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Deliverable 2.6

Proposed acceptance procedures for different wheel and track types

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EXECUTIVE SUMMARY

This deliverable details procedures for the prediction of rolling noise for different wheel types using TWINS. Wheel types covered are

- axi-symmetric monobloc wheels
- (weakly) non axi-symmetric wheels,
- wheels with braking discs,
- resilient wheels, and
- wheels with dampers.

TWINS predictions allow absolute levels of sound power emitted from wheels and track to be determined, as well as corresponding track-side sound pressure levels. However, these levels depend highly on input parameters that are independent of the rolling stock, such as rail roughness and track decay rates. 'Wheel acceptance procedures' are therefore dependent on the context of each virtual testing scenario. For this reason, the current deliverable also outlines several virtual testing scenarios in addition to the purely technical procedures mentioned above.

Throughout this deliverable, several open questions are highlighted that need deeper investigations. Some of these will be performed within work package 5 permitting to determine the final recommendations from the ACOUTRAIN project.

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1. INTRODUCTION

It is well known that rolling noise is generally the predominant noise source for a broad range of vehicle speeds. Its correct prediction is therefore crucial for any virtual homologation process in regard to pass-by levels. Rolling noise depends on both wheel and track design, therefore the track is mentioned in the title of the present deliverable as well. Indeed, the track will be considered here in respect to track parameters to be used for computations. The acceptance procedures dealt with, however, concentrate on the rolling stock [N.B.: At present, no acoustical homologation of tracks is foreseen in the relevant legislation].

“Wheel acceptance” is a part of a virtual homologation procedure. It can be based on absolute predictions or on a relative assessment using a reference wheel. Often, “wheel acceptance” will be directly equivalent to ensuring compliance with pass-by level limits. However, in certain cases the contribution of traction noise may need to be considered. There are indeed different possible scenarios of wheel acceptance in the context of a virtual homologation procedure. Several of such scenarios are discussed in section 2.

The virtual testing scenarios mentioned are useful in order to understand the context of wheel acceptance procedures. The main scope of this deliverable, however, is on the technical details of acceptance procedures. These are detailed in section 3 for different wheel types, including the discussion of measurements that have been performed on a wheelset with braking discs.

Related documents:

The current **Technical Specifications for Interoperability (TSI)** (partial revision of 2011) (1) is the reference for (classical) homologation procedures and sets the limit levels that apply.

The simplest virtual homologation procedure in respect to rolling noise is a relative assessment of a new wheel with respect to a reference wheel. This case is included here; however, it has already been described in **ACOUTRAIN deliverable D1.1: Clarification of the simplified method in the partial revision of the TSI** (2).

The simplified procedure described in the TSI also refers to **EN standard 13979-1:2003** (3) that contains a technical approval procedure for monobloc wheels (both mechanical and acoustical). It includes a description of the computational approach by the use of TWINS that covers even non axi-symmetric wheels and wheels equipped with dampers. The scope of EN 13979 is in principle not limited to distinct rolling stock; however, the current TSI only refers to EN 13979 in respect to the simplified evaluation of freight wagon wheels. Note that this deliverable contains recommendations that do not entirely comply with EN 13979.

ACOUTRAIN deliverable D1.2 (4) contains “First recommendations for a future certification process”. This document defines a framework for virtual testing and introduces a nomenclature for virtual testing scenarios which is used in the current deliverable.

ACOUTRAIN deliverable D2.1 (5) is a “User guide describing procedures for modelling wheels and tracks”. It covers only axi-symmetric monobloc wheels without damping devices and serves as a basis for the current deliverable in terms of general calculation settings (especially concerning the track).

2. VIRTUAL TESTING SCENARIOS

The scenarios described here are split into the main categories "Full Virtual Testing" (Full VT), "Hybrid Virtual testing" (Hybrid VT), and "Extension of Approval" (EoA), as introduced in deliverable D1.2 (4). The aim of the present deliverable is to propose acceptance procedures covering the widest possible range of scenarios. As a consequence, Full VT is also discussed here although it might not be proposed by Acoutrain in the near future.

All approaches proposed in the present deliverable will be subject to validation which is performed within WP 5. Several open questions raised in this report will be dealt with in WP 1 and WP 5 again in order to determine the final procedures to be proposed (deliverable 1.8).

2.1 ABSOLUTE PREDICTIONS FOR ROLLING STOCK WITH WHEEL/RAIL NOISE AS THE ONLY SOURCE (FULL VIRTUAL TESTING)

Two major parameters, both independent of the rolling stock, largely determine the result of absolute predictions: track decay rates (TDR) and rail roughness.

Three different choices seem possible with respect to TDRs to be used for computation:

1. The use of the TSI limit (minimum) TDR,
2. the use of a new "reference TDR" (representing a typical TSI compliant track), or
3. the use of a measurement performed on a particular TSI compliant track.

The direct use of the 'TSI decay rate (limit)' may lead to an over-estimation compared to real TSI test tracks ('worst case'). However, it seems difficult to obtain a consensus about the use of a different (more favourable) track decay rate. Clearly, this is an open question.

A very similar problem arises in relation to rail and wheel roughness. Possible scenarios are:

1. The use of the TSI limit roughness as total roughness,
2. the use of TSI limit roughness + typical wheel roughness (from database),
3. the use of TSI limit roughness + measured wheel roughness,
4. the use of a new reference rail roughness + database wheel roughness,
5. the use of a new reference rail roughness + measured wheel roughness, or
6. the use of measured rail and wheel roughness.

As for the TDR, use of the limit curve for rail roughness may lead to an over-estimation compared to real TSI tracks. Solutions that require the measurement of wheel roughness appear unrealistic in most cases because these require a set of run-in wheels. The most advantageous approach appears the use of a database containing typical roughness spectra for different wheel and brake types. However, the construction of an agreed database with reliable and representative roughness spectra is a difficult mission, especially if different brake types should be treated in a satisfactory way.

As for track decay rates, the definition of roughness spectra is an open question.

2.2 HYBRID VIRTUAL TESTING

The term “Hybrid Virtual Testing” is used here for procedures that use directly measured sound power or sound pressure levels for certain noise sources, while predicting other noise sources through computations. Although measured input data is used for most of the rolling noise computations as well, these are as such not qualified as hybrid methods here.

2.2.1 Measurement of standstill noise and simulation of pass-by without contribution of sources other than those that can be measured at standstill

This scenario applies to rolling stock for which sources other than rolling noise can be neglected at pass-by or can be assessed at standstill. [N.B.: If no other noise sources are present at all, the scenario described in section 2.1 applies.]

Typically this will be the case for trailer vehicles (passenger or freight) that contain auxiliary systems such as HVAC that only produce noticeable emissions at standstill. Pass-by and standstill cases can then be separated and assessed independently using different approaches (i.e. simulation for pass-by and measurement for standstill). Most often, the measured standstill noise (L_{pAeq},T = 20 s) will remain well below the predicted rolling noise (L_{pAeq}, T = duration of pass-by). For this case a possible approach could be to detemine the energetic sum of both to be used as an estimation for pass-by noise. Note that auxiliary sources must be operated as during rolling.

If the contribution of auxiliary sources to pass-by levels is negligible, the rolling noise computation can in principle also be performed as a relative assessment with respect to a reference wheel. However, strictly speaking this check already requires an absolute prediction of rolling noise levels. For coaches and EMUs the difference between standstill and pass-by noise limits seem high enough (15 and 13 dB(A)) to guarantee that auxiliary sources have no influence on pass-by noise. However, it is an open question whether the contribution of these sources to pass-by noise may be neglected *a priori* for certain types of rolling stock.

For pass-by at maximum speed, rolling noise will most often remain dominant for powered vehicles as well. For pass-by at 80 km/h, however, traction noise may be significant in certain cases, and the scenario described in section 2.2.2 applies. For powered vehicles, traction noise should therefore always be estimated by means of laboratory measurements and/or simulations in order to determine whether it has to be taken into account for pass-by simulations.

2.2.2 Measurement of standstill noise and simulation of pass-by with contribution of sources other than wheel-rail noise to pass-by noise

This scenario applies to powered vehicles and most importantly to locomotives and DMUs.

The pass-by noise simulation has to be an absolute prediction because of the presence of other noise sources. The use of the ACOUTRAIN tool (or an equivalent tool such as VAMPPASS, BRAINS, SITARE) is necessary in order to combine the different noise sources.

The assessment of traction noise in a laboratory is not trivial because engine test benches do not generally fulfil the requirements of an acoustic lab. The separation of rolling and traction noise

can alternatively be made by performing successive pass-by measurements with the traction system on and off (if possible). This can of course be done on a non TSI-compliant track.
A more detailed description of this issue is beyond the scope of WP 2 and will be dealt with in WP 3.

2.2.3 Transposition procedures

One typical transposition case is the estimation of pass-by levels that would be obtained on a TSI compliant track on the basis of pass-by measurements on a non TSI-compliant track. Such procedures are described in ACOUTRAIN deliverable D2.5 (TNO) and will not be detailed here.

Transposition between different vehicle configurations would be another case; however, such cases are detailed in section 2.3 (Extension of Approval) as far as rolling noise is considered.

2.3 Extension of Approval

2.3.1 Relative assessment between a new wheel and an approved reference wheel

This type of virtual testing is already accepted today and described in EN 13979-1 (3) for standard axisymmetric monobloc wheels. For other wheel types this standard mentions the possibility of using an experimental modal analysis (informative annex E) for updating of the wheel model, however, clear procedures are missing.

Section 3 of this deliverable contains such procedures for different wheel types and gives information of how to determine whether the computational approach is acceptable for a given wheel.

In contrast to EN 13979, powered wheels are equally considered here.

In the following, several recommendations for the relative assessment between two wheels are given. The authors are aware of the fact that these partly disagree with EN 13979.

The wheel component of sound power should be used as criterion whenever possible. Additionally the use of total sound power is needed for wheels that can significantly modify the track response (resilient wheels). [N.B.: EN 13979 proposes the use of (total) SPL at 3 m from the track, but this is less sensitive to a change in wheel radiated power and strongly depends on the track used. Moreover, SPL is disadvantageous as a criterion as it calls for the additional computation of sound propagation which is another source of uncertainty.]

A new wheel should be accepted if its A-weighted sound power is below the A-weighted sound power of the reference wheel. For resilient wheels, the same must apply for total A-weighted power. EN 13979 also demands that the new wheel should be less noisy in every one-third octave band. This is critical because the spectral content can differ significantly if highly contributing modes pass from one frequency band to another. We propose to remove or soften this spectral criterion.

If wheel sound power is used as the acceptance criterion, the analysis is largely independent of the track behaviour. Roughness, however, has a certain influence on the result because it introduces a weighting of frequency bands. For simplicity, the use of the TSI roughness limit and the 'Remington' contact filter is recommended.

2.3.2 Absolute assessment of a new wheel, based on a virtual reference vehicle validated experimentally

This scenario requires a virtual reference vehicle that has been validated with respect to pass-by measurements on a TSI-compliant track. As rolling noise is highly dependent on track properties, this also means that a TWINS model has been built on the basis of input data measured on the real test track. Measured rail and wheel roughness should be used. The wheel model can then be modified to the new design and the TWINS model be used for absolute predictions in order to show the compliance with TSI noise limits.

This procedure, of course, requires the reference vehicle to be validated against measurements. The maximum admissible deviation between reference measurement and reference simulation remains to be defined. Especially, the measured level must not be under-predicted but the model should rather represent a worst case. Another open question is how to deal with the uncertainty that can be ascribed to the calculated output. This question will be addressed in the validation phase of Acoutrain.

Example: An EMU equipped with wheel dampers has run through the classical homologation procedure and has been shown to remain significantly below TSI limits. A relative assessment ("simplified procedure") has shown that a geometrically optimised wheel without dampers produces a wheel sound power slightly above the reference wheel; therefore the new wheel is not admissible on the basis of this simplified procedure. An absolute assessment, however, can possibly prove the compliance with TSI limits for the EMU without wheel dampers.

3. ACOUSTIC CALCULATION PROCEDURES

This section contains technical details on how to compute sound power and sound pressure levels radiated by wheel and rail. Recommendations on general calculation settings are given in section 3.1. Modelling of the rail does not depend on the wheel and is dealt with in section 3.2 for all wheel types. The subsequent sections are dedicated to the modelling of different wheel types. All calculation procedures considered are based on TWINS (6). The recommendations given in the present deliverable largely base on ACOUTRAIN deliverable D2.1 (5): “User guide describing procedures for modelling wheels and tracks”. However, only axi-symmetric monobloc wheels are considered in deliverable D2.1. These procedures shall be enlarged here in order to cover

- (weakly) non axi-symmetric wheels (section 3.4),
- wheels with braking discs attached (section 3.5),
- resilient wheels (section 3.6), and
- wheels with dampers (section 3.7).

The wheel input for TWINS is obtained using a finite element (FE) model of the wheel. For wheels other than axi-symmetric monobloc wheels, the FE-calculated wheel modal basis is generally updated according to experimental modal analysis results. Recommendations on both computation and measurement procedures will be given in the following.

3.1 General calculation settings

For general calculation settings such as roughness spectra, contact filter and other parameters we refer to ACOUTRAIN deliverable D2.1 (5).

The roughness input obviously has a big influence on the results for any absolute prediction. Relative predictions are less critical, but the spectral content has still an effect on results (e.g. it determines the relative importance between track and wheel). The roughness spectra to be used depend on the scenario. Section 2 has mentioned several possibilities and certain recommendations have been given; however, the discussion within Acoutrain has not yet led to commonly agreed recommendations.

3.2 Track modelling

Deliverable D2.1 (5) devotes an important part on the modelling of tracks. In the context of virtual homologation, the reproduction of a given real track will generally be of minor interest. If absolute predictions are considered, however, the choice of the track is of high importance (see discussion in section 2.1.

3.3 AXI-SYMMETRIC MONOBLOC WHEELS

Deliverable D2.1 (5) contains all necessary information for modelling of axi-symmetric monobloc wheels: meshing of the wheel, choice of response nodes and the setting of calculation parameters in TWINS.

Note that a purely computational approach (without experimental modal analysis) has widely been accepted as reliable for axi-symmetrical monobloc wheels (see also EN 13979 (3)). Indeed, mode shapes and natural frequencies can be predicted by the use of a FE-model with a good level of confidence. Modal damping is very low and differs little from one wheel design to another. Standard values, depending on the number of nodal diameters, can be used. Indeed, under rolling conditions the railway wheel is subject to internal losses (modal damping) as well as to a loss of energy due to the contact with the rail (so called “damping when rolling”). Whenever the “damping when rolling” is higher than the wheel modal damping, an exact measurement of the latter is not necessary. This is the case for (bare) monobloc wheels that have very low modal damping ratios.

3.4 NON AXI-SYMMETRIC WHEELS

Non axi-symmetric wheels cannot in principle be modelled in TWINS. The wheel modal basis that is used as an input for TWINS only contains the modal displacements of a wheel section (at different radii r) together with the number of nodal diameters n of each mode. This means that all wheel modes are assumed to have a cos ($n\theta$) dependence in circumferential direction (θ being the circumferential angle of the wheel). This is obviously not the case for non axi-symmetric wheels. However, certain non axi-symmetric wheels in fact behave nearly as axi-symmetric and may therefore be modelled in TWINS. This section gives guidelines how to determine these wheels and how to obtain realistic wheel modal data input for TWINS.

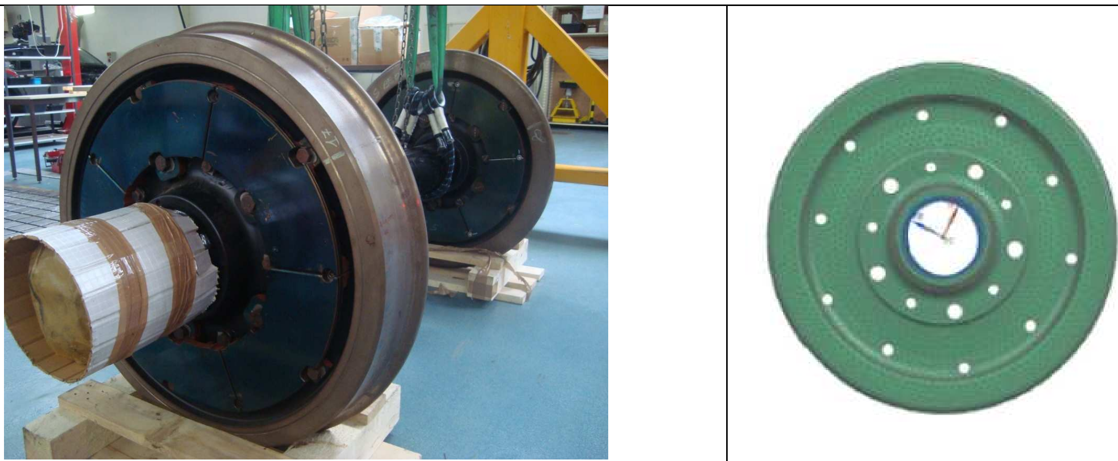
Only wheels with a web curvature that is constant over the wheel circumference are considered here. Wheels with a “double curved web” are assumed to have a too specific behaviour to be modelled in TWINS. Wheels that can potentially be treated with TWINS include wheels with a web thickness that is varying over the circumference and wheels with holes. Both web thickness and holes can be seen as patterns on the web with a certain number of lobes. This number is determining for the influence on wheel modes. Holes may also be needed in order to fix braking discs. The special case of wheels with wheel-mounted discs is treated in section 3.5. Also wheel dampers may introduce a slightly non axi-symmetric behaviour; these are dealt with in section 3.7. Measurements on wheels with attached discs have been performed within this task, using a sample provided by ALSTOM. Details about these measurements can be found in section 3.5.2.

3.4.1 Modelling procedure

Figure 3-1 (a) shows the wheelset supplied by ALSTOM for the ACOUTRAIN project. This wheelset is an EMU type wheelset with 850 mm diameter wheels and straight web. Each wheel has a mass of 308 kg. It features two circles of 10 holes each for the attachment of braking discs. Note that in this section, only the bare wheel is considered. The wheel equipped with discs is discussed in section 3.5.

The holes of the present wheel can be seen as a pattern with 10 lobes (or 5 diameters). This pattern couples with modes of 5 nodal diameters (and multiples of 5, which are beyond the frequency range of interest, however). The two solutions (rotated by 18°) obtained for each (n = 5) mode correspond to one mode with nodal diameters across the holes and one mode with nodal diameters between holes. The mode shapes and frequencies of all other mode pairs are identical (a frequency shift of less than 0.1 % is observed here, related to numerical errors). Mode pairs with 5 nodal diameters still show a shift of less than 1% in the present example. This low influence of the holes can most likely be explained by the fact that the stiffness of higher order modes is mainly controlled by the wheel tread.

Note that besides patterns with large numbers of holes, prime numbers of holes also lead to a behaviour close to an axi-symmetric wheel.



(a) Photograph of the wheelset equipped with discs	(b) FE-mesh of single wheel (without discs)

Figure 3-1: ETR wheelset supplied by ALSTOM for Acoutrain

The present wheel can clearly be modelled in TWINS. The modal basis that is needed for TWINS computations can be extracted from the 3D model over a section with holes or a section without holes. However, the wheel modal input file will then contain all 'double modes', half of them having a negligible modal displacement at the contact point. As the number of admissible modes is limited in TWINS, these modes must be removed (possibly manually) from the wheel modal input file.

As the wheel behaviour is very close to an axi-symmetric wheel it can alternatively be modelled in 2D. The procedure would then be the following:

1. Use a 2D model (wheel without holes) to compute the modal basis and extract an mp-file,
2. create a 3D model with holes and compute the modal basis,
3. adjust natural frequencies in the mp-file according to 3D computations.

Indeed, natural frequencies are slightly shifted due to the presence of holes. In the present example, the natural frequencies of radial modes decrease about 4 to 5% due to the softening of the wheel web. One nodal circle axial modes (1Ln) are less affected and decrease around 2% in frequency.

The reduction of the web surface area due to the holes should not be taken into account in TWINS computations. This introduces a certain security margin.

The presented case is relatively non-ambiguous. In the case of bigger holes, the behaviour of the wheel will diverge from that of an equivalent axi-symmetric wheel. Further research is necessary in order to establish a criterion and a commonly agreed limit for this criterion permitting to determine whether a given wheel may be treated as axi-symmetric.

As the direct analysis of mode shapes is relatively costly, the definition of a frequency criterion would be preferable. The frequency shift between ‘mode pairs’ of the same type would be a possible criterion. It could easily be measured by comparing receptances measured at different wheel sections. The comparison may be reduced to the most important modes, e.g. radial and 1-axial modes of order 2 to 4. Of course, the relation between frequency shift and consistency of mode shapes remains to be investigated in order to define a reasonable limit for an allowed maximum frequency shift. The preoccupation with such a criterion within ACOUTRAIN would be desirable as it applies for all types of non axi-symmetric wheels, including wheels with dampers or attached discs.

Note that prime numbers of holes on one circle are favourable for an axi-symmetric behaviour. A pattern of 5 holes for example does not couple with wheel modes $n < 5$, therefore the first ‘double modes’ should be found for $n = 5$.

Modal damping is expected to be very close to that of axi-symmetric monobloc wheels, i.e. the standard values can be used (see deliverable D2.1 (5)).

3.5 Wheels with wheel mounted braking discs

3.5.1 Modelling procedure

Wheels with flanged braking discs represent a special (but frequent) case of non axi-symmetric wheels. First, this non axi-symmetry is caused by the attachment holes in the wheel. However, typically these holes remain sufficiently small so that the wheel can be modelled as axi-symmetric (see section 3.4). Second, the braking discs themselves can be non axi-symmetric as is the case for the ALSTOM wheelset tested in ACOUTRAIN (Figure 3-1). This can slightly increase the ‘degree of non axi-symmetry’. As mentioned above, a criterion permitting assessment of whether the wheel concerned may be modelled as axi-symmetric still needs to be defined.

Irrespectively of the wheel symmetry, braking discs always affect the wheel modes in two ways:

1. Braking discs lead to some frequency shift of the wheel modes
2. Braking discs lead to increased modal damping of the wheel modes due to friction losses between discs and wheel

The effect of braking discs on wheel mode shapes is expected to be low. Therefore, a bare wheel (2D) FE-model is recommended for the computation of the wheel modal basis. The wheel modal input for TWINS computations is then obtained as follows:

1. Use a 2D model (wheel without discs) to compute the modal basis and extract an mp-file,
2. perform an experimental modal analysis (EMA) to measure natural frequencies and modal damping (see details in section 3.7.2), and
3. adjust natural frequencies and damping in the mp-file according to measurements.

Comments:

The presented procedure is identical to the one that is used for wheels equipped with dampers. However, braking discs are generally more massive than wheel dampers and can therefore be expected to have a slightly bigger influence on wheel mode shapes and frequencies. This is especially the case for discs that are directly fixed on the wheel web.

Within ACOUTRAIN, measurements have only been performed on the complete ATSA wheelset with discs because the discs could not be removed for the project. For industrial projects the situation is often similar and experimental modal analyses concentrate on the complete assembly for efficiency reasons. This is another reason why testing the influence of discs on wheel modes for each specific case does not seem realistic. It would therefore be desirable to produce more experimental proof that typical braking discs have negligible influence on wheel mode shapes.

Note that the introduction of braking discs in a 2D model is not recommended, even if the discs are mainly axi-symmetric. Indeed, the experience of VIBRATEC is that natural frequencies computed with such models are likely to show a higher discrepancy with measurements than those obtained with a bare wheel model. Even if a 3D model is used, the modelling of discs and their attachment on the wheel is relatively difficult and does not necessarily lead to better results than a wheel without any discs. The next section discusses different modelling procedures for wheels with attached discs in some detail. These include a full 3D model as well, illustrating the relatively low benefit from such a sophisticated model.

3.5.2 Example of a wheel with braking discs: ATSA ETR

Effect of braking discs on wheel natural frequencies

The complete ATSA ETR wheelset with discs has been displayed in Figure 3-1. Each wheel is equipped with one disc on either side with a mass of 63 kg. As visible on the photograph, the discs are separated in 5 parts; however, the connections between these parts are relatively strong. Indeed, the discs show similar disc modes as discs without partitions. Often, braking discs are mounted on the wheel web using connections that are located on one circle. This permits a radial thermal expansion of the discs without damaging the attachments. In the current example, the discs are fixed via two circles of connections; however, connections between wheel and discs are limited to the inner circle (near the wheel hub). The holes in the wheel web on the outer circle permit the discs to be connected to each other without contact to the wheel.

As mentioned, only measurements with the assembled braking discs have been performed within ACOUTRAIN because the discs could not be removed from the wheelset. Therefore the effect of the discs on the wheel modes can only be quantified by comparison with the FE-model. However, the model of the bare wheel can be used with good confidence. The complete model with discs bears higher uncertainties because of the connections between wheel and discs.

The main issue when constructing an FE-model of a wheel equipped with braking discs is the correct modelling of the connection between discs and wheel. The present model (as well as previously performed studies) shows that the detailed modelling of screws is not necessary. In contrast, the assumption of a perfect connection between wheel and discs at all contact surfaces leads to better results than the use of bars for the simulation of screws. [N.B.: the connections between the two discs through the holes in the web on the outer circle have been modelled using bar elements].

<table>
<tr><td rowspan="2">Mode type</td><td rowspan="2">FE-model</td><td colspan="3">Number of nodal diameters n</td></tr>
<tr><td>2</td><td>3</td><td>4</td></tr>
<tr><td rowspan="2">1 axial modes</td><td>w/o discs</td><td>-1.6 %</td><td>-1.3 %</td><td>-4.4 %</td></tr>
<tr><td>with discs</td><td>-5.0 %</td><td>-3.0 %</td><td>-1.7 %</td></tr>
<tr><td rowspan="2">Radial modes</td><td>w/o discs</td><td>7.1 %</td><td>1.7 %</td><td>-1.4 %</td></tr>
<tr><td>with discs</td><td>3.2 %</td><td>1.5 %</td><td>-1.0 %</td></tr>
</table>

Table 3-1: ATSA ETR wheel: Difference in wheel natural frequencies between FE-computations (model without and with discs) and measurement (complete wheelset with discs)

The observed shifts between calculated and measured natural frequencies are summarised in Table 3-1 for the most important wheel modes. Both the bare wheel model results as well as the complete model results are compared to measurements. Even though the maximum deviation is slightly reduced, there is no clear improvement of the model due to the addition of the discs. Note that only one result is given per mode because the frequency shift between the two modes of the same mode pair is extremely low.

Even though the scope is on rolling noise (i.e. wheel modes), computed and measured disc modes have also been compared. As the discs are much lighter than the wheels, the disc modes are much more sensitive to the attachment than the wheel modes and therefore represent another way to judge the quality of the model. Results are given in Table 3-2. Only axial disc modes are given because radial (in plane) modes are high in frequency and difficult to separate from wheel modes. The overall fit is acceptable. In fact, a better fit could be obtained with a slightly different attachment of the discs on the wheel, but at the expense of a worse correlation of wheel modes.

<table>
<tr><td>n</td><td></td><td>0</td><td>1</td><td>2</td><td>3</td><td>4</td><td>5</td><td>6</td></tr>
<tr><td rowspan="3">0 axial modes</td><td>Meas.</td><td>not meas.</td><td>615</td><td>734</td><td>1042</td><td>1346</td><td>1656
2127</td><td>2576</td></tr>
<tr><td>Calc.</td><td>713</td><td>683</td><td>774</td><td>1061</td><td>1458</td><td>1760
1874</td><td>2410</td></tr>
<tr><td>Diff.</td><td></td><td>11.0 %</td><td>5.5 %</td><td>1.8 %</td><td>8.4 %</td><td>6.3 %
- 11.9 %</td><td>6.9 %</td></tr>
</table>

Table 3-2: ATSA ETR wheel: Difference in disc natural frequencies between FE-computations and measurement

It is likely that the observed discrepancies between FE-computations and measurements may be reduced further by using a more sophisticated FE-model. For example, the current disc models are based on plans because CAD data of the discs was not available. However, the FE model shows a quite low effect of the discs on natural frequencies. Calculated mode shapes have also shown to be virtually identical.

The presented example, and in particular the relatively small incidence of the discs on wheel natural frequencies, corroborates the assumption that wheels equipped with discs can be treated as described in section 3.5.1 above (i.e. similarly to wheels with dampers). Indeed, a frequency shift of the same order of magnitude (around 5 %) can commonly be observed between bare wheels and wheels equipped with dampers.

Effect of braking discs on modal damping

The second and most important effect of braking discs on wheel modes is the increase of modal damping due to friction between wheel and discs. Table 3-3 summarises natural frequencies and damping of wheel modes measured on the ATSA ETR wheelset. Damping loss factors of the most important modes (1L and R of order $n \geq 2$) are of the order of 0.5 to 1%. Note that corresponding bare wheel modes show a damping of around 0.02%.

n=	0		2		3		4	
type of mode	f [Hz]	$\eta = 2C/C_c$ [%]	f [Hz]	$\eta = 2C/C_c$ [%]	f [Hz]	$\eta = 2C/C_c$ [%]	f [Hz]	$\eta = 2C/C_c$ [%]
0L	324	0.2	437	0.05	1116	0.1	1983	0.01
1L	1842	0.4	2481	1.4	3078	1.2	3935	0.6
R			2057	0.6	2741	0.9	3477	0.4

Table 3-3: Measured natural frequencies and damping loss factors

These increased modal damping loss factors lead to a reduction in rolling noise. In the present example, the reduction of wheel radiated power calculated with TWINS is 2.5 dB(A) at a speed of 80 km/h and 3.0 dB(A) at 250 km/h. This is based on the use of the TSI roughness limit as the total roughness input. Note that the reduction of total sound power (or sound pressure) will generally be much lower, depending on the relative importance of wheel and track.

The damping determined for the ETR wheel is relatively high compared to wheels that have been dealt with by VIBRATEC in the past. Most likely it is highly dependent on the nature of the attachment between wheel and discs (location, surface of contact, clamping torque, disc geometry, etc).

Effect of braking discs on wheel radiation

Shielding and re-radiation of discs is neglected in the proposed procedure. If the discs are fixed in the centre of the web, their vibration levels are close to the levels of the web itself. Therefore shielding and re-radiation will cancel out. If the discs are fixed near the hub only, the lateral vibration levels on the discs may be smaller than on the wheel and shielding could be more important than radiation – in this case the computation gives a worst case estimation.

3.6 RESILIENT WHEELS

Resilient wheels are characterised by a rubber layer between the web and the tread, as shown in Figure 3-2. They were initially designed to prevent or reduce squealing noise emission in tight curves and reduce unsprung masses. Resilient wheels are often claimed to be efficient for the reduction of rolling noise as well; however, this depends on the choice of the resilient layer dynamic stiffness and damping. Indeed, the resilient ring modifies the mechanical behaviour of the wheel in various ways, and not all of these modifications are favourable for a reduction of rolling noise. A discussion of the acoustics of resilient wheels can be found in reference (7).

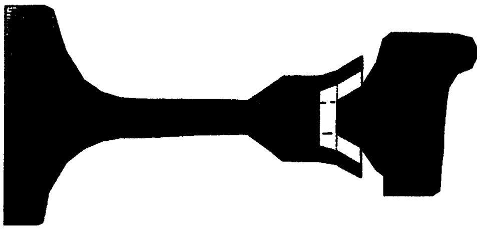


Figure 3-2: Sketch of a resilient wheel section

The main effects occurring are:

- Reduction of wheel web vibration due to uncoupling between wheel web and wheel tread. In order to be effective over a broad frequency range, the (radial) decoupling frequency should be below approximately f_0= 1.2 kHz (i.e. below the frequency range where the wheel is the dominant source of noise).
- Reduction of rail vibration. Uncoupling of the wheel tread also leads to an increased radial wheel receptance. A bigger fraction of the relative displacement between rail and wheel induced by the roughness is then absorbed by the wheel. The decoupling of the wheel tread has to be effective in the frequency range where the rail is usually dominant (i.e. in the range from 600 Hz to 1.2 kHz). Due to the effect on track vibration, total radiated power should be used instead of wheel sound power as an acceptance criterion for resilient wheels.
- Increase of wheel tread vibration. The absorption of roughness excitation in the mid-frequency range in favour of rail vibration may lead to an increase of wheel tread vibration. Additionally, radial wheel modes are shifted to lower frequencies. This leads to increased excitation of the first radial modes and to an increased modal density within the relevant frequency range (0 to 5 kHz).
- Reduction of wheel vibration due to the increase of modal damping. The internal damping of the resilient material is much higher than that of steel and therefore increases the modal damping of the wheel. However, the effect on modal damping also depends on the stiffness of the resilient layer. In order to maximise modal damping, the radial stiffness value of the resilient layer should be chosen close to that of the wheel web, i.e. the frequency f_0 should be set above approximately 1.8 kHz. Obviously, a trade-off between modal damping and uncoupling of the wheel tread has to be found.

3.6.1 Modelling recommendations

The following recommendations are based on VIBRATEC experience with modeling of resilient wheels from different manufacturers.

The key parameters to account for the dynamic behaviour of a resilient wheel are radial and axial dynamic stiffness and damping loss factors of the resilient layer.

Rolling noise predictions for trains with resilient wheels can therefore be performed with a good level of confidence if the radial and axial dynamic stiffness and damping of the rubber ring can be known.

Radial dynamic stiffness of the resilient layer – K_r

Certain resilient wheels have a strongly V-shaped gap between tread and web which permits the resilient material to shear rather than compressing it. Like this, a static radial stiffness as low as 20 MN/m can be obtained, resulting in a dynamic stiffness ranging from ***20 to 40 MN/m***. The uncoupling frequency of the tread mass on the rubber stiffness - mode R1 - will range between 70 and 100 Hz.

Since this 20 to 40 MN/m stiffness is much lower than the wheel web radial stiffness, tread radial modes will not involve the wheel web. The wheel radial modes will depend mainly on wheel tread geometry. These modes will occur at much lower frequencies than on monobloc wheels and will be slightly more damped due to the damping effect of the rubber. However, this damping effect remains low because the potential energy of the radial modes will mainly be concentrated in the wheel tread (and not the resilient material - see equation 3-1 below).

Standard resilient wheels will exhibit a rubber radial static stiffness ranging from 100 to 400 MN/m. The rubber dynamic stiffness will therefore range from about ***200 to 1 000 MN/m***. In this case, the mode in which the tread uncouples from the web in the radial direction - mode R1 – can be more difficult to extract from an experimental modal analysis.

Radial modes of type R2, R3 and R4 can be significantly damped – modal loss factor η ranging from 1 to 6% - due to the effect of rubber deformation - see equation 3-1 below.

Estimation of K_r from measurements

The radial dynamic stiffness K_r is often difficult to obtain with a reliable accuracy from the wheel manufacturer. In general, only the static value can be obtained.

For an existing wheel, a reasonable estimate of K_r in term of stiffness can be obtained by using the following procedure:

1. Identify the modal parameters of the first wheel tread radial modes (at least R2, R3, R4 modes and possibly R0 and R1), by means of an experimental modal analysis (see also the measurement recommendations for damped wheels – section 3.7.2).
2. In parallel, identify the acceleration ratio between radial vibration levels on both sides of the resilient layer, close to the excitation point (see Figure 3-3).
3. Build an FE model of the wheel and adjust the radial dynamic stiffness K_r to get a good tuning between measured and computed modes and to get a good correspondence between radial acceleration ratio on both sides of the rubber ring (tread side and web side).

4. After tuning of numerical modes to experiments, adjust the hysteretic damping loss factor η_{rubber} related to K_r, to obtain a good agreement between measured and computed modal damping. The modal damping of modes R1 to R4 can be computed using a potential energy approach from the following formula:

$$\eta = (\eta_{rubber}E_{p-rubber})/E_p \text{ or } \eta_{rubber} = (\eta E_p)/E_{p-rubber} \qquad \textbf{3-1}$$

η being the modal damping of the mode under study, extracted from the experimental modal analysis,

η_{rubber} being the hysteretic damping of the rubber**,**

$E_{p-rubber}$ being the potential energy of the rubber for the mode under study (given by the FE model), and

E_p being the total potential energy of the mode under study (given by the FE model).

Inversely, if the radial stiffness and damping value of the rubber is known with a reasonable accuracy from the manufacturer, the effective damping of the wheel radial modes can be obtained numerically using equation 3-1 (but the FE model is required to compute the potential energies needed in equation 3-1).

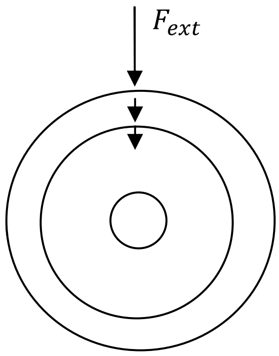


Figure 3-3: Determination of radial acceleration on both sides of the resilient ring

Axial dynamic stiffness of the resilient layer – K_a

A similar approach to the one applied in the radial direction should be applied for the axial direction.

The first axial modes of the tread from 0L2 to 0L5, and possibly 0L0 (umbrella mode) with the two wheel treads vibrating axially out-of-phase, should at least be identified.

In parallel, the ratio between axial vibration levels on both sides of the resilient ring should be measured and analysed.

The axial dynamic stiffness amplitude K_a can then be adjusted by tuning the computed axial natural frequencies to measured ones, and the rubber damping can also be determined by using equation 3-1.

3.7 Monobloc wheels with dampers

Wheel dampers are solutions that allow the wheel modal damping to be increased without affecting too much the functional requirements of the wheel. Damped monobloc wheels can indeed be used for applications that are unsuitable for resilient wheels, such as high speed traffic. The fact that wheel dampers have a relatively low impact on wheel modes except their damping ratios also facilitates rolling noise computations using TWINS.

Note that the dampers considered here are devices that modify the modal damping of a wheel. Devices that mainly act through shielding cannot be treated with the presented procedure.

Procedures for the assessment of damper efficiencies have recently been developed within the Franco-German research project STARDAMP (7).

3.7.1 Modelling recommendations

Monobloc wheels with dampers are treated very similarly to ‘undamped’ wheels in terms of finite element modelling. Indeed, the damper effect is accounted for by updating the calculated wheel modal basis with measured damping values and natural frequencies (see section 3.7.2). This is based on the hypothesis that mode shapes remain unaffected by the dampers. This is generally the case; however, very massive and possibly stiff devices may have some effect. Similarly to the case of wheels with braking discs, a maximum admissible frequency deviation between wheels with and without damper could be defined. However, the definition of this value would need more research effort. Measuring the incidence on mode shapes directly seems too time consuming for an industrial context.

The same recommendations as for standard monobloc wheels apply. We refer to deliverable D2.1 (5) for details.

3.7.2 Measurement recommendations

Detailed measurement procedures are given in the Final Report of the STARDAMP project (reference (7)). Two procedures are discussed there, the classical experimental modal analysis (EMA) and a simplified experimental analysis (SEMA). We recommend that an EMA should be performed, implying the use of dedicated software for the post-processing of recorded FRFs. Indeed, this method delivers a better estimation of modal damping than the “half power bandwidth method” used in the SEMA. This applies in particular to highly efficient dampers that lead to a higher modal overlap.

The most important points discussed in reference (7) are reproduced in the following.

Test setup

It is recommended to use a complete wheelset for measurements instead of a single wheel. Both wheels shall be equipped with dampers. The wheelset should be freely suspended by using elastic ropes or springs. The presence of axle boxes is no constraint for these measurements. The suspension can be fixed on the outer extremities (on the axle boxes if present) or on the axle between the wheels; however, the bouncing mode of the wheelset on its suspension should be below 30 Hz (if elastic ropes are used, a decoupling as low as 2 Hz can generally be achieved).

Preliminary tests

An important issue to be controlled is the representativeness of the tested sample. Indeed, the efficiency of a wheel damper can highly depend on its mounting conditions. These do not, of course, only depend on acoustical considerations but above all on safety. It is important to respect the correct mounting conditions in order to obtain a wheel that is representative for operating conditions.

- Ideally, several samples should be tested to produce mean modal damping values to be used for TWINS computations. If only one wheelset is available, at least both wheels of this wheelset should be measured.
- Tests after dismantling and remounting the dampers can be compared to be sure that the mounting conditions are representative and that the dispersion is not too high.

Certain wheel dampers can exhibit nonlinear behaviour, i.e. their efficiency depends on the excitation level. It is therefore recommended to control the wheel damper behaviour at different excitation levels. Ideally, the excitation is displacement-controlled for this type of test because roughness excitation is a displacement type excitation. Testing with a displacement amplitude of 1 μm is proposed as a realistic excitation. However, such excitation levels may already be difficult to obtain in practice because sufficiently powerful shakers will often not permit a frequency range up to 6 kHz to be covered. [N.B.: Around the main radial modes, the receptance of a wheel equipped with absorbers typically lies in the range of 10^{-9} to 10^{-8} N/m, corresponding to a stiffness of 10^{8} to 10^{9} m/N. A displacement-controlled excitation of 1 μm then leads to an injected force of 100 N to 1 kN.] The corresponding low amplitude excitation can be applied by the use of a shaker or hammer; ideally the same excitation will be chosen as the one that is planned to be used for the rest of the experimental modal analysis. If the damper has been shown to behave linearly, the remaining testing can then be performed at low excitation levels (and notably by the use of an impact hammer).

Certain types of wheel dampers can have a large surface covering an important part of the wheel web. As mentioned earlier, possible shielding effects cannot be taken into account by this procedure. However, it should be verified that axial vibration levels of the damper (due to radial excitation of the wheel) remain below or close to the levels measured directly on the wheel web; this procedure delivers a worst case estimation.

Test equipment

The accelerometer shall have a mass of less than 10 g and a diameter less than 15 mm. The accelerometer and its mounting must not have a resonance below 10 kHz. This can usually be achieved by attaching the accelerometer using a ceramic or Cyanoacrylate glue.

Excitation shall be provided by an instrumented hammer containing an integral force gauge. A suitable hammer would normally have a mass of around 100-200 g and a titanium tip. It shall be demonstrated that it gives a sufficiently flat input spectrum up to at least 7 kHz (i.e. the power spectrum of the force shall be flat within 20 dB in this range). It is important to tap the wheel very gently and to avoid double strikes.

Data shall be recorded using a multi-channel Fourier analyser. For example, a 16 channel analyser permits the signals of 5 tri-axial accelerometers to be recorded simultaneously (+ one channel for the force sensor), which is sufficient for one wheel section (see EMA procedure below).

For the measurement of FRFs ISO 7626-5 shall be used (8). The following measurement parameters shall be used:

- FRF measurement may be accelerance or mobility;
- The hammer signal shall be used as the reference to determine the transfer functions;
- An average of at least five impacts shall be used for each FRF measurement;
- The coherence function shall be used as an indication that the signal is not contaminated by background noise. A coherence of at least 0.8 is required apart from at isolated dips in the FRF[1]
- The analysis shall be triggered by the force pulse. At least 10 ms of pre-trigger shall be used. The pre-trigger shall not exceed 15% of the analysis window;
- A rectangular window shall be used (i.e. no window). For the force signal a 'force' window may be used instead. . An exponentially decaying window can be used for undamped wheels in order to keep the window at a reasonable length. For shaker excitation windowing has to be used (e.g. Hanning);
- The analysis window shall be long enough to capture the whole of the decaying response due to an impact. For a strongly damped wheel it is expected that 1 s will be sufficient, implying a frequency resolution of 1 Hz. For less efficient dampers an analysis window of 2 s (leading to a frequency resolution of 0.5 Hz) may be necessary. A sufficient frequency resolution is indeed necessary for the correct estimation of modal damping. The determination of modal damping for wheels without dampers will require a minimum frequency resolution of 0.25 Hz; however, the exact measurement of damping is generally not necessary because default damping values are used for bare wheels. Nevertheless, recording at least a reduced set of FRFs for the undamped wheel can help to identify the wheel modes and provide validation of the FE model.
- The maximum frequency of analysis shall be at least 5.62 kHz (for the 5 kHz one-third octave band);
- The sample rate shall be at least 2.5 times the maximum frequency of interest (i.e. at least 14 kHz to obtain measurements up to the 5 kHz one-third octave band. A higher sample rate can be used if available.

[1] It is accepted that the signal may drop into the background noise at anti-resonances of the wheel. The criterion is included to ensure that the majority of the signal within a frequency band is not contaminated by noise.

A measurement of background (instrumentation) noise shall also be taken to verify that both the force and the response signal is at least 10 dB above the background noise in each one-third octave band. To achieve this, the data acquisition system shall have sufficient resolution (normally at least 16 bits; 24 bit resolution is preferable). Parallel data acquisition across channels shall be used to avoid any phase shift between the signals.

EMA procedure

An experimental modal analysis (EMA) is a procedure that permits the (measured) vibration response of an object to be decomposed into modes of vibration and frequency, shape and damping of each mode to be identified. This has to be performed by the use of specific software for post-processing of the measured FRF. This procedure also requires an experienced user and its detailed description goes beyond the scope of this report.

Excitation can be provided by either an electromagnetic shaker or an impact hammer. In the case of shaker excitation, the above listed acquisition parameters (windowing, etc) have to be adapted. Hammer excitation, however, is generally sufficient if the above measurement recommendations are respected.

The procedure suggested here is based on the measurement of ‘direct FRFs’ (as opposed to reciprocal measurements) but reciprocal measurements are also possible. Two excitation positions are used, sufficient for the excitation of radial and axial mode types. The wheel response is recorded at a more important number of points. The measurement grid has to be chosen sufficiently fine in order to identify all relevant mode shapes. An example is given in Figure 3-4 which constitutes a sufficient mesh for a standard wheel.

The use of tri-axial accelerometers is preferable because the post-processing of recorded FRFs becomes more straightforward. Meshing of half a wheel is sufficient because of its axi-symmetry. Only one wheel of a wheelset is meshed in detail; the opposing wheel, as well as the axle are meshed in a much coarser way. This permits modes to be identified that involve the axle or both wheels simultaneously (e.g. bending modes of the axle or wheel modes that occur twice with both wheels moving in phase or out of phase). Five measurement positions per section are used (two on the tread and three on the web) and 13 sections, resulting in a total of 65 positions. In the example given three measurement points are also placed on each damper (green part of the mesh), however, this is not necessary for the determination of wheel modes and their corresponding modal damping.

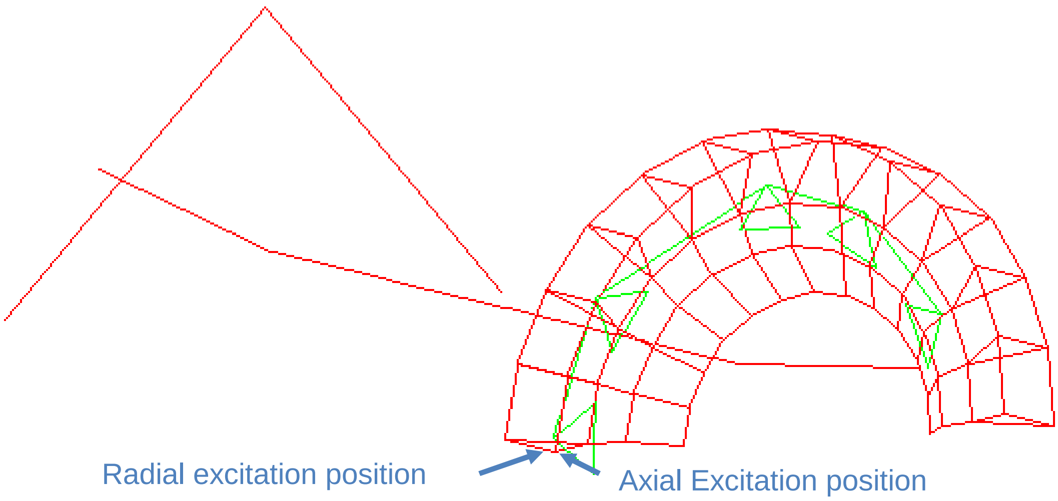


Figure 3-4: EMA mesh example

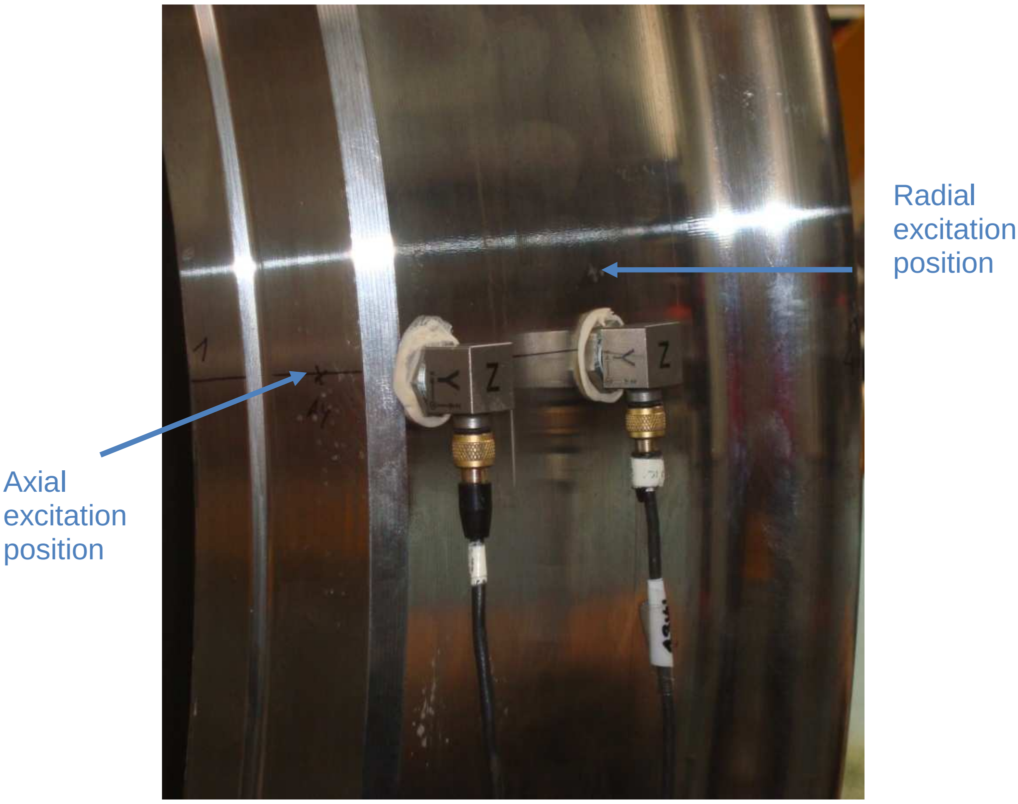


Figure 3-5: Position of accelerometers

Two positions for force excitation are recommended (as indicated in Figure 3-4 and Figure 3-5):

- Radial excitation at the outer side of the tread, for *Rn* and 1-*Ln* modes),
- Axial excitation on the tread, for 0-*Ln* modes.

Figure 3-5 shows that the radial excitation position is 'occupied' by the accelerometer. The excitation position should therefore be shifted in the circumferential direction (and not axially!). Similarly, the tri-axial accelerometer could be fixed laterally, in which case the axial excitation has to be shifted in the circumferential direction.

The resulting FRFs are analysed with dedicated software which allows an extraction of modal parameters (notably damping). Based on the experimental mesh, mode shapes can be visualised and mode types can be identified.

3.7.3 Tuning of the wheel modal input file for TWINS

Modal damping and natural frequencies are replaced in the wheel modal input file (mp-file) by the measured values[2]. Initially, unidentified modes may be left with the standard modal damping corresponding to undamped wheels. It is then highly recommended to compute wheel receptances using this updated wheel model and to compare them with directly measured receptances. Generally some discrepancies are observed that can be minimised by correcting the initial damping values (frequencies are most often correctly detected). This procedure is repeated until a sufficient fit is obtained. It is important to consider the radial receptance at mid-tread (nominal contact point) and at the outer extremity of the tread (approximately +60mm axially relative to the nominal contact point) as well as axial receptance. This permits radial, 1-axial as well as 0-axial modes to be excited. Modes that have not been identified during the EMA will remain visible as sharp peaks in the receptance. These may then be tuned to fit with the measured receptance through an iterative process.

[2] Note that the mp-file uses the damping ratio $\xi = C/C_c$ instead of the loss factor $\eta = 2C/C_c$ used above.

4. SOURCES OF UNCERTAINTY IN WHEEL VIRTUAL TESTING

The idea of comparing two different wheel designs using the computational approach has largely been accepted and is described in EN standard 13979 (3). One of the reasons is undoubtedly that no measurement approach allows two different wheels to be tested under exactly the same conditions in terms of wheel roughness and track properties. In this sense, the computational approach can be beneficial for the control of uncertainties. At the same time it has to be guaranteed that the computational approach leads to similar results to direct measurements. Deliverable D2.2 (9) therefore summarised comparisons of available measurements and TWINS predictions. One outcome of this analysis was that the main source of uncertainty is found in the input parameters such as roughness and track decay rates. Measured modal damping factors are also input parameters that represent a source of uncertainty. Deliverable D2.2 includes one wheel damper case from the STARDAMP project. The reduction of track-side SPL due to the use of wheel dampers was relatively well predicted. However, one such comparison does not represent an analysis of uncertainties.

As stated above, the main issue in terms of uncertainty is that laboratory measurements are always performed on few (or even one single) samples. It is therefore crucial to take the necessary precautions in order to assure that the tested sample is representative for the wheels that will be present on the real trains. A similar issue exists, of course, with classical testing, because the few samples that are used for pass-by noise measurements have to be representative for entire series of rolling stock to be produced. For laboratory testing of wheel dampers the risk is higher because often modal damping is assessed using one single sample. A thorough analysis of uncertainty related to the dispersion between different samples of wheels or wheel dampers would be extremely costly, and will most probably not be available in the near future.

Another source of uncertainty can be seen in the use of TWINS for wheels that have a slightly non axi-symmetric behaviour or wheels where the mode shapes may be affected by adding devices such as dampers or braking discs. Several recommendations have been given with respect to criteria that may be used in order to determine whether a certain wheel design can be treated with the computational approach. A more detailed investigation would need far more research effort.

Acoutrain deliverable D2.3 (10) quantifies the sources of uncertainty and their effect on the outcomes of virtual testing. It represents a basis for the decisions to be made within WPs 1 and 5. Similarly, the present deliverable provides elements that shall serve for the definition of commonly agreed acceptance procedures. To the author’s opinion, all cases presented (wheel types) are in principle suitable for rolling noise computations with TWINS, subject to a wise use of the tool. However, there remains a lack of precise rules that permit unambiguous separation of designs for which the computational approach shall be allowed from designs that are not suitable for this approach.

5. REFERENCES

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