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PROJECT OF A VIBRATION PASSIVE CONTROL SYSTEM FOR THE MAIN NOISE SOURCE OF A SMALL TRACTOR WITHOUT CAB

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Abstract. *The increasing concern about worker well-being with increased productivity has led to studies in the area of ergonomics and causes of occupational diseases. The present work studies one of the critical factors in the insalubrity of operation of agricultural tractors: the noise. A previous study of a small agricultural tractor performed to characterize the behavior of noise at the tractor operator's ears has shown that the roof causes a considerable increase in the sound pressure level. This has been proven through road test measurements and numerical simulations. Thus, this work was proposed to design a vibrations passive control system, through constrained layers with metallic and viscoelastic materials, in order to reduce the noise coming exclusively from this source. Simulations by finite element method and measurements in the field and laboratory, with and without the presence of these control devices were done. The proposed solution with constrained layers has shown to be an effective solution to control the vibration of a large plate with a high number of natural frequencies in the frequency range of interest (1000 Hz to 1200 Hz). This solution reduced the vibration significantly, up to 18 dB.*

Keywords: *passive control, vibrations, noise, tractor.*

1. INTRODUCTION

Agricultural mechanization has brought many growth opportunities for the field workers, reducing their body fatigue and increasing their productivity. With the passage of time, the concern with the worker has increased and studies were performed in order to improve their lives and to avoid work-related illnesses.

Unhealthy working environment is a critical factor that can lead to illnesses, reducing the quality of people's lives. According to Miyakita and Atsushi (1997), among the number of people in Japan with hearing loss greater than 40 dB in the 4 kHz range for various economy sectors, agriculture was the third sector with the largest number of people with hearing damage due to work, falling below the construction and manufacturing sectors, which remained the most critical ones. In spite of this, the percentage of people with hearing damage due to the work of agriculture was superior to all sectors.

In general, noise control can be done in three ways: in the source, in the trajectory and in the receiver. The design phase is the best opportunity to analyze and control the vibrations of a product. When a vibration problem is not predicted in the design, the system or structure can be modified in order to reduce the level of vibration and radiated noise. One way to accomplish this modification is by adding devices to the structure to reduce the level of vibration. Such devices control vibration at source and can be commonly found in the form of vibration isolators, viscoelastic links, dynamic neutralizers and constrained layers. When vibration control is not possible at source and transmission, the control at the receiver is used through personal protective equipment (PPE).

One type of passive vibration control is the constrained layer. This device is formed by layers of metallic material and viscoelastic material and aims to introduce damping to the primary system by shearing the viscoelastic material caused by the vibration restriction of the metallic layer. Ross, Ungar and Kerwin (1959) were some of the precursors in the study of this type of device, in their research, theoretically and experimentally, were analyzed several types of configurations of layers of metallic and viscoelastic materials.

A study made by Subramanian et al. (2003) proposed a methodology for optimization of constrained layer dampers (CLD) in order to reduce the noise coming from the structure. For this, they have used the finite element method to identify the points of the structure of greater energy of modal deformation and then determine the location and optimal sizes of the device. Another approach was used in the research conducted by Hujare and Sahasrabudhe (2014) in which the influence of the thickness of the viscoelastic material on the modal loss factor of a vibrating structure was studied. In that research, numerical and experimental analyzes were performed on a beam element to evaluate the effectiveness of the control device.

Madeira et. al (2017) have presented a multi-objective optimization approach to minimize weight and maximize modal damping in laminated composite panels with CLD treatments. The design variables were the number and position of the constrained layers patch treatments on the surface of the laminated plate.

Bakar et al (2017) have investigated the effectiveness of CLD in reducing disc brake squeal noise. In their research a maximum noise reduction of 11.3 dB(A) was achieved.

The purpose of this research is to identify the main noise source of a small tractor without cab, project some vibrations control devices, manufacture and test the proposals and verify their efficiency. It is expected an attenuation in vibration of at least 6 dB.

2. METHODOLOGY

The case studied in this work requires an easy to implement vibration control in the field. The first alternative evaluated was the introduction of cushions (a type of vibration insulation) at the entrance of the system, before the roof support, but the insertion of this type of device would reduce the stiffness of the whole structure of the roof, which could lead to loss of security against roll over. Thus, constrained layers were designed as they would not influence the safety of tractor operation and would be easy to implement. They were designed, simulated and experimentally validated to check their contribution in reducing the vibration of the roof.

The numerical modal analyses were done by the finite element method with the software package Hyperworks. The model was made with shell elements and the spot welds with beam elements, the base of the roof support was constrained in all degrees of freedom. After the numerical modal analyses were done, the experimental modal analyses were done in order to validate the numerical model. In these analyses were used the hardware Scadas of LMS, the software Impact Test of the same manufacturer, twenty uniaxial ICP accelerometers and one small modal ICP hammer.

Some initial dynamic tests were done in a flat asphalt track, in all gear combinations, varying the engine speed from its minimum to its maximum. In these tests were used the hardware Scadas of LMS, the software Test Lab of the same manufacturer, two ICP microphones positioned near the operator's ears, the same twenty accelerometers of the experimental modal analyses, but one of them positioned at the roof support input, and a speed sensor near engine crankshaft pulley. By means of these tests it was possible to perform operational modal analyses in order to verify the main natural frequencies of the roof and identify the main vibration source through the mapping of the engine and transmission orders.

After the identification of the main natural frequencies related to the noise at operator's ears, a project of passive vibration control was done with constrained layers with butyl rubber, used to introduce damping in the primary system.

A numerical analysis was performed at LAVIBS using Ansys software in order to determine the positions of the constrained layers. The proposal was validated experimentally at LAVIBS, with the roof in a free-free boundary condition, calculating the Inertance (point acceleration / excitation force) in some roof points without the constrained layers, with four layers, with six layers and then with nine layers and the results were evaluated and compared.

2.1 Viscoelastic Material mathematical model

The viscoelastic materials have mixed characteristics, presented by two classes of well-defined materials: liquid and crystalline solids, that is, materials that simultaneously have elastic and viscous deformations. The relation between tension and deformation of this type of material is not obtained directly through a constant of proportionality as in the highly elastic ones, but through a constitutive equation involving several terms of derivatives of different orders relating the tension and deformation.

Bagley and Torvik (1983) have proposed a way of describing the behavior of these materials, in the relation between tension and deformation, in a simpler way, using a fractional order model. A general form of the one-dimensional fractional derivative model for viscoelastic materials can be described as follows:

$$\sigma(t) + \sum_{m=1}^M b_m D^{\beta_m} \sigma(t) = E_0 \varepsilon(t) + \sum_{n=1}^N E_n D^{\alpha_n} \varepsilon(t). \quad (1)$$

Thus the complex modulus of elasticity, $E_c(\Omega)$, for the Zenner model can be determined by the four-parameter fractional derivative model:

$$E_c(\Omega) \approx \frac{E_0 + E_\infty (i\Omega b)^\alpha}{1 + (i\Omega b)^\alpha} = \frac{E_0 + E_\infty \phi (i\Omega_r)^\alpha}{1 + \phi (i\Omega_r)^\alpha}, \quad (2)$$

where E_0 is the asymptotic value of the elastic modulus for very low frequencies and E_∞ for very high frequencies, b the material relaxation time constant and α is the fractional order of the respective derivative to the constitutive differential equation of the viscoelastic material. The right-hand side of Eq. (2) shows the equation as a function of the reduced frequency Ω_r . The parameters of this equation are determined as follows:

$$b = b_0 s(T); \quad (3)$$

$$\Omega_r = \Omega s(T); \tag{4}$$

$$\phi = b_0^\alpha; \tag{5}$$

$$\log_{10} s(T) = \frac{-\theta_1(T-T_0)}{(\theta_2+T-T_0)}. \tag{6}$$

The Equation (3) shows that the constant b depends of b_0 , that is b computed at reference temperature T_0 , and of $s(T)$, that is a material parameter determined by Eq. (6). In this equation, on the other hand, θ_1 and θ_2 are material parameters determined experimentally and T is the operation temperature.

3. RESULTS

The Figure 1 shows the left hand side (LHS) noise distribution when the tractor is moving in a flat surface with the 4th gear and 2nd range coupled, that is the most critical gear combination in terms of overall noise pressure level. This noise color map is in function of the frequency (horizontal axis) and the engine speed (vertical axis). It is possible to observe the main noise order (27.88), the critical frequency and the engine speed in which it is excited the most. The order of 27.88 corresponds precisely to the order of the first gear pair of the range, when the 4th gear is coupled.

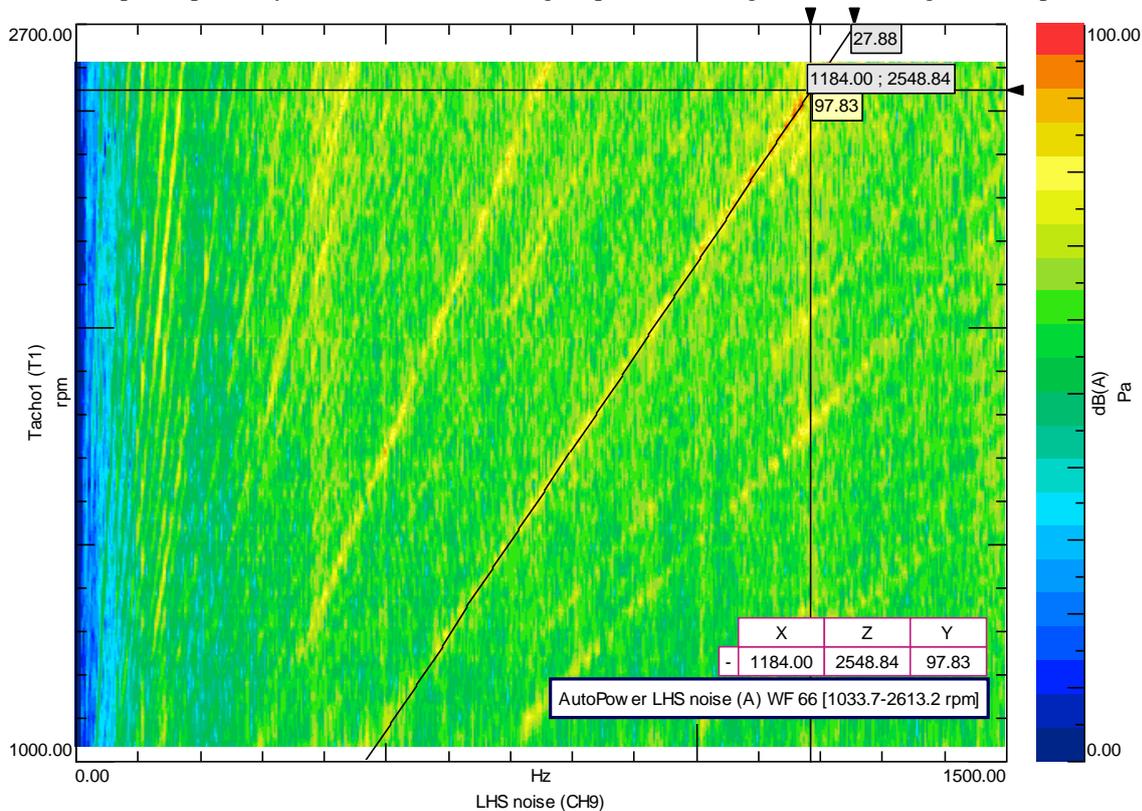


Figure 1 – LHS noise autopower with metallic roof.

With the vibration source identified, a dynamical test was done in the same condition of the previous test but without the roof and other structure points were instrumented in order to identify other structure borne noise sources beyond the roof. The noise color map is showed in the Fig. 2.

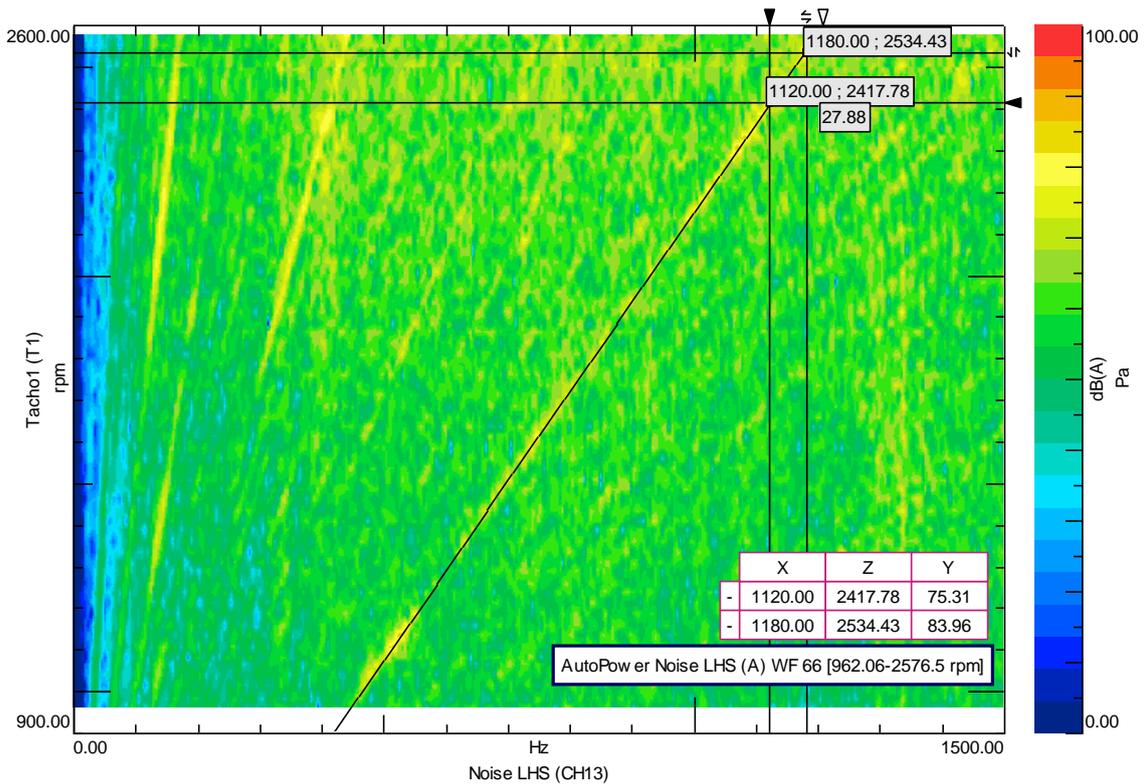


Figure 2 – LHS noise autopower without roof.

Through this graph it can be seen that there is still the noise coming from the transmission (order 27.88), but at a much lower level compared to the system with the roof (from 97.83 dB(A) to 83.96 dB(A)). The other points instrumented with accelerometers have showed very low levels of acceleration compared to the vibration signals of the roof.

Thus, it was done a numerical modal analysis of roof structure and several natural frequencies were found in the range from 1100 Hz to 1200 Hz. Two of the vibration modes are presented in Fig.3.

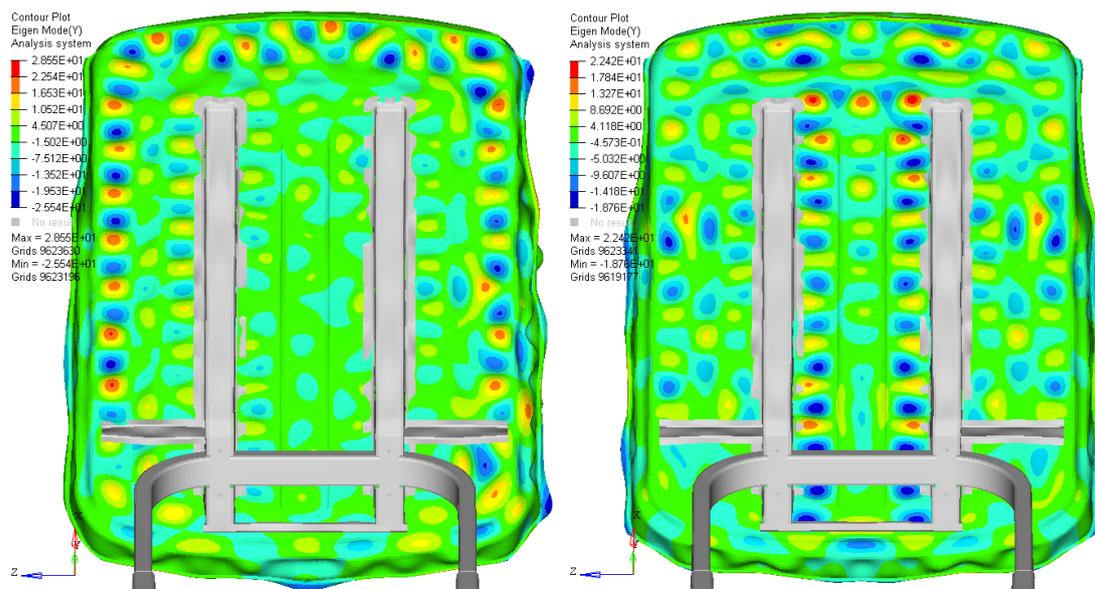


Figure 3 – Two of the several roof vibration modes next to 1180 Hz frequency.

This finite element model was validated by an operational modal analysis. Figure 4 shows the high modal density in the main frequency range of the noise at the operator's ears.

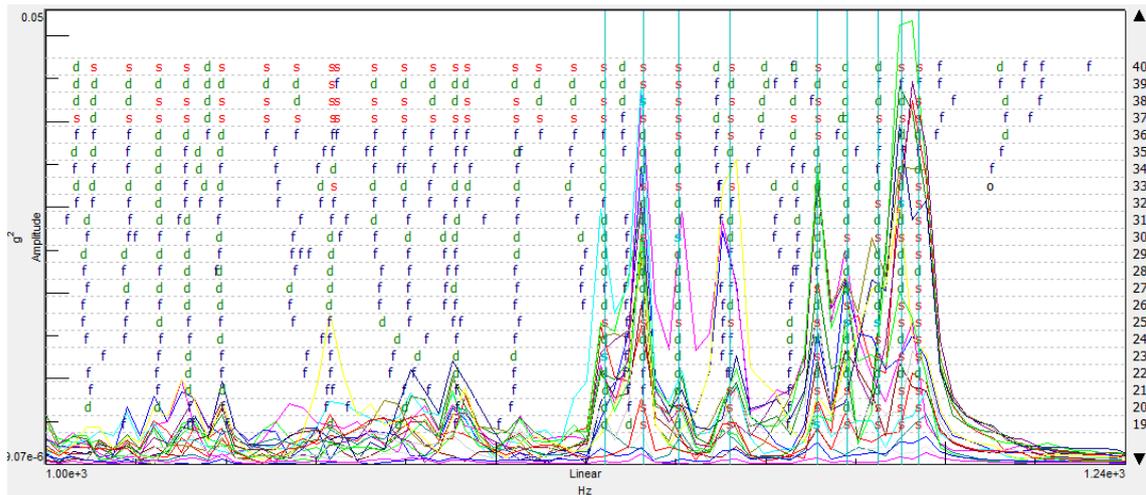


Figure 4 – Cross power of the acceleration points of the roof in relation to the structure input accelerometer.

With these results, it can be concluded that the main source of noise of the tractor is the roof that resonates when it is excited by the vibration coming from the transmission of the tractor.

A proposed solution to the problem of excessive vibration of the roof would be to introduce damping to the primary system through a set of sheet metal and viscoelastic layer called the constrained layer. A project of nine 210 x 100 x 6 mm sheets connected to a 6 mm butyl rubber layer of the BT-806/55 type was done. A numerical analysis was also done with the composite model to evaluate the position configuration of the nine constrained layers, shown in Fig. 5. This figure shows that the maximum deformations of the vibration modes were transferred to the constrained layers.

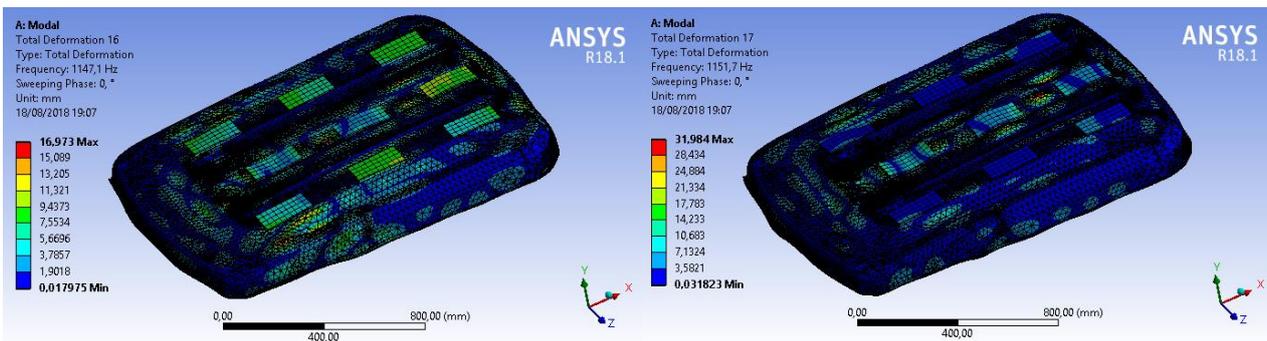


Figure 5 – Composite system vibration modes with constrained layers.

For the validation of this project, the constrained layers were manufactured and these were fixed to the roof for the physical test. The test was performed without any constrained layer, with four layers, with six layers and then with the nine layers, to evaluate the progressive reduction of vibration when introducing them to the system. The last two configurations of constrained layers are shown in Fig. 6.



Figure 6 – The left image of the test performed with six constrained layers and the right with nine constrained layers.

The Figure 7 shows a comparative chart of the Inertance at a point far from the positioning of the constrained layers, with no layer added and with the sets of constrained layers added to the roof.

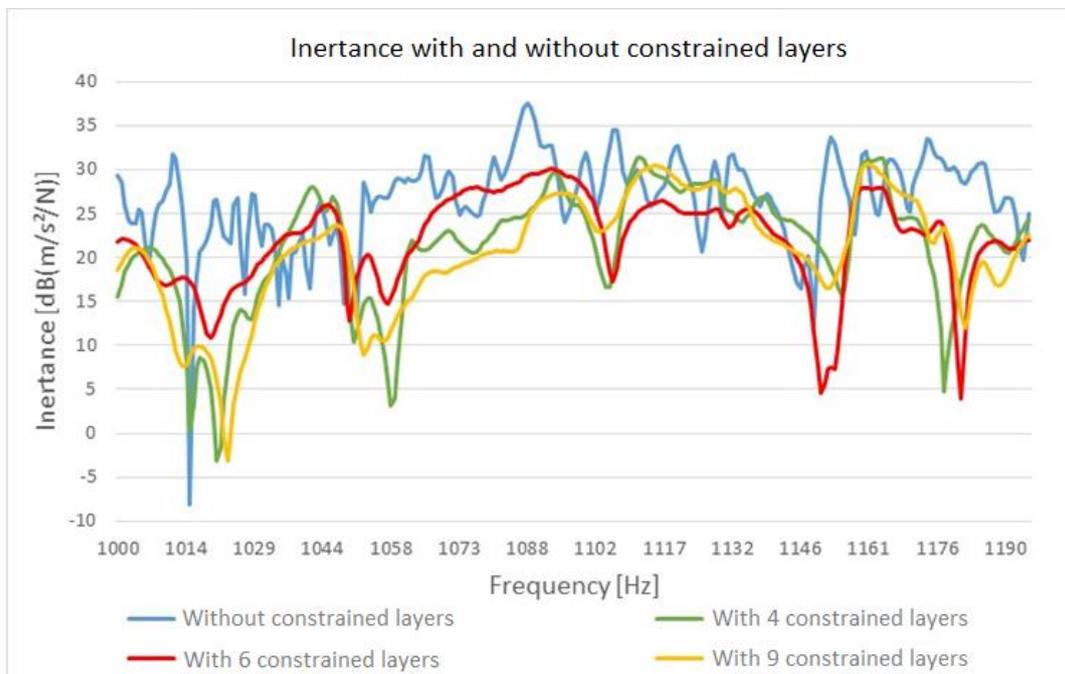


Figure 7 – Comparison of Inertance with and without constrained layers.

By comparing the roof Inertance values without the layers and with six layers, a reduction of at least three dB and a maximum of seventeen dB in the frequency range of interest can be observed. By comparing roof Inertance without the ensembles and the set of nine layers, it is possible to observe a reduction of at least three dB and a maximum of eighteen dB in the frequency range of interest. Comparing this result to the test result with six layers added, we conclude that there is not a large gain in terms of vibration reduction by adding more than six constrained layers to the primary system.

4. CONCLUSION

In this work it was possible to observe, according to the previously shown results, that the main source of the noise is the roof that resonates when it is excited by the vibration coming from the transmission of the tractor. The solution with constrained layers was effective, in this case of high modal density in the frequency range of interest. By adding these sets to the primary system it was possible to achieve up to 18 dB reduction in vibration in the roof.

A next step would be to test this solution with the moving tractor in the field and laboratory to verify its effectiveness in noise reduction because the attenuated noise would be only the noise transmitted by the roof structure and not the noise that is reflected in it from other sources.

For future researches some optimization could be implemented to optimize the position and size of the vibration control devices.

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6. RESPONSIBILITY NOTICE

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