but with the complication that Re is frequency dependent. There is no assumption about a plane far field for cases 2 and 3; in actual fact Re describes a transfer from a cylindrical near field (below ca 600 Hz) via a resonant region at about 800 Hz to the cylindrical far field (above ca 1000 Hz). Please c.f. Fig. 1 for a frequency plot of Re.

The analysis is performed in 1/3 octave bands to capture the frequency behaviour.

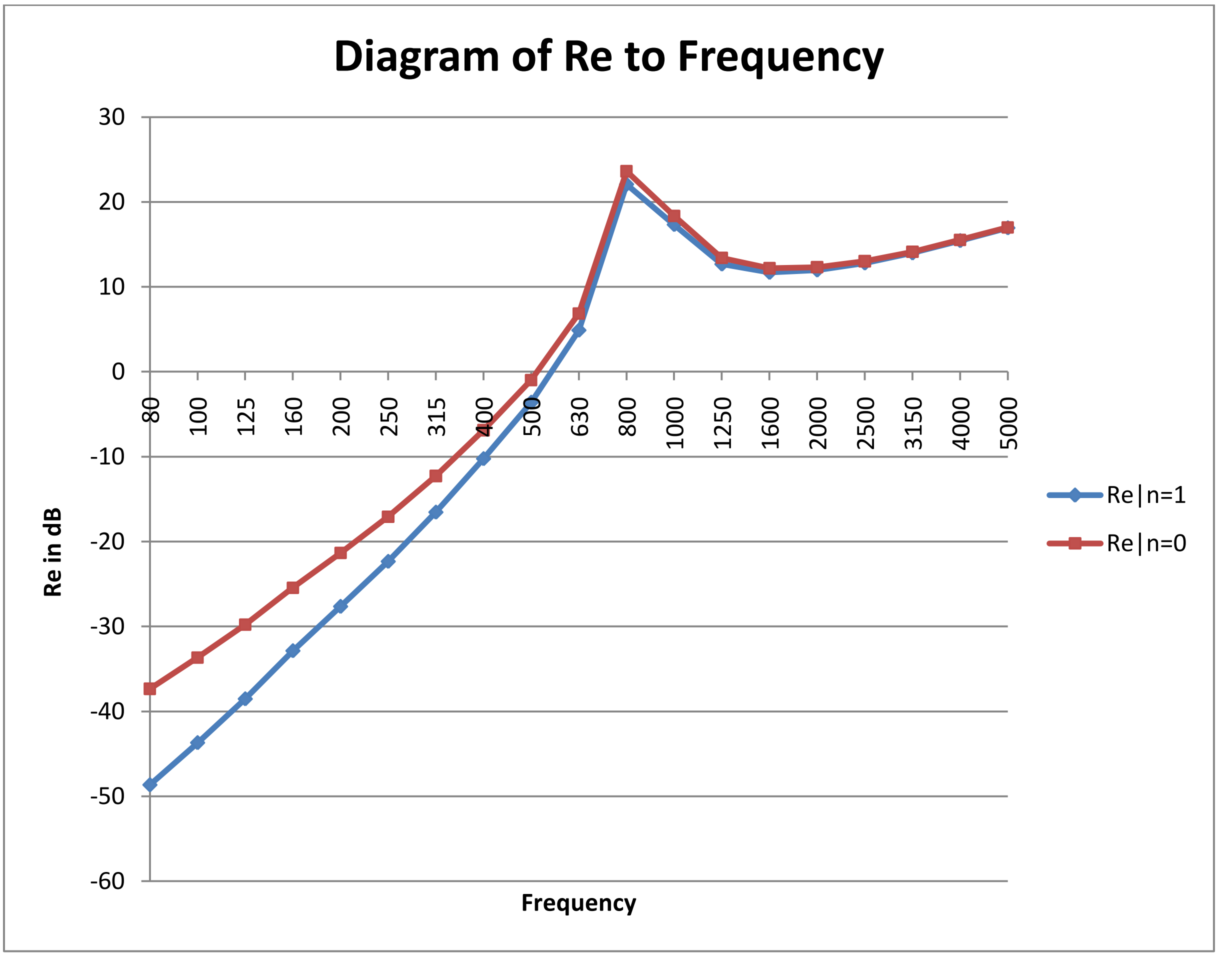


Fig. 1 The frequency behaviour of Re.

Radiation Efficiency Re increases Lw for frequencies bigger than f_1=~500 Hz for case 2 (red line) and f_2=~550 Hz for case 3(blue line), while for lower frequencies Re decreases Lw. On the diagram can be noticed, if the dominating part of the spectrum is above f_1 (or f_2) then Re increase Lw while decrease one for frequencies lower than f_1(or f_2). Therefore the frequency dependence of the spectrum of the source affects whether Re would increase or decrease the total level of Lw.
Since the A-filter affects the frequency spectrum the effect of Re could be different on total linear levels and total A-weighted levels.

The measurement results

The frequency spectrum of case 1 was used in the measurements. To measure with a spectrum corresponding to Lw for cases 2 and 3 would have required filtering of the measured Lv to get a control signal to the amplifier feeding the loudspeaker. As a consequence, Lw-Lp in total level is different for the three cases, respectively.

Below in Table 1 and Table 2, total levels are presented for the sound power level of the exhaust piping and the generated sound pressure level in the cabin.

Case	Lw (dB)	Lp (dB)	Lw-Lp (dB)
1. Plane far field	53.4	48.6	4.8

2. Cylindrical monopole	52.5	21.2	31.3
3. Cylindrical dipole	51.0	19.2	31.8

Table 1. Linear total levels in 1/3 octave bands 80 Hz to 5000 Hz.

Case	**LwA (dBA)**	**LpA (dBA)**	**LwA-LpA (dBA)**
1. Plane far field	38.6	26.7	11.9
2. Cylindrical monopole	51.8	19.9	31.9
3. Cylindrical dipole	50.3	18.4	31.9

Table 2. A-filtered total levels in 1/3 octave bands 80 Hz to 5000 Hz.

f	Lw,case1	Lw,case2	Lw,case3	Lp,case1	Lp,case2	Lp,case3	A-filter	LwA,case 1	LwA,case2	LwA,case3	LpA,case1	LpA,case2	LpA,case3
Hz	dB	Re|n=0	Re|n=1	dB	Re|n=0	Re|n=1	dB	dBA	Re|n=0	Re|n=1	dBA	Re|n=0	Re|n=1
80	50,6	13,2	1,9	48,2	10,8	-0,5	-22,5	28,1	-9,3	-20,6	25,7	-11,7	-23,0
100	47,9	14,3	4,2	37,4	3,8	-6,3	-19,1	28,8	-4,8	-14,9	18,3	-15,3	-25,4
125	39,7	9,9	1,1	24,5	-5,3	-14,0	-16,1	23,6	-6,2	-15,0	8,4	-21,4	-30,1
160	43,2	17,7	10,3	25,2	-0,3	-7,7	-13,4	29,8	4,3	-3,1	11,8	-13,7	-21,1
200	28,9	7,6	1,3	13,5	-7,8	-14,1	-10,9	18,0	-3,3	-9,6	2,6	-18,7	-25,0
250	25,5	8,5	3,2	-5,7	-22,7	-28,0	-8,6	16,9	-0,1	-5,4	-14,3	-31,3	-36,6
315	33,3	21,1	16,8	7,1	-5,2	-9,5	-6,6	26,7	14,5	10,2	0,5	-11,8	-16,1
400	39,3	32,4	29,0	12,6	5,7	2,3	-4,8	34,5	27,6	24,2	7,8	0,9	-2,5
500	26,1	25,1	22,5	-3,9	-4,8	-7,5	-3,2	22,9	21,9	19,3	-7,1	-8,0	-10,7
630	26,1	33,0	31,0	-6,9	-0,1	-2,0	-1,9	24,2	31,1	29,1	-8,8	-2,0	-3,9
800	28,6	52,2	50,6	-4,3	19,3	17,8	-0,8	27,8	51,4	49,8	-5,1	18,5	17,0
1000	21,1	39,4	38,4	-4,8	13,6	12,5	0	21,1	39,4	38,4	-4,8	13,6	12,5
1250	17,3	30,8	30,0	-18,6	-5,2	-5,9	0,6	17,9	31,4	30,6	-18,0	-4,6	-5,3
1600	11,4	23,6	23,1	-26,0	-13,8	-14,3	1	12,4	24,6	24,1	-25,0	-12,8	-13,3
2000	2,4	14,7	14,4	-35,9	-23,6	-23,9	1,2	3,6	15,9	15,6	-34,7	-22,4	-22,7
2500	-4,3	8,8	8,5	-46,1	-33,1	-33,3	1,3	-3,0	10,1	9,8	-44,8	-31,8	-32,0
3150	-9,3	4,8	4,7	-55,5	-41,4	-41,5	1,2	-8,1	6,0	5,9	-54,3	-40,2	-40,3
4000	-12,1	3,5	3,4	-60,2	-44,7	-44,8	1	-11,1	4,5	4,4	-59,2	-43,7	-43,8
5000	-19,9	-2,9	-3,0	-65,8	-48,8	-48,9	0,5	-19,4	-2,4	-2,5	-65,3	-48,3	-48,4
total	53,4	52,5	51,0	48,6	21,2	19,2		38,6	51,8	50,3	26,7	19,9	18,4
dominating	53,2	52,5	51,0	48,6	20,6	19,1		36,4	51,7	50,2	26,6	19,8	18,4

Analysis of the measurement results

We have discussed above that Lw-Lp in total levels is different for the three cases. Below we shall give a closer examination of this difference and explain why Lw-Lp is so much greater for cases 2 and 3 than for case 1. The key to the explanation is that the frequency bands that dominate the total levels are different for the three cases.

To this end, the dominating 1/3 octave bands are identified and are marked by blue/dark shading in Table 3. Dominating frequency band are 5 the highest neighbouring frequencies. The corresponding total levels are presented in Table 4 and 5.

Case	Sound power level		Sound pressure level		Lw-Lp (dB)
	1/3 octave bands	Lw (dB)	1/3 octave bands	LpA (dB)	
1	80 – 200 Hz	53.2	80 – 200 Hz	48.6	4.6
2	400 – 1000 Hz	52.5	400 – 1000 Hz	20.6	31.9
3	630 – 1600 Hz	51.0	400 – 1000 Hz	19.1	31.9

Table 4. Linear total levels for dominating 1/3 octave bands.

Case	Sound power level		Sound pressure level		LwA-LpA (dBA)
	1/3 octave bands	LwA (dBA)	1/3 octave bands	LpA (dBA)	
1	315 - 800 Hz	36.4	80 – 200 Hz	26.6	9.8
2	630 - 1600 Hz	51.7	400 – 1000 Hz	19.8	31.9
3	630 – 1600 Hz	50.2	400 – 1000 Hz	18.4	31.8

Table 5. A-weighted total levels for dominating 1/3 octave bands.

We observe that the total levels for the dominating bands in Table 4 and Table 5 are close to the corresponding values in Table 1 and Table 2. Thus, it is possible to analyze differences in Lw-Lp based on where the dominating frequency bands are located in the spectrum.

Let us start with the linear levels. Here, there is no significant shift of the location of the dominating bands when Lw is transformed to Lp. Case 1 is dominated by low frequencies whereas cases 2 and 3 are dominated by relatively high frequencies. The reason for the difference in Lw-Lp is the relatively higher radiation efficiency at high frequencies compared to low frequencies; this feature is ignored in case 1.

So for linear levels, Lw-Lp in total level is determined by the conditions that are prevailing at the frequencies in the dominating bands. Therefore, Lw-Lp is larger for cases 2 and 3 where relatively high frequencies are dominating.

Cases 2 and 3 have a similar behaviour for the A-weighted levels. Thus Lw-Lp are approximately the same, about 32dB, for linear and A-weighted levels.

Case 1 has a slightly different behaviour. For A-weighted levels, LwA is dominated by relatively high frequencies, which is caused by the A-filter. LpA, on the other hand, is dominated by low frequencies, which is a consequence of that the effect from the low transmission loss at low frequencies is stronger than the effect of the A-filter. This means that Lw-Lp, here about 12 dBA, is larger than for the linear case, but not as large as for cases 2 and 3.

Summary

In summary, the reason for the big difference for case 1 on one hand and cases 2 and 3 on the other hand in Lw-Lp measured as total level, is the following. The radiation efficiency, affecting both level and frequency spectrum, is ignored for case 1 whereas it is included in cases 2 and 3.

Lnu.se

School of Engineering
351 95 Växjö
tel 0772-28 80 00, fax 0470-76 85 40