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# EXPERIMENTAL DETERMINATION OF PASS-BY NOISE CONTRIBUTIONS FROM THE BOGIES AND SUPERSTRUCTURE OF A FREIGHT WAGON

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Pass-by noise from freight trains on straight tracks is dominated by rolling noise. The main contributions are generally accepted to originate from vibrations of the track and the wheels. However, there is uncertainty about the contributions from the bogie and the wagon superstructure. This article describes experimental methods to determine these. The methods combine vibration measurements on a rolling wagon and measurements of structural and of vibro-acoustical transfer functions under static conditions. Firstly, an inverse method is used to determine the equivalent structural excitation of a wagon. Secondly, two reciprocity methods are described for detailed vibro-acoustic transmission path analysis, with the wagon parts excited by the equivalent forces. As an example, some data will be shown from experiments on a widely used freight wagon. It appears that the contributions from two bogies and from the wagon superstructure are respectively 20 and 30dB(A) lower than the measured pass-by noise levels. Detailed information is obtained about the ranking of sub-areas which contribute to the sound radiation. The results illustrate the practical value of the methods even in the case of weak partial sources.



## 1. INTRODUCTION

Railway freight noise on straight tracks is mainly caused by the so-called "rolling noise". The roughness on the wheels and rails causes them to vibrate, these vibrations in turn being transmitted into the adjacent structural components. All these vibrating components radiate sound and therefore contribute to the pass-by noise. Models and software for the prediction of the noise radiated by a given combination of wheel and track, are relatively well developed and validated [1, 2]. In contrast, the prediction of noise radiation by wagons remains poorly developed. Therefore, as part of the Brite-Euram project *Silent Freight*, both modelling [3] and experimental methods have been investigated to improve this situation. In this article research on experimental analysis is presented.

Within the Silent Freight project the use of both microphone arrays and vibro-acoustic transfer path analysis have been considered as complementary approaches. This paper concentrates on the latter approach. It has the disadvantage of considerable experimental effort that is required, but since antennae methods have limited spatial resolution at low frequencies and cannot fully resolve weak source components in the presence of strong sources, this paper can be seen to provide useful additional information. In this article, the methodology is described and its application on a Tombereau wagon is demonstrated. This is generally considered to be a noisy type of freight wagon. Therefore, not only the methodological aspects but also the specific results of the analysis are of interest.

## 2. ANALYSIS METHODS

### 2.1. TRANSMISSION PATH MODEL

Equation (1) is a simple mathematical representation of a superposition model for rolling noise due to $m$ partial sources:

$$O_{total} = \sum_{j=1}^{m} O_j = \sum_{j=1}^{m} TF_j \times I_j. \tag{1}$$

The system inputs $I_j$ genuinely characterize the partial sources themselves. The output–input ratios $O_j/I_j$ form the transfer functions $TF_j$, which characterize the transmission paths. Figure 1 is a schematic presentation of the structure-borne sound transmission in freight wagons. In the present study, the focus is on the sound radiation by the bogies and by the wagon superstructure. Therefore, in the following paragraphs the terms of equation (1), which belong to these partial sources, will be considered in detail.

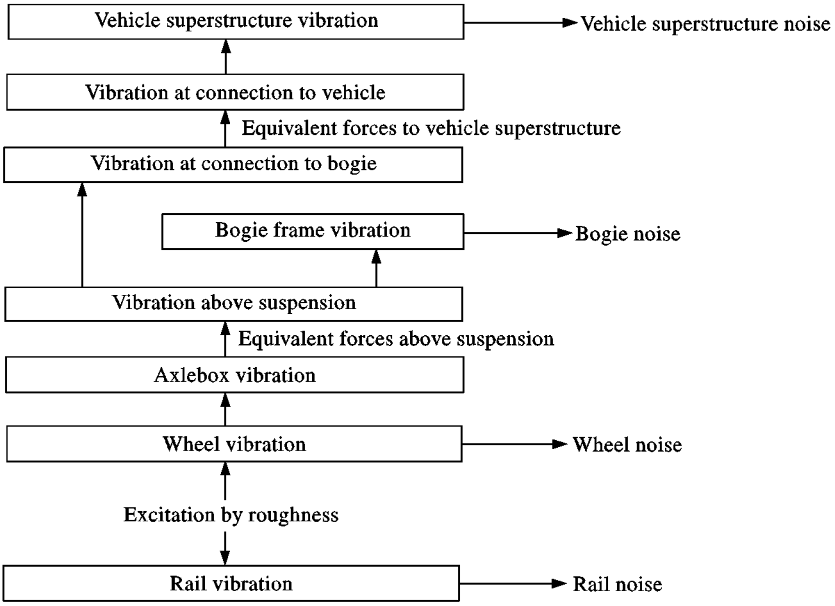


Figure 1. Schematic presentation of structure-borne sound paths in freight wagons (source: D. J. Thompson).

### 2.2. EQUIVALENT FORCES

For both the bogie and the wagon superstructure the source terms in equation (1) are sets of equivalent forces. These are sets of fictitious external point forces. When applied to a static bogie or wagon structure, together they would reproduce the accelerations on these structures, which have been measured during rolling. There exist many sets of forces (location, magnitude and phase), which would fulfil the requirements of equivalence. Therefore, for practical applications, a creative task of decisive importance is the choice of a proper multiple-input–multiple-output system for equivalent forces and vibration responses. The three steps for determining the equivalent forces are as follows [4–6]:

*Step* 1: Measure on the structure of interest (i.e., bogie or wagon superstructure) accelerations at $m$ positions, including their relative phase, when the train is running at a normal speed. These accelerations may be characterized by a vector $\mathbf{a}$.

*Step* 2: Select $n$ positions on the bogie or on the wagon superstructure, where external forces $F_j$ can be applied when the vehicle is standing still and with bogie and superstructure uncoupled. Measure the $m \times n$ accelerance matrix $\mathbf{A}$, which contains the complex frequency response functions $a_i/F_j$.

*Step* 3: Analytically determine the vector $\mathbf{F}_{eq}$ of $n$ equivalent forces (including relative phase), using either an inverse or an pseudo-inverse of $\mathbf{A}$ as follows:

$$\mathbf{F}_{eq} = \mathbf{A}^{-1}\mathbf{a} \quad \text{or} \quad \mathbf{F}_{eq} = \mathbf{A}^{+}\mathbf{a}. \tag{2a, b}$$

The above procedure is applied for each frequency component of the signals. More details on the modelling aspects are discussed in sections 3.1 and 3.2.

### 2.3. STRUCTURAL-ACOUSTIC TRANSMISSION

To determine the partial output terms on the right-hand side of equation (1), transfer functions are needed in addition to the equivalent forces. These are measured with bogie and wagon superstructure uncoupled. Two variants of equation (1) are discussed. In one method, the phase angle between the forces is taken into account and in the other mean square forces are used as inputs.

#### 2.3.1. *Reciprocity method*

The analysis method which preserves phase information needs complex-valued transfer functions $p_2/F_1$. It is convenient to measure them reciprocally [5, 6] according to

$$\frac{p_2}{F_1} = \frac{a'_1}{j\omega Q'_2}. \tag{3}$$

A monopole sound source with volume velocity $Q'_2$ is positioned at the receiver position at the trackside. On the vehicle structure, accelerations $a'_1$ are measured at the positions and in the directions of the corresponding equivalent forces. From an experimental point of view this type of analysis is very simple and straightforward. However, a disadvantage is that no detailed information is obtained on the partial contributions from distinct parts of the bogie or wagon superstructure. A method which provides this detailed information better, is discussed next.

### 2.3.2. *Hybrid method*

In the hybrid method, the input terms in equation (1) are mean-square equivalent forces, thus ignoring their correlations. Then the corresponding transfer functions are of the type $p_2^2/F_1^2$. These are determined in two steps. In the first step, the transfer functions $P_{1,k}/F_{1,j}^2$ are determined. To determine these, the vehicle structure is excited by point forces at each of the equivalent force positions in turn. During each point force excitation, the radiated sound power $P_{1,k}$ of $m$ partial surfaces $S_{1,k}$ is measured, using sound intensity scanning close in front of each surface.

In the second step, the transfer functions $p_2^2/P_{1,k}$ are determined from the power radiated by each sub-area to the mean-square pressure at the receiver location. This gives the following:

$$\frac{p_2^2}{F_{1,j}^2} = \sum_{k=1}^{m} \frac{P_{1,k}}{F_{1,j}^2} \times \frac{p_2^2}{P_{1,k}}. \tag{4}$$

To determine the transfer functions given by the second term on the right-hand side of equation (4), a reciprocity measurement is performed as follows [5–8]:

$$\frac{p_2^2}{P_{1,k}} = \frac{p_2^2}{nRQ_{1,k}^2} = \frac{1}{n} \sum_{i=1}^{n} \frac{p_{1,ik}'^2}{RQ_2'^2}. \tag{5}$$

In the denominator of the middle term of equation (5), the sound power radiation of partial area $S_{1,k}$ is modelled as that of $n$ fictitious uncorrelated monopoles of equal strength and evenly distributed over the surface area. Their equivalent volume velocity is $Q_{1,k}$, whereas $R$ is their radiation resistance. The fictitious monopole sources are considered to lie against an acoustically hard surface and to radiate approximately into a half-space. Therefore, their radiation resistance is approximately twice the free-field value of monopoles, i.e.,

$$R \approx \frac{\rho\omega^2}{2\pi c}. \tag{6}$$

In the actual measurement represented by the right-hand term of equation (5), $n$ microphones are placed against the partial surface $S_{1,k}$. They measure the blocked pressures $p_{1,ik}'$ caused by a monopole sound source with volume velocity $Q_2'$ placed at the receiver position on the trackside. If the partial surface $S_{1,k}$ is an acoustically hard part of the wagon, the only requirement is that the microphones are very close to the surface. If $S_{1,k}$ is an open area, for example, between wagon bottom and rail, then during the reciprocity experiment this opening is closed with an acoustically hard plate against which the microphones are placed. Of course, they have to be placed on the appropriate side of the plate, where the fictitious substitution monopoles would radiate sound into the direction of the outgoing sound intensity of the real source which is replaced.

## 3. EXPERIMENTS

### 3.1. EQUIVALENT FORCES ACTING ON BOGIE ABOVE WHEEL SUSPENSION

Measurements on a running freight wagon were performed with the assistance of SNCF on a test track in France, at a speed of 100 km/h. Accelerations on the two-axle bogie were measured on two-axle boxes of the same axle and above wheel suspensions of three wheels. On the basis of these data and using the symmetry of the bogie structure (see Figure 2) the following conclusions were drawn with respect to the equivalent force modelling of the bogie excitation. Firstly, the four wheels may be considered as uncorrelated vibration sources of equal strength. Secondly, on the beams directly above the wheel suspension the vibration levels in the $z$ direction (parallel to the track) are much lower than in the $x$ and $y$-directions. Therefore, it seems unnecessary to include them in the model. On this basis, it was assumed that the bogie vibrations can be reproduced by eight equivalent forces $\mathbf{F}_i^{\mathrm{T}} = \{F_{ix}, F_{iy}\}$, with $i = 2, 3, 4, 5$ (see Figure 2), for which the following properties of auto- and cross-power spectral densities hold:

$$\langle F_{ix}F_{ix}^*\rangle = \langle F_{jx}F_{jx}^*\rangle, \qquad \langle F_{iy}F_{iy}^*\rangle = \langle F_{jy}F_{jy}^*\rangle,$$

$$\langle F_{ix}F_{iy}^*\rangle = \langle F_{jx}F_{jy}^*\rangle,$$

$$\langle F_{ix}F_{jx}^*\rangle = \langle F_{iy}F_{jy}^*\rangle = \langle F_{ix}F_{jy}^*\rangle = 0 \quad \text{for} \quad i \neq j \tag{7a–d}$$

In words, equation (7) states that at different positions, 2–5, the forces are equal, on average, for the same (local) direction and that for the different positions they are uncorrelated and also that at all positions the cross-spectra of the two perpendicular forces are the same.

The accelerations $\mathbf{a}_i$ above the primary suspension (f.i. suspension number $i = 3$) can be derived from the forces $\mathbf{F}_i$ and the accelerances $\mathbf{A}_{3,i}$. These accelerances are

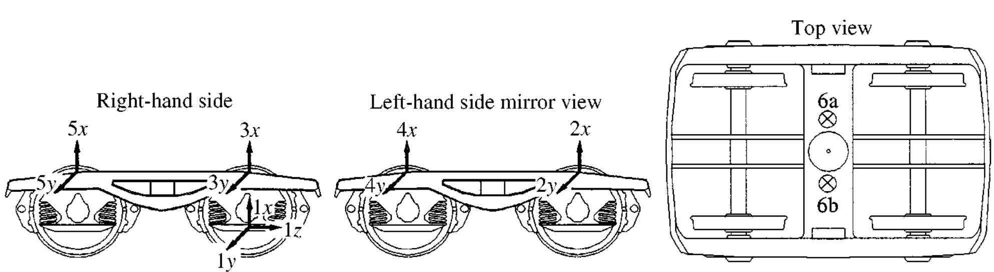


Figure 2. Measurement locations and directions on Y-25 bogie.

the ratios of the accelerations at position 3 in $x$ and $y$ directions and the point forces at each of positions 2–5:

$$\mathbf{a}_3 = \sum_{i=2}^{5} \{\mathbf{A}_{3,i}\mathbf{F}_i\}. \tag{8}$$

The auto- and cross-power spectral densities $\langle \mathbf{a}_3\mathbf{a}_3^{\mathrm{H}} \rangle$ of the accelerations $\mathbf{a}_3$ can be rewritten using equations (7a–d):

$$\langle \mathbf{a}_3\mathbf{a}_3^{\mathrm{H}} \rangle = \sum_{i=2}^{5} \{\mathbf{A}_{3,i}\langle \mathbf{F}_i\mathbf{F}_i^{\mathrm{H}} \rangle \mathbf{A}_{3,i}^{\mathrm{H}}\}, \tag{9}$$

where H denotes Hermitian transpose. The right-hand side of equation (9) gives the auto- and cross-spectral densities of the forces $\langle \mathbf{F}_i\mathbf{F}_i^{\mathrm{H}} \rangle$ and the $2 \times 2$ matrices of accelerances $\mathbf{A}_{3,i}$ which were measured during the static tests. The left-hand side is the matrix of auto- and cross-spectral densities of accelerations during the running test. Now the eight forces can be derived by solving equation (9) for $\mathbf{F}_i$ again using equations (7a–d).

For the purpose of the experiment it was important to determine a set of equivalent forces that solely excites the bogie structure. Therefore, the accelerance measurements were performed with the bogie uncoupled from the wagon superstructure, with the wheels and axles removed and the bogie mounted on resilient supports at the proper height above the track.

### 3.2. EQUIVALENT FORCES ACTING ON SUPERSTRUCTURE AT CONNECTION WITH BOGIE

Figure 3 shows a sketch of the wagon superstructure. Equivalent forces in the vertical direction were modelled at positions 7a and 7b at a stiff part of the bottom structure. These are adjacent to the connection point with the bogie and opposite to positions 6a and 6b in Figure 2. At the same positions the sum and difference of the accelerations, i.e., $(a_{7a} + a_{7b})$ and $(a_{7a} - a_{7b})$, were measured during running tests

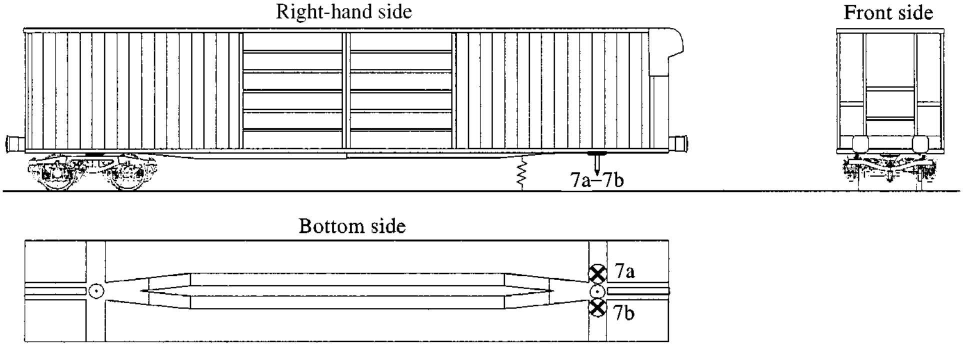


Figure 3. Measurement locations and directions on superstructure of Tombereau wagon.

and for the accelerance measurements. Two equivalent forces were defined. One is the sum of the two forces in the vertical direction which are equal in magnitude and phase at positions 7a and 7b. The other is the difference between two such forces. This force difference is proportional, at low frequencies, to a moment about an axis parallel to the track. These equivalent forces were calculated using a variant of equation (2a).

### 3.3. RECIPROCAL MEASUREMENT OF VIBRO-ACOUSTIC TRANSFER FUNCTIONS

Reciprocal measurements according to the right-hand side of equation (3) were performed with loudspeakers at different positions at the trackside. Because the bogie was standing free, a wooden plate was installed above its upper side to simulate the acoustical shielding due to the wagon superstructure. The distance of the loudspeakers from the track centre was 7.5 m and the height 1.5 m above the rail head. Strictly speaking, for the "reciprocity" experiment to be valid the loudspeakers should have had the same directivity pattern as an omnidirectional microphone in a "direct" experiment. The problem is that a broadband loudspeaker of sufficient strength cannot be made with these properties. However, because there were no reflecting surfaces and obstacles nearby except the ground, this requirement could be relaxed. As long as the loudspeaker directivity pattern is spherical over a sufficiently large solid angle in the direction of the freight wagon, both for the direct sound and for the ground reflection, the measurement results will not differ from those with a real omnidirectional source. Two loudspeakers were used to cover frequency ranges of 80–600 and 500–5000 Hz respectively. They have a smooth spherical directivity pattern with a coverage angle of at least 100° between the $-3$ dB points. This angle was considered to be sufficiently large in view of the length of the wagon and the distance between loudspeakers and track. The high-frequency loudspeaker was specially designed for this type of experiment. Its box is a large diameter sphere (Figure 4). A heavy-duty horn driver is mounted inside this sphere to generate sound through a small circular aperture. The directivity pattern of such a loudspeaker box is very smooth and spherical over a certain angle in front of the diaphragm opening. Its design was based upon available theory [9, 10] and its designed performance was confirmed in anechoic room tests.

### 3.4. HYBRID MEASUREMENT OF VIBRO-ACOUSTIC TRANSFER FUNCTIONS

For the measurement of the first term on the right-hand side of equation (4), the structures were excited with a 200 N electrodynamic shaker and the sound power was determined using a sound intensity probe, scanning over the radiating surfaces by hand. The bogie was standing freely mounted in the same way as for the accelerance measurements. Measurements on the bogie were performed only for excitations with $F_{3x}$ and $F_{3y}$. For all the other forces the transfer functions follow from symmetries. For the measurements on the wagon superstructure a single


Figure 4. High-frequency loudspeaker in front of Tombereau wagon.

exciter was used at the connection point, that is, in the middle between positions 7a and 7b (Figure 3). These measurements determine the transfer functions from squared sum force $(F_{7a} + F_{7b})^2$ to sound powers $P_{1,k}$. The transfer functions from squared force difference $(F_{7a} - F_{7b})^2$ to sound power is reconstructed using the transfer accelerences from sum force and from force difference to wagon body accelerations, which were measured as well.

The static tests were performed in a different location from that used for the running tests, with different acoustic properties. Therefore, all acoustic transfer functions from vehicle and bogie to sound pressure at the track side have been corrected for differences in attenuation. These differences were determined from transfer functions between the loudspeakers at the trackside and microphone positions above the track measured at both test sites. The differences were usually smaller than 1.5 dB in each 1/3 octave band.

## 4. RESULTS

### 4.1. BOGIE

The results in this section are primarily meant to illustrate the practical value of the analysis methods.

In Figure 5, one-third octave band levels are presented of the equivalent forces derived for the free-standing bogie. The equivalent forces were determined in

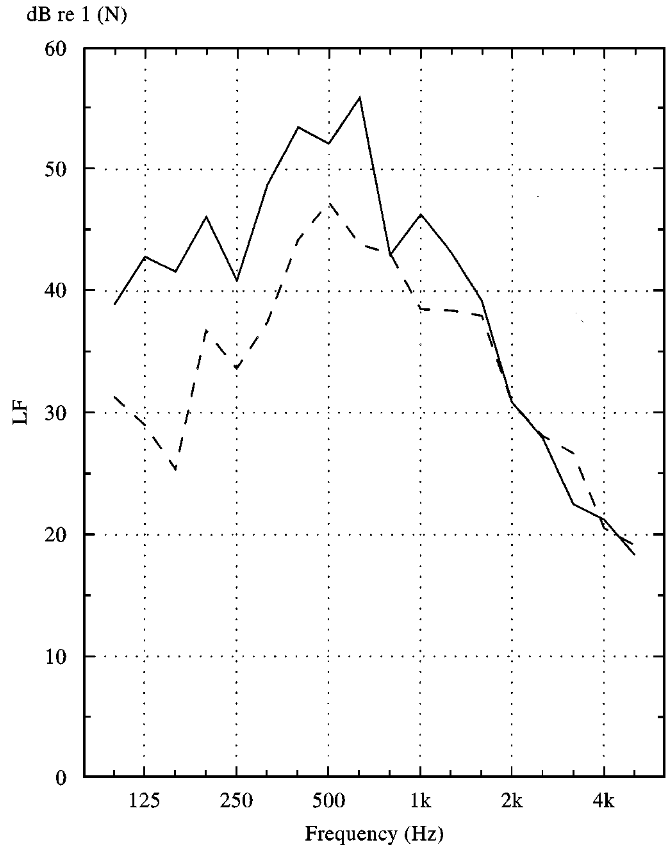


Figure 5. Third-octave band levels of equivalent forces $F_{3x}$ and $F_{3y}$ acting on the bogie above the primary suspension: ——, $F_{3x}$; - - - - $F_{3y}$.

narrow-frequency bands including phase and then converted to 1/3 octaves for presentation purposes. It is seen that the force in vertical direction is about 5–10 dB stronger than in transverse direction for most bands below 1.6 kHz.

In Figure 6, the estimated spectrum of pass-by sound pressure level radiated by two bogies is shown, together with the contributions due to the vertical and the horizontal equivalent forces. It can be seen that the transverse force, i.e., the smallest of the two forces, is nevertheless the most important one for the sound generation. This type of information can be used for acoustic redesign of the bogie.

Figure 7 shows the total pass-by sound pressure levels and the contributions from two bogies, estimated with the reciprocity method and the hybrid method respectively. Two observations can be made. Firstly, the overall contribution of the bogies is approximately 20 dB(A) lower than the measured pass-by noise, and the difference in 1/3 octave bands is at least 10 dB. Secondly, the results for both methods are approximately equal. This demonstrates that the transfer functions measured with two different methods are consistent.

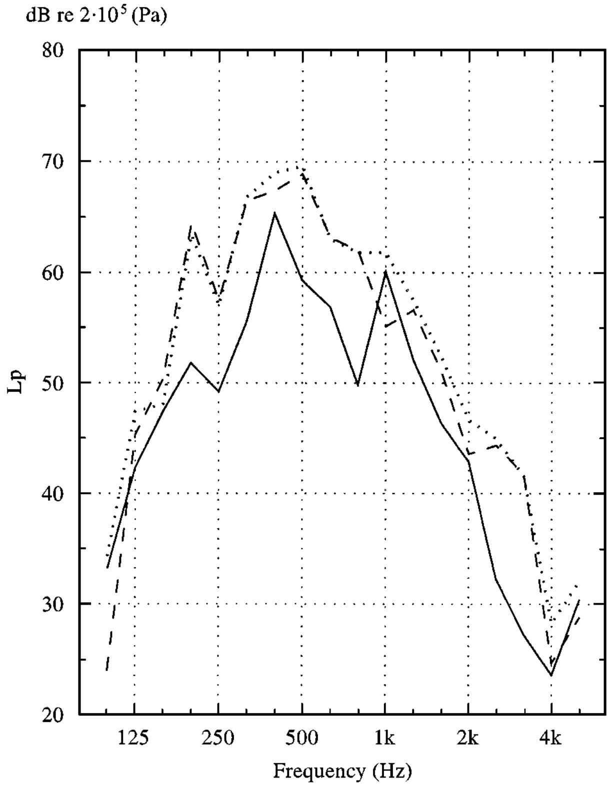


Figure 6. Sound pressure levels from two bogies by all four vertical "$F_x$" and horizontal forces "$F_y$" per bogie: ——, 65.2 dB(A) bogies by $F_x$; - - - - 70.2 dB(A) bogies by $F_y$; · · · · · 71.3 dB(A) bogies in total.

### 4.2. WAGON SUPERSTRUCTURE

Figure 8 shows the contribution from the wagon superstructure to the pass-by sound pressure level estimated using the reciprocal method for the transmission measurements. Also shown are the estimated contributions from the vertical equivalent force and from the force difference (moment). The contribution from the vertical force excitation is at least 10 dB stronger at frequencies below 1.25 kHz.

Figure 9 shows the contributions from the wagon superstructure to the pass-by sound estimated with the two methods for transmission measurements. Again it is seen that both methods give results which are approximately equal. The superstructure contribution is about 30 dB(A) under measured pass-by level and thus well below the contribution by the bogie in this case as well. It is close only to the measured spectrum below 200 Hz.

Figure 10 shows the contributions from different parts of the superstructure. The side-wall of the wagon closest to the pass-by microphone and the wagon floor are responsible for almost all of the superstructure noise.

Next to the Tombereau wagon, the contributions of an empty tank wagon and a flat container wagon without containers were measured. There were no

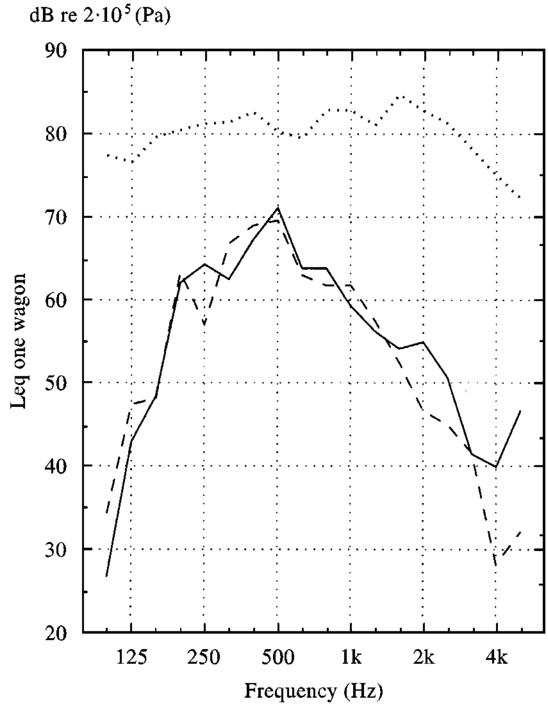


Figure 7. Pass-by sound pressure levels and contributions of two bogies (hybrid and reciprocal methods): ——, 71.7 dB(A) hybrid; - - - - 71.3 dB(A) recip.; · · · · · 92.3 dB(A) pass-by.

significant differences between tank and the Tombereau wagon superstructures. The flat wagon was 6 dB(A) less noisy.

### 4.3. DISCUSSION

The accuracy of the above results was checked, and these checks provided independent support for the validity of the presented results. Firstly, it could be shown both for the bogie and for the wagon superstructure, that there was an accurate reproduction by the equivalent forces of vibration levels at positions which were not used in the inversion process. Secondly, for the vehicle side-wall a simple calculation of radiated sound levels, using measured accelerations during running in combination with estimated radiation efficiencies and a simple propagation model, agreed within a few dB with the reconstructed sound levels of Figure 10.

## 5. CONCLUSIONS

The feasibility of both variants of the analysis method studied has been demonstrated under representative circumstances on a freight wagon.

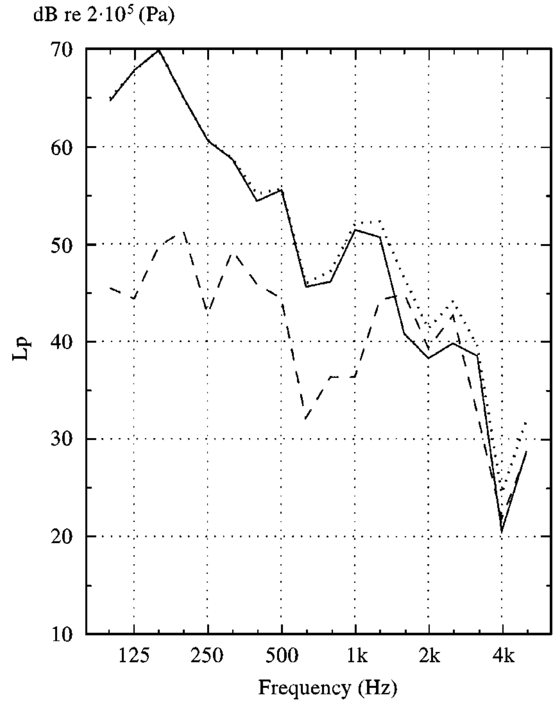


Figure 8. Sound pressure from superstructure by sum "$f_{7a}+f_{7b}$" and force difference "$f_{7a}-f_{7b}$" (reciprocal). ——, 62.7 dB(A) sum force; - - - -, 52.6 dB(A) diff. force; · · · · ·, 63.1 dB(A) total.

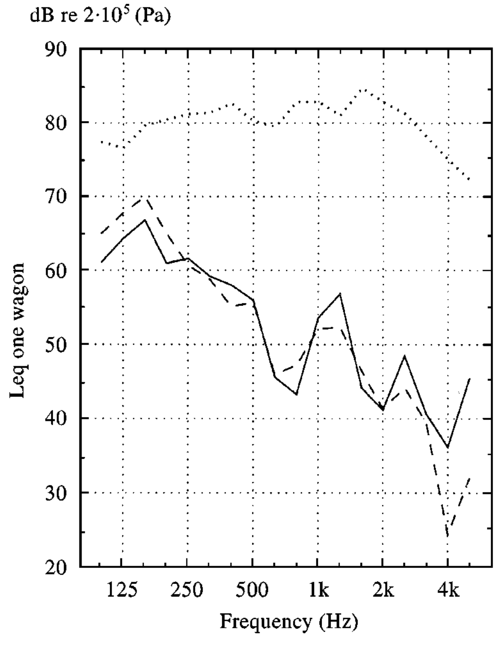


Figure 9. Pass-by sound pressure levels and contributions of superstructure (hybrid and reciprocal methods). ——, 63.4 dB(A) hybrid; - - - -, 63.1 dB(A) recip.; · · · · ·, 92.3 dB(A) pass-by.

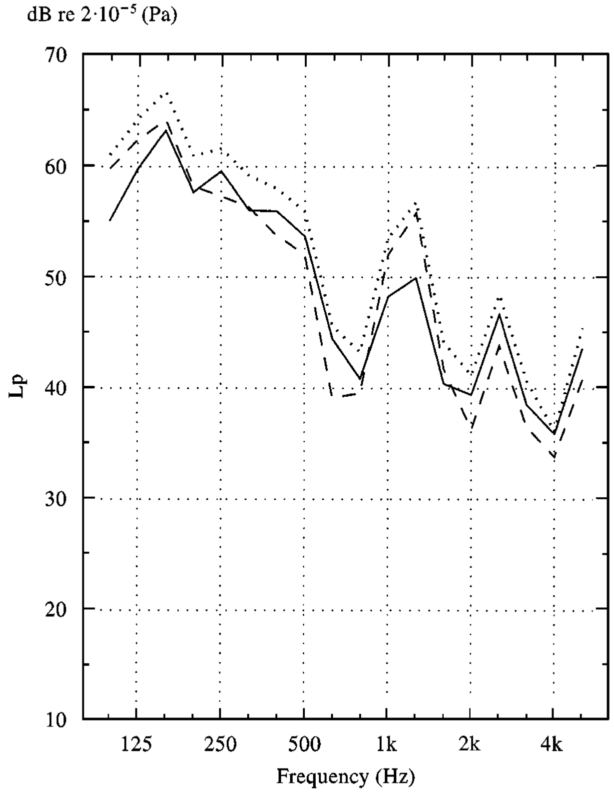


Figure 10. Sound pressure levels from vehicle bottom, walls and roof (hybrid method). ——, 59.9 dB(A) walls + roof; - - - -, 61.0 dB(A) bottom, · · · · ·, 63.4 dB(A) wagon body.

An important and new result is the determination of the partial contributions from the bogies and the wagon superstructure to the total pass-by noise. These contributions are respectively 21 and 29 dB(A) lower than the total sound level. This proves the ability of the method to quantify partial source contributions, which are very weak compared with those of the predominant sources. Moreover, the method can determine the contribution from weak sources in close proximity to strong sources. This is, for example, the case with the wagon bottom which is close to the track, which has a very weak and a relatively strong source respectively.

A warning, however, is appropriate with respect to aerodynamic noise. The method studied in this article cannot take this into account. Therefore, depending on the train speed and on the relative strength of the partial sources, the method may underestimate the noise generation by the wagon superstructure.

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## REFERENCES

1. D. J. THOMPSON, B. HEMSWORTH and N. VINCENT 1996 *Journal of Sound and Vibration* **193**, 123–135. Experimental validation of the TWINS prediction program for rolling noise, part 1: description of the model and method.
2. D. J. THOMPSON, P. FODIMAN and H. MAHÉ 1996 *Journal of Sound and Vibration* **193**, 137–147. Experimental validation of the TWINS prediction program for rolling noise, part 2: results.
3. N. S. FERGUSON 2000 *Journal of Sound and Vibration* **231**, 791–803. Modelling the vibrational characteristics and radiated sound power for a Y25 type bogie and wagon.
4. J. W. VERHEIJ, M. H. A. JANSSENS and D. J. THOMPSON 1993 *Noise* 93, *St. Petersburg*, Vol. 2, 55–60. An equivalent forces method for quantification of structure-borne sound paths.
5. J. W. VERHEIJ 1997 *International Journal of Acoustics and Vibration* **2**, 11–20. Inverse and reciprocity methods for machinery noise source characterization and sound path quantification. Part 1: sources.
6. J. W. VERHEIJ 1997 *International Journal of Acoustics and Vibration* **2**, 103–112. Inverse and reciprocity methods for machinery noise source characterization and sound path quantification. Part 2: transmission paths.
7. J. W. VERHEIJ 1992 *Proceedings of the 2nd International Congress on Recent Developments in Air- and Structure-borne Sound and Vibration*, Auburn, Al, Vol. 1. 591–598. Reciprocity method for quantification of airborne sound transfer from machinery.
8. J. ZHENG, F. J. FAHY and D. ANDERTON 1994 *Applied Acoustics* **42**, 333–346. Application of a vibro-acoustic reciprocity technique to the prediction of sound radiation by a motored IC engine.
9. PH. M. MORSE and K. U. INGARD 1968 *Theoretical Acoustics*, 341–343. New York: McGraw-Hill.
10. E. SKUDRZYK 1971 *The Foundation of Acoustics*, 401–407. New York: Springer-Verlag.