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OPTIMIZATION OF RESILIENT WHEELS FOR ROLLING NOISE CONTROL

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Resilient wheels are currently used on light rail systems such as tramways to prevent squealing noise and to reduce impact noise. On the other hand, they are rarely found on main lines (passenger rolling stock and freight rolling stock). Although manufacturers often claim that resilient wheels are favourable for rolling noise control, no extensive theoretical investigation confirming this statement has been published to date. In this paper, it is shown how resilient wheels can be effectively optimised in order to reduce rolling noise emission, compared to a conventional monobloc wheel. A preliminary analysis of the physical phenomena accounting for rolling noise generation emphasizes the key design parameters affecting both wheel and radiation. These parameters are the radial dynamic stiffness and damping loss factor of the rubber layer. The tread mass is also relevant. The influence of these design parameters is then qualified by a parametric study performed with the TWINS software. An optimum radial dynamic stiffness of the resilient layer is found which depends on operating conditions. Reductions in overall rolling noise up to 3 dB (A) are calculated for the configurations investigated. However, poor selection of the design parameters can lead to a noise increase compared to a standard monobloc wheel. It is also shown that a proper design for rolling noise control will not affect wheel efficiency with regard to squeal noise.

1. INTRODUCTION

Resilient wheels are characterized by inserting a rubber layer between the web and the tread. They were initially designed to prevent or reduce squealing noise emission in tight curves and dynamic stress levels on unsprung masses. Although these wheels are often claimed to be efficient for rolling noise reduction, no theoretical investigation confirming this statement has been published to date. The purpose of this paper is to show how resilient wheels can be effectively optimized in order to reduce rolling noise emission, compared with a conventional monobloc wheel.

Initially, the potential positive effects on noise are investigated; the ability of the main wheel design parameters to reduce wheel and track radiation is discussed. These parameters are the dynamic stiffness and damping loss factor of the rubber, and to a lesser extent, the tread mass.

Secondly, the method for modelling the resilient wheel is discussed. This method combines finite element calculations, to derive the wheel modal basis, and TWINS† calculations, to estimate the effect on the overall radiated noise. Parametric studies are carried out to quantify the effect of variations in wheel parameters on radiated noise, and to derive the optimum wheel parameters. Several configurations of operating conditions (track type, roughness spectra) are considered.

Finally, the compatibility between a rolling noise design and a squeal noise design is discussed.

The results presented in this paper concern freight rolling stock (920 mm wheel diameter). However, similar results have been obtained with tramway wheels, characterized by a smaller diameter (660 mm).

2. RESILIENT WHEELS: MECHANISMS FOR ROLLING NOISE REDUCTION

Three positive effects on rolling noise control can be expected from the resilient wheel concept: uncoupling of the web and tread leads to a reduction of web vibration and radiation; damping reduces vibration levels on the whole wheel; reduction in mechanical impedance can reduce track vibration and radiation. These positive effects can be partially cancelled out by a fourth effect related to the increase in the number of wheel modes liable to be excited during rolling.

2.1. ROLLING NOISE MECHANISM

Wheel and rail roughness introduces a vertical relative displacement between wheel and rail. The extent to which wheel or rail moves in response to this relative displacement depends on their respective mechanical receptances in the vertical direction. These receptances (calculated with TWINS) are compared in Figure 1 for a freight configuration (UIC60 rail with bi-bloc concrete sleepers, SNCF 9054 type freight wheel).

Below 1600 Hz, the rail vertical receptance is much higher than for the wheel: the relative displacement is mainly converted into rail vibration, the wheel vibration remaining very low. Consequently, with conventional freight wheels, the track turns out to be the major radiator over the 100–1600 Hz range (Figure 2), with a high contribution in the 800–1000 Hz one-third octave bands. The wheel response is essentially related to excitation of its natural frequencies with a significant radial amplitude at the contact point. For conventional freight wheels, these modes generally occur at frequencies higher than 1600 Hz (radial modes and

† Track Wheel Interaction Noise Software, developed and validated with the support of the European Rail Research Institute, ERRI.

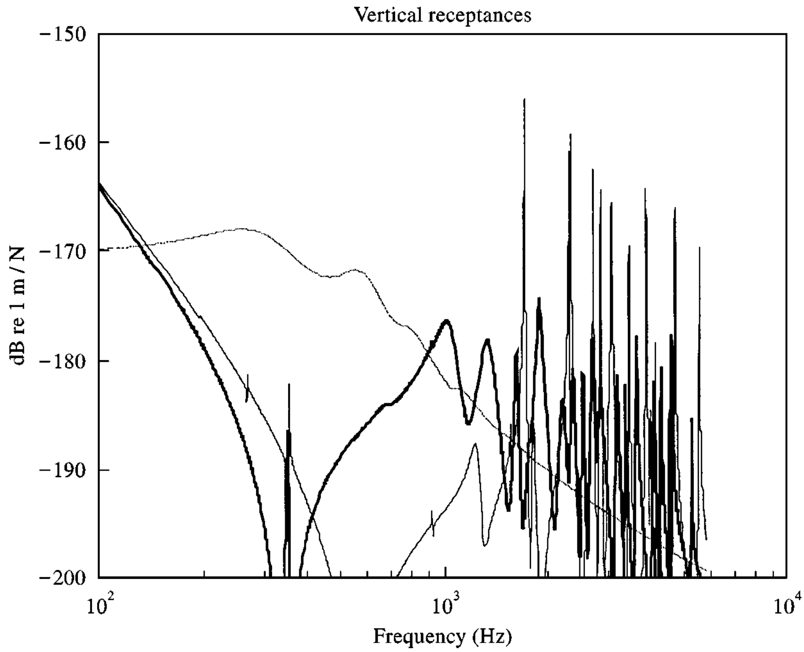

Figure 1. Vertical wheel, rail and contact receptances calculated with TWINS wheel: SNCF 9054 type (920 mm freight wheel) track: UIC60 rail on bibloc or monobloc concrete sleeper. – –, rail; ——, monobolic wheel; ———, resilient wheel 800 Hz.

1-axial modes). In terms of wheel radiation, radiation from the web (related to axial vibration) is usually higher than from the tread (Figure 2).

2.2. DISCUSSION OF THE MECHANISM FOR ROLLING NOISE REDUCTION

2.2.1. *Uncoupling effect*

On conventional freight wheels, the major part of the wheel sound power is radiated by the web. Therefore, uncoupling the tread from the web, at a frequency f_0 below the radiating frequency range 1800–5000 Hz, is liable to suppress almost all the web vibration and radiation. To take advantage of this uncoupling effect, the f_0 frequency should be set below about 1200 Hz (tread uncoupling in the radial direction).

2.2.2. *Damping effect*

The damping effect is related to the damping properties of the rubber, which is much higher than those of steel (rubber loss factor η_e ranging between 0·1 and 0·4, compared with 10^{-4} for the steel). In order to maximize the modal damping values η_m for those modes contributing to noise emission (modes over the 1800–3000 Hz for a speed of 100 km/h), the frequency f_0 should be set above 1800 Hz. This involves a rubber radial stiffness about three times the value required to optimize

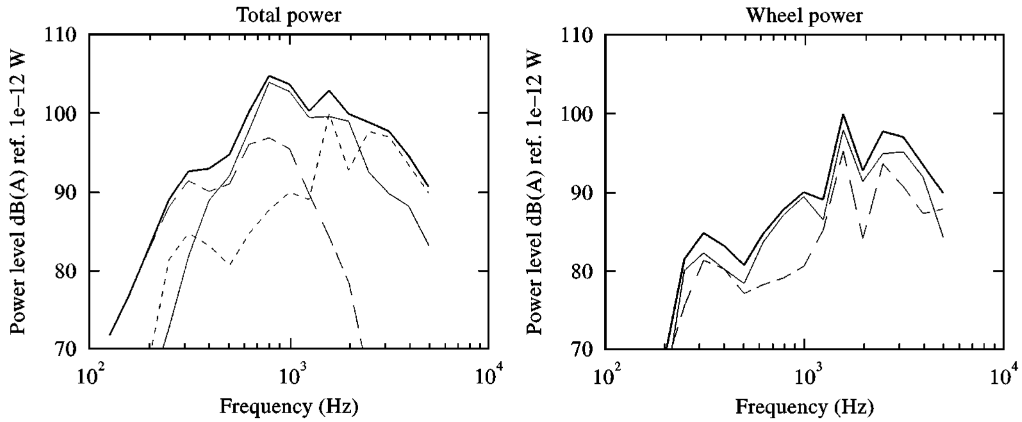

Figure 2. From TWINS calculations. Bibloc track with UIC60 rail, 100 km/h. Left: respective track, wheel and sleeper weight in overall radiated noise, Right: wheel total axial and radial radiation. (a) ——, total: 111·1 dB (A); ——, rail: 109·1 dB (A); – –, sleep: 102.6 dB (A); - -, wheel: 104·7 dB(A); (b) ——, wheel total: 104·7 dB (A); ——, wheel axial: 102·6 dB (A); – –, wheel radial: 99·6 dB (A).

the uncoupling effect. Consequently, a compromise between these two effects should be found.

2.2.3. *Reduction of mechanical impedance effect* (*or receptance increase effect*)

The uncoupling of the wheel tread at frequencies around 600–1200 Hz will induce a significant increase in mechanical receptance of the wheel at the contact point (Figure 1) and may lead to a reduction in track vibration and radiation. In other words, a resilient wheel may act as a *vibration absorber* with regard to the rail. The roughness amplitude tends to be converted into a wheel vibration rather than a rail vibration. However, whether the reduction in track noise is compensated by an equivalent rise in wheel radiation (see the next Section) remains to be determined.

3.2.4. *Increased tread radiation effect*

The uncoupling of the tread by means of a resilient layer will increase the number of radial modes of the tread; the first radial mode will occur at frequencies lower than 1800 Hz (typical first radial mode of a Ø 920 mm conventional monobloc wheel) and a higher modal density will be observed on the 0–5000 Hz.

2.3. SYNOPSIS

This first analysis shows that the resilient wheel is a promising concept for rolling noise reduction. In the following section, parametric studies are performed to derive the best compromise between the different positive and negative effects of the resilient wheel. It must be noted that resilient wheels can be designed to have an effect on both wheel and track radiation.

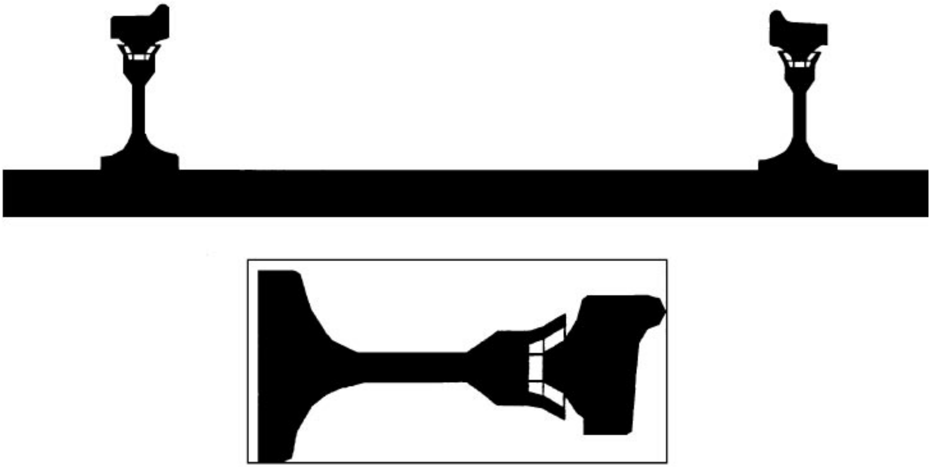

Figure 3. Mesh of a resilent wheel: upper: including the axle for 1 nodal diameter modes; lower: with a clamped hub for $n = 0$ and $n \geqslant 2$ modes.

TABLE 1

Resonance frequency of R1 mode, influence of coupling to the axle and corresponding radial stiffness of the rubber ring

Radial uncoupling frequency (R1 mode) (Hz)					
With a clamped hub (without axle)	230	480	700	810	910
With coupling to the axle	286	600	860	1020	1100
Resilent layer radial dynamic stiffness (N/m)	$4{\cdot}3 \times 10^8$	19×10^8	40×10^8	54×10^8	68×10^8

3. MODELLING OF THE WHEELS

The parametric studies have been performed with the TWINS software [1–5]. Prior to the noise calculation in TWINS itself, the modal behaviour of the wheel is required (resonance frequencies and associated modal shapes, masses and damping). These data are usually produced with a finite element (FE) package [5]. The basic rules to follow in performing the FE analysis of resilient wheels are briefly discussed in the following section. The wheel and track parameters used for the calculations are then discussed.

3.1. RESILIENT WHEEL MODELLING

The wheel design is shown Figure 3. It is derived from a monobloc wheel with a straight web. The resilient layer is added by cutting the web near the tread. The wheel is meshed with axisymmetric solid elements. The resilient layer can be modelled with shell axisymmetric elements, or with solid elements; the first method allows for a separate tuning of axial and radial stiffnesses of the resilient layer, which is suitable for a pre-design study, while the latter is more representative of an actual wheel design.

A FE model of the wheel without the axle and with a clamped hub is used to calculate the modal parameters for nodal diameters $n = 0$ and $n \geqslant 2$. However, the prediction of the resonance frequencies and modal mass of 1-nodal diameter modes is poor, as the coupling with the axle is not taken into account in such a model. Therefore, the $n = 1$ modes are derived from a FE model including the two wheels and the axle (see Figure 3 and Table 1).

The damping η_m of the m-th mode is estimated by the relation (see reference [6] p. 459)

$$\eta_m = (W_e/W)\eta_e$$

where W_e, η_e are the elastic energy and damping loss factor of the resilient layer and W the elastic energy of the whole wheel (resilient layer + steel).

3.2. PARAMETERS USED FOR CALCULATIONS

3.2.1. *Wheel parameters*

The relevant parameters for the uncoupling of the tread are its mass (uncoupled mass), the dynamic stiffness and the loss factor of the rubber. For a given tread mass, the uncoupling can be characterized by the natural frequency of the $R1$ mode (radial mode with 1 nodal diameter), for which the tread moves in the radial direction, in out of phase with the web and the axle (Figure 4). The resilient wheel used for computations has a mass of 386 kg (new wheel profile) and a diameter of 920 mm.

The set of parameters considered are shown in Table 2.

In the following sections, the *tuning frequency* will refer to the one with a clamped hub. Two damping loss factors are considered: $\eta = 0{\cdot}2$ and $0{\cdot}3$.

The influence of the mass of the tread has not been studied, as it is not a parameter compatible with operational constraints: the shape of a new tread must usually allow 25 mm wear, so that its initial design cannot be extensively modified.

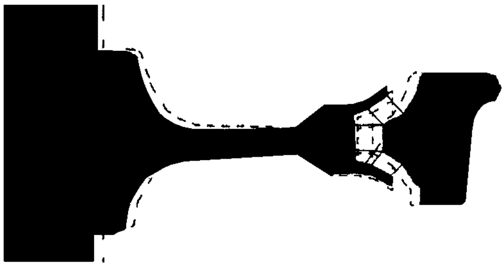

Figure 4. R1 modal shape.

TABLE 2

Optimum resilient wheel parameters depending on the track type 100 *km/h—rubber loss factor*: 0·2—*tread braked roughness spectrum. Parentheses*: *disc-braked roughness spectrum*

	Track 1 (soft rail pad)	Track 2 (medium rail pad stiffness)	Track 3 (stiff rail pad)
Optimum uncoupling frequency (Hz)	700 (800)	800 (900)	900 (900)
Total sound reduction (dB (A))	2.9 (2·6)	2·7 (3·1)	2·8 (3·6)
Track sound reduction (dB (A))	3 (2·1)	2·3 (1·5)	1·6 (1·26)
Wheel sound reduction (dB (A))	1·6 (6·2)	4·5 (7·7)	6·2 (7·8)

However, a reduction in tread mass leads to an increase in wheel receptance, which favours track noise reduction.

3.2.2. *Track parameters*

As it is expected that the resilient wheel will have an effect on track radiation, three track configurations have been considered in the calculations. They are based on a ballasted track, with UIC 60 rail on monobloc concrete sleepers, but with different rail pad dynamic stiffnesses K in the vertical direction:

— a soft rail pad: $K_{vertical} = 80$ MN/m,
— a medium rail pad stiffness: $K_{vertical} = 400$ MN/m,
— a rather high rail pad stiffness: $K_{vertical} = 800$ MN/m.

This parameter accounts for the decay rates of the wave travelling along the rail: an increase in rail pad dynamic stiffness leads to an increase in waves decay rates, and hence reduces the effective radiating length of the rail [7].

These tracks are described in TWINS using a model of a Timoshenko beam on continuous elastic foundations, which gives a reasonable representation of the average rail vibration, although it does not include the specific effects of the pinned–pinned resonance and of the rail cross-section deformations.

3.2.3. *Other parameters*

A rolling speed of 100 km/h has been considered for the calculation. Two combined wheel rail roughness spectra have been used, relevant for tread- and disc-braked wheels (see Figure 5). The contact patch filtering effect is taken from reference [8].

4. RESULTS OF PARAMETRIC STUDIES

4.1. EFFECT OF THE UNCOUPLING FREQUENCY (RADIAL STIFFNESS)

The calculations are first performed with a rubber damping loss factor of 0·2, a medium rail-pad stiffness ($K_v = 400$ MN/m), and the tread braked roughness

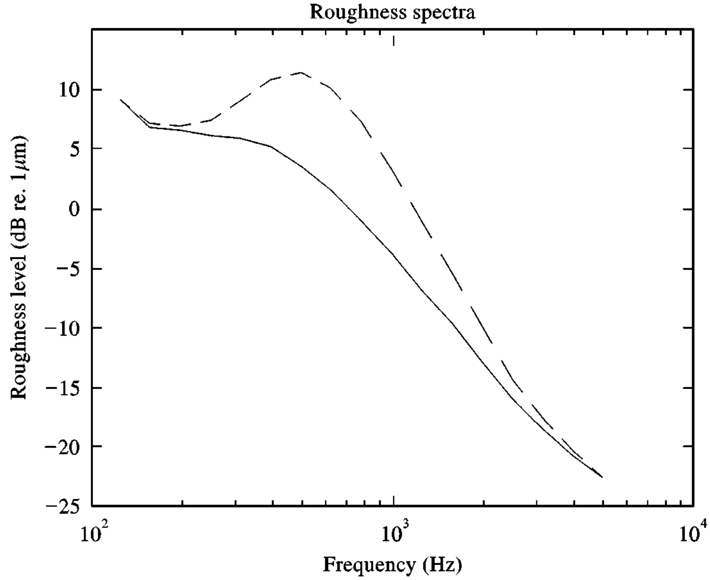

Figure 5. Wheel-rail roughness spectra, including the contact filtering effect. ——, disc braked; – –, tread braked.

spectrum. Figure 6 summarizes the effect of the uncoupling frequency on the wheel noise, the track noise and the overall noise.

4.1.1. *Wheel radiation*

Compared with the reference monobloc wheel, a low-tread uncoupling frequency (230 Hz) leads to an increase of around 2 dB (A) in wheel radiated power.

Then, as the uncoupling frequency increases, wheel radiation is reduced (since modal damping increases); for an uncoupling frequency of 900 Hz, a wheel power reduction of about 6 dB (A) can be obtained compared with the reference freight wheel.

It is likely that, in terms of *wheel sound power reduction*, the uncoupling frequency at 900 Hz is not optimal and that further reduction could be obtained by increasing the uncoupling frequency. However, this may not be realistic as a very high stiffness value will then be required. Furthermore, an increase of the rubber loss factor from 0·2 to 0·3 brings an additional wheel sound reduction of approximately 1·5 dB (A).

4.1.2. *Effect on track radiation*

Track radiation is clearly affected by the resilient wheel, and an optimum is found for a tread uncoupling frequency of 800 Hz; in comparison with the reference freight wheel, a track sound power reduction of 2·3 dB (A) is achieved. This acoustic gain is directly related to the reduction in rail vertical component, which is initially

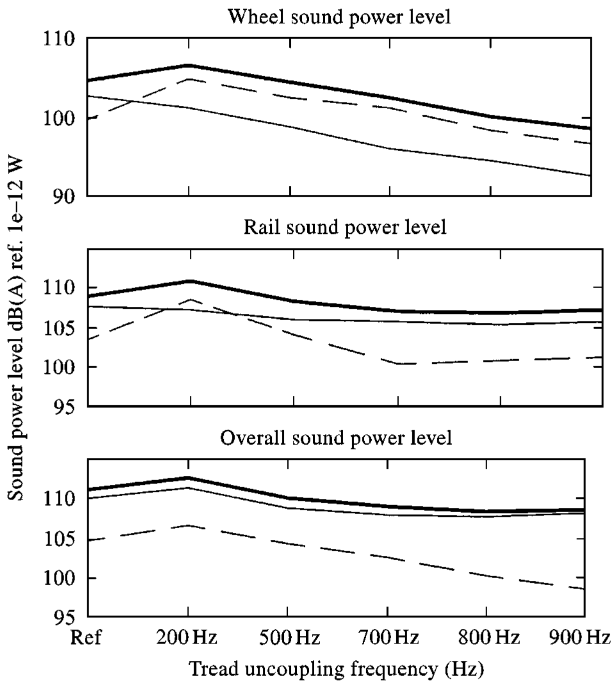

Figure 6. Effect of the tread radial uncoupling frequency: upper: upper: wheel radiated power (axial + radial); middle: rail radiated power (vertical + lateral); lower: overall noise (track + wheel). 100 km/h, medium rail pad stiffness, tread-braked roughness rubber damping loss factor: 0·2. upper: (a) ——, wheel total; ——, wheel radial; – –wheel axial. Middle: (b) ——, rail total; ——, rail vertical; – –, rail lateral. Lower: (c) ——, total power; ——, rail power; – –, wheel power.

maximum in the frequency range 800–1000 Hz (see Figure 2). Note that a low-tread uncoupling frequency (230 Hz) can lead to an increase of track radiation.

4.1.3. *Effect on overall noise*

In terms of overall sound power (track + wheel), the same optimum tuning frequency is found (800 Hz), as the track is the major contributor to overall noise: the resilient wheel with an uncoupling frequency of 800 Hz brings about a track noise reduction of 2·3 dB (A) and a wheel noise reduction of 4·5 dB (A), leading to an overall noise reduction of 2·7 dB (A).

Once this optimum is achieved, the track is the major acoustic source; its contribution to the overall noise is 6·6 dB (A) greater than that of the wheel.

Figure 7 presents a comparison of the overall radiated noise between the reference wheel and the optimum resilient wheel, and the contributions of each component (rail, sleeper and wheel) to the overall noise, for the optimum resilient wheel configuration.

These spectra confirm that the overall sound reduction of 2·7 dB (A) is mainly due to the effect of the resilient wheel on the track between 1000 and 2000 Hz. With the optimized resilient wheel, the rail is the major contributor to the overall noise over the 800–5000 Hz frequency range.

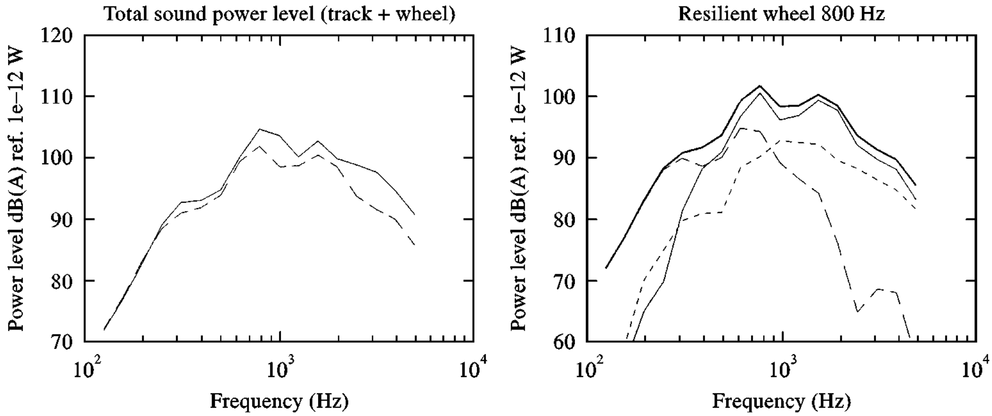

Figure 7. Overall noise—100 km/h — medium rail pad stiffness — tread braked roughnessleft; left: overall radiated noise—comparison of reference freight wheel and optimum resilient wheel. Right: overall radiated noise with optimum resilient wheel; contribution of sleeper, rail and wheel. (a) ——, track + ref. wheel 111·1 dB(A); – –, track + res. wheel 800 Hz: 108.4 dB(A). (b) ——, total: 108·4 dB(A); ——, rail: 106·8 dB(A); – –, sleeper: 100.6 dB(A); - -, wheel: 100·2 dB(A).

4.2. EFFECT OF THE TRACK TYPE AND THE ROUGHNESS SPECTRUM SHAPE

The track type can affect the relative weight of the track and the wheel in the overall noise. As the resilient wheels modify the track and the wheel radiation, differences in the optimum resilient wheel parameters can be found, depending on the track type. The results are shown in Table 2.

As the rail pad stiffness increases, the track weight in the overall noise decreases. Hence, the optimum tread uncoupling frequency increases with the rail pad stiffness: the track power reduction becomes lower, but the wheel power reduction increases (due to higher wheel modal damping). However, although the wheel and track sound reduction are not the same depending on the rail pad stiffness, the overall sound reduction is not affected by the rail pad stiffness variation (2·8 dB (A)).

Note that the optimum resilient wheel parameters are only slightly modified by a change in the roughness excitation spectra.

Figure 8 presents for each track configuration, the overall sound power levels obtained with the reference wheel and the optimum resilient wheels.

4.3. SQUEAL NOISE

The squeal noise is related to a stick-slip phenomena in narrow curves, which can lead to the excitation of wheel modes having an important axial modal amplitude at the wheel/rail contact point (mainly the 0-axial modes, $0Ln$). One of the most successful treatments for squeal noise is to damp the wheel. Table 3 gives the calculated modal loss factor of the $0Ln$ modes, for the resilient wheel tuned at 800 Hz and a loss factor of the resilient layer equal to 0·2. The axial stiffness of the rubber layer is approximately 10 times lower than the radial stiffness.

These modal damping are of the same order of magnitude than those of resilient wheels presented in reference [9], intended to suppress completely the squeal noise.

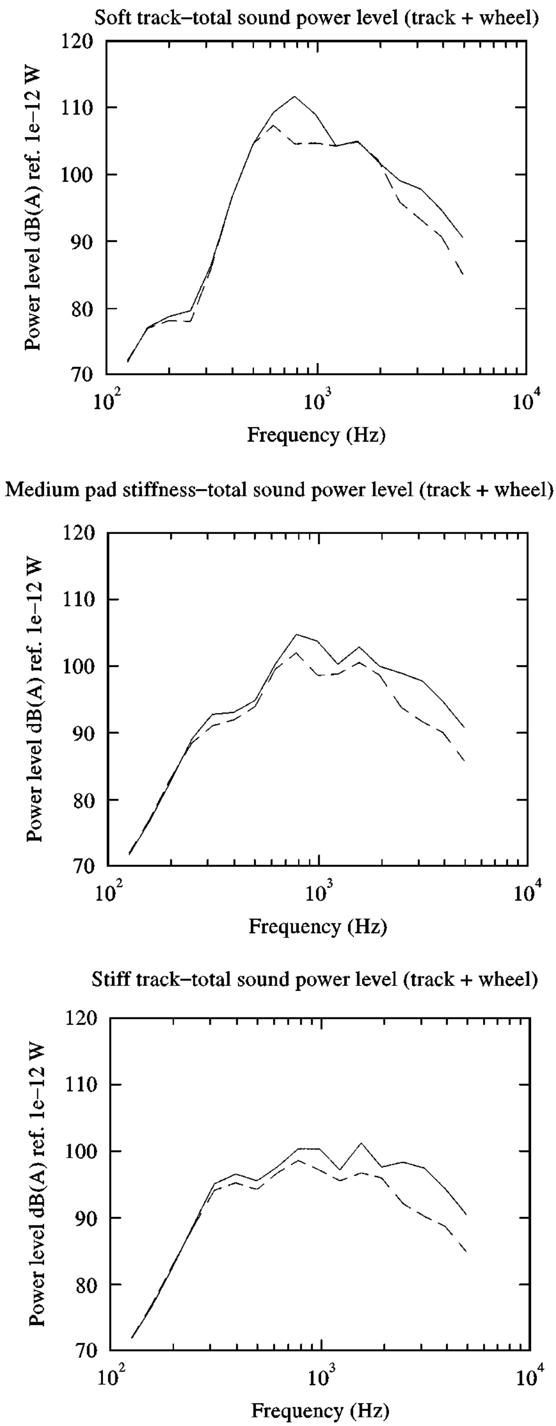

Figure 8. Effect of the resilient wheel on the overall noise, depending on the rail pad stiffness 100 km/h—tread-braked roughness. Upper: soft rail pad; middle: medium rail pad stiffness; lower: stiff rail pad. (a) ——, track + ref. wheel 116·5 dB (A); – –, track + res. wheel 700 Hz: 113.6 dB (A); (b) ——, track + ref. wheel: 111·1 dB (A); – –, track + res. wheel 800 Hz: 108.4 dB (A); (c) ——, track + ref. wheel: 109·0 db (A); – –, track + res. wheel 900 Hz: 106.3 dB (A).

TABLE 3

Modal loss factor of 0*ln modes—resilient wheel* 800 *Hz*

Mode	0*L*2	0*L*3	0*L*4	0*L*5	0*L*6
Frequency (Hz)	350	930	1630	2380	3200
Loss factor	0·009	0·004	0·017	0·012	0·007

Although the tread uncoupling frequency has been selected to minimize the rolling noise, the wheel design can also be efficient against squealing noise.

5. CONCLUSION

The resilient wheel concept remains a promising concept for rolling noise reduction, as significant wheel and track sound power can be achieved.

- In terms of overall sound reduction (track + wheel), the optimum tread radial uncoupling frequency is found to be between 700 asnd 900 Hz, depending on the track type and the shape of the excitation spectra. This optimum tuning frequency minimizes the track radiation (between 1·3 and 3 dB (A)), which is initially maximum in that frequency range. An overall sound reduction ranging from 2·6 to 3·6 dB (A) can be expected.
- For a tuning frequency of 800 Hz, the corresponding wheel sound power reduction is about 4·5 dB (A) with an elastomeric loss factor of 0·2. However, the optimum is not achieved in terms of wheel sound power reduction, and further reduction should be obtained:

— *by increasing the tread radial uncoupling frequency, which will increase the wheel modal damping*: an increase from 800 to 900 Hz leads to an additional wheel sound reduction of 1·7 dB (A) (from 4·5 to 6·2 dB (A)), and the optimum is still not achieved (probably around 1200 Hz). However, this increase of tuning frequency will lead to a lower track sound reduction and therefore to a lower overall sound reduction.

— *by increasing the elastomeric loss factor*: an increase from 0·2 to 0·3 can bring an additional wheel sound reduction of 1·5 dB (A).

Consequently, an overall rolling noise reduction exceeding 6 dB (A) could be obtained for cases where the wheel radiation exceeds the track radiation.

- The optimum wheel parameters do not depend strongly of the track type and the shape of the roughness spectra. Moreover, tread wear does not modify the wheel efficiency significantly. This indicates that the resilient wheel provides an interesting concept for noise control.

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