DEVLOPMENT OF TUNED VIBRATION ABSORBERS FOR AN AIRCRAFT FUSELAGE

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1 ABSTRACT:

Recent studies on the C-130J-30 Hercules have revealed the need for vibration and sound reduction within its fuselage. The propellers of the C-130J have a blade pass frequency (BPF) of 102 Hz at cruising speed that creates a noise problem in the fuselage from propeller wash. Tuned Vibration Absorbers (TVAs) were designed to reduce vibration, especially in its lower modal frequencies corresponding to the BPF. After fitting these absorbers to a scale model of the fuselage, testing revealed by Vipac Engineers and Scientists revealed that vibration was suppressed at frequencies lower than the designed tuning frequency. The aim of this project was to analyse this frequency-shifting phenomenon and find an accurate method of tuning the TVAs.

As part of the process to identify this problem several experiments were conducted. This paper begins by introducing the basics of TVA operation including some theoretical models that were constructed. This is followed by experimental work carried out on a fully clamped flat plate on which TVAs were attached to analyse their effect on vibration attenuation and frequency shifting.

2 KEYWORDS:

Tuning Frequency, Vibration Attenuation, Frequency-Shifting

3 BACKGROUND:

3.1 Tuned Vibration Absorbers

A passive Tuned Vibration Absorber (TVA) is a mass coupled with a spring-damping system. When applied to a structure in an anti-node position the TVA reduces the amplitude of vibration of the base structure at the designed frequency. This type of vibration absorber is particularly useful when the noise being generated is tonal and therefore the absorber can be designed to target the frequency of interest. At the tuning frequency of the absorber, the absorber will vibrate intensely, thereby dissipating vibration energy from the base structure. Passive vibration absorbers do not require external power supplies therefore they are normally chosen for simple applications, especially concerning narrow bandwidth vibration. A schematic diagram of a tuned

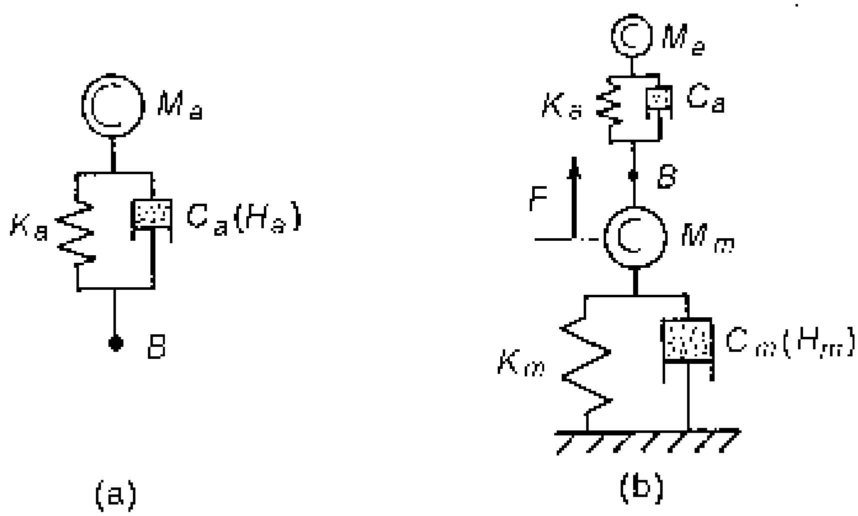


Figure 1 **[1]**

a) A schematic diagram of a simple passive Vibration Absorber
b) The absorber attached to a simple base structure

vibration absorber attached to a simple base structure is depicted in Figure 1. The absorbers may be tuned to the desired frequency by altering their stiffness or mass. The fundamental natural frequency of any single degree of freedom object is defined as:

$$\omega = \sqrt{\frac{k}{m}} \qquad [2]$$

3.2 Theoretical Work

In order to examine the reasons behind the frequency-shifting phenomenon, theoretical studies were conducted on the TVAs. The first task was to model the cylindrical C-130J fuselage as a ring-stiffened cylinder. Using equations from Leissa [3] the cylinder section was analysed to find the resonant modes. This was then used to design a flat plate that has the same modal density and wavelengths at the frequency of interest as the cylinder model (in the 125Hz one-third octave band). The major assumption in using this analogy is that the effect of stiffening is far greater than the effect of the curvature.

Free vibration of plates is a well-explored field. Leissa [4] provides tabulated results for the natural frequency of plates of various aspect ratios. The plate used in this experiment was close to an a/b ratio of 2/3. When designing the plate, the impedance of the absorber had to be much greater than that of the plate, i.e.

$$Z_{absorber} >> Z_{plate}.$$

This is to ensure that the vibration response of the absorber is much greater than the vibration response of the plate.

$$Z_{absorver} = Q_d \; m_a \; w_a \approx (17)(0.4)(2\pi)(132) \qquad [5]$$

$$= 5639.8 \text{ where } Q_d \text{ is found to be} \approx 17$$

$$Z_{plate} \approx 2.3\rho c_s t^2 = 2.3(2700)(3145)t^2 \qquad [5]$$

Where ρ is density of aluminium and c_s is the transverse shear wave speed of aluminium.

$$2.3(2700)(3145)t^2 << 5639.8$$

$$t << 0.017\text{m}$$

This is the criteria for designing the plate, in practice the thickness of the plate is much smaller than the value calculated.

The size of plate is restricted to the size of the aperture between the reverberation chambers that is 1600 x 1100 mm^2**.** A plate size size of a = 1000mm and b = 1500mm was therefore the amount of free plate that could vibrate (allowing for the clamping space allocation)**.** Using equations from Leissa and solving it with Matlab, the following tabulated results were found:

m	n	wa^2(rho/D)^0.5	E(Pa)	rho	v	h	a(m)	w(rad/s)	w(Hz)
1	6	195	7E+10	2700	0.33	0.0025	1	759.080637	120.8114
2	6	230.04	7E+10	2700	0.33	0.0025	1	895.481588	142.5203
3	4	193.24	7E+10	2700	0.33	0.0025	1	752.229447	119.721
4	1	206	7E+10	2700	0.33	0.0025	1	801.90057	127.6264
4	2	218	7E+10	2700	0.33	0.0025	1	848.613224	135.061

Table 1 Natural Frequency calculations for designed plate

The designed plate was designed to have five modes in the 125 Hz one-third-octave band so that the effect of the TVAs attenuating broadband vibration could be observed. Table 1 shows that this was achieved. Checking the designed plate thickness,

$$h = 0.0025\text{m} \ll 0.017\text{m (by a factor of 7)}$$

The final design of the plate had dimensions, **h = 0.0025m**, **w x l = 1000 x 1500 mm^2** and **aluminium** chosen as its material.

4 EXPERIMENTAL WORK

A simple experiment was designed initially to analyse the tuning frequencies of the absorbers provided by Vipac. This experiment involved exciting the TVAs directly with a shaker and measuring the vibration response with one accelerometer mounted on the bracket and another on the TVA. After the two accelerations were measured, a transfer function was derived of input: output and the effect of the absorbers was analysed (measure response ratio X/Y). The experimental arrangement is shown in Figure 2 and an example result for TVA 1 is shown in Figure 3. The results for TVA2 and TVA 3 were similar to TVA1.

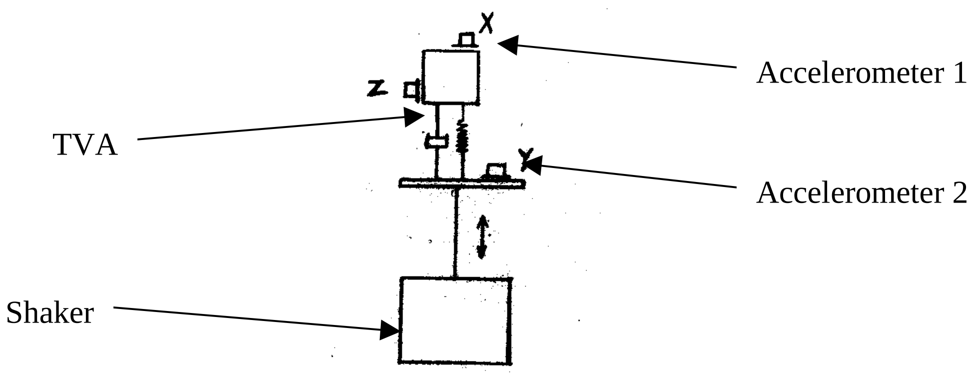


Figure 2 Shaker Test Schematic Diagram

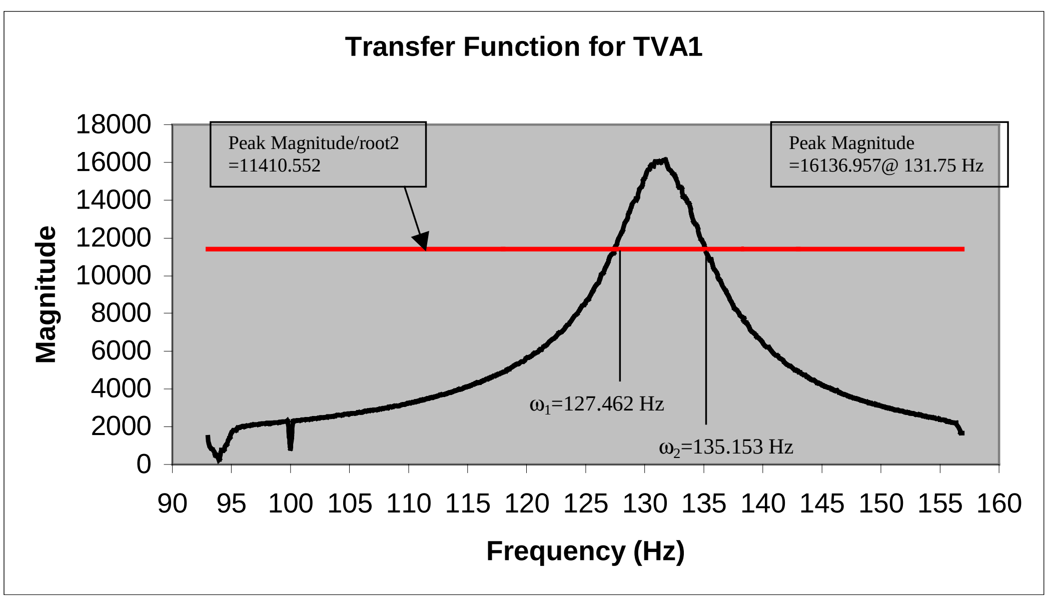


Figure 3 Transfer Function X/Y for TVA1

From the measurements and plots of the transfer functions the following quantities were calculated:

$$\text{Quality Factor, } Q=\omega/\Delta\omega=17.131 \Rightarrow \text{Loss Factor, } \eta=1/Q=0.0584$$

It was found that the supplied elastomeric absorbers were tuned to a frequency of approximately 132Hz. The quality factors of the absorbers were found to be approximately 17. A more detailed analysis on the TVAs involved analysing their effect on a vibrating base structure.

For the experiment a 1600mm by 1100mm (50mm around each edge for clamping) by 2.5mm thick Aluminium sheet was used to produce the desired resonant modes close to 136 Hz. A laser scanning vibrometer was used to analyse the velocity distribution across the flat panel in a reverberation chamber separating wall. A shaker was located in the source room, which excited the panel with white noise. By attaching a force transducer directly to the stinger of the shaker we were able to calculate the direct transfer function of Force input/ Velocity output. Four different cases were studied on the plate including the response of the free plate, one TVA attached, two attached and finally all three attached. The force input location and location of TVAs were placed at arbitrary positions on the plate but well spaced from each other. The TVAs were attached to the plate using brackets that were made extremely stiff so that they minimised any lateral vibration or 'rocking' of the TVA. A flow diagram of the experimental setup is shown in Figure 4.

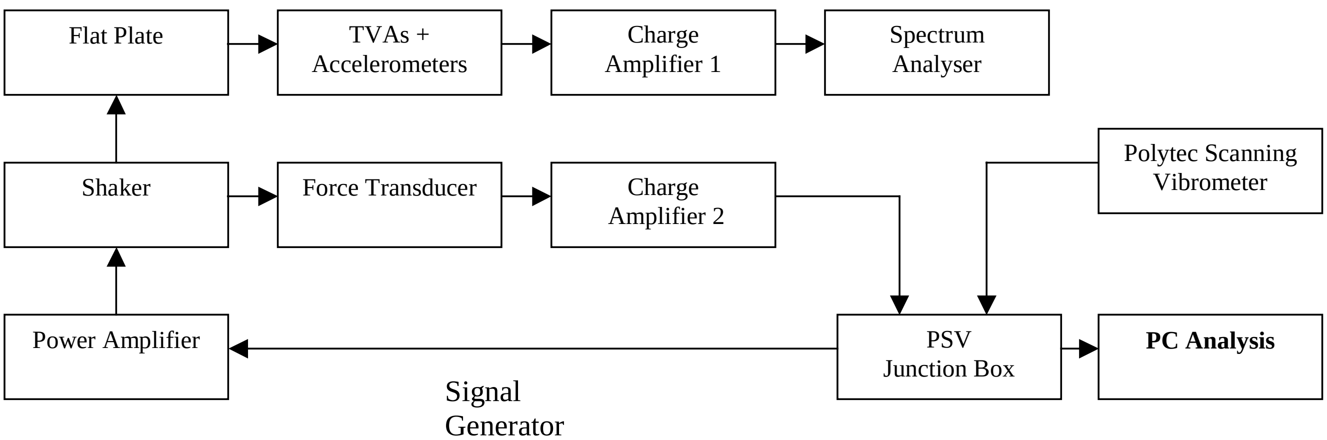


Figure 4 Flat Plate Experiment Flow Diagram

The results of the experiment are shown in Figure 5, which clearly shows that the TVAs reduced vibration in the plate significantly especially at the resonant frequency of 136Hz. The graph also shows however, that as more TVAs are added to the panel (and therefore more mass) the frequency shifting of the resonant modes can be seen. For instance, the mode at 136Hz with no TVAs attached has shifted to around 131Hz with all three TVAs attached.

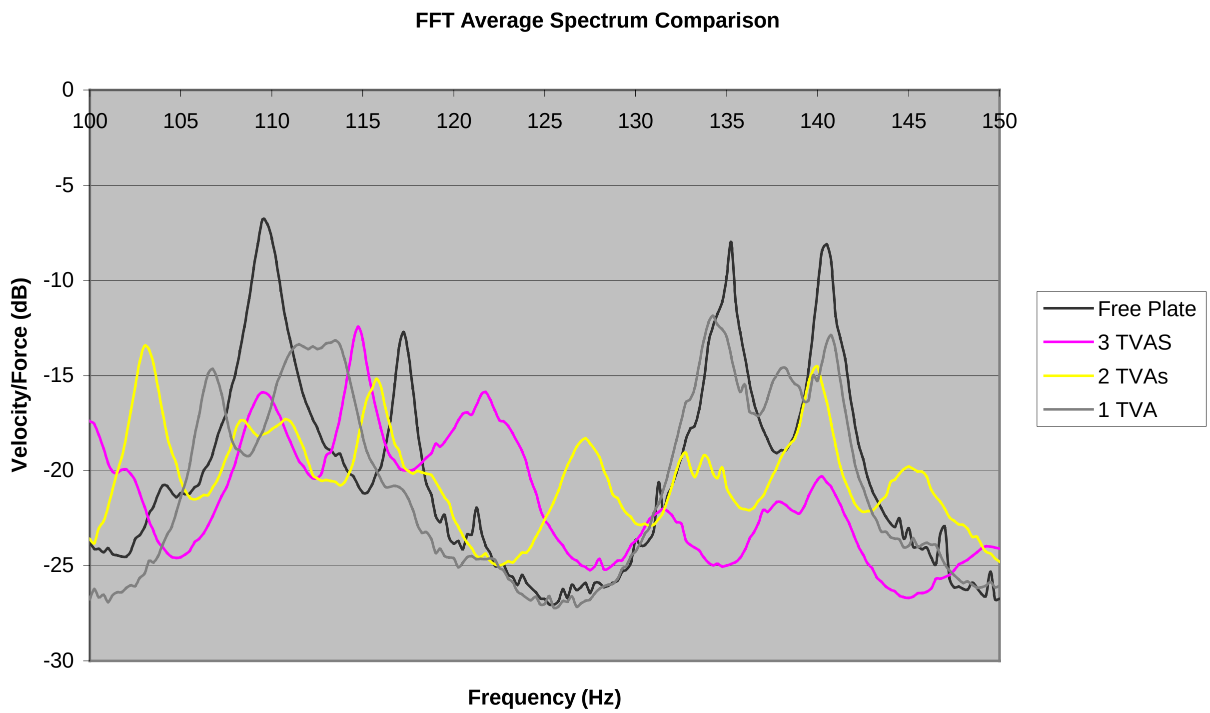
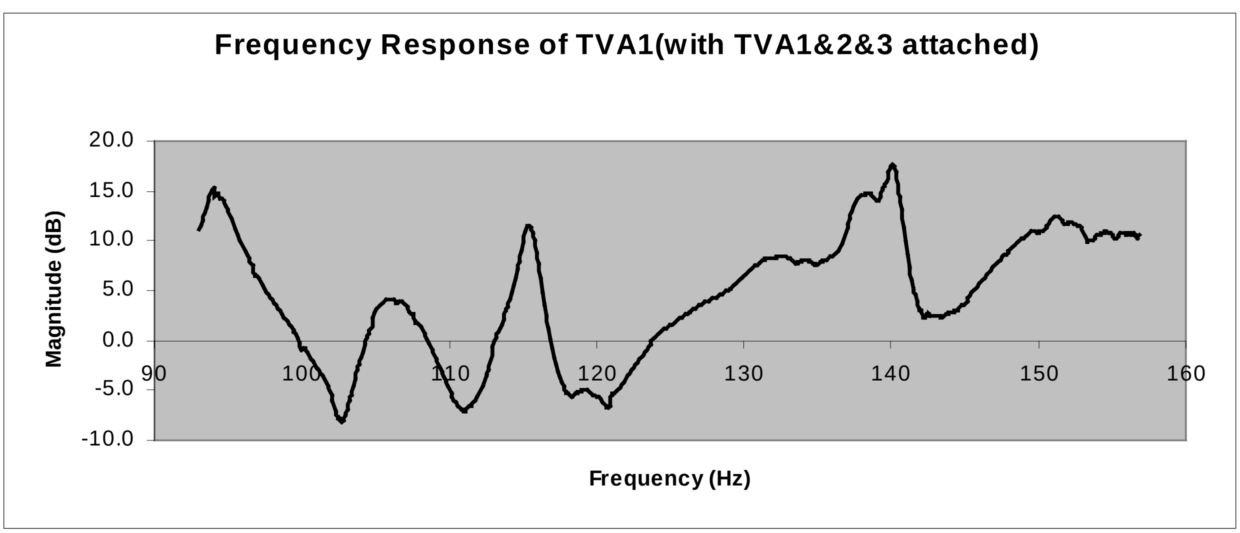


Figure 5 Average Spectrum (Vel./Force) of plate using 4 different configurations

To understand how much vibration the TVA undergoes and to observe how the TVA behaves when attached to a base structure, accelerometers were placed on the TVAs themselves. This test was conducted in the same way as the shaker test described earlier except the TVAs are excited by the plate (base structure) rather than a shaker. Two accelerometers were used again, one on the base of the TVA bracket and one mounted directly on the TVA. It was found that modes that were resonant in the panel were transferred into the TVA, which indicates there may have been some "rocking" (lateral vibration of brackets). This is shown in the graph of Figure 6. It was expected that these graphs would be analogous to the graphs obtained as shown in Figure 3. It can be seen however, that the peaks in the graphs of Figure 6 also correspond to some of the peaks (resonant modes) found in the plate (Figure 5). The tuning frequency of 132Hz measured in the shaker test has shifted when attached to the plate. From the following plots it appears that the TVA may be tuned to close to 115Hz or close to 140Hz. This shifting phenomenon of the absorber tuning frequency is not fully understood at this stage.

Figure 6 Frequency responses of TVA1 mounted on flat plate



5 FUTURE WORK:

The last experiment that remains to be conducted will look at Transmission Loss of the flat plate between the walls of the two adjacent rooms. Loud speakers will be used to introduce noise and panel vibration into one of the rooms and part of the sound energy would be transmitted through the test partition. The space averaged sound pressure levels in both rooms are measured with the condition that points of measurement are well away from the source in the source room (make sure it is not in the near field of the source). The receiver 'room constant' can be easily determined by measurements of reverberation decay time, that is time taken for the reverberation field to be reduced by 60dB or using a standard sound power source. The difference in sound pressure levels between both rooms would be the Noise Reduction, NR. Transmission loss is related to both NR and the room constants and test partition's surface area. Once again two sets of measurements will be required: one set or measurements with just the flat plate and one set with the TVAs attached to the plate.

Future work on this project should look at using adjustable (tuneable) TVAs to target the particular resonant peaks. Also resonant peak shifts when extra mass is added to the plate should be analysed (when TVAs are introduced) so that maximum attenuation can be achieved using the TVA. Also, to remove any possibility of the absorber rotating and producing vibrations in the lateral direction, symmetrical absorbers should be used (possibly two absorbers fixed back to back).

6 CONCLUSIONS:

The results obtained in these experiments show that TVAs can have a significant effect on reducing vibration at and near the tuned frequency. This indicates that TVAs can be used effectively over a relatively large bandwidth effectively. There is also noticeable shift of the tuning frequency of the TVAs when attached to the plate. The reason behind this phenomenon is yet to be analysed. By extending work currently undertaken from these findings it should be possible to isolate the reason of the TVA tuning frequency shifting when attached to a base structure.

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