

VISCOELASTIC DAMPING TECHNOLOGIES FOR STRUCTURAL ACOUSTICS CONTROL OF RAILWAY WHEELS

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ABSTRACT

Nowadays increased levels of acoustic comfort in our everyday lives must be achieved in order to assure improved quality of human life. In view of this, vibration and structural acoustics control has become an important issue with a significant impact in transportation noise side effects, such as the ones originating from train wheels. This article addresses the use of viscoelastic damping technologies in a standard train wheel and presents a qualitative analysis of their effects on the vibroacoustic response and radiated sound power.

1 INTRODUCTION

Exposure to high levels of noise is often cited as a major factor that may contribute to people dissatisfaction, discomfort and health problems. Among the several sources of noise that we all are often exposed in our everyday life, one important source is related to transportation noise, in general, and to radiated noise from trains, in particular. Trains have been used as one of the more appellative and efficient solutions of transportation in big cities because they are less aggressive to the environment and general quality of human life. However, their side effects are not completely absent from the chaos and routines of big cities and with the ever increasing need of transportation of more people in less time, we are progressively being more and more affected by the exposure to noise radiated from trains and consequently by the underlying side effects, as we “passively” assist to the increasing of trains speed and volume traffic. Hence, it is important to determine the real acoustic level of sound radiation exposure that we

are dealing with and to develop ways of reducing noise radiation.

In general a train can be seen as composed by several sources of noise, be they of aerodynamic and/or mechanical nature. While the former is generated by unsteady airflow, generated for example over the nose, inter-carriage joints or roof-mounted equipment such as pantographs, the latter is generated at the wheel-rail contact, and is generally accepted to somewhat involve rolling noise, caused by vibrations of the wheel and track, impact noise, due to discontinuities in the wheel or rail surface, squeal noise, which occurs in sharp curves and is induced by unsteady friction forces at the wheel-rail contact, and ground-borne vibration and noise effects. Other mechanical sources of noise exist, though, which may suggest some concern, such as the intake and exhaust effects of the train engines that do not significantly depend on the vehicle speed depending instead more on the tractive effort required. Of special concern is also interior noise, which is transmitted

from each of these sources to the interior by both air-borne and structure-borne paths, which should be properly limited to assure the comfort of the passengers.

From a noise radiation viewpoint, a distinction is nowadays made between high speed trains (e.g. inter-city/region trains), used for passenger transportation across big distances and at a lower which may go above 300 km/h, and low speed strains (e.g. suburban trains and light rail vehicles, freight trains, trams and underground trains) which operate at smaller velocities. Among the different sources of noise, it is well known and accepted that aerodynamic noise effects become dominant, or at least as important, only above 300 km/h, where the track-wheel interaction effects start to become less significant. Below this speed the primary source of noise in low speed trains is the rolling noise. In order to tackle this problem, a considerable amount of research efforts have been devoted to the development and testing of noise reduction techniques (Thompson and Gautier 2006; Thompson et al. 2009). These techniques to reduce rolling noise are diverse and a subset is related with wheel-based solutions (Thompson and Jones 2006).

From the wheel-based solution perspective, an alternative is increasing the wheel damping. Railway wheels, without any damping treatment, usually behave as very lightly damped resonant bodies and efficient noise radiators. Various devices have been developed to increase the damping of railway wheels by absorbing energy from their vibrations and thereby reducing the noise produced (Thompson and Jones 2006). Constrained layer damping treatments, consisting of a thin layer of viscoelastic material applied to the wheel covered by a thin stiff constraining layer, have been successfully used to tackle severe curve squeal problems. However, by careful design, sufficient damping can be also achieved with viscoelastic damping technologies making significant reductions in rolling noise (Jones and Thompson 2000). Interior noise and vibration reduction using constrained-layer and free-layer damping treatments to luxury sleeper carriages has also been experimentally investigated by Fan et al. (2009), that have employed three new damping materials able to reduce the

internal vibration and noise and to provide a more comfortable travelling environment relative to motion and sound for the passengers. Due to its mechanical dissipative behaviour, viscoelastic materials have also been employed in track mini-barriers or vibration absorbers, as presented by Thompson and Gautier (2006).

This article addresses the use of passive viscoelastic damping technologies applied to train wheels. Thus, a constrained viscoelastic damping layer is considered mounted in the external side of the wheel. The damping solution is qualitatively assessed considering different sizes of external viscoelastic layers with a constant thickness and the potential of this damping technology is critically evaluated. After a brief physical description of the problem, a case study is performed. First, an uncoupled vibro-acoustic system is considered and the frequencies and mode shapes are determined. The mode shapes of the uncoupled system constitute a first analysis to envision the effect of the individual modes to the net sound radiation and might be used to assist the designer to decide a more appropriate location and extent of the viscoelastic damping treatment. The vibro-acoustic system, constituted by the train wheel, constrained viscoelastic layer damping treatment and acoustic media (air), is modelled by means of finite and infinite element technology using the commercial vibro-acoustics software ACTRAN/VA (FFT 2007). Lastly, the vibro-acoustic problem is solved and the harmonic forced response of the system with and without the damping treatment is assessed and critically analyzed.

2 PHYSICAL DESCRIPTION OF THE PROBLEM

2.1 Railway Wheel Noise Sources

Train wheel designs may have various shapes, geometries and sizes and also different types of materials and functional solutions. As obvious, these characteristics strongly affect the spatial and modal dynamics of the wheel and as a consequence the type and strength of noise generation mechanisms. Wheels are usually axisymmetric structures that, similarly to flat circular plates, have out-of-plane normal modes

of vibration which can be described in terms of the number of nodal diameters and nodal circles. Additionally the wheel (and disk) also has in-plane normal modes which usually lie well outside the frequency range of interest of the analysis of railway wheels.

As discussed by Barron (2003), there are four main contributions to rail and wheel noise generation induced by the wheel/rail contact interaction (e.g. rolling, impact and squeal noise) of low speed trains: the noise produced by (i) the rail roughness, (ii) the flat spots on the railroad car wheels, (iii) the gaps in the rail joints and (iv) the rubbing of the wheel flange and the rail. Rolling noise is broadband in nature and, compared with other sources, has been the subject of the greatest amount of research over the years (Thompson et al. 2009). In order to control rolling noise, and other types of noise, many solutions have been devised, somewhat playing with the geometry, shape, size and material of the wheel or by adding damping treatments to the structure or incorporating vibration absorbers (Jones and Thompson 2000; Cervello et al. 2001; Thompson and Gautier 2006). As far as rolling noise is concerned, noise control has been mainly performed by controlling the surface roughness and playing with the braking system, minimizing the vibration response of wheels and tracks by adding damping treatments, performing shape optimization or by introducing vibration isolation/absorbers, or by preventing sound radiation, for example, by using local shielding measures (Vincent 2000; Thompson and Jones 2006).

As previously referred, the most important mechanical noise sources from a train are generated at the wheels by the rail contact. Rolling noise is caused through vibrations of the wheel and track structures; it is induced at the wheel/rail contact point as a result of vertical irregularities in the wheel and rail surfaces and it is caused by the interaction between the wheel and the rail roughness. The static loading due to the vehicle causes local elastic deformation of the wheel and rail and as a result a contact occurs over a small area; another important variable to rolling noise is the contact patch between the wheel and the track.

The irregularities of the wheel and rail surface induce dynamic forces at the con-

tact region that excite the wheel and rail into vibration and as a result sound radiation occurs. When roughness wavelengths are short compared with the contact patch length, their effect on the rail system is attenuated. Thus, an alternative to reduce rolling noise is reducing the wheel diameter. Additionally, the level of the noise depends also on the wheel and rail mechanical receptances in the vertical direction and since waves are propagated by the rail, which works as an infinite waveguide, the distances between the sleepers also interfere with the level of sound radiation. Theoretical models accounting for these phenomena are well developed and have been extensively validated (Bouvet et al. 2000; Jones and Thompson 2000).

The friction between the wheel and the track produce an intense squealing noise and this noise is predominant in tight curves, where the solid bodies will be loaded together and the friction will be maximum among them. It is mostly induced by the unsteady friction forces at the wheel rail contact, the stick-slip phenomenon. The fundamental frequency of such squeal noise corresponds to a natural frequency of the wheel, generally at high frequencies. In Thompson and Jones (2006), it can be seen that the flange contact reduces the squeal noise, but generates a different form of noise with high frequency. In the same work, the authors present the main factors which influence the mechanism and the levels of squeal noise radiation; humidity, temperature, wheel and rail wear, train speed and the track geometry. Another type of noise generation occurs as a result of the ground-borne vibration, where the noise is caused by track and wheel irregularities and by the movement of the set of axle loads along the track (Olofsson and Lewis 2006).

2.2 Wheel Types and Design Approaches

Some modifications are generally used in the design of train wheels to help reducing noise. The so-called *resilient wheels* are designed to prevent or reduce squealing noise emission in tight curves (Bouvet et al. 2000; Cigada et al. 2008). They are characterized by inserting a rubber layer between the web and the tread, as a consequence of

the major part of the sound power being radiated by the web. These embody the wheel with a high degree of noise and vibration suppression capabilities especially in the domain of urban transport. Apart from the noise-reducing advantage, these wheels also considerably reduce wheel and rail wear and the overall life cycle costs. A good efficiency in the rolling noise reduction and an optimum study of this type of wheel with viscoelastic treatment is present in (Bouvet et al. 2000). The vibro-acoustic characterization of different kinds of wheels (both solid and resilient ones) has been carried out in (Cigada et al. 2008).

The use of elastomeric materials in resilient wheels has been shown to yield good damping efficiency as is the case when used in the solid wheel with viscoelastic layers. Jones and Thompson (2000) considered different rubbers to demonstrate that the increase of the damping reduce the sound composed by the sleeper, wheel and rail sound radiation. They have assumed that the normal modes and natural frequencies are little affected by the addition of damping layers, turning a finite element model for an undamped structure useful to study the coupled system. They present a damping treatment as a manner to reduce efficiently the sound power in higher frequency bands, but to lower frequencies, below 2000 Hz, where the wheel damping has little effect on the sound radiation, the treatment was not so efficient. They have analyzed the effectiveness of the damping treatment and the influence of the constraining layer stiffness in the noise control.

2.3 Viscoelastic Damping Technologies

In the aircraft industry, viscoelastic materials have been used for a long time and nowadays some special viscoelastic polymers with constraining layers are commercially available. In the past some techniques of squeal noise control were developed based in the reduction of the friction between wheel and rail with water and grease. Nowadays, to reduced carriage noise and the different sources of sound radiation, viscoelastic damping technologies have been suggested and frequently used because of their high damping efficiency, low weight and reduced cost. To reduce the roll-

ing noise, the use of the viscoelastic layers both inside and outside of the wheel is recommend, converting vibration energy into heat; the reduction in squeal noise can be as much as 30 dB, as referred in (Heathcote Industrial Plastics 2009). Regarding the temperature range over which the constrained layer damping technology should work, for the best noise reduction the polymer is optimized between 10 and 40°C but reasonable performance is maintained from -10 to 60°C.

In principle, the technology employed to control radiated noise must not interfere with the operation of the appliance or machine. Therefore, the use of surface mounted viscoelastic damping technologies on the structure is a good alternative to noise control. In recent years, with the development of new technologies, this kind of passive damping technology with viscoelastic materials has been properly used in the structure surfaces with a secure adhesion also supporting a large temperature range. These materials have large fabrication, versatility and can be easily used in many specific applications with different shapes and sizes.

3 CASE STUDY: RAILWAY WHEEL

3.1 Description of the System

In general, damping treatments can be applied in the external sides of the web and/or between the tread and the web of the wheel (Cervello et al. 2001). Attention here is devoted to the use of constrained layer damping on one side of the wheel and to the analysis of the vibration and structural acoustics responses in order to infer about their real behavior considering more realistic and complex load conditions involving high support “static” loads and impacts suffered by the wheel due to the track-wheel interaction.

A coupled vibro-acoustic finite element (FE) modeling approach using the software ACTRAN/VA (FFT 2007) is considered here and a FE model is created to analyze the influence a viscoelastic damping treatment applied in a standard 920 mm diameter cast iron train wheel would have in its vibro-acoustic response. Both mean square (MS) velocity frequency responses and the radia-

tion sound power of damped and undamped wheel are analyzed for non-rotating wheels excited by a distributed pressure. The wheel and constrained layer damping treatment are modeled with solid and solid-shell FEs, respectively. The near field fluid medium is modeled and discretized with 3D finite fluid FEs and in the boundary of the acoustic medium the far field medium is modeled through the use of infinite element technology available in ACTRAN/VA. For simplification a single standard train solid wheel isolated from the rest of the train is considered, the gyroscopic and centrifugal effects are neglected, only one side of the wheel is coupled with the acoustic medium, the mesh geometry of the wheel is somewhat simplified, isothermal conditions are assumed, clamped boundary conditions are considered in the inner cylindrical surface of the wheel (hub), the fluid medium is considered homogeneous, isotropic, inviscid and compressible and the solid elastic medium is also homogeneous and isotropic.

The study has been carried out on a standard 920 mm outer diameter solid wheel with a curved web, similar to the one considered by Cigada et al. (2008). Approximate dimensions and shape of the cross-section were extracted from a drawing of the wheel. The structural and fluid medium meshes are generated with the

complete system is modeled through FEs; consequently, some geometric adaptations are made in the primary format of the wheel to permit the generation of a uniform size element mesh.

The material of the solid wheel is cast iron and the material properties of both the solid and fluid medium (air) are presented in Table 1. The original design of the wheel and the plane mesh which is revolved to generate the solid railway wheel mesh is shown in the Figure 1. A coarse mesh with a limited number of FEs is used to reduce the computational cost. The solid mesh is generated through the revolution of the plane mesh of the cross-section, yielding a discretization of 50 elements in the circumferential direction (Figure 1); the clamped boundary condition of the inner cylindrical surface of the wheel is also represented in the figure. The solid FE used (HEX08) is a linear eight-noded element with three translation degrees of freedom (DoFs) per node (FFT 2007).

The solid wheel mesh is coupled with a semi-spherical fluid mesh with coincident nodes at the fluid-structure interface (Figure 2). The FE applied to build the fluid mesh is an acoustic element (HEX08) with eight nodes and one degree of freedom per node, which is pressure (FFT 2007). The FE mesh for the fluid consider an enclosed fluid,

Table 1. Material and geometric properties of the vibro-acoustic system.

STRUCTURAL MEDIUM						
	Young's Modulus / GPa	Poisson's ratio	Density / kg m ⁻³	Inner radius / mm	Outer radius / mm	Thickness / mm
Wheel	197	0.27	7200	89	460	135
Constraining layer	210	0.3	7850	(128,162)	400	2
Viscoelastic layer	Table 2	0.49	1140	(128,162)	400	1
ACOUSTIC MEDIUM						
	Speed of sound / m s ⁻¹			Density / kg m ⁻³		
Air	340			1.225		

commercial software FEMAP and afterwards exported to, and manipulated, with the pre/post-processing software ACTRAN/VI using the solving capabilities of ACTRAN/VA. The cross-section and 3D meshes generated with FEMAP are presented in Figure 1. The

which means that the boundary conditions need to be changed to consider the free field

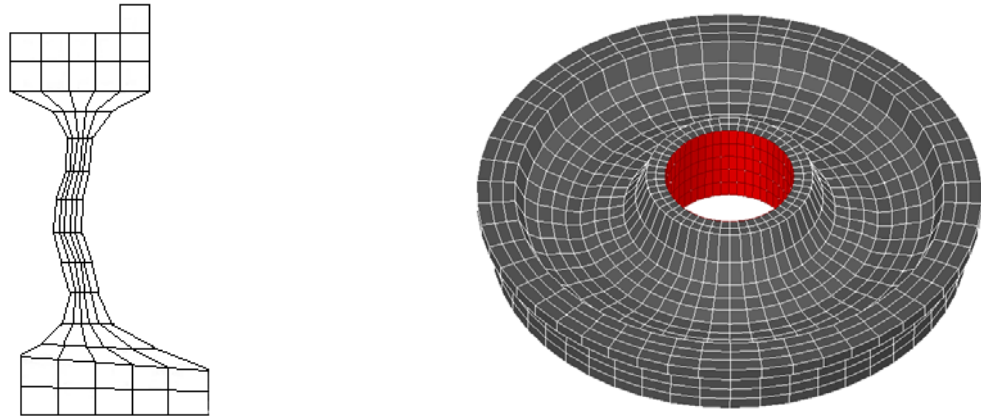


Figure 1. Cross-section and 3D FE mesh of the standard 920 mm diameter rail wheel.

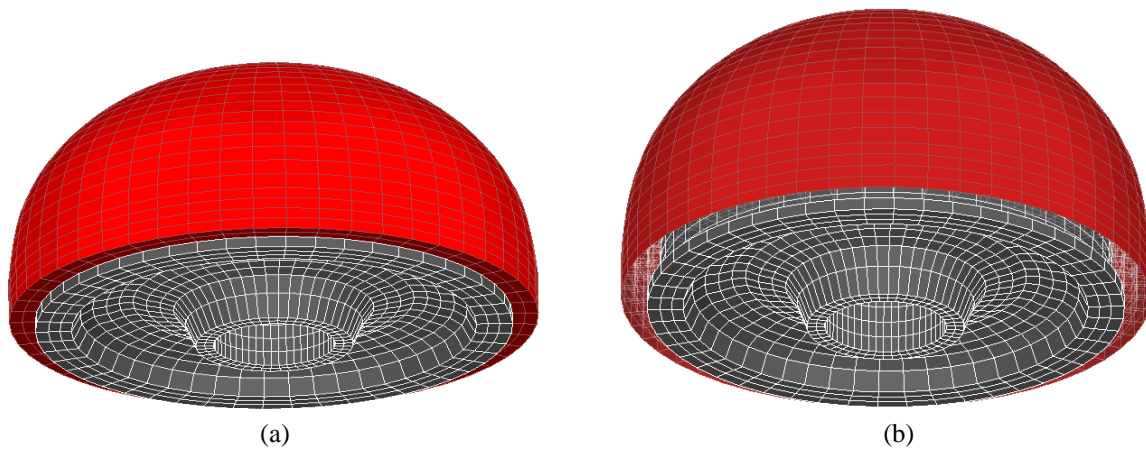


Figure 2. (a) 'Near field' fluid and solid train wheel meshes; (b) Far field infinite fluid mesh boundary and solid train wheel mesh.

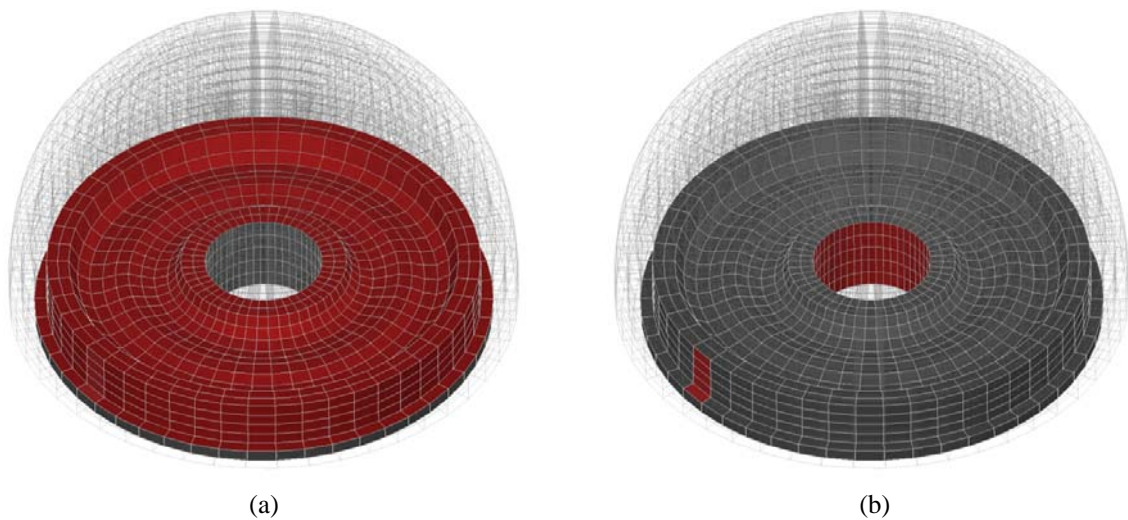


Figure 3. (a) Radiating surface mesh; (b) clamped surface mesh and loaded elements (distributed pressure).

radiation condition, which is made by the use of infinite element technology. Thus, the boundary of the FE fluid mesh is considered as a free field with the use of infinite elements (Figure 2b). The particularity

of infinite elements is the use of exponential functions multiplying the shape functions which is able to reproduce the decay of the

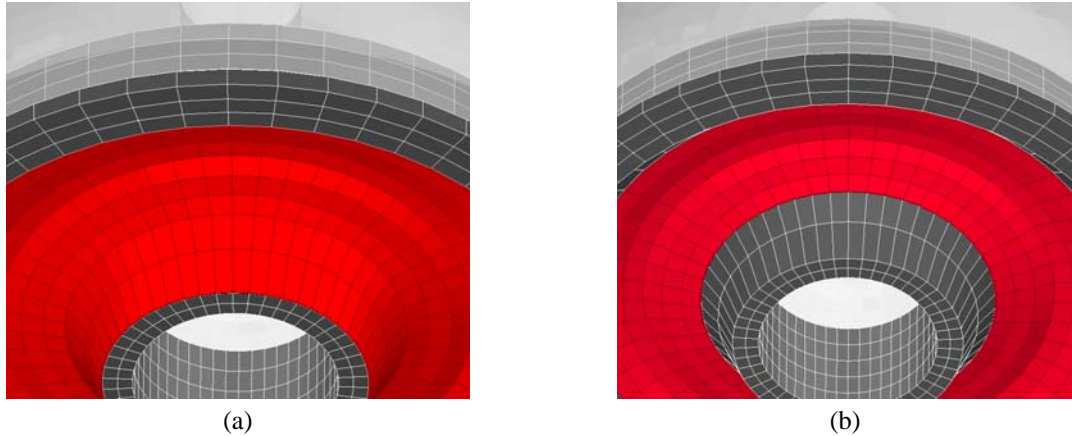


Figure 4. (a) Damping treatment mesh fully covering the hub, web and the zone under the rim of the inner wheel surface; (b) damping treatment mesh without covering the hub of the inner wheel surface.

pressure in the infinite (far) field (Bettess 1992; Astley 2000). The ACTRAN/VA (FFT 2007) uses an infinite element as a plane acoustic element (QUA04) with four nodes per element and with a pressure degree of freedom in all nodes.

A surface is modeled between the fluid and the solid mesh because the nodes at the interface must have both pressure and displacement DoFs (Figure 3a). The software ACTRAN/VA automatically assumes DoFs of pressure to both meshes if a surface is generated between a solid and a fluid mesh with coincident nodes. The acoustic element has four nodes per element and a pressure degree of freedom per node (FFT 2007). To simplify the problem, the noise radiation is analyzed only in the external face of the wheel.

The forced responses are calculated considering a unitary distributed pressure applied in a section of the tread and flange of the wheel (Figure 3b) and the inner diametrical surface of the wheel is considered as a clamped surface (Figure 3b).

The constrained layer damping treatment is composed by a 1 mm thick viscoelastic layer and a 2 mm thick steel constraining layer. The rigid constrained layer used in the damping treatment shifts the neutral axis towards the center of the viscoelastic layer increasing the effect and efficiency of the viscoelastic damping treatment yielding higher amount of shearing and energy dissipation in the damping layer. The constraining (steel) and viscoelastic damping

layer material and geometrical properties are presented in Table 1.

The mesh generated for the annular constrained layer damping treatment, comprising the constraining and constrained steel and viscoelastic layers, respectively, are generated with solid-shell elements (HEX8) available in ACTRAN/VA. This solid-shell element has eight nodes and three displacement DoFs per node (FFT 2007). Due to the thinness of the plate and damping layers, problems related with shear locking can compromise and deteriorate the accuracy of the results. However, the element HEX08 has some built-in remedies to avoid numerical pathologies such as locking. Figure 4 presents the two damping treatment meshes used in the analysis.

3.2 Constitutive Model of the Viscoelastic Material

Viscoelastic materials are very useful materials for vibration control purposes. The constitutive mechanical properties of such materials are mainly frequency and temperature dependent (Nashif et al. 1985; Jones 2001). Many authors have proposed different mathematical models to represent such constitutive behavior. Here, an *anelastic displacement fields* (ADF) constitutive representation is used to represent the properties of the viscoelastic material (Vasques et al. 2007a; Vasques et al. 2007b).

The values of the complex shear modulus in the Laplace domain, $G(s)$, can be obtained from

$$G(s) = G_{\infty} \left(1 + \sum_{i=1}^n \frac{\Delta_i s}{s + \Omega_i} \right), \quad (1)$$

or, alternatively, considering a pure imaginary variable, $s = j\omega$, in the frequency domain the complex shear modulus, $G(j\omega)$, is given by

$$G(j\omega) = G_{\infty} \left(1 + \sum_{i=1}^n \Delta_i \frac{\omega^2 + j\omega\Omega_i}{\omega^2 + \Omega_i^2} \right), \quad (2)$$

where G_{∞} is the so-called relaxed (also known as static) modulus, ω and j are the circular frequency and imaginary unit, respectively, and Ω_i is the inverse of the characteristic relaxation time at constant strain and Δ_i the correspondent relaxation resistance. To take into consideration the relaxation behaviour, the entire ADF model itself may be comprised of several individual fields, where n series of ADFs are used to describe the material behaviour. Given a set of measured values of the shear modulus in the form of a frequency dependent complex modulus, $G(j\omega)$ is determined through curve fitting techniques. The number of series of ADF parameters determines the accuracy of the matching of the measured material data over the frequency range of interest. The identified value of these parameters for the viscoelastic material used in this work, with commercial designation 3M ISD112, at 27°C is presented in Table 2. Additionally, this material has mass den-

viscoelastic materials can be improved by using a constraining layer, which shifts the neutral axis towards the center of the viscoelastic layer (Nashif et al. 1985; Mead 1998; Jones 2001).

3.3 Analysis and Results

The natural frequencies and the correspondent normal modes extracted with the undamped modal analysis solver of ACTRAN/VA are shown in Figure 6. For the development of good damping technologies for noise control in train wheels, it is important to first perform a modal analysis in order to investigate and critically evaluate the radiation efficiency of the different normal modes, as presented and discussed for example in (Thompson and Jones 2000; Thompson and Jones 2002; Thompson 2007). In view of this, all natural modes may be evaluated to assess the most appropriate locations of the viscoelastic damping treatment with the greatest deformations or displacements and to determine the position of the vibration control system.

The radiated sound power and MS velocity frequency response for the undamped and damped wheels are presented in Figure 6 and Figure 7. For the MS velocity frequency response determination, the excitation considered is a unitary distributed pressure over a few elements of the tread as shown in Figure 3b and the response is the mean square velocity over the area of the same elements. The acoustic and structural responses are calculated for the damped and undamped wheel assuming the two damping treatments depicted in Figure 4 and with

Table 2. ADF parameters for material 3M ISD112 at 27°C using three series (Vasques et al. 2007a).

ADF MODEL			
G_{∞} / MPa	i	Δ_i	$\Omega_i / \text{rad s}^{-1}$
0.1789	1	3.5286	504.20
	2	8.7533	4282.5
	3	60.324	39313

sity 1140 kg m^{-3} and a frequency independent Poisson's ratio equal to 0.49 is considered.

In practice, viscoelastic materials dissipate more energy when the predominant deformations are shear. Thus, the use of

the varying inner radius defined in Table 1. The inner radius of the damping treatment is changed to see the influence of the size of the viscoelastic treatment in the vibroacoustic behavior of the wheel. Since the use of viscoelastic materials in ACTRAN/VA

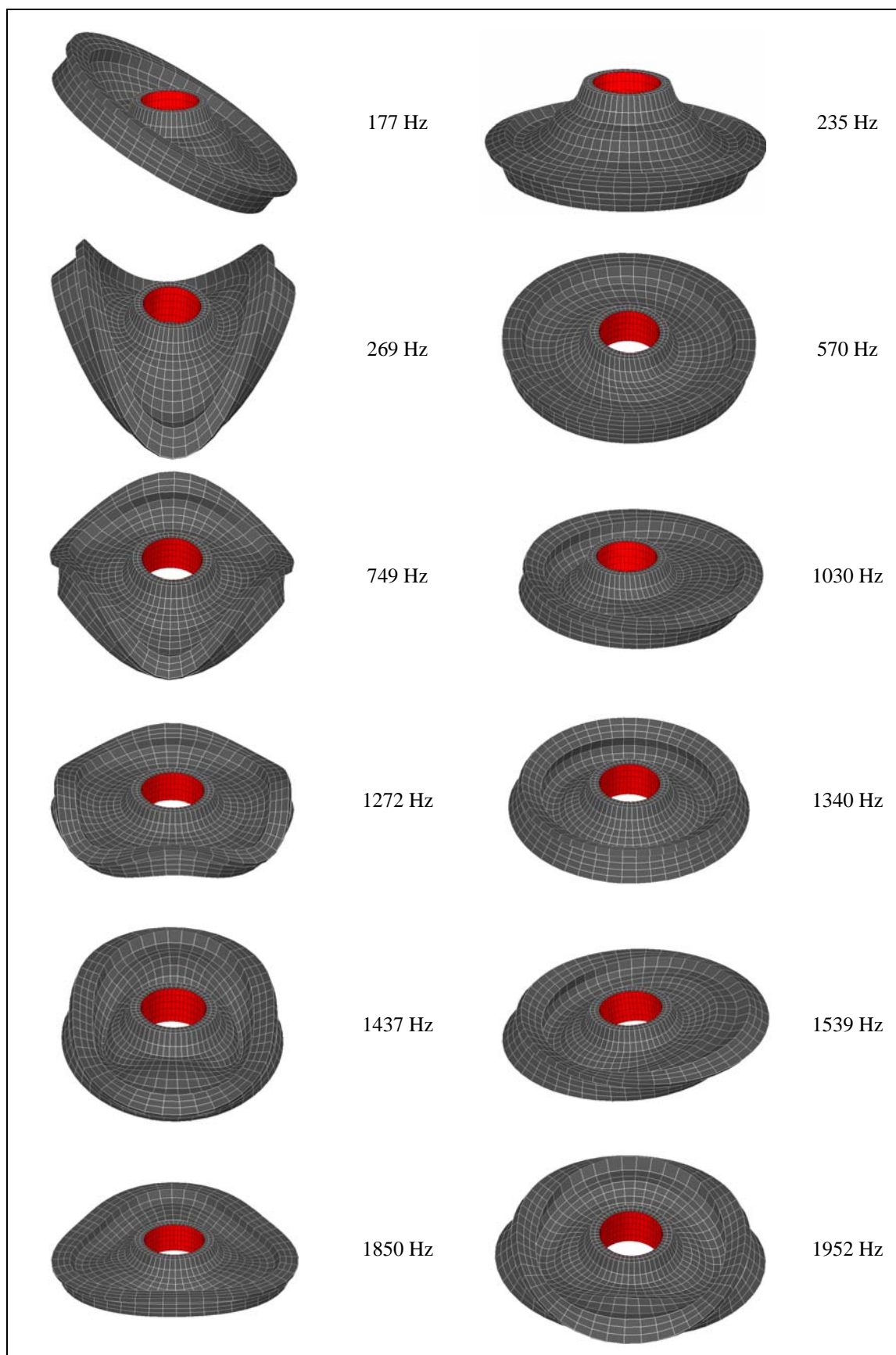


Figure 5. Natural frequencies and mode shapes of the undamped train wheel.

demands the use of a direct frequency response solution method (ACTRAN/VA does not make a modal analysis to coupled vibro-acoustic systems), the responses were evaluated over a frequency bandwidth of [0-1500] Hz with a frequency resolution of 1/4 Hz.

As can be seen, to some modes the reduction in the radiated sound power and MS velocity is more effective, but to all

tively compared to gain preliminary insight into the problem in order to proceed to a more detailed research of the use of the full capabilities of state-of-the-art viscoelastic damping technologies applied to railway wheels to reduce both rolling noise and curve squeal noise effects.

The use of viscoelastic materials in direct solution methods of the model demand a problem resolution in the frequency do-

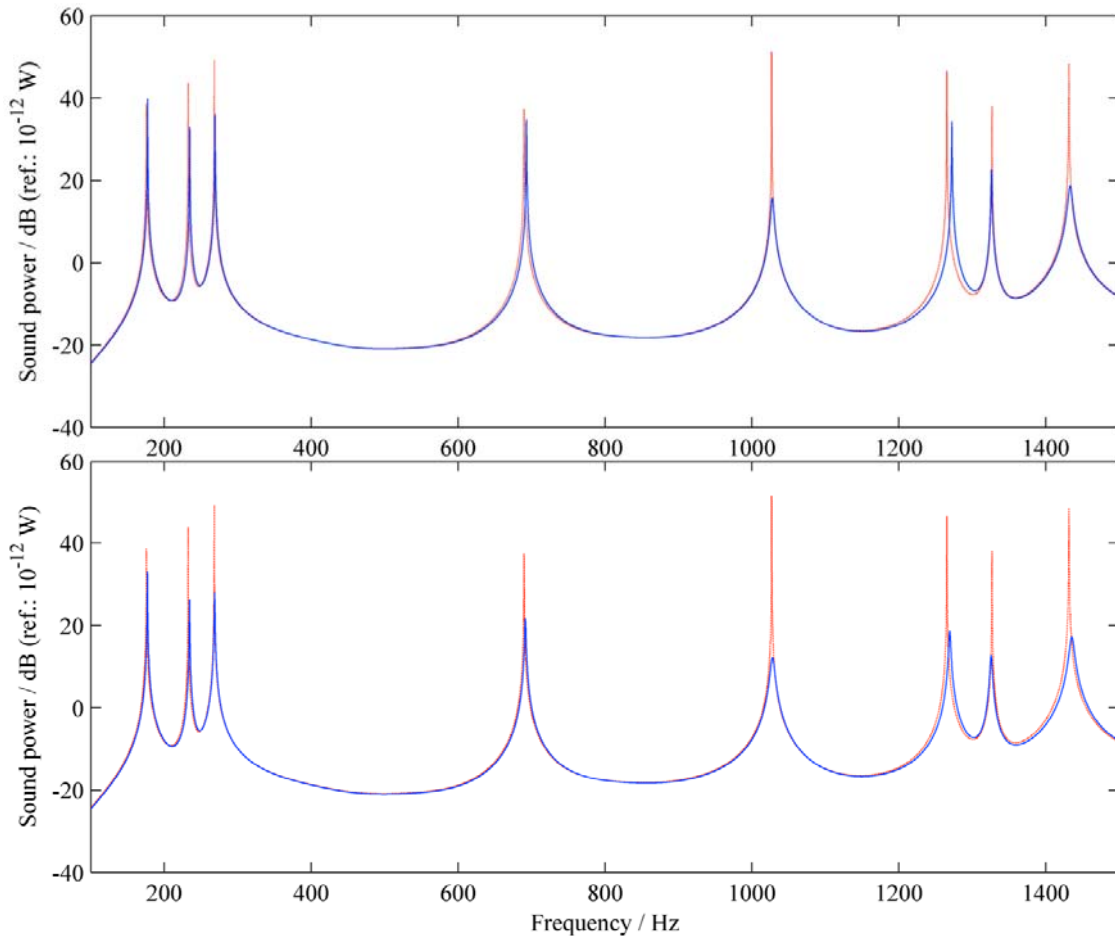


Figure 6. Radiated sound power of the undamped (dotted red line) and viscoelastically damped (solid blue line) wheel for different inner radius of the annular constrained damping treatment: inner radius equal to 128 and 162 mm (from top to bottom figure).

modes it can be observed that more than 10 dB sound power reduction is obtained.

4 CONCLUSION

The comparison among the vibro-acoustic behavior of a train wheel with and without viscoelastic damping treatments for a non-rotating wheel is presented, and the reduction of the radiated sound power is shown to yield good high frequency attenuation. With this analysis the damped and undamped performance are qualita-

main, which is supported by ACTRAN/VA. The comparison among the vibro-acoustic behavior of a train wheel with and without viscoelastic damping treatments for a non-rotating wheel is presented, and the reduction of the radiated sound power is shown to yield good high frequency attenuation. With this analysis it is expected to be able to qualitatively compare the damped and undamped performance and to gain preliminary insight into the problem in order to

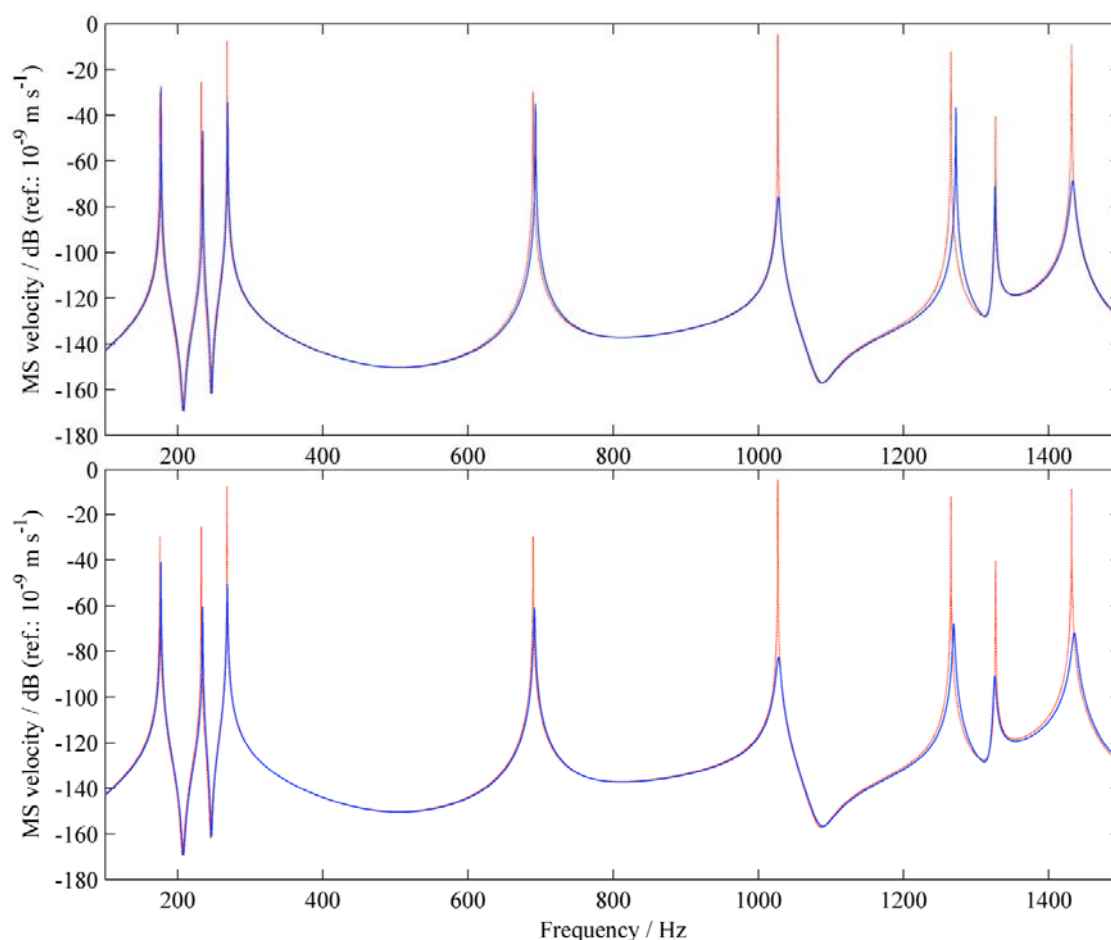


Figure 7. Mean square (MS) velocity over the area where the distributed excitation is applied of the undamped (dotted red line) and viscoelastically damped (solid blue line) wheel for different inner radius of the annular constrained damping treatment: inner radius equal to 128 and 162 mm (from top to bottom figure).

proceed to a more detailed research of the use of the full capabilities of state-of-the-art viscoelastic damping technologies applied to railway wheels to reduce rolling noise and curve squeal noise effects.

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