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INGÉNIEURS DE L'AUTO

JANVIER 2022 # 875

Interview

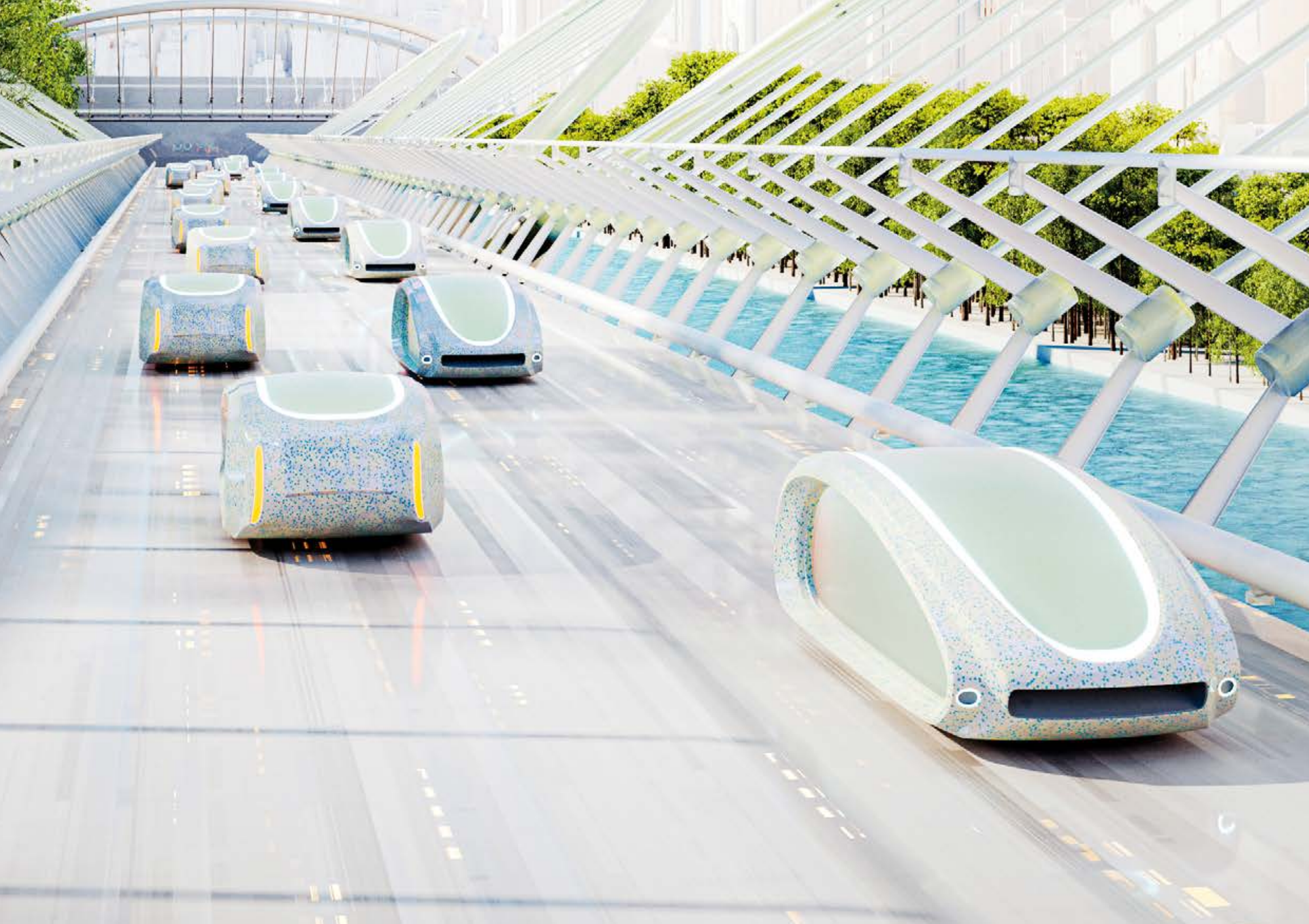
Vitesco Technologies, nouvel équipementier majeur

Pleins Feux

Nouvelles technologies et distribution automobile, 2 mondes liés

Dossier

Conduite automatisée sur terre, mer et dans les airs



Congrès SIA-CTTM NVH Confort

Journée d'Étude SIA-FIEV Management des connaissances

Conférence SIA : Comment faire confiance à l'IA dans l'automobile ?

SIA SOCIÉTÉ DES INGÉNIEURS DE L'AUTOMOBILE

Sommaire

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Edito

Luc Marbach
Président de la SIA

En ce début d'année, je vous présente au nom de la SIA mes meilleurs vœux pour 2022, de santé pour vous et vos proches, et de plein succès dans vos projets personnels et professionnels.

Souhaitons également que notre filière automobile, fortement secouée par la crise sanitaire et par la crise des semi-conducteurs, retrouve, grâce aux brillantes compétences de ses acteurs, les ressources indispensables pour mener à bien les nombreuses transformations engagées.

Ce numéro reflète la multiplicité de ces évolutions, voire révolutions.

Sur **l'électrification**, vous découvrirez au travers de l'interview de son CEO, Stéphane Fregosi, comment Vitesco Technologies France s'appuie sur ses technologies dans le domaine des motorisations électrifiées pour se développer, notamment en France.

La synthèse sur les **semi-conducteurs** – comment ils sont fabriqués et testés – vous permettra de mieux appréhender la complexité de cette industrie, devenue vitale pour l'automobile.

La C.E. Electronique et Architectures Electroniques et Logicielles vous fera découvrir les **futures architectures hardware et software** des véhicules.

Si l'électrification va réduire le besoin d'entretien des véhicules, de nouveaux besoins considérables émergent, comme la maintenance des calculateurs et logiciels embarqués. Le CNPA et ses adhérents préparent cette **révolution des métiers de l'après-vente.**

Nous connaissons moins le monde du **poids lourd** ; dans ce domaine le jeu est plus ouvert entre les solutions de **réduction de l'empreinte carbone** que dans l'automobile.

Plutôt que le grand tsunami annoncé il y a encore peu, c'est à la faveur de multiples avancées techniques que les **véhicules automatisés** progressent. Notre grand dossier vous fera découvrir les **technologies** mises en œuvre sur les VA, et pas seulement sur route : sur **rails, dans les airs, sur et sous la mer** également.

L'Intelligence artificielle est essentielle pour les fonctions autonomes, comment faire pour **avoir confiance dans l'IA** ? Deux experts font le point.

Et comme il y aura encore longtemps un conducteur, **Core for Tech** innove avec une solution d'alerte de perte de vigilance, simple et efficace.

Améliorer le **confort acoustique intérieur, diminuer le bruit émis…,** quels étaient les points clefs de la dernière édition du congrès Confort NVH ?

Alors, branchez vos tablettes ou ouvrez vos revues, je suis certain que vous serez, comme nous, enthousiastes devant toutes ces transformations, engagées ou à venir !

Un congrès Confort NVH qui redéfinit le développement de l'acoustique

Dynamisme, enthousiasme et ouverture vers de nouvelles données acoustiques marquent l'édition 2021 du congrès Confort NVH organisé par la SIA en partenariat avec le CTTM. Un événement qui s'est tenu dans une période de contraintes et d'exigences fortes pesant sur la définition des futurs véhicules.

Organisé les 13 et 14 octobre derniers au Mans par la SIA et le CTTM (Centre de Transfert de Technologie du Mans), le congrès Confort NVH avait à traiter de sujets majeurs en raison des évolutions technologiques et réglementaires.

Ce congrès intervient dans une période où l'acoustique automobile doit répondre à deux enjeux : la place du confort acoustique avec l'émergence des véhicules électriques, et la prise en compte de l'automobile dans l'environnement urbain. Le premier impose un changement de méthode de développement, explique Jean-François Rondeau, Directeur de l'innovation de Faurecia Clarion Electronics et président de la Communauté d'Expert NVH de la SIA : « *L'acoustique doit être traitée dès le début d'un projet de véhicule et définie selon des critères centrés sur l'utilisateur.* »

La prise en compte de l'automobile dans son environnement, l'autre grand enjeu du congrès, a notamment été largement présentée dans le dossier central de notre numéro de septembre dernier. Il est désormais fondamental de définir les réels contributeurs aux bruits dans les villes et de sortir de décisions plutôt politiques. Ce thème révolutionne par ailleurs les approches traditionnelles des ingénieurs et nécessite une collaboration avec de nouveaux acteurs, ce qui s'est traduit par un programme plus ouvert que celui des précédents congrès.

Ainsi, la présence de Bruitparif, considéré comme le champion européen des observatoires du bruit, a apporté une nouvelle dimension de la vision de cette problématique et de la façon de l'aborder. Sa compétence et sa richesse d'information, grâce notamment à sa capacité de mesure et d'analyse de l'environnement sonore urbain, ont éclairé de nombreux spécialistes lors de la table ronde. « *Les données des radars de Bruitparif nous permettent de documenter la réalité de l'environnement sonore et de démontrer que le bruit des véhicules n'est pas la source dominante de pollution sonore* », constate Thomas Antoine, Groupe Renault et VP de la Communauté d'Experts NVH de la SIA.
Les constructeurs, équipementiers, enseignant-chercheur et prestataires de solutions numériques ont présenté des papiers d'une grande valeur scientifique, portant entre autres sur les problématiques vibratoires, les contrôles actifs, les solutions innovantes d'isolation ou la simulation.

Les missions attribuées aux acousticiens ne concernent plus vraiment une réduction du niveau de bruit mais l'élaboration d'un confort acoustique. Par exemple, si la réglementation impose un niveau de décibels à l'avertisseur AVAS obligatoire sur les véhicules électriques, le travail des ingénieurs porte principalement sur un design sonore représentatif d'une identité de marque.
Ce congrès a également permis de réunir à nouveau le monde des acousticiens. Leur plaisir à se retrouver s'est manifesté par l'enthousiasme des participants comme des exposants

Yvonnick Gazeau

160 participants

15 exposants

3 sessions parallèles et **2** tables rondes, ***soit 60 interventions***

Le papier « Identification of damping added by a trim foam on a steel plate with an inverse vibration problem » par Meryem Le Deunf, est à découvrir page suivante

Identification of damping added by a trim foam on a steel plate with an inverse vibration problem

M. Le Deunf[1,2], C. Pézerat[2], F. Ablitzer[2], N. Merlette[1]

1 : CEVAA, 2 rue Joseph Fourier 76800 St-Etienne du Rouvray
2 : LAUM Laboratoire d'Acoustique de l'Université du Mans - UMR CNRS 6613, avenue Olivier Messiaen 72085 Le Mans cedex 09

Abstract: The lightweight of vehicles is a main target for the automotive industry. Researches are done to find new materials or methods to optimize the NVH comfort solutions. In vehicles, trim foams are classically used for sound insulation. Some previous researches show that trim foams can dampen vibrations under particular conditions. To understand which physical phenomena allow vibrations of panels to be reduced by foam, a new measurement test bench is made. This bench is made of a specific suspended steel plate, with a foam placed on it, where different cases of contact at the foam-plate interface (glued or not) are studied. A scanning laser vibrometer is used to measure the displacement field of the plate. With an inverse method, it is then possible to extract the global stiffness and damping of the system from the measured displacement field of the plate. Two inverse methods named the Force Analysis Technique (FAT) [1] and the Corrected Force Analysis Technique (CFAT) [2], are used to compute the characteristics of the sandwich structure. Finite Element Analysis is used to quantify the damping given by the foam, due to viscothermal losses and/or skeleton damping. Results on a specific foam show how the behavior can be very different if the foam is glued or not and how the added damping becomes optimum in the different frequency bands.

Keywords: Inverse method, damping, foam.

Introduction

The ecological impact of vehicles is a major issue of OEM's and changes in motorization push to reinventing the future cars. To decrease power consumption and reject, weight reduction can be a good part of the solution. New composite materials are created with high rigidity and damping, but it's not enough. Another solution is to limit the heavy bitumen pads used for vibration damping and replaced them with trim foam. These foams are present in vehicles for acoustic comfort. But due to their viscoelastic capacity, they can have a non-negligible effect on the damping. To know if this solution is sufficient. A typical bench is a set-up to quantify the damping provided by a trim foam on a steel plate. Different types of interfaces between foam and plate are tested to highlight several damping phenomena. An inverse method is used to extract structural parameters, like damping and elastic modulus, from the displacement fields of the system. This inverse method based on the verification of the plate motion equation is presented below. All results for various foam-plate interfaces are compared to determine the characteristics of the damping mechanisms involved.

Process to determine the damping of a two-layer structure

Set-up

A test bench is a set-up to characterize the global damping of the foam-steel plate system. The vibrating structure is a steel plate suspended at the four corners by nylon string. The foam plate is placed on it with a different interface (glue or not). The experimental set-up is present in Figure 1 and the plate in Figure 2.

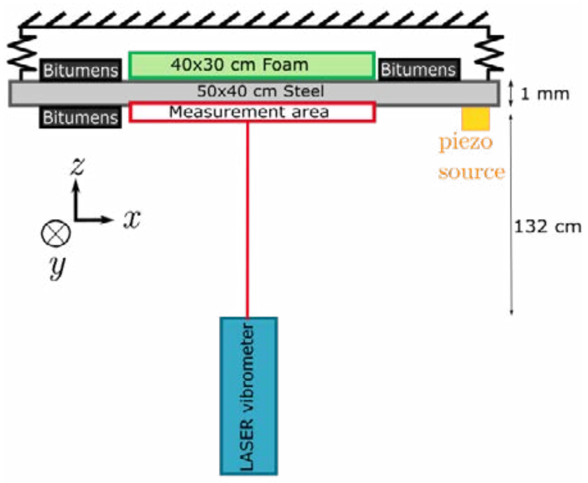

Figure 1.Schematic of the experimental set-up.

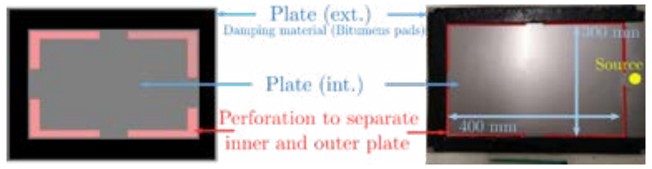

Figure 2.Plate to avoid boundary condition effects.

To limit the influence of the environment measurement, the steel plate, present in Figure 2, have an external part with bitumen pads to reduce global modes of the structure, and an internal part to quantify the damping. Only the inner plate is characterized because the foam is placed on this area. The study concerns a single foam. This foam is a polyurethane type used for sound insulation of vehicles. The out-of-plane displacement of the inner plate (in the z direction) is the quantity measured by a 1D scanning laser vibrometer. The used injected load is obtained by a piezoelectric source that is located outside the measuring area.

With CFAT and FAT methods, it is then possible to characterize the structural parameters of the sandwich panel, like the storage modulus and damping.

Inverse methods

The CFAT and FAT inverse methods consist in verifying locally the flexural equation of motion for a thin isotropic plate in a harmonic regime. The analysis is made in an area without sources so that the equation of motion describes equality between the stiffness and the mass terms:

$$\frac{D}{\rho h}\left(\frac{\partial^4 w}{\partial x^4}+\frac{\partial^4 w}{\partial y^4}+2\frac{\partial^4 w}{\partial x^2 \partial y^2}\right)=\omega^2 w. \qquad [1]$$

where D is the bending stiffness, ρ the density, h the thickness, ω the angular frequency, w(x,y) the out-of-plane displacement field. The flexural rigidity,

$$D=\frac{E'(1+j\eta)h^3}{12(1-\nu^2)}, \qquad [2]$$

introduces the Poisson coefficient ν and the complex modulus E'(1+jη) where η is the loss factor, which corresponds to an equivalent damping factor of the sandwich system. The equation of motion [1] describes a local equilibrium, it is then valid at any point in the structure, regardless of boundary conditions.

Knowing the displacement field w(x,y) of the structure and its spatial derivatives $\frac{\partial^4 w}{\partial x^4}, \frac{\partial^4 w}{\partial y^4}$, and $\frac{\partial^4 w}{\partial x^2 \partial y^2}$, it is possible to identify the term $\frac{D}{\rho h \omega^2}$, which may vary with the frequency. The real and imaginary parts of this term give the equivalent stiffness and damping of the sandwich. The characteristics of the structure can then be determined from the measured displacement field and the estimation of its spatial derivatives, The FAT and CFAT methods consists in approximate these derivatives by finite difference schemes so that the value of a local parameter can be obtained from measured neighboring displacements around the identification point [3], [4] and [5].

The FAT method is used to regularize errors due to measurement noise using a low-pass filter in wavenumber. The regularized equation with the filtering and the windowing is,

$$\frac{D}{\rho h}\left[\left(\left(\delta_{ij}^{4x}+2\delta_{ij}^{2x2y}+\delta_{ij}^{4y}\right).\psi_{ij}^{2D}\right)*H_{ij}\right] = \omega^2\left[\left(w_{ij}.\psi_{ij}^{2D}\right)*H_{ij}\right]. \qquad [3]$$

First, it is necessary to window (ψ_{ij}^{2D}) the signal to soften the discontinuities at the limits and avoid the negative effects of the filter (Gibbs phenomenon). This is done using a Tukey window.

The filter (H_{ij}), which removes high wavenumbers, is weighted by a Hanning window to keep the local aspect of the method. This filter eliminates the amplification of errors associated with the inverse problem. It needs a regulation parameter depending on the measured are noisy (equal to 1 for noisy measurements to 4 for very good measurement).

The CFAT method is a modification of FAT that regularizes the inverse resolution by controlling the filtering provided by the discretization of the motion equation. This method consists in introducing correction coefficients into the finite difference schemes, to benefit from a low-pass wavenumber filtering effect correcting the errors introduced into the resolution. The corrected discretized equation of motion is written as,

$$\frac{D}{\rho h}\left(\tilde{\mu}^4\delta_{ij}^{4x}+2\tilde{\nu}^4\delta_{ij}^{2x2y}+\tilde{\mu}^4\delta_{ij}^{4y}\right)=\omega^2 w_{ij} \qquad [4]$$

and corrective coefficients,

$$\tilde{\mu}^4=\frac{\Delta^4 k_f^4}{4\left[1-\cos(k_f\Delta)\right]^2},$$

$$\tilde{\nu}^4=\frac{\Delta^4 k_f^4}{8\left[1-cos\left(\frac{k_f\Delta}{\sqrt{2}}\right)\right]^2}-\tilde{\mu}^4.$$

The corrective terms require knowing the natural bending wavenumber of the plate

$$k_f^4=\frac{\rho h}{D}\omega^2 \qquad [5]$$

In our study, this wavenumber k_f is unknown because it depends on the characteristics of the sandwich panel (plate+foam). Characteristics are identified iteratively, with a first iteration without correction (i.e. by imposing $\tilde{\mu}^4=\tilde{\nu}^4=1$), to provide a first initial value of k_f. At each iteration, the value of $\frac{D}{\rho h}$ is identified, which provides a new estimate of k_f. According to the system, the number of iterations is not the same, in our case 5 iterations are necessary to ensure the convergence of the inverse problem.

The CFAT method allows characterizing structural parameters for high frequencies depending on the spatial step. To identify the characteristics of the plate over a large frequency range, it is necessary to use it in combination with the FAT method for medium frequencies.

Results

Multiple types of interfaces (glued or not) are tested to understand the effect of the interface on damping.

The damping is identified with the CFA, for high frequencies and FAT methods, for medium frequencies.

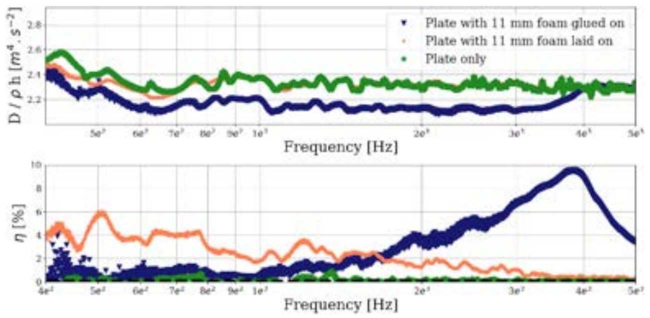

Figure 3. Elastic (D/ph) and damping (η) properties obtained from FAT/CFAT for a foam (11mm) glued or not to the plate and the plate.

In Figure 3, rigidity divided by the mass per unit and loss factor is presented according to frequency.

The foam has an impact on the rigidity depending on the type of contact with the steel plate. When the foam is glued on the plate, represented by the blue triangles, the rigidity is lower than when the foam is simply laid down. At 3 000 Hz, the rigidity increases until it reaches that of the other two measures. This variation is at the same frequency that the damping reaches a maximum. However, when the foam is laid on the plate, symbolized by the orange dots, the values are close to the plate alone. For this configuration, the foam seems not to provide any rigidity to the plate.

For the foam-steel plate structure, the foam brings damping, but depending on the interface, the damping is maximum for different frequency ranges. The damping is important between 2 000 Hz and 5 000 Hz when the foam is glued to the plate. For this interface, the damping reaches a maximum of 9.5% at 3 900 Hz. On the other hand, when the foam is not glued to the plate, the damping is high between 400 Hz and 1 500 Hz. The damping is high at low frequency and decreases when the frequency increases. The damping phenomena are different depending on the interface between foam and plate.

When the foam is laid on the plate, the rigidity does not change compared to the plate alone, but the damping contribution is non-negligible. The friction at the interface may be the mechanism that provides damping to the foam-plate structure. In the case of the foam glued on the plate, the damping peak is linked to the variation of rigidity. This fluctuation of damping and rigidity is typical of a viscoelastic effect of the foam, explain by Ungar et al. [6].

To understand, what mechanisms are present when the interface between foam and plate is simply laid on, other tests are done. The first assumption is that a thin layer of air is compressed between the foam and the plate [7]. It is necessary to change the disposition of the set-up to have a gap of air between the plate and the foam. The plate is positioned vertically to hook the foam to it, as shown in Figure 4.

Figure 4. Photography of the foam suspended very close to the vertical plate.

Another assumption is the energy transfer of the plate to the foam. By constraining the foam to the plate, the transfer may be maximized. A cardboard plate is used to have a three-layer structure, as presented in Figure 5: steel plate, foam, and cardboard plate. No glue is used for the two interfaces of the foam. Of course, the system is not the same, because the cardboard provided more rigidity to the system.

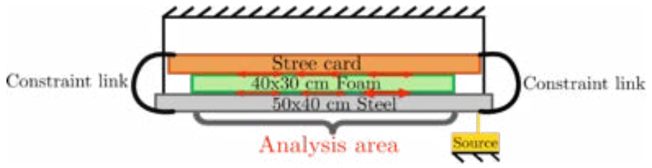

Figure 5. Schematic of the three-layer structure consisting of a cardboard plate, a foam, and a steel plate.

This configuration constraining the foam, but micro-friction may be present at each interface.

The results for all types of contact at the interface foam-steel plate are presented in the following figure.

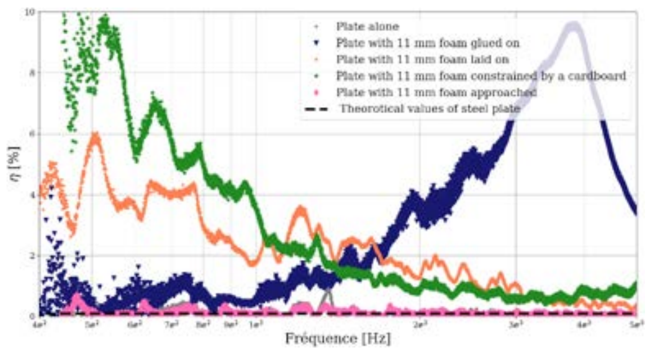

Figure 6. Damping (η) propertie depending on the frequency for various types of contact at the interface between foam and steel plate.

In Figure 6, four types of contact are compared to the plate alone. When the structure is vertical as in Figure 4, the foam is approached to the plate but is not in contact, it's represented by pink diamonds. In this case, the foam does not influence the damping of the plate. On other hand, when the foam is compressed by the cardboard, represented by the green stars. The damping is important at a low frequency between 400-1 000 Hz, more than 10%. But at high frequency, between 1 000-3 500 Hz, the damping is less than the result from the configuration of the foam placed on the plate, illustrated by the orange dot. Finally, after 3 500 to 5 000 Hz, the damping increase. The first result indicates there is no damping provided by a thin layer of air. Then, the assumption of an added damping due to the layer of air can be rejected. But, when the foam is constrained, the damping is more important at low frequency and is superior to the damping for the configuration when the foam is glued to the plate, represented by the blue triangle.

All configurations provide pieces of information on the damping sources brings by the foam depending on interface contacts. According to the frequency band to be damped, a type of contact is more interesting than another.

Conclusion

The test bench allows measuring the displacement of the multi-layer damping structure. By applying inverse methods named CFAT and FAT, it is possible to calculate the global damping of the structure. To understand the origin of the damping, several types of interfaces are tested. In the case of glued interface, the damping is linked to the visco-elastic comportment of the foam-plate structure. In the case of a not glued interface, the first hypothesis is a thin layer of air brings viscous effects. But the test of the foam approached to plate shows it is not this mechanism that provides dissipation to the plate. A second hypothesis is a friction at the interface. Another assumption is the energy transfer of the plate to the foam, with a viscoelastic dissipation of the foam. When the foam is not glued, it is not constraint by the movement of the plate. But when cardboard compresses the foam, it is more dependent on the motion of the plate. With CFAT and FAT methods, it is then possible to quantify the damping of a complex structure independent of the source of dissipation. The results show dependence between the type of contact and the frequency. More research is necessary to understand exactly what is the phenomena at the origin of the dissipation when the foam is laid on the plate ●

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