ON THE OPTIMIZATION OF (GENERALIZED) IMPEDANCE FOR ACOUSTIC LINERS

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Abstract

The acoustic liner is an important technology to reduce noise transmission in ducts, for applications ranging from heating and air-conditioning ventilation systems, to aircraft turbofan engines. In particular, the new generation of Ultra-High-By-Pass-Ratio turbofan engine while reducing fuel consumption, threatens higher noise levels at low frequencies because of its larger diameter, lower number of blades and rotational speed. Moreover, this is accompanied by a shorter nacelle, leaving less available space for acoustic treatments. This need of higher performances, has encouraged research in the field of innovative acoustic liners. In this paper, we review some guidelines generally employed for the design of acoustic liners, along with their substantial limitations. Then, the state of art of the research in innovative liners is presented, including both passive and active solutions. This work defines the background for our following quest towards the employment of reinforcement-learning algorithms for the optimization of the liner technology.

Keywords: Acoustic Liners, Impedance Optimization, Noise Reduction

1. Introduction

1.1 The Theoretical Problem Of Boundary Treatment For Noise Mitigation

First of all, it is important to resume the mathematical problem of noise transmission attenuation. In general, the wave control by treating the boundaries of propagative domains is a large area of research encompassing all fields from electromagnetics to solid mechanics and acoustics. In this work we are dedicated to acoustics, but some concepts could be applied to other fields as long as the interdisciplinary analogies are properly defined. In acoustics a typical boundary treatment problem is the room modes damping, where the objective is to damp the acoustic modes in an enclosed cavity. P. Morse [1], in 1939, provided analytical solutions for the modes in an enclosed cavity with walls uniformly treated with materials characterized by the so-called *normal surface impedance*. Morse affirms that such quantity univocally identifies the absorptive properties of a surface, as it "depends only on the material and not on the incident wave (except for the variation with frequency)" [1]. The surface normal acoustic impedance $Z(\omega)$, is a complex quantity defined as the ratio between the local acoustic pressure $p(\omega)$ and the local normal acoustic velocity $v_n(\omega)$ in the frequency domain (ω being the angular frequency). Morse was analysing so-called locally-reacting surfaces, whose reaction depends only on the local sound pressure, and can therefore be characterized by the surface normal impedance. But Morse identified the concept of locally reacting wall as a degeneration of a

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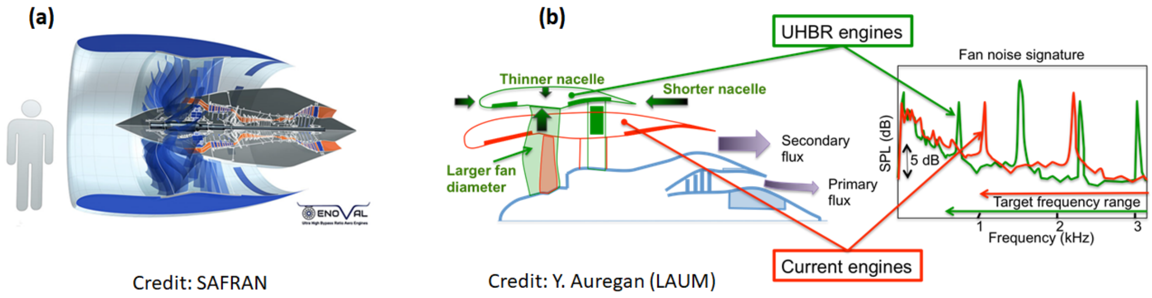

Figure 1 – Size of the UHBR turbofan compared to an average man in **(a)**, and nacelle length evolution along with the noise signature shift in **(b)**.

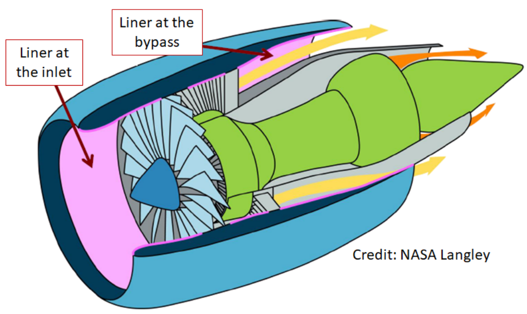

Figure 2 – Sketch of the interior of a turbofan engine with inlet and bypass liners in pink.

more general case, where propagation could happen inside the wall, according to an anisotropic wave propagation equation with normal and tangential phase speeds (or refractive indexes) different from each other. By reducing the tangential phase speed to zero, the propagation inside the wall would produce a locally-reacting behaviour at the wall interface with air, represented by the normal surface impedance. A non-zero tangential phase speed "inside the wall", characterizes the so-called *non-locally reactive* surfaces, where the wall reaction does not depend only upon the local pressure, but also upon its spatial distribution. Typical non-locally reacting surfaces are indeed the interfaces with porous materials, which can be modelled as propagative homogeneous media in the equivalent fluid formulations [2].

Another typical boundary treatment problem in acoustics is the noise transmission mitigation in an open duct, by treating its parietal walls with the so-called liners. Examples of industrial fields where this problem is particularly felt are the Heating and Ventilation Air-Conditioning Systems (HVAC) and the aircraft turbofan engines. The latter field in particular, on the one hand has to face significant restrictions on fuel consumptions and pollutant emissions, and on the other hand is tightened by the increasingly restrictive regulations on noise pollution. These two constraints are unfortunately in conflict with each other. As the fuel restrictions demand larger turbofan diameters, less number of blades and higher by-pass-ratios, the engine noise signature is shifted toward lower frequencies, which are much more challenging to mitigate by parietal treatments than high frequencies. It is the case for the Ultra-High-By-Pass-Ratio turbofans, depicted in Fig. 1. Preliminary studies of an acoustic waveguide without flow are common practice [3] as liners are usually meant to modify the acoustic field without significantly impacting on the aerodynamic one. Such hypothesis is nowadays very much debated [4], and in any case theoretical and experimental testing with flow constitutes an important successive step for the industrial implementation of any liner technology.

The greatest difficulty for the parietal treatment of a waveguide is, antonomastically, that it applies on the parietal walls $\partial\mathscr{A}$ (see Fig. 3), whereas the noise propagates along the longitudinal axis x which is clearly parallel to $\partial\mathscr{A}$. Such problem is usually referred to as *grazing incidence* problem. It is very well known that if a plane wave field is incident on an infinite surface with an elevation angle $\bar{\theta} \in (0, \pi)$,

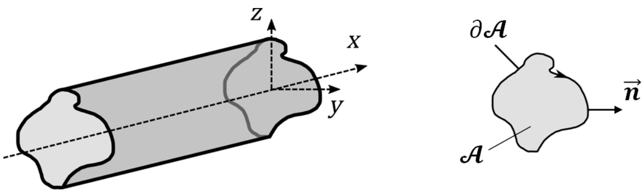

Figure 3 – A cylindrical waveguide along coordinate x, with cross section of arbitrary shape $\mathscr{A}$. Left: overview of the waveguide. Right: detail of the cross-section and its contour $\partial\mathscr{A}$. $\vec{n}$ is the local exterior normal at each point of the contour.

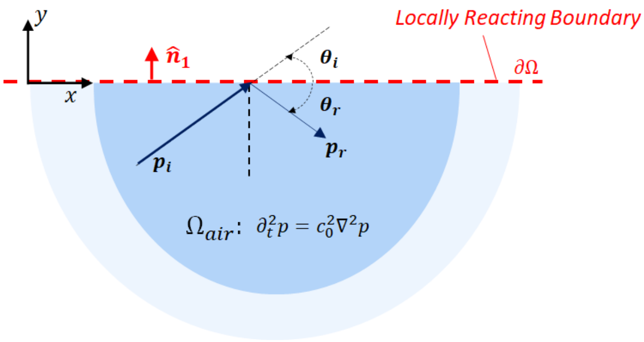

Figure 4 – Semi-infinite acoustic domain Ω_{air}, expanding indefinitely along $\pm x$ and $-y$, bounded on $y = 0$ by a locally-reacting boundary $\partial\Omega$, and definition of the incidence and reflection angles according to the convention adopted in this text.

where $\bar{\theta}$ is taken from the tangential boundary coordinate (see Figure 4), then perfect absorption is achieved when the surface acoustic impedance is $Z = \rho_0 c_0/\sin\bar{\theta}$, where ρ_0 and c_0 are the air density and sound speed in the steady state. For $Z = \rho_0 c_0/\sin\bar{\theta}$, a plane wave propagating at a certain angle θ_i, *sees* a frontal impedance $Z_f = \rho_0 c_0 sin\bar{\theta}/\sin\theta_i$ along its propagation direction, when it impacts the boundary. Therefore, if $\theta_i = \pm\bar{\theta}$ the frontal impedance encountered by the wave is equal to the characteristic impedance of air $\rho_0 c_0$, and no reflection occurs.
The problem arises when $\theta_i = 0$ or π. C. Bardos, G. Lebeau and J. Rauch [5] demonstrated that a sufficient condition for the boundary to fully control the wave propagation is that every ray of the acoustic field must interact with the boundary. But in case of the *grazing incidence* problem, i.e. in case of an acoustic field propagating in a waveguide, there will always be some rays not directly interacting with the boundary, therefore not *controllable*. This is also the reason why the effectiveness of any liner in noise transmission attenuation, degrades if the cross-section area of the waveguide increases, as less number of acoustic rays will directly interact with the boundary.

Nevertheless, even if the "grazing incidence problem" is not fully controllable, it should still be possible to determine an optimal liner behaviour which achieves the maximum noise transmission attenuation. The first work in this sense was done by L. Cremer [6] who searched for the optimal *local impedance* $Z_{cre}(\omega)$ producing the highest attenuation rate of the *least-attenuated* duct mode in a 2D infinite acoustic waveguide lined on one side, see Fig. 5, without mean flow. His work is indeed rooted in the so-called duct-modes solution, i.e. the solution of the dispersion problem in terms of axial wavenumbers $k_{x,m}(\omega)$ and duct modes $\psi_m(y,z)$, corresponding to the eigenvalues and eigenfunctions of the transverse Laplace operator reduced to the waveguide cross section.

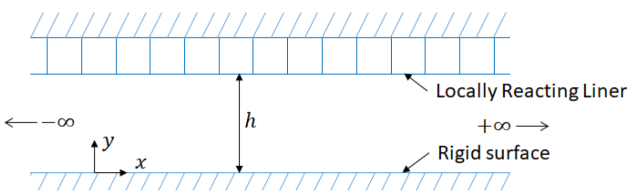

Figure 5 – Infinite 2D lined duct considered in Cremer's work [6].

Such optimal impedance is the one providing the coalescence of the first two duct modes, which means that for $Z(\omega) = Z_{cre}(\omega)$ the first and second wavenumber solutions of the dispersion problem merge. For $Z(\omega)$ slightly varied with respect to $Z_{cre}(\omega)$, either the first or the second duct mode presents an attenuation rate which is lower with respect to the case of $Z_{cre}(\omega)$. The duct modes are arranged in "first", "second", "third", etc. according to the number of node lines in the shape of $\psi_m(y,z)$, in ascending order. In the complex wavenumber space (Re$\{k_x\}$, Im$\{k_x\}$), this coalescence happens at the so-called *exceptional points*.

In an infinite waveguide, the transmission between two points along the longitudinal axis is fully determined by the attenuation rate (of each duct mode involved in the noise propagation) times the distance between the two points. As the first duct modes are the ones presenting the lowest attenuation rate $\mathrm{Im}\{k_x(\omega)\}$, by maximizing it then the transmission will be minimized.

B. Tester [3] reformulated the optimization problem based upon a non-conventional Green's function, allowing to slightly correct the Cremer value, and to determine the optimal impedances not only for the first couple of modes, but also for higher order couples of modes and for different geometries of the duct cross section. The tube was still considered as infinite and the optimal impedances were still producing the coalescence of the couple of modes to be mostly attenuated.

In [7], the optimal impedance based upon exceptional points, is found also for the spinning and radial modes, typical of cylindrical ducts. According to [7], shear flow, Mach number and boundary layer thickness strongly affect the optimal impedance values based upon exceptional points. Moreover, a slight variation of the impedance with respect to the optimal one, can significantly impact the attenuation performances.

Such impedances obtained from exceptional points, have all the same form: $Z_{opt}(\omega) = \rho_0 c_0 (a - \mathrm{j}b) k_0 h/\pi$, where a and b are constant positive real numbers, $k_0 = \omega/c_0$ is the wavenumber of the plane mode in hard-walled ducts and h is a characteristic dimension of the duct cross-section, normal to the liner. It is interesting to notice that while for free plane waves with an angle of incidence $\theta_i \in (0,\pi)$, the optimal impedance is purely resistive, for grazing incidence ($\theta_i = 0$ or π), $Z_{opt}(\omega)$ has a reactive component which has the same order of magnitude as the resistive one. Tester [8] provided a physical qualitative explanation for the specific value obtained for $Z_{opt}(\omega)$, by considering a line source applied along the centre longitudinal axis of the waveguide. He found that "the optimum impedance is roughly that value which allows the amplitude and phase of the singly-reflected field to cause the most efficient destructive interference with the direct field". It becomes then more understandable the importance of the phase change of the reflected field with respect to the incident one, produced by a complex impedance.

Another issue of such optimal complex impedance function $Z_{opt}(\omega)$ is that its inverse Fourier transform does not correspond to any real function in time domain, as it does not satisfy the so-called *reality condition* $Z^*(\omega) = Z(-\omega)$. This is so because the real part of $Z_{opt}(\omega)$ is not a pair function, which means that such optimal impedance is not physically realisable if not for single discrete frequencies $\omega = \omega_{target}$, but not on a continuous frequency span.

Almost contemporary to the work of Tester, are the contribution of Rice [9], who identified the cut-off ratio first, and the modal propagation angle then, as parameters which very well correlated with the optimal impedance. In particular, the modal propagation angle provided also a nice physical perspective upon the optimization guideline. As modal propagation angle Rice considers the angle between the modal wavenumber vector $\vec{k}_m$ and the longitudinal axis of the duct, at the boundary. Such angle defines the number of interaction of a plane wave with the duct lateral boundaries. Such plane wave approximate the duct-mode propagation at the boundaries, in a simplified geometric approach proposed in [10], and retrieved in [9]. In case of mean-flow, Rice suggests the employment of the group velocity, rather than the wavenumber vector, to identify the modal propagation angle.

Nevertheless, both the exceptional points and elevation angles approaches, do not consider the finiteness of the lined segment in the duct. Indeed, according to Tester [3], the exceptional point entails also a linear increase in the mode amplitude, which becomes negligible only for very long ducts. Moreover, the correlation between the dispersion solutions, classically employed for impedance optimization in liners, and the scattering performances (in terms of reflection, absorption and transmission) is not yet fully developed.

In the next sections, we go through various major liner concepts of the actual state of art in this field of research.

2. Passive Boundary treatments for noise mitigation

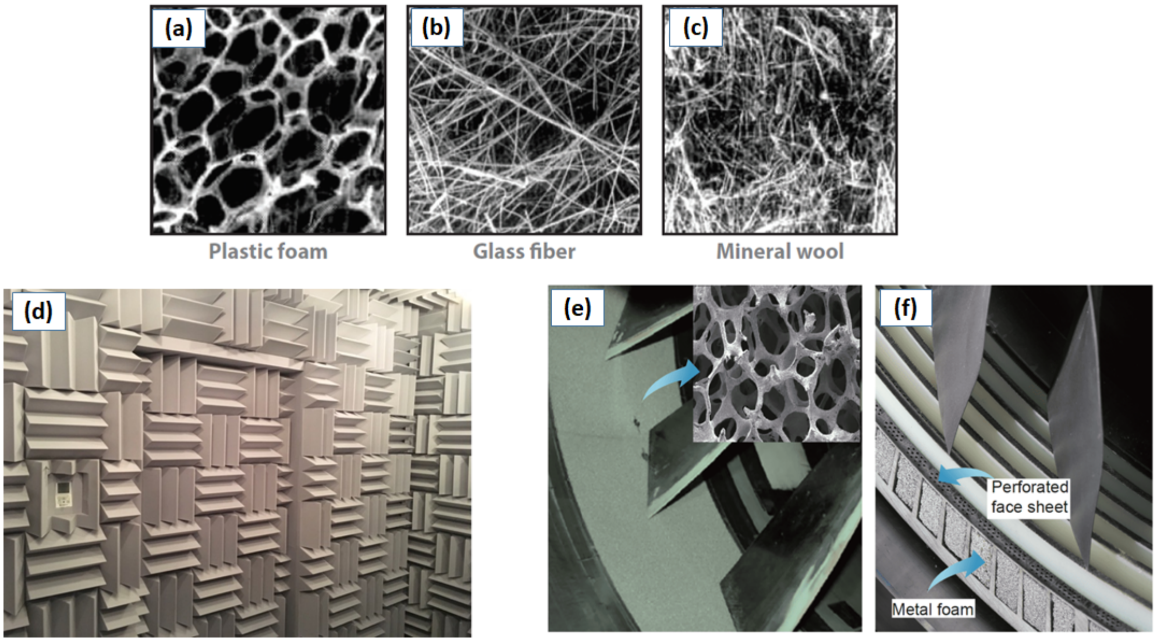

Figure 6 – Types of porous materials microstructures: plastic foam **(a)**, glass fiber **(b)** and mineral wool **(c)**, from [11]. Applications of porous media in room acoustics (absorbing wedges in an anechoic chamber **(d)**) and in turbofan over-the-rotor nacelle (liner of metal foam **(e)**, with groves **(f)** to keep the aerodynamic performances, from [12])

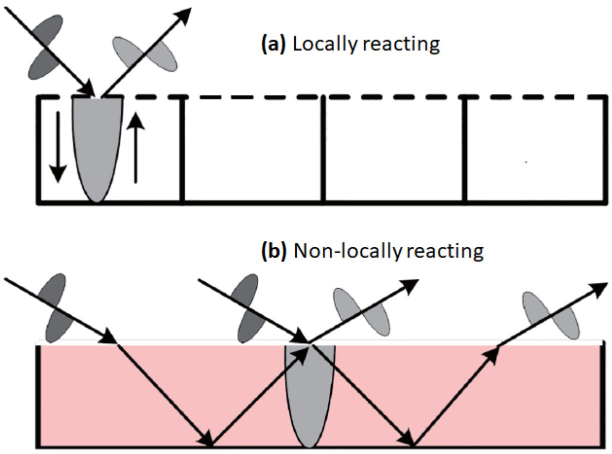

Figure 7 – Sketch of locally-reacting **(a)** and non-locally reacting **(b)** liner (from [12])

The problem of noise attenuation can be addressed by two main general philosophies: one is to trap or reflect the sound energy by means of some resonant element, and the other one is to dissipate it through viscous and heat conduction mechanisms at the solid-air interface. Clearly, the first idea does not exclude the second one, as dissipation inevitably occurs in passive systems, but, in resonant devices, dissipation is a secondary phenomenon respect to resonance itself. The two phenomena of sound attenuation have also been targeted at the same time in order to exploit each other advantages.

Porous materials are renowned for the significant sound absorption they can achieve especially at

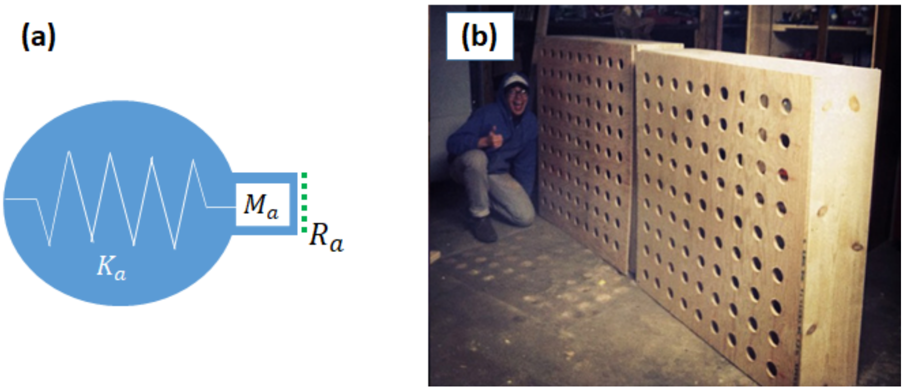

Figure 8 – Sketch of a Helmholtz resonator in **(a)** with the mechanical mass-spring-resistance in-series analogy, with the equivalent acoustical mass M_a (given by the air in the neck), equivalent acoustical spring K_a (the compressibility of the cavity) and resistance R_a (given by an additional perforated sheet place at the inlet, and/or by the visco-thermal exchanges in the narrow neck). Typical application of the Helmholtz resonator in room acoustics **(b)**, from www.andymacdoor.com.

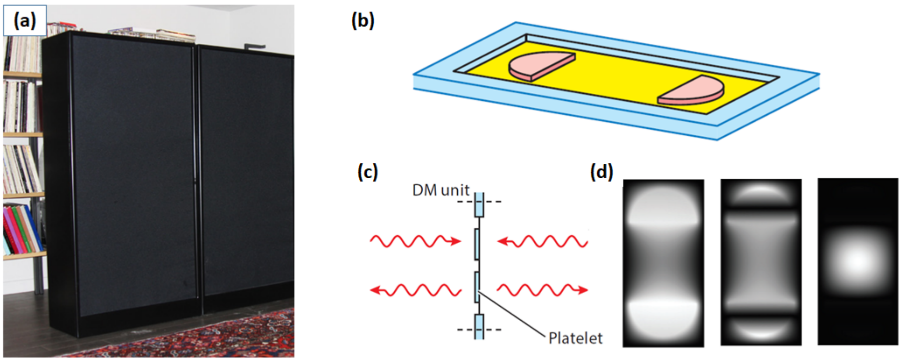

Figure 9 – Membrane bass-trap for room acoustics **(a)** from www.andymacdoor.com; sketches of the decorated membrane (DM) **(b)**, **(c)** with the additional masses (platelets), providing the multimodal behaviour **(d)**, from [11].

high frequencies. Thanks to the large air-solid interface area obtained by fibrous and permeable microstructures (see Figures 6a,b,c), the viscosity and heat conduction mechanisms produce large dissipation. As dissipative forces vary linearly with the rate (e.g. viscous acoustic damping force is proportional to velocity), it follows that sound dissipation is a quadratic function of frequency. For this reason, the porous materials tend to improve their sound absorption potential as frequency increases. Moreover, as the dissipative forces mainly depend upon the acoustic velocity, the absorption is maximum at a $\lambda/4$ (λ being the wavelength of sound waves) distance from a rigid back-wall (where the acoustic velocity becomes zero). This explains the very large space required by the so-called absorbing wedges, classically used in room acoustics applications (see Figure 6d). Rigid porous materials, such as metal foams, are currently under investigation for applications as acoustic liners also in a harsh environment as the over-the-rotor region of the nacelle (see Figure 6e,f). Because of high percentage of void into their structure, porous materials allow sound propagation into their volume in both normal and tangential directions with respect to the interface with air. For this reason, they are defined as *non-locally* or *bulk* reacting liners, as opposed to the locally-reacting ones (see Figure 7).
Another classical sound-proofing device is the perforated, and micro-perforated, plate. It consists of a thin, rigid (usually metallic) plate with straight holes, usually backed by a cavity in order to increase the wave amplitude at the perforated plate location through the constructing interference effect. The sound energy dissipation mechanisms through the holes are indeed the same as in porous materials.
The classical sound-proofing device based upon the resonance principle is the famous Helmholtz resonator which is composed of a thin neck connecting the interface with a back-cavity. The air in

the narrow neck moves as a whole, as if it were a mass, which is connected to a spring given by the back-cavity compressibility. Slight dissipative effects occurs in the neck. Another basic resonant absorber is the membrane, which is classically applied in room acoustics, where it takes the name of bass-trap (see Figure 9a).
All such resonant-based acoustic devices mainly suffer from the narrowness of the efficient absorption bandwidth.

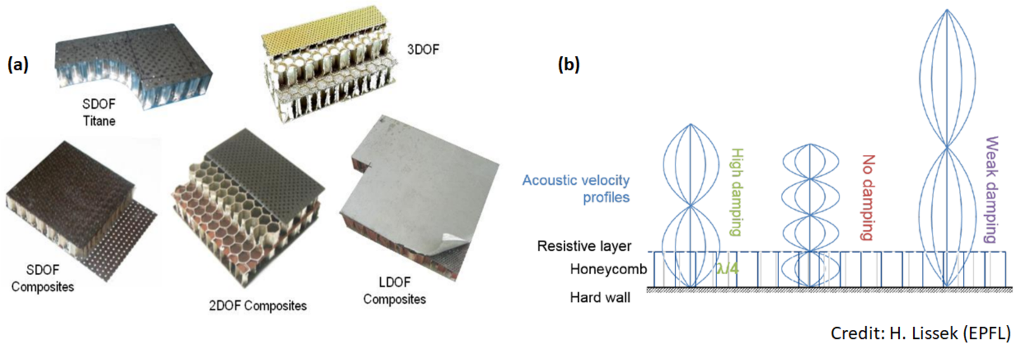

Figure 10 – Single and multi-degree-of-freedom liners **(a)**, whose functioning is based upon the quarter-wavelength principle **(b)**.

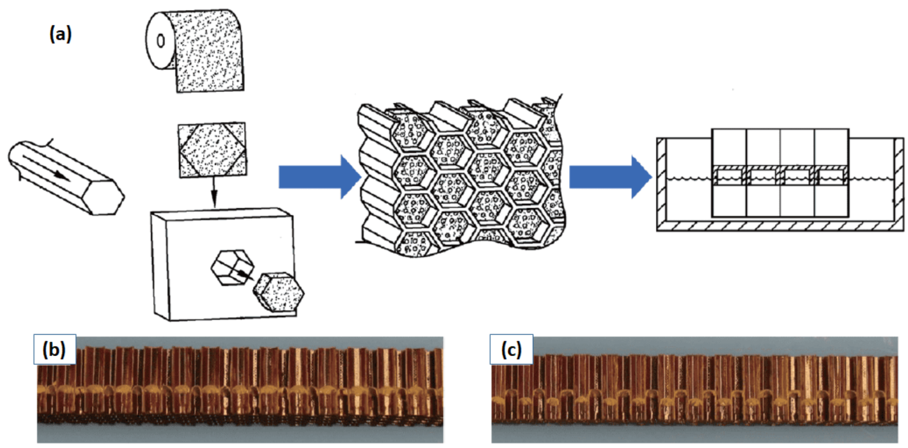

Figure 11 – Schematics of manufacturing procedures **(a)** and photographs of mesh-cap honeycomb with **(b)** uniform depth and **(c)** variable depth, from [12].

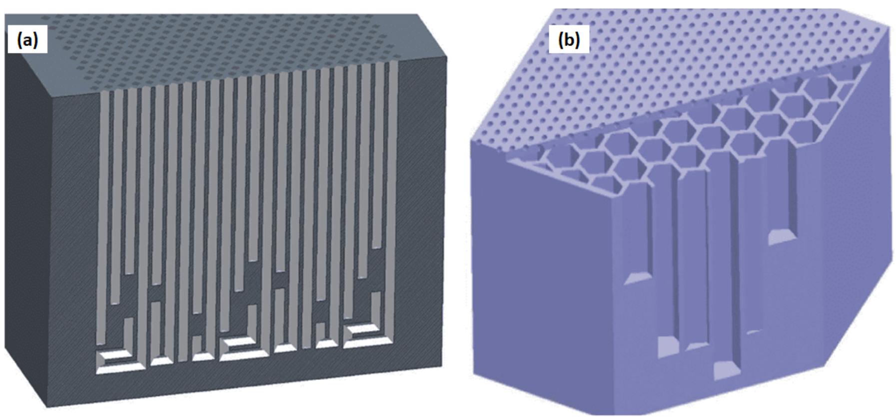

Figure 12 – Sketch of variable-depth liners with narrow **(a)** and wide chambers **(b)**, from [12].

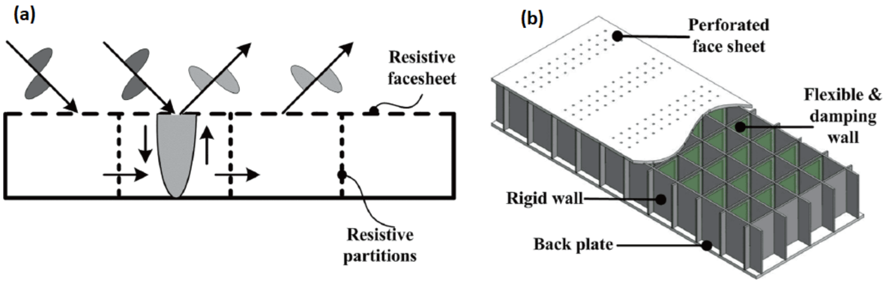

Figure 13 – Sketch of hybrid locally/non-locally reacting liners with perforated **(a)** and flexible **(b)** cavity walls, from [12].

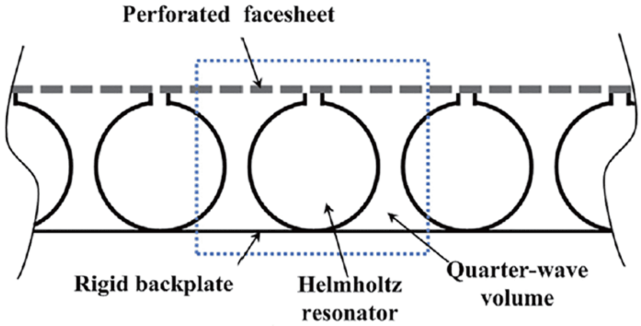

Figure 14 – Double resonant liner [13].

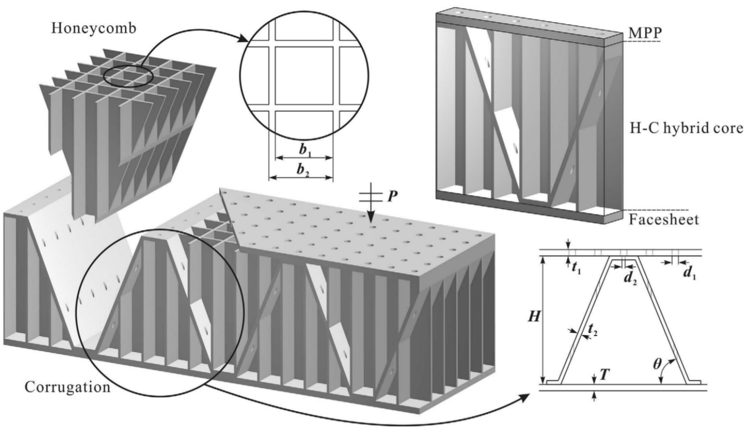

Figure 15 – Liner with inclined microporous septa in the honeycomb, producing the so-called honeycomb-corrugation hybrid structure, but with microperforations on both top facesheet and corrugations [14].

The traditional acoustic liner technology still applied nowadays for noise transmission attenuation at the inlet and outlet portions of turbofan engines is the so-called Single-Degree-of-Freedom (SDOF) liner. It is made of a closed honeycomb structure and a perforated plate which is used to provide the dissipative effect, to add mass in order to decrease the resonance frequency, and also to maintain the aerodynamic flow as smooth as possible on the internal wall of the nacelle. As the honeycomb structure is impervious, propagation is prevented transversely to the wall, therefore it can be considered as *locally reacting* as long as the incident field wavelength is much larger than the size of the honeycomb cells. The traditional SDOF liner is a quarter-wavelength resonator with a narrow absorption spectrum around $\lambda/4$, see Figure 10. Guess [15] has provided a method in three steps to evaluate the geometrical parameters of a SDOF liner, from a specified acoustic resistance and reactance, which can be tuned, for example, to the Cremer's values at a target frequency. The difficulty stays in the fact that the SDOF liner parameters affect both the resistance and reactance terms at the same time, making the tuning not so straightforward.

In order to broaden the noise absorption bandwidth, a septum is used to separate the honeycomb into two or more parts, forming the so-called double (DDOF) or multi-degree-of-freedom (MDOF) configurations. They provide on the one hand larger bandwidth, but on the other excessive weight and bulky structures. Other improvements of the SDOF liner technologies have been developed, such as the mesh-cap liner [12] where porous sheets were inserted into the honeycomb cavities, see Figure 11. Compared to the integral septum in traditional MDOF liners, the mesh caps were only anchored to the cavity walls. Different number and types of mesh caps (with various resistances) could placed at different depths, in order to adjust the liner impedance. Another variant of the SDOF liner is the variable depth liner, with either narrow or wide chambers [16], see Figure 12. In order to exploit the benefits of a non-local reaction, the hybrid acoustic liner was conceived, where the cavity walls of the SDOF liner were substituted by perforated panels (see Figure 13). Another proposed alternative was to use flexible walls as partitions in order to profit from the visco-elastic dissipation.

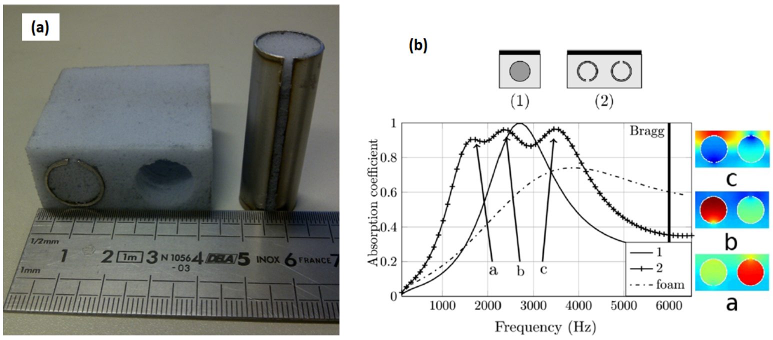

Figure 16 – Meta-porous with resonant-inclusions photo **(a)** and normal absorption performances **(b)** [17].

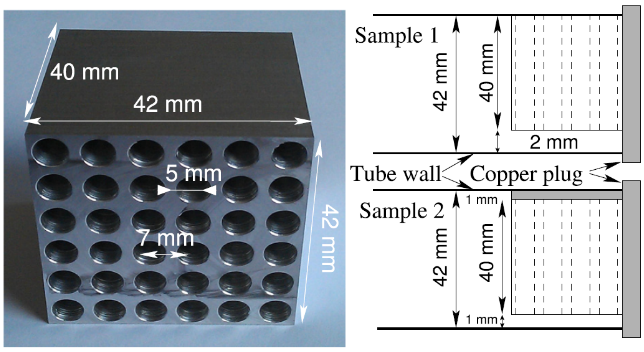

Figure 17 – Slow-sound absorber by periodical structure of narrow slits with quarter-wavelength inclusions [18].

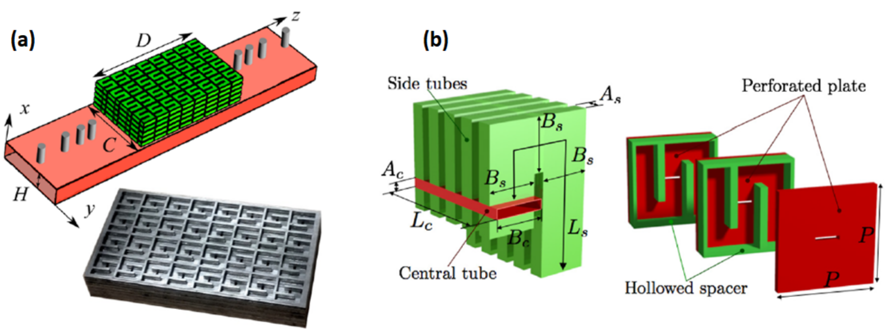

Figure 18 – Slow-sound liner **(a)** with folded branches obtained by hollowed plates **(b)**, [19].

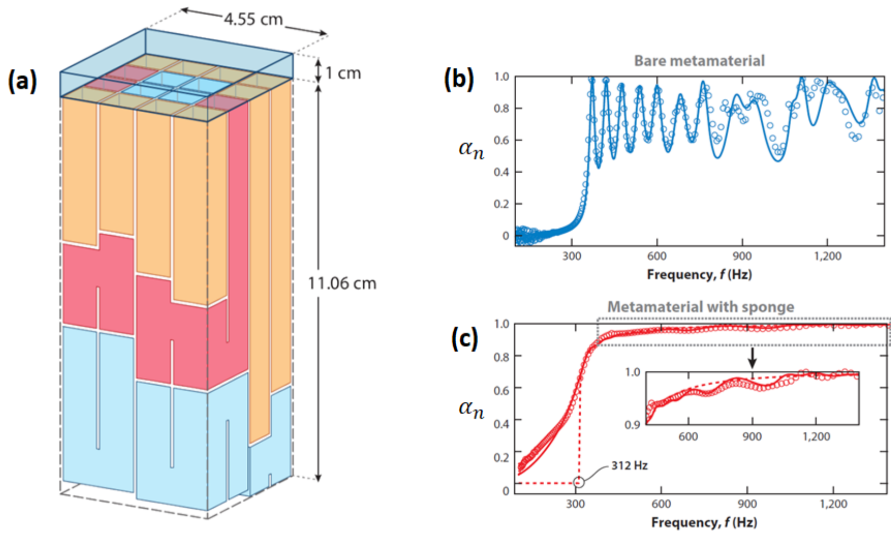

Figure 19 – Optimal absorber designed by [20], made of an array of channel-resonators **(a)**, and its normal absorption α_n performance without **(b)** and with **(c)** the addition of a sponge layer 1 cm thick.

Apart from the challenging environments that liners for turbofans applications must confront with, the main issue of classical boundary treatments, both for grazing incidence or room acoustics problems, are the low frequencies, especially among 100 and 1000 Hz, where the traditional soundproofing techniques described above would require excessive thickness. From here, it comes the great interest for acoustic metamaterials, and their quest toward *subwavelength* dimensions, i.e. to achieve picks of noise attenuation with thicknesses as much as possible below the quarter-wavelength. Most of such metamaterials combine together different traditional absorbers (such as porous materials, Helmholtz and/or quarter-wavelength resonators and membranes) in such a way to improve the overall performance. Beck et al. [13] has improved the SDOF traditional liner with the inclusion of a Helmholtz resonator in the honeycomb cavity, which produced an additional low-frequency resonance, see Figure 14. Tang et al. [14] have inserted inclined microporous septa in the honeycomb, producing the so-called honeycomb-corrugation hybrid structure, but with microperforations on both top facesheet and corrugations, see Figure 15. The asset of such metamaterial is its high mechanical stiffness and a broader low-frequency bandwidth respect to traditional quarter-wavelength absorbers.
The insertion of resonators inside porous materials has given rise to the so-called meta-porous absorbers [17] (see Figure 16). Another interesting perspective is the design of porous microstructures in order to achieve targeted acoustic performances [21], [22]. While very well suited for room acoustics applications though, porous non-rigid materials would hardly resist harsh environments such as the nacelle of turbofans.
The problem of lowering the frequency of maximum absorption (usually around $c_0/4d$ with d the thickness of the absorber) have also been tackled by targeting the reduction of the *effective sound speed* in the material. In traditional porous materials the sound speed is usually the same as in air, as it tends asymptotically to $c_0/\sqrt{\alpha_\infty}$, with α_∞ the so-called tortuosity of the porous material which is around unity. In [18] a periodic structure composed by narrow slits with quarter-wavelength inclusions reduces the effective sound speed. By properly designing the slow-sound effect along with the associated dissipation, subwavelength absorption performance could be achieved (see Figure 17). The slow-sound effect can be obtained also by folding branch tubes as proposed in the subwavelength liner of [19], see Figure 18. The space-coiling is indeed a very much pursued technique to achieve the quarter-wavelength performance with subwavelength thickness. Li et al. [23] combined a perforated plate with a coiled coplanar quarter-wavelength resonator allowing to match the air impedance at one frequency, while Chen et al. [24] coiled two tubes, axially coupled in series, with different diameters, allowing to reproduce the subwavelength equivalent of a DDOF resonator.
Also membranes have been exploited in order to produce subwavelength metamaterial absorbers. The team of Hong Kong University of Science, Department of Physics, has provided several contributions to the so-called decorated-membrane (DM) metamaterial [25], [26], [27]. A DM comprises

an elastic membrane with one or more rigid weights attached on its surface, as in Figure 9b. Rigid platelets were added to the membrane in [26] leading to flapping modes of the DM under normally incident plane waves. These flapping modes are responsible of high energy concentration along the perimeter of the platelets. As such modes couple only to evanescent acoustic modes in the plane wave regime, the incident wave energy is dissipated in the displacement profile of the DM. The performance of the DM has then been improved by placing an air-cavity cushion behind, leading to the so-called “hybrid” DM [27] allowing to match the normal incidence air impedance at several frequencies.

3. Active Boundary treatments for noise mitigation

Active acoustical devices can be defined as those requiring some “active component”, i.e. an external energy source, in order to achieve a certain desired acoustic behaviour. In [28] active devices are split into two categories according to weather the external source provides energy to the acoustic domain or not. The latter can therefore be defined as acoustically active, the former as acoustically passive [29].
The reason for employing some active device is to overcome the limitations of the passive ones. First of all, a passive device performance is fixed by its own geometry and material. As the noise frequency range of aero-engines significantly varies with the flight phases (such as take-off, climb, cruise, approach and landing), it could be very advantageous if the target frequency bandwidth for noise mitigation could adapt accordingly. In addition, the aforementioned challenges presented by the new ultra-high bypass ratio (UHBR) turbofan engines (smaller thickness and less surface area available for acoustic treatments along with low frequency noise signature) conflicts with the integral constraint relating bandwidth and thickness demonstrated by [20]. Therefore, several active strategies have been proposed in the attempt to better *adapt* the liner behaviour to the desired acoustic performance, meanwhile satisfying stricter spatial constraints at low frequencies.

3.1 Adaptive Resonators

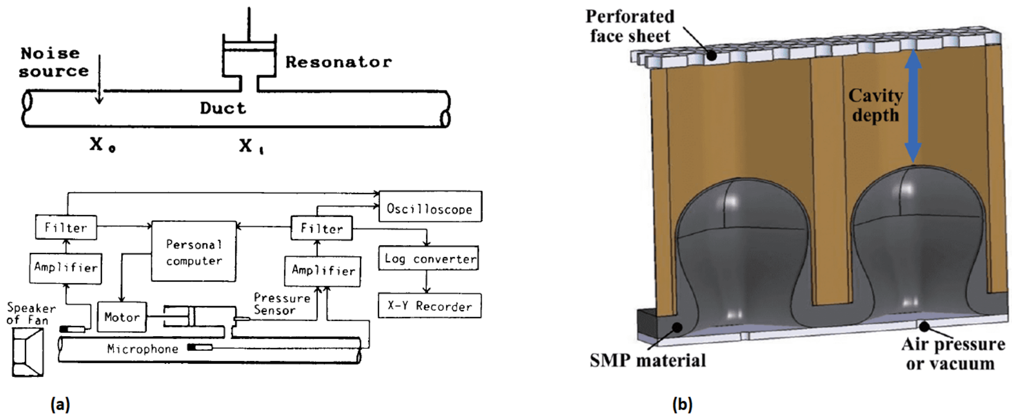

Figure 20 – Helmholtz resonator liner examples with variable cavity volume: the “semi-active” control concept proposed by [30] **(a)**, and the “morphing resonator” by shape-memory-polymer of [31] (image from [12]) **(b)**.

The most intuitive *active* solution might be to simply adjust the geometry of classical resonant absorbers in order to change the resonance frequency per the need. This has been accomplished by varying either the acoustic stiffness, i.e. the cavity ([33], [30], [34], [35], [36], [31]), see Figure 20, or the acoustic mass (i.e. the orifice area , [37], [38], [39], [40], [32] etc.) of Helmholtz resonators, see Figure 21, but both these techniques tended to present complex structure, excessive weight and high energy consumption [12]. In [41], an adaptive Helmholtz resonator liner segment was used with the objective to scatter or redistribute energy among modes, so that to maximize the effectiveness of a neighbour passive liner segment.

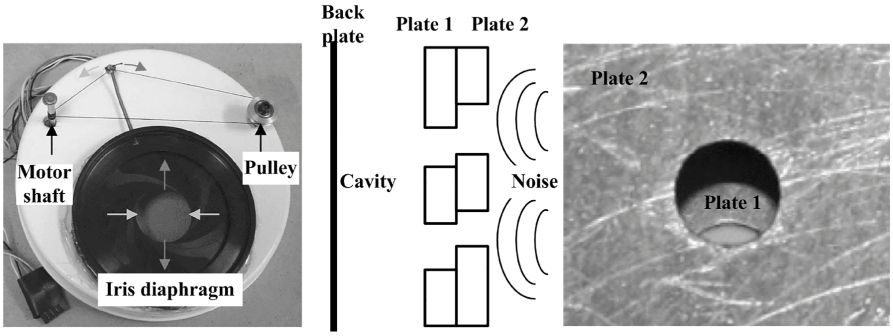

Figure 21 – Helmholtz resonator with variable orifice area of the neck, controlling an iris diaphragm [32].

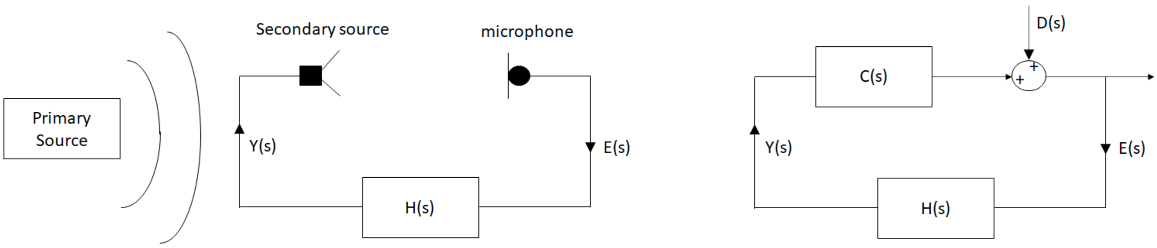

Figure 22 – Idealised physical illustration (left) and its equivalent block diagram (right) of a SISO feedback secondary source approach, from [45].

An alternative to geometry adjustment, is to couple the Helmholtz resonator in series with a compliant piezoelectric composite diaphragm which substituted the rigid back plate. This way, a Double-Degree-Of-Freedom (DDOF) resonator was obtained ([42], [43]), and the piezoelectric could also work as energy harvester [44].

3.2 Active Noise Control: from the secondary source to the impedance control concept

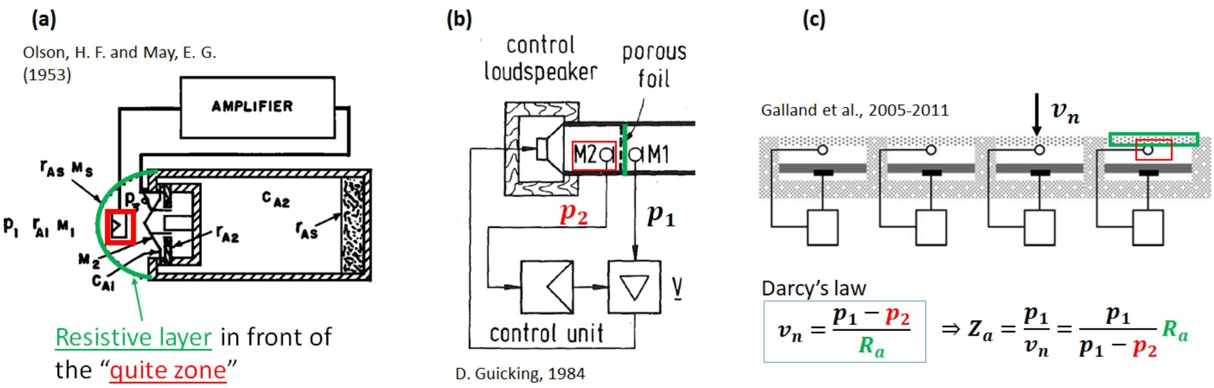

Figure 23 – Evolution of the "hybrid" strategy (secondary source behind a resistive layer) for impedance control: the "electronic sound absorber" of [46] and [47] **(a)**, the "active equivalent of the quarter wavelength resonator" of [48] **(b)**, and the "hybrid" liner controlling the pressure behind the porous layer [49], to achieve a target impedance Z_a [50] **(c)**.

Instead of adjusting the Helmholtz resonator acoustic behaviour, Active Noise Control (ANC) researchers proposed to make use of an electro-mechano-acoustical system (a loudspeaker for example) as either a *secondary source*, therefore acoustically active, or as an *Electroacoustic Absorber* (EA) which is supposed to feature acoustical passivity. The objective of the *secondary source* was to achieve a "quite zone", i.e. a zero (or minimum) sound pressure at one specific location (or in

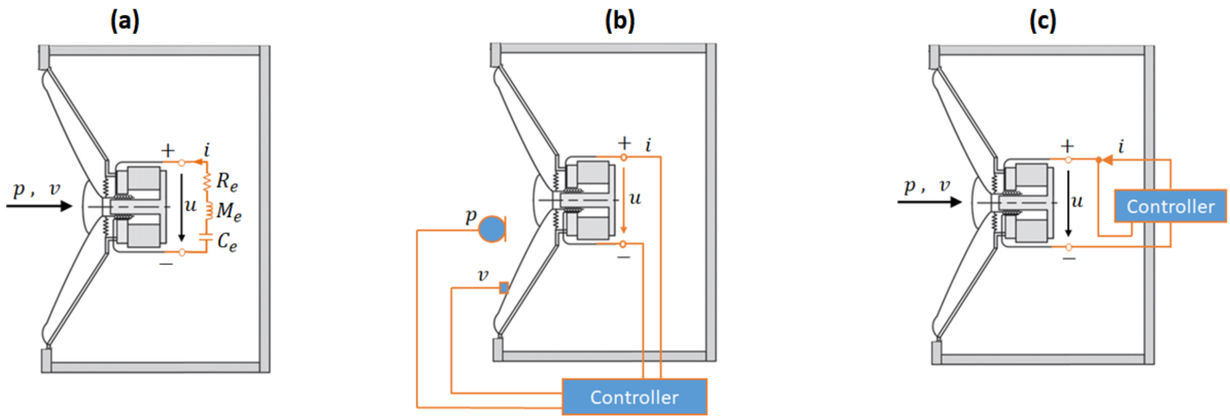

Figure 24 – Evolution of the Electroacoustic Absorber (EA) concept for impedance control: the shunting techniques of [51], **(a)**, the direct impedance control of [52] **(b)**, and the self-sensing strategy of [53] and [54] **(c)**.

a certain area), by the destructive interference principle (or upon a "total acoustic potential energy" minimization) [45]. Such techniques made use of one or several sensors (microphones) coupled with one or more actuators (loudspeakers) by a feedback and/or feedforward control. The main difference between the feedforward and the feedback approaches is that in the first case a separate reference signal, well correlated with the primary noise meant to be cancelled, is used to drive the secondary source. The destabilizing elements of the secondary source strategy (either feedback or feedforward), might be resumed and simplified in: unexpected and undesired behaviours of the acoustic surrounding environment and/or of the control architecture itself. Examples of unexpected behaviour of the acoustic environment are: modifications of the primary source contents and position, changes in the geometry of surrounding cavity, presence of flow-induced measurement noise, insufficient spatial modelling of the acoustic domain where noise reduction must be achieved, etc. Undesired conditions of the control architecture are: unexpected loudspeaker dynamics, time-delay (if digital) and other signal perturbations. All these issues of the secondary source approach, concern the so-called *secondary path*, which is the path travelled by the signal from the secondary source input, to the microphone output [45]. The secondary path differs from the primary one as the latter is the one travelled by the signal from the primary noise source to the microphone. In the simplest Single-Input-Single-Output digital feedback control architecture for example, the transfer function between the disturbance (coming from the primary source) $D(s)$ and the error signal $E(s)$ (see Figure 22), is:

$$\frac{E(s)}{D(s)} = \frac{1}{1 - C(s)H(s)}, \tag{1}$$

where s is the Laplace variable, $H(s)$ is the corrector and $C(s)$ is the secondary path transfer function. From Eq. (1), a high phase shift introduced by $C(s)$, can impact the stability margin, according to the well-known Nyquist criterion [55].
In order to improve global noise reduction in acoustic cavities, *compensating filters* [45], such as phase correction [56] and velocity compensation [57], have been introduced in the feedback controllers. Adaptive filters, on the other hand, rapidly evolving thanks to the digital technology, very much supported the feedforward strategies [58].

Already in the first paper on Active Noise Control, Olson and May [46] suggested to collocate sensor and actuator in order to reduce the risk of having constructive, instead of destructive, interference. The collocation of sensor and actuator, and the idea to use a resistance in front of their "spot type noise reducer" (see Figure 23a), was seminal in the active *impedance control* strategy. Guicking first [48], and then Galland [49], developed the idea of [46] of an "active equivalent of the quarter wavelength resonance absorber" in normal and grazing incidence respectively (see Figure 23b,c).

The “quite zone” (of zero pressure) behind a resistive layer allowed to increase the pressure gradient across the layer and improve the effectiveness of the porous material at lower frequencies. A FX-LMS adaptive filter was employed in order to achieve such hybrid behaviour in [49]. Analogously to the adaptive Helmholtz resonators of Section 3.1 where active means were used to adjust the resonator behaviour, here a secondary source was used to modify the effectiveness of a quarter wavelength resonator equipped with a resistive layer. The same technique was slightly modified by [50] in the attempt to reproduce the Cremer's liner optimal impedance for the first duct modes pair [6], [3] (see Figure 23c). As the optimal impedance could not be achieved in a broadband sense, this interesting approach remained limited to monotonal applications.
These are examples of impedance control achieved through secondary source approaches combined with passive liners, but the collocation of sensor and actuator suggested also another avenue: the modification of the actuator (loudspeaker or else) own mechano-acoustical impedance.

Instead of using the loudspeaker as a secondary source, or to use active means to adjust the impedance of a passive absorber, one can envisage to adjust the loudspeaker own mechano-acoustical response by controlling its electrical dynamics. The objective shifts from creating a “quite zone” at a certain location, to achieving an optimal impedance on the loudspeaker diaphragm. That is why this concept can be referred to as Electroacoustic Absorption [47]. Clearly, the electroacoustic absorption becomes equivalent to the secondary source approach in case of collocation between sensor and actuator, with a target impedance set to zero [59].

Electroacoustic absorption can either be realized by electrically shunting the loudspeaker terminals, therefore modifying the electrical dynamics through passive analogical circuits (see Figure 24a), or by using one or more sensor information (on pressure and/or velocity) to feed into a control algorithm [47] (see Figure 24b). Electrical shunting techniques were investigated by [51] and [47] among others. The main advantage of the electrical shunting strategy is the assurance of *acoustical passivity* (and therefore stability [60]) as the acoustic energy was transferred through the loudspeaker diaphragm vibration, to the electrical passive shunted circuit and dissipated in heat. The so-called direct-impedance control instead, was based upon the direct measurement of both pressure and velocity on the loudspeaker diaphragm. The measured impedance then could be adjusted to the desired one, by minimizing the error signal [52]. Equivalently, a certain reflection coefficient was targeted thanks to the two-microphone-method in [61] and [62]. The addition of intrusive and bulky sensors, such as accelerometers or frontal microphones, and the complexities related to the necessary adaptive filters, make these direct impedance control techniques not ideal for compact and light implementations, such as liners.
An alternative to the use of external sensors was offered by self-sensing strategies (see Figure 24c), as in [63], [53], [64], [65] and [54]. [47] introduced an unified formulation of the EA by demonstrating the equivalence between direct impedance control (based upon pressure and velocity sensing) and the shunting techniques. They highlighted how, on the one hand, the direct impedance control strongly depends upon the neutralization of the loudspeaker electrical impedance, therefore is limited by stability issues coming from wrong modelling of the electrical inductance. On the other hand, the electrical shunting equivalent, even though always stable, is often not easily realizable.
The idea was then to increase the flexibility of the EA by substituting the electrical shunt with a synthesized digital corrector in [51] and [66]. In particular, [66] proposed a mechanical-model-inversion-based control architecture. It detected pressure by one or more microphones placed as close as possible to the loudspeaker to achieve quasi-collocation, and drove the current (thanks to a Howland current pump [67]) in order to equal out the own mechanical dynamics of the loudspeaker, and reproduce a desired acoustical impedance at the speaker diaphragm. The Howland current pump on the one hand, and the use of a microphone on the other, allowed to restrict the model inversion to the mechanical dynamics only, getting rid of the electrical inductance modelling issues of the direct impedance control, at the same time providing a flexible controller thanks to the digital implementation. As the mechanical model uncertainties are much less critical than the electrical ones, it was possible to enlarge the frequency bandwidth of absorption. This control architecture proved great ver-

satility and it has been employed in various applications. Rivet et al. [66], [68] proved the efficiency of such EA concept for room modal equalization. The target impedance was set to a SDOF resonator or to the equivalent impedance of a MDOF resonator composed by more SDOFs in parallel. Such EA strategy was originally designed in order to achieve the characteristic impedance of air $\rho_0 c_0$ in an as large as possible frequency bandwidth, for total absorption in normal incidence. In order to optimize the target impedance assigned on the EAs for room modal equalization, a numerical study has been conducted in [69] by considering purely resistive impedances and discarding the effect of the EA reactance. A plane wave approximation of the cavity modes [70] has recently inspired a fast technique to reconstruct the pressure field in rooms [71] which could be adopted to envisage more physically-grounded optimizations of the EA target impedance, taking into account both its resistance and reactance.
In [72] the EA strategy has been implemented in a liner made up of small unit cells, and tested in an acoustic waveguide with and without flow. The main interest of such EA-liner is its tunability which makes it particularly attractive for reducing noise radiation from turbofan engines during different flight phases. The nacelle parietal walls at the inlet and bypass regions (see Figure 1) are the most suitable for such electro-active treatments.
The same control strategy has also been proposed to obtain a phase gradient reflection coefficient, by grading the complex target impedance on an array of EAs, so that to control the reflected wavefront direction [73]. The interest in wave-steering ranges from carpet cloaking to noise transmission reduction for grazing incidence in an acoustic waveguide.

The EA technology also allows for innovative, generalized impedance concepts, such as the Advection Boundary Law [29, 74, 75, 76, 77, 78, 74, 79, 80, 81, 76, 82, 83, 84, 85]. The optimization of generalized impedance concepts will require to define new analytical tools. Reinforcement learning, on the other hand, might help in such non-convex optimization problems.

4. Conclusions

In this paper, we have introduced the general problem of acoustic liners. The classical analytical optimization strategies are reviewed, along with their limitations and the space for improvement. Then, we have gone through various technologies employed for liners innovation, both by passive and active means. The horizons opened by active impedance control strategies, demands for further efforts towards new guidelines for the generalized impedance synthesis. In this sense, reinforcement learning algorithms can provide a significant help. Such innovative tools, can indeed be exploited in active control systems to overcome the limitations of defining optima conditions a-priori. Rather, the optimization can be achieved in loco, therefore avoiding the loss of performances due to deviations from analytical optima, as those due, for example, to the uncertainties in the flow field and its related quantities.

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6. Copyright Statement

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