Vibration Suppression of Electronic Box by a Dual Function Piezoelectric Energy Harvester-Tuned Vibration Absorber

SAJID RAFIQUE*, AND SADIQ ALI SHAH**

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ABSTRACT

Over the past few years, remarkable developments in piezoelectric materials have motivated many researchers to work in the field of vibration energy harvesting by using piezoelectric beam like smart structures. This paper aimed to present the most recent application of a dual function piezoelectric device which can suppress vibration and harvest vibration energy simultaneously and a brief illustration of conventional mechanical and electrical TVAs (Tuned Vibration Absorber). It is shown that the proposed dual function device combines the benefits of conventional mechanical and electrical TVAs and reduces their relative disadvantages. Conversion of mechanical energy into electrical energy introduces damping and, hence, the optimal damping required by this TVA is generated by the energy harvesting effects. This paper presents the methodology of implementing the theory of 'electromechanical' TVAs to suppress the response of any real world structure. The work also illustrates the prospect of extensive applications of such novel "electromechanical" TVAs in defence and industry. The results show that the optimum degree of vibration suppression of an electronic box is achieved by this dual function TVA through suitable tuning of the attached electrical circuitry.

Key Words: **Vibration Control, Energy Harvesting and Vibration Absorber.**

1. INTRODUCTION

Vibration is a global phenomenon and is present in all engineering structures experiencing dynamic loading. Regular occurrence of vibrations beyond a certain level can have detrimental effects on these structures such as deterioration in performance and efficiency which can eventually lead to catastrophic failure. The situation becomes more critical where sensitive electronic equipment is needed to be isolated from harsh vibrations of its base. Conventionally, viscoelastic dampers are used to suppress the base vibration by dissipating vibration energy into heat, maintaining the vibration levels in the desired limits. However, it is not so straight forward to control the required optimum damping in the viscoelastic dampers, exhibiting the major shortcoming of this kind of dampers.

* School of Mechanical, Aerospace & Civil Engineering, University of Manchester, M13 9PL, UK.
** Assistant Professor, Department of Mechanical Engineering, Mehran University of Engineering & Technology, Shaheed Zulfiqar Ali Bhutto Khairpur Mirs Campus.

Over the last few years, notable research has been carried out to develop such smart dampers which can damp the unwanted vibrations by converting vibration energy into electrical energy instead of wasting it away into heat. Such dual function vibration dampers are called "energy harvesting vibration absorbers" or "electromechanical" vibration absorbers [1]. In the following sections, the working of Mechanical and Electrical TVAs is presented prior to illustrating the details of the proposed "electromechanical" TVA.

1.1 Mechanical TVA

Typically, a mechanical TVA can be defined as an auxiliary system whose parameters, such as mass, stiffness and damping, may be tuned to suppress the vibration of its host structure [2,3]. The auxiliary system can be represented as a spring-mass-damper system as shown in Fig. 1.

It was illustrated in [4] that the mass of a mechanical TVA, as shown in Fig. 1(b), can be divided into two parts, (i) effective mass ma,eff and (ii) the redundant mass mred. It is the effective part of the TVA mass which is responsible for vibration suppression of a structure at the point of attachment. The redundant part of the TVA mass does not take part in vibration suppression and is simply added to the host structure. For the TVA under consideration, the total mass of the TVA is 5.5 gms, the effective mass is 3.3 grams and the rest of 2.2 gms is redundant mass and simply added to the structure. The total modal mass of the electronic box at the point of attachment is 276 gms. Neglecting the redundant mass of the TVA and any damping in the original host structure, the optimal tuning condition can be represented as [1-4]:

$$\frac{\omega_a}{\Omega_s} = \frac{1}{1+\mu} \tag{1}$$

where

$$\mu = \frac{m_{a,eff}}{M_A^{(s)}} \tag{2}$$

Where ω_a is the first resonance frequency of the TVA, Ω_s is the troublesome excitation frequency, μ is the ratio of the effective mass $m_{a,eff}$ of the absorber to the mass $M_A^{(s)}$

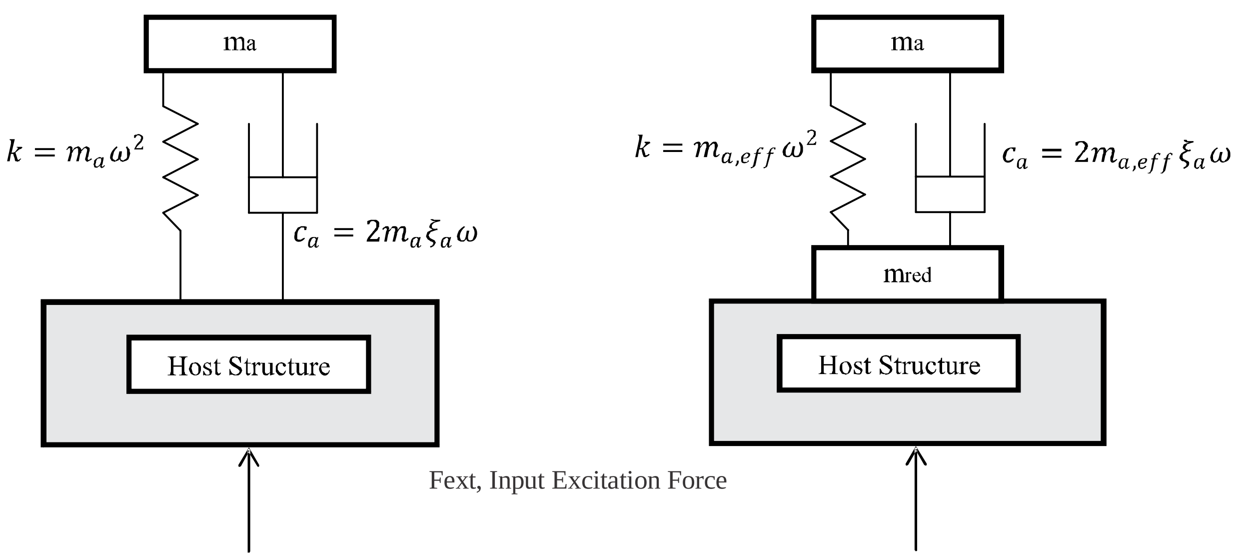

FIG. 1(a). ACTUAL ARBITRARY TVA

FIG. 1(b). EQUIVALENT TWO DEGREE OF FREEDOM MODEL OF TVA associated with the host structure's targeted mode at the targeted degree of freedom. A TVA requires an optimal amount of damping in order to suppress the contribution of the targeted mode of the vibration frequency response at the point of attachment over a wide band of excitation frequencies [1-3] as shown in Fig. 1(b). The required optimal viscous damping ratio can be calculated as [1,3]:

$$\xi_{opt} = \sqrt{\frac{3\mu}{8(1+\mu)^3}} \tag{3}$$

The practical implementation of the exact amount of damping in conventionally-damped TVAs creates a design challenge. Additionally, if once it's designed and implemented, such damping may be difficult to re-tune in response to a deviation in the system parameters [1]. The inability to efficiently control and adjust the damping in mechanical TVA is one of its main limitations.

1.2 Electrical TVA

In order to sort out the above mentioned shortcomings of mechanical TVAs, it was demonstrated by Flotow, et. al. [5] that a TVA can also be realised with other physics also known as 'electrical' TVA [5-6]. In such a device the auxiliary system is a piezoelectric shunt circuit which is analogue to a spring-mass-damper system of a mechanical TVA. In electrical TVA, a piezoelectric patch is bonded directly to the host structure and connected across an external inductor-resistor circuit as shown in Fig. 2(a-b) [6].

The piezoelectric patch is used to convert the vibration energy of the host structure into electrical energy, introducing electrical damping [1,6]. As shown in Fig. 2, a capacitor can also added into the circuit, turning it into an RLC circuit. The electrical energy produced in this way is then dissipated most efficiently as Joule heating through the resistor when the electrical resonance produced by the LC components is close to the frequency of the targeted mode [1,6]. In such RLC (Resistor Inductor Capacitor) circuit, the optimal resistance value ensures that the contribution of the targeted mode to the vibration frequency response at a selected location is suppressed over a wide band of excitation frequencies [1]. In this kind of TVA, exact amount of optimal damping can be added or the same can be amended according to changing dynamic conditions, by changing the value of the resistor [1].

It is important to note that the electrical TVA is more compact and its performance is relatively less temperature dependent as compared to viscoelastic TVA [6-9]. Furthermore, the damping level can be easily controlled in electrical TVA. Besides these advantages, electrical TVA also holds certain disadvantages. The analysis required to calculate the optimal parameters of the electrical TVA is not tractable to complex generic host structures [1]. In contrast, the classical theory of the conventional mechanical TVA is readily applicable to any arbitrary host structure of any complexity as the only host structure data it requires are the target frequency Ω_s and the modal mass $M_A^{(s)}$ of the targeted mode [1-3].

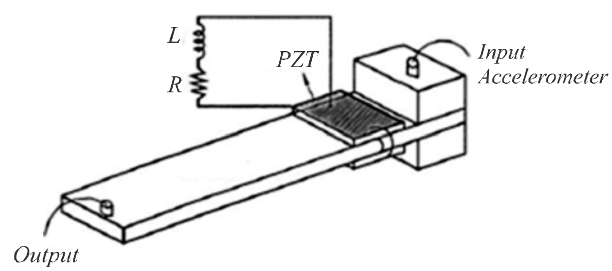

FIG. 2(a). AN EXAMPLE OF AN "ELECTRICAL" TVA APPLIED TO HOST STRUCTURE (CANTILEVER) WITH SERIES RL

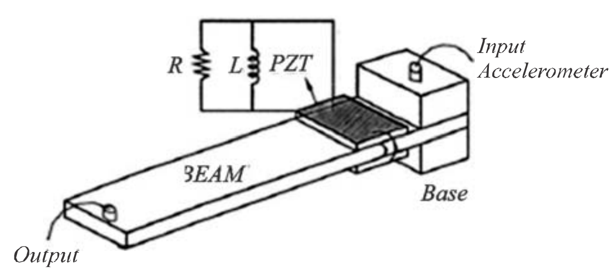

FIG. 2(b). WITH PARALLEL RL CIRCUIT [6]

1.3 The Proposed "Electromechanical" TVA

Having discussed the workings of mechanical and electrical TVAs and their respective limitations, the model of an "electromechanical" TVA is presented here. The electromechanical TVA can be realised if the damping element in the Fig. 1 may be replaced by the damping produced due to energy harvesting [10-11]. The modal parameters of the target mode can be calculated using the method given in [12]. It is important to note that, unlike in mechanical TVA, the damping level in the electromechanical TVA can easily be controlled by adjusting the value of electrical load connected to the energy harvesting circuit. This paper presents the model and analysis of such dual function energy harvesting/ TVA or EH/TVA beam device, in which a suitably shunted piezoelectric beams are used as a TVA to suppress the vibration of an electronic box as shown in Fig. 3. The proposed 'electromechanical' TVA combines the advantages of mechanical and electrical TVAs and, thereby reducing their disadvantages.

Before embarking into the details of the methodology part of the electromechanical TVA, it is worthwhile to see the power output by various vibration-based energy harvesting techniques such as piezoelectric, electrostatic and electromagnetic transduction in Tables 1-2 [1].

In Table 2 [1] below is presented the power consumption survey by some of the portable devices in order to visualise the future usage of vibration energy harvesting.

2. METHODOLOGIES

As mentioned in preceding sections, the aim of the study is to dampen the targeted resonance peak of the FRF (Frequency Response Function) (FRF or receptance) of

TABLE 1. POWER GENERATED BY VARIOUS VIBRATION TRANSDUCTION METHODS

No.	Vibration Conversion Mechanism	Energy Generated (mW/cm^3)
1.	Piezoelectric	100
2.	Electromagnetic	0.5-8
3.	Electrostatic	8-42

TABLE 2. POWER REQUIREMENT OF VARIOUS MODERN ELECTRONIC DEVICES

No.	Electronic Device	Power Consumption
1.	Electronic Watch or calculator	1mW
2.	Implanted medical devices	10mW
3.	Hearing aids	100mW
4.	Bluetooth transceiver	45mW
5.	Palm MP3	100mW
6.	Touch phone, active mode without GPS system on	25mW

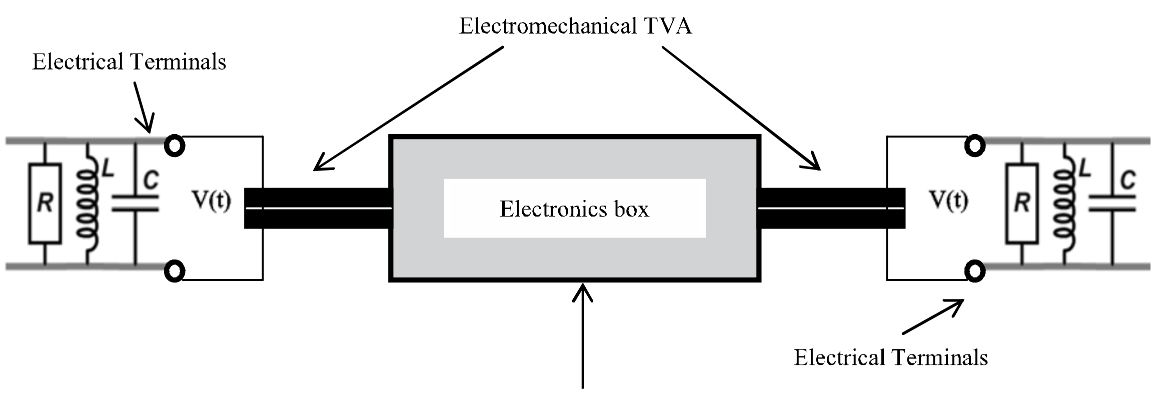

FIG. 3. WORKING PRINCIPLE OF THE PROPOSED 'ELECTROMECHANICAL' TVA

an electronic equipment box, to the external excitation F_{ext} over the range of troublesome excitation frequencies ω, in the vicinity of a natural frequency. The frequency responses of the electronic box with and without the proposed TVA attached are calculated by following the theory mentioned in reference Bonello, et. al. [10], and Rafique, et. al. [11]. Furthermore, the classical theory of Hartog, [3] is adapted in order to derive the optimal level of hypothetical viscous damping that will be used as the benchmark for the performance of the electromechanical TVA[1].

The effective mass of the cantilevered piezoelectric beam can be calculated by following the technique of Kidner, et. al. [4]. The proposed theory is verified by considering the test case whose parameters are given in Tables 3-4. As described earlier, in order to design a TVA to suppress the vibration of any structure, the only information it requires is the target frequency and the modal mass of the structure for the desired DOF (Degree of Freedom).

3. RESULTS AND ANALYSIS

For this test case the mass ratio was calculated as μ=1.86%. The optimal damping ratio ζ_{aopt}(=8.24%) is calculated using Equation (3). The black thick solid dotted curve in Fig. 4 shows the equivalent lumped-parameter TVA receptance with optimal parameters.

The excellent agreement between the thick black dotted and the thin red solid lines validate the theory of the proposed 'electromechanical' absorber as shown in Fig. 4. The black solid line in Fig. 4 shows the receptance of the electronic box without the TVA attached. It can be observed that the response of the electronic box before and after the TVA has been suppressed significantly. Another major advantage of the proposed TVA is that its weight is less than 2% of the effective modal mass of the electronic box, showing its compactness and agility.

It is also important to note that the actual viscous damping present in the TVA is 1%. However, the optimum damping required for the TVA to optimally tune itself is ζ_{aopt}=8.24%. This shows that the remainder of the damping (7.24%) is provided by the energy harvesting effect of the TVA. In Fig. 4, the thick solid blue line shows the response of the 'electromechanical' TVA without the electrical damping effects. This demonstrates the significance of using the accurate values of resistor, inductor and the capacitor in the attached circuit, in order to precisely tune the TVA of Fig. 3.

As presented in [1], there can be various electronic configurations of the RLC circuits to attain the tuned conditions. However, a parallel resistor, inductor and capacitor configuration is used in this work with both

TABLE 3. MODAL PARAMETERS OF ELECTRONIC BOX ASSEMBLY

$\Omega_1/(2\pi)$(Hz)	0	$M_A^{(1)} = \dfrac{1}{\{\hat{\phi}_A^{(1)}\}^2}$ (gram)	276
$\Omega_2/(2\pi)$(Hz)	127.65	$M_A^{(2)} = \dfrac{1}{\{\hat{\phi}_A^{(2)}\}^2}$ (gram)	470

TABLE 4. PARAMETERS OF EITHER BEAM OF THE ELECTROMECHANICAL TVA [10]

Property	Units	Value
Young's Modulus of the Piezoelectric, Y_p	GPa	66
Young's Modulus of the shim, Y_{sh}	GPa	72
Density of the piezoelectric material	kg/m³	7800
Density of the shim material	kg/m³	2700
Piezoelectric constant, d_{31}	pm/volt	-190
Relative dielectric constant (at constant stress)	-	1800
Overhanging length of the beam, l	mm	58.75
Width of the beam, b	mm	25
Thickness of each piezoelectric layer, h_p (upper & lower layers)	mm	0.267
Thickness of the shim (substrate), h_{sh}	mm	0.285

sides of the piezoelectric beams attached to a similar circuit as shown in Fig. 3. The optimum values of the attached RLC components were determined through a MATLAB optimization program. For the present study, the optimum value of resistor was 44.8KΩ, inductor of 13.5 Henry and the external capacitor of 76nF was used to generate the optimum damping as well as to optimally tune the TVA to suppress the target excitation frequencies.

It is also important to note that the above mentioned optimum values (of RLC) are not unique and these can vary if the value of any of the component in the RLC combination changes. In that case, the optimization program will generate totally a new set of values of the RLC circuit. The results in Fig. 4 clearly demonstrated that the electromechanical TVA is capable of producing a frequency response that closely mimics the benchmark response (shown by the black solid dotted line). In addition to achieve the optimum vibration attenuation, the proposed 'electromechanical' TVA is also generating useful electrical energy which can be conditioned and stored by adding the appropriate circuitry in the model. However, the scope of this non-linear analysis is beyond the scope of the present work.

5. CONCLUSIONS

This paper presented an analytical study on the most recent application of a dual function piezoelectric device (i.e. electromechanical TVA) which can suppress vibration and harvest vibration energy simultaneously. It was demonstrated that the novel 'electromechanical' TVA can optimally suppress the vibration of an electronic box over a range of troublesome excitation frequencies. This device was formed from two symmetric dual-function energy harvesting/TVA beams which were suitably shunted in order to attenuate a vibration mode of an electronic box. The optimised damping of this TVA device was generated by the piezoelectric energy harvesting mechanism of the vibrating beams by using the optimised values of RLC

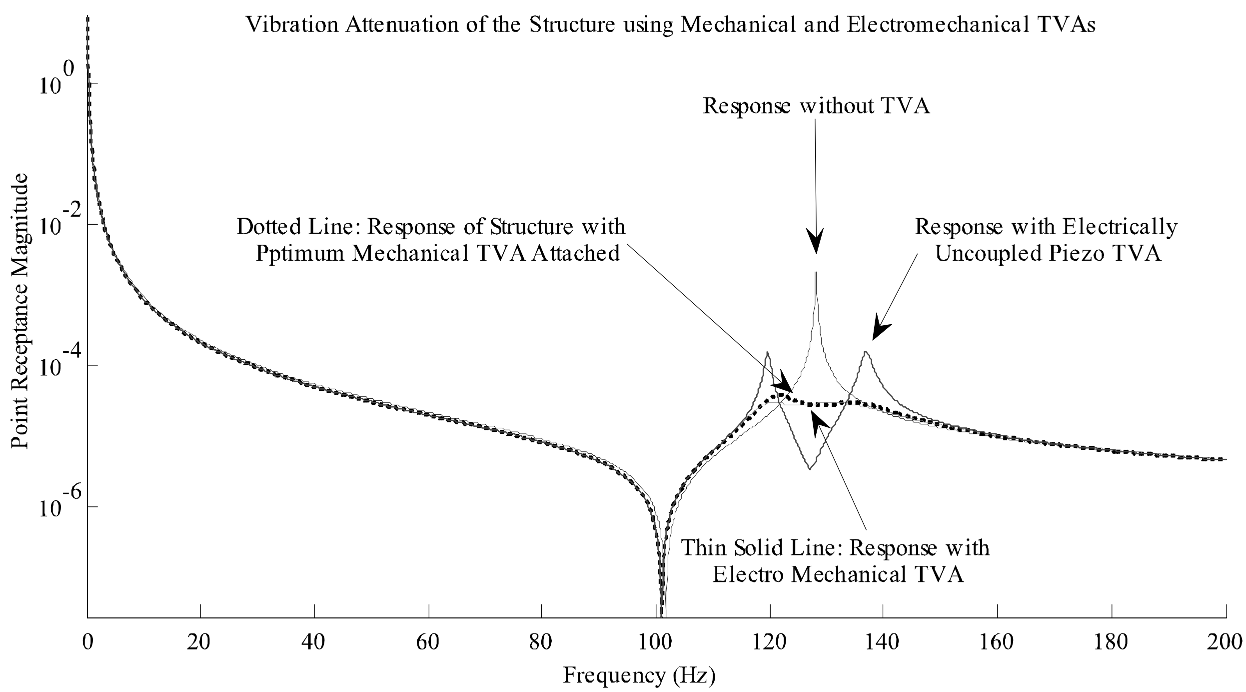

FIG. 4: RESULTS OF OPTIMUM VIBRATION SUPPRESSION OF AN ELECTRONIC BOX WITH AND WITHOUT ' MECHANICAL' AND ' ELECTROMECHANICAL' TVAS AT THE POINT OF ATTACHMENT

circuit. The proposed electromechanical TVA shared the advantages of the classical mechanical TVA and its electrical analogue (i.e. a shunted piezoelectric patch bonded directly to the host structure), eliminating their relative disadvantages and combining their advantages. It is important to note that the effective mass of less than 2% dampened the vibration mode of the electronic box, exhibiting the compactness of the proposed absorber. In conventional TVAs the effective mass of the absorbers are normally around or more than 10%.

The work successfully demonstrated the methodology of implementing the theory of 'electromechanical' TVAs to suppress the response of any real structure. It is envisaged that the proposed 'electromechanical' TVA can have enormous applications in many commercial and defence applications as vibration are present in majority of engineering systems. It is worth mentioning that, though, the energy generated by these TVA is very small but still it hold potential to power various modern electronic applications as the consumption of many wireless electronic nodes is in micro to milliwatts.

6. NOMENCLATURES

ω_a	First resonance frequency of TVA
ω	Excitation frequencies
Ω_s	Troublesome excitation frequency
$m_{a,eff}$	Effective mass of absorber
$m_{a,red}$	Redundant mass of absorber
m	Total mass
μ	Mass ratio
$M_A^{(s)}$	Modal mass
Ω	Unit symbol of electrical resistance
ζ_{opt}	Optimum damping
ζ_a	Damping of absorber
c_a	damping co-efficient
RLC	Resistor Inductor Capacitor
$\hat{\phi}_A^{(1)}$	Mass normalised modal coordinate of first mode at point A
nF	Nano-Farad
w	Width of the absorber
l	Length of the absorber
t	Thickness of the absorber, shim and piezoelectric with subscripts

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