MATEMATICA PER L'AUTOMAZIONE INDUSTRIALE

Sistemi avanzati per l'orientamento corretto di pezzi di produzione: l'esperienza del progetto PoCN di AREA Science Park

DSMIF, Università di Udine

Dipartimento di Scienze Matematiche, Informatiche e Fisiche

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SWITCHING AND SWEEPING CONTROL UNITS FOR VIBRATION AND SOUND RADIATION CONTROL

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OVERVIEW

Part A) *Switching – Sweeping Tuneable Vibration Absorbers*

- Background on tuneable vibration absorbers
- Single switching and sweeping vibration absorber
- Experimental implementation
- Multiple sweeping vibration absorbers

Part B) *Sweeping Shunted Piezoelectric Patches*

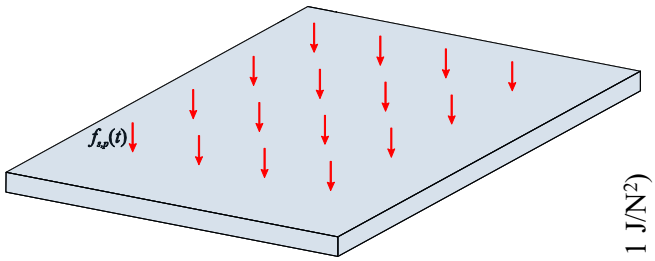
- System description
- Single mode model
- Tuning laws and control effects produced by different types of shunts
- Sweeping shunt operation principle
- Control effects produced by multiple sweeping shunts

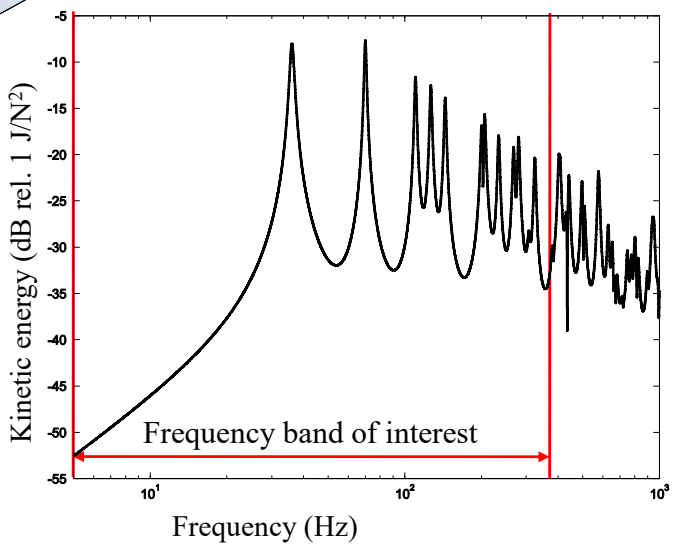
PART A

SWITCHING AND SWEEPING TUNEABLE VIBRATION ABSORBERS

BACKGROUND

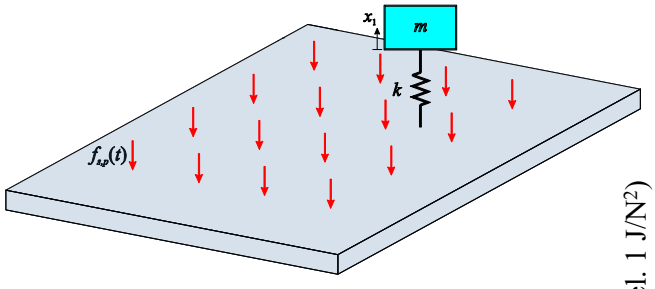
- The structural response of a lightly damped structure is characterised by well separated resonances at low frequency

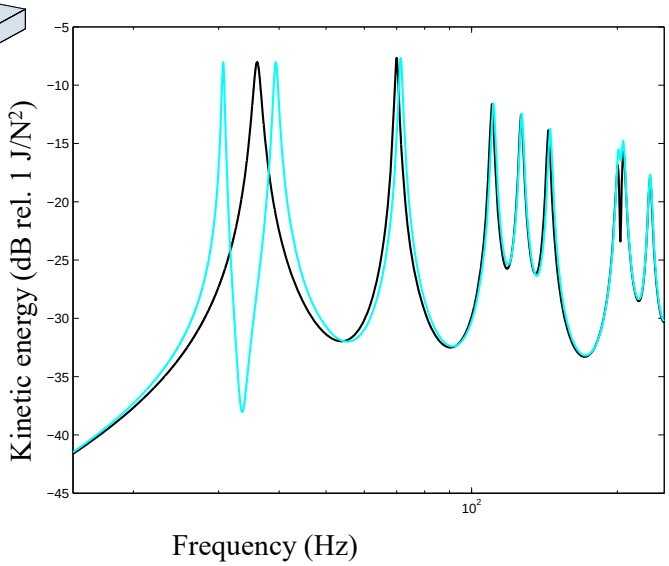




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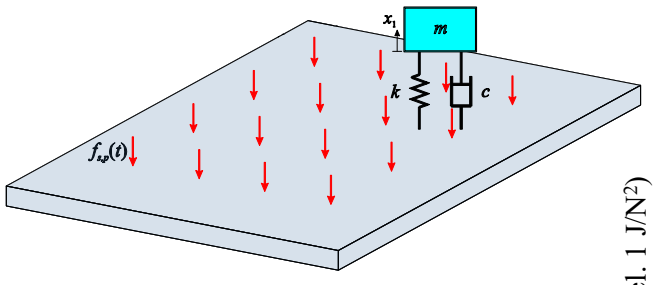
- An undamped vibration absorber is able to reduce the structural response at a single resonance frequency but introduces a new resonance peak

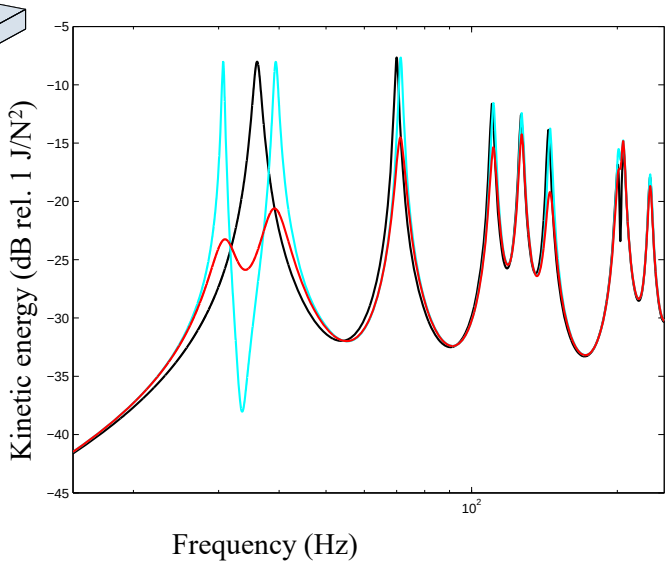




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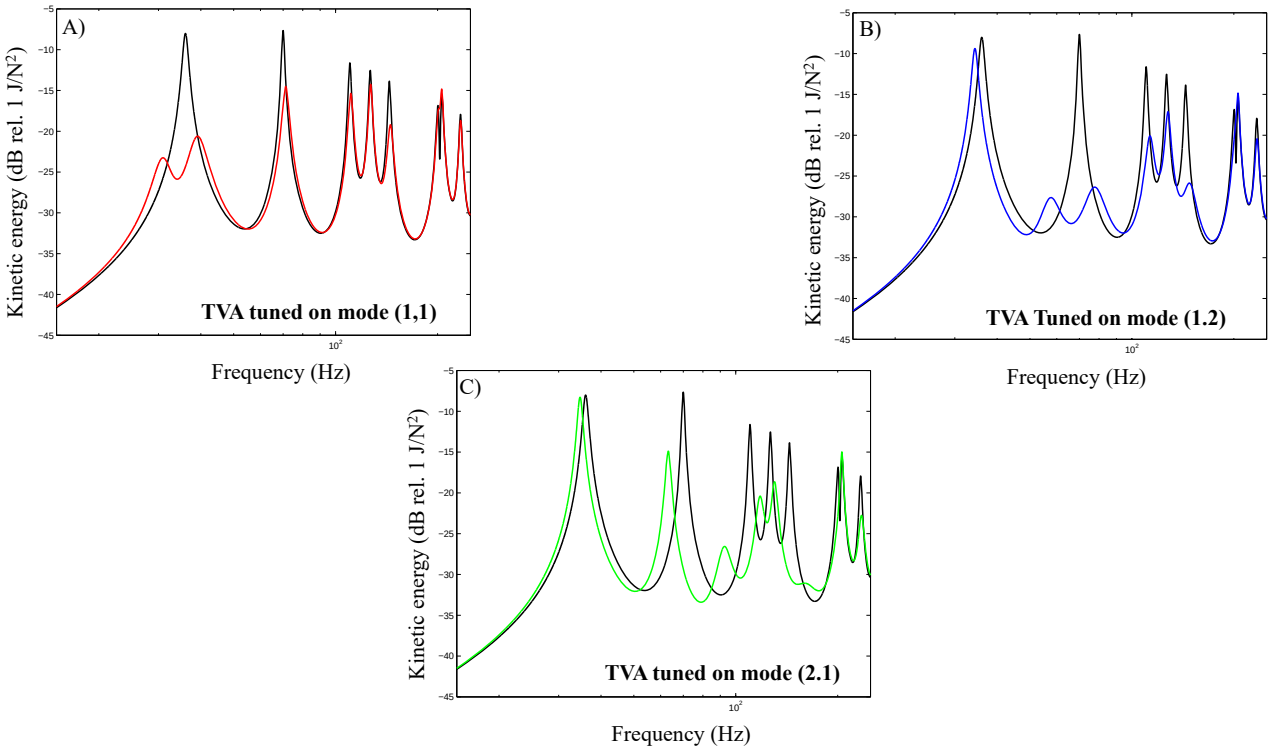
- By increasing the internal damping of the device it is possible to reduce the vibration of the structure in a narrow frequency band





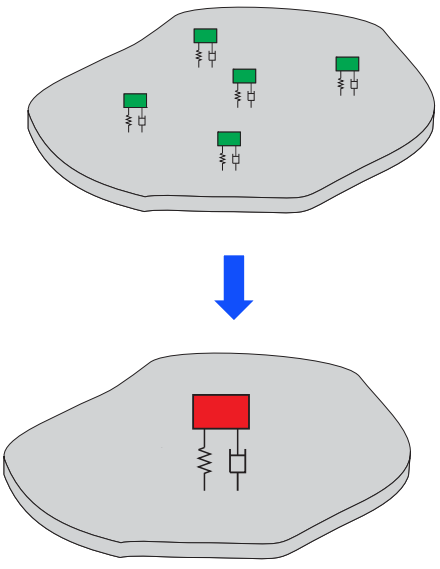
BACKGROUND

- The TVA can be tuned to control the resonant response due to one flexural mode at the time



INTEGRATED VIBRATION ABSORBER

- To control the response at multiple resonance frequencies multiple absorbers are required

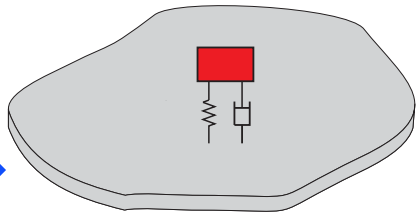


Traditional approach
Requires large number of units

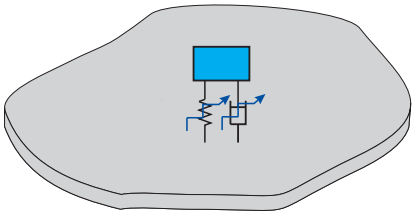
Proposed approach
Integrated absorbers

SWITCHING AND SWEEPING VIBRATION ABSORBERS

Proposed approach
Integrated absorbers

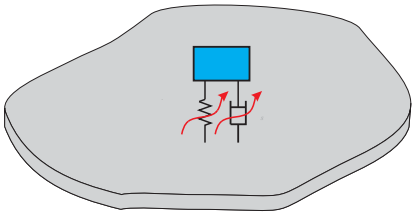


Switching operating mode



suspension stiffness and damping can be selected among three different **<u>discrete</u>** pairs of values, called states

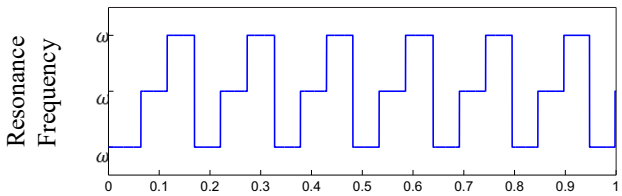
Sweeping operating mode



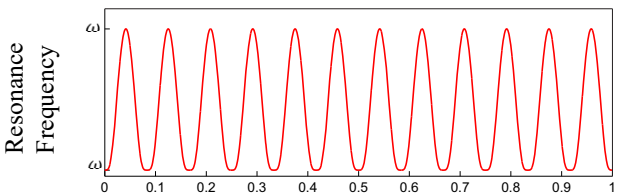
suspension stiffness and damping varies **<u>continuously</u>** between two values

SWITCHING AND SWEEPING VIBRATION ABSORBERS

Switching operating mode



Sweeping operating mode



Switching law

$$\omega_{switching} = \begin{cases} \omega_A \\ \omega_B \\ \omega_C \end{cases}$$

$\omega_{A,B,C}$ = target structural resonance frequency

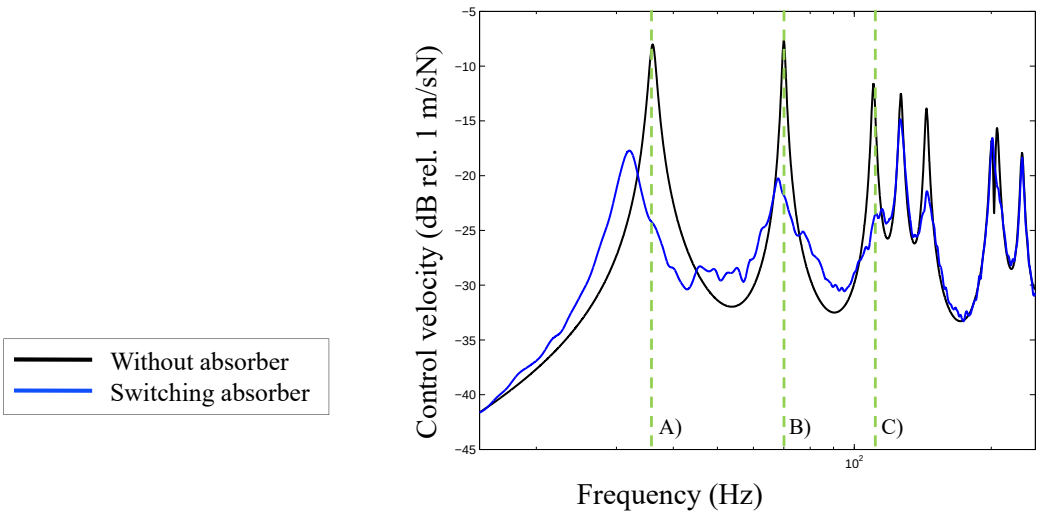
Sweeping law

$$\omega_{sweeping} = \omega_n + \omega_u \sin^2(\omega_s t)$$

ω_n = absorber resonance frequency
ω_u = upper sweeping frequency

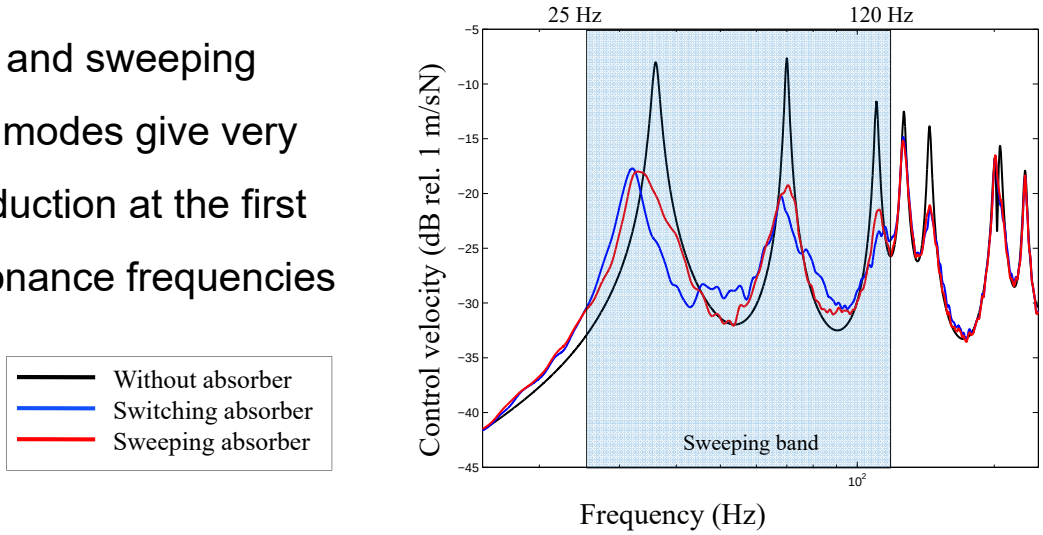
SIMULATION OF SWITCHING VIBRATION ABSORBER

- Switching versus sweeping vibration absorbers



SIMULATION OF SWITCHING AND SWEEPING VIBRATION ABSORBERS

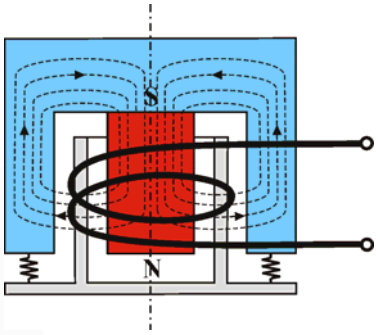
- Switching versus sweeping vibration absorbers
- Switching and sweeping operating modes give very similar reduction at the first three resonance frequencies



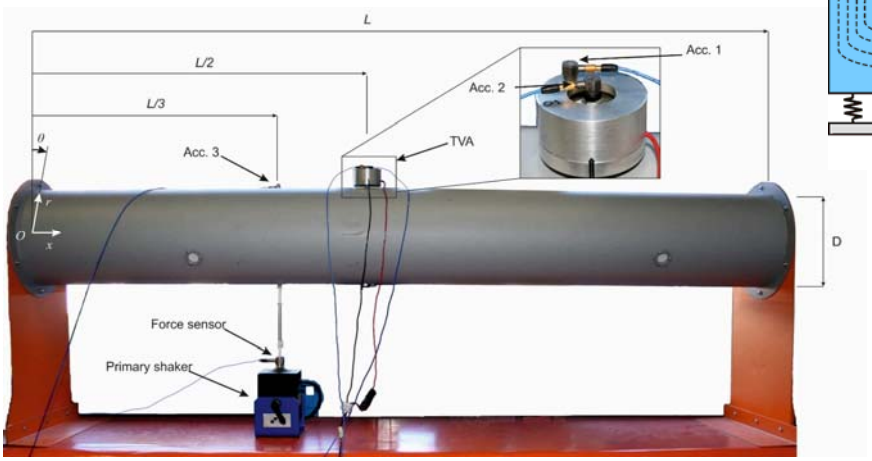
(dB)	Vibration reduction around resonance frequency			Broadband vibration reduction
	$f_1 = 91$ Hz	$f_2 = 141$ Hz	$f_3 = 296$ Hz	60-300 Hz vibration reduction
Switching TVA	16.3	14.1	12.5	5.3
Sweeping TVA	12.2	11.7	10.2	4.6

EXPERIMENTAL TUNEABLE VIBRATION ABSORBER

- Test structure: lightly damped pipe supported at the two ends and excited by a broadband force
- The tuneable vibration absorber consists of a coil magnet actuator equipped with two accelerometers



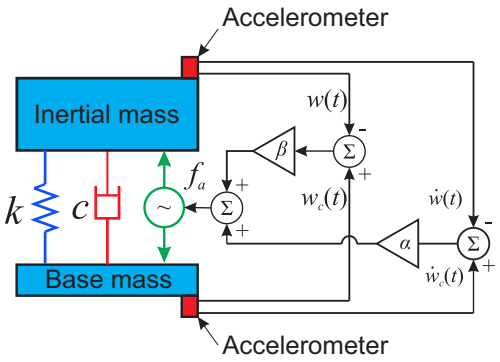
Experimental setup



EXPERIMENTAL TUNEABLE VIBRATION ABSORBER

- A relative displacement and a relative velocity feedback is used to tune the resonance and damping of the absorber

Scheme of the absorber



Active force in parallel with the suspension

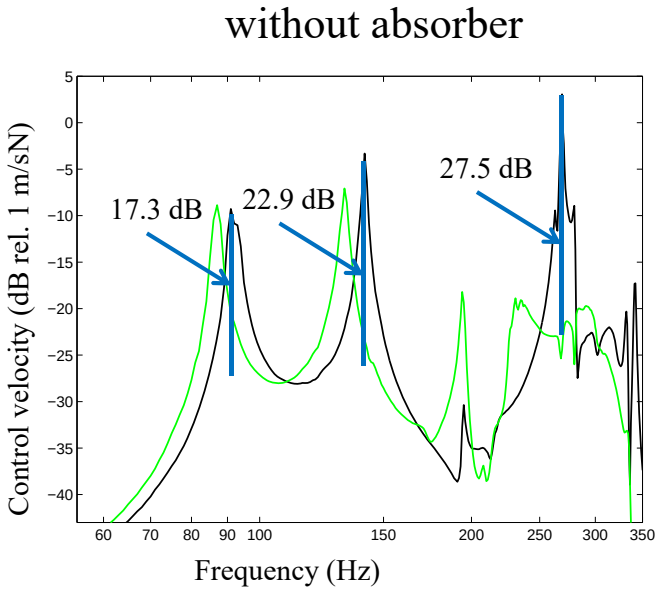
Displacement gain | Relative displacement

$$f_a = Bl[\alpha\, w_r(t) + \beta\, \dot{w}_r(t)]$$

Transduction coefficient | Velocity gain | Relative velocity

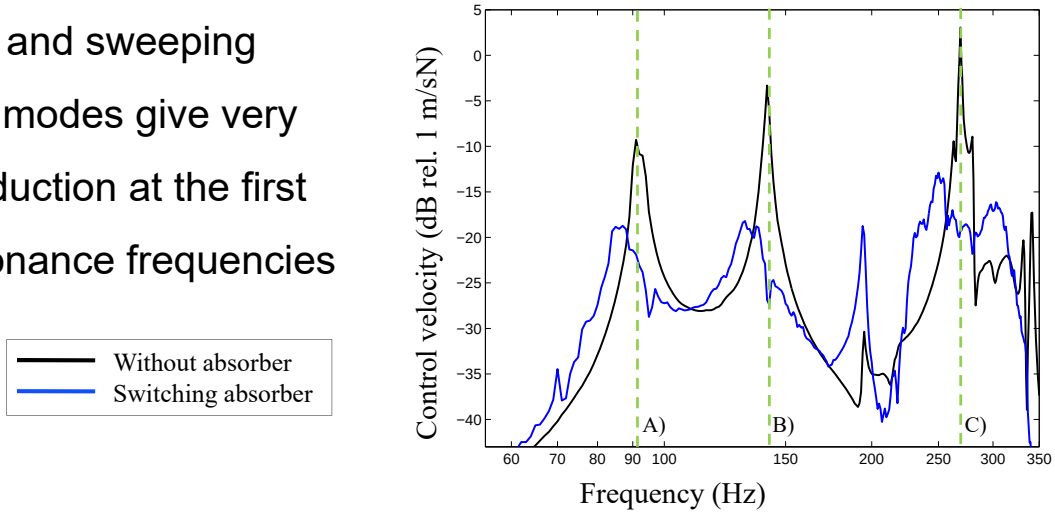
EXPERIMENTAL RESULTS FOR CLASSICAL TVA

- Measured control velocity



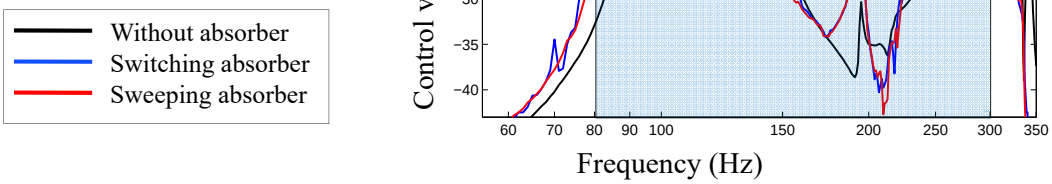
EXPERIMENTAL SWITCHING VIBRATION ABSORBER

- Switching versus sweeping vibration absorbers
- Switching and sweeping operating modes give very similar reduction at the first three resonance frequencies



EXPERIMENTAL SWITCHING AND SWEEPING VIBRATION ABSORBERS

- Switching versus sweeping vibration absorbers
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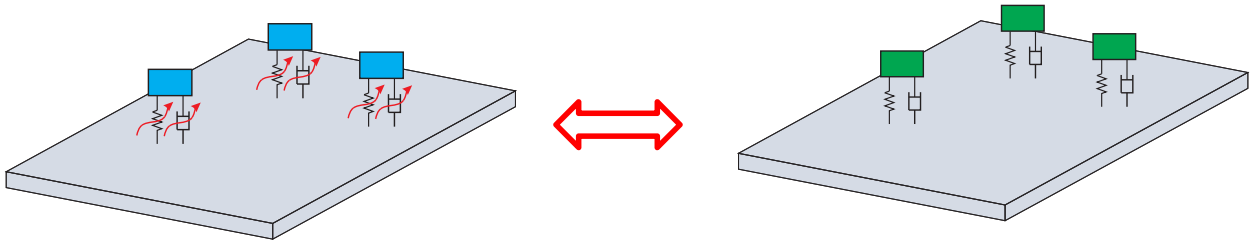
(dB)	Vibration reduction around resonance frequency			Broadband vibration reduction
	$f_1 = 91$ Hz	$f_2 = 141$ Hz	$f_3 = 296$ Hz	60-300 Hz vibration reduction
Switching TVA	9.8	15.7	15.9	6.7
Sweeping TVA	6.3	11.2	18.1	6.4

MULTIPLE SWEEPING VIBRATION ABSORBERS

- Test structure: lightly damped simply supported structure excited by multiple uncorrelated random forces
- Control: There sweeping absorbers versus three classical absorbers

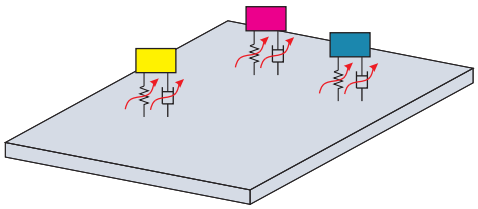
Sweeping absorbers

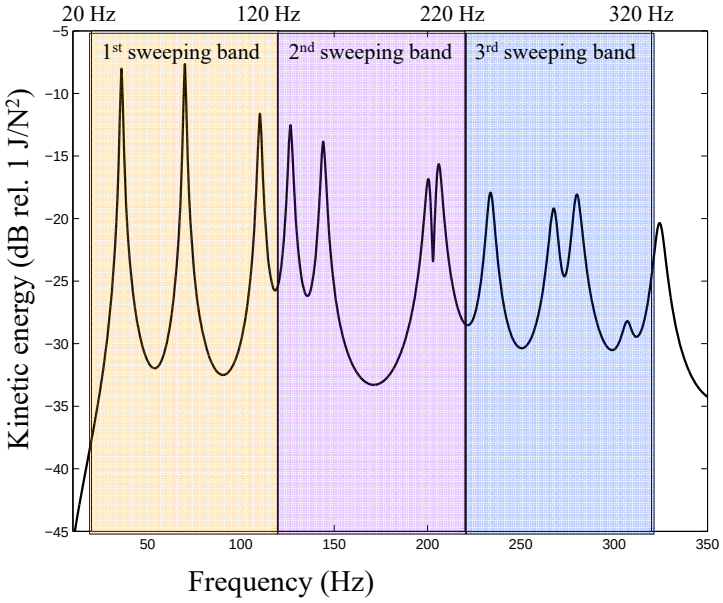
Classical absorbers



MULTIPLE SWEEPING VIBRATION ABSORBERS

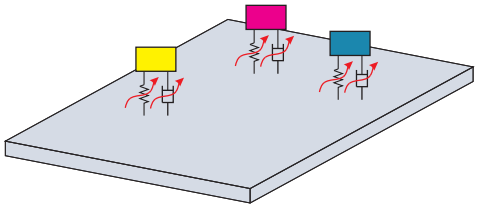
- Response of a plate excited by multiple uncorrelated random forces and controlled by three sweeping absorbers





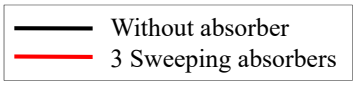
MULTIPLE SWEEPING VIBRATION ABSORBERS

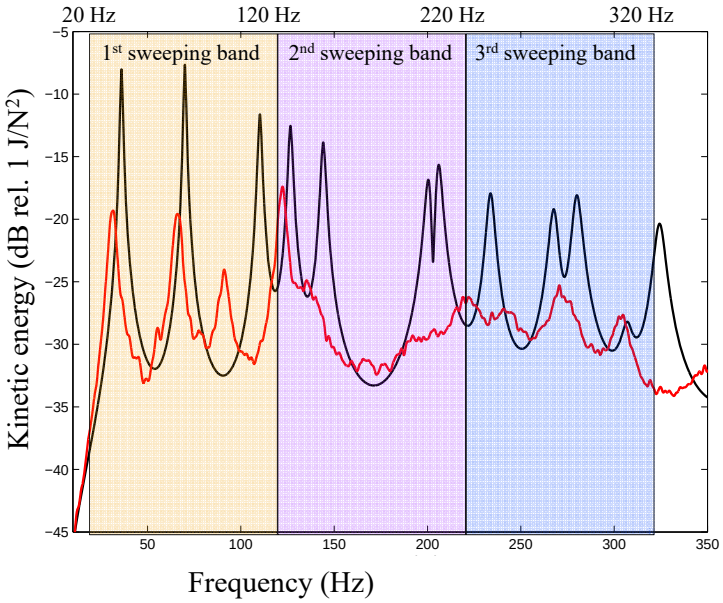
- Good reduction is obtained in all three sweeping bands



Flexural vibration reductions in (dB)

Frequency band (Hz)	20-120	120-220	20-320	20-320
Un-tensioned panel	6	5	4	5.2





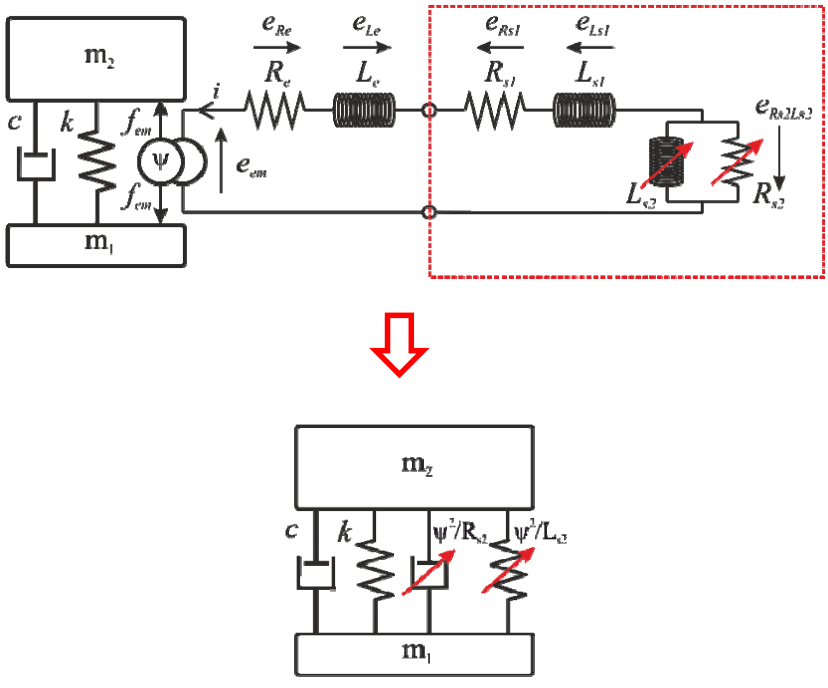
CONCLUSIONS PART A

- Switching and sweeping absorbers are able to control the structural vibration due to three structural modes

- The sweeping operation mode is particularly appealing since it does not require an exact tuning and are robust with respect of changing of the structural response

- Multiple sweeping vibration absorbers can be used to control the vibration of a structure subjected to broadband excitation

CURRENT WORK



Applications



PART B

SWEEPING

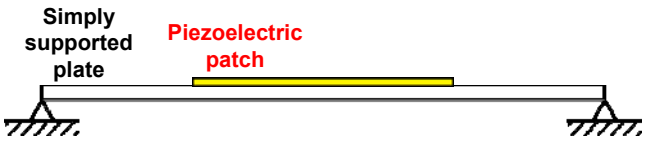
SHUNTED PIEZOELECTRIC ABSORBER

SYSTEM DESCRIPTION



Parameters of the plate	
Parameter	**Value**
Material	*Aluminium*
Dimensions	*414x314x1 mm*
Damping ratio	*2%*

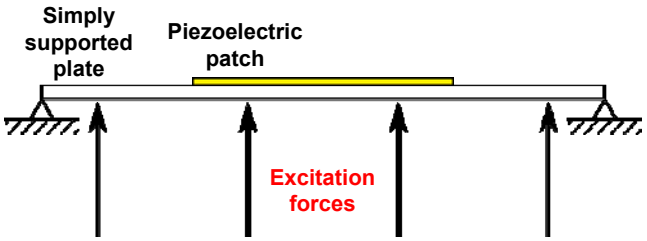
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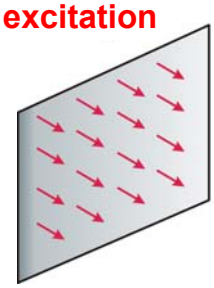
Parameters of the plate	
Parameter	**Value**
Material	*Aluminium*
Dimensions	*414x314x1 mm*
Damping ratio	*2%*

Parameters of the patch	
Parameter	**Value**
Material	*PZT*
Dimensions	*80x80x1 mm*
Strain/charge constant	$d_{31} = 400 \times 10^{-12}$ *m/V* $d_{32} \approx d_{31}$
Capacitance	$C_{pe} = 540 \times 10^{-9}$ *F*

SYSTEM DESCRIPTION



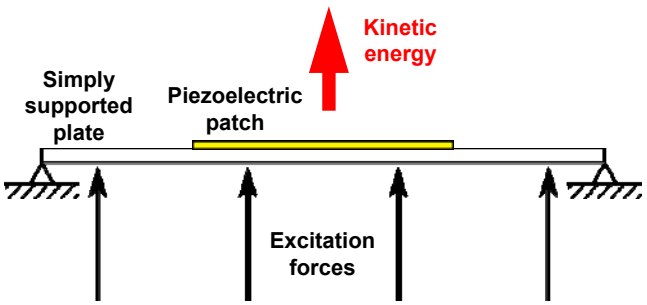
Rain-on-the-roof excitation



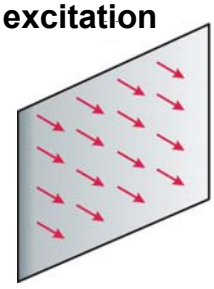
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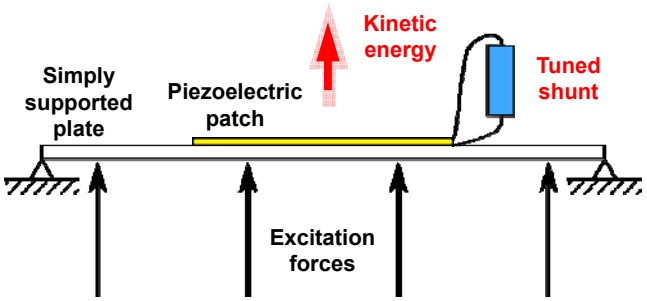
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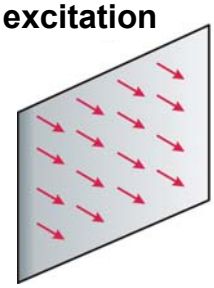
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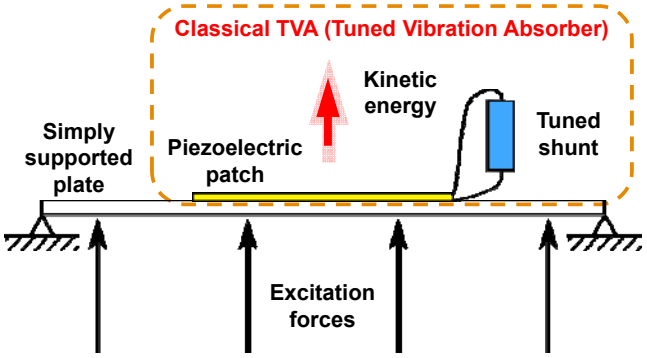
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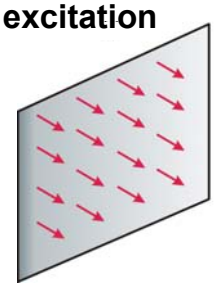
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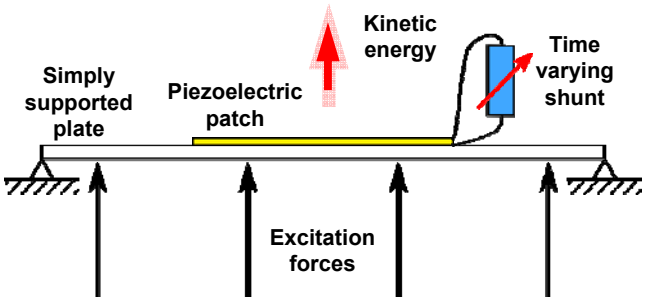
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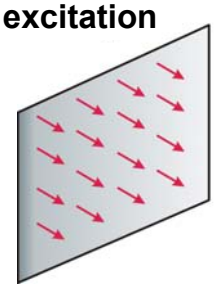
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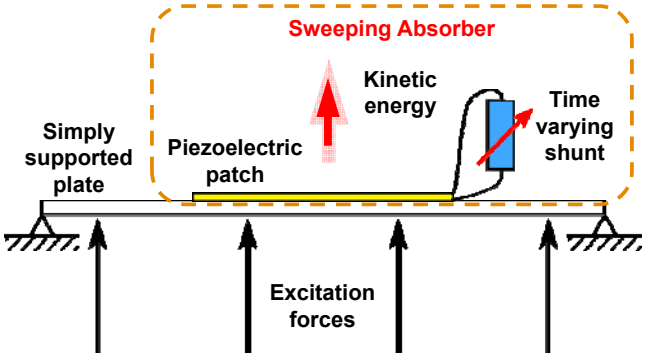
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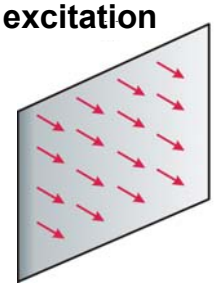
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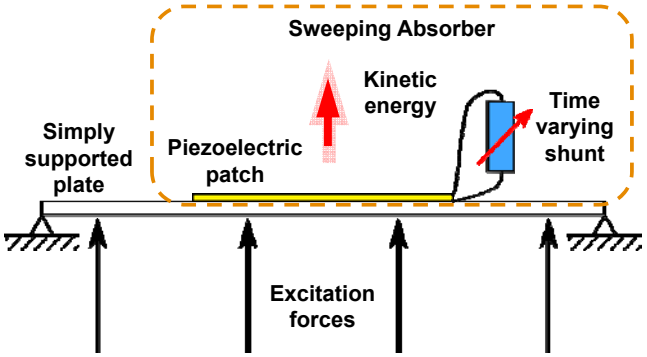
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Damping ratio	*2%*

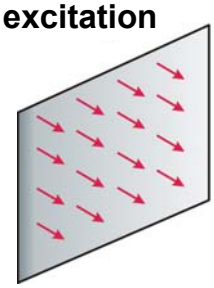
Parameters of the patch	
Parameter	**Value**
Material	*PZT*
Dimensions	*80x80x1 mm*
Strain/charge constant	$d_{31} = 400 \times 10^{-12}$ *m/V* $d_{32} \approx d_{31}$
Capacitance	$C_{pe} = 540 \times 10^{-9}$ *F*

SYSTEM DESCRIPTION

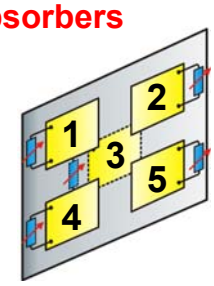


Parameters of the plate	
Parameter	**Value**
Material	*Aluminium*
Dimensions	*414x314x1 mm*
Damping ratio	*2%*

Rain-on-the-roof excitation



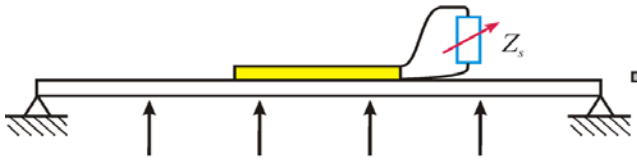
Array of absorbers



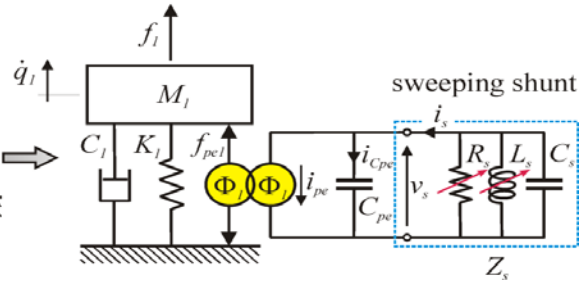
Parameters of the patches	
Parameter	**Value**
Material	*PZT*
Dimensions	*80x80x1 mm*
Strain/charge constant	$d_{31} = 400\ x10^{-12}\ m/V$ $d_{32} \approx d_{31}$
Capacitance	$C_{pe} = 540\ x10^{-9}\ F$

SINGLE MODE MODEL

Cross-section of the panel



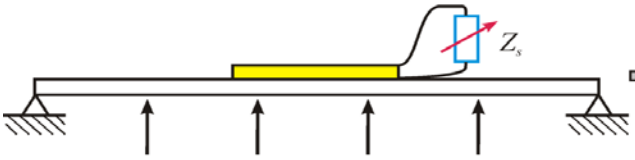
Electromechanical model



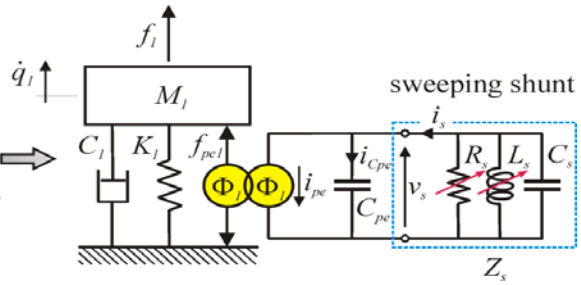
Note: sub index 1 refers to the first flexural mode of the plate.

SINGLE MODE MODEL

Cross-section of the panel



Electromechanical model



Mechanical equilibrium eq. $\left(M_1 s + C_1 + \frac{K_1}{s}\right)\dot{Q}_1 = F_1 + \Phi_1 V_s$,

Where:

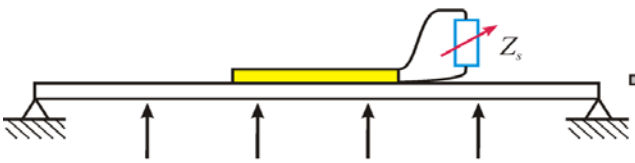
$\dot{Q}_1(s)$ is the flexural response of the panel,

Φ_1 is the EM transduction coefficient

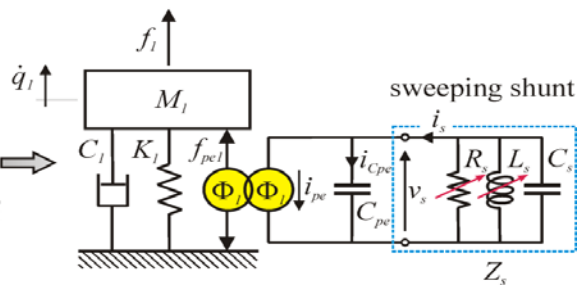
Note: sub index 1 refers to the first flexural mode of the plate.

SINGLE MODE MODEL

Cross-section of the panel



Electromechanical model



Mechanical equilibrium eq. $\left(M_1 s + C_1 + \frac{K_1}{s}\right)\dot{Q}_1 = F_1 + \Phi_1 V_s$,

Transducer electrical eq. $I_s = sC_{pe}\left(1 - k_{pe}^2\right)V_s(s) + \Phi_1 \dot{Q}_1(s)$,

Where:

$\dot{Q}_1(s)$ is the flexural response of the panel,

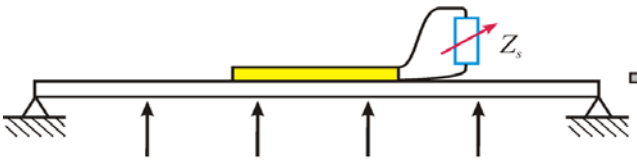
Φ_1 is the EM transduction coefficient, and

k_{pe}^2 the EM coupling factor.

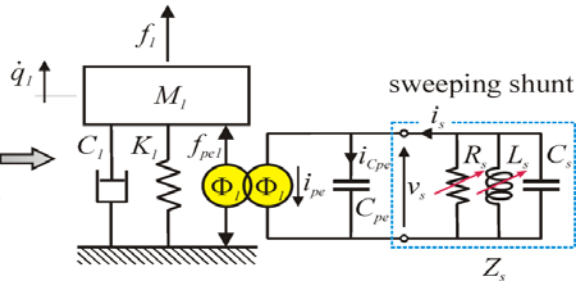
Note: sub index 1 refers to the first flexural mode of the plate.

SINGLE MODE MODEL

Cross-section of the panel



Electromechanical model



Mechanical equilibrium eq. $\left(M_1 s + C_1 + \frac{K_1}{s}\right)\dot{Q}_1 = F_1 + \Phi_1 V_s$,

Transducer electrical eq. $I_s = sC_{pe}\left(1 - k_{pe}^2\right)V_s(s) + \Phi_1 \dot{Q}_1(s)$,

Shunt electrical eq. $V_s = -\frac{1}{\frac{1}{R_s} + C_s s + \frac{1}{L_s s}} I_s$,

Note: sub index 1 refers to the first flexural mode of the plate.

Where:

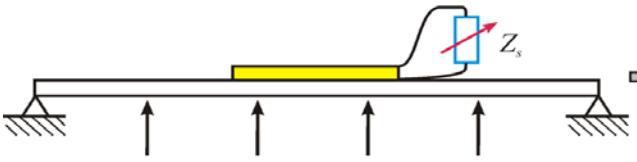
$\dot{Q}_1(s)$ is the flexural response of the panel,

Φ_1 is the EM transduction coefficient, and

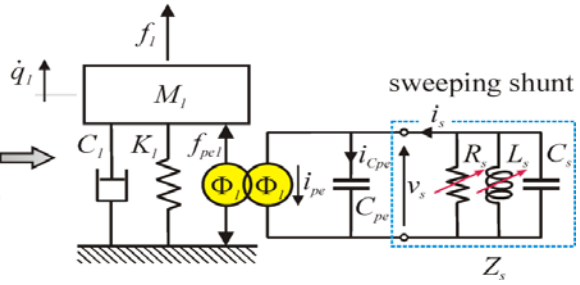
k_{pe}^2 the EM coupling factor.

SINGLE MODE MODEL

Cross-section of the panel



Electromechanical model



Z_m

Mechanical equilibrium eq. $\left(M_1 s + C_1 + \frac{K_1}{s}\right)\dot{Q}_1 = F_1 + \Phi_1 V_s$,

Transducer electrical eq. $I_s = sC_{pe}\left(1 - k_{pe}^2\right)V_s(s) + \Phi_1 \dot{Q}_1(s)$,

Shunt electrical eq. $V_s = -\frac{1}{\frac{1}{R_s} + C_s s + \frac{1}{L_s s}} I_s$,

Note: sub index 1 refers to the first flexural mode of the plate.

Where:

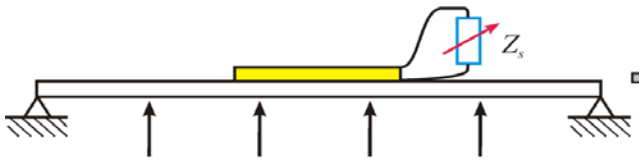
$\dot{Q}_1(s)$ is the flexural response of the panel,

Φ_1 is the EM transduction coefficient, and

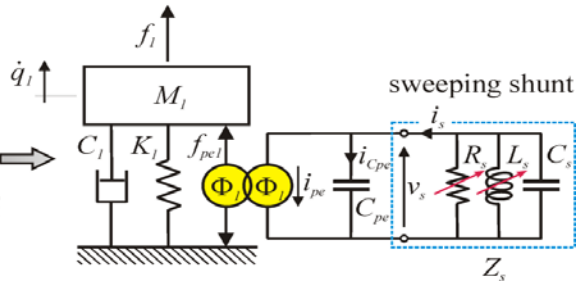
k_{pe}^2 the EM coupling factor.

SINGLE MODE MODEL

Cross-section of the panel



Electromechanical model



Z_m

Mechanical equilibrium eq. $\left(M_1 s + C_1 + \frac{K_1}{s}\right)\dot{Q}_1 = F_1 + \Phi_1 V_s$,

Transducer electrical eq. $I_s = sC_{pe}\left(1-k_{pe}^2\right)V_s(s) + \Phi_1 \dot{Q}_1(s)$,

Shunt electrical eq. $V_s = -\frac{1}{\frac{1}{R_s} + C_s s + \frac{1}{L_s s}} I_s$,

Z_s

Note: sub index 1 refers to the first flexural mode of the plate.

Where:

$\dot{Q}_1(s)$ is the flexural response of the panel,

Φ_1 is the EM transduction coefficient, and

k_{pe}^2 the EM coupling factor.

SINGLE MODE MODEL

Modal response of the plate with the shunt absorber

$$H(\omega) = \frac{\dot{Q}_1}{F_1} = \frac{1}{Z_m + Z_s \dfrac{\Phi_1^2}{1 + sC_{pe}\left(1-k_{pe}^2\right)Z_s}}$$

SHUNT TUNING LAWS

Modal response of the plate with the shunt absorber

$$H(\omega) = \frac{\dot{Q}_1}{F_1} = \frac{1}{Z_m + Z_s \dfrac{\Phi_1^2}{1 + sC_{pe}\left(1 - k_{pe}^2\right)Z_s}}$$

1. $C_1 = 0$

1. Neglecting structural damping



SHUNT TUNING LAWS

Modal response of the plate with the shunt absorber

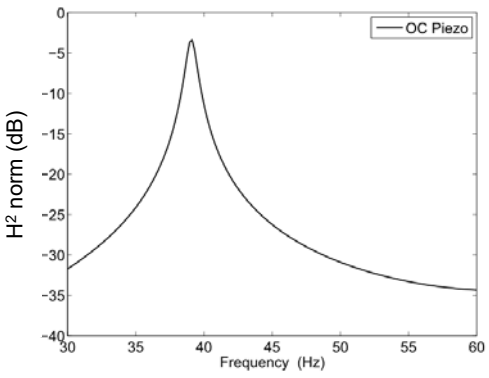
$$H(\omega) = \frac{\dot{Q}_1}{F_1} = \frac{1}{Z_m + Z_s \dfrac{\Phi_1^2}{1 + sC_{pe}\left(1 - k_{pe}^2\right)Z_s}}$$

1. $C_1 = 0$
2. $\min\left(\int_{-\infty}^{\infty} |H(\omega)|^2 d\omega\right)$

1. Neglecting structural damping



2. Minimizing the H^2 norm



SHUNT TUNING LAWS

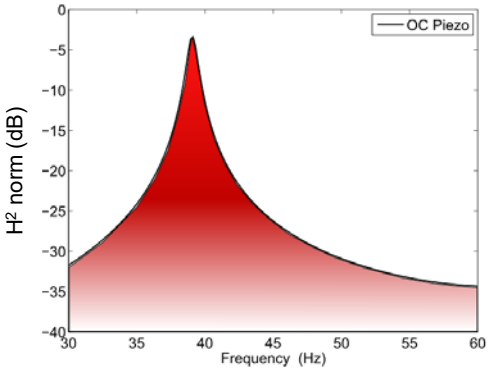
Modal response of the plate with the shunt absorber

$$H(\omega) = \frac{\dot{Q}_1}{F_1} = \frac{1}{Z_m + Z_s \dfrac{\Phi_1^2}{1 + sC_{pe}\left(1 - k_{pe}^2\right)Z_s}}$$

1. $C_1 = 0$
2. $\min\left(\int_{-\infty}^{\infty} |H(\omega)|^2 d\omega\right)$

1. Neglecting structural damping

2. Minimizing the H^2 norm



SHUNT TUNING LAWS

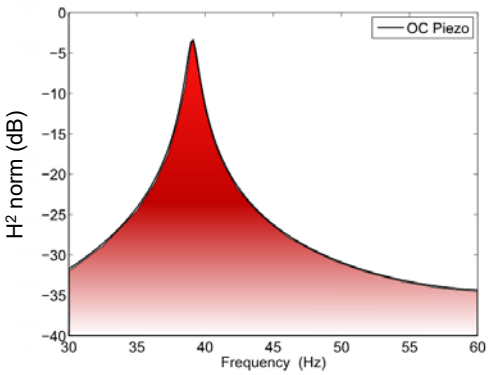
Modal response of the plate with the shunt absorber

$$H(\omega) = \frac{\dot{Q}_1}{F_1} = \frac{1}{Z_m + Z_s \dfrac{\Phi_1^2}{1 + sC_{pe}\left(1 - k_{pe}^2\right)Z_s}}$$

1. $C_1 = 0$
2. $\min\left(\int_{-\infty}^{\infty} |H(\omega)|^2 d\omega\right)$

1. Neglecting structural damping

2. Minimizing the H^2 norm

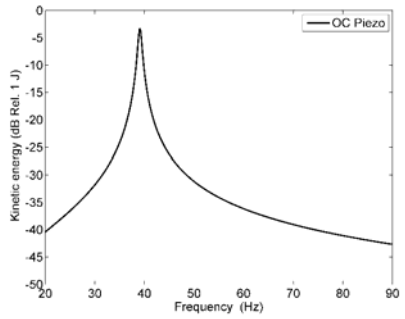


Optimal tuning values

$$L_{s1} = \frac{1}{\omega_1^2 C_{pe}(1 - k_{pe}^2)}$$

$$R_{s1} = \sqrt{\frac{M_1}{\Phi_1^2 C_{pe}(1 - k_{pe}^2)}}$$

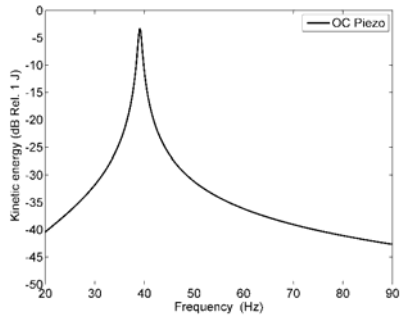
EFFECT OF DIFFERENT SHUNTS

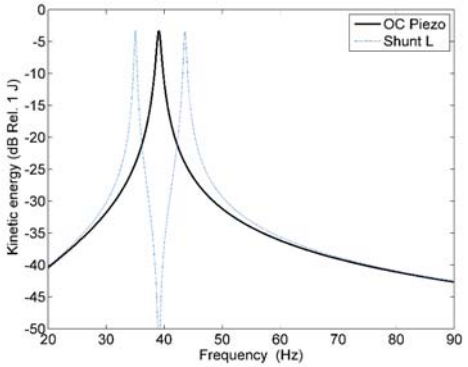


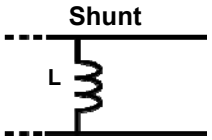
System FRF – 1st mode

EFFECT OF DIFFERENT SHUNTS

Shunt L



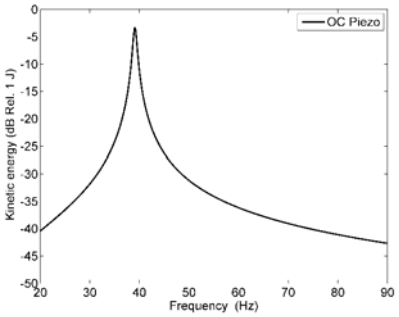




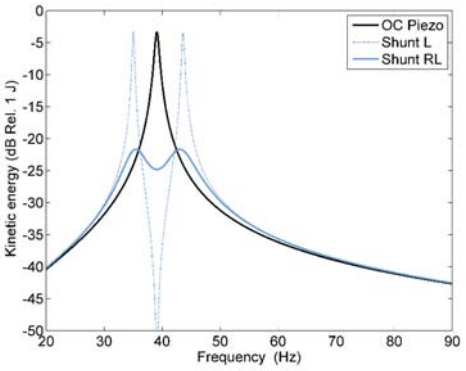
System FRF – 1st mode

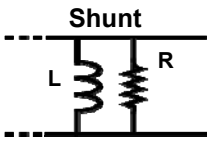
EFFECT OF DIFFERENT SHUNTS

Shunt RL



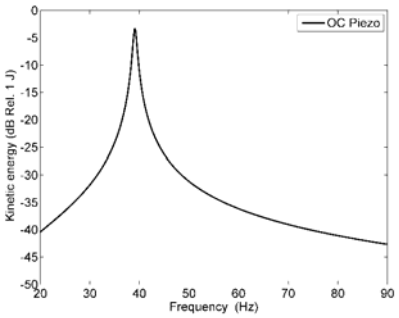
System FRF – 1st mode



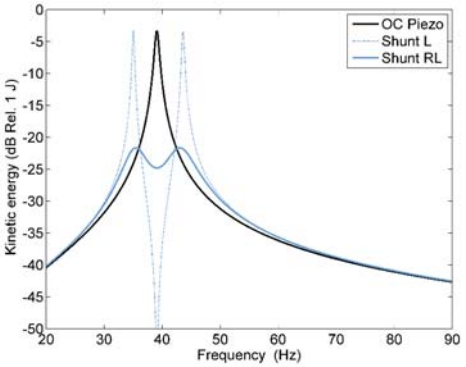


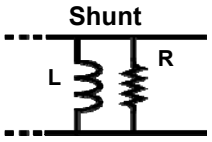
EFFECT OF DIFFERENT SHUNTS

Shunt RL

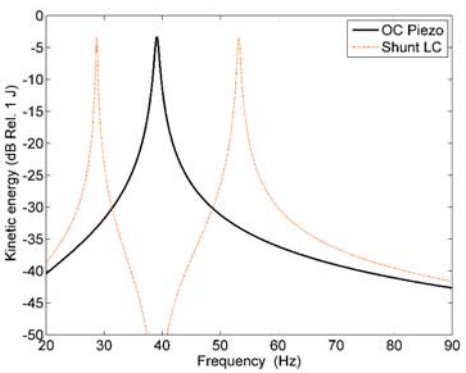


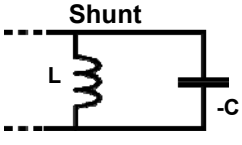
System FRF – 1st mode



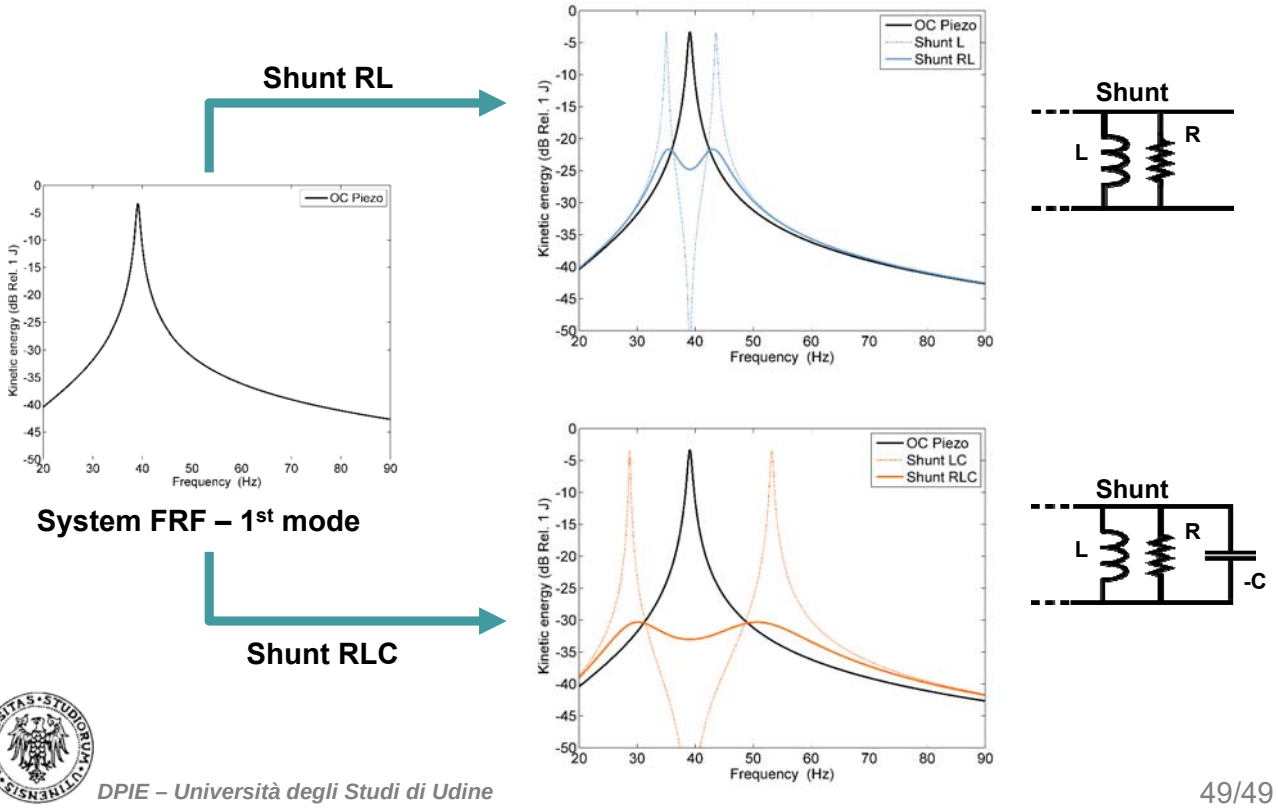


Shunt LC

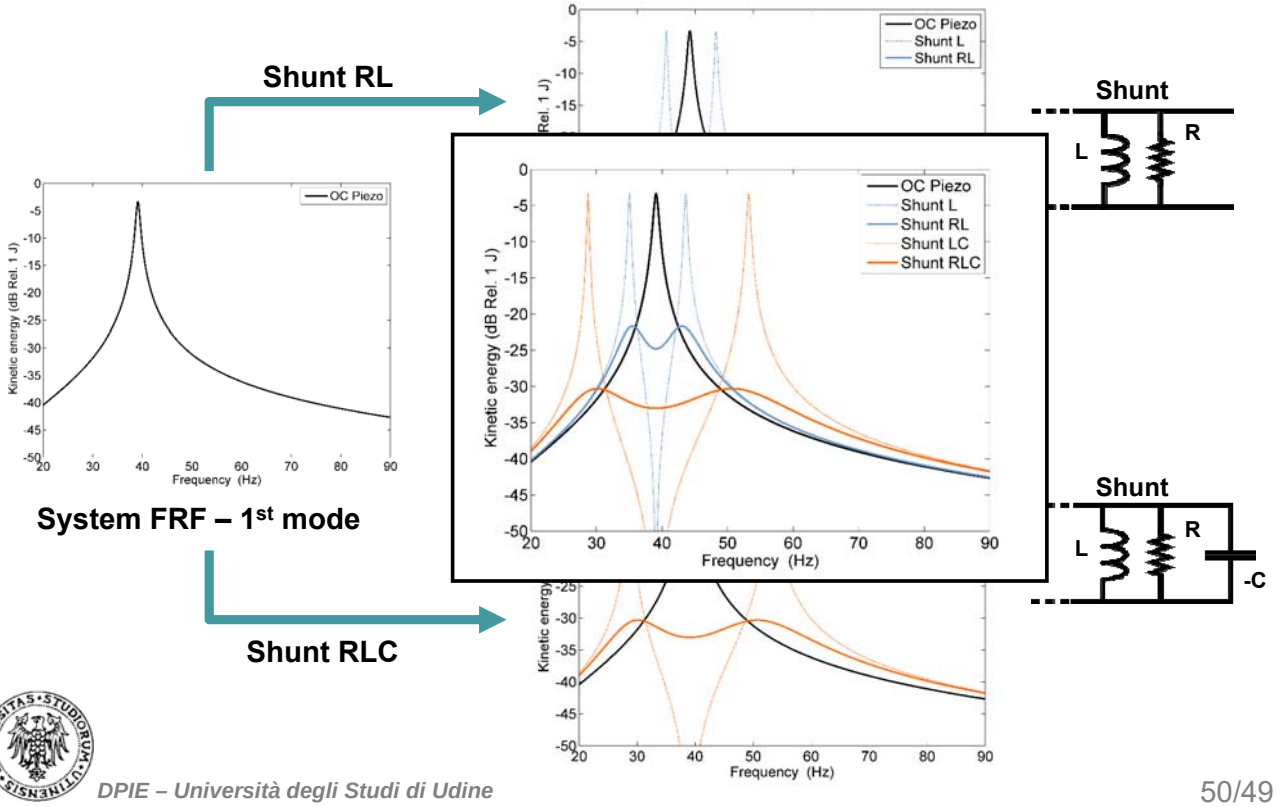




EFFECT OF DIFFERENT SHUNTS



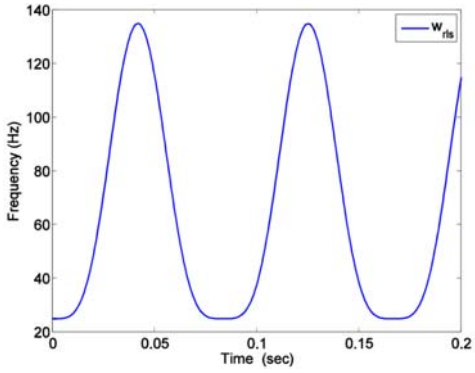
EFFECT OF DIFFERENT SHUNTS



OPERATING PRINCIPLE

Natural frequency of the shunt

$$\omega_{RLS} = \omega_o + (\omega_f - \omega_o)\sin^4\left(\frac{2\pi t}{T}\right)$$

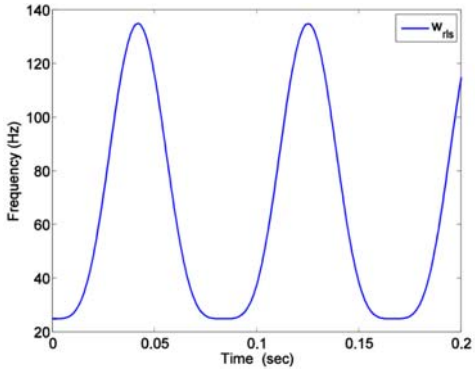


OPERATING PRINCIPLE

Natural frequency of the shunt

$$\omega_{RLS} = \omega_o + (\omega_f - \omega_o)\sin^4\left(\frac{2\pi t}{T}\right)$$

$$L_s = \frac{1}{\omega_{RLS}^2 C_{pe}(1-k_{pe}^2)}$$



OPERATING PRINCIPLE

Natural frequency of the shunt

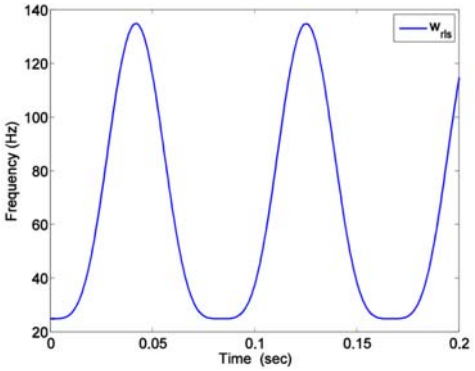
$$\omega_{RLS} = \omega_o + (\omega_f - \omega_o)\sin^4\left(\frac{2\pi t}{T}\right)$$

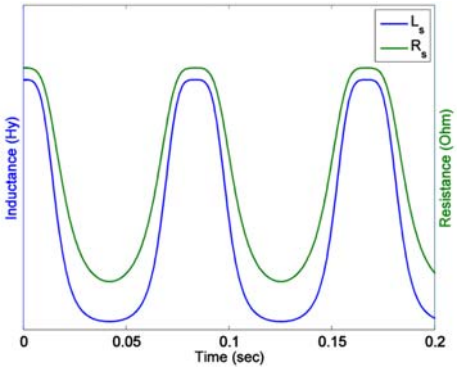
$$L_s = \frac{1}{\omega_{RLS}^2 C_{pe}(1-k_{pe}^2)}$$

Fixing the damping ratio

$$\zeta_{RLS} = 0.3$$

$$R_s = \frac{1}{2\zeta}\sqrt{\frac{L_s}{C_{pe}(1-k_{pe}^2)}}$$

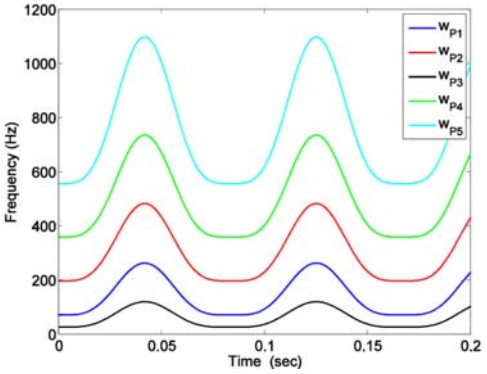




FIVE ABSORBERS OPERATION MODES

Series mode

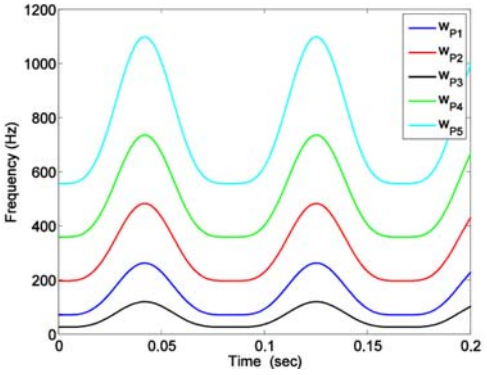
$$\omega^i_{RLS_{Series}} = \omega^i_o + (\omega^i_f - \omega^i_o)\sin^4\left(\frac{2\pi t}{T}\right)$$



FIVE ABSORBERS OPERATION MODES

Series mode

$$\omega^i_{RLS_{Series}} = \omega^i_o + (\omega^i_f - \omega^i_o)\sin^4\left(\frac{2\pi t}{T}\right)$$



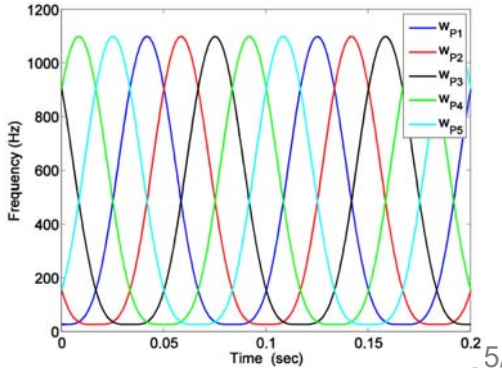
Asynchronous mode

$$\omega^i_{RLS_{Async.}} = \omega_o + (\omega_f - \omega_o)\sin^4\left(\frac{2\pi t}{T} - (i-1)\frac{\pi}{5}\right)$$

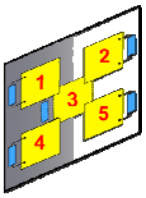
With:

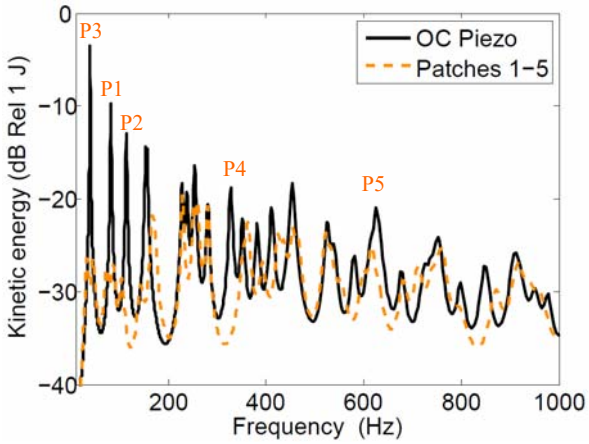
$$\omega_o = 30\ Hz$$

$$\omega_f = 1000\ Hz$$



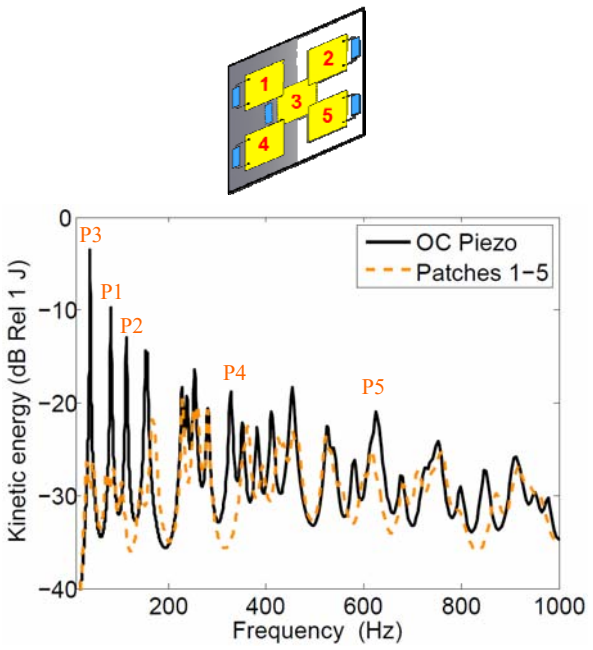
Multiple shunted absorbers comparison



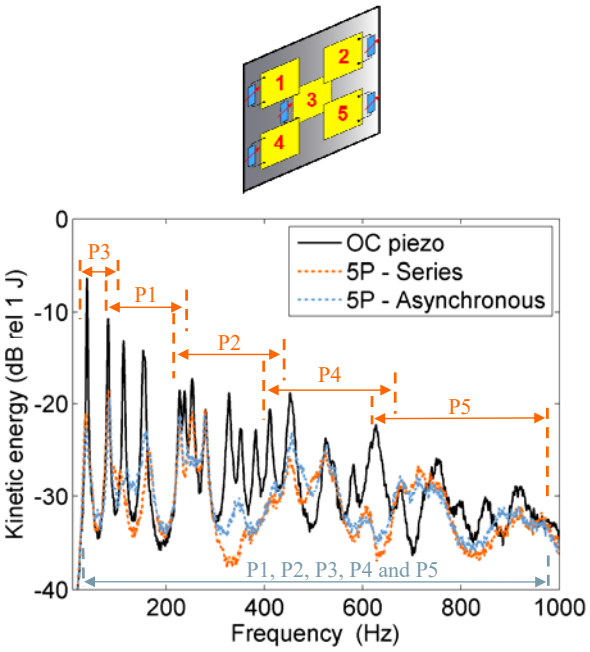


FRF of the system with five classical shunted absorbers

Multiple shunted absorbers comparison



FRF of the system with five classical shunted absorbers



FRF of the system with five sweeping shunted absorbers

Concluding Remarks

The proposed sweeping absorbers:

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Kinetic energy reduction	*7 to 18 dB*	*6 to 15 dB*	*5 to 14 dB*	*3 to 6 dB*

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The proposed sweeping absorbers:

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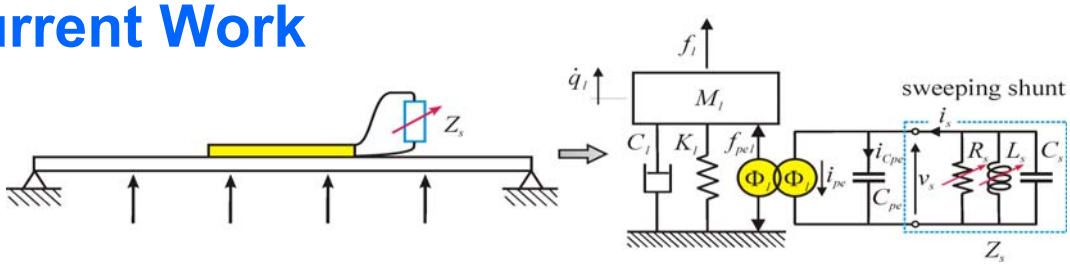
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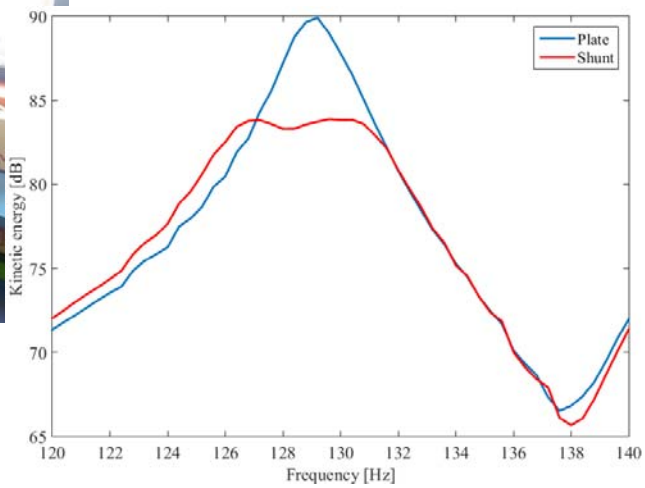
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Future work: experimental implementation of the TVA and STVA using piezoelectric transducers.

Current Work







BIBLIOGRAPHY

M. Zilletti, S.J. Elliott, E. Rustighi, Optimisation of Dynamic Vibration Absorbers to Minimise Kinetic Energy and Maximise Internal Power Dissipation. *Journal of Sound and Vibration*, 2012, 331(18): p. 4093-4100

P. Gardonio, M. Zilletti, Integrated tuned vibration absorbers: a theoretical study. *Journal of the Acoustical Society of America*, 2013, 2013, 134(5): p. 3631–3644

M. Zilletti, P. Gardonio, Experimental implementation of switching and sweeping tuneable vibration absorbers for broadband vibration control. *Journal of Sound and Vibration*, 2015, 334: p. 164-177

P. Gardonio, M. Zilletti. Sweeping tunable vibration absorbers for low and mid frequencies broad band vibration control. *Journal of Sound and Vibration*, 2015, 354.

D. Casagrande, P. Gardonio, M. Zilletti. Sweeping piezoelectric patch vibration absorbers. *MoViC–RASD Thirteen International Conference on Motion and Vibration Control and Twelve International Conference on Recent Advances in Structural Dynamics*, Southampton, UK, 3-6 July 2016.

E. Turco, P. Gardonio, L. Dal Bo. Time-varying shunted electro-magnetic tuneable vibration absorber. *MoViC–RASD Thirteen International Conference on Motion and Vibration Control and Twelveth International Conference on Recent Advances in Structural Dynamics*, Southampton, UK, 3-6 July 2016. pp 1–16.