Virtual test method of structure-borne sound for a metro bogie

Gang Xie[1], Martin Rissmann[2], Pascal Bouvet[2], Xiaowan Liu[3], David Thompson[3]

Luis Baeza[3], Juan Moreno[4], Julián Martín Jarillo[4],

Francisco D. Denia[5], Juan Giner-Navarro[5], Ines Lopez Arteaga[6]

[1] CDH-AG, Competence Center Vibration & Acoustics, 70565 Stuttgart, Germany
Email: gang.xie@cdh-ag.com
[2] Vibratec, Chemin du Petit Bois BP 36, F-69131 Ecully Cedex, France
[3] Institute of Sound and Vibration Research, University of Southampton, Southampton SO17 1BJ, United Kingdom
[4] Servicio de Ingeniería de Material Móvil, Área de Ingeniería, Metro de Madrid
[5] Centro de Investigación en Ingeniería Mecánica (CIIM), Universitat Politècnica de València
[6] Department of Aeronautical and Vehicle Engineering, KTH Royal Institute of Technology, SE-100 44 Stockholm, Sweden

Abstract. This paper presents a virtual test method for structure-borne noise generated from railway running gear. This method combines a number of existing tools to form a system approach. The wheelset and bogie frame are modelled using FEM software Nastran to include details of their construction. The primary springs are simplified to standard CBUSH elements in Nastran with point and transfer stiffness modelled by frequency-dependent complex stiffness, which are tuned against measurements. The wheel-rail contact forces due to roughness excitation are obtained by the wheel-rail interaction tool TWINS. The vibration of the full running gear is simulated in Nastran by applying the wheel-rail contact forces. The forces transmitted to the vehicle body through traction bars and dampers are calculated for predicting structure-borne noise.

Keywords: bogie, structure-borne sound.

1 Introduction

The concern of environment, climate change and increased living standards have driven road, railway and air industries to provide innovative technologies to develop better and more sustainable transport systems. The need to design quieter and more comfortable trains has never been higher than in the present time.

The most important source of noise from the rail transport system is rolling noise caused by wheel and rail vibrations induced at the wheel/rail contact [1]. This is particularly the case for low-speed metros and light rail systems. The mechanism of rolling noise has been well understood through many years of research and a validated modelling tool, TWINS was also developed by Thompson et al. [2] to predict wheel/rail contact forces and radiated noise of wheel and rail. Rolling noise is transmitted to the interior of vehicles through not only airborne paths, but also structure-borne paths, which include the whole running gear system. Airborne transmission tends to be dominant at higher frequencies while structure-borne transmission at lower frequencies. The difference between the mechanisms of airborne and structure-borne transmission also leads to different modelling approaches [1]. This paper focuses on the structure-borne noise. To control the structure-borne noise and vibration inside the vehicle, the whole running gear system including the bogie, suspensions and wheelsets must also be considered. However, a whole-system approach intended for use in design has not been reported before.

In the automotive industry, the full vehicle structure is digitized and its operational conditions are simulated by computers to predict interior acoustic responses and to approve vehicle designs. The modelling approach is mainly based on approaches such as the finite element method (FEM). As many similarities for the noise and vibration problems are shared between railway and automobile industries, this type of virtual test process used in the automobile industry can also be applied to rail vehicles for structure-borne noise. The key to build a good virtual test process is to have a validated engineering method and efficient computational techniques. This paper presents a comprehensive virtual test method for the structure-borne noise inside railway vehicles.

2 Methodology

The structure-borne path starts at the wheel/rail contact and propagates through the suspensions and bogie frame into the car body. The proposed method combines FEM for the running gear and the established TWINS method for the wheel/track interaction, which together lead to a whole system approach. A trailer bogie of a metro train is used as an example for the proposed method. Structural components of the running gear, including the bogie frame, traction bars, dampers, wheelsets and axle boxes are modelled using FEM. Special attention must be given to modelling suspension elements as they often show frequency-dependent characteristics. The rubber bushings of the traction bar and dampers are modelled with complex stiffnesses, which have been obtained from laboratory measurements. The primary springs have been modelled as a series of lumped masses and springs. The stiffness of the springs also takes account of the internal mass effect of the rubber elements and has been tuned with the measured dynamic stiffness. The wheel/rail contact forces can be calculated using TWINS with measured wheel and rail roughness. For the full bogie system, each wheel is excited by the wheel/rail forces and frequency response analyses are then carried out in FE software. Dynamic forces transmitted to the train body through the connection points obtained from frequency response analyses can then be used for transfer path analysis (TPA). The structure-borne noise can then be predicted by using these forces together with noise transfer functions from forces at these positions to the interior sound pressure. These could be obtained either through simulations or measurements.

For frequencies up to 1 kHz, the number of acoustic modes is very high due to the size of the interior cavity of the railway vehicle. It is therefore very rare that the full vehicle body is modelled using FEM. In the present project measured transfer functions have been used. The model has been validated against field-test data obtained from an experimental campaign.

3 Structure-borne noise transmission model

3.1 FE model

In the present study, a trailer bogie of a Metro de Madrid vehicle is modelled using the FE package Nastran. The full FE model includes the bogie frame, a front wheelset, a rear wheelset, axle boxes, primary suspension springs, lateral dampers and traction bars, as shown in Fig. 1. The wheelsets, bogie frame and axle boxes are modelled using solid elements. The structural part of dampers and traction bars could also be modelled using solid elements or appropriate one-dimensional beam elements; the latter approach is used here. They are connected by bushing elements to the bogie. The primary suspension spring is modelled using spring elements with frequency-dependent complex stiffness without FE representation of its structure. The whole FE model consists of over 1.36 million nodes.

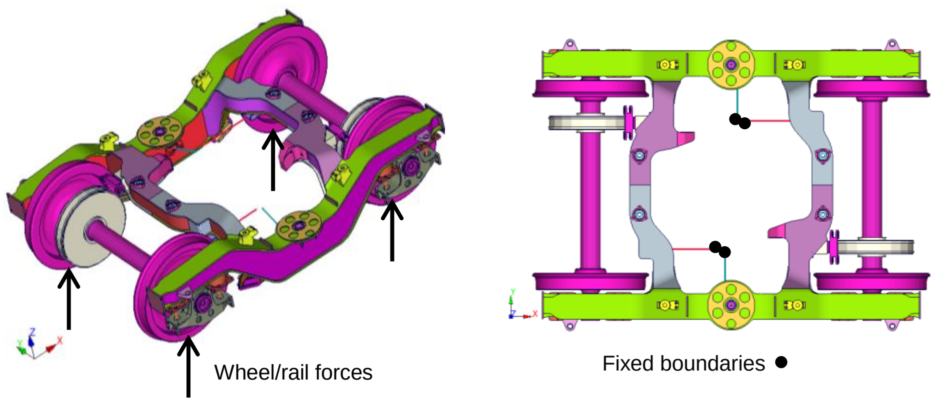

Fig. 1. The FE model for the bogie with dampers and traction bars modelled by 1D bar elements.

3.2 Frequency-dependent stiffness

The primary suspension spring is built with four layers of metal and rubber, as shown in Fig. 2. The dynamic model is composed of a series of masses and springs [3]. The mass-spring model was extended to include wave motion of rubber elements. As result, each spring has frequency-dependent point and transfer stiffness and this model gave the best correlation with dynamic tests carried out in laboratory. To simplify the global FE model, the series of masses and springs was reduced to a single frequency-dependent spring for each direction, connected between wheelset and bogie frame. The stiffness matrix is expressed as

$$K = \begin{bmatrix} k_{\mathrm{po}} & -k_{\mathrm{tr}} \\ -k_{\mathrm{tr}} & k_{\mathrm{po}} \end{bmatrix} \quad (1)$$

Note that the point stiffness k_{po} on the bogie side is different from the wheelset side due to the asymmetric build of primary spring whereas the transfer stiffnesses k_{tr} are equal by reciprocity. To represent this stiffness matrix in Nastran, CBUSH element was used [4]. CBUSH element is a general spring and damper element with the capability of including frequency dependent stiffness and damping. A conventional CBUSH element, connecting bogie and wheelset, and two grounded springs on both bogie and wheelset sides have to be combined. These three springs together give the stiffness matrix as

$$K = \begin{bmatrix} k_{\mathrm{tr}} & -k_{\mathrm{tr}} \\ -k_{\mathrm{tr}} & k_{\mathrm{tr}} \end{bmatrix} + \begin{bmatrix} k_{\mathrm{po}} - k_{\mathrm{tr}} & 0 \\ 0 & 0 \end{bmatrix} + \begin{bmatrix} 0 & 0 \\ 0 & k_{\mathrm{po}} - k_{\mathrm{tr}} \end{bmatrix} \quad (2)$$

where the first part is a normal CBUSH and other two are grounded springs.

For the traction bar and lateral damper, the measured dynamic stiffnesses of the bushing are applied to local coordinates which are defined to be the same as that used in the measurements.

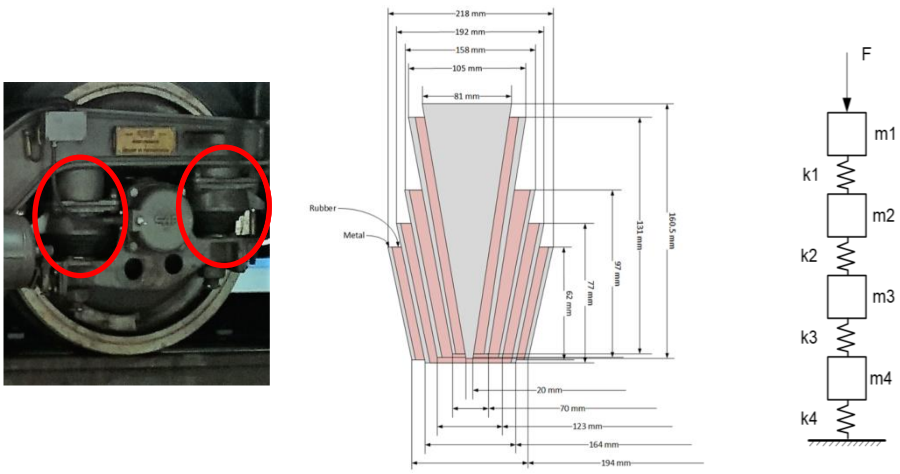

Fig. 2. Left, primary spring. Middle, cross section layout of rubber and metal. Right, the model.

The frequency response calculation in FEM is usually carried out using a modal summation approach to reduce the computational cost. The modal equation is given by

$$(-\omega^2[m] + i\omega[b] + [k])\{q(\omega)\} = f(\omega) \quad (3)$$

where $[m]$,$[b]$ and $[k]$ are modal mass, damping and stiffness matrices of the FE model and $q(\omega)$ contains the modal generalised coordinates. This approach can also be used for a system which contains frequency-dependent stiffness and damping. In this case, the mode shape obtained from the constant stiffness is used and a modal correction to the modal damping and stiffness matrices is carried out to account for the frequency-dependent part of the stiffness. Eq.(3) becomes

$$(-\omega^2[m] + i\omega[b + \Delta b] + [k + \Delta k])\{q(\omega)\} = f(\omega) \tag{4}$$

where Δk and Δb are modal correction terms which are due to ΔK and ΔB, the changes of stiffness and damping in physical coordinates, given by

$$\Delta b = [\phi]^T[\Delta B][\phi] \tag{5}$$

$$\Delta k = [\phi]^T[\Delta K][\phi] \tag{6}$$

where $[\phi]$ is the matrix of mode shapes.

3.3 Wheel/rail contact force

The contact force $\vec{F}_{\text{contact}}$ resulting from the wheel-rail interaction is calculated using a TWINS-like approach [2] as follows. For this, the roughness excitation $\vec{R}$, and the receptance matrix $[A]$ of the wheel, rail and contact at the contact point are required. The contact force $\vec{F}_{\text{contact}}$ is then calculated according to

$$\vec{F}_{\text{contact}} = \begin{pmatrix} F_z \\ F_y \end{pmatrix} = ([A]_w + [A]_r + [A]_c)^{-1}\vec{R} \tag{7}$$

The roughness excitation $\vec{R}$ is actually an imposed displacement in the vertical direction, such that $\vec{R} = (R_z, 0)$. Furthermore, the input roughness is filtered by the contact patch. This filtering effect depends on speed and wheel load. A Remington contact filter will be used here [1].

The receptance matrices $[A]$ of the wheel, rail and contact are calculated by different means [3]. In the present paper, $[A]_w$ is calculated with the FE method; $[A]_c$ and $[A]_r$ are calculated with TWINS.

4 Results

4.1 Contact forces

Fig. 3 shows the lateral and vertical contact forces calculated from filtered roughness excitation. The wheel load is 6 tons and the vehicle speed is 60 km/h. The track dynamics was modelled with TWINS. Wheels 3 and 4 are the rear left and rear right wheels as shown in Fig. 1. Due to asymmetry of the wheelset, the left and right wheels of a wheelset have different contact forces.

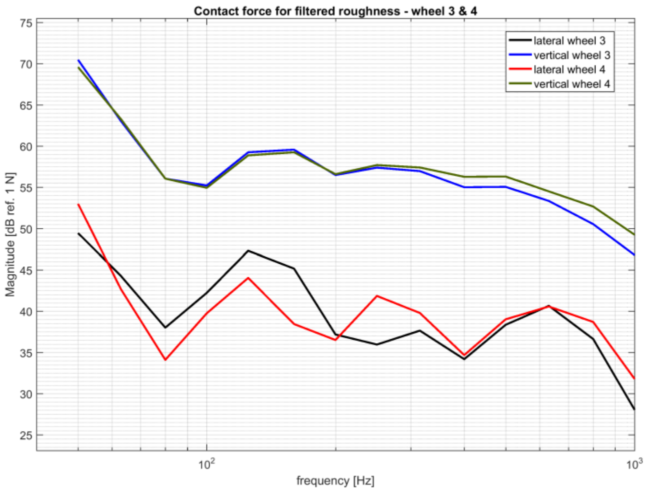

Fig. 3. Wheel/rail contact forces.

4.2 Forces acting on vehicle body

Since the vehicle body was not considered, the so called blocked-force method was used. Therefore the connecting points on the bolster, to which the traction bars and lateral dampers are attached, are fixed in the Nastran calculations. The forces acting on these four nodes are forces transmitted to the vehicle. For each wheel, vertical F_z and lateral F_y contact forces are correlated (longitudinal contact forces are negligible). However, contact forces on different wheels can be regarded as uncorrelated. Therefore, the forces acting on the vehicle body can be obtained by superposition of the force calculated for each of the four wheels individually. Such calculation can be readily carried out with the random analysis capability available in Nastran.

Fig. 4 (top figure) shows the resultant forces acting on the vehicle body by the traction bars and lateral dampers. The lateral dampers contribute more to the resultant force than the traction bars, in particular between 200 and 500 Hz.

4.3 Interior noise

The interior noise due to a single wheel is calculated by multiplying the transmitted forces obtained from above with transfer functions.

$$p_i = \sum_{j,k} H_{kj} F_{kj} \tag{11}$$

where i is 1 to 4 wheels, k is 1 to 4 corresponding to four connection positions to the vehicle body and j is 1 to 3 DoFs. The total structure-borne noise is finally given by

$$p = \sqrt{\sum_i |p_i|^2} \tag{12}$$

The transfer functions obtained through measurements and the predicted structure-borne noise are shown in Fig. 4 (middle and bottom figures). The main transfer paths can be identified as the longitudinal direction of the traction bar and the lateral direction of the lateral damper.

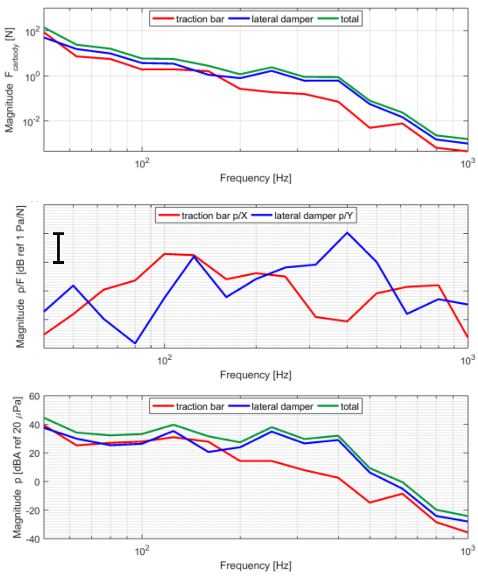

Fig. 4. Transmitted forces, measured noise transfer functions and predicted structure-borne noise from traction bars and lateral dampers

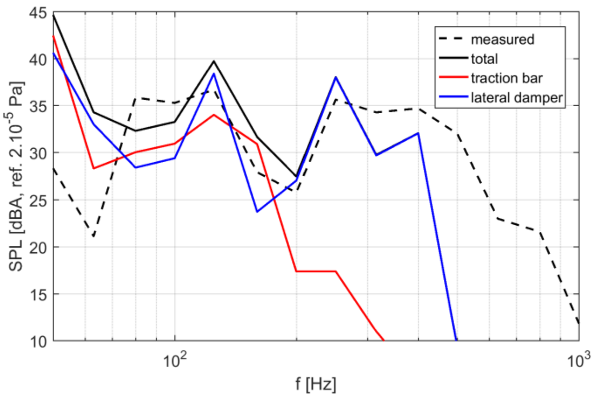

Fig. 5. Comparison of structure-borne noise levels inside the vehicle for a microphone above a trailer bogie.

Fig. 5 compares the total structure-borne noise predicted for one microphone position inside the vehicle above a trailer bogie with the result from the experimental TPA as well as predicted contributions of the traction bar and the lateral damper. A very satisfactory agreement in the total structure-borne noise is observed between 80 and 400 Hz, which corresponds to the frequency range for which structure-borne noise was found to be significant during tests.

At low frequencies (below 50 Hz), discrepancies could be related to the measured rail roughness or track impedance. Above 400 Hz, the computations under-estimate the structure-borne noise contribution, which is likely to be due to inconsistent stiffness for the rubber bushings at the extremities of the lateral dampers and traction bars.

5 Conclusions

A virtual test method for the structure-borne noise of railway running gear has been presented in this paper. The bogie, wheelsets and suspensions have been modelled using FEM including dynamic properties measured in the laboratory. The excitation force on the wheel-rail contact has been calculated using TWINS. The blocked forces method was used to obtain the transmitted forces to the vehicle body. A case study implementing this method has been carried out on a metro vehicle using Nastran. The structure-borne noise inside the vehicle has been predicted using measured noise transfer functions. The developed model for the metro bogie requires validations against measurements, which will lead to further refinement and update to the model. Having validated the model, the developed virtual test model can be used to evaluate the impact of design changes at early stage, such as new material, component or noise and vibration countermeasure. Industry can potentially significantly shorten the design cycle and reduce costs of hardware tests.

Acknowledgement

The work presented in this paper has received funding from the Shift2Rail Joint Undertaking under the European Union’s Horizon 2020 research and innovation programme under grant agreement no. 777564. The contents of this publication only reflect the authors’ view and the Joint Undertaking is not responsible for any use that may be made of the information contained in the paper.

Provision of vehicle data from CAF Spain is gratefully acknowledged.

References

1. Thompson, D. Railway noise and vibration: mechanism, modelling and means, 1st Edition, Elsevier, Oxford, UK (2009).
2. Thompson, D., Hemsworth, B. and Vincent, N., Experimental validation of the TWINS prediction program for rolling noise, part 1: description of the model and method. Journal of Sound and Vibration, 193, 123-135 (1996).
3. Report: Deliverable 4.2 – Complete virtual test method for structure-borne and airborne noise transmission, EU Shift2Rail/H2020 project RUN2rail, 2019.
4. MSC NASTRAN Quick reference guide, 2018