

DESIGN OF ‘QUIET’ CIRCULAR SAW BLADES THROUGH ADDED DAMPING

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ABSTRACT

A great deal of information on damping technologies is nowadays available for the practical reduction of machinery noise, in general, and circular saw blades, in particular, for wood-working operations. However, some of these noise reduction techniques are not gaining wide-spread acceptance, which possibly can be explained by lack of information/interest of the end-users. The need for continuing dissemination of more efficient and appellative noise reduction techniques for low-noise woodworking circular saw blades immediately emanates from the foregoing. The use of viscoelastic damping technologies is one possibility to reduce the radiated noise levels from circular saw blades. These technologies are analyzed in this article in order to gain a preliminary insight into the possible interest of this noise control solution to further continue developing more refined and efficient viscoelastic-based noise reduction designs for widespread use of low-noise (‘quiet’) circular sawing industrial practices for woodworking.

1 INTRODUCTION

The environmental awareness and the ever increasing demand for technology improvement is nowadays raising the concerns of people to promoting more appropriate machines and components which do not cause health damage to its operators. One such example occurs in woodworking companies, where excessive workers exposure to noise is a very important public health issue. In these companies, band and circular sawing is one of the most common wood-working operations. Although large band saws can still produce significant levels of noise, circular saws are in general responsible for the great majority of the employee noise overexposures in woodworking companies. Therefore, nowadays, whenever

possible we must use proven damping technology, scientific methods and common sense to achieve a beneficial relationship between environmental protection, public health and economically technological sustainable growth. In view of this, this article addresses the development and application of noise reduction techniques to circular saw blades employing constrained viscoelastic damping layers.

Noise radiation of annular plates representing circular saws has been widely studied by many researchers and in different ways. Lee and Singh (1994) studied the modal radiation efficiency considering a rotating frame. The rotation effect in the modal radiation efficiency was considered through a model of centrifugal forces (Sinha 1987). More recently, Côté et al.

(1998) considered a non-rotating frame where the effects of rotation are considered through the gyroscopic and centrifugal effects in a forced analysis of the radiation. Vibration control strategies were reported by Nishio and Marui (1996) that studied the effect of the slots in the vibration of a circular saw. The noise control or the vibration control of circular saws concerning experimental prediction and active noise and vibration control was analyzed by Wang et al. (1999).

The aim of this article is to study the vibroacoustic behavior of a damped circular saw and qualitatively assess the radiated noise reduction that occurs when employing a constrained viscoelastic damping layer. Circular saws are generally thin structures which in operation are usually subjected to elevated rotating speeds which strongly dictate the dynamic characteristics of the system due to gyroscopic and centrifugal stiffening effects. These effects, though, must be carefully taken into account at the modeling and design stages of saws with viscoelastic damping solutions (Côté et al. 1998, p. 204), at the cost of increasing the complexity of the analysis. For simplicity, these effects are usually neglected which, for a preliminary qualitative analysis, is a reasonable and admissible modeling shortcoming.

This article starts with a brief physical description of the problem and some theoretical details regarding the finite and infinite element based vibroacoustic modeling strategies are outlined. Next, the viscoelastic constitutive behavior of the material used in this study as damping treatment is discussed and the article ends up with a case study considering a vibroacoustic FE annular plate model representing a standard circular saw blade.

2 PHYSICAL DESCRIPTION OF THE PROBLEM

Noise produced in circular saw blades is generally accepted to involve two main sources: (i) aerodynamic sources, involving the tooth and gullet area of the blade during idle and (ii) structural vibration noise, that is produced by blade and work-piece related sources (Nielsen and Stewart 2007). The understanding of the noise generation

mechanisms and strategies to reducing the noise produced by circular saw blades has for a long time motivated a great deal of research efforts. On the one hand, research has dealt with the aerodynamic aspects, mainly consisting of studies on airflow disturbances created by rotating disks with various types of openings (gullets) cut into the periphery; little agreement, though, has been reached among researchers on how forces are generated and how disk geometry and gullet's geometry affect the nature of the noise produced. On the other hand, research efforts have been directed towards the vibration behavior of the saws and the resultant noise radiated by the blade.

There are many different geometries of circular saws in the market nowadays, with all of them being somewhat prone to induce noise problems. As previously discussed, the generated noise can be a consequence of many different aspects and physical phenomenon, like the vibration of work-piece, the cut impact, the vibration of the saw or aerodynamic sources during idle. The focus of this article is put on the noise generated by the vibration of the saw. Among the many sources causing vibration of the saw, the most important are the aerodynamic fluctuations (Cho and Mote Jr. 1979), and the self excited vibration (Tian and Hutton 2001) caused by the cut. These sources excite the structure, and the resulting vibrating structure radiates unwanted sound.

Since in woodworking, the work-piece is wood, which in general has a significant amount of damping that makes them inefficient noise radiators and vibration conveyors to the supporting and fixing apparatus, work-piece related sources are considered not too much relevant. Regarding the aerodynamic ones, they are not of concern in this article and the attention is driven to structure-borne noise generated by the saw blade.

Blade vibration response may be of a *resonant type*, whether resulting from aerodynamic and other excitation sources during idle or tooth impact in cutting, or of a *forced-response type*, which is caused by tooth impact occurring only during cutting. Therefore, in idle, the radiated noise of a 'whistling' blade typically consists in an intense pure tone noise resulting from blade resonant vibration which results from exci-

tation forces providing energy in a frequency range where at least one easily excitable mode exists; in this case, damping plays a central role, which for lightly damped saws may result in extremely high level of pure tone noise usually of an efficient structural acoustics mode. In a different way, during cutting, the tooth impact excitation produces broadband vibrational energy as well as energy at the rotating circular frequency (and harmonics) of the saw. Therefore, resonant response during cutting may involve more than one resonant frequency. Lastly, considering forced-blade vibration, the response at non-resonant frequencies is not as sensitive to damping as the resonant case, turning resonant reduction techniques not appropriate, and the noise is characterized by strong frequency peaks at the tooth passage frequency and harmonics (Nielsen and Stewart 2007).

‘Low-noise’ saw blades designs available in the market are most effective in reducing resonant blade vibration noise during idle and have mainly been based in trying to somewhat introduce damping; the use of laser cut slots and plugs causing localized ‘disruption’ of wave propagation of certain modes and damping through the scrubbing that occurs in the laser slots and plugged holes has been proposed. Alternatively, laminated blade designs making use of the principle of constrained layer damping have been proposed since the 1970s as effective means to reduce both resonant vibration during idle and blade vibration noise during cutting. Although not so effective, free layer damping treatments have also been considered (Szymani and Mote 1977; Nielsen and Stewart 2007). However, regarding the forced-blade structure-borne noise, aside from the highly effective laminated blade construction discussed above, typical damping treatments were reported to provide only minor noise reduction under those forcing conditions (Nielsen and Stewart 2007).

3 VIBROACOUSTIC FINITE / INFINITE ELEMENT MODELING

In a vibroacoustic analysis the governing system of equations is particularly different because each individual problem, namely the acoustic and the structural problems,

need also to consider the interaction that occurs between the two mediums. For that purpose, weak and strong structural-acoustic coupling strategies have been proposed somewhat combining the equations governing the physics of the two mediums. In a finite element context, such a model is usually described in terms of displacements for the structure and acoustic pressures for the fluid. Additionally, in general terms, the resultant coupled system may be formulated in terms of the displacement field and pressure variables, yielding a non-symmetric coupled system, or in terms of a pressure-related potential function which, although not allowing a direct determination of the pressure, it may yield a more convenient symmetric coupled matrix governing equation (Ohayon and Soize 1998). Modal solvers are usually employed to solve weak or fully coupled system of equations; as the degree of coupling increases, the higher the computational cost is and more robust and sophisticated modal solvers are required. Alternatively, besides allowing the fully coupling being considered and the consideration of structural-acoustic damping effects, a direct frequency response analysis may be more straightforward to use and implement, benefiting also from the high computational efficiency of algebraic solvers such as the Krylov solver. The latter solver is a direct sequential solver aimed to be used to quickly calculate the frequency response of a system where all materials and boundary conditions have a simple relation with the frequency (FFT 2007).

Some difficulties arise with the use of finite elements for the modeling of the acoustic infinite free field medium. For finite element analysis of free fields it would be necessary to model the entire field, resulting in a prohibitive computational cost. Alternatively, infinite element technology comprising finite elements which extend from any surface to infinity have been successfully reported as a means to overcome this difficulty. The particularity of infinite elements is the use of exponential functions multiplying the shape functions which is able to reproduce the decay of the pressure in the infinite (far) field (Bettess 1992; Astley 2000).

In this study the vibroacoustic finite element code ACTRAN/VA is used to model

a structure ‘immersed’ in air. The structural and fluid medium meshes are generated with the commercial software FEMAP and afterwards exported to, and manipulated, with the pre/post-processing software ACTRAN/VI using the solving capabilities of ACTRAN/VA. Infinite element technologies available in ACTRAN/VA are used in the fluid mesh and the effect of far field acoustic attenuation can be reproduced with a good accuracy.

4 CASE STUDY: CIRCULAR SAW BLADE

A fully coupled vibroacoustic finite element (FE) model is generated using the software ACTRAN/VA. The annular plate (saw) and constrained layer damping treatment are modeled with solid-shell FEs; the surrounding acoustic fluid (air) is modeled with acoustic fluid FEs and in the boundary of the acoustic medium an infinite fluid is considered through the use of infinite element technology. At this preliminary stage of the analysis, the rotating (centrifugal) effects, which might be considered through an equivalent initial stress field (pre-stress effects), are not taken into account. Furthermore, for simplification, the teeth of the saw are not considered, gyroscopic effects

with viscoelastic materials may present some advantages over other solution methods (Vasques et al. 2007a; Vasques et al. 2007b) and demands a problem resolution in the frequency domain, which is the solution method supported and preferred by ACTRAN/VA. Results are thus presented in terms of frequency response functions (FRFs) of the velocity response and radiated sound power for a non-rotating undamped and damped saw blade in order to assess the effects of the damping treatment.

4.1 Material and Geometric Properties of the Saw (Annular Plate)

A circular saw is considered as an annular plate in this section. Simulations are made and the results obtained for an annular plate with the material and geometric properties described in Table 1 are presented. The geometry of the constrained layer and the viscoelastic layer are also described.

For the simulation and analysis, the plate is considered clamped in the inner radius surface and the excitation typically produced by a circular saw in operation is simplified to a harmonic force represented by a unitary transverse point load applied at a FE node in one side of the plate. The saw is

Table 1. Material and geometric properties of the vibroacoustic system.

STRUCTURAL SYSTEM						
	Young's Modulus / GPa	Poisson's ratio	Density / kg m ⁻³	Inner radius / mm	Outer radius / mm	Thickness / mm
Plate	215	0.32	7800	63.5	173	3
Constraining layer	70	0.3	2700	72.2	(90,108,126)	0.254
Viscoelastic layer	Table 2	0.49	1140	72.2	(90,108,126)	0.127
ACOUSTIC SYSTEM						
	Speed of sound / m s ⁻¹			Density / kg m ⁻³		
Air	340			1.225		

are neglected, only one side of the saw is coupled with the acoustic domain, the plate is considered to be in an infinite baffle, isothermal conditions are assumed and the inner circle of the saw is considered as clamped. The use of direct solution methods when considering a structural model

considered homogeneous and to be made of an isotropic elastic material, in this case steel. The air properties are also presented in Table 1. Further, gyroscopic and centrifugal forces effects are neglected in this preliminary analysis.

4.2 Constrained Viscoelastic layer damping treatments

Viscoelastic materials are very useful materials for vibration control purpose. The constitutive mechanical properties of such materials are mainly frequency and temperature dependent. 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 as presented by Vasques et al. (2007a).

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	Ω_i / rad s ⁻¹
0.1789	1	3.5286	504.20
	2	8.7533	4282.5
	3	60.324	39313

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

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 behavior, the entire ADF model itself may be comprised of several individual fields, where n series of ADFs are used to describe the material behavior. 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 are presented in Table

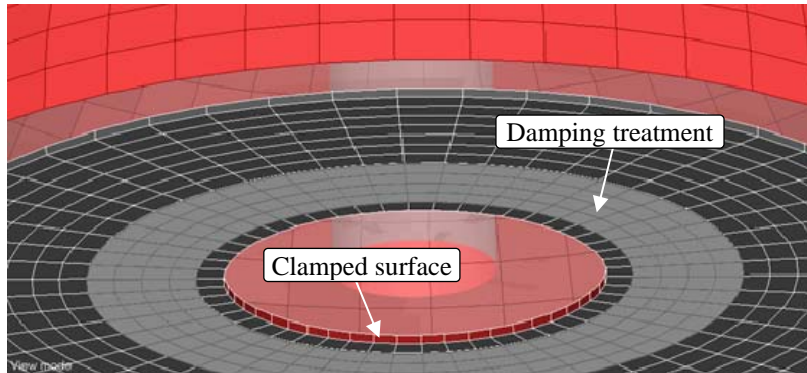


Figure 1. Clamping surface and viscoelastic constrained layer mesh representation.

domain the 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

2. Additionally, this material has mass density 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 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).

The application of the annular constrained viscoelastic damping treatment and the correspondent FE mesh is depicted in Figure 1.

The mesh generated for the viscoelastic

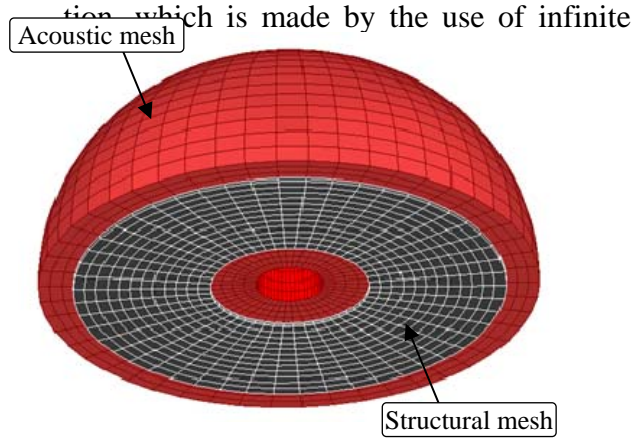
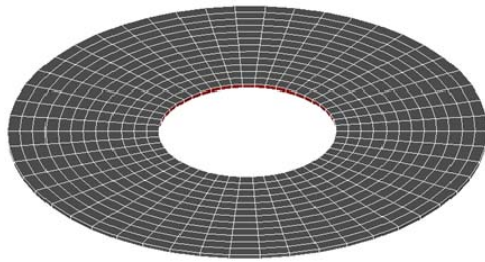


Figure 2. Representation of the structural and fluid-structure meshes.

constrained layer has the same elements used for the plate, namely eight-noded solid-shell elements (HEX08). As obvious, the meshes need to be coincident, which means the nodes at the interfaces plate/viscoelastic layer and viscoelastic layer/constrained layer, must be the coincident to enforce displacement continuity.

4.3 Fluid Media Model and Discretization

The main difficulty concerning the fluid mesh generation is that the nodes at the interface between the fluid medium and the structural medium need to be coincident. This means that the nodes at the interface must have both pressure degrees of freedom (DOFs) and displacement DOFs. In view of this, the fluid mesh was generated taking into account the geometry of the structural mesh. Considering that structures vibrating in free field suffer a small influence by the fluid, in this case air, to reduce the computational time of analysis only the acoustic medium in one side of the plate was modeled for this qualitative analysis. Due to the spherical wave propagation nature of the sound radiation, the fluid mesh is modeled as a semi-sphere (Figure 2).

The finite element 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 finite element mesh for the fluid consider an enclosed fluid, which means that the boundary conditions need to be changed to consider the free field radiation condition which is made by the use of infinite

element technology.

4.4 Structural Model and Discretization

The geometry of circular saws is very similar to the geometry of annular plates. To make this simplification, the teeth need to be neglected which, according to Côté et al. (1998), is an admissible simplification. Thus, an annular plate is considered representative of a circular saw and is discretized with solid elements as shown in Figure 2. The structural element type used (HEX08) is a solid-shell element with 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.

The mesh is regular, in the sense that the mediums are discretized with an equally spaced discretization step of the radial and angular coordinates, containing twelve elements in the radial direction and fifty divisions in the circumferential coordinate. Due to the small thickness of the plate, only one finite element is considered through the thickness. It is worthy to mention that the structural mesh should have a more refined discretization than the fluid mesh, due to the propagating wave speed magnitude of the solid structural medium which, in general, is higher than the one in the air. Indeed, it is in fact the regular structural mesh discretization radial and angular steps that dictate the useful bandwidth of the analysis (the finest the mesh is the higher the maximum frequency of the bandwidth would be). However, in order to obtain an admissible computational effort, the mesh size was limited to a comfortable level of discretization. Regarding the boundary conditions considered, a simplification is adopted considering the inner radius clamped, which agrees well with the physical phenomenon observed in saw blades in operation.

Since the aim is to simulate a saw blade in free field, as previously discussed, the use of infinite elements is necessary to reproduce the decaying effect of the far field pressure. Figure 3 shows the coupled system with the different finite and infinite domains and meshes.

4.5 Modal Analysis of the Saw (Annular Plate)

A preliminary modal analysis of the annular plate is carried out in order to deter-

mine the regions of larger deformations. In principle, these regions are the most attractive places to locate the constrained viscoelastic damping layer treatment. However, it must be considered that a circular saw has regions which need to be free, such as the cut region and the clamping region (Figure 1), which puts some limitations in the choice of the treatment location and covered surface.

A modal analysis was performed for the uncoupled undamped annular plate with the aim of determining and evaluating the structure undamped natural frequencies and mode shapes. This information is important to critically judge which modes might be prone to be more efficient noise radiators and the frequencies involved. The first nine modes of the annular plate are represented in Figure 4. It can be observed that the modes with radial nodal lines (nodal diameters, m) have the lower frequencies, while modes with circumferential nodal lines (nodal circles, n) have higher frequencies. The mode shapes are denoted here by (m,n) .

4.6 Frequency Analysis of the Saw (Annular Plate)

Due to the fact that ACTRAN/VA does not allow making a modal analysis of a coupled vibroacoustic system, a forced frequency analysis is performed in this section. Two types of curves of frequency response are presented, namely forced frequency response in terms of (i) radiated sound power and (ii) displacement (*receptance*) of the undamped and damped annular plate (saw), comparing both the struc-

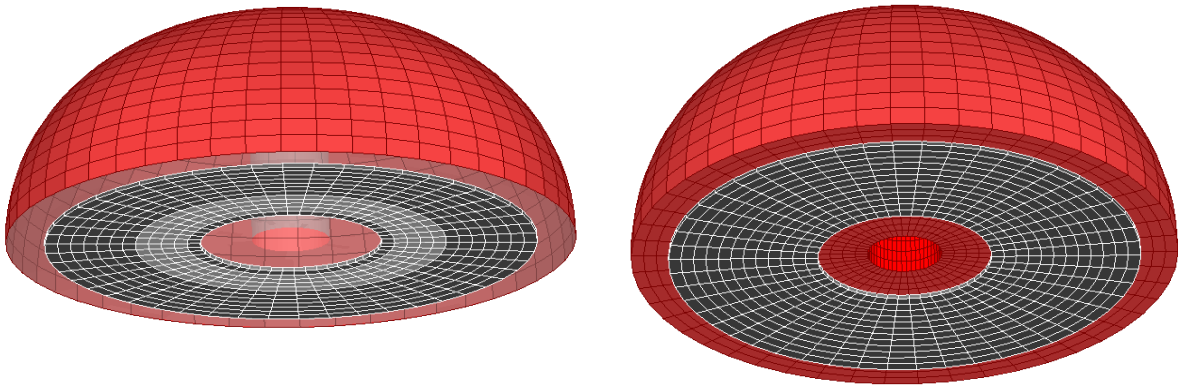


Figure 3. Left: Representation of the infinite fluid mesh surface; right: finite fluid meshed volume.

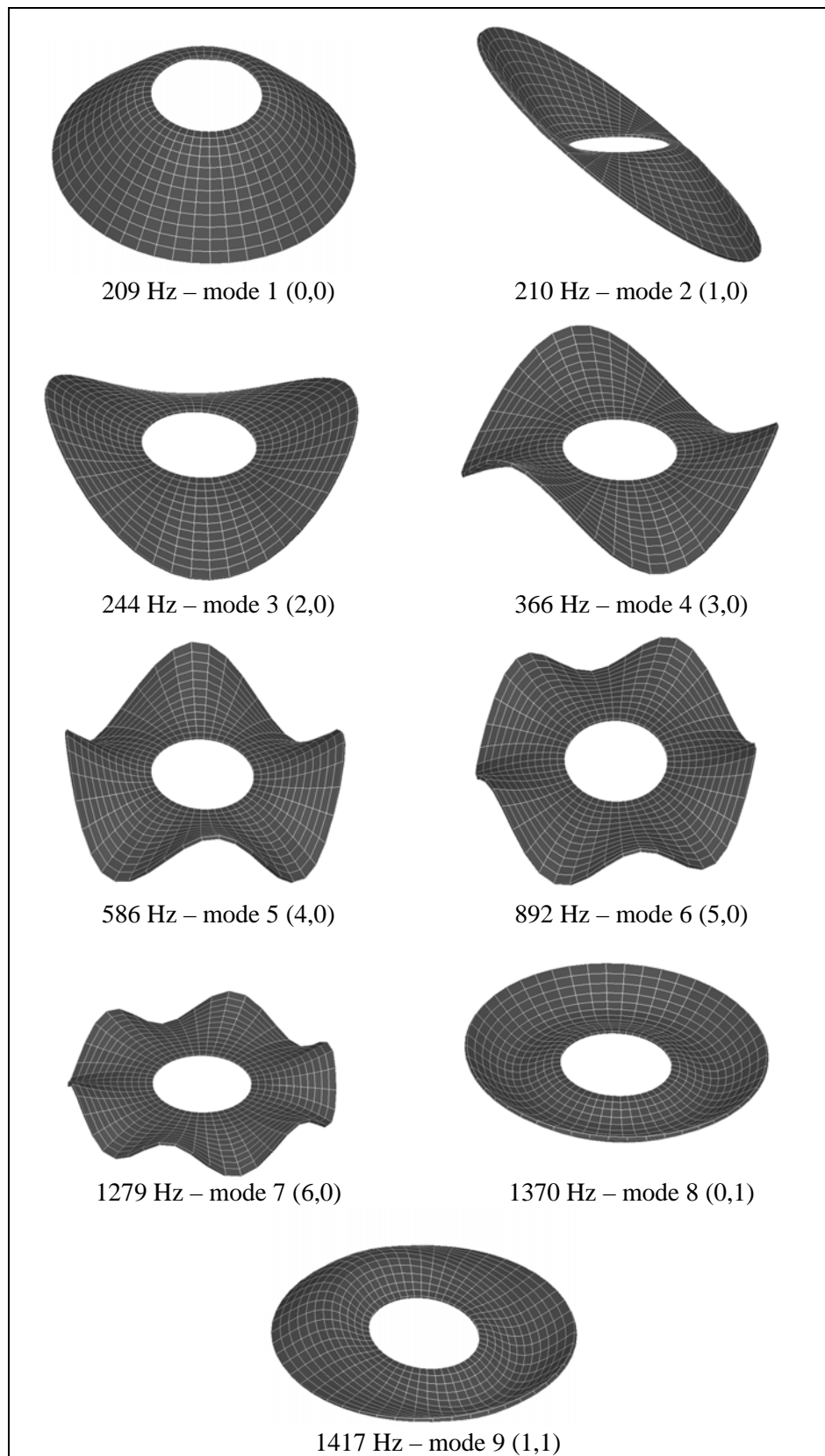


Figure 4. Undamped mode shapes and natural frequencies.

tural acoustics and vibratory behaviour with and without a viscoelastic constrained layer damping treatment for various outer diame-

ters of the constrained viscoelastic damping treatment.

4.6.1 Excitation model

A forced frequency analysis, representative of the real response observed due to the in-operation excitation, demands a model for the input force (or excitation). Some authors have considered a rotating force and a non-rotating plate (circular saw). Conversely, other analyses are made which consider a non-rotating force, but a rotating plate. As previously discussed, the excitation can originate from many different sources, but the principal sources usually considered are the self-excited vibrations due to tooth impact occurring only during cutting and the aerodynamic fluctuations occurring during idle. The self-excited vibrations are caused by the phenomenon called *stip-slick*. The stip-slick phenomenon is caused by a variation of the coefficient of friction, between the saw and the cut piece. This variation generates a variation of the cut force applied. It can be noted that this phenomenon is self generated, which is a characteristic of unstable systems. These self-generated forces excite the resonant frequencies of the saw, producing sound radiation. The other source of forces is due to the aerodynamic fluctuation, which thanks to the variation of the pressure on the tooth cause an excitation force which excites the natural frequencies of the saw.

Considering that in this article the rotating force is considered to be non-rotating, the force is modeled as a harmonic force applied at the board of the plate and, as shown in Figure 5, it is represented here by an out-of-plane point load on a FE node present on the outer diameter. This model of force is an elementary attempt to approximate the physics of the underlying excitation during cutting.

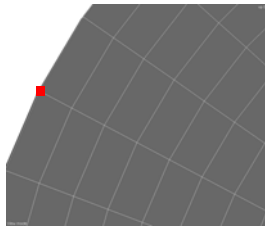


Figure 5. Location of the nodal input force (red square) represented in a small portion of the plate.

4.6.2 Radiated sound power

The interest of this analysis is to qualitatively assess the reduction of the radiated noise generated by the vibrating saw with and without the viscoelastic damping treatment and to critically evaluate the influence of the covering area of the treatment (diameter) on the vibroacoustic response. For that purpose, the simplified excitation model comprising a harmonic force described in the previous section is applied to determine the FRFs for both structures (with and without viscoelastic constrained layers) in order to evaluate the radiated sound power, W_{rad} . By definition it is given by (FFT 2007, Sec. 30.5)

$$W_{\text{rad}} = \frac{1}{2} \int_S p_{\text{tot}} v_{\text{tot}}^* dS, \quad (3)$$

where p_{tot} and v_{tot}^* are the total acoustic pressure and complex conjugated velocity fields over the radiating structural surface, S . However, classically, for an arbitrary structure, with some time- (angle brackets) and space-averaged (over bar) mean-square vibrational velocity, $\overline{v^2}$, the radiated sound power can also be defined as (Wallace 1972; Norton and Karczub 2003, p. 204)

$$W_{\text{rad}} = \sigma \rho_0 c S \overline{v^2}, \quad (4)$$

where ρ_0 is the density of the fluid medium into which the structure radiates, c is the speed of sound in the fluid medium (in this case air) and σ is the global (in opposition to the modal one) radiation ratio. The radiation ratio thus provides a powerful and useful relationship between the structural vibrations and the corresponding radiated sound power. Indeed, Equation (4) is perfectly generic and can be used to describe any radiating structural system, provided the radiation ratios of different systems can be established and known. Then, estimating the subsequent radiated noise is a relatively easy process which is performed directly in terms of the radiating surface vibration levels, determined either theoretically (analytically or numerically) or experimentally.

It is well known from structural acoustics theory (Junger and Feit 1986; Cremer et al. 2005; Fahy and Gardonio 2007) that the radiation ratio, σ , of an arbitrary structure is defined as the sound power radiated by

the structure into half space (i.e. one side of the structure) divided by the sound power radiated by a large piston with the same surface area and vibrating with the same RMS velocity as the structure (Wallace 1972). Thus, it describes the efficiency with which the structure radiates sound as compared with a piston of the same surface area with a unity radiation ratio, and can be either greater or less than unity.

A radiating surface is a set of finite element faces on which radiated acoustic power is calculated with ACTRAN/VA (Figure 6). Results are thus presented in terms of frequency response functions of the velocity response and radiation sound power for the undamped and damped saw blades in order to assess the effects of the damping treatment. The radiated sound power is determined assuming the annular plate (saw) in an infinite baffle and is determined for a damped and undamped cases. The results are presented in Figure 7 and Figure 8.

It is well known that the radiation efficiency depends not only on the size and shape of the radiating structure, as compared with the structural vibration wavelength, but also on the manner the body is vibrating. The previous insight into the typical vibroacoustic behaviour of radiating structures demonstrates that there are critical frequencies values below which the radiated modes are inefficient. Additionally, the net sound radiation efficiency accounts for the individual and cross-coupled influence of all modes, with the coupling effect being more pronounced below the critical frequency value, with a somewhat destructive or constructive ‘interference’ of the modes. The previous frequency response functions demonstrate that the viscoelastic damping treatments are able to attenuate both the radiated noise and receptance over all modes considered. The attenuations are evident for the treatment with the larger outer diameter but are still significant with the treatment with smaller diameter (let us recall that a power reduction of 3 dB corresponds to half the radiated noise power).

Off-resonance, where more than one mode is significant and where the radiated power cannot be simply calculated using individual radiation ratios, attenuation is not obtained, making the damping treatment

only efficient at the resonances. This also may turn the treatment inefficient when considering a tonal disturbance due to the rotating speed (and harmonics) of the rotating saw; those effects are not considered in this study.

It is also known that the constructive and destructive ‘interference’ of neighbouring ‘cells’ of each individual mode dictate the individual mode radiation ratio (or efficiency). Therefore, at the design stage of the damping treatment, the modal interference of the vibration modes should be considered to attenuate the resultant radiation modes.

CONCLUSION

Vibroacoustic systems are multiphysics problems, which means that different physical domains are involved and correlated in the analysis. In this case, there are different mediums which need to be discretized, namely the structural (circular saw blade) and acoustic fluid (air) media. The circular saw is considered to be in free field, so that the far field wave attenuation effect needs to be considered in the analysis. This latter issue is tackled by modeling the far field acoustic fluid medium with infinite element technology, while the remainder proximity fluid field and structural system are modeled through solid and solid-shell three-dimensional elements available in ACTRAN/VA.

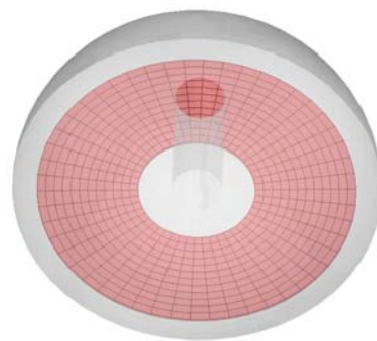


Figure 6. Surface used to determine the radiation sound power in ACTRAN/VA.

Constrained viscoelastic damping layers are considered in order to attenuate sound radiation and to reduce the “kerf” losses due to the blade thickness and vibration. Constrained layers are more effective than

pure viscoelastic layers due the increase of the shear deformation in the viscoelastic

and without viscoelastic damping treatments in terms of the reduction of the radi-

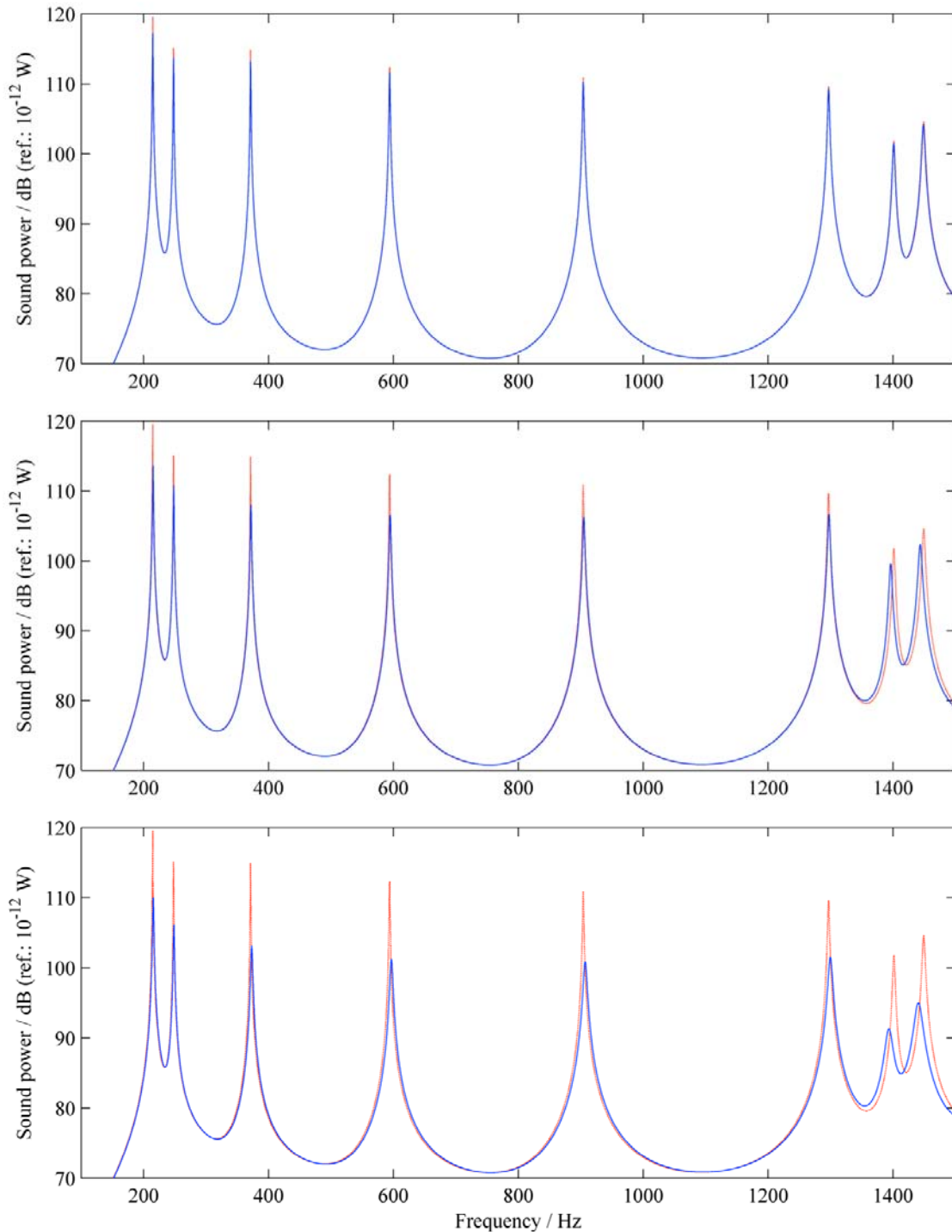


Figure 7. Radiated sound power of the undamped (dotted red line) and viscoelastically damped (solid blue line) saw for different radius of the annular constrained damping treatment: outer radius equal to 90, 108 and 126 mm (from top to bottom figure).

material. Thus, the effect of a commercial constrained layer applied to the circular saw is considered and assessed.

The comparison of the vibroacoustic behavior of a standard circular saw blade with

ated sound power and vibratory response is shown to yield in general good attenuation of all modes, being more significant in the higher ones and for larger annular diameters of the treatment. The variation of the outer

diameter of the viscoelastic constrained layer generates a great variation concerning

to a more detailed research of the use of viscoelastic damping technologies applied

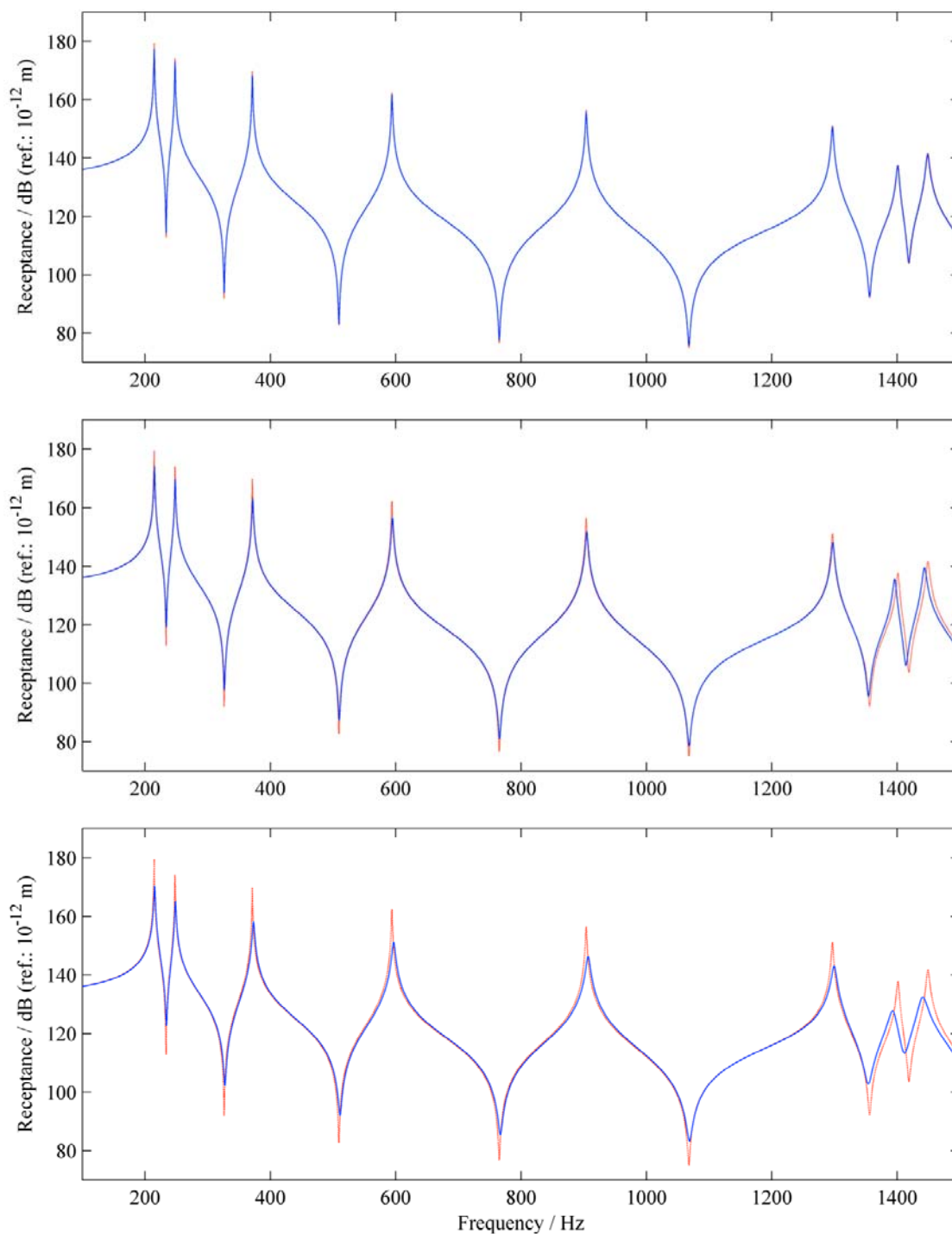


Figure 8. Driving point receptance determined at the outer radius of the undamped (dotted red line) and viscoelastically damped (solid blue line) saw for different radius of the annular constrained damping treatment: outer radius equal to 90, 108 and 126 mm (from top to bottom figure).

the reduction of the sound radiation.

With this preliminary simple analysis the damped and undamped performance are qualitatively compared to gain preliminary insight into the problem in order to proceed

to general circular tooling. Effects due to the gyroscopic and centrifugal nature of the rotating system due to the rotation of the saw may originate significant modifications

of the natural frequencies and damped vibroacoustic behavior.

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