Acoustic Metamaterials: A Potential for Cabin Noise Control in Automobiles and Armored Vehicles

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Received 26 February 2016

Revised 26 June 2016

Accepted 1 July 2016

Published 25 August 2016

The aim of this paper is to provide an overview of the existing industrial practices used for cabin noise control in various industries such as automotive, marine, aerospace, and defense. However, emphasis is placed on automobiles and armored vehicles. Generally, automobile cabins usually constitute of thin structural panels, where the fundamental frequency typically falls below 200 Hz. If a specific structural mode couples with a specific acoustic mode of the cabin, booming noise occurs. As such, discomfort may be felt by the occupants. Fundamentally, vibroacoustics problems may be minimized if the acoustic modes and the structural modes are decoupled, which is achieved usually by structural modifications or acoustical treatments. However, if excessively performed, the weight limitation of an automobile design will be exceeded; not to mention the adverse effect of increased weight on several factors such as fuel efficiency, mileage life of tires and acceleration of the vehicle. Moreover, current solutions have several drawbacks in low frequency noise control. In light of this, it is of great interest to explore the feasibility of acoustic metamaterials as an alternative with hope to improve cabin noise.

Keywords: Cabin noise; interior noise; noise control; vibroacoustics; review; automobile; armored vehicle; acoustic metamaterials; defense industry; automotive industry.

1. Introduction

In the automobile industry, both mechanical and electrical technologies have evolved dramatically since nineteenth century. Continuous and rapid cutting-edge

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developments allow manufacturers to gain technological advantage and stand out from their competitors. The rate of these developments has, however, decreased and more or less arrived at a plateau lately. Existing technology and specifications are now simply a baseline in the viewpoint of most consumers. As such, greater demands have grown among other factors for purchase consideration between manufacturers. One of such factors includes passenger comfort [Shende, 2014]. This aspect is related to the consideration of human factors engineering and the study of noise and vibration within automobile cabins. Moreover, the continuous work of enhancing current automobile technologies drives the need for better noise solutions in automobile cabins for minimal weight penalty [European Automotive Research Partners Association, 2013].

Apart from the commercial consideration, the defense industry also emphasizes the need in improving passenger or occupant (referred as crew in the defense industry) comfort in the design phase of armored vehicles. This has been widely discussed as the amplitudes of cabin noise in armored vehicles — especially tracked vehicles — are way higher than commercial vehicles. In fact, safety design requirement is defined with reference to defense standards to impose a limit (< 85 dBA) for maximum allowable cabin noise under a continuous exposure of eight hours [Department of Defense, 2015]. The reason is to not adversely affect the operational readiness of the crew during a mission or any day-to-day operation. As such, crew comfort has become a marketing asset for a defense manufacturer to stand on a higher ground than its competitors.

The most common and easiest approach in reducing noise exposure by the crew is to provide them with Combat Vehicle Crewman's (CVC) helmet. These CVC helmets generally come with an in-built active noise control (ANC) module for enhanced noise reduction. In some cases, they are also advised to equip themselves with earplugs for enhanced noise attenuation if necessary. These solutions, however, are more effective in the higher frequency range (> 200 Hz). Therefore, the crew may still be exposed to excessive cabin noise especially when a phenomenon known as booming noise is present.

In general, automobile cabins constitute of thin structural panels such as doors, windscreen, and windows. As such, the fundamental frequency of these structural members usually falls below 200 Hz although the cabin shape of each automobile may be unique. Cabin noise is highly dependent on the behavior between acoustic modes of the cavity and structural modes of the encompassing panels. If a specific structural mode couples with a specific acoustic mode, cabin noise is likely to amplify significantly as a result of vibroacoustics; earlier referred to as booming noise. This phenomenon is usually perceived by auditory system of humans as a loud low pitch monotone. Depending on the sensitivity of an individual to a specific booming frequency, discomfort such as nausea or giddiness may be experienced, which is evidently undesirable.

Fundamentally, vibroacoustics problems may be minimized if the acoustic modes and the structural modes are decoupled, which is achieved usually by structural

modifications or acoustical treatments. This is seemingly easy as such studies can be performed numerically during the design phase. However, it is possible that excessive treatments may be required on specific panels due to closely-spaced acoustic modes. Inevitably, the weight limitation of an automobile will be exceeded; not to mention the adverse effect of increased weight on several factors such as fuel efficiency, mileage life of tires/track pads and acceleration of the vehicle [Vige, 2010]. In light of these, it is of great interest to explore the feasibility of an emerging research — acoustic metamaterials — with hope to improve cabin noise of automobiles and armored vehicles.

This paper aims to provide an overview of the existing industrial practices used for cabin noise control in various industries namely automotive, aerospace, and defense. Several review papers can be found in the literature pertaining to cabin noise control in automobiles but only a few highlighted the drawbacks of existing methodologies. Moreover, none highlighted the potential of applying acoustic metamaterials in the cabins of automobiles and armored vehicles [Elliott, 2008; Genuit, 2004; Hansen, 2005; Lalor and Prietsch, 2007; Rao, 2003; Sestieri and Carcaterra, 2013]. In Sec. 2, the current industrial practices pertaining to cabin noise control are discussed. In Sec. 3, the potential of acoustic metamaterials is highlighted. In Sec. 4, a conclusion is made based on the earlier sections.

2. Current Practices in the Industry

Extensive studies have been conducted over the span of past decades due to the rising demand for enhanced passenger comfort in various industries namely automotive, aerospace, and defense. These studies are categorized accordingly based on the nature of their approach and discussed in the following order:

- (1) Psychoacoustics
- (2) Acoustical Treatments
 - (a) Helmholtz Resonator
 - (b) Sound Absorption Material
 - (c) Viscoelastic Damping Treatment
 - (d) Dynamic Vibration Absorber
- (3) Active Controls
 - (a) Active Noise Control
 - (b) Active Structural Acoustic Control
- (4) Numerical Methods
 - (a) Finite Element Method and Boundary Element Method
 - (b) Statistical Energy Analysis
- (5) Dimensional Analysis

2.1. Psychoacoustics

Conventionally, cabin noise is measured from a specific field point using microphones. Through post-processing, this measurement is then matched against safety regulation; typically A-weighted. Conversely, it is believed that such approach does not account for the subjective behavior of an individual to the perception of noise. Thus, psychoacoustics aims to consider this area by studying noise perception of the humans' auditory system. This is usually practiced in the automotive industry to *shape* cabin noise signature and is becoming more popular recently especially for luxury vehicles. Analyses are presented in terms of sound quality evaluation (SQE), which is expressed in terms of loudness, sharpness, roughness, fluctuation strength, speech intelligibility, and/or tonality. In the automotive industry, different methods have been proposed to improve the perception of cabin noise.

A potential approach is to consider the effect of masking by balancing interior noise components such as engine noise, booming noise, road noise, and wind noise, by the presence of another sound [Hoshino, 2000]. The focus is placed on only wind and road noise as they are claimed to be dominant cabin noise contributors at high travelling speed ($> 100\text{ km/h}$). However, the travelling speed of automobiles is usually much slower in an urban area with well-developed infrastructure due to regular intervals of traffic light. As such, booming and engine noise are likely to be the dominant cabin noise contributors for a huge percentage of the commute time. For armored vehicles, travelling at such high speed is even more unlikely due to their massive tonnage. Some investigated the combined effect of vibration and cabin noise as a result of chassis components [Genuit, 2004; Radic *et al.*, 2011] and road profile [Nopiah *et al.*, 2014, 2015].

Some works specifically targeted the sound evaluation of booming noise by proposing booming index [Hatano and Hashimoto, 2000; Shin *et al.*, 2009]. This index is used for objective evaluation of the booming sound quality within an automobile cabin. However, correlation may sometimes be poor between the index and field measurements depending on the studied frequency range. In light of this, numerous attempts can be found to improve this approach by applying artificial neural network [Lee and Lee, 2009; Lee *et al.*, 2006]. Concurrently, the artificial neural network is improved on by proposing to combine different methods [Paulraj *et al.*, 2010; Wang *et al.*, 2007; Yildirim and Eski, 2008]. Typically, psychoacoustics studies require a jury or human subject evaluation, which is time consuming since it involves surveying a specific sample audience size and compiling the feedback may be tedious. In light of this, new models and methodologies are proposed to omit the process [Allman-Ward *et al.*, 2004; Nor *et al.*, 2008]. Apart from automobiles, Hu *et al.* [2014] and Liu *et al.* [2013] performed similar studies on train cabins.

These works clearly showed that psychoacoustics may be a good approach for sound quality evaluation of cabins. However, such approach is only feasible when the cabin noise is not excessively high such that safety exposure limit is exceeded.

This is likely the case in armored vehicles since crew noise is usually on or near the borderline of safety exposure limit. As such, higher priority should always be placed on crew noise reduction rather than perception studies and/or noise signature shaping.

2.2. Acoustical treatments

2.2.1. Helmholtz resonator

More than two decades ago, Fahy and Schofield [1980] highlighted several disagreements between literatures pertaining to the coupling effect of Helmholtz resonator (HR) in an enclosure. In light of this, they attempted to resolve the disagreements with an investigation to understand the influence of a HR on the acoustic mode of a rectangular enclosure. To their surprise, an interesting phenomenon is observed when the natural frequency of the HR is tuned to the frequency of an acoustic mode. Two new coupled modes are generated on each side of the tuned frequency. The frequency separation between each new coupled mode and the tuned frequency is governed by a coupling parameter and the position of the resonator with relative to the pressure anti-node of the acoustic mode. The coupling parameter widens the separation when the volume ratio of cavity-to-resonator increases. It is emphasized that the tuning of a HR should be carefully performed with strong understanding of the environment of its intended application. For instance, if only an exciting frequency exists, tuning the HR exactly to that particular frequency will be beneficial. However, if there exist more than one exciting frequencies (varying excitation), careful tuning will be necessary to achieve acceptable noise reduction. Subsequently, their work has been referenced widely as a baseline for fundamental knowledge relating to the interaction between an HR and a closed cavity. Their work is later reasoned by Cummings [1992] to be infeasible for practical applications since the likelihood of having more than one acoustic mode is high. Apart from that, these acoustic modes may not be well-separated due to high modal density in certain cases. As such, an analytical model is proposed with the consideration of more HRs in a rectangular enclosure. However, he highlighted the tremendous amount of computational effort required for the proposed model. Furthermore, the consideration of a parametric study is too tedious analytically due to a vast amount of possible variables. Answering his concerns, Driesch [2002] considered to optimize several design parameters of multiple syringe-type resonators including chamber volume, internal damping and opening area numerically, which is more efficient. These optimally-designed resonators are then applied on a tractor cabin to validate their acoustical performance. Despite her efforts, optimal acoustical performance is only achieved if the targeted acoustic mode remains at the same frequency, which is pointed out earlier by Fahy and Schofield [1980]. This clearly implies that the design of HR has to be unique for each cabin and cautions have to be taken for optimal acoustical performance.

Franco [1991] considered the concept of HR in an automobile to reduce cabin noise in the low frequency range (< 200 Hz) by adding a long rectangular slit on the topmost panel of the boot compartment. As such, a connection is established between the respective air volumes — passenger and boot compartments. However, the cabin noise is reduced by only a negligible amount of about 1 dB in the studied frequency range. The work is limited to the tuned frequency of the HR, which is dependent on the volume of the boot compartment and the opening area of the slit. Optimization studies are conducted to account for these parameters [Kamci, 2009; Kopuz and Lalor, 1995]. Generally, in automobile design, the volume of the boot compartment is usually optimized for maximum room space. This, in turn, will become a design issue if a smaller boot compartment is required to tune its natural frequency upwards to match a targeted acoustic mode of the passenger compartment. Or in another case, a bigger boot compartment to match a low frequency acoustic mode. Either ways, the approach may not be that effective since the amount of cabin noise reduction may still be insignificant. Furthermore, debates will arise whether to compromise storage space for improved cabin noise. Nonetheless, one can also consider enlarging the slit to have an opposite effect on the natural frequency of the HR. Despite the fact, there is always a limit to the largest allowable slit area since the effect of a HR may diminish when the slit area becomes too large. In most armored vehicles, a boot compartment may even be unavailable since the crew are expected to carry their own belongings. On the contrary, one may still suggest the possibility of adding empty compartments within the crew cabin for HR effect. Hence, the overall crew space will be affected unnecessarily, which leads to difficulty in the ingress/egress movement of the crew. On the other hand, if slits are made on the cabin walls, crew survivability may be compromised.

From here, it is evident that multiple HRs may indeed be more effective for noise reduction in a large enclosure as mentioned by Cummings [1992]. As ideal as it may sound, it is worth noting that the addition of more HRs into an automobile cabin may raise design issues pertaining to space and maybe even weight constraints. Both are certainly undesirable. In light of this, Doria [1995] put forward an analytical solution by connecting two expansion chambers in series in the form of a linear duct assembly. The opening of the resonator is then exposed to the enclosure wall. The key idea is to achieve equivalence to one resonator yet possessing the characteristics of two. Again, the volume of each expansion chamber has to be carefully tuned to achieve optimum noise reduction in the rectangular enclosure. On top of that, the positioning of the resonator on the enclosure wall is also important as a poorly-placed resonator may adversely influence the acoustic characteristics of the enclosure. The challenge of tuning the volume of each expansion chamber accurately is highlighted; particularly in realizing them experimentally. This is due to geometric deviation as a result of inevitable machining tolerance. Nonetheless, such is no longer a concern with the current manufacturing technology. In an automobile, limited positions may be available for proper placement due to the inherent positions

of other structural features or electrical components. This issue worsens in armored vehicles as even more vetronics systems are present. Evidently, the consideration of practical installation by earlier works seems to be lacking.

Recently, Chintapalli and Padmanabhan [2014] proposed the use of multiple smaller resonators; each tuned similarly to the targeted acoustic mode of a rectangular enclosure. They showed significant noise reduction albeit it becomes negligible beyond three resonators. Hence, the acoustical performance of their resonators may deteriorate further if extended to larger enclosures. Separately, Yu *et al.* [2009] proposed T-shaped resonators that aims to remain compact in geometry yet not compromising acoustical performance. They started off by establishing an analytical model [Li *et al.*, 2007; Li and Cheng, 2007] for multiple resonators within an enclosure based on an earlier work [Fahy and Schofield, 1980]. Concurrently, their analytical model is validated experimentally by several designs of T-shaped resonators. Likewise, they showed the importance of proper positioning of the resonators within an enclosure for optimum noise control. This is consistent with the findings by Doria [1995]. Moreover, they also highlighted the influence of internal resistance on the acoustical performance of the resonators. These issues are discussed in detail subsequently [Yu and Cheng, 2009; Yu *et al.*, 2008]. To demonstrate the effectiveness and feasibility of their proposed resonators, further work [Li *et al.*, 2010] is carried out by incorporating the resonators into a double-leaf panel to attain improved narrowband noise attenuation at targeted low frequencies (< 400 Hz) depending on the tuning of the resonators. Although the spatial requirement due to its small cross-sectional area is attractive, the physical length of the resonator increases as the targeted frequency gets lower (up to about 40 cm for noise attenuation at 200 Hz). Similarly, in the aerospace industry, the concept of HRs is also studied by different research groups [Esteve and Johnson, 2002, 2005; Howard *et al.*, 2005; Lane *et al.*, 2005; Park and Sohn, 2010; Park and Sohn, 2009; Park, 2013a, 2013b; Sohn and Park, 2011].

In some works [Fenech *et al.*, 2006; Liu and Herrin, 2010; Wang *et al.*, 2010; Yang *et al.*, 2015], the use of micro-perforated panels (MPP) are proposed, which are based on the fundamentals of HR. In this case, each perforation effectively represents a resonator with the backing air cavity representing the expansion chamber of a conventional resonator [Arenas and Crocker, 2010]. The potential of MPP is recently discussed in detail by Maa [2015]. In this paper, the key design parameter is the influence of open-area ratio on the acoustical performance of a MPP, where an optimum sound absorption is achieved at specified open-area ratio. This is aligned with the findings of other recent works [Bravo *et al.*, 2012, 2013; Zhao *et al.*, 2015]. However, the difficulty in realizing MPP with much smaller orifices (< 0.3 mm in diameter) is a concern due to limitations of current manufacturing techniques; including additive manufacturing methods. It should be noted that this requirement may only be necessary if one is able to compromise for a narrower bandwidth of sound absorption with larger orifices. On the other hand, Yu *et al.* [2014] showed otherwise by considering a multi-layered MPP configuration arranged in a staggered formation. Despite the success of MPP in duct applications [Cox and D'Antonio,

2009], the depth of the backing air cavity has to be sufficiently generous for effective low frequency attenuation. Again, cabin space is affected. The inevitability of a backing air cavity could, however, be potentially minimized or omitted based on the recent work by Toyoda *et al.* [2015] albeit extensive further studies are necessary for such realization in panels. From these works, MPP seems to be an improved concept of HR albeit the minute orifices may likely trap dust particles. As such, one should also account for the ease in cleaning the panels especially if they are installed in the cabin interior or at tight corners.

It is evident that the application of resonators into automobiles and armored vehicles is likely to raise inevitable issues pertaining to (i) spatial requirement, (ii) the increase in vehicular weight with increasing number of resonators, (iii) the effort required for design optimization, and particularly for MPP, (iv) the ease of cleaning. The lack of literature on actual application for cabin noise control in automobiles and armored vehicles is evident that such approach is not practical enough to be accepted by manufacturers. Moreover, the total number of resonators increases with cabin size as well. Aside from these apparent concerns, extensive effort will most probably be necessary to incorporate multiple resonators in a cabin while maintaining interior aesthetics to the potential buyers.

2.2.2. Sound absorption material

Sound absorption material — a common and traditional passive noise solution found in the industry — is usually available in sheets made of a porous medium such as fiberglass or polyurethane foam. As sound waves propagate within the porous network, part of the sound energy is dissipated due to viscous and thermal losses [Arenas and Crocker, 2010; Cox and D'Antonio, 2009]. As such, the transmitted or reflected sound waves are reduced in sound energy. Depending on the area of application, several parameters, such as porosity, density and, thickness, can be varied to achieve intended acoustical performance. Being easily implementable and cheap makes them attractive as a preliminary measure for cabin noise control. As the discussion of a detailed theoretical background is beyond the scope of this paper, interested readers can refer to specialized literature for further perusal [Arenas and Crocker, 2010; Cox and D'Antonio, 2009; Egab *et al.*, 2014].

There are numerous analytical works that demonstrated the influence of dissipative material on the walls of an enclosure [Bruneau *et al.*, 1985; Du *et al.*, 2011; Lanoye *et al.*, 2008; Martin and Bodrero, 1997; Pan *et al.*, 1999; Suzuki *et al.*, 1989]. They showed that the interior noise of an enclosure can be reduced especially if the placement of sound absorption materials is optimized. Later, Sarigul and Kiral [1999] extended these analytical concepts to a two-dimensional model for cabin noise control in a truck cabin. Sound absorption materials are considered only on the floor and rear wall for practicality as the sides are usually the doors. Although considerable noise reduction is presented at 95 Hz, they provided no concise details on the acoustical characteristics and thickness of the modeled material. As such,

the results may be questionable since it is known that a reasonably thick layer of sound absorption material is required for noise attenuation at such low frequency to be at least noticeable. More realistic results are demonstrated in an excavator cabin [Willemssen *et al.*, 2009] and a van [Cherng *et al.*, 2002]. Apart from automobile applications, this solution is also considered in aircraft fuselage [Boily and Charon, 1999] and train cabin [Hu *et al.*, 2014]. Exclusively, active sound absorption material, known as smart foam, is also considered in aircraft fuselage [Gentry *et al.*, 1997; Guigou and Fuller, 1999; Kim *et al.*, 1999].

The necessity of having a thick layer of sound absorption material is proven otherwise by a research group that proposed a so-called “double porous material” [Atalla *et al.*, 2001]. The material is constructed simply by introducing periodic arrays of cylindrical cut-outs similar to that of MPP albeit in the macroscale for this case. Despite the difference in the shape of the cut-outs (square) considered in their numerical model, they showed enhanced low frequency absorption coefficient of up to 0.9 — specifically between 200 and 300 Hz — with a thickness of 11.5 cm. This finding is noteworthy due to the fact that a higher sound absorption coefficient is achieved in the lower frequency range as compared to the higher frequency range. This differs from the sound absorption profile of conventional sound absorption materials. In light of this, they performed a parameter study to understand the influence of cut-out size and its distribution. It is concluded that a smaller size is preferred for low frequency noise control and sound absorption performance is observed to deteriorate when periodicity is removed. Several subsequent works in all aspects — numerical, experimental and analytical — are performed by the same research group and is later compiled as a review paper, which serves as a design guideline for interested researchers [Sgard *et al.*, 2005]. The success of sound absorption material in mid and high frequency noise attenuation is well-known albeit documentation of the case studies is usually presented in company websites, brochures, and catalogs. As such, limited reported literature can be found pertaining to their applications in the industry.

Recently, the inclusion of HRs in porous material is proposed by arranging them similarly in a periodic manner [Doutres *et al.*, 2015]. Expectedly, excellent sound attenuation is achieved but limited to the tuned resonance frequency of the resonators. Owing to the acoustical performance of the porous material in the higher frequency range, the obvious advantage of such design is to also have an enhanced yet limited bandwidth of low frequency absorption. For example, they achieved an insertion loss of up to 8 dB at 145 Hz in one of their configurations. From their findings, it is evident that the proposed design may be more suitable in applications where the control of tonal noise is of concern. On the side note, the construction of this design may pose numerous challenges such as the alignment (angle) of each resonator with respect to the incident wave and maintaining a uniform depth of placement for each resonator within the porous material.

Apart from this recent work, hardly any reported studies can be found on the potential of sound absorption material for low frequency (< 200 Hz) noise control

without compromising weight gain as a result of increased thickness. Moreover, the degradation of their composition with time is another issue that affects reliability and acoustical performance in the long term basis. This is especially the case in armored vehicles as their mobility is usually in harsh environment. Frequent maintenance is definitely not favored.

2.2.3. Viscoelastic damping treatment

The benefits of viscoelastic damping treatments are comparable to sound absorption materials; particularly relating to cost, simplicity, reliability, and stability. However, their working principle is unlike sound absorption materials. They are typically applied to the targeted panel to reduce vibrational energy and, in turn, reduce sound radiation into the enclosure. In general, there are three main types of such damping treatments — free-layer damping (FLD), constrained-layer damping (CLD) and tuned viscoelastic damper (TVD).

FLD is applied by spray-coating onto a targeted panel. Vibrational energy is dissipated by its extension and compression along the plane of the vibrating panel. CLD consists of a layer of viscoelastic material sandwiched in between two outer constraining layers. When a panel vibrates, vibrational energy dissipation occurs due to shearing deformation as a result of the outer constraining layers. TVD acts like a mass-spring system, where a tuning mass is attached to the targeted panel by a layer of viscoelastic material. This viscoelastic material effectively acts as a spring. The natural frequency of the system is then tuned to an excitation frequency. As such, prior knowledge of the vibrational profile in the base structure is essential. Similar to the coupling effect of HR, the excitation frequency is separated into two sidebands if properly tuned.

Out of these, CLD is commonly used in the industry due to the presence of additional damping as a result of shearing deformation as compared to FLD of identical weight [Rao, 2003]. Supporting this statement, the effectiveness of CLD is evident from literature with it being considered in various transport systems such as ship [Pang *et al.*, 2014], tractor [Cao, 2005], truck [Subramanian *et al.*, 2004], aircraft [Boily and Charron, 1999], and helicopter [Caillet *et al.*, 2012]. Some even attempted to introduce active control system into CLD [Poh *et al.*, 1996]. On the other hand, TVD is less popular albeit applications can still be found in aircraft [Halvorsen and Emborg, 1989] and payload fairing [Howard *et al.*, 2005]. This trend is possibly explained by the added advantages of dynamic vibration absorber (DVA), which is technically simpler in design. Nonetheless, it is noted that viscoelastic damping materials are sensitive to temperature changes and may, in turn, exhibit changes in inherent damping properties. As such, damping performance may no longer be optimal in a harsh environment encountered by armored vehicles; implied by the lack of application in this area.

2.2.4. Dynamic vibration absorber

Generally, the underlying concept of DVA is similar to TVD in the sense that both are made up of a mass-spring system. However, they should not be confused as identical since DVA does not require damping to achieve intended functionality. This implies the preference of DVA over TVD due to very limited literature for the latter. Moreover, discrepancy between numerical and experimental results can be reduced as the need for modelling viscoelastic damping material in DVA is not required. Being a passive solution, it has similar benefits with those as discussed earlier.

The use of DVA is demonstrated by Huang and Fuller [1997, 1998] on the walls of a cylindrical enclosure to reduce vibration magnitude. Depending on the type of excitation, optimum performance is achieved by either placing numerous lightweight absorbers in close proximity to a heavyweight absorber or distributing them in a scattered manner on the enclosure walls. On the contrary, a later work by the same group realized further noise reduction by mistuning the DVAs [Fuller *et al.*, 2000]. Consistently, they indicated the influence of positioning the DVAs on optimum noise reduction, which is extended by Huang and Chen [2000] albeit the consideration of heavyweight absorbers is not included. However, their findings are based on numerical and analytical predictions without any experimental validation. In another work, Esteve and Johnson [2002, 2005] proposed the combination of DVAs and HRs to enhance noise reduction but in this case, their predictions are validated experimentally though in a cylindrical enclosure instead of an actual aircraft. Sacarcelik [2004] performed a more comprehensive study with validation of his predictions on an actual payload fairing.

All these works showed the effectiveness of DVA in reducing cabin noise by controlling panel vibration. However, similar to HR, they are only effective when the excitation is constantly maintained at the same frequency. Any deviation from the tuned frequency will cause a reduction in the overall performance. Moreover, depending on the size of the enclosure, multiple DVAs are required and strategically placed to achieve effective cabin noise reduction. With a smaller cabin size in automobiles and armored vehicles as compared to aircrafts, one may relate to proportionality and suggest lesser required DVAs. This may not be the case since vibration input into these vehicles is varying. As such, more DVAs may be required instead to achieve broadband control. However, to place them optimally, identical issues will arise as highlighted earlier for the case of using HR.

2.3. Active controls

The benefits of conventional passive treatments, such as cost and simplicity of installation, are undebatable. However, the acoustical performance of passive controls may be limited to high frequency with a narrow low frequency bandwidth. For an excitation that varies in frequency, passive control is still possible as highlighted earlier

but proper considerations must be emphasized during the design phase. Active control is an improvement to passive control for its ability to sense vibration or sound pressure level of a specific field point and generate an output that is of opposing magnitude or phase to counter them. Cabin noise can either be controlled directly by an array of secondary sources or by reducing structural vibration with some form of active actuators. Generally, the former is commonly known as active noise control (ANC) and the latter as active structural acoustic control (ASAC).

2.3.1. Active noise control

Typically, microphones are mounted in the cabin to measure interior noise. The measured noise signal is then fed into a controller, where an anti-phase signal is then generated to several loudspeakers for transmission. The loudspeakers are positioned strategically in the cabin; usually in close proximity to the control region. In reality, perfect correction is impossible due to the possibility of phase lag or other contributing factors. As such, the microphones concurrently measure the corrected cabin noise and repeat the whole process. This process allows ANC to be achieved. In control theory, the microphones and loudspeakers are referred to as error sensors and secondary sources respectively. The number of error sensors and secondary sources is dependent on the cabin size. This implies that the number of error sensors and secondary sources is increased for the case of a large cabin. Conversely, increasing the number of error sensors and secondary sources in a fixed cabin size may yield better noise control. The discussion of a detailed theoretical background is beyond the scope of this paper. Interested readers can refer to the literature for further perusal [Muller and Moser, 2013].

There are two kinds of controller — namely feedback and feedforward [Sutton *et al.*, 1994; Tichy, 1991]. The former is a very simple control system, where it involves secondary sources and error sensors as described above. Despite its simplicity, prior knowledge of the primary source is mandatory and the secondary sources must be placed in close proximity to the error sensors to prevent phase lag, which eventually drives the system into instability. Hence, robustness becomes a problem. Feedforward control is more complex, where an additional reference sensor is required to measure the primary source and subject the signal to an algorithmic filtering. The filtered signal is then transmitted through the secondary sources. Concurrently, the error sensors will pick up the combined sound field of the primary and secondary sources and adjust the filtering accordingly. In a way, this may be known as adaptive control, which is desirable for frequency dependent primary sources. The advantage of such approach is that cancellation only focuses on the reference signal rather than the global noise signature, which may include other components such as speech or music. However, for this to be ideal, the correlation of the reference

signal with the primary source must be near unity. Understandably, this is a huge challenge.

In the aerospace industry, countless research efforts are placed on improving the interior noise comfort of a fuselage, specifically in the passenger compartment, since 1990s. On a scaled-down basis, the feasibility of ANC is considered in a cylindrical enclosure using a feedback control system [Silcox *et al.*, 1989]. Conversely, another research group demonstrated likewise on a full-scale twin turbo-prop aircraft [Dorling *et al.*, 1989]. Targeting the tonal excitation of the propeller (88 Hz), a maximum reduction of up to 11 dB is achieved. Concurrently, another work demonstrated likewise on a different aircraft but with lesser secondary sources [Elliott *et al.*, 1989]. They claimed that the sum of squared measured sound pressure can be minimized in this way. Similarly, they considered a tonal excitation source (88 Hz) and achieved a maximum reduction of up to 14 dB. Evidently, these preliminary works showed the potential of ANC for low frequency cabin noise control. With the potential of ANC shown experimentally, Bullmore *et al.* [1990] put forward an analytical model to validate the results of earlier work [Elliott *et al.*, 1989]. Good agreement is observed in their validation. To their surprise, they noticed a high level of local reduction around the error sensors. A later paper compiled all these works and investigated their observations [Elliott *et al.*, 1990]. Later, the dependency of global cabin noise reduction on the number of secondary sources is investigated in two different aircrafts [Martin *et al.*, 1994]. Numerically, an incremental gain in the global reduction is computed, which agrees with experimental results up to fifteen secondary sources. The computed reduction remains constant beyond fifteen secondary sources. Although modelling assumptions may be the cause, no clear explanations are provided.

To achieve global attenuation within a cabin, an extensive amount of sensors is necessary. This is inevitable though undesirable since the bulkiness of the control system naturally increases as well; not to mention the increase in complexity as well. Weight is of high design priority in the aerospace industry since it affects fuel efficiency and payload capacity. Taking into consideration of the findings by Bullmore *et al.* [1990], a possible solution is to adopt a local control by placing error sensors near the head of a flight crew or passenger. This solution is somewhat reasonable since the motion of a flight crew or passenger usually remains around the same spatially. However, a new problem arises as the control region generated by the error sensors is usually insignificant that no noticeable improvement can be felt. In light of this, virtual sensors are proposed not only to solve this limitation but also to broaden the control bandwidth [Kestell, 2000]. Additionally, they are designed to track the motion of one's head and translate the control region accordingly. This is done by predicting the forward wave. However, the work is limited to a one-dimensional field and is highlighted of the potential challenges in extending it into a three-dimensional field. Lane *et al.* [2000] considered otherwise by attempting to improve the performance of the controller for better cabin noise control with more

complex algorithm. Apart from commercial aircrafts, similar studies are done on a military cargo aircraft using a feedforward control system [Gerner *et al.*, 2003; Kochan *et al.*, 2010, 2011].

In the automotive industry, ANC is also reported in numerous works. The feedforward control system is demonstrated on the front seats of a commercial car by Sutton *et al.* [1994]. Despite the effectiveness of the control system below 200 Hz, they highlighted that it is not sufficiently robust and reliable as instability occurs even with a slightest delay of milliseconds. Lee *et al.* [2014] and Oh *et al.* [2002] performed similar studies by considering, instead, improved algorithms. Evidently, earlier works validated the potential of ANC but proper consideration of distributing the secondary sources and error sensors in the cabin has yet to be considered. Stanef *et al.* [2004] demonstrated an optimization study in a mining vehicle. They found that significant cabin noise reduction can be achieved if the secondary source is placed in close proximity to the primary source. Their findings resonate with the earlier observations by Elliott *et al.* [1989]. If this distance exceeds half the wavelength of interest, negligible reduction will be noticed due to high modal overlapping. Moreover, this issue cannot be solved by simply introducing more secondary sources or error sensors. As such, they concluded that global control is not feasible in a three-dimensional field especially for scenarios where modal density is high in the low frequency range. This clearly identifies the limitation of ANC for cabin noise control in spite of the decent acoustical performance in the lower frequency range.

Till date, the idea of ANC is not as well-received as expected in the automotive industry due to high cost of implementation and complexity of cabin sound field [Rao, 2003]. This especially repels consumers away in the purchase consideration of low- and mid-end cars, where extra cost is added into the total purchasing price. A brilliant attempt to reduce the installation cost of feedback control system is proposed by incorporating it into the in-car entertainment system [Cheer and Elliott, 2013]. They developed three different kinds of feedback control systems of varying complexity and validated their performance in a commercial car. The idea is certainly innovative by exploiting existing built-in audio systems, which is almost a necessity in any typical commercial car for musical pleasure. However, no details are provided pertaining to the estimated cost of each controller.

The drawbacks are clearly identified by these works with more evidences from review papers [Elliott, 2008; Hansen, 2005] and magazine articles [Hughes, 2009; Noise & Vibration Worldwide, 2002] highlighting numerous hurdles to overcome before ANC is welcomed with open arms by the automotive industry. Some of these reasons include complex hardware configurations, high installation/implementation cost, non-user-friendly software, complex algorithms, unique controller for different cabins, need for external power source, and non-ruggedized control systems. It is highly recommended that one should consider ANC only if attempts in implementing passive solutions are exhausted or impractical since the return of investment

for ANC is highly uncertain. Noting the last point from the list of challenges, the application of ANC in armored vehicles is even more challenging if the control system is not sufficiently ruggedized to withstand harsh environmental conditions experienced commonly during mobility.

2.3.2. Active structural acoustic control

ASAC is more advantageous as compared to ANC due to its characteristic in controlling both cabin noise and structural vibration simultaneously. ANC does not influence structural vibration and hence if cabin noise is dominated by sound radiation of the panels, the acoustical performance of the control system may not be that effective. Generally, the working principle is similar to ANC. Accelerometers are attached onto the targeted panel to measure the vibration amplitude due to a sound source. The measured signal is then transmitted to a controller to generate an anti-phase signal, which becomes the vibration output of a mechanical actuator attached to the same panel. As such, a secondary vibration input onto the panel to reduce its vibrating amplitude and minimize sound radiation into the cabin. Accelerometers are placed on the panel to capture and feedback the corrected vibration signal to the controller. This approach is more common in the aerospace industry with applications extending to space shuttles; whereas literature is very limited in automotive applications.

Fuller and Jones [1987] first demonstrated the potential of ASAC on a cylindrical enclosure experimentally. Their control system involves a mini-shaker and several accelerometers mounted onto a targeted panel. An external sound source simulates tonal engine noise into the enclosure, which in turn excites the encompassing panels into vibration. Accelerometers pick up and transmit the vibration signal to a controller, which eventually generates an anti-phase signal to the mini-shaker. As a result, sound radiation from the targeted panel is minimized. To their surprise, global interior noise control is observed albeit only one actuator is used. They then extended the work to a full-scale aircraft for validation [Simpson *et al.*, 1991]. Concurrently, another research group demonstrated analytically and experimentally at a higher level of detail in a rectangular enclosure; including optimizing the position of the actuator [Pan *et al.*, 1990; Pan and Hansen, 1991; Pan, 1991]. They found that the optimum performance can be achieved from more than one position.

Earlier works proved the feasibility of ASAC for cabin noise control but mechanical shakers are typically bulky and heavy. Hence, piezoelectric patches are proposed in place of mechanical shakers to achieve the same functionality [Fuller *et al.*, 1992; MacMartin, 1996; Mathur and Troi, 1993]. The performance of piezoelectric patches are proved to be comparable to mechanical shakers if properly designed, which brings the technology of ASAC to a whole new level with reduced bulkiness [Sun *et al.*, 1996]. Apart from being compact, piezoelectric patches require lesser external power to be driven as compared to mechanical shakers [Balachandran *et al.*, 1996]. Expectedly, piezoelectric patches are in the limelight and mechanical shakers are

replaced gradually in ASAC systems with time. This is evident from later literature found on aircrafts [Grewal *et al.*, 2000; Kincaid *et al.*, 1997; Palumbo *et al.*, 2000], payload fairing [Niezrecki and Cudney, 2001], and commercial cars [Sas and Dehandschutter, 2009; Song *et al.*, 2003].

At this juncture, it is understood that cabin noise can either be achieved by ANC or ASAC. However, there is no comparative work to distinguish them clearly with regards to their respective benefits. In light of this, Sas and Dehandschutter [2009] attempted to answer this with a demonstration on a commercial car. Based on their numerical and experimental results, they found that ANC is able to achieve higher cabin noise reduction but limited to the vicinity of the error and reference microphones. This is in agreement with earlier observations by Bullmore *et al.* [1990]. Conversely, ASAC provides a more consistent global cabin noise reduction. From here, light is shed for researchers to consider each approach depending on their objective. Misol *et al.* [2012] considered differently another comparative study to understand and distinguish the acoustical performance of feedback and feedforward controllers by implementing them on a windshield of a commercial car experimentally. Although comparable results are achieved by both types of controllers, the effectiveness reduces when the vibrating pattern of the windshield becomes more complex. This is somewhat similar to one of the drawbacks of ANC with reduced performance in complex cabin sound field.

Among the various active solutions, ASAC still stands out as it is able to kill two birds with one stone by reducing both structural vibration and cabin noise simultaneously. However, similar to other active control systems, they do have their shortcomings. Also, ASAC may be detrimental to the structure as secondary vibration input may induce additional fatigue damage [Hirsch *et al.*, 1998]. This shortens the life of the structure, which eventually leads to increased maintenance cost. In the perspective of armored vehicles, the consideration of ASAC may not be well-accepted than ANC since increased fatigue damage into the hull may indirectly affect crew survivability.

2.4. Numerical methods

First and foremost, it should be noted that the emphasis of this paper is to provide an overview of the current practices used in the industry for the study of cabin noise. As such, a detailed account of past analytical studies is of irrelevance to this paper. Moreover, the industry relies dominantly on the usage of computational software rather than analytically obtaining results from mathematical formulations. A tremendous amount of time will be consumed in reality since cabins are usually irregularly shaped with complex boundary conditions [Atalla and Bernhard, 1994]. Conversely, one should not forget the importance of analytical frameworks developed by early researchers since the success of computational software is built based on them. Extensive literature can be referred to for interested readers to understand the analytical aspect of numerical methods [Dhandole and Modak, 2007; Dowell

et al., 1977; Everstine, 1997; Kipp and Bernhard, 1987; Kopuz and Lalor, 1995; Missaoui and Cheng, 1997, 1999; Nefske *et al.*, 1982; Pan *et al.*, 1999; Petyt and Koopmann, 1977; Richards and Jha, 1979; Sestieri *et al.*, 1984; Shuku and Ishihara, 1973; Succi, 1987; Wu *et al.*, 2000].

Generally, there are three main numerical methods that are adopted widely by the industry for cabin noise prediction. They are namely finite element method (FEM), boundary element method (BEM), and statistical energy analysis (SEA). Geometrical acoustics techniques, such as ray tracing or beam tracing, are typically used for architectural (or room) acoustics. Although a room is essentially an enclosure, these techniques are only accurate if the targeted wavelength is way smaller as compared to the dimensions of the enclosure [Lalor and Priebisch, 2007]. Thus, they are unsuitable for typical cabins, which are much smaller in size than a room or hall. As such, it is beyond the relevancy of this paper to include geometrical acoustics techniques.

2.4.1. Finite element method and boundary element method

From literature, it is evident that analytical works have declined since the late 1990s as later reported works start to make use of computational software; developed based on the analytical efforts of early researchers. Kim and Lee [1998] attempted to develop a FEM software to predict cabin noise of a scaled commercial car. They validated their software experimentally by considering several boundary conditions and showed the feasibility of their software for such study. However, Cherng *et al.* [2002] argued otherwise with the importance of using formulations to predict acoustic modes of a given cabin analytically using classical formulations. Despite their argument, it is evident in their work that analytical accuracy drops with increasingly complex cabin shape; prominently at higher-order modes. In support of their standpoint, they stated that accurate predictions are not really essential during the design phase of an automobile, where approximated results are reasoned to be sufficient. In argument, their views may no longer be valid when the coupling effect of vibroacoustics is considered. Moreover, a physical full-scale prototype is usually unavailable for validation in industrial reality. As such, accuracy of the model is mainly governed by the working experience of the analysis engineer to define appropriate boundary conditions and assumptions. Hence, if the deviation of the model is too large, drastic design changes may be necessary after the validation stage, which is most probably too late. Even if it is still possible in the prototype stage, higher cost will be incurred for design changes.

As a rule-of-thumb, the recommended number of elements per wavelength is ten for an accuracy of about 10% [Nefske *et al.*, 1982]. Later, this is debated in detail by Marburg [2002] with six being reasonably acceptable. This greatly reduces computational time for larger models. Since the frequency range of interest is generally below 200 Hz for cabin noise studies, coarse mesh is more than sufficient for accurate prediction of acoustic modes. However, to consider the coupling effect of

vibroacoustics, finer mesh is more favorable. This is usually done by combining FEM and BEM (FEM-BEM). Unlike FEM, BEM (based on Helmholtz integral equation) requires only the boundary conditions and discretization of encompassing panels to predict either interior or exterior noise at an arbitrary acoustic field point. As such, the original three-dimensional FEM problem is reduced to a two-dimensional BEM problem. In this manner, preparation and computational time are greatly reduced [Atalla and Bernhard, 1994]. BEM can be further categorized into two different types — namely direct (DBEM) and indirect (IBEM). The former considers a closed volume and predicts either the interior or exterior noise at one time while the latter is able to predict either side simultaneously and does not require a closed volume.

An example is demonstrated on a harvester cabin by Desmet *et al.* [2003]. They considered between coupled, semi-coupled, and uncoupled models and matched the results against field measurements to understand the accuracy of each model. They found that the consideration of either semi-coupled or coupled models is crucial in achieving accurate predictions. Their observation is further supported by the results of Liu *et al.* [2006] with their comprehensive study on an armored tracked vehicle. They considered a semi-coupled model as the influence of structural modes on acoustic modes is claimed to be more dominant than the other way round. The agreement of their numerical and experimental results validated their assumption. Kim *et al.* [2012] and Wang *et al.* [2010] demonstrated similarly on construction vehicles and bus respectively.

Liu *et al.* [2006] proposed the use of structural intensity to visualize vibrational power flow in the encompassing panels of a scaled automobile in the form of streamlines; analogous to that used in fluid dynamics for fluid flow studies. From the visualization, they proposed the placement of a damper at critical spot to channel the flow of vibrational energy towards it. Seemingly, vibration amplitude at selected panels is reduced. However, their work remains as a numerical study without any further work in the same industry. A more efficient FEM-BEM approach is proposed using acoustic transfer vectors (ATV) and modal acoustic transfer vector (MATV) [Citarella *et al.*, 2007]. ATV allows the generation of a transfer function between a structural field point and an acoustic field point in the cabin. This is advantageous because the recalculation of sound pressure level at a specific field point in the acoustic field is no longer necessary since the transfer function is valid as long as the acoustic field remains unmodified. This transfer function is not affected by any change in excitation forces on the structure or modifications. As such, computational time is reduced. The approach based on MATV works on a different concept where the transfer function is generated based on the modal response at a structural field point and an acoustic field point in the cavity. This provides a convenient approach as modal response of the structure, which is obtained either from modal test or numerical prediction, can be imported into the model. Hence, the need to compute normal surface velocities of the structure is not required. The

drawback of this approach is its dependency on modal response, where the transfer functions become invalid upon any structural changes.

The focus from most of these works tends towards the prediction of cabin noise or acoustic/structural modes. Minimal attention is placed on improving cabin noise subsequently with the predicted results. In light of this, researchers took a step forward mainly by structural modifications. This is preferred in the design phase since the process of remodeling is relatively easy. In contrary, the process may be iterative if the analysis engineer has insufficient working experience. Generally, it involves the addition of stiffeners to increase the vibrating frequencies of selected panels. On a downscale basis, Demirkan *et al.* [2010] and Park *et al.* [2010] demonstrated separately on an enclosure whereas others demonstrated on commercial vehicles [Olatunbosun *et al.*, 2011; Sung *et al.*, 1994] and tractor [Cao, 2005]. On the side note, some proposed differently by modifying the trim in commercial cars [Choi *et al.*, 1997; Kang *et al.*, 2000] or by optimizing the boundary conditions of stiffened panels [Chandra *et al.*, 2015; Zheng and Wei, 2013]. All these works showed the effectiveness of structural modification in decoupling structural modes from acoustic modes. However, the added stiffener may not be always effective if its position on the targeted panel is not properly considered and optimized.

Lately, panel acoustic contribution analysis (PACA) is used as an additional tool of the FEM-BEM approach. It determines the significance of a panel's contribution to a specific acoustic field point in the cabin by categorizing the panel as positive, neutral or negative. Positive indicates an increase in sound pressure at an acoustic field point when the vibration amplitude of a panel increases. Negative indicates likewise for the sound pressure but when the vibration amplitude of a panel decreases. Neutral is rather self-explanatory, which simply indicates no influence on the sound pressure with any modification to the panel. The focus of structural modifications or acoustical treatments must be placed on the positive panels since their vibration amplitude directly increases the pressure level at an acoustic field point in the cabin. Its benefits and conveniences have been proven by recent works mainly on commercial vehicles [Guo *et al.*, 2012; Han *et al.*, 2012; Jahani *et al.*, 2013; Xu *et al.*, 2015; Yuksel *et al.*, 2012]. The optimization of cabin shape may also be considered as proposed by Zhang and Cheng [2015].

2.4.2. Statistical energy analysis

Being a statistical approach, the underlying concept of SEA is dissimilar from deterministic approaches like FEM and BEM. The acoustic cavity and encompassing structures are divided into network of coupled subsystems; each having similar number of same mode types. Solutions are obtained based on the principle of energy conservation. This computation is usually faster than deterministic approaches since the number of degree of freedoms is further reduced [Vige, 2010]. As such, the use of SEA is beneficial either for acquiring quick results for an iterative design process or matching cabin noise against safety regulations for the auditory system of humans.

One such example is demonstrated on a commercial car by Chen *et al.* [2011]. They showed good agreement against experimental results from 200 Hz to 20 kHz and concluded that the feasibility of SEA for high frequency cabin noise analysis. Their conclusion resonates with that of Atalla and Bernhard [1994]. Marzbanrad and Beyg [2012] attempted likewise on another commercial car albeit they lacked validation of their numerical results. SEA is also applied for similar intention on ship [Pang *et al.*, 2014] and trains [Forssen *et al.*, 2012; Zhao *et al.*, 2013]. Interest in improving the methodology of SEA is still ongoing; evident from recent literature [Putra *et al.*, 2015]. On the other hand, others attempted a different approach in improving SEA by combining it with FEM to achieve higher prediction accuracy in the lower frequency range [Blanchet and Caillet, 2014; Chen *et al.*, 2011; Lalor and Priebisch, 2007; Zhao *et al.*, 2013]. As such, the entire audible range of hearing can be covered (20 Hz to 20 kHz). At this point, one may notice the lack of literature pertaining to armored vehicles since, factually, concerns are always in the lower frequency range (< 200 Hz).

2.5. Dimensional analysis

This subsection is included for the intent of coverage since some researchers have been attempting to predict cabin noise using a different approach. In the industry, physical prototypes are usually not available in the design phase and justifying for a mock-up to be fabricated solely for noise and vibration studies may sometimes be difficult due to additional design cost. With this in mind, researchers [Georgiev *et al.*, 2006a, 2006b; Krylov *et al.*, 2003; Krylov, 2002, 2004] proposed to apply dimensional analysis to obtain a simpler downscale model of an automobile, which possesses a different cabin shape as the original. The downscale model allows them to perform prediction of cabin noise and correlate the results back to the full-scale cabin. There are two main considerations in performing cabin noise prediction in downscale models — degree of model's complexity and accuracy of prediction. One should note that increasing the degree of model's complexity may indeed yield more accurate results but in the expense of computational cost. A balance has to be achieved between these two factors.

In their work, the downscaled model is known as quarter-scale interior cavity acoustic rig (QUASICAR). First, an analytical model is derived based on the principles of Green's function, Helmholtz equation and acoustic wave equation to predict the interior noise level of the QUASICAR. In their analytical model, they assumed that the feedback of acoustic response has no influence on the structural response of the boundary and smallest radius of the plate curvature is large in comparison with the flexural wavelengths of interest so that shell-type behavior shall not be accounted for. Subsequently, they extended to experimental investigation and found that the measured acoustic modes are of good agreement with their analytical prediction but is otherwise for the structural modes. This disagreement is claimed to be caused by the negligence of structural inconsistency. In theory, scaling of pure acoustic

modes is usually accurate; if the original cabin is downscaled by a specific numeric value, the original frequency of each acoustic mode will generally shift upwards by the multiple of this numeric value. This is similar in the context of a simple panel. However, in complex structures, it is not that simple because structural vibration fields are a combination of different elastic waves with some being dispersive and some being otherwise. Hence, it is difficult to establish a linear relationship.

Evidently, numerous analytical, numerical, and experimental considerations are crucial to arrive at the proposed QUASICAR model. This, in turn, makes the whole process laborious for the industry to even consider. Therefore, practicality of this approach in automobiles or armored vehicles is still far away since much time may be consumed to fine-tune the model.

3. The Emergence of Acoustic Metamaterials

Acoustic metamaterial (AM) is a recent research topic and is still in its infancy stage. Since the start of the current millennium year, there has been a rapid rise in research interest on AMs due to the exceptional phenomena exhibited for the first time in history. The mechanical properties of such metamaterials are basically governed by their carefully engineered physical structure rather than their chemical composition, which is the case in natural materials for the latter. Their construction is analogous to traditional honeycomb panels, which consists of periodic unit cells. In addition, each unit cell has a locally resonant structure, which exhibits exceptional phenomena at resonance; either negative effective bulk modulus or negative effective mass density. If properly designed, both characteristics can be achieved and the propagation of sound waves will be prohibited through the AM.

Dating back to 1960s, Veselago [1968] first demonstrated the preliminary concept analytically, where a negative refraction index is attainable if a medium exhibits both negative permeability and negative permittivity simultaneously. Unfortunately, his mathematical formulations remained as an abstract for over three decades before being realized experimentally by Pendry [2000]. In his work, he challenged the limitations of conventional lens by proposing a so-called nanoscale “superlens”, which exhibits negative refraction index in the frequency bandwidth of visible light. At that moment, such phenomenon is only observed in the frequency bandwidth of microwave. During that period, most works focused on a frequency range that is in orders of magnitude higher than the audible frequency range (20 Hz to 20 kHz) of humans. Liu *et al.* [2000] deviated from the mainstream and proposed an AM that attenuates noise in the mid frequency range (400 and 1100 Hz) experimentally. The AM composes of periodically arranged high density spherical cores (lead) uniformly coated with a layer of soft and elastic material (silicone rubber). By specifying the size and geometry of each unit cell, noise attenuation at targeted frequencies is achieved. Their work was, however, a small-scale study in the laboratory. Their pioneering work brought forth a new field of research in various disciplines such as transformation optics [Jiang *et al.*, 2009; Kinsler and McCall, 2015; Lu *et al.*, 2009],

energy harvesting [Chen *et al.*, 2014], noise barriers [Garcia-Chocano *et al.*, 2012; Garcia-Chocano and Sanchez-Dehesa, 2013; Ho *et al.*, 2003; Lagarrigue *et al.*, 2013; Sanchez-Dehesa *et al.*, 2011] and vibration/shock mitigation [Akl and Baz, 2012; Baravelli and Ruzzene, 2013; Huang and Sun, 2010; Huang *et al.*, 2009; Huang and Sun, 2009; Kim *et al.*, 2015; Pai *et al.*, 2014; Peng and Pai, 2014; Reynolds and Daley, 2014; Sharma and Sun, 2015; Tan *et al.*, 2012].

The consideration of using AMs specifically for cavity noise control has been reported recently in two separate works. The first work proposed a one-dimensional AM, which constitutes of a five-mass array with each attached to a slender stem mounted onto the bottom surface of a small rectangular enclosure [Islam and Newaz, 2012]. The other work fabricated a cubic enclosure with its interior surfaces made up of periodically arranged locally resonant structures [Claeys *et al.*, 2014]. Both successfully demonstrated the excellent acoustic attenuation of AM in the mid and high frequency range. Recent works moved to explore the possibility of low frequency noise. Two of such examples include the work of Naify *et al.* [2010, 2011a, 2011b, 2012, 2013] and Mei *et al.* [2012]. In general, their proposed AM constitutes of each unit cell being a thin membrane with a single or multiple platelets attached to its surface. This is better named as a membrane-platelet assembly. By tuning the mass of the platelet, noise attenuation can be achieved at targeted frequency below 200 Hz. Yang *et al.* [2010] demonstrated noise attenuation at as low as 50 Hz.

Till date, no existing natural materials are able to exhibit similar behavior at such small physical dimensions. This becomes advantageous when design requirements turn out to be so stringent in certain scenarios that no natural materials meet the necessary acoustical properties. From literature, it is evident that all these works remained as a laboratory study. As such, an extensive amount of exploratory works is definitely imperative to realize the full potential of AMs. Moreover, although positive results are observed from these reported studies, one should also include other vital factors in their design such as product reliability and ease of manufacturing. If the design is unreliable in practical applications or under certain environmental conditions, an increase in maintenance rate is likely to be inevitable. Clearly, this is undesirable for any potential investors. Additionally, the production of highly complex AMs may be challenging despite the availability of existing additive manufacturing techniques. Apart from this, the selection of the printing material must be cautiously considered as well since it affects the overall weight of the design. Generally speaking, designing an AM to meet industrial requirements — particularly for the defense platforms — may likely be a huge hurdle to overcome mainly due to the exposure to harsh environmental conditions and possible restriction of troops mobility within the vehicle.

4. Conclusion

From literature, it is evident that there are numerous approaches for tackling such problems; each having its own benefits and drawbacks. Psychoacoustics may be a

good approach but is only feasible when the cabin noise is not excessively high such that safety exposure limit is exceeded. This is likely the case in armored vehicles since crew noise is usually on or near the borderline of safety exposure limit of 85 dBA. As such, higher priority should always be placed on crew noise reduction rather than perception studies and/or noise signature shaping.

Dimensional analysis can be a great tool in predicting cabin noise and structural/acoustic modes but when considered in detail, it is clearly time consuming and tedious to fine-tune the downscale model. Instead, these predictions can be performed more efficiently and effectively by numerical methods — namely FEM and BEM. Furthermore, the addition of PACA seems to be a highly reliable approach in identifying problematic panels that are contributing largely to the cabin noise. However, its functionality is only limited to source detection and hence further structural modifications or acoustical treatments are still necessary to improve cabin noise. These are generally done by passive or active solutions.

For active controls, their acoustical performance in the low frequency range (< 200 Hz) is evident as proven by numerous works. However, they are yet to be well-accepted by the automotive and defense industry due to several challenges such as complex hardware configurations, high installation/implementation cost, nonintuitive software, complex algorithms, unique controller for different cabins, need for external power source, and nonruggedized control systems. As such, passive controls are still more reliable and stable in terms of acoustical performance. This is analogous to the preference of a fully-mechanical system as compared to a mechatronics system in the context of industrial product design. Active controls will be even more challenging to be implemented in armored vehicles as compared to commercial vehicles since they have to be sufficiently ruggedized to withstand harsh environmental conditions experienced commonly during mobility. Clearly, more time and research efforts are required to prove its practicality.

For passive controls, they are widely used in various industries, which is evident from their reliability and stability over active controls. However, they are not as effective in attenuating low frequency noise (< 200 Hz). For example, a typical sound absorption material will require a very thick depth to at least reduce cabin noise noticeably at such long wavelength. As such, cabin space and vehicular weight are affected. Moreover, they deteriorate at a faster rate with time when exposed to harsh environmental conditions. This adversely affects their optimum acoustical performance and lead to higher maintenance cost, which is typically the case in armored vehicles. As also highlighted, other passive solutions face similar issues.

In light of the drawbacks of current industrial practices, the potential of AM is highlighted as a possible alternative for low frequency cabin noise control due to its exceptional phenomena. As highlighted, numerous works have demonstrated its acoustical performance in the lower frequency range without requiring a very thick specimen that is comparable to the wavelength of interest. The main challenge

is the extension of existing works out of the laboratory to explore the feasibility of practical applications. As low frequency cabin noise is still a concern to date, preliminary works on AM may shed some light on its potential as an alternative solution to improve cabin noise in automobiles and armored vehicles; leading to better passenger comfort. Apart from merely achieving attractive results in the low frequency range (< 500 Hz), the consideration of a few essential factors pertaining to manufacturability and reliability should also be considered for realization. Consequently, if these factors are considered in the design process of an AM, a potential approach for cabin noise control in automobiles and armored vehicles may emerge in the future.

Acknowledgments

The authors gratefully acknowledge the financial support provided by Singapore Economic Development Board (EDB) and Singapore Technologies Kinetics Ltd (ST Kinetics) under the EDB Industrial Postgraduate Program (EDB-IPP) (Grant No. COY-15-IPP/120005).

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