Investigations of a Tuned Vibration Absorber with High Damping for Increasing Acoustic Panels Sound Transmission Loss in Low Frequency Range

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ABSTRACT

Many commercial partitions such as walls, windows and doors can provide reasonable sound insulation in medium and high frequency ranges. Those structures often perform in the mass control region of the acoustic spectrum and follow the "Mass Law" model. However, the Mass Law often does not provide adequate noise insulation at low frequencies. Hence other methods of increasing the sound insulation are required. One of those methods involves attaching to the vibrating structure a tuned mass or Tuned Vibration Absorber (TVA).While the TVA was invented over a century ago and was used successfully to reduce structural vibrations, applications to acoustics problems are relatively new. Most of researches in the last decade focused on TVA tuned to the excitation frequencies and the TVAs were not always optimised or they have been designed with little damping. This paper investigates the effect of attaching a highly damped TVA on sound transmission loss of panels. The numerical simulations show that significant reduction in vibration and sound transmission is achieved at the first panel resonance.

INTRODUCTION

Many researches show the detrimental effect of noise on people's health and well being even at levels well below potentially damaging for hearing (Birgitta Berglund, 1999). In addition recent researches show that the negative effects of low frequency noise on people may have been underestimated while the number of low frequency noise sources such as low flying airplanes and wind turbines has increased. Most common sound insulating partitions such as walls, windows, doors do not perform well in the low frequency ranges and most manufacturers do not supply sound attenuation information in low frequency region e.g. below 100 Hz.

The physical mechanism of the sound transmission through acoustic partition involves acoustic waves in the fluid (e.g. air) originating from one or more sound sources, impinging on a partition and forcing it to vibrate. The vibration of the partition forces the air on the other side of the partition to vibrate, which is then further radiated as sound into the quiet side. The level of the Sound Transmission through solid walls and enclosures is determined by the level of structural vibrations and the efficiency of the sound radiation from the vibrating panels(Wallace, 1972) .

Sound insulation performance of acoustic partitions is commonly quantified by the Sound Transmission Loss (STL or TL) defined as the ratio of the incident Sound Power to the Sound Power transmitted by the acoustic partition.

One of the methods used to increase the STL of a partition is to increase its mass.. Doubling the mass of the partition in most cases will increase the STL by 6 dBs, according to the "Mass Law". However the Mass Law also describes a 6 dB reduction in the STL when the frequency is halved, resulting in low STL levels at low frequencies for common partitions. There are other more efficient ways to increase the STL such as double partitions with a gap between the panels (double glazing for windows). However in order to achieve considerable improvements in the STL of double panels in the low frequencies range the size of the gap should be in the order of hundreds of mm. Also the structural coupling (rigid connections) between the panels should be avoided or minimised and acoustic absorptive materials should be placed in the gap (Bies and Hansen, 2003). Researches indicate that the performance of triple partitions in low frequency ranges is not better than double partitions and even worse if they have a symmetrical configuration (Vinokur, 1996)

The first mode of a simply supported panel mode is the most efficient sound radiator and the natural frequency of the first mode corresponds to the lowest dip in the STL curve(Bies and Hansen, 2003).

The Tuned Vibration Absorber (TVA) is a reactive device invented over 100 years ago to reduce vibrations of structures (Frahm, 1911). In its simplest form a TVA consist of a mass, stiffness and a damping element.

Den Hartog (Hartog, 1956) and others developed formulae for optimising the TVA to minimise displacement in the structures it is attached to.

TVAs have been successfully implemented in a variety of industries used for reducing the vibration response excitation of structures such as pipes and oil lines, overhead power lines, engines, pumps machinery and machinery enclosures, ships and airplane structures, household appliances, earth moving machinery, tall buildings oscillation caused by wind or earthquakes, etc.(J. Q. Sun 1995, Chen et al., 2009).

Another successful area for TVA application has been the sport industry where TVAs have been applied to tennis racquets, golf clubs(Meyer, 2002), skis and snow mobiles among others.

Previous attempts to use TVA for increasing the STL of panels had mixed success(Kuik et al., 2009). Some of those works dealt with Tuned Vibration Neutralisers TVNs which have very little damping and optimised for a very narrow frequency band around the excitation frequency. Other works deal with means and results of using variable TVA(Carneal et al., 2008). While many TVAs have been successfully used in the low frequency range for reducing structural oscillation and vibrations, there have not been many investigations in using TVAs for increasing the STL in low frequency range.

The purpose of this research is to investigate methods to optimise TVAs to commonly used acoustic insulating panels, in order to achieve maximum STL increase in the low frequency range.

GLASS PANEL CASE STUDY

For investigating the effects of adding a TVA to a common acoustically insulating panel (e.g. window), a 3mm thick (h), by 1m wide (L_x) and 1.5m height glass sheet, was selected. The material properties for Silica Glass (commonly used for windowpanes) are given below:

Table 1. Silica Glass Mechanical Properties (Matweb[1])

Parameter	Value
Density ρ	2180 Kg/m^3
Young's Modulus E	68.0 GPa
Poisson's ratio ν	0.17
Loss factor η	0.01

[1] from (Bauccio, 1994)

The mass of the panel is calculated as: 9.81kg.

A modal analysis of this panel with simply supported edges was performed in Ansys (Figure 1) and results were compared with the theoretical results obtain from known formulae (Bies and Hansen, 2003):

$$f_{i,n} = \frac{\pi}{2}\sqrt{\frac{B}{m}}\left(\frac{i^2}{L_x} + \frac{n^2}{L_y}\right) \quad (1)$$

$$B = \frac{Eh^3}{12(1-\nu^2)} \quad (2)$$

Table 2. First natural modes of the glass panel

i	*n*	*f [Hz]*
1	*1*	*11.2*
1	*2*	*21.5*
2	*1*	*34.5*
1	*3*	*38.8*
2	*2*	*44.8*
2	*3*	*62.0*

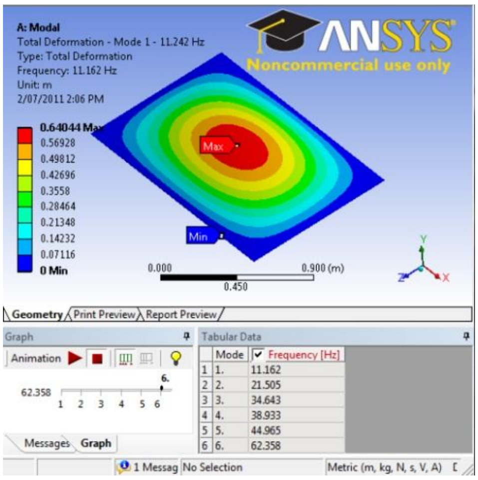

Figure 2- First natural modes and the mode shape of the first mode of a simply supported glass panel - Ansys simulation.

As it can be seen from above table and figure, a good match was obtained between the theoretical calculated and Ansys FEA simulation.

Ansys solution also includes the calculation of the "Effective Mass" of each mode and for the first mode this value is 6.45kg. This will be used in the following section to simulate the plate vibration behaviour at its first mode as a single degree of freedom system with an equivalent mass, stiffness and damping.

The equivalent stiffness of the panel can be then found from the simple formula:

$$k = (2\pi f_n)^2 m = 32\text{kN/m or } 32\text{N/mm} \quad (3)$$

A Harmonic analysis of the glass plate was also performed in Ansys. The glass panel was subjected to a 1Pa harmonic pressure, equally distributed over the face of the plate in a frequency range containing the first panel resonance. The maximum displacement – in the middle of the panel plotted against the excitation frequency is given in -Fig 3 below

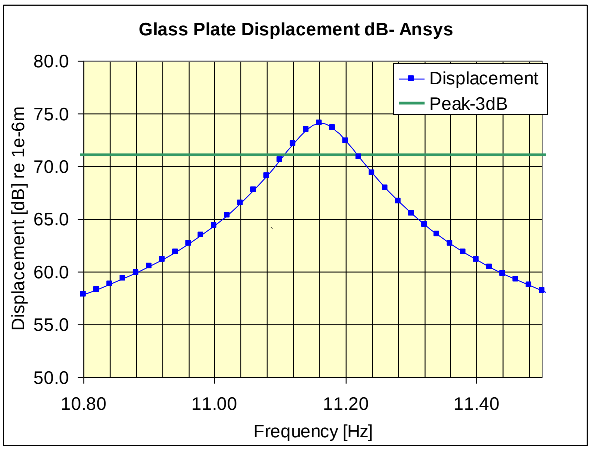

Figure 3- Glass plate maximum surface displacements as a function of frequency–Ansys FEA results

As expected there is a big peak in the displacement response at the natural frequency of the panel. From the results an estimation of the modal damping of the first mode was checked using the half power points method as approximately: ζ_1=0.005

This is expected because of the relation

$$\zeta = \eta/2 = 0.005 \quad (4)$$

From

$$\zeta = \frac{c}{c_{cr}} \quad (5) \quad \text{and} \quad c_{cr} = 2\sqrt{km} \quad (6)$$

mathematical relations we can calculate the damping:

$$c_1 = 4.54N{*}sec/m \quad (7)$$

This value was also used in the following sections.

Impedance of a single degree of freedom Structure

In classical applications of TVA for reducing the vibrations of structures, such as an out of balance machinery, the main structure is modelled as a rigid mass connected to the ground with a spring and damper:

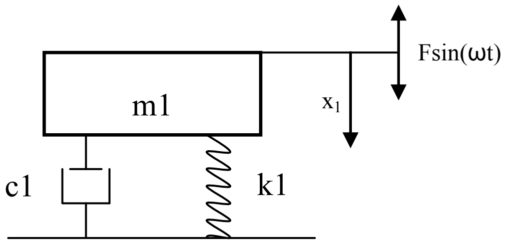

Figure 4 – Main spring and damper with a sinusoidal force excitation

The mass is excited by a sinusoidal force. Instead of following the normal equation of motion approach, an alternative method based on summation of individual impedances of elements, was followed. This approach is based on electrical analogies(Olson, 1958)

The Impedance of the panel can be first simply modelled as a consisting of a single mass, stiffness and damping elements:

$$Z_m = i\omega m \qquad Z_k = -ik/\omega \qquad Z_c = c \tag{8}$$

The panel total impedance is the sum of all three impedances:

$$Z_{panel} = Z_{m1} + Z_{k1} + Z_{c1} \tag{9}$$

For simulating our glass panel the previous calculate values: m1=6.45kg, k1= 32kN/m and c1=4.91N*s/m, are used. The impedance diagram of the components and main structure (simulating the the panel) is given below in Figure 5

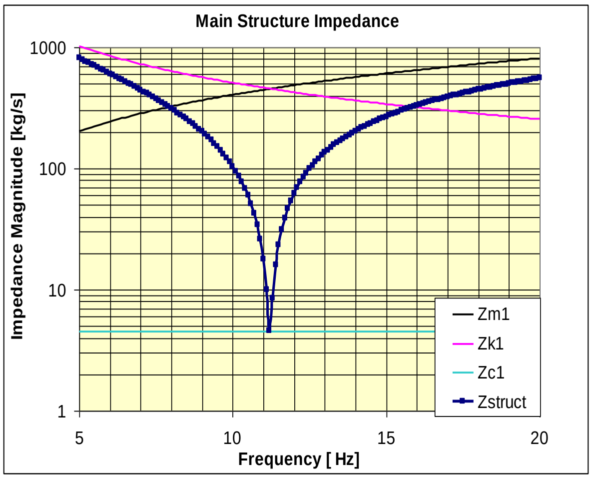

Figure 5- Impedance Magnitude of the glass panel modelled as a simple mass connected to the ground through a spring and damper

As expected the impedance has a minimum at the natural frequency of 11.2Hz and the value of the minimum impedance is equal to the damping value.

Impedance, Mobility and Compliance of a Structure with an optimised attached TVA

The TVA also consists of same three elements but the excitation force acts on the base of the TVA, not on its mass as in the main structure case.

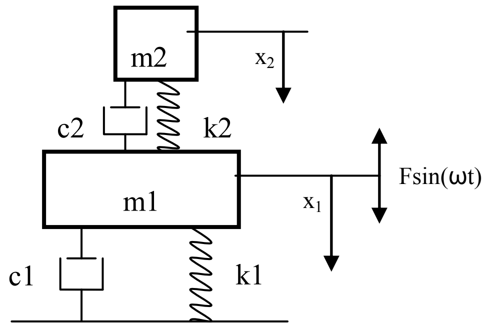

Figure 6- a TVA consisting of a mass (m2), spring (k2) and a damper (c2) attached to a main structure

Using the common electrical analogy (Force=Voltage, Displacement=Current), the sum of the spring and damping impedances is in series which each other and in parallel with the mass impedance. The reason for that is that c2 and k2 share the same displacements and together they are subjected to the same force with the mass (m2).

Therefore the total impedance of a TVA is given as:

$$Z_{TVA=} = \frac{Z_{m2}(Z_{k2}+Z_{c2})}{Z_{m2}+Z_{k2}+Z_{c2}} \tag{10}$$

The formula to optimise a TVA was developed by Den Hartog (Hartog, 1956) to a given structure, require the natural frequency of the TVA to be related to the natural frequency of the main structure f_{n1} and the mass ratio μ- the ratio of the TVA mass (selected to be 200gr) to the main structure mass as following:

$$\mu = m2/m1 = 0.031 \tag{11}$$

$$f_{n2} = f_{n1}\frac{1}{1+\mu} = 10.9Hz \tag{12}$$

and the optimal TVA damping:

$$\zeta = \frac{c}{c_{cr}} = \sqrt{\frac{3\mu}{8(1+\mu)^3}} = 0.103 \tag{13}$$

Note: This level of damping may not be achievable by common damping materials and more advance viscoelastic materials or a viscous damper may need to be used.

Therefore the optimal TVA parameters are:

m_2= 0.2kg, k_2= 932N/m and c_2=2.8 N*s/m

The impedance diagram of the components and the TVA is given below in Figure 7.

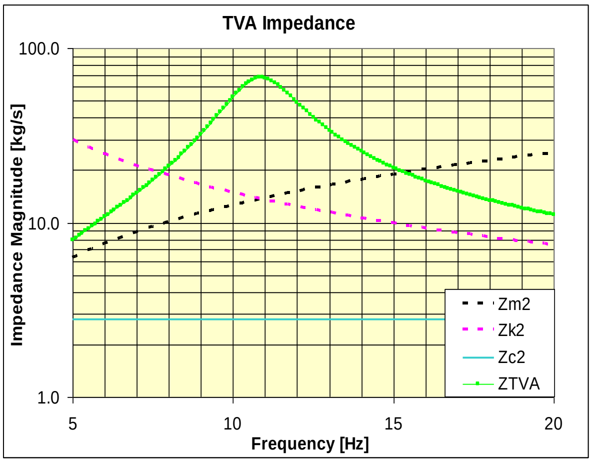

Figure 7-Impedance Magnitude of the TVA

The Impedances of the main structure and the TVA are added together, to give the impedance of the system at the point of the force excitation: Figure 8.

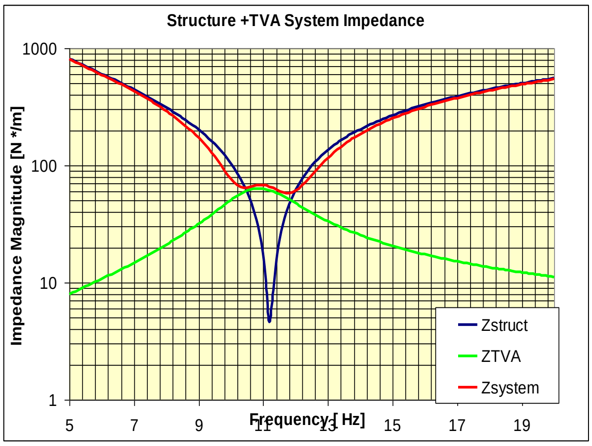

Figure 8-Impedance of the system simulating the glass plate with the TVA attached.

The effect of the TVA can be seen as "pushing up" the dip of the original structure impedance and therefore increasing the Impedance of the structure to which is attached to. Interesting to notice that the new two resulting minima of the system impedance, correspond to the two new natural frequencies of the main structure with TVA (a 2 degrees of freedom system) (Hartog, 1956).

The frequency response to unit force is the Mobility which is the inverse of the Impedance. It is displaying peak responses –peak velocities per unit excitation force, at the natural frequencies of the systems: Figure 9

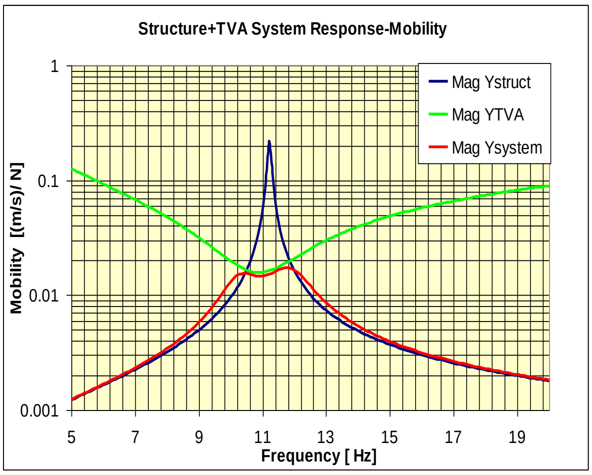

Figure 9-Mobility (FRF) of the simulated glass plate with a TVA attached

As it can be seen in Figure 9 the original maximum surface velocity was considerably reduced-by a factor of 6.34, which is a 16 dB reduction (using the squared velocity ratio).

At this point it is important to remember that the optimisation formulas used above for the TVA parameters are based on minimising the displacement not the velocity. The optimisation ensures that the two new displacement resonances peaks of the system will have equal magnitudes. Looking at the displacement system response (Compliance or Admittance) in Fig 10 below, we can see that indeed the magnitude of the Compliance peaks of the system, is the same.

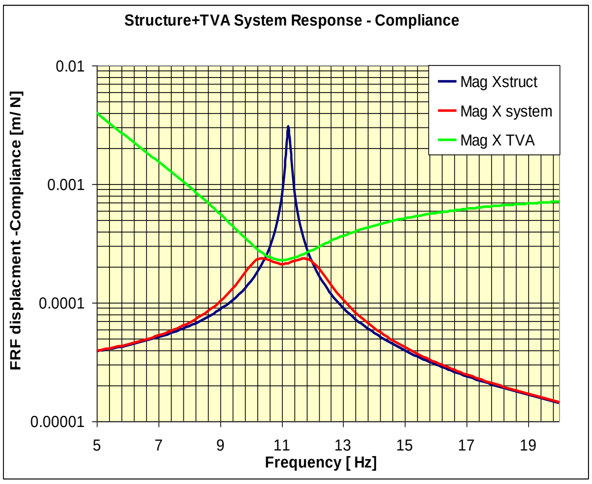

Figure 10- Displacement Response –Compliance of the simulated glass plate with a TVA attached

As the sound radiated by a vibrating panel is proportional to the velocity (squared) it will be useful to develop a formula for designing a TVA to achieve minimisation of the Mobility of the main structure. In this case study the differences in the natural frequencies of the two peaks are quite low and so the difference in the magnitude of the Mobility peaks, caused by multiplication by a different jω, is also not great. Therefore this TVA is close to optimal in respect to structural velocity minimisation.

STL of a Glass Panel and the Insertion Loss (IL) of the TVA - FEA simulations

Comsol FEA s/w was chosen to perform the STL analysis of the glass panel and the TVA because it has a ready to use Acoustic Module. Some of the capabilities of this s/w will be demonstrated below.

The same 3mm thick and 1m wide by 1.5m length, simply supported glass panel was analysed in Comsol. The glass panel was modelled with triangular shell elements and the air domain adjacent to the panels on both sides was modelled with tetrahedral acoustic elements.

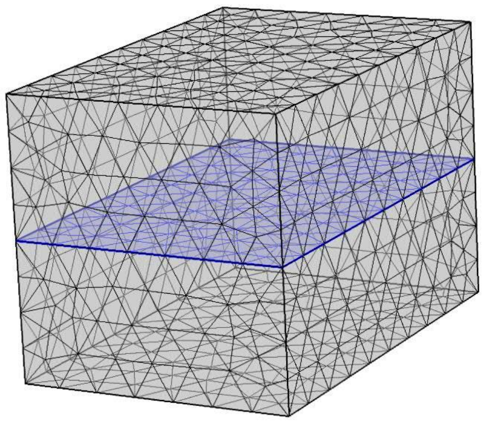

Figure 11-Comsol FEA model of the glass panel and the surrounding air.

The glass panel was excited by a normal incidence plane wave of 1Pa amplitude originating from the bottom plane of the bottom half of the air domain. The vertical walls of the air domains are completely reflective so no acoustic energy is lost while the top and bottom surfaces are anechoic terminated. The panel is coupled to the air domain through Fluid Structure Interaction (FSI) equations and the program calculates the panel displacement and velocities as well as the sound pressure and sound intensity through-out the air domains.

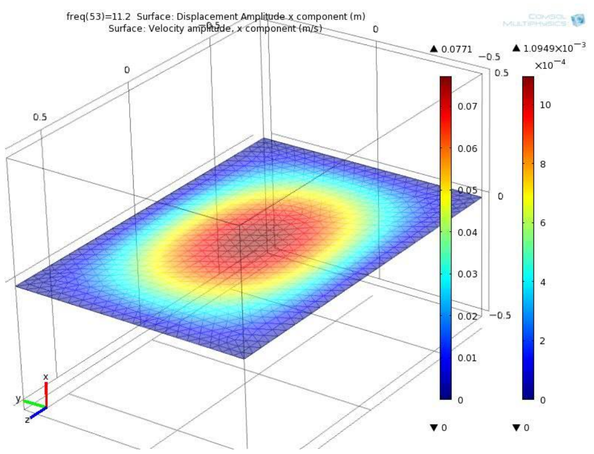

Figure 12 Glass panel normal displacements and velocities at first resonance (11.2 Hz)

To compare the FEA glass panel surface velocities to the structural velocity of the rigid mass the Mobility was multiplied by the same excitation force that the FEA panel was subjected to, namely- 1.5N (1Pa x 1.5m^2.) The two models give similar results –Figure 13 below.

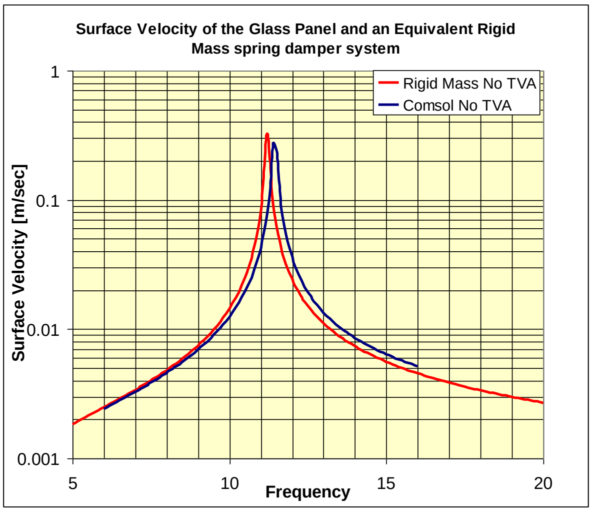

Figure 13- Maximum Surface velocity of the panel and the rigid structure velocity - frequency responses

To calculate the panel STL the intensity over the bottom and top surfaces is integrated to give the input sound power (W_in) and the output sound power (W_out) .

The STL is then calculated by:

$$STL = 10\log\left(\frac{W\_in}{W\_out}\right)\text{dB} \tag{14}$$

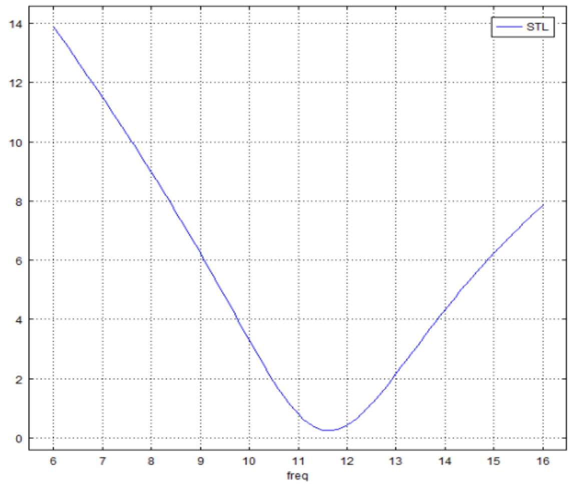

Figure 14- Glass Panel STL - Comsol simulation

There seems to be very little sound attenuation at the glass panel resonance- less than 0.5 dB STL.

Glass Panel with TVA

The best location for attaching a TVA is obviously the panel middle point where displacements and accelerations are at maximum.

To simulate a TVA attached to the panel the Impedance of TVA as was previously developed is inserted in form of force equation at a "load point" in the middle of the panel:

$$F_p = -Ztva*u*j*omega \qquad (15)$$

Where u is the calculated panel velocity and *u*j*omega* is the velocity at the TVA attachment point to the panel.

Comparing the main structure velocity reduction achieved by the TVA Figure 16 below it seems that the TVA Comsol model is giving a higher velocity reduction at the point of attachment than the one predicted by the simple two degrees a freedom model.

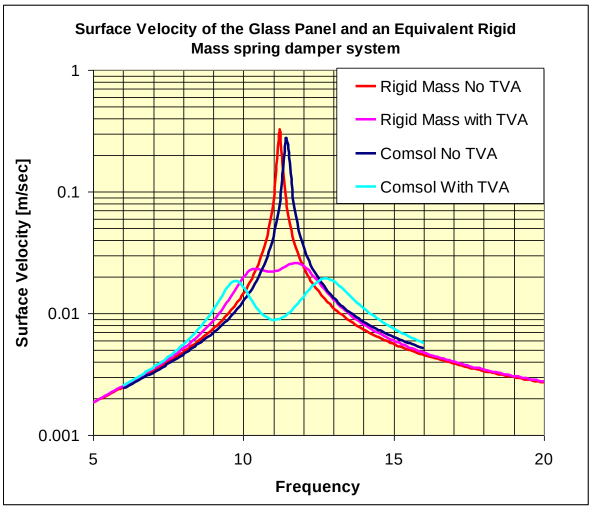

Figure 15- Maximum Surface velocity of the panel and the Rigid structure velocity with and w/o TVA-frequency responses

Examining the STL of the glass panel with the TVA - Figure 16 below, the lowest STL with the TVA is almost 8 dB which is significantly higher than without the TVA but still lower than predicted by the velocity reduction.

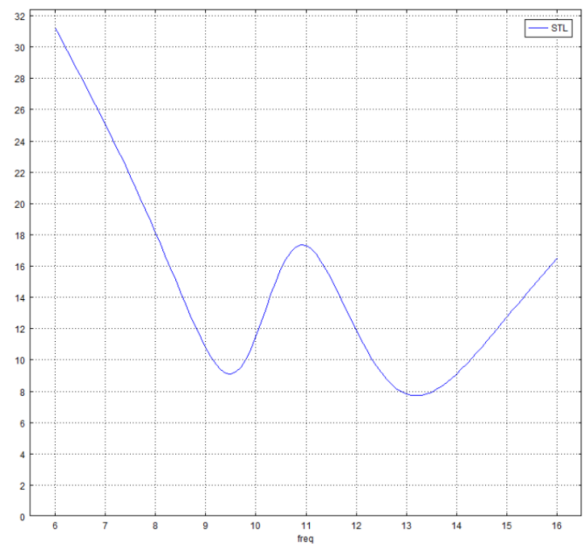

Figure 16- STL of glass panel with the TVA

A TVA with a higher level of damping (c2=6N*s/m) was also analysed and the STL results show a greater level of noise reduction (Figure 17).

To get a better understanding of the noise reduction effect of the TVA, the transmitted Sound Pressure Level (SPL) was calculated by subtracting the calculated STLs data from discrete tones or equal amplitude swept sine sound source input on the other side of the glass panel.

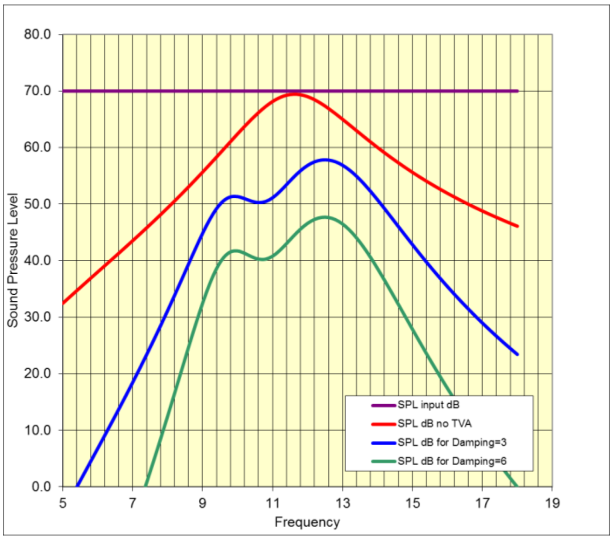

Figure 17-Simulated incident and transmitted noise

The sound source had a constant value of 70 dBs ("SPL input dB" in the graph) at each of the frequency points of the STL data. By subtracting the STL of each of the configurations predicted output SPL were obtained.

The STL of a higher level of TVA damping was also performed and the results show a greater level of noise reduction.

The overall SPL levels over the frequency range of 5Hz to 18 Hz are given in table 3 below

SPL input dB Total**:** 88.8dB

Table 3. TVA effect on the glass panel SPL transmission in the frequency range containing the first natural frequency of the panel.

Parameter	Value	TVA-IL
SPL transmitted w/o TVA	80.3dB	-
SPL transmitted with TVA c2=3 N*s/m	68.5dB	**11.8dB**
SPL transmitted with TVA c2=6 N*s/m	58.0dB	**22.3dB**

The Insertion Loss of the TVA was calculated by subtracting the transmitted SPL with the TVA and without.

DISCUSSION AND FURTHER WORK

The selected panel size and thickness had a natural frequency of 11.2 Hz. Thicker and/or smaller size glass panels will have higher natural frequencies well into the audible frequency range. The effect of attaching one or more optimised TVAs on other panel of various sizes and materials will be investigated in the future.

The required TVA damping levels to achieve the predicted high level of STL may require some viscous damping mechanism. A prototype of a TVA with similar characteristic as given in this paper was built and test results are expected soon.

CONCLUSIONS

The results from two different simulation methods have predicted that adding even a light mass TVA with high damping tuned to the first structural resonance of a panel a large velocity reduction can be achieved. The increase in the noise insulation capabilities of the panel, while not as large as the surface velocity reduction, it is still higher than achieved by doubling the panel mass.

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