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Active noise and vibration control systems for passenger transport vehicles: an overview of concepts and applications

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ABSTRACT

The new generation of passenger vehicles, especially cars and aircraft, require innovative solutions for reducing interior noise that can meet new design criteria. In particular, given the increasing cost of fuel caused by declining oil supplies and the ever-increasing demand to reduce pollutant gas emissions into the air, there is a great interest in lightweight, fuel-efficient vehicles. Active noise and vibration control techniques offer interesting solutions that can be effectively combined with lightweight passive treatments to control noise at low and medium audio frequencies. The introductory part of this article gives an overview of the principles of active control using feedforward and feedback systems for variously controlling tonal and stochastic noise. The core of the paper then provides a review of key concepts and applications of active control systems to the reduction of noise and vibration in passenger transport applications. Finally, an extensive summary of active noise and vibration control in passenger transport is presented with an outlook on future directions of research and development.

1. Introduction

Over the last four decades, research into noise and vibration control has been given a strong impetus by a distinct methodology known as ‘active noise and vibration control’. The basic concepts of active acoustic control were introduced by Lueg in 1936 [1], who submitted a patent for feedforward active control of tonal acoustic waves propagating in both one-dimensional wave guides and three-dimensional free field environments. However, the first experimental studies with simple analogue control circuits were not presented until about two decades later by Olson and May [2] and Conover [3]. It then took another two decades or so for research and development to begin on the first feasible applications, which utilised more complex control systems based on the digital signal processing technology emerging at that time [4–6].

During the 1980s, several researchers began investigating the feasibility of Active Noise Control (ANC) technology to reduce the strong tonal noise in aircraft cabins produced by propeller blades [7–11] and to reduce the tonal noise inside cars generated by internal combustion engines [12,13]. Later on, experimental demonstrations were carried out on full-scale vehicles and the first trials were conducted to test the effectiveness of this technology in propeller-driven aircraft [14–17], helicopters [18], passenger cars [19,20] and more recently on maritime vessels [21,22]. Typically, these types of control system utilise multichannel controllers that drive loudspeakers positioned behind the interior panels of the cabin (especially the side panels and those below the luggage compartments of the

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aircraft cabin or the car door panels) in order to reduce noise in the vicinity of the passengers' heads. This type of control is typically realised with an adaptive feedforward algorithm that relies on a reference signal, which is normally measured using a tachometer positioned on the rotor of one of the propellers of the aircraft or on the crankshaft of the car, and on error signals provided by microphones positioned on the headrests of the passengers' seats [4,5]. More recently developed aircraft cabin noise control systems utilise inertial electrodynamic actuators [23] or actively tuned resonators [24] that are mounted on the fuselage structure so that the acoustic control field is generated by radiation from the fuselage panels themselves rather than via loudspeakers; this provides a greater level of integration into the trim panels.

The first experiments on propeller-driven aircraft gave very encouraging results and so, a few years after the initial tests, the first commercial applications were realised and developed [23,25,26]. To date, more than a thousand cabin noise control systems have been installed in propeller-driven aircraft by various manufacturers. For example, this type of system is installed in all Bombardier Q400 DASH-8 passenger airplanes [27]. In contrast, in the automotive sector active control technology took significantly longer to reach widespread commercial use. This slower uptake was a result of both the cost of the required hardware components (microphones, loudspeakers, controller) and the cost of integrating the active system components (electrical connections, control software) into the vehicle architecture; these additional costs lead to a significant reduction in the already low profit margins typical in the automotive sector in comparison to the aerospace sector. For example, the first automotive production active engine noise control system was implemented in 1992 by Nissan, but employed loudspeakers, amplifiers and digital signal processors separate from the car audio system and was expensive due to this lack of integration [28]. More recently, however, it has become possible to integrate the active control system into the digital audio and telecommunications systems commonly available in cars and thus considerably reduce both component costs and installation costs. In addition, fuel efficiency requirements, related to the reduced future availability of oil and the problems due to CO₂ emissions, have forced the automotive industry to produce increasingly lightweight cars with low-displacement engines that present major noise problems. Active noise control systems make it possible to partially solve these problems (mainly engine noise and partly low-frequency tyre rolling noise) without significantly affecting the weight of the car and these factors have led to wider adoption of active sound control technologies in the automotive sector [29]. Finally, due to its adaptive nature, a given control system can be installed on several car models and offer a considerable reduction in vehicle design and development costs.

The success achieved with active noise control has driven research into new solutions based on active control. For example, Active Vibration Control (AVC) systems have been developed to mitigate vibration transmission through the structural elements connecting the rotor to the helicopter cabin structure [30,31], and adaptive dynamic absorbers have been used to control the vibro-acoustic noise generated by jet engines mounted on the rear of the fuselage of McDonnell Douglas' DC-9 and MD-80 aircraft [32]. In parallel, active mounting systems have also been developed for car engines, controlled by a feedforward controller that minimises noise inside the car [33,34] and active suspension systems have also been developed for application in rail vehicles [35–37]. In the maritime sector, AVC technologies have been developed and applied to control various problems [38], including propeller shaft vibration [39], cruise ship funnel vibration [40], machinery rafts [41] and hull radiated noise [42].

Since the first half of the 1990s, there has been a great deal of research activity to develop so-called 'smart structures' for Active Structural Acoustic Control (ASAC) [43–46], with a particular focus on aeronautical applications [47–54] although there have also been some studies with application focused on cars [55–57], rail [35] and maritime vessels [42]. In the aeronautical application, these systems consist of sensors and actuators directly connected to the aircraft structure, which provides a light and compact solution [58–62] that is able to reduce the transmission of vibro-acoustic noise into the cabin. The interest into smart structures has also been motivated by the fact that, by using pairs of collocated sensors and actuators [61,62], feedback control systems can be implemented that allow the control of stationary stochastic (random) disturbances [63–65] such as the noise transmitted to the cabin of the aircraft by turbulent air flow over the fuselage surface or by jet engine noise. The first practical large-scale smart structures application has been developed for the Airbus A400M transport vehicle [66]. This system used a large number of inertial electrodynamic actuators mounted directly on the fuselage of the aircraft in order to reduce the tonal noise generated by the four engines attached to the aircraft wings, noise which is effectively transmitted into the aircraft via the thin aluminium panels (approximately 1.3–1.5 mm) of which the fuselage is composed.

This article¹ introduces the main aspects that characterise active noise and vibration control systems utilising feedforward and feedback architectures. In addition, it presents a broad overview of the most successful applications that have been studied and developed to date in the passenger transportation fields, including aeronautical, automotive and maritime applications. The remainder of the article is organised into four sections. Section 2 introduces the basic concepts of feedforward and feedback control architectures. Sections 3 and 4 provide a review of the most notable systems that have been researched and developed for active noise control and active vibration control respectively. The final section provides conclusions and an outlook on the future potentials of active noise and vibration control technologies within transportation applications. The article provides a rather vast bibliography, however, given the large number of works carried out on ANC and AVC systems for application to passenger transportation, some references have inevitably been overlooked.

¹ Part of this article takes its cue from reference [67], however, the scope has been significantly broadened and extended.

2. Control architectures

In general, active noise and vibration control system architectures can be divided into two main categories: feedforward and feedback. Both approaches can be implemented with a single sensor-actuator pair and a single-channel controller, or with multiple sensors and actuators and a multichannel controller. The basic principles of these two approaches are briefly summarised in the following two subsections to help contextualise their application in passenger transportation vehicles, rather than to provide an exhaustive review of control algorithms. The interested reader is referred to key texts on this topic and the contained extensive list of references [4–6].

2.1. Feedforward control architectures

Feedforward control systems utilise a reference signal(s), correlated with the unwanted disturbance signal, to drive a control filter that is either designed offline or adapted online to minimise the error signal(s) measured by error sensors. A block diagram showing the general architecture of a feedforward control system in the frequency domain is shown in Fig. 1. In this case, a multichannel control system is assumed with multiple error signals, $\mathbf{e}(\omega)$, disturbance signals, $\mathbf{d}(\omega)$, control signals, $\mathbf{s}(\omega)$, and signals generated by the control sources at the error sensors, $\mathbf{d}_c(\omega)$. As a result, the plant response, $\mathbf{G}_{es}(\omega)$, is a matrix of responses between the control sources and error sensors and the controller, $\mathbf{h}(\omega)$, is also multichannel, although the block diagram assumes a single reference signal and so the controller has a single-input, multiple-output architecture.

In general, feedforward control systems are suited to problems where reference signals are available that are both correlated and time-advanced with respect to the error signals. Accordingly, feedforward systems are well suited to the control of deterministic signals, such as those generated by harmonic or periodic disturbances, since a single reference signal can usually be measured relatively straightforwardly. For example, considering the problem of controlling the tonal noise produced by aircraft propellers, a car engine or reciprocating machinery in general, a suitable reference signal can be measured using some form of tachometer. As a result, feedforward control will be introduced considering a tonal control problem. As shown schematically in Fig. 1, the reference signal, $r(\omega)$, is passed through a bank of filters $h_1, h_2, \dots, h_m$ which generate the control signals $s_1, s_2, \dots, s_m$. These signals drive the control sources (actuators or loudspeakers) to achieve the desired control. Theoretically, in the noise control case, the control filters should be chosen or adapted to minimise the whole acoustic field in an enclosed space (given for example by the total acoustic potential energy of the enclosure), however, in practice the control sources are typically driven to minimise the sum of the squares of the mean values of the error signals $e_1, e_2, \dots, e_n$ measured by error microphones. Similarly, in the context of a practical active vibration control system, rather than minimising the whole vibration field of the structure (given for example by the total flexural kinetic energy of the structure), the control sources are driven to minimise the sum of the squares of the mean values of the error signals measured typically by accelerometers, for example.

If the primary disturbance is initially assumed to be harmonic with frequency ω_0 , then, as shown schematically in the block diagram in Fig. 1, the vector with the n complex phasors of the signals measured by the error sensors $\mathbf{e}(\omega_0)$ can be expressed as the sum of the vector $\mathbf{d}(\omega_0)$ with the signals $d_1, d_2, \dots, d_n$ generated by the primary disturbance at the error sensors and the vector $\mathbf{d}_c(\omega_0)$ with the signals

$d_{c1}, d_{c2}, \dots, d_{cn}$ generated by the control sources also at the error sensors. The vector $\mathbf{d}_c(\omega_0)$ can be derived from the product of the matrix $\mathbf{G}_{es}(\omega_0)$, with dimensions $n \times m$ whose elements are formed by the frequency responses between each error sensor and each control source, and the vector with the signals driving the control sources $\mathbf{s}(\omega_0)$, i.e. $\mathbf{d}_c(\omega_0) = \mathbf{G}_{es}(\omega_0)\mathbf{s}(\omega_0)$. Thus, the vector of error signals can be expressed as

$$\mathbf{e}(\omega_0) = \mathbf{d}(\omega_0) + \mathbf{d}_c(\omega_0) = \mathbf{d}(\omega_0) + \mathbf{G}_{es}(\omega_0)\mathbf{s}(\omega_0). \quad (1)$$

The signals $s_1, s_2, \dots, s_m$ that drive the m control loudspeakers are generated by passing the reference signal related to the primary disturbance $r(\omega_0)$ through a set of m control filters $h_1, h_2, \dots, h_m$; therefore, the vector $\mathbf{s}(\omega_0)$ can be expressed as

$$\mathbf{s}(\omega_0) = \mathbf{h}(\omega_0)r(\omega_0), \quad (2)$$

where $\mathbf{h}(\omega_0)$ is the vector containing the m control filters. Thus, by substituting Eq. (2) into Eq. (1), the following cost function must be minimised to reduce the sum of the mean square values of the error signals

$$J = \sum_{r=1}^n |e_r|^2 = \mathbf{e}^H \mathbf{e} = \mathbf{r}^H \mathbf{h}^H \mathbf{G}_{es}^H \mathbf{G}_{es} \mathbf{h} \mathbf{r} + \mathbf{r}^H \mathbf{h}^H \mathbf{G}_{es}^H \mathbf{d} + \mathbf{d}^H \mathbf{G}_{es} \mathbf{h} \mathbf{r} + \mathbf{d}^H \mathbf{d}. \quad (3)$$

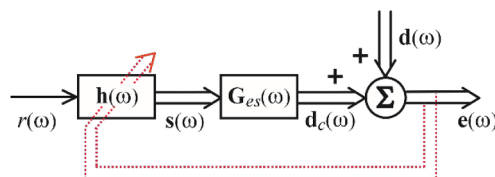


Fig. 1. Block diagram of a multichannel active control system with a feedforward control architecture.

This is a quadratic Hermitian function whose minimum is given by the following vectors with the optimal filters

$$(a) \mathbf{h}_{opt} = -[\mathbf{G}_{es}^H \mathbf{G}_{es}]^{-1} \mathbf{G}_{es}^H \mathbf{d} \quad (b) \mathbf{h}_{opt} = -\mathbf{G}_{es}^{-1} \mathbf{d} \quad (c) \mathbf{h}_{opt} = -\mathbf{G}_{es}^H [\mathbf{G}_{es}^H \mathbf{G}_{es}]^{-1} \mathbf{G}_{es}^H \mathbf{d} \quad (4)$$

which hold respectively for (a) the over-determined case with more sensors than control sources ($n > m$); (b) the fully-determined case with equal numbers of sensors and control sources ($n = m$) and (c) the under-determined case with fewer sensors than control sources ($n < m$) [4]. Based on these equations, the optimal control filters $h_1, h_2, \dots, h_m$ can be derived by measuring the vector of primary disturbance spectra at the error sensors, $\mathbf{d}$, and the matrix, $\mathbf{G}_{es}$, containing the frequency responses between the inputs to the m control sources and the outputs from the n error sensors. In practice, measurements of the primary disturbance spectra at the error sensors and measurements of the frequency responses between the error sensors and the control sources will contain uncertainties and may also change over time, which means that calculation of the control filters using Eqs (4) will in practice not actually result in the optimal performance. Therefore, an iterative approach is normally implemented which, as shown schematically in Fig. 1, can follow non-stationary effects present in the primary disturbance and adapt the control filters [5]. This is most commonly achieved using the Filtered-Reference Least Mean Squares (FxLMS) algorithm [5].

For the sake of completeness, it should be noted that control systems with a feedforward architecture can also be used to control disturbances with stochastic (random) characteristics in either temporal or spatial domains. If the stochastic disturbance is generated by a single identifiable source then it may be possible to realise an effective feedforward controller with a single reference signal, but in many practical applications disturbances are spatially stochastic, or equivalently may be generated by multiple partially, or uncorrelated sources and then it is necessary to use a large number of reference signals, $r_1, r_2, \dots, r_v$, and a multi-input multi-output control filter array, $\mathbf{H}_{opt}$ [4]. In this case, the feedforward controller must be causally constrained and so the reference signals must be time-advanced compared to the error signals. The potential performance of the controller will also be fundamentally limited by the correlation between the reference and error signals [5], this is before consideration of the physical limitations due to error sensor and control source positioning and the acoustics or dynamics of the system under control. In addition to these considerations, the computational requirements of a stochastic feedforward control system are significantly higher than a deterministic system, not only because of the need to include multiple reference signals, but also because the length of the required control filters significantly increases from the deterministic case, where a single channel system controlling a single tone requires only two coefficients. Nevertheless, multiple applications of feedforward control to stochastic problems have been explored and the transportation relevant applications will be described in the following sections.

2.2. Feedback control architectures

Feedback control systems do not utilise a reference signal, but instead the controller is directly driven by the error signal(s), as shown in Fig. 2. Feedback control systems for disturbance rejection are mainly used to control stochastic or random disturbances, particularly where it is not possible to obtain a time-advanced and coherent reference signal. For example, feedback control systems may be applied to control the noise generated by jet aircraft engines, air turbulence exciting aircraft fuselage or car body panels and tyre-road noise, which will be variously discussed in the following sections. This section will review the basic principles and architecture of a feedback active control system for reduction of noise or vibration, but more detailed exposition on feedback control systems in this context is provided in various texts [5,6,58,59,68–71].

As shown for the multi-input multi-output feedback control system presented in Fig. 2, the $s_1, s_2, \dots, s_m$ signals that drive the control sources are obtained by feeding back the $e_1, e_2, \dots, e_n$ signals measured by the error sensors via a group of control functions $h_{11}, h_{12}, \dots, h_{mn}$. As for feedforward control, the vector with the n complex amplitudes of signals measured by the error sensors $\mathbf{e}(\omega)$ can be expressed as the sum of the vector $\mathbf{d}(\omega)$ with the signals generated by the primary disturbance at the error sensors and the vector $\mathbf{d}_c(\omega)$ with the signals generated by the control sources also at the error sensors:

$$\mathbf{e}(\omega) = \mathbf{d}(\omega) + \mathbf{d}_c(\omega) = \mathbf{d}(\omega) + \mathbf{G}_{es}(\omega) \mathbf{s}(\omega). \quad (7)$$

In this case, the vector of control signals is given by

$$\mathbf{s}(\omega) = -\mathbf{H}(\omega) \mathbf{e}(\omega), \quad (8)$$

where $\mathbf{H}(\omega)$ is a sparse matrix with control functions. Thus, by substituting Eq. (8) into Eq. (7), the vector of error signals with closed

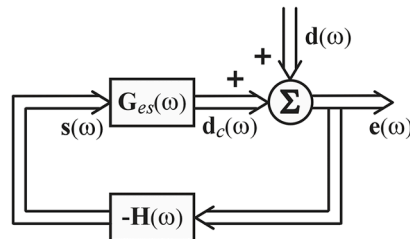


Fig. 2. Block diagram of a multichannel Active Noise Control system with a feedback control architecture for stochastic noise reduction.

feedback loops is given by

$$\mathbf{e}(\omega) = [\mathbf{I} + \mathbf{G}_{es}(\omega)\mathbf{H}(\omega)]^{-1}\mathbf{d}(\omega), \quad (9)$$

where the matrix $[\mathbf{I} + \mathbf{G}_{es}(\omega)\mathbf{H}(\omega)]$ is called the 'return matrix' [5]. The design of the matrix with the control functions is rather complex and requires a model of the frequency responses between the output signals of the n error sensors and the input signals to the m control sources contained in the matrix $\mathbf{G}_{es}(\omega)$ [5,69–71]. In designing the controller, it is also necessary to not only consider the level of disturbance signal reduction, but also the stability of the controller. Further complicating the design of feedback controllers in practice is the need to consider the robust stability of the system, that is, the stability of the controller when it is faced with the real-world practical variations in the controlled system that lead to response uncertainties. Since there are many entire textbooks on the subject of feedback control design, the intricacies will not be included here and the interested reader is referred to classic texts on the subject, which include [5,6,58,59,68–71].

To avoid the complexities of designing multichannel feedback controllers, simplified control systems are often implemented. This includes the use of decentralised systems that have an equal number of sensors and actuators arranged in pairs so that independent single-channel feedback loops can be closed with generally good stability conditions. The control function for each of these loops can then be designed following classical methods for the design of feedback loops, which are normally based on the analysis and synthesis of the frequency response between the sensor and the open-loop actuator [5,60–62,68]. Specific applications of feedback control architectures will be discussed variously in the following sections in relation to the active control of noise (Section 3) and active control of vibration (Section 4).

3. Active noise control

Passenger transportation vehicles are all affected by noise, whether this is generated by the power train, or the motion of the vehicle, either over a surface (road or rail) or through a medium (air or water). In passenger transport applications, excess noise mainly leads to issues of passenger comfort, however, in extreme situations can also pose a threat to hearing. Passive noise control measures are clearly widely applied and generally lower cost than active solutions, however, they are typically limited in performance at low frequencies due to inherent size and weight constraints, which are particularly relevant in transport applications. As a result, active noise control systems have been widely explored and implemented across various modes of transport.

3.1. ANC system designs and limitations

3.1.1. Global noise control

In general, passenger transport vehicles are characterised by an enclosed cabin space. In an ideal scenario, any noise control treatment would limit unwanted noise throughout the enclosure, however, the frequency range over which global active noise control can be achieved is limited by the complexity of the enclosed acoustic field. To provide insight into this limitation, the performance of an ideal controller on the acoustic field generated in a rectangular enclosure by an ideal acoustic monopole source positioned near the bottom right-hand corner of the enclosure is analysed. The assumed enclosure has dimensions $l_x \times l_y \times l_z = 6 \times 2.5 \times 1.8 \text{ m}^3$ and modal damping $\zeta = 0.03$. It is initially assumed that the controller consists of ideal sensors and actuators that can measure and excite exactly the following groups of natural acoustic modes of the cavity: (1), (1 – 2), (1 – 3), (1 – 4), ..., (1 – 20). Fig. 3 shows the resulting control performance in terms of the total acoustic potential energy within the enclosure. When only the first mode is cancelled, (mode type (0, 0, 0) purely volumetric mode of the air in the cavity), the total acoustic potential energy of the cavity is significantly reduced at frequencies below the first resonance due to the second natural acoustic mode type (1, 0, 0). When the ideal control system cancels both the first and second natural acoustic modes of the cavity, the first acoustic resonance due to the second natural acoustic mode of type (1, 0, 0) does not appear on the potential energy spectrum of the cavity. Similarly, when the first R acoustic modes are cancelled with the ideal control system, the first $R - 1$ resonances disappear from the potential energy spectrum of

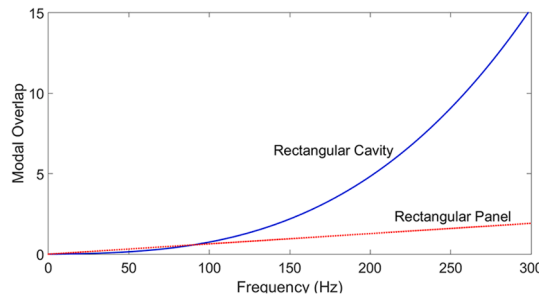


Fig. 3. Acoustic potential energy of a cavity excited by an acoustic monopole. Solid blue line, no control. Faint green lines, cavity controlled by 1, 2, 3, 4, 5, 10, 20 ideal modal sensor-actuators that progressively cancel the first 1, 2, 3, ... 20 modes of the cavity. Dotted red line, cavity controlled by a system with 20 acoustic monopoles driven in such a way as to minimise the mean square value of the error signals measured by 20 acoustic pressure sensors.

the cavity. The graph in Fig. 3 shows that, for example, with an ideal 5-channel controller, the acoustic response of the cavity is reduced for frequencies up to 3 times the first resonance ω_1 , which for the cavity under consideration has a frequency of approximately 28.6 Hz. Instead, a control system with 20 channels is required to control the acoustic field in the cavity up to a frequency of approximately 5 times the frequency of the first resonance.

This relationship between the required number of channels and the upper frequency of control can be related to the number of significantly excited modes. For example, as shown in Fig. 4, for a rectangular cavity the number of significantly excited modes at each excitation frequency, i.e. the so-called 'modal overlap factor', increases rapidly with frequency. In fact, for rectangular cavities, the modal overlap factor is given by [60,72]

$$M_c(\omega) = 2\zeta\omega n_c(\omega) = 2\zeta\omega \left(\frac{V\omega^2}{2\pi^2c^3} + \frac{A\omega}{8\pi c^2} + \frac{P}{16\pi c} \right), \quad (10)$$

where $n_c(\omega)$ is the so-called 'modal density', and c , ζ , V , A , P are respectively the speed of sound, the damping ratio, the volume, the lateral surface area of the cavity and the length of the perimeter formed by the cavity edges. This expression indicates that, at high frequencies, the number of significantly excited modes at each frequency increases with the cube of the excitation frequency. The modal overlap factor can be effectively used to estimate the complexity of the modal response of the controlled system. Thus, it can be used to estimate the number of control sources required to produce the desired control action in a given frequency band, or to calculate the control frequency band for a given number of control sources [5]. If, for example, one wished to control the vibration of an aluminium panel of 1 mm thickness and size $l_x \times l_y = 1 \times 0.5 \text{ m}^2$, the number of control sources required to produce the control action in a given frequency band would be much smaller than that found for the rectangular cavity. In fact, the modal overlap factor for thin plates is given by [60,72]

$$M_p(\omega) = \eta\omega n_p(\omega) = \eta\omega \frac{S}{4\pi} \left(\frac{\rho h}{D} \right)^{1/2}, \quad (11)$$

where $n_p(\omega)$ is the modal density of the plate, η is the structural damping factor, $D = \frac{Eh^3}{12(1-\nu^2)}$, S and h are the flexural stiffness, surface area and thickness of the plate and ρ , E and ν are the density, Young's modulus and Poisson's ratio of the plate material. This expression indicates that, as can be seen in the graph in Fig. 4, the modal overlap factor for rectangular plates increases linearly with frequency. Thus, when control is desired up to medium-high audio frequencies, it may be convenient to control the vibration of the cavity walls, and thus the transmission of sound into the cavity, rather than the acoustic field in the cavity itself. This should be particularly applicable to situations where external noise is mainly transmitted from some of the cavity side walls and will be discussed in Section 3.1.3.

In practice, active noise control systems normally use loudspeakers as control actuators and microphones as error sensors. The cost function to be minimised is then estimated by a number of microphones measuring the acoustic field at a finite number of positions. In addition, the acoustic control field is generated by a number of loudspeakers that can only partially reproduce the optimal acoustic control field that minimises the cost function estimated using the error microphones. In this case, the optimal filters are derived from Equations (4a-c) of the formulation presented above. If, for example, a practical system is studied that has twenty control loudspeakers uniformly distributed throughout the enclosure (for simplicity modelled as monopoles) and twenty error microphones (modelled as ideal pressure sensors) positioned in the vicinity of the control loudspeakers, then, as shown by the red dotted line in Fig. 3, the desired reduction in the cost function, i.e. the acoustic potential energy of the cavity, is only achieved at low frequencies up to 2.5 times the first resonance of the cavity ω_1 . At higher frequencies, the approximations introduced by the error microphones to estimate the optimal cost function and by the loudspeakers to generate the optimal acoustic control field can even lead to an increase, rather than a reduction, in the noise level in the enclosure. The result presented in Fig. 3 was derived by assuming that the control system microphones and loudspeakers are evenly distributed throughout the cavity volume. Although this may seem like a good distribution, in practice, particularly when more complicated realistic vehicle cabins are considered, the position of sensors and control actuators can be determined by optimisation routines, such as the genetic algorithm, that identify the best configuration of transducers with

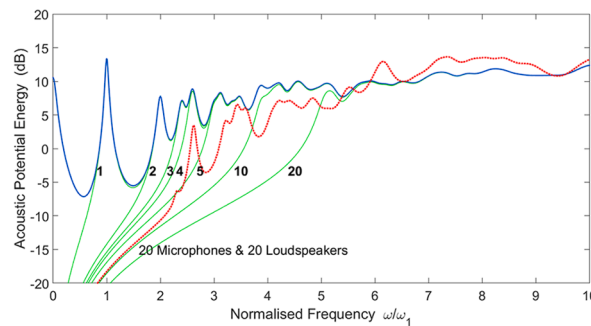


Fig. 4. Modal overlap factor for a cavity with dimensions $6 \times 2.5 \times 1.8 \text{ m}^3$ (coarse-blue line) and for an aluminium panel with thickness 1 mm and dimensions $1 \times 0.5 \text{ m}^2$ (dotted-red line).

reference to the cost function [5] although the physical limitations on control bandwidth remain.

3.1.2. Local noise control

The bandwidth over which unwanted noise can be controlled globally throughout an enclosure is clearly limited by the physics of the problem and the practical number of control sources and error sensors that can be installed. Although the noise transmitted into an enclosure may be more readily controlled via the structure of the enclosure due to the lower modal overlap, this may not always be possible due to installation constraints and requirement for structural control actuation over a large number of vibrating structures. An alternative approach to achieving broadband active noise control for passengers is to utilise a so-called local active noise control system.

Local active noise control systems typically consist of loudspeakers and error microphones located in close proximity to the head and ears of the user, as shown in Fig. 5. Headrest local active noise control systems have been implemented using both feedback [73, 74] and feedforward [75] control architectures, primarily depending on whether or not time-advanced reference signals are available in the considered application. In any case, the physical principles of local control are consistent and the loudspeakers are utilised to create a 'zone of silence' centred on the person's ears, which, for a noise reduction of 10 dB in a diffuse sound field, has been shown to have a diameter proportional to $\lambda/10$, where λ is the wavelength [76]. Fig. 6 shows the size and shape of the zone of quiet at two normalised frequencies when a single monopole acoustic source is used to cancel the pressure at a point in a diffuse sound field, as investigated variously [75–79]. Thus, for noise at 500 Hz, the characteristic size of the zone of silence is approximately 7 cm. Although the control bandwidth can therefore exceed that achieved by global active noise control systems, the size of the zone of quiet inherently limits the bandwidth of local active control. Furthermore, since the location of the zones of silence are centred around the error microphones, unless the error microphones are located inside the user's ears, which is generally impractical, the zones of silence at higher frequencies will not encompass the ears. This issue has been partially solved via virtual sensing techniques, which estimate the pressure at a location that is remote from the physical error microphone [80]. If the remote location is designed to coincide with the person's ears, then the active headrest can be driven to reduce the acoustic pressure experienced by the user and locate the zones of silence around their ears [75]. However, when the position and orientation of the person's head change, then their ears will again not coincide with the zone of silence generated around the virtual microphone location. As a result, methods of tracking the head location and updating both the virtual sensing algorithm and the controller have been developed, which allow a further increase in the controllable bandwidth [81]. Recent developments in this area have considered how the accuracy of head-tracking influences the control performance [82] and how interpolation of the plant responses can improve performance [83]. Nevertheless, head-tracked local active noise control comes at relatively significant expense in terms of both the physical hardware and computational requirement, and a number of practical challenges remain outstanding.

3.1.3. Active structural acoustic control

An alternative approach to overcoming the physical limitations of global ANC is to utilise Active Structural Acoustic Control. Instead of controlling the noise in the vehicle cabin directly, ASAC systems focus on controlling noise transmission into the vehicle cabin via the enclosing structure. This approach is well-suited to transportation applications, since the principal contribution to the interior noise often comes from external sources (e.g. engine, propeller, airflow and tyre noise) [84–86]. Moreover, whilst global ANC in a three-dimensional enclosure is limited to low frequencies due to the acoustic modal overlap increasing with the cube of the excitation frequency, in the case of structures the modal overlap generally increases more gradually allowing a greater control bandwidth, as discussed in relation to Fig. 4. For example, in the case of thin bi-dimensional structures, such as the fuselage skin and interior trim panels in aircraft or the firewall and roof panels in cars, the flexural vibration modal overlap increases linearly with excitation frequency, thus offering the potential for a much broader frequency range of control [59]. As a result, over the past three decades or so, there has been a significant level of research and development activity in the field of ASAC systems [5,6,58–60].

Indeed, the idea of reducing the noise transmission to the interior of enclosures by actively controlling the sound radiation/

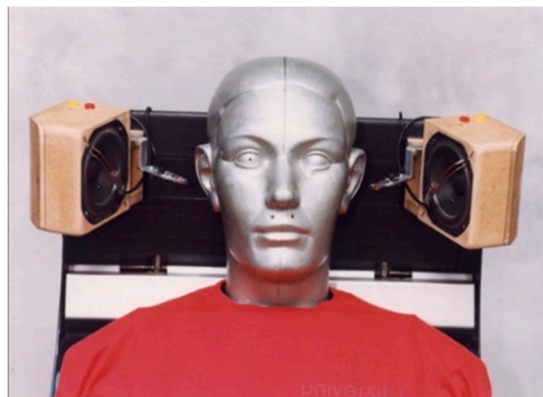


Fig. 5. Local active noise control headrest, showing loudspeakers and multiple error microphone.

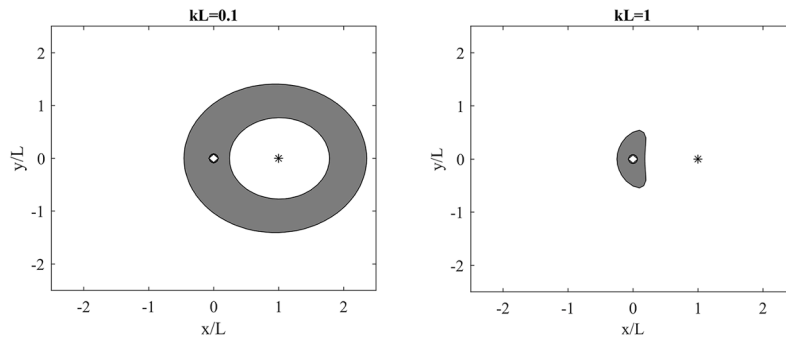


Fig. 6. Cross section of the spatial extent of the zone of quiet within which the average pressure is reduced by 10 dB from simulations of cancellation of the pressure in a diffuse field at a single point, o, using a single monopole secondary source, *, at two different normalized frequencies.

transmission through the walls dates back to the early 1990s [43–46,63,64]. To start with, structural actuators were integrated into the walls of the enclosure and driven by a centralised controller, provided with a reference signal well correlated to the external disturbance, to reduce the interior sound field measured using acoustic sensors scattered throughout the volume. Although the actuators were integrated into the walls, this solution required a large number of error sensors suitably positioned in the enclosure to accurately estimate the “complex” three-dimensional acoustic field generated by the flexural vibrations of the walls, as for the ANC systems discussed in Section 3.1.1. A step forward was made when the control system was set to minimise the sound transmission/radiation by the panels rather than the whole acoustic field in the interior volume. Overall, this approach became known as Active Structural Acoustic Control [44] and, at first, was developed for the control of tonal disturbances, but has now also been developed for broadband control [87] and realised using both feedforward [88–91] and feedback architectures [88,92–95]. In the following sections, various applications of ASAC to passenger transportation vehicles are discussed; however, readers interested in the fundamental working principles of the various configurations and control architectures of smart structures for vibro-acoustic control can refer to the comprehensive works presented in [61,62].

3.2. Active noise control in aircraft

One of the earliest applications of active noise control within the passenger transportation sector was to propeller aircraft. The internal noise control problem associated with propeller-driven aircraft is dominated by relatively low frequency tones, which are directly related to the rotational speed of the propellers. In general, rather bulky and heavy passive control treatments are required to mitigate the high-level tones that arise in the cabin [86]. Instead, active noise control offers a solution to this noise control problem based on a feedforward controller that relies on tachometer-based reference signals. Theoretical work in this area began in the mid-1980s with numerical studies designed to develop a fundamental understanding of the physical control problem [96,97], before moving forward to offline experimental studies using data from full-scale trials [17,98] and in-flight experimental implementations [16,99,100]. Ultimately, this has resulted in various production systems [23,25,101], with the first one fitted as standard to the Saab 2000 in 1994 [23].

Early implementations of propeller aircraft active noise control systems utilised loudspeakers as the control sources driven to minimise the pressures measured by an array of microphones distributed throughout the aircraft cabin at frequencies associated with the blade passing frequency and its harmonics [16]. These systems utilised the standard FxLMS algorithm for tonal control [5] as described in Section 2.1. For the control system reported in [16], which is one of the earliest in-flight demonstrations, 16 loudspeakers were utilised to minimise the pressures due to the first three harmonics of the blade passing frequency, corresponding to 88, 176 and 264 Hz, measured at 32 microphones distributed throughout the cabin. This system was able to achieve reductions at the fundamental blade passing frequency of 13 dB and reductions of about 12 dB at the first two harmonics, which correspond to overall reductions of about 7 dBA. Early experiments were also conducted on helicopters, for example, Boucher *et al.* [18] conducted flight trials on prototype EH101 helicopter, a large, 30-seater aircraft made by Agusta and Westland Helicopters. The control system encompassed 32 error microphones and 16 loudspeaker secondary sources, with attenuation in the mean squared sound pressure level of up to 12 dB at the fundamental tonal excitations.

Controlling noise globally throughout the aircraft cabin clearly requires a large number of control sources and error sensors, since the physical limits associated with the number of acoustic modes excited are approached, as discussed in Section 3.1.1. When there is a significant structure-borne noise component, it may be possible to achieve significant reductions in the internal cabin noise utilising active vibration control and a study to this effect is reported in [17]. The investigated system achieves comparable reductions in the sound pressure levels associated with the blade passing frequency and its harmonics, but only requires two control shakers located near to the aircraft pylon attachment. The performance of the system, however, is not reported for in-flight trials and the disturbance sound field is generated artificially by shakers rather than by the propellers. The performance of this system also relies on the proximity of the control forces to the disturbance forces and, as a result, the control forces required are relatively high. A similar system was reported for helicopter applications in [102], where flight trials were conducted to demonstrate the viability of this approach.

Active noise control systems utilising structural actuation, or active structural acoustic control systems, have been variously researched and developed to production levels. These systems effectively replace the loudspeakers utilised in [16] with structural actuators, with the objective of improving cost efficiency through reductions in the installation cost, channel count and channel power requirements [99]. Grewal et al. [48] described a large-scale experimental implementation of an ASAC system applied to a deHavilland Dash-8 turboprop aircraft, with the propeller-induced noise simulated using an external array of loudspeakers. This feedforward control system used piezoelectric actuators mounted on the frames of the fuselage wall structure to control the tonal airborne propeller noise generated by the loudspeaker array, showing reductions of up to 8.4 dB at the seats of the passengers located in the vicinity of the propellers. Palumbo et al. [99] reported the results of flight trials of a feedforward ASAC system applied to a Raytheon 1900D aircraft to control propeller induced tonal noise. The ASAC system in this case utilised 21 inertial electromagnetic actuators mounted on the fuselage frames to control the pressures measured at 32 error microphones distributed throughout the aircraft cabin. The results of these trials demonstrated reductions of up to 15 dB in the fundamental harmonic generated by the propeller and also highlighted the low electrical power requirements, with each actuator only consuming around 7 W. More recently, Haase et al. [103] presented a study on the implementation of a multichannel feedforward controller with 12 inertial actuators and 24 accelerometer sensors on the fuselage of a Dornier Do728 aircraft in a large-scale grounded trial with the counter-rotating open propeller noise generated by a large external loudspeaker array. The implemented system was able to realise a mean sound power reduction of 3.4 dB in the cabin for the first five harmonics of the counter rotating open rotor propellers.

Beside controlling the sound transmission through the fuselage walls, a few attempts have been made to control the sound transmission through the double wall partition, formed by the external fuselage wall and the internal trim panel furnished with porous liners on the internal side, which are used for both thermal and noise insulation [86]. Lyle and Silcox [104] developed active trim panels whose outer surface was equipped with piezoelectric transducers driven to minimise the sound transmission. Overall, they measured a 5 dB reduction in the noise transmission to the interior of the aircraft. Active trim panels have also been investigated by Tran and Mathur [105] with full scale experiments on a McDonnell Douglas DC-9 aircraft. The feedforward control system explored in this work used 16 piezoelectric patch actuators bonded to the trim panels in the aft section of the fuselage and 32 microphone error sensors scattered throughout the same aft section of the fuselage. This solution, however, was found to be less effective than the system presented in [106] for the same aircraft; possibly because the sound field produced by the active trim panel did not couple effectively with the sound field transmitted to the interior. This may have been the result of the poor coupling between the piezoelectric patch actuators and the “floppy” trim panel, which therefore were not able to generate the destructive sound fields necessary to cancel the primary disturbances. More recently quite an extensive bulk of work has been performed by Misol et al. [103,107–110] on the design and implementation of ASAC systems on aircraft sidewalls for the reduction of noise transmission. This work was based on both laboratory setups encompassing a section of the fuselage wall of the Airbus A350 aircraft [103,107–110], as well as a full-scale fuselage test using a Dornier Do728 aircraft [52] and a Dassault Falcon 2000LX MSN 006 “iSTAR” business jet [111]. As shown in Fig. 7a, these studies were focused on trim panels equipped with low-weight and low-cost electrodynamic inertial actuators and accelerometer sensors [107,108]. Also, the advantage of using virtual microphone sensors such that the radiated sound field was reconstructed from microphones located on the trims themselves was exploited [109,110]. Overall, the idea was to develop modular sections of the fuselage double-wall construction with a local multichannel feedforward control system set to minimise the vibration response of the trim panel using the accelerometer error sensors and the electrodynamic inertial actuators [52]. As shown in Fig. 7b, control performance of between 5 and 10 dB was achieved at the fundamental and higher order harmonics produced by the counter rotating open rotor propeller, although a control spillover effect can also be noticed at two harmonics since the error sensors were not filtered to measure the radiation modes.

Another practical application of ASAC has been investigated at the Helmut-Schmidt-University facilities [66], where a multi-channel feedforward ASAC system was developed and tested for the control of the high level tonal disturbances generated by the four propellers of the Airbus A400M aircraft shown in Fig. 8a, which are effectively transmitted to the cabin interior through the fuselage section centred on the rotational plane of the propellers. As shown in Fig. 8b,c,d, the system employs twin coil-magnet electrodynamic seismic actuators, which can be used in- and out-of-phase to generate transverse force or bending moments on the fuselage framework structure. Microphone error sensors were used to detect the sound transmitted into the cabin and an FxLMS [5] controller was

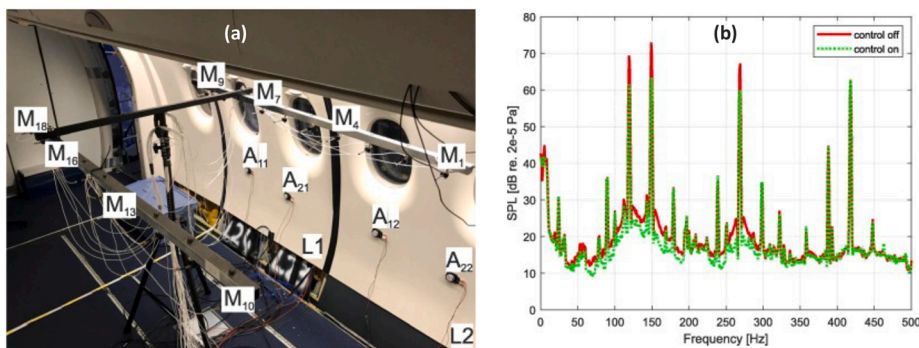


Fig. 7. ASAC trim panel (a) and noise reductions (b) at the tonal excitations generated by a counter rotating open rotor propeller (From Ref. [52]).

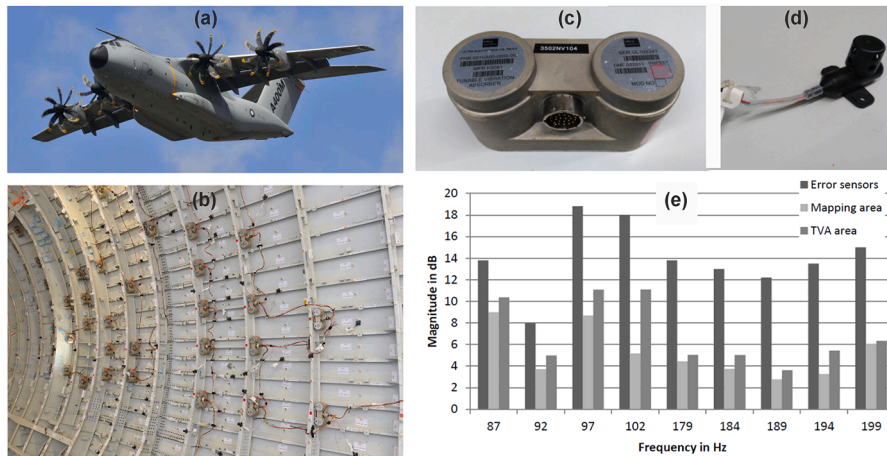


Fig. 8. (a) A400M turboprop (b) Fuselage section with (c) inertial actuators and (d) microphone sensors. (e) Measured noise reductions at the error sensors and mapping areas. (From Ref. [66]).

developed to minimise the sound pressure measured at the error microphones. Fig. 8e shows the noise attenuation produced in the cabin measured at the microphone error sensors of the controller as well as over two mapping areas. The plot confirms the potential of this ASAC system, which generates very large reductions comprised between 8 and 19 dB, at the error sensors and significant reductions comprised between 3 and 9 dB at the monitor positions.

One of the more commercially successful implementations of active control in the aircraft application is the system developed by Ultra Electronics and installed in over 1200 aircraft including the Bombardier Q400 shown in Fig. 9a [27]. Although this system can be related back to the early research reported in [16,96], it utilises inertial actuators to realise the control action [101] and as such may be considered an ASAC system. As shown in Fig. 9b, the system installed in the Bombardier Q400 utilises 42 inertial actuators, 84 sensors (including 80 microphones) and can control a weighted sum of the signals related to eight tones harmonically related to the engine speed [101]. The noise level distribution measured in the cabin of a propeller-driven aircraft at the fundamental blade passing frequency of 88 Hz is shown in Fig. 9d, which confirms how effectively the control system can achieve global control at this low frequency. Also, Fig. 9c shows the spectrum of the sound pressure inside the aircraft cabin.

In addition to the active control systems discussed up until this point, which have largely focused on the control of tonal components produced by propeller engines, preliminary investigations have also been conducted into broadband stochastic control systems. Of practical note is a laboratory-based study into the implementation of a smart trim panel for the control of the transmission of stochastic broadband disturbances through an aircraft fuselage double wall construction using a multichannel feedforward controller [51,112]. As shown in Fig. 10a, reference accelerometers were mounted to the fuselage skin and the control actuators and error accelerometers were mounted to the interior trim panel. As depicted in Fig. 10b, reductions of the order of 2 to 4.5 dB were observed at the lower order

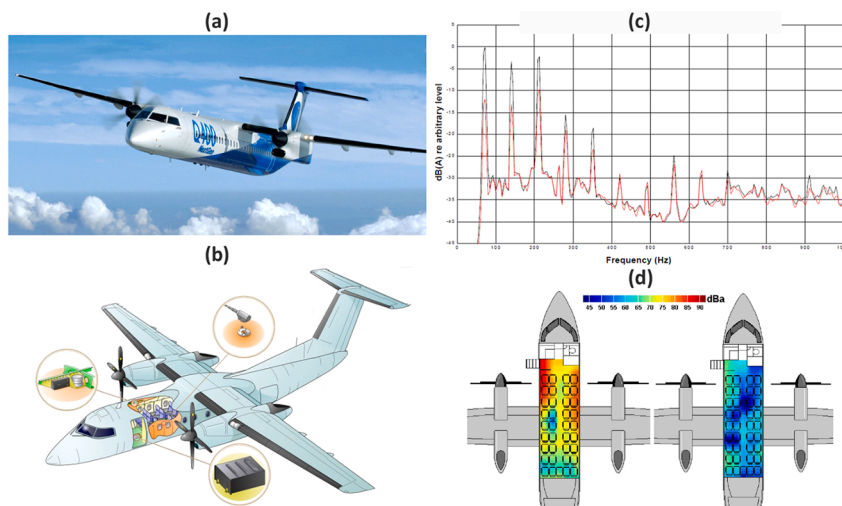


Fig. 9. (a) Bombardier Q400 propeller aircraft (b) Control system components, showing actuators, error sensors and control unit. (c) spectrum of cabin noise without (black line) and with (red line) ANC on (d) sound pressure map without control (left) and with control (right) (Courtesy of Ultra Electronics Ltd).

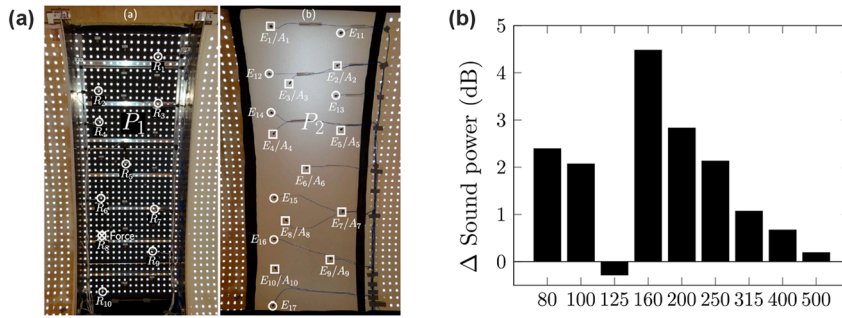


Fig. 10. (a) Smart trim panel for the control of stochastic excitations showing the positions of the error (E) and reference (R) accelerometers and the control actuators (A). (b) Reduction of the sound power level at low audio frequencies (From Ref. [51]).

third-octave bands. The aim of the active system was indeed to control the sound transmission through the double wall construction at low frequencies, where the double wall layout is less effective due to the well-known mass-air-mass resonance effect [60].

When controlling broadband stochastic disturbances, it is often challenging to obtain a time-advanced reference signal and, as a result, feedback control architectures are well-suited to this problem. In the aircraft application, this challenge arises when, for example, attempting to control the noise transmitted into the cabin due to Turbulent Boundary Layer (TBL) excitation of the fuselage. To this end, researchers at the Institute of Sound and Vibration Research investigated the potential performance of decentralised feedback control applied to the trim panel of a Bombardier Dash-8 Q400 fuselage section [113], as shown in Fig. 11. This figure shows the decentralised system with five feedback control units, which use seismic electromagnetic actuators developed by Paulitsch et al. [114] with an accelerometer sensor at the centre of its footprint. The plot in Fig. 12 indicates that the 5 decentralised control units generate reduction in the nearfield sound radiation of between 3 and 10 dB. There is also, however, a control spillover problem at low frequencies, which is due to the out-of-phase electro-mechanical response of the actuator below its fundamental resonance. Nevertheless, as discussed in [115–126] this problem can be variously mitigated. The idea of implementing decentralised instead of centralised vibration control is very appealing for its simplicity and effectiveness, which can lead to the development of self-contained modular control units that can be mass fabricated and mounted on vehicles without the need of specific design and development work (see Refs. [61,62,95,127–131] for overviews on decentralised vs. distributed vibration control). Indeed, recently, quite a lot of research has been variously dedicated to develop effective control units that present interesting solutions to enhance the robust stability of the feedback loop, to overcome undesirable non-linear effects and to guarantee self-tuning features such that the unit can be installed straightforwardly on a variety of vehicles and can be operated on a wide range of disturbances (e.g. see [115–126,132,133]). In this respect, Algermissen and Monner [134] have worked on a self-contained feedback control unit specifically designed for aircraft trim panels vibration control, which encompasses a very small sized inertial actuator equipped with accelerometers at the base footprint and inertial mass and a digital controller that can be conveniently used to implement a frequency-shaped control function that guarantees robust feedback control.

To extend the performance of active noise control systems in aircraft applications to higher frequencies, it is possible to resort to local control systems, as described in Section 3.1. The cost of installing such systems for every passenger in a large aircraft is probably prohibitively expensive, however, various investigations have more recently been focused on local active noise control headrests [135–138]. For example, in [135] a local active noise control system is implemented on an aircraft-like seat in a small aircraft cabin mock-up and the performance of a delayless subband adaptive filter control implementation is explored. In [136] an active noise control system is installed in a typical economy class aircraft seat, as shown in Fig. 13, and used to control noise recorded on a Boeing 777 reproduced in a laboratory environment. Although the research on local active noise control for aircraft applications reported in [135–138] makes useful technology development, none report realistic in-flight application and this appears to remain an on-going challenge.

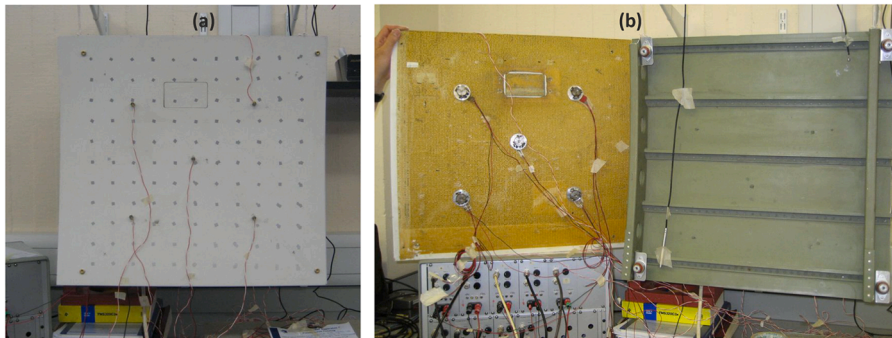


Fig. 11. (a) Front view of a fuselage section of a Bombardier Dash-8 Q400 turboprop aircraft whose trim panel (c) is equipped with five feedback control units using an electromagnetic inertial actuator with an accelerometer sensor at the centre of its footprint. (From Ref [113]).

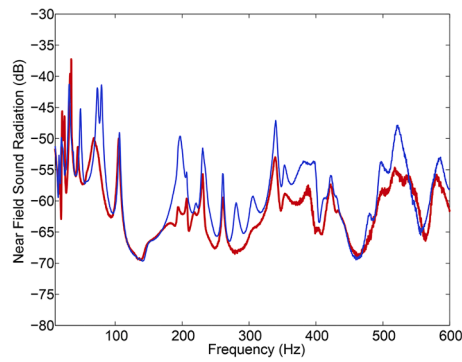


Fig. 12. Near field sound radiation when the fuselage skin is excited by a shaker. Thin blue line: without control units; thick red line: with closed loop control units. (From Ref [113]).

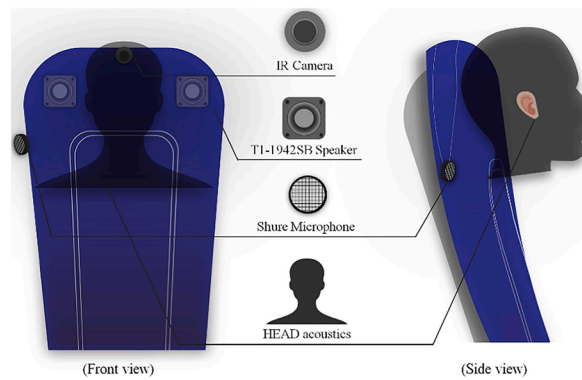


Fig. 13. Aircraft seat active noise control system (From Ref [136]).

3.3. Active noise control in cars

One of the most commercially widespread applications of active noise control within the passenger transport sector is in the automotive environment. The technology around this application was developed in parallel to the propeller aircraft systems discussed in Section 3.2. A very early, single-input single-output feedforward active noise control system implemented by General Motors and applied to control the engine orders associated with a diesel engine was reported in 1984 [12,19]. This system was already able to show significant levels of attenuation at the error microphone for various engine orders at a range of engine speeds at frequencies below around 200 Hz. However, the level of analysis is somewhat limited in terms of the robustness of the system to changes in engine speed during engine run-up for example, which is the primary challenge in the automotive engine noise control application compared to the relatively stationary nature of the propeller aircraft application. Subsequent research into the development of active engine noise control systems was reported in the late 1980s [19], which demonstrated effective noise control using a multichannel feedforward control implementation applied to three different vehicles. The system utilised a reference signal provided by an engine tachometer, two control loudspeakers and four error microphones and achieved a convergence speed of 0.1 seconds, which allowed the system to track changes in the noise due to engine speed and load variations. Despite only utilising error microphones mounted to the front headrests, the system was shown to also achieve significant reduction at the rear seats of the vehicle due to control of the first longitudinal acoustic mode of the cabin. This global control can be related to both the modal nature of the cabin and the relatively low frequency control bandwidth, with an upper frequency of 200 Hz, which relates directly to the physical limitations discussed in Section 3.1.1.

A typical active engine noise control system installation in a vehicle is shown in Fig. 14, which consists of 6 control loudspeakers, 8 error microphones and the reference signal provided directly from an engine speed sensor. One significant advantage regarding potential commercial exploitation of such active engine noise control systems is the economy of implementation. If the control systems were integrated into the car audio system, then very little additional hardware would be required – particularly in modern vehicles where microphones are in place for the telecommunication systems and the DSP associated with the audio system could in principle be leveraged for realisation of the relatively efficient FxLMS algorithm, for example. However, the first commercial system implemented by Nissan in their Bluebird vehicle in 1992 [28] in fact utilised an entirely separate set of loudspeakers, amplifiers and processors from the audio system and so was expensive and added weight to the vehicle. Perhaps due to these integration challenges, but also potentially due to the subjective performance of relatively low frequency control, it was not until the 2000's that a larger number of automotive manufacturers commercially exploited active engine noise control systems. For example, Honda implemented a

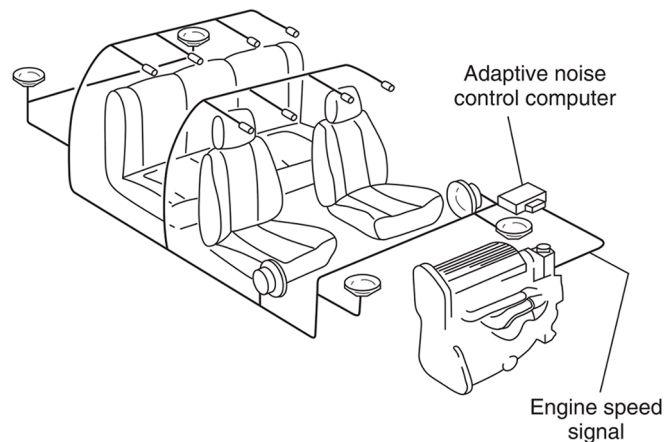


Fig. 14. Active engine noise control system with 6 loudspeakers, 8 error microphones and a reference signal provided by the engine speed (From Ref. [139]).

production active noise control system for both hybrid vehicles and vehicles utilising variable cylinder management [140]. In these applications, the active noise control system was able to compensate for acoustic issues associated with the operational characteristics of these specific vehicle powertrain systems, which were implemented to improve fuel efficiency. This trend was seen across a number of manufacturers including Toyota [141], Nissan [142] and Audi [143] for example and a large range of production systems now exist on the market [29].

Evolving from standard active engine noise control systems that aim to maximally attenuate specific engine orders, active control systems have also emerged that selectively attenuate or enhance features of the engine sound. These systems utilise the active noise equalizer [144] or sound profiling control strategies [145] to actively manipulate the engine sound with the primary objective of tailoring the driver experience. For example, Honda developed an active sound control system to enhance the “sporty” sound of the car during acceleration, whilst reducing engine boom during idle and cruise conditions [146]. More recently, research in the area of active engine noise control has generally considered some aspects related to the subjective impact of active control, with [147] investigating how active engine order cancellation influences the perceived loudness in a super sports car and [148] presenting a subjective study into the perceived performance of a global engine noise control system, reporting improvement in the acoustic comfort.

Actively controlling engine noise in cars is a problem well-suited to the feedforward control architecture described in Section 2.1 due to the availability of a single reference signal that is both time-advanced and correlated with the noise to be controlled. In the case of road noise control, obtaining reference signals that provide a time-advanced and coherent relationship with the error signals in the car cabin is more challenging due to the complex and unpredictable nature of road noise [149]. However, despite these challenges, feedforward active road noise control systems were demonstrated shortly after the first engine noise control systems in the early 1990s [150]. These road noise control systems typically utilise multiple loudspeakers and error microphones positioned in the car cabin, in a similar way to the engine noise control system shown in Fig. 14, but the reference signals are instead provided by multiple accelerometers that directly measure the vibration induced by road-tyre interactions. Fig. 15 shows the typical accelerometer locations utilised to provide suitable reference signals.

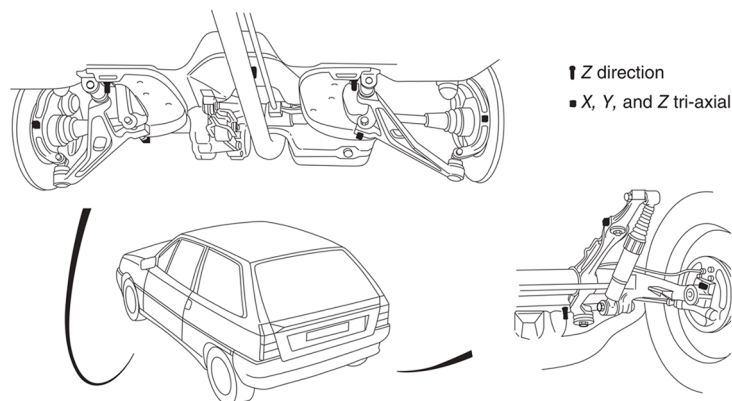


Fig. 15. Typical positioning of accelerometer reference signals located around the vehicle suspension system (From Ref. [139]).

The selection of reference signals in feedforward road noise control system is not a straightforward task due to the complex interactions between the tyre and the road and the often-nonlinear propagation paths through the structure of the vehicle to the interior cabin acoustics [151]. This complexity also means that multiple accelerometer reference signals are required, for example to capture paths in the vertical, lateral and longitudinal directions. In [150] it is noted that the reference accelerometer locations are vehicle specific, but an approach to select an appropriate subset of accelerometers from a larger set is proposed based primarily on maximising the multiple coherence between the reference signals and the error microphones in the cabin. Although this provides a convenient and efficient selection method, it does not directly consider the causality and the required time-advance between the reference and error signals was also evaluated to ensure this was sufficient considering the practical system delays. A similar reference accelerometer selection process was utilised in [152] and both of these systems achieve similar levels of control at frequencies below 300 Hz. The challenges associated with reference sensor selection remain and recent work has explored optimal approaches to reference sensor selection [153], although the underlying principles are still based on the fundamental methods outlined in [150].

To avoid some of the challenges associated with positioning and selecting reference accelerometers and the high costs of installing them in the somewhat hostile environment around the suspension systems, several alternative global active road noise control strategies have been proposed. Of note is the mass production system described in [33] and produced by Honda, which, as shown in Fig. 16, utilises a single channel fixed feedback controller to attenuate noise at the front seats associated with the first longitudinal mode of the car cabin. The feedback system is coupled with a fixed feedforward controller that is designed to ensure that the sound pressure level in the rear seats, which approximately coincide with the nodal line of the targeted mode, is not increased. A more complex multichannel feedback active road noise control system has also been proposed in [154], which uses microphones at both the headrests and positioned on the floor of the car cabin as feedback error signals to realise global control using an Internal Model Control architecture. This system is able to achieve similar performance to the feedforward control systems reported in [150,152] without the complexity of positioning reference accelerometers, however, potentially due to challenges of performance and robust stability this system design has not been developed for mass production.

To increase the bandwidth of achievable road noise control it is necessary to move to a local active noise control system, as discussed in Section 3.1. Although many practical local active noise control systems have been reported in the literature, with a large number of fundamental studies being published in the 1990s, the application to road vehicles was not thoroughly explored until relatively recently. This is perhaps due to both the engineering, safety and cost challenges of installing loudspeakers and error microphones in each headrest. However, specific investigation of local active noise control for the reduction of road noise inside a vehicle was reported in [155]. The investigated system utilised two headrest mounted loudspeakers, as shown in Fig. 17, to control the pressures at the ears of a dummy head, which were estimated using the remote microphone technique [156] and an array of monitoring microphones, some of which are also noted in Fig. 17 by the numbers in blue circles. A standard feedforward control strategy was utilised, with the reference signals being provided by 8 sensors installed around the four wheels [155]. The system is shown to be able to achieve very wide-band attenuation, with significant attenuation up to around 1 kHz, as shown in Fig. 17. That said, the performance is shown to only be maintained over this wide frequency range when the remote microphone processing and plant responses used in the feedforward controller are updated based on knowledge of the head position. To address this challenge, integration of head-tracking technology into local active noise control systems has been proposed [81] and in the automotive environment it is possible that this head-tracking capability could be integrated, at least for the driver, with driver monitoring systems. In addition to head-tracking, the robustness of feedforward local active noise control in passenger vehicles to practical uncertainties has also been investigated, particularly in relation to the performance of different virtual microphone techniques [157,158]. This work explored how different uncertainties can be combined [157] and how different virtual sensing algorithms are robust to either response uncertainties or changes in the properties of the reference signals [158]. An alternative local active noise control system based on feedback control has recently been proposed [159], which avoids the complexity and cost of installing multiple reference sensors in the vehicle. This feedback headrest local active noise control system is shown to achieve comparable performance to the feedforward systems discussed previously, but the robustness in this case to head movements, speech in the car cabin and changes in the environment have not yet been explored.

In addition to the active noise control systems discussed to this point, the potential of ASAC systems in the automotive environment

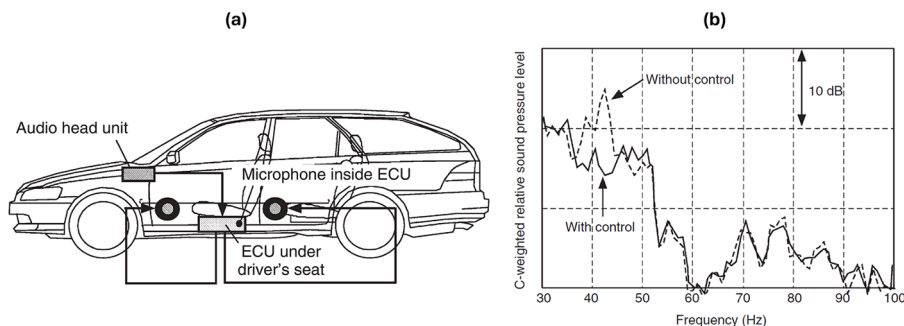


Fig. 16. (a) Feedback active road noise control system described in [33] and (b) the corresponding performance achieved at the front seats (From Ref. [139]).

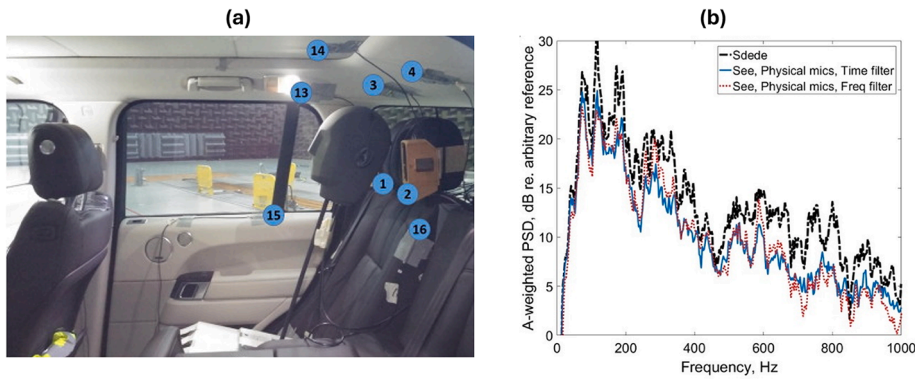


Fig. 17. Local active road noise control system described in [155] (left) and the corresponding performance achieved at the right ear error microphone (From Ref. [155]).

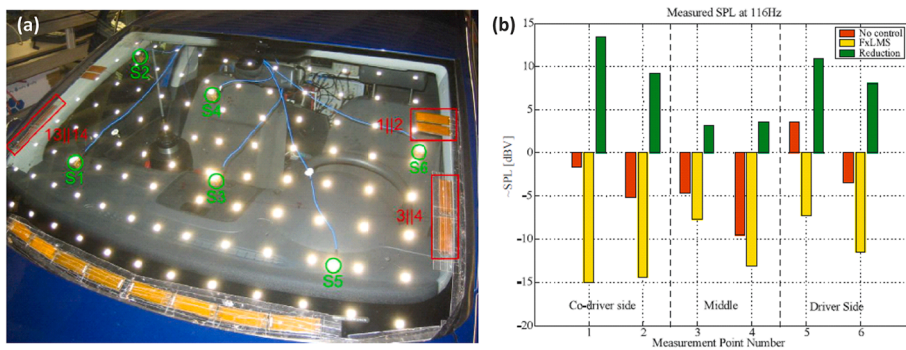


Fig. 18. Glass pane equipped with piezoelectric actuators and sensors for the implementation of a multichannel feedforward ASAC controller (a) and measured differences in SPL at the second structural resonance of the windscreen (b) (From Ref. [55]).

have also been explored for the control of broadband interior noise. For example, Misol *et al.* [55,56] proposed an active windscreen, which as shown in Fig. 18a, was constructed from a standard laminated glass pane equipped with piezoelectric patch actuators bonded to the blackened rim of the windshield. According to Fig. 18b, the laboratory experiments showed reductions of the sound pressure level of between 3 and 13 dB at the second structural resonance of the windscreen.

3.4. Active noise control in maritime transportation

The application of active noise control systems to maritime transportation is much less common than for other transportation applications. This is most likely due to a lower level of standardisation and perhaps expectation in the maritime environment compared to other sectors. This distinction has been discussed in [160] for acoustic comfort in general via a comparison between standardisation in the building and yachting sectors. This study highlights the need for further development and acceptance of acoustic quality indicators within maritime transportation. Nevertheless, a number of studies focused on the application of active noise control to luxury yachts have been published, partially due to comfort classifications [160] and partially due to customer expectations.

In [21] a multichannel feedback active noise control system is presented which utilises four error microphones mounted in the headboard of the bed and two control loudspeakers positioned under the bed around the head position to control low frequency noise in the cabin. Significant levels of narrowband noise attenuation are reported at frequencies below 100 Hz for noise generated by air conditioning units, power generators and the main engines. In [22] a feedforward active noise control system is also explored for reducing power generator noise in the master cabin of a luxury yacht. In this case, three loudspeakers and four error microphones, as shown in Fig. 19, are used to attenuate 19 engine orders using a FXLMS controller with the reference signal provided by a tachometer. This system, therefore, is similar to the feedforward control systems utilised to control the tones associated with propeller noise in aircraft (see Section 3.2) and engine orders in a car (see Section 3.3). The distinction here is that although the tones are largely time-invariant, it is necessary to control a very large number of orders to achieve a perceptual improvement in the acoustic comfort.

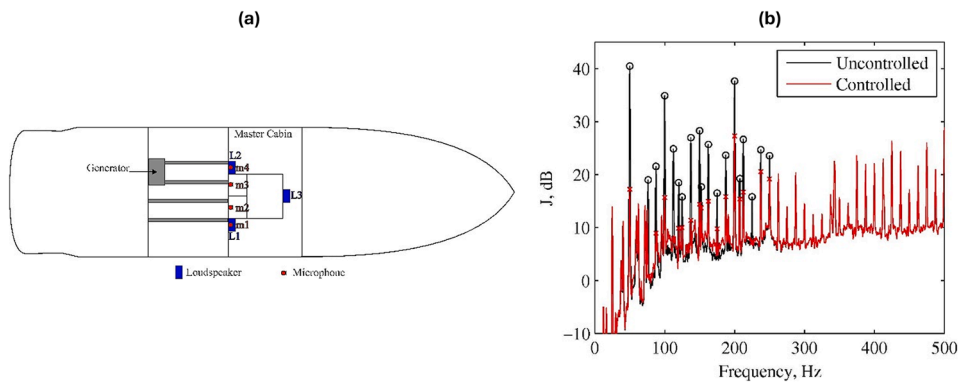


Fig. 19. (a) The layout of microphones (m1-m4) and control loudspeakers (L1-L2) used in the active noise control system described in [22] and (b) the corresponding control performance achieved at the error microphones (From Ref. [22]).

4. Active vibration control

In addition to high noise level disturbances, transportation vehicles are also commonly affected by significant levels of vibration. The typical sources are similar to those generating the noise control challenges, with either the power-train or the motion of the vehicle being the primary fundamental sources [84–86,161–163]. A wide range of passive control treatments have been developed to mitigate both sources of vibration. In general, vibration control can be realised at three points: firstly, by neutralising the source of vibration; secondly, by isolating the transmission of vibration through junctions; thirdly, by controlling the vibration at the receiver body or structure. However, controlling the vibration of the source is quite a challenging task since it requires passive devices (normally vibration absorbers) capable of producing large control forces. Isolating the vibration transmission through mounts offers an appealing option, which, nevertheless is limited by the fact that the joining elements should comply with two contrasting requirements: to connect, they must be statically as rigid as possible, whereas, to isolate, they should be dynamically as soft as possible [164]. The control of vibrations at the receiver structure has the advantage of controlling exactly the component to be insulated. However, in most cases, receiving structures are made of complex frameworks of stiffeners and panels, whose responses are characterised even at low frequencies by a reach density of resonant modes that need to be controlled [72]. As a whole, passive treatments offer a viable solution at mid to high frequencies, but at low frequencies they lead to bulky solutions, that require significant space and weight, limiting their applicability to transport vehicles [84–86]. Moreover, passive solutions cannot adapt to changes in the vehicle operational condition, which might include the variable speed of powertrain or machinery, or the change in the dynamic response due to changes in, for example, aircraft altitude and thus cabin pressurization (which defines fuselage tensioning effects), vehicle speed (which defines the loads on the vehicle bodywork) sea temperature or air temperature and humidity (which may generate significant structural tensioning effects). To overcome these limitations, like in the case of active noise control discussed in Section 3, active vibration control systems have been variously researched and developed for application in passenger transport vehicles.

Although the concept of active vibration control does not perhaps extend as far back as the first patent on active noise control [1], the technology has several decades of history. Indeed, in 1968 a review paper already contained 59 prior publications on the fundamental concept [165]. However, the implementation of practical solutions was limited to simple systems encompassing electromechanical actuators (mainly servo-hydraulic mounts [166–168]) and elementary analogue controllers [5]. This section firstly provides an overview of the fundamental principles of active vibration isolation at junctions between structures and active vibration control of whole distributed structures. Then it presents an extensive overview of the more recent practical applications of active vibration control systems for passenger transportation vehicles with a focus on aircraft, cars, trains and maritime vessels.

4.1. Active vibration control system designs and limitations

The active control of vibrations is in its own way more involved than the active control of noise. Indeed, here, beside the control architectures (feed-forward vs. feedback), also the type (point or distributed, strain or inertial), the number and position (collocated/dual) the electro-dynamic response (stiff vs. soft, low vs. high energy density) of the control sensors and actuators play a key role on the performance of the whole control system. In fact, often active vibration control systems are presented as smart systems or smart structures [58,62]. These terms are used to emphasise the level of integration between the structure under control, the sensors-actuators transducers and the controller. The following two subsections present brief introductions to the key features of the two principal active control strategies, that is: the active vibration isolation at junctions between structures and the active vibration control of whole distributed structures. The readers wishing to delve deeper should refer to the milestone references reported below and to the classical textbooks on AVC [5,6,58,59,68,169,170].

4.1.1. Active vibration isolation

To start with, the implementation of active vibration isolation systems is revised considering the simplified source-mount-receiver

models depicted in Fig. 20(a-d). As for passive systems, three active control approaches are normally employed [171]: firstly, the neutralisation of the source vibration; secondly, the isolation of the vibration transmission through the junction; and thirdly the control of the receiver vibration. As depicted in Fig. 20(a), source control requires a sky-hook control force that counteracts the primary force disturbance. Naturally, electro-mechanical transducers generate pairs of reactive forces [169,172–174]; hence, to have the desired force action, the actuator is normally reacted off a seismic mass mounted on a spring, which leads to the so-called inertial actuator shown in Fig. 20(e). As depicted in the graph of Fig. 20(e), this solution cannot provide the required constant force at low frequencies [169]. Indeed, below the fundamental natural frequency ω_a , the actuator generates increasingly smaller control forces with negative phase, which may lead to control spillover with direct feedback control architectures [115,175,176]. Alternatively, as shown in Fig. 20(b, c), mount control implements a reactive control force on the junction between the source and receiver components. Two configurations can be envisaged: the so-called soft and hard active mounts [177], where the actuator is mounted respectively in parallel and in series with the passive mount [178,179]. Here, the isolators are deemed to copy the oscillations of the free-source element, which corresponds to softening the dynamic stiffness of the mount such that low levels of vibration are transmitted to the receiver structure. Hence, as depicted in Fig. 20(b, c), in these cases, large control forces are required at low frequencies below the fundamental resonance frequency of the elastically mounted source ω_s . This is particularly true for the hard mount configuration shown in Fig. 20(c), where the transducer must overcome its own hard stiffness. Finally, as reported in Fig. 20(d), receiver control relies on a sky-hook control force at the base where the mount is connected to the receiver structure. In this case too, the sky-hook force is obtained with the inertial actuator shown in Fig. 20(e). According to the graph in Fig. 20(d), quite a large control force is needed at the fundamental resonance frequency of the mount-source assembly ω_s . Here too, the inertial actuator should be designed with a much lower fundamental natural frequency, ω_a , than that of the elastically suspended source, ω_s , although this may lead to other complications such as saturation effects due to the proof mass hitting the end stops of the elastic suspension [119,120], which have been tackled in various ways [121–126,132,133].

In general, the choice of the control actuators and error sensors, and thus of the cost function being minimised by the controller, play an important role for the success of an active isolation system. Two aspects are central for the choice of the actuators: the stress vs. strain property and the energy-density vs. frequency-band of the actuation transducer [169,180,181] which are depicted qualitatively in Fig. 21 for the most common electro-mechanical transducers employed in active vibration control. Normally stiff actuators, with high stress to strain ratios, are employed for the hard mounts (Fig. 20(d)), whereas, compliant actuators, with low stress to strain ratios, are used in soft mounts (Fig. 20(c)) and in inertial actuators (Fig. 20(a, b)), which require large strokes of the suspended mass to generate a substantial base control force. As highlighted in Fig. 20(e), this requirement can be partially relaxed by using a leverage that magnifies the oscillation of the suspended mass (normally this accomplished with fluids [182–185]). The control performances of active isolators strongly depend on the appropriate choice of the actuator, which must guarantee over the whole frequency band of the primary disturbance the control force and control stroke deemed by the dynamic response of the source-mount-receiver assembly [164,178]. Moreover, its energy density must be sufficient to guarantee low weight and small dimension practical designs.

In most cases, the controller is set to minimise either the force through the mount or the velocity at the base [178]. An interesting option combines these two parameters in terms of minimisation of the vibration power transmission [186–189]. This cost function can incorporate both the linear and rotational Multi Degrees Of Freedom (MDOF) vibration transmission at each mount and can synthesise the vibration transmission through multiple mounts [186,190–192]. Nevertheless, it must be handled with care since it may lead to vibration absorption from uncontrolled flanking paths [58,68,187].

In general, to control tonal disturbances feed-forward control algorithms offer the best option; in particular, adaptive feedforward controllers. Here the concept of quality plays a key role on the definition of a minimum phase plant response between the sensor(s)-actuator(s) [58,178], which is a key factor for the implementation of fast and stable adaptive feedforward controllers [4,59,178]. Alternatively, to control stochastic disturbances, feedback control provides a viable solution. Albeit the control literature proposes

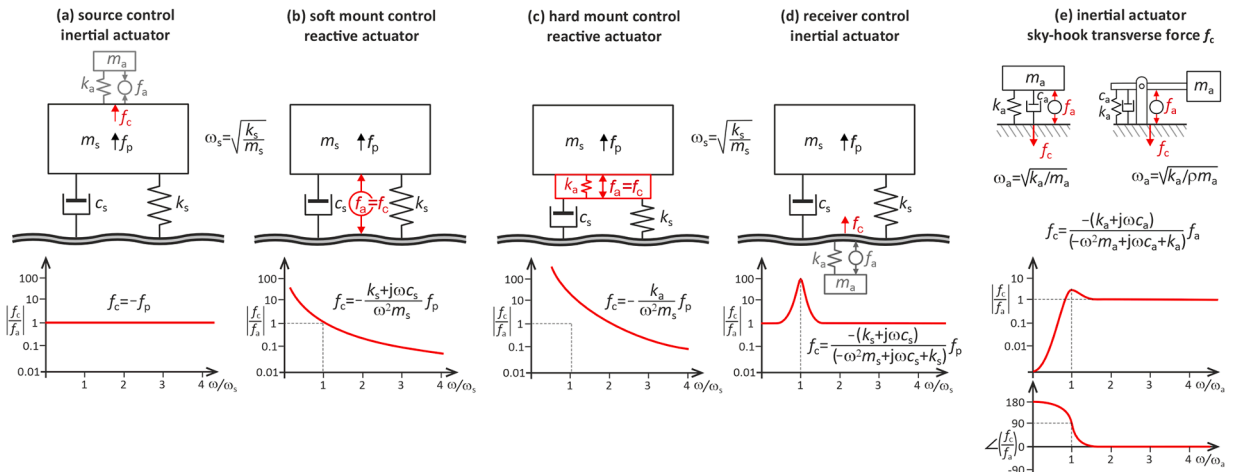


Fig. 20. (a) Source, (b) soft-mount, (c) hard-mount, (d) receive control vibration isolation, (e) Inertial actuator control force.

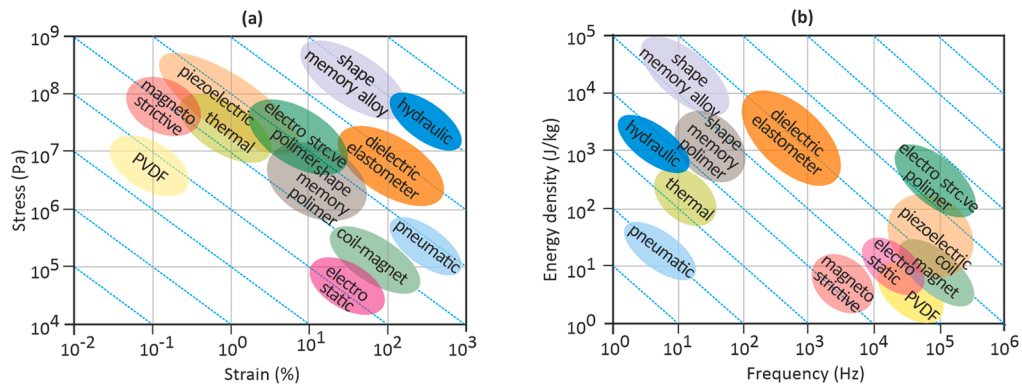


Fig. 21. Qualitative diagrams for (a) stress vs. strain and (b) energy density vs. frequency constitutive properties of the most common transducers employed in active isolation and active vibration control systems.

quite advanced multi-channel optimal feedback control approaches (e.g. see [58,59]), much simpler solutions have been adopted in practice like direct velocity feedback control, which mimic the so-called sky-hook active damper [118,179,193,194]. These features have brought to the further developments, in particular the implementation of decentralised active mounts that, indeed, implement local velocity feedback loops, that is sky-hook active dampers [195–197]. Here too, collocation and duality of the sensor-actuator pairs [58,68–71,178,198] is vital to guarantee stable feedback loops with large control gains [199–204].

4.1.2. Active vibration control of distributed structures

In parallel to active isolation, active vibration control offers a valuable option to control the vibration of distributed structures. As discussed in [129] and shown in Fig. 22(a), several control layouts can be implemented on structures, which can be categorized with respect to: a) the control layout (centralised vs. distributed), b) the cost function (global or local) and c) the optimisation (holistic or modular). For instance, a centralised layout (CN-1) can be used where a single multi-channel controller processes all sensors signals to generate the control signals to all actuators. Alternatively, distributed solutions can be envisaged encompassing numerous local controllers having the ability to link with other controllers in the system (CN-2,3,4). In the special case where there is no communication between local controllers, the active system is normally referred to as decentralised control (CN-4a). Both centralised and

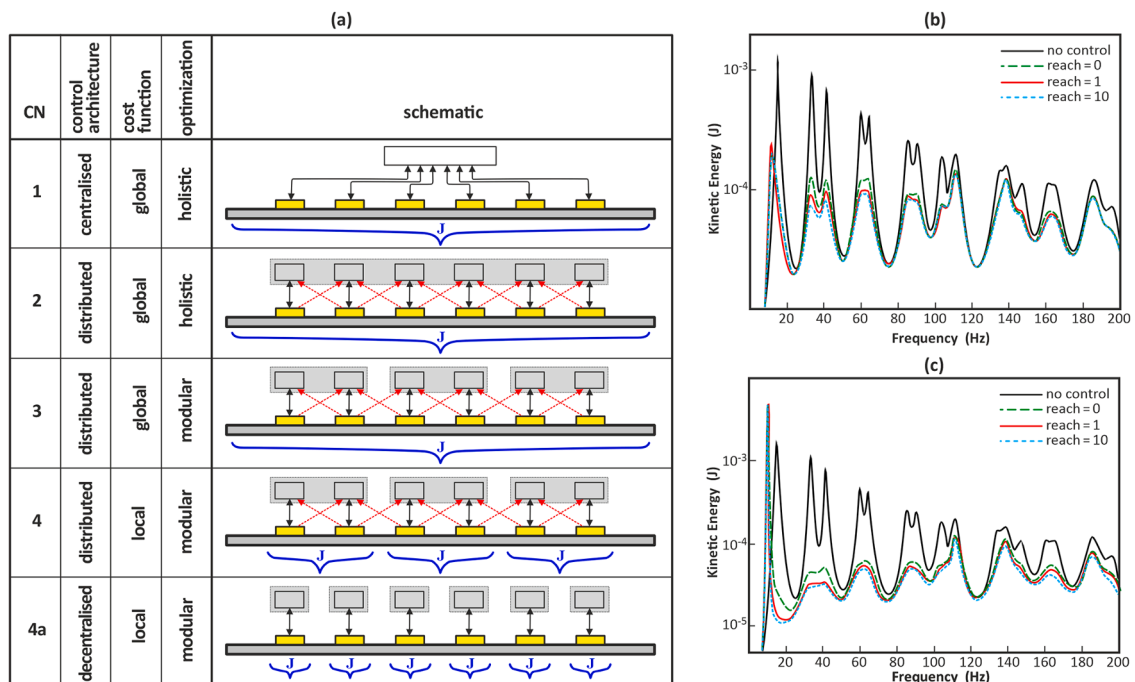


Fig. 22. (a) Active vibration control systems. Plate flexural kinetic energy for (b) centralised (CN-1) vs. distributed (CN-2) controllers using holistic optimisation and for (c) centralised (CN-1) vs. distributed controllers (CN-3,4,5) using modular optimisation (Adapted from [129]). In (b,c) reach indicates the number of units each unit communicates with.

distributed controllers can be set with respect to a global or a local cost function, like the minimisation of the spatially-averaged vibration of the whole structure or, alternatively, the cancellation of the vibration at the control units or the maximisation of the vibration power absorbed by the control units (CN-4,4a). Finally, the design of the compensators in the units of the distributed control systems, can be done holistically or modularly, that is with or without knowledge of the mutual effects between units.

The study presented in [129] compared the flexural vibration control effects generated on a thin plate by the control layouts shown in Fig. 22(a) using the inertial force actuators depicted in Fig. 20(e). As can be seen in Fig. 22(b, c), for broadband excitations, centralised feedback control systems that implement global-holistic control (CN-1) offer marginal advantages compared to distributed system implementing either global-holistic control (CN-2) or global-modular (CN-3) and global-local (CN-4,4a) control. However, as can be noticed in Fig. 22(c), the distributed systems implementing global-modular and global-local controls (CN-3,4,4a) are characterised by significant control spillover effects at low frequencies close to the fundamental resonance frequency of the actuator [115–117,119,120] which are due to the non-perfect duality between the sensor-actuator pair of each unit [58,68–71,198,205]. Nevertheless, the study in [129], showed that, when the control systems are set to control the response of a single or a small group of modes, which is typical in the feedforward control of tonal disturbances, then there is a net advantage either for centralised or distributed systems that implement global holistic control (CN-1 and CN-2).

The practical implementation of distributed control requires simple, small-scale, vibration control units that guarantee robust stability and significant vibration control performance. In this respect, the works presented in [61,95,206–208] showed that direct velocity feedback control [199,201–204,209] offers a viable solution since it generates sky-hook active damping [68,95,200] that guarantees both vibration absorption and control stability. However, to properly implement a direct velocity feedback loop, the unit should have a dual sensor-actuator pair [61,62,198,201–203,205]. In this respect, a number of solutions have been proposed, like the collocated point accelerometer seismic sensors and electromagnetic or magneto-strictive inertial actuator [115,118,175,176,210–213] shown in Fig. 23(a), the piezoelectric patch actuator and accelerometer sensor displayed in Fig. 23(b) [62,130,131,214] and the piezoelectric patch sensor and actuator pairs shown in Fig. 23(c, d, e) [62,215–218]. Each of these solutions present limitations with respect to collocation and duality and thus for the stability of the feedback loop. For example, as shown in Fig. 20(e), the proof mass actuator in the unit shown in Fig. 23(a) generates a point force above the fundamental resonance frequency ω_a of the actuator [117,128,219,220]. However, at lower frequencies than ω_a the control force is actually out-of-phase with the driving signal such that a positive velocity feedback loop effect is generated at these frequencies by the unit, which produces control spillover and can even lead to instability [118,169,175,176,221]. The extent of this phenomenon can be mitigated by lowering the fundamental resonance frequency $\omega_a = \sqrt{k_a/m_a}$ of the actuator, but this solution tends to increase the static displacement of the proof-mass m_a [118,175,176], which, in presence of shocks, may hit the actuator stop ends and trigger instability effects [119,120,175,212,222]. To circumvent the contrasting actuator requirements of a low fundamental resonance frequency and a low static deflection, various approaches have been proposed. This includes solutions with relative displacement, relative velocity and force feedback loops between the coil-magnet components of the actuator [121–123,213]; the use of a compensator to electronically lower the fundamental resonance frequency of the actuator [124,223]; the use of a nonlinear internal controller to reduce excessive oscillation in the moving mass and thus avoid hitting the end stops [125,132]; and the use of a flywheel inertial actuator to augment the inertia effect for a given weight of elastically suspended actuator mass [126,133]. The piezoelectric patch actuator with the accelerometer sensor at its centre shown in Fig. 23(b) offers a rather simple design, which, however, guarantees a collocated sensing-actuation effect at low frequencies such that the flexural wavelength of the structural vibration is significantly greater than the dimension of the patch [62,130,131,209,214,218]. This issue is normally solved by low-pass filtering the error signal, which avoids also the problem of positive feedback at higher frequencies than the accelerometer working bandwidth [130,214]. The piezoelectric patch sensor and actuator pair shown in Fig. 23(c) presents an even simpler unit, which however must be carefully designed since the two transducers are coupled via both bending and axial deformation of the main structure. This can generate stability issues since the two coupling effects are actually out-of-phase and thus can lead to a positive feedback loop effect at higher frequencies where the axial coupling becomes relevant [214]. This problem can be minimised by choosing sufficiently small size patches and adopting yet again a low-pass filter in the loop. Alternatively, the solution proposed in

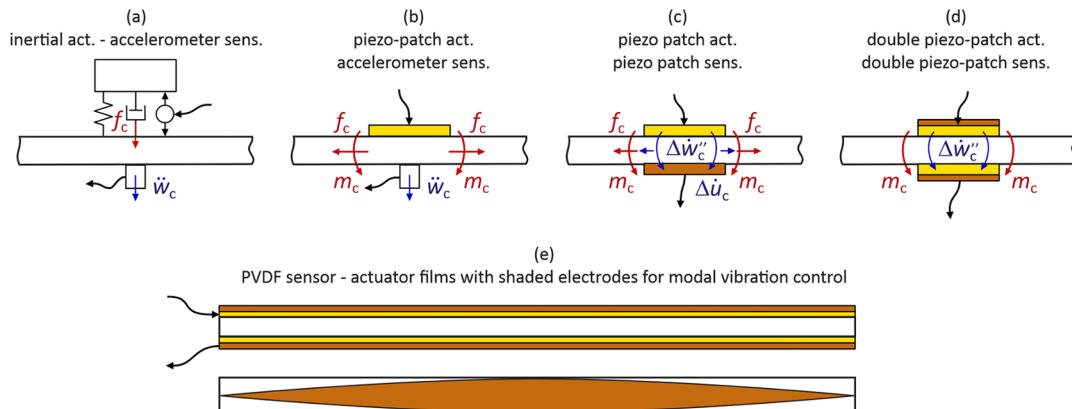


Fig. 23. (a-d) Typical sensor-actuator pairs used for modular point control units. (e) distributed sensor-actuator pairs for global vibration control.

Fig. 23(d) can be adopted where pair of patches are used to generate and detect bending vibration only. This design can be employed with larger transducers that cover the whole structure such that the vibration control can be targeted to specific vibration modes, or sound radiation modes by properly shaping the electrodes of the transducers [60–65,214].

Beside these general considerations, the implementation of active vibration systems presents few more features. For example, multiple wave-types (e.g. flexural and longitudinal waves) can co-exist in beam-like structures, which cannot be controlled independently since they are normally coupled at junctions or terminations of the structures [224,225]. This aspect can be effectively tackled by implementing controllers set to minimise the vibration power flow due to all waveforms [188,224]. The idea of minimising the vibration energy was extended to the control of flexural vibrations in two-dimensional structures too [226], although it was limited to the implementation of feed-forward control of power input at specific points rather to the power flow. Here, the problem of control spillover is central, and the cost functions must be carefully chosen in such a way as to avoid vibration energy absorption from flanking paths [187].

4.2. Active vibration control in helicopters

Helicopters are characterized by a harsh vibroacoustic environment, primarily caused by the powertrain components (including the engines and main gearbox) and tonal excitations from the rotor blades. This can lead to very high levels of structure-borne noise in the cabin and undesirably high levels of vibration of the pilot and passenger seats, cabin floor, and the instrumentation used to operate the vehicle, which can pose both operational and safety challenges. The most significant impact on passenger comfort in helicopters stems from the vibration induced by the main rotor. This vibration results from blade aerodynamic interactions and is transmitted to the cabin through the rotor hub. Although this vibration is characterised by a series of harmonics, the dominant vibration frequency in helicopters typically falls between 10 and 30 Hz, depending on the size of the helicopter. Passenger comfort is particularly sensitive to vibrations below 80 Hz, with discomfort increasing at lower frequencies. To maintain both aircraft operability and the comfort of passengers and crew, vibration levels must be kept within defined limits, which depend on factors such as exposure duration and frequency content, as specified in various civilian [227] and military [228] standards. Initial efforts to reduce helicopter vibration led to the development of various passive vibration control strategies, which, despite their effectiveness, resulted in significant weight penalties. As a result, from the mid-1970s onward, AVC systems began to be recognized as a promising approach to fulfilling the rigorous vibration standards while maintaining a tolerable increase in weight. Although various AVC architectures have been conceptualized, they all share the same fundamental components: actuators, sensors, and a controller. The specific arrangement and, notably, the placement of actuation can differ considerably between systems. For classification, the actuation location can be used to define three principal categories of AVC system for helicopter applications:

- rotor actuation (addressing vibration at the source)
- rotor/airframe interface actuation (providing vibration isolation)
- fuselage actuation (controlling vibration at the crew, passengers and equipment locations)

Fig. 24 illustrates some of the principal AVC systems developed over recent years, categorized into these three groups and Table 1 catalogues whether these systems are at a prototype stage or have been implemented in production helicopters. The following subsections will cover the foundational principles behind the various AVC solutions summarised by Fig. 24. The intent here is not to exhaustively review the literature on helicopter AVC systems, but rather to highlight significant examples that help to contextualise the field.

4.2.1. Active vibration control at the source

The basic principle of applying AVC to the source in the helicopter application is to actuate and control the main rotor to prevent the vibration from being transmitted into the airframe. Different types of active rotor control have been proposed by both helicopter industry and research institutions since the initial development of active control systems. The main architectures are schematically shown in Fig. 25 and are described in the following subsections.

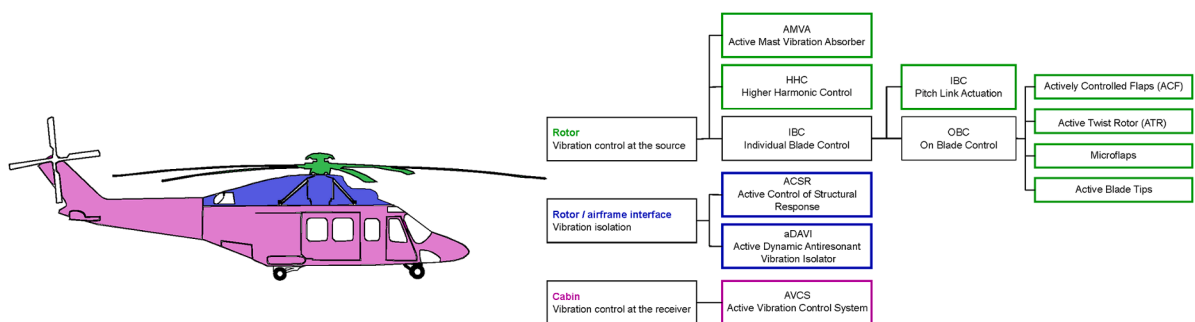
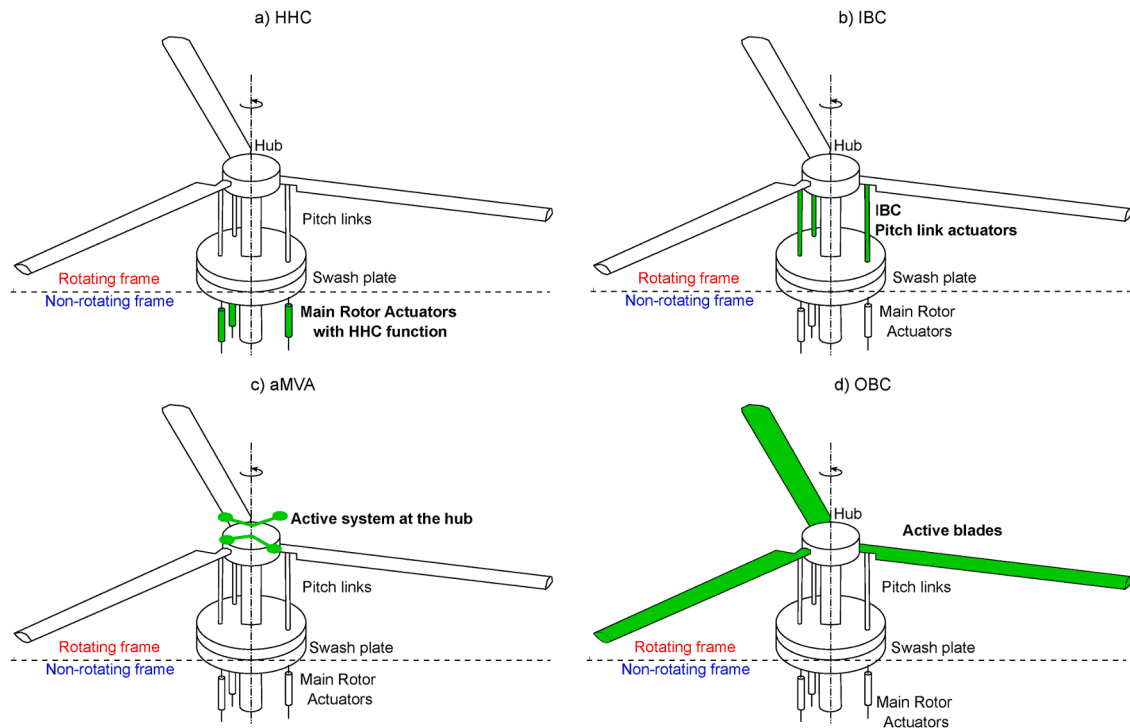


Fig. 24. Schematic diagram of helicopter zones – rotor (green), rotor/airframe interface (blue), and cabin (purple) – with the corresponding active vibration control strategies.

Table 1

Active vibration control systems categorised by the actuation location.

Active vibration control system	Production\R&D	Actuation location
Higher Harmonic Control (HHC)	R&D	Rotor Vibration control at the source
Individual Blade Control (IBC) pitch link actuated		
Active mast vibration control		
Actively control flaps		
Active Blade Twist		
Microflaps	Production	Rotor\airframe interface Vibration isolation
Active blade tips		
Active Control of Structural Response (ACSR)		
Active DAVI	Production	Cabin Vibration control at the receiver
Active Vibration Control System (AVCS)		

**Fig. 25.** Schematic diagrams showing the arrangement of actuators (green) for a) Higher Harmonic Control b) Individual Blade Control c) active Mast Vibration Absorption d) On Blade Control.

4.2.1.1. Higher Harmonic Control. The initial theoretical investigations on active rotor control technology were based on so-called Higher Harmonic Control (HHC). As shown in Fig. 25a, HHC uses servo-hydraulic actuators attached directly to the swashplate. These actuators are normally driven at the lower frequencies required to adjust the pitch angle of the rotor blades to control the flight dynamics of the aircraft. The concept of HHC is based on using the same actuators to control vibration, which means that the actuators must be able to produce the necessary forces over a much wider frequency bandwidth than required for aircraft manoeuvres. To control vibrations at rotor harmonic frequencies, specifically defined as multiples of the number of blades (N) times the rotational speed of the rotor (rev), in the non-rotating frame (as shown in Fig. 25a), Higher Harmonic Control (HHC) must counteract loads generated in the rotating frame and transmitted to the fixed frame. Consequently, the actuators must be capable of generating control forces at least at the N/rev harmonic frequency.

Initial investigations into HHC date back to 1957 but focused more on enhancing rotor performance than on vibration reduction, [229–231]. Since then, however, HHC has undergone extensive testing, including numerous wind tunnel experiments. For example, in the late 1970s, NASA launched a wind tunnel research programme focused on HHC [232]. In this programme the rotor was a 4-blade articulated aero-elastically scaled model in which the swashplate actuators were used to generate input forces at $4/\text{rev}$ frequency alongside the regular flight control inputs. Initially, the tests were executed in open-loop and later in closed-loop using an adaptive control algorithm to minimise the load measured in the fixed frame or the measured acceleration [233]. Since 1980, various wind tunnel tests were carried out to explore and refine the concepts of HHC. These tests examined new digital control algorithms applied to different rotor configurations, including hingeless four-blade rotors, three-blade rotors, and tiltrotor systems [234–240].

The first flight test to evaluate the effectiveness of HHC in reducing vertical cabin vibration was carried out by Bell on a UH-1A helicopter with a 2-bladed, semi-rigid rotor [241]. The rotor was equipped with HHC actuators, installed in the non-rotating system and replacing the conventional rod-end links between the control mixer and the stationary swashplate. Although this flight test did not completely fulfil all the initial objectives, it demonstrated that HHC could achieve some reduction in vertical vibration and also highlighted that the optimal actuation force phasing depends on the flight conditions and the objective function to be minimised. Subsequently, HHC systems have been investigated by Hughes Helicopters on OH-6A [242,243]. Fig. 26(a) shows the installation of the HHC actuator on the four-blade rotor. Vibrations were measured at the pilot's seat in all three directions. Fig. 26(b) presents the 4/rev vibration levels as a function of airspeed, both with and without closed-loop HHC control. The results indicate that vibrations at the pilot's seat are significantly reduced when HHC is active at airspeeds above 65 knots, except for the longitudinal component. However, the baseline longitudinal vibration level is already lower than the other directional components. Other implementation of HCC control have been made by Sikorsky [182–185,244,245] and Aerospatiale [246,247]. These various studies have demonstrated the potential of HHC, showing significant reduction in vibration at the pilot's seat across various flight speeds [243].

Despite over half a century of research and development on HHC, no known serial production helicopters currently employ this technology. The factors contributing to this absence include the stringent demands for system reliability, the critical need for fail-safe mechanisms, complex certification hurdles, and the necessity for control effectiveness balanced against the constraints of minimal weight and cost. One of the major impediments is the requirement for actuators to generate high amplitude and frequency loads and the challenges associated with applying these forces to the rotor components and blades effectively. Furthermore, a fundamental limitation of HHC in conventional helicopters is that the number of actuators is limited to three. This limitation means that HHC can only manage three out of the six hub loads (which include three forces and three moments), inherently restricting its vibration suppression capabilities. Additionally, the actuation system, typically located in the fixed frame, can only generate forces in the rotating frame at specific frequencies. This means that for rotors with more than three blades some harmonics cannot be transferred to the blades, potentially diminishing the control system effectiveness. For example, it could hinder the ability of the system to control 2/rev harmonics, significantly contributing to the rotor noise emission. For an in-depth analysis and contextual understanding of HHC, its applications and limitations with respect to helicopter vibration control, the reader is referred to [248,249].

4.2.1.2. Individual blade control. Individual Blade Control (IBC) presents a sophisticated alternative to HHC and was proposed in 1976 [250]. IBC utilises dedicated actuators for the independent actuation and control of each blade, which aims to overcome some of the constraints of HHC. IBC replaces conventional pitch links, which can be seen in Fig. 25a, with actuators that react off of the rotating part of the swash plate to control the blade pitch angle, as illustrated in Fig. 25b. Extensive research into IBC has been carried out and is summarised in [251,252]. The practical application of IBC has been initially tested and verified using a wind tunnel model tailored for this specific control methodology. Given the additional control freedom provided by the IBC compared to the HHC, other control objectives, in addition to vibration control, can be implemented to mitigate undesirable dynamic effects. Some examples are gust alleviation [253], lag damping augmentation [254], stall flutter suppression [255] and flapping stabilization in forward flight [256]. Depending on the objective of the controller, the actuation frequency band may be required in a low-frequency region ranging from

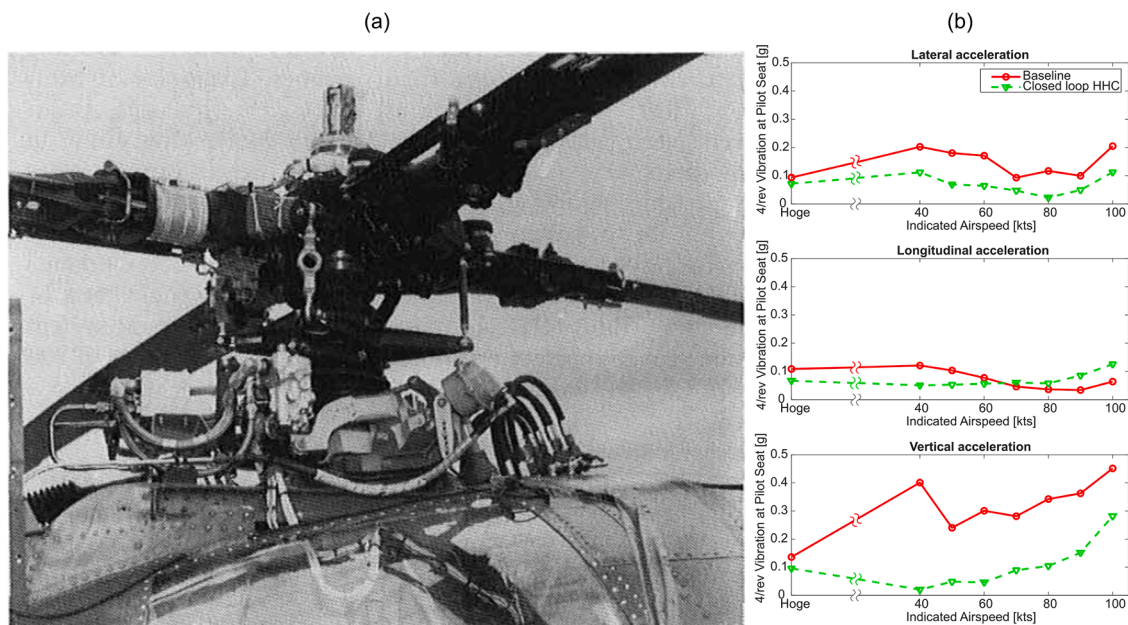


Fig. 26. (a) HHC system on OH-6A, right lateral actuator installed between mixer and swashplate [239]; (b) Pilot's seat vibrations with and without HHC in level flight (adapted from [242]).

0 to 1/rev for gust alleviation, flying handling qualities, blade instability improvement and ground resonance avoidance. The high-frequency range above 1/rev is required to improve vibration control and stall flutter suppression. Regardless of the specific application, the design of the IBC system must prevent any interference with the cyclic pilot control at the 1/rev frequency.

The first full-scale wind tunnel test was conducted on a 4-bladed Bo 105 rotor with servo-hydraulic IBC at the NASA Ames wind tunnel in 1993 and 1994 [257,258]. The experiment aimed to reduce vibration and Blade-Vortex Interaction (BVI) noise and enhance flight performance. Meanwhile, the initial practical application of IBC on a full-scale helicopter was realized in 1990 [259]. The control system incorporated actuators between the swashplate and the rotor blades, effectively replacing the conventional pitch links. This configuration allowed for the superimposition of individual blade control forces over the standard 1/rev swashplate flight control inputs. The IBC system was integrated into the MBB BO-105 hingeless rotor helicopter and first evaluated on a whirl tower. Subsequently, in 1990 and 1991, flight tests were conducted at Eurocopter's facility in Ottobrunn, employing low-imposed control authorities [259,260]. During the test campaign, a closed-loop IBC explored the vibration reduction capability using error signals measured in the rotating frame and a disturbance rejection controller type [261]. Minimisation of 4/rev fuselage vibrations was achieved by eliminating 4/rev hub force and moment excitations. Results shown in reference [261] compare the cabin vibration spectra obtained during level flight at 60 kts with and without the IBC system. The data clearly demonstrate that the IBC effectively reduces the 4/rev vibration components at the copilot seat location along all three spatial axes. The theoretical background and design of the controller are outlined in [262]. In this application, of the three forces and three moments exciting the fuselage, only three were retained for the demonstration campaign: M_x , M_y and F_z .

A flight test of an IBC system was carried out by ZFL on a 6-bladed CH-53G helicopter in open-loop [263] and closed loop with full authority [264]. Using single harmonic IBC inputs, the average 3-axis acceleration reduction measured in the cabin at specific locations was 60%, while using multi-harmonic control achieved 84% reduction for the same flight conditions.

Theoretical applications of IBC have also been considered for vibration reduction in coaxial rotor helicopters. Since these aircraft can achieve higher forward flight speeds than a traditional helicopter and use a stiff rotor blade design, the main rotor is prone to generate high vibratory loads, requiring an active vibration control system. The vibration reduction feasibility of individual blade pitch control on the lift-offset coaxial main rotor of the XH-59A has been investigated in [265–267].

From this brief discussion on IBC, it has been demonstrated that the system can obtain effective vibration reduction, noise reduction, and performance enhancement on conventional helicopters. Nevertheless, IBC has yet to reach the level of maturity to be implemented on a production helicopter due to the stringent hardware requirements. If, on the one hand, IBC gives more actuation freedom compared to the HHC, on the other hand, it requires a significant amount of power to the actuators in the rotating reference frame by using slip rings. Moreover, the actuators are part of the primary flight control system and, therefore, are located on a critical load path, enhancing their safety requirements. For an in-depth analysis of IBC, readers are encouraged to consult the comprehensive discussions presented in references [268,269].

4.2.1.3. Active mast vibration absorbers. An alternative method to control the rotor vibration consists of counteracting the loads at the hub before they propagate through the rotor mast and gearbox into the airframe. The actuation in this case is located at the hub centre, as schematically shown in Fig. 25c. A traditional hub vibration control strategy uses hub-mounted passive vibration absorbers. These devices typically feature a mass mounted on a spring element, with their resonance finely tuned to the N/rev frequency and minimal damping to present maximum mechanical impedance at the N/rev frequency and effectively absorb vibrational energy. Many commercially available mast vibration absorbers are engineered to diminish the in-plane loads. While these systems can effectively attenuate vibrations at the N/rev frequency, they are often criticised for their relatively high weight and limited operational scope, being most effective at nominal RPM and potentially increasing vibration levels at other RPM regimes.

Since in-plane forces in the rotating reference frame are characterized by $(N-1)/\text{rev}$ and $(N+1)/\text{rev}$ frequencies, which in the fixed reference frame contribute to the N/rev in-plane forces, a hub-mounted active vibration absorber aiming to reduce the in-plane transmitted loads into the fixed frame must produce dual-frequency in-plane forces. A device consisting of two independent active subsystems, each targeting the $(N-1)/\text{rev}$ and $(N+1)/\text{rev}$ harmonics, has been proposed by Lord Corporation, in partnership with Sikorsky Aircraft Corporation, in reference [270]. Fig. 27 schematically shows the subsystem targeting the $(N-1)/\text{rev}$ harmonic, which generates a force, f_{r1} , given by the sum of the centrifugal forces of two coplanar eccentric masses rotating at $(N-1)/\text{rev}$ in a 1/rev

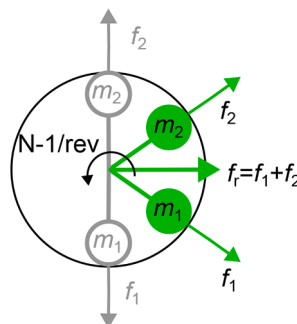


Fig. 27. Schematic diagram showing the exciter targeting $(N-1)/\text{rev}$ harmonic.

rotating reference system. The resulting force is zero if the masses are aligned, (i.e. 180-degree angle shift as shown by the grey masses in the figure). If the relative angle between the masses decreases, the resulting force generated by the subsystem increases reaching a maximum when the shift angle is 0 degree. Adjusting the relative angle between the two masses modulates the amplitude of the resultant force of the exciter, while the phase can be modified by shifting both masses by the same angle. The subsystem targeting the $(N+1)/\text{rev}$ harmonic is identical but has a rotational speed set to $(N+1)/\text{rev}$ in the $1/\text{rev}$ rotating reference system.

In the prototype described in [270], the force generated by one unit is provided by the rotation of two eccentric masses driven by two independent electrical motors. The relative angle between the eccentric masses and their absolute rotational position are used to adjust the modulus and phase of the force produced by each exciter. Fig. 28 shows the installation of the device tested on a Sikorsky Black Hawk helicopter to reduce the cabin vibration measured by accelerometers, with the controller using an adaptive LMS feed-forward algorithm to minimise the mean squared vibration at the blade passing frequency. Fig. 29 compares the acceleration recorded in flight without any control system installed (red bars), a passive control system (blue bars) and the active control system. The plot shows that the active system can reduce the measured acceleration levels in multiple cabin locations and outperform the passive vibration absorber [271].

The Hub-Mounted Vibration Suppressor (HMVS) has been also tested in conjunction with an active vibration control system (AVCS) acting in the fixed frame and discussed in the next section [272]. Other companies have patented analogous technologies [273–275], for example, Leonardo Helicopters designed and tested a prototype of an active mast vibration absorber [276,277]. The fundamental actuation mechanism is again similar to the HMVS, with two identical subsystems, each addressing the $(N-1)/\text{rev}$ and $(N+1)/\text{rev}$ frequencies. Each subsystem is set into rotation by a mechanical transmission connected to the helicopter mast, ensuring the forces generated are consistently at the correct frequency regardless of the angular velocity of the rotor. Individual electrical motors for each of the four masses are used to control the positions of the masses, but not to impose the main rotational speed. The control algorithm is based on Enhanced Phase Locked Loop (EPLL) harmonic tracking to identify the amplitude and phase of the disturbance, with the positioning algorithm allowing the masses to be placed at the correct angular positions, whilst avoiding collisions between the masses. The device has been built and tested on a dedicated test bench, measuring the generated forces and comparing them with the reference forces.

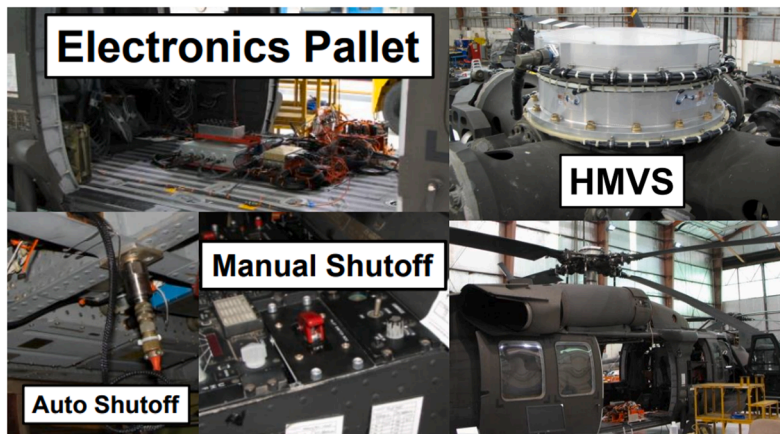


Fig. 28. Hub-Mounted Vibration Suppressor (HMVS) as installed on a UH-60A BLACK HAWK helicopter (From Ref. [271]).

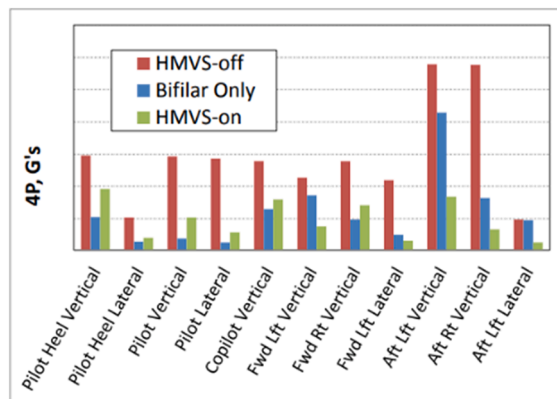


Fig. 29. Hub-Mounted Vibration Suppressor (HMVS) performance (From Ref. [271]).

4.2.1.4. On blade control. The On-Blade Control (OBC) strategy aims to improve cabin vibrational comfort by controlling the vibration of the blades, utilizing actuators mounted directly onto the blades. OBC achieves vibration reduction by altering the unsteady aerodynamic loads on the blades, which directly impacts the vibrational characteristics of the rotor system addressing the vibration at its origin. Fig. 24 lists four distinct implementations of OBC, each achieving varying degrees of maturity: 1) Active Flap Control (AFC), 2) Active Twist Rotor (ATR), 3) Microflaps, and 4) Active Blade Tips. Given the extensive body of research on these control strategies, conducting a comprehensive review of all theoretical, simulation-based, and experimental studies significantly exceeds the scope of this paper. Nonetheless, a selection of illustrative examples for each implementation is provided below for further insight. OBC devices have several advantages compared to HHC or pitch link actuated IBC. They require significantly less power for actuation and allow high-frequency broadband actuation. Moreover, the actuation is not directly located on a critical load path as seen for the HHC and pitch actuated IBC where a failure can directly affect the vehicle airworthiness. Another advantage of the OBC systems is that they can provide additional degrees of freedom as multiple actuation systems can be distributed along the blade, allowing multiple modes to be controlled targeting both vibration reduction, rotor performance enhancement, and noise emission reduction.

Active flaps

The Actively Controlled Flap (ACF) is the most developed among the OBC devices. It features a partial span trailing edge flap, typically positioned near the 0.75 blade span. The foundational concept of using an active flap on the trailing edge of the rotor blade for vibration mitigation originated from research conducted at the University of California, Los Angeles. The capacity of the ACF to significantly reduce vibrations has been confirmed through numerous simulation studies, highlighting its efficacy and potential for vibration reduction [278–281].

From 1992 to 1995, a pioneering project aimed at demonstrating the concept of an active flap rotor through wind tunnel tests was undertaken by McDonnell Douglas Helicopter Systems and NASA Langley's Acoustics Division. The project yielded promising results for vibration reduction, showcasing the potential effectiveness of active flap rotors. However, it also revealed limitations related to the actuation and control system, highlighting areas requiring further development [282–284]. Consequently, following research on AFC focussed on three distinct but complementary streams: actuation, computational simulation development, and practical implementation through wind tunnel and in-flight experimental tests. The first research stream aimed to extend the stroke limits of piezoelectric actuators by exploring various architectural mechanisms. These studies were crucial for achieving the necessary flap deflection angles despite the considerable centrifugal forces experienced by the actuator, as reviewed in [285,286]. The second research stream focused on developing computational simulation models to gain fundamental insights into the behaviour of ACF systems. These models were essential for conducting design and test studies and integrating such systems into practical rotorcraft configurations and explored a range of challenges related to the alleviation of vibrations caused by dynamic stall [287,288]; aeroelastic stability analysis of active flap rotors, considering the effect of the actuator dynamics [289,290]; reduction of Blade-Vortex Interaction noise [291]; simultaneous vibration and noise reduction [292]; optimization of rotors equipped with active flaps for vibration and noise reduction [293]; and integrated design approaches for active flap rotors [294].

Scaled wind tunnel tests played a critical role in the development and validation of Actively Controlled Flap (ACF) technology, despite the complexity limiting the number of experiments conducted. Early scaled wind tunnel models demonstrated the feasibility of ACF in hover and forward flight conditions [295,296]. More refined tests further assessed control authority using trailing-edge flaps actuated by piezoelectric actuators [295,297,298]. The results showed significant reductions in rotor hub loads through neural network control algorithms. Subsequent full-scale wind tunnel tests, such as the DARPA/NASA/Army-funded Smart Material Actuated Rotor Technology (SMART) project [299] and the U.S. Army-led Active Flap Demonstration Rotor test [300–303], demonstrated the effectiveness of active flap systems in reducing noise and vibrations while improving rotor performance. The SMART rotor achieved vibration reductions of up to 95% and blade-vortex interaction noise reductions of 6 dB. Further advancements in ACF technology included Leonardo Helicopters full-scale active rotor demonstrator, developed under the Clean Sky 1 initiative [304], which retrofitted the AW139 5-blade rotor with two distinct technology strands: an active Gurney flap intended to address performance enhancement (briefly described in section iii. Microflap below) and active trailing edge flaps intended to address vibration control [298]. Fig. 30 shows the AFC assembly consisting of composite flap with a foam core (Fig. 30a) and a composite hinge. While Fig. 30b shows a

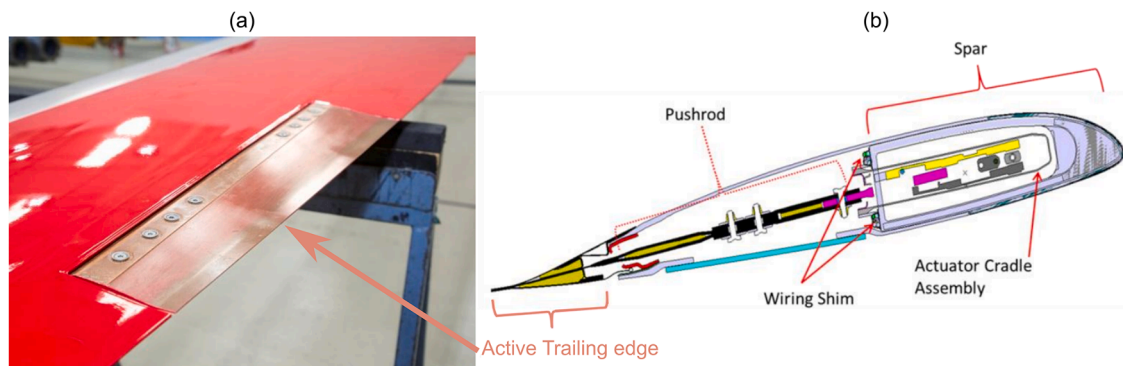


Fig. 30. a) Active trailing edge flap and b) actuation mechanism schematic (From Ref. [305]).

schematic of the actuation where the actuator drives the pushrod and induces an angle change on the trailing edge.

The first flight test by Eurocopter is ADASYS rotor, tested in Germany on a BK117, which demonstrated vibration reductions of 50% to 90% during flight tests [306–308]. These advancements were followed by the Eurocopter Blue Pulse demonstrator, integrating the ADASYS active rotor with a new electronic architecture to improve vibration suppression, noise cancellation, and aerodynamic control [309]. Fig. 31 shows a detail of the active trailing flaps of the ADASYS and Blue Pulse demonstrator. These studies collectively highlight the potential of smart materials and active control systems in enhancing rotorcraft performance, reducing noise, and mitigating vibrations, paving the way for future advancements in actively controlled rotor technology. Despite these advancements, the implementation of ACF on production helicopters has not been pursued due to the complexity of the actuation system operating in harsh environments, such as the rotor blades, and the challenges associated with delivering the necessary power from the fixed frame. Overcoming these technical hurdles will be essential for transitioning ACF technology from experimental demonstrators to widespread operational use.

Active twist rotor

The Active Twist Rotor (ATR) implements distributed twists along the blade by utilizing piezoelectric fibres embedded within the composite structure of the blade. The advancement of the ATR was closely connected with the progress in active materials and structures technology, commonly referred to as smart structures, which occurred in the early 1990s [285]. Kessler provided a comprehensive survey detailing various aspects of the evolution and development of the ATR in [248].

Bent and Hagood at MIT pioneered piezoelectrically actuated fibre composites with integrated electrodes [310], enabling blade twist via embedded active fibres, unlike earlier methods using surface-mounted piezoceramic actuators [311]. This technology provided significantly greater control authority and spurred various ATR research efforts [312–316]. Closed-loop tests showed up to 90% single-component vibration reduction at 4/rev [316], but no further successful testing has been reported since.

An alternative actuation approach based on macro-fibre composites (MFC) motivated a renewed interest in developing an ATR solution in Europe [317,318]. In this case, the active fibres are not integrated into the blade spar but are instead integrated into the upper and lower blade skin. The rationale behind this approach is that the active material can cover a larger area than spar-integrated fibres [315]. The ATR concept is appealing because it eliminates moving parts, guarantees aerodynamic smoothness of the blade surface, and ensures structural efficiency by integrating actuators into the structure. Nonetheless, new challenges have to be faced in the implementation of ATR, as noted in [248]. Firstly, there is uncertainty surrounding the ability of ATR blades to endure prolonged exposure to harsh environmental conditions, considering that composite rotor blades are typically engineered for extended lifespan. Another concern pertains to the maintenance of embedded fibre actuators and repair methods in cases of actuation damage. Furthermore, the cost of the active materials for blade actuation is considerable and the ATR performance in vibration and noise reduction has yet to be demonstrated in flight.

Microflaps

Another OBC approach uses microflaps, or deployable Gurney flaps, which consist of a small tab typically 1.5–2.0% of the blade chord located near the trailing edge of the blade. The microflap is derived from the Gurney flap, originally conceived by Gurney to increase the downward force generated by the spoiler on his race car. Liebeck first studied the aerodynamics of a Gurney flap in a wind tunnel [319], observing the production of two counterrotating vortices behind the flap. Wind tunnel tests and CFD simulations [320–322] indicate that Gurney flaps placed on the lower surface of the airfoil enhance lift and increase the nose-down pitching moment. Microflaps have been studied for various applications such as flutter suppression of high aspect ratio flexible wings, wing trailing edge vortex alleviation, aerodynamic load control for wind turbine blades, rotorcraft performance enhancement, and



Fig. 31. Active rotor ADASYS vs Blue Pulse (From Ref. [309]).

rotorcraft vibration and noise reduction [323–329].

Extensive theoretical studies examining the potential of microflaps for rotorcraft vibration and noise reduction using closed-loop HHC control were conducted at the University of Michigan [328,329]. These studies were conducted on a helicopter configuration resembling the full-scale four-bladed BO-105 hingeless rotor. Single, dual and give-microflap configurations under closed-loop HHC control achieved significant vibration reductions of 73%, 84% and 92%, respectively, demonstrating strong control authority. The excellent control authority of the microflap combined with its size advantage over the conventional plain trailing edge flaps indicates the potential of the microflap as a viable active OBC device for future rotorcraft. Given the small size of the microflaps, their practical implementation can pose various challenges. An ideal actuation system for the microflaps needs to be compact and lightweight with the capability to provide high-frequency actuation with low-power requirements. Various actuation methods based on electromagnetic devices, such as DC motors and voice coil actuators, are under development by various research groups [330,331]. However, these developments are in preliminary stages and need more time to mature into devices suitable for practical rotorcraft configurations.

Active blade tips

A fourth OBC device is known as the active tip, where approximately 10% of the blade tip region is used as an all-movable control surface for vibration reduction as demonstrated in [332]. In this study a small-scale four-bladed bearingless rotor mounted on a Bell-412 hub was equipped with blade tips that can be independently pitched from the main blade using piezoelectric actuation. The basic concept this Smart Active Blade Tip rotor focuses on vibration reduction through an adaptive control algorithm utilizing a neural network. This research demonstrated the technical feasibility of the smart active blade tip concept, achieving over 85% reduction in oscillatory root flap bending of the active blade or fixed-frame thrust. However, the active tip blade is not a mature technology and no further work on this configuration has been pursued. For an in-depth analysis of OBC, readers should refer to the comprehensive discussions presented [248,333].

4.2.2. Active vibration isolation

In the 1990s, Westland introduced the Active Control of Structural Response (ACSR) [334]. This approach consists of isolating the fuselage from the rotor, which represents the major vibration source, by placing actuators between the rotor and fuselage, as shown in Fig. 32a. Using a feedforward control algorithm, a controller drives the actuators to minimize the cabin vibration measured by accelerometers. The ground and flight tests performed on a Westland 30 four-bladed hingeless helicopter are described in [335]. The system consists of four control actuators mounted between the fuselage and the structure supporting the gearbox mounted in parallel to the passive struts, as schematically shown in Fig. 32a. Twenty-four sensors at different control locations inside the cabin were used for vibration measurements. The cost function to be minimized consisted of a combination of the weighted sum of measured vibrations and actuation requests. The open-loop control achieved an 80% reduction from baseline vibration levels. Flight tests on the Westland 30 with the ACSR system operating in the closed-loop mode at various flight conditions were also performed, providing remarkable vibration reduction throughout the whole flight envelope explored. Preliminary tests aimed at implementing an ACSR system on a Sikorsky S-76B helicopter were also conducted at Sikorsky, and a substantial reduction in vibration levels at the blade passing frequency was observed [336]. The ACSR is currently in-service on AW101 helicopters.

Alternative approaches to active isolation of the main gearbox have focused on augmenting the performance of a passive Dynamic

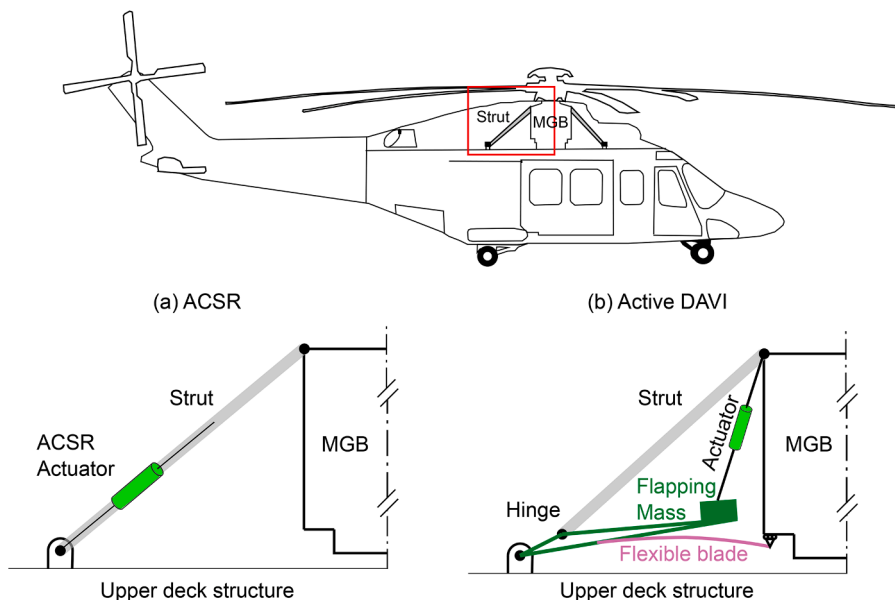


Fig. 32. Schematic diagrams showing the arrangement of (a) ACSR and (b) Active DAVI. The scheme shows the Main Gear Box (MGB) strut installation.

Antiresonant Vibration Isolator (DAVI) [337] with some form of actuation. For example, in [338] a hybrid system is proposed as shown in Fig. 32b, in which an electromechanical actuator is incorporated into the DAVI in order to actively control the suspension transmissibility and enlarge the effective frequency band such that the device can cope with changes in RPM. The main difficulty when designing hybrid vibration absorbers is the choice of actuator technology and integration in order to avoid downgrading the performance of the passive system. The issue observed during the development of the active DAVI shown in Fig. 32b is the introduction of friction due to the presence of the actuator which reduces the effectiveness of the passive system. However, simulations coupled with the results obtained experimentally confirmed the efficiency of a resonant vibration absorber with dynamic amplification tuned using an active controller to minimise the force transmitted to the fuselage via the MGB strut for the 15–20 Hz bandwidth. Moreover, using a resonant vibration absorber to achieve control at frequencies around its nominal resonance frequency gives a clear advantage in terms of power consumption compared to classical active vibration control systems such as the ACSR (with the weight penalty of the suspended mass). It has, however, been reported that the proposed active DAVI needed more development to provide a lower passive friction solution before it could be tested in flight.

4.2.3. Active vibration control at the receiver

Active vibration control systems designed to reduce cabin vibrations directly at passenger and crew locations have been developed and implemented in production aircraft. As shown in Fig. 33, these systems consist of actuators distributed around the helicopter cabin fed by a controller, which is tuned to minimize the overall vibration measured by distributed sensors at the blade passing frequency using the rotor speed as a reference signal. Lord Corporations has developed this feedforward control system, which uses both linear and circular actuators and the FxLMS algorithm to implement control [339]. Leonardo Helicopters install these systems on AW139, AW169 and AW189 for enhanced vibration comfort. As an example, Fig. 34 compares the controller On and Off conditions in two-level flights at 30 and 160 KIAS. The graphs show that the active control system can reduce the vibration globally inside the cabin for both flight conditions.

4.3. Active vibration control in aeroplanes

Like helicopters, aeroplanes are also affected by significant vibration disturbances, which arise from a number of structure-borne and air-borne sources. The NASA Technical Report by Hubbard [340] provides a full account of the many vibration excitation mechanisms that develop in aircraft, which for passenger transport vehicles can be categorised into those due to the propulsion system and those due to either the aerodynamic or aeroelastic responses of the fuselage. These various vibration excitation mechanisms generate periodic tonal disturbances, as well as both narrowband and broadband stochastic disturbances. Readers interested in further investigating these noise and vibration generating mechanisms are directed to specialist publications, for example [65,340–346]. The following subsections present practical examples where active vibration control systems have been developed to mitigate the effects of specific structure-borne and air-borne disturbances and transmission mechanisms, as well as the implementation of active systems for vibration control at receiving subsystems in aircraft, such as delicate equipment and sensors.

4.3.1. Active vibration control at the source

The active control of vibration generation in aeroplanes has been focussed on two areas of the engines. Firstly, the control of imbalance effects and the flexural response of the fast-rotating shafts, and secondly, the vibration control of propeller or fan blades. Considering first the control of shaft vibrations, Vance *et al.* [347–349] investigated how electro-rheological dampers could be exploited to implement active vibration control in turbo engines. As shown in Fig. 35a, the fast-rotating shafts of turboprop and turbofan aircraft engines are supported by rolling bearings, which generate some level of squeeze film damping, that is normally strengthened by specific squeeze film damper components. Nikolajsen and Hoque [350,351] were among the first to show that Electro-Rheological (ER) dampers can be effectively used to develop controllable bearing dampers. As shown in Fig. 35b, these

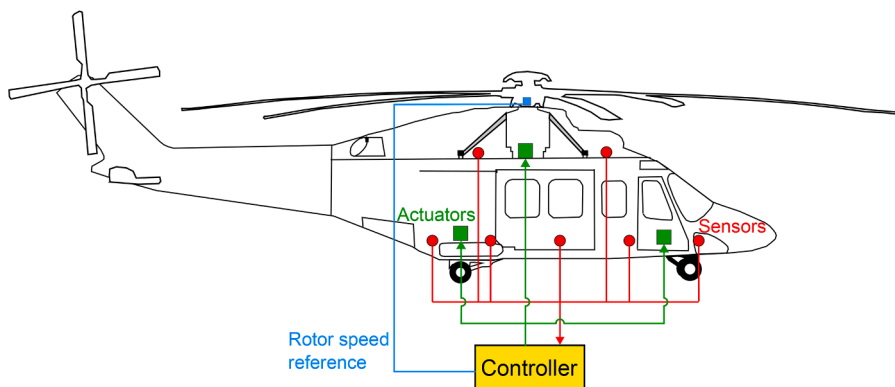


Fig. 33. Schematic diagram showing the arrangement of actuators (green), error (red) and reference (blue) sensors for active vibration control system.

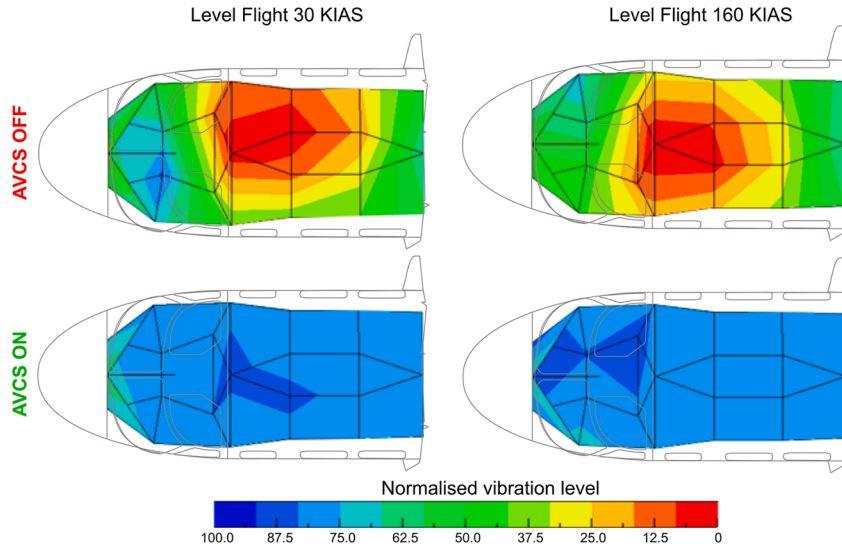


Fig. 34. normalised acceleration map in the cabin with the active vibration control system OFF top maps and ON bottom maps for two flight conditions: level flight condition at 30 KIAS (left maps) and 160 KIAS (right maps). (Courtesy of Leonardo Helicopters).

dampers rely on fluids that under the action of an electric field can convert a Newtonian fluid to a Bingham plastic. They showed that the voltage set across the disks of the damper can be used to actively control a Coulomb-type friction damping effect. Overall, they demonstrated experimentally that the ER damper can significantly reduce the peak rotor vibration at low critical speed. Moreover, the actively controlled ER damper can be employed to change the effective stiffness of the system at specific rotor speeds during run-up. The ER damper was selected as the most suitable active damper from seven active damper concepts: 1) Squeeze-Film Dampers SFD, 2) Hydrostatic SFD, 3) Electro-Rheological (ER) damper, 4) Piezoelectric Pusher Damper, 5) DC Electromagnetic Damper, 6) AC Electromagnetic Damper and 7) Gas Damper. Ref. [349] provides a detailed overview of the operation principles and of the technical advantages and disadvantages of all of these solutions, while Fig. 35c,d shows the operation scheme for the DC electromagnetic damper and the Eddy current damper. Overall, the ER dampers were found to be the most appropriate solution. In general, the ER devices can be operated to generate parametric control where the viscosity of the damper is adapted at regular intervals of time with reference to intermittent measurements of the rotor vibration (for example every rotor revolution) or via real-time feedback control

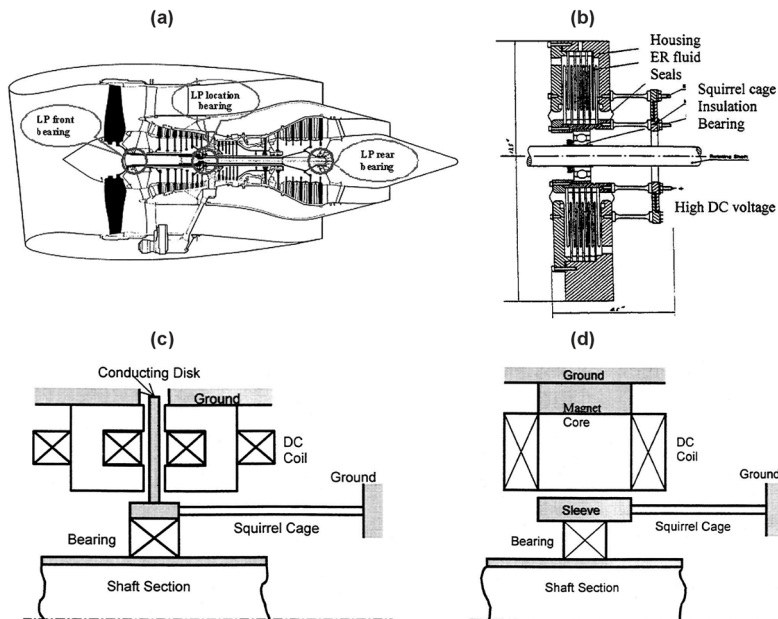


Fig. 35. a) Example of a gas turbine engine with LP front/location/rear bearings, b) Cross section of the ER damper studied in Ref. [350], c) DC electromagnetic damper operating principle. d) Eddy current damper operating principle (picture (a) from Ref. [353], (b, c, d) from Ref. [349]).

where the device produces a damping force proportional to the vibration of the rotor.

Beside ER dampers, Active Magnetic Bearings (AMBs) have also been considered for the active vibration control of high-speed rotors. These systems can be very effective in controlling rotor vibrations, but they present quite a significant limitation in terms of the static and dynamic stiffness they can generate. With AMBs there is no need for the conventional rolling bearings, which constitute both part of the vibration generation mechanism and the main vibration transmission path. However, the load carrying capacity as well as the stiffness and damping effects are produced by the electromechanical interaction. In particular, the load carrying capacity is linked to the mass and size of the coils necessary to generate the magnetic fields, which at present are too large for practical applications. Nevertheless, they offer some appealing advantages too, such as the lack of fatigue and tribology issues that arise from contacts, the low sensitivity to operational conditions (e.g. temperature, loads, etc.), the wide range of tuneability and the robust open loop response. Kasarda *et al.* [352] proposed an interesting solution using electromagnetic dampers, where the load carrying capacity is ensured by retaining the conventional rolling elements. Here, the electromagnetic damper is installed in parallel with a classical squeeze film damper encompassing a squirrel cage device and is operated by a proportional-derivative feedback vibration control system that produces the desired stiffness and damping vibration control effects. Tonoli *et al.* [353] brought forward this idea and presented a feasibility study for both a fully active and a semi-active version of the magnetic damper [354] suitable for aero-engines using a position feedback control loop.

An alternative route to actively mitigating vibration in aircraft engines shafts due to imbalance has also been proposed, which, in place of the squeeze film dampers, uses piezoelectric stack actuators at the bearing locations [355–358]. Borsdorf *et al.* [358] considered an integral force feedback control to drive the piezoelectric stack actuators [68,359], while Simoes *et al.* [357] employed a LQR optimal controller. Significant reductions in the vibration induced by imbalance under gyroscopic effects was demonstrated in [355] and future work was proposed “towards implementation in a whole aircraft engine involving certification processes for aviation applications”. More recent fundamental research has focused on the implementation of active control systems based on the flexible deformation of the rotor rather than its absolute displacement [360] and Zhao *et al.* [361] have explored the use of piezo-based rotating inertial actuators for the suppression of vibration from rotating machinery using a feedforward FxLMS controller.

The bladed compressor and turbine stages of both turboprop and turbofan engines are characterised by very harsh loading conditions, which may result in high levels of vibration on the whole bladed disks that could lead to damage of the blades. This is a relevant problem, which may lead to significant imbalance effects that could even result in the mechanical failure of the whole engine. This problem has become even more relevant with the advent of integrally bladed structures that guarantee significant weight savings but display a much lower damping effect since there is no friction at the blade mounting point. For this reason, specially dedicated friction damping solutions must be employed at the roots of the blades, or between the blades themselves or at the shrouds [362,363]. Despite their simplicity, these solutions are not reliable as the friction damping may vary significantly depending on load and temperature variations, as well as wear and aging effects. In principle, viscoelastic fillers could be integrated into the structure, but this presents some inherent problems related to temperature and wearing effects, which nevertheless can be overcome to some extent with polymeric, metallic or ceramic oxide coating [364]. More recently, the idea of using shunted piezoelectric elements to mitigate the vibration associated with the ringing of blades has been investigated. For instance, Tang and Wang suggested bonding shunted piezoelectric patches on each blade, whose effect was augmented with a charge current feedback loop [365]. Several studies followed this idea, where, as depicted in Fig. 36, piezoelectric patches connected to fully passive or active shunts were employed either on the supporting ring structure or directly on the blades [366–374].

In general, the piezoelectric patches are connected to shunts formed by series or parallel Resistive-Inductive (RL) electric components [375,376], which together with the capacitive (C) effect of the patch generates a RLC resonant circuit. This RLC electric

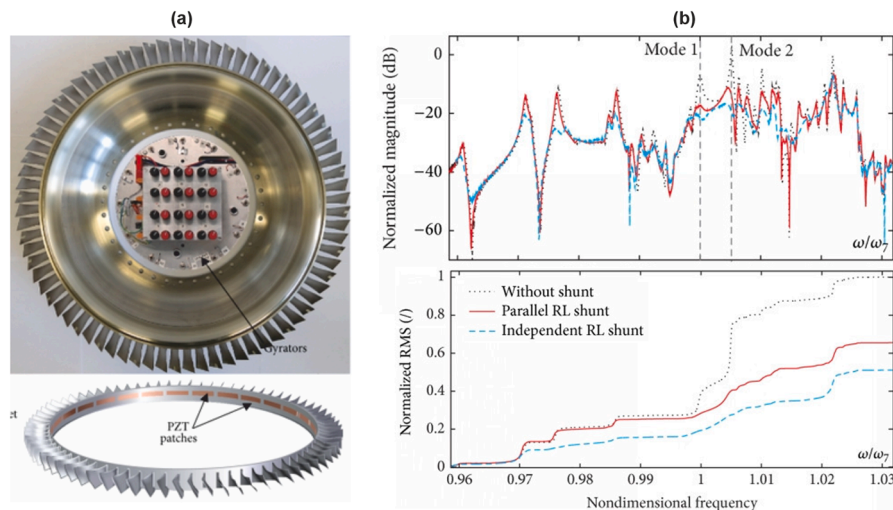


Fig. 36. (a) Aircraft turbine element equipped with shunted piezoelectric patches on the bladed drum. (b) Narrow band spectrum and frequency-cumulative RMS of the flexural vibration of one palette. (From Ref [370]).

loading is converted by the piezoelectric patch respectively into concentrated damping-inertia-stiffness loading effects [377,378], which resemble classical spring-mass-damper Tuned Vibration Absorbers [379]. Hence, the shunted piezoelectric patch can be used to control the resonant response of a target mode of the structure [380]. Here, the resonance frequency of the shunt should be tuned to the resonance frequency of the target mode and the energy dissipation should be maximised [381]. In general, the higher the mass of a TVA, the higher its vibration control performance [380]. Hence, for a shunted piezoelectric patch, a higher shunt inductance will give a higher vibration control effect [381]. However, having large inductances presents two problems – firstly, large passive inductors require unreasonably bulky (i.e. heavy and large) solenoid electrical components [382]; and secondly, since the resonance frequency of the shunt must match the excitation or resonance frequency of the host structure, the capacitance would need to be very small and typically much smaller than the inherent capacitance of the piezoelectric element, which must therefore be compensated for [381]. These two problems can be circumvented using active shunt circuits, which use operational amplifiers to synthesise the large inductance [383] and the negative capacitance [384]. Both fully analogue [383] and digital [385] active shunts have been studied. These circuits require a power supply and should be carefully designed and operated to avoid stability issues; nevertheless, they offer the possibility of developing self-tuning shunts [386–389], which can track time-varying tonal excitations or changes in the resonant response of the host structure, as well as changes in the electric response of the shunt itself [390–392]. Although these systems do not comply with the classical architecture of active systems, which encompass distinct sensor, controller and actuator components, they operate as if they were active feedback systems with self-sensing actuators [393] and they even encompass self-tuning properties that resemble the adaptive architectures of active systems [394].

4.3.2. Active vibration isolation

In addition to the active control systems conceived to directly address the imbalance and flexible response of the fast-rotating shafts and the vibration of the turbine and compressor elements of turboprop and turbofan engines, significant research has also been dedicated to the development of active mounts to isolate the structure of the aircraft from the vibration of the engine. Aircraft engine mounts must in general accomplish two tasks: a) to constrain rigid body motion due to weight, propulsion torque and thrust and manoeuvring g-loads; b) to provide vibration isolation at the structural mounting points as well as at junctions with auxiliary components. For the first task the mounting system should be statically rigid, whereas for the second task it should be dynamically very soft [164,395] and it is generally necessary to opt for a compromise between these two requirements, whilst also ensuring long service life and low production, installation and maintenance costs. These contrasting requirements cannot be easily fulfilled with classical elastomeric [396–398] or wire-mesh [395] mounts and they can only partially be met with fluidic mounts [399–402]. Thus, there has been a strong interest in alternative solutions, including active mounts [166,178,403–410].

Normally, for tonal disturbances, such as those produced by imbalances of the rotor assembly or by the blades of the compressors, turbines, propeller, or fan in turboprop and turbofan engines, feedforward adaptive controllers are employed [4,58,59,178]. These controllers rely on a reference signal, taken from the revolution speed of the engine shaft using a tachometer, and use accelerometers or force vibration error sensors [178,411,412], which are normally located at the base of the mount and used to adapt the control filter. Less frequently, full feedback controllers have also been employed, which could in principle be used to control broadband stochastic disturbances, but normally provide lower control performance for the tonal components of the engine excitations [59].

Quite a few active mount technologies have been conceived and studied over the years (e.g. see Refs. [169,180,181]). However, despite the simplicity of the proposed solutions, their practical implementation has not been straightforward, primarily because of a) force/stroke limitations of the transducers [169,180,181], b) the need for bulky and heavy transducers [167,180,189,413], c) the high levels of electrical power consumption and d) the comparatively higher installation and maintenance costs. Nevertheless, a successful active mount for aircraft applications was developed at Lord Corporation [166,167,403,405,407,408], which, as shown in Fig. 37a, encompasses a classical fluidic mount equipped with an electro-magnetic (coil-magnet type) actuator that excites the fluid in the chamber so as to reproduce the inertial active force effect depicted in Fig. 20e.

In general, a classical resilient mount would offer elastic and dissipative effects, which generate a dynamic stiffness that, as shown by the solid green line in Fig. 37b, in contrast to vibration isolation requirements, tends to grow with frequency. Instead, a fluid mount

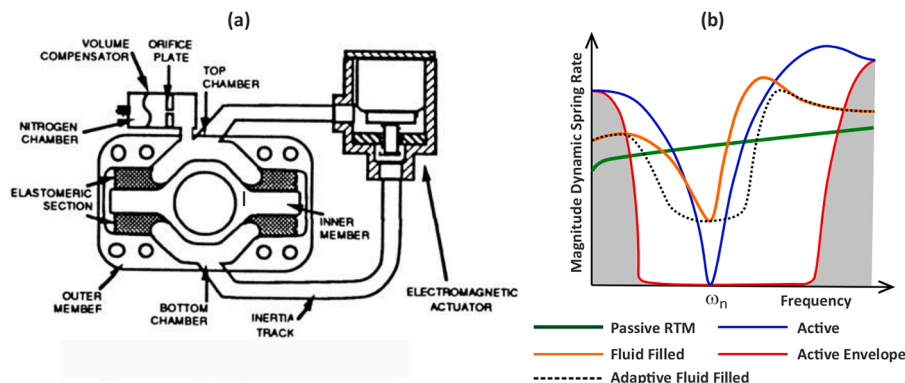


Fig. 37. (a) Active fluidic mount (from Ref. [403]). (b) Dynamic stiffness of passive, semi-active, active mounts (adapted from Ref. [403]).

would produce a tuned mass damper effect in parallel to the resilient mount, such that, as depicted by the orange line in Fig. 37b a narrow-band notch effect in the dynamic flexibility is generated [399–401]. As highlighted by the back dotted line, this notch effect can be swap over a given frequency band. More importantly, as highlighted with the blue line and gray-shaded region delimited by the red line, the notch effect can be both enhanced and tuned over a wider frequency band through the electro-magnetic actuator, such that the active mount can effectively reduce the vibration transmission over a wide range of rotational speeds [166,167,403,405,407, 408]. Compared to classical viscoelastic mounts, these mounts can provide a comparatively high static stiffness and yet have a much lower dynamic stiffness at the notch frequency.

As shown in Fig. 37a, the mount proposed by Lord Corporation [166,167,403,405,407,408] is realised by connecting a coil-magnet actuator to the fluid chamber of a classical fluidic mount to control the pressure on the fluid and thus tuning of the notch effect. More specifically, as described in Ref. [403], the inner part of the mount can be moved relative to the outer part by controlling the pressure in the top and bottom chambers. This pressure differential is set by the small piston of the coil-magnet transducer that produces a force excitation proportional to the current fed to the coil. Actually, this force effect is magnified since the effective piston area in the mount is many times larger than the piston area of the vibrating coil piston. The coil piston is mounted via a spring such that, together with the stiffnesses offered by the fluid in the two chambers of the mounts it produces a resonance that lowers the dynamic stiffness at the resonance frequency and the actuator power requirements. The frequency band of the notch effect can be raised or lowered by the controller within the limits highlighted by the envelope. In this way the active mount can implement the so-called frequency-shaped control [168] to successfully prevent the vibration transmission of time-varying tonal excitations, which is indeed the case for aircraft engines that operate at given rotation speeds. More specifically, the active envelope depicted in Fig. 37b is obtained by adjusting the magnitude and phase of the current to the transducer coil with a feedforward controller that exploits the reference signal from a tachometer mounted on the engine shaft, whose filter is continuously adapted in such a way as to minimise the force measured by the error sensor mounted at the base of the mount. This will allow lower transmitted dynamic forces while maintaining a high static stiffness, resulting in lower vibrations in the structure. As reported in [403], this active mount has been designed to be compatible with the isolation requirements for turbofan engines mounted on commercial aircraft such as a Boeing 737 and 767, Airbus A320, McDonnell Douglas MD80. The details of specific applications studies can be found in Refs. [403,405]. A key feature of this active mount is that it can provide the large forces and strokes necessary to isolate the vibration transmission of large turbofan or turboprop engines with a relatively low power supply. In general, as can be noticed in the graph of Fig. 37b, the mount can reduce the dynamic stiffness within the control envelope by 100 times, that is 40 dB, with respect to the static stiffness. The mount could be actuated hydraulically too, by replacing the coil-magnet transducer with a hydraulic actuator. In this case the active mount can bear much higher loads, although it would need a servo-hydraulic actuation setup, which, nevertheless, is bound to use a larger amount of power due to the intrinsic inefficiency of the servo-valves.

Overall, Miller and Ahmadian [403] listed the following advantages of active mounts for engine vibration transmission to the airframe of aircraft:

- 1) reduced noise and vibration, which result into a greater customer satisfaction, lower maintenance costs, weight savings by reducing amount of vibration and sound insulation treatments;
- 2) higher static stiffness, which results into longer isolator lifetime, lower low frequency motion of the engine, better engine alignment, reduced engine packaging and use of a common isolation system for multiple types of vehicles;
- 3) adaptive algorithms provide a self-tuning capability, which can adapt the active mount to changes in the dynamic response of the airframe and engine due to exogenous effects (e.g. variations of temperature, pressure, payload, fuel, age, etc.).
- 4) the ability to compensate vibration transmission through flanking paths such as thrust and torque links, which ensures standard control effects for a wide variety of engines, installations and aircraft vehicles.

In parallel to active mounts, adaptive tuned vibration absorbers have been employed too to control the vibration transmission at the footprints of hard joints [32]. Although these systems are not characterised by a fully active control architecture [414], in practice they closely replicate the action of an inertial actuator except that the mechanical properties of the spring-mass system are continuously adapted by the controller in such a way as to set the system to resonate at the excitation frequency and thus maximise the vibration absorption. Practical systems were successfully developed to isolate the vibration transmission of engines mounted in the aft of the aircraft cabin [415,416].

4.3.3. Active vibration control at the receiver

As noted in the introduction to this section, the fuselage structures are affected by many structure-borne and air-borne excitations, which are due to the engines vibration and landing gearbox shocks transmitted to the fuselage airframe structure via the wings, and to the engine noise the aero-dynamic excitations that develop directly on the skin of the fuselage structure. These vibration effects can be an issue for delicate electronic equipment increasingly used for aircraft operation and control, particularly during shock events associated with landing (e.g. see review paper in [417]). While classical vibration control solutions and specially designed robust electronics have been applied to protect sensitive electronic equipment in aircraft, active vibration control of these systems has been explored. For example, Aglietti *et al.* [418–420] investigated the possibility of equipping PCB boards or their enclosures with active vibration control systems employing piezoelectric actuators. Alternatively, Kim *et al.* [196] developed the decentralised feedback control units shown in Fig. 38a, which employ reactive electro-magnetic actuators [169,172–174], and could be used to minimize the vibration transmitted to small bodies such as the racks encasing delicate electronic circuits [196,421]. These units implemented direct velocity feedback control [169,199] in such a way as to generate an active damping effect on the box structure [95,169,200], which is

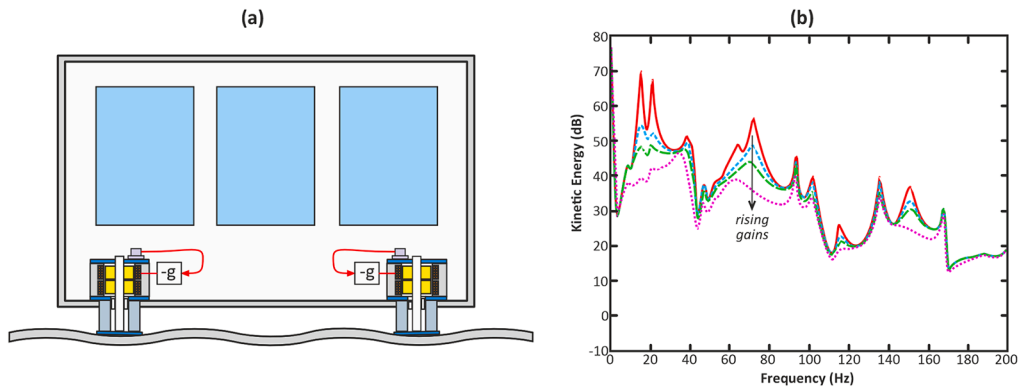


Fig. 38. (a) Experimental apparatus composed of a base vibrating structure and a rack with DSP boards on four decentralised velocity feedback vibration control units using electro-magnetic actuators. (b) measured vibration kinetic energy of the rack with respect to incremental feedback control gains g in the four units (adapted from Ref. [422]).

ideal for broadband vibration control, in particular for the control of shocks. Although the reactive force actuation does not ensure full collocation and duality with the sensor located at the end connected to the board [196,201–204], the proposed design proved quite robust to local and cross talk instability effects such that rather high feedback gains could be implemented in each unit so that significant vibration reduction of the case structure could be achieved, as can be seen in Fig. 38b [196,421].

4.4. Active vibration control in cars and trains

Like all forms of passenger transport vehicle, excessive vibration in land vehicles is undesirable both in terms of excessive exposure of passengers to direct vibration, and in terms of exposure to noise through radiation from vibrating structures. Although there are many forms of land vehicle, we focus here primarily on applications of active vibration control to cars and trains, since they have formed the core of the research in this area. In both cases, to varying degrees of importance, vibration sources can once again be categorised as the onboard, including powertrain and auxiliary machinery components, such as Heating, Ventilation and Air Conditioning (HVAC), and outboard, including vibration due to road-tyre interaction or wheel-rail interaction, and aerodynamically induced structural vibration. Control of these various vibration sources will be covered in the following subsections, taking the same structure as in the previous sections.

4.4.1. Active vibration control at the source

Active control at the source is generally the most effective mechanism of controlling noise and vibration throughout a structure or environment. However, due to the complexity of real-world vibration sources (such as an internal combustion engine) it is often not feasible and instead focus has been paid to isolating the vibration sources or controlling at the receiver location, and these two approaches are discussed in the following subsections. For completeness, however, we will discuss some specific applications where control of vibration at the source has been considered in the context of automotive and railway applications.

One area where active vibration control at source has been explored, which has links to the same problem in the aircraft application discussed in Section 4.3 is the control of aerodynamically induced vibration of the vehicle exterior skin panels. Aerodynamically induced vibration of vehicle panels is a significant contributor to interior cabin noise when the vehicle is travelling at higher speeds and, therefore, is of interest in automotive applications and high-speed rail applications. This problem has, for example, been explored in [423,424] from a fundamental perspective through an investigation into the active control of TBL noise transmission through a cavity-backed double panel. The proposed system utilises a velocity feedback control approach to control the vibration of the double panel system and thus mitigate against noise transmission into the cavity and provides insight into the physical limitations of the control approach determined by the dynamics of the controlled system. A more practical investigation of a similar system has been presented in [403], where either modal control or proportional feedback control using a PVDF actuator and collocated accelerometer are utilised to reduce vibration in car body panels as shown in Fig. 39. Despite the potential performance of actively controlling car body panels to reduce, for example, TBL excited vibration in automotive and rail applications, it does not appear to have been as widely explored as in the aircraft application, perhaps due to differences in passenger expectations, but more likely due to the high cost of the technology which would require multiple actuators and sensors to be distributed across the exterior skin panels.

Another area where active vibration control at source has been explored is in the control of HVAC units, which are now ordinarily installed in land transport vehicles. HVAC units in commercial or rail vehicles can be quite large and can cause annoying interior and exterior noise as well as significant structural vibrations. In order to reduce vibration transmission from a HVAC system compressor to the bodywork of a city public transport tramway and the structure-borne noise in its interior, active tuned vibration absorbers were designed, [57].

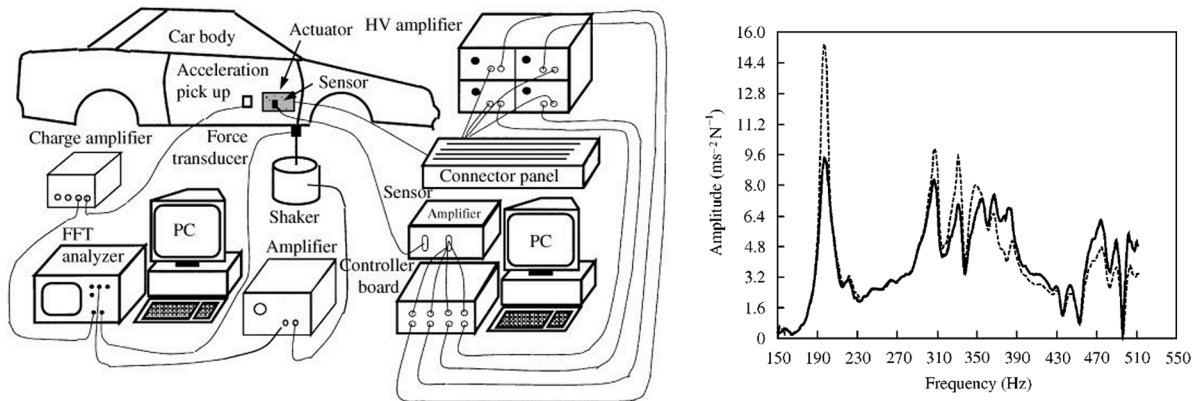


Fig. 39. System for the demonstration of active vibration control of a car body panel and an example of the measured FRF before (dashed) and after control (solid) (From Ref. [425]).

4.4.2. Active vibration isolation

Active vibration isolation has typically been applied in two key areas in the automotive environment: isolation of powertrain vibration, and isolation of road-induced vibration. In the rail sector it appears that less work has focused on active powertrain isolation, perhaps due to its distance from the passengers or perhaps due to the more widespread usage of electric power, but active isolation has been applied variously to railcar suspension systems.

Active vibration isolation of internal combustion engines has focused on the realisation of active engine mounts, which help to reduce the transmission of engine-induced vibration into the chassis of the vehicle. Significant research has focused on this problem, with the first system described in 1986 [427], and commercial systems being utilised since the 1990s. For example, in 1998 Nissan utilised two electromagnetic engine mounts to reduce vibration in a four-cylinder direct injection diesel [426,428]. This system utilised a feedforward control strategy that used the rotational speed of the crankshaft measured with a tachometer as a reference signal. The active suspension is driven in such a way as to minimise both vibration at the connection points (measured with accelerometers) and noise inside the car (measured with microphones placed near the passenger seats). As with all active mounts, the system is combined with a passive mount, typically realised using rubber and Fig. 40 shows the active engine mount developed by Nissan. Fig. 41 shows the noise reduction measured inside a car equipped with this type of active engine mount at the most critical 2nd and 4th engine orders under the most critical operating conditions in each case. Similar systems have also been developed or utilised by Honda [34], as well as Jaguar and Audi.

In a similar vein to active engine mounts, significant research has been paid to active suspension designs, which can be designed to address ride and handling, but also vibration and, therefore, noise transmission into the passenger cabin. In this case, since the vibration has a broadband stochastic characteristic and a straightforwardly available reference signal is not available, feedback control design strategies are typical utilised. The first active suspension system was developed by Lotus [429] and applied to their F1 racing car and subsequently the Esprit experimental vehicle [430]. However, as noted in the review of advanced suspension systems by Hrovat [430], these systems were based on early contributions to optimal control in the aerospace industry and early investigations into active suspensions using single degree of freedom and quarter car models [431]. The Lotus developed system primarily focused on ride and handling requirements, however, later systems have also explored the vibration and consequent noise control aspects. For example, in

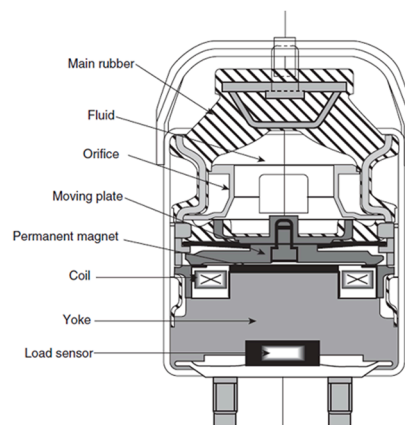


Fig. 40. Components of an active engine mount described in [426] (From Ref. [139]).

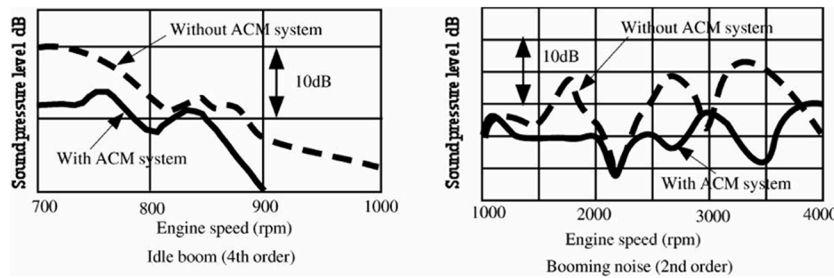


Fig. 41. Sound pressure level measured in a car with a diesel engine fitted with the active engine mounts shown in Fig. 40 (From Ref. [426]).

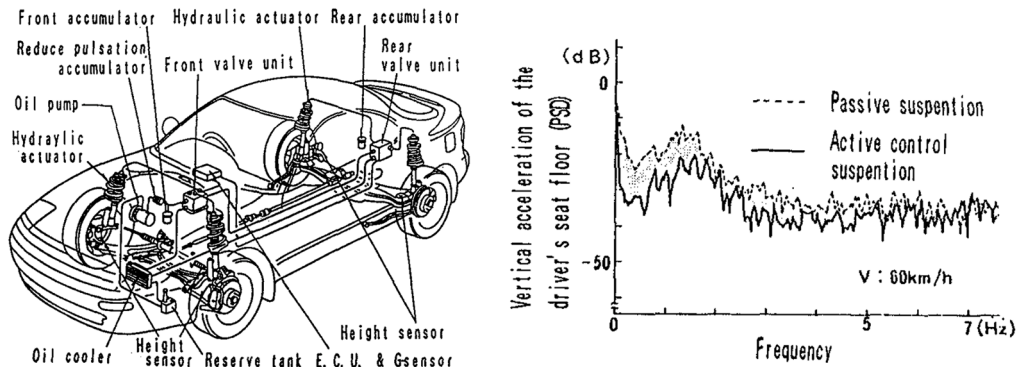


Fig. 42. Active suspension system developed by Toyota showing an overview of the system and the low frequency vibration performance in terms of the acceleration measured at the driver's seat floor (From Ref. [433].)

[432] an active suspension system utilising hydraulic actuators and a robust feedback control strategy is utilised to reduce body displacements and in the early 1990s both Toyota [433] and Nissan [434] produced commercial systems that showed vibration reduction. An overview of the Toyota system is shown in Fig. 42 along with a measurement showing its low frequency vibration control performance. A more recent review of active suspension systems is presented in [435], which discusses a range of commercial systems being produced by Cadillac, Michelin, Mercedes, Audi and Bose, demonstrating the widespread adoption of this technology.

In the rail industry, active isolation has also been widely investigated over a similar period, utilising similar approaches to the automotive application discussed above. In 1997 an initial study was presented that demonstrated how an active vibration isolation system applied to a German Intercity Express train railcar was able to achieve significant noise attenuation within the railcar cabin [35]. The active control system in this case used shakers driven to minimise the pressures measured inside the railcar cabin operating in combination with the passive suspension system. Interestingly, compared to the common use of feedback control in the automotive application, a feedforward control strategy was utilised in this case due to the primary excitation being related to the higher harmonics of the wheel rotational speed. Despite demonstrating high levels of noise control performance, this system was not tested on-track, with the primary disturbance instead generated using an additional shaker. Many other active suspension systems have been reported in the literature, as discussed in [36], however, most of these works take a model-based or test-rig approach to exploring the problem. On-track testing of an active suspension system has, however, been reported in [37]. This work reports the implementation and testing of a skyhook damping control strategy, utilising a hydraulic actuator for control, with the active system operating in parallel with a conventional air spring. Compared to a hydraulic mount, the active mounting system is shown to offer significant improvement to the measured comfort level defined according to [436].

4.4.3. Active vibration control at the receiver

As in all passenger transportation applications the, the ultimate receiver is the passenger. Although not common in cars, active seat installations have been widely developed for heavy goods vehicles and similar applications [437–440]. As shown in Fig. 43, these systems utilise some form of actuation between the floor and the seat in order to reduce the vibration experienced by the seat occupant. It is important in these applications to consider the dynamics of the human body when designing the active control system, hence the inclusion of a dummy during testing in Fig. 43. These active seat systems have been designed using both feedback [437] and feed-forward [438] control strategies, with a wide variety of actuators including electromagnetic linear actuators [438], rotary motors [439] and electrohydraulic actuators [440].

A more niche, but interesting application of active vibration control to the receiver problem in the automotive context focuses on controlling steering wheel vibration [441]. This system aims to reduce the engine idle vibrations transmitted to the steering wheel via the steering column using piezoelectric actuators and sensors arranged in a feedback control structure. Although only presenting

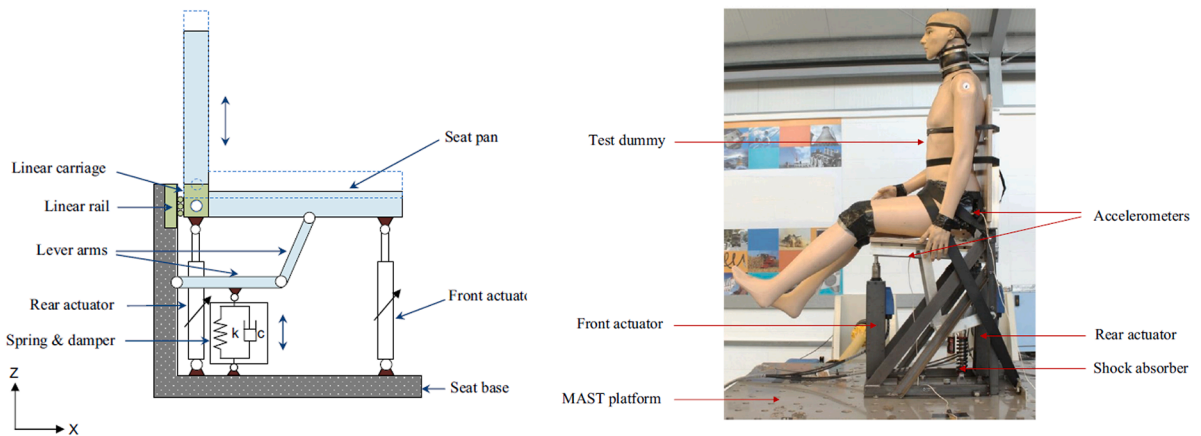


Fig. 43. Diagram and experimental realisation of an active vibration control seat design (From Ref. [438]).

numerical results, significant performance is reported, but the cost of the active system is noted as a significant barrier in this application. However, since this work was published in 2005, it is not uncommon for steering wheels to be actuated to provide haptic feedback to the driver [442] and this opens future opportunities for this type of technology to be more easily adopted.

Another interesting application of active vibration control in automotive industry, which can also be considered as control of the receiving structure, is related to deficiencies in the vibroacoustic characteristics of convertible cars (cabriolets). The principal deficiency in this case is the reduced torsional stiffness of the bodywork in comparison to sedans, which reduces the typical natural frequency of the fundamental torsional mode from about 50 Hz in a sedan, to about 20 Hz in a convertible [57]. This is important because the driver's comfort is strongly related to the dynamic stiffness of the car bodywork at frequencies below 30 Hz. Passive stiffeners or vibration absorbers can be utilised, although they add significant weight to the vehicle. An active approach to reduce torsional vibrations is to integrate actuators within the diagonal stiffening struts of the car underbody and significant vibration reductions have been reported using various actuation and control approaches. However, to commercialize this active concept, system cost, size, complexity and power consumption must be further reduced.

4.5. Active vibration control in maritime transportation

Active vibration control in the maritime passenger transport environment is generally a more recent exploration than in the other modes of passenger transport considered in this review. This is possibly due to the natural harshness of the ocean environment and therefore the higher threshold for acceptable noise and vibration performance or due to the cost of implementing these active systems. However, predominately this century, various strategies have been explored and developed for the active control of vibration onboard marine vessels. Key applications of active vibration control to maritime transportation will be reviewed in the following sections, focusing on control at source, isolation, and control at receiver.

4.5.1. Active vibration control at the source

Active control of vibration at source in the maritime transportation sector has primarily focused on controlling vibration related to the propeller shaft. This is probably due to the recognition that longitudinal vibrations of the propeller shaft are a primary mechanism

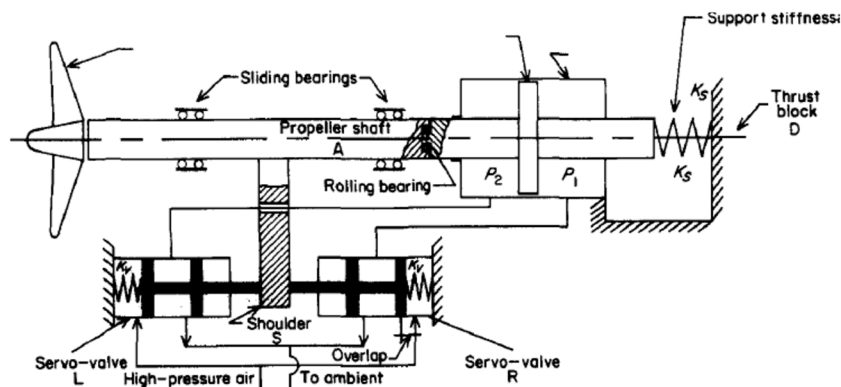


Fig. 44. Diagram of an active control system for propeller shaft vibration control (From Ref. [39]).

for the excitation of the hull of the ship [443] and onward radiation of noise into the ocean, disturbing the marine environment, and throughout the vessel, disturbing its occupants. It is interesting to note that this application of active control differs from many of those applications discussed in this review article, in that it not only aims to improve the noise and vibration environment for the occupants, but also to reduce the noise radiated into the wider environment. This distinction perhaps relates to the fact that the radiated noise can be directly related to the key excitation mechanism provided by propeller shaft vibration, which can be actively controlled, or perhaps to the fact that the acoustic wavelength in water is more than four times longer than in air for the same frequency, which helps with the pressure matching requirement associated with active noise control. Nevertheless, active vibration control of the propeller shaft is one of the earliest documentations of active control in the maritime transportation sector [39,444]. The motivation for active control in this application is consistent with all other successful applications – passive control alternatives are limited in frequency range due to their fixed tuning, they are limited at low frequencies and they add considerable weight [39].

Although the first active treatment for the propeller shaft vibration control problem was proposed in [444], this was a purely theoretical study and as noted in [39] was subject to significant practical complexities and cost due to the use of actively controlled magnetic bearings. The active propeller shaft vibration control system proposed in [39] is shown in Fig. 44 and is realised using a self-contained pneumatic servo-controller powered by high-pressure air. The system is shown to actively resist longitudinal vibration in the propeller shaft using a scale-model and achieves significant reductions to both step and sinusoidal excitation of the shaft.

Since the first proposal to actively control propeller shaft vibration, there has been a very large number of publications related to this problem application, for example [445–451]. These publications have explored different actuation mechanism, including electromagnetic [447] and piezoelectric [445] based mechanisms, and different control strategies, including feedback [445–450] and feedforward [451] approaches. Model-based studies have been extensively explored [446–449,451], providing new physical insight,

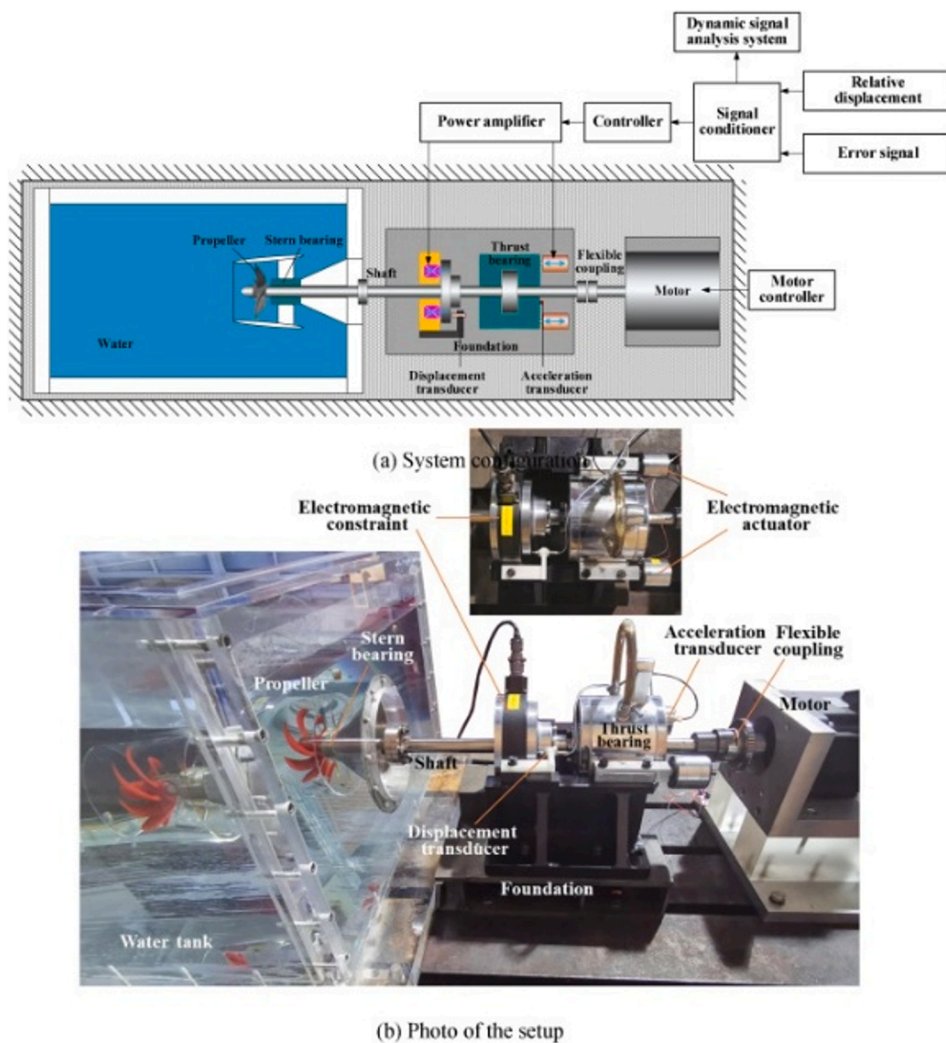


Fig. 45. Diagram and photograph of an experimental setup used to assess the performance of an active control system for propeller shaft vibration control (From Ref. [449]).

and variously more elaborate laboratory-based experimental studies have also been presented [446,448–451]. For example, although presenting a model-based study, [447] explores the effect of realistic ship motion on propeller vibration and its control. In terms of experimental validation, there have been various levels of complexity included, with [448,449] including the effects of the fluid loading on the propeller, as shown in Fig. 45. Despite the extensive body of research in this area, there is limited evidence of full-scale implementation perhaps due to the practical challenges of applying this at scale. Reference [445] presents a brief demonstration of the active propeller control technology on board a small motor yacht and reports reduction in the measured torque and interestingly notes that the reduced torsional vibration leads to a 6 dB reduction in the 3rd order of excitation in the interior of the yacht.

Another application of active vibration control at source that has been explored for application in maritime transportation has focused on active damping of the funnel in a cruise ship [40]. It is noted in [40] that the funnels provide a dominant path for energy transmission throughout the cruise ship and, therefore, reducing the vibration of the funnels helps to control unwanted noise and vibration at source. Passive control of vibration transmission from the funnels into the rest of the ship is typically realised by ensuring that the required connections between the funnel and the casing are made using passive isolators located at the nodes of vibration. However, as shown in the left-hand image in Fig. 46, the modal behaviour of the funnels is complex and, therefore, ensuring the connection points are at nodal positions for all modes of vibration is non-trivial. Moreover, the performance of the passive isolators is limited at low frequencies, and these factors motivated the exploration of an active solution. Due to the size of the funnel system on a cruise ship, as well as competing engineering requirements, installation and wiring of a centralised active vibration control system may be impractical. Therefore, a decentralised active vibration control strategy is proposed in [40], which utilises the ‘smart active damper’ shown in the right-hand side of Fig. 46. This self-contained unit contains an inertial mass actuator, accelerometer and simple velocity feedback control strategy. The proposed system is experimentally realised and applied to control the vibration of a large steel structure, representative of the funnel vibration control, and using a single instance of the ‘smart active damper’ shows effective damping of two resonances in this structure. Subsequent work by the authors have demonstrated the use of multiple units on a simple beam structure [452], but a distributed rather than fully decentralised control strategy is applied. Clearly there is scope for building on this work to explore application to the large and complex structures found in the maritime environment.

4.5.2. Active vibration isolation

Active vibration isolation of machinery on-board marine vessels has been widely investigated. Some of the earliest work in this area focused on fully active solutions, where the machinery raft was isolated from the hull via electromagnetic levitation [41]. This concept, however, required a large number of actuators, high levels of power and, in the event of power failure, would be left with no isolation due to its purely active design. As a result, subsequent work focused on the realisation of combined active-passive isolation systems. Early practical work on an active-passive isolation mounting system was then conducted in the early 2000’s and resulted in the ‘smart spring concept’ [453–455]; this system combines active control in parallel with a passive isolator. In the initial prototype active control capability was provided by a single electromagnetic actuator and the passive isolation was provided by steel coil springs, as shown in the left-hand image in Fig. 47; later realisations, as shown in the right-hand image in Fig. 47, combined a standard elastomeric mount with 6 electromagnetic actuators arranged in a hexapod arrangement to enable control in all 6 degrees of freedom [453–455]. This system has been shown experimentally to provide significant levels of vibration control isolation, in particular the active part has been shown to be able to provide significant levels of control over both a broad bandwidth focusing on structural resonances [453] and at discrete frequencies associated with the harmonics of reciprocating machinery [455].

Since the ‘smart spring’ concept, there has been significant research into the application of active vibration isolation to the floating raft systems quite widely utilised in maritime applications. These rafting systems isolate machinery using a two-stage isolation system, as shown in Fig. 48. Floating raft systems are effective at isolating machinery from the hull structure, and thus reducing structurally radiated noise into the ocean, as well as providing protection to the machinery when subject to external loads and shock [456].

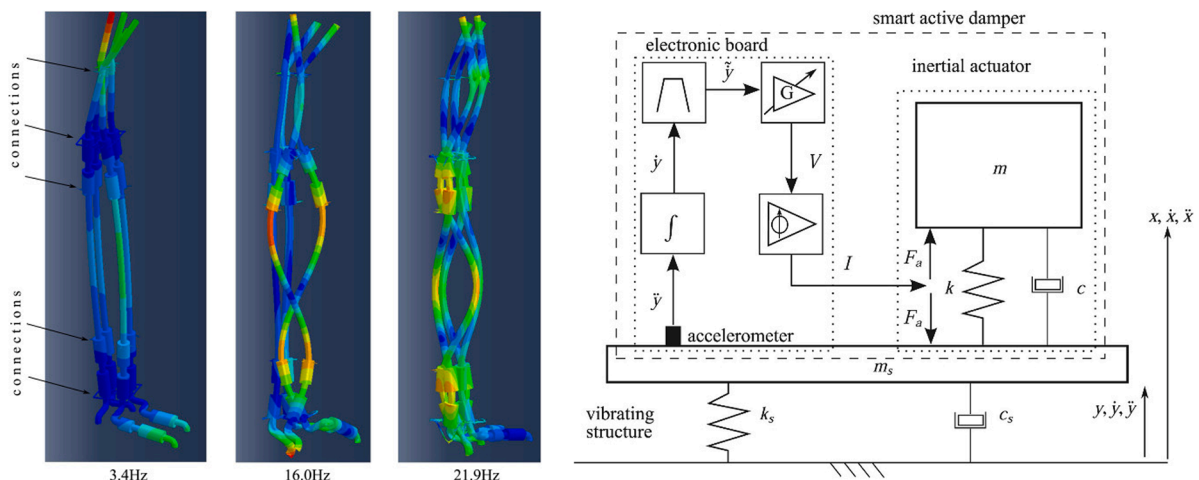


Fig. 46. Vibration modes of the funnel system in a cruise ship based on finite element analysis and a smart active damper (From Ref. [40]).

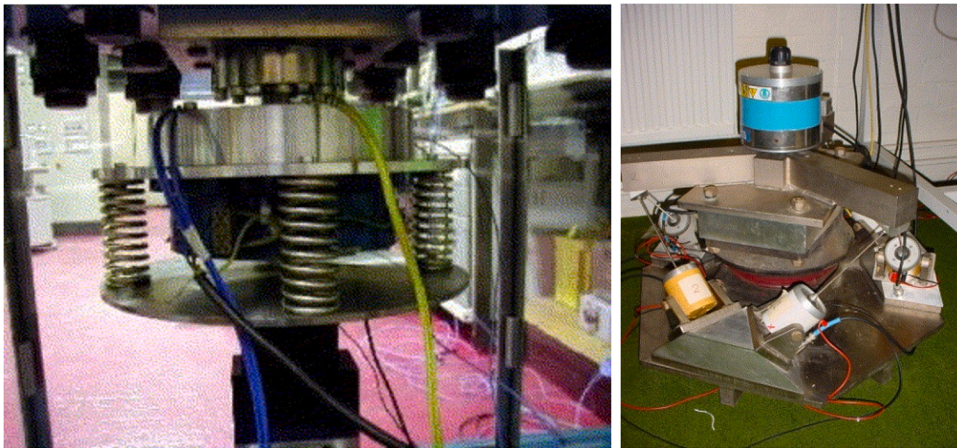


Fig. 47. 'Smart spring' mounting system concepts (From Ref. [454]).

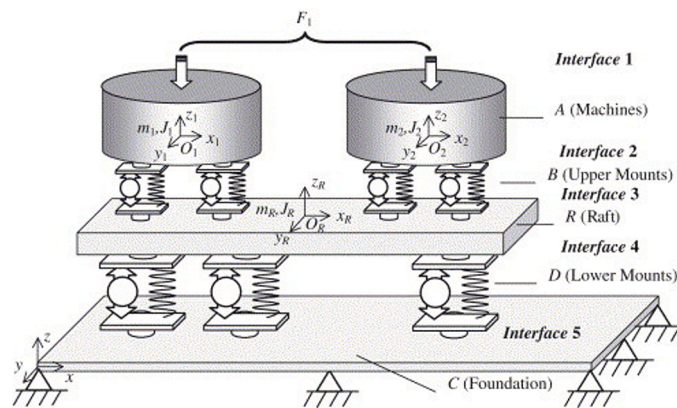


Fig. 48. Analytical model of an active floating raft isolation system (From Ref. [456]).

However, the performance of passive raft isolation systems, which typically utilise resilient elastomeric mounts, is limited at low frequencies and at structural resonances [455]. To overcome these challenges, various studies into active-passive rafts for vibration isolation in maritime applications have been explored. The first analytical investigation into an active-passive isolation raft was presented in [456], with a comparison between the different raft configurations shown in Fig. 49 providing insight into practical design considerations. In particular, this work demonstrated how introducing active capabilities into either the machinery mounts (Fig. 49b), the raft mounts (Fig. 49c), or in both locations (Fig. 49a) influences control performance and efficiency.

Following this early work into active-passive raft systems, many more studies into the design and realisation of such systems for the maritime environment have been published [455,457–464]. This research has developed and demonstrated active-passive isolation systems for the maritime application on large-scale laboratory-based demonstrators of varying degrees of complexity [457–463] and has been applied to an operating marine vessel to demonstrate the performance at sea [464]. There are many more publications in this area, but this selection demonstrates the breadth of research in this area, as well as the relatively high level of maturity of this technology in the maritime sector.

4.5.3. Active vibration control at the receiver

In the maritime space there are two potential key 'receivers' – occupants of the vessel and the wider oceanic environment, which particularly focuses on disturbance to marine life and stealth of naval vessels [465]. The main focus of active vibration control at the receiver in the maritime transportation environment has focused on controlling the vibration in the hull, which acts as the receiver to, for example, machinery and power train excitation, and radiates into the ocean environment. The majority of research focusing on controlling vibration in the hull structure has focused on naval applications [38,42,466–470] and specifically submarines. More recently, however, interest in underwater radiated noise has become broader due to the size and number of commercial vessels [465], and the extensive body of research demonstrating the impacts of anthropogenic noise on marine life [470].

Much of the published research into active control of hull vibration has been analytical or numerical in nature [42,466,468,469], which is perhaps due to the complexity and scale or the real-world problem or perhaps due to the secure nature of much of the early research in this area. Scale experimental implementations have been reported [38,467] and full-scale trials on a 4500-tonne vessel

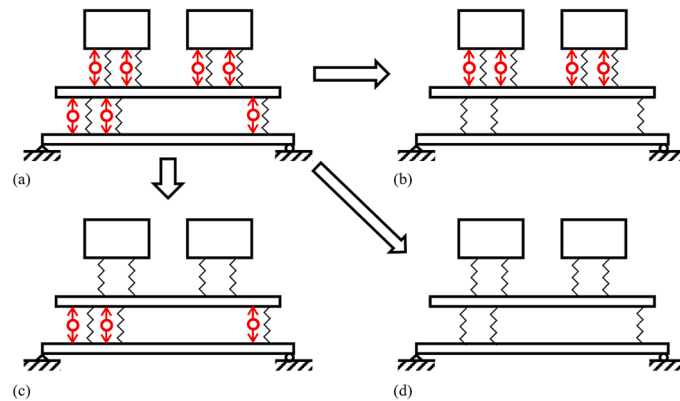


Fig. 49. Isolation raft configurations investigated in [456] showing active and passive isolation components. (a) Shows the system with full active control; (b) shows the system with active machinery control; (c) shows the system with active raft control and (d) shows the purely passive system.

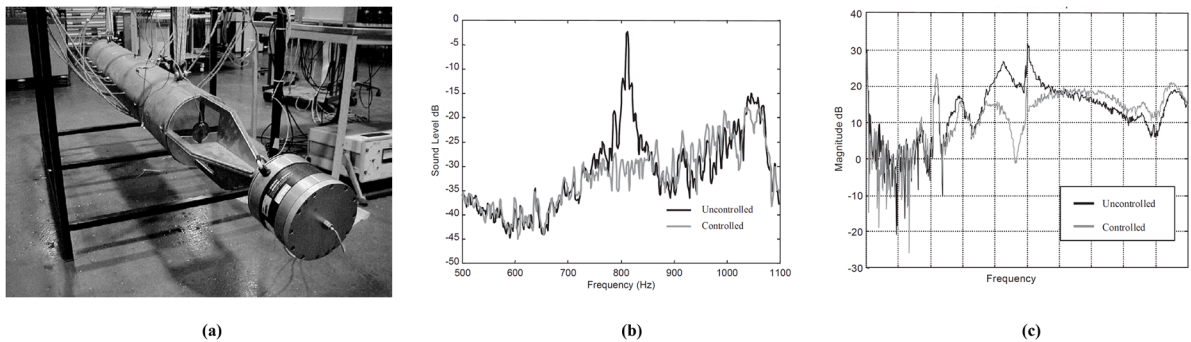


Fig. 50. (a) Laboratory scale rig used to investigate 'selective damping' technology for hull vibration control, (b) far-field acoustic power radiated by structure in (a) with and without control; (c) shows the suppression of thrust block resonance in full-scale application of 'selective damping' (From Ref. [38]).

briefly described in [38]. Fig. 50a shows the laboratory scale rig used to investigate the 'selective damping' technology reported in [38] and Fig. 50b shows the far field sound power measured for this system in air with and without the active vibration control system applied. The 'selective damping' approach uses arrays of sensors and actuators to actively damp selected modes that radiated into the far field. Fig. 50c shows an example result from the full-scale trials of the 'selective damping' technology, demonstrating the effective reduction of modes at the thrust block. Although there has been limited published research directly focusing on wider maritime transportation applications of active vibration control in the receiving structure. There is clearly a move towards requiring such high-performance noise and vibration control technologies in commercial shipping applications due to emerging policies focused on reducing commercial shipping noise [471–473].

5. Summary

This article has provided a thorough overview of the fundamental concepts and applications of both active noise and vibration control technology within passenger transportation vehicles. This is the first comprehensive review covering the application of active control technologies to passenger transportation, including road, rail, air and maritime. Although the level of adoption of active control systems in these different transportation domains varies, the technological and physical insights provided by this article are enhanced by reviewing all domains. For example, although the commercial exploitation of active control is more advanced in the automotive and aeronautical sectors, there are strong links between both the physical basis and the active control systems utilised in the rail and maritime sectors. Therefore, this article has not only provided an extensive review of active noise and vibration control systems for passenger transportation vehicles, but also enabled synergies between the various domains to be highlighted. This aims to not only benefit the reader's understanding of the current state-of-the-art, but also to support future research and development in the field.

Following a brief introduction, which lays out the overall historical arc through which active control technologies have been developed, a brief introduction to the feedforward and feedback control architectures that form the basis for realising active control systems has been presented. The remainder of the article is split into two sections – one covering active noise control systems and one covering active vibration control systems. In both cases, the review presents a summary of the key contributions to the development, realisation and application of active control systems to passenger transportation vehicles, and an extensive reference list is provided,

including over 400 references to publications of note. Due to the extensive research conducted in this area, this reference list is not exhaustive, but covers the most significant contributions and, critically, includes many references of historical importance that may otherwise be lost.

This review article demonstrates the wide variety of active noise and vibration control solutions that have been developed for passenger transportation vehicles, with commercial applications in the automotive and aerospace applications being presented alongside more research focused realisation in the rail and maritime applications. It has been demonstrated that there has now been long-term interest in the development of active noise and vibration control systems from both academia and industry, however, there is still significant work to be done to fully exploit the potentials of active noise and vibration control in all passenger transportation applications. Typically, active control systems are more costly and complex to realise than passive systems, but as performance requirements increase in all domains and the need for lightweight noise and vibration control systems increase, the future presents significant potential for further development of active control systems. This is perhaps particularly true as novel passenger transportation vehicles emerge, including self-driving vehicles, which will change the expectations of the user.

There are several research areas that require more exploration to further develop active technology for application across civil transportation vehicles. Fundamental work is needed to functionally miniaturise, in both size and weight, and reduce the power consumption of the sensors, actuators and electronics used to realise the controllers. In many applications it would also be desirable for the control system components to be self-powered, with wireless connectivity, to reduce the necessary cabling for installation and the associated integration challenge; this would require synergetic design with various energy harvesting and storage technologies. Overall, it would be desirable for modular ANC or AVC units to be developed, which could be easily installed across a wide range of vehicles as add-ons with no need for significant modifications to the vehicle design or production processes. Another key area of emerging research interest is in the integrated design of active-passive control systems, which aims to ensure that the active and passive components are designed in a holistic manner to maximise the overall performance. At present, noise and vibration control in vehicles is still typically based on passive designs, with some a posteriori augmentation with active technologies to help solve specific vibro-acoustic problems. Emerging active-passive designs include tuneable metamaterials, whose development is being supported by additive manufacturing technologies, including the printing of functional materials for actuation, sensing and electronics. Tuneable active metamaterials are well-suited to the noise and vibration control challenges associated with transportation applications due to their high-performance and lightweight characteristics. In the vehicle environment, active control systems are likely to benefit from additional technology introduced into the vehicle for other purposes, such as infotainment and safety. For example, while head-tracking has already been utilised to increase the performance of in-car active noise control systems, research is required to explore broader video monitoring and analysis could be used to tune local and global ANC systems. Beside these technological aspects, there are also some fundamental challenges related to the control of non-linear systems, which is an area of research with significant current interest motivated by the utilisation of machine learning. Similarly, there are also significant challenges remaining to achieve effective active control of transient disturbances and broadband stochastic disturbances in large-scale practical applications. Finally, within the context of passenger transportation vehicles, the need to utilise alternative energy sources will present new noise and vibration control challenges due to the introduction of new sources of noise and vibration generation, and active control solutions are likely to be an invaluable mitigation tool.

CRediT authorship contribution statement

Paolo Gardonio: Writing – review & editing, Writing – original draft. **Jordan Cheer:** Writing – review & editing, Writing – original draft. **Michele Zilletti:** Writing – review & editing, Writing – original draft.

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.

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Data availability

Data will be made available on request.

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