Determination of the Vibration and Acoustic Radiation of a Concrete Railway Sleeper

C. Bosquet, N.S. Ferguson and D.J. Thompson

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Determination of the Vibration and Acoustic Radiation of a Concrete Railway Sleeper

by

C. Bosquet, N.S. Ferguson and D.J. Thompson

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December 1997

Authorized for issue by
Dr R J Pinnington
Group Chairman

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Preface

This memorandum forms the result of a project carried out by Cédric Bosquet in the summer of 1997. The author completed the M.Sc. in Sound and Vibration Studies as a visiting exchange student and as such was not obliged to complete a project as a thesis. He performed the project described here voluntarily in order to take full opportunity of his visit to the ISVR to widen his experience.

Unfortunately due to the need to return to France to complete his course in Lyon, the work has been left with some loose ends, but it has been decided to publish it in its current state so that the progress which has been made is at least recorded.

D.J. Thompson
N.S. Ferguson
(supervisors)
ACKNOWLEDGEMENTS

Many thanks to my supervisors, Dr. D. J. THOMPSON and Dr. N. FERGUSON, for their help and encouragement throughout the project.

Thanks to the structural dynamics group for letting me use their instrumentation. In particular, I would like to thank Richard Grice for helping me to get started with the analyser and for his helpful comments in ways of doing the experiment.

Thanks also to the technical staff for their availability and kindness. In particular, I would like to thank Mr. C. Chalk for always making sure that my sleeper was moved in time and at the right place.

Finally, thank to the Institute of Sound and Vibration Research for accepting me to follow the MSc course during this European exchange program with INSA de Lyon (Institut National des Sciences Appliquées de Lyon). Especially, I would like to thank Professor F. J. FAHY for organising my arrival here in the ISVR and for always advising me throughout the twelve months of my stay.

It was a great pleasure for me to spend one full year in the ISVR where I could follow a high quality and professional course and obtain great experience during the project as well as improve my level in English.
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1-INTRODUCTION

1.1 Description of the problem

The noise from trains is a major problem for the expansion of the railway network. In particular, controversy often arises from the proposed high-speed French TGV lines. Although the expansion of the high-speed French train is part of the modernisation of the country, major objections are raised on environmental grounds, and in particular due to the noise that would be generated by the trains.

This rolling noise from trains comes mainly from wheel/rail interactions that generate the vibrations of these two components. The wheel-rail rolling noise is generally attributed to structural vibrations of the wheels and rails, excited from the contact patch area. This contact patch area is the surface of contact between the wheels and the rail during the motion of the train. Indeed, the main excitation is caused by the wheel and rail roughness (undulations), which introduce a relative vibration between the wheel and the rail. See reference [1].

Previous research has led to the construction of mathematical models for the wheel and rail radiation to predict their relative contribution (see reference [2]), and then to the construction of models described in references [3] and [4], based respectively on the finite element and periodic structure method. These models have been included within the software package TWINS (references [1] and [5]). This Track-Wheel Interaction Noise Software package was developed for the C163 Expert Committee of the European Rail Research Institute (ERRI) and is intended as a tool with which different designs of wheel and track can be assessed in terms of their noise generation. The figure 1 shows an overview of these TWINS models.
The rail is essential to understand the dynamics of the other components such as the wheel and are more interested in the sleeper vibration and noise radiation, it is nevertheless needed to study the interaction models of these components. Although in this project we need to know the contribution of each component to the rolling noise, we have been interested up till now.

Besides, the possible radiation from the ballast (some chippings on which the track is considered in the models, see references [6] and [7]) and has not been validated, sleepers (pieces of wood or concrete that support the rail) has only been recently modelled with a lot of improvement. For a number of years, the radiation from the wheel is modelled and vibration of the wheel and the rail have been although the radiation and vibration of the sleeper were not.
1.2 Background knowledge

1.2.1 noise radiation

Early studies by Thompson [2] considered the mathematical models of the radiation from each component (wheel and rail), from which the relative contributions could easily be seen. These models were constructed because the current measurement techniques were inadequate to resolve this. Reference [2] also describes the results of applying these models to experimental data.

It could be seen that the prediction and measurement were in good agreement for all the conditions of the experiment between 630 Hz and 2500 Hz, and for most conditions over the full frequency range of the experiment of 250 Hz to 5000 Hz. From these models the relative importance of wheel and rail in producing the total radiated sound was established. They were both found to be important at and above 1250 Hz. Below this frequency the rail was the major radiator. Unfortunately, this paper does not explain in detail the models used to predict the sound radiated from the wheel and the rail, concentrating on the relative importance of the wheel and the rail in the rolling noise.

New models for the sound radiation from the wheel and the track, described in reference [1], have been implemented in TWINS. The sound power is calculated by combining predicted vibration spectra with radiation efficiencies in one-third octave bands. Results in [5] show once more the wheel to be dominant at high frequencies and the track at low frequencies. However unlike the earlier work, the sleeper is now shown to be important at frequencies up to 250-500 Hz, or for a wooden sleeper even 1 KHz. Different models of radiation are available for the wheel, the rail and the sleepers. Much more detail is given in [6] for these radiation models. It is important to note that the radiation from the vibration of the ballast is completely ignored.
The free vibration of a wheel is also modelled in [1] by using finite elements.

Wheel types and reasonable agreement has been found with finite element predictions.

2.2 Wheel vibration

rail cross section is divided into four quadrilaterals.

Field of motion are distinguished: axial, radial and rotational (torque of the wheel).

Formulae for the radiation efficiency of various types of motion, known in three
1.2.3 rail vibration

While the wheel is a finite body, the rail is effectively infinite. It therefore acts as wave guide to structural waves which propagate away from the excitation point. At any given frequency, various different waves are permitted. At low frequency these are simple bending and torsional waves, but at higher frequencies the cross-section deforms allowing other wave types to propagate. Moreover, a rail has greater damping than a wheel, due to its supports (rail pad, sleepers and ballast) and this leads to a decay of the vibration with distance along the rail.

Unlike the wheel, the finite element method cannot be used directly to represent an infinite rail. However, it is possible to use periodic structure theory (see references [3] and [4]) to model an infinite beam-like structure such as a rail by considering it as a periodic structure of arbitrary periodic length. Finite element models can then be used to define the periodic element. The sleepers, railpad and ballast were also included, although only as equivalent continuous layer of mass/stiffness.

The receptance can be calculated from this periodic structure theory model by assembling the response at each frequency from the components of the various permissible waves. Finally predicted receptances were compared with published experimental results and good agreement was generally found.

Three alternative models, as described in [9], are available in TWINS for the track dynamics and are shown on figure 2:

* Track model 1: Continuously supported beam model.
  The track is considered as a Timoshenko beam on a continuous support. The support consists of a resilient layer (the railpads), a mass layer (the sleepers) and a second resilient layer (the ballast). It is similar to the continuous model of Grassie [10],
Experiments and simulations: Generally good agreement is found and the validity of the models is assessed by means of comparisons between experimental results obtained on several European tracks and presented in [11].

Figure 2: Model for track vibration: (a) Continuously supported beam; (b) Continuously supported track model including cross-section deformation.

In [3] and described above, Track model 3: Rail model including cross-section deformation developed and ballast and mass representing the sleeper.

systems, with hysteretic damping springs representing pad
track. The support consists, as above, of spring-mass-spring
The second track model includes the periodic nature of the

* Track model 2: Periodically supported beam model.

* Hysteretic loss factor rather than viscous dampers.

Although the damping of the pads and ballast is modelled by
1.2.4 wheel-rail interaction

The interaction between wheel and rail is modelled introducing a relative displacement input between these two components due to the roughness on the wheel and rail surfaces. The model, described in [3], has been constructed to couple the wheel and the rail receptances, allowing six coupling co-ordinates. It is an improvement on earlier models by Remington [8] and that also used by Grassie [10].

The rolling noise is generated by surface irregularities ("roughness") on the wheel and/or rail running surface. Therefore, roughness needs to be measured and included in TWINS. The input data consists of the spatial data on a series of parallel lines across the contact patch. A model is included to calculate the average roughness input to the system from these data in the spatial domain. The details of the input roughness are given in [1].

The roughness induces a vertical relative displacement between the wheel and the rail or in the Hertzian contact spring, the motion of each depending on the relative amplitudes (and phase) of their receptances. Coupling in the lateral direction is represented by a creep force element. The model of this wheel-rail interaction is described in [1] and shown on figure 3.

Figure 3: Details of the wheel-rail interaction.
receiving point and measuring the sound pressure \( p(X) \) on the sub-surface \( \Delta \). Place a small, omni-directional and calibrated microphone at the point \( X \) and use the reciprocity principle, by

\[
\frac{(X)\Delta}{(\lambda)d} = sp(t)n = \frac{(X)\Delta}{(X)d} = \frac{(X)\Delta}{(X)d}
\]

Principle of a vibro-acoustic reciprocity

The rate of sound from a vibro-acoustic source is determined by reciprocity. The rate of sound pressure and volume velocity, we would like to apply this method to.

The equation, \( (X)\Delta = \frac{(X)\Delta}{(X)d} \), is the fundamental principle that a number of the measurements presented in this report make use of the measurement of the vibration and acoustic radiation of a composite membrane sphere.
running machine to obtain the acoustic transfer function. The pressure was measured by a microphone.

It has been demonstrated that it was not only possible to predict the sound pressure at a receiver point but it follows from the results that the relative contribution of the sub-components of a vibrating engine to the sound pressure may be determined as well. Indeed, the sound pressure is obtained by integrating the contribution of all the sub-areas. Therefore, it is clear that the contribution of one component can be found by integrating only the area of this component. This is therefore quite interesting because it could be possible to try this technique on a track and determine at low frequency the relative contribution of the sleeper.

1.4 Description of the sleeper

The sleeper studied here is a re-inforced concrete design, manufactured by TARMAC. Its main dimensions are shown in figure 5.

![Diagram of a sleeper showing dimensions](image)

**Figure 5**: half a sleeper showing dimensions
Finally in chapter 4, the results are combined to derive the radiation efficiency of the heater. Chapter 3, the radiated sound power due to a unit force is determined reciprocally. Chapter 2, a modal analysis is performed on the heater. In the second, described in the work reported here consists of two experiments. In the first, described in 1.5 Work carried out.
2- MODAL ANALYSIS EXPERIMENT

2.1 Presentation of the experiment

The aim of this experiment is to excite the sleeper with an instrumented hammer to obtain the velocity distribution of the sleeper within a frequency range of 0-3200 Hz. Then, thanks to a modal analysis software, it is also possible to determine the natural frequencies, the mode shapes and the damping of the sleeper.

Then, in order to get the velocity distribution, the sleeper has to be meshed by dividing the structure into small areas delimited by points marked on the sleeper (node points). This mesh, once created on the structure, is re-created in the modal analysis software MESHGEN™ from ICATS. The co-ordinates of the nodes are listed in Appendix A and the mesh is shown in figure 6. It has to be noted that only half of the sleeper is considered in the study because of the symmetry of the structure and the assumption that both sides of the sleeper behave identically.

![Figure 6: Mesh of half a sleeper used in the modal analysis sofware.](image-url)
foundation is soft enough, the first bending mode occurs at about 120 Hz.
The first bending mode will be shown later to be about 22 Hz, which indicates that the
resonance frequency is too low for the resonator to be used effectively. The pieces of cork
provide a resilient foundation. The resonator is required to drive the amplifier and to store the
data.

Typewriter available.

Different types of accelerometers were used because not enough of a single

Remarks:

- pieces of cork
- PC Elora
- Analyzer HP3566A/3567 A
- 5 channel amplifiers BK 7635
- 1 accelerometer BK 470
- 3 accelerometers BK 493
- Hammer BK type 8202/1271063

In figure 2, the equipment used during this experiment is the following and was set-up as

22 SET-UP
2.3 Experiment

To obtain the velocity distribution, normally the structure has to be excited at one or a few points and the velocity (or acceleration) measured on all the structure. Nevertheless, the principle of reciprocity says that the transfer function from a source at one point to a response at a second point is the same as the transfer function with the source and the response positions interchanged. Therefore, as the number of accelerometers is limited compare to the number of nodes of the sleeper mesh, it is very time-consuming to use the direct measurement, i.e. one fixed excitation and several responses. On the other hand, by using a hammer for the excitation, it is relatively easy to move this from one point to another. Therefore four nodes have been
and the four transducers are each connected to one channel of the analyzer. Then, for
the Y-direction, which is supposed to be the side which resists the most. The hammer
The first set of accelerometers was fixed at nodes 3, 16, 27 and 40, all four in

Figure 10: Input power of steel tip

Figure 9: Input power of soft rubber tip

Figure 8: Input power of soft rubber tip

Study; this has therefore been used.
has the smallest decrease of power within the frequency range (0-3200 Hz) of the
lip was measured and recorded. See figure 8, 9 and 10. If occurring that the steel lip
power within the frequency range of the analyzer. Therefore, the input power of each
and soft rubber. In order to have good results, it is essential to have a good input
These different hammer tips are available for the hammer (steel, stiff rubber,
node have been excided by the hammer.
been chosen to be response points where the transducers are fixed, and the rest of the
each excitation with the hammer at any node of the sleeper, the Frequency Response Functions (FRF) between the velocity of the response points and the force input were recorded (after an average of 5) thanks to the analyser. It has to be noted as well that a uniform window was used within a frequency range of 0-3200 Hz with a resolution of 3201 lines (i.e. one line per Hz).

The same experiment was carried out with a second set of accelerometers fixed now at nodes 4, 18, 28, 42 in the z-direction. Once again, the FRF were recorded with the analyser.

2.4 Results and modal analysis

2.4.1 Frequency response function and data processing

Figure 11 below shows the FRF (Frequency Response Function) of the node 4 (channel 2 of the analyser during the first measurement) when excited at node 9. As the number of FRF is about a hundred, only one typical graph is displayed in this report.

![Example of a Frequency Response Function](image)

Figure 11: Example of a Frequency Response Function
...are recorded below. Which allows the analysis to be quick and accurate. The results of this modal analysis and chosen at random among the 120 files. The number of files used in this analysis is 17. Then, a multiple analysis is processed within the software using several FRF files. The first step of the analysis is needed to acquire a reasonable idea of the results. This model analysis was carried out first on several single analyses, i.e., with only one FRF model analysis was carried out on several files required by the software, the model analysis was created first on several single analyses. Once each file was compatible with the format required by the software, the

Thus, each FRF file was edited to add the relevant information in the excitation and the response, type of the FRF (in this case it is inertance), etc. These files needed a lot of information within each of the FRF files, such as position of software MODERN from ICATS. However, the model analysis within the software MODERN was still incomplete. After obtaining of the velocity distribution of the decker, it is possible to show

2.4.2 Modal analyses

From a combination of those files, the list of those files is given in Appendix B. Each of these files represents the FRF of a particular response point to a particular excitation point. The velocity distribution of the decker can be determined and written into four separate text files with "DIARY" and were called "FRF". Frequency response functions (FRF) were then imported into MATLAB. The data were separated into MATLAB form and was inserted into MATLAB form, and has to be divided into representing the coherence, cross-spectra, and period power. The data files "DAT" were separated into MATLAB form. "MAT" files for this software delivered with the package (.SHFTOML.EXT) was used.

Four accelerometers used simultaneously. Four different files for each of the Frequency response functions (one for each of the measurement has to be transferred into MATLAB form and was divided into transferred into files that can be read by the package. Each data file obtained during primarily, because using the modal analysis software, the data files have to be...
2.4.3 Results of modal analysis

The 17 FRF used within the modal analysis package are displayed in figure 12 and represents the inertance.

![Graph showing FRF](image)

Figure 12: The 17 FRF used in Modent

The natural frequencies and the damping loss factors of the mode of the sleeper are listed below in the Table 1 and 2.*

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency in Hz</th>
<th>Damping</th>
</tr>
</thead>
<tbody>
<tr>
<td>mode 1</td>
<td>21.5</td>
<td>11.04 %</td>
</tr>
<tr>
<td>mode 2</td>
<td>38.5</td>
<td>8.82 %</td>
</tr>
<tr>
<td>mode 3</td>
<td>119.6</td>
<td>0.79 %</td>
</tr>
<tr>
<td>mode 4</td>
<td>351.8</td>
<td>1.00 %</td>
</tr>
<tr>
<td>mode 5</td>
<td>382.6</td>
<td>1.42 %</td>
</tr>
<tr>
<td>mode 6</td>
<td>681.5</td>
<td>1.25 %</td>
</tr>
<tr>
<td>mode 7</td>
<td>1036.4</td>
<td>1.48 %</td>
</tr>
<tr>
<td>mode 8</td>
<td>1054.3</td>
<td>0.79 %</td>
</tr>
<tr>
<td>mode 9</td>
<td>1070.7</td>
<td>1.45 %</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency in Hz</th>
<th>Damping</th>
</tr>
</thead>
<tbody>
<tr>
<td>mode 10</td>
<td>1090.1</td>
<td>1.62 %</td>
</tr>
<tr>
<td>mode 11</td>
<td>1482.5</td>
<td>1.09 %</td>
</tr>
<tr>
<td>mode 12</td>
<td>1526.7</td>
<td>1.04 %</td>
</tr>
<tr>
<td>mode 13</td>
<td>1943.0</td>
<td>2.02 %</td>
</tr>
<tr>
<td>mode 14</td>
<td>1984.7</td>
<td>0.88 %</td>
</tr>
<tr>
<td>mode 15</td>
<td>2488.7</td>
<td>2.00 %</td>
</tr>
<tr>
<td>mode 16</td>
<td>2496.4</td>
<td>1.59 %</td>
</tr>
<tr>
<td>mode 17</td>
<td>2951.2</td>
<td>1.76 %</td>
</tr>
<tr>
<td>mode 18</td>
<td>2986.2</td>
<td>1.28 %</td>
</tr>
</tbody>
</table>

Table 1

Table 2

* See chapter 5 for comparison with results from another study by Vibratec.
Figure 13: Relevant systems used for reciprocal measurement

Forms, but for the current situation, the relevant systems are shown in Figure 13.

The principle of reciprocity dictates an equivalence between two systems:

\[ \text{Reciprocity Relation:} \]

\[ \text{Stepper: } W/F \]

To obtain a relation between the radiated sound power and the force input to the aim of this experiment is to carry out a reciprocal measurement on the stepper in order. The second experiment took place in the reverberation room of the ISVR.

3.1 Preceding

3.1 Presentation of the experimental and theory.
In system 1, a sound pressure is generated at B, \( p_B \), due to a force input on the structure at A, \( F_A \), whereas in system 2, an acceleration is generated on the structure at A, \( a_A \), due to a volume acceleration sound source at B, \( U_B \). The reciprocity relation in this case is:

\[
\frac{p_B}{F_A} = -\frac{a_A}{U_B} \tag{2}
\]

Use of such a relation allows measurements of the transfer function from force to radiated sound to be made indirectly by measuring the acceleration on a structure due to a known external sound pressure. This has the advantage that a mechanical shaker does not need to be used, which thus avoids noise radiation by the shaker, as well as difficulties of introducing a force in a given direction.

\* Sound power in a reverberation room

In order to apply this principle to measurement in a reverberation room, a relation is first required between an r.m.s sound pressure measurement, represented here by \( s \), and the overall sound power, \( W \), in the room. This is given as

\[
W = s^2 \frac{1382 V}{\rho c^2 T_{60}} \tag{3}
\]

where \( V \) is the volume of the room, \( T_{60} \) is the (frequency dependent) reverberation time (i.e. the time taken for a signal to decay to -60 dB of its level), \( \rho \) is the density of air, and \( c \) is the speed of sound in air.

It is assumed here that the sound pressure level has a uniform distribution through the room, which is true for a sufficiently high acoustic modal density. In practice, it would be measured as an average over a number of locations, or more practically by spatially averaging using a revolving microphone (on a boom).
(6) \[ \frac{d^t}{d^t} = \frac{d^t}{d^t} \]

(8) \[ \frac{d^t}{d^t} = \frac{d^t}{d^t} \]

(7) \[ \frac{d^t}{d^t} = \frac{d^t}{d^t} \]

(9) \[ \frac{d^t}{d^t} = \frac{d^t}{d^t} \]

Finally, these equations can be assembled to give a relation between a measureable quantity and the sound pressure in the room (\( \varphi \)) and the distance between the source and the microphone: the ratio of the acceleration on the microphone to the sound pressure.

\[ \frac{d^t}{d^t} = \frac{d^t}{d^t} \]

A Reciprocal Relation for the Reverberant Room

\[ \frac{d^t}{d^t} = \frac{d^t}{d^t} \]

It should be noted that this relation is independent of the position of the source, \( U \).

\[ \frac{d^t}{d^t} = \frac{d^t}{d^t} \]

From the above equation (3), this means that for a reverberant room, the source can be replaced by:

\[ \frac{d^t}{d^t} = \frac{d^t}{d^t} \]

A volume acceleration source of r.m.s. amplitude \( U \) generates a sound power

\[ \frac{d^t}{d^t} = \frac{d^t}{d^t} \]

A Sound pressure in a reverberant room due to a source \( U \)
Taking $\rho = 1.21 \text{ Kg/m}^3$, and $c = 340 \text{ m/s}$, this can be expressed in terms of decibels, as

$$10 \log \left( \frac{W}{W_0} \right) = 10 \log \left( \frac{a^2}{p^2} \right) + 84.5$$

(10)

where $W_0 = 10^{-12} \text{ Watt}$ so that the result is in dB re $10^{-12} \text{ Watt/N}^2$.

3.2 Set-up

The instrumentation used during the experiment is as follows:

- Accelerometer B&K 4378/1187979 of sensitivity 31.7 pC/ms$^2$.
- Charge amplifier B&K type 2635.
- Microphone B&K 4133/873080.
- Preamplifier B&K type
- Rotating boom B&K type 3923.
- Measuring amplifier B&K type 2609 20-20000 Hz.
- Oscilloscope OS306 Gould 20 MHz.
- Analyser HP3566A/3567A.
- Noise generator (module of the analyser).
- Power amplifier HP400 high performance dual.
- PC Elonex
- Two Lifters, one Clarke-strong arm and one Weber Hydraulik
- Loudspeaker

The set-up is as following (see figure 14).
The stepper was measured using a sensitive transducer. Several measurements were made around the measurement of the acceleration of the stepper, the power spectrum of the pressure of the power amplifier was taken. The noise generated by the noise floor inside the reverberation room. The white noise was generated by the noise source inside the reverberation room. To generate a uniform distribution of the sound pressure, the chamber facing the wall was excited by a white noise from a loudspeaker placed in a corner of the room. The floor to avoid the phenomena of standing waves underneath the structure. The inside reverberation room, the stepper was placed 1.5 meters from the medallion.

3.3 EXPERIMENTAL

Figure 14: Set-up of the reverberation chamber experiment.
carried out for different positions of the transducer on the sleeper. Both of the power spectrum were recorded with the analyser.

3.4 Calibration

The results were stored for the first experiment in different files. The same processing as in section 2 was applied to these files to transfer them into MATLAB format and separate the different channels of the analyser. Then the calibration can be processed in MATLAB software.

The calibration of the power spectrum of the pressure inside the chamber was performed using a calibrator of 94 dB. The output voltage read on the measuring amplifier was recorded while the calibrator was on the microphone. It has to noted that the calibration must be done with the complete set-up identical to the one used during the real measurement.

\[ x \text{ Calibration of pressure:} \]

\[ 94 \text{dB} = 20 \log \left( \frac{p}{p_0} \right) \quad \text{with} \quad p_0 = 20 \times 10^{-6} \text{ Pa} \]

which gives \[ \Rightarrow \quad p = 1.002374 \text{ Pa} \]

The output voltage during the calibration was \( V = 13.5 \text{ mV} \) which is related to the pressure \( p \). Then, the pressure \( P_c \) related to a 1V output is:

\[ P_c = \frac{p}{V} = \frac{1.002374}{0.0135} = 74.25 \text{ Pa/V} \]

Thus, to find the power spectrum of the pressure, the voltage has to be multiplied by \( P_c \):

\[ < p^2 > = < v^2 \cdot (74.25)^2 > \]
\[<v_2^2> = <v_3^2> = \frac{\Delta v}{0.1 \text{ m/s}}\]

By 0.1 m/s

Thus, to find the power spectrum of the acceleration, the voltage has to be multiplied by \(0.1 \text{ m/s}^2\)

1000 mV = 0.1 m/s^2

The gain on the charge amplifier was set at 1000 mV/1000 mV/s, until output:

1 unit output = 0.1 m/s^2
10 unit output = 1 m/s^2

Therefore:

Too high to be set on the charge amplifier, the sensitivity was set at 3.17 pC/m/s^2. But as this sensitivity is charge amplifier and the sensitivity of the transducer:

The calibration of the power spectrum of the acceleration is related to the

\[X \text{ Calibration of acceleration}\]
### 3.5 Results

The power spectrum of the acceleration and the pressure are displayed below on figure 14 and 15.

![Power spectrum of pressure](image1)

**Figure 14**

![Power spectrum of acceleration](image2)

**Figure 15**

Then, knowing the power spectrum of the pressure and the acceleration, the relation between the radiated sound power and the force applied to the structure can be found using equation (9). Figure 16 shows $W/F^2$ in logarithmic scale (using equation (10)) with $re \ dB = 1W/N^2$.

![Logarithmic scale](image3)

**Figure 16**
The radiation efficiency for each mode, the results is as follows:

\[ \frac{\epsilon \omega}{k^2} \ln \left( \frac{\epsilon H}{k^2} \right) = \omega \frac{\epsilon H}{k^2} = \frac{\epsilon H}{k^2} \]

This is done using the following formula, which is derived from the incident field. Therefore, in order to find the radiation efficiency, the velocity distribution of the incident field is used. The velocity distribution obtained after the hammer measurement is in fact the hammer excitation, as follows:

\[ \left( \frac{c}{W} \right) \sum \frac{c}{d} \frac{c}{W} = \frac{< \omega > S_{pc}}{W} = \sigma \]

The radiation efficiency of the steeper will be calculated by the following formula:

4.1 Radiation efficiency

4. DATA PROCESSING AND RESULTS
4.2 Mode shape

The following mode shapes were obtained. The fundamental mode (= 22 Hz) shown is a rigid body mode of the sleeper on the cork supports. The other modes are primarily flexural bending along the length of the sleeper.
5. DISCUSSION

5.1 Comparison of natural frequencies

In the reference [16], Vibratec has carried out an experimental modal analysis on a monobloc concrete free sleeper type B 70-W. The results found in this project on a free reinforced concrete sleeper are recalled from chapter 2 in table 1 and 2. For comparison, the results of Vibratec are displayed in table 3.

<table>
<thead>
<tr>
<th>Mode</th>
<th>Frequency (Hz)</th>
<th>Damping (in %)</th>
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<tr>
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**Table 1**

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**Table 2**

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<td>3rd bending</td>
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<td>4th bending</td>
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<td>8th bending</td>
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**Table 3: Vibratec modal analysis results**

Comparison between the results is reasonable if one considers modes 3, 4, 6, 7, 11, 13, 15 and 17 and the 8 bending modes of the Vibratec tested sleeper. The additional modes on the Tarmac sleeper could be the results of coupling with torsional...
from the results of data processing error.

A|80, it is still unclear whether this error comes from a calibration mistake (a

non-calibrated hammer or an error in the microphone calibration) or whether it comes
about 20 dB at high frequencies, which is unreasonable.

frequencies. However, the results displayed in Figure 18 show a radiation efficiency
result of chapter 2 and 3, appears quite unreasonable. Indeed, the radiation efficiency
results of chapter 2 and 3, appears quite unreasonable. Indeed, the radiation efficiency

The final results of the radiation efficiency found in chapter 4, combining the

5.2. Radiation efficiency

are probably not significantly different than the B 70 W speaker.

modes of the speaker. The variation of damping is wide in the result of this study but
6. RECOMMENDED FURTHER WORK

Having obtained the radiation efficiency and the modal characteristics of the free sleeper, it is also important to check the effect on the noise radiation due to the fact that the sleepers, in real conditions, are in situ in the ballast.

It could then be possible to characterise the effect of the ballast on the sleeper (e.g. natural frequencies, mode shapes and damping as well as radiation efficiency) by comparing lab measurement on a free sleeper (that has been done during this project) and field experiment. This latter experiment is to be carried on a test track.

The installation of this railway track has been arranged by British Steel Track Products and installed by Grant Rail as subcontractors to British Steel for the ISVR at Chilworth.

The experiment to be carried out is an intensity measurement over the sleeper (uncoupled to the rail) and the ballast surrounding it, in order to get the sound power radiated from each item. The excitation of the structure would be an electrodynamic shaker (positioned near the rail footing) covered by an absorbent sealed box to reduce subsequently the extraneous noise transmitted through it. Also, it is important to obtain the velocity distribution of the sleeper in situ. This could be done by using acceleration transducers over the sleeper. Then, by integrating the intensity (see equation (13),

\[ \frac{W}{F^2} = \int \frac{I}{F^2} ds \]  \hspace{1cm} (13)

the sound power radiated can be found and then combined with the velocity distribution to calculate the radiation efficiency of the sleeper in situ using equation (11) described in chapter 4. From the sound power radiated from the sleeper and the
Consequence of this relationship could also be investigated.

It could include a phase relationship between direct and reflected rays. The terms of absorption or in terms of acoustic impedance. Additionally, the reflection of absorption or in terms of absorption. Therefore, it could also be interesting to study the ballistic in the ballistic is available. Therefore, it could also be interesting to study the ballistic over the frequency range.

Finally, the noise coming from the walls measured in the field with ballast. It should also be possible to compare their contribution to the sound radiation
APPENDIX I

Coordinates of the nodes of the sleeper mesh.

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<td>48</td>
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</table>
\( \chi \) and \( \nu \) are the direction of measurement of the acceleration.

\( \chi \) where \( \nu \) is the position of the acceleration during the measurement in the room.

\( I \) is the position of the acceleration in the room.

**Powerlaw**

- File containing the power spectrum information (exponent in Revere\(er\) chamber):

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\( \chi \) and \( \nu \) are the direction of excitation (1 = \( x \), 2 = \( y \), 3 = \( z \)).

\( \chi \) where \( \nu \) is the excitation point (with hammer).

\( \chi \) where \( \nu \) is the number of the node where the accelerometer is fixed.

Sweep

\( \chi \) where \( \nu \) means the first experiment (i.e., position of accelerometer on the top of the column).

Format of file: \( \text{AM-FTH} \)

List of the frequency response function files used to find the radiation efficiency:

**APPENDIX II**
REFERENCES:


Experimental model analysis.

Experimental characterization of ballast properties. See Chapter 4.3.1.

16. N. PREMONT, J. P. GOUTARD and N. VINCENET 1996 VIBRATIONS REPORT

Transmission measurement using a reciprocity technique.


Sound radiated by a muffled IC engine

348. Application of a visco-elastic reciprocity technique to prediction of


noise transmission through aircraft fuselages.

43-52. Development of a reciprocity technique for the prediction of propeller


2: Experimental results and comparisons with theory.

Supplement 24, 100. 114 Track dynamic behaviour at high frequencies. Part

11. N. VINCENET and D. THOMPSON 1996 Vehicle System Dynamics

of the railway track to high frequency vertical excitation.


Theoretical models and laboratory measurements.
