



POLITECNICO DI MILANO
DEPARTMENT OF MECHANICAL ENGINEERING
DOCTORAL PROGRAMME IN 2026

SILESIAIAN UNIVERSITY OF TECHNOLOGY

Faculty of Automatic Control, Electronics and Computer Science
Department of Measurements and Control Systems

Active device casings with an open part or made from a different material

Doctoral Dissertation

by

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A dissertation submitted for the award of Doctor of Philosophy under a double-degree programme performed with Politecnico di Milano, Italy.

2026
Gliwice, POLAND

Abstract

Noise pollution has become a major concern across many modern engineered systems, from transportation and industrial equipment to everyday household appliances. Devices in these various domains must comply with strict noise-emission requirements, which has driven extensive research into reducing the noise radiated from their structures and enclosures. To satisfy weight and dimensional constraints, lightweight casings are widely used. These structures are not only efficient from a design perspective but also provide the possibility of applying active control to improve their acoustic performance.

In vibro-acoustic coupled systems, openings in plate structures serve several practical functions across automotive, aerospace, marine, and naval engineering applications, such as facilitating heat dissipation, routing electrical cables, enabling personnel access, and reducing weight. These openings create a mixed interface—vibrating structure plus openings—so total sound comes from two coupled paths. The low-mid band is critical yet hard to treat passively under weight, space, and transparency constraints.

The dissertation develops a design-oriented analytical framework with three novelties. First, the model is parametric in opening shape and location for systematic exploration and optimization. Second, plate, enclosure, and free-field are coupled in one formulation: Rayleigh-Ritz plate with hyperbolic-trigonometric admissible functions for thin, clamped panels with openings; enclosure modal expansions augmented at the interface to enforce pressure/velocity continuity; radiated power via a radiation-resistance matrix for parametric studies. Third, total sound power combines plate radiation with sound transmission through the opening, clarifying how geometry redistributes energy between paths.

Combined finite-element and boundary-element modelling visualizes the coupled fields and verifies the theory. Laboratory tests on a concrete-frame enclosure with a thin steel plate confirm natural frequencies, mode shapes, vibration spectra, and sound power. The models reproduce the measured shifts and redistribution of vibro-acoustic responses due to the geometry changes.

On this basis, this work establishes a deterministic opening-design workflow. Treating geometry as decision variables (area, placement, and orientation) shapes the response and reduces low-band emissions. Optimized designs achieve up to 12 dB peak reduction in total sound power within 100–350 Hz. The results revealed that having discontinuities in the opening contour leads to higher resonances in the acoustic response of the system in the mid-range of the considered frequencies. Also, if the opening is placed asymmetrically, the system shows additional resonances at these frequencies.

Active strategies under broadband excitation are evaluated. A sound-power-aware actuator-placement method is introduced for inertial actuators on plates with openings; it uses sound-power sensitivities and opening pressure to target leakage-influenced regions with high authority. Experiments compare Active Structural Acoustic Control (ASAC) on the plate, Active Noise Control (ANC) at the opening, and a dual-actuator-type arrangement. The dual case gives the greatest attenuation at error microphones—averaging around 15 dB and peaks exceeding 30 dB—outperforming ASAC-only and ANC-only while remaining compatible with compact device-casing constraints.

The framework is applied to transparent-patched plates to separate radiation and transmission in multi-material panels. A frequency-dependent transmission factor represents the patch, distinguishing plate-borne radiation from through-opening leakage and guiding actuator layout

when transparency is required. Comparisons show how patch areal mass and stiffness modify the mixed interface and how active placement should adapt.

Overall, this research links opening geometry, actuator layout, and mixed-interface physics to measurable vibro-acoustic response. It delivers validated models, an opening-design methodology with demonstrated passive reductions, and active strategies—including a dual-actuator configuration—that achieve substantial broadband attenuation in the low-mid frequency band.

Streszczenie

Hałas emitowany przez współczesne systemy inżynierskie stanowi istotne ograniczenie projektowe w wielu zastosowaniach transportowych, przemysłowych oraz konsumenckich. W celu spełnienia wymagań dotyczących emisji hałasu powszechnie stosuje się lekkie obudowy zawierające otwory funkcjonalne, które jednocześnie prowadzą do powstania złożonego sprzężenia wibroakustycznego oraz zwiększonej transmisji dźwięku. Szczególnie istotne znaczenie ma zakres niskich i średnich częstotliwości, w którym metody pasywne są ograniczone przez wymagania masowe, przestrzenne oraz funkcjonalne.

W rozprawie opracowano zorientowane projektowo ramy analityczne umożliwiające modelowanie oraz optymalizację układów wibroakustycznych z otworami. Zaproponowany model jest sparametryzowany względem kształtu i położenia otworu oraz uwzględnia sprzężenie pomiędzy płytą, wnęką akustyczną oraz polem swobodnym. Dynamikę płyty opisano metodą Rayleigha–Ritza z zastosowaniem funkcji hiperboliczno-trygonometrycznych, natomiast moc promieniowaną wyznaczono z wykorzystaniem macierzy oporu promieniowania. Całkowitą moc akustyczną zdefiniowano jako sumę promieniowania strukturalnego oraz transmisji przez otwór, co umożliwiło analizę redystrybucji energii akustycznej w funkcji parametrów geometrycznych.

Poprawność modelu zweryfikowano z wykorzystaniem modelowania numerycznego FEM–BEM oraz badań eksperymentalnych przeprowadzonych na układzie obudowy z cienką płytą stalową. Uzyskano zgodność częstotliwości własnych, postaci drgań oraz charakterystyk mocy akustycznej. Na tej podstawie opracowano deterministyczną procedurę projektowania otworów, w której parametry geometryczne traktowane są jako zmienne decyzyjne. Zoptymalizowane konfiguracje umożliwiły redukcję szczytowej mocy akustycznej do około 12 dB w zakresie 100–350 Hz. Wykazano ponadto, że nieciągłości konturu otworu oraz jego asymetryczne położenie prowadzą do wzrostu rezonansów w średnim paśmie częstotliwości.

W pracy przeanalizowano również strategię aktywnej redukcji hałasu w warunkach szerokopasmowego wymuszenia. Zaproponowano metodę rozmieszczenia inercyjnych wzbudników drgań opartą na kryteriach mocy akustycznej oraz wpływie wycieku przez otwór. Badania eksperymentalne obejmowały porównanie aktywnej strukturalnej kontroli akustycznej (ASAC), aktywnej redukcji hałasu (ANC) oraz konfiguracji hybrydowej typu dual-actuator. Konfiguracja hybrydowa umożliwiła uzyskanie średniej redukcji sygnału błędu rzędu 15 dB oraz maksymalnej przekraczającej 30 dB w wybranych zakresach częstotliwości.

Zaproponowane podejście zastosowano również do płyt wielomateriałowych z transparentnymi wstawkami, wprowadzając zależny od częstotliwości współczynnik transmisji umożliwiający rozdzielenie składowej promieniowania strukturalnego i wycieku akustycznego. Wyniki wskazują na istotny wpływ właściwości materiałowych oraz rozmieszczenia elementów wykonawczych na skuteczność sterowania.

Przeprowadzone badania wykazują ścisły związek pomiędzy geometrią otworów, rozmieszczeniem elementów wykonawczych oraz odpowiedzią wibroakustyczną układu. Opracowane modele, procedury projektowe oraz strategię sterowania aktywnego umożliwiają skuteczną szerokopasmową redukcję emisji hałasu w lekkich strukturach z otworami funkcjonalnymi.

Riassunto

Il controllo del rumore rappresenta un aspetto fondamentale nella progettazione di numerose applicazioni ingegneristiche, tra cui sistemi di trasporto, apparecchiature industriali e dispositivi domestici. L'esposizione prolungata a elevati livelli sonori può ridurre il comfort dell'utente e contribuire a effetti negativi sulla salute, rendendo necessarie normative sempre più restrittive in materia di emissioni acustiche. Per soddisfare simultaneamente requisiti acustici e vincoli strutturali, sono ampiamente impiegate strutture di rivestimento leggere. La ridotta massa di tali strutture offre vantaggi in termini di integrazione e produzione, consentendo al contempo l'applicazione di strategie di controllo attivo per migliorare l'attenuazione sonora complessiva.

Nei sistemi vibro-acustici accoppiati, le aperture presenti nelle strutture a piastra svolgono diversi ruoli funzionali in ambiti applicativi quali il settore automobilistico, aerospaziale e navale. Esse favoriscono la dissipazione termica, il passaggio di cablaggi elettrici, l'accesso per la manutenzione e la riduzione della massa complessiva. Tuttavia, introducono un'interfaccia mista costituita da regioni strutturali vibranti e porzioni aperte, per cui l'emissione sonora totale deriva da percorsi di trasmissione accoppiati. La gamma di frequenze medio-basse risulta particolarmente critica, poiché difficilmente mitigabile mediante soluzioni passive in presenza di vincoli di massa, spazio o trasparenza.

La tesi sviluppa un quadro analitico orientato alla progettazione caratterizzato da tre contributi principali. In primo luogo, viene introdotta una rappresentazione parametrica della geometria e della posizione delle aperture, che consente analisi sistematiche e procedure di ottimizzazione. In secondo luogo, l'accoppiamento tra la piastra, la cavità interna e il campo acustico esterno è descritto mediante una formulazione unificata basata su un modello Rayleigh-Ritz con funzioni ammissibili iperbolico-trigonometriche, un'espansione modale della cavità con continuità imposta di pressione e velocità normale, e una matrice di resistenza alla radiazione per la stima della potenza sonora irradiata. In terzo luogo, la potenza sonora totale è definita come contributo combinato della radiazione strutturale e della trasmissione diretta attraverso l'apertura, evidenziando il ruolo della geometria nella redistribuzione dell'energia vibro-acustica.

La modellazione agli elementi finiti e agli elementi al contorno è impiegata per visualizzare i campi accoppiati e validare le previsioni teoriche. Le verifiche sperimentali su un involucro con piastra sottile in acciaio confermano le variazioni delle frequenze naturali e delle risposte vibro-acustiche dovute a modifiche geometriche. Configurazioni ottimizzate consentono riduzioni fino a 12 dB della potenza sonora totale irradiata nella banda 100–350 Hz. Inoltre, contorni discontinui o aperture disposte in modo asimmetrico introducono risonanze aggiuntive nella banda intermedia, evidenziando l'influenza della geometria sul comportamento acustico globale.

È stato inoltre analizzato il controllo attivo in condizioni di eccitazione a banda larga. È stata sviluppata una strategia di posizionamento degli attuatori basata su criteri di potenza sonora, finalizzata all'impiego di attuatori inerziali per una riduzione efficace del rumore. Confronti sperimentali tra Active Structural Acoustic Control (ASAC), Active Noise Control (ANC) e una configurazione ibrida mostrano che l'approccio duale garantisce la maggiore attenuazione, con valori medi intorno a 15 dB e picchi superiori a 30 dB a specifiche frequenze, mantenendo al contempo un'implementazione compatta. Il quadro teorico è stato infine esteso a piastre con inserti trasparenti, introducendo fattori di trasmissione dipendenti dalla frequenza per descrivere l'influenza dei materiali e supportare il corretto posizionamento degli attuatori nei casi in cui sia richiesta accessibilità ottica.

Nel complesso, la tesi mette in relazione la geometria delle aperture, la fisica delle interfacce miste e le strategie di attuazione con la riduzione misurabile della potenza sonora. Essa fornisce modelli validati e linee guida progettuali per il controllo passivo e attivo del rumore in sistemi vibro-acustici leggeri operanti nella gamma di frequenze medio-basse.

Acknowledgements

I would like to express my sincere gratitude to my supervisors, Prof. Marek Pawełczyk and Prof. Stanisław Wrona from the Silesian University of Technology, Poland, and Prof. Hamid Reza Karimi and Prof. Francesco Ripamonti from Politecnico di Milano, Italy, for their guidance, encouragement, and insightful feedback throughout this work.

This dissertation was carried out as a double-degree doctorate under the agreement between the Silesian University of Technology, Poland, and Politecnico di Milano, Italy. A part of the work was supported by the project IN-NOVA: Active reduction of noise transmitted into and from enclosures through encapsulated structures, which has received funding from the European Union's Horizon Europe program under the Marie Skłodowska-Curie grant agreement no. 101073037, coordinated by the Silesian University of Technology.

I sincerely thank Prof. Marek Pawełczyk for his exceptional scientific guidance, steady career support, and unfailing commitment, which enabled the realization of my PhD within the international framework of the IN-NOVA project and will continue to inform my future research and professional growth. His dedicated involvement was fundamental to establishing the double-degree doctoral program.

I also express my gratitude to Prof. Stanisław Wrona for his warm hospitality in Poland, his constant technical assistance, and his many insightful scientific discussions. His daily support fostered a productive environment that significantly advanced this work and will meaningfully contribute to our future collaboration.

I gratefully acknowledge Prof. Hamid Reza Karimi, who has supervised my research since my master's studies and continued to guide me throughout my PhD. His long-term mentorship and ongoing collaboration have played a pivotal role in shaping my research direction and academic development.

I am likewise grateful to Prof. Francesco Ripamonti for his academic supervision and the hospitality he extended during my stay at Politecnico di Milano. His support was instrumental in the successful implementation of the double-degree doctoral program.

I am also grateful to Prof. Jian Kang and Prof. Francesco Aletta from University College London, UK, for hosting me during my secondment at their esteemed institute, and to Dr. Rolf Schirmacher and Dr. Jan Kirchhof for their cooperation and hospitality during my secondment at Müller-BBM Active Sound Technology GmbH, Germany. My appreciation further extends to all members of the International Supervisory Council of the IN-NOVA project.

I am deeply thankful to my family for their unwavering support, especially my parents, Said and Mervat, whose love and patience made this journey possible.

Khaled Maamoun
Gliwice, 2026

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Outline of the thesis

The thesis follows the **article-based** format and is organized into two main parts: a cover essay introducing the research background, motivation, and overall contributions, followed by the appended papers listed below.

Paper A K. S. A. Maamoun, S. Wrona, M. Pawelczyk, and H. R. Karimi, “Shaping of the frequency response of vibrating plates with openings for vibro-acoustic systems,” *Mechanical Systems and Signal Processing*, vol. 218, p. 111539, 2024.
<https://doi.org/10.1016/j.ymssp.2024.111539>

Paper B K. S. A. Maamoun, M. Pawelczyk, and S. Wrona, “Analyzing the impact of sound source positioning on frequency response of vibrating plates with openings in vibro-acoustic systems,” in *Proceedings of the 30th International Congress on Sound and Vibration (ICSV30)*, Amsterdam, IIAV, July 2024.

Paper C K. S. A. Maamoun, S. Wrona, M. Pawelczyk, and H. R. Karimi, “Optimizing design of openings in vibrating plates for enhanced vibro-acoustic performance using a genetic algorithm approach,” *Journal of Sound and Vibration*, vol. 598, p. 118847, 2025.
<https://doi.org/10.1016/j.jsv.2024.118847>

Paper D K. S. A. Maamoun, C. K. Lai, S. Wrona, M. Pawelczyk, and H. R. Karimi, “Dual-actuator-type active noise control in vibro-acoustic systems with openings,” *Scientific Reports*, vol. 15, no. 1, p. 32278, 2025.
<https://doi.org/10.1038/s41598-025-17810-8>

Paper E K. S. A. Maamoun, A. Aboutiman, and A. M. Ali, “Synergistic use of inertial actuators for advanced noise control in transparent-patched vibrating plates,” *Measurement*, vol. 257, p. 118701, 2026.
<https://doi.org/10.1016/j.measurement.2025.118701>

List of Abbreviations

ANC	Active Noise Control
ASAC	Active Structural Acoustic Control
FEM	Finite Element Method
BEM	Boundary Element Method
PABE	Pressure Acoustics, Boundary Elements
RRM	Rayleigh–Ritz Method
MITC	Mixed Interpolation of Tensorial Components
ANM	Analytical (Rayleigh–Ritz) Model
GA	Genetic Algorithm
FxLMS	Filtered-x Least Mean Squares
NLMS	Normalized Least Mean Squares
ADC/DAC	Analog-to-Digital / Digital-to-Analog Converter
MUMPS	Multifrontal Massively Parallel Sparse (solver)
PVC	Polyvinyl Chloride

List of Symbols

x, y, z	Cartesian coordinates
L_x, L_y, L_z	Enclosure/plate spans in x, y, z
a, b, h	Plate length, width, thickness
ζ, η	Non-dimensional coordinates ($\zeta = x/L_x, \eta = y/L_y$)
S_p	Structural (plate) interface area
S_{op}	Opening interface area
S_{mix}	Mixed interface ($S_p \cup S_{op}$)
S_a	Actuator contact areas on the plate
ρ_a	Air density
c	Speed of sound in air
ρ_p	Plate material density
E	Young's modulus of the plate material
ν	Poisson's ratio of the plate material
$D(x, y)$	Flexural rigidity
m_{pt}	Areal mass of transparent patch (PVC)
$w(x, y, t)$	Plate transverse deflection
$\Phi(\zeta, \eta)$	Row vector of admissible (basis) functions for w
$p(x, y, z, \omega)$	Acoustic pressure (frequency domain)
ω	Angular frequency
$k = \omega/c$	Acoustic wavenumber
M_p, K_p	Plate mass and stiffness matrices
M_c, K_c	Enclosure mass and stiffness matrices
C_c	Opening-loss matrix of the enclosure
$C_{c\&p}$	Coupling matrix between cavity and plate
$M_{\text{total}}, K_{\text{total}}, C_{\text{total}}$	Coupled mass, stiffness, and damping matrices

V, Q	System response vector and generalized external forcing vector
A_s	Surface element area in patch discretizations
$\mathbf{R}(\omega)$	Radiation-resistance matrix of the plate
$\mathbf{q}_r$	Vector of plate normal velocities at surface patches
$\mathbf{q}_p$	Vector of acoustic pressures at opening patches

Chapter 1

Introduction

1.1 Background and motivation

Across modern engineering domains, the pursuit of quieter and lighter products has become a defining design challenge. Industrial machinery, medical devices, and mobility vehicles now operate within stricter acoustic comfort targets, while consumer expectations demand transparency, both optical and metaphorical, in product performance. To achieve compactness, efficiency, and visual appeal, many enclosures integrate openings for ventilation, maintenance access, or signal transmission, and increasingly employ composite or transparent materials. Yet, these same features disrupt the balance between structural rigidity and acoustic isolation, introducing complex vibro-acoustic interactions that amplify noise emissions.

Conventional Passive Noise Control (PNC) treatments, once sufficient for sealed casings, struggle to remain effective without adding mass or volume [1]. In this context, active strategies emerge as essential complements. Active Noise Control (ANC) mitigates airborne propagation through adaptive acoustic cancellation, whereas Active Structural Acoustic Control (ASAC) [2, 3] employs inertial actuators to modify the plate’s dynamic response, effectively turning the structure into a controllable acoustic radiator capable of reducing total sound emission. Together, these approaches provide a unified framework for addressing the acoustic challenges of modern lightweight and multifunctional structures.

1.2 Foundational concepts

1.2.1 Passive approaches and their limits

Thick, double-walls, and sandwich panels are standard for noise reduction enhancement. At low frequencies, however, sound level gains rely mainly on added surface density and thickness, which conflicts with stringent weight, space, and transparency constraints typical of modern casings [1, 4].

An additional technique is to reshape the plate dynamics by adding discrete masses, ribs, or stiffeners to shift resonances and reduce radiation. Classic and modern studies show that added elements can target dominant modes and improve noise reduction, with analytical, finite element method (FEM)–based, and radiation-matrix analyses used to guide layout and sizing [5, 6, 7]. Yet these benefits are often narrowband and sensitive to placement tolerances; optimization helps, but complexity grows with the number of added elements [1]. Double walls exploit an intervening air cavity that increases the transmission loss (TL) above the mass–air–mass resonance, but TL collapses around that resonance unless the cavity spacing and damping are tuned. Porous fills can help for light panels while having limited effect for heavy ones [8, 9]. Practical constraints (cavity depth, mounting, added weight) further limit applicability in compact devices.

Sandwich plates increase the bending stiffness-to-mass ratio and can shift structural resonances upward, with well-established sound transmission loss (STL) formulations and extensive experimental validation [10, 11]. Recent work embeds periodic resonators or optimizes core topology to extend performance without prohibitive mass, but added manufacturing complexity and band-limited effects remain trade-offs [12, 13].

Finally, when transparency or airflow must be preserved (e.g., monitoring ports, vents), laminated or lightweight panels still underperform below a few hundred hertz; increasing thickness helps but clashes with design constraints, reinforcing the need for complementary active solutions [14, 15, 16].

1.2.2 Modeling vibro–acoustic systems

Analytical vibro–acoustic modeling formulates the coupled dynamics of elastic structures and acoustic domains within a single framework of field equations and interface conditions. A standard method employs Rayleigh–Ritz expansions for the plate together with a modal representation of the cavity; coupling is enforced through continuity of normal velocity/displacement and pressure at the plate–fluid boundary. Radiated sound power to the free field is then obtained via radiation–resistance matrix [17, 18].

The same analytical framework admits extensions for varied boundary conditions, geometric features, and material inhomogeneities by adapting the admissible functions and interface terms, while retaining close correspondence with measurements [19, 20]. For complex layouts or detailed verification, numerical methods complement these models—most commonly the finite element method (FEM) for structural/acoustic domains and the boundary element method (BEM) for exterior radiation; wave-based techniques provide an additional option when separable domains are available [21].

1.2.3 Active noise control for wall apertures

For apertures in rigid walls, two ANC layouts are prevalent. Boundary arrays distribute secondary sources and error microphones along the frame to impose a desired boundary condition (e.g., reduced acoustic pressure/particle velocity) while keeping the opening usable. Planar arrays span the aperture with a grid of sources/sensors to synthesize a virtual acoustic barrier across the opening, offering higher spatial control at the cost of partial occlusion and maintenance access [22, 23]. In both cases, actuator spacing sets the controllable bandwidth (a spatial-sampling limit), and sensor placement upstream/downstream of the aperture shapes the protected zone.

Hybrid concepts embed micro-perforated layers or other porous inserts within the opening so that passive absorption carries mid–high frequencies while ANC targets the low-frequency leakage; this extends usable bandwidth and can reduce the control effort required from the loudspeakers [24, 25]. Practical trade-offs include added build complexity, small increases in thickness, and sensitivity to installation tolerances.

Most implementations adopt feedforward control with a reference microphone and multichannel filtered-x LMS, which explicitly models the secondary path and maintains stability/robustness under plant variations; early adaptive ANC developments established this pathway from concept to practice [26, 27, 28]. In lightweight vibrating plates the structure-borne path can be dominating. Active Structural Acoustic Control (ASAC) is therefore complementary and is discussed elsewhere [29, 30]. However, to our knowledge, in-depth analyses for ASAC under broadband noise excitation with laboratory validation for lightweight panels with openings have not been reported in the literature.

1.2.4 Actuator placement and optimization

Actuator placement is a main determinant of achievable attenuation in ASAC [2]. Candidate sites are typically ranked using modal controllability measures, energy weights, or radiation-aware sensitivities when the objective is acoustic output [7, 31]. For plates with material inhomogeneities (e.g., patches or laminates), these radiation-aware criteria remain applicable provided the spatial variation of properties is included in the model [7].

For plates with openings, however, the literature offers limited placement guidance; leakage paths are often not modeled explicitly, leaving placement rules insufficiently developed [32]. Because actuator locations are limited to a finite set by mounting, wiring, and clearance, selecting the best subset is a discrete optimization problem. Practical solvers therefore use evolutionary or hybrid greedy-evolutionary searches to explore this space [33]. To keep layouts implementable and reliable, the optimization adds terms or constraints that cap control effort and reduce sensitivity to small placement errors and parameter drift.

1.3 Research gap

Sound power can be estimated reliably, yet what is missing is a design-oriented workflow that treats the opening shape and location as decision variables, predicts the resulting vibro-acoustic response and radiated sound power across frequency, and is validated for diverse geometries. Existing studies provide reusable radiation-resistance operators and vibration-pattern-based estimators, but they do not quantify how specific geometric features—such as contour discontinuities or asymmetric placement—map to measurable acoustic outcomes across the design space [34, 35].

For opening design, an optimization tool that provides the flexibility to select opening shape and location under functional constraints, while simultaneously accounting for coupling and transmission at the interface, is still lacking [36, 37]. In addition, for plates composed of different materials (for example, transparent or lightweight patches), most modeling and control studies still adopt homogeneous-plate assumptions; the combined effects of spatially varying properties and acoustic leakage on actuator placement remain insufficiently addressed [4, 1].

In terms of control, ANC at openings in rigid panels has advanced, whereas active control for lightweight plates with openings remains dominated by tonal or simulation-based studies; broadband, laboratory-validated evaluations are still limited [30, 38].

Actuator placement methods are established for vibrating plates using vibration and sound power metrics. Explicit placement rules and optimization workflows that incorporate leakage paths and design changes introduced by openings remain underdeveloped [31, 32].

1.4 Aims and scope

This dissertation develops a design-oriented framework for vibro-acoustic systems that include openings and, where relevant, plates made from different materials. The aims are to: (i) formulate an analytical model that couples plate and cavity dynamics with free-field while remaining explicitly parametric in opening shape and location; (ii) create an opening-design procedure that optimize shape and placement under system constraints and predicts the resulting sound power response; (iii) devise an actuator placement methodology for inertial actuators that is informed by sound power metrics and accounts for leakage and local stiffness changes introduced by openings and material variations; and (iv) evaluate active control strategies at laboratory scale under broadband excitation, including active noise control, active structural acoustic control, and their combined use.

The scope focuses on sound power and vibration responses. System requirements such as airflow, visibility, and mass are respected through geometric and placement limits. Validation

combines analytical predictions with selected finite element simulations, supported by experimental measurements. Performance is assessed using sound power, selected-point sound pressure levels, and control effort.

1.5 The thesis of the dissertation

The main thesis is as follows: it is possible to efficiently reduce noise emitted from vibro-acoustic systems with structural openings or with areas made of a different material. This can be achieved by optimally designing the shape and location of the openings, using inertial actuators arranged based on sound power metrics, and combining inertial and loudspeaker actuators in a hybrid configuration.

1.6 Organization of the dissertation

The dissertation is structured as an article-based dissertation with two main components. The first part is a cover essay that outlines the research context, motivation, and contributions. The second part consists of the appended journal articles that collectively form the core of the work.

Chapter 1 contains the introduction. Chapter 2 reviews passive and active noise-reduction methods, with emphasis on systems featuring openings. Chapter 3 presents the methodologies implemented in this work: models, the optimization framework, control methods, and the experimental setup. Chapter 4 presents the results. In Chapter 5, conclusions, the author's contributions, summary of the attached articles, and future work are presented. Appendix A contains the five articles included in the dissertation.

Chapter 2

State of the art

2.1 Passive noise control

Passive noise control (PNC) denotes noise mitigation strategies that require no external power. Transmission loss improves with surface density in the mid band, but below a few hundred hertz the mass and thickness required become impractical for compact or transparent casings. Recent lightweight designs confirm that purely mass-based strategies cannot satisfy low-frequency targets under modern form-factor constraints [1, 4]. Double-wall partitions can raise transmission loss above the mass–air–mass resonance; however, performance deteriorates near resonance unless the cavity depth and damping are tuned carefully. Porous fills mitigate this to some extent for light skins, while the benefit for heavy panels is limited, and device form factors often restrict cavity depth [8, 9].

Sandwich and laminate constructions improve the stiffness-to-mass ratio and shift structural resonances upward, with canonical formulations and extensive validation available [10, 11]. More recent core designs extend performance in selected bands, but the gains are typically narrowband and introduce manufacturing complexity, especially in the sub-kilohertz range [12]. Recently, the authors of [13] optimized the double panel configuration to achieve the maximum STL. The design focuses on the core between the panels, with the panels themselves considered as passive domains. They achieved STL improvements of over 15–40 dB in targeted frequency ranges from 1000 Hz to 3000 Hz.

Another route is to reshape the structural dynamics using ribs, stiffeners, or discrete masses. When tuned to dominant bands, these additions can reduce radiation; layouts are guided by analytical criteria, finite-element studies, and radiation-based evaluations. The benefits, however, tend to be narrowband and sensitive to placement tolerances, which complicates application in small enclosures [39, 6]. In [40], a detailed modeling framework for this approach treats a rectangular plate with elastically restrained edges (modeled as translational and rotational springs), additional masses, and arbitrarily oriented ribs based on Rayleigh–Ritz method. By optimizing the positions of these additional elements, the author extends their results to shape the frequency response of the vibrating plate using a memetic algorithm [1]. The study identifies practical limits that matter for product casings: feasible mounting area, allowable added mass and rib geometry, sensitivity to adhesive bonding, and the need to pre-select the maximum number of elements, all of which are reflected explicitly in the optimization constraints. This framework extends naturally to lightweight casings, where reduced surface density and bending inertia improve the effectiveness of wall-mounted ASAC (see Section 2.2).

However, the optimization above does not account for functional openings in the vibrating plate. Such openings are common in engineering applications—including automotive, aerospace, marine, and industrial equipment—for heat dissipation, cable routing, personnel access, and weight reduction; they introduce airborne leakage and alter local stiffness and boundary compliance. Accordingly, Section 2.3 reviews the state of the art on the modeling of thin plates with

openings and discusses the implications for vibro-acoustic response and control design. In view of these considerations, the literature frequently adopts a thin vibrating plate as a baseline barrier for lightweight device casings in the low-to-mid frequency range, since it satisfies mass and functional constraints, remains controllable, and provides a suitable substrate for the active strategies discussed in Section 2.2.

2.2 Active noise control

The passive control methods are effective in noise reduction for wide range of frequency, but their systems get more complicated for low frequencies [41]. Thus, active methods can be employed in such cases. In 1936, ANC technology was invented and patented [26]. It's based on the concept of canceling the noise by sound waves that have the same amplitude with an opposite phase. With the development of the digital signal processing hardware components, the adaptive ANC systems were first implemented effectively using least-mean-square (LMS) algorithm [27]. However, the physical path between the actuator and the error sensors can cause instability, which motivated researchers to develop the filtered-x least-mean-square (FxLMS) algorithm [27, 28, 42]. Hence, the reference signal to the controller is filtered through the model of the secondary path between the actuator and the error sensors, making the FxLMS the most popular ANC algorithm due to its stability and simple structure [43, 44, 45].

Different authors implemented and evaluated multi-channel versions of these algorithms in [46, 47, 48] and [49]. However, the multi-channel FxLMS-based algorithms, such as in [50, 51], are still preferred in a wide variety of applications due to their stability and flexibility of implementation. A multi-channel FxLMS with online secondary path modeling (SPM) using the auxiliary random noise technique and variable step sizes for each filter was shown to have a high convergence rate [52]. In [53], the authors addressed the increased residual error from auxiliary signal injection for online secondary path estimation to enhance the system performance. The interaction between decentralized controllers in ANC system was studied in [54]. The increased complexity of ANC systems with a larger number of actuators, controllers, and sensors elevated the need for distributed control algorithms. The authors analyzed the performance of decentralized architectures by deriving stability condition based on Gerschgorin circle theorem, which was proved to be sufficient for stabilizing small systems. Along with the development of wireless communication technology between various ANC system components, the authors proposed an ANC system over a network of distributed acoustic nodes [55, 56]. Recently, an augmented diffusion FxLMS algorithm with neighborhood-based adaptation and node-based combination was proposed in [57].

Active Structural Acoustic Control (ASAC) is mostly performed using a passive noise barrier, vibration actuators, sensors and a control system, as in [30, 58]. ASAC can reduce noise transmission through a structure by controlling its vibrations. With careful implementation, ASAC not only reduce the noise in specific local areas but can achieve overall global reduction [59]. In 1994, the implementation of ASAC in device casings was first introduced and patented [60]. In general the noise casings can be categorized into rigid casings which have steel rigid frame and lightweight casings. The control for the whole device casing, named Noise-Controlling Casings, has been significantly developed and implemented with various control algorithms for the rigid casing [61, 62], and for a light-weight casing [63, 64]. Additionally, it can be applied to off-the-shelf devices as long as their casings' walls can be subjected to vibration forces [65]. In addition to PNC and ANC, semi-active barrier technology can also be applied to noise-controlling casings. This technology utilizes the inherent characteristics of the structure to meet the energy requirements for operation, typically needing only a small additional external energy supply [66]. Recently, the authors introduced semi-active actuators with a tunable mass moment of inertia, as detailed in

[67, 68].

2.3 Modeling of vibrating plates with openings

Vibrating plates that include openings require (i) a structural domain description that captures stress concentration and mode changes, and (ii) a consistent coupling to the surrounding acoustic domain across a mixed interface comprising a structural and open segment. Analytical formulations are well suited for design-oriented studies because they remain explicitly parametric in geometry and boundary impedance. The first approach presents an assumed-mode formulation with energy subtraction for plates with openings then contrasts its admissible functions with those used in an improved Fourier-series approach. Using this formulation, natural frequencies and mode shapes for rectangular plates with circular, rectangular, or oval openings show close agreement with finite-element predictions over the first several modes [69].

A key modeling choice in [69] is the use of polynomial admissible functions for the plate kinematics. By contrast, in [19], improved Fourier series augmented by supplementary trigonometric terms are employed to accelerate uniform convergence and to better resolve boundary-layer behavior and curvature near restraints; this supplementary basis has been shown to reduce the number of terms required to achieve a given accuracy in coupled problems. In short, while the assumed-mode, energy-subtraction approach with polynomial admissible functions provides a compact route to free-vibration analysis of plates with openings, the improved Fourier-series framework with supplementary trigonometric terms achieves faster and more uniform convergence in coupled problems [69, 19]. The formulation in [19] was later extended to a mixed interface with opening; the extension couples two acoustic cavities on either side of the plate and does not include exterior free-field radiation [35].

A complementary method introduces local coordinates within the opening via the Independent Coordinate Coupling Method (ICCM). Because the system functions of the opening domains are assembled in their natural (local) coordinates, numerical integration remains straightforward for off-centre or rotated openings, and published comparisons report close agreement with finite-element solutions and bench-scale experiments across boundary cases [70, 71]. The formulation, however, is posed for free vibration; vibro-acoustic coupling and multi-opening can be challenging [70].

In [72], a discrete Rayleigh–Ritz formulation with variable-stiffness mapping is adopted to model the structural dynamics of plates with local material variations or openings. The plate is described within a mapped computational domain, where spatial changes in thickness and bending rigidity account for geometric discontinuities. Legendre-based admissible functions together with Gauss–Legendre numerical integration are employed to assemble the structural operators, enabling accurate prediction of natural frequencies and mode shapes for plates with complex boundary conditions using compact approximation bases.

In vibro-acoustic systems where a vibrating structural segment is adjacent to an opening, the acoustic transmission mechanism becomes mixed, as part of the acoustic energy is radiated through structural vibration while another portion is directly transmitted through the opening. Depending on the acoustic size of the opening and the required modelling fidelity, different approaches can be employed. High-resolution representations based on Rayleigh-type radiation formulations allow consistent prediction of exterior sound fields and associated radiation loading [73, 74], whereas reduced-order impedance descriptions provide efficient approximations for acoustically compact openings [75, 76].

For the structural radiation component, it is advantageous to employ radiation operators that condense the acoustic kernel into algebraic matrix forms suitable for numerical implementation. By discretizing the radiating surface into elemental patches, the radiated sound power can be

evaluated efficiently during parametric studies and actuator-placement optimization procedures. Such matrix-based radiation representations have been widely adopted in panel-radiation and active control investigations, offering a practical balance between computational cost and physical accuracy [77, 67].

To summarize, polynomial assumed modes can converge slowly, and ICCM is primarily formulated for free vibration and poses challenges when extended to strong vibro–acoustic coupling. For systematic studies, a practical workflow is to employ the improved Fourier–series basis with supplementary trigonometric terms for the structural displacement [19], paired with the opening sound–power estimation for rapid acoustic–power predictions. This combination enables fast and reliable assessment of radiated sound power across varying opening shapes and locations, as discussed further in this thesis.

Chapter 3

Methodology

3.1 The proposed model-based framework for thin plates with openings

In this work, the frequency response of a vibrating plate with openings is analyzed by varying the geometry and placement of the apertures on a mixed vibro–acoustic interface enclosing a rectangular cavity, as in Fig. 3.1. An analytical model presented in Paper A [78] employs hyperbolic and trigonometric admissible functions to achieve faster convergence than polynomial formulations [72]. Sound radiation is evaluated using the radiation resistance matrix, while complementary FEM and FEM–BEM models are implemented to visualize the coupled acoustic fields and to validate analytical predictions against experimental vibration and sound power measurements. Moreover, the effect of the noise source locations on the system response is addressed in Paper B [79].

The novelty of the proposed approach lies in establishing a direct link between an adaptable analytical model—capable of representing different configurations of the opening interface—and the corresponding acoustic response of the system. This framework is integrated with a COMSOL Multiphysics[®] FEM environment, enabling detailed visualization of the interaction between structural vibration and acoustic pressure fields, thereby enhancing both physical insight and design potential.

To illustrate the properties adopted in the analysis, a representative vibro–acoustic model is defined, reflecting typical configurations found in engineering applications. The system consists of a thin steel plate mounted over an air-filled rectangular enclosure. The plate material has a density of $\rho = 7850 \text{ kg/m}^3$, Young’s modulus $E = 210 \times 10^9 \text{ Pa}$, and Poisson’s ratio $\nu = 0.3$. Its dimensions are $a = 0.42 \text{ m}$, $b = 0.39 \text{ m}$, and $h = 0.001 \text{ m}$, corresponding to the length, width, and thickness, respectively. The plate is assumed to be fully clamped along its perimeter, while the remaining five walls of the enclosure are treated as acoustically rigid (sound-hard) boundaries. The enclosure cavity has a depth of $d = 0.4 \text{ m}$ and is filled with air of density $\rho_a = 1.2 \text{ kg/m}^3$ and sound speed $c = 343 \text{ m/s}$. Through the analytical formulation (the full derivation is provided in Paper A [78]), the total radiated sound power is expressed as

$$\bar{\Pi}_{\text{total}}(\omega) = \bar{\Pi}_{\text{rad}}(\omega) + \bar{\Pi}_{\text{loss}}(\omega), \quad (3.1)$$

where $\bar{\Pi}_{\text{rad}}(\omega)$ denotes the sound power radiated by the vibrating plate, and $\bar{\Pi}_{\text{loss}}(\omega)$ represents the component associated with acoustic leakage through the opening. The developed FEM model provides a detailed numerical counterpart to the analytical formulation, enabling both visualization and quantification of the coupled vibro–acoustic behavior. The model integrates the structural and acoustic domains through multiphysics coupling, allowing direct assessment of vibration patterns, sound pressure distribution, and radiation effects. The use of shell elements for the plate and pressure acoustics for the surrounding air ensures computational efficiency while preserving

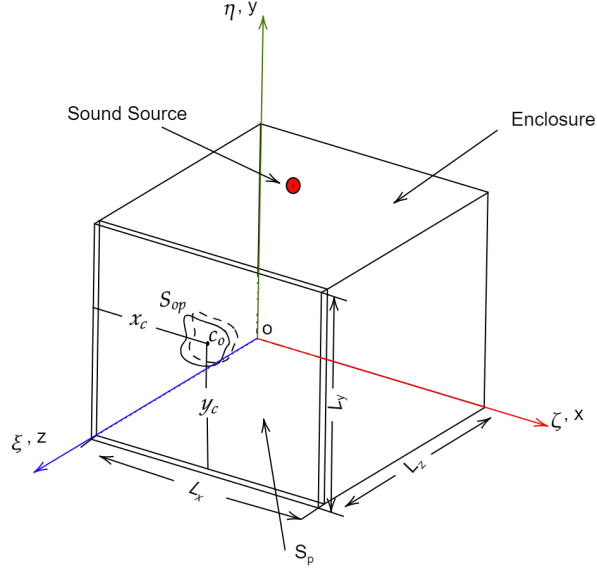


Figure 3.1: Enlarged system of the enclosure and coupled with a mixed interface of a vibrating rectangular plate with an arbitrary shaped opening interface S_{op} , where $(x_{c_o}, y_{c_o}) \in S_{op}$ and S_p is the structural plate interface with the primary noise source at (0.12 m, 0.29 m, 0.1 m). Reproduced from [78].

physical accuracy within the low- to mid-frequency range. Furthermore, the inclusion of boundary element representations extends the acoustic domain, eliminating reflections and approximating free-field conditions. Overall, the FEM framework serves as both a validation tool for the analytical model and a diagnostic platform for investigating how opening geometry influences structural response and sound radiation characteristics, as in Fig. 3.2.

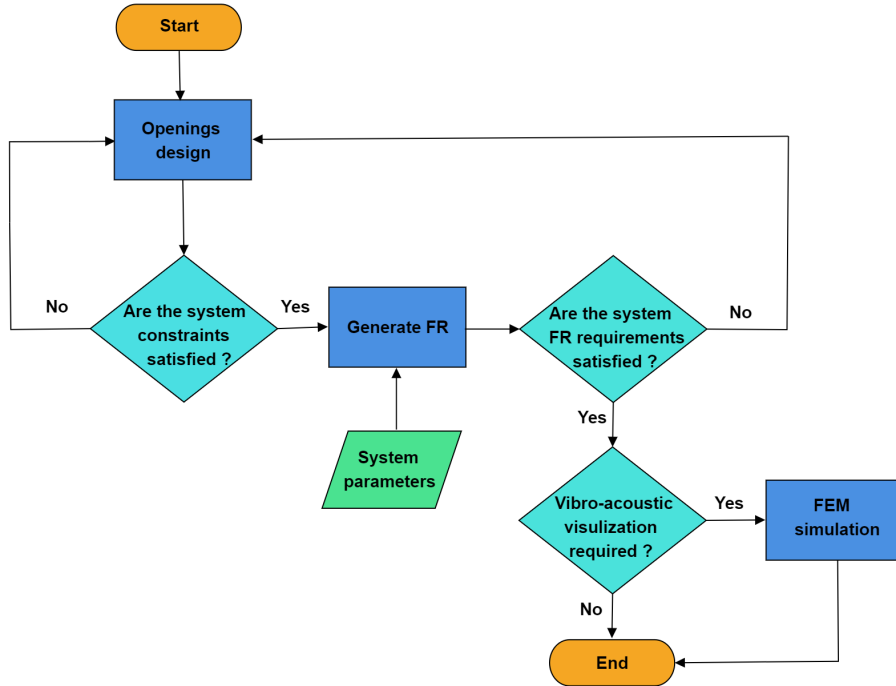


Figure 3.2: Schematic of the proposed shaping frequency response (FR) method of the vibrating plate with openings, correlating opening configurations with FR to satisfy system requirements within set constraints. Reproduced from Paper A [78].

3.2 The proposed optimization method

Optimizing structural openings in vibro-acoustic systems is essential for balancing functional requirements with acoustic performance across engineering applications. This work proposes an efficient optimization approach for designing openings in plate structures within vibro-acoustic systems [80], with the objective of enhancing acoustic performance. A genetic-algorithm framework is formulated to determine the optimal opening shapes and locations simultaneously, leveraging the analytical model presented in Section 3.1 within a master-slave software architecture. Additionally, the optimization problem is extended to include the localization of the inertial actuator on the mixed interface of the vibro-acoustic systems.

Significant efforts have been undertaken to optimize the wall structure design for vibro-acoustic systems. Using FEM, authors in [81] employed the plate stiffness as a variable to optimize the sound and vibration responses of the plate. Meanwhile, a multi-material topology optimization with variable thickness method was developed in [82] based on an alternating active-phase algorithm. In [32], the authors implemented a memetic algorithm (MA) to find the optimal positions of additional masses and ribs to shape the frequency response of the vibrating plate. The particle swarm optimization algorithm was applied by the authors in [33] to determine the best dimensions of the embossed panel section to reduce the sound power. Additionally, composite plates have been strongly considered for optimization, serving as noise barriers and vibration suppressors in vibro-acoustic systems in [83] and [84].

Recently, in [13], the authors utilized the Method of Moving Asymptotes (MMA) to optimize the topology of the core between two plates to increase the Sound Transmission Loss (STL). In [85], the authors employed a genetic algorithm (GA) with a surrogate model to construct a multi-objective optimization for rectangular plates with acoustic black holes (ABH) technique, which is efficient in passive vibration control for mid-higher frequency ranges [86, 87]. However, the literature reveals a gap in the optimal design of openings—allowing for variations in shape and location—on plate structures within vibro-acoustic systems. Openings introduce additional complexity in shaping the frequency response of the mixed interface [88] and must be treated explicitly to achieve high-performance designs. Additionally, to our knowledge, this is the first work to present an optimization tool for positioning inertial actuators on vibrating plates that contain structural openings in a vibro-acoustic system.

3.2.1 Optimization problem formulation

The optimization framework reformulates the analytical vibro-acoustic model to handle the inputs as a variable vector $\vec{v}_o$, which flexibly encodes the geometric parameters of the openings or actuators. Each opening is defined by a set of deterministic functions that describe its domain as

$$S_{op}(x, y) = \text{function}(\vec{v}_o), \quad (3.2)$$

where $\vec{v}_o$ includes the parameters corresponding to the position, orientation, and characteristic dimensions of the opening. For elliptical openings, $\vec{v}_o = [x_c, y_c, \theta_c, l_c]$, with (x_c, y_c) denoting the center, θ_c the rotation angle, and l_c a representative axis length. The same formalism allows switching between different deterministic shape functions (elliptic, rectangular, or triangular) depending on the design requirements.

The model outputs are reformulated to extract the maximum and average sound power in a defined frequency range (100–350 Hz), representing the system’s key acoustic indicators. These quantities form the basis of the cost function:

$$f_m(\vec{v}_o) = w_1 \cdot g_1(\vec{v}_o) + w_2 \cdot g_2(\vec{v}_o), \quad (3.3)$$

where g_1 and g_2 correspond to the maximum and average sound power, respectively, and $w_1 = 0.85$, $w_2 = 0.15$ balance their relative importance. The objective is to minimize $f_m(\vec{v}_o)$, thereby

reducing both the dominant peaks and overall levels of sound radiation. Geometric and structural constraints, such as the minimum spacing between openings and prescribed opening-to-solid ratios, are enforced to ensure feasibility.

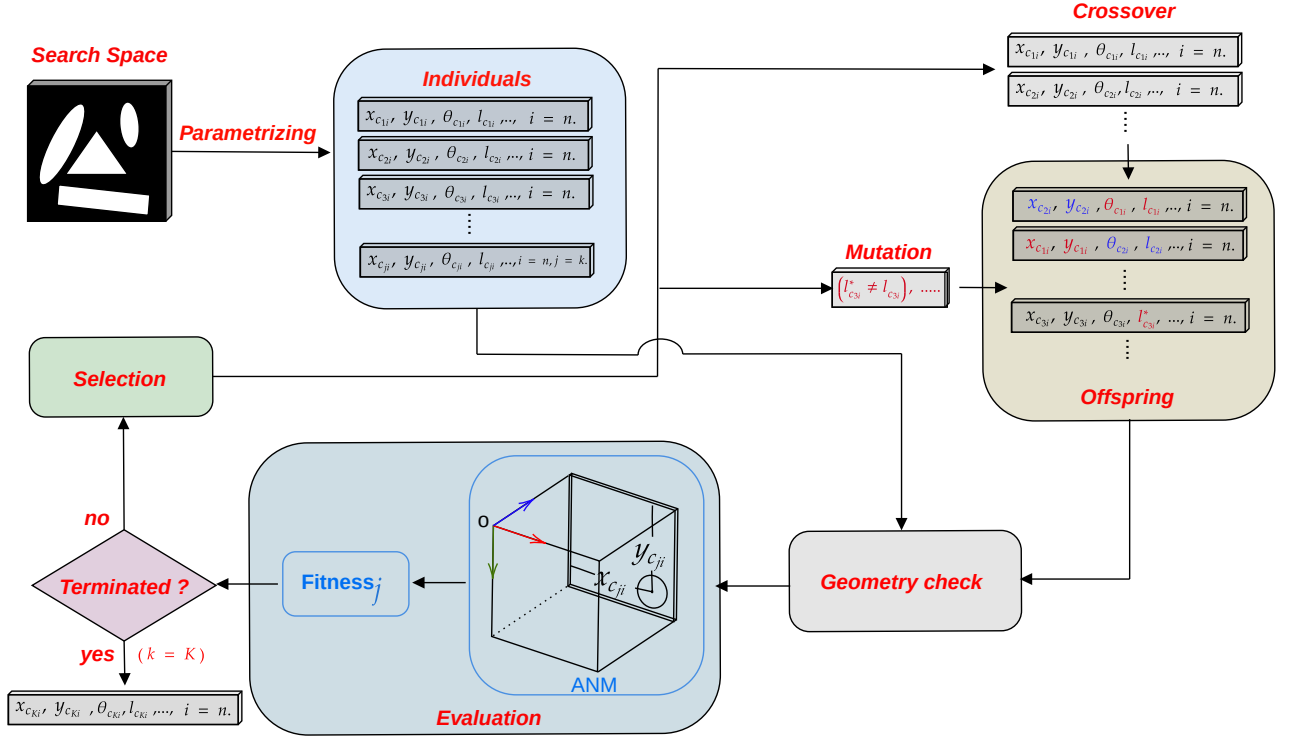


Figure 3.3: Genetic algorithm implementation. Reproduced from Paper C [80].

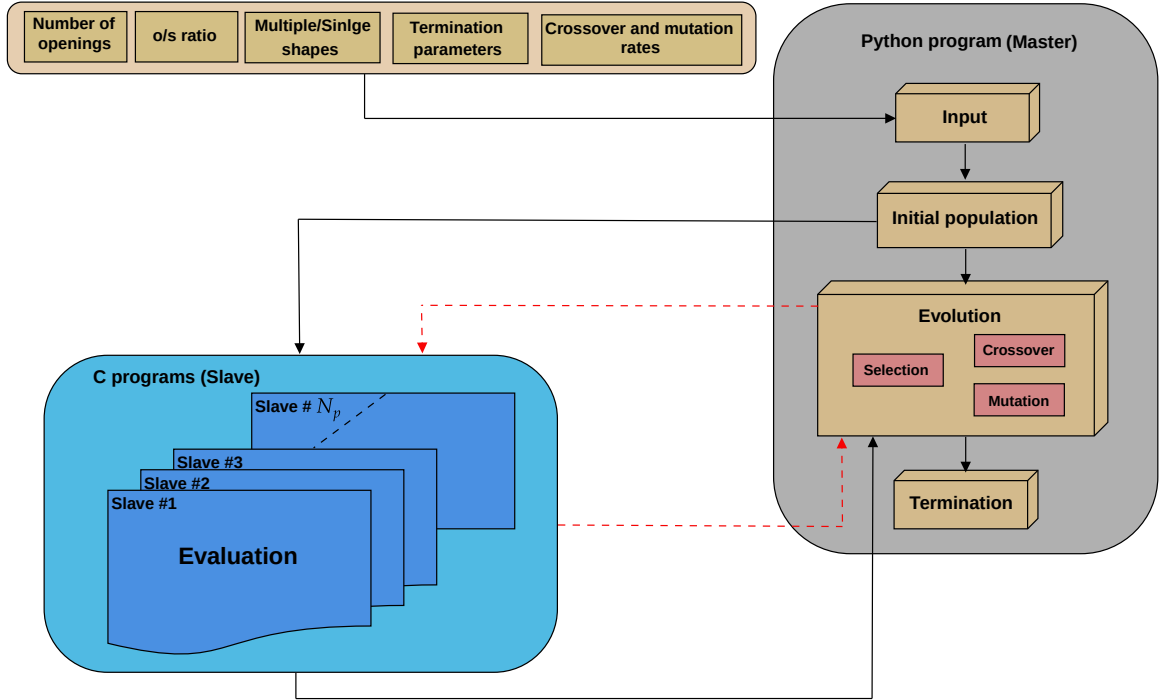


Figure 3.4: Master-slave GA architecture. Here, N_p is the number of slaves delegated by the master. Reproduced from Paper C [80].

The optimization employs a genetic algorithm (GA) (see Fig. 3.3) implemented in a master–slave software architecture (see Fig. 3.4). The GA, written in Python, governs the evolutionary process (selection, crossover, and mutation), while compiled C modules perform high-speed evaluations of the analytical model across multiple CPU cores in parallel. Convergence is reached when no improvement occurs over 20 of the 50 allowed generations, with crossover and mutation rates set to 0.3. This formulation enables efficient exploration of high-dimensional design spaces, producing optimal opening geometries and configurations that enhance the vibro–acoustic performance.

3.3 The proposed actuator placement methodology

ASAC systems have been extensively investigated in the literature to enhance noise reduction through optimized actuator placement. However, these studies typically neglect the influence of structural openings. Such openings modify the coupled structural–acoustic behavior by introducing acoustic leakage and additional transmission paths, thereby significantly affecting the radiated sound power of the system. Consequently, existing actuator placement strategies remain limited when applied to vibro–acoustic systems incorporating structural openings.

To address this gap, the work presented in Paper D [89] introduces an optimization framework for the placement of inertial actuators in vibro–acoustic systems with structural openings. The approach integrates an analytical sound power estimation model with linear–quadratic optimal control (LQOC) theory to compute the optimal actuator force gains. Each actuator configuration is then treated as an individual within a genetic algorithm (GA) framework, which optimizes the actuator locations based on the resulting sound power performance. The framework explicitly incorporates the mixed structural–acoustic interface and the acoustic leakage introduced by openings, providing a physically consistent and computationally efficient means of predicting and enhancing actuator effectiveness.

3.3.1 Optimizing the estimated sound power under actuator forces

Building upon the analytical vibro–acoustic framework described in Section 3.1, the model is extended to include inertial actuators mounted on the vibrating plate, treating it as a controlled acoustic barrier. The formulation accounts for both radiated and transmitted sound power through the mixed structural–acoustic interface, where the actuators are represented as locally concentrated mass regions ideally bonded to the plate surface. Using LQOC theory, the total sound power—comprising structural radiation and acoustic leakage—is minimized with respect to the actuator force gains, yielding a configuration for the optimal complex amplitudes. This allows each configuration to be evaluated within a genetic algorithm framework to determine optimal placement on the mixed interface.

The optimization framework proposed in Paper C [80] is extended to determine the optimal locations of inertial actuators mounted on the vibrating plate with the opening included. In this case, the design vector $\vec{v}_o$ represents the actuator configuration,

$$\vec{v}_o = [x_{a_1}, y_{a_1}, x_{a_2}, y_{a_2}, \dots, x_{a_n}, y_{a_n}], \quad (3.4)$$

where (x_{a_i}, y_{a_i}) specifies the coordinates of the i -th actuator on the plate surface. The analytical model integrates the structural and acoustic domains to evaluate how actuator placement influences the total radiated sound power under active excitation.

The optimization objective remains the minimization of the sound power level, which indirectly guides the actuators toward regions of high structural–acoustic coupling efficiency. The genetic algorithm follows the same evolutionary structure described earlier, ensuring that the actuator spacing satisfies geometric constraints and avoids boundary proximity that may affect

physical implementation. This formulation enables the derivation of actuator layouts that achieve broadband noise reduction through efficient excitation of key vibration modes.

The approach provides a theoretical means of identifying actuator locations that minimize the sound power and improve active control performance in systems incorporating structural openings. Although experimental one-to-one validation is challenging, this method offers valuable design guidance for practical implementation, as demonstrated in Paper D [89]. The optimal locations of the inertial actuators were evaluated using the ASAC approach and compared against alternative actuator configurations, as discussed in Section 4.2.

3.4 Experimental setup and model validation

The laboratory room measures 5.8 meters in length and 3.5 meters in width. While the walls are lined with sound-absorbing materials, the presence of equipment and other objects within the room gives it acoustic characteristics more representative of a typical real-world environment rather than those of a dedicated acoustic chamber. The experimental rig consists of a rectangular steel plate mounted to a concrete enclosure and equipped with a loudspeaker inside the enclosure. The loudspeaker, a Behringer EUROLIVE B208D, serves as the primary excitation source and is used to generate band-limited random noise.

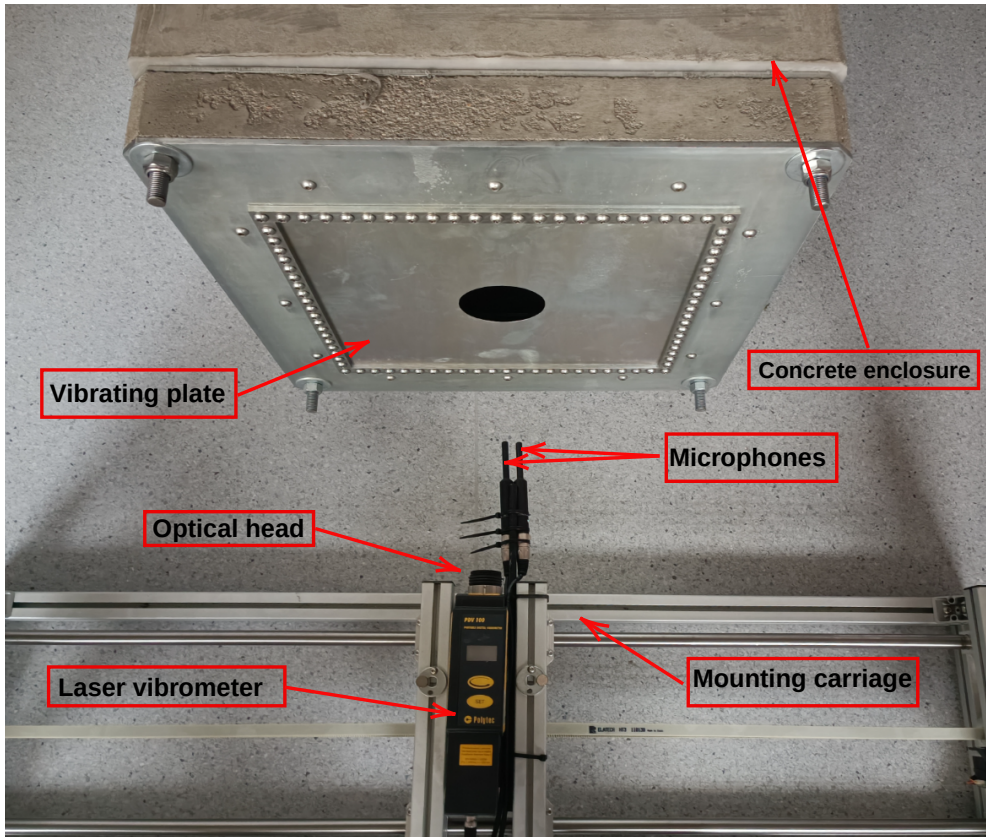


Figure 3.5: Overview of the experimental setup with a vibrating plate with a central circular opening, a laser vibrometer, and measurement microphones mounted on a carriage for two-directional positioning. Adapted from [78].

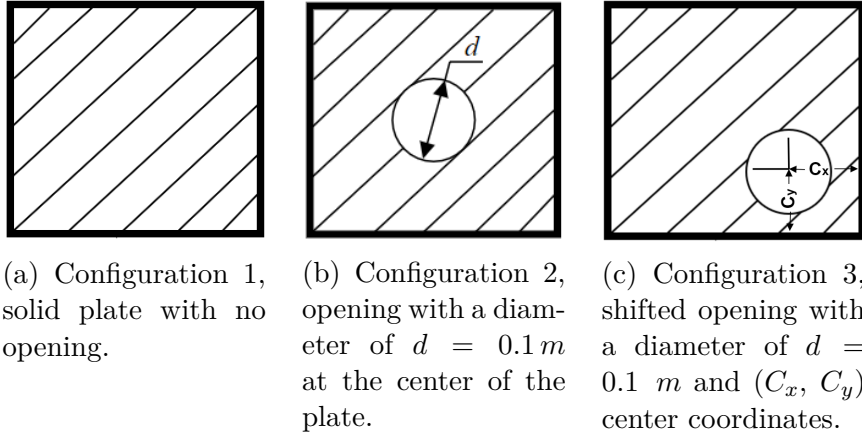


Figure 3.6: Vibrating plate with different opening configurations. Reproduced from [78].

The concrete walls of the enclosure provide a high degree of sound attenuation; hence, the majority of the acoustic energy that exits the enclosure is transmitted through the steel plate or the opening, depending on the studied configuration. Two Beyerdynamic MM1 measurement microphones are used to capture the sound pressure at a distance of 0.1 m from the plate and mounted on a carriage along with a laser vibrometer in Fig. 3.5. The carriage is designed to control the movement of the vibrometer along with the microphones in the horizontal and vertical directions to capture the measurements at the required positions.

Table 3.1: The first 11 natural frequencies comparison in (Hz), Exp: Experimental results, ANM: Analytical Model, FEM: Finite Element Model, and NaN: stands for a mode not distinguished in the experiment. Reproduced from [78].

Mode	Configuration 1			Configuration 2			Configuration 3		
	Exp	ANM	FEM	Exp	ANM	FEM	Exp	ANM	FEM
1	54	54.9	58.2	58	57.4	55.9	51	52.9	52.4
2	103	106.8	105.6	NaN	101.9	101.7	97	104.1	106
3	117.5	116.9	115.5	111	112.1	110.8	112	116.1	116.7
4	163.5	165.1	162.5	149	151.4	156.8	154	163.8	163.4
5	184.5	189.9	187.5	180	183.2	186.2	183	186.7	187.7
6	214	212.9	211.5	220	228.5	225.1	206	212.3	213.9
7	242	244.3	240.6	225	229.1	234.5	241	237.3	243.9
8	259.5	258.9	255.1	247	244.6	249.1	280	272.8	277.2
9	295	299.6	298.1	280	288.7	284.7	290	295.2	295.9
10	337	335.5	329.7	328	330.9	325.1	329	329.2	335.5
11	343	341.7	339.9	333	334.1	340.9	342	338.2	342.2

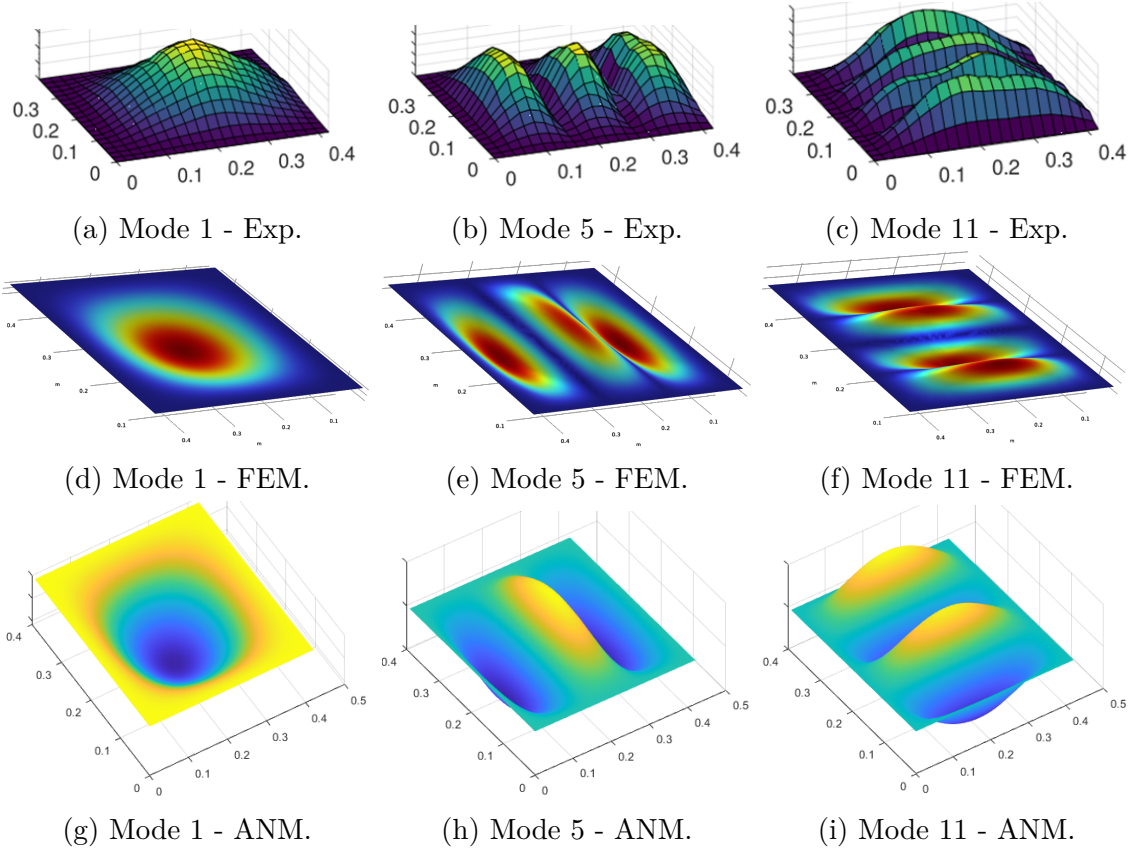


Figure 3.7: The 1st, 5th, and 11th mode shapes for Configuration 1, dimensions in [m]. Reproduced from [78].

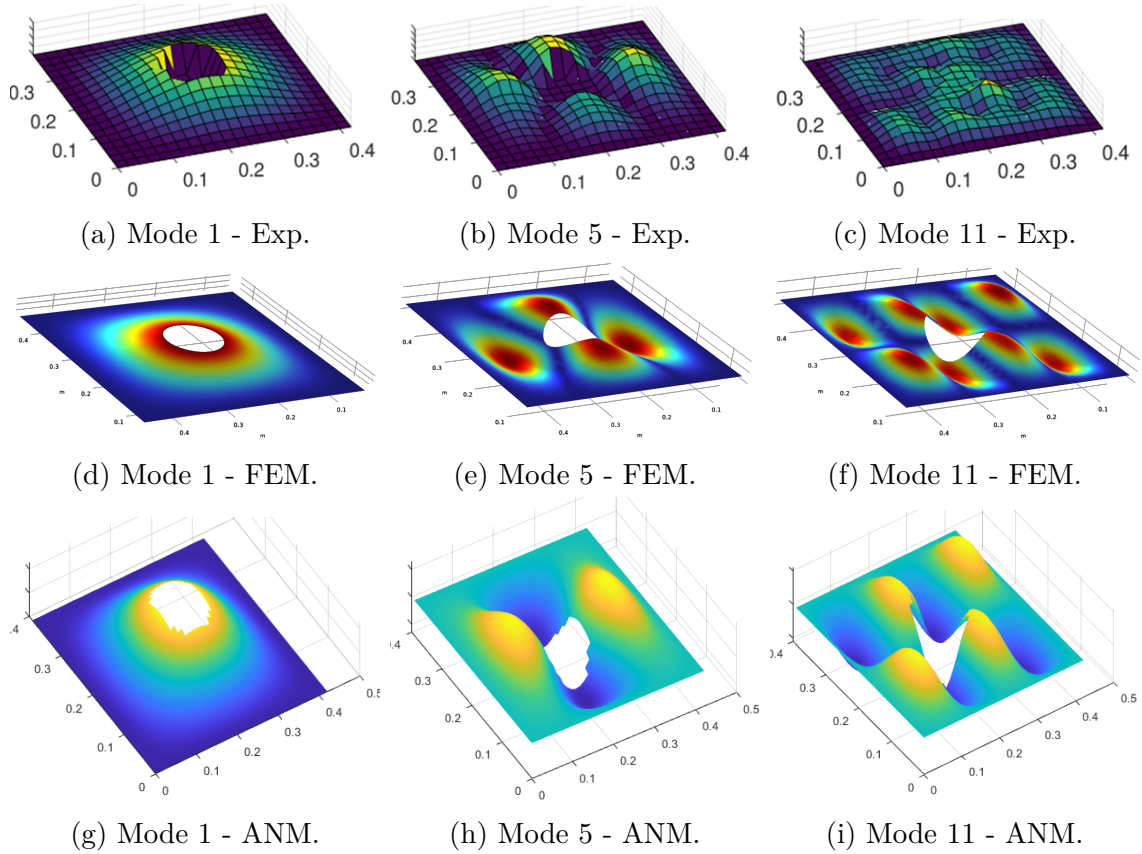


Figure 3.8: The 1st, 5th, and 11th mode shapes for Configuration 2, dimensions in [m]. Reproduced from [78].

The plate's vibration was measured across a consistent grid of 22×20 points, or 440 total points, spaced at intervals of 0.02 m. The estimated signal energy obtained for specific frequencies of structural resonances at all measurement locations was aggregated to extract the mode shapes. The results obtained from the simulation of FEM and calculations of analytical models are compared to the measurements from the experimental rig for the three different configurations, as shown in Fig. 3.6. In configuration 1, the solid plate acts as a benchmark for maximum sound insulation, making it ideal for applications demanding high sound reduction. As a benchmark, it is also suitable for comparing configurations within the proposed approaches and the experimental results. Configuration 2 places a circular opening with a diameter of $d = 0.1$ m in the center of the plate, which is an advantage for accommodating central tubes or facilitating symmetric interaction with the surroundings, which is necessary in different engineering applications. In configuration 3, the opening's center has $C_x = 0.105$ m and $C_y = 0.0975$ m as coordinates relative to the adjacent corner of the opening.

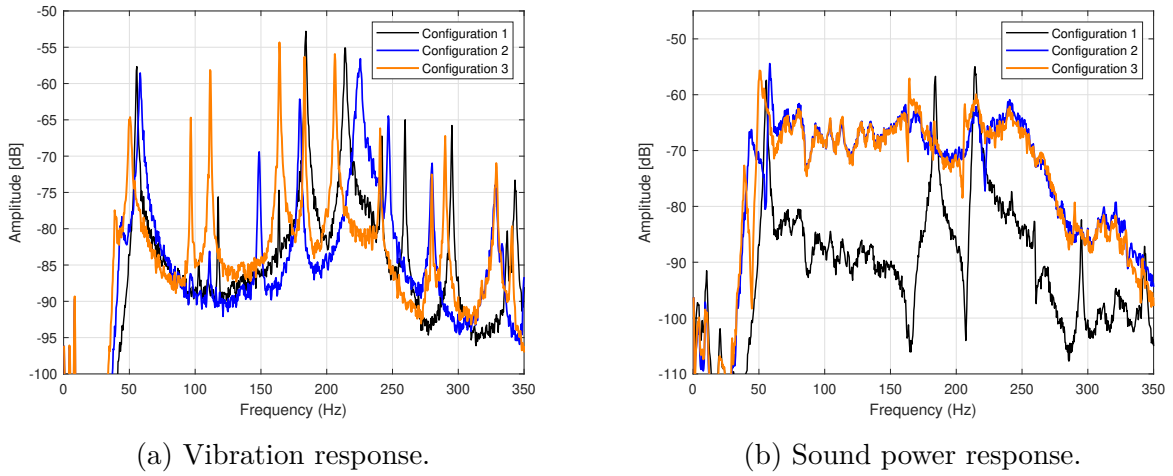


Figure 3.9: Sound power and vibration response from the experimental measurements. Reproduced from [78].

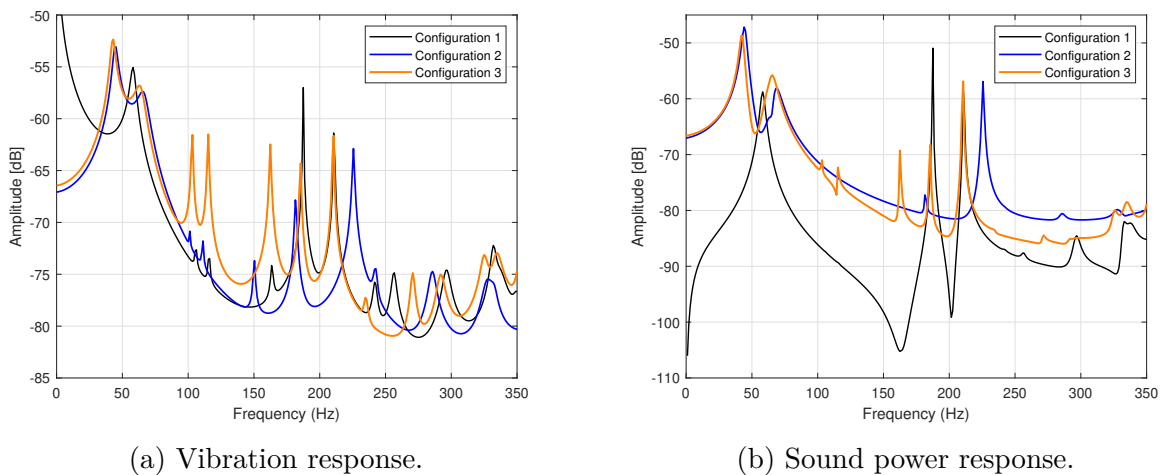


Figure 3.10: Sound power and vibration response from the analytical model. Reproduced from [78].

This configuration enhances the uniform distribution of stress across the plate for applications that incorporate additional components, such as volumetric actuators in active noise control [32]. Table. 3.1 compares the natural frequencies of the first 11 modes, covering a frequency range

characteristic of many noise-generating devices. However, this assumption does not limit the considerations; the number of modes can be increased if needed. Figs. 3.7 and 3.8 present the 1st, 5th, and 11th mode shapes for configurations 1 and 2, respectively. The mode shapes of the additional configuration, as well as the coupled system mode shapes obtained from the FEM, are reported in Paper A [78]. In addition, the vibration and sound power responses obtained from the experiment and the analytical model are shown in Figs. 3.9 and 3.10, respectively. The results reveal a substantial correlation between the experimental measurements and the computed values, covering aspects such as natural frequencies, mode shapes, and both vibration and sound power responses.

3.4.1 Laboratory setup for active control

The laboratory setup is adjusted to be suitable for active control experiments, as shown in Fig. 3.11. Three Beyerdynamic MM1 error microphones are used for the active control system with three DAEX32EP-4 inertial actuators.

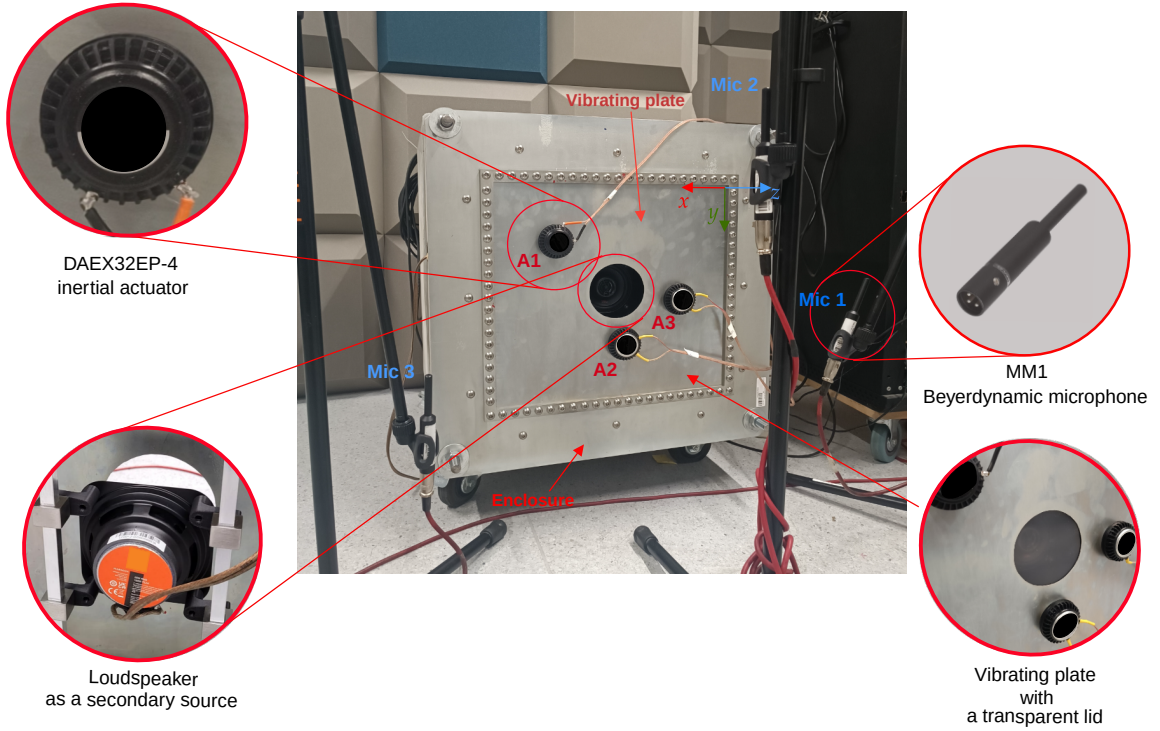


Figure 3.11: Overview of the experimental rig with active control components showing a concrete enclosure with a vibrating plate that contains a central circular opening of radius 0.05 m. The steel plate spans 0.42 m in x , 0.39 m in y , and has a thickness of 0.001 m in $-z$. Actuators (A1, A2, and A3) are mounted on the plate at (0.32 m, 0.1 m), (0.2 m, 0.29 m), and (0.1 m, 0.2 m), respectively. Microphones (Mic 1, Mic 2, and Mic 3) are positioned at the coordinates $(-0.18$ m, 0.12 m, 0.25 m), $(0.21$ m, -0.1 m, 0.4 m), and $(0.53$ m, 0.24 m, 0.25 m), respectively. The loudspeaker, used as a secondary source, is centered behind the opening. Reproduced from [89].

For configurations incorporating a secondary loudspeaker source, a JBL Stage2 424 speaker is used and positioned centrally behind the opening, as in Fig. 3.11. To test the system with a reduced acoustic leakage effect, a plastic lid is adhesively bonded to the steel plate. The lid is made of semi-rigid polyvinyl chloride (PVC) with a density of approximately 1300 kg/m^3 . In this work, the Analog Device SC598 with SHARC+ core is used along with EVSOMCRR-EZKIT development board [90]. Thus, the system is equipped with the ADAU1979, a 4-channel, 24-bit

Sigma-Delta ADC, and the ADAU1962, a 12-channel, 24-bit differential output DAC. For the selected ADC and DAC with a sampling frequency of 48 kHz, the group delay is 1 ms. Anti-aliasing IIR filters are designed for downsampling from 48 kHz to the downsampling frequency ($F_{s,D}$) of 3 kHz, followed by upsampling. Based on the analysis of the IIR filter characteristics, a 20-sample group delay is introduced into the system during both downsampling and upsampling, which is equivalent to 0.83 ms. The Synchronous Serial Peripheral Port (SPORT) is used for data communication protocol. The SPORT non-blocking read/write operation utilizes the direct memory access (DMA) peripheral, allowing data to be transmitted from one peripheral to another without additional CPU load. The buffer must remain inaccessible during the DMA transfer to prevent data corruption and can only be accessed once it is fully filled. When the SPORT finishes filling the buffer with ADC data, it triggers an interrupt, which is handled by the SPORT callback function. To align with the downsampling/upsampling architecture, each buffer contains 16 samples per channel, causing the callback function to be invoked consistently at a rate of 3 kHz. Thus, the output buffer being generated by the callback function will only be sent to the DAC in the next timeframe. This will cause an additional delay of $\frac{1}{F_{s,D}}$ to be introduced in the system, which is approximately 0.33 ms. This makes the total latency of the system (from ADC to system to DAC) to be around 2.16 ms, which is borderline acceptable for active control implementation [2].

For active control, a feedforward multichannel FxLMS algorithm with a leakage factor is used due to its simplicity and robustness [2]. Furthermore, the control system incorporates normalization based on the power of the single reference signal. For more details on the hardware implementation, refer to Paper D [89].

Chapter 4

Results and discussion

This chapter examines how opening design and actuator placement shape the performance of vibro-acoustic systems, with emphasis on noise barriers that include openings or plates made of different materials. First, the passive behavior is analyzed and optimized to quantify how the size, shape, and location of openings redistribute stiffness and mass on the mixed interface and, through the associated acoustic leakage, modify the vibration and sound-power response. Next, active strategies are evaluated: (i) Active Structural Acoustic Control using inertial actuators placed at the optimized locations identified in Section 3.3, and (ii) conventional Active Noise Control applied at the opening. The comparative tests motivate a dual-actuator-type architecture that combines both paths. Finally, a transparent-patched barrier is assessed, and the placement optimization is extended to one, two, and three inertial actuators on that panel, highlighting actuator synergy and the influence of spatial layout on broadband performance.

4.1 Passive noise reduction

Impact of system design variation

The vibro-acoustic behavior of the system was analyzed to understand the effect of the opening shape and location on both vibration and sound power responses. As shown in Fig. 4.1a, in contrast to the solid plate (configuration 1), configuration 4 is a benchmark for comparison, indicative of the highest sound power leakage in scenarios where the plate is absent.

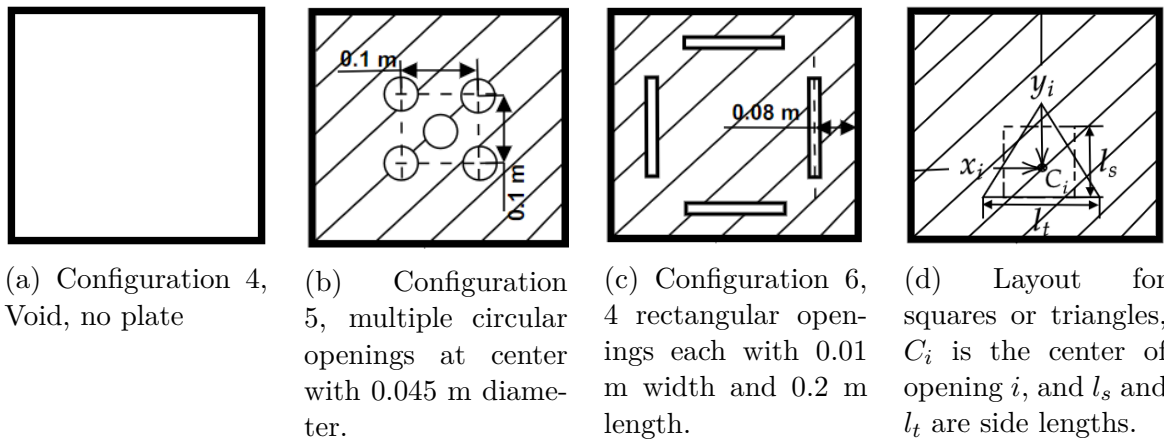


Figure 4.1: Various plate configurations evaluated for vibro-acoustic performance. Reproduced from [78].

In Fig. 4.1b, configuration 5 features five centralized circular openings with a total opening area equal to this of configurations 2 and 3. Fig. 4.1c presents configuration 6 with four rectangular

openings, each equidistant from the edges and aligned with the center of the corresponding edge, to examine the influence of different opening interface contours and their effects near the edges. Configurations 5 and 6 can have an advantage for applications where precise cable routing is essential, allowing for better organization and airflow around the cables for enhanced cooling performance.

Fig. 4.2 shows how the plate vibrates at different frequencies from 1 to 350 Hz. Compared to Configuration 1, Configuration 2 shows a shift in resonant frequencies due to the reduction in stiffness. Configuration 3's vibration resonances vary slightly in amplitude, as the shifted opening affects the plate's modal behavior unevenly, causing a significant increase in the resonance between 100 and 200 Hz.

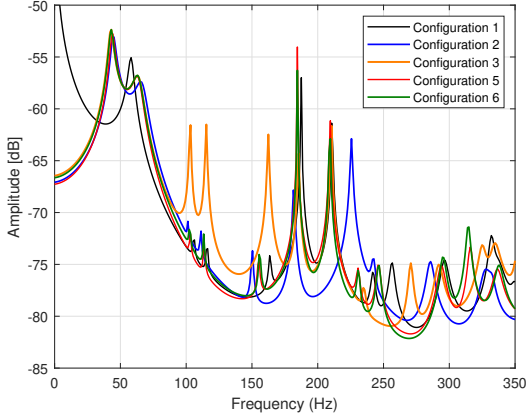


Figure 4.2: Vibration response of the plate with different configurations. Reproduced from [78].

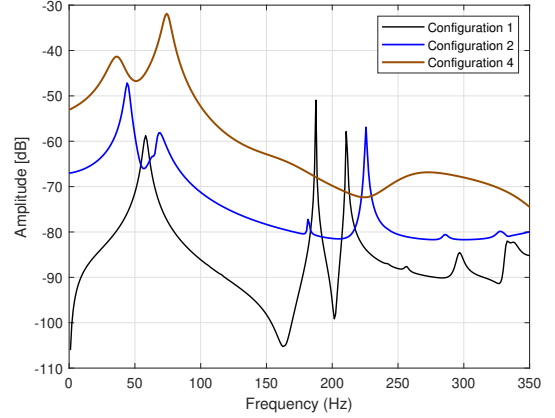
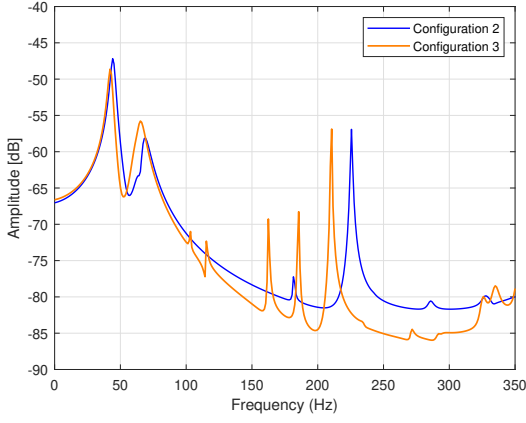


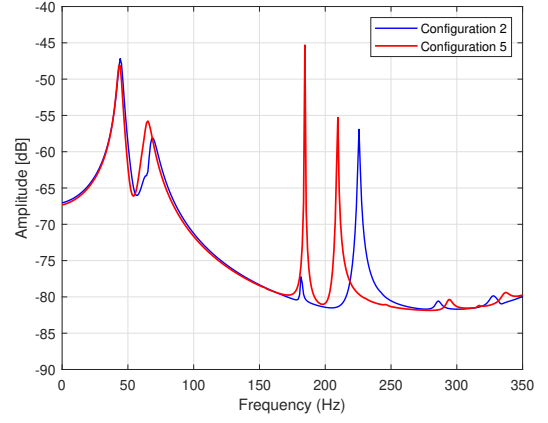
Figure 4.3: Sound power response of the system for Configuration 1, 2, and 4. Reproduced from [78].

Configuration 5, characterized by multiple circular openings, presents closely spaced resonances due to the stiffness characteristics of the plate's central region. The results for Configuration 6 suggest it achieves damping effectiveness comparable to Configuration 1. This is because, within the specified frequency range, the mode shapes of the plate remain largely uninfluenced by the location of the openings. These openings are situated close to the nodal lines where the mode shapes exhibit minimal movement, thus not significantly affecting the vibrational characteristics of the plate.

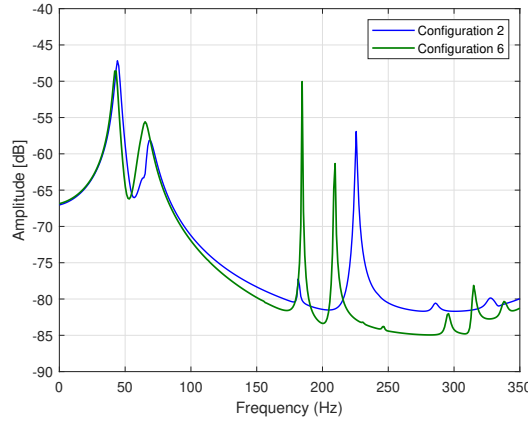
The sound power response, as presented in Fig. 4.3, shows that in Configuration 4, the system has no plate, hence it exhibits the highest sound power levels across the frequency spectrum. Meanwhile, Configuration 1 has no leakage and displays the lowest sound power levels, with a reduction of around 35 dB for frequencies below 187 Hz and approximately 25 dB for higher frequencies. Comparing the configurations with openings, each of them has a total opening area of 0.007854 m^2 , hence the plate mass reduction is the same in all configurations but the stiffness distribution will be different. Configuration 2 shows a middle performance between Configurations 1 and 4, with a reduction around 17 dB across all the spectrum response, and it is used as a benchmark to test the other configurations with the openings in Fig. 4.4. In Fig. 4.4a, the sound power response for configurations 2 and 3 reveals the impact of opening placement in relation to the system mode shapes. Configuration 2's central opening aligns more with the plate's anti-nodal areas for different mode shapes, increasing the leaked sound power, while Configuration 3's shifted opening interacts more with nodal points, reducing the sound power at frequencies above 100 Hz but also amplifying the resonance amplitudes for both vibration and sound power response.



(a) Sound power response for configurations 2 and 3.



(b) Sound power response for configurations 2 and 5.



(c) Sound power response for configurations 2 and 6.

Figure 4.4: Sound power responses of the plate for various configurations. Reproduced from [78].

Fig. 4.4b shows that the concentration of multiple openings in Configuration 5's central area does not impact the sound power response across the frequency range. However, this concentration increases the stiffness of the central region, which amplifies the resonance in the mid-range. In Fig. 4.4c, the rectangular openings of Configuration 6 lead to lower sound power across most frequencies, but this configuration amplifies the resonance around 185 Hz. The location of the openings allows the plate to vibrate in a way that is similar to Configuration 1, as the stiffness is spread out more evenly and the openings align more with the anti-nodal areas of the plate for the majority of the system's mode shapes. Additionally, the layout, as described in Fig. 4.1d and Table. 4.1, is designed to accommodate multiple openings. The shape of the openings can be made by combining squares, as seen in Configurations 7 and 8, or equilateral triangles, as in Configurations 9 and 10.

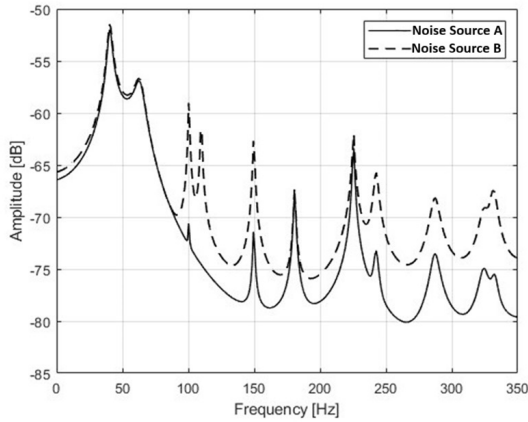
The above results indicate that, with the assumption of equal opening area, the configurations with a continuous perimeter have their maximum resonating frequency at a lower amplitude than the other configurations. As shown in Table. 4.1, Configurations 2, 3, 8, and 10 resonate with less amplitudes compared to the others. In addition, maintaining the same opening area - excluding the solid and void configurations - results in similar average sound power levels indicating the impact of sound power loss. The configurations examined-so far-are examples of how the design of openings—both their shape and location on vibrating plates—can effectively shape the frequency responses (FR) to comply with the requirements of vibro-acoustic systems while respecting the opening constraints, such as the total opening area.

This demonstrates the potential of implementing the proposed models into a methodology to

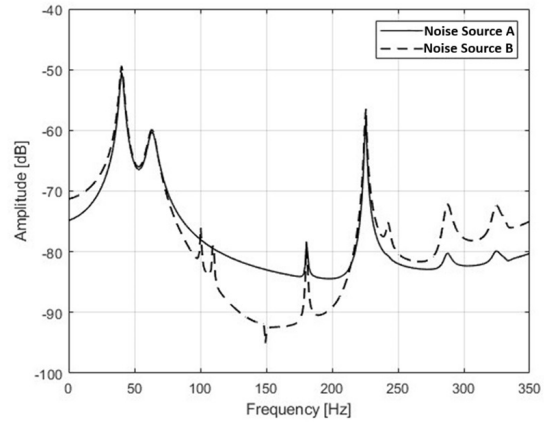
Table 4.1: Comparison table of configurations, PA: denotes the peak amplitude at resonance frequency (RF) in the sound power response above 80 Hz, AVG: represents the average of this response for the same range. Reproduced from [78].

Configuration	Opening	Design dimensions [m]				RF [Hz]	PA [dB]	AVG [dB]
		(x_1, y_1)	(x_2, y_2)	(x_3, y_3)	l_s, l_t			
1		See Fig. 3.6a				188	-50.9	-88.11
2		See Fig. 3.6b				226	-56.9	-78.08
3		See Fig. 3.6c				211	-56.87	-80.28
4		See Fig. 4.1a				80	-37.30	-65.15
5		See Fig. 4.1b				185	-45.31	-77.93
6		See Fig. 4.1c				185	-50.05	-79.88
7		(0.07,0.195)	(0.21,0.195)	(0.35,0.195)	0.051	185	-54.62	-80.69
8		(0.21,0.195)	(0.261,0.195)	(0.195,0.195)	0.051	217	-61.4	-80.18
9		((0.07,0.195)	(0.21,0.195)	(0.35,0.195)	0.0775	185	-51.376	-80.69
10		(0.21,0.195)	—	—	0.1344	222	-59.03	-80.1

shape the frequency response of vibro-acoustic systems in various engineering applications that use vibrating plates with openings as noise barriers. For additional information, refer to Paper A [78].



(a) Vibration response.



(b) Sound power response.

Figure 4.5: Comparison of vibration and sound power response for noise source A vs noise source B. Adapted from [79].

Employing an optimization algorithm to fine-tune the design parameters of the openings is a viable approach. To examine the effect of the relative position of the opening to the primary noise source inside the enclosure, Configuration 2 is evaluated under several noise source locations: A (0.16, 0.35, 0.1) m, B (0.36, 0.33, 0.34) m, and C (0.1, 0.06, 0.2) m (for the coordination system, refer to Fig. 3.11). Fig. 4.5 illustrates the consequence of moving the noise source closer to the vibrating plate at location B, manifesting in amplified vibration response amplitude across all modes, along with a reduction in average sound power below 226 Hz. In Fig. 4.6, a general location C was selected, which exhibited amplified vibration amplitudes across the entire frequency range, accompanied by an increase in sound power above 226 Hz. These results underscore the influence of the noise source's relative position on the system's ability to excite different modes.

Moreover, they highlight how the relative positioning of the opening and the noise source impacts the system's sound power response.

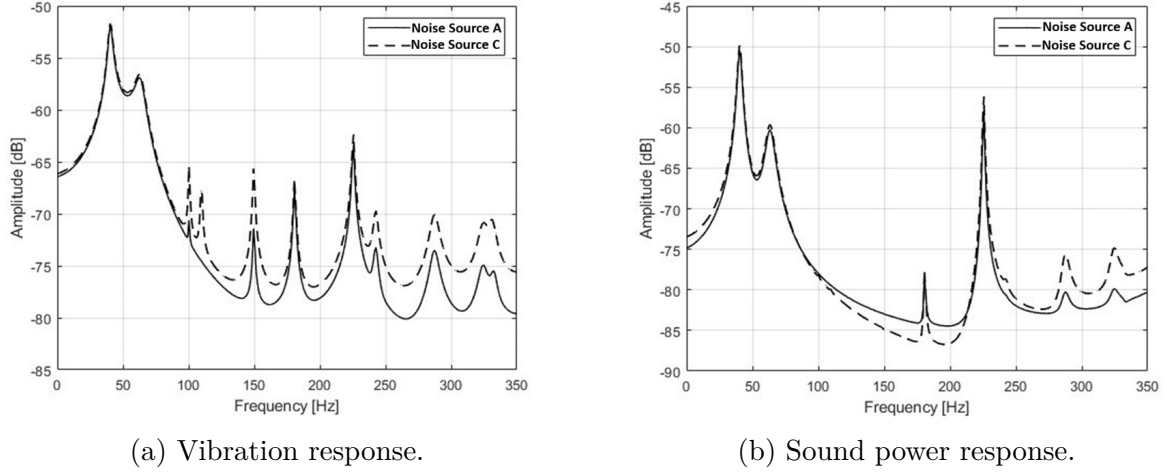


Figure 4.6: Comparison of vibration and sound power response for noise source A vs noise source C. Adapted from [79].

4.1.1 Optimal opening design

Vibro-acoustic systems can be defined with different opening requirements. Systems with varied opening to solid (o/s) ratios, with a different number of openings, or with diverse opening shapes are addressed in this section. For instance, various engineering applications require different levels of ventilation while maintaining specific noise levels. By adjusting the (o/s), the airflow can be controlled, and the proposed method can be used to determine the optimal design parameters for the openings, allowing for passive reduction of sound power. This section analyzes nine different systems to address a variety of opening requirements. Three (o/s) are examined: 30%, 15%, and 5%. Each has a benchmark of a least fitness individual with one opening. The optimization method is tested to find the optimal opening design in case of one, two or three openings. Dividing the required opening area into more than one opening is beneficial for engineering systems that require device monitoring openings or enhanced cooling distribution.

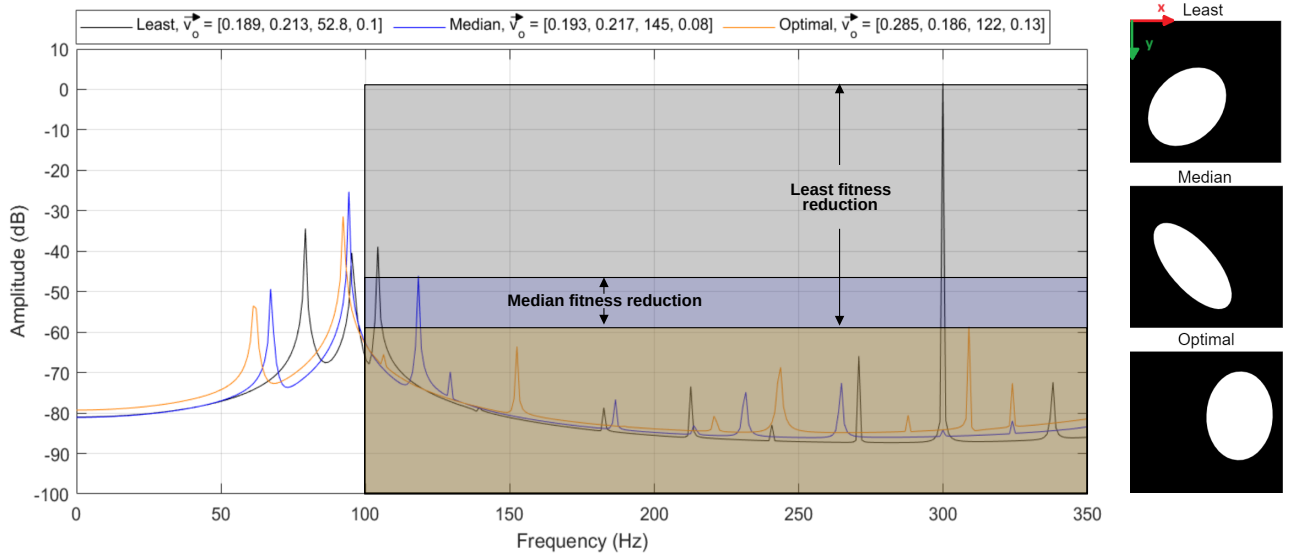


Figure 4.7: Sound power response for least, median, and optimal fitness individual opening's design with $n = 1$ and $o/s = 30\%$. Reproduced from [80].

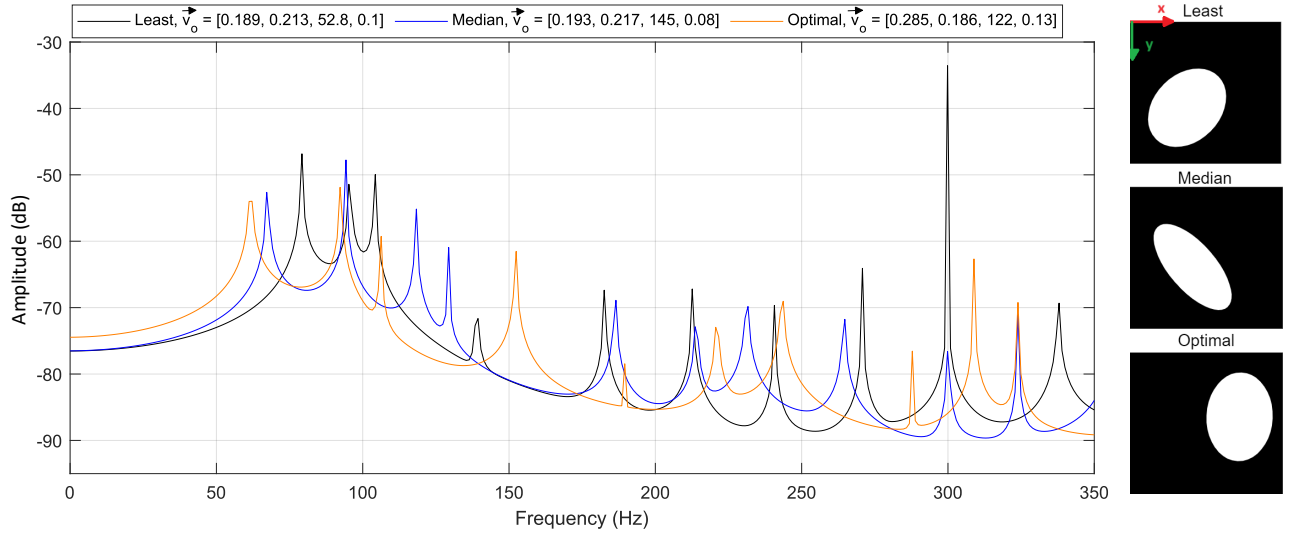


Figure 4.8: Vibration response for least, median, and optimal fitness individual opening's design with $n = 1$ and $o/s = 30\%$, calculated by averaging the displacement over the vibrating plate. Reproduced from [80].

The elliptical shape is favorable for openings due to its manufacturing advantages, particularly the avoidance of sharp corners that are difficult to fabricate. Fig. 4.7 shows the impact of implementing the design parameters obtained from the GA algorithm for a system with $n = 1$ and $o/s = 30\%$ design requirement, resulting in a 12 dB reduction in sound power amplitude compared to the median individual design and 60 dB compared to the least fitness design. The sound power response is the combination of the radiated sound from the plate vibration and the leakage sound from the openings interface. However, the maximum peaks of sound power responses are more related to the radiation component than the leakage power. This is due to the high vibration amplitudes at the resonance frequencies of the system response. Fig. 4.8 depicts the vibration response of the openings under consideration. This highlights that maintaining the same total opening area for the least, median, and optimal designs ensures consistent mass reduction, while the distribution of stiffness varies.

Table 4.2: Comparison table of optimization results for different design requirements. Reproduced from [80].

n	1			2			3		
	o/s [%]	Design	Fitness (dB)	o/s [%]	Design	Fitness (dB)	o/s [%]	Design	Fitness (dB)
30			-11.1	30		-11.1	30		-11.1
			-51.55			-57.5			-48.8
			-62.12			-62.49			-55.5
15			-35.2	15		-35.2	15		-35.2
			-60.06			-51.3			-62.3
			-72.1			-68.48			-71.26
5			-27.7	5		-27.7	5		-27.7
			-59			-53.98			-56.9
			-71.4			-71			-70.83

The placement of an opening in the anti-nodal area alters the plate's stiffness distribution, reducing the local resistance to deformation and leading to higher displacements. As a result,

the plate exhibits higher vibration amplitudes under the same force [91], particularly at modes coinciding with the anti-nodal area containing the opening. This leads to increased amplitudes in the system’s sound power response. Thus, the optimization is likely finding the best locations of the openings that give the best distribution of the stiffness to satisfy the objective of the cost function. Several systems with a variety of requirements have been studied in Paper B [80]. Table. 4.2 presents an overview of the fitness values for the median and optimized designs and the least fitness individuals for the proposed systems. By defining different systems’ requirements, including opening area, number of openings, and shapes, the introduced optimization method can effectively reduce the sound power response of vibro-acoustic systems in engineering applications containing vibrating plates with openings for various purposes. Additionally, if for a certain application a more regular arrangement of the opening is demanded, this can be easily obtained using the presented method. However, the pursuit of a greater reduction in the system’s sound power response presents a challenge. Further exploration of technologies such as active control may address this challenge.

4.2 Active solution

The active evaluation starts with ASAC. Using the placement methodology in Section 3.2, inertial actuator locations are optimized for the first time on vibrating plates that incorporate an opening. The optimal locations for the inertial actuators, as mentioned in the Section 3.3, are assessed using the ASAC approach and compared against alternative actuator arrangements as illustrated in Fig. 4.9. Experimental investigation of the three actuator sets demonstrates that Set 1 yields the best performance, as indicated by the mean squared error (MSE) measured at the three microphones and illustrated in Fig. 4.10.

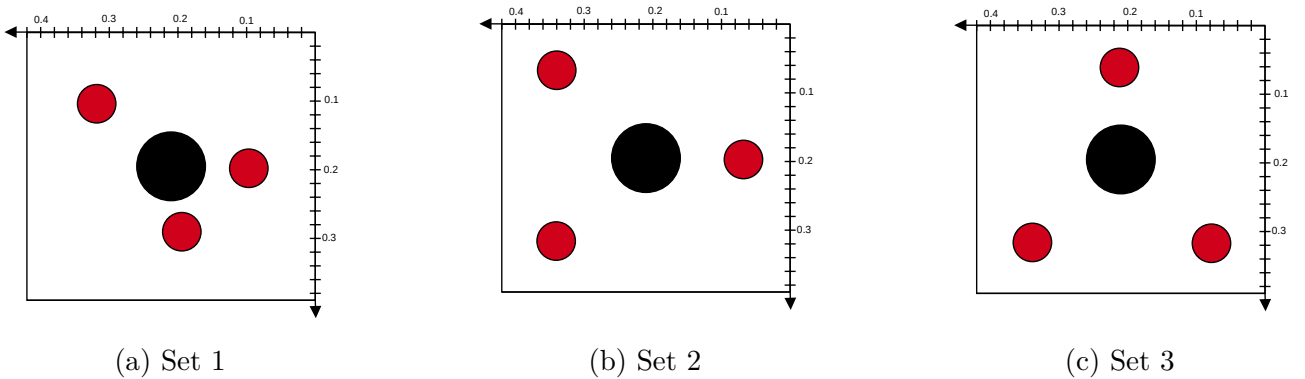


Figure 4.9: **(a)** Set 1: Actuators positioned based on the analytical model. **(b)** Set 2: Actuators localized toward the left half of the plate. **(c)** Set 3: Positioned with a similar concentration toward the bottom half of the plate. Reproduced from [89].

In passive configurations, placing actuator masses near the opening reduces MSE below 200 Hz but leads to increased peaks between 200 and 300 Hz (Fig. 4.10a). This behavior is attributed to the opening’s location, which coincides with regions of high modal displacement at low frequencies [80]. Adding mass in this region increases the local inertia and alters the modal participation of these modes, making them less efficiently excited [91]. If a purely passive solution is required, the model can be adapted by eliminating the actuator forces and optimizing the locations of the added masses. However, this approach is primarily effective for shifting acoustic resonances from one frequency band to another. In such cases, the cost function may be modified to target narrowband noise reduction objectives [32]. This change leads to a reduction in vibration energy near the opening and thus a decrease in the MSE associated with sound radiation at lower

frequencies. When ASAC is applied, Set 1 consistently yields the lowest MSE across nearly the entire frequency range. In the 100–200 Hz band, it achieves an average reduction approximately 5 dB greater than that obtained with Sets 2 and 3. This frequency range is particularly critical, as it corresponds to dominant the acoustic leakage through the opening, which is discussed further in this section. The placement of actuators in Set 1, near the edge of the opening, facilitates a greater displacement range and thus enhances control effectiveness. In contrast, Sets 2 and 3 with actuators positioned near the clamped edges of the plate, where higher structure stresses result in limited displacement. These factors collectively reduce their capacity to control vibrations, suppress radiated sound, and generate effective anti-noise signals. These findings indicate the efficacy of the proposed analytical model in guiding optimal actuator placement for improved noise reduction. Based on this optimization, the inertial actuators' arrangement in Set 1, also shown in the experimental setup (Fig. 3.11), is hereafter referred to as the ASAC configuration throughout this work. To assess the performance of conventional active control strategies, comparative experiments are conducted under identical acoustic and structural conditions. The enclosure was excited using a loudspeaker generating band-limited white noise in the 50–300 Hz range. The adaptive control algorithm employs control filters of length 400 and secondary path models of length 300, with a step size of $\mu = 1 \times 10^{-6}$ and a regularization factor of $\beta = 1 \times 10^{-6}$.

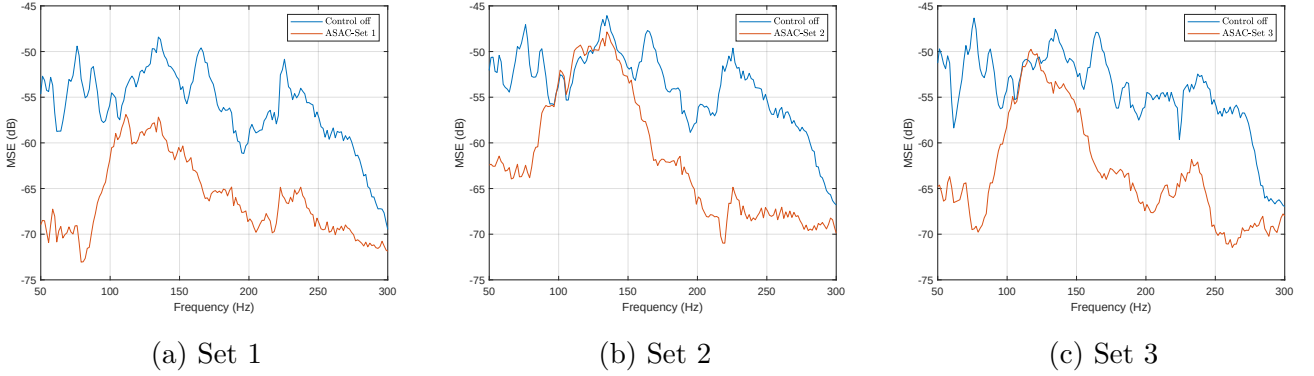


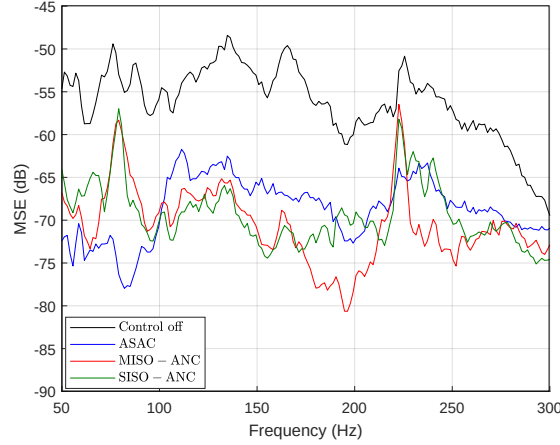
Figure 4.10: Experimental mean squared error (MSE) at the microphones for (a) Set 1, (b) Set 2, and (c) Set 3 actuator arrangements, with and without applying ASAC. Reproduced from [89].

These parameters were experimentally tuned and remain fixed across all control configurations unless stated otherwise. The first configuration, ASAC, employs three error microphones and three inertial actuators, arranged according to the optimized layout. The second, Multiple-Input Single-Output Active Noise Control (MISO-ANC), follows a conventional overdetermined approach using a single loudspeaker as the actuator with three error microphones. The third, Single-Input Single-Output Active Noise Control (SISO-ANC), represents a single-channel scheme using Mic 2 as the error microphone, while Mic 1 and 3 are used only as observation microphones. In the frequency range below 100 Hz, the ASAC configuration exhibits lower MSE values compared to the loudspeaker-based systems, as in Fig 4.11a. However, this band is particularly challenging to control, as it contains the system's fundamental natural frequencies [80].

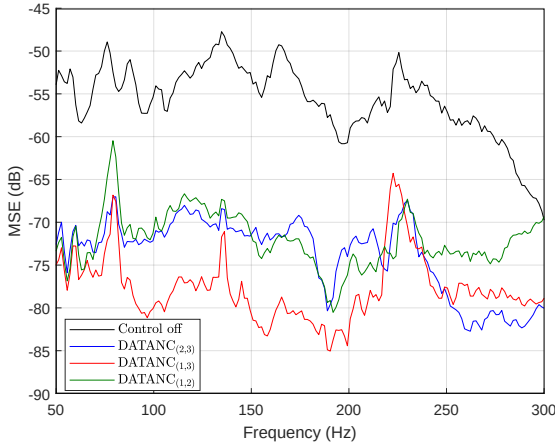
Moreover, this frequency range is generally less critical in practice, as sounds below 100 Hz tend to have lower perceptual impact, particularly at moderate listening levels [92, 93]. While the methodology proposed in this work can be extended to different frequency ranges, the selected range remains relevant for various engineering applications, such as noise control casings and household appliances [61, 94, 95], where dominant noise energy is often concentrated above 100 Hz, making attenuation in this band more critical than at lower frequencies. The secondary loudspeaker in the MISO-ANC and SISO-ANC configurations is also less effective in this band, with a typical lower operating limit near 65 Hz. Thus, although ASAC shows better performance at this range, the perceptual and practical impact of this advantage remains limited. In the frequency

range dominated by the acoustic leakage through the opening (100–200 Hz), the loudspeaker-based configurations (MISO-ANC and SISO-ANC) demonstrate advanced performance compared to the ASAC configuration. This improvement is due to the reduced vibro-acoustic coupling efficiency of the system in this band.

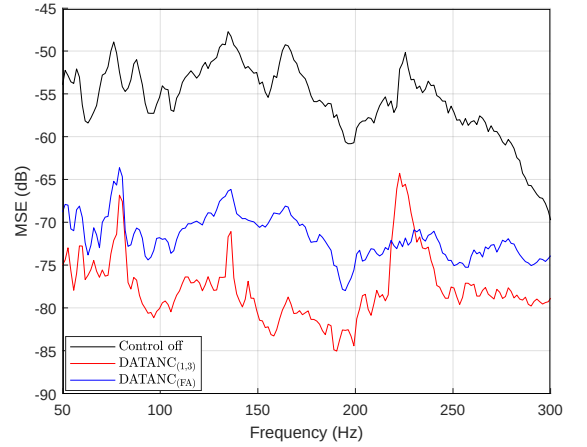
Nevertheless, ASAC achieves an average noise reduction of approximately 12 dB, which remains significant for applications where only inertial actuators can be used. Loudspeaker-based configurations, however, achieve greater reduction in this frequency range, with MISO-ANC outperforming SISO-ANC by more than 2 dB on average.



(a)



(b)



(c)

Figure 4.11: MSE performance comparison across different active noise control configurations. **(a)** Conventional ANC methods: ASAC (three inertial actuators), SISO-ANC (a single loudspeaker with one error microphone), and MISO-ANC (a loudspeaker with three microphones). **(b)** Proposed DATANC configurations in which individual inertial actuators are selectively deactivated while the secondary loudspeaker remains active. **(c)** Comparison between $\text{DATANC}_{(1,3)}$, which uses the loudspeaker along with actuators A1 and A3, and $\text{DATANC}_{(\text{FA})}$, which includes all actuators. Reproduced from [89].

This additional performance comes at the cost of higher computational complexity, as MISO-ANC processes multiple reference signals and adapts multiple control paths simultaneously. Notably, despite relying on a single error microphone, the SISO-ANC configuration demonstrates a considerable global noise reduction across all three error microphones. Despite not being included in the control filter update, Mic 1 and Mic 3 exhibit significant reductions in acoustic power,

comparable to their respective performance in the MISO-ANC configuration, as shown in the corresponding sound power spectrograms reported in Paper D [89].

A distinct resonance near 225 Hz, corresponding to a natural frequency of the system dominated more by the vibro-acoustic coupling, is observed due to the excitation from the primary source. As shown in Fig. 4.11a, loudspeaker-based configurations exhibit limited control performance at this frequency. This discrepancy is due to the placement of the loudspeaker near the enclosure opening, which limits its ability to effectively excite the structural modes.

Consequently, the secondary source shows weak coupling with the vibrating structure, reducing its effectiveness in canceling the noise associated with this dominant resonance. The analysis of the system response using conventional control methods reveals that different actuator types exhibit superior performance in distinct frequency bands; hence, a hybrid approach that combines inertial and loudspeaker actuators may offer enhanced broadband control. This approach, referred to as Dual-Actuator-Type Active Noise Control (DATANC), is proposed based on insights from the performance of conventional control methods. The initial DATANC implementation employs a fully determined configuration with three error microphones and a combination of two inertial actuators and one loudspeaker, where actuator A1, A2, or A3 is deactivated in separate setups. These configurations are denoted as $\text{DATANC}_{(2,3)}$, $\text{DATANC}_{(1,3)}$, and $\text{DATANC}_{(1,2)}$, respectively, with the deactivated actuator remaining physically present. Their system responses are presented in Fig. 4.11b. The underdetermined configuration ($\text{DATANC}_{(\text{FA})}$) uses all four actuators simultaneously to evaluate performance under increased actuation power, as shown in Fig. 4.11c.

Dual-Actuator-Type Active Noise Control

Among the fully determined DATANC configurations, $\text{DATANC}_{(2,3)}$ and $\text{DATANC}_{(1,2)}$ exhibit acceptable performance in the 100–200 Hz frequency range, achieving an average MSE reduction of approximately 18 dB across the error microphones.

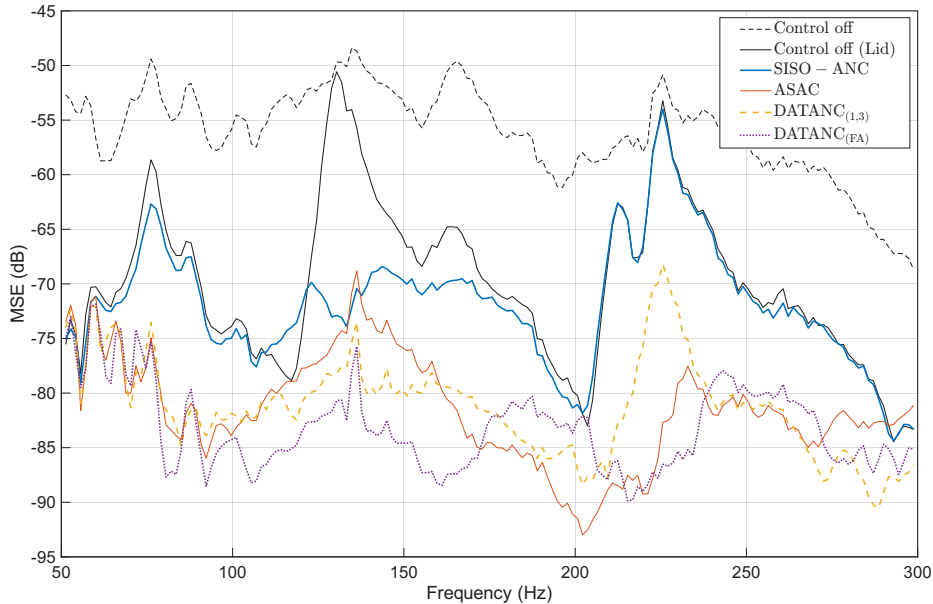


Figure 4.12: MSE performance for different active noise control configurations in the presence of a structural opening sealed with a transparent lid. “Control off” denotes the baseline configuration with the opening, while “Control off (Lid)” indicates the same system with the lid applied. For ASAC, parameters were set to $\mu = 1 \times 10^{-6}$ and $\beta = 1 \times 10^{-6}$; for all other configurations, $\mu = 1 \times 10^{-7}$ and $\beta = 1 \times 10^{-5}$. Reproduced from [89].

This level of attenuation is comparable to that obtained with the MISO-ANC configuration, as shown in Fig. 4.11b. However, $\text{DATANC}_{(1,3)}$ outperforms both configurations, providing over 8 dB

of additional MSE reduction relative to $\text{DATANC}_{(2,3)}$ and $\text{DATANC}_{(1,2)}$ within the same frequency band. The spatial proximity between the loudspeaker and actuator A2 near the enclosure opening, negatively affects their mutual coherence. Thus, when both sources are active simultaneously, their acoustic fields interact destructively in this frequency band. This phenomenon is also observed in the $\text{DATANC}_{(\text{FA})}$ configuration, as in Fig. 4.11c. This spatial analysis of the relative positions of the loudspeaker, inertial actuators, and vibrating plate is reflected in the transfer functions of the primary and secondary paths associated with the four actuators (see Paper D [89] for an analysis of the transfer functions).

A2 also shows an abrupt deviation from its prior phase trend beyond 100 Hz. This irregular and weakly correlated phase behavior reduces overall phase consistency across the control channels, thereby limiting the coherence required for effective noise cancellation in the $\text{DATANC}_{(\text{FA})}$ configuration. To summarize, the superior noise reduction performance of $\text{DATANC}_{(1,3)}$ can be explained by the spatial arrangement of the actuators and the characteristics of their transfer paths: $\text{DATANC}_{(1,3)}$ avoids destructive spatial coupling and benefits from more consistent phase relationships across the control channels, which together account for its superior attenuation performance. To assess the contribution of the sound transmission through the opening to the overall system response, the system is tested using a plate structure with a transparent plastic lid in the place of the former opening. In practice, this is also beneficial for enclosures where a view of the interior is required, or where a different material is needed for part of the wall due to various reasons (e.g., casings of household appliances often use different materials).

As shown in Fig. 4.12, the passive system response highlights the impact of the structural opening: without the lid (Control off), the average sound pressure level is approximately -56 dB, whereas with the lid in place (Control off (Lid)), it is reduced to around -70 dB, clearly demonstrating the effect of sound transmission through the opening. In the 100–200 Hz band, the loudspeaker-based configuration (SISO-ANC) no longer outperforms the ASAC approach in the presence of the transparent lid. For reference, see Fig. 4.11a for the corresponding comparison in the case of the system with acoustic leakage. The ASAC system achieves over 30 dB attenuation at certain frequencies and an average reduction of approximately 12 dB across this frequency range, compared to only 4 dB for the SISO-ANC configuration. Additionally, the previously observed destructive acoustic coupling between inertial actuator A2 and the loudspeaker is no longer present, which enhances the performance of the full actuation configuration ($\text{DATANC}_{(\text{FA})}$), yielding an average reduction of 15 dB compared to 12 dB for $\text{DATANC}_{(1,3)}$. At higher frequencies, particularly around 225 Hz where the dominant vibro-acoustic resonance of the system occurs, the SISO-ANC configuration achieves no or minimal noise reduction. This underscores its limited effectiveness in addressing strongly coupled vibro-acoustic behavior at those frequencies. This limitation in performance also points to the need for further parameter tuning in the SISO-ANC configuration.

Transparent-patched plates

This section studies vibrating plates in which the opening is patched with transparent material so that visual access is maintained while the vibro-acoustic behavior of the wall is altered. Prior studies have considered transparent structures mainly in passive configurations or under tonal excitation, while systematic ASAC evaluation on transparent-patched plates under broadband excitation with placement optimization remains limited [96, 31]. ASAC is therefore applied using inertial actuators mounted on the patched plate, where actuator locations are selected by the previously introduced optimization tool. Configurations employing one, two, and three actuators are examined to assess their influence on noise reduction performance, as shown in Fig. 4.13. The analysis focuses on the synergistic effects of multiple actuators and the influence of their spatial arrangement including mutual positioning, placement on the plate, and relative location to the noise source. Thus, the optimization method presented in Section 3.3 can guide the strategic

placement of actuators to enhance the overall performance of transparent-patched plates (see Paper E [97]).

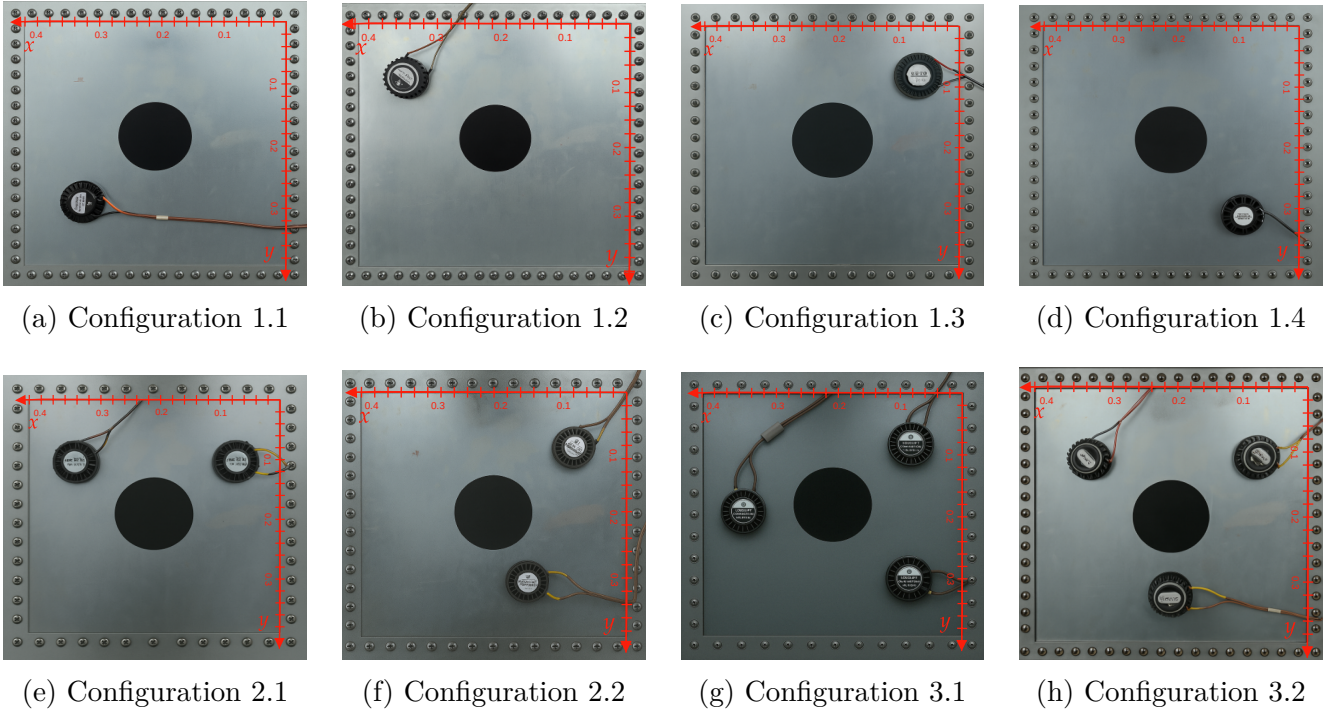


Figure 4.13: The investigated configurations. Reproduced from [97].

At first glance, using additional actuators may appear to demand more power while providing a highly improved performance. However, this is not necessarily true, as the synergy among actuators plays a critical role. Proper spatial placement is essential to enable a synergistic interaction that enhances overall system effectiveness.

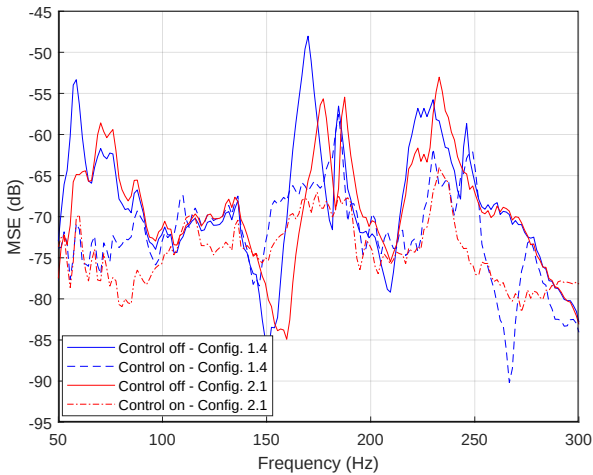


Figure 4.14: MSE of the microphones in configurations 1.4 and 2.1. Reproduced from [97].

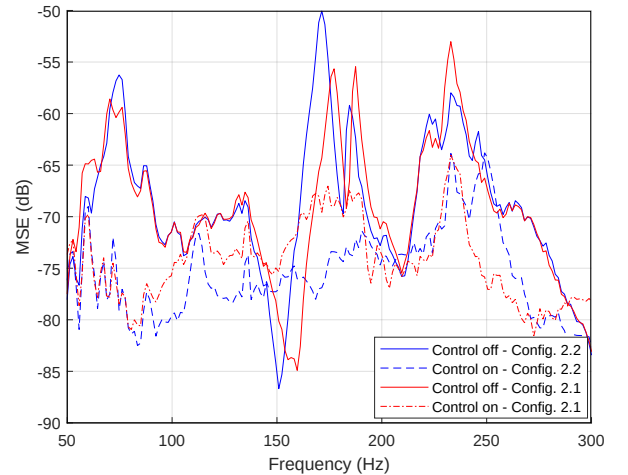


Figure 4.15: MSE of the microphones in configurations 2.1 and 2.2. Reproduced from [97].

Fig. 4.14 compares the MSE of the microphones in Configurations 1.4 and 2.1, where Configuration 1.4 uses a single actuator and Configuration 2.1 uses two. Although the use of two actuators in Configuration 2.1 results in better attenuation of resonances around 180 Hz and 240 Hz, it does not lead to significant improvement across the full frequency range or in the overall average sound power level. Furthermore, this limited gain comes with a 45% increase in the average

RMS voltage drawn by the actuators compared to Configuration 1.4, as in Table. 4.3. The performance of the systems with two actuators can be improved by properly selecting the position of the actuator, as the optimized arrangement shown in Configuration 2.2. In Fig. 4.15, Configuration 2.2, using ASAC, demonstrates superior performance in reducing sound power across the three microphones compared to Configuration 2.1. Resonance attenuation in the 140–215 Hz range is nearly doubled, and the average reduction across the 75–300 Hz range increases by approximately 50%. This improvement is attributed to the more effective placement of the two actuators in Configuration 2.2. Both actuators are positioned away from regions of high mechanical stress, and one is located near the PVC patch, where reduced stress concentration allows for greater actuation flexibility. However, all configurations discussed thus far (Configurations 1.1 to 2.2) tend to converge to similar sound power levels within the 200–240 Hz range.

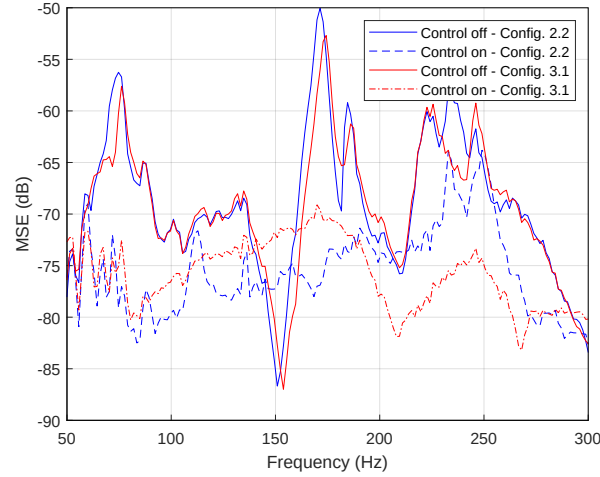


Figure 4.16: MSE of the microphones in configurations 2.2 and 3.1. Reproduced from [97].

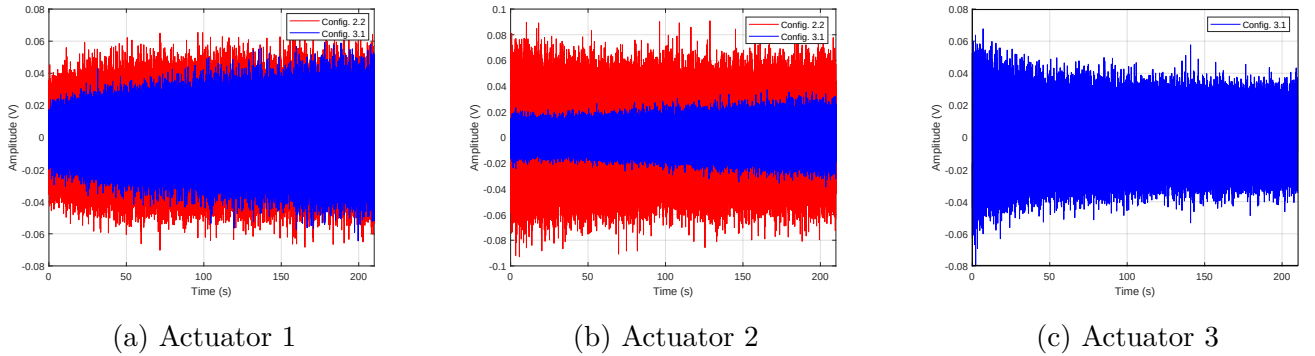


Figure 4.17: Control signals of the actuators in configurations 2.2 and 3.1. Reproduced from [97].

To address this limitation, systems with three actuators are investigated. With the actuator arrangement used in Configuration 3.1 (Fig. 4.13g), the system achieved an approximate 5 dB improvement in performance across the 215–275 Hz range, effectively overcoming the limitation previously observed, as in Fig. 4.16. This enhancement was accomplished with only a 2.5% increase in the average RMS voltage of the control signal. However, the third actuator, positioned in a region of higher mechanical stress, required more actuation power than the second actuator, as shown in Fig. 4.17.

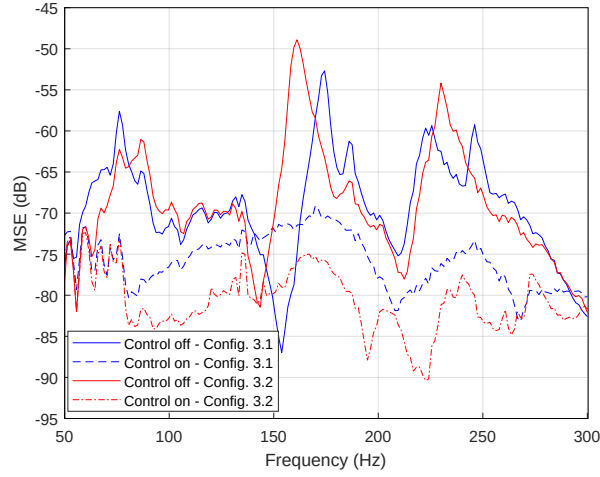


Figure 4.18: MSE of the error microphones in configurations 3.1 and 3.2. Reproduced from [97].

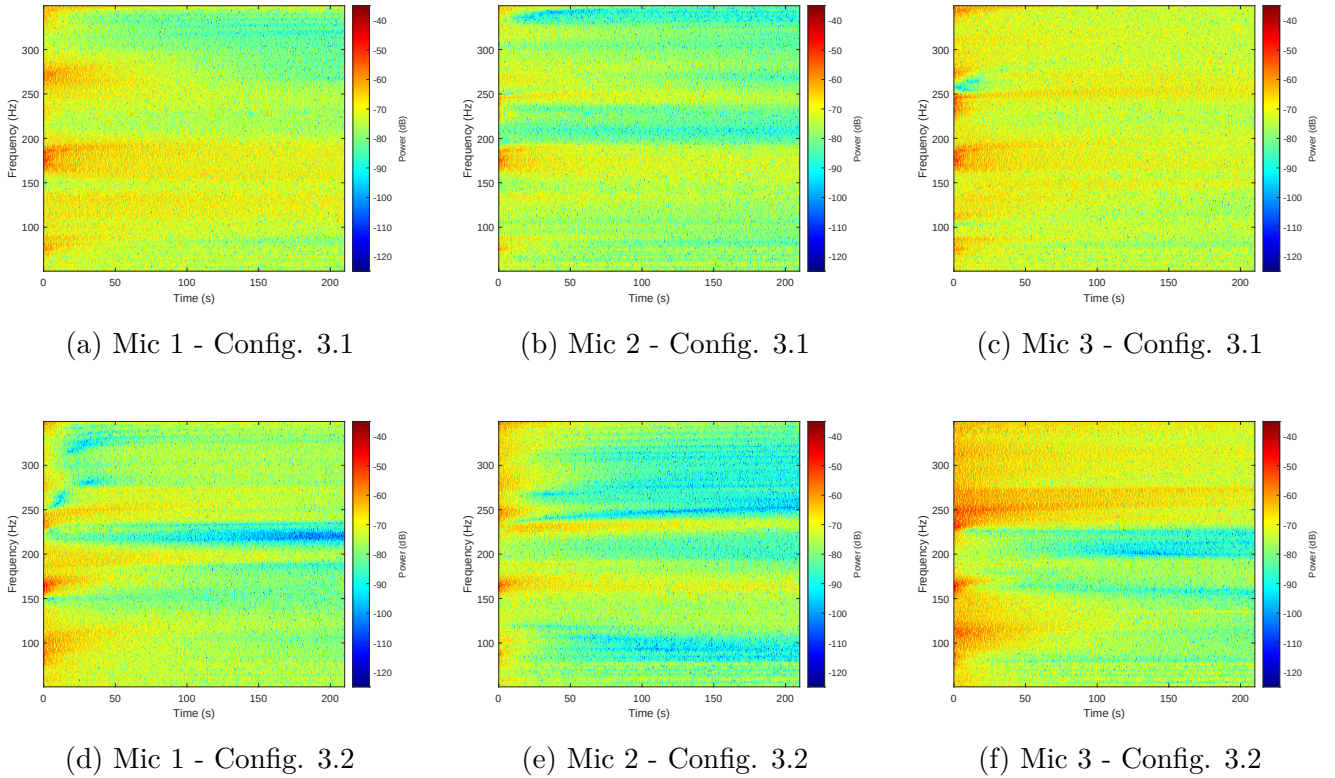


Figure 4.19: Microphone signal power over time for configurations 3.1 and 3.2. Reproduced from [97].

Configuration 3.1 failed to match the performance of configurations with fewer actuators in the lower half of the considered frequency range. This limitation is primarily attributed to the suboptimal placement of the first and second actuators, which are both concentrated in the right half of the plate near regions of high mechanical stress around the clamped edges. In contrast, Configuration 3.2, which features a more balanced spatial distribution of the actuators, achieves improved performance across the entire frequency range. This arrangement provided the system with greater flexibility to introduce actuation power. As a result, the configuration led to doubling the average RMS voltage of the control signals. Consequently, it achieved a doubling in average noise reduction, exceeding 12 dB, and an additional reduction of more than 8 dB in the system's resonant peaks. This enhanced performance is also clearly reflected in the spectrogram of sound

Table 4.3: Summary of the evaluation metrics for all tested configurations. Reproduced from [97].

Config	Reduction (dB)	Frequency range (Hz)				Average RMS (V)
		75–140	140–215	215–275	75–300	
1.1	Resonance	5.9	8.3	2.8	6.3	0.0077
	Average	2.9	-1.7	4.5	1.9	
1.2	Resonance	5.5	9.1	-0.2	-0.2	0.0080
	Average	1.2	-1.6	5.3	1.8	
1.3	Resonance	5.5	12.9	-0.3	8.2	0.0091
	Average	1.5	0.12	4.2	1.5	
1.4	Resonance	5.1	9.5	5.9	9.5	0.0081
	Average	0.6	-0.2	7.8	2.4	
2.1	Resonance	10.4	11.5	10.9	10.9	0.0118
	Average	5.3	1.1	9.1	4.5	
2.2	Resonance	14.9	21.2	5.8	13.7	0.0118
	Average	8.5	5.3	5.9	6.2	
3.1	Resonance	14.4	16.4	14	16.4	0.0121
	Average	6.5	3.6	11.8	6.5	
3.2	Resonance	11.9	26	23.2	24	0.0251
	Average	12.6	12.4	16.4	12.6	

power across the three error microphones, with particularly notable improvements observed on microphones 1 and 3, as shown in Fig. 4.19. Although microphones 1 and 3 show higher local attenuation due to their spatial proximity to actuators 1 and 3 in Configuration 3.2, global noise reduction is still achieved. This outcome is consistent with the spatial control criteria described in the Section 3.4, where the microphone arrangement was designed to satisfy established spacing constraints. The results highlight that placing inertial actuators in regions of lower mechanical stress is critical, as improper placement can significantly degrade system functionality and lead to inefficient use of available resources. Moreover, increasing the number of actuators does not necessarily result in better noise reduction; in some cases, a single actuator can deliver acceptable performance compared to configurations with two or three times as many when the objective is to minimize specific resonances in the system’s acoustic response.

Chapter 5

Conclusion and future work

The work demonstrates that the design of openings on a vibrating plate can be used to shape the vibro-acoustic response of a mixed interface. Comparative studies across configurations relevant to engineering applications show that discontinuities in the opening contour lead to higher resonances in the mid-frequency range and that asymmetric placement introduces additional resonances. The novelty lies in directly linking an adaptable analytical model—parameterized by opening shape and location—to the acoustic response and integrating it with a COMSOL Multiphysics® FEM model for visualization of plate–pressure interaction. Building on this, a novel opening design optimization method is presented, combining a genetic algorithm with the analytical model within a master-slave software architecture. Within 100–350 Hz, optimally designed openings achieve reductions of about 5–15 dB in peak sound power amplitude compared with median-fit layouts, and even larger improvements relative to least-fit designs.

On active control, the study introduces a reformulation of the vibro-acoustic analytical model to account for transmission through the opening and uses it to guide inertial-actuator placement by jointly considering radiation and leakage. The proposed Dual-Actuator-Type Active Noise Control (DATANC) concept integrates loudspeakers and inertial actuators. Experiments show that, among ASAC layouts, placing actuators near the opening edge yields the lowest MSE and an average reduction of about 12 dB in 100–200 Hz (with alternative layouts giving about 5 dB and 8 dB). Loudspeaker-based ANC (SISO/MISO) outperforms ASAC in the 100–200 Hz band dominated by through-opening transmission, whereas inertial actuators are particularly effective at attenuating strong vibro-acoustic resonances (e.g., around 225 Hz). The study also investigates a transparent-patched plate, showing that optical accessibility can be preserved while enabling effective ASAC; actuator performance depends strongly on spatial placement, and suitably optimized multi-actuator arrangements can achieve sound–power reductions exceeding 12 dB.

The results presented throughout the dissertation confirm the main thesis formulated in Section 1.5. The research demonstrates that optimal design of openings and actuator placement can effectively reduce noise emission from vibro-acoustic systems with mixed structural–acoustic interfaces. Through analytical modeling, numerical optimization, and experimental validation, it has been proven that the geometry and positioning of openings govern the redistribution of vibro-acoustic response, while the optimized placement of inertial actuators enhances the active control performance. Moreover, the introduction of hybrid configurations combining inertial and acoustic actuators validates the feasibility and benefits of broadband noise reduction. For systems with noise barriers made of different materials, this study clarifies how sound leakage through openings and the radiation component jointly shape the sound power response, and proves that conventional active control achieves greater noise reduction when actuators follow optimized spatial layouts.

5.1 Basic framework and motivations of appended papers

The appended papers collectively establish a unified framework for modeling, optimizing, and actively controlling vibro–acoustic systems with openings. The motivation arises from the need to predict and reduce noise emissions in lightweight structures where openings are required for ventilation, access, or other design purposes, such as in device casings. Although many analytical and numerical models exist, the influence of openings has been largely overlooked in studies focused on optimization or active control under broadband excitation. The main goal of this research is therefore to develop an integrated framework that combines modeling, optimization, and experimental validation to advance active noise control in such systems.

Paper A develops the analytical foundation, formulating a model that captures the vibro–acoustic interaction of plates with openings using trigonometric and hyperbolic functions while accounting for acoustic leakage. The model is validated through FEM/BEM simulations and experiments, providing a consistent basis for analyzing how opening geometry and placement affect sound radiation. Paper B investigates the influence of the primary noise source location on the vibro–acoustic response, demonstrating how source–opening positioning alters the radiated sound power and field distribution—key insights for subsequent optimization and control.

Paper C introduces a genetic–algorithm–based optimization integrated with the analytical model to minimize sound power by refining opening geometry and placement. The master–slave optimization structure enables efficient parametric analysis, marking the transition from analytical modeling to model–based design optimization. Paper D extends the framework to active control using linear–quadratic optimal control for inertial actuators combined with genetic–algorithm placement optimization. It introduces and experimentally validates the Dual–Actuator–Type Active Noise Control (DATANC) system, which merges structural and acoustic actuation to mitigate both leakage and vibration–induced noise.

Finally, Paper E extends the framework to systems incorporating panels made of different materials, such as transparent PVC patches on metallic plates. By including a mass–law–based transmission coefficient, the analytical model accounts for sound transmission through material interfaces. Experimental results confirm the active control performance across different configurations, demonstrating the applicability of the proposed method to realistic multi-material structures.

In summary, the appended papers present a coherent progression from modeling to experimental validation, establishing a comprehensive methodology for the modeling, optimization, and active control of vibro–acoustic systems with openings or made of different materials.

5.2 Summary of Paper A

Paper A establishes the analytical foundation of this research by developing a unified model for predicting the vibro–acoustic behavior of plate–cavity systems incorporating structural openings. The study addresses the lack of adaptable analytical formulations capable of capturing both the radiated and transmitted sound power components caused by openings in the vibrating plate. The proposed analytical model employs trigonometric and hyperbolic admissible functions for describing both the structural displacements and the acoustic pressure field inside the cavity, offering superior convergence and computational efficiency compared to traditional polynomial formulations. Sound radiation is evaluated through the radiation resistance matrix, while a complementary finite–element (FEM) model is used to visualize and validate the analytical predictions. Experimental validation is carried out in laboratory conditions to confirm the model’s capability to represent real vibro–acoustic coupling phenomena.

The main contributions of this study are: (i) the introduction of a new analytical framework

capable of modeling coupled structural–acoustic behavior in systems with arbitrary openings, bridging the gap between theoretical prediction and experimental observation; (ii) the decomposition of the total radiated sound power into physically interpretable components representing plate radiation and acoustic leakage; and (iii) the integration of analytical and numerical models, enabling detailed visualization of pressure and vibration fields. These developments establish a methodology for shaping the vibro–acoustic response, forming the basis for the optimization and active control analyses presented in the subsequent papers.

5.3 Summary of Paper B

Paper B builds directly upon the analytical formulation introduced in Paper A, focusing on the influence of the noise source position on the vibro–acoustic coupling and the overall system response. The study investigates how relocating the primary acoustic excitation relative to the opening and vibrating plate alters the structural vibration modes, sound power distribution, and resonance characteristics. A combination of analytical prediction and experimental verification is used to establish the understanding of source–structure–cavity interactions. The model captures how certain source positions can either excite or suppress specific structural modes, depending on their spatial relationship to the vibrating plate.

The study demonstrates that the vibro–acoustic response is strongly influenced by the relative position of the noise source, the opening, and its proximity to the vibrating plate. When the source faces the opening, the system exhibits greater sound power at low frequencies. In contrast, moving the source closer to the plate amplifies vibration amplitudes across all modes. These findings establish the source location as a critical parameter governing the system’s dynamic behavior and provide an essential foundation for the optimization and active control methodologies developed in subsequent papers

5.4 Summary of Paper C

Paper C advances the research framework by introducing a comprehensive optimization methodology for structural openings in vibro–acoustic systems. While previous studies have addressed geometric optimization for plates or cavities, the explicit optimization of openings—both in shape and placement—in vibro–acoustic systems remains largely unexplored. To address this, a genetic–algorithm–based optimization framework is proposed, reformulating the analytical model developed in Paper A as the evaluation engine. The framework operates under a master–slave software architecture, where the analytical solver serves as the slave module to compute objective functions for candidate geometries generated by the optimization master. The primary objective is to minimize total sound power while satisfying geometric and functional constraints, making the method directly applicable to engineering structures such as enclosures and casings with ventilation or access openings.

The main contributions of this paper are threefold: i) It proposes a novel optimization methodology for openings in vibrating plate structures within vibro–acoustic systems to enhance acoustic performance; ii) It employs a genetic algorithm framework integrated with an analytical vibro–acoustic model using a master–slave software architecture; and iii) It investigates the influence of opening location, number, and shape on the sound power characteristics of vibro–acoustic systems, providing valuable insight to prevent the selection of suboptimal design parameters that may lead to undesirable acoustic behavior.

5.5 Summary of Paper D

Paper D extends the framework into the domain of active control, focusing on actuator placement optimization in vibro–acoustic systems with openings. The study proposes an analytical model that integrates sound power estimation with the effect of the inertial actuators to determine their optimal positions. Each actuator configuration is evaluated within a genetic–algorithm environment, treating actuator locations as optimization variables. The work investigates the effectiveness of active noise control (ANC) and active structural acoustic control (ASAC) configurations in mitigating noise in such systems. It also introduces the Dual–Actuator–Type Active Noise Control (DATANC) concept, combining loudspeaker and inertial actuators to achieve broadband attenuation by addressing both the acoustic leakage through openings and the structure-borne radiation from the vibrating plate. The framework is validated through extensive experiments involving adaptive control schemes and comparative evaluation against conventional ANC and ASAC systems.

The contributions of this work are diverse. It introduces the first integrated methodology for optimizing inertial actuator placement on vibrating plates with openings, bridging passive and active noise control paradigms. It experimentally demonstrates the efficiency of the DATANC approach, achieving superior broadband attenuation through hybrid acoustic–structural actuation. Furthermore, it quantifies how acoustic leakage and actuator spatial distribution influence control performance, showing that hybrid actuation provides significant benefits over single-actuator systems. This paper motivated further analysis of actuator placement and active control in structures made of different materials, as these configurations offer a balance between functional requirements—such as visual accessibility—and effective passive noise reduction by limiting acoustic leakage through openings.

5.6 Summary of Paper E

Paper E broadens the application of the developed methodologies to vibro–acoustic systems incorporating hybrid materials, specifically panels composed of metallic plates with openings covered by transparent plastic patches. The study introduces an extended analytical model that accounts for sound transmission through the patch by incorporating a mass–law–based transmission coefficient. Experimental and analytical investigations are conducted on a steel–PVC plate serving as an active acoustic barrier. The system’s total sound power is expressed as the sum of the radiated component from the plate and the transmitted component through the patch, allowing explicit characterization of the multi-material coupling. The active control methodology developed in Paper D is adapted to this hybrid system, optimizing inertial actuator placement and control parameters to achieve maximum sound power reduction under broadband excitation.

The main achievements of this study are threefold. It extends the analytical and experimental framework to hybrid multi-material structures, enabling accurate prediction of sound transmission through transparent patches. It experimentally demonstrates the effectiveness of active control in such systems, confirming that transparency and acoustic insulation can coexist within a single design. Finally, it highlights broader design implications for practical applications, such as vibro–acoustic casings that must balance visual access and noise suppression at the expense of airflow.

5.7 The future work

Future work will focus on extending the DATANC concept to commercial appliances that employ lightweight panels and structural openings for airflow, monitoring, and maintenance, where noise leakage through necessary openings makes hybrid active control strategies particularly advantageous.

Building on the current optimization framework, further investigations should establish quantitative relationships between plate size, the number of openings, and the required number of actuators to guide practical design decisions. For transparent-patched systems, examining the relationship between plate dimensions and patch size will also be essential.

Because placing physical microphones around casings is often challenging, particularly in compact or uncontrolled environments, virtual sensing techniques can be employed as alternatives using remote measurements or advanced filtering methods. Moreover, integrating deep-learning-based online path identification represents a promising approach to enhance robustness under varying operating conditions. Comprehensive research and experimental validation are required to assess the reliability of these solutions.

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A. Attached papers

Five papers listed below are attached in the appendix:

Paper A K. S. A. Maamoun, S. Wrona, M. Pawelczyk, and H. R. Karimi, “Shaping of the frequency response of vibrating plates with openings for vibro-acoustic systems,” *Mechanical Systems and Signal Processing*, vol. 218, p. 111539, 2024.

<https://doi.org/10.1016/j.ymssp.2024.111539>

Paper B K. S. A. Maamoun, M. Pawelczyk, and S. Wrona, “Analyzing the impact of sound source positioning on frequency response of vibrating plates with openings in vibro-acoustic systems,” in *Proceedings of the 30th International Congress on Sound and Vibration (ICSV30)*, Amsterdam, IIAV, July 2024.

Paper C K. S. A. Maamoun, S. Wrona, M. Pawelczyk, and H. R. Karimi, “Optimizing design of openings in vibrating plates for enhanced vibro-acoustic performance using a genetic algorithm approach,” *Journal of Sound and Vibration*, vol. 598, p. 118847, 2025.

<https://doi.org/10.1016/j.jsv.2024.118847>

Paper D K. S. A. Maamoun, C. K. Lai, S. Wrona, M. Pawelczyk, and H. R. Karimi, “Dual-actuator-type active noise control in vibro-acoustic systems with openings,” *Scientific Reports*, vol. 15, no. 1, p. 32278, 2025.

<https://doi.org/10.1038/s41598-025-17810-8>

Paper E K. S. A. Maamoun, A. Aboutiman, and A. M. Ali, “Synergistic use of inertial actuators for advanced noise control in transparent-patched vibrating plates,” *Measurement*, vol. 257, p. 118701, 2026.

<https://doi.org/10.1016/j.measurement.2025.118701>



Shaping of the frequency response of vibrating plates with openings for vibro-acoustic systems

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ARTICLE INFO

Communicated by S. Marburg

Keywords:

Vibrating plates with openings
Structure optimization
Sound power estimation
Shaping frequency response

ABSTRACT

The frequency response of vibrating plates with openings can be tailored through the shape and location of these openings in the mixed interfaces of vibro-acoustic systems. This paper introduces a novel method for designing the characteristics of openings in mixed interfaces to achieve the desired vibrational and acoustic responses. The core of this methodology is an analytical model that utilizes the Rayleigh–Ritz method, augmented by a radiation resistance matrix for precise estimation of sound power in a free field. This approach provides the essential flexibility to alter the shapes and locations of the openings. A Finite Element Method (FEM) is specifically employed to enhance the visualization of the interactions within the multiphysics domains of the vibro-acoustic systems. This FEM model, developed for the enclosure and coupled with the vibrating plate according to Kirchhoff's thin plate theory, is complemented by the Boundary Element Method (BEM) for the sound pressure calculations. The methodology is further substantiated by experimental validation, which confirms the theoretical predictions and enhances the credibility of the proposed models.

1. Introduction

The openings in the plate structure of a vibro-acoustic coupled system serve multiple practical functions in various engineering applications, including automotive, aerospace, marine engineering, and naval architecture. These functions include facilitating heat dissipation, accommodating electrical cable routing, providing personnel access, and weight reduction. In these areas, effectively managing acoustic properties while maintaining structural integrity and functionality is essential [1–4]. The mixed vibro-acoustic interface exhibits a complicated frequency response, which can make it difficult to achieve the required level of noise attenuation. The process of shaping the plate's frequency response is inherently complex, as noted in [5]. The introduction of openings adds further complexity to this task. Consequently, the ability to effectively shape the frequency response of the mixed interface is highly beneficial [6–8]. By carefully selecting the shape and location of these openings, it is possible to enhance the passive transmission loss, thereby improving its effectiveness.

The starting point to analyze the vibro-acoustic coupled system is the modeling of the vibrating plate with openings. The presence of openings generally makes it challenging to derive an analytical solution for the problem under consideration [9]. The existing research mainly uses semi-analytical and numerical methods. Among these, semi-analytical approaches are frequently characterized by techniques based on the Rayleigh–Ritz method (RRM). RRM was applied to the prediction of natural frequencies of rectangular plates [10] with rectangular openings [11,12].

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In [13,14], the authors used the Chebyshev–Lagrange and the spectro-geometric methods to study the free vibration of the rectangular plate with openings under general boundary conditions. These two methods are both improvements of RRM. An independent coordinate coupling method (ICCM) was introduced in [15], and then it was applied to point support perforated rectangular plates [16] as well as rectangular plates with multiple openings [17]. In [18], the authors indicated that using variable stiffness to incorporate openings in vibrating plates limits frequency analysis to below the critical frequency. However, the developed methodology in [19] counters this by applying a meshfree approach, the discrete Ritz method (DRM), featuring variable stiffness at the openings. This is particularly relevant for metal thin plates, where the critical frequency often exceeds the working frequency range.

The mentioned literature is devoted to vibro-acoustic systems, encapsulating the plate vibration, enclosure internal sound field, and the mixed interface that embodies the vibro-acoustic coupling, along with the sound transmission through the openings. In addition, the authors in [20] modeled the sound field in the acoustic opening as the superposition of plane waves, and in [21], the authors applied the same concept to a diffused field, and both are effective only below the cut-off frequency and suitable for openings smaller than the wave length under low frequency conditions. For larger openings relative to wavelength, [22,23] utilized the Kobayashi potential method for its computational efficiency compared to the superposition technique. In [24], the authors treated the opening as a massless membrane within the mixed interface, and [25] unified the sound transmission through structures and openings in a singular theoretical acoustic model.

In [26], researchers conducted vibro-acoustic analysis on a 3D acoustic field enclosed by a flexible plate under varied elastic boundary constraints. The analysis involved representing plate displacement and cavity sound pressure using the Fourier cosine series, where the series' coefficients served as generalized coordinates. Similarly, the authors in [27], used the same methodology to explore sound transmission through a vibro-acoustic interface dividing two subrooms with openings, where they applied the Lagrangian and Rayleigh–Ritz methods to assess how opening placement and size affect modal behavior but did not consider opening shapes other than rectangular or the sound transmission to free field.

The sound radiation from a vibrating plate can be estimated using the Kirchhoff–Helmholtz integral equation [28–30]. In [31], the authors used the resistance radiation matrix for sound radiation estimation. This matrix does not depend on the velocity distribution on the plate and thus can be reused for parametric studies, significantly reducing the computation time. In [32,33], the authors used the resistance radiation matrix and validated the results experimentally, where the authors in [34] extended the sound radiation experimental estimation to the usage of photogrammetry. In [35], the authors introduced an alternative for classical microphone measurements with the use of the vibrational pattern to determine the radiated sound power.

Different studies implemented the Finite Element Method (FEM) for sound power estimation from vibro-acoustic systems [36–38], while others combined the FEM with the Boundary Element Method (BEM) to address the low-frequency restriction of the FEM and the high computational time [39,40]. The comparison between the Wave Based Method (WBM) and FEM for sound power radiation from vibro-acoustic systems was addressed by [41,42], where the WBM proved to be an alternative technique for the FEM.

In this paper, the frequency response of a vibrating plate with openings is shaped by designing the shape and location of these openings on the mixed interface, which covers a rectangular enclosure. An analytical model is developed for the vibro-acoustic system based on the work introduced by [26] with the implementation of hyperbolic and 2D trigonometric structure admissible functions to overcome the polynomials admissible functions need for a large number of terms to represent the vibrating plate [19]. Also, the radiation resistance matrix is used to extend the sound power estimation in the free field. Additionally, a FEM model is developed for the system to better visualize and analyze the results based on the FEM-BEM technique. For different case studies, the analytical and FEM models have been validated against experimental setup for vibration and sound power measurements.

The paper consists of 5 sections. Section 2 describes a FEM model with the selected physics, mesh, and boundary conditions. In Section 3, an analytical model is described with the vibro-acoustic coupled system and the sound power radiation. In Section 4, the models' results are compared with the measurements from the vibrometer and two Beyerdynamic MM1 microphones in the experimental setup. Additional discussion for different shapes and locations of the opening on the mixed interface is introduced in Section 5.

2. FEM modeling of a vibro-acoustic system

2.1. 3D model

In this section, the main objective is to model the vibro-acoustic interaction between a thin vibrating plate with an opening and the enclosed sound source. The considered enclosure has inner dimensions of 0.47 m × 0.44 m × 0.4 m. The domain of the enclosure and the infinite void of the surrounding domain contain air with density and speed of sound as, respectively,

$$\rho_a = 1.2 \text{ kg/m}^3, c = 343 \text{ m/s}, \quad (1)$$

An excitation mono-pole sound source is placed in the enclosure with a reference power of $P_{rms} = 1 \text{ W}$. The source is located at coordinates $x = 0.24 \text{ m}$, $y = 0.43 \text{ m}$, and $z = 0.3 \text{ m}$ to align with the experimental setup for model validation. The enclosure is constructed with a concrete frame with a thickness of 0.025 m to be consistent with the experimental setup, as illustrated in Figs. 1 and 2. A vibrating plate is overlapping the opening wall of the enclosure with a size of a , b , and h as 0.42 m, 0.39 m, and 0.001 m, respectively, made of steel with density ρ , Young's modulus E , and Poisson ratio ν as indicated in Eq. (2).

$$\rho = 7850 \text{ kg/m}^3, E = 210 \times 10^9 \text{ Pa}, \nu = 0.3, \quad (2)$$

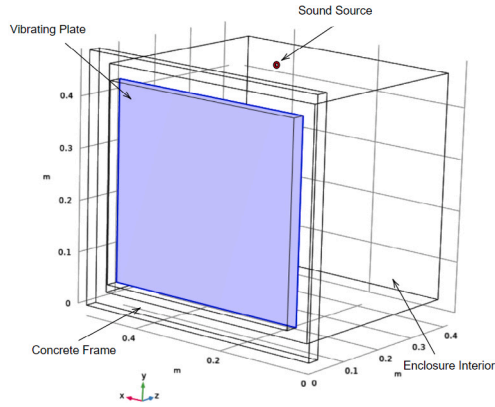


Fig. 1. Enclosure mounted with the vibrating plate.

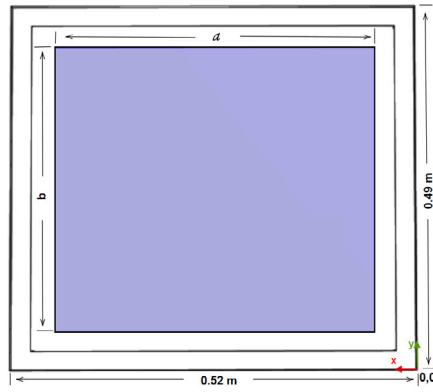


Fig. 2. 2D view of the enclosure mounted with the vibrating plate.

2.2. Physics, mesh, and boundary condition considerations

The FEM model utilizes COMSOL Multiphysics® as the commercial software, ensuring that the physics, mesh, and boundary conditions are implemented according to the software's specifications.

(1) Model physics:

The plate model is built in 3D space under the 'Shell' interface, as the thickness-to-span ratio h/a in the considered case allows the shell interface to be representative [43]. The shell elements in the model are quadratic and conform to the Mindlin-Reissner type, as set by default. The Mixed Interpolation of Tensorial Components (MITC) approach is selected. As mentioned in [44], the MITC formulation adapts the elements for both thick shells, as per Mindlin theory, and thin shells, following Kirchhoff theory, based on the ratio of h/a . For the dimensions considered in this model, they will automatically converge towards the solution proposed by Kirchhoff theory. The enclosure domain is represented using the Pressure Acoustics, (Frequency Domain) interface. This interface specifically addresses the Helmholtz equation in the frequency domain [45].

Moreover, the enclosure's surroundings are modeled as Pressure Acoustics, Boundary Elements (pabe), which solve the Helmholtz eigenvalue equation on boundaries. This method searches for the out-of-plane wave numbers at a given frequency, making the implementation of the pabe physics interface convenient for calculating the sound pressure characteristics in the infinite void. The considered cases use COMSOL Multiphysics®, which provides both the Acoustic-Structure Boundary multiphysics coupling feature and the Acoustic BEM-FEM Boundary multiphysics coupling.

(2) Meshing:

To effectively resolve the propagation of the acoustic wave in space, a minimum of 7 to 10 mesh elements per wavelength (at the maximum frequency) is required, as suggested in [46]. The maximum mesh size is calculated by

$$h_{\max} = \frac{\lambda_{\min}}{N} = \frac{c_{\min}}{f_{\max} N}, \quad (3)$$

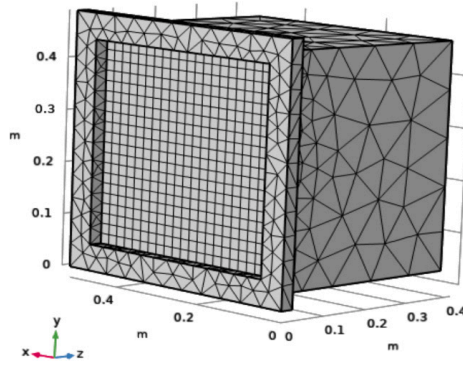


Fig. 3. Meshing of the enclosure with solid plate.

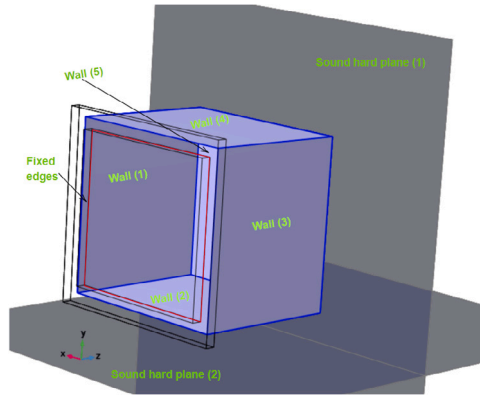


Fig. 4. Infinite void includes two perpendicular planes Sound hard (1) and Sound hard (2), for the enclosure walls, wall (1), wall (2), wall (3), wall (4), and wall (5) sound-hard condition has been set, and red frame shows the fixed edges.

This calculation is for the air domain, using a free tetrahedral mesh type. For the plate, the maximum element size is selected to be 0.039 m, which is 1/10 the plate's width, and the mesh type is a mapped mesh, as shown in Fig. 3.

(3) Boundary conditions:

The five walls of the enclosure have been set as sound-hard rigid walls, and the infinite void contains two sound-hard planes to replicate the features of the surrounding in Fig. 4. Moreover, the edges of the plate have been set to be fixed to represent the fully clamped plate.

(4) Solver and study:

For each opening configuration, two studies have been implemented: the eigenfrequency and frequency domain. The first study is used to extract information about the natural frequencies of the plate and its mode shapes, while for the frequency domain, the frequencies have been selected to sweep from 1 Hz to 350 Hz with step 3 Hz, and it is used to calculate the vibration and radiation response of the plate under the mono-pole source excitation. The model is computed by MUMPS (multifrontal massively parallel sparse) direct solver.

3. Analytical modeling of the vibro-acoustic system

3.1. Modeling of free vibrating plate

The Kirchhoff–Love theory of thin plates is used to model the free vibrating rectangular plate [47]. The plate is assumed to be isotropic and homogeneous, excluding the region of the opening in which the thickness of the plate will be zero. The Rayleigh–Ritz method (RRM) is used to define an approximate solution to the eigenvalue equation to calculate the natural frequencies and mode shapes of the plate vibration. Considering a rectangular plate with side lengths L_x in the x -direction and L_y in the y -direction, see Fig. 5. The Lagrange functions of the Plate can be calculated as

$$L_P = U_P - T_P, \quad (4)$$

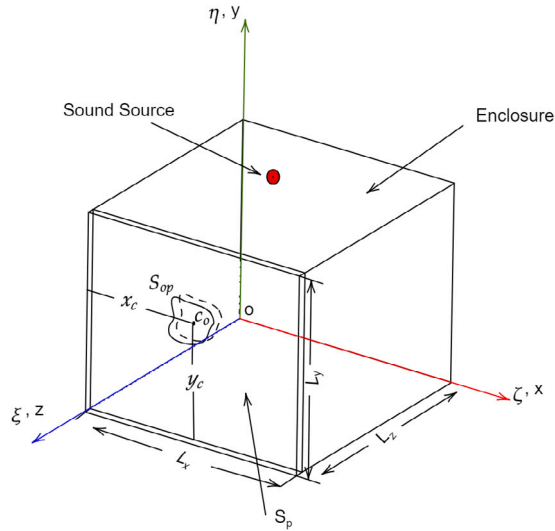


Fig. 5. Enlarged system of the enclosure and coupled with a mixed interface of a vibrating rectangular plate with an arbitrary shaped opening interface S_{op} , where $(x_c, y_c) \in S_{op}$ and S_p is the structural plate interface.

The kinetic and potential energies, T_p and U_p respectively, of the rectangular plate can be expressed as

$$T_p = \frac{1}{2} \rho \int_0^{L_x} \int_0^{L_y} h(x, y) \dot{w}^2 dx dy, \quad (5)$$

$$U_p = \frac{1}{2} \int_0^{L_x} \int_0^{L_y} D(x, y) \left[\left(\frac{\partial^2 w}{\partial x^2} \right)^2 + \left(\frac{\partial^2 w}{\partial y^2} \right)^2 + 2\nu \left(\frac{\partial^2 w}{\partial x^2} \frac{\partial^2 w}{\partial y^2} \right) + 2(1-\nu) \left(\frac{\partial^2 w}{\partial x \partial y} \right)^2 \right] dx dy, \quad (6)$$

where

$$h(x, y) = \begin{cases} h, & \text{if } (x, y) \notin S_{op}, \\ 0, & \text{if } (x, y) \in S_{op}, \end{cases} \quad (7)$$

in which S_{op} is the defined domain of the opening. For circular opening shape, the point C will be the center of the circle and S_{op} can be expressed as

$$S_{op}(x, y) = \left(\frac{x}{r} \right)^2 + \left(\frac{y}{r} \right)^2 - 1 \quad (8)$$

where r is the circle radius, $D(x, y) = \frac{E}{12(1-\nu^2)} h(x, y)^3$, and $w = w(x, y, t)$ represents the deflection of the plate in z -direction at time t . The deflection in z -direction can be expressed as

$$w(x, y, t) = \Phi(x, y)Q(t), \quad (9)$$

By using the non-dimensional variables $\zeta = \frac{x}{L_x}, \eta = \frac{y}{L_y}$, $\Phi(x, y)$ can be expressed as $\Phi(\zeta, \eta) = [\phi_1, \phi_2, \dots, \phi_{n_p}]$, an $1 \times n_p$ matrix consisting of the admissible functions, and $Q(t) = [q_1, q_2, \dots, q_{n_p}]^T$ is an $n_p \times 1$ vector consisting of general coordinates, in which n_p is the number of admissible functions to approximate the deflection and $h(\zeta, \eta) = h(x, y)$, $D(\zeta, \eta) = D(x, y)$. By substituting Eq. (9) into Eqs. (5) and (6) results in Eq. (10).

$$T_p = \frac{1}{2} \dot{Q}^T M_p \dot{Q}, \quad U_p = \frac{1}{2} Q^T K_p Q, \quad (10)$$

$$M_p = \rho L_x L_y M_d, \quad K_p = \frac{L_y}{L_x^3} K_d, \quad (11)$$

$$M_d = \int_0^1 \int_0^1 \Phi^T \Phi h(\zeta, \eta) d\zeta d\eta, \quad (12)$$

$$K_d = \int_0^1 \int_0^1 \left[\frac{\partial^2 \Phi^T}{\partial \zeta^2} \frac{\partial^2 \Phi}{\partial \zeta^2} + \alpha^4 \frac{\partial^2 \Phi^T}{\partial \eta^2} \frac{\partial^2 \Phi}{\partial \eta^2} + \nu \alpha^2 \left(\frac{\partial^2 \Phi^T}{\partial \zeta^2} \frac{\partial^2 \Phi}{\partial \eta^2} + \frac{\partial^2 \Phi^T}{\partial \eta^2} \frac{\partial^2 \Phi}{\partial \zeta^2} \right) + 2(1-\nu) \alpha^2 \frac{\partial^2 \Phi^T}{\partial \zeta \partial \eta} \frac{\partial^2 \Phi}{\partial \zeta \partial \eta} \right] D(\zeta, \eta) d\zeta d\eta, \quad (13)$$

where $\alpha = L_x/L_y$ is the aspect ratio of the plate. The integration will be calculated using the numerical integration method, hence the functions $h(\zeta, \eta)$ and $D(\zeta, \eta)$ will be treated as factors according to considerations in Eqs. (7) and (8). The admissible function matrix $\Phi(\zeta, \eta)$ can be expressed by the admissible functions in x -direction as ψ_ζ and in y -direction as ψ_η

$$\Phi_i(\zeta, \eta) = \psi_{\zeta_i}(\zeta)\psi_{\eta_i}(\eta), \quad i = 1, 2, \dots, n_p. \quad (14)$$

The admissible functions are selected to be a combination of hyperbolic and trigonometric functions as described in [16], which satisfy the clamped–clamped boundary conditions required for the analysis in this paper.

3.2. Enclosure and coupled system modeling

The model has been enlarged to include the enclosure physics, see Fig. 5. The sound pressure inside the enclosure can be described as a combination of rigidly walled enclosure modes. These modes are the product of cosine functions in a three-dimensional space with one supplementary 2D expansion, which overcomes the discontinuity issues at the opening of the plate, creating the pressure admissible functions, as described in [26,27]. The Lagrange functions for the enclosure can be calculated as follows:

$$L_c = U_c - T_c - W_c - W_{c\&p} \quad (15)$$

where T_c , the kinetic energy of the enclosure, is calculated as follows:

$$T_c = \frac{1}{2\rho_a\omega^2} \int_0^{L_x} \int_0^{L_y} \int_0^{L_z} \left[\left(\frac{\partial p}{\partial x} \right)^2 + \left(\frac{\partial p}{\partial y} \right)^2 + \left(\frac{\partial p}{\partial z} \right)^2 \right] dx dy dz \quad (16)$$

where ω is the angular frequency and p is the pressure admissible function. U_c , the potential energy of the enclosure, can be calculated as follows:

$$U_c = \frac{1}{2\rho_a c^2} \int_0^{L_x} \int_0^{L_y} \int_0^{L_z} p^2 dx dy dz \quad (17)$$

W_c , the loss energy from the enclosure at the opening, can be calculated as follows:

$$W_c = -\frac{1}{2} \int_{S_{op}} \frac{\left(p \frac{\partial p}{\partial z} \right) \Big|_{z=L_x}}{i\omega(-\rho_a i\omega)} dS_{op} \quad (18)$$

where i is the imaginary number $\sqrt{-1}$. The coupling between the acoustic pressure inside the enclosure and the plate can be formed using the work done principle as follows:

$$W_{c\&p} = \int_{S_p} \Phi p dS_p \quad (19)$$

where Φ represents the deflection of the plate in the z -direction as in Eq. (9). Eq. (4) can be adapted to include the coupling term as follows:

$$L_p = U_p - T_p + W_{c\&p} \quad (20)$$

After substituting the deflection expression of the plate from Eq. (9) and the pressure field expression inside the enclosure into the Lagrangian energy functions (Eq. (16) to (19)), and implementing the Rayleigh–Ritz procedure, the dynamical system can be described in the harmonic form as

$$(-\omega^2 M_{\text{total}} + K_{\text{total}})V = Q \quad (21)$$

where

$$K_{\text{total}} = \begin{bmatrix} K_c - C_c & 0 \\ C_{c\&p} & K_p \end{bmatrix}, \quad M_{\text{total}} = \begin{bmatrix} M_c & -\rho C_{c\&p} \\ 0 & M_p \end{bmatrix} \quad (22)$$

The matrices K_c and M_c denote the stiffness and mass matrices of the enclosure, while C_c and $C_{c\&p}$ are associated with the energy loss at the opening interface and energy coupling at the plate interface, respectively. The detailed computations for these matrix elements are described in [48]. V and Q represent the system response and the generalized external pressure vectors, respectively, and Q can be expressed as

$$Q = p \Big|_{\substack{x=x_s \\ y=y_s \\ z=z_s}} P_{ex},$$

where P_{ex} is the pressure harmonic excitation as the sound source at (x_s, y_s, z_s) inside the enclosure. To better represent the real system, the total damping matrix C_{total} is introduced and estimated using the formula $C_{\text{total}} = 2\zeta_d \sqrt{K_{\text{total}} M_{\text{total}}}$, where ζ_d is the damping ratio. For the coupled model, the selected ζ_d is 0.0002 which better represents the system compared to the experimental results. The final dynamical equation of the system can be expressed as

$$(-\omega^2 M_{\text{total}} + K_{\text{total}} + i\omega C_{\text{total}})V = Q \quad (23)$$

3.3. Sound power estimation

The total sound power has been estimated using the radiation resistance matrix for the plate structure and the pressure at the opening interface. The radiation resistance, as described in [31,32], is used to estimate the sound power, $\bar{I}_{\text{rad}}$, at each frequency, ω (rad/s), and can be expressed as

$$\bar{I}_{\text{rad}}(\omega) = \Psi^H \mathbf{R}(\omega) \Psi, \quad (24)$$

where superscript H denotes the complex conjugate transpose, Ψ is the plate velocity, and $R(\omega)$ is the radiation resistance matrix at the frequency ω . Dividing the S_p into $N_{x_p} \times N_{y_p}$ elements, Ψ can be expressed as a matrix of $[\Psi_{s1}, \Psi_{s2}, \Psi_{s3}, \dots]$ in which Ψ_{sm} is the plate velocity calculated at the location s of the m th element in the plate, and the size of the matrix is $1 \times (N_{x_p} \times N_{y_p})$. Ψ_{sm} can be expressed as

$$\Psi_{sm} = -i\omega V_p \sum_{i=0}^n \sum_{j=0}^n \Psi_{\zeta i} \Psi_{\eta j} |_{s} \quad (25)$$

where V_p is the last n^2 elements of the system response V . The radiation resistance matrix $R(\omega)$ can be expressed as a matrix of $N_{x_p} \times N_{y_p}$ elements in which the $n_{x_p} n_{y_p}$ th element is

$$R_{n_{x_p} n_{y_p}}(\omega) = \frac{\omega \rho_a (A_s)^2}{4\pi} \cdot \frac{\sin(kr_{n_{x_p} n_{y_p}})}{r_{n_{x_p} n_{y_p}}} \quad (26)$$

where $r_{n_{x_p} n_{y_p}}$ is the distance between n_{x_p} th and n_{y_p} th elements, $k = \omega/c$ is the wavenumber, and A_s is the element area. The sound leakage has been estimated using the pressure information at the opening interface. The S_{op} interface is divided into N_c elements. For consistency, the elements' numbers for both S_p and S_{op} are selected to keep the same element size in both cases. The pressure at the opening interface can be expressed as

$$P_{n_c} = V_c p|_{s_c} \quad (27)$$

where s_c is the location of the n_c element on the S_{op} interface, and V_c is the first n_c^3 elements of the system response V ; hence, the sound power loss at the opening can be expressed as

$$\bar{I}_{\text{loss}}(\omega) = \frac{\sum_{n_c=0}^{N_c} P_{n_c, rms}^2}{\rho_a c} A_s, \quad (28)$$

where $P_{n_c, rms} = \frac{1}{\sqrt{2}} P_{n_c}$ is the root mean square of P_{n_c} . From Eqs. (24) and (28), the total sound power can be estimated as follows:

$$\bar{I}_{\text{total}}(\omega) = \bar{I}_{\text{rad}}(\omega) + \bar{I}_{\text{loss}}(\omega), \quad (29)$$

4. Experimental rig and model validation

The experimental rig consists of a rectangular steel plate mounted to a concrete enclosure and equipped with a loudspeaker inside the enclosure. The loudspeaker is generating a random, broadband noise. The concrete walls of the enclosure provide a high degree of sound attenuation; hence, the majority of the acoustic energy that exits the enclosure is transmitted through the steel plate or the opening, depending on the studied configuration. Two Beyerdynamic MM1 measurement microphones are used to capture the sound pressure at a distance of 0.1 m from the plate and mounted on a carriage along with a laser vibrometer in Fig. 6. The carriage is designed to control the movement of the vibrometer along with the microphones in the horizontal and vertical directions to capture the measurements at the required positions.

The plate's vibration was measured across a consistent grid of 22×20 points, or 440 total points, spaced at intervals of 0.02 m. The estimated signal energy obtained for specific frequencies of structural resonances at all measurement locations was aggregated to extract the mode shapes. The results obtained from the simulation of FEM and calculations of analytical models are compared to the measurements from the experimental rig for the three different configurations, as shown in Fig. 7. In configuration 1, the solid plate acts as a benchmark for maximum sound insulation, making it ideal for applications demanding high sound reduction. As a benchmark, it is also suitable for comparing configurations within the proposed approaches and the experimental results. Configuration 2 places a circular opening with a diameter of $d = 0.1$ m in the center of the plate, which is an advantage for accommodating central tubes or facilitating symmetric interaction with the surroundings, which is necessary in different engineering applications. In configuration 3, the opening's center has $C_x = 0.105$ m and $C_y = 0.0975$ m as coordinates relative to the adjacent corner of the opening.

This configuration enhances the uniform distribution of stress across the plate for applications that incorporate additional components, such as volumetric actuators in active noise control, see [5]. Table 1 compares the natural frequencies of the first 11 modes, covering a frequency range characteristic of many noise-generating devices. However, this assumption does not limit the considerations; the number of modes can be increased if needed. To maintain the paper's structural clarity, Figs. 8, 9, and 10 illustrate the 1st, 5th, and 11th mode shapes of the analyzed cases. The coupled system mode shapes from the FEM are presented in Appendix.

In addition, the vibration and sound power responses of the experiment, analytical model, and FEM model are depicted in Figs. 11, 12, and 13, respectively. The results reveal a substantial correlation between the experimental measurements and the computed values, covering aspects such as natural frequencies, mode shapes, and both vibration and sound power responses.

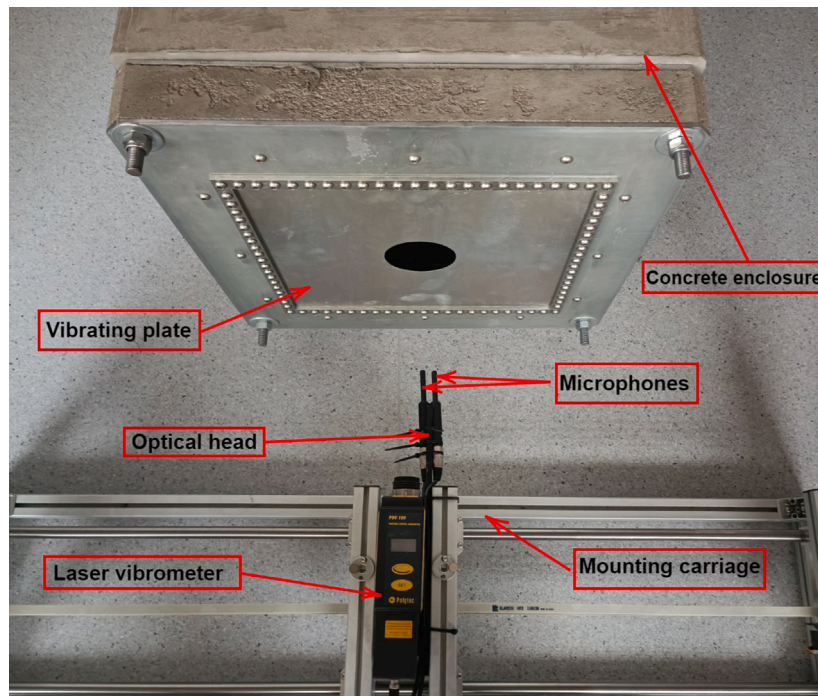


Fig. 6. Overview of the experimental setup with a vibrating plate with central circular opening, a laser vibrometer, and measurement microphones mounted on a carriage for two-directional positioning.

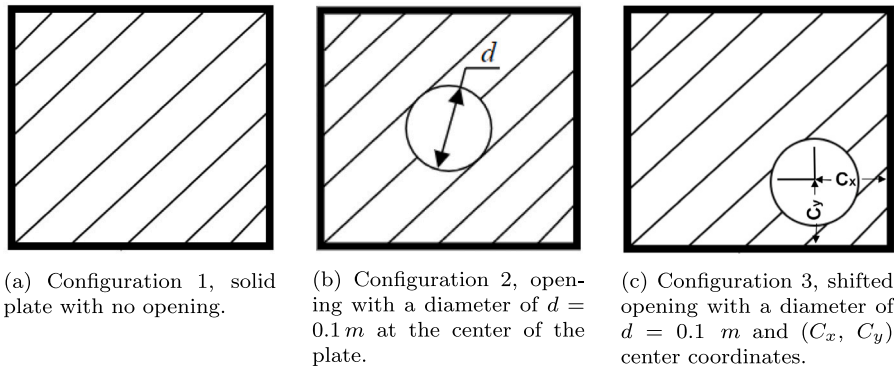


Fig. 7. Vibrating plate with different opening configurations.

Table 1

The first 11 natural frequencies comparison in (Hz), Exp: Experimental results, ANM: Analytical Model, and FEM: Finite Element Model.

Mode	Configuration 1			Configuration 2			Configuration 3		
	Exp	ANM	FEM	Exp	ANM	FEM	Exp	ANM	FEM
1	54	54.9	58.2	58	57.4	55.9	51	52.9	52.4
2	103	106.8	105.6	NaN	101.9	101.7	97	104.1	106
3	117.5	116.9	115.5	111	112.1	110.8	112	116.1	116.7
4	163.5	165.1	162.5	149	151.4	156.8	154	163.8	163.4
5	184.5	189.9	187.5	180	183.2	186.2	183	186.7	187.7
6	214	212.9	211.5	220	228.5	225.1	206	212.3	213.9
7	242	244.3	240.6	225	229.1	234.5	241	237.3	243.9
8	259.5	258.9	255.1	247	244.6	249.1	280	272.8	277.2
9	295	299.6	298.1	280	288.7	284.7	290	295.2	295.9
10	337	335.5	329.7	328	330.9	325.1	329	329.2	335.5
11	343	341.7	339.9	333	334.1	340.9	342	338.2	342.2

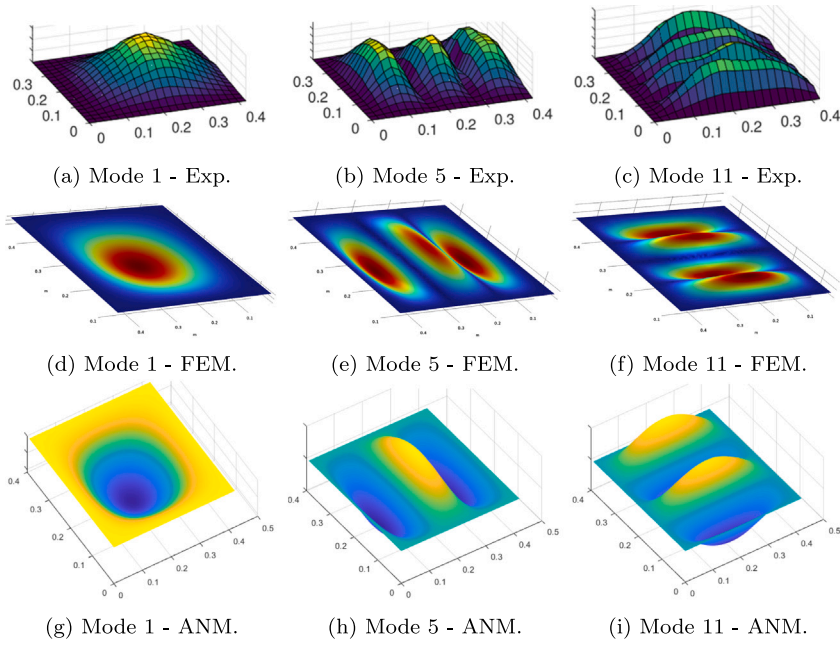


Fig. 8. The 1st, 5th, and 11th mode shapes for Configuration 1, dimensions in [m].

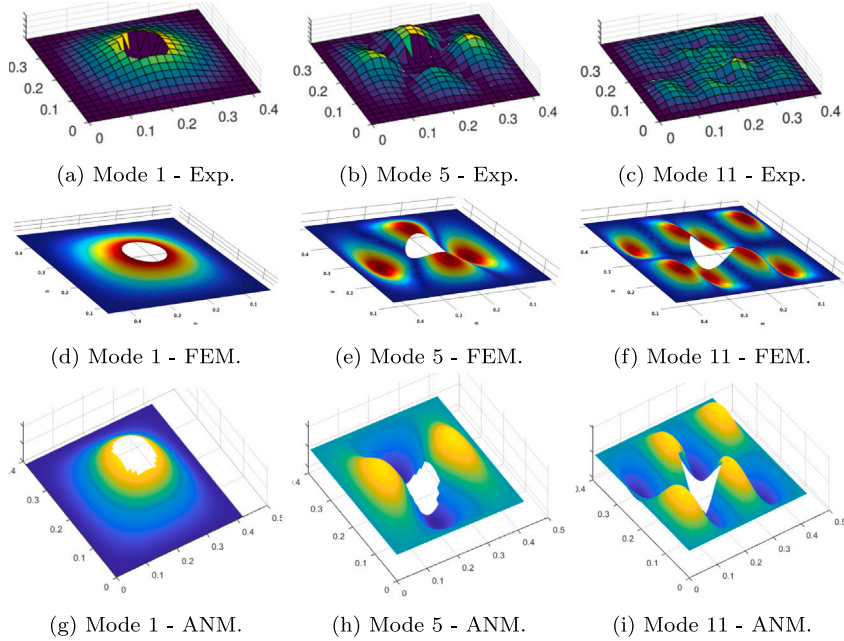


Fig. 9. The 1st, 5th, and 11th mode shapes for Configuration 2, dimensions in [m].

5. Simulation and results

The vibro-acoustic behavior of the system was analyzed to understand the effect of the opening shape and location on both vibration and sound power responses. As shown in Fig. 14(a), in contrast to the solid plate in configuration 1, configuration 4 is a benchmark for comparison, indicative of the highest sound power leakage in scenarios where the plate is absent. In Fig. 14(b), configuration 5 features five centralized circular openings with a total opening area equal to this of configurations 2 and 3. Fig. 14(c) presents configuration 6 with four rectangular openings, each equidistant from the edges and aligned with the center of the corresponding edge, to examine the influence of different opening interface contours and their effects near the edges. Configurations

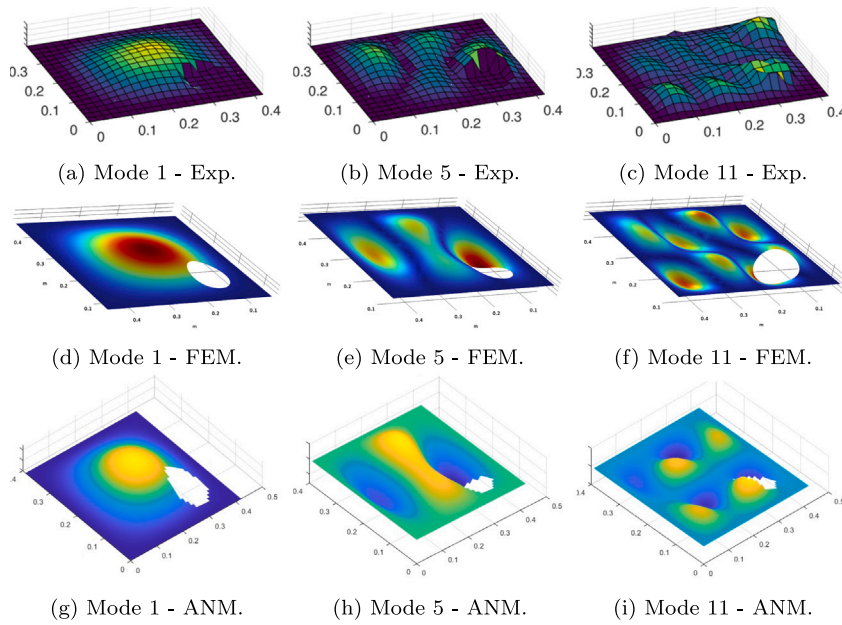


Fig. 10. The 1st, 5th, and 11th mode shapes for Configuration 3, dimensions in [m].

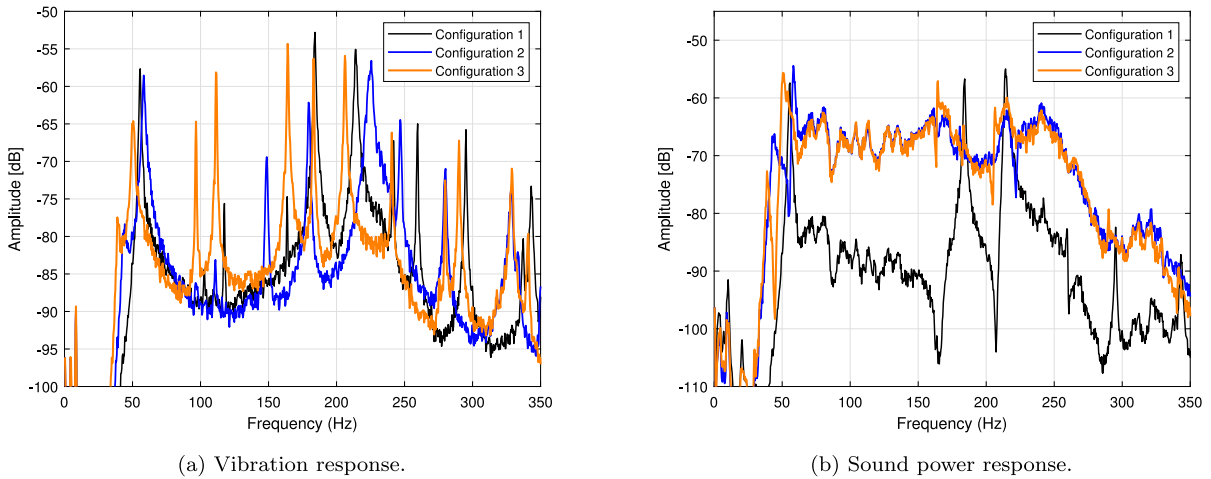
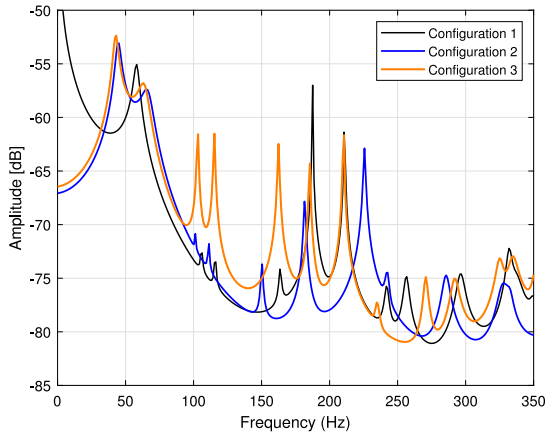


Fig. 11. Sound power and vibration response from the experimental measurements.

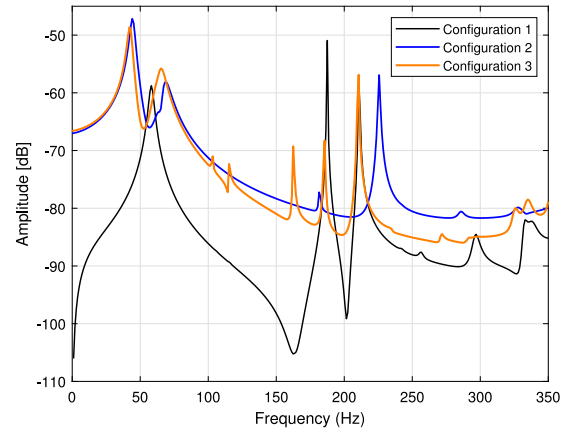
5 and 6 can have an advantage for applications where precise cable routing is essential, allowing for better organization and airflow around the cables for enhanced cooling performance.

Fig. 15 shows how the plate vibrates at different frequencies from 1 to 350 Hz. Compared to Configuration 1, Configuration 2 shows a shift in resonant frequencies due to the reduction in stiffness. Configuration 3's vibration resonances vary slightly in amplitude, as the shifted opening affects the plate's modal behavior unevenly, causing a significant increase in the resonance between 100 and 200 Hz.

Configuration 5, characterized by multiple circular openings, presents closely spaced resonances due to the stiffness characteristics of the plate's central region. The results for Configuration 6 suggest it achieves damping effectiveness comparable to Configuration 1. This is because, within the specified frequency range, the mode shapes of the plate remain largely uninfluenced by the location of the openings. These openings are situated close to the nodal lines where the mode shapes exhibit minimal movement, thus not significantly affecting the vibrational characteristics of the plate. The sound power response, as presented in Fig. 16, shows that in Configuration 4, the system has no plate, hence it exhibits the highest sound power levels across the frequency spectrum. Meanwhile, Configuration 1 has no leakage and displays the lowest sound power levels, with a reduction of around 35 dB for frequencies below 187 Hz and approximately 25 dB for higher frequencies. Comparing the configurations with openings, each of them has a total opening area of 0.007854 m^2 , hence the plate mass reduction is the same in all configurations but the stiffness

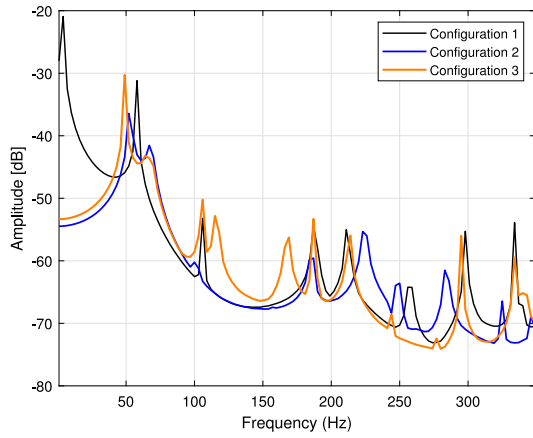


(a) Vibration response.

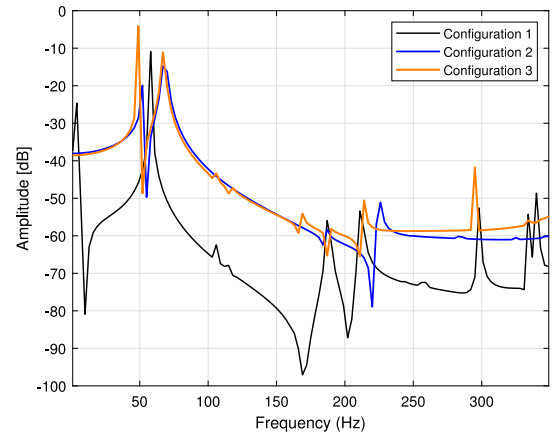


(b) Sound power response.

Fig. 12. Sound power and vibration response from the analytical model.

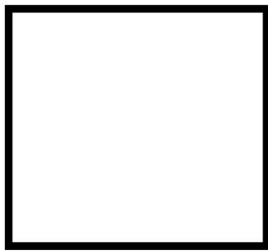


(a) Vibration response.

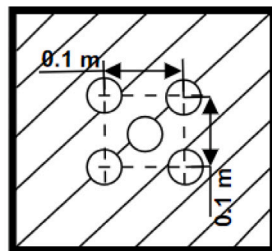


(b) Sound power response.

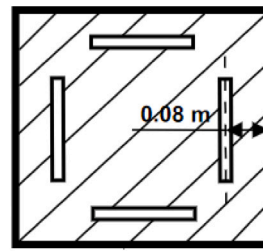
Fig. 13. Sound power and vibration response from the FEM model.



(a) Configuration 4, Void, no plate



(b) Configuration 5, multiple circular openings at center with 0.045 m diameter.



(c) Configuration 6, 4 rectangular openings each with 0.01 m width and 0.2 m length.

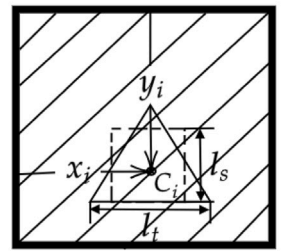
(d) Layout for squares or triangles, C_i is the center of opening i , and l_s and l_t are side lengths.

Fig. 14. Various plate configurations evaluated for vibro-acoustic performance.

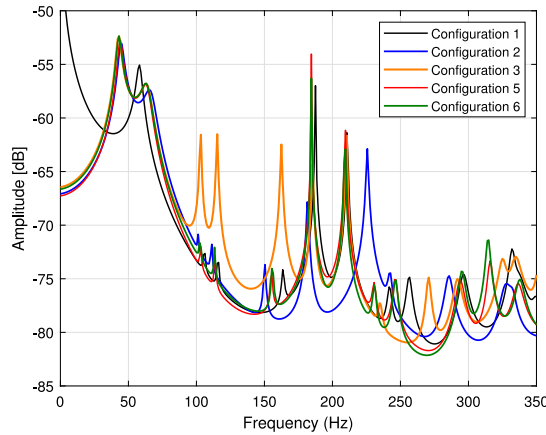


Fig. 15. Vibration response of the plate with different configurations.

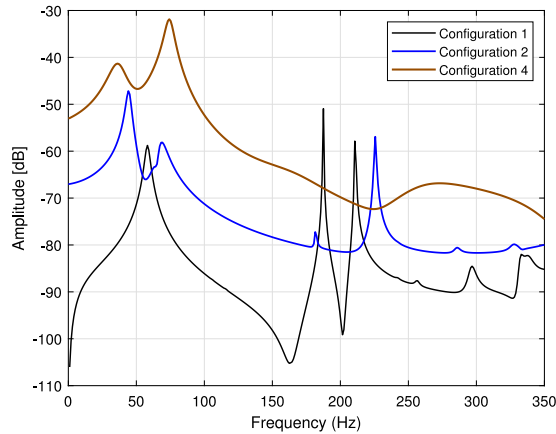


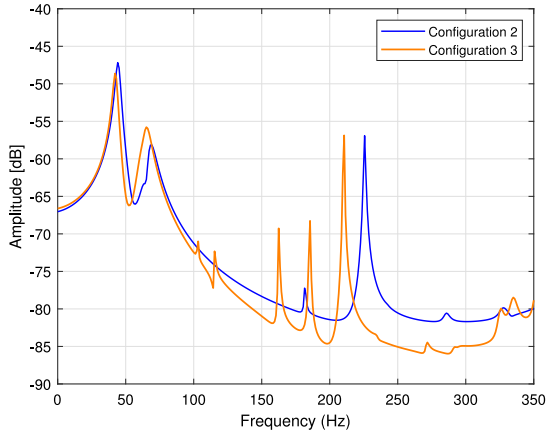
Fig. 16. Sound power response of the system for Configuration 1, 2, and 4.

distribution will be different. Configuration 2 shows a middle performance between Configurations 1 and 4, with a reduction around 17 dB across all the spectrum response, and it is used as a benchmark to test the other configurations with the openings in Fig. 17.

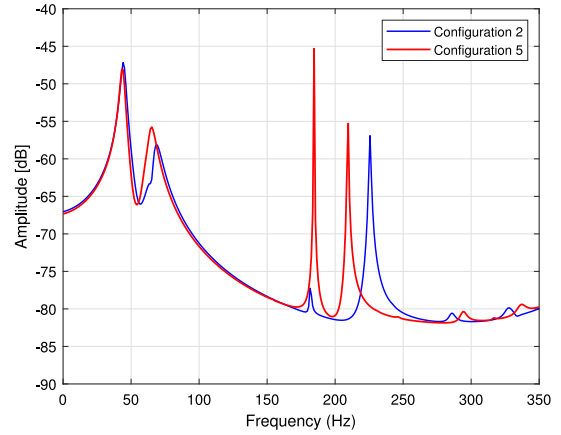
In Fig. 17(a), the sound power response for configurations 2 and 3 reveals the impact of opening placement in relation to the system mode shapes. Configuration 2's central opening aligns more with the plate's anti-nodal areas for different mode shapes, increasing the leaked sound power, while Configuration 3's shifted opening interacts more with nodal points, reducing the sound power at frequencies above 100 Hz but also amplifying the resonance amplitudes for both vibration and sound power response. Fig. 17(b) shows that the concentration of multiple openings in Configuration 5's central area does not impact the sound power response across the frequency range. However, this concentration increases the stiffness of the central region, which amplifies the resonance in the mid-range.

In Fig. 17(c), the rectangular openings of Configuration 6 lead to lower sound power across most frequencies, but this configuration amplifies the resonance around 185 Hz. The location of the openings allows the plate to vibrate in a way that is similar to Configuration 1, as the stiffness is spread out more evenly and the openings align more with the anti-nodal areas of the plate for the majority of the system's mode shapes. Additionally, the layout, as described in Fig. 14(d) and Table 2, is designed to accommodate multiple openings. The shape of the openings can be made by combining squares, as seen in Configurations 7 and 8, or equilateral triangles, as in Configurations 9 and 10.

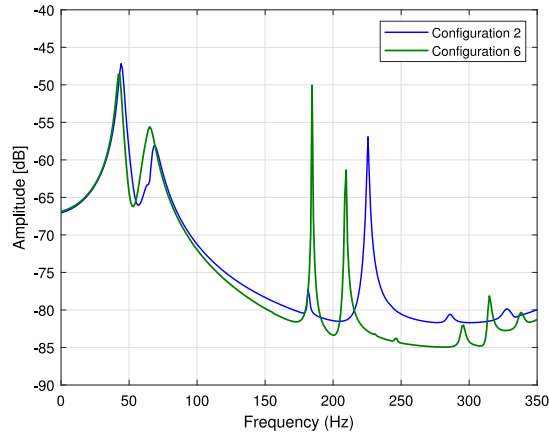
The above results indicate that, with the assumption of equal opening area, the configurations with a continuous perimeter have their maximum resonating frequency at a lower amplitude than the other configurations. As shown in Table 2, Configurations 2, 3, 8, and 10 resonate with less amplitudes compared to the others. In addition, maintaining the same opening area – excluding the solid and void configurations – results in similar average sound power levels indicating the impact of sound power loss. The



(a) Sound power response for configurations 2 and 3.



(b) Sound power response for configurations 2 and 5.



(c) Sound power response for configurations 2 and 6.

Fig. 17. Sound power responses of the plate for various configurations.

configurations examined in this paper are examples of how the design of openings – both their shape and location on vibrating plates – can effectively shape the frequency responses (FR) to comply with the requirements of vibro-acoustic systems while respecting the opening constraints, such as the total opening area. This demonstrates the potential of implementing the proposed models into a methodology to shape the frequency response of vibro-acoustic systems in various engineering applications that use vibrating plates with openings as noise barriers. Fig. 18 graphically summarizes the proposed shaping FR method, ensuring that the design of openings meets both the acoustic performance and the opening constraints.

6. Conclusion

The comprehensive analysis of vibration and sound power responses for the mixed interface of the vibro-acoustic system shows how the design of the openings on the vibrating plate can be employed to shape the system frequency response. This paper compares examples of different configurations that could be used in a variety of engineering applications that require the use of vibrating plates with openings as noise barriers. These configurations revealed that having discontinuities in the opening contour leads to higher resonances in the acoustic response of the system in the mid-range of the considered frequencies. Also, if the opening is placed asymmetrically, the system shows additional resonances at these frequencies.

The novelty of the proposed method lies in directly linking an adaptable analytical model, which allows for varied designs in the shape and location of the opening interface, to the system's acoustic response. It is integrated with a COMSOL Multiphysics® FEM, providing visualization of the interaction between the plate vibration and the acoustic pressure and increasing the potential of the proposed method.

Table 2

Comparison table of configurations, PA: denotes the peak amplitude at resonance frequency (RF) in the sound power response above 80 Hz, AVG: represents the average of this response for the same range.

Configuration	Opening	Design dimensions [m]				RF [Hz]	PA [dB]	AVG[dB]
		(x_1, y_1)	(x_2, y_2)	(x_3, y_3)	l_s, l_t			
1		See Fig. 7(a)				188	-50.9	-88.11
2		See Fig. 7(b)				226	-56.9	-78.08
3		See Fig. 7(c)				211	-56.87	-80.28
4		See Fig. 14(a)				80	-37.30	-65.15
5		See Fig. 14(b)				185	-45.31	-77.93
6		See Fig. 14(c)				185	-50.05	-79.88
7		(0.07,0.195)	(0.21,0.195)	(0.35,0.195)	0.051	185	-54.62	-80.69
8		(0.21,0.195)	(0.261,0.195)	(0.195,0.195)	0.051	217	-61.4	-80.18
9		((0.07,0.195)	(0.21,0.195)	(0.35,0.195)	0.0775	185	-51.376	-80.69
10		(0.21,0.195)	-	-	0.1344	222	-59.03	-80.1

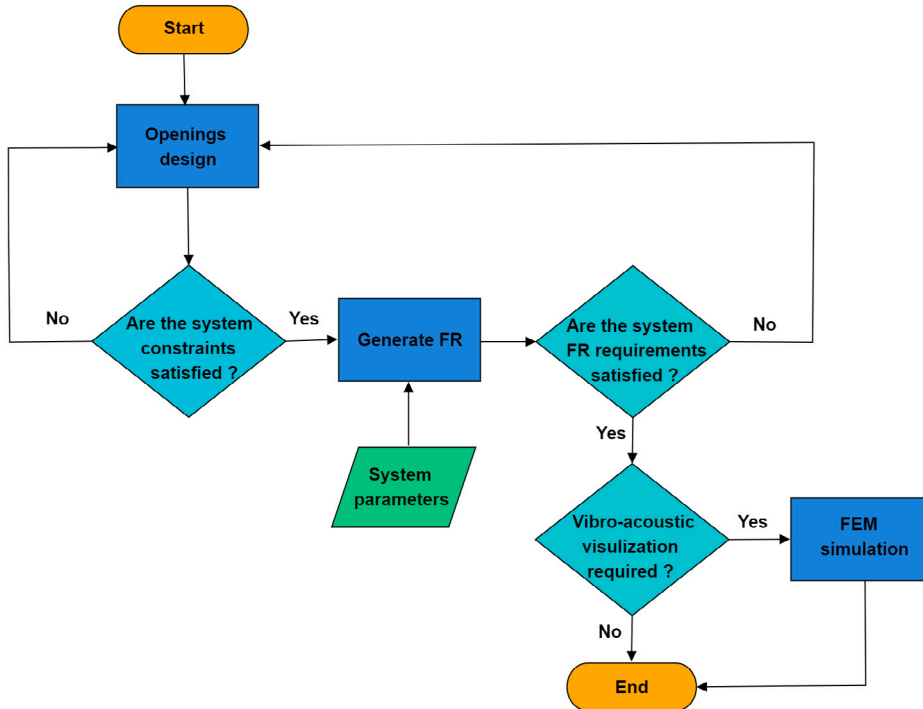


Fig. 18. Schematic of the proposed shaping frequency response (FR) method of the vibrating plate with openings, correlating opening configurations with FR to satisfy system requirements within set constraints.

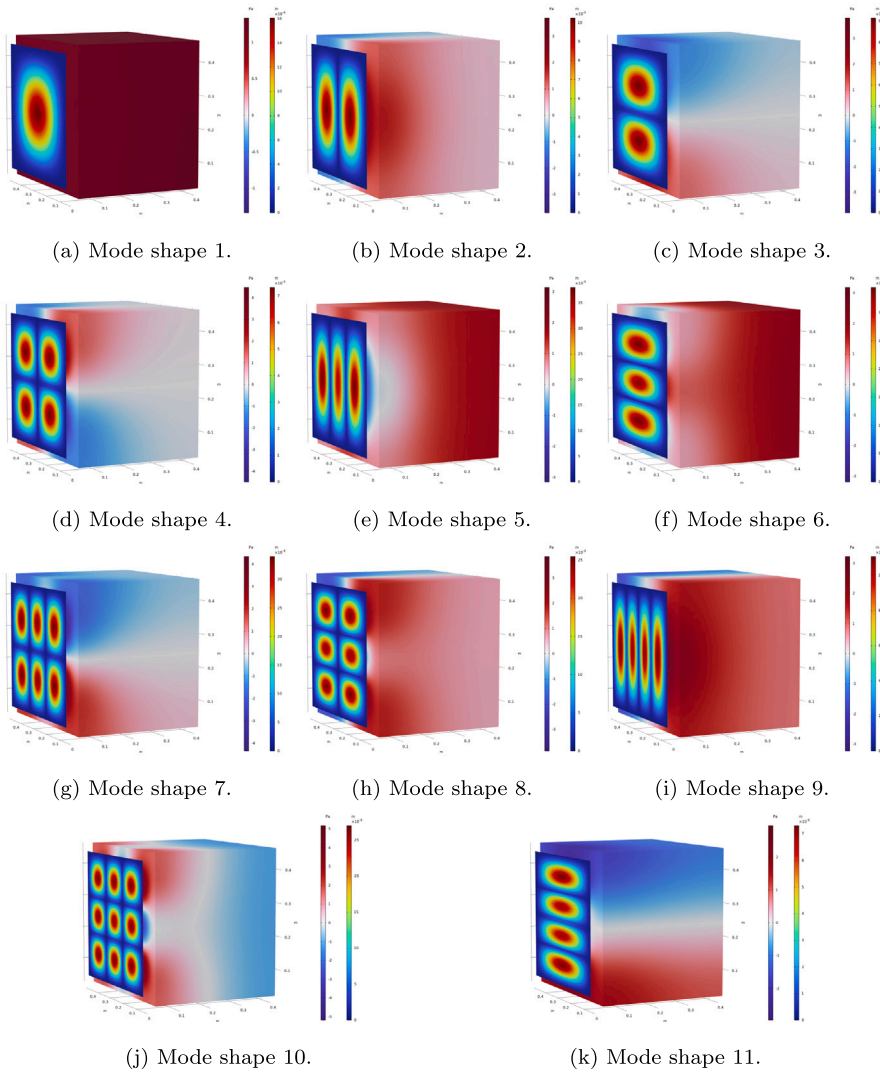


Fig. 19. Mode shapes of Configuration 1.

However, the method has some limitations. It can be challenging to determine the optimal shape and location for the openings to achieve specific acoustic performance. Employing an optimization algorithm to fine-tune the design parameters of the openings is a viable approach. Additionally, considering this method as a passive tool for shaping frequency response in vibrating plates with openings creates opportunities to integrate it with active control techniques, enhancing its effectiveness in various applications, such as noise reduction.

CRediT authorship contribution statement

Khaled Said Ahmed Maamoun: Writing – review & editing, Writing – original draft, Visualization, Validation, Software, Resources, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Stanislaw Wrona:** Writing – review & editing, Writing – original draft, Validation, Supervision, Resources, Formal analysis, Data curation, Conceptualization. **Marek Pawelczyk:** Writing – review & editing, Writing – original draft, Validation, Supervision, Project administration, Funding acquisition, Conceptualization. **Hamid Reza Karimi:** Writing – review & editing, Validation, Supervision, Formal analysis, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

Acknowledgments

The work was supported by the project IN-NOVA: Active reduction of noise transmitted into and from enclosures through encapsulated structures, which has received funding from the European Union's Horizon Europe program under the Marie Skłodowska-Curie grant agreement no. 101073037 and in part by the Italian Ministry of University and Research under grant 'Learning-based Model Predictive Control by Exploration and Exploitation in Uncertain Environments' (PRIN PNRR 2022 fund, ID P2022EXP2W).

Appendix. FEM coupled system modes shapes

See Figs. 19–21.

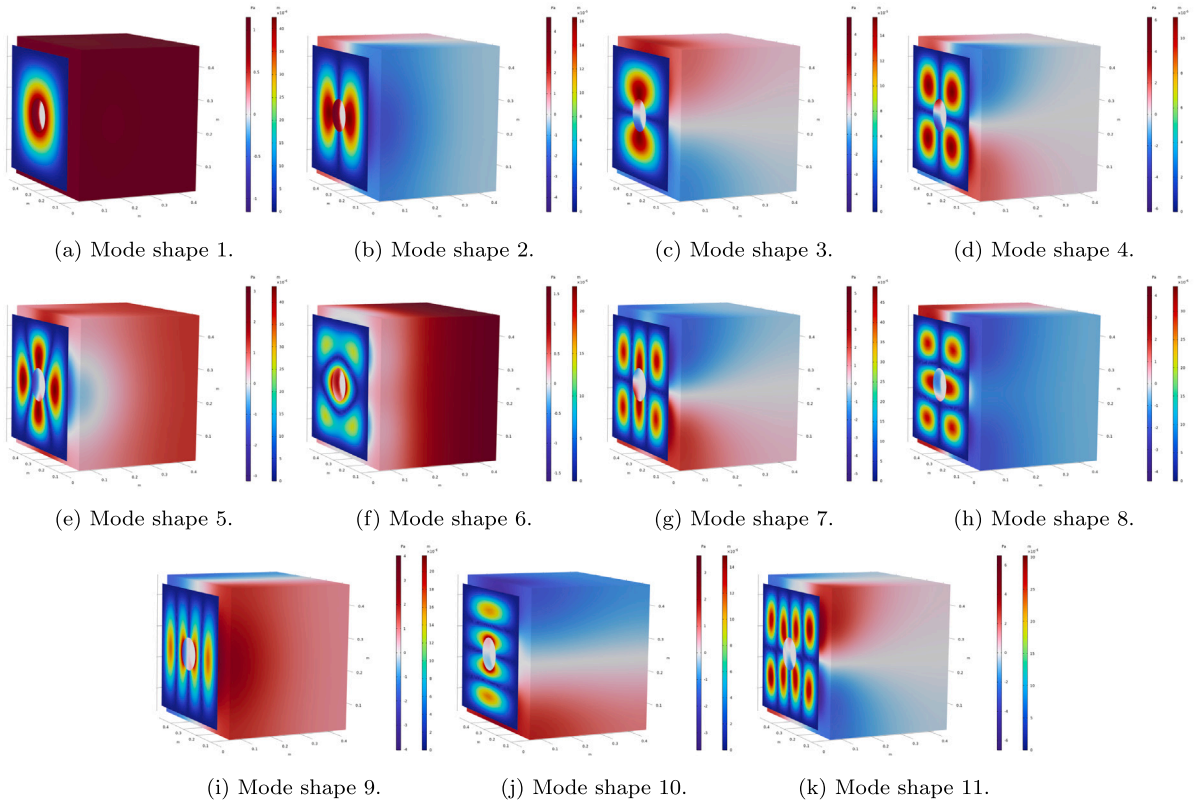


Fig. 20. Mode shapes of Configuration 2.

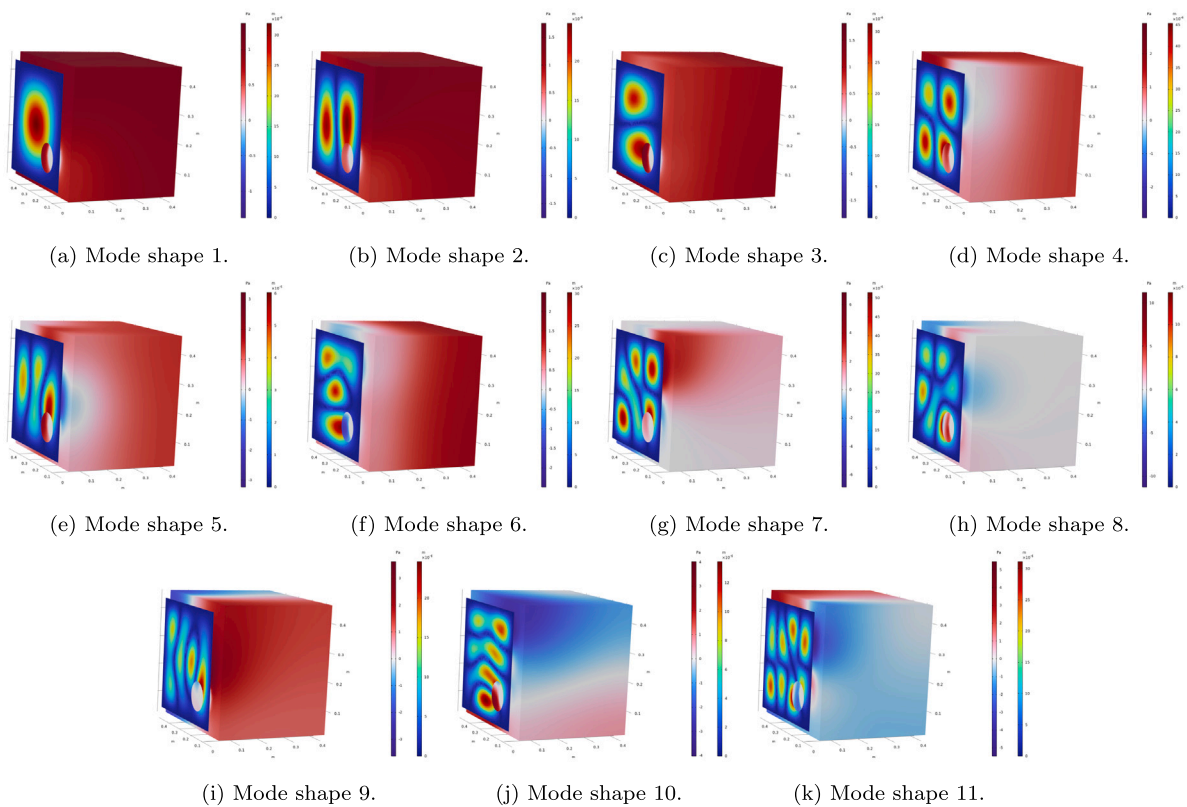
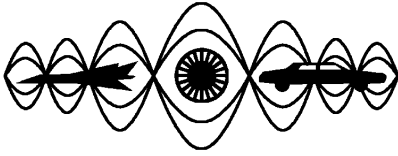


Fig. 21. Mode shapes of Configuration 3.

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ANALYZING THE IMPACT OF SOUND SOURCE POSITIONING ON FREQUENCY RESPONSE OF VIBRATING PLATES WITH OPENINGS IN VIBRO-ACOUSTIC SYSTEMS

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In various engineering applications, the openings in the plate structure of a vibro-acoustic coupled system have a crucial role for various functions. This paper explores the influence of sound source positioning inside an enclosure on the vibrational and acoustic responses of a vibrating plate with openings, which is used as a noise barrier in vibro-acoustic systems. We implement an analytical model based on the Rayleigh-Ritz method to estimate sound power in a free-field environment. The precision of our results is verified through experimental testing. Our findings offer valuable insights into the behavior of vibro-acoustic systems and highlight their practical applications in engineering design.

Keywords: Vibrating plates with openings; Vibro-acoustic systems; Sound power estimation; Sound source positioning

1. Introduction

In a coupled vibro-acoustic system, the openings in the plate structure play vital roles in various engineering fields like automotive, aerospace, marine engineering, and naval architecture. These roles include heat dissipation, routing electrical cables, providing access to personnel, and reducing weight. It's crucial to control acoustic characteristics without compromising the structure's functionality and integrity [1, 2]. In practice, structural plates face challenges from noise sources' locations, affecting their ability to reduce noise. Investigating noise source locations and their impact on plate responses is crucial [3]. Strategic placement of the openings relative to the noise source can enhance passive transmission loss, improving system functionality. Effectively shaping the frequency response of the interface is highly beneficial [4]. Exploring vibro-acoustic systems starts with modeling vibrating plates with openings, posing analytical challenges noted by [5]. Research mainly relies on semi-analytical and numerical methods. The Rayleigh-Ritz method (RRM) is frequently used to predict natural frequencies, even for plates with rectangular openings [6]. Progress in studying free vibration of rectangular plates with openings is notable. Chebyshev-Lagrange and spectro-geometric methods, refining RRM, have advanced understanding under general boundary conditions [7]. The independent coordinate coupling method (ICCM) has also contributed, applied to diverse configurations like point support perforated rectangular plates and those with multiple openings [8]. Literature acknowledges frequency analysis limitations in plates with openings, notably below critical frequency, due to variable stiffness. A meshfree approach, the discrete Ritz

method (DRM), was developed, accommodating variable stiffness at openings. Particularly beneficial for metal thin plates, where critical frequencies exceed working ranges [9]. Research on vibro-acoustic systems covers plate vibrations, internal sound fields of enclosures, and the mixed interface, including sound transmission through openings. Some studies model acoustic fields in openings as a superposition of plane waves, effective for smaller openings in low-frequency conditions [10]. The Kobayashi potential method is preferred for larger openings due to its computational efficiency over superposition [11]. Additionally, treating openings as massless membranes within the mixed interface and integrating sound transmission into a unified theoretical model represent significant advancements [12].

In the study presented in [13], authors developed a mathematical model for a vibro-acoustic system comprising six walls and twelve plates, using coupled partial differential equations. Their findings emphasize that the source location significantly impacts fluid-structure interaction, affecting the system's overall response. In [3], authors introduced a finite element model for an enclosure with six aluminum vibrating plates. They investigated effects of varying noise source locations, confirming alterations can enhance system functionality, aiding complex noise control operations.

In this paper, the effect of the noise source locations is studied for a vibro-acoustic system consists of a rectangular enclosure with a single vibrating plate with circular opening. The results have been validated against experimental measurements for the vibration and sound power. The paper consists of 4 sections. Section 2 describes the analytical model with the vibro-acoustic coupled system and the sound power radiation. In Section 3, the models' results are compared with the measurements from the vibrometer and two Beyerdynamic MM1 microphones in the experimental setup. Different case study for the noise source locations are introduced in Section 4.

2. Analytical modeling of the vibro-acoustic system

2.1 Modeling of free vibrating plate

The Kirchhoff-Love theory of thin plates is used to model the free vibrating rectangular plate [14]. The plate is assumed to be isotropic and homogeneous, excluding the region of the opening in which the thickness of the plate will be zero. The Rayleigh-Ritz method (RRM) is used to define an approximate solution to the eigenvalue equation to calculate the natural frequencies and mode shapes of the plate vibration. Considering a rectangular plate with side lengths L_x in the x -direction and L_y in the y -direction, see Fig. 1. The Lagrange functions of the plate can be calculated as

$$L_P = U_P - T_P, \quad (1)$$

The kinetic and potential energies, T_P and U_P respectively, of the rectangular plate can be expressed as

$$T_P = \frac{1}{2} \rho \int_0^{L_x} \int_0^{L_y} h(x, y) \dot{w}^2 dx dy, \quad (2)$$

$$U_P = \frac{1}{2} \int_0^{L_x} \int_0^{L_y} D(x, y) \left[\left(\frac{\partial^2 w}{\partial x^2} \right)^2 + \left(\frac{\partial^2 w}{\partial y^2} \right)^2 + 2v \left(\frac{\partial^2 w}{\partial x^2} \frac{\partial^2 w}{\partial y^2} \right) + 2(1-v) \left(\frac{\partial^2 w}{\partial x \partial y} \right)^2 \right] dx dy, \quad (3)$$

where $h(x, y)$ is the plate thickness adjusted for the opening by becoming zero within S_{op} , as in Fig. 1. $D(x, y) = \frac{E}{12(1-v^2)} h(x, y)^3$, and $w = w(x, y, t)$ represents the deflection of the plate in z -direction at time t . The deflection in z -direction can be expressed as

$$w(x, y, t) = \Phi(x, y)Q(t), \quad (4)$$

By using the non-dimensional variables $\zeta = \frac{x}{L_x}, \eta = \frac{y}{L_y}$, $\Phi(x, y)$ can be expressed as $\Phi(\zeta, \eta) = [\phi_1, \phi_2, \dots, \phi_{n_p}]$, an $1 \times n_p$ matrix consisting of the admissible functions, and $Q(t) = [q_1, q_2, \dots, q_{n_p}]^T$ is an $n_p \times 1$ vector consisting of general coordinates, in which n_p is the number of admissible functions to approximate the deflection and $h(\zeta, \eta) = h(x, y), D(\zeta, \eta) = D(x, y)$. By substituting Eq. 4 into Eqs. 2 and 3 results in Eq. 5.

$$T_P = \frac{1}{2} \dot{Q}^T M_p \dot{Q}, \quad U_P = \frac{1}{2} Q^T K_p Q, \quad (5)$$

$$M_p = \rho L_x L_y M_d, \quad K_p = \frac{L_y}{L_x^3} K_d, \quad (6)$$

$$M_d = \int_0^1 \int_0^1 \Phi^T \Phi h(\zeta, \eta) d\zeta d\eta, \quad (7)$$

$$K_d = \int_0^1 \int_0^1 \left[\frac{\partial^2 \Phi^T}{\partial \zeta^2} \frac{\partial^2 \Phi}{\partial \zeta^2} + \alpha^4 \frac{\partial^2 \Phi^T}{\partial \eta^2} \frac{\partial^2 \Phi}{\partial \eta^2} + v \alpha^2 \left(\frac{\partial^2 \Phi^T}{\partial \zeta^2} \frac{\partial^2 \Phi}{\partial \eta^2} + \frac{\partial^2 \Phi^T}{\partial \eta^2} \frac{\partial^2 \Phi}{\partial \zeta^2} \right) + 2(1-v) \alpha^2 \frac{\partial^2 \Phi^T}{\partial \zeta \partial \eta} \frac{\partial^2 \Phi}{\partial \zeta \partial \eta} \right] D(\zeta, \eta) d\zeta d\eta, \quad (8)$$

where $\alpha = L_x/L_y$ is the aspect ratio of the plate. The integration will be calculated using the numerical integration method. The admissible function matrix $\Phi(\zeta, \eta)$ can be expressed by the admissible functions in x -direction as ψ_ζ and in y -direction as described in [15], and set to satisfy the clamped-clamped boundary conditions required for the analysis in this paper.

2.2 Enclosure and coupled system modeling

The model has been enlarged to include the enclosure physics, see Fig. 1. The sound pressure inside the enclosure can be described as a combination of rigidly walled enclosure modes. These modes are the product of cosine functions in a three-dimensional space with one supplementary 2D expansion, which overcomes the discontinuity issues at the opening of the plate, creating the pressure admissible functions, as described in [16], which implements the Lagrange functions for the kinetic and potential energy of the system.

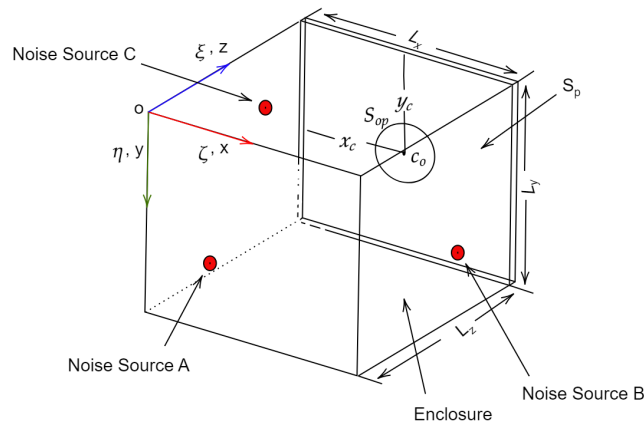


Figure 1: Enlarged enclosure system coupled with mixed interface of vibrating rectangular plate with circular opening S_{op} . S_p is the structural plate interface. Noise sources: A (0.16, 0.35, 0.1) m, B (0.36, 0.33, 0.34) m, and C (0.1, 0.06, 0.2) m.

The loss energy from the enclosure at the opening, W_c , can be calculated as follows:

$$W_c = -\frac{1}{2} \int_{S_{op}} \frac{\left(p \frac{\partial p}{\partial z}\right) \Big|_{z=L_x}}{i\omega(-\rho_a i\omega)} dS_{op} \quad (9)$$

where i is the imaginary number $\sqrt{-1}$, ω is the angular frequency, and p is the pressure admissible function. The coupling between the acoustic pressure inside the enclosure and the plate can be formed using the work done principle as follows:

$$W_{c\&p} = \int_{S_p} \Phi p dS_p \quad (10)$$

where Φ represents the deflection of the plate in the z -direction as in Eq. 4. After substituting the deflection expression of the plate from Eq. 4 and the pressure field expression inside the enclosure into the Lagrangian energy functions, and implementing the Rayleigh-Ritz procedure, the dynamical system can be described in the harmonic form as

$$(-\omega^2 M_{\text{total}} + K_{\text{total}})V = Q \quad (11)$$

where

$$K_{\text{total}} = \begin{bmatrix} K_c - C_c & 0 \\ C_{c\&p} & K_p \end{bmatrix}, \quad M_{\text{total}} = \begin{bmatrix} M_c & -\rho C_{c\&p} \\ 0 & M_p \end{bmatrix} \quad (12)$$

The matrices K_c and M_c denote the stiffness and mass matrices of the enclosure, while C_c and $C_{c\&p}$ are associated with the energy loss at the opening interface and energy coupling at the plate interface, respectively. The detailed computations for these matrix elements are described in [4]. V and Q represent the system response and the generalised external pressure vectors, respectively, and Q is expressed as

$$Q = p \Big|_{\substack{x=x_s \\ y=y_s \\ z=z_s}} P_{ex},$$

where P_{ex} is the pressure harmonic excitation of a mono-pole noise source at (x_s, y_s, z_s) inside the enclosure. To better represent the real system, the total damping matrix C_{total} is introduced and estimated using the formula $C_{\text{total}} = 2\zeta_d \sqrt{K_{\text{total}} M_{\text{total}}}$, where ζ_d is the damping ratio. For the coupled model, the selected ζ_d is 0.0002 which better represents the system compared to the experimental results. The final dynamical equation of the system can be expressed as

$$(-\omega^2 M_{\text{total}} + K_{\text{total}} + i\omega C_{\text{total}})V = Q \quad (13)$$

The total sound power is estimated using the radiation resistance matrix for the plate structure and the pressure at the opening interface. The radiation resistance, as described in [3], is used to estimate the sound power, $\bar{\Pi}_{\text{rad}}$, at each frequency, ω (rad/s), and can be expressed as

$$\bar{\Pi}_{\text{rad}}(\omega) = \Psi^H \mathbf{R}(\omega) \Psi, \quad (14)$$

where superscript H denotes the complex conjugate transpose, Ψ is the plate velocity, and $R(\omega)$ is the radiation resistance matrix at the frequency ω .

The sound power loss at the opening can be expressed as

$$\bar{\Pi}_{\text{loss}}(\omega) = \frac{\sum_{n_c=0}^{N_c} P_{n_c, rms}^2}{\rho_a c} A_s, \quad (15)$$

where s_c is the location of the n_c element on the S_{op} interface, and V_c is the first n_c^3 elements of the system response V . $P_{n_c,rms} = \frac{1}{\sqrt{2}}P_{n_c}$ is the root mean square of P_{n_c} and P_{n_c} is the pressure at the opening interface and is expressed as

$$P_{n_c} = V_c p|_{s_c} \quad (16)$$

From Eqs. 14 and 15, the total sound power can be estimated as follows:

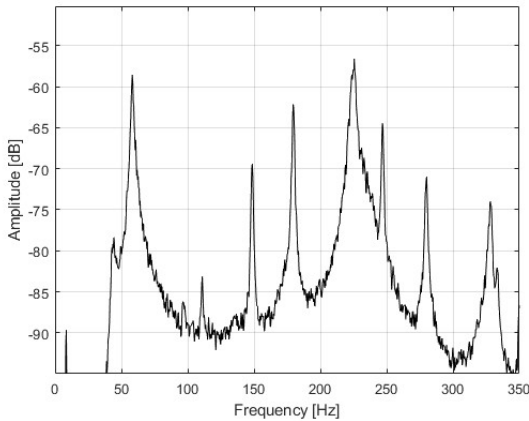
$$\bar{\Pi}_{total}(\omega) = \bar{\Pi}_{rad}(\omega) + \bar{\Pi}_{loss}(\omega), \quad (17)$$

3. Experimental rig and model validation

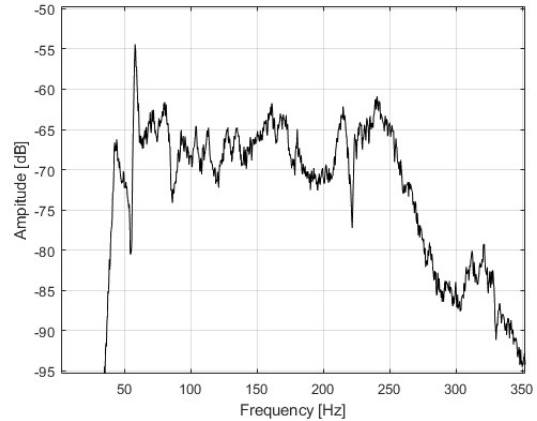
The experimental setup includes a steel plate mounted to a concrete enclosure housing a loudspeaker generating random, broadband noise. The concrete walls of the enclosure minimize sound transmission, directing most energy through the steel plate or the opening. Sound pressure measurements utilize two Beyerdynamic MM1 microphones positioned 0.1 m from the plate on a movable carriage, alongside a laser vibrometer shown in Fig. 2. This carriage enables precise positioning horizontally and vertically for data collection. Vibration measurements were taken across a 22×20 grid totaling 440 points, each spaced 0.02 m apart. Comparison between model and experimental rig measurements, presented in Table. 1, covers natural frequencies of the first 11 modes, typical for noise-generating devices, with flexibility to increase mode considerations. Furthermore, Figs.3 and 4 illustrate vibration and sound power responses, highlighting a significant correlation between experimental and computed values, including natural frequencies and power responses.

Table 1: Comparison of the first 11 natural frequencies (Hz) between Experimental (Exp) and Analytical Model (ANM).

Mode	Natural Frequencies (Hz)										
	1	2	3	4	5	6	7	8	9	10	11
Exp	58	NaN	111	149	180	220	225	247	280	328	333
ANM	57.4	101.9	112.1	151.4	183.2	228.5	229.1	244.6	288.7	330.9	334.1



(a) Vibration response.



(b) Sound power response.

Figure 3: Vibration and sound power response from the experimental measurements.

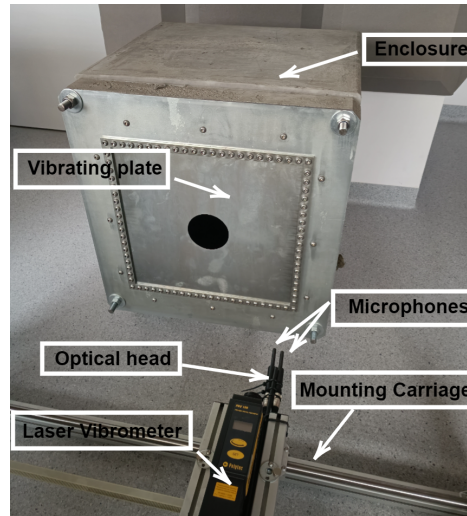
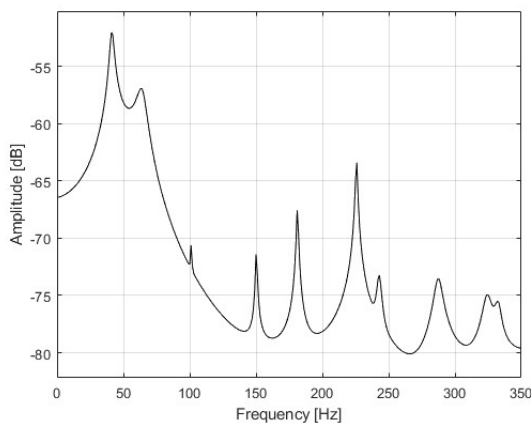
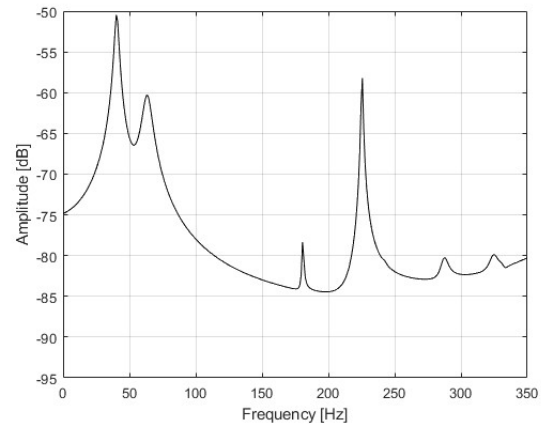


Figure 2: Overview of the experimental setup with a vibrating plate with central circular opening, a laser vibrometer, and measurement microphones mounted on a carriage for two-directional positioning.



(a) Vibration response.

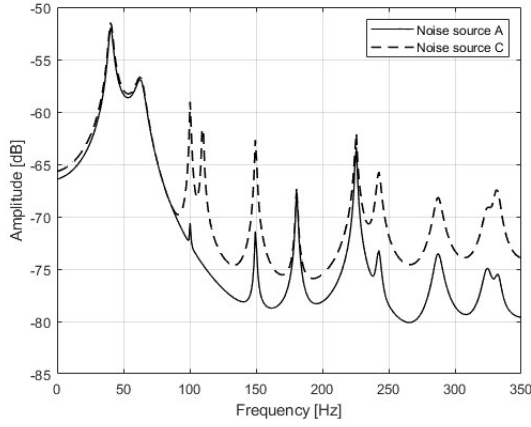


(b) Sound power response.

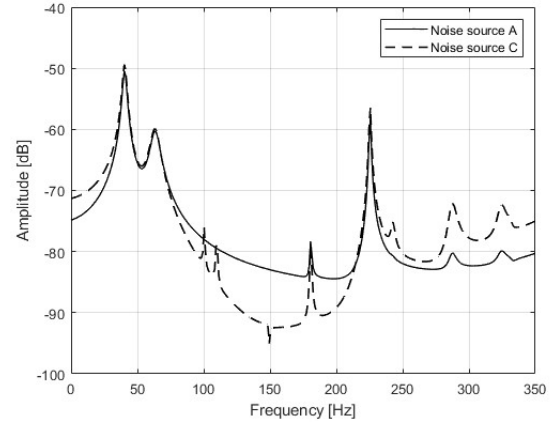
Figure 4: Model vibration and sound power response with noise source A.

4. Simulation and Results

The analysis delved into the vibro-acoustic behavior of the system to assess how the noise source's location impacts both vibration and sound power responses. Fig.5 illustrates the consequence of moving the noise source closer to the vibrating plate at location B, manifesting in amplified vibration response amplitude across all modes, along with a reduction in average sound power below 226 Hz. In Fig. 6, a general location C was selected, which exhibited amplified vibration amplitudes across the entire frequency range, accompanied by an increase in sound power above 226 Hz. These results underscore the influence of the noise source's relative position on the system's ability to excite different modes. Moreover, they highlight how the relative positioning of the opening and the noise source impacts the system's sound power response.

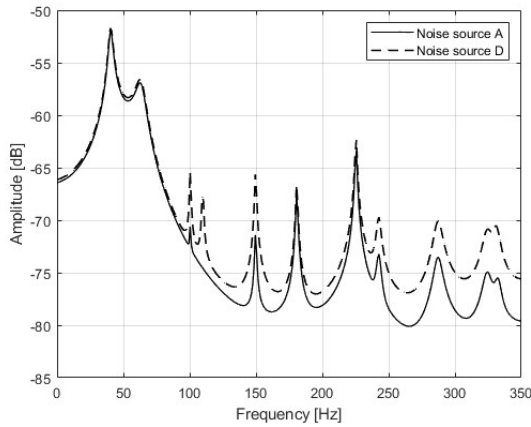


(a) Vibration response.

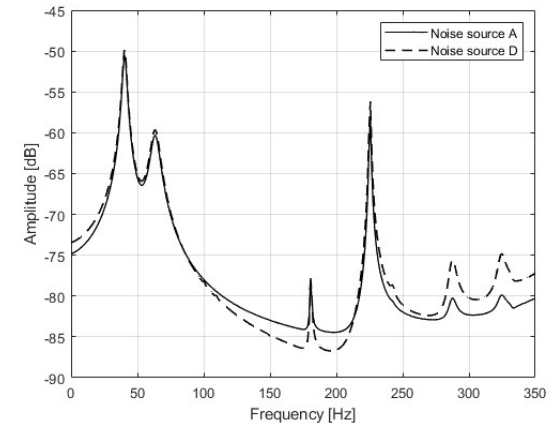


(b) Sound power response.

Figure 5: Comparison of vibration and sound power response for noise source A vs noise source B.



(a) Vibration response.



(b) Sound power response.

Figure 6: Comparison of vibration and sound power response for noise source A vs noise source C.

5. Conclusions

This paper offers insights into noise source positioning within an enclosure and its influence on vibrational and acoustic responses of a plate with openings, vital for vibro-acoustic systems. Employing an analytical model based on the Rayleigh-Ritz method and validated through experimental testing. The results highlight noise source placement's impact on vibration excitation and sound power distribution. Relocating the noise source within the enclosure led to distinct changes in vibration amplitudes and sound power levels, with notable variations across frequencies. These findings hold implications for engineering design and applications. Understanding the relationship between noise source positioning and system responses allows engineers to optimize systems for enhanced noise control and vibration mitigation.

6. Acknowledgments

The work was supported by the project IN-NOVA: Active reduction of noise transmitted into and from enclosures through encapsulated structures, which has received funding from the European Union's Horizon Europe program under the Marie Skłodowska-Curie grant agreement no. 101073037.

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Optimizing design of openings in vibrating plates for enhanced vibro-acoustic performance using a genetic algorithm approach

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ARTICLE INFO

Keywords:

Vibrating plates with openings
Genetic algorithm optimization
Vibro-acoustic system enhancement
Structural-acoustic coupling
Sound power reduction

ABSTRACT

Optimizing structural openings in vibro-acoustic systems is essential for balancing functional needs with acoustic performance in various engineering applications. This paper presents an efficient optimization approach for designing openings in plate structures within vibro-acoustic systems, aimed at enhancing acoustic performance while maintaining structural integrity. A genetic algorithm framework is developed to determine the optimal shapes and locations of openings simultaneously, utilizing an analytical model based on hyperbolic and trigonometric admissible functions within a master-slave software architecture. The proposed method accommodates a variety of opening requirements for different systems and employs a cost function that evaluates both the average and peaks of sound power responses to identify the designs with the highest fitness. This results in significant reductions in sound power levels compared to median and worst-case designs. Additionally, the method analyzes how variations in opening locations, numbers, and shapes impact acoustic performance, providing guidance for opening designs that avoid undesired acoustic outcomes and improve noise barrier performance in vibro-acoustic systems.

1. Introduction

Incorporating openings in plate structures is crucial in several engineering applications, especially those involving vibro-acoustic coupled systems. These structures are utilized in various fields, such as household appliances, manufacturing and vehicle engineering. They serve essential functions such as heat dissipation, routing electrical cables, providing personnel access, and weight reduction [1,2]. Effectively managing acoustic properties while maintaining structural integrity and performance is crucial, as this not only ensures the efficiency and longevity of various systems but also has significant implications for ecological sustainability and human health [3,4]. Thus, significant effort has been made by researchers to design and control the sound power characteristics between plate structures and the internal sound field in vibro-acoustic systems [5].

This interest is reflected by different studies employing semi-analytical approaches [6,7], finite element method (FEM) [8,9], and recent works utilizing active control [10,11]. Consequently, significant efforts have been undertaken to optimize the wall structure design for vibro-acoustic systems. Using FEM, authors in [12] employed the plate stiffness as a variable to optimize the sound and vibration responses of the plate. Meanwhile, a multi-material topology optimization with variable thickness method was developed in [13] based on an alternating active-phase algorithm. In [14], the authors implemented a memetic algorithm

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<https://doi.org/10.1016/j.jsv.2024.118847>

Received 19 September 2024; Received in revised form 14 November 2024; Accepted 15 November 2024

Available online 25 November 2024

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(MA) to find the optimal positions of additional masses and ribs to shape the frequency response of the vibrating plate. The particle swarm optimization algorithm was applied by the authors in [15] to determine the best dimensions of the embossed panel section to reduce the sound power. Additionally, composite plates have been strongly considered for optimization, serving as noise barriers and vibration suppressors in vibro-acoustic systems in [16,17]. Recently, in [18], the authors utilized the Method of Moving Asymptotes (MMA) to optimize the topology of the core between two plates to increase the Sound Transmission Loss (STL). In [19], the authors employed a genetic algorithm (GA) with a surrogate model to construct a multi-objective optimization for rectangular plates with acoustic black holes (ABH) technique, which is efficient in passive vibration control for mid-higher frequency ranges [20,21]. Additionally, in active control of the sound power response, the authors in [22] used a modified heuristic genetic algorithm (MHGA) to place piezoelectric actuators on plate structures to suppress the vibration modes. Meanwhile, in [23], the authors utilized a memetic algorithm to increase the controllability of the plates, reducing the acoustic response by 5 dB. However, the literature shows a gap in finding an optimal design for openings that can differ in shapes and locations on plate structure for vibro-acoustic systems, as the openings add further complexity to effectively shape the frequency response of the mixed interface [24] and need to be carefully addressed to reach the best possible design.

To determine the optimal designs for openings on vibrating plates, the modeling of vibro-acoustic systems with mixed interfaces needs to be flexible enough to accommodate different shapes and various locations of the openings [25]. The literature clarifies that the semi-analytical approach is frequently characterized by techniques based on the Rayleigh–Ritz method (RRM), which is used for the prediction of natural frequencies of rectangular plates [26] with rectangular openings [27,28]. However, a Rayleigh–Ritz multiple terms method was suggested to overcome the lack of the prediction accuracy in case of large openings. The Chebyshev–Lagrange and spectro-geometric methods were employed as enhancements of the Rayleigh–Ritz method [29,30]. The method showed good agreement with the FEM in ANSYS software for both transverse and in-plane vibration predictions and demonstrated flexibility in changing the opening parameters and boundary conditions. However, the work was not extended to predict the sound radiation characteristics of the plates. Furthermore, an independent coordinate coupling method was introduced and applied to perforated rectangular plates with point support, as well as those with multiple openings [31,32]. The results showed a reduction of the computational time by avoiding complicated numerical integration but did not give flexibility to change the opening parameters. In [33], the authors used Nitsche based isogeometric analysis (IGA) to study the free vibration of vibrating plates with openings. Addressing limitations related to incorporating openings in vibrating plates, previous studies have indicated that using variable stiffness may restrict frequency analysis below the critical frequency [34]. However, a mesh-free approach known as the discrete Ritz method (DRM) has been developed to counter this issue, particularly for metal thin plates where the critical frequency often exceeds the considered frequency range [35]. Regarding the sound estimation, appropriate Green’s function can be employed, as in [36,37]. In [38,39], the authors used the radiation resistance matrix to extend the sound power estimation in the free field. In [40], the authors recently proposed a spectral-geometry method (SGM) to study sound radiation from vibrating plates with arbitrary-shaped openings. However, they neglected the structure-fluid coupling, which is a crucial factor for different applications. In [41], sound transmission through structures and openings was unified into a singular theoretical acoustic model. In [42], the authors treated the opening as a massless membrane on the vibrating plate. For larger openings relative to the wavelength, the Kobayashi potential method was utilized in [43,44] due to its computational efficiency compared to the superposition technique. The authors in [45] applied the concept of superposition of plane waves to a diffused field, building on the approach modeled in [46], which focused on the acoustic opening. Finally, the multi-dimensional Fourier series method is used to describe the sound pressure in a rectangular [47] and a trapezoidal enclosure [48] with different impedance boundary conditions. Using the Legendre polynomial series, the authors in [49,50] studied the acoustic characteristics of a rectangular cavity packed by a vibrating plate. Unlike the system addressed in [40], in recent work, the authors in [51] considered both the structure-acoustic coupling and the transmission loss at the opening interface. They revealed that the shape and location of the openings strongly influence the stiffness characteristics of vibrating plates. However, they did not provide a deterministic optimization procedure to find the optimal opening designs for systems with different requirements.

To fill the gap of designing the vibro-acoustic systems with optimal parameters for the openings on their structure-acoustic interface, this paper proposes an optimization methodology for determining the shapes and locations of these openings. The analytical model used in this paper is based on hyperbolic and trigonometric admissible functions for both plate displacements and sound pressure in the enclosure. This reduces the computational effort required by other methods in [49,50]. It also accounts for the sound radiation as well as the transmission coupling with the ability to accommodate a variety of opening shapes. The model is validated experimentally by the authors in [51], and reformulated in this paper to fit the GA framework in terms of input–output structure and the openings’ deterministic functions handling. This reformulation is designed to accommodate the GA cost function within the proposed master–slave software architecture. For systems with different opening requirements, the optimized designs meet the objective of reducing sound power responses, making the vibrating plates with openings effectively function as noise barriers.

The main contributions of this paper are threefold: (i) Proposing a novel optimization method for openings in vibrating plate structures within vibro-acoustic systems to enhance acoustic performance; (ii) Utilizing a GA framework, incorporating an analytical model for the vibro-acoustic system within a master–slave software architecture; (iii) Investigating the effects of varying the locations, numbers, and shapes of openings on the sound power characteristics of vibro-acoustic systems, which helps in avoiding the improper selection of design parameters that could lead to undesired sound power characteristics. The remainder of this paper is structured as follows: Section 2 addresses a description of the analytical model of the vibro-acoustic system and the sound power estimation. In Section 3, the genetic algorithm framework and the cost function are developed with system architecture. The discussion and the optimization results for a variety of systems are presented in Section 4. Finally, the paper is concluded by Section 5.

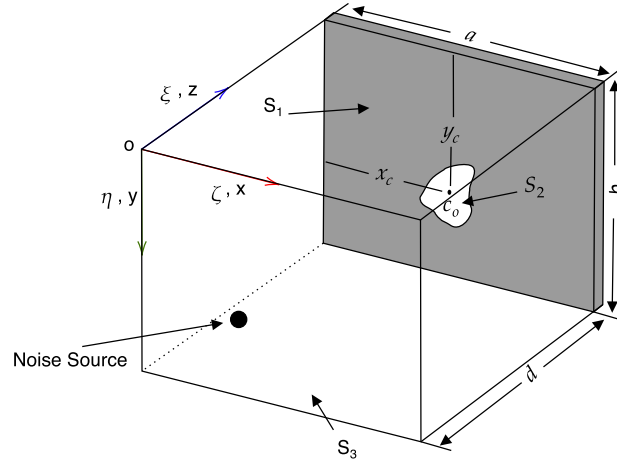


Fig. 1. Analytical model configuration including the enclosure domain S_3 , the mixed interface of a vibrating plate with structural interface S_1 , and arbitrary shaped opening interface S_2 . The position (x_c, y_c) is within S_2 and the noise source is positioned at (x_s, y_s, z_s) .

2. Modeling of the vibro-acoustic system

The configuration of the analytical model and the coordinate system under investigation are shown in Fig. 1. It consists of a rectangular enclosure capped with a freely vibrating rectangular plate that accommodates an arbitrary opening interface. The boundary conditions are such that the plate is assumed to be fully-clamped and the other five walls of the enclosure are sound hard, although other boundary conditions can be considered in a similar way.

2.1. Modeling of the plate vibration

The freely vibrating plate is modeled using the Kirchhoff-Love theory of thin plates [52] and assumed to be isotropic and homogeneous. Hence, considering only transverse motion and neglecting the effect of rotary inertia, the kinetic energy of the plate, T_p , can be expressed as:

$$T_p = \frac{1}{2} \rho \iint_{S_1} h(x, y) \dot{w}^2 dx dy, \quad (1)$$

where ρ is the plate density, $h(x, y)$ is the plate variable thickness, and equals zero if the position (x, y) lies within the opening domain S_2 . The deflection of the plate in the z -direction at time t is denoted as $w(x, y, t)$. The initial conditions are given by

$$w(x, y, t_0) = 0, \quad \left. \frac{\partial w(x, y, t)}{\partial t} \right|_{t=t_0} = 0, \quad t > t_0 > 0. \quad (2)$$

The strain energy of the plate, U_p , can be expressed as:

$$U_p = \frac{1}{2} \iint_{S_1} \left\{ D(x, y) \left[\left(\frac{\partial^2 w}{\partial x^2} \right)^2 + \left(\frac{\partial^2 w}{\partial y^2} \right)^2 + 2\nu \left(\frac{\partial^2 w}{\partial x^2} \frac{\partial^2 w}{\partial y^2} \right) + 2(1-\nu) \left(\frac{\partial^2 w}{\partial x \partial y} \right)^2 \right] \right\} dx dy, \quad (3)$$

where E is the Young's modulus, ν is the Poisson ratio, and $D(x, y) = \frac{E}{12(1-\nu^2)} h(x, y)^3$. The model is reformulated in terms of handling the inputs as a variable vector $\vec{v}_o$ to fit the GA framework with the flexibility to switch between different deterministic functions for the opening domains, as discussed in Section 3. Thus, S_2 can be expressed as follows:

$$S_2(x, y) = \text{function}(\vec{v}_o), \quad (4)$$

In the case of an elliptic opening with $\vec{v}_o = [x_c, y_c, \theta_c, l_c]$, Eq. (4) can be expressed as follows:

$$S_2(x, y) = \frac{[(x - x_c) \cos(\theta_c) + (y - y_c) \sin(\theta_c)]^2}{a_c^2} + \frac{[(x - x_c) \sin(\theta_c) - (y - y_c) \cos(\theta_c)]^2}{b_c^2} - 1, \quad (5)$$

where (x_c, y_c) represents the center coordinates of an ellipse, a_c denotes its semi-major axis, and b_c is the semi-minor axis. The element l_c in $\vec{v}_o$ can represent either a_c or b_c , with the other axis being derived from the required opening area. The angle θ_c denotes the angle formed by l_c with the x -axis of the plate in the counterclockwise direction. The function $w(x, y, t)$ can be expressed as follows:

$$w(x, y, t) = \Phi(x, y) V_g(t), \quad (6)$$

where $\Phi(x, y) = [\phi_1, \phi_2, \dots, \phi_{n_p}]$ is an $1 \times n_p$ matrix consisting of the admissible functions, and $V_g(t) = [v_{g1}, v_{g2}, \dots, v_{gn_p}]^T$ is an $n_p \times 1$ vector consisting of general coordinates, in which n_p is the number of the deflection admissible functions. Using non-dimensional variables ζ and η and substituting Eq. (6) into Eqs. (1) and (3), T_p and U_p can be expressed as

$$T_p = \frac{1}{2} \dot{V}_g^T M_p \dot{V}_g, \quad U_p = \frac{1}{2} V_g^T K_p V_g, \quad (7)$$

$$M_p = \rho ab M_{p_l}, \quad K_p = \frac{b}{a^3} K_{p_l}, \quad (8)$$

where M_{p_l} and K_{p_l} are the mass and stiffness matrices of the plate, respectively, in the dimensionless format and can be driven from Eqs. (1) and (3). The admissible function matrix $\Phi_i(\zeta, \eta)$ can be expressed by the admissible functions in x -direction as ψ_ζ and in y -direction as ψ_η

$$\Phi_i(\zeta, \eta) = \sum_{i=0}^{n_p} \sum_{j=0}^{n_p} \psi_{\zeta i} \psi_{\eta j}. \quad (9)$$

The admissible functions are selected to be a combination of hyperbolic and trigonometric functions as described in [53], which satisfy the clamped-clamped boundary conditions required for the analysis in this paper.

2.2. Coupled system modeling

The sound pressure inside the enclosure can be described as a combination of rigidly walled enclosure modes. These modes are the product of cosine functions in a three-dimensional space with one supplementary 2D expansion, which overcomes the discontinuity issues at the opening of the plate, creating the pressure admissible functions, as described in [47,54]. The Lagrange functions of the coupled system require the definition of the loss energy from the enclosure at the opening, W_e , which can be calculated as follows:

$$W_e = -\frac{1}{2} \int_{S_2} \left(p \frac{\partial p}{\partial z} \right) \Big|_{z=d} \frac{1}{\rho_a \omega^2} dS_2, \quad (10)$$

where ω is the angular frequency and p is the pressure admissible function. The coupling between the acoustic pressure inside the enclosure and the plate can be formed using the work done principle as follows:

$$W_{e\&p} = \int_{S_1} w \cdot p dS_1, \quad (11)$$

where w represents the deflection of the plate in the z -direction as in Eq. (6). By implementing the Rayleigh–Ritz procedure, the dynamical system can be described in the harmonic form as

$$(K_{\text{total}} - \omega^2 M_{\text{total}})q = Q, \quad (12)$$

where

$$K_{\text{total}} = \begin{bmatrix} K_e - C_e & 0 \\ C_{pe} & K_p \end{bmatrix}, \quad M_{\text{total}} = \begin{bmatrix} M_e & -\rho C_{ep} \\ 0 & M_p \end{bmatrix}. \quad (13)$$

The matrices K_e and M_e denote the stiffness and mass matrices of the enclosure, respectively, and C_e is associated with the energy loss at the opening interface. These three matrices are of the size $n_e^3 \times n_e^3$ where n_e is the number of admissible pressure functions used to approximate the pressure distribution in the enclosure in each direction. C_{ep} is related to the energy coupling at the plate interface and of the size $n_e^3 \times n_p^2$ and C_{pe} is its transpose. The detailed computations for these matrices elements can be derived from the work described in [55]. Q and q represent the generalized external pressure and the system response vectors, respectively, and Q can be expressed as

$$Q = p \Big|_{\substack{x=x_s \\ y=y_s \\ z=z_s}} P_{ex},$$

where P_{ex} is the pressure harmonic excitation as the sound source at (x_s, y_s, z_s) inside the enclosure.

2.3. Sound power estimation

The estimation of total sound power utilizes the radiation resistance matrix for the plate structure and the pressure at the opening interface. Described in [11,38], the radiation resistance, denoted as $R_{\text{rad}}(\omega)$, is employed to compute the sound power, $\bar{\Theta}_{\text{rad}}$, at each angular frequency, ω , expressed as:

$$\bar{\Theta}_{\text{rad}}(\omega) = \Psi^H \mathbf{R}(\omega) \Psi, \quad (14)$$

where the superscript H denotes the complex conjugate transpose, Ψ is the plate velocity, and $R(\omega)$ is the radiation resistance matrix at the frequency ω . Dividing the S_1 into $N_{x_p} \times N_{y_p}$ elements, Ψ can be expressed as a matrix of $[\Psi_{s1}, \Psi_{s2}, \Psi_{s3}, \dots]$ in which Ψ_{sm} is the plate velocity calculated at the location s of the m th element in the plate, and the size of the matrix is $1 \times (N_{x_p} \times N_{y_p})$. Ψ_{sm} can be expressed as

$$\Psi_{sm} = -i\omega \left[\sum_{i=0}^{n_p} \sum_{j=0}^{n_p} \psi_{\zeta i} \psi_{\eta j} \Big|_s \right] q_p, \quad (15)$$

where q_p is the last n_p^2 elements of the system response q , and i is the imaginary number $\sqrt{-1}$. The radiation resistance matrix $R(\omega)$ can be expressed as a matrix of $N_{x_p} \times N_{y_p}$ elements in which the (n_{x_p}, n_{y_p}) element is

$$R_{n_{x_p}, n_{y_p}}(\omega) = \frac{\omega \rho_a (A_s)^2}{4\pi} \cdot \frac{\sin(kr_{n_{x_p}, n_{y_p}})}{r_{n_{x_p}, n_{y_p}}}, \quad (16)$$

where $r_{n_{x_p}, n_{y_p}}$ is the distance between $n_{x_p}^{\text{th}}$ and $n_{y_p}^{\text{th}}$ elements, $k = \omega/c$ is the wavenumber, and A_s is the element area. The sound leakage has been estimated using the pressure information at the opening interface S_2 which is divided into N_c elements. The pressure at the opening interface can be expressed as

$$P_{n_c} = q_c \cdot p|_{s_c}, \quad (17)$$

where s_c is the location of the n_c element on the S_2 interface, and q_c is the first n_c^3 elements of the system response q ; hence, the sound power loss at the opening can be expressed as

$$\bar{\Theta}_{\text{loss}}(\omega) = \frac{\sum_{n_c=0}^{N_c} P_{n_c}^2}{\rho_a c} A_s, \quad (18)$$

From Eqs. (14) and (18), the total sound power can be estimated as follows:

$$\bar{\Theta}_{\text{total}}(\omega) = \bar{\Theta}_{\text{rad}}(\omega) + \bar{\Theta}_{\text{loss}}(\omega). \quad (19)$$

However, for the optimization purposes, the output of the model need to be reformulated to calculate the maximum sound power peak in the frequency range from ω_1 to ω_2 as follows:

$$\bar{\Theta}_{\text{max}} = \max_{\omega_1 \leq \omega \leq \omega_2} \{ \bar{\Theta}_{\text{total}}(\omega) \}, \quad (20)$$

and the average sound power in the same frequency range as follows:

$$\bar{\Theta}_{\text{avg}} = \frac{1}{N_\omega} \sum_{i=1}^{N_\omega} \bar{\Theta}_{\text{total}}(\omega_i), \quad (21)$$

where N_ω is the number of discrete frequency points in the range of ω_1 to ω_2 . The following exemplary model properties are considered as they are widely used in different engineering applications. The plate is made of steel with a density $\rho = 7850 \text{ kg/m}^3$, Young's modulus $E = 210 \times 10^9 \text{ Pa}$, and Poisson's ratio $\nu = 0.3$. The dimensions of the plate are $a = 0.42 \text{ m}$, $b = 0.39 \text{ m}$, and $h = 0.001 \text{ m}$, representing its length, width, and thickness, respectively. The enclosure domain has a depth of $d = 0.4 \text{ m}$, air density $\rho_a = 1.2 \text{ kg/m}^3$, and speed of sound $c = 343 \text{ m/s}$ with the noise source at $(0.16 \text{ m}, 0.35 \text{ m}, 0.1 \text{ m})$.

3. Optimization problem

The utilized variables for the optimization problem are corresponding to the shape, location and number of the openings. These variables are optimized in the sense of minimizing a chosen cost function. Various opening to solid (o/s) ratios describing the total opening area on the plate structure are addressed and the considered optimization problem can be categorized into three cases depending on the optimization variable vector $\vec{v}_o$ as follows:

- (i) and (ii) Given the number of openings $n = 1$ and 2 respectively, the geometric parameters of elliptically shaped openings, as described in Section 2.1, are considered as the optimization variables. Hence, the variable vector with $n = 2$ is $\vec{v}_o = [x_{c_1}, y_{c_1}, \theta_{c_1}, l_{c_1}, x_{c_2}, y_{c_2}, \theta_{c_2}, l_{c_2}]$. In this case, the optimization algorithm is required to find a solution in a $(4 \times n)$ dimensional space. For each opening, four variables are required. The first two variables are corresponding to the x and y coordinates of the opening center, defining its position on the plate. The third variable represents the rotation angle, indicating the orientation of the opening. Finally, the fourth variable denotes the semi-major or minor axis.
- (iii) Defining the number of openings $n = 3$, the shapes can be elliptic with the same variables in (i) and (ii). They can also be rectangular with center coordinates (x_{r_i}, y_{r_i}) for the opening i , a side length l_{r_i} , and θ_{r_i} denoting the angle formed by l_{r_i} with the x -axis. Additionally, they can be equilateral triangles with centroid coordinates (x_{t_i}, y_{t_i}) , side length l_{t_i} , and θ_{t_i} denoting the angle formed by l_{t_i} with the x -axis. However, in the case of different shapes, the variable vector contains 5 elements for each opening, where the 5th element can be ≤ 1 , between 1 and 2, or ≥ 2 . The switching between the deterministic functions that describes the opening domains, as the given elliptic example in Section 2.1, accounts for these cases as follows: if the element is ≤ 1 , the shape is elliptic and the 4th element is l_{c_i} ; if the element is between 1 and 2, the shape is rectangular and the 4th element is l_{r_i} ; if the element is ≥ 2 , the shape is an equilateral triangle and the 4th element is l_{t_i} . The algorithm, in this case, can generate an individual with either all openings in the same shape or a combination of the three possible shapes. This depends on the value of the 5th element in the subset that describes an opening in the variable vector. Additionally, it works on finding the optimal solution in (5×3) dimensional space.

Thus, nine systems with different opening requirements is addressed in the discussion, as in Section 4.

3.1. Cost function

The cost function, denoted as $f(\vec{v}_o)$, evaluates the quality of an individual j represented by the variable vector $\vec{v}_o$. It maps the search space to a scalar value, indicating the fitness or score of the individual and is represented as follows:

$$f(\vec{v}_o) = \sum_{k=1}^{N_o} w_k \cdot g_k(\vec{v}_o), \quad (22)$$

where:

- N_o is the number of objectives or components in the cost function.
- w_k represents the weight associated with the k th component.
- $g_k(\vec{v}_o)$ denotes the k th component function, which represents a specific aspect of the individual's quality.

The proposed cost function offers versatility for application across various domains. For instance, it can be utilized to increase sound power levels, adjust resonance frequencies, or function as a noise barrier. For the scope of this article, the overall objective is to minimize the maximum sound power peak in the range between 100 and 350 Hz and the average of the sound power response in the same range. While not conservative and subject to adjustment according to specific requirements, this range is relevant for various engineering applications, such as noise control casings for household appliances, as outlined in [2,4,56]. By adjusting the cost function, Eq. (22), as follows:

$$f_m(\vec{v}_o) = w_1 \cdot g_1(\vec{v}_o) + w_2 \cdot g_2(\vec{v}_o), \quad (23)$$

where $g_1(\vec{v}_o)$ and $g_2(\vec{v}_o)$ are the maximum peak and the average of the sound power response, respectively. The symbols of these quantities are changed from $\bar{\Theta}_{\max}$ and $\bar{\Theta}_{\text{avg}}$ in Eqs. (20) and (21) to indicate their dependency on the variable vector. This highlights that these quantities can reflect different systems, where each system represents a variant of the opening designs according to the variable vector that is used as an input to the reformulated analytical model in Section 2. In addition, this highlights the generality of the utilized framework to accommodate different components of the cost function. The weights of $g_1(\vec{v}_o)$ and $g_2(\vec{v}_o)$ are $w_1 = 0.85$ and $w_2 = 0.15$, respectively, and the objective is to minimize $f_m(\vec{v}_o)$. The given weights are selected to give priority to lowering the highest peak amplitude of the frequency response and to reduce the possibility of other peaks amplification. However, it is possible to modify these weights to suit other objective needs.

3.2. Optimization algorithm

The genetic algorithm utilized in this work follows a systematic approach to searching for optimal solutions within a given search space, as shown in Fig. 2. The algorithm begins by randomly generating a number of individuals, representing potential solutions, to form the initial population. Then, a geometry check is performed to first eliminate individuals with an o/s ratio less than the user requirement. Additionally, it verifies that the smallest distance between the domains of the openings, or between the domains of the openings and the plate edges, is greater than 0.01 m, which is 10 times the plate thickness. Thus, the geometry check is fundamental to keep the structural integrity. Subsequently, each individual's fitness is evaluated by calculating the cost function, which typically involves the analytical model (ANM) simulation, encompassing the sound power response of the system. After evaluating the fitness of each individual, the population is sorted based on their fitness values. The algorithm then terminates if the maximum number of generations is reached or there is a lack of improvement in the cost function over a predefined number of recent generations, which can be defined as convergence. If a termination criteria is met, the optimization process ends; otherwise, the algorithm proceeds. In the selection step, a subset of the population is chosen for reproduction based on their fitness values. Typically, the fittest individuals are selected to survive, while the others are rejected. The crossover step involves generating offspring by applying a crossover operator to selected pairs of individuals (parents). Offspring inherit characteristics from their parents, with individuals of higher fitness more likely to be chosen as parents. The mutation step introduces diversity into the population by randomly mutating individuals with a predefined rate. Finally, the offspring individuals are evaluated after passing the geometry check. In this work, the maximum number of generations is set to 50, and termination is triggered if no improvement is observed after 20 generations. The crossover and mutation rates are selected as 0.3. These values show stable optimization performance and can be adjusted if needed. The software is based on master-slave architecture, as shown in Fig. 3.

The architecture implements the GA in Python (the “master”) and integrates C programs (the “slaves”) for evaluating the fitness of individuals. The master generates and evolves a population of potential individuals by applying selection, crossover, and mutation operations. To evaluate the fitness of individuals, the master delegates the execution to N_p slaves by leveraging the available CPU cores. For each individual, the master creates a unique file containing the individual's parameters using UUIDs (Universally Unique Identifiers) to ensure that each file has a distinct name, preventing any overlap or conflict between files. The master then runs the C programs in parallel across the CPU cores. Finally, it checks the termination criteria to end the program and return the optimal solution or evolve more.

Fig. 4 shows an example of the learning process along with the fitness value distribution over all the generations of the design optimization of the opening with $n = 1$ and $o/s = 5\%$. This highlights the balance between exploration and exploitation, which leads to an effective optimization within the search space. The individual with the optimal fitness and the one with the worst fitness are highlighted with the optimal and least lines in Fig. 4(b), respectively. These individuals, along with the median fitness individual, will be utilized for further analysis in Section 4.

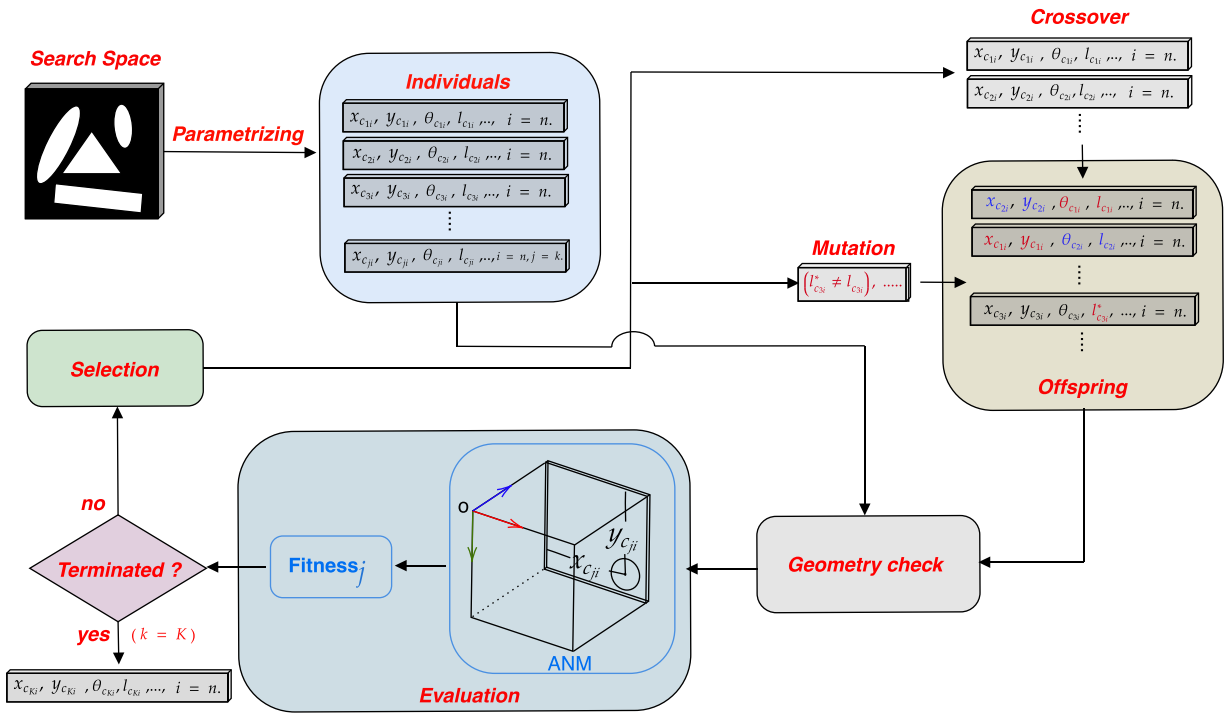
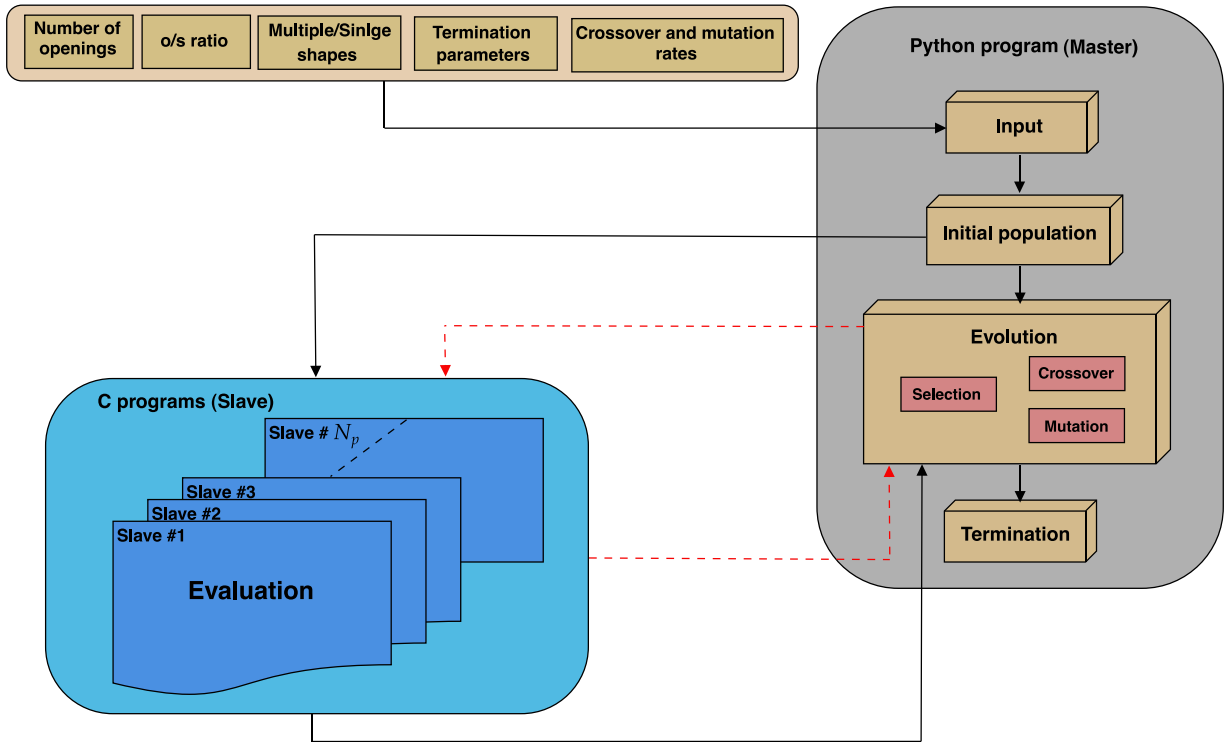


Fig. 2. Genetic algorithm implementation.

Fig. 3. Master-slave GA architecture. Here, N_p is the number of slaves delegated by the master.

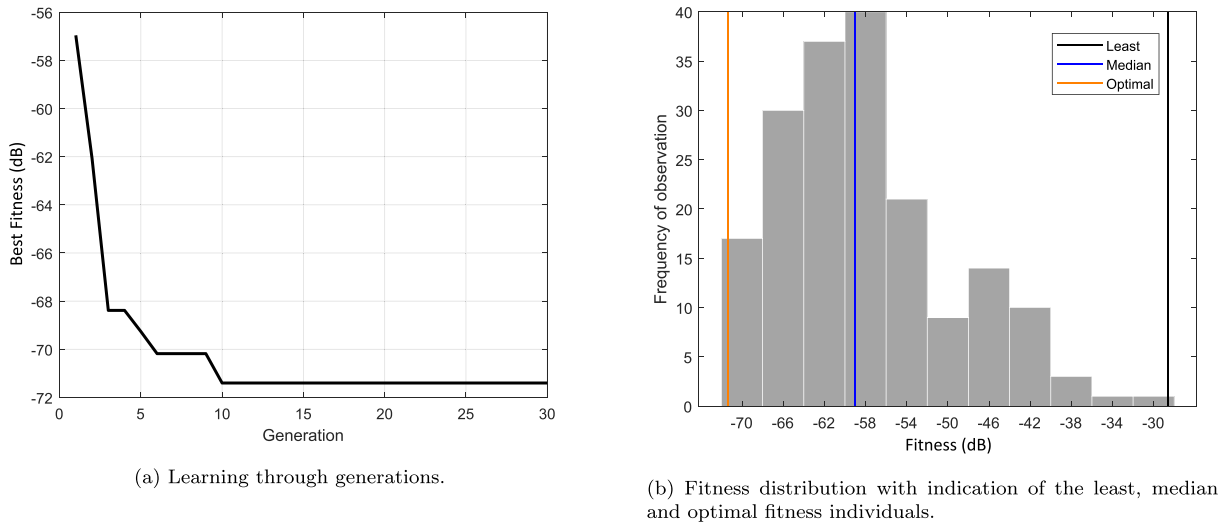


Fig. 4. Genetic algorithm performance.

4. Results and discussion

The size and arrangement of openings on the mixed interface of a vibrating plate affects the distribution of mass and stiffness, thereby influencing the sound power levels of the vibro-acoustic system. The introduced optimization method of the frequency response is applicable to various systems. These systems can be defined with different opening requirements. Systems with varied opening to solid (o/s) ratios, with a different number of openings, or with diverse opening shapes are addressed. For instance, various engineering applications require different levels of ventilation while maintaining specific noise levels. By adjusting the (o/s), the airflow can be controlled, and the proposed method can be used to determine the optimal design parameters for the openings, allowing for passive reduction of sound power. This section analyzes nine different systems to address a variety of opening requirements. Three (o/s) are examined: 30%, 15%, and 5%. Each has a benchmark of a least fitness individual with one opening. The optimization method is tested to find the optimal opening design in case of one, two or three openings. Dividing the required opening area into more than one opening is beneficial for engineering systems that require device monitoring openings or enhanced cooling distribution.

4.1. Elliptical-shaped opening

The elliptical shape is chosen for the openings due to its manufacturing advantages, especially in avoiding sharp corners that are difficult to manufacture.

Fig. 5 shows the impact of implementing the design parameters obtained from the GA algorithm for a system with $n = 1$ and $o/s = 30\%$ design requirement, resulting in a 12 dB reduction in sound power amplitude compared to the median individual design and 60 dB compared to the least fitness design. The sound power response is the combination of the radiated sound from the plate vibration and the leakage sound from the openings interface. However, the maximum peaks of sound power responses are more related to the radiation component than the leakage power. This is due to the high vibration amplitudes at the resonance frequencies of the system response. Fig. 6 depicts the vibration response of the openings under consideration. This highlights that maintaining the same total opening area for the least, median, and optimal designs ensures consistent mass reduction, while the distribution of stiffness varies. The placement of an opening in anti-nodal area alters the plate's stiffness distribution, reducing local resistance to deformation due to the fact that anti-nodal areas experience high stress and displacement. As a result, the plate exhibits higher vibration amplitudes under the same force [57], particularly at modes coinciding with the anti-nodal area containing the opening. This leads to increased amplitudes in the system's sound power response. Thus, the optimization is likely finding the best locations of the openings that give the best distribution of the stiffness to satisfy the objective of the cost function.

For systems with $n = 1$ but different o/s ratios, similar reductions are observed, as depicted in Figs. 7 and 8, and detailed in Table 1.

Smaller openings size leads to lower average sound power due to the fact that less sound is passing through the openings. However, this is not preserved for all the studied systems as the relative location between the opening and the noise source in the enclosure is affecting the system response. Combining two openings while preserving the total opening area presents benefits for applications such as optimizing the airflow around the devices components or enhancing ventilation efficiency, while the vibrating plates are required to serve as noise barriers.

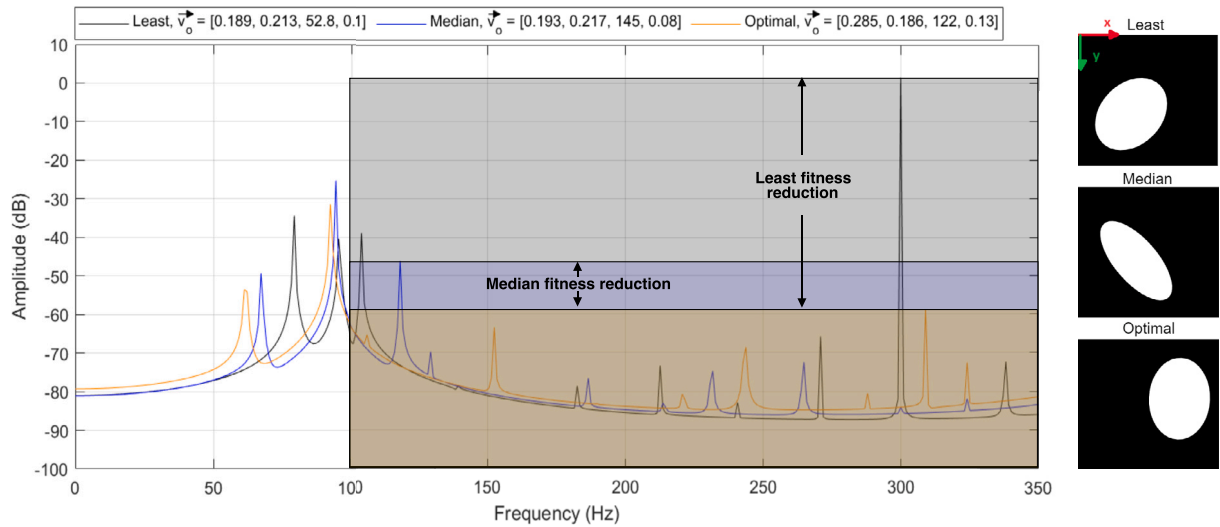


Fig. 5. Sound power response for least, median, and optimal fitness individual opening's design with $n = 1$ and $o/s = 30\%$.

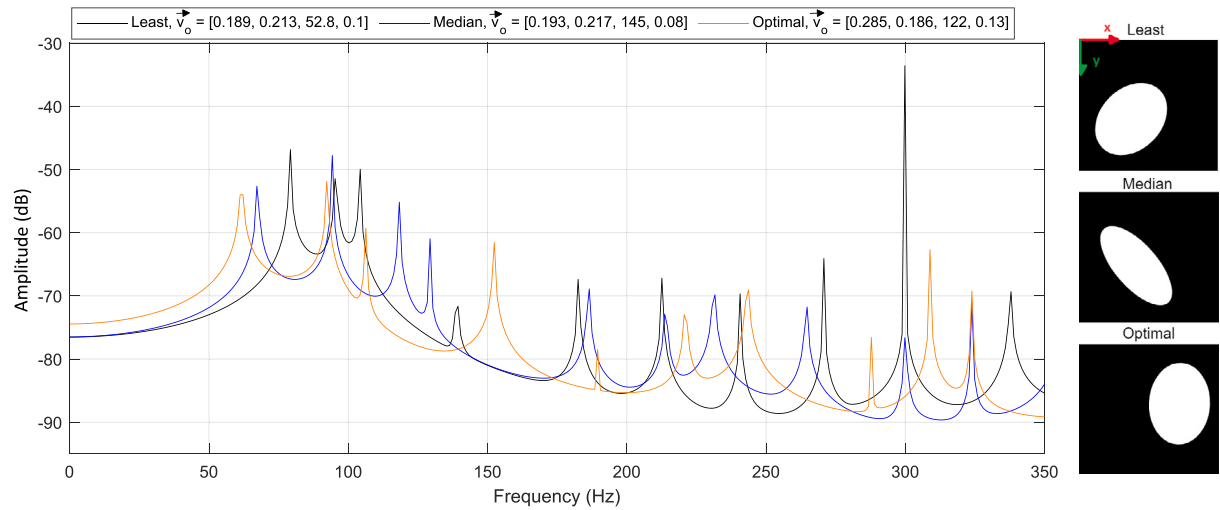


Fig. 6. Vibration response for least, median, and optimal fitness individual opening's design with $n = 1$ and $o/s = 30\%$, calculated by averaging the displacement over the vibrating plate.

For systems with more than one opening, Figs. 9, 10, and 11 compare o/s of 30%, 15%, and 5%, respectively. However, the design with the least fitness value is maintained with one opening, as it is considered the intuitive solution and provides a coherent basis for comparison with the other systems. For the system with $o/s = 30\%$, the optimization faces challenges in finding varied designs for the opening, as the search space is limited due to the large size required. Hence, only a reduction of 5 dB is achieved in the optimal individual compared to the median one, as shown in Fig. 9. In Fig. 10, when the required opening area is reduced to $o/s = 15\%$, more solutions could be generated, thus a reduction of 12 dB was achieved in this case. The optimization had a chance to find an individual with the openings located far from direct exposure to the noise source, while taking into account the effect of the stiffness distribution and the physics considered in the model. More individuals in the search space allows for better optimization, hence the reduction in the systems with $n = 2$ and $o/s = 5\%$ reached to 14 dB compared to the median fitness design, as in Fig. 11. The comparison between the systems with multiple openings reflects the nature of the search space; for higher o/s ratios, the possibility to generate individuals that offer significant variations in system characteristics is limited, resulting in less observed reductions of the sound power response.

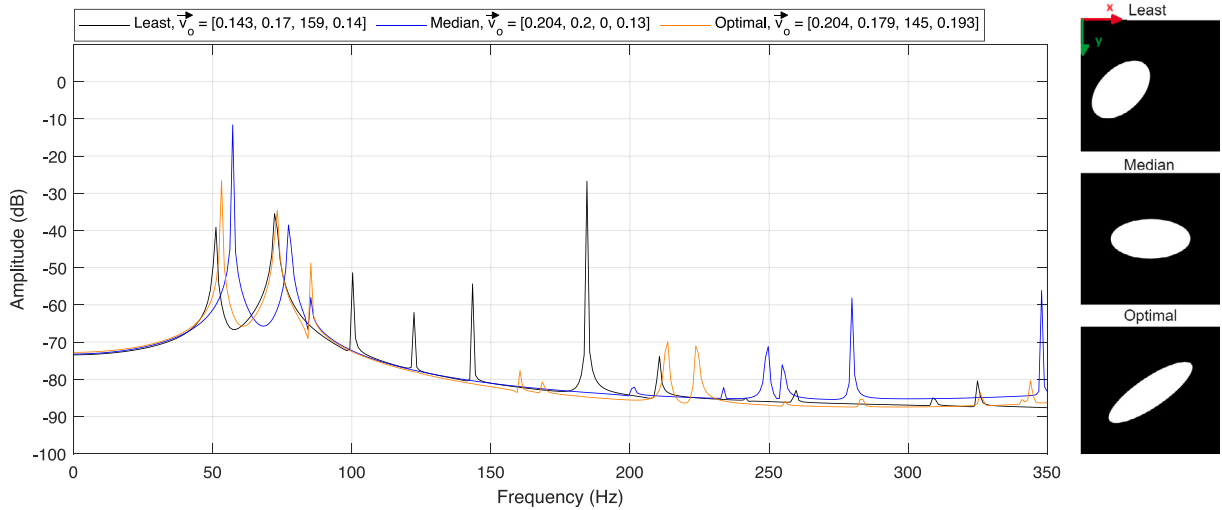


Fig. 7. Sound power response for least, median, and optimal fitness individual opening's design with $n = 1$ and $o/s = 15\%$.

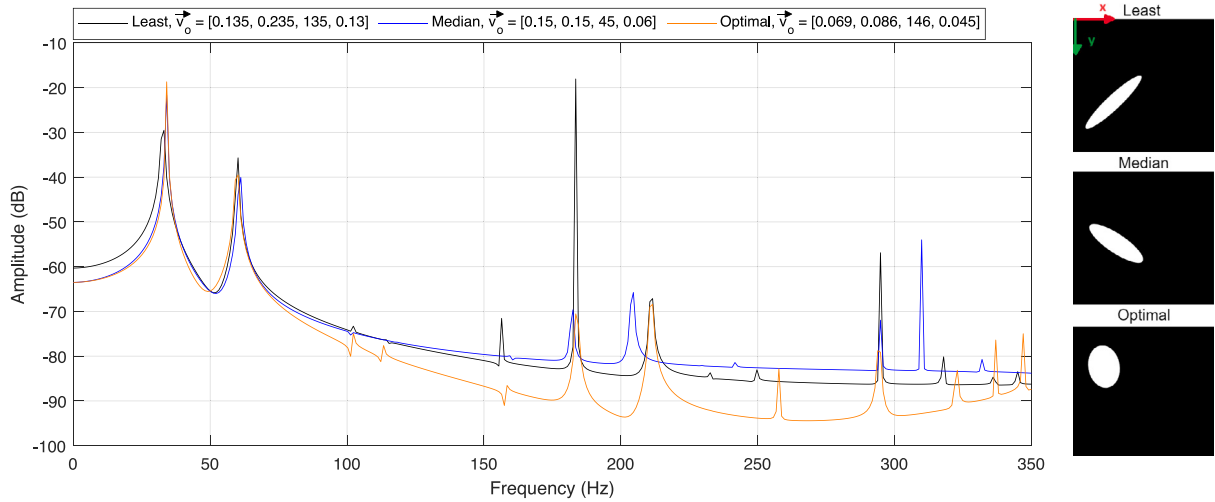


Fig. 8. Sound power response for least, median, and optimal fitness individual opening's design with $n = 1$ and $o/s = 5\%$.

4.2. Mixed opening shapes

The investigation is expanded to include diverse shapes of the openings, which add flexibility to distribute the stiffness across the plate by tailoring the structural stress concentrations. Fig. 12 indicates the designs for a system with $n = 3$ and $o/s = 30\%$, with the possibility of incorporating different opening shapes. Hence, the diffraction effects at the edges of the openings are different. The results show a reduction gain of 10 dB in the sound power response for the optimal individual compared to the median one.

Fig. 13 compares the optimal configurations for $o/s = 30\%$ with different numbers and shapes of the openings. Systems with one and three openings exhibit lower average sound power compared to the two openings configuration. This difference is due to the relative location of the source with respect to the openings, the positioning of the openings relative to each other, and the diffraction effects at the edges of the openings.

In addition, Fig. 14 emphasizes that incorporating rectangular and triangular shapes offers versatility in arranging the openings, thereby increasing the search space and efficiently minimizing the cost function, which results in a reduction of 10 dB in the highest resonance of the sound power response.

For the system with $n = 3$ and $o/s = 15\%$, the optimization has larger search space due to the smaller opening area, hence, better reduction is achieved compared to the previous system, as indicated in Fig. 15. The shape of openings significantly influences the diffraction of sound waves at their edges. This phenomenon affects how sound waves bend around the openings, altering their

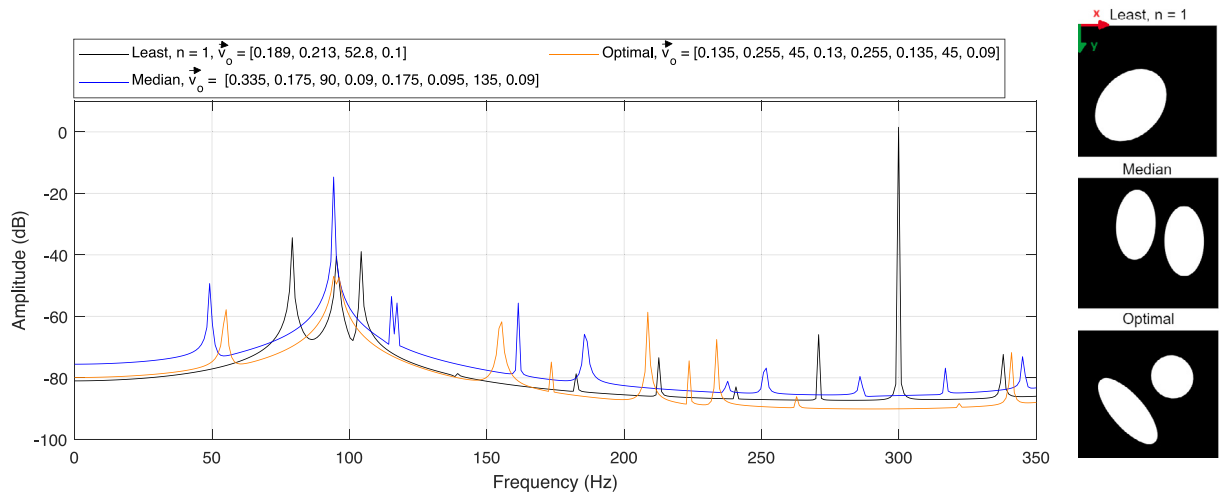


Fig. 9. Sound power response for median and optimal fitness individual opening designs with $n = 2$ and $o/s = 30\%$, and the least fitness individual design with $n = 1$ and $o/s = 30\%$.

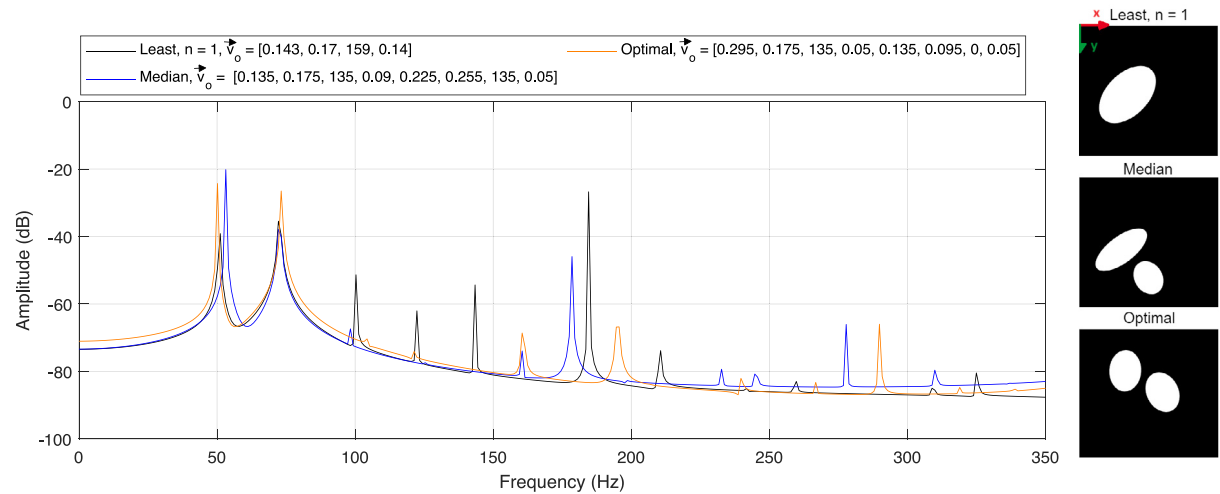


Fig. 10. Sound power response for median and optimal fitness individual opening's design with $n = 2$ and $o/s = 15\%$, and the least fitness individual design with $n = 1$ and $o/s = 15\%$.

direction and intensity. Consequently, different openings' shapes lead to varied patterns of sound diffusion, impacting the plate's vibrational behavior and sound pressure distribution.

Introducing different shapes adds more complexity in determining which physical factors are most responsible for the response variation, highlighting the importance of the proposed method. The optimization considers this multiphysics problem to find a compromise between the effects of the opening shapes, their positions on the vibrating plate, their relation to the stiffness distribution, and the relative positions to the noise source as well as among the openings themselves. Table 1 presents an overview of the fitness values for the median and optimized designs and the least fitness individuals for the proposed systems.

By defining different systems' requirements, including opening area, number of openings, and shapes, the introduced optimization method can effectively reduce the sound power response of vibro-acoustic systems in engineering applications containing vibrating plates with openings for various purposes. Additionally, if for a certain application more regular arrangement of the opening is demanded, this can be easily obtained using the presented method. Moreover, this work not only presenting novel optimization methodology for the openings designs but also highlights the important physical factors that should be considered while designing the openings interfaces in vibro-acoustic systems.

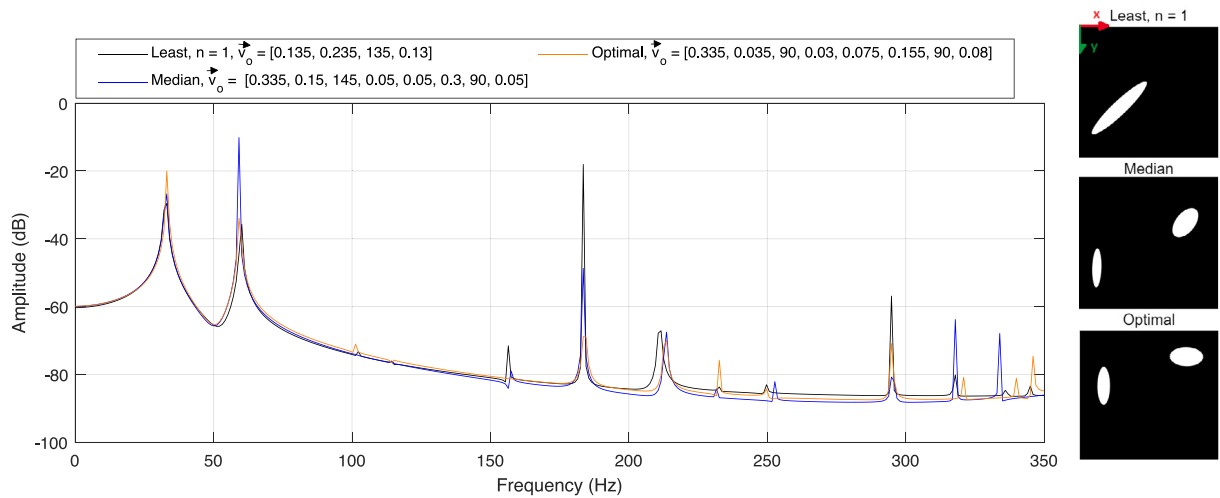


Fig. 11. Sound power response for median and optimal fitness individual opening's design with $n = 2$ and $o/s = 5\%$, and the least fitness individual design with $n = 1$ and $o/s = 5\%$.

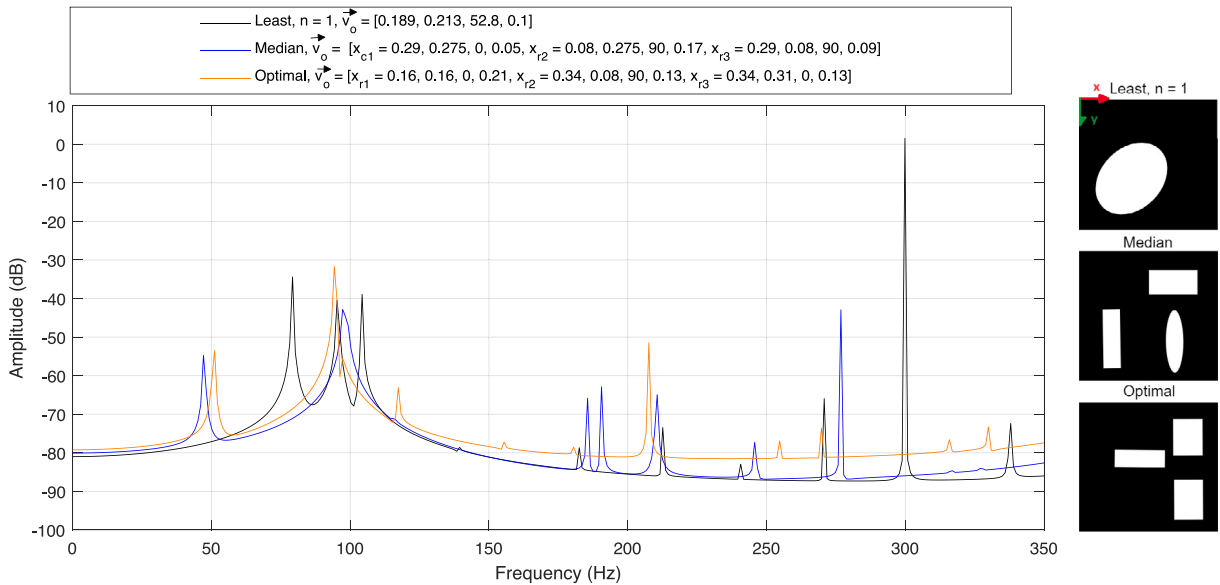


Fig. 12. Sound power response for median and optimal fitness individual opening's design with $n = 3$ and $o/s = 30\%$, and the least fitness individual design with $n = 1$ and $o/s = 30\%$.

5. Conclusion

The comprehensive analysis of the frequency response for the mixed interface of the vibro-acoustic system reveals the potential of optimizing the design of openings on the vibrating plate to align with the system's response objectives. This paper presents a novel opening optimization method applicable to diverse engineering applications employing vibrating plates with these openings as noise barriers. The shape and relative location of the openings influence the vibration and sound power response of the system. Consequently, carefully designing these openings is essential, as improper design parameters can lead to unintended sound power response. For instance, smaller opening areas generally yield lower average sound power when positioned farther away from the noise source. Placing openings on the vibrating plates in areas of high stress will enhance vibration resonance and increase the

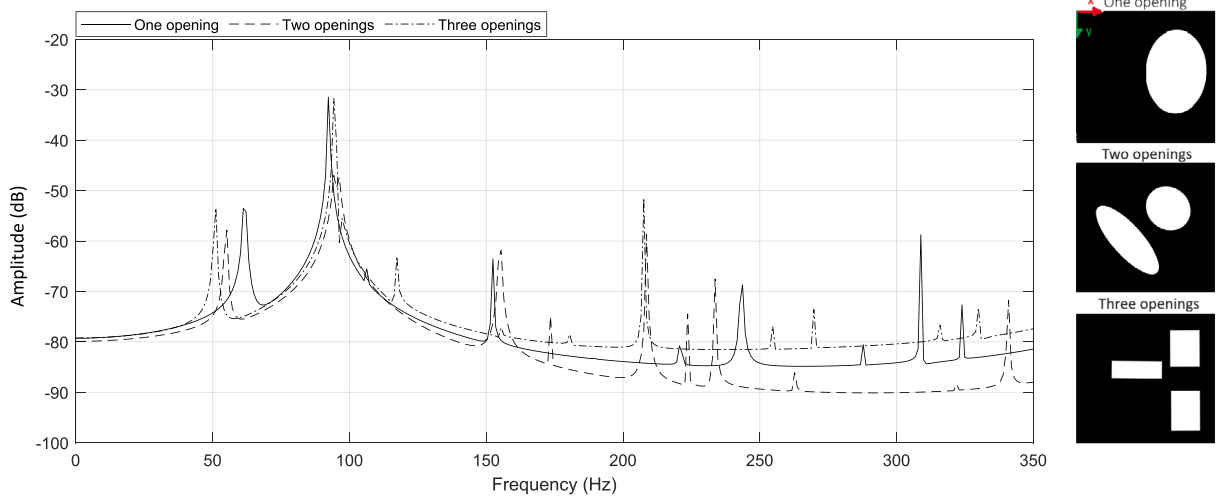


Fig. 13. Comparison of the sound power responses of optimal designs with an o/s ratio of 30% and varying numbers and shapes of openings.

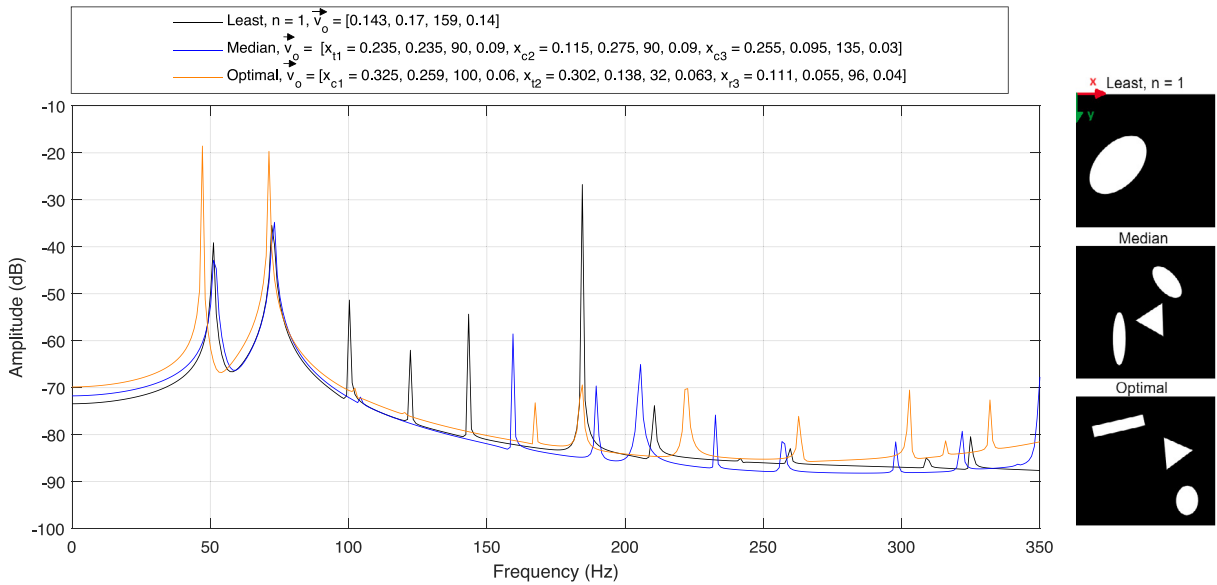


Fig. 14. Sound power response for median and optimal fitness individual opening's design with $n = 3$ and $o/s = 15\%$, and the least fitness individual design with $n = 1$ and $o/s = 15\%$.

system's sound power response. This is underscoring the importance of tailored opening designs. The investigation, conducted within a framework encompassing various systems requirements, including different sizes of openings and various configurations of their shapes and numbers, showcases the capability of achieving reductions in the system's sound power response. These reductions range from 5 dB to 15 dB in peak amplitude of the sound power response in the frequency range between 100 and 350 Hz, compared to median-fit openings' designs and even higher compared to the least-fit openings' designs. The novelty of this optimization method lies in its integration of a new approach for optimizing openings in vibrating plate structures within vibro-acoustic systems, with a genetic algorithm framework and an analytical model within a master-slave software architecture. However, the pursuit of a greater

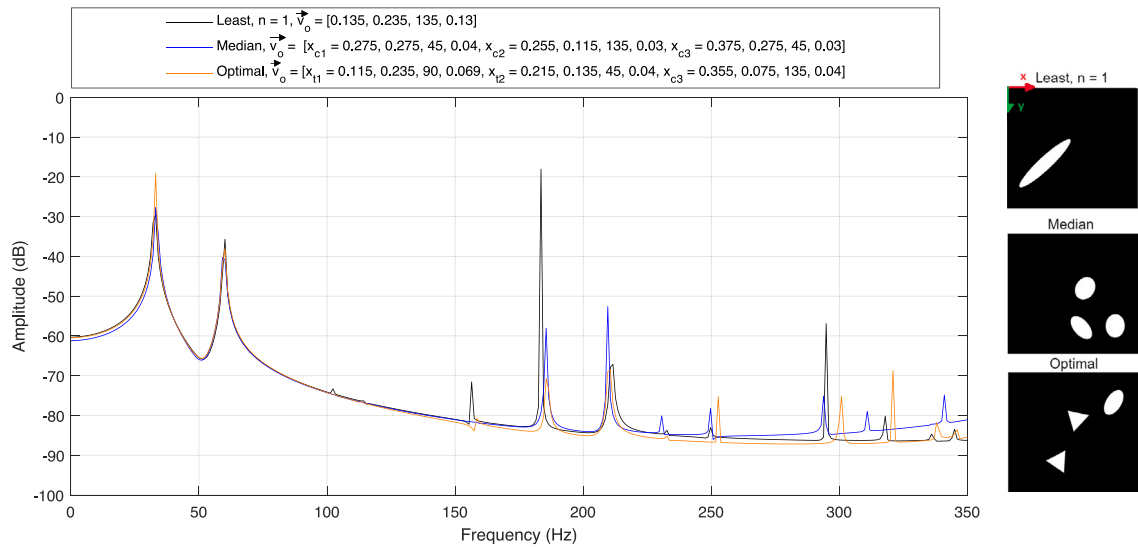


Fig. 15. Sound power response for median and optimal fitness individual opening's design with $n = 3$ and $o/s = 5\%$, and the least fitness individual design with $n = 1$ and $o/s = 5\%$.

Table 1
Comparison table of optimization results for different design requirements.

n	1		2		3			
	o/s [%]	Design Fitness (dB)	o/s [%]	Design Fitness (dB)	o/s [%]	Design Fitness (dB)		
30		-11.1	30		-11.1	30		-11.1
		-51.55			-57.5			-48.8
		-62.12			-62.49			-55.5
15		-35.2	15		-35.2	15		-35.2
		-60.06			-51.3			-62.3
		-72.1			-68.48			-71.26
5		-27.7	5		-27.7	5		-27.7
		-59			-53.98			-56.9
		-71.4			-71			-70.83

reduction in the system's sound power response presents a challenge. Further exploration of technologies such as active noise control may address this challenge.

CRedit authorship contribution statement

Khaled Said Ahmed Maamoun: Writing – review & editing, Writing – original draft, Visualization, Validation, Software, Resources, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Stanisław Wrona:** Writing – review & editing, Writing – original draft, Validation, Supervision, Resources, Formal analysis, Data curation,. **Marek Pawelczyk:** Writing – review & editing, Writing – original draft, Validation, Supervision, Project administration, Funding acquisition, Conceptualization. **Hamid Reza Karimi:** Writing – review & editing, Validation, Supervision, Formal analysis, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgments

The work was supported by the project IN-NOVA: Active reduction of noise transmitted into and from enclosures through encapsulated structures, which has received funding from the European Union's Horizon Europe program under the Marie Skłodowska-Curie grant agreement no. 101073037.

Data availability

Data will be made available on request.

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OPEN Dual-actuator-type active noise control in vibro-acoustic systems with openings

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Openings in plate structures are essential in various engineering applications, particularly in vibro-acoustic systems where airflow is required. This paper investigates noise control in vibro-acoustic systems with noise barriers incorporating structural openings, focusing on active noise control and Active Structural Acoustic Control (ASAC). It also introduces a novel approach, Dual-Actuator-Type Active Noise Control (DATANC), which combines loudspeakers and inertial actuators into the same barrier to address the challenges of noise reduction. A sound power estimation method is proposed to account for sound transmission through the opening and is integrated into an analytical model for optimizing actuator placement; predictions show strong agreement with observed behavior. Among the ASAC configurations, experimental analysis shows that actuators placed near the edge of the opening achieve the greatest noise reduction in the 100–200 Hz range, where acoustic leakage is dominant. DATANC consistently outperformed all single-actuator configurations, delivering superior attenuation of dominant vibro-acoustic resonances while maintaining reasonable computational complexity. The analysis is extended to a plate with a transparent lid over the opening to evaluate the contribution of acoustic leakage to the system performance. The findings of this study demonstrate that optimized actuator placement, combined with DATANC, provides a practical solution for noise control in systems where structural openings are required.

Keywords Active noise control, Active structural acoustic control, Virtual sound barrier, Vibrating plates with openings, Sound power reduction

The rapid development of industrial equipment, household devices, and the automotive and transportation industries has significantly contributed to the rise in global noise pollution levels. Due to its significant impact on both physiological and psychological health, extensive research efforts have focused on mitigating noise levels. Openings in the plate structure of vibro-acoustic systems serve essential functions, including heat dissipation, routing electrical cables, providing personnel access, and weight reduction^{1,2}. However, the shape and location of the openings, as well as their relative position to the noise source in such systems, have a significant impact on the system's sound power response³. Significant effort has been made by researchers to control the sound power characteristics of vibro-acoustic systems. Passive noise control (PNC) provides solutions that do not require an external energy supply, typically achieving optimal noise reduction through modifications to the physical properties of the barriers separating the noise source from the surroundings^{4,5}. PNC is effective for noise attenuation in a considerably high-frequency range compared to active noise reduction strategies, which are generally more effective below this range⁶. Passive noise barriers are typically thick, heavy, and often provide significant heat insulation, which can create additional challenges in certain applications. As a result, actively controlled barriers present a more suitable alternative. These barriers integrate control sources, which may be acoustic, such as loudspeakers, or structural, such as inertial actuators⁷.

Based on the concept of canceling noise using sound waves of equal amplitude and opposite phase, Active Noise Control (ANC) was first proposed and patented by Lueg in 1936⁸. Since then, ANC systems have become essential tools for noise reduction in various engineering applications, including automotive cabins, aircraft interiors, and domestic appliances^{9–12}. The filtered-x least-mean-square (FxLMS) algorithm is the most popular ANC algorithm due to its stability and simple structure as the reference signal to the controller is filtered through the model of the secondary path between the actuator and the error sensors^{13,14}. In practice, the output amplitude

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of the control filter must be constrained to limit the maximum output level and prevent actuator overload^{15,16}. Thus, the leaky FXLMS algorithm is one of the methods used to limit output power by introducing a leakage factor that constrains the control effort and stabilizes the adaptation process^{17,18}. ANC is also employed to establish a Virtual Sound Barrier (VSB) in enclosed systems with openings. Secondary actuators can be placed along the boundary of the opening, a technique referred to as boundary control¹⁹. This approach allows for improved airflow since the speakers do not obstruct the opening. Nonetheless, the boundary length can impose physical constraints on controllability and requires sufficient thickness to mount the actuators securely. In the contrary, planar control creates a grid of the secondary actuators on the opening plane which limits the accessibility to the noise source^{20,21}. ANC has been shown to achieve up to a 10 dB reduction in broadband urban noise through a fully opened window²². However, placing a planar loudspeaker array directly across the opening may obstruct accessibility. Additionally, ANC can be combined with micro-perforated panel absorbers (MPPA) to enhance broadband noise reduction in enclosed spaces where high sound energy dissipation is required²³.

On the other hand, Active Structural Acoustic Control (ASAC) is typically implemented using passive noise barriers, vibration actuators, sensors, and a control system^{24,25}. It's a well-established technique for reducing noise and structural vibrations. With careful implementation, ASAC can achieve not only local noise attenuation but also global noise reduction^{26,27}. ASAC was implemented in a closed passenger car equipped with an active windshield, achieving up to 15 dB reduction in interior sound pressure levels. The study focused on a fully sealed cabin, without addressing the effects of openings such as partially open windows on acoustic performance²⁸. The performance of a tonal ASAC strategy that utilizes an experimentally identified radiation matrix to control sound power radiation was investigated, demonstrating superior performance compared to the active vibration control method²⁹. The study was limited to tonal excitation, with broadband disturbances not addressed. A feedforward ASAC algorithm was also tested on the lining panels of an aircraft cabin using a low-cost microcontroller, achieving a tonal noise reduction of 23 dB³⁰. A radiation-mode-based ASAC system for an enclosure with vent openings has been theoretically evaluated³¹. However, the model was validated using COMSOL-based finite element (FE) and boundary element (BE) simulations, and the analysis was limited to tonal disturbances. Existing studies reveal a gap in ASAC research on systems with openings compared to fully enclosed systems, highlighting the need for further investigation. The placement of inertial actuators on vibrating plates is a critical factor influencing system performance. When properly positioned, actuators can significantly enhance both the vibration and acoustic responses. Various studies have explored actuator placement optimization using cost functions derived from vibration characteristics^{32,33} and modal radiation analysis³⁴. Although the active force of inertial actuators can be evaluated using linear quadratic optimal control theory applied to analytical models^{35,36}, these approaches do not account for systems with openings, which introduce additional complexity due to the acoustic leakage.

This paper introduces the Dual-Actuator-Type Active Noise Control (DATANC) system, which combines loudspeaker and inertial actuators to address noise reduction challenges in vibro-acoustic systems with openings. To investigate its effectiveness, the DATANC system is experimentally evaluated alongside conventional ANC and ASAC configurations with a single type of actuators. The present study experimentally evaluates ASAC in these systems under random noise excitation, extending beyond prior work focused on tonal disturbance excitation³¹. To enable a fair comparison between different control configurations, the locations of the inertial actuators are optimally determined using an analytical model that accounts for sound transmission through the enclosure opening. This approach builds on previously established control strategies^{35,36}. Additionally, the study evaluates the impact of this acoustic leakage on the overall sound power response by analyzing a vibrating plate fitted with a transparent lid over the opening. The system, with a transparent lid, itself serves as an effective solution to enhance noise barrier functionality. Nevertheless, the transparency of the material allows for continued device monitoring. The main contributions of this paper are as follows: i) the optimal placement of inertial actuators using an analytical model that accounts for sound transmission through the enclosure opening; ii) the development of the DATANC system, which integrates loudspeakers and inertial actuators to achieve enhanced noise control performance in these systems, outperforming conventional ANC and ASAC configurations under broadband random noise excitation; iii) a spatial analysis of the relative placement of actuators and its influence on system performance, including both frequency response behavior and control effectiveness; iv) an investigation of the acoustic transmission effect by introducing an enclosure opening patched with a transparent lid.

Methods

A previously introduced vibro-acoustic system¹ is modified to incorporate inertial actuators, treating the vibrating plate as an actively controlled acoustic barrier. This modification is primarily aimed at enabling the optimal placement of inertial actuators by accounting for both radiated sound and acoustic transmission through structural openings. The system consists of a rectangular enclosure with a freely vibrating, fully clamped rectangular plate that includes a circular opening interface, as illustrated in Fig. 1. The remaining five walls of the enclosure are modeled as acoustically rigid (sound hard).

Analytical model

The Kirchhoff–Love theory of thin plates³⁷ is employed to model the vibrating plate in free space, under the assumptions of isotropy and homogeneity. The analysis considers only transverse motion, while the effects of rotary inertia are neglected. The kinetic energy of the plate, T_p , is expressed as:

$$T_p = \frac{1}{2} h \iint_{S_{mix}} \rho(x, y) \dot{w}^2 dx dy, \quad (1)$$

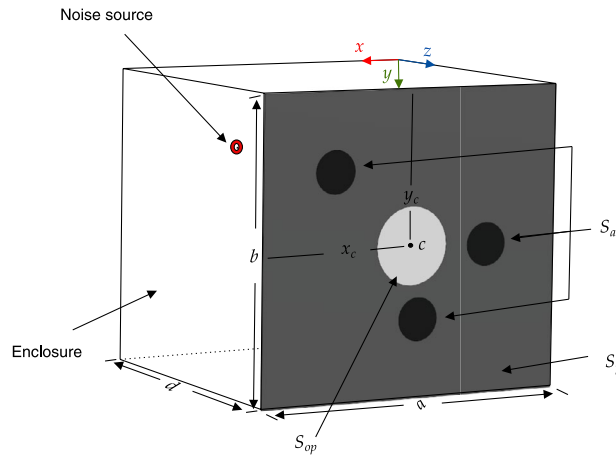


Fig. 1. Analytical model configuration showing the enclosure domain, the vibrating plate S_p , a circular opening interface S_{op} with a radius of 0.05 m , and the actuators domain, S_a , is a combination of circles with a radius of 0.02 m for each actuator. The position (x_c, y_c) is the center of S_{op} and the primary noise source is positioned at $(x_s = 0.3\text{ m}, y_s = 0.1\text{ m}, z_s = -0.2\text{ m})$.

where h is the thickness of the plate, and $\rho(x, y)$ represents the varying density of the mixed interface, S_{mix} . The density function is defined as:

$$\rho(x, y) = \begin{cases} \rho_p, & \text{if } (x, y) \in S_p, \\ \rho_{act}, & \text{if } (x, y) \in S_a, \\ 0, & \text{if } (x, y) \in S_{op}, \end{cases} \quad (2)$$

where ρ_p is the density of the plate material, and ρ_{act} is a theoretical density value representing the mass effect of the actuators on the system kinetic energy. The actuators, being small relative to the dimensions of the plate, have a negligible effect on the strain of the plate surface. By assuming ideal bonding and neglecting the stiffness of the actuators^{34,38,39}, the total energy contribution from the actuators can be represented by ρ_{act} in the plate kinetic energy function, as in Eq. (1, 2). The deflection of the plate in the z -direction at time t is denoted as $w(x, y, t)$. The initial conditions are given by

$$w(x, y, t_0) = 0, \quad \left. \frac{\partial w(x, y, t)}{\partial t} \right|_{t=t_0} = 0, \quad t \geq t_0 > 0. \quad (3)$$

Sound power estimation

The estimation of sound power is based on radiation from the vibrating plate and the sound transmission through the opening. The total sound power is given by

$$W_{\text{total}}(\omega) = W_{\text{rad}}(\omega) + W_{\text{loss}}(\omega), \quad (4)$$

where $W_{\text{rad}}(\omega)$ is the radiated sound power from the plate and can be expressed in vector form as

$$W_{\text{rad}}(\omega) = q_r^H \mathbf{R}(\omega) q_r, \quad (5)$$

where the superscript H denotes the complex conjugate transpose. The vector q_r represents the plate velocity and is obtained from the last n_p^2 elements of the system response vector q . Here, n_p is the number of modes considered in one direction of the 2D vibrating plate, and $\mathbf{R}(\omega)$ is the radiation resistance matrix at frequency ω , computed analytically using surface discretizations and mutual element interactions^{1,36,40}. $W_{\text{loss}}(\omega)$ is the leakage sound power and can be estimated as follows:

$$W_{\text{loss}}(\omega) = \frac{q_p^H q_p}{\rho_a c} A_s. \quad (6)$$

Where q_p is the pressure at the opening interface and can be calculated from the first n_c^3 elements from the system response, n_c is the number of modes considered in one direction of the 3D enclosure, A_s is the element area, ρ_a is the density of the air, and c is the speed of sound in the air. By substituting Eqs. (5, 6) in Eq. (4), the total sound power in vector form is expressed as

$$W_{\text{total}}(\omega) = q_r^H \mathbf{R}(\omega) q_r + \frac{q_p^H q_p}{\rho_a c} A_s. \quad (7)$$

Based on the additive theory⁴¹, the plate velocity q_r can be expressed as

$$q_r = q_{r,\text{pri}} + \sum_{k=1}^K q_{r,\text{unit},k} f_{c,k}, \quad (8)$$

where $q_{r,\text{pri}}$ is the velocity due to the primary source, $q_{r,\text{unit},k}$ is the velocity due to the k -th unit control source, and $f_{c,k}$ is the complex amplitude of the k -th control source. Similarly, the pressure q_p at the opening interface can be written as

$$q_p = q_{p,\text{pri}} + \sum_{k=1}^K q_{p,\text{unit},k} f_{c,k}, \quad (9)$$

where $q_{p,\text{pri}}$ is the pressure due to the primary source, and $q_{p,\text{unit},k}$ is the pressure due to the k -th unit control source. The quantities $q_{r,\text{unit},k}$ and $q_{p,\text{unit},k}$ are unique for each actuator as they differ due to the different location of the actuators on the plate. Substituting the expansion of q_r into the radiated sound power $W_{\text{rad}}(\omega)$, we obtain:

$$\begin{aligned} W_{\text{rad}}(\omega) &= q_{r,\text{pri}}^H \mathbf{R}(\omega) q_{r,\text{pri}} + \sum_{k=1}^K \sum_{j=1}^K f_{c,k}^H q_{r,\text{unit},k}^H \mathbf{R}(\omega) q_{r,\text{unit},j} f_{c,j} \\ &+ \sum_{k=1}^K f_{c,k}^H q_{r,\text{unit},k}^H \mathbf{R}(\omega) q_{r,\text{pri}} + \sum_{k=1}^K (q_{r,\text{unit},k}^H \mathbf{R}(\omega) q_{r,\text{pri}})^H f_{c,k}. \end{aligned} \quad (10)$$

For the leakage sound power $W_{\text{loss}}(\omega)$, substituting the expansion of q_p , we get:

$$\begin{aligned} W_{\text{loss}}(\omega) &= \frac{1}{\rho_a c} A_s \left(q_{p,\text{pri}}^H q_{p,\text{pri}} + \sum_{k=1}^K \sum_{j=1}^K f_{c,k}^H q_{p,\text{unit},k}^H q_{p,\text{unit},j} f_{c,j} \right. \\ &\left. + \sum_{k=1}^K f_{c,k}^H q_{p,\text{unit},k}^H q_{p,\text{pri}} + \sum_{k=1}^K (q_{p,\text{unit},k}^H q_{p,\text{pri}})^H f_{c,k} \right). \end{aligned} \quad (11)$$

Hence, the total sound power can be expressed as

$$\begin{aligned} W_{\text{total}}(\omega) &= q_{r,\text{pri}}^H \mathbf{R}(\omega) q_{r,\text{pri}} + \frac{q_{p,\text{pri}}^H q_{p,\text{pri}}}{\rho_a c} A_s + \sum_{k=1}^K \sum_{j=1}^K f_{c,k}^H q_{r,\text{unit},k}^H \mathbf{R}(\omega) q_{r,\text{unit},j} f_{c,j} \\ &+ \sum_{k=1}^K f_{c,k}^H q_{r,\text{unit},k}^H \mathbf{R}(\omega) q_{r,\text{pri}} + \sum_{k=1}^K (q_{r,\text{unit},k}^H \mathbf{R}(\omega) q_{r,\text{pri}})^H f_{c,k} \\ &+ \frac{1}{\rho_a c} A_s \left(\sum_{k=1}^K \sum_{j=1}^K f_{c,k}^H q_{p,\text{unit},k}^H q_{p,\text{unit},j} f_{c,j} + \sum_{k=1}^K f_{c,k}^H q_{p,\text{unit},k}^H q_{p,\text{pri}} + \sum_{k=1}^K (q_{p,\text{unit},k}^H q_{p,\text{pri}})^H f_{c,k} \right). \end{aligned} \quad (12)$$

Using the linear quadratic optimal control theory, we minimize $W_{\text{total}}(\omega)$ by setting its gradient to zero. The optimal gain $f_{c,k}^{\text{opt}}$, can be calculated by solving the linear system:

$$\mathbf{f}_c^{\text{opt}} = -\mathbf{H}^{-1} \mathbf{b}. \quad (13)$$

where:

- $\mathbf{f}_c^{\text{opt}} = [f_{c,1}^{\text{opt}}, f_{c,2}^{\text{opt}}, \dots, f_{c,K}^{\text{opt}}]^T$ is the vector of control amplitudes.
- $\mathbf{H}$ is a Hermitian matrix of size $K \times K$, with elements:

$$H_{k,j} = q_{r,\text{unit},k}^H \mathbf{R}(\omega) q_{r,\text{unit},j} + \frac{1}{\rho_a c} A_s q_{p,\text{unit},k}^H q_{p,\text{unit},j}. \quad (14)$$

- $\mathbf{b}$ is a vector of size K , with elements:

$$b_k = - \left(q_{r,\text{unit},k}^H \mathbf{R}(\omega) q_{r,\text{pri}} + \frac{1}{\rho_a c} A_s q_{p,\text{unit},k}^H q_{p,\text{pri}} \right). \quad (15)$$

In this study, three actuators are considered ($K = 3$). Accordingly, the system of equations can be expressed as

$$\begin{aligned}
 H_{1,1}f_{c,1}^{\text{opt}} + H_{1,2}f_{c,2}^{\text{opt}} + H_{1,3}f_{c,3}^{\text{opt}} &= -b_1, \\
 H_{2,1}f_{c,1}^{\text{opt}} + H_{2,2}f_{c,2}^{\text{opt}} + H_{2,3}f_{c,3}^{\text{opt}} &= -b_2, \\
 H_{3,1}f_{c,1}^{\text{opt}} + H_{3,2}f_{c,2}^{\text{opt}} + H_{3,3}f_{c,3}^{\text{opt}} &= -b_3.
 \end{aligned} \tag{16}$$

Solving this system yields the optimal gains $f_{c,\text{opt},1}$, $f_{c,\text{opt},2}$, and $f_{c,\text{opt},3}$. A genetic algorithm³ is then used to search for the actuator locations that minimize the system's acoustic response, with the search space defined by the possible positions of the three actuators. It should be highlighted that this is a theoretical calculation of the sound power, and it would be difficult to obtain one-to-one validation with real control setup. However, this method offers a valuable guidance for identifying optimal actuator locations to achieve effective noise reduction, as discussed in the Results section.

The following model properties are chosen due to their widespread use in various engineering applications and to reflect the experimental setup illustrated in Fig. 2. The plate is composed of steel with a density $\rho_p = 7850 \text{ kg/m}^3$, Young's modulus $E = 210 \times 10^9 \text{ Pa}$. The dimensions of the plate are $a = 0.42 \text{ m}$, $b = 0.39 \text{ m}$, and $h = 0.001 \text{ m}$, representing its length, width, and thickness, respectively. The density of the actuator can be estimated based on its mass (150 g) and geometric dimensions, assuming a radius of 0.02 m and a fixed height of 0.001 m, which corresponds to the uniform thickness of the mixed interface between the plate and the actuators. The enclosure domain has a depth of $d = 0.4 \text{ m}$, air density $\rho_a = 1.2 \text{ kg/m}^3$, and speed of sound $c = 343 \text{ m/s}$ with the primary noise source at (0.3 m, 0.1 m, -0.2 m). Using this system parameters, the optimal positions of the three inertial actuators were determined to be $(x = 0.32 \text{ m}, y = 0.10 \text{ m})$, $(x = 0.20 \text{ m}, y = 0.29 \text{ m})$, and $(x = 0.10 \text{ m}, y = 0.20 \text{ m})$. The resulting actuator positions are evaluated and compared with alternative actuator arrangements in the Results section, where the effectiveness of the analytical model as a tool for guiding the optimal placement of inertial actuators is discussed.

Experiment

The laboratory room measures 5.8 meters in length and 3.5 meters in width. While the walls are lined with sound-absorbing materials, the presence of equipment and other objects within the room gives it acoustic characteristics more representative

of a typical real-world environment rather than those of a dedicated acoustic chamber. The experimental setup consists of a rectangular steel plate mounted on a concrete enclosure, within which a loudspeaker is installed. The material and dimensions of the plate are as described in the Analytical model section. The concrete walls of the enclosure provide a high degree of sound attenuation; hence, the majority of the acoustic energy

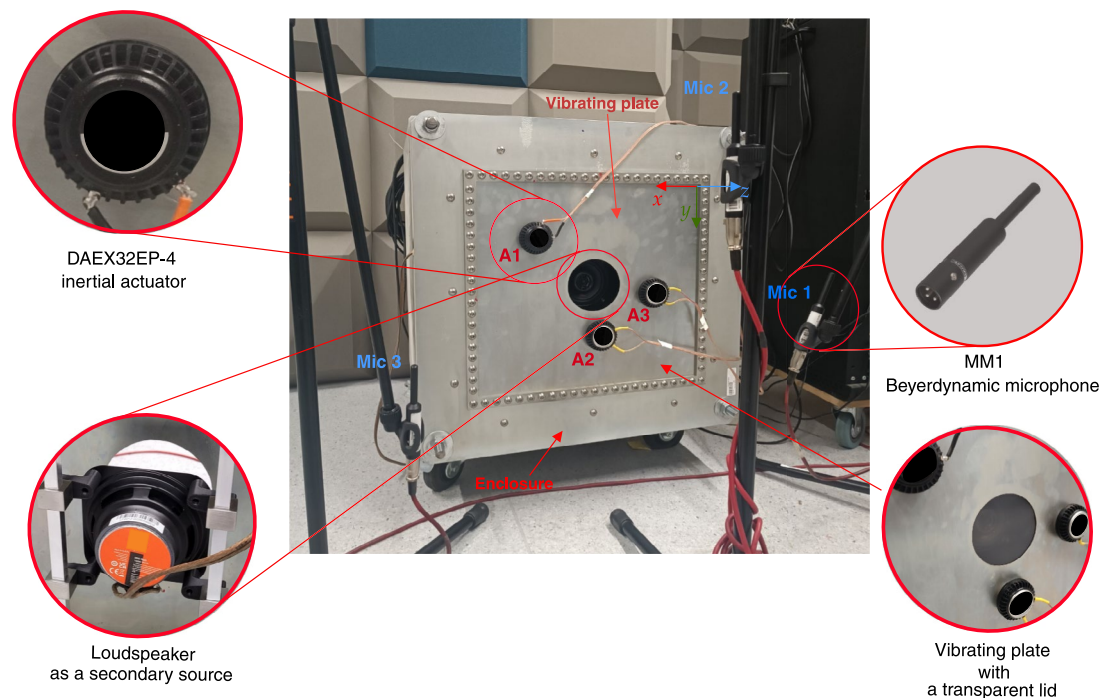


Fig. 2. Overview of the experimental rig showing a concrete enclosure with a vibrating plate that contains a central circular opening of radius 0.05 m. The steel plate spans 0.42 m in x , 0.39 m in y , and has a thickness of 0.001 m in $-z$. Actuators (A1, A2, and A3) are mounted on the plate at (0.32 m, 0.1 m), (0.2 m, 0.29 m), and (0.1 m, 0.2 m), respectively. Microphones (Mic 1, Mic 2, and Mic 3) are positioned at the coordinates (−0.18 m, 0.12 m, 0.25 m), (0.21 m, −0.1 m, 0.4 m), and (0.53 m, 0.24 m, 0.25 m), respectively. The loudspeaker, used as a secondary source, is centered behind the opening.

that exits the enclosure is transmitted through the plate and the circular opening. The loudspeaker, a Behringer EUROLIVE B208D with a rated maximum sound pressure level (SPL) of 113 dB, serves as the primary excitation source and is used to generate band-limited random noise. Three Beyerdynamic MM1 error microphones are used for the active control system with three DAEX32EP-4 inertial actuators. Although the actuator placement was optimized using theoretical sound power calculations, the experimental evaluation will be based on fixed microphone positions using adaptive active noise control. These positions are selected to capture the dominant acoustic field from the noise barrier, where the transmitted sound pressure is highest, and are spaced such that the distance between them, as well as their distance from the vibrating panel, remains small enough to allow for global noise reduction within the control region^{26,42,43}. While variations in microphone placement may affect local details of the measured response, the overall global performance is not expected to be significantly influenced. A detailed investigation into the optimal placement of error microphones is considered beyond the scope of this study. For configurations incorporating a secondary loudspeaker source, a JBL Stage2 424 speaker is used and positioned centrally behind the opening, as in Fig. 2. To test the system with a reduced acoustic leakage effect, a plastic lid is adhesively bonded to the steel plate. The lid is made of semi-rigid polyvinyl chloride (PVC) with a density of approximately 1300 kg/m³. In this work, the Analog Device SC598 with SHARC+ core is used along with EVSOMCRR-EZKIT development board⁴⁴. Thus, the system is equipped with the ADAU1979, a 4-channel, 24-bit Sigma-Delta ADC, and the ADAU1962, a 12-channel, 24-bit differential output DAC. For the selected ADC and DAC with a sampling frequency of 48 KHz, the group delay is 1 ms. Anti-aliasing IIR filters are designed for downsampling from 48 kHz to the downsampling frequency ($F_{s,D}$) of 3 kHz, followed by upsampling. Based on the analysis of the IIR filter characteristics, a 20-sample group delay is introduced into the system during both downsampling and upsampling, which is equivalent to 0.83 ms. Additionally, the system has a delay of 0.33 ms due to the output buffer handling. This makes the total latency of the system (from ADC to system to DAC) to be around 2.16 ms, which is borderline acceptable for active control implementation⁶. For active control, a feedforward multichannel FxLMS algorithm with a leakage factor is used due to its simplicity and robustness⁶, as illustrated in Fig. 3. Furthermore, the control system incorporates normalization based on the power of the single reference signal.

$X(n)$ represents the artificially generated reference signal. However, a reference microphone can be used if necessary. The primary path of the system is denoted by the matrix $P(n)$, which characterizes the acoustic propagation from the noise source to the error microphones. The active control filters are represented by the vector $W(n)$, consisting of K adaptive FIR filters $W_k(z)$, each responsible for generating a component of the control signal vector $y(n)$. The vector $\hat{y}(n)$ represents the anti-noise signals applied through the actuators. The matrix $S(n)$, which models the secondary path dynamics between actuators and microphones, is generally not accessible in real-time operation and is therefore substituted with its estimate $\hat{S}(n)$. The $K \times M$ secondary paths of the plant is modeled as the impulse response of FIR filters using the normalized Least Mean Square (NLMS) algorithm⁴⁵. This algorithm updates the coefficients of the K adaptive filters $W_k(n)$ in the controller as follows:

$$W_k(n+1) = (1 - \mu_n(n)\beta)W_k(n) + \mu_n(n) \sum_{m=1}^M x'_{km}(n)e_m(n), \quad (17)$$

where $x'_{km}(n)$ is the filtered reference signal, $e_m(n)$ is the error signal, and $\mu_n(n)$ is the normalized step size given by

$$\mu_n(n) = \frac{\mu}{\sum_{k=1}^K \sum_{m=1}^M \|x'_{km}(n)\|^2}, \quad (18)$$

where μ is the fixed step size, $\|\cdot\|$ represents the L^2 -norm and β is a regularization factor that applies a leakage effect.

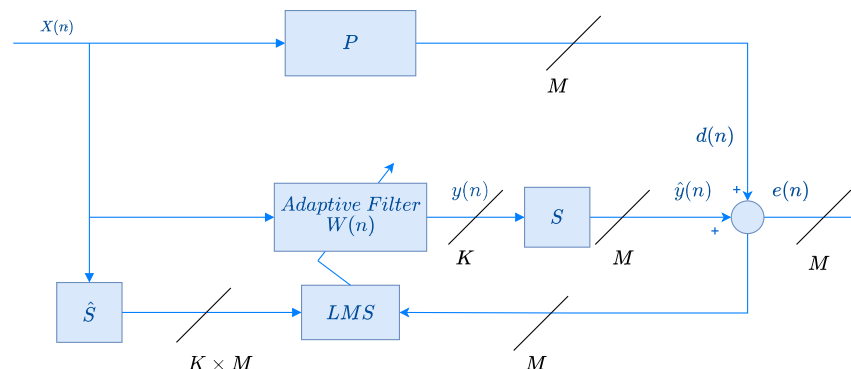


Fig. 3. Block diagram of a single-reference multichannel feedforward FxLMS algorithm. M and K refer to the number of the output and secondary source, respectively.

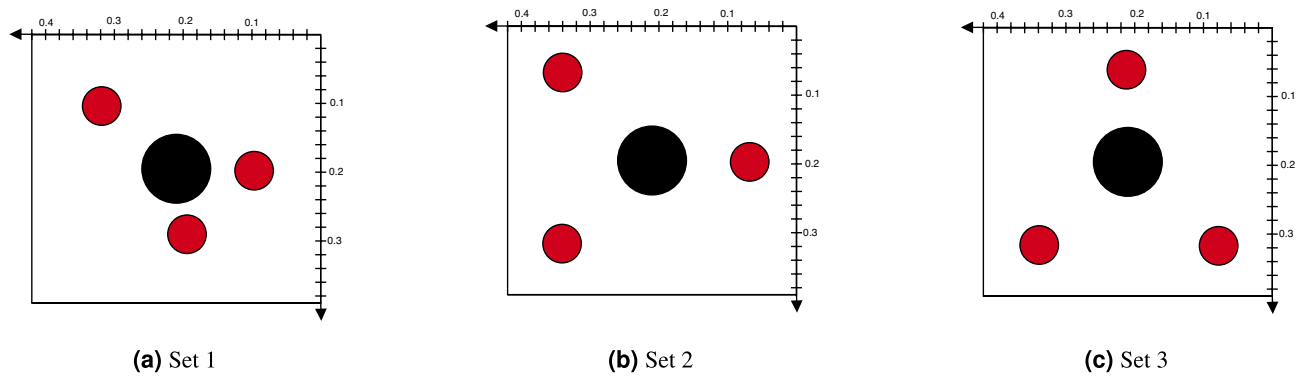


Fig. 4. (a) Set 1: Actuators positioned based on the analytical model. (b) Set 2: Actuators localized toward the left half of the plate. (c) Set 3: Positioned with a similar concentration toward the bottom half of the plate.

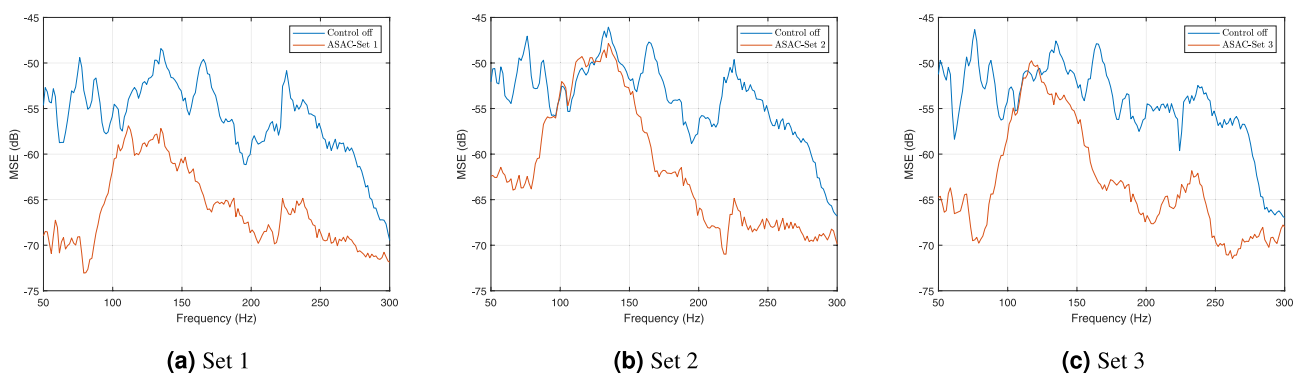


Fig. 5. Experimental mean squared error (MSE) at the microphones for (a) Set 1, (b) Set 2, and (c) Set 3 actuator arrangements, with and without applying ASAC.

Results

The optimal locations for the inertial actuators, as identified in the Methods section, are assessed using the ASAC approach and compared against alternative actuator arrangements as illustrated in Fig. 4. Experimental investigation of the three actuator sets demonstrates that Set 1 yields the best performance, as indicated by the mean squared error (MSE) measured at the three microphones and illustrated in Fig. 5. In passive configurations, placing actuator masses near the opening reduces MSE below 200 Hz but leads to increased peaks between 200 and 300 Hz (Fig. 5a). This behavior is attributed to the opening's location, which coincides with regions of high modal displacement at low frequencies³. Adding mass in this region increases the local inertia and alters the modal participation of these modes, making them less efficiently excited⁴⁶. If a purely passive solution is required, the model can be adapted by eliminating the actuator forces and optimizing the locations of the added masses. However, this approach is primarily effective for shifting acoustic resonances from one frequency band to another. In such cases, the cost function may be modified to target narrowband noise reduction objectives³⁸. This change leads to a reduction in vibration energy near the opening and thus a decrease in the MSE associated with sound radiation at lower frequencies. When ASAC is applied, Set 1 consistently yields the lowest MSE across nearly the entire frequency range. In the 100–200 Hz band, it achieves an average reduction approximately 5 dB greater than that obtained with Sets 2 and 3. This frequency range is particularly critical, as it corresponds to dominant the acoustic leakage through the opening, which is discussed further in this section. The placement of actuators in Set 1, near the edge of the opening, facilitates a greater displacement range and thus enhances control effectiveness. In contrast, Sets 2 and 3 with actuators positioned near the clamped edges of the plate, where higher structure stresses result in elevated resonance levels and limited displacement. These factors collectively reduce their capacity to control vibrations, suppress radiated sound, and generate effective anti-noise signals. These findings indicate the efficacy of the proposed analytical model in guiding optimal actuator placement for improved noise reduction.

Based on this optimization, the inertial actuators' arrangement in Set 1, also shown in the experimental setup (Fig. 2), is hereafter referred to as the ASAC configuration throughout this work. To assess the performance of conventional active control strategies, comparative experiments are conducted under identical acoustic and structural conditions. The enclosure was excited using a loudspeaker generating band-limited white noise in the 50–300 Hz range. The adaptive control algorithm employs control filters of length 400 and secondary path models of length 300, with a step size of $\mu = 1 \times 10^{-6}$ and a regularization factor of $\beta = 1 \times 10^{-6}$. These parameters

were experimentally tuned and remain fixed across all control configurations unless stated otherwise. The first configuration, ASAC, employs three error microphones and three inertial actuators, arranged according to the optimized layout. The second, Multiple-Input Single-Output Active Noise Control (MISO-ANC), follows a conventional overdetermined approach using a single loudspeaker as the actuator with three error microphones. The third, Single-Input Single-Output Active Noise Control (SISO-ANC), represents a single-channel scheme using Mic 2 as the error microphone, while Mic 1 and 3 are used only as observation microphones. In the frequency range below 100 Hz, the ASAC configuration exhibits lower MSE values compared to the loudspeaker-based systems, as in Fig 6a. However, this band is particularly challenging to control, as it contains the system's fundamental natural frequencies³. Moreover, this frequency range is generally less critical in practice, as sounds below 100 Hz tend to have lower perceptual impact, particularly at moderate listening levels^{47,48}. While the methodology proposed in this work can be extended to different frequency ranges, the selected range remains relevant for various engineering applications, such as noise control casings and household appliances^{2,24,49}, where dominant noise energy is often concentrated above 100 Hz, making attenuation in this band more critical than at lower frequencies. The secondary loudspeaker in the MISO-ANC and SISO-ANC configurations is also less effective in this band, with a typical lower operating limit near 65 Hz. Thus, although ASAC shows better performance at this range, the perceptual and practical impact of this advantage remains limited. In the frequency range dominated by the acoustic leakage through the opening (100–200 Hz), the loudspeaker-based configurations (MISO-ANC and SISO-ANC) demonstrate advanced performance compared to the ASAC configuration.

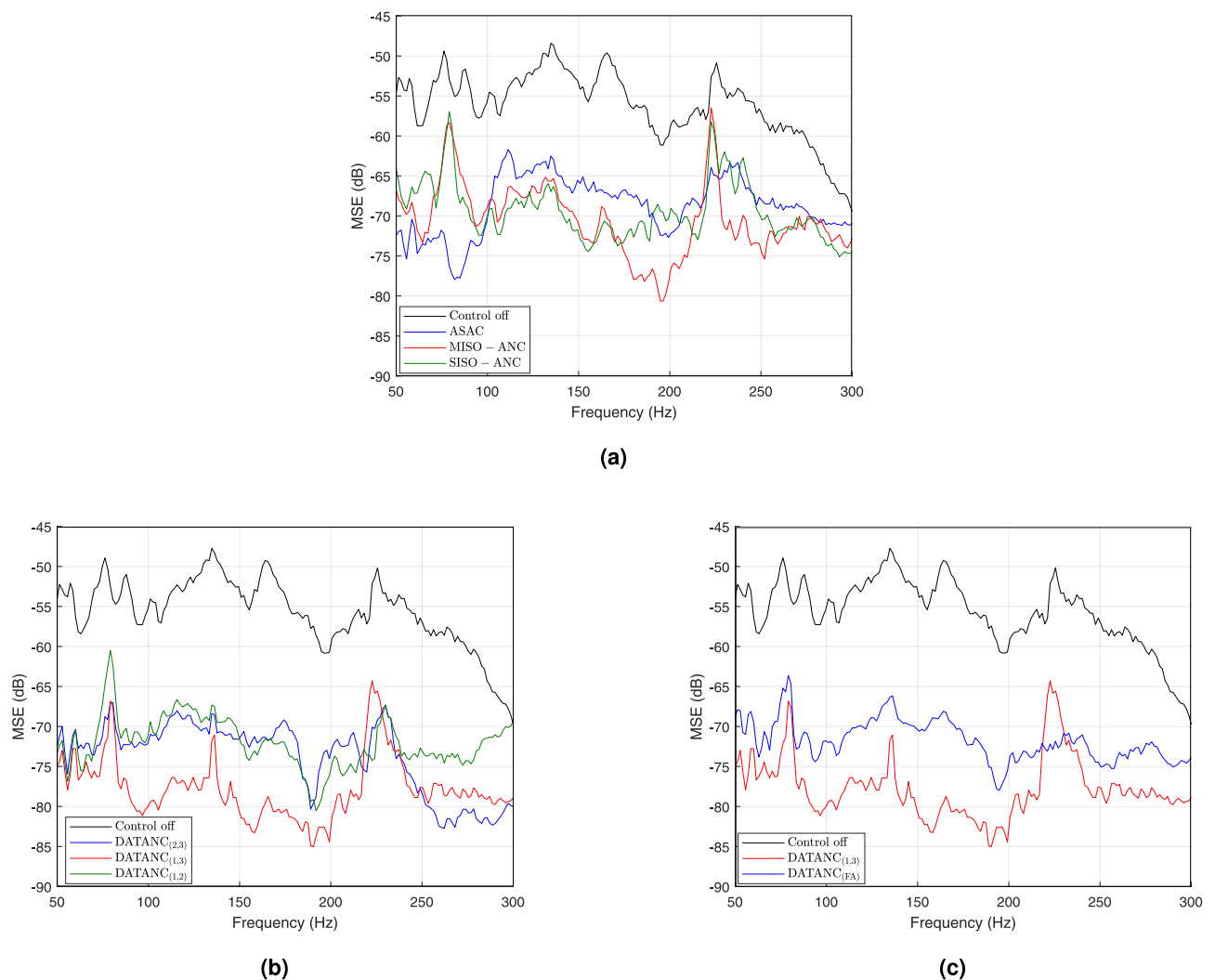


Fig. 6. MSE performance comparison across different active noise control configurations. **(a)** Conventional ANC methods: ASAC (three inertial actuators), SISO-ANC (a single loudspeaker with one error microphone), and MISO-ANC (a loudspeaker with three microphones). **(b)** Proposed DATANC configurations in which individual inertial actuators are selectively deactivated while the secondary loudspeaker remains active. **(c)** Comparison between DATANC_(1,3), which uses the loudspeaker along with actuators A1 and A3, and DATANC_(FA), which includes all actuators.

This improvement is due to the reduced vibro-acoustic coupling efficiency of the system in this band. Nevertheless, ASAC achieves an average noise reduction of approximately 12 dB, which remains significant for applications where only inertial actuators can be used. Loudspeaker-based configurations, however, achieve greater reduction in this frequency range, with MISO-ANC outperforming SISO-ANC by more than 2 dB on average. This additional performance comes at the cost of higher computational complexity, as MISO-ANC processes multiple reference signals and adapts multiple control paths simultaneously. Notably, despite relying on a single error microphone, the SISO-ANC configuration demonstrates a considerable global noise reduction across all three error microphones (for microphones power spectrograms see Supplementary information). A distinct resonance near 225 Hz, corresponding to a natural frequency of the system dominated more by the vibro-acoustic coupling, is observed due to the excitation from the primary source. As shown in Fig. 6a, loudspeaker-based configurations exhibit limited control performance at this frequency. This discrepancy is due to the placement of the loudspeaker near the enclosure opening, which limits its ability to effectively excite the structural modes. Consequently, the secondary source shows weak coupling with the vibrating structure, reducing its effectiveness in canceling the noise associated with this dominant resonance. The analysis of the system response using conventional control methods reveals that different actuator types exhibit superior performance in distinct frequency bands; hence, a hybrid approach that combines inertial and loudspeaker actuators may offer enhanced broadband control. This approach, referred to as Dual-Actuator-Type Active Noise Control (DATANC), is proposed based on insights from the performance of conventional control methods. The initial DATANC implementation employs a fully determined configuration with three error microphones and a combination of two inertial actuators and one loudspeaker, where actuator A1, A2, or A3 is deactivated in separate setups. These configurations are denoted as $\text{DATANC}_{(2,3)}$, $\text{DATANC}_{(1,3)}$, and $\text{DATANC}_{(1,2)}$, respectively, with the deactivated actuator remaining physically present. Their system responses are presented in Fig. 6b. The underdetermined configuration ($\text{DATANC}_{(\text{FA})}$) uses all four actuators simultaneously to evaluate performance under increased actuation power, as shown in Fig. 6c.

Dual-actuator-type active noise control

Among the fully determined DATANC configurations, $\text{DATANC}_{(2,3)}$ and $\text{DATANC}_{(1,2)}$ exhibit acceptable performance in the 100–200 Hz frequency range, achieving an average MSE reduction of approximately 18 dB across the error microphones. This level of attenuation is comparable to that obtained with the MISO-ANC configuration, as shown in Fig. 6b. However, $\text{DATANC}_{(1,3)}$ outperforms both configurations, providing over 8 dB of additional MSE reduction relative to $\text{DATANC}_{(2,3)}$ and $\text{DATANC}_{(1,2)}$ within the same frequency band. The spatial proximity between the loudspeaker and actuator A2 near the enclosure opening, negatively affects their mutual coherence. Thus, when both sources are active simultaneously, their acoustic fields interact destructively in this frequency band. This phenomenon is also observed in the $\text{DATANC}_{(\text{FA})}$ configuration, as in Fig. 6c. The resonance observed around 225 Hz lies outside this band. As a result, it leads to a reduction of more than 20 dB compared to the scenario without active control. This spatial analysis of the relative positions of the loudspeaker, inertial actuators, and vibrating plate is reflected in the transfer functions of the primary and secondary paths associated with the four actuators. These paths are identified using the normalized least mean square (NLMS) algorithm. Fig. 7 shows the average magnitude response of the primary and secondary paths across the three microphones.

The secondary loudspeaker exhibits the highest transfer function gain in the 100–200 Hz range, outperforming the inertial actuators. This higher response highlights the loudspeaker's dominant role in compensating for acoustic leakage through the enclosure opening. A3 demonstrates superior magnitude performance over the other inertial actuators, particularly in the 100–120 Hz and 180–200 Hz bands, making it a strong candidate for effective noise reduction within this frequency range (see Supplementary information for individual path responses).

In the phase responses shown in Fig. 8a and Fig. 8c, the paths associated with actuator A2 at Mic 1 and Mic 3 exhibit the greatest mismatch, not only with the primary path but also relative to the other secondary paths. A2 also shows an abrupt deviation from its prior phase trend beyond 100 Hz. This irregular and weakly correlated phase behavior reduces overall phase consistency across the control channels, thereby limiting the coherence required for effective noise cancellation in the $\text{DATANC}_{(\text{FA})}$ configuration. To summarize, the superior noise reduction performance of $\text{DATANC}_{(1,3)}$ can be explained by the spatial arrangement of the actuators and the characteristics of their transfer paths: $\text{DATANC}_{(1,3)}$ avoids destructive spatial coupling and benefits from more consistent phase relationships across the control channels, which together account for its superior attenuation performance. As shown in Fig. 9, the $\text{DATANC}_{(\text{FA})}$ configuration demonstrates the fastest convergence rate among all tested configurations. This rapid adaptation is due to the additional degrees of freedom provided by the underdetermined structure. However, this improvement comes at the cost of increased computational complexity, as more channels are used and solve an underdetermined system during adaptation. $\text{DATANC}_{(1,3)}$ configuration also performs better than the conventional control methods, achieving both faster convergence and a lower steady-state error.

Acoustic leakage

To assess the contribution of the sound transmission through the opening to the overall system response, the system is tested using a plate structure with a transparent plastic lid in the place of the former opening. In practice, this is also beneficial for enclosures where a view of the interior is required, or where a different material is needed for part of the wall due to various reasons (e.g., casings of household appliances often use different materials). As shown in Fig. 10, the passive system response highlights the impact of the structural opening: without the lid (Control off), the average sound pressure level is approximately -56 dB, whereas with the lid in place (Control off (Lid)), it is reduced to around -70 dB, clearly demonstrating the effect of sound transmission through the

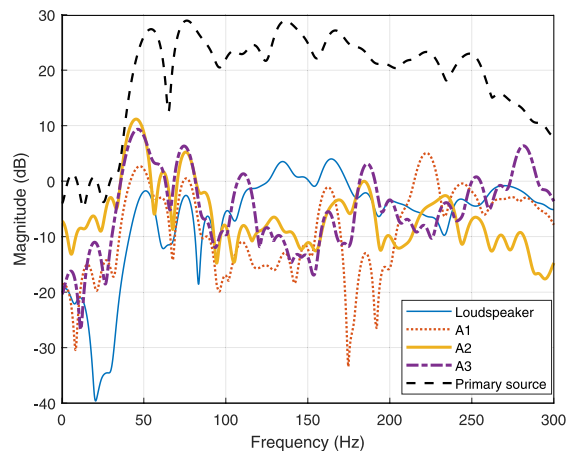
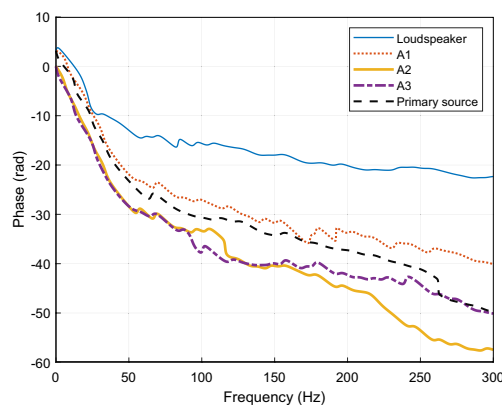
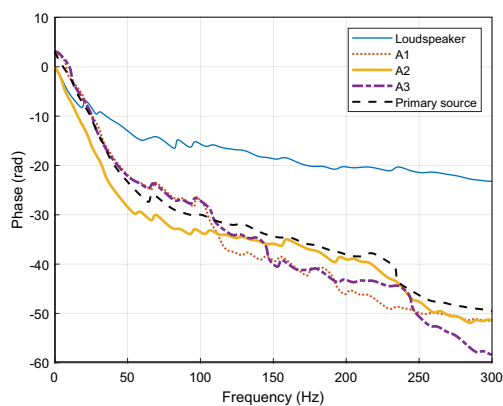


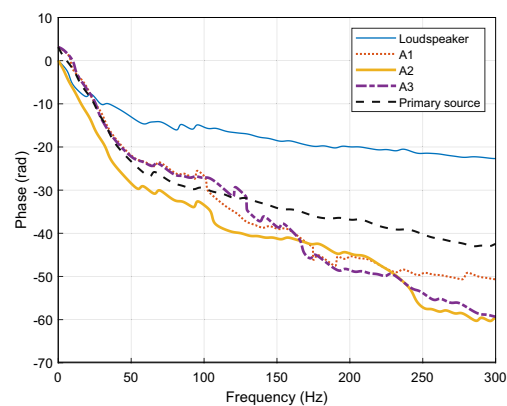
Fig. 7. Average magnitude responses of the acoustic transfer functions from the primary source and each secondary actuator (Loudspeaker, A1, A2, A3) to the microphones. The responses are averaged across the three microphones.



(a) Mic 1



(b) Mic 2



(c) Mic 3

Fig. 8. Phase responses of the acoustic transfer functions from each secondary source (Loudspeaker, A1, A2, A3) and the primary noise source to the three microphones: **(a)** Mic 1, **(b)** Mic 2, and **(c)** Mic 3.

opening. In the 100–200 Hz band, the loudspeaker-based configuration (SISO-ANC) no longer outperforms the ASAC approach in the presence of the transparent lid. For reference, see Fig. 6a for the corresponding comparison in the case of the system with acoustic leakage.

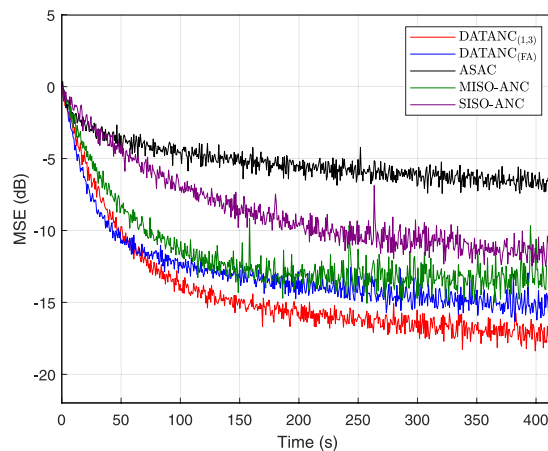


Fig. 9. MSE comparison over time for the conventional and the proposed active control configurations.

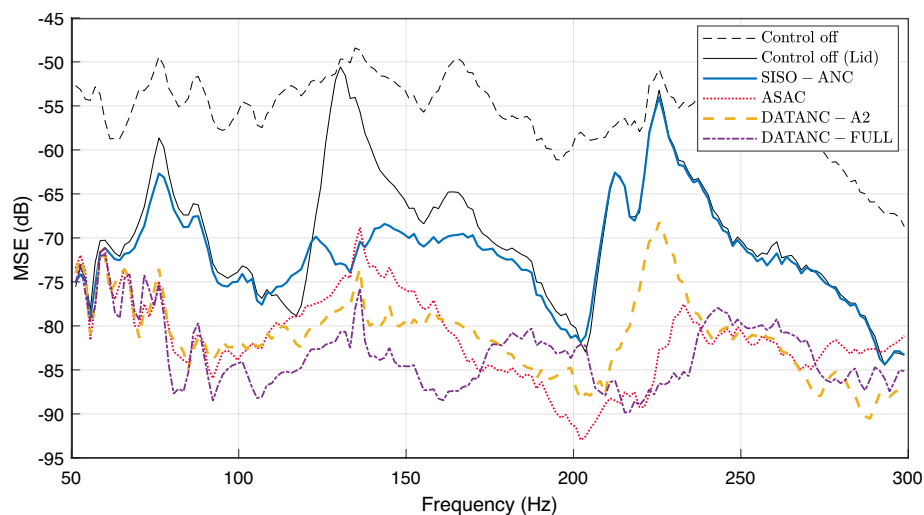


Fig. 10. MSE performance for different active noise control configurations in the presence of a structural opening sealed with a transparent lid. “Control Off” denotes the baseline configuration with the opening, while “Control Off (Lid)” indicates the same system with the lid applied. For ASAC, parameters were set to $\mu = 1 \times 10^{-6}$ and $\beta = 1 \times 10^{-6}$; for all other configurations, $\mu = 1 \times 10^{-7}$ and $\beta = 1 \times 10^{-5}$.

The ASAC system achieves over 30 dB attenuation at certain frequencies and an average reduction of approximately 12 dB across this frequency range, compared to only 4 dB for the SISO-ANC configuration. Additionally, the previously observed destructive acoustic coupling between inertial actuator A2 and the loudspeaker is no longer present, which enhances the performance of the full actuation configuration ($\text{DATANC}_{(\text{FA})}$), yielding an average reduction of 15 dB compared to 12 dB for $\text{DATANC}_{(1,3)}$. At higher frequencies, particularly around 225 Hz where the dominant vibro-acoustic resonance of the system occurs, the SISO-ANC configuration achieves no or minimal noise reduction. This underscores its limited effectiveness in addressing strongly coupled vibro-acoustic behavior at those frequencies. This limitation in performance also points to the need for further parameter tuning in the SISO-ANC configuration. In the absence of acoustic leakage, the actuator's ability to influence the acoustic field is reduced, making the system more sensitive to the adaptation parameters. For example, the SISO-ANC setup becomes unstable after 166 s when using $\mu = 1 \times 10^{-6}$ and $\beta = 1 \times 10^{-6}$, whereas improved stability is observed with $\mu = 1 \times 10^{-7}$ and $\beta = 1 \times 10^{-5}$, as shown in Fig. 11.

Discussion

This study investigates the performance of various active noise control strategies in vibro-acoustic systems that include openings in their noise barriers. Such systems are common in engineering applications where additional functions, such as airflow, are required. The analysis focuses on both conventional Active Noise Control (ANC) and Active Structural Acoustic Control (ASAC) approaches. A modification of a vibro-acoustic analytical model is proposed to account for sound transmission through the opening. Consequently, the optimization of the

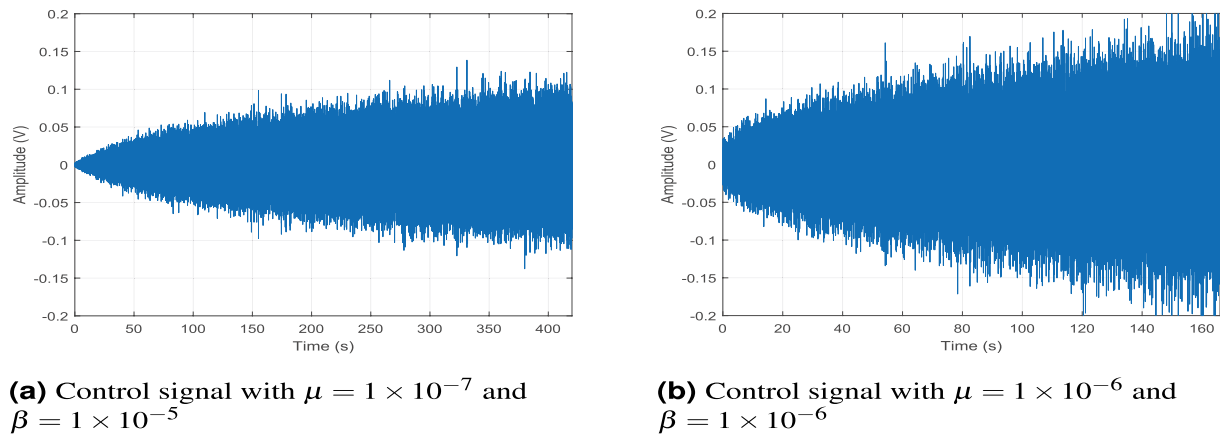


Fig. 11. Control signals for the actuator in a single-channel ANC system with a transparent lid. Panels **(a)** and **(b)** show signals generated using $\mu = 1 \times 10^{-7}$, $\beta = 1 \times 10^{-5}$, and $\mu = 1 \times 10^{-6}$, $\beta = 1 \times 10^{-6}$, respectively.

Configuration	Average MSE [100–200 Hz] (dB)	MSE near 225 Hz (dB)	MPI
Control off	–53.90	–50.84	–
DATANC _(1,3)	–79.73	–64.28	9900
DATANC _(1,2)	–71.94	–67.30	9900
DATANC _(2,3)	–71.48	–67.45	9900
MISO–ANC	–71.32	–56.47	3300
DATANC _(FA)	–70.84	–70.83	13200
ASAC	–66.37	–63.31	9900
SISO–ANC	–65.08	–58.20	1100
Control off (Lid)	–68.61	–50.58	–

Table 1. Summary of performance metrics across passive and active control configurations for the system with an opening. The reported average MSE values are first computed as the mean across the three microphones, and then averaged over the 100–200 Hz frequency band. The MSE near 225 Hz corresponds to the dominant vibro-acoustic resonance of the system. MPI (Multiplications Per Iteration) indicates the computational complexity required for each control update.

inertial actuator placements is guided by both sound radiation and acoustic leakage, demonstrating reliable guidance for actuator arrangement that captures the key physical characteristics of the system and results in more effective ASAC performance in experimental testing compared to alternative configurations. It also analyzes the proposed approach, Dual-Actuator-Type Active Noise Control (DATANC), which integrates the advantages of loudspeakers and inertial actuators. Among the tested ASAC configurations, the setup with actuators positioned near the edge of the structural opening (Set 1) achieves the lowest MSE, with an average reduction of 12 dB in the 100–200 Hz band. In comparison, Sets 2 and 3 achieve average reductions of 5 dB and 8 dB, respectively. This enhanced performance is attributed to the increased local inertia from the added mass and the actuators’ ability to induce greater structural displacement in the vibrating plate. Sound transmission through the system’s structural opening predominantly affects the 100–200 Hz frequency range. This behavior is confirmed experimentally by evaluating the system’s performance with and without a transparent PVC lid used to minimize acoustic leakage. As shown in Table 1, the passive configuration with the lid (Control off (Lid)) achieves a substantially lower average MSE of –68.61 dB, compared to –53.90 dB for the open configuration (Control off).

Thus, in the 100–200 Hz frequency range where sound transmission through the opening dominates, loudspeaker-based control methods (MISO-ANC and SISO-ANC) outperform ASAC, which relies only on inertial actuators. However, inertial actuators are particularly effective at attenuating the system’s strong vibro-acoustic resonances, where this coupling coupling is high (e.g., around 225 Hz in the system under consideration). These observations highlight the need for a hybrid control strategy that integrates structural and acoustic actuation to ensure effective broadband attenuation. Nonetheless, such configurations must be carefully designed, as interaction between closely positioned actuators near the opening can degrade control performance. This is evidenced by the performance of DATANC_(1,3), which achieves better overall noise reduction than the full actuation configuration DATANC_(FA) while incurring lower computational cost. In contrast, DATANC_(FA) exhibits superior performance in reducing the resonance around 225 Hz, along with an enhanced convergence rate, due to the increased actuation availability. The computational complexity associated with each configuration

by calculating the multiplications per iteration (MPI) using the formula $MPI = M_r \cdot P_a \cdot (2N_c + L_s)$, where M_r denotes the number of input reference signals, P_a represents the number of actuators or control signals, N_c is the length of each control filter, and L_s is the length of the secondary path model. The table also evaluates the computational complexity associated with each configuration by calculating the multiplications per iteration (MPI) using the formula $MPI = M_r \cdot P_a \cdot (2N_c + L_s)$, where M_r denotes the number of input reference signals, P_a represents the number of actuators or control signals, N_c is the length of each control filter, and L_s is the length of the secondary path model. Overall, this study underscores the importance of optimally placing the system actuators to enhance functionality, as demonstrated through analytical and experimental investigations. It also highlights the effectiveness of combining loudspeakers and inertial actuators for active noise control in systems with structural openings, offering a practical compromise between performance and computational cost. Future work will explore the application of DATANC in commercial appliances where lightweight panels and structural openings are essential for airflow, monitoring, and maintenance. These systems often suffer from noise leakage through necessary opening, making them ideal candidates for hybrid active control strategies that balance noise reduction with functional design constraints. Additionally, in more sophisticated systems, such as those requiring multiple openings and additional secondary loudspeakers, the flexibility of the analytical model and its proposed modifications will allow these configurations to be examined. Building on the current optimization framework, future investigations could establish relationships between plate size, the number of openings, and the required actuators to further inform practical design decisions.

Data availability

The datasets used and/or analysed during the current study available from the corresponding author on reasonable request.

Received: 14 June 2025; Accepted: 27 August 2025

Published online: 02 September 2025

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Acknowledgements

The work was supported by the project IN-NOVA: Active reduction of noise transmitted into and from enclosures through encapsulated structures, which has received funding from the European Union's Horizon Europe program under the Marie Skłodowska-Curie grant agreement no. 101073037.

Author contributions

KSA.M. and CK.L. conducted the main work of this manuscript under the supervision of S.W., M.P., and HK.R. KSA.M. derived the formulations and performed the experiments. CK.L. implemented the real-time control algorithm under the guidance of S.W. and M.P. KSA.M. wrote the initial draft of the manuscript under the supervision of HK.R. All authors reviewed the final manuscript.

Funding

The work was supported by the project IN-NOVA: Active reduction of noise transmitted into and from enclosures through encapsulated structures, which has received funding from the European Union's Horizon Europe program under the Marie Skłodowska-Curie grant agreement no. 101073037.

Declarations

Competing interests

The authors declare no competing interests.

Additional information

Supplementary Information The online version contains supplementary material available at <https://doi.org/10.1038/s41598-025-17810-8>.

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Synergistic use of inertial actuators for advanced noise control in transparent-patched vibrating plates

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ARTICLE INFO

Keywords:

Active structural acoustic control
Actuator placement optimization
Sound insulation
Multifunctional integrated design
Transparent structures

ABSTRACT

Transparent and lightweight structures are increasingly used in applications where visual access and weight reduction are critical, yet they offer poor acoustic insulation—particularly at low frequencies. This study introduces a novel Active Structural Acoustic Control (ASAC) strategy for transparent-patched vibrating plates for advanced noise reduction. The system is experimentally tested under broadband excitation from 50 to 300 Hz, focusing on how the quantity and placement of inertial actuators affect system performance. To support optimal actuator configuration, this work introduces a novel analytical model that enables optimized actuator placement, incorporating structural dynamics and acoustic transmission loss through the transparent patch. Experimental results show that optimized single-actuator setups achieved up to a 5 dB reduction in resonance peaks, while well-positioned multi-actuator configurations yielded over 12 dB average sound power reduction in the considered frequency range and notable suppression of dominant resonances. However, the use of multiple actuators requires greater attention to spatial distribution, as poor placement can diminish their synergistic effect, increase control effort, and limit overall performance. This work introduces advanced methodologies in the application of Active Structural Acoustic Control for transparent-patched lightweight structures, providing new insights into how actuator configuration influences the acoustic performance.

1. Introduction

Exposure to elevated noise levels is widely acknowledged as a significant risk to human health, comfort, and safety [1]. The increasing prevalence of industrial machinery, household appliances, and transportation vehicles has contributed to higher noise levels, potentially undermining the health and compromising operational safety [2,3]. Lightweight structures, especially thin plates used in aerospace, automotive, and domestic devices, often suffer from poor acoustic insulation and are prone to vibration-induced noise [4,5]. In such applications, visual access through structural elements is crucial for effective system monitoring, vehicle operation, and user comfort [6–8]. For passive noise insulation, laminated glass is well recognized for its sound insulation capabilities [9,10]. This performance is often enhanced by incorporating inorganic fillers, which increase the path length for sound wave propagation, thereby improving acoustic attenuation [11,12]. However, insufficient filler content can lead to the formation of micro-cracks and a significant reduction in the mechanical integrity of the composite structure. Absorption efficiency increases with structural thickness [13], leading to poor low-frequency performance in thin

structures. This limitation is particularly significant for many domestic appliances, industrial machines, and transportation vehicles, where the dominant noise components typically lie below 500 Hz. Rigid panels typically exhibit superior sound reflection in accordance with the mass law, where sound transmission loss (STL) increases with frequency and surface density. However, this approach is incompatible with lightweight design requirements. A recent study [14] demonstrates that lightweight retrofit solutions can achieve up to 5 dB noise reduction without significant mass penalties, reinforcing the viability of low-mass strategies in noise-sensitive aerospace applications. However, further reduction of cabin noise also depends on suppressing fuselage panel vibrations, which remains challenging, as conventional vibration isolators and insulation materials can increase panel weight [15]. The limitation in material choices imposed by transparency and lightweight requirements often demands the integration of more advanced noise reduction strategies to enhance the acoustic performance. Semi-active control concepts have emerged as intermediate solutions, offering improved adaptability with lower energy demands compared to fully active systems. For instance, tunable-inertia actuators have been proposed

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to enhance vibration isolation performance under changing dynamic conditions [16,17]. More recently, piezoelectric meta-plates with non-linear semi-active electrical interfaces have demonstrated broadband vibration suppression at low frequencies while maintaining structural lightness [18]. However, as the method lacks cooperative active control among periodic cells, it may suffer from reduced control synergy across the structure. Active Structural Acoustic Control (ASAC) is a proven technique for mitigating noise transmission by actively suppressing structural vibrations in various engineering systems [19,20]. ASAC is implemented using lightweight panels, inertial actuators, sensors, and a control system, as outlined in [21,22].

The placement of inertial actuators on vibrating plates plays a critical role in determining the effectiveness of active noise control systems, as it directly influences the system's ability to attenuate structural-acoustic responses [23]. Several studies have investigated actuator placement strategies. For example, some approaches optimize placement based on the vibration response, employing techniques such as a modified heuristic genetic algorithm (MHGA) for positioning piezoelectric actuators to suppress dominant vibration modes [24]. Other studies extend the cost function to include the radiated sound field, aiming to reduce acoustic emissions more effectively [5]. In [25], a memetic algorithm was applied to enhance structural controllability, achieving a reduction of up to 5 dB in acoustic response. However, there remains a gap in the literature regarding the influence of inertial actuator placement on the measured sound power radiated from vibrating plates incorporating transparent patches—an area that this study aims to address. Recently, authors in [26] proposed an intelligent control strategy for helicopter cabin noise reduction, using a neural network-based model combined with genetic algorithms to identify dominant acoustic radiation panel elements and optimize the placement of sensors and actuators.

Since its introduction in 1936, the filtered-x least-mean-square (FxLMS) algorithm has become the most widely used approach in active control due to its stability and simple structure [27,28]. In this algorithm, the reference signal is filtered through a model of the secondary path — between the actuator and error sensors — to ensure accurate adaptation [29,30]. In practical implementations, it is essential to constrain the control filter's output amplitude to avoid actuator overload [31,32]. To address this, the leaky FxLMS algorithm introduces a leakage factor that limits the control effort and enhances adaptation stability [33,34]. The authors in [35] conducted a theoretical evaluation of a radiation-mode-based ASAC system for an enclosure with vent openings, focusing solely on tonal disturbances without addressing real-world applicability. A feedforward ASAC algorithm was subsequently implemented on aircraft cabin lining panels using a low-cost microcontroller, achieving a 23 dB reduction in tonal noise [36]. In [37], a tonal ASAC strategy utilizing an experimentally identified radiation matrix was shown to outperform conventional active vibration control methods in regulating sound power radiation. These studies were limited to tonal excitations and did not address broadband noise scenarios. However, the application of ASAC to transparent structures remains insufficiently investigated, primarily due to the mechanical limitations of these materials, which often reduce the efficiency of interaction with actuation forces.

This gap in the literature underscores the need for ASAC strategies specifically tailored to structural systems incorporating transparent components. Unlike the work presented in [25,26], the present study investigates the implementation of ASAC in transparent-patched vibrating plates, aiming to achieve both optical transparency and the mechanical performance required for effective active noise reduction. The main contribution of this paper is threefold: First, it introduces a novel ASAC system that combines inertial actuators with transparent-patched plates. Second, it develops an analytical optimization method for actuator placement that explicitly accounts for the acoustic transmission loss characteristics of the transparent patch, enabling more effective control strategies. Third, it experimentally evaluates various

multiactuator configurations under broadband excitation, using sound power estimation techniques to demonstrate the impact of actuator positioning on measured acoustic performance.

The remainder of this paper is organized as follows: Section 2 presents the analytical model used for estimating total sound power with the aim of achieving optimal noise reduction. Section 3 details the experimental setup developed for the active control system. Section 4 discusses the experimental results and evaluates system performance with varying numbers of actuators. Finally, Section 5 concludes the paper and summarizes the key findings.

2. Analytical modeling

The vibro-acoustic system introduced in [38] is modified to incorporate a hybrid-material noise barrier, in which a steel vibrating plate is patched with semi-rigid polyvinyl chloride (PVC) to cover a circular opening at its center. Within the framework of this article, the noise barrier is actively controlled, necessitating an accurate estimation of the actuators' effects. While providing exact experimental validation of this method is challenging, the experimental results presented in Section 4 demonstrate that the proposed approach offers a promising pathway for identifying effective actuator placements.

2.1. Vibrating noise barrier

The dimensions of both the patch and the inertial actuator are relatively small compared to those of the steel plate, as in Fig. 1. Assuming ideal bonding, their influence on the plate's strain and added stiffness can be considered negligible. Consequently, their primary contribution is attributed to the kinetic energy of the system. Thus, the kinetic energy of the system, denoted as T_s , need to be adjusted compared to the model in [38]. The vibrating plate, assumed to be isotropic and homogeneous, is modeled using the Kirchhoff–Love theory for thin plates [39], considering only transverse motion and neglecting the effects of rotary inertia. The total kinetic energy of the system is expressed as:

$$T_s = \frac{1}{2} \iint_{\Omega_s} \rho(x, y) H(x, y) \dot{w}^2 dx dy, \quad (1)$$

where $\rho(x, y)$ and $H(x, y)$ are the varying density and thickness, respectively, defined as:

$$H(x, y) = \begin{cases} h_p, & \text{if } (x, y) \in \Omega_p, \\ h_{at}, & \text{if } (x, y) \in \Omega_{at}, \\ h_{pt}, & \text{if } (x, y) \in \Omega_{pt}, \end{cases} \quad \rho(x, y) = \begin{cases} \rho_p, & \text{if } (x, y) \in \Omega_p, \\ \rho_{at}, & \text{if } (x, y) \in \Omega_{at}, \\ \rho_{pt}, & \text{if } (x, y) \in \Omega_{pt}, \end{cases} \quad (2)$$

where h and ρ refer to the local thickness and material density, respectively, while the subscripts p , at , and pt indicate the steel plate, actuator, and transparent patch region. The domains Ω_p , Ω_{at} , and Ω_{pt} define the spatial regions occupied by the steel substrate, the actuator, and the PVC patch, respectively. The plate is assumed to be fully clamped and its deflection in the z -direction at time t is denoted as $w(x, y, t)$. The initial conditions are given by

$$w(x, y, t_0) = 0, \quad \dot{w}(x, y, t_0) = \left. \frac{\partial w(x, y, t)}{\partial t} \right|_{t=t_0} = 0, \quad t > t_0 > 0. \quad (3)$$

2.2. Sound power estimation

In the case of a vibrating plate with an opening, the total sound power comprises two components: the power radiated by the vibrating plate and the power lost as acoustic leakage through the opening. It can be expressed as:

$$\bar{\Theta}_{\text{total}}(\omega) = \bar{\Theta}_{\text{rad}}(\omega) + \bar{\Theta}_{\text{loss}}(\omega). \quad (4)$$

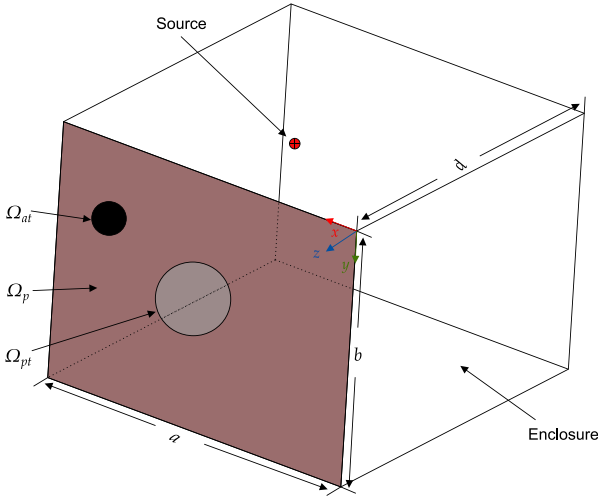


Fig. 1. Schematic of the analytical model, including the enclosure. The domains Ω_p , and Ω_{pt} represent the steel plate and the transparent patch, respectively. The actuators domain, Ω_{at} , is a combination of circles with a radius of 0.02 m for each actuator. The primary source is located at the coordinates (0.3 m, 0.1 m, -0.3 m).

In this article, the sound power consists of two primary components: the sound radiated by the vibrating plate and the sound transmitted through the PVC patch. The total sound power can be expressed as:

$$\bar{\Theta}_{\text{total}}(\omega) = \bar{\Theta}_{\text{rad}}(\omega) + \tau_{\text{pt}}(\omega) \cdot \bar{\Theta}_{\text{loss}}(\omega), \quad (5)$$

where τ_{pt} is the transmission coefficient of the PVC patch, which is defined in detail later in Eq. (8), and $\bar{\Theta}_{\text{rad}}(\omega)$ is the radiated sound power from the plate, expressed as:

$$\bar{\Theta}_{\text{rad}}(\omega) = u_r^H R(\omega) u_r, \quad (6)$$

where the superscript H denotes the complex conjugate transpose, u_r is the plate velocity vector, and can be extracted from the last N_p^2 elements of the system response u . Here, N_p is the number of modes considered in one direction of the 2D vibrating plate, and $R(\omega)$ is the radiation resistance matrix at frequency ω . The term $\bar{\Theta}_{\text{loss}}(\omega)$ represents the sound power transmitted through the patch and is given by:

$$\bar{\Theta}_{\text{loss}}(\omega) = \frac{u_p^H u_p}{\rho_a c} A_s, \quad (7)$$

where u_p is the acoustic pressure at the patch interface, obtained from the first N_c^3 elements of the system response. N_c is the number of modes considered in one direction of the 3D enclosure, A_s is the surface element area, ρ_a is the air density, and c is the speed of sound in air. The novelty here is the introduction of the transmission coefficient of the PVC patch, τ_{pt} , into the sound power estimation. It is computed based on the mass-law approximation as:

$$\tau_{\text{pt}}(\omega) = \frac{1}{1 + \left(\frac{\rho_a c}{2m_{\text{pt}}\omega} \right)^2}, \quad (8)$$

where m_{pt} is the areal mass density of the PVC patch. Substituting Eqs. (6) and (7) into Eq. (5), the total sound power is given by:

$$\bar{\Theta}_{\text{total}}(\omega) = u_r^H R(\omega) u_r + \tau_{\text{pt}}(\omega) \cdot \frac{u_p^H u_p}{\rho_a c} A_s. \quad (9)$$

To estimate the effect of the inertial actuator forces on the total sound power, the additive synthesis method [5] is utilized. The plate velocity u_r and the acoustic pressure at the patch interface u_p are expressed as:

$$u_r = u_{r,\text{pri}} + \sum_{k=1}^K u_{r,\text{unit},k} f_{c,k}, \quad (10)$$

where $u_{r,\text{pri}}$ represents the response due to the primary source alone, and $u_{r,\text{unit},k}$ represents the response due to the k th unit control source.

$$u_p = u_{p,\text{pri}} + \sum_{k=1}^K u_{p,\text{unit},k} f_{c,k}, \quad (11)$$

where $u_{p,\text{pri}}$ and $u_{p,\text{unit},k}$ represent the pressure responses due to the primary source and the k th unit control source, respectively. The complex scalar $f_{c,k}$ is the amplitude of the k th actuator force.

Substituting into the total sound power expression, the resulting form becomes:

$$\begin{aligned} \bar{\Theta}_{\text{total}}(\omega) = & u_{r,\text{pri}}^H R(\omega) u_{r,\text{pri}} + \tau_{\text{pt}}(\omega) \cdot \frac{u_{p,\text{pri}}^H u_{p,\text{pri}}}{\rho_a c} A_s \\ & + \sum_{k=1}^K \sum_{j=1}^K f_{c,k}^H u_{r,\text{unit},k}^H R(\omega) u_{r,\text{unit},j} f_{c,j} + \sum_{k=1}^K f_{c,k}^H u_{r,\text{unit},k}^H R(\omega) u_{r,\text{pri}} \\ & + \sum_{k=1}^K \left(u_{r,\text{unit},k}^H R(\omega) u_{r,\text{pri}} \right)^H f_{c,k} \\ & + \tau_{\text{pt}}(\omega) \cdot \frac{1}{\rho_a c} A_s \left(\sum_{k=1}^K \sum_{j=1}^K f_{c,k}^H u_{p,\text{unit},k}^H u_{p,\text{unit},j} f_{c,j} + \sum_{k=1}^K f_{c,k}^H u_{p,\text{unit},k}^H u_{p,\text{pri}} \right. \\ & \left. + \sum_{k=1}^K \left(u_{p,\text{unit},k}^H u_{p,\text{pri}} \right)^H f_{c,k} \right). \end{aligned} \quad (12)$$

The quantities $u_{r,\text{unit},k}$ and $u_{p,\text{unit},k}$ are influenced by both the number and spatial placement of the actuators on the plate. Consequently, the total sound power $\bar{\Theta}_{\text{total}}(\omega)$ can be minimized by optimizing the actuator configuration in terms of quantity and location. The genetic algorithm framework proposed in [7] is adapted in this work to optimize the spatial arrangement of inertial actuators, based on the minimization of total sound power. While the original formulation in [7] focuses on optimizing opening shapes and locations, our implementation modifies the variable vector definition to suit actuator placement requirements. This framework utilizes a parallel master-slave architecture distributed across multiple CPU cores. Computations are performed on a workstation equipped with a 13th Gen Intel Core i5-13500 processor (14 cores) and 32 GB of RAM. While this setup is efficient for medium-scale panel size, larger systems may require additional computational resources.

It should be emphasized that this represents a theoretical estimation of the sound power, and achieving a one-to-one validation with an actual control setup would be challenging. Nonetheless, the proposed approach provides valuable guidance for determining optimal actuator placements for effective noise reduction [5,17].

The model parameters selected in this study reflect commonly used values in engineering applications. The vibrating plate is made of steel, with a density of $\rho = 7850 \text{ kg/m}^3$ and a Young's modulus of $E = 210 \times 10^9 \text{ Pa}$. Its dimensions are $a = 0.42 \text{ m}$ (length), $b = 0.39 \text{ m}$ (width), and $h_p = 0.001 \text{ m}$ (thickness). The enclosure is defined with a depth of $d = 0.4 \text{ m}$, and is filled with air of density $\rho_a = 1.2 \text{ kg/m}^3$ and sound speed $c = 343 \text{ m/s}$. The theoretical height of each actuator is $h_{\text{at}} = 0.006 \text{ m}$, corresponding to an individual actuator mass of approximately 150 g. While the thickness of the PVC patch $h_{\text{pt}} = 0.5 \text{ mm}$, with density $\rho_{\text{pt}} = 1300 \text{ kg/m}^3$.

3. Experimental setup

The experimental setup consists of a rectangular steel plate mounted to a concrete enclosure with a loudspeaker inside. The primary loudspeaker is set to generate a random, band-limited noise between 50 Hz to 300 Hz with a position fixed at (0.3 m, 0.1 m, -0.3 m) for all the considered configurations. While the methodology proposed in this work can be extended to other frequency ranges, the selected band remains highly relevant for a variety of engineering applications, such as noise control casings for household appliances and consumer electronics [40–42]. These applications often require casings to incorporate

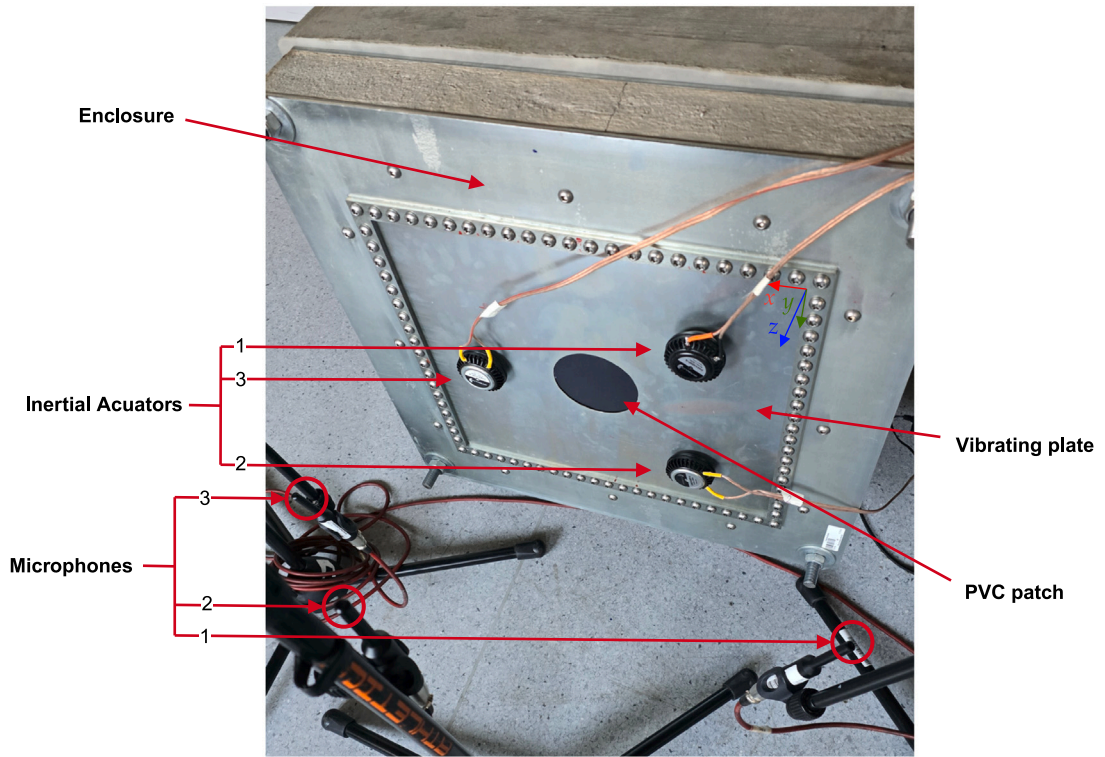


Fig. 2. Overview of the experimental setup showing a concrete enclosure with a vibrating plate that contains a PVC patch of a radius 0.05 m. Three actuators mounted on the plate at (0.11 m, 0.1 m), (0.11 m, 0.32 m), and (0.29 m, 0.2 m). The microphones are positioned at the coordinates (−0.18 m, 0.12 m, 0.25 m), (0.21 m, −0.1 m, 0.4 m), and (0.53 m, 0.24 m, 0.25 m).

transparent sections for device monitoring or lightweight materials for improved portability and efficiency, which can compromise acoustic isolation. As a result, attenuation within this frequency range becomes particularly important, as it typically corresponds to dominant structural resonances and contributes significantly to the perceived noise level. Frequencies below this range are generally less critical in practice, as they tend to have lower perceptual impact [43,44]. The concrete walls of the enclosure provide a high degree of sound attenuation; hence, the majority of the acoustic energy that exits the enclosure is transmitted through the plate and the circular patch, as shown in Fig. 2. Three Beyerdynamic MM1 error microphones are used for the active control system, along with one to three DAEX32EP-4 inertial actuators, whose positioning in different configurations will be discussed in this section. The actuator placement is selected using theoretical sound power calculations, whereas the experimental measurements relies on fixed microphone positions within an adaptive active noise control framework. These positions are carefully selected to capture the dominant acoustic field. The microphones are spaced such that both their distances from each other and from the vibrating panel remain sufficiently small to enable global noise reduction [45–47]. While small changes in microphone placement may affect local response characteristics, the overall global performance is not expected to be significantly impacted. A detailed investigation into the optimal positioning of error microphones is considered beyond the scope of this study. The laboratory room has dimensions of 5.8 m in length and 3.5 m in width. Despite the walls being treated with sound-absorbing materials, the presence of various equipment and objects within the space results in acoustic properties that more closely mimic a real-world environment than those of a purpose-built acoustic chamber. The setup for the hardware is equipped with the Analog Devices SC598 with a SHARC+ core, along with the EVSOMCRR-EZKIT development board [48].

A feedforward multichannel FxLMS algorithm with a leakage factor is employed for the active structure acoustic control (ASAC), due to its simplicity and robustness [28]. In the adaptive filtering process, the algorithm updates the coefficients of the K adaptive filters $W_k(n)$ in the controller as follows:

$$W_k(n+1) = (1 - \alpha(n)\lambda) W_k(n) + \alpha(n) \sum_{j=1}^J \tilde{R}_{kj}(n) \varepsilon_j(n), \quad (13)$$

where $\tilde{R}_{kj}(n)$ is the filtered reference signal, $\varepsilon_j(n)$ is the error signal from the j th sensor, and $\alpha(n)$ is the normalized step size defined by:

$$\alpha(n) = \frac{\alpha_n}{\sum_{k=1}^K \sum_{j=1}^J \|\tilde{R}_{kj}(n)\|^2}, \quad (14)$$

where α_n is the fixed step size, $\|\cdot\|$ denotes the L^2 -norm, and λ is the leakage factor that helps prevent coefficient drift. The length of each of the control filters is set to 400, while the secondary paths consist of 300 coefficients each, with $\alpha_n = 1 \times 10^{-6}$ and $\lambda = 1 \times 10^{-6}$.

4. Results and discussion

In this study, configurations with one, two, and three actuators are investigated to evaluate their impact on noise reduction performance, as in Fig. 3. The analysis focuses on the synergistic effects of multiple actuators and the influence of their spatial arrangement including mutual positioning, placement on the plate, and relative location to the noise source. This section also demonstrates how the analytical optimization method presented in Section 2 can guide the strategic placement of actuators to enhance the overall performance of the system.

Fig. 4 compares the mean square error (MSE) of the three microphones in each configuration in which a single actuator is used. It is clear that the positioning of the actuator, even with no control

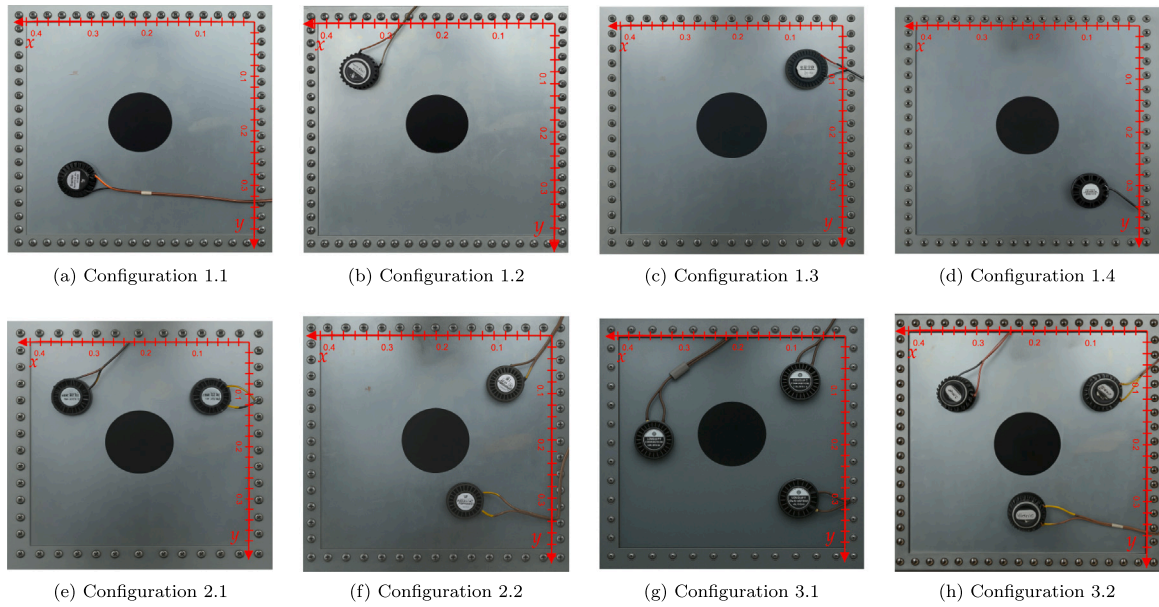


Fig. 3. The investigated configurations.

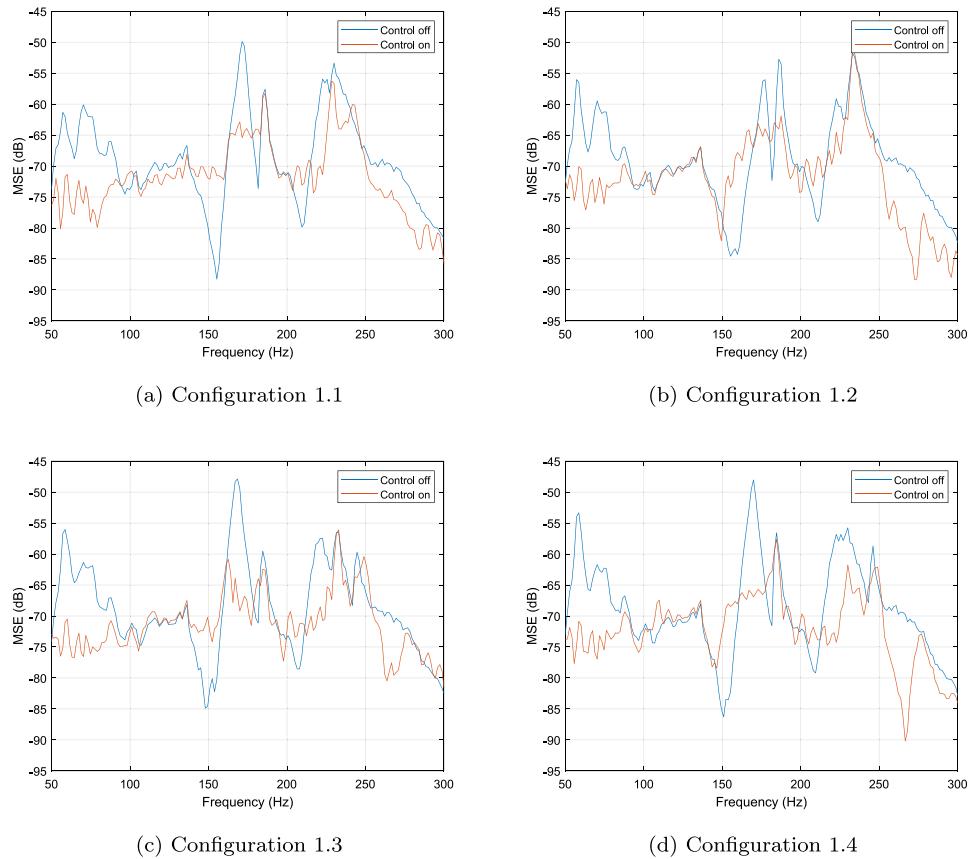


Fig. 4. MSE of error microphone signals comparing different configurations for a single actuator.

effort, affects the system response. Previous studies have shown that the system's high-frequency resonances are primarily attributed to acoustic radiation from the vibrating panel rather than to the sound leakage. This understanding is supported by previous studies [7,38], which showed that variations in leakage primarily influence the average sound power level, while the resonance remains largely governed by the structural dynamics of the panel itself. For instance, Configuration

1.2 has the additional mass close to the noise source, which acts like direct additional damping on the plate and results in a 5 dB reduction in the resonances between 150 Hz and 200 Hz compared to the other configurations, as in Fig. 4(a). Placing an actuator in an anti-nodal region alters the local inertia of the plate and increases its resistance to deformation, since these areas are subject to high displacement and mechanical stress. Consequently, the plate exhibits lower vibration

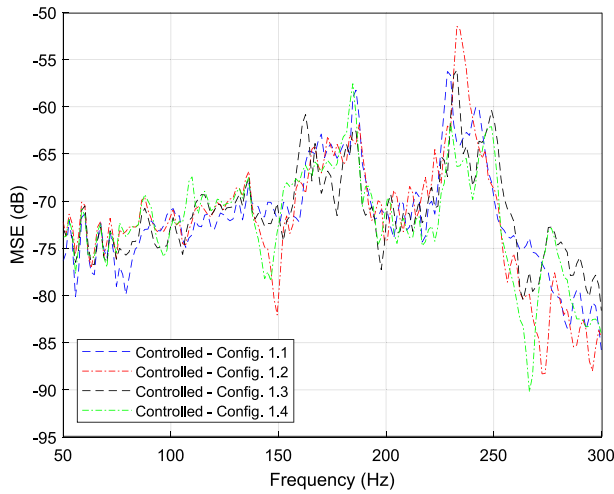


Fig. 5. Comparison of the noise signals in case of active control for configurations with single actuator.

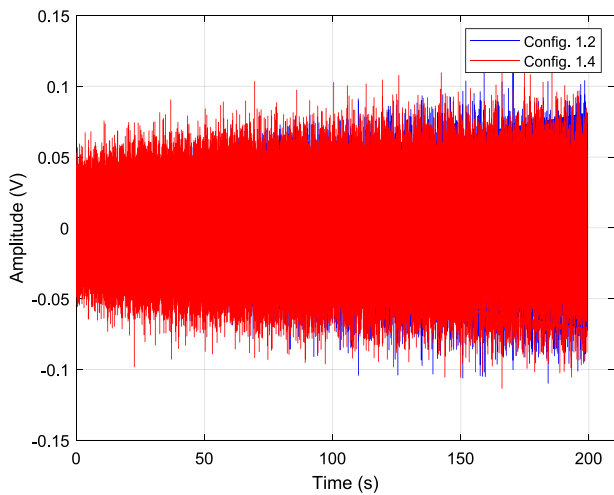


Fig. 6. Control signal comparison between configurations 1.2 and 1.4.

amplitudes under the same actuation force [49], particularly at modes where the anti-nodal region aligns with the actuator mass. This leads to a reduction in the radiated sound power [7].

However, this could limit the possible control effort of the actuator, as it will be difficult to introduce more displacement in the region close to the primary noise source. In contrast, the optimization method recommends placing the actuator in the corner opposite the primary noise source, as seen in Configuration 1.4.

Furthermore, comparing Configurations 1.3 and 1.4 reveals that the actuator in Configuration 1.4 is positioned farther from the clamped edge, which typically experiences higher mechanical stresses and may restrict the actuator's ability to induce additional vibrations in the plate. To better compare the single-actuator configurations under ASAC control, Fig. 5 is presented.

Configuration 1.4 demonstrates superior performance compared to the other configurations, particularly in the 200–250 Hz range, where it effectively suppresses the highest resonance observed across all setups. Remarkably, this improved performance does not come at the cost of increased control effort; Configuration 1.4 requires no significantly greater control input to suppress its lowest resonance than Configuration 1.2, which exhibits the highest peak.

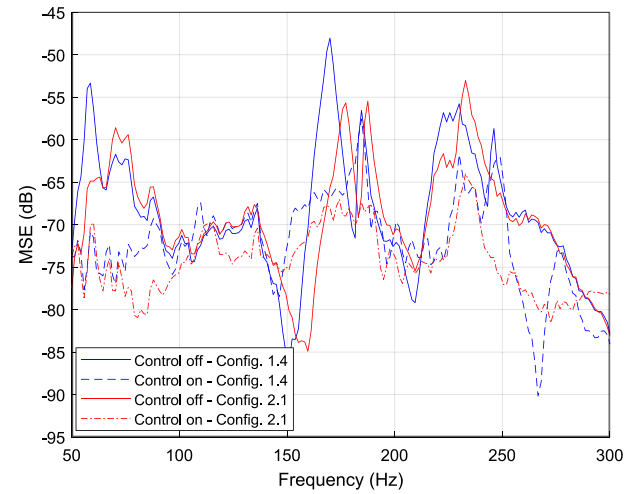


Fig. 7. MSE of the microphones in configurations 1.4 and 2.1.

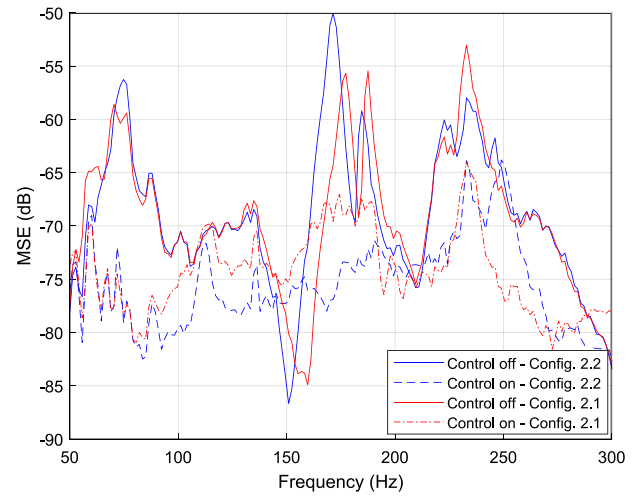


Fig. 8. MSE of the microphones in configurations 2.1 and 2.2.

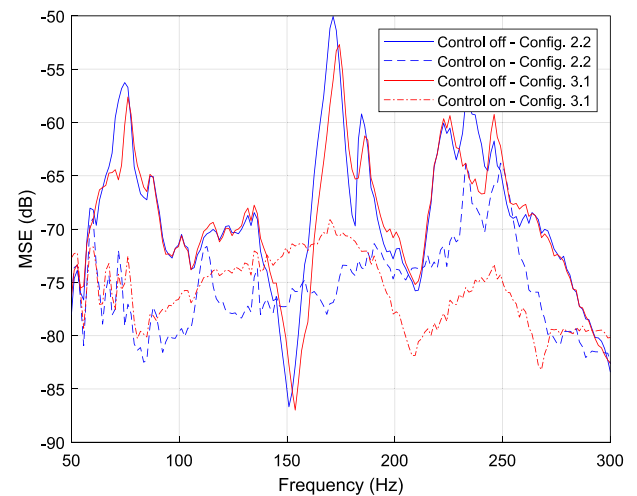


Fig. 9. MSE of the microphones in configurations 2.2 and 3.1.

The control efforts of the actuators in Configurations 1.2 and 1.4, representing the highest and lowest resonance peaks within the considered frequency range, are shown in Fig. 6. Additionally, the average

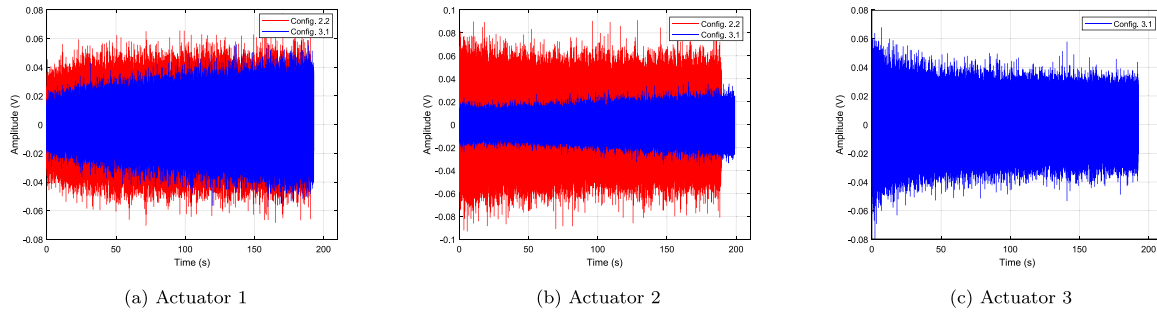


Fig. 10. Control signals of the actuators in configurations 2.2 and 3.1.

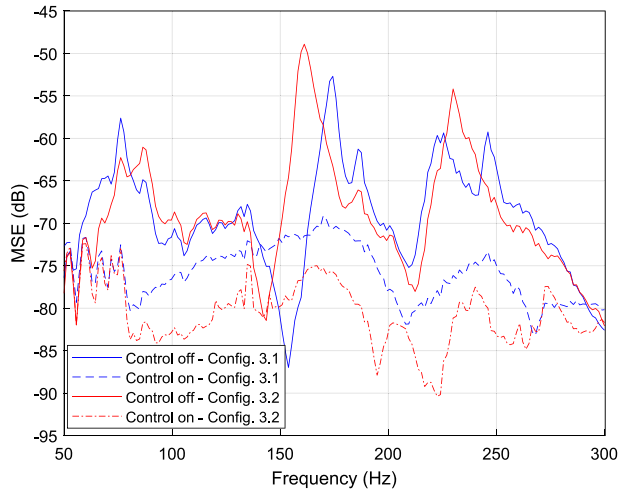


Fig. 11. MSE of the error microphones in configurations 3.1 and 3.2.

RMS voltage of the control signal in Configuration 1.4 is not the highest among the four single-actuator configurations (see Table 1), which also summarizes the evaluation metrics, including resonance behavior and average system performance across four frequency sub-ranges for detailed comparison.

At first glance, using additional actuators may appear to demand more power while providing a highly improved performance. However, this is not necessarily true, as the synergy among actuators plays a critical role. Proper spatial placement is essential to enable a synergistic interaction that enhances overall system effectiveness. Fig. 7 compares the MSE of the microphones in Configurations 1.4 and 2.1, where Configuration 1.4 uses a single actuator and Configuration 2.1 uses two. Although the use of two actuators in Configuration 2.1 results in better attenuation of resonances around 180 Hz and 248 Hz, it does not lead to significant improvement across the full frequency range or in the overall average sound power level. Furthermore, this limited gain comes with a 45% increase in the average RMS voltage drawn by the actuators compared to Configuration 1.4, as in Table 1.

The performance of the systems with two actuators can be improved by properly selecting the position of the actuator, as the optimized arrangement shown in Configuration 2.2. In Fig. 8, Configuration 2.2, using ASAC, demonstrates superior performance in reducing sound power across the three microphones compared to Configuration 2.1. Resonance attenuation in the 140–215 Hz range is nearly doubled, and the average reduction across the 75–300 Hz range increases by approximately 50%. This improvement is attributed to the more effective placement of the two actuators in Configuration 2.2.

Both actuators are positioned away from regions of high mechanical stress, and one is located near the PVC patch, where reduced stress concentration allows for greater actuation flexibility. However,

all configurations discussed thus far (Configurations 1.1 to 2.2) tend to converge to similar sound power levels within the 200–240 Hz range. To address this limitation, systems with three actuators are investigated. With the actuator arrangement used in Configuration 3.1 (Fig. 3(g)), the system achieved an approximate 5 dB improvement in performance across the 215–275 Hz range, effectively overcoming the limitation previously observed, as in Fig. 9. This enhancement was accomplished with only a 2.5% increase in the average RMS voltage of the control signal. However, the third actuator, positioned in a region of higher mechanical stress, required more actuation power than the second actuator, as shown in Fig. 10.

Configuration 3.1 failed to match the performance of configurations with fewer actuators in the lower half of the considered frequency range. This limitation is primarily attributed to the suboptimal placement of the first and second actuators, which are both concentrated in the right half of the plate near regions of high mechanical stress around the clamped edges. In contrast, Configuration 3.2, which features a more balanced spatial distribution of the actuators, achieves improved performance across the entire frequency range (see Fig. 11). This arrangement provided the system with greater flexibility to introduce actuation power. As a result, the configuration led to doubling the average RMS voltage of the control signals.

Consequently, it achieved a doubling in average noise reduction, exceeding 12 dB, and an additional reduction of more than 8 dB in the system's resonant peaks. This enhanced performance is also clearly reflected in the spectrogram of sound power across the three error microphones, with particularly notable improvements observed on microphones 1 and 3, as shown in Fig. 12. Although microphones 1 and 3 show higher local attenuation due to their spatial proximity to actuators 1 and 3 in Configuration 3.2, global noise reduction is still achieved. This outcome is consistent with the spatial control criteria described in Section 3, where the microphone arrangement was designed to satisfy established spacing constraints. The results highlight that placing inertial actuators in regions of lower mechanical stress is critical, as improper placement can significantly degrade system functionality and lead to inefficient use of available resources. Moreover, increasing the number of actuators does not necessarily result in better noise reduction; in some cases, a single actuator can deliver acceptable performance compared to configurations with two or three times as many, when the objective is to minimize specific resonances in the system's acoustic response.

5. Conclusion

This study presented a novel approach to active noise control in lightweight, transparent-patched vibrating plates through the synergistic use of inertial actuators. By integrating transparent patches into the plate structure, the system maintains optical accessibility while enabling effective implementation of Active Structural Acoustic Control (ASAC). An analytical optimization framework was developed to strategically place actuators, explicitly considering the acoustic transmission loss of the transparent patch. Experimental investigations under

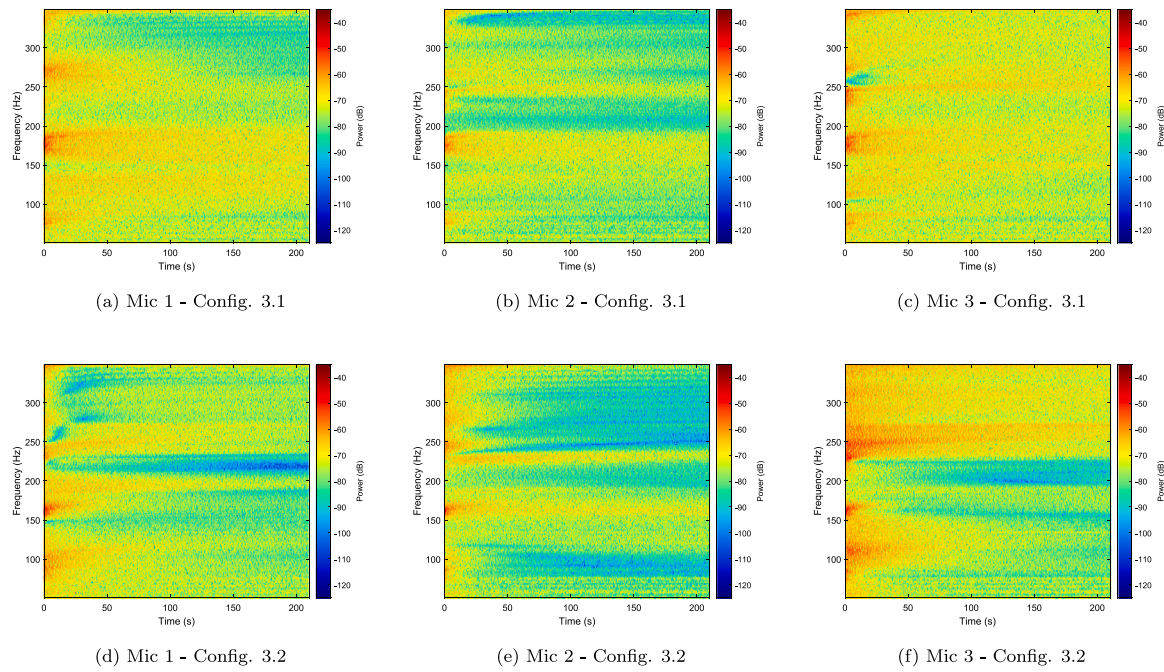


Fig. 12. Microphone signal power over time for configurations 3.1 and 3.2.

Table 1

Summary of the evaluation metrics for all tested configurations.

Config	Reduction (dB)	Frequency range (Hz)				Average RMS (V)
		75–140	140–215	215–275	75–300	
1.1	Resonance	5.9	8.3	2.8	6.3	0.0077
	Average	2.9	−1.7	4.5	1.9	
1.2	Resonance	5.5	9.1	−0.2	−0.2	0.0080
	Average	1.2	−1.6	5.3	1.8	
1.3	Resonance	5.5	12.9	−0.3	8.2	0.0091
	Average	1.5	0.12	4.2	1.5	
1.4	Resonance	5.1	9.5	5.9	9.5	0.0081
	Average	0.6	−0.2	7.8	2.4	
2.1	Resonance	10.4	11.5	10.9	10.9	0.0118
	Average	5.3	1.1	9.1	4.5	
2.2	Resonance	14.9	21.2	5.8	13.7	0.0118
	Average	8.5	5.3	5.9	6.2	
3.1	Resonance	14.4	16.4	14	16.4	0.0121
	Average	6.5	3.6	11.8	6.5	
3.2	Resonance	11.9	26	23.2	24	0.0251
	Average	12.6	12.4	16.4	12.6	

broadband noise excitation demonstrated that actuator performance is highly dependent on spatial placement, particularly in relation to mechanical stress zones and proximity to the noise source. While configurations with additional actuators offered the potential for improved performance, the results showed that poorly positioned actuators could lead to inefficiencies, increased power consumption, and limited noise attenuation. Additionally, optimized placements enabled synergistic interactions between actuators, achieving sound power reductions of more than 12 dB. Future work can investigate the relationship between plate size, transparent patch dimensions, and the number of required actuators. As plate dimensions increase, maintaining effective acoustic performance may necessitate a higher number of actuators, along with careful optimization of their spatial distribution to ensure efficient control. Overall, the findings address a critical gap in the literature concerning the limited application of ASAC to transparent structural systems. By demonstrating that effective noise control can be achieved

without compromising optical transparency, this work highlights the viability of ASAC in domains where visual access and lightweight design are essential.

CRediT authorship contribution statement

Khaled Said Ahmed Maamoun: Writing – review & editing, Writing – original draft, Visualization, Validation, Software, Resources, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Alkafh Aboutiman:** Writing – review & editing, Writing – original draft, Validation, Resources, Formal analysis, Data curation, Conceptualization. **Ali Mohamed Ali:** Writing – review & editing, Writing – original draft, Validation, Formal analysis, Data curation.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgments

The work was supported by the project IN-NOVA: Active reduction of noise transmitted into and from enclosures through encapsulated structures, which has received funding from the European Union's Horizon Europe program under the Marie Skłodowska-Curie grant agreement no. 101073037.

Data availability

Data will be made available on request.

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