Industrial Methodologies for the Prediction of Interior Noise Inside Railway Vehicles: Airborne and Structure Borne Transmission

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Abstract. This paper presents computational methods that are used by rolling stock manufacturers to predict noise inside vehicles. For airborne transmission, which dominates in the medium-high frequency range, a four-step procedure is applied: source description by their emitted sound power level, propagation of noise to the train's exterior surface, panel transmission loss and acoustic response of the interior cavity. Reasonable agreement between computations and measurements is usually obtained, and the method makes it possible to rank the different source contributions and airborne transmission paths. Structure borne noise dominates in low frequencies. Finite Element models are used to improve car body design (dynamic stiffness at input points and carbody vibroacoustic transfers), but they do not cover the whole problem since the modelling of excitation from the bogie is not included. Recent research allowing the computation of blocked forces at car body input points and starting with wheel/rail interaction is briefly presented. Concerning source modelling, a focus is made on traction noise, including electromagnetic excitations in electric motors and mechanical excitations due to the meshing process inside gearboxes. Efficient computational methods and validation examples are presented. The coupling of these methods with optimization methods has great potential for improvement of motor noise and vibration design.

Keywords: Acoustic comfort · Airborne and structure borne transmission · Acoustic sources · Gearbox noise · Traction motor noise

1 Introduction

The comfort inside trains is one of the most important issues for railway operators, and studies have shown the importance of the acoustic aspect for passengers. The interior noise levels of new rolling stock are specified by railway operators from the tender stage, for various train operating conditions (standstill, acceleration and braking, running at constant speeds).

For interior noise prediction during the different vehicle development phases, rolling stock manufacturers need efficient and accurate modelling tools, taking into account the different sources, transmission paths, environmental (free field or tunnel circulation) and

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operating conditions. A description of computational methods for airborne and structure borne noise transmission in use in the railway industry is presented in Sect. 2 below.

Section 3 is dedicated to source modelling, with a focus on recent developments to model traction motor noise due to electromagnetic excitation and gearbox noise due to the meshing process.

2 Modelling Methods for Interior Noise Prediction

2.1 Relative Contributions from Airborne and Structure Borne Transmission Paths

Usually, at locations close to bogies, the contribution of airborne noise transmission is dominant at high frequencies whereas structure borne noise transmission dominates at lower frequencies (typically below 200–300 Hz).

Figure 1 gives an example of structure borne contribution to the total interior noise. In this case, the structure borne contribution was assessed using Experimental Transfer Path Analysis (TPA) and in-situ blocked force measurement at the bogie attachment point to the vehicle body [1]. Several examples of TPA analysis on different types of railway vehicles have shown a predominance of structure borne transmission in the low frequency range [2, 3].

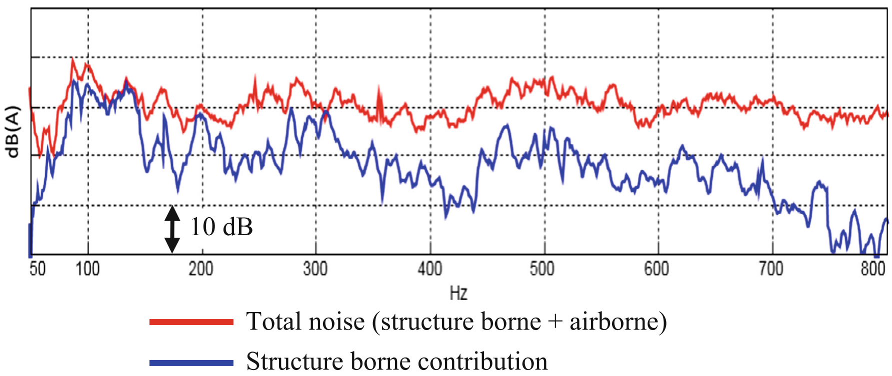

Fig. 1. Sound pressure levels inside an urban railway vehicle

2.2 Airborne Noise Model

For more than 15 years, rolling stock manufacturers have implemented and regularly improved computational methods to estimate the airborne contribution inside vehicles. A global overview and critical analysis of methods that can be used for interior noise prediction is given in ref. [4, 5].

For the prediction of interior noise, different sources (rolling noise, traction noise, equipment noise and aeroacoustic excitation), transmission paths and operating conditions must be taken into account. Airborne noise models are built from the tendering phase to roughly assess the acoustic solutions to be used in order to meet interior noise requirements. When possible, this first model is based on input data from previous or similar vehicles. In a second step, the model is used to establish acoustic target settings for each component (panel transmission loss (TL), source sound power, etc.). It is then regularly updated throughout the project when new acoustic data are made available from suppliers or from acoustic tests on components. The methodology applied to calculate the air-borne noise component is composed of four steps as shown in Fig. 2:

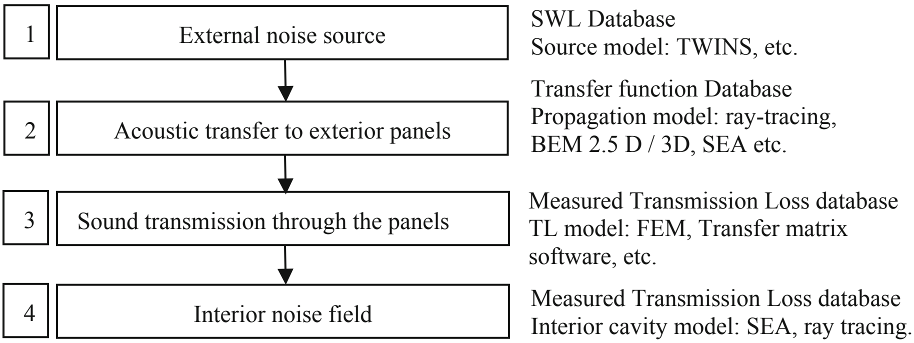

Fig. 2. Railway vehicle. airborne noise computation method

Step 1 - Noise source description of: Globally, external sources cover

- Mechanical sources (wheel/rail noise, traction noise, auxiliary noise, etc.);
- Interior sources (HVAC, ventilation, etc.)
- For high speeds: aerodynamic sources (such as turbulent flow generated around the bogie and inter-coach spacing), Turbulent Boundary Layer (TBL) around the coach.

Depending on source types, the source sound power levels can either be computed (TWINS calculations for wheel/rail noise, Computational Fluid Dynamics (CFD) computations for aerodynamic sources or TBL, Finite Element Method (FEM) computations for gearbox and traction motor noise) or extrapolated from measurement databases.

An extension of the validation of TWINS for cases like resilient wheels or track radiation in the low frequency range is still required [5]. For fans, acoustic source data can be affected by their installation on the vehicle, because of flow disturbances at the inlet that may be different between laboratory tests and vehicle conditions.

Step 2, Noise transmission to coach panels: The sound transmission from sources to the coach's external envelope is described by means of (Lp_i - Lw) transfer functions, Lp_i being the parietal sound pressure at any point i of the coach envelope and Lw being the source sound power. These transfer functions must be determined for various environmental conditions (free field and tunnel).

A global overview of methods that can be used for in ref. [6]. Several computational tools such as 3D Boundary Element Method (BEM) or ray tracing are sometimes used to compute these transfer functions [7]: this remains quite challenging with regard to the necessity to cover the whole 100 Hz – 5 000 Hz frequency range, the whole geometry of the vehicle and very different acoustic environments (reverberation in tunnels, free field diffraction effects, etc.), while keeping reasonable accuracy and computational time. Because of these difficulties, experimental databases of transfer functions measured with artificial sources at standstill or in rolling conditions on similar coach geometries are still used, with the main drawback that in the case of a new rolling stock geometry they can hardly be transposed from one vehicle to another.

Consequently, this is a field where research efforts are still needed: more recently a 2.5D boundary element method was used to predict the exterior sound pressure spectrum on the train sides when the train was in running operation. Reasonable agreement was found with measurements [8].

Steps 3 and 4, Transmission through panels and interior noise field: panel Transmission Loss (TL) is usually taken from a database, either from laboratory measurements or from in-situ measurements on the vehicle. Simple computational tools such as transfer matrix methods are often used to estimate tendencies, but with limited accuracy regarding the complexity of the car body's structure and trim panels. FE models are still too heavy in terms of time to set up models and compute acoustic TL. TL computation of panels under TBL excitation is not covered at this moment.

For acoustic propagation inside cavities, Statistical Energy Analysis (SEA) (Fig. 3 and ref [4, 9]) or ray-tracing methods [10] are classically used with a good level of accuracy and short computational time. Here, the main parameter to control is sound absorption inside the vehicle, which poses no specific difficulty.

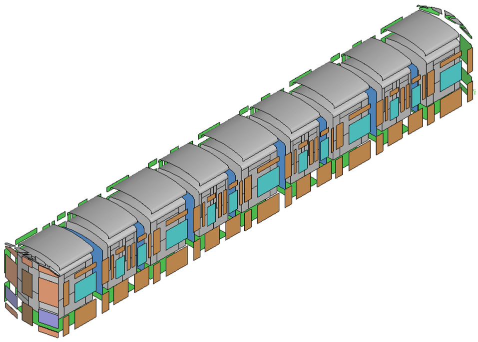

Fig. 3. Example of an SEA model for interior airborne noise prediction

The global accuracy of such models is quite good, with the capability to predict global interior noise levels with a maximum difference of 1 to 2 dB depending on the input parameters' quality. Post-processing of model results leads to a ranking of source and panel contributions to interior noise (Fig. 4). Even more important, the model is robust and fast for parametric studies: the effect of low noise solutions simulated with the model is usually in line with that obtained after implementation on the vehicle.

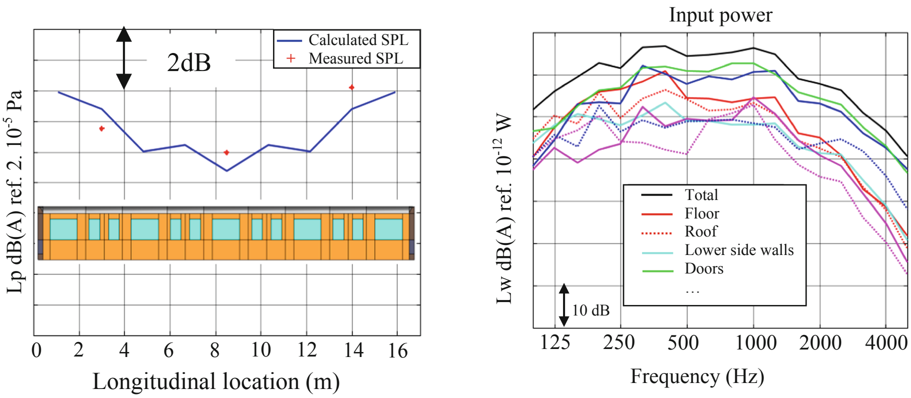

Fig. 4. Interior airborne noise prediction for Metro rolling stock. ***Left:*** comparison between measurements and computations ***Right:*** Panel contribution analysis

2.3 Structure Borne Noise Model

Structure borne noise is transmitted from the bogie to the car body through all the mechanical links, mainly dampers and traction bars. Secondary suspensions are usually a transmission path of lesser importance.

Structural Finite Element Models (FEM) are used by rolling stock manufacturers to control and optimize car body design. The quantities that are computed and compared with targets are the dynamic stiffness at input points (the real part of the mobility is sometime used), and the car body's vibroacoustic transfer functions P/F. Most of the time, simple models are used to take the acoustic cavity into account (hybrid FEM-SEA or analytical formula). Computations can then be done up to 1 kHz without any major difficulty, with a very good level of confidence for dynamic stiffness and an acceptable level of confidence for vibro-acoustic transfers. An example for a Light Rail Vehicle is given in Fig. 5. Full vibroacoustic models including a Finite Element model of the cavity can also be used to improve accuracy in the low frequency range (below 300 Hz) by taking into account the cavity's modal behavior, but this is rarely done in railway projects at this moment.

This type of computation is very useful to ensure a "healthy" car body Noise-Vibration-Harshness (NVH) design, with low vibroacoustic transfer functions. In a second step, these transfer functions must be combined with input dynamic forces at connecting points to predict the interior noise levels. The blocked force approach is more and more used, these blocked forces being usually obtained from in situ tests [1, 2] on vehicle or laboratory tests [12, 13] on components.

A virtual test method for structure borne noise generated from railway running gear is in progress to compute blocked forces transmitted to the car body starting with wheel/rail contact forces. A complete description of a method under assessment is given in [13]; only a brief summary is presented here. The Finite Element model includes the wheelsets, axle boxes and bogie (Fig. 6). Traction bars and dampers are represented by equivalent beam elements. All suspension elements and rubber bushings are modeled with frequency variable stiffness elements, the dynamic stiffness of these elements being

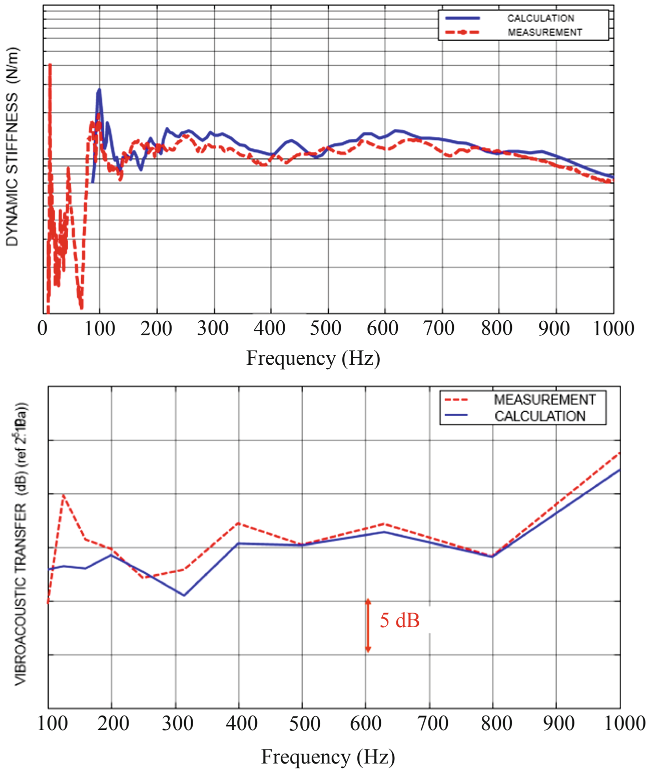

Fig. 5. Car body Finite Element Analysis computation results for structure borne noise control ***Upper:*** Input point dynamic stiffness ***Lower:*** P/F transfer function estimated with an analytical formula

previously measured in laboratory conditions. Wheel/rail contact forces are computed, taking into account the interaction with the track, wheel/rail roughness and contact filtering effect. They are then introduced into the FE model to compute blocked forces at the car body attachment points. In a final step, these blocked forces are combined with measured car body vibroacoustic transfers P/F to estimate structure borne interior noise.

3 Traction Noise

Different physical phenomena are at the origin of traction noise: electromagnetic excitation for electric motors, mechanical excitation due to the meshing process between pairs of teeth in a gearbox and aeroacoustics sources due to cooling unit fans. This section focuses on electric motor noise generated by electromagnetic excitation and gearbox noise generated by the meshing process.

3.1 Electric Motor Noise

Modelling Noise Due to Electromagnetic Excitation: The main excitation phenomenon is the Maxwell magnetic pressure acting in the air gap, on the stator and

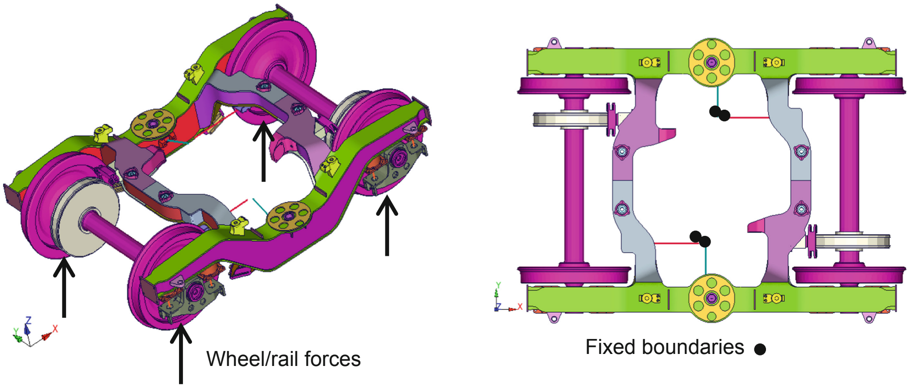

Fig. 6. The FE model of a complete bogie with dampers and traction bars, from ref. [13]

on the rotor. Many parameters influence the content of the excitation, such as the motor topology (induction or synchronous machine, number of slots and poles) or the motor drive (current shape including Pulse-With Modulation (PWM) effects).

Noise computation is a multiphysical problem for which it is necessary to combine electromagnetic, dynamic and acoustic Finite Element calculations (Fig. 7).

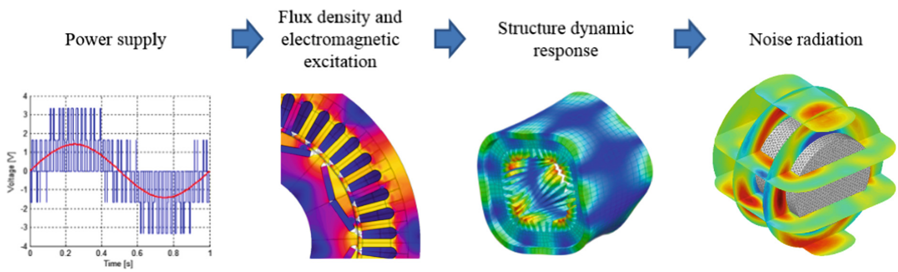

Fig. 7. Noise generation process for electric motor

This type of calculation has been presented by several authors e.g. [14], and has recently been validated against measurements in industrial applications [15]. A typical example of comparison between measurement and computation is presented Fig. 8: in this case the model gives representative order of magnitude of vibration levels, and is able to capture the dominant physical phenomenon (here the excitation of a stator mode by the electromagnetic excitation at a speed of about 4500 rpm).

Noise Minimization Strategies. Several low noise strategies are possible and can also be combined:

- Optimization of the structural design and/or of the PWM strategy, in order to avoid resonance phenomena, characterized by a spatial and frequency coupling between the electromagnetic excitation and the modal behavior of the stator [16].
- Optimization of the electromagnetic design in order to minimize Maxwell pressure inside the airgap of the machine. An example is given below.

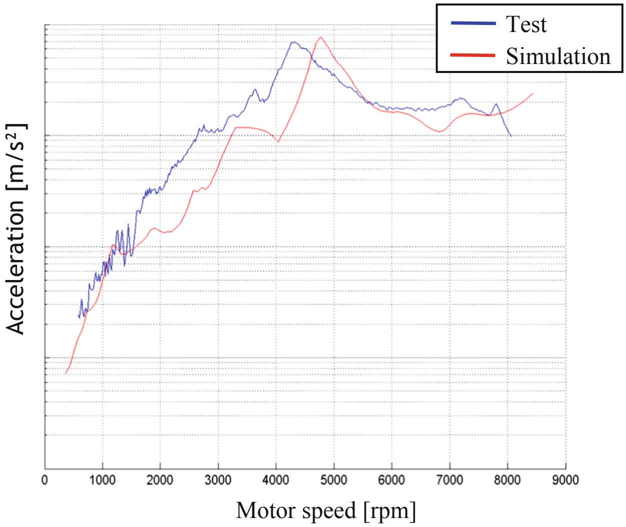

Fig. 8. Comparison of measured/computed normal mean squared acceleration of stator housing for the dominant engine order excitation during a run-up

Electromagnetic Design Optimization. Electric motor design optimization currently aims at minimizing a cost function corresponding to the acoustic power level of the machine, while respecting constraint functions so as not to deteriorate the motor's overall performance. In ref [17], the computational method presented in Fig. 7 is driven with an optimization software and different case studies are presented, including an Interior Permanent Magnet Synchronous Machine (IPMSM, Fig. 9).

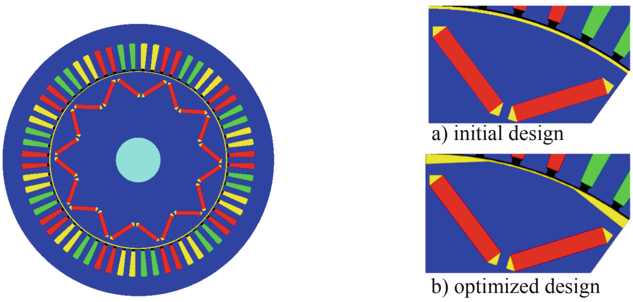

Fig. 9. Geometry of the initial and optimized Interior Permanent Magnet Synchronous Machine (IPMSM), from ref. [17]

The results after optimization show a very significant 14 dB decrease of the maximum Sound Power Level during an engine run-up (Fig. 10). Some inequality constraints are also defined, so that the mean torque produced by the motor is not reduced and torque ripple is not increased in comparison with the initial design. This reduction is achieved by small changes in the shape of the rotor poles depicted in Fig. 9 that do not affect the motor's price or weight.

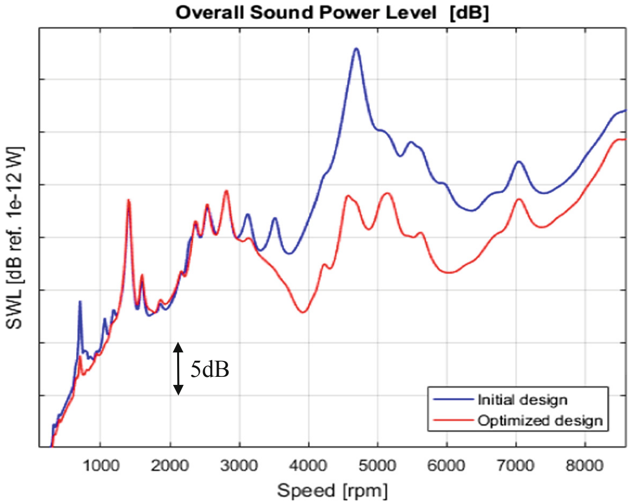

Fig. 10. Comparison of initial and optimized Interior Permanent Magnet Synchronous Machine (IPMSM) design SWL, from ref. [17]

3.2 Gearbox Noise

Gearbox Whining Noise Modelling: the gear meshing process is usually the main excitation source in gearboxes. Indeed, it is commonly assumed that Static Transmission Error (STE) and gear mesh stiffness fluctuations are responsible for radiated gearbox noise. They generate dynamic mesh forces which are transmitted to the housing through wheel bodies, shafts and bearings. Housing vibration is directly related to the noise radiated from the gearbox (whining noise).

The computational scheme is based on a 2-step procedure (Fig. 11):

- Step 1: computation of the excitation (transmission errors and mesh stiffness fluctuations), the input parameters being the teeth macro and microgeometries. Note that in certain situations, the static deflection of the shafts and housing can strongly influence STE computations and has to be taken into account. This can be done by means of static deflection computations including the complete model (gears, shafts and housing) to determine the equivalent helix deviation function of input torque: a unit torque is applied and therefore permit to determine the static deflection and the equivalent helix error as illustrated in Fig. 12.
- Step 2: computation of the vibration response of the gearbox, based on a Finite Element model of the gearbox and a modal approach. For gearbox whining, specific frequency algorithms (spectral iterative method) can be used to solve the dynamic equations with an iterative procedure [18]. These algorithms are much faster than time domain algorithms proposed in multibody software.

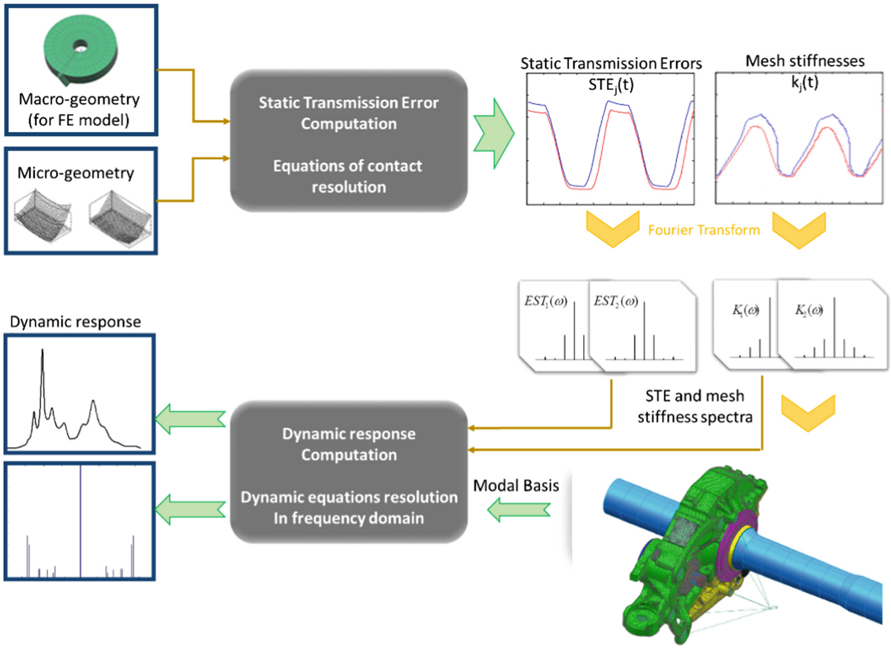

Fig. 11. Overview of the computational scheme

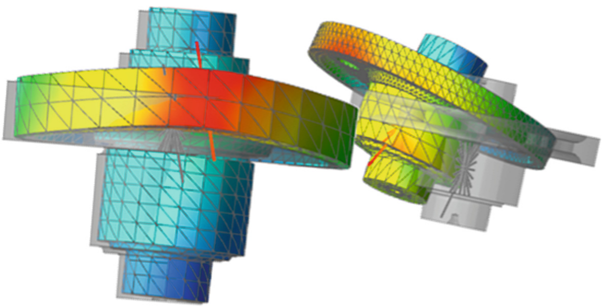

Fig. 12. Equivalent helix deviation due to static deflection generated by the transmitted torque

Ref [19, 20] respectively give examples of validation on an automotive and a railway application: a reasonable agreement between computations and measurements is achieved.

Figure 13 shows a comparison between measured and computed housing vibrations for a railway gearbox, under high torque condition. In this case, two STE computations are done, with and without the effect of static deflection: better vibration predictions are usually obtained when the effect of static deflection on STE is taken into account. Despite the uncertainty sources, a correct order of magnitude is provided for almost all measurement points on the housing.

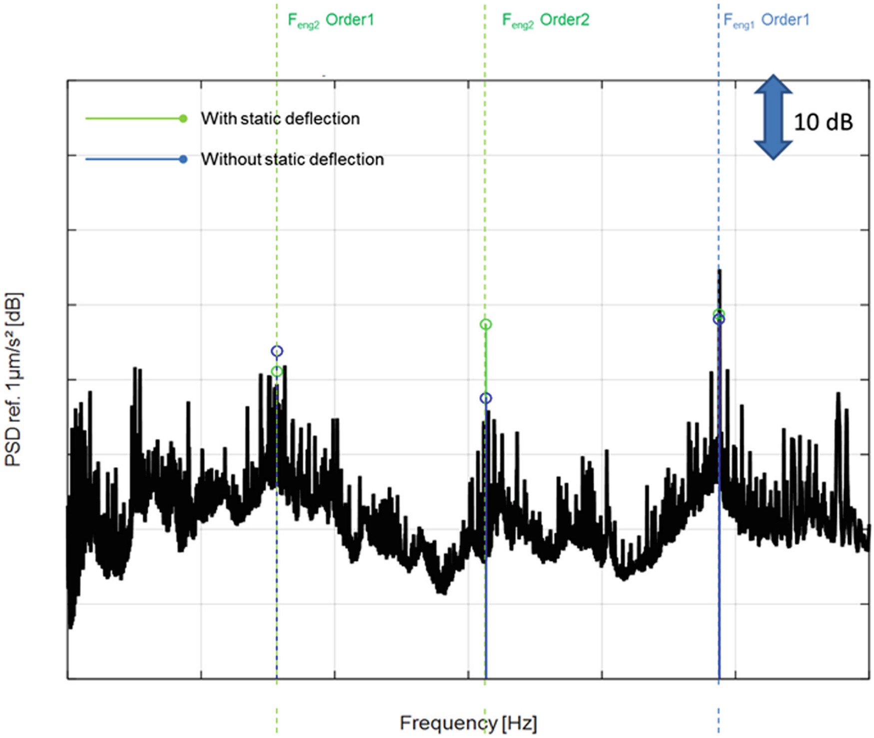

Fig. 13. Railway gearbox. Power spectral density of the housing normal mean squared acceleration (from ref. [12])

Noise Minimization: Gearbox noise can be minimized in different ways:

- Optimization of the housing's structural design, to avoid the excitation of local modes by meshing frequencies. This type of optimization is rather classical in NVH engineering and is not detailed here.
- Tooth microgeometry optimization in order to minimize STE. An example is given below.

Tooth microgeometry optimization in simple mesh gear systems for a given torque has been studied by many authors. However, for railway and automotive applications, this may not be sufficient as the torque transmitted by the gearbox is not constant. To overcome this limitation, an optimization strategy has been developed in order to minimize the STEs of multi-mesh gearboxes for a wide operating torque range [21].

An example of peak-to-peak STE values as a function of the torque is given in Fig. 14, before/after optimization. For the reference case, the STE was already minimized for medium and low torques (200–600 Nm). After optimization, different tooth microgeometries are proposed: for low and medium torque, they reach the same level of performance as the reference microgeometry, but they bring significant improvement for higher torque values (above 1000 Nm).

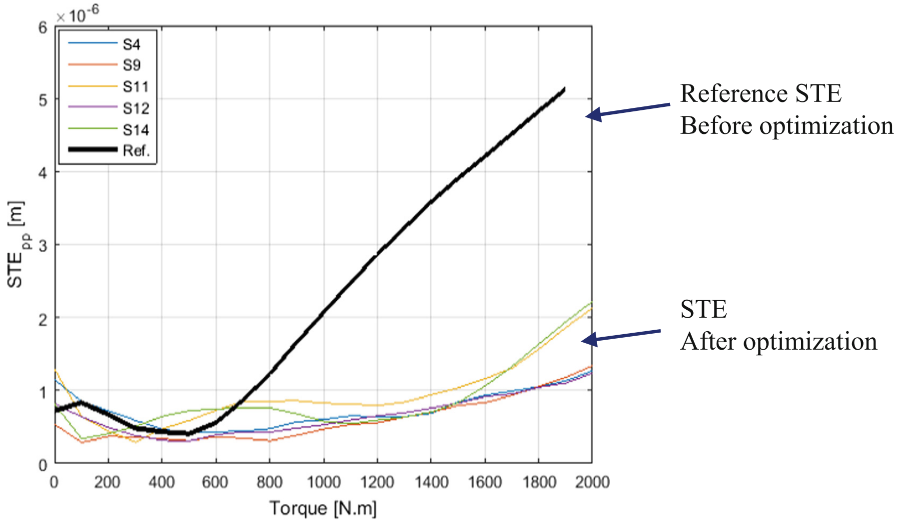

Fig. 14. Static transmission error computation (peak to peak values) function of torque before/after tooth microgeometry optimization.

A robustness analysis can then be carried out in order to select, among the different possible optimized microgeometries, the more robust with regard to manufacturing tolerances. This will ensure that noise reduction will still be obtained even if the selected optimized microgeometry will not be perfectly manufactured.

This robustness study is done with a Monte-Carlo method, i.e. 10000 STEs are computed, chosen randomly optimized teeth microgeometry parameters limited by the manufacturing tolerance intervals. This allows the establishment of the STE probability function of each possible optimized solution and the computation of statistical variables such as mean value and standard deviation of STE.

Figure 15 shows an example of STE probability density functions for the reference microgeometry and three possible optimized solutions. This illustrates how the final optimized solution is selected. The solution S1 has the smallest mean and maximum values. Although the mean value for the optimized solution S2 is also low, it is less interesting than the solution S1 because its STE maximum value is higher: the risk of deterioration of the noise because of manufacturing tolerances is higher. From a practical point of view, the mean value is the first criterion to consider. If the mean values are of the same order of magnitude (up to a difference of 15%), the solution with the smallest maximum value of the STE probability density function should be retained.

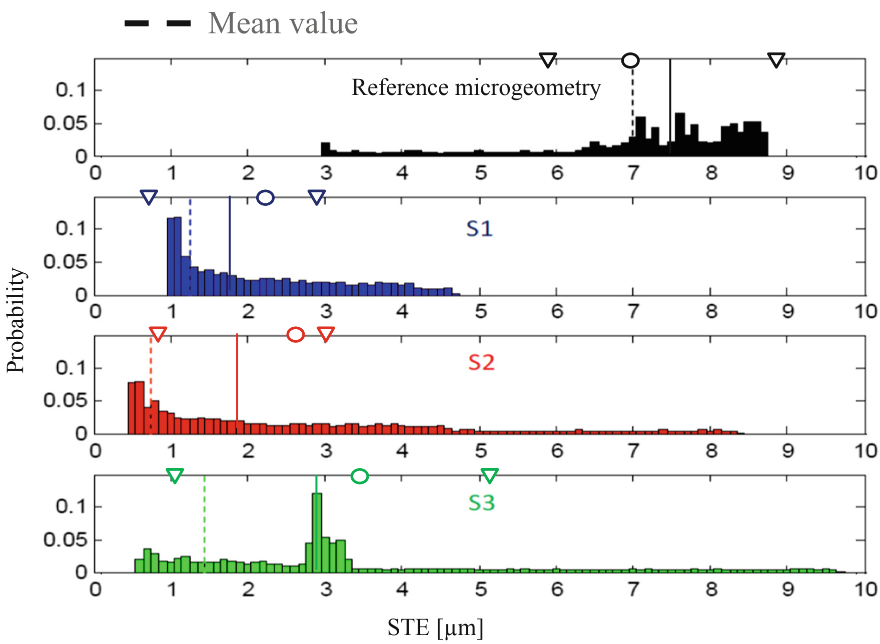

Fig. 15. STEs Probability density functions for the reference microgeometry and three optimized solutions S1, S2, S3 (from ref. [13])

4 Conclusions

Industrial methods used by rolling stock manufacturers to predict interior noise levels inside vehicles have been presented. For airborne transmission, global and pragmatic modelling strategies have been developed over the past 15 years, mixing experimental databases and acoustic computations with reasonable levels of confidence for most situations, while keeping low computational times compatible with railway projects. Some (non-exhaustive) fields of improvement have been mentioned in this paper.

Modelling structure borne noise transmission remains a challenge and a field of applied research for railway companies. Up to now, most of the work consists in controlling the car body design (dynamic stiffness and vibroacoutic transfer) independently from structural excitation coming from the bogie. Research on this topic is in progress, and validation and optimization of computational procedures are required.

Concerning traction motor noise, efficient computational methods are now available, with a sufficient level of accuracy to be used for industrial applications. The coupling of these methods with optimization software has a great potential for motor NVH design.

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