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Noise control design of railway vehicles–Impact of new legislation

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Abstract

Type testing specifications can affect the design considerations for a rail vehicle. It is shown with illustrative examples how low-noise design features optimized for the type test condition may have very limited effect for normal operation. The type test conditions in the new European legislation Technical Specification for Interoperability-Conventional Rail (TSI-CR) for noise emission from interoperable rail vehicles is used as a basis. The purpose of a pass-by noise type test is to certify the rolling stock but the track is in many situations both the dominant excitation and radiation source. In the TSI-CR a stringent track specification is defined to keep the track noise contribution low and to promote reproducibility between test sites. It is shown by simulations with the TWINS software that the ranking of noise sources may be different on a type test track than on a typical operational track. This may lead to a demand to introduce noise reduction features on the vehicles that have a small effect on operational track. As two examples, the introduction of wheel absorbers and bogie skirts is investigated. The inaccuracies in the present standards for rail roughness measurement and the consequence in terms of uncertainties in noise emission are highlighted. For rail vehicle type tests under stationary and accelerating conditions, the track properties are irrelevant but the operating modes of auxiliary equipment and cooling fans are crucial. It is shown that the consequences of either specifying a typical load cycle or a worst-case scenario are considerable due to different ranking of sources. Consequently, the focus for the engineering work can in such cases be devoted to systems that only dominate in extreme cases and do not contribute to the noise emission during normal operation.

1. Introduction

Railway noise is a multi-faceted phenomenon, in which the infrastructure and the rolling stock interact in a complex way. This fact complicates the split of responsibilities between the stakeholders in the business: operators, infrastructure owners, vehicle and track product manufacturers and governmental bodies for transport and environment. With this fragmentation of responsibility there is an obvious risk that effort and money is spent on noise control measures in the wrong places, especially when several kinds of sources are involved (e.g. wheel, track, aero-acoustics, cooling fans). The elementary rule in noise control that it is fruitless

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to reduce non-prominent sources must be observed. This is, of course, common knowledge within the community of railway noise experts but it is sometimes difficult to communicate to non-specialists.

Rolling stock suppliers are normally responsible for the sound emission from vehicles in a type test. The test conditions and the setting and compliance to limit values have in the past been defined in commercial contract between supplier and customer. Now, with the upcoming Technical Specification for Interoperability (TSI) in Europe this situation is about to change. The noise creation limits with associated test conditions will be stipulated by legislation.

Leth has described noise legislation from the perspective of a rolling stock supplier [1]. The focus then was on the high-speed range whereas the present paper has a stronger focus on the so-called "conventional" speed range (up to 190 km/h). The current paper will take into account the substantial changes in the proposed TSI-Conventional Rail (CR) that have occurred since the earlier paper and will also have a wider scope—including noise at standstill conditions and a discussion on the required measurement accuracy of rail roughness.

2. Future noise legislation

The TSI will have a great impact on the harmonization of the European railway system. The objective is to stimulate the growth of the railway business by defining a common standard which should allow an open market and facilitate cross-border traffic. Limit values on noise creation were first included in the TSI for high-speed rolling stock (speeds greater than 250 km/h) that has been in force since December 2002 [2]. There is a currently ongoing revision of the TSI-HS that will include speeds down to 200 km/h and a general adjustment of limit values and track specifications, taking into account the results from a large scale test campaign NOEMIE [3] on existing high-speed rolling stock.

Next to come is the TSI-CR that will cover railway vehicles operating at speeds up to 190 km/h. In the TSI-CR, the noise emission part [4] has been drafted by an expert group within the Association Européenne d'Interopérabilité Ferroviaire (AEIF). Formally, however, it is the European Commission (EC) that presents the TSI to the EU member states for their approval, which means that the EC can make changes to the TSI drafted by the AEIF. The TSI-CR includes limit values for noise creation during pass-by, standstill and acceleration plus noise inside the driver's cab. The limit values proposed by AEIF are summarized in Table 2 in the appendix. This was approved in November 2004 and the TSI is expected to be officially published in 2005. The ambition is to have as many common elements as possible in the TSI-HS and TSI-CR, such as track specification, indicators and operating conditions.

It is foreseen that the TSI will have a great impact on the market and become the de facto standard for the majority of new rolling stock contracts, even those that are not primarily intended for interoperable service. Also European countries outside the EU are expected to be influenced by the noise specifications in the TSI.

3. Trains on reference track and on normal operation track

For the speed range relevant to the TSI-CR (up to 190 km/h) rolling noise plays an important role. To ensure reproducibility between tests performed at different test sites it is necessary to have an acoustically stringent specification for the track. Such a track, on the other hand, would be very different from a "normal" track. A compromise between reproducibility and availability has to be found. The existing TSI-HS track specification prescribes a roughness spectrum (see Fig. 1), and a *static* pad stiffness $k_{\mathrm{vert}} \geqslant 500\,\mathrm{MN/m}$ at a preload of 60 kN (the *dynamic* pad stiffness is normally found to be a factor 2-4 higher than the static stiffness). During the course of the work with the TSI-CR the need for a functional track specification was identified and an alternative proposal (sometimes referred to as TSI+) for a track specification has emerged from the NOEMIE project [3] and was accepted for the TSI-CR. This specification uses track decay rate spectra instead of pad stiffness and also has a different rail roughness spectrum (see Fig. 1). The proposed vertical track decay rate curve is shown in Fig. 2 together with decay rates calculated with TWINS ("Rodel" module [5]) for various vertical pad stiffness values. The proposed TSI-CR track specification is considered to be representative of a newly ground 60 kg/m rail on well-maintained ballast foundation, concrete sleepers and with medium pad stiffness ($k_{\mathrm{vert,dyn}} \approx 400\,\mathrm{MN/m}$). The TSI-CR proposed track specification should thus represent a "best case" scenario in the existing network provided that it is maintained with great care.

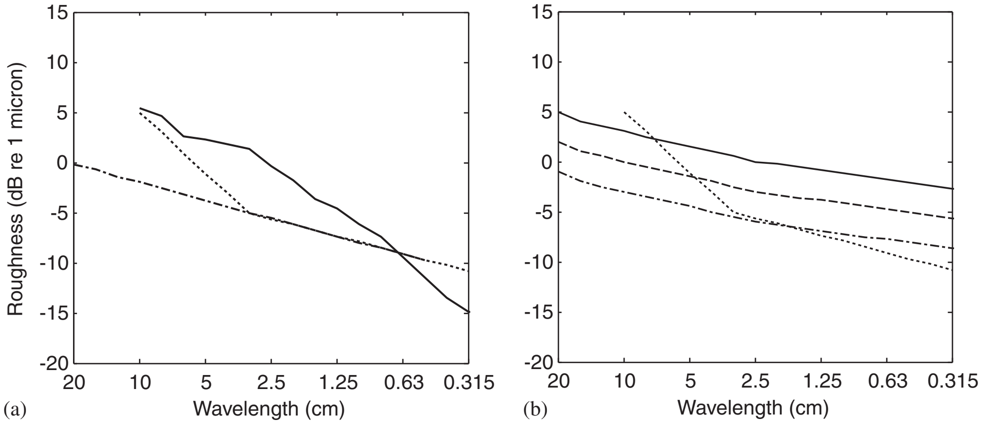

Fig. 1. (a) Rail roughness spectra: — Lundby measurement (average operational track quality); -·-·- TSI-HS specification December 2002; · · · · TSI-CR specification. (b) Wheel roughness spectra used in the analysis. — "average" wheels; - - - "good" wheels; -·-·- "excellent" wheels; · · · · TSI-CR track.

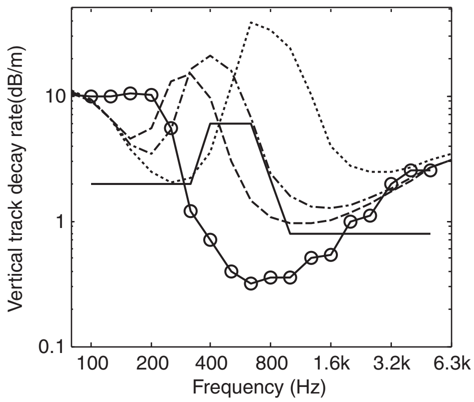

Fig. 2. Vertical track decay rate spectra: — TSI-CR specification; – – – calculated $k_{\mathrm{pad,vert}} = 200\,\mathrm{MN/m}$; -·-·- calculated $k_{\mathrm{pad,vert}} = 400\,\mathrm{MN/m}$; · · · · calculated $k_{\mathrm{pad,vert}} = 1600\,\mathrm{MN/m}$; ○-○-○ measured Silent Track project reference track.

Notably, the rail roughness on the reference track is considerably lower than on an average operational track (see Fig. 1). The track decay rates depend mainly on the pads and the preload from the fastening system. There is a considerable variation of properties between operational tracks in the network. The reference track is expected to have a higher decay rate than an average operational track. For instance, a measured decay rate from a representative European track in the Silent Track/Silent Freight projects [6] is compared in Fig. 2 with the TSI-CR decay rate spectrum. This track corresponds to a dynamic pad stiffness of $k_{\mathrm{vert,dyn}} \approx 80\,\mathrm{MN/m}$.

In the following, a numerical study will demonstrate the consequences of having different combinations of roughness, pad stiffness and decay rates for a two car electric multiple unit (EMU) trainset. It will be shown how different noise control options on the vehicle (wheel dampers, bogie skirts) have very different effects on the total noise reduction depending on the type of track. For example, noticeable effect when the vehicle is run on a "low-noise" reference track may be negligible on an operational track.

3.1. Rolling noise—definition of baseline numerical model

The numerical model used in the study is based on one-third octave data tuned to the Regina tests treated in Ref. [1]. Sound power spectra and decay rates have been calculated with the TWINS software and combined with a Bombardier in-house software to calculate $L_{A\mathrm{eq,TP}}$ pass-by noise levels. Although, the train unit has only two cars, the results are applicable for longer train compositions as well. Simulations show that the number of cars (n_{car}) has a marginal influence on the $L_{A\mathrm{eq,TP}}$ level at a 7.5 m microphone distance when $n_{\mathrm{car}} \geqslant 2$. Compared with $n_{\mathrm{car}} = 2$ the asymptotic limit for an infinite number of cars only gives an additional +0.5 dB. The wayside properties also have some influence on the results. The calculation model used allows the ground level and ground impedance to be varied. In the analyses below, the ground level was assumed to be 1 m below railhead and the flow resistivity of the ground surface 1000 rayls (corresponding to "sandy silt" or gravel [7]). By changing the ground parameters within reasonable values, the absolute $L_{A\mathrm{eq,TP}}$ value can differ typically in the range ±1 dB but the relation between source contributions, and consequently the conclusions of this study, will remain.

Three different levels of wheel roughness have been assumed (see Fig. 1(b)). The "average" curve represents the typical roughness for Regina EMU wheels measured during a campaign in 2002. The two curves "good" and "excellent" have the same shape but are offset by −3 and −6 dB, respectively. In Fig. 1(b) the TSI-CR rail roughness has also been included for comparison.

As a first step, the two-car trainset is considered to be running at 80 km/h on a track with TSI-CR roughness. Noise from traction gear and cooling fans is neglected. This implies that the TSI limit value for "coaches" should be met (i.e. $L_{A\mathrm{eq,TP}}$ below 80 dB). In Fig. 3 the calculated wayside sound pressure level $L_{A\mathrm{eq,TP}}$ is shown for wheel roughness qualities "average" and "good" and a range of different pad stiffness values. The TSI-CR reference track corresponds to a dynamic pad stiffness of 350–400 MN/m and the TSI-HS track to a dynamic pad stiffness in the range 800–1600 MN/m ($k_{\mathrm{pad,vert}} = 1600$ MN/m has been stated as a practical upper limit for pad stiffness [6]). For this assumption of a TSI-CR track it is seen that the wheel roughness spectrum "good" is required to not exceed 80 dB. It is clearly seen how the track noise and also the total noise rapidly grows for softer pads. The noise levels in Fig. 3(b) are 2.2 dB lower than in Fig. 3(a). The reason why the 3 dB reduction in wheel roughness in Fig. 3(b) has such a high impact on the noise levels is because the wheel roughness dominates over the rail roughness in Fig. 3(a).

According to the TSI-CR the train shall comply with the limit value both at 80 km/h and also at its maximum speed (for which the limit is adjusted using the well-established regression law $30\log_{10}(v_{\max}/80)$.) In Fig. 4, the effect of increasing the speed to 200 and 250 km/h is illustrated. Not surprisingly, there is a shift towards more wheel noise than at 80 km/h but it is interesting to see that, for 200 km/h, the track noise is still roughly equal to the wheel noise for a pad stiffness of 350 MN/m. By comparing Figs. 4 and 3(b) it appears also that the $30\log_{10}(v)$ regression law is justified at least up to 200 km/h. The dotted line represents the aerodynamic noise contribution, which increases rapidly with speed and becomes noticeable at 250 km/h. In the numerical model, the aerodynamic contribution has been derived from pass-by measurements of high-speed trains running at speeds 250–320 km/h (see details in Section 3.3).

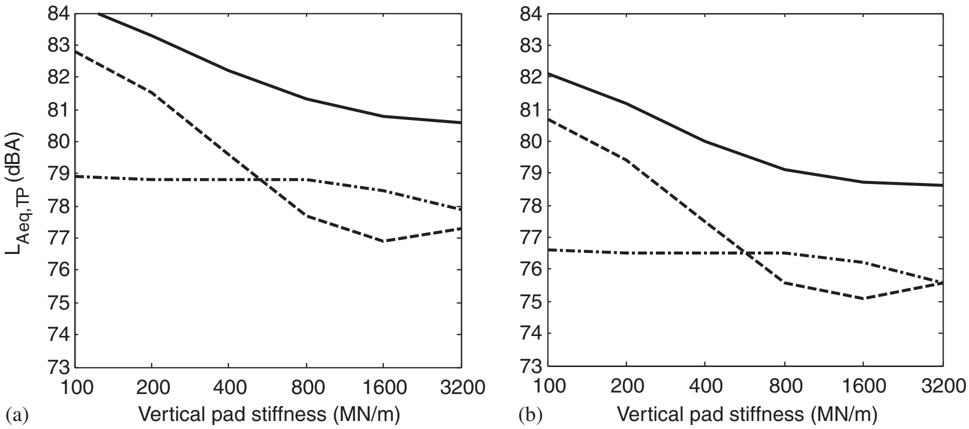

Fig. 3. Calculated pass-by sound pressure level at 80 km/h: — total; – – – track; - · - · - wheel. (a) TSI-CR rail roughness and "average" wheel roughness and (b) TSI-CR rail roughness and "good" wheel roughness.

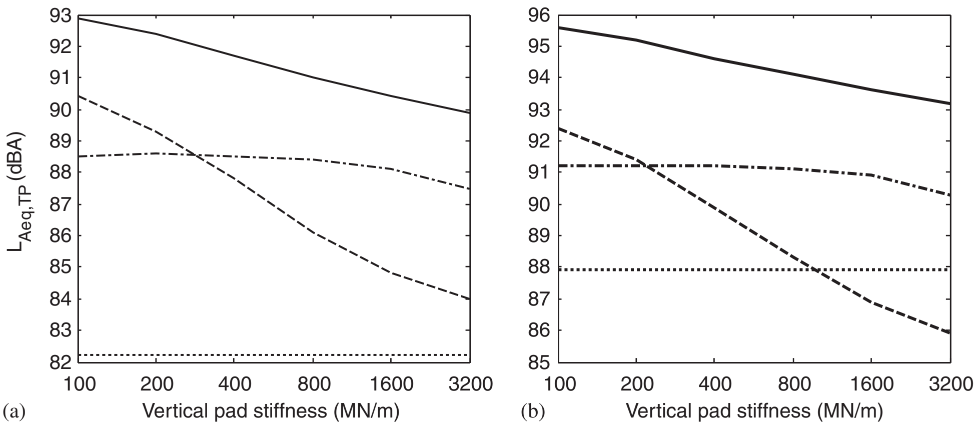

Fig. 4. Calculated pass-by sound pressure level for TSI-CR rail roughness and "good" wheel roughness: — total; – – – track; -·-·- wheel; · · · · aerodynamics. (a) 200 km/h and (b) 250 km/h.

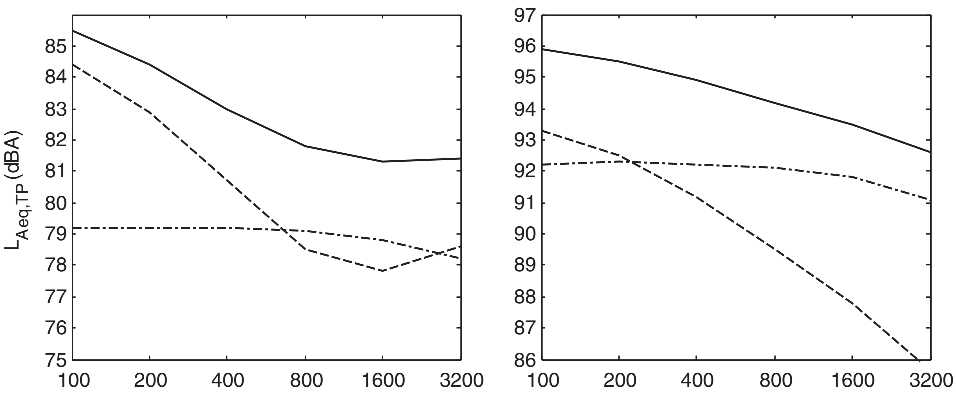

Fig. 5. Calculated pass-by sound pressure level for average (Lundby) rail roughness and "average" wheel roughness: — total; – – – track; -·-·- wheel. (a) 80 km/h and (b) 200 km/h.

The next step is to see what happens when these results are transferred to "normal" operational conditions. Now the "Lundby" measured rail roughness spectrum in Fig. 1(a) and the "average" wheel roughness spectrum in Fig. 1(b) are used. From the results shown in Fig. 5(a) the noise in operational conditions (i.e. a soft track and high roughness) could easily exceed 84 dB for this train configuration. This should be compared with the TSI limit of 80 dB. Note that in Fig. 5(b) the track noise is somewhat underestimated for high pad stiffness values (1600–3200 MN/m) due to the lack of rail roughness data at long wavelengths for the Lundby spectrum. The long wavelengths have some importance due to the low-frequency character of the sleeper noise, which becomes important for the case with very stiff pads.

3.2. Rolling noise—efficiencies of noise control options on the vehicle

In the numerical calculation, two noise control options are considered: wheel dampers and bogie covers. The effect of wheel damping is taken into account by using the "factor damping" feature in TWINS [5]. This means that all modal damping ratios are multiplied by the same factor. Unless a full-scale experimental modal analysis of the damped wheel is carried out, this is the best option. The initial damping ratios are the recommended default values of 10^{-4} for modes with 2 or more nodal diameters, larger for modes with 0 or 1

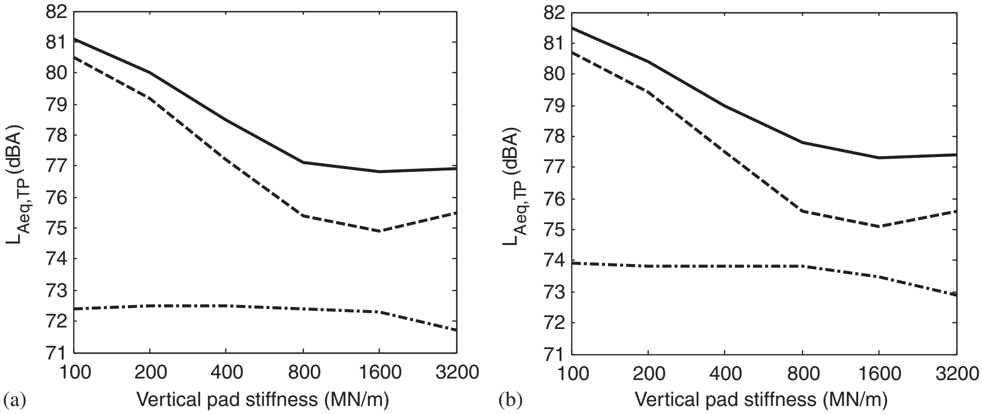

Fig. 6. Calculated pass-by sound pressure level for same case as in Fig. 3(b): — total; – – – track; - · - · - wheel. (a) Highly damped wheels and (b) effective bogie skirts.

nodal diameter [5]. A factor of 15 has been assumed for a highly efficient wheel damper like the GHH plate absorbers used for instance on the Norwegian Airport Train [8]. Using wheel-mounted brake discs has been shown to introduce some damping to the wheels; in TWINS analyses a damping multiplication factor 7 has been found typical for such wheel types. The effect of bogie skirts is derived from the scale model study in Ref. [9]. For the wheel noise, the skirt insertion loss is taken as 0 at 50 Hz, 0.5 dB at 400 Hz, 5 dB at 10 kHz and with linear interpolation in-between. This particular design is believed to be an extremely good bogie screen with a complete coverage all the way down to the wheel axle height. The bogie skirts are assumed to have negligible influence on the track noise.

Fig. 6 shows the influence of wheel damping and bogie skirts at train speed 80 km/h for TSI-CR rail roughness and 'good' wheel roughness. Comparison with Fig. 3(b) shows that the introduction of wheel damping gives a reduction of 3.5–4 dB in wheel noise but a total reduction of only 1–1.5 dB. It should be observed that this comparison is made between *undamped* wheels and *fully damped* wheels. Considering the widespread use of wheels with cheek-mounted brake discs it is also relevant to mention that the reduction obtained when switching from such wheels to fully damped wheels is 1.5–2 dB on wheel noise and 0.5–1 dB on the total noise.

Similarly, the bogie skirts give a reduction in wheel noise of 2.5 dB but a total noise reduction of 0.5–1 dB. Calculations for higher speeds would show that even at 200 km/h the noise reduction is limited to a meagre 1–2 dB. For medium-to-soft track it is also evident that further improvements in wheel damping and bogie skirt design (or a combination of the two measures) would be futile as the track now dominates strongly.

It is now interesting to switch to the operational conditions and consider the effectiveness of the wheel dampers and bogie skirts. The TSI track and 'good' wheel roughness spectra behind the results in Fig. 3(b) have been used for Fig. 7(a), while the operational track and wheel from Fig. 5(a) are used for Fig. 7(b). The total effect of the two measures and also the effect of the two together is displayed in Fig. 7. Assuming that the operational track may very well have more unfavourable decay rates than the TSI + track (equivalent to a lower pad stiffness in this analysis), what is seen as a 1–2 dB reduction on the type test track (Fig. 7(a)) may in reality mean a 0.5–1 dB reduction on operational track (Fig. 7(b)). When looking at the noise reduction potential it is also worth pointing out that the efficiencies of the wheel dampers and bogie skirts in this study are at the very high end of what is practically achievable.

There is at present a proposal to introduce a second step of long-term limit values in the TSI legislation on vehicle noise emission. The intention is to foster the development of lower noise levels. The present study, however, clearly displays that only marginal reductions can be achieved unless measures are taken on the infrastructure. A very clear illustration of this is to study Fig. 3(b). By completely removing the noise radiation from the vehicle the total pass-by noise is only brought down 2 dB on a track with TSI + equivalent pad stiffness (400 MN/m).

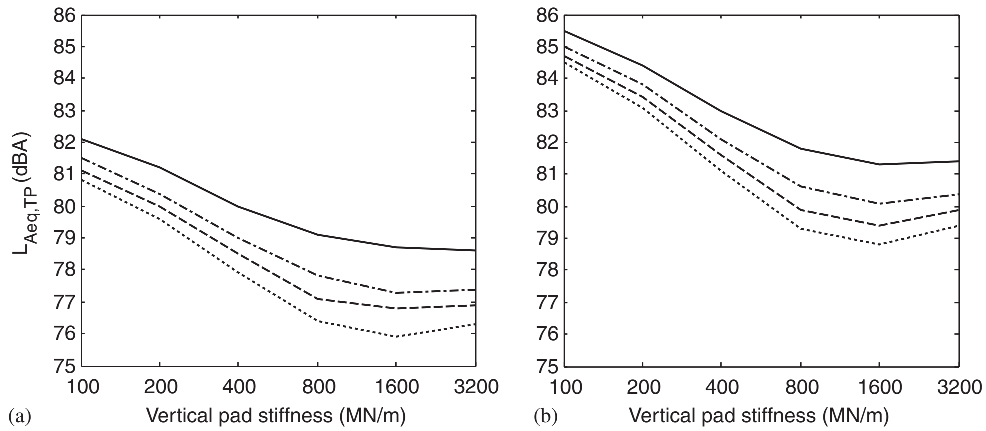

Fig. 7. Calculated total pass-by sound pressure level at 80 km/h for different combinations of noise control measures: — baseline vehicle with undamped wheels; — — — highly damped wheels; - - - - - efficient bogie skirts; · · · · damped wheels and bogie skirts. (a) TSI-CR type test roughness conditions and (b) operational roughness conditions.

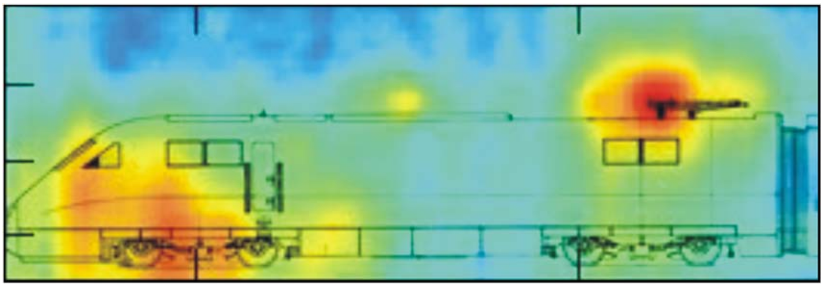

Fig. 8. Sound pressure distribution for an ICE-S power head at 290 km/h derived from a Bombardier 96 microphone array measurement.

3.3. Aeroacoustic noise—efficiencies of noise control options

The relative importance of aeroacoustic sources is dependent on the level of excitation for rolling noise, namely the wheel roughness and the rail roughness. For a rail–wheel combination with low roughness, such as type testing on a reference track with a new vehicle, the aeroacoustic sources may have to be reduced to achieve a low total level. As shown in the previous section, normal operation will often produce rolling noise levels that are considerably higher than on a reference track with a new vehicle. Hence, the design changes giving reduced levels from aeroacoustic sources may never be heard or even not clearly measurable in normal operation.

The example below is extrapolated from the two-car EMU of the analysis in the previous section. The aerodynamic source strengths were adjusted (together with rolling noise) to the measured wayside sound pressure levels from pass-by tests with high-speed trains at speeds of 250–320 km/h. As expected, a speed exponent of 6 was found over a wide frequency range for the aerodynamic sources. In the model, these sources have been placed at both high and low positions, typically representing pantographs and bogie cut-outs. The dominance of these two aeroacoustic sources is visualized in Fig. 8, which shows the result from a test recently performed on an ICE-S trainset running at 290 km/h measured with a 96 microphone array developed at Bombardier Transportation. Fig. 9 shows the calculated pass-by noise levels when running at 320 km/h. For the TSI-CR track, Fig. 9(a), the aeroacoustic noise now dominates whereas for the operational track, Fig. 9(b), rolling noise still dominates. The effect of a hypothetical 3 dB reduction of the aeroacoustics contribution at this speed would mean a 1 dB total reduction on TSI-CR (type test) track and 0.5 dB on operational track. Note that the pad stiffness has little influence on the total noise at this speed compared with the results at lower speeds in the previous section.

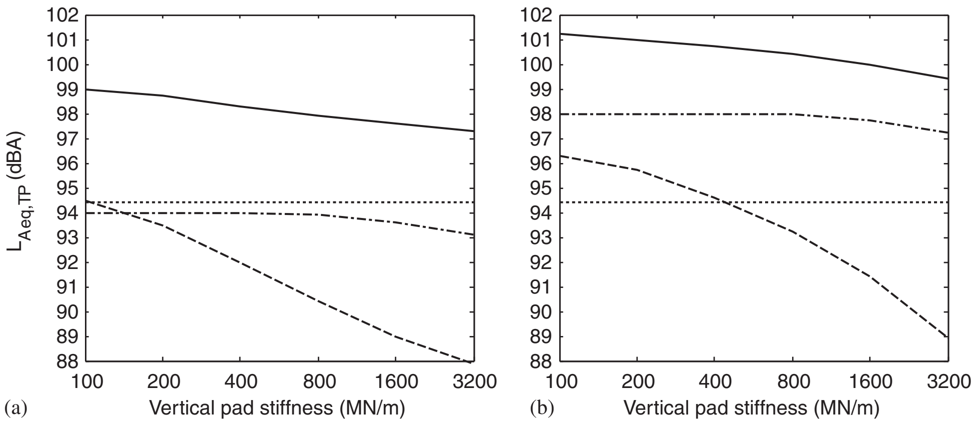

Fig. 9. Calculated total pass-by sound pressure level at 320 km/h: — total; — — — track; - · - · - wheel; · · · · aeroacoustics. (a) TSI-CR type test roughness conditions and (b) operational roughness conditions.

3.4. Feasible wavelengths for rail roughness limit specification

In standards and proposals for new legislation, different limits for rail roughness have been put forward. Apart from the roughness spectrum limit values, another important question concerns the range of wavelengths to be included in these spectra. First of all the range should cover the wavelengths corresponding to the frequencies where rolling noise is the dominant noise source. Another important issue is that the roughness can be measured with acceptable accuracy. If too wide a range is chosen there is a big risk of disqualifying proper test tracks whereas a too narrow range could result in problems with reproducibility. In TSI-HS the long wavelength limit is 200 mm and in the proposed TSI-CR it is 100 mm. The corresponding frequency excited by these wavelengths at the respective TSI-HS and TSI-CR maximum speeds is roughly the same: 500 Hz. Below this frequency there should be negligible rolling noise contribution unless the track is equipped with very stiff pads.

The measurement accuracy of the roughness spectrum can be analysed in the same way as in conventional frequency analysis by exchanging frequency, f, with wavenumber, $k = 1/\lambda$, and measurement time, T, with measurement length, L. From textbooks such as Newland [10] the ratio of the standard deviation, σ, to the mean, m, can be calculated as

$$\frac{\sigma}{m} = \frac{1}{\sqrt{B_e L}}, \tag{1}$$

where B_e is the effective bandwidth. In order to get reliable results the product $B_e L$ should be much greater than unity. To achieve a confidence of 90% that the measured value should not deviate more than ± 2 dB it is required that $B_e L > 50$.

In the draft prEN ISO3095 the method for rail roughness measurement is standardized and states that the roughness should be measured over a length of at least 1 m in 12 different positions along both rails. If the running surface on the railhead is wide enough three parallel traces should be measured. The estimation of the roughness spectrum could thus be made on as much as 36 records each 1 m long. An optimistic estimate of the measurement distance could thus be $L = 36$ m. With this length a bandwidth is required of $B_e > 1.4\,\mathrm{m}^{-1}$. For a one-third octave band spectrum the bandwidth is 23% of the centre wavenumber requiring the wavenumber to be at least $6\,\mathrm{m}^{-1}$ and the corresponding wavelength should be less than 160 mm. The conclusion is that with 90% confidence a measurement of rail roughness one-third octave spectrum will have the following uncertainties: 200 mm: ± 2.2 dB; 100 mm: ± 1.6 dB; 50 mm: ± 1.1 dB.

Table 1
Summations of total sound power in a typical case with one fan and a compressor at standstill condition

	Fan full, L_{wA}	Fan half, L_{wA}	Compressor, L_{wA}	Total, L_{wA}	Reduction, L_{wA}
Baseline case	90	–	80	90.4	–
	–	75	80	81.2	–
Case 1: Fan reduced	85	–	80	86.2	4.2 (“extreme”)
	–	70	80	80.4	0.8 (“normal”)
Case 2: Compressor reduced	90	–	75	90.1	0.3 (“extreme”)
	–	75	75	78.0	3.2 (“normal”)

The fan can operate at full or half-speed. The figures in the column “reduction” are obtained by comparing with the baseline cases. The phrases “extreme” and “normal” refer to how often these conditions occur in service.

4. Operating conditions at standstill and acceleration

Noise emission from vehicles at standstill and acceleration will be included in the TSI-CR. Under these conditions the noise from cooling systems for propulsion equipment and heating, ventilation and air-conditioning (HVAC) units is important. An analogous situation to comparing type test conditions with operational conditions for pass-by noise can be said to exist for the standstill and acceleration cases. The noise from cooling fans depends heavily on their rotational speed. If the noise requirements prescribe that all fans should run at full speed then the vehicle will be optimized for an *extreme* situation occurring only for a fraction of the train’s service time whereas the noise at more normal operating conditions is unchanged. This is demonstrated in the following example.

Two sources are assumed to be present at standstill: (1) a noisy cooling fan that can run at full/half- speed, (2) a compressor (or any arbitrary source) that has a single operating condition. The sound power for the sources and the total sum of them are listed in Table 1 as “baseline case”. In the same table is also presented the effect on the total noise from reducing either of the two sources by 5 dB. It is found that the noise reduction of the fan has a strong effect for the extreme full-speed case whereas for the “normal” case, it is more effective to make the reduction on the compressor.

Several measures are known to reduce the noise emission from fans: improved inlet/outlet flow conditions, blade shape optimization, optimization of tip clearance, etc. One of the most effective measures is to reduce the blade tip speed, with given constraints on required air flow, as the fan noise is proportional to the tip speed raised to the power of five [11]. Hence the difference between full-fan speed and half-fan speed is 15 dB. One obvious noise reduction strategy is only to run the fans at the minimum possible rotational speed required by the cooling demand. The role of the thermal control system therefore is important. Fans with continuous speed control thus have a higher potential for a “minimum cooling demand” strategy than the conventional fans with half/full-speed switch and would result in a lower average blade tip speed and hence reduced noise. An intelligent control system could also increase the cooling power at uninhabited sections of the track in order to be able to relax the cooling power when approaching stations and at platforms. This will be temporarily allowed by the thermal inertia of the cooled down systems.

5. Conclusions

Currently the competence and capability exist to design and build low-noise trains if the associated costs are accepted. Ideally the same low-noise trains should also reduce the noise annoyance from railways for the population but it is not necessarily the case with existing legislation because track input may dominate. A system approach including both rolling stock and infrastructure and finding a proper balance between the two is crucial to achieve further progress beyond the introduction of composite blocks for freight wagons. This is well-known to be the obvious first step to take in tackling the railway noise problem on a European level and is not treated in this paper.

Reducing only the vehicle noise part will normally not mean much on the total level due to a high contribution from the track noise part. The results from TWINS calculations and tests that form the basis of this paper show that the introduction of noise control means on the vehicle such as wheel absorbers and bogie skirts will mostly give a reduction in total levels of 1–2 dB on TSI-CR type test track but only 0.5–1 dB on operational track. This should be compared with the accuracy of pass-by noise measurements that is in practice 1–2 dB. If, however, the track noise is reduced, for example by introducing rail dampers, a significant effect on the total levels would result and the full benefit of a low-noise vehicle will show up.

Taking into account the practically achievable measurement accuracy for the roughness of the rail is another important factor. The uncertainty in a roughness measurement can be considerable especially for long wavelengths. This may lead to a disqualification of a proper test track or the acceptance of an improper track. If the requirement is to achieve a confidence of 90% that the measured value does not deviate more than ± 2 dB, then wavelengths longer than 160 mm cannot be used unless the measurement length is increased.

To reduce noise radiation during standstill and acceleration, fan noise and engine noise should be tackled. Setting the requirements for a normal operation condition as in the TSI-CR proposal instead of a rarely occurring full load condition will be a more effective way to reduce the noise disturbance. A low level at full load condition does not necessarily mean that the noise is also reduced at a normal condition. An example has been shown where improper setting of operational conditions to full load instead of normal could direct the noise control efforts to the wrong source.

Appendix—exterior noise limit values for TSI-CR

Table 2 lists the exterior noise limit values proposed for the TSI-CR. The columns labelled ‘‘ordinary’’ apply for all new vehicle designs approved after the TSI-CR has entered into force. The columns labelled ‘‘transition 0–2 years’’ apply to options of additional vehicles within existing contracts or new contracts with vehicles of existing design during this period. The column labelled ‘‘transition 0–5 years’’ applies strictly to DMUs with engine power greater than 500 kW and is valid regardless of whether it is new or an existing design.

The pass-by noise shall be measured at the train's maximum service speed (but not higher than 190 km/h) and at 80 km/h. The microphone shall be positioned 7.5 m from the track centre and 1.2 m above the railhead.

For stationary and starting noise the measurement positions specified in prEN ISO3095 shall be used (7.5 m from the vehicle centreline). For stationary noise the energy average of all microphone positions shall be taken and for starting noise the maximum of the microphones. The cooling fans shall in both cases operate at their normal condition at an ambient temperature of 20 °C.

Table 2
Noise limit values for TSI-CR. Values in parentheses are exceptions (‘‘specific case’’) for the UK and Irish network

Operating condition	Pass-by ($L_{pAeq,TP}$)		Standstill ($L_{pAeq,60s}$)		Starting (L_{pAFmax})		
	Ordinary	Transition 0–2 years	Ordinary	Transition 0–2 years	Ordinary	Transition 0–2 years	Transition 0–5 years
New wagons (APM < 0.15)	82	84	65	67	–	–	–
Upgraded wagons (APM < 0.15)	84	86	–	–	–	–	–
New wagons (0.15 < APM < 0.275)	83	85	65	67	–	–	–
Upgraded wagons (0.15 < APM < 0.275)	85	87	–	–	–	–	–
New wagons (APM > 0.275)	85	87	65	67	–	–	–
Upgraded wagons (APM > 0.275)	87	89	–	–	–	–	–
Electric locos < 4.5 kW	85	87	75	77	82 (84)	84 (86)	–
Electric locos ⩾ 4.5 kW	85	87	75	77	85	87	
Diesel locos < 2 kW	85	87	75	77	86 (89)	88 (91)	–
Diesel locos ⩾ 2 kW	85	87	75	77	89	91	
EMUs	81	83	68	70	82	84	–
DMUs < 500 kW/engine	82	84	73 (77)	75 (79)	83 (85)	85 (87)	–
DMUs ⩾ 500 kW/engine	82	84	73 (77)	75 (79)	85	–	87
Passenger coaches	80	82	65	67	–	–	–

The abbreviation APM stands for "axles per metre" and is used for freight wagons. The number of wheel axles shall be divided by the length of the wagon between buffers. The values in parentheses apply for vehicles specifically built to operate on the UK and Irish networks. The reason for this exception is the narrow loading gauge and the consequences this space limitation has on noise control measures. Levels for freight wagons are not valid for Finland, Sweden, Estonia, Latvia and Lithuania.

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