Vibration control
with air film damper

Master’s Thesis in the Master’s programme in Sound and Vibration

GREGOIRE LEPOITTEVIN

Department of Civil and Environmental Engineering
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CHALMERS UNIVERSITY OF TECHNOLOGY
Göteborg, Sweden 2008

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Abstract

In the automotive industry a common way to control panel vibrations is by means of the application of layers of viscoelastic materials. This thesis investigates the possibility to damp the vibrations of automotive panels with an alternative solution, based on an air film damper: in this case the vibration energy is dissipated due to the viscosity of the air.

In order to estimate its potentials, a simplified model of an air film damper is analysed and compared to other damping devices. After this, measurements are realised to identify the driving parameters that govern the reduction of vibration obtainable using the air film damper. Finally, to validate the ideas and hypotheses coming from the interpretation of the measurements a finite element analysis has been employed.

Key words: damping mechanisms, passive vibration control, air viscosity and transfer function measurement.
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Notations

Symbols

A  Surface area \([m^2]\)
C  Speed of sound in air \(c=343 \text{ m.s}^{-1}\)
E  Young’s modulus \([Pa]\)
f  Frequency \([s^{-1}]\)
I  Inertia \([m^4]\)
\(\mu\)  Viscosity of air \(\mu=1.8 \times 10^{-5} \text{ Pa.s}\)
\(\rho_0\)  Density of air \(\rho_0=1.21 \text{ kg.m}^{-3}\)
\(\omega\)  Angular frequency \(\omega=2\pi f \text{ [rad.s}^{-1}\)\]

Abbreviations

AFD  Air Film Damper
FRF  Frequency Response Function
PP  Poly Propylene
1 Introduction

In the automotive industry a common way to control panel vibrations is by means of the application of layers of viscoelastic materials. The energy vibration is dissipated due to the viscoelastic phenomena taking place inside the material when the material itself is periodically deformed.

This thesis investigates the possibility to damp the vibrations of automotive panels by means of an alternative solution based on an air film damper: in this case, the vibrational energy is dissipated due to the viscosity of the air. This solution is currently used to add passive damping on gas turbine blades. Its advantages consist in the fact that it does not significantly change the dimensions, the mass and the natural frequencies of the system to damp. The goal of this thesis is to study the possibility to implement such a solution on an automotive structure.

To study the air film damper, a simple two-degree of freedom model is first considered. In order to estimate its potential, such a device damper is then practically implemented on a flat plate and compared through tests to other damping mechanisms. A further measurement campaign is performed to identify the parameters that govern the vibration reduction obtainable with the air film damper. Within this activity, also the dynamic behaviour of the plate used to generate the air gap is investigated.

After this, a finite element model is built up in order to analyse the physical phenomena that take place in the interaction between the primary plate (i.e. the plate that has to be damped) and the plate on top of it that is used to generate the air film. This analysis is useful to try and to interpret the test results collected in the previous phase of the activity.
2 Air film damper

2.1 Principle

A thin layer of air is squeezed between two plates vibrating independently. The vibration of the two plates generates a pumping effect of the air layer and energy is dissipated because of the inherent viscosity of the air.

Two approaches can be considered in the analysis of an air film damper (AFD). The first one concerns the fluid motion in the air film: its objective is the evaluation of the air velocity, in order to estimate the amount of energy dissipated. The second one deals with the structural dynamics of the plates that delimit the air film. Its objective consists in evaluating the effects of the air film on the vibrations of these plates, when an external force excites them. This thesis work focuses on this latter aspect.

Figure 1 shows a simplified representation of an AFD. A thin air cavity is partially constrained between two parallel plates. Both the bottom plate (that represents the “primary” structure to damp) and the top plate are considered as simply supported. The air is assumed as a viscous incompressible fluid.

Figure 1: Top and side views of a simplified AFD
2.2 Two-degree of freedom model

In order to investigate the influence of the air layer on the vibrations of the primary panel, a two-degree of freedom model is used [1].

More in detail, both the primary plate and the top plate are assumed to vibrate in one of their natural modes, giving the possibility to model them as a single-degree of freedom system, using an effective modal mass $M$ and stiffness $K$. The dissipation due to the air is modelled as a linear viscous dashpot with a coefficient $C$. The mass of the air is neglected as well as the internal dissipation mechanisms of the plates.

The equations of motion for the system presented in Figure 2 are:

$$
\begin{align}
M \frac{d^2 X(t)}{dt^2} + C \left( \frac{dX(t)}{dt} - \frac{dx(t)}{dt} \right) + k(\dot{X}(t) - \dot{x}(t)) &= F(t) \\
m \frac{d^2 x(t)}{dt^2} + C \left( \frac{dx}{dt} - \frac{dX(t)}{dt} \right) + k(x(t) - \dot{X}(t)) &= 0
\end{align}
$$

(1a, 1b)

In the frequency domain, $F(t)$, $X(t)$ and $x(t)$ can be written as:

$$
F(t) = F_0 e^{-j\omega t}, \quad X(t) = X_0 e^{-j\omega t}, \quad x(t) = x_0 e^{-j\omega t}
$$

(2,3,4)
It is convenient to introduce a dimensionless frequency defined as:

$$\Omega = \frac{\omega}{\omega_0} \text{ with } \omega_0^3 = \frac{K}{M} \quad (5,6)$$

Assuming that both the primary and the top panel are simply supported around their edges and vibrating in a beam-like mode, their natural frequencies are defined as:

$$\omega_S^2 = \frac{E_S I_S}{\rho_S A_S} \left( \frac{m\pi}{2L} \right)^4 \text{ and } \omega_{TP}^2 = \frac{E_{TP} I_{TP}}{\rho_{TP} A_{TP}} \left( \frac{n\pi}{2l} \right)^4 \quad (7,8)$$

In equations 7 and 8, n and m are the mode numbers of the primary panel and of the top panel. Both of them are assumed to be made of the same material. The reference frequency is $\omega_0 = \omega_S$.

An important parameter that describes the degree of coupling between the two plates and that, as we will see later on, strongly influences the damping induced by the air film is the ratio between the two natural frequencies given in formulas 7 and 8:

$$\epsilon = \frac{\omega_{TP}^2}{\omega_S^2} = \frac{n^4 L^4 h^2}{m^4 l^4 H^2} \quad (9)$$

For $\epsilon=1$ (tuned damper), the primary panel and the upper panel can be described as “fully coupled”. The mass ratio of the top plate over the structural plate is defined as:

$$r = \frac{m}{M} = \frac{lhw}{LHW} \quad (10)$$

Furthermore, the stiffness ratio is defined as:

$$k = \frac{m \omega_{TP}^2}{M \omega_S^2} = \frac{M \omega_{TP}^2}{M \omega_S^2} = \frac{n^4 wh^3 L^3}{m^4 WH^3 l^3} \quad (11)$$

A dimensionless value of damping is introduced:

$$\gamma = \frac{C}{\omega_0 M} = \frac{rC}{\omega_0 m} \quad (12)$$
The equations of motion can be reformulated as:

\[
\begin{cases}
(1 - \Omega^2)X_0 + j\gamma\Omega(X_0 - x_0) + r\varepsilon(X_0 - x_0) = \frac{F_0}{K} \\
- r\Omega^2 x_0 + j\gamma\Omega(x_0 - X_0) + r\varepsilon(x_0 - X_0) = 0
\end{cases}
\]

(13a, 13b)

The damping value, \( \gamma \), depends on the effective mass of the structural plate. Assuming that the kinetic energy of the plate corresponding to a certain vibration amplitude \( A \) is the same as the kinetic energy of a beam of length \( 2L \), vibrating in its first simply supported mode at frequency and peak amplitude \( A \) one obtains:

\[
M_{\text{eff}} = \frac{M_s}{2}
\]

(14)

In the same way, an equivalent frequency independent dashpot is defined assuming that the energy dissipated per cycle in the air film at a relative displacement \( \Delta \) and frequency \( \omega \) is equal to the one dissipated per cycle by a linear viscous dashpot, \( C_{\text{eff}} \), at vibration amplitude \( \Delta \) and frequency \( \omega \). Then, for an open-air cavity in the first simply supported mode under the assumption of high viscosity and incompressibility, the energy dissipated is:

\[
D = \pi C_{\text{eff}} \omega \Delta^2 = 8\pi \mu \omega \frac{lW^3}{t^3} = 8\pi \mu \omega \Delta^2
\]

(15)

The parameter \( \mu \) is the viscosity of air. The dimensionless value of damping becomes:

\[
\gamma = \frac{C_{\text{eff}}}{\omega \cdot M_{\text{eff}}} = 16\mu \frac{lW^3}{t^3 \omega \cdot M_{\text{eff}}}
\]

(16)

By assuming that the AFD is tuned (\( \varepsilon=1 \)), the response of the structural plate is then:

\[
\frac{X_0}{F_0/K} = \frac{(r + j\gamma\Omega) - r\Omega^2}{(1 - \Omega^2)(r + j\gamma\Omega) - r^2(1 - \Omega^2 + r + j\gamma\Omega)}
\]

(17)
2.3 Implementation on a simple structure

In order to study the performance of an AFD, a measurement device called ‘RTC-III’ (see Figure 3) is used. RTC-III stands for ‘Rayonnement des Tôles de Carrosserie’ and it is employed in a lab environment to reproduce the dynamic phenomena that can be observed on a vehicle floor. A panel mounted on the RTC-III is excited by an enforced motion generated by a shaker attached to a frame that is connected to the panel itself around its boundary. The damping performances are investigated by comparing the transfer functions between the shaker and some points on the plate when different configurations are measured.

![RTC-III device](image)

Figure 3: RTC-III device

Figure 4 shows the experimental set up of an AFD system described by the two-degree of freedom model discussed in the previous paragraph.

![Steel plate suspended over the structure](image)

Figure 4: Steel plate suspended over the structure
2.3.1 Investigations about the thickness of the air film damper

In order to have a meaningful experimental set up, the two-degree of freedom model has been used to set the thickness of the air gap in such a way to make it effective (in terms of damping added to the primary structure) in a frequency range that is strictly included inside the measurement frequency range. Using equation 17, the response of the structure is computed with the following parameters for the bottom panel:

- Material: Steel
- M: 2.3 kg
- 2L: 573 mm
- 2W: 450 mm
- H: 1 mm

The parameters used for the top panel are the following:

- Material: Steel
- m: 1.2 kg
- 2l: 400 mm
- 2w: 400 mm
- h: 1 mm

The excitation is 1 N and the AFD is assumed to be tuned (ε=1). Figure 5 presents the response of the structure at 96 Hz for different (normalized) frequencies and air gap thickness values.

![Figure 5: Response of the structure at 96 Hz, (Q amplification at the resonance)](image)
In Figure 5 the response of the structure is represented in terms of $Q$, the vibration amplification at resonance. This quantity is linked to the damping by the following relationship:

$$\delta = \frac{1}{\max(|Q|)}$$  \hspace{1cm} (18)

From Figure 5, one sees that for a very small air gap, the system behaves as a single-degree of freedom model. The resonance frequency of the system does not differ from the one of the structural plate. For large air gap, it behaves as a double-degree of freedom system and in this case one of the peaks corresponds to the panels moving in phase and the other to the panels moving out of phase. The damping reaches its maximum value in the transition region between the single-degree of freedom behaviour and the two-degrees of freedom behaviour.

By calculating the response of the structure for each frequency, the damping coefficient can be evaluated as a function of the air gap thickness and frequency. The result of such calculation is reported in Figure 6.

![Figure 6: Damping as function of frequency and air gap thickness](image)

Considering the system under study, a high damping level is provided between 150 Hz and 350 Hz with an air gap of 0.7 mm.
2.3.2 Measurement set-up

The transfer function between the shaker and the primary plate is measured and averaged over 4 points, presented in Figure 7.

![Figure 7: Accelerometers positions](image)

The accelerometers are placed on the side facing the shaker (see Figure 3). The measurement devices are:

- 4 accelerometers *Piezotronics*, Type: 352C34, Serial numbers: 39542, 39534, 39539, 39538
- 1 Shaker
- 1 force transducer, *Kistler*, Type: 9311 A, Serial number: 136071
- 1 amplifier, *B&K*, Type: 2618, Serial number: 814606
- 1 signal conditioner, *Tescon*, Type: 4000
- Acquisition station: *LMS mobile*, Type: SC31, Serial number: 41000605

The measurement chain is presented in Figure 8.

![Figure 8: Measurement chain](image)
The structure is excited with a random noise from 20 Hz to 1200 Hz. The acquisition settings are the following:

- Number of spectral lines: 1024
- Resolution: 1 Hz
- Number of averages: 20

2.3.3 Results

Figure 9 presents the coherence and the averaged FRF of the bare plate up to 1 kHz.

![Figure 9: Bare measurement](image)

The coherence is globally acceptable. The deterioration of the coherence that appears between 600 Hz and 900 Hz could be due to the interaction between the frame and the plate. Concerning the transfer function, from 100 Hz to 500 Hz, the resonance peaks can be clearly identified. Above this frequency range, there is a drop of approximately 10 dB that could be related to a strong resonant behaviour of the frame.
Figure 10 presents the effect of suspending a panel over the structure. The air gap thickness is 0.7 mm, according to the calculation reported in paragraph 2.3.1. The result is presented in two plots, the first one focusing on the frequency range where the effects of the air film damper are expected (from 0 Hz to 400 Hz) and the second one focusing on the full measured frequency range (up to 1 kHz).

![Figure 10: Effect of the air film damper on the structure](image)

The measurements qualitatively confirm the model predictions. A vibration reduction of about 5 dB on the peaks is observed between 100 Hz and 350 Hz. Compared to the bare configuration, the peaks are shifted down in frequency. Above 400 Hz, there is no clear effect. At some frequencies, vibrations of the structure are even amplified compared to the bare measurement.
2.4 Comparison with other damping mechanisms

2.4.1 Viscoelastic damping and friction damping

The first results show that the presence of an air film reduces the vibrations of the primary structure. It is important to evaluate the efficiency of an AFD in comparison to other damping devices usually employed. Moreover, also in a solution exploiting the AFD, other damping mechanisms could be simultaneously included.

These mechanisms are:

- Viscoelastic damping
- Friction damping

The viscoelastic damping [2] occurs by using polymeric materials. The main property of these materials is the broad area of transition between a crystalline behaviour and a fluid behaviour. In this area, the long molecule chains become flexible and relaxation processes leads to damping.

Friction damping takes place when a plate is laid on the structure to damp. The relative motion in between the two structures leads to friction and then dissipation of energy.

In order to reproduce these mechanisms, a damping layer is used in two different configurations. It has the following characteristics:

- Length: 0.4 m
- Width: 0.3 m
- Thickness: 4.3 mm
- Weight: 0.64 kg

The viscoelastic dissipation is reproduced by gluing the damping layer on the primary structure and the friction damping by simply laying it on the same primary structure.
Figure 11 presents the comparison between viscoelastic damping and friction damping.

Globally, the performances of the viscoelastic damping layer look better than those obtained by means of friction damping. This is particularly evident by comparing the amplitude vibrations at the resonance peaks below 500 Hz. In the case of a glued damping layer, the peaks seem to be shifted up in frequency. It means that gluing the damping makes the structure stiffer.
The comparison between AFD results and friction damping results is presented in Figure 12.

![Figure 12: Air film damping compared to friction damping](image)

The vibration reduction achieved by means of the AFD is generally similar to the one obtained by means of the friction damping. In the comparison, it has to be considered that the surface treated with friction damping is 0.3 m by 0.4 m while the air film damper covers a surface of 0.4 m by 0.4 m. Between 400 Hz and 1 kHz, the response of the structure is smoother in the case of friction damping.
2.4.2 Combined mechanisms

The results reported in the previous paragraphs show that viscoelastic dissipation is the most efficient mechanism (among the investigated ones) for the reduction of panels vibrations. The next step is to combine friction damping and air film damping, and to compare the results so obtained with viscoelastic damping.

Requirements from a previous study [3] are taken into account:

- Intricated air cavities are not important for the AFD efficiency
- Variable air gap thickness increases the AFD performance

Two samples are realised taking into account these requirements. Both of them are made from a damping layer, have the same size (0.3 m × 0.4 m) and weight (0.64 kg). The air gap thickness is between 0.7 mm and 0.9 mm. Damping is only due to different boundary conditions. The first one is called ‘Embossed damping layer laid-on’. It has the following characteristics:

- Large contact area
- Low intrication
- Constant air gap thickness

Its profile is presented in Figure 13.

![Figure 13: Profile of the embossed damping layer laid-on](image)

The second one is called ‘Pyramidal damping layer laid-on’. It has the following characteristics:

- Small contact area
- Low intrication
- Variable air gap height

Its profile is presented in Figure 14.

![Figure 14: Profile of the pyramidal damping layer laid-on](image)
The comparison between the two configurations and the viscoelastic damping is presented in Figure 15.

**Figure 15: Combined configurations compared to viscoelastic damping**

The combination of air film damping and friction damping does not reach the efficiency of the viscoelastic damping. Concerning the effects of a variable air gap thickness and intricated air cavities, it seems that they have a limited influence on the air film damping performance.
2.5 Evaluation of the driving parameters

The measurements show that an AFD is not as efficient as a viscoelastic damping layer, even if it is combined with friction damping. For this reason, it is important to investigate which are the parameters that drive the performance of the AFD, in order to have a better understanding of its behaviour and to be able to optimise it.

The parameters responsible for the dissipation of energy in the AFD can be identified from Equation 16 that for convenience we report again here below:

\[
\gamma = 16\mu \frac{lw^3}{t^3 \omega_0 M}
\]

(19)

As one can see from this equation, the parameters responsible for the dissipation are:

- Air gap thickness, t
- Top panel dimensions, l and w
- Air viscosity and primary structure natural frequency, \( \mu \) and \( \omega_0 \)
- Mass of the structure to damp, M

Only the influence of the air gap thickness is here investigated. The impact of the weight of the upper panel is also studied. It is an important aspect as a steel top panel is not a realistic solution.
2.5.1  Air gap thickness

Three different air gap thickness values are tested: 0.7 mm, 1.4 mm and 2.1 mm. The measurement set-up is the same as in Figure 4. The results are presented in Figure 16.

Below 200 Hz, the measured configurations behave according to the model prediction (see Equation 19): the smallest thickness is the most efficient. Above this frequency, the effect of the air film thickness on the plate vibration is almost negligible (this is also in agreement with the two-dofs model previously presented).
2.5.2 Top panel material and weight

In order to investigate the influence of the top panel material and weight, a Poly Propylene (PP) plate of 0.3 kg is suspended over the structure. The corresponding results are presented in Figure 17.

Below 400 Hz, the weight and the material of the top panel do not have almost any impact on the performance of the AFD. This is in line with the predictions of the two-dofs model, since the mass of the top plate is not taken into account in the dimensionless value of damping. It could be noticed anyway that above 400 Hz, the vibration reduction achieved with the PP plate is slightly better than with the steel plate. This might be due to the fact that the intrinsic damping of PP is likely to be higher than that of steel.
2.6 Energy flow analysis

The results presented in the previous paragraphs show that the dynamic behaviour of the top plate used to form the air film has an influence on the energy dissipation process. This means that if on one side the viscosity of the air allows to dissipate energy on the other side the elasticity of the same air allows that a part of vibration energy is transferred to the top plate. In order to verify this, three accelerometers are placed on the top plate to evaluate its averaged FRF. In all these measurements, the PP plate was considered, with an air gap of 0.7 mm.

Figure 18 presents the comparison between the primary plate vibration and the top plate vibration.

![Comparison between the structure and the top plate vibration](image)

Although there is no contact between the structure and the top plate, vibrations of the top plate are measured. It confirms that the elasticity of the air film allows for some vibration transmission.

Below 500 Hz, the top plate vibration is remarkably lower than that of the structural panel. From 500 Hz to 800 Hz, the vibration level is similar for the structure and the top plate. Out of the working range of the AFD, the vibration transmission reaches a high level.
2.7 General analysis

First of all, concerning the efficiency of an AFD, the presence of the air film reduces the vibrations of the primary structure. Compared to friction damping, the AFD provides similar result. Nevertheless, the viscoelastic damping seems to be the most efficient of the three mechanisms overall the frequency range. The combination of air film damping and friction damping damps the structure below 500 Hz. The intrication of the air cavities and a variable air gap have a limited effect.

About the driving parameters, the air gap thickness and the top panel material have a slight influence on the damping performances.

Concerning the dissipation mechanism, the viscosity of air is responsible. Nevertheless, a part of the energy goes to the upper panel, but in a relatively small proportion.
3 Top plate vibration

In part 2, the results show that the presence of a constrained air film reduces the vibration of the primary structure. Nevertheless, vibrations of the top plate are observed. As this panel is facing the car compartment, it is important to understand its dynamic behaviour and to reduce as much as possible its vibration level.

In the first part of this chapter, measurements with a supported top plate put over the primary structure are carried out and compared with the configuration in which a plate was suspended above the primary structure itself. This comparison is considered because in an application of the AFD on a vehicle, design solutions that include supports cannot most probably be avoided. Then, different parameters are studied in order to identify the ones that could reduce the top panel vibration. The output of this investigation is a list of requirements.

According to this list, measurements are realised with a scanning laser head in order to have precise quantification of the reduction of the vibration of the top plate (with respect to the vibration of the primary structure).
### 3.1 Supported configuration

The top panel is made of PP and the supports of steel. The thickness is 0.7 mm to keep the AFD effective between 150 Hz and 350 Hz. The results are presented from 0 Hz to 400 Hz and from 0 Hz to 1 kHz.

Up to 400 Hz, the presence of supports seems to reduce the performances of the AFD; this result can be explained by the fact that the supports change the energy coupling between the top and the primary plate, compared to hypothetical design solutions that do not exploit supports.

Above 400 Hz, the presence of the supports does not seem to have relevant effects. In order to minimise the vibration of the top plate, a study aimed at identifying the optimal supports configuration is presented in the following section.
3.2 Optimisation of the supports

The parameters that are investigated are the following:

- Material
- Effect of gluing the supports
- Thickness
- Area of contact and position

They have been selected according to the possibility to control them in production. Other parameters have been neglected. For all the measured configurations, the top plate is made of PP.

3.2.1 Material

Three different materials have been selected and tested according to their stiffness: steel, rubber and non-woven tissue. The obtained results are presented in Figure 20.

![Material supports](image)

Figure 20: Material supports

It can be observed that the supports made of steel are the least effective. Below 400 Hz, the non-woven tissue is more efficient than the rubber supports with a difference around 5 dB at the peaks.

Between 400 Hz and 600 Hz, the supports made of rubber are the most performing solution allowing to achieve a reduction of about 20 dB compared to the vibration level of the bare plate. Above 600 Hz, the highest vibration reduction is achieved by the non-woven tissue supports. The analysed
configurations suggest that (as it had to be expected) mid-low stiffness materials should be chosen for the supports, to get a lower vibration of the top plate.

### 3.2.2 Effect of gluing the supports

The impact of gluing the supports, either to the top panel or both to the primary and to the top panel is an important parameter because it may influence the vibration transmission.

Three configurations are compared: supports not glued, neither to the structure nor to the top panel, supports glued only to the top panel and glued to both panels. For these three configurations, the supports are made of rubber. The results obtained are shown in Figure 21.

![Figure 21: Effect of gluing the supports](image)

Overall the frequency range, the vibration level between the three configurations is similar, except between 400 Hz and 600 Hz, where the configuration without glued supports shows much better performance. Overall, anyway, the effect of gluing/not gluing the supports cannot be considered as very relevant.
3.2.3 Thickness

Two thickness values are tested: a small and a large value. In both cases, the supports are made of rubber.

For the selected material, it can be observed that above 500 Hz the thick supports are more efficient than the thin ones. Below 500 Hz, the thick supports are more efficient than the thin ones only between 200 Hz and 350 Hz. These observations suggest that in the upper part of the frequency range thick supports provide better performances than thin ones. But at low frequency, further investigations would be necessary to get clear indications.

Figure 22: Thickness of the supports
3.2.4 Contact area and position

These two parameters are investigated at the same time. Rubber supports are tested in three configurations: high number of supports, mid-number of supports and low number of supports. The result is presented in Figure 23.

![Figure 23: Effect of the number of supports](image)

It can be observed that the configuration with a high number of supports is the least efficient compared to the other two. The comparison between the configuration with a low number and a mid-number of supports is less clear. At high frequency (above 500 Hz) the low number configuration seems to behave better, but at low frequency this is not always the case. These results suggest that there exists an optimum configuration to minimise the vibration of the top plate and that this optimum might also depend on the targeted frequency range.
3.2.5 Effect of the supports on the structure

To complete the investigation concerning the supports, the vibrations of the primary plate are compared to the vibrations of the top plate and the vibrations of the bare configuration. Then, the FRF of the structure and the top plate is acquired at the same time. For this test, the supports are made of rubber.

It can be observed that the primary structure is almost not damped compared to the bare configuration. On the other side, for the top plate, the vibration reduction goes up to 30 dB from 100 Hz to 1 kHz, when compared to the primary plate. These results suggest that for the considered thickness the effects of the AFD on the structure are negligible; concerning the top panel vibration, three potential reasons can explain the vibration reduction.

The first one is that a large amount of energy is dissipated in the supports because of their viscoelastic properties. The second one is the small area of contact between the plates that reduces the amount of transmitted energy. The third one is that the rubber supports behave as a vibration isolator and decouple the top plate from the primary structure.
To verify this last possibility, the system consisting in the supports and the top plate is assumed to behave as a mass-spring system. The resonance frequency is then:

\[ f_{res} = \frac{1}{2\pi} \sqrt{\frac{EA}{tm}} \]  

(20)

\( E \) is the Young’s modulus of the supports, \( A \) is the total area of the supports, \( t \) is the thickness of the supports and \( m \) is the reduced mass of the two plates. According to the material data, the resonance frequency turns out to be:

\[ f_{res} = 5 \text{ kHz} \]  

(21)

This means that for rubber supports, the top plate is not isolated from the structure and that an explanation for the vibration isolation of the top plate has to be searched in the other mechanisms mentioned above.

### 3.3 Requirements

For the measured configurations and the selected materials, the material properties and the thickness of the supports are important parameters.

A material stiffness between the one of the rubber and the one of the non-woven tissue gives a reduction up to 30 dB compared to the bare configuration.

In the case of the rubber supports, above 500 Hz it seems that the thicker are the supports, the lower are the vibrations of the top plate.

Gluing the supports to the top plate has a limited impact on its vibrations.

Concerning the position of the supports and the contact area, it seems that there is an optimum configuration. This configuration could be identified by means of the analysis of the velocity distribution of the primary structure. This analysis would allow to position the supports in regions of the primary structure that have a relatively low level of vibration.
3.4 Estimation of vibration reduction

The measurements reported in the previous section have been done to observe the physical phenomena related to the presence of the thin air film between the primary plate and the top plate and to identify the parameters that influence the performance of the AFD and the vibrations of the top plate. In this part, the focus is to quantify the vibration reduction achieved by means of the supported PP plate. In order to do this, two configurations with different supports are tested and compared to a reference solution. The number of measurement points is also increased with respect to the previous measurements.

3.4.1 Measurement set-up

The RTC-III is still used for this measurement campaign. The measurements are realised with a scanning laser head, shown in Figure 25. It allows to have quantification of the vibration level of the top plate that does not depend on the position of the measurement points.

![Figure 25: Scanning laser head](image)

The measurement devices are:

- 1 scanning laser head, Polytec, Type: PSV 400, Serial number: 6042326
- 1 vibrometer controller, Polytec, Type: OFV 500, Serial number: 5042089
- 1 pre-amplifier, Tescon, Type: 1027B, Serial number: 3766
- 1 amplifier, B&K, Type: 2618, Serial number: 814606
- 1 shaker
- 1 force transducer, Kistler, Type: 9311 A, Serial number: 136071
The averaged FRF is evaluated over 36 measurement points showed in Figure 26.

![Measurement points](image)

**Figure 26: Measurement points**

The structure is excited via a periodic chirp, on the frequency range from 20 Hz to 1200 Hz. The acquisition is done between 20 Hz and 1 kHz, with a resolution of 0.7813 Hz.

Three configurations are measured. Two of them follow the requirements previously established. In one case, the supports are made of rubber. In the other case, they are made of porous material. The thickness of the rubber supports is smaller than the one of the porous material supports.

In the third configuration that is used as a reference, a damping layer is glued to the structure. This damping layer has the same dimension and weight than that of the PP plate.
3.4.2 Results

The comparison between the results obtained in the two “supported” configurations is presented in Figure 27. The comparison with the reference solution is presented in Figure 28.

On average, the vibration of the top plate is lower when porous material supports are used. This is true in particular above 200 Hz. Below this frequency, at the resonance peaks the vibration levels of the top plate with porous material supports are similar to those with rubber supports. It can be observed that the top plate looks more damped when supports made of rubber are employed.

Figure 27: Comparison between the supported configurations
The “supported” configuration with porous material supports shows an average vibration that is lower also than that obtained in the reference configuration (Figure 28). It has to be observed again, though, that in this supported configuration the levels at the resonance peaks are even a bit higher than those obtained in the reference configuration below 250 Hz. In general, in this supported configuration the peaks look quite “sharp” (i.e. with low damping) even when their level is relatively low.

Figure 28: Comparison with a reference solution
4 Simulation

To validate the assumptions and guesses done in the interpretation of the results of the experimental activity, a finite element analysis of the supported configuration has been performed. With this analysis, it is meant to have just a qualitative value since it is not possible, for time reasons, to carry out a full simulation of the RTC-III device. The value of the analysis consists in giving the possibility to better understand the phenomena involved in the transfer of energy from the primary plate to the top plate and also in giving the possibility to understand which is the role of the thin layer of air in this same transfer of energy.

4.1 Description of the FE model

Three configurations are modelled: the bare plate, the “supported” configuration with PP plate having rubber supports and the “supported” configuration with PP plate having porous material supports.

The structure and the top plate are modelled as shell elements. The supports are modelled as solid elements. The air between the two plates is modelled using fluid solid elements. On the free side faces of the air layer, open-air boundary conditions are defined by means of the characteristic impedance $\rho_0c$. The calculation is done up to 600 Hz with a direct analysis using the commercial software MSC/Nastran.

The loss factor of the thin air layer is estimated from the dimensionless value of damping coefficient (see Equation 16). Since this damping coefficient is air gap thickness and frequency dependent, it is first calculated for the first 4 plate modes for both supported configurations and then an average is taken. The resulting average loss factor for the case of rubber supports and for the case of porous material supports is respectively 0.34 and 0.0055.
4.2 Results

Figure 29 presents the results from the simulation of the three configurations presented above.

![Simulation results](image)

**Figure 29:** Simulation results

The model seems to qualitatively reproduce well the phenomena observed during the measurements. For both supported configurations, the resonance peaks have similar amplitude but they are less damped in the case of porous material supports.
4.2.1 Effect of the air cavity on the upper panel vibration

In the case of rubber supports, it can be observed that neglecting the air between the two plates leads to an increase of the vibration level up to 15 dB. Then, it is important to take into account the air film in the model to have relevant results.

Figure 30: Effect of the air film on the top plate
4.2.2 Comparison between the primary plate and the top plate vibrations

In Figure 31, it can be observed that the top plate and the primary plate vibrate at the same level up to about 300 Hz. On the other hand, the reduction of the vibration on the primary plate (with respect to the bare case) is extremely relevant (up to 20 dB).

Figure 31: Primary structure and top plate vibration with rubber supports
From Figure 32, it seems that the upper plate with porous material supports is decoupled from the primary plate already above 50-100 Hz. For this supported configuration, it seems that the porous material supports act as vibration isolators. Concerning the primary plate, there is at maximum a 10 dB difference between the peak vibration levels in the bare and in the “supported” configuration with porous material supports.

Figure 32: Structure and upper panel vibration with porous supports

From Figure 32, it seems that the upper plate with porous material supports is decoupled from the primary plate already above 50-100 Hz. For this supported configuration, it seems that the porous material supports act as vibration isolators. Concerning the primary plate, there is at maximum a 10 dB difference between the peak vibration levels in the bare and in the “supported” configuration with porous material supports.
5 Conclusion

The two-degree of freedom model of the air film damper gives relevant information and helps to find the optimum air gap thickness for a given frequency range to control. The implementation on a simple structure shows that vibrations are reduced at the expected frequencies. A comparison with friction damping reveals that it allows to obtain a similar vibration reduction. But the viscoelastic dissipation is the most efficient mechanism, even compared to a solution combining friction damping and air film damping.

The performance of the air film damper deteriorates when the air gap thickness is increased. On the other side in such case a strong reduction of the vibrations of the top plate has been noticed. This phenomenon is quite interesting, because the top plate is the interface between the structure and the cavity.

The investigation of the top plate vibrations, in a supported configuration, shows that the material properties and the thickness of the supports are important parameters. The comparison with a reference damping solution reveals that at low frequencies the supported configuration in general needs to be improved.

The FE model could qualitatively reproduce the observed phenomena. They showed that the fluid of the air gap has to be modelled to have a relevant estimation of the vibration level of the top plate. The model also reveals that choosing the right material data for the supports allows decoupling the top plate from the structure already at very low frequencies.

Three aspects of the study require further investigations. The first one concerns the measurement activity. A possibility to improve the behaviour of the supported plate in the low frequency domain is to test a material with higher loss factor and a smaller E-modulus than the rubber. The second aspect is about the simulation. As the model behaves in a similar way than the measured configuration, it may be used to find a better configuration for the position and the size of the supports than the one currently used. The last aspect concerns the acoustic performance of the system under study that, for time reasons, was not investigated during the activity here reported. This acoustic performance is of course an important point, since the final goal is always to reduce the sound pressure level in the car compartment.
6 Bibliography


