Acceptance test for damped railway wheels

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Abstract

In this paper a procedure for quality acceptance of low-noise railway wheels is presented. The low-noise feature is obtained by applying a constrained layer damping through the use of a viscoelastic polymer and a pre-formed constraining plate.

The proposed procedure is based on the analysis of the vibration amplitude in resonance conditions and does not use any acoustical measurement, whose use is particularly not advised in industrial workshops. Frequencies to be considered have been chosen with both the analysis of the sound power emitted by the wheel in a semi-anechoic chamber and the analysis of all acoustic frequency responses measured during type tests.

The methodology, verified on a reduced set of damped wheels, has shown a sufficient sensibility and proved to be potentially able to highlight the changes in the vibroacoustics characteristics of wheels due to the intrinsic repeatability of the application of the damping panel.

1 Introduction

The problem of noise has become central for railways, a transportation mode that is nevertheless characterized by the lowest environmental impact.

The excitation source for rolling noise, that is certainly the most important component for electric train in a particularly wide range of speed (approximately between 50 km/h and 250 km/h), is due to the combined wheel and rail roughnesses. This roughness is not simply the energy sum of wheel and rail roughnesses considered separately (and uncorrelated) but is modified by the *contact patch filter* as the contact area has finite dimensions that reduce the overall excitation at the higher frequencies (i.e. at the shorter wavelengths).

As pointed out at European level, the highest priority is currently the reduction of wheel roughness where it is dominating on rail roughness, i.e. for freight wagons with cast iron brake blocks. Researches and simulations lead to an estimation of a reduction of around 8÷10 dB of the night equivalent noise level by using sintered or synthetic blocks or by modifying the wagons to accommodate any type of disc braking.

Passenger rolling stock started their migration to disc braking in the ‘70s and it can be said that all vehicles material are now disc braked. Also motor units, as high power locomotives or Electrical Multiple Units, have special mechanical arrangements (brake discs mounted on the wheel web) that, although originally designed for space reasons and braking performances, have also limited to a minimum the wheel roughness.

As a result, for low-roughness disc braked wheels it is not straightforward to assign only to the wheel or only to the rail the origin of the noise pollution. Studies and models validated during the last 10÷15 years show that the relative importance of the wheel and of the rail can considerably change from case to case. As falling in the competence of a wheel manufacturer, Lucchini Sidermeccanica (formerly Lucchini CRS) started, in the mid '90s, a series of activities that allowed to develop some solutions for both the block braked wheels and the disc braked wheels sectors. For the latter, Lucchini Sidermeccanica has developed a treatment, named *Syope*, that is based on the constrained layer damping technique [1].

It is in fact clear that the noise generation is due to the wheel and rail surface vibrations that are excited by the extremely high contact stiffnesses also for wheel and rail roughnesses not visible to the naked eye. The *Syope* treatment aims at reducing wheel vibrations, and therefore the emitted noise, *under unchanghed excitation conditions*.

As shown in many circumstances, for example in the use on the Italian high speed train ETR500 [2, 3], on the narrow-gauge Circumvesuviana railway in Naples [4] or on the Ferrovia Merano-Malles [5], such treatment proved to be safe and durable, but also showed high acoustic performances with an almost negligible extra cost in the Total Life Cycle Cost of the vehicle and without any additional maintenance burden. Moreover, it can be applied to *any* wheel mounted on a wheelset with disc braking as it has been certified that the treatment has no structural influence.

After prototyping, the treatment has become a mass-produced item; it has been applied, for example, on Czech Republic Pendolino and will also be mounted on new Cisalpino and Trenitalia tilting trains currently under manufacturing. The *Syope* will also be mounted, as a retrofit, on *the whole fleet* of Circumvesuviana.

The application of the treatment to the wheel web is made by rolling a steel disc preliminarily covered with the adhesive damping polymer and pre-formed. Obviously the results of the this manufacturing process vary, as those of any other process, in a statistical way. It was therefore fundamental to guarantee the characteristics of the application (uniformity of the panel adhesion) and the corresponding acoustical properties (noise reduction) by the definition of an acceptance procedure with adequate simplicity, robustness and significance.

This paper shows the results obtained by the definition, the tuning and the application of the acceptance procedure of the *Syope* treatment to be applied in the mounting workshop (and not in the laboratory).

2 Preliminary considerations

Mechanical tests and acceptance criteria are normally defined during the commissioning phase of a wheelset. Obviously some tests are performed on samples (for example the analysis of the chemical composition of some heats) or on a particularly small set of samples in the case of complex, long or expensive analyses (for example, full scale fatigue tests).

In the vibroacoustics field, also considering the persistent absence of specific standards, univocal evaluation and acceptance procedures on treated (or untreated) wheel have not been defined yet.

For example, Lucchini Sidermeccanica installed in its laboratories a semi-anechoic chamber fulfilling the requirements of ISO 3744 standard, that nevertheless *is not required* by any standard on design, manufacturing or operation of railway wheels. Lucchini is probably, therefore, the only manufacturer that is able to make a set of experimental evaluations on the acoustic "efficacy" of a new design wheel.

As a standard, Lucchini measures, with an internal protocol, the sound power emitted by the wheel during an impact test with an instrumented hammer; such sound power spectrum allows to evaluate the vibration modes which are more responsible of the emitted noise in the lab. It is worth to highlight that rolling introduces a further and often prevailing source of damping, whose entity is not known to the wheel manufacturer; nevertheless, through this test it is possible to make sure that the treatments applied on the wheel lead to a sufficient damping, hopefully greater than the *rolling damping*.

The sound power test, to be performed on a single wheel and that therefore is a *type test*, is so long, complicated and needs a special environment that it is not even possible to think about its possible use in production to describe the variance of the behaviour obtained during the normal manufacturing of treated wheels. From this considerations arises the need to measure the real performance of a wheel *after* the application of the treatment with an *acceptance test* to be made on the entire batch or on a sufficient number of wheels defined by a sampling plan.

This acceptance test must fulfil the typical requisites of process tests: it should be relatively unaffected by external factors, it should be meaningful (observability criterion), it should be relatively easy to be used by personnel with low training, it should give simple and univocal results.

These requests led to discarding the option of acoustic tests in the workshop, even if they would have had the maximum meaning, as it is not easy to get there a sufficient level of cleanliness and low background noise to use measurement microphones. It was therefore necessary to make use of an *indirect* measurement of the acoustic behaviour by means of the measurements of the vibration behaviour, obviously ensuring the meaningful of the obtained results.

3 Measurements needed for the procedure

The optimal test that satisfies all the mentioned criteria is a *single* inertance (acceleration/force) frequency response function. This type of measurement is possible with sensors which are sufficiently rugged to be used in an industrial environment, is relatively insensible (with some precautions on boundary conditions on wheel constraints) to disturbances and is sufficiently simple to be adopted systematically on a great number of wheels. With the considerations described hereinafter, it proved to be a good candidate to replace similar acoustical tests.

The set up and tuning of the procedure needed numerous preliminary tests that are described here with reference to a wheel of diameter 890 mm with a symmetric (straight) web. The conclusions reached are therefore general but the results obtained here are clearly relevant only to this specific wheel.

The wheel, shown in Figure 1, is freely suspended with its axis vertical to get the usual free-free excitation and response conditions. It was observed that the highest rigid frequency of the wheel on the support is at approximately 50 Hz, well below the frequency range of emission of the wheel.

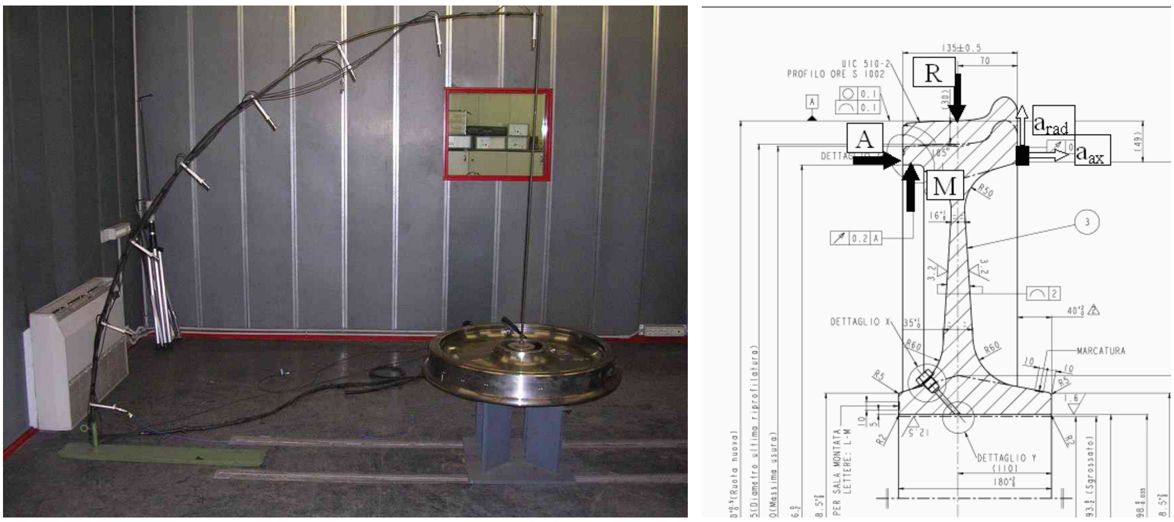

Figure 1: Sound power measurement setup in the semi-anechoic chamber (left). Excitation and response positions (right).

The wheel has been excited with an instrumented hammer varying the angular position between an arc of measurement microphones, evenly spaced on an angle of 90° and numbered starting from the one on the wheel axis, and the excitation section. Angular positions of 0°, 30°, 45°, 60°, 90° and 150° have been considered. This choice comes from a preliminary set of measurements specifically made to verify the excitability and acoustic response conditions of the wheel, including the possible symmetries and antisymmetries of the acoustic field and of the vibration response of the wheel.

Always with reference to Figure 1, several excitation conditions were considered, i.e. radial at the centre of the wheel tread (R), axial just under the maximum wear groove (A), and in the area used for turning on the lathe (M). Response were measured in terms of sound pressures in the aforementioned positions and in terms of accelerations only in radial and axial directions (a_{rad} and a_{ax} respectively), as the circumferential direction is not excited and that in the literature it is considered relatively unimportant for noise emission.

For limits on the available number of acquisition channels (eight), tests were made in two distinct phases:

a) excitation with instrumented hammer and response of the 7 measurement microphones. The 4 most important microphones were therefore selected;

b) excitation with the instrumented hammer and response of a triaxial accelerometer + the 4 microphones previously selected.

Obviously in case of a larger availability of transducers and channels it would be possible to make all the measurements simultaneously; in that case 20 channels (1 for the hammer, 7 for the microphones and 12 for the axial/radial accelerometers) would be needed. The set of FRF obtained in two times is nevertheless meaningful as they are normalized to the measured input.

4 Description and application of the procedure

The measurement procedure has been formalized in a document delivered to Lucchini Sidermeccanica including some short operational rules. Instead of including the synthetic protocol, it is preferred here to define the steps and to describe the results obtained on the practical case that helped to the tune the procedure.

4.1 Determination of the excitation direction

Although it is conceivable to use for the type test *all* the excitation directions previously mentioned, and even others at different angles due to the specific wheel geometry (chamfers, fillets, etc.), it is evident that the acceptance test will be possible only with a monoaxial excitation impulse.

In the case described here, the wheel section with the straight web resulted in higher importance of the excitations in axial (A) and turning (M) directions. In particular, the axial input position has been chosen below the maximum wear groove as this area is available also at the end of the useful wheel life, allowing further checks of the efficacy of the damping treatment.

Wheel tread geometry is such that both (A) and (M) inputs are able to excite flexural and torsional eigenmodes of the tread area, while (A) excites more global flexural eigenmodes affecting the whole tread / web / hub complex.

It was therefore decided to use excitation (A) as the *only* excitation to be used for all subsequent analyses. What said must be obviously revised case by case depending on the geometry and on the peculiarities in the vibroacoustic response of each specific wheel.

4.2 Collection of a sufficient number of vibroacoustic FRFs

Although it is possible, as aforementioned, to measure the entire set of frequency responses simultaneously, from a logical point of view it is preferable to analyse the microphone responses first.

The choice of the set of measurements is justified by considerations on processing and on observability (deviations) of the results. First of all, it is necessary to note (Figure 2) that acoustic FRFs are generally affected by a higher background noise level compared to the corresponding vibration FRFs. This is due both to a low noise emission of the wheel in some directions and to the position of the semi-anechoic chamber which is close to other manufacturing departments. This leads to the conclusion that trying to get a level of detail much higher than that offered by the chosen 7 microphones evenly spaced on 90° can be illusory.

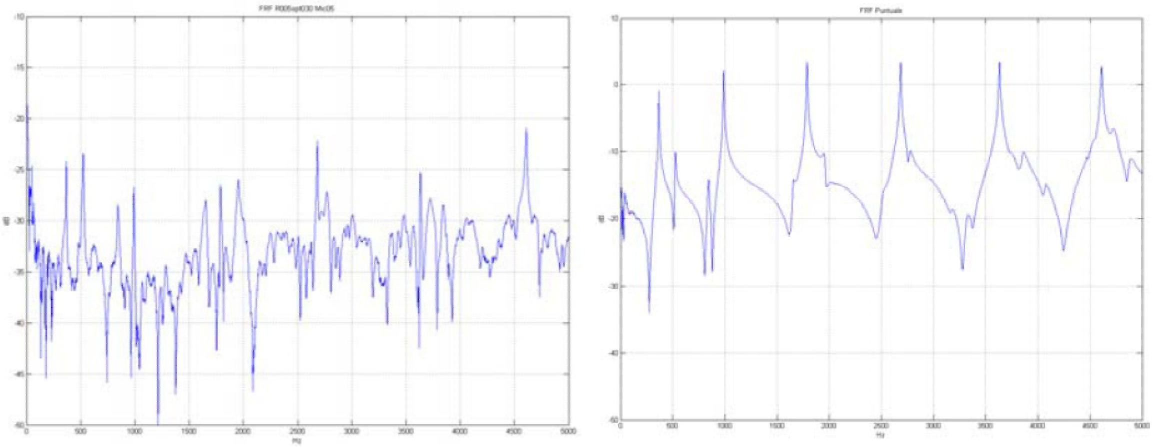

Figure 2: Left: microphone/hammer FRF for axial excitation with a relative angle of 0° (microphone #5, 60° wrt to the vertical axis). Right: point FRF for axial excitation

About the number of excitation positions on the circumference to be used for a complete characterization of the sound emission, in the case under study it was possible to make use of geometric symmetry of the wheel. During some preliminary analyses two symmetric positions (90° and 270°) were used; once the symmetry in the acoustic response of the wheel was verified, such duplication was removed introducing other previously non considered point.

4.3 Identification of most important frequencies and of the optimal response

The vibration response of the wheel is dominated, also in the case of relatively high damping, by its modal behaviour. The acoustic response is strictly derived from it but it is affected also by the fluid-structure coupling and by the following combination of the sound pressure fields originating from the different portions of the wheel. As an example, in the wheel under test the microphone placed on the wheel axis (mic #1) always showed levels much lower than the others and, on the basis of previously mentioned considerations, was excluded by the selected set of microphones for signal/noise reasons.

The frequencies of interest are clearly all those that can contribute in some way to find out possible non conformities in the application of the damping treatment. As it can be easily argued from Figure 2, the number of eigenmodes that can be extracted from vibration FRFs is very high (greater than 25) but some of them have levels that may be critical for their identification during an acceptance test; moreover, they risk to be linked to local conditions that are poorly representative of the behaviour in service of the wheel.

To identify the frequencies of interest, a combined approach based only on acoustic FRFs was used as follows:

a) use of the total normalized sound power (given by the sum of $|FRF|^2$ multiplied by the relevant areas), identifying the frequencies relative to the peaks in the sound power function that contribute to a relevant portion (90%÷95%) of the sound power normalized to the input;
b) use of the entire set of acoustic FRFs as follows:
   1. selection of the peaks in the FRFs that are due to eigenfrequencies and pick of the amplitude in resonance conditions. For high damping responses, it is advisable to select the frequencies by using an energy criterion including only those peaks that contribute to a relevant portion (90%÷95%) of the energy of the acoustic response for the case of a flat spectrum excitation;
   2. filling of a matrix containing the amplitude of the modes identified for each FRF;

3. reduction of the matrix discarding the values that appear only in some FRFs and vice versa retaining the frequencies that appear in *all* the FRFs. At the end of the reduction, the set of frequencies to be considered is identified.

The set of frequencies to be used in the remaining part of the procedure is obtained by considering the sets of frequencies found at step a) *and* b3). If the frequencies found at step b3) are not fully included in the frequencies at step a), reintegrate the reduced matrix found at step b3) inserting the corresponding frequencies.

The advantage of using a combination of methods a) *and* b) is that while with the first the global response is taken in consideration, thereby guaranteeing the maximum physical meaning, with the second it is possible to observe the behaviour at each frequency. It is in fact possible that some responses have peaks with high amplitude that cancel out for phase reasons in the global sound power, losing the corresponding frequency.

As an example, Figure 3 shows the normalized sound power for the wheel considered as an application of the procedure and an example of the identification of peaks in a single acoustic FRF. Part of the full matrix of amplitudes for each frequency and each input/output combination obtained by the automatic extraction process (95% of the energy of each acoustic FRF) is shown in Figure 4. It is evident that some frequencies are more representative in terms of appearance and in terms of relative amplitude of the peaks at those frequencies, this implying that the response is not dominated by a specific mode but that vice versa all the modes are almost equally represented in the response.

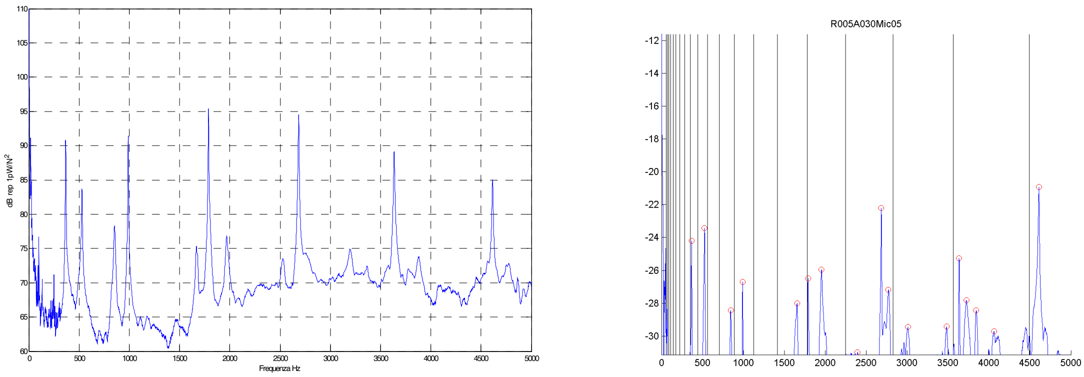

Figure 3: Left: normalized emitted sound power for axial excitation. Right: example of automatic identification of peaks on an acoustic FRF (axial excitation at 30°, mic #5) using the 95% energy criterion.

Frequenze Modali [Hz]	365	508	825	988	1658	1787	1956	2514	2591	2659	2684	2734	2949	3175	3271	3296	3366	3631	3729	3774	3825	3876	3953	4242	4493	4611	4691	4724		
	Ampiezze [dB]																												STD	N° Vuoti
R005A000Accx						0.0					3.6							3.9								4.9		0.2	2.3	23
R005A000Accy				-0.9		1.4					2.8						-5.5	3.9		-4.4			-5.2	-5.2		1.9		-4.9	3.8	18
R005A000Accz						0.8					3.2							3.3								2.3			1.1	24
R005A000Mic02	-22.3										-16.9							-16.7								-22.8			3.3	24
R005A000Mic03	-25.6	-26.4	-28.3	-28.8	-31.0	-30.4	-29.1	-29.1			-23.1		-30.9	-30.5	-31.0	-28.9	-28.9	-22.0		-29.3	-29.1	-29.1	-30.6			-21.0			3.0	8
R005A000Mic04	-30.1	-22.9	-28.5	-22.8	-27.1	-23.6	-26.1	-28.5	-27.8		-21.0		-30.0	-29.0	-29.5		-29.6	-20.7	-29.2		-30.9	-30.9				-27.4		-30.1	3.3	8
R005A000Mic05	-24.4	-24.2	-28.2	-25.2	-29.5	-23.7	-25.4	-29.9	-28.6		-20.3		-30.0	-29.8	-29.8		-29.7	-23.9	-27.9				-30.1			-20.9		-30.2	3.3	9
R005A030Accx				-0.2		2.2					4.9							3.3			0.9	0.9				5.6		4.3	2.1	20
R005A030Accy											5.3							2.8								2.6		0.6	1.9	24
R005A030Accz	-0.9			2.2		3.4					3.3							3.3								2.8			1.7	22
R005A030Mic02	-23.0	-27.6		-24.1	-27.0	-27.5	-25.4	-27.3			-16.9					-27.6	-27.6	-16.6							-27.2	-23.7	-27.4	-27.4	3.7	13
R005A030Mic03	-24.9	-25.6	-28.3	-25.2		-27.8	-28.9	-28.9			-20.7			-28.0	-28.0	-28.4	-28.4	-20.6		-29.2		-29.4		-29.0		-21.0		-28.2	3.0	10
R005A030Mic04	-28.1	-23.1	-26.8	-25.8	-26.3	-25.5	-25.3	-28.0	-28.0		-22.2		-30.6	-28.6	-28.6		-28.0	-20.1			-28.6								2.7	12
R005A030Mic05	-24.2	-23.4	-28.4	-26.7	-28.0	-26.5	-26.0	-31.0			-22.2	-27.2	-29.4				-29.4	-25.3	-27.8		-28.4	-28.4	-29.7	-29.7		-20.9			2.7	9
R005A045Accx				-1.1		3.7	0.5				2.2							2.6			-1.5	-1.5				5.8		1.6	2.5	19
R005A045Accy				-1.5		5.1	0.2				-0.6							0.0			-3.2	-3.2				3.8		-0.4	2.8	19
R005A045Accz	-1.4	-10.1		0.9		1.1					-1.8							-5.6								-6.2			4.1	21
R005A045Mic02	-23.4			-24.2	-26.7	-25.4	-25.8				-17.1							-16.5								-23.2			3.9	20
R005A045Mic03	-25.2	-25.7	-28.5	-24.7		-26.9	-28.3	-28.3			-19.9		-28.5	-29.7	-30.1			-20.7	-27.3						-27.6	-21.0		-29.3	3.3	12
R005A045Mic04	-27.6	-23.3	-27.6	-26.2	-26.4	-23.9	-26.1	-28.4	-28.2		-23.2		-29.5				-29.4	-20.5	-28.1		-29.9		-29.6			-25.5	-30.6		2.8	10
R005A045Mic05	-24.0	-24.0	-27.7	-25.6	-28.0	-27.4	-26.2	-30.2			-23.3	-27.0							-26.5		-29.4		-29.0	-29.6		-22.8		-30.1	2.5	12
R005A060Accx						-1.7	-0.7				-0.4							1.9							-4.0	0.4		0.9	1.9	21
R005A060Accy				-0.8		1.4	-3.8				3.5							3.2								-0.7		-0.4	2.6	21
R005A060Accz						0.2					2.7							3.3								2.2			1.3	24
R005A060Mic02	-23.5			-24.7	-26.9	-26.7	-25.1				-17.7						-27.3	-16.7		-27.0					-27.8	-22.4			3.8	17
R005A060Mic03	-25.0	-25.4	-28.5	-25.5		-25.2	-28.4	-28.4			-20.2		-29.0					-21.4						-29.1	-27.5	-20.2		-28.0	3.2	14
R005A060Mic04	-27.2	-23.4	-27.6	-28.3	-26.0	-27.5	-26.3	-28.5	-28.5		-25.7		-29.0				-28.0	-20.5			-30.4		-30.1			-25.6	-29.1	-30.0	2.5	10
R005A060Mic05	-23.8	-23.9	-27.7	-25.9	-28.0	-27.1	-25.1				-24.2	-27.5					-29.5	-27.8	-27.6	-28.0	-29.7		-30.4	-30.0		-21.8		-29.8	2.5	10
R005A090Accx				-0.3		-0.8					1.3							5.2			0.9	0.9				2.0		2.1	1.8	20
R005A090Accy						2.5					7.5																		3.5	26
R005A090Accz				2.1		0.6					0.5							3.3								-0.5			1.5	23
R005A090Mic02	-22.4			-26.6	-26.7	-27.2	-25.1	-27.9			-18.4							-17.0							-27.7	-23.9		-27.8	3.8	17
R005A090Mic03	-24.8	-25.4	-28.4	-25.6	-30.6	-28.1	-29.4	-29.4	-28.8		-22.3		-29.8	-28.6		-28.5	-28.5	-22.1		-29.2	-29.1	-29.1	-30.3	-30.1	-28.3	-20.1		-28.4	2.8	5
R005A090Mic04	-28.0	-23.0	-28.9	-24.7	-26.6	-25.6	-26.6	-27.2	-27.0		-23.2		-29.3	-28.3		-28.8	-27.4	-20.8	-29.8							-25.0		-29.1	2.5	10
R005A090Mic05	-24.6	-24.0	-29.4	-24.5	-29.3	-24.7	-25.0	-29.5	-29.2		-25.2	-27.8	-29.1		-30.8		-30.4	-27.8	-28.8	-29.4	-30.3			-30.4		-23.2		-29.0	2.5	7
R005A270Accx				-0.2		-0.4					-2.3	-3.3						3.2			-0.7	-0.7			-3.1	-0.1		1.6	2.0	18
R005A270Accy						1.9					4.5							4.1			-1.3	-1.3				0.6		-2.6	2.8	21
R005A270Accz				[illegible]		[illegible]					[illegible]							[illegible]								[illegible]			[illegible]	[illegible]

Figure 4: Matrix of the amplitude at each resonance frequency in the full set of acoustic FRFs. In the rightmost columns the standard deviation (of data expressed in dB) and the number of empty spaces ("vuoti") are shown.

The full matrix is reduced by using criteria of numerousness of the frequencies and uniformity of amplitudes leading to the final matrix shown in Figure 5.

Frequenze Modali [Hz]	365	508	988	1787	2684	3631	4611			
	Ampiezze [dB]							STD	N° Vuoti	Media
R005A000Accx				0.0	3.6	3.9	4.9	2.2	3	3.1
R005A000Accy			-0.9	1.4	2.8	3.9	1.9	1.8	2	1.8
R005A000Accz				0.8	3.2	3.3	2.3	1.1	3	2.4
R005A000Mic02	-22.3				-16.9	-16.7	-22.8	3.3	3	-19.7
R005A000Mic03	-20.8	-25.7	-17.0	-21.2	-23.1	-22.0	-21.0	2.6	0	-21.6
R005A000Mic04	-30.1	-22.9	-22.8	-23.6	-21.0	-20.7	-27.4	3.4	0	-24.1
R005A000Mic05	-24.4	-24.2	-25.2	-23.7	-20.3	-23.9	-20.9	1.9	0	-23.2
R005A030Accx			-0.2	2.2	4.9	3.3	5.6	2.3	2	3.2
R005A030Accy					5.3	2.8	2.6	1.5	4	3.6
R005A030Accz	-0.9		2.2	3.4	3.3	3.3	2.8	1.7	1	2.4
R005A030Mic02	-23.0	-27.6	-24.1	-27.5	-16.9	-16.6	-23.7	4.5	0	-22.8
R005A030Mic03	-24.9	-25.6	-25.2	-27.8	-20.7	-20.6	-21.0	2.9	0	-23.7
R005A030Mic04	-28.1	-23.1	-25.8	-25.5	-22.2	-20.1	-24.4	2.6	0	-24.2
R005A030Mic05	-24.2	-23.4	-26.7	-26.5	-22.2	-25.3	-20.9	2.2	0	-24.2
R005A045Accx			-1.1	3.7	2.2	2.6	5.8	2.5	2	2.6
R005A045Accy			-1.5	5.1	-0.6	0.0	3.8	2.9	2	1.4
R005A045Accz	-1.4	-10.1	0.9	1.1	-1.8	-5.6	-6.2	4.1	0	-3.3
R005A045Mic02	-23.4		-24.2	-25.4	-17.1	-16.5	-23.2	3.8	1	-21.7
R005A045Mic03	-25.2	-25.7	-17.8	-21.0	-19.9	-20.7	-21.0	2.8	0	-21.6
R005A045Mic04	-27.6	-23.3	-26.2	-23.9	-23.2	-20.5	-25.5	2.3	0	-24.3
R005A045Mic05	-24.0	-24.0	-25.6	-27.4	-23.3	-28.2	-22.8	2.1	0	-25.0
R005A060Accx				-1.7	-0.4	1.9	0.4	1.5	3	0.0
R005A060Accy			-0.8	1.4	3.5	3.2	-0.7	2.1	2	1.3
R005A060Accz				0.2	2.7	3.3	2.2	1.3	3	2.1
R005A060Mic02	-23.5		-24.7	-26.7	-17.7	-16.7	-22.4	3.9	1	-22.0
R005A060Mic03	-25.0	-25.4	-25.5	-25.2	-20.2	-21.4	-20.2	2.5	0	-23.3
R005A060Mic04	-27.2	-23.4	-28.3	-27.5	-25.7	-20.5	-25.6	2.7	0	-25.4
R005A060Mic05	-23.8	-23.9	-25.9	-27.1	-24.2	-27.8	-21.8	2.1	0	-24.9
R005A090Accx			-0.3	-0.8	1.3	5.2	2.0	2.4	2	1.5
R005A090Accy				2.5	7.5			3.5	5	5.0
R005A090Accz			2.1	0.6	0.5	3.3	-0.5	1.5	2	1.2
R005A090Mic02	-22.4		-26.6	-27.2	-18.4	-17.0	-23.9	4.2	1	-22.6
R005A090Mic03	-24.8	-25.4	-25.6	-28.1	-22.3	-22.1	-20.1	2.7	0	-24.0
R005A090Mic04	-28.0	-23.0	-24.7	-25.6	-23.2	-20.8	-25.0	2.3	0	-24.3
R005A090Mic05	-24.6	-24.0	-24.5	-24.7	-25.2	-27.8	-23.2	1.5	0	-24.9
R005A270Accx			-0.2	-0.4	-2.3	3.2	-0.1	2.0	2	0.0
R005A270Accy				1.9	4.5	4.1	0.6	1.8	3	2.8
R005A270Accz			2.1	0.8	0.2	3.1	0.1	1.1	2	1.2

Figure 5: Reduced matrix of amplitudes at each resonance frequency in the full set of acoustic FRFs. In the rightmost column the average amplitude is shown.

The acoustic FRF that contains all the frequencies identified as important, that has the maximum mean value and the minimum dispersion of amplitude values is defined as "optimal" and represents the sound pressure / force transfer function that can be retained as most representative of the vibroacoustic behaviour of the wheel. In the case of the wheel under test, the optimal FRF is the one shown in Figure 3, where almost all the peaks are present in a particularly limited amplitude range.

Although this result is not directly implemented in the acceptance test protocol, it was fundamental to choose the frequencies of interest and it can help if it is desired to check with a single acoustic measurement the *overall* efficacy of the treatment. Obviously it requires a specific environment, not compatible with manufacturing departments, and the criterion based exclusively on vibration FRF described in the following paragraph was preferred. Nevertheless, the position where the accelerometer was placed during vibration tests corresponds to that of "optimal" excitation. Using in fact the point FRF and the related transfer functions, it is possible to estimate the contribution of any input to the "optimal" response.

As a confirmation of the validity of the proposed approach, the application of the criterion of the normalized sound power led to the identification of 5 important frequencies, while the extension to the analysis of all FRFs led to the reintroduction of two other important frequencies (at 508 Hz and 4611 Hz respectively) that were not evident in the normalized sound power.

4.4 Identification of the equivalent vibration test

Once the accelerometer is placed in the point described above, it is possible to collect the vibration FRFs exciting the wheel in the points used for the vibroacoustic characterization obtaining a set of amplitudes in resonance conditions at the identified set of important frequencies. For the wheel under test the experimental FRFs are shown in Figure 6, where two undamped wheels, five damped wheels and one wheel with only one panel are compared.

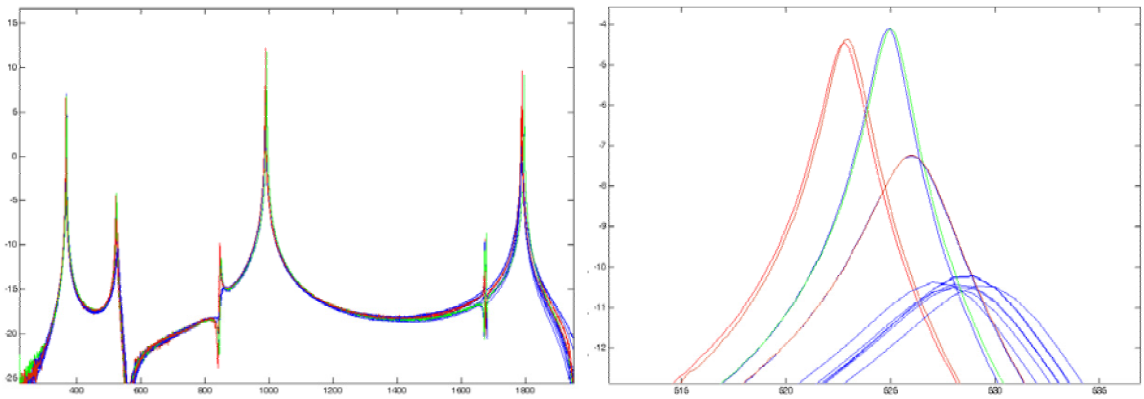

Figure 6: Vibration FRFs for the wheel under study (left) and close-up around the peak at 525 Hz (right). Upper curves are relative to two undamped wheels from different batches; mid curves are relative to the wheel with only one panel; lower curves are relative to five damped wheels. Some measurements are duplicated to check repeatability.

The amplitudes in resonance conditions can be plotted as in Figure 7. Amplitudes are particularly repeatable for point FRF (30° angle with the arc of microphones), except for an untreated wheel that had some peculiarities that influenced high frequency results.

With an angle of 15° between the hammer and the accelerometer great dispersions are observed at 2684 Hz (Figure 8), while with an angle of 30° the dispersions are at 988 Hz (Figure 9). These dispersions can be easily justified by observing the eigenmodes at the corresponding frequencies that suggest, for future tests, to avoid to put the accelerometers at angular positions that are an integer fraction of the circumference.

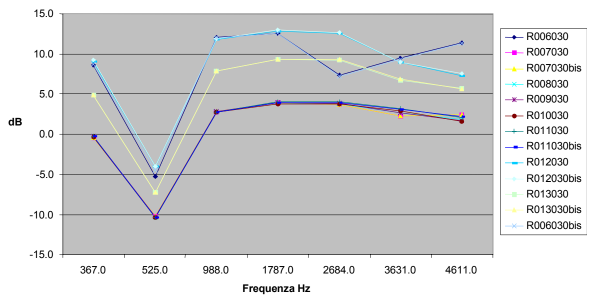

Figure 7: peaks amplitude of point FRFs for untreated wheels, one panel, two panels (in descending order)

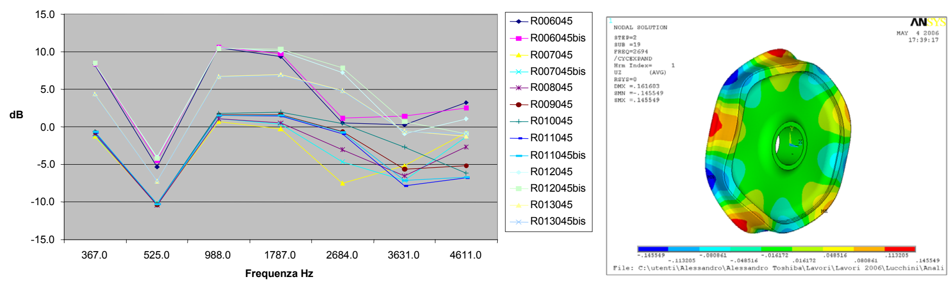

Figure 8: peaks amplitude of FRFs with an angle accelerometer / hammer of 15° (left). Eigenmode calculated with undamped FEA at 2694 Hz.

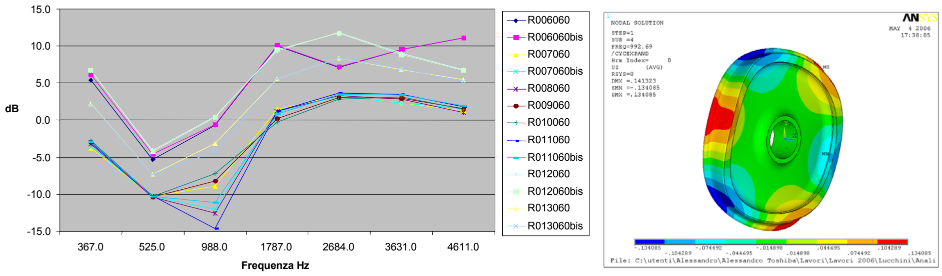

Figure 9: peaks amplitude of FRFs with an angle accelerometer / hammer of 30° (left). Eigenmode calculated with undamped FEA at 992 Hz.

5 Conclusions and further developments

A methodology to perform vibration measurements aimed at the acceptance test of damped wheel was developed and presented. It was proved that, suitably using type tests conducted in a semi-anechoic chamber, *it is not strictly necessary* to make acoustic tests but that a simple vibration test can give all the information needed to assess the quality of the application of the damping treatment.

It is not possible at the moment to define completely the acceptance criteria that shall be evaluated on the basis of the following considerations:

1) amplitudes in resonance conditions *should* vary slightly from wheel to wheel, at least at the final set of frequencies selected;
2) it could be interesting to verify the behaviour at some eigenfrequencies of the untreated wheel that are completely missing in the damped wheel and that could be indicative of the quality of the application of the treatment;
3) a specific weighting function will probably need to be defined that takes into account the difference to the average response *and* the average values (possibly in an inversely proportional way);
4) both the average values and the acceptability levels shall be defined *dynamically* and shall be updated on the basis of historical data.

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