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ACOUSTIC PERFORMANCE EVALUATION OF A PANEL UTILIZING NEGATIVE STIFFNESS MOUNTING FOR LOW FREQUENCY NOISE CONTROL

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Abstract. *In order to encounter the challenges of low frequency noise absorption below 100 Hz, primarily emitted from the engines in air and ground vehicles, a method of acoustic panel mounting is proposed based on the KDamper concept which utilizes the negative stiffness damping effect. The practical constraints in such applications relating to the thickness of the panel for adequate noise absorption, are compensated through the unique damping properties of the KDamper and the ability to attenuate low frequency excitation in a considerable range. Therefore, a preliminary implementation of an acoustic panel with appropriately designed elastic mounts incorporating negative stiffness elements, is evaluated in terms of acoustic performance. During the design stage of the mounting, the problem is approached as in the case of classic vibration isolation of a rigid mass, due to the nature of the low frequency excitation. However, the elasticity of the deformable thin plate is taken into account by modeling its dynamic behavior using generalized modal values. The investigation consists in establishing the theoretical framework of the problem for the examination of metrics such as the sound transmission loss (STL) and the optimization of the mounting in order to maximize the STL performance, accompanied by FEM simulations. Additionally, comparisons with conventionally mounted panels is provided for reference and to highlight the advantages of the proposed solution. Early results regarding the noise control capabilities of such a panel configuration, indicate the potential of this proposition in specific applications.*

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1 INTRODUCTION

Researchers have reported several hazardous impacts of low-frequency noise on human health such as annoyance, headache, city-fatigue [5], etc. To attenuate low frequency sound, passive solutions such as foam layers, sound diffusing panels and blocks, sandwich panels and many others [14], constitute the most widespread means for acoustic treatment in a vast range of applications from room acoustics, to loudspeaker enclosures etc.. However, these means present certain limitations in the low frequency range, namely below 500 Hz approximately, which is very important regarding noise control in transportation media and inside buildings, among other applications. Traditional materials follow the mass density law for sound shielding. According to the mass-density law, sound transmission through a substance [19] is given by $T = \frac{1}{\rho t f}$, where ρ is density, t is thickness of the medium, and f is the sound wave frequency. Thus, for a particular frequency, doubling the thickness of the sound insulator would result in an increment of only 6 dB STL. Therefore, a thicker porous material is required for shielding of low frequency noise [11], but even a thick concrete wall fails to attenuate low levels of low frequency noise. To encounter these issues, other types of solutions have been proposed in the form of local dampers [13], perforated plates [4], meta- surfaces or meta-diffusers [8] and acoustic/elastic meta-materials [3] among others.

Lightweight single or double walls are very common solutions for partitioning in dwellings. They are typically composed of two leaves mounted on a wood or steel frame. The cavity between these leaves is filled by absorbing material to improve the acoustic performance in mid-to-high frequencies. The limited capability of the aforementioned conventional passive means at low frequencies, introduces a dilemma, provided that improvement of their performance is generally associated with an increase in weight and volume. Additionally, other solutions such as conventional passive vibration dampers have the ability to absorb at very specific frequencies, and they can also be very heavy.

Owing to the need of low frequency sound absorption, it is desired to develop a new generation of partitions capable of attenuating noise in low-to-high frequency range. To contribute to the solution the paper considers the application of the KDamping concept towards the design of highly dissipative low-frequency mounts. The KDamper is a novel passive vibration isolation and damping concept, based essentially on the optimal combination of appropriate stiffness elements, which include a negative stiffness element [1, 2]. In order to propose a solution to this problem, the STL of panels is initially examined, considering the sound propagation through a single, finite, rectangular thin plate. The acoustic behavior of a simply supported plate in this low frequency range, may be treated as a case of classic vibration control. The mounting of this plate is investigated and the way it participates in its STL performance. An equivalent analytical framework is presented describing the behaviour of a conventional panel supported by KDamper mounts incorporating the effect of the panel flexural stiffness. it is worth

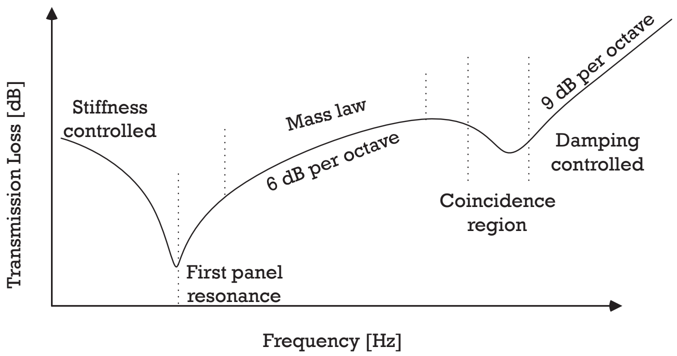


Figure 1: Typical STL profile of a panel.

noting that, the previous approaches considered the problem as in the case of classical vibration of a rigid mass ignoring any contribution of the plate's modes [17].

2 SOUND TRANSMISSION LOSS OF PANEL

In this section the theory of Sound Transmission Loss (STL) through a single panel is briefly presented. Various ways have been proposed regarding the modeling and simulation of the sound propagation through obstacles and the resulting STL [20, 15, 9]. In this work, the fundamental physics of the problem are investigated via an impedance based approach. According to this approach Transmission loss of sound occurs when there is an impedance mismatch between the propagation media of the traveling sound waves, thus sound is reflected and/or absorbed [9]. Direct sound transmission through thin panels depends on the mass, stiffness and damping of the system plate-support.

Figure 1 shows a typical form of the STL curve for sound propagation through a thin plate [7]. This curve presents five distinct regions in corresponding frequency ranges, depending on the properties of the system.

In a general case as in Fig. 2a, the STL is investigated via the assumption of a propagating plane wave between two different media. The impedance mismatch comes from the difference of impedance that the propagation medium of the incident waves Z_0, has from the propagation medium of the transmitted waves Z_1. In order to quantify the consequences of this mismatch, the transmission coefficient τ is introduced, defined as the ratio of the complex amplitude of the transmitted sound pressure to the complex amplitude of the incident sound pressure:

$$\frac{\tilde{p}_t}{\tilde{p}_i} = \tau = 2\frac{\tilde{Z}_1}{\tilde{Z}_1 + \tilde{Z}_0} \tag{1}$$

Figure 2 illustrates a wall, a rigid flat surface that consists only on mass, oscillating in an infinite rigid baffle with air on both sides, subjected to normal incident plane waves $\mathbf{p}_i = \tilde{p}e^{\jmath(\omega t - \kappa_0 z)}$. This particular case is refereed to as the ”limp wall” assumption and the resulting frequency response of the STL is known as the ”mass law”.

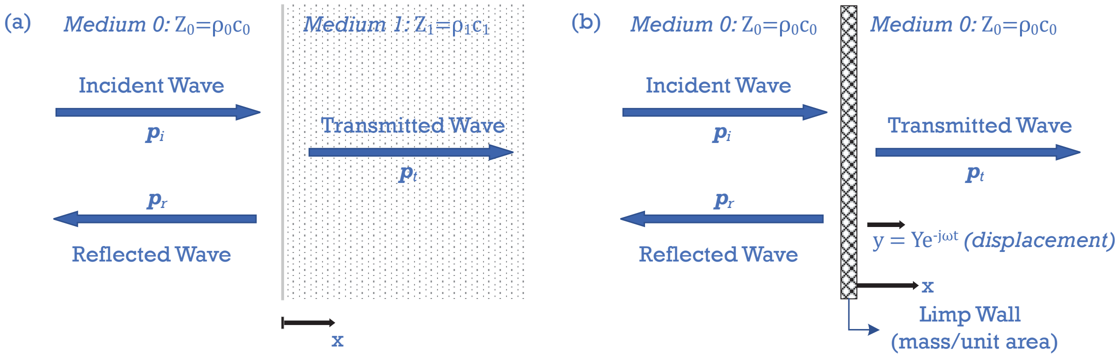


Figure 2: (a) Wave propagation in the presence of impedance mismatch (b) Limp wall assumption.

For frequencies higher than the natural frequency f_0, the STL is controlled by the mass per unit area of the panel. This corresponds to a 6 dB per octave increase of STL, while doubling the mass per unit area leads to a 6 dB per octave increase. Specifically, for a thin panel, neglecting stiffness and damping (limp wall):

$$STL = 10log\Big[\frac{\omega^2\overline{m}^2}{4(\rho_0 c_0)^2/cos^2\theta}\Big] \tag{2}$$

where θ is the angle of incidence of the propagated sound waves, c_0 is the speed of sound in air, ρ_0 the air density and $\overline{m}$ $[kg/m^2]$ is the mass per unit area of the panel. In the case of normal incident waves, $\theta = 0$.

Now, assuming that the same laws apply in case of a partition as illustrated in Fig. 3, by applying pressure and velocity continuity conditions at the interface of air and plate, the transmission coefficient comes as

$$\tau = \frac{2\tilde{Z}_0}{-\jmath(\omega m - k/\omega) + (2\tilde{Z}_0 + c)} \tag{3}$$

The imaginary part in the denominator of Eq. (3), indicates that the mass contribution ωm and the spring contribution $-k/\omega$ have a phase difference of 180° while the term $2\tilde{Z}_0$ indicates the radiation at both sides of the wall. The transmission coefficient can also be written as

$$\tau = \frac{\tilde{Z}_f}{\tilde{Z}_p + \tilde{Z}_f} \tag{4}$$

where $\tilde{Z}_p = -\jmath(\omega m + \jmath c - k/\omega)$ is the partition (mechanical) impedance and $\tilde{Z}_f = 2\tilde{Z}_0$ is the fluid loading impedance in both directions [9]. The

For the calculation of the Sound Transmission Loss (STL) in the case of a thin plate using the single DOF model as in Fig. 3, the expression of the mechanical impedance

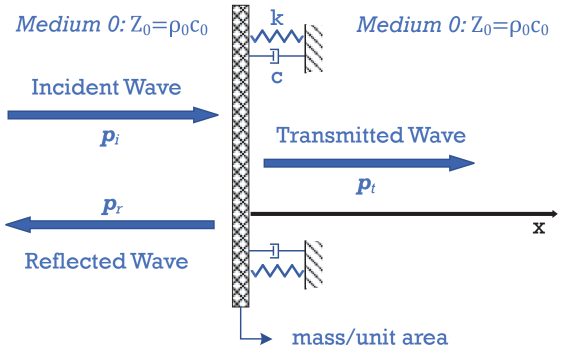


Figure 3: Wave propagation at a partition.

$\tilde{Z}_p$ can be rearranged to include the transfer function of the displacement as to the excitation, namely

$$\tilde{Z}_p = -\frac{1}{\jmath\omega}\frac{|\tilde{T}_{XF}|^{-1}}{A} \tag{5}$$

where A is the surface of the plate subjected to the excitation. Therefore, the STL is calculated as

$$STL = 10\log_{10}\frac{1}{|\tau|^2}\,[dB] \tag{6}$$

3 THE KDAMPER CONCEPT

The KDamper, which can be considered as an extension of the traditional Tuned Mass Damper (TMD), is a novel passive vibration isolation and damping concept, based essentially on the optimal combination of appropriate stiffness elements, including a negative stiffness element [1]. The KDamper supplements the inertial forces of the added TMD mass with the stiffness force of the negative stiffness element. This negative stiffness element can be realized by a number of ways such as pre-stressed disc (Belleville) springs, arranged in appropriate geometrical configurations [3]. Among others, this can provide significant comparative advantages, especially in the very low frequency range, such as better damping characteristics without the need of heavy additional masses.

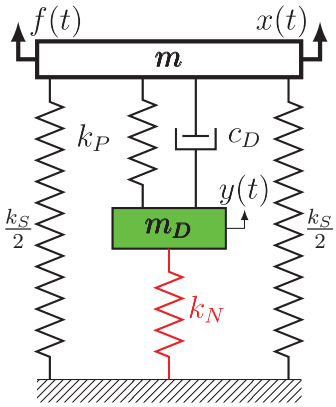


Figure 4: Schematic diagram of the KDamper oscillator.

The KDamper elements are calculated for a specific mass around a desired frequency f_0 or $\omega_0 = 2\pi f_0$ and the overall static stiffness comes as $k_0 = m\omega_0^2$, which also equals to

$$k_0 = \frac{k_P k_N}{k_P + k_N} + k_S \tag{7}$$

The overall static stiffness k_0 of the system is maintained.

Next, the basic parameters of the KDamper concept μ,κ and ρ, are defined as

$$\mu = m_D/m \tag{8}$$

$$\rho = \omega_D/\omega_0 \tag{9}$$

$$\kappa = -k_N/(k_P + k_N) \tag{10}$$

Finally, the non-dimensional parameters of the stiffness elements of the KDamper are defined as

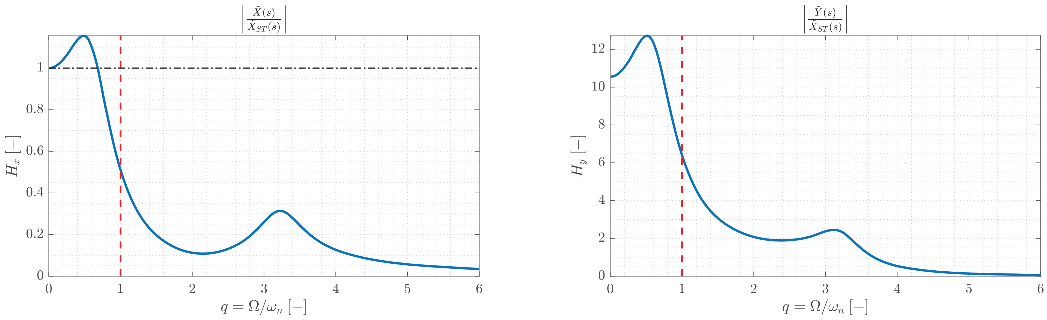
$$\kappa_S = \frac{k_S}{k_0} = 1 + \kappa(1+\kappa)\mu\rho^2 \tag{11}$$

$$\kappa_P = \frac{k_P}{k_0} = (1+\kappa)\mu\rho^2 \tag{12}$$

$$\kappa_N = \frac{k_N}{k_0} = -\kappa\mu\rho^2 \tag{13}$$

leading to the calculation of the stiffness elements (k_S, k_P, k_N) of the oscillator. Furthermore k_D is defined as $k_D = k_P + k_N$, thus, for a specific damping ratio ζ_D, the damping constant is calculated as:

$$c_D = 2\zeta_D\sqrt{k_D m_D} \tag{14}$$



(a) Frequency response of $|\tilde{T}_{XK}|$. (b) Frequency response of $|\tilde{T}_{YK}|$.

Figure 5: Frequency response of of the transfer functions of the KDamper.

Figures 5a and 5b, present the frequency response of the transfer functions of the dynamic magnification factor for both degrees of freedom (DOFs) x and y of the KDamper respectively, for an indicative set of parameters μ, κ and ρ. More information regarding the design of the KDamper can be found in [16].

4 SOUND TRANSMISSION LOSS OF DEFORMABLE THIN PLATE

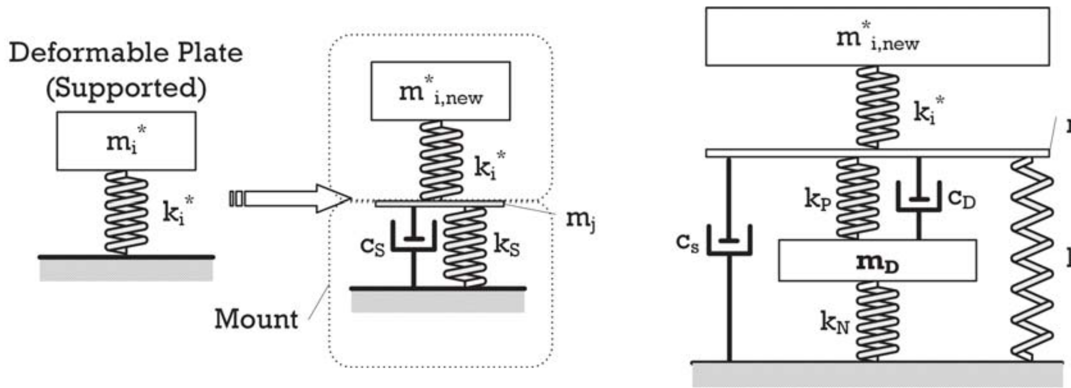
The investigation in this paper is an extension of previous work [17, 16]. However, a slightly different approach is taken here, where the thin plate is now considered as elastic/deformable instead of rigid as before. The goal is to examine how this property of the plate, participates in the dynamics of the system and consequently, the acoustic performance.

In order to do that, the assumption that the bending stiffness of the deformable plate is in a way in series with the stiffness of the mounting, is formulated. This assumption is also investigated through modal analysis and frequency response with FEM. This is achieved by utilizing the generalized values of the structure, namely generalized mass and the corresponding generalized stiffness for the appropriate modes, calculated from modal analysis and FEM simulations, in order to construct a corresponding single degree of freedom (SDOF) model representing the dynamic behavior of the deformable thin plate.

4.1 THEORETICAL FORMULATION

In this section the KDamper theoretical framework is extended to include the case where the rigid mass is replaced by a deformable plate. Figure 6a, shows the equivalent dynamic SDOF model of a deformable plate and the approximate modification when the plate is mounted on elastic mounts.

Figure 6b, shows the equivalent model when the simple elastic mount is replaced with an elastic mount based on the design of the KDamper.



(a) Deformable plate on elastic mounts with stiffness k_S.

(b) Implementation of the mounting based on the KDamper.

Figure 6: Modelling of a mounted deformable plate.

The equation of motion of the equivalent SDOF model is

$$m_i^* \ddot{u} + c_i^* \dot{u} + k_i^* u = f(t) \tag{15}$$

while in the case of the simple elastic mount becomes

$$m_{i,new}^* \ddot{u}_1 + k_i^* (u_1 - u_2) = f(t) \tag{16}$$

$$m_j \ddot{u}_2 + c_S \dot{u}_2 + k_S u_2 + k_i^* (u_2 - u_1) = 0 \tag{17}$$

where m_j is the connecting mass which is assumed as neglible in order to act just as a connector, namely an additional DOF and k_S is the stiffness of the elastic mount which needs to be calculated.

In the case of the KDamper elastic mount, the equation of motion in matrix formulation comes as

$$\mathbf{M}\, \vec{\ddot{u}}(t) + \mathbf{C}\, \vec{\dot{u}}(t) + \mathbf{K}\, \vec{u}(t) = \vec{F}\, f(t) \tag{18}$$

where the mass ($\mathbf{M}$), damping ($\mathbf{C}$) and stiffness ($\mathbf{K}$) matrices, come as

$$\mathbf{M} = \begin{bmatrix} m_i^* & 0 & 0 \\ 0 & m_j & 0 \\ 0 & 0 & m_D \end{bmatrix}, \; \mathbf{C} = \begin{bmatrix} 0 & 0 & 0 \\ 0 & c_S + c_D & -c_D \\ 0 & -c_D & c_D \end{bmatrix}$$

$$\mathbf{K} = \begin{bmatrix} k_i^* & -k_i^* & 0 \\ -k_i^* & k_i^* + k_P' + k_S' & -k_P' \\ 0 & -k_P' & k_P' + k_N \end{bmatrix}, \; \mathbf{F} = \begin{bmatrix} 1 \\ 0 \\ 0 \end{bmatrix} \tag{19}$$

At this point, it should be highlighted that the Dashpots ($\mathbf{c_s}$,$\mathbf{c_D}$) shown in Figure 6a denote schematically the systems' hysteretic damping and they should not be confused with the parallel dashpots that are usually employed when viscosity is considered. The loss factor η, can represent more accurately the dynamic response of nonlinear systems compared with the damping ratio which is defined on the grounds of the linear single degree of freedom (SDOF) viscous model [10, 6].

Therefore, the damping matrix is empty $\mathbf{C} = \begin{bmatrix} 0 \end{bmatrix}$ $(\mathbf{c_s} = \mathbf{c_D} = 0)$ and the hysteretic damping is introduced indirectly in the stiffness matrix $\mathbf{K} = \begin{bmatrix} \end{bmatrix}$,

where

$$k'_S = k_S\,(1 + \jmath n) \tag{20}$$

$$k'_P = k_P\,(1 + \jmath n) \tag{21}$$

The transfer matrix relative to the excitation, for each frequency ω, is calculated from the following expression:

$$\mathbf{TRF} = (-\omega^2\mathbf{M} + \mathbf{K})^{-1}\mathbf{F} \tag{22}$$

where

$$\mathbf{TRF} = \left\{ \tfrac{\tilde{u}_1}{\tilde{F}} \quad \tfrac{\tilde{u}_2}{\tilde{F}} \quad \tfrac{\tilde{u}_3}{\tilde{F}} \right\}^T \tag{23}$$

Finally, the transfer function of the deformable plate mounted on KDamper mounts, is taken as $T_{XF_{KD}} = \frac{\tilde{u}_1}{\tilde{F}}$.

4.2 OPTIMIZATION OF THE KDAMPER PARAMETERS

The aim is the selection of the optimal values for the KDamper elements, in order to obtain the maximum possible STL performance, especially in the resonance region of the system in comparison to the initial case where the plate is simply supported on its edges.

4.2.1 OPTIMIZATION ALGORITHM

For the optimization, the "$fmincon$" function of the MATLAB software for local minimization of non-linear functions and constraints, is utilized. The optimization algorithm is formulated as follows:

$$\min_{\mathbf{x}} f(\mathbf{x})\ s.t. \begin{cases} c(x) \leq 0 \\ c_{eq}(x) = 0 \\ \mathbf{A}\,\mathbf{x} \leq b \\ \mathbf{A_{eq}}\,\mathbf{x} = b_{eq} \\ lb \leq \mathbf{x} \leq ub \end{cases} \Bigg\| \tag{24}$$

where $\mathbf{x}$ is the vector containing the optimization variables and $f(\mathbf{x})$ is the objective function to be minimized. In this case the vector $\mathbf{x}$ is

$$\mathbf{x} = \begin{Bmatrix} k_N & k_P & k_S \end{Bmatrix}^T \tag{25}$$

Regarding the objective function, the algorithm maximizes the minimum value of the STL of the model in Fig. 6b, namely

$$f(\mathbf{x}) = -\min(STL_{KD}) \tag{26}$$

Initially, the only constraint to which the optimization is subjected has to do with the static stability margin of the KDamper, which is defined as

$$\epsilon = \frac{k_{N_{lim}} - k_N}{k_N} \tag{27}$$

where

$$k_{N_{lim}} = \frac{k_P k_S}{k_P + k_S} \tag{28}$$

and is set to have a minimum value of 10%.

Therefore, the optimization problem becomes

$$\min_{\mathbf{x}} f(\mathbf{x}) \; s.t. \begin{cases} 0.1 - \epsilon \leq 0 \\ lb \leq \mathbf{x} \leq ub \end{cases} \Bigg\| \tag{29}$$

with the optimization variables between the lower bounds

$$\mathbf{lb} = \begin{Bmatrix} -1 \times 10^6 & 0 & 0 \end{Bmatrix}^T \tag{30}$$

and the upper bounds

$$\mathbf{ub} = \begin{Bmatrix} 0 & 15 \times 10^5 & 1 \times 10^7 \end{Bmatrix}^T \tag{31}$$

The starting point $\mathbf{x}_0$ of the algorithm, is selected such that the stiffness element k_S is of the same order of magnitude as the stiffness element k_i^* of the deformable plate model.

4.2.2 CASE EXAMINATION: RECTANGULAR DRYWALL

A 0.625×1.6 m rectangular drywall is considered, according to real-life masonry applications. The width corresponds to the horizontal distance between the upright supporting beams providing the stable frame of the drywall. Typically, the height of such a drywall is around 3.2 m, meaning that it can be covered by two pieces with the

Table 1: Drywall properties.

$\overline{m}\ [kg/m^2]$	$W\ [m]$	$H\ [m]$	$t\ [m]$	$m\ [kg]$	$E\ [MPa]$	ν
8.35	0.625	1.6	12.5×10^{-3}	8.35	2900	0.31

dimensions consider here. The selected dimensions correspond also to a frontal surface of $1\ m^2$. The relevant properties of the drywall are summarized in Table 1.

Modal analysis via the finite element method (FEM), assuming simply supported boundary conditions as in Fig. 7a, gives the following generalized values for the fundamental mode of the plate, shown in Fig. 7b:

$$f_1^* = 31.39\ [Hz] \quad m_1^* = 2.82\ [kg] \tag{32}$$

In case of a simply supported rectangular thin plate this frequency may also be estimated accurately by the theoretical expression

$$\omega_{mn} = \sqrt{\frac{D}{\rho t}}\Big[\Big(\frac{m\pi}{l_x}\Big)^2 + \Big(\frac{n\pi}{l_y}\Big)^2\Big] \tag{33}$$

for $m = n = 1$ which corresponds to the 1^{st} mode and

$$D = \frac{Et^3}{12(1-\nu^2)} \tag{34}$$

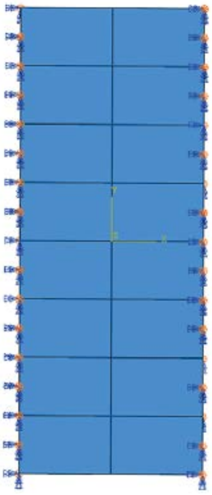
where D is the plate flexural rigidity, t is the plate thickness, E the Young's modulus and ν is Poisson's ratio. The corresponding generalized mass and stiffness [18] are defined as

$$m_{11}^* = m/4 \tag{35}$$

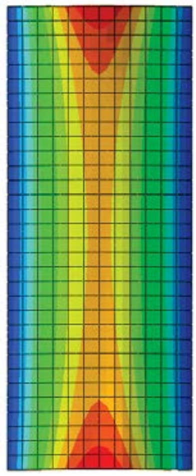
$$k_{11}^* = \frac{1}{4} l_x l_y D \pi^4 \Big(\frac{1}{l_x^2} + \frac{1}{l_y^2}\Big)^2 \tag{36}$$

Based on these values an equivalent generalized stiffness is calculated for the formulation of the corresponding SDOF model. Based on expression (36), it can be easily shown that the generalized stiffness is minimized when $l_x = l_y$, i.e for a square plate. However, as it will be shown hereinafter, acoustic performance is optimized when stiffness is maximized; in that terms the square plate constitutes the less attractive configuration .

Next, it is assumed that the plate will be supported on fifteen (15) elastic mounts in a configuration that essentially divides the plate into two equal parts of $0.3125\ m \times 1.6\ m$, as presented in Fig. 8a. Initially, this configuration as a simply supported plate



(a) Initial boundary conditions.



(b) Mode under consideration.

Figure 7: Simply supported drywall $0.625 \times 1.6\ m,\ A = 1\ m^2$.

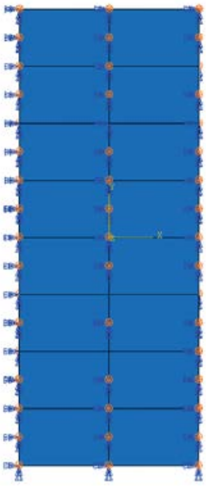
is considered, in order to obtain a more indicative set of generalized values for the following case of the mounted plate, which come as:

$$f_1^* = 196.92\ [Hz] \quad m_1^* = 1.91\ [kg] \tag{37}$$

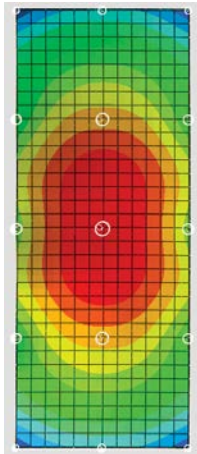
Using the objective function of Eq. (26), the algorithm searches for a solution in the frequency range $f_e = 1 : 1000\ Hz$. After the optimal set of parameters is calculated, the resulting stiffness k_S is divided into 15 springs and the FEM simulation is repeated in order to calculate the new generalized mass $m^*_{i,new}$. This procedure repeats a few times until there is a convergence of this value within a margin of error where any deviations become insignificant to the final results.

Then, for the generalized mass $m^*_{1,new}$ of the mounted plate, the corresponding element values of the KDamper come as in Table 2. The effect of the hysteretic behavior of KDamper is investigated parametrically for four (4) different loss factor η values and the results are illustrated in Fig. 9. The total loss factor η of a structure is the ratio of lost to reversible mechanical energy per radian cycle in a dynamic system. Nowadays, someone can easily find industrially produced rubber materials with specified loss factor values higher than $\eta = 0.4$ [12].

Figure 10, essentially compares the STL performance of the panel when utilizing elastic mounts - red-dashed line for simple elastic mounts and blue line for the optimized KDamper mounts - with the initial case as in Fig. 7, which is represented by the green line. Additionally, the mass law curve is represented for reference with the black-dashed line. When using elastic mounts the new generalized mass of the corresponding mode is higher, namely $m^*_{new} = 4.156\ kg$ compared to the initial $m^* = 2.819\ kg$. The same



(a) Mounting points for the elastic mounts.



(b) Mode under consideration.

Figure 8: Drywall supported by 15 mounts.

η	μ	$f_i^*\ [Hz]$	$m_{i,new}^*\ [kg]$	κ	$k_S\ [N/m]$	$k_P\ [N/m]$	$k_N\ [N/m]$
0.05	0.01	42.73	4.159	5.247	5.280×10^5	4.348×10^4	-3.652×10^4
0.1	0.01	43.72	4.159	5.526	5.291×10^5	3.894×10^4	-3.297×10^4
0.2	0.01	26.76	4.154	1.478	5.225×10^5	2.739×10^5	-1.634×10^5
0.4	0.01	20.10	4.156	0.376	5.301×10^5	1.234×10^6	-3.371×10^5

Table 2: Values of KDamper parameters for assuming loss factor values $\eta = 0.05 : 0.4$.

mass m_{new}^* is also used for the mass law. Therefore, this fact alone is expected and it shows to improve the STL performance of the initial panel.

5 DISCUSSION

An increase in the loss factor, improves the acoustic performance of the panel at the design frequency and the uniformity of STL at the resonance frequency range. However, the design frequency shifts to lower values and at the same time, higher damping also translates to smaller oscillation amplitude for the added mass. Of course, arrangements with extraordinary high loss factors are not easily realized practically and that is something that has to be considered. For example, in this contribution the loss factor is selected η=0.2.

The resulting STL curves for the four cases are compared in Fig. 10. The STL curve of the ”mass law” (limp wall approximation) is presented as a reference for the performance in these cases. The obvious advantage of the proposed KDamper configuration is

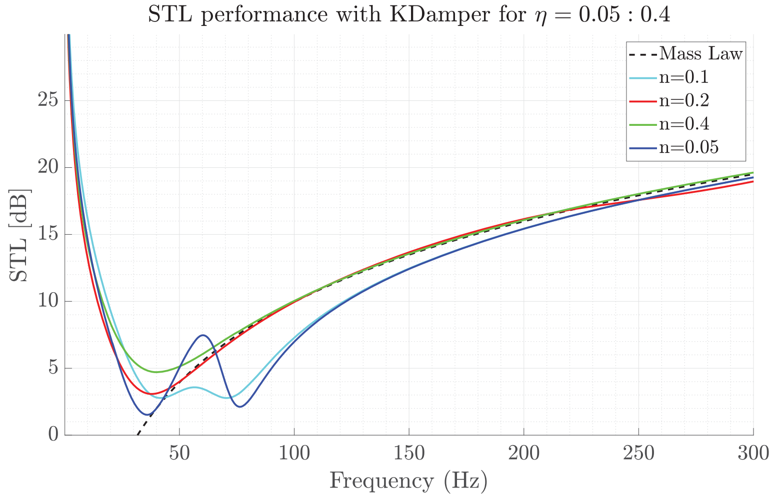


Figure 9: Resulting STL curves comparison with optimized $KDamper$ mount for different Loss factor η values.

the higher level of STL in the resonance region and the coincidence with the "mass law" curve for higher frequencies. On the other hand, the simple elastic mount configuration revealed little or no improvement in low frequencies compared with the simply supported panel, especially in the case of low damping but exhibited a similar behavior with the KDamper mounted panel for higher frequencies.

This impedance based approach for the calculation of the STL performance of a panel, demonstrates the viability and advantages of a KDamper based, mounted plate over conventional mounting mechanisms. The addition of the elastic mounts on a deformable plate, increases the generalized mass, therefore improves its acoustic performance. Compared to the simple elastic mounts, the KDamper provides an additional improvement in the resonance region. The main observation as shown in Fig. 6, is that the equivalent dynamic stiffness of the deformable plate and the stiffness of the mounting, operate approximately in series. Consequently, as it comes from the expression

$$k_{tot} = \frac{k_i^* k_S}{k_i^* + k_S} \tag{38}$$

the total stiffness of the model is restricted by the initial equivalent dynamic stiffness of the plate. An indicative graphical representation of Eq. (38), is shown in Fig. 11.

This implies that in order to utilize stiffer mounting -therefore increasing the eigenfrequency of the model where the total STL level is higher and the effect of the KDamper is more prevalent- requires a stiffer plate. To achieve that, either the mounting has to be more dense, namely dividing the plate into smaller "pieces" or the material of the

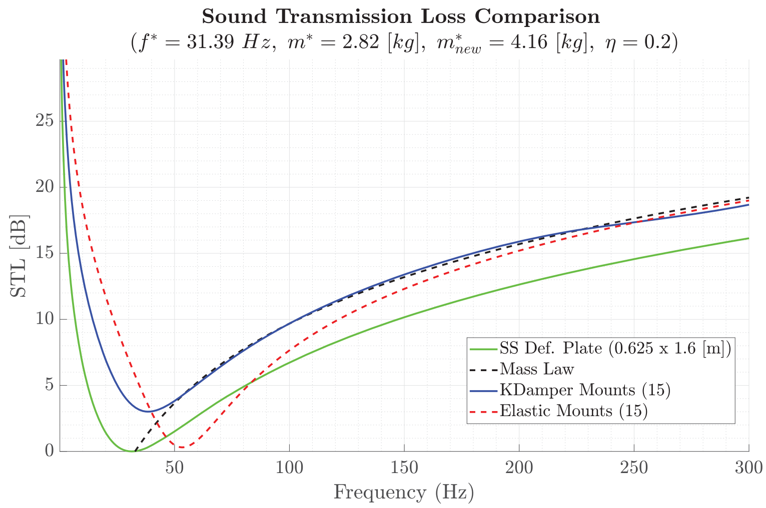


Figure 10: Comparison of the resulting STL curves with the optimized KDamper mount.

plate itself has to have higher Young's Modulus, thus higher bending stiffness D, or/and higher mass per unit surface.

6 CONCLUSIONS

A novel mount for acoustic panels was presented based on the KDamper concept, incorporating a negative stiffness element. The application of the mount in flexible panels was investigated based on analytical formulas and an equivalent "KDamper-Flexible panel model" was developed, to approximate more accurately the acoustic performance of the configuration compared to the "rigid mass" approach. The optimization algorithm of the system's parameters was described and an iterative procedure was followed to define the optimized set of parameters aiming to maximize the STL.

Then, a real-case scenario of a $0.625 \times 1.6\ m$ rectangular drywall was studied, comprising the reference arrangement for all the subsequent comparisons. The hysteretic behavior of the mount was examined for a range of loss factor (η) values, which control the design frequency, revealing the importance of a suitable rubber material to the system's performance. Finally, the superiority of the KDamper mounts was shown by comparison with the Simply-Supported panel and the simple elastic mounts. In case of the elastic supports an increase was observed at oscillating panel effective mass, resulting to an improved sound attenuation. The KDamper mounts proved advantageous compared with the simple elastic mounts improving at least four (4)dB the STL curve in the low frequency region. To the authors knowledge this improvement has not been observed in any other product being available on the market. Additionally, it was observed that the equivalent dynamic stiffness of the panel and the stiffness of the mounting, operate approximately in series. Hence, the total stiffness of the model is bounded by the

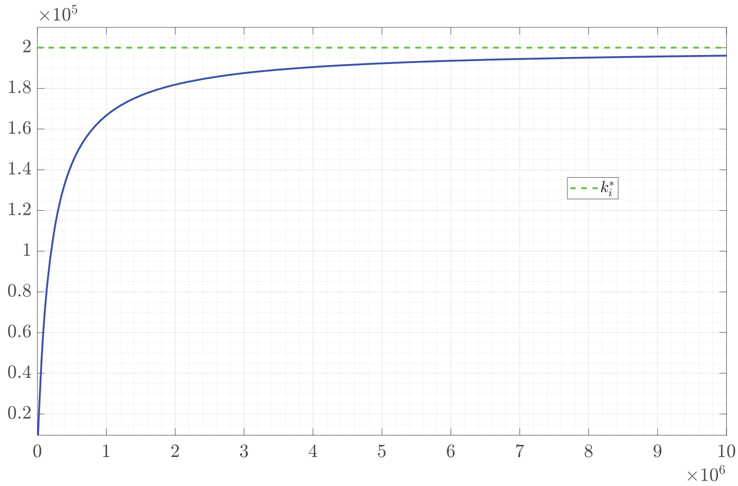


Figure 11: Springs in series configuration.

initial equivalent dynamic stiffness of the panel.

It should be noted that the approach followed in this paper, considers an equivalent mount-flexible panel model where the total system stiffness is calculated by applying a heuristic approach. Therefore, a topic for future work, is the calculation of the STL via a full numerical coupled structural-acoustic FE model and validate the performance experimentally.

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