A review of noise and vibration control technologies for rotorcraft transmissions

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ABSTRACT

An expanded commercial use of rotorcraft can alleviate runway congestion and improve the accessibility of routine air travel. To date, commercial use has been hindered by excessive cabin noise. The primary noise source is structure-borne vibration originating from the main rotor gearbox. Despite significant advancements in gear design, the gear mesh tones generated often exceed 100 dB. This paper summarizes the findings of a literature survey of vibration control technologies that serve to attenuate this vibration near the source, before it spreads into the airframe and produces noise. The scope is thus limited to vibration control treatments and modifications of the gears, driveline, housing structures, and the strut connections to the airframe. The findings of the literature are summarized and persistent and unresolved issues are identified. An emphasis is placed on components and systems that have been demonstrated in flight vehicles. Then, a discussion is presented of emerging technologies that have the potential to make significant advancements over the state of the art.

Keywords: cabin noise, gear vibration I-INCE Classification of Subjects Number(s): 13.1.4, 40

1. INTRODUCTION

Civilian rotorcraft can improve the accessibility of routine air travel by providing point-to-point transportation to locations without runways. Their short field length also gives rotorcraft the potential to reduce airport congestion. However, rotorcraft are not widely utilized due to limitations in reliability, payload capacity, and noise levels inside the cabin. Advanced driveline noise and vibration control technologies can improve the passenger acceptance of rotorcraft while reducing dynamic loading, thereby increasing service life. Alternatively, these technologies can be viewed as mass reductions for a constant noise and vibration design if cabin noise treatments can be replaced by lighter weight driveline solutions. Figure 1 depicts a typical cabin noise spectrum. Tonal noise levels can exceed 110 dB, which can damage hearing and make it impossible to communicate. Shaft and blade passage harmonics of the main and tail rotors produce tones below 200 Hz. Gear mesh tones, which dominate between 500-4000 Hz, are more objectionable to humans, because they reside in a more sensitive range of human hearing and overlap with the speech band. Further, gear mesh frequencies are modulated by rotor frequencies (1), resulting in rough, "grinding-like" sounds.

In rotorcraft, the majority of power flows from the gas turbine engines, through the gearbox – where speed is reduced by about 60:1 to 100:1 – and mast, to the main rotor. The main rotor suspends the weight of the aircraft through the mast bearing's connection to the fuselage. Vibration generated at the gear meshes transfers through the gears, shafts, and bearings into the transmission housing, which is directly "hard" mounted or strut mounted to the fuselage. As shown in Figure 2, the transmission mounts provide a structure-borne path for the vibration to reach the fuselage and generate noise. If the cabin is not fully sealed, the housing vibration can also cause cabin noise by direct radiation.

Table 1 summarizes cabin noise measurements in military (M), civil (C), civil-utility (U), and civil-research (R) helicopters. The peak level and associated octave within the gear-dominated

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frequency range are reported. One limitation of this dataset is the age of the measurements. Four key observations can be made. First, maximum noise levels fall within the 500 or 1000 Hz octave bands. Second, military and utility vehicles (85-110 dB) are often louder than civil vehicles (86-94 dB). Third, military vehicles tend to get louder with size. This is hypothesized to result from increased power and vibration levels within the powertrain. Fourth, larger civil rotorcraft are quieter than smaller ones. Larger civil vehicles may have more effective sound proofing in the cabin, possibly due to having more physical space or a higher mass allowance for passive cabin treatments. This claim is supported by direct comparisons of military-civil equivalents, which are listed in the same row: -4 dB change in the small weight class (Lynx vs. WG30), -11 to -14 dB change in the large weight class (Sea King vs. VIP Commando and S-61N), and -34 dB change in the very large weight class (CH-53A vs. CHRA).

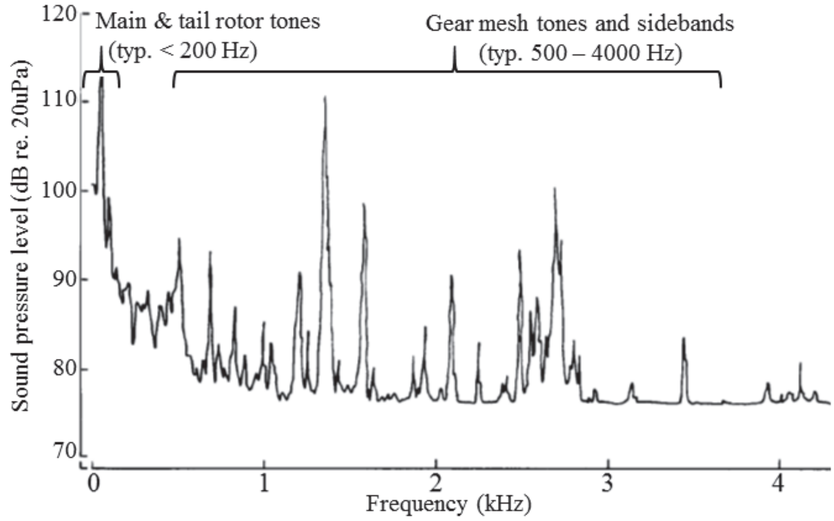

Figure 1 – Example measured noise inside of a helicopter cabin (2)

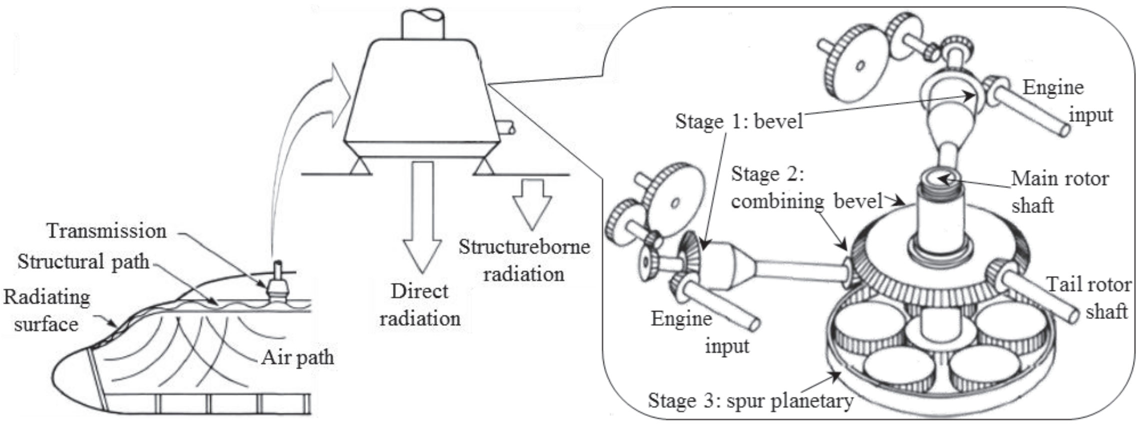

Figure 2 – Schematic of helicopter transmission noise paths; schematic adapted from (3, 4)

A typical main rotor powertrain consists of 1 or 2 engines mounted horizontally and connected to a multi-stage speed reduction transmission. At least one stage is a bevel or spiral bevel gear set to convert roll rotation into yaw rotation. The final stage is a bull gear set or planetary system to withstand high torque loads. To identify the gearing stages and types that cause the most significant cabin noise, Table 2 summarizes the mesh frequencies and dominant cabin noise tone for a variety of helicopters. In 7 out of the 8 cases, the lowest gear mesh frequency is responsible for the peak noise. This is true, because the vibro-acoustic transfer path from the gearbox to the cabin has a low pass characteristic (3, 5). These dominant tones also have the most harmonics within the speech band. Consequently, the low-speed final stage gearing, either planetary or bull gear sets, has the most significant impact on the cabin noise environment.

The scope of the literature review is limited to vibration control treatments and modifications of the gears, driveline, transmission housing, and the strut or mount connection to the airframe. Technologies that are implemented in the rotor blades, rotor hub, gas turbine engines, or fuselage are not considered. Solutions that address gear-mesh-induced noise and vibration are strongly emphasized. Since rotor-induced vibrations propagate to the cabin through the transmission and its mounts, select rotor vibration technologies are briefly discussed.

Table 1 – Cabin noise induced by gearing; peak levels (dB re. 20uPa) and associated octave bands are shown

Vehicle	Ref.		Max gross weight, kg	Peak, dB	Band, Hz	Comment
Bell OH-58C	1987 (3)	M	1,451	85	573	Single tone measurement
Augusta A-109	1980 (6)	C	2,559	90	1000	
Westland Lynx	1980 (6)	M	3,291	98	500	Common powertrain
Westland WG30		C	5,806	94		
Sikorsky S-76A	1986 (5)	U	4,587	105	1000	
Bell 212	1980 (6)	U	5,080	103	1000	
Aérospatiale Puma	1980 (6)	M	7,000	103	1000	
Sikorsky Sea King	1980 (6)	M	10,000	102	500	
Westland VIP Commando		C	9,707	89		Common platform
Sikorsky S-61N		C	8,620	86		
Sikorsky CH-53A	1977	M	15,876	110	1000	CHRA was a CH-53A w/
NASA-Sikorsky CHRA	(2, 7)	R	15,876	76		custom sealed cabin

Table 2 – Mesh frequencies associated with helicopter gear stages (most dominant cabin noise tones in bold)

Vehicle	Mesh frequencies, Hz	Vehicle	Mesh frequencies, Hz
Aérospatiale Puma	Spur 2-pair Helical 2-pin/1-gear: ~4550 Bevel pair: ~1750 Planetary stage 1: ~1600 Planetary stage 2: ~550, ~**1100**	Sikorsky Sea King S-61N Westland VIP Commando	Spur 2-pair: 13,325 Helical 2-pair: 5,968 Spiral bevel: 1,369 Planetary: **683**
Augusta 109	Combining stage Bevel pair: 1850 Planetary: **820**	Sikorsky UH-60 Black Hawk (8)	Bevel 2-pair Bevel 2-pin/1-gear: **1628** Planetary: **980**
Westland Lynx WG30	Combining stage Spiral bevel 2-pair: ~2150 Conformal 2-pin/1-bull: ~**450**	Sikorsky CH-53A NASA-Sikorsky CHRA	Bevel pair: 2710 Planetary stage 1: **1370** Planetary stage 2: 527
Sikorsky S-76A	Helical 2-pair Spiral bevel 2-pair: 1221 2-Spur/1-bull: **727.5**	Bell OH-58C	Spiral bevel pair: 1919 Planetary: **573**

The paper is organized as follows. First, the noteworthy technologies identified during the literature review are described and the key findings from their development are presented. Next, a summary of the technologies is given. Then, conclusions and our recommendations are enumerated.

2. NOISE AND VIBRATION CONTROL TECHNOLOGIES

This section presents a literature review of the noise and vibration control technologies for rotorcraft drivelines that have been reported in journal and conference papers, NASA and U.S. Army technical reports, and U.S. patents. The technologies are categorized as either "mature" or "emerging" based on their current level of development. Here, a mature designation is given to technologies that appear in production rotorcraft or have been demonstrated on rotorcraft or transmission test stands. The technologies are sub-categorized depending on whether they provide source control, i.e., transmission technologies (e.g., gears, shafts, bearings, and housing), or path control, i.e., the structure-borne vibration transmission path to the cabin (e.g., transmission mounts and struts).

2.1 Mature Technologies

2.1.1 Source Control

Over the past several decades, many gear technology advancements have reduced the four key sources of gear-induced vibration in rotorcraft: static transmission error (STE), mesh stiffness variation, friction, and the changing location of planets during rotation. Industry has minimized STE by implementing state-of-the-art gear tooth profiles. This includes involute spur and helical gears with topological modifications and Litvin's bevel and spiral bevel profiles (9), which reduce the influence of shaft misalignment and torque-dependent shaft deflection. Gear design must now consider the remaining sources to further reduce noise levels. Mesh stiffness variation is an inherent feature of toothed gears. In the 1st stage of the McDonnell Douglas Advanced Rotorcraft Transmission (ART), this issue was mitigated by using a face gear rather than a spiral bevel gear, resulting in a lower noise sensitivity to misalignment and a higher mesh frequency that was pushed outside the speech band (10). More generally, this variation can be reduced by improving the load sharing between teeth through increases in tooth contact ratio (CR). Oswald et al. (11) studied the effect of gear type and gear parameters on the noise produced by a 134 kW (180 hp) gearbox and concluded that total (profile plus face) CR was the most influential factor for reducing noise. Single helical gears were about 2-17 dB quieter than spur gears and about 4 dB quieter than double helical gears. Friction forces are produced in the off-line-of-action direction as gear teeth slide toward and away from each other, changing direction at the pitch point. These forces can be elevated when tip relief is used to reduce STE (12) and when thin lubrication films result from high stress and low entraining velocity conditions, which most likely occur in the final gearing stages. Hansen et al. (13) installed isotropically superfinished gears into the final two stages of a S-76C+. Laboratory measurements demonstrated vibration reductions of 7 and 3.7 dB in the 1st harmonic of the bull gear and spiral bevel gear meshes, respectively. Contrary to theory (12) and prior laboratory measurements (14), there was no improvement in the higher harmonics of the bull gear mesh. Planet phasing in planetary gears refers to the cancellation of mesh forces and sliding impulses that results from the selection of the planet configuration and tooth numbers. Kahraman (15) predicted the spectral content for a generic planetary design. Parker (16) established design rules for suppressing certain translational or rotational modes. Planet phasing in spur planetary systems has reduced noise by about 11 dB (17). This technology was state-of-the-art in the early 90s and had one of its early appearances in the McDonnell Douglas ART (10).

2.1.2 Path Control

A very influential class of transmission mounts is anti-resonant isolators, which isolate the cabin from low frequency rotor tones. These isolators, which act like series vibration absorbers, contain a mass whose inertial force introduces an anti-resonance in the isolator's force transmissibility. Flannelly (18) introduced this concept as the dynamic anti-resonant vibration isolator (DAVI), which uses mechanical amplification of the isolator's inertia as shown in Figure 3(a). This isolator has a low static deflection and weight and its anti-resonant frequency is independent of the source and receiver masses. However, the mechanical amplifier requires bearings and has a limited amplification ratio (5:1 to 10:1 (19)). Halwes and Simmons (20) patented the liquid inertia vibration eliminator (LIVE), which used hydraulic amplification to achieve ratios of 20:1 (20) to 50:1 (19) in a more compact and lighter form factor (Figure 3(b)). LIVE systems and its derivatives have been put into production on the Bell Models 427, 429, and 430, Agusta A109E, Eurocopter EC135, and Airbus H135 (19, 21). A variety of semi-passive fluid-based anti-resonant mounts have been proposed. More recently, Smith et al. (22) introduced the Smart Link, which is a "rigid" active LIVE system containing a piezoelectric actuator in parallel with the primary load-carrying spring. Full-scale laboratory testing demonstrated vibration reduction of 99.4% (44 dB) at the N/rev tone and 90% (20 dB) over the N/rev ± 15.5% frequency range using 135-215 W (0.18-0.29 hp) of power.

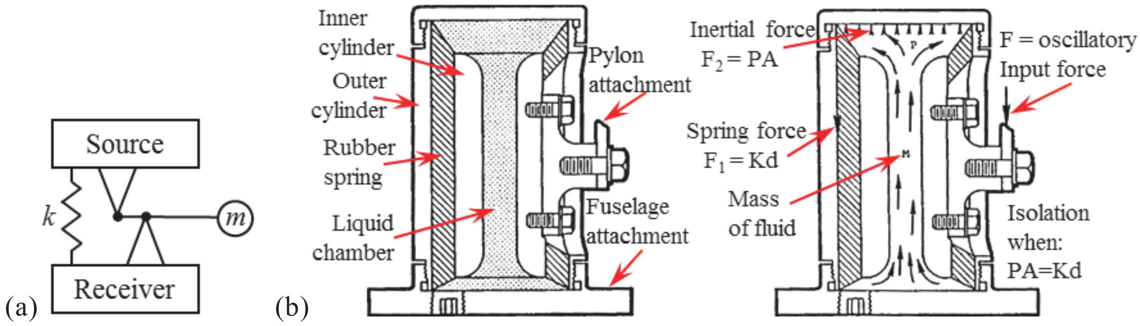

Figure 3 – (a) Flannelly's DAVI (18) and (b) Halwes and Simmons' LIVE system (20)

Active gearbox struts have been introduced for attenuating gear mesh tones. Sutton et al. (23) used three axially-oriented magnetostrictive inertial actuators to actively counteract the longitudinal and bending modes of a gearbox strut. Full-scale laboratory testing for small velocity disturbances demonstrated about 30-40 dB attenuations of the kinetic energy at the strut's fuselage mount over a 250-1250 Hz range when elastomeric (rather than steel) spherical bearings were used at the ends of the strut. Boundary conditions were found to have a strong impact on the open loop dynamic response of the experimental assembly. Maier et al. (24) flight tested a helicopter outfitted with a complete set of seven active gearbox struts. Each actuator was a single piezoelectric cylindrical shell bonded around a conventional strut. Testing during hover and forward flight at different cruising speeds showed mean cabin noise reductions of about 11 dB at the primary gear-mesh frequency (1895 Hz). Performance degraded when the actuator's stroke limit was reached at high cruising speeds. Ground testing showed 3-8 dB reductions when simultaneously controlling tones at 1485 Hz and 1895 Hz. Hoffmann et al. (25) developed an active gearbox strut that contained three axially-oriented piezoelectric actuators controlled with a narrowband, multi-channel variant of the FX-LMS algorithm. Flight testing demonstrated up to 19.5 dB reductions in the first gear-mesh frequency for all flight conditions and condition-dependent reductions of 4 to 8 dB in higher frequency gear-mesh tones. Performance was limited by decreased actuator performance at low frequencies and insufficient modal separation at higher gear-mesh frequencies. It was noted that without passive cabin treatment, airborne transmission of gear-mesh noise was significant.

Due to the difficulty of implementing active noise cancellation of gear mesh tones using speakers in the cabin, researchers have considered using actuation of the fuselage at the transmission mounting points. Millott et al. (26) and Welsh and Yoerkie (27) flight tested such an active noise control system on a Sikorsky S-76. The results demonstrated an average reduction of 18 dB in the primary gear mesh tone throughout the cabin during all steady flight conditions and an 8-14 dB reduction during maneuvers (acceleration from and deceleration to hover). It was noted that this technology would be used in the Sikorsky S-92 Helibus. Under the Army Combat Tempered Platform Demonstration, Sikorsky currently plans to use unbalanced mass actuators at the four corners of a UH-60A Black Hawk to actively reduce the vibration transmitted into the cabin.

2.2 Emerging Technologies

2.2.1 Source Control

Magnetic gear technology offers non-contact (friction- and wear-free) and relatively smooth motion transfer through magnetic fields (28). As a result, it has the potential to dramatically reduce gear vibrations, increase reliability, and reduce vulnerability to torque overload and loss of lubrication. Early designs, which mimicked traditional configurations (e.g., spur, planetary, etc.) with teeth replaced by magnets, had very limited torque capacity due to the limited number of engaged teeth (28). Newer coaxial magnetic gear designs (29) utilize magnetic poles (teeth) around the full circumference of the input and output rotors to increase torque capacity. Jian and Chau (30) used Halbach magnet arrays to further increase torque capacity while reducing torque ripple and eddy current losses (Figure 4(a)). Recently, Paden (31) manufactured and tested a 15 kW (20 hp) version of this design that had a specific torque (torque per unit mass) of 17 N-m/kg (68 lbf-in/lb), which is better than any prior prototype, but lower than an equivalent, non-magnetic planetary gearbox that was optimized (50 N-m/kg (201 lbf-in/lb)). When scaled to 300 kW (402 hp) for the Bell OH-58 helicopter's final reduction stage, the specific torque (112 N-m/kg (450 lbf-in/lb)) improves relative to a non-magnetic planetary design (136 N-m/kg (547 lbf-in/lb), calculated using empirical scaling laws (32) to predict weight reductions resulting from the use of modern technology).

Some references have proposed passive or active vibration control treatments for gears. Kish (33) developed a gear containing a circumferential elastomeric band to add torsional compliance for torsional isolation and ensuring even torque distribution in a split torque 2nd stage. Relative to an all-steel design, testing of a half-size gearbox demonstrated lower noise levels and 0.3-3.3 g (3-7 dB) reductions in housing vibration at the 1st and 2nd harmonics of the gear mesh frequency. The elastomeric spur gear exhibited lower dynamic transmission error (DTE) than a steel one, but the DTE of a helical variant was similar to its steel counterpart. However, temperature cycling caused changes to the elastomer's material properties and operational failure. Sammataro et al. (34) patented an isolated ring gear for planetary stages that contains circumferentially-oriented flexures to provide a torsionally-soft load path between the teeth and transmission housing. Simple modeling suggested that a 16 dB reduction in force transmissibility is possible at the mesh frequency. The gear meshes do not appear to be modeled and measurements were not presented. Guan and DeSmidt (35) studied the passive vibration reduction of a planetary gear stage using discrete support struts equally spaced around the ring gear's boundary. For input speeds of 0-4000 rpm, simulations showed that dynamic ring stresses and mesh forces reduce with the number of struts and that changes in the struts' stiffness can shift resonances, but not alter their magnitude. Chen and Brennan (36) considered a gear with three circumferentially-oriented inertial actuators in its body for actively controlling dynamic mesh forces caused by mesh stiffness variation. A model was used to show that the actuation force needed to eliminate torsional and translational vibration is similar in magnitude to the dynamic mesh force and depends on the CR of the gears and the linear speed and transmitted power at the mesh contact. Measurements demonstrated about 7.5 dB reduction in the angular acceleration of the gear at the fundamental mesh tone using model-based control signals calculated a priori. Guan et al. (37) modeled and theoretically compared four actuation concepts for actively reducing gearbox housing vibrations caused by STE excitation: (a) on-gear, inertial, (b) gear-shaft torsional, (c) direct active bearing, and (d) shaft transverse vibration. The required actuation effort, robustness of the FX-LMS control algorithm to measurement noise, and electrical power requirements were studied. Control of the shaft's transverse vibration was the most promising concept due to its relatively low sensitivity to noise and required electrical power, low to moderate actuation force, and improved ability to retrofit.

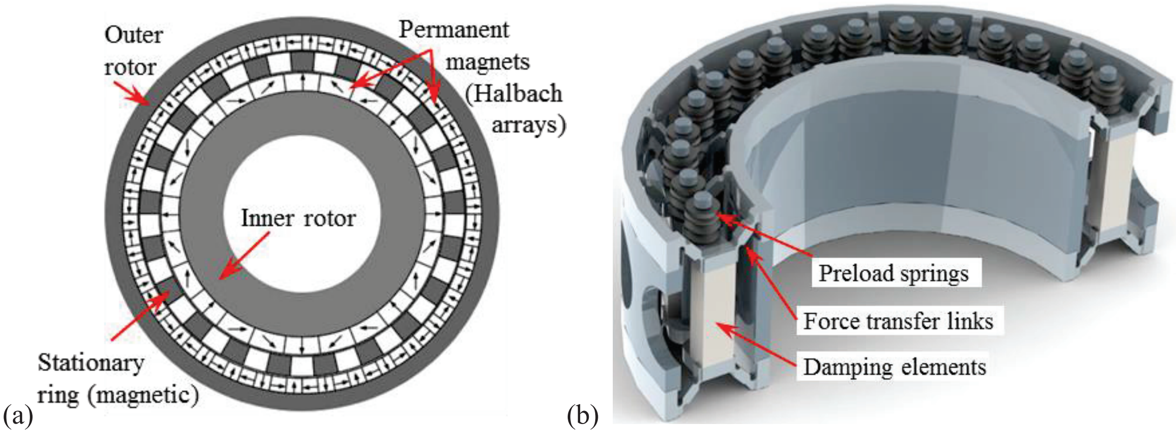

Figure 4 – (a) Jian and Chau's magnetic coaxial gear (30), (b) Asnani et al.'s piezoelectric vibration ring (45)

Active control of a gearbox shaft's transverse vibration by exciting auxiliary bearings along the line-of-action (LOA) direction of the gear mesh has been investigated by Montague et al. (38), Rebbechi et al. (39), and Guan et al. (40). Montague et al. (38) observed up to 75% (12.4 dB) reductions in housing vibration at a 4500 Hz spur gear mesh tone when transmitting 25-55 kW (34-74 hp) through the mesh and using feed-forward control of two piezoelectric actuators. Yet, housing vibration increased up to 208% (6.4 dB) at certain locations and shaft speeds. Rebbechi et al. (39) used FX-LMS control of two magnetostrictive actuators to demonstrate housing acceleration attenuations of the 1st (20-28 dB), 2nd (5-13 dB), and 3rd (0-2 dB) harmonics of the gear mesh frequency when transmitting 0-6 kW (0-8 hp) of power. Guan et al. (40) controlled a single piezoelectric actuator using an enhanced delayed-x LMS algorithm to reduce housing LOA acceleration by up to 18 dB when controlling the fundamental gear mesh tone and up to 2-6 dB when controlling the 1st and 2nd harmonics. However, the algorithm elevated the modulation sidebands.

Multiple researchers have used piezoelectric materials within bearings or shaft attachments for active control or to convert vibration energy into electrical energy, which is then dissipated in resistive circuits. The latter technique is called piezoelectric shunt damping. Zhao et al. (41) installed on a shaft a rotating, ring-like mass that excited a piezoelectric shunt damper with its inertial force. The tonal force transmitted to a benchtop driveline's housing was reduced by 7-9 dB when two devices were used on a shaft rotating at 60 rpm under no load. Two devices were needed to provide attenuation that was independent of angular position. With a total mass equal to 14% of the other rotating components, this design had a relatively high weight penalty. In another study, Zhao et al. (42) operated their piezoelectric device as an inertial actuator using the FX-LMS control algorithm. Transmission of a gear tone was attenuated by 9-17 dB while modulation sidebands were either amplified up to 7 dB or attenuated by 0-24 dB. Atzrodt et al. (43) developed a bearing housing in which the bearing forces are transmitted to the housing through four radially-oriented and equally-spaced piezoelectric shunt dampers that were tuned for narrow-band damping. Shaker testing of a prototype demonstrated up to 17.5 dB reduction in the velocity transmitted through the housing. Pinte et al. (44) developed an active bearing containing two preloaded piezoelectric stacks (each used as a sensor and actuator) oriented orthogonal to each other. Adaptive repetitive feed-forward control was used along with a conventional feed-back controller. Benchtop testing of a shaft supported by two conventional bearings and one active bearing was conducted while disturbing a passive bearing using a shaker. After a significant transient amplification, the steady-state force transmitted to the bearing's housing was reduced by 5-45 dB between 400-900 Hz, with peak performance at 700 Hz. Asnani et al. (45) created the vibration ring – a ring-shaped mechanical spacer containing many piezoelectric shunt dampers that can dissipate a portion of the radial vibration energy acting between any two driveline components (Figure 4(b)). The vibration ring can also harvest energy for rotating sensors. A lumped parameter model supplied with the manufacturer's electrically-measured piezoelectric properties suggested that the vibration ring's loss factor could range from 0.28-0.75. However, Asnani et al. (46) experimentally showed that the loss factor of piezoelectric stacks is significantly lower than that predicted using the manufacturer's data. Using realistic stack properties, finite element simulations of a refined design indicate that the mechanism can provide a system loss factor of about 0.05 when installed between a spur gear and a shaft and subjected to realistic boundary conditions and radial forcing. Deng et al. (47) have designed a magnetostrictive vibration ring; a prototype is currently being fabricated.

Periodic shafts have been proposed for isolating the transmission housing from gear mesh tones. Periodic structures are composed of repeating cells, each containing a material or geometric discontinuity. These structures act as mechanical notch filters, because the impedance mismatch at each discontinuity causes a portion of the vibration waves within a frequency stop band to be reflected back toward the source. Richards and Pines (48) modeled and tested shafts with geometric periodicity (discrete diameter changes). The model showed that the number and length of cells respectively determine the stop band's attenuation and spectral location. Vibration power reductions of 65-98% (9.1-34 dB) (over 0-2 kHz) relative to a uniform shaft were measured during laboratory testing of a 0.69 m (27 in) long shaft up to 0.75 kW (1 hp). The measurements also suggest that the periodic shaft may have reduced the vibration at the end of the shaft located adjacent to the excitation.

2.2.2 Path Control

Keningsberg and Eastman (49) and Yoerkie et al. (50) introduced elastomeric mounts to provide vibration isolation or damping of rotor vibration tones for hard-mounted helicopter transmissions. Although elastomers have a very low Young's modulus relative to metals, they are incorporated in thin, constrained layers to create sufficiently high axial stiffness (51). Using a very stiff test rig, Yoerkie et al. (50) demonstrated that the mount could attenuate vibration by 0-60 dB over 0-5.5 kHz; however, the performance is predicted to be considerably lower in a helicopter due to the softer boundary conditions. Millott et al. (26) noted that these technologies raised flight certification issues and were difficult to retrofit due to their significant impact on many system-level design considerations.

Periodic isolation struts and mounts have been developed for attenuating gear mesh vibrations. Asiri et al. (52) measured the vibration transmission through struts with geometric and material discontinuities. Distinct stop bands with attenuations of about 20-85 dB were observed during shaker testing of individual struts. Performance reduced to about 0-40 dB during isolation testing of four struts used to connect an electric motor/gearbox to a base. Asiri et al. (53) studied an active periodic strut composed of metal layers and piezoelectric actuators, which were controlled using velocity feed-back. Velocity transmissibility of a single active strut was 20-40 dB below unity from 650-4000 Hz (and 10-20 dB lower than a passive periodic strut up to about 1750 Hz). The velocity transmissibility through four active struts that supported a small gearbox was 10-30 dB lower than the passive strut from 0-500 Hz and 1100-1500 Hz. Using an analytical model of conventional periodic mounts containing elastomer and metal layers, Szefi et al. (54) showed that increases in diameter reduce the frequencies and span of the stop band, whereas increases in stiffness increase the stop band frequencies. A design optimization technique was used to illustrate that stop band frequency targets could not always be met when realistic mass, stiffness, and diameter constraints were imposed. Consequently, Szefi et al. (54) theoretically evaluated the use of vibration absorbers and inertial amplifiers (anti-resonant isolators) for passively enhancing performance. They concluded that embedded anti-resonant isolators were effective and had a small weight penalty. Szefi et al. (55) developed a compact periodic isolator containing fluidic anti-resonant isolators (similar to the LIVE system) within each metal layer. Velocity transmissibility measurements demonstrated that the embedded inertial amplification successfully reduced the stop band frequencies so that the dominant gear mesh tones could be attenuated. However, this comes at the expense of a considerably higher stop band transmissibility. Szefi et al. (56) refined this design for implementation in a Bell Model 427 by considering realistic transmissibility, modal, and quasi-static stiffness constraints (Figure 5). Transmissibility below -40 dB over 500-2000 Hz was predicted, but measurements were not reported. Dylejko and MacGillivray (57) theoretically considered an aperiodic elastomeric isolator containing metal layers, to which a vibration absorber or inertial amplifier was attached. The absorbers and amplifiers were shown to suppress the internal resonances of the elastomer layers, thereby extending the bandwidth over which ideal single-degree-of-freedom isolation is achieved. Le Hen et al. (58) converted Szefi et al.'s (55) periodic isolator into an active isolator by incorporating a piezoelectric actuator into one of the embedded anti-resonant isolators. Velocity transmissibility of a 2-celled isolator concurrently excited by white noise and three sinusoidal tones was measured while controlling the actuator with a version of the FX-LMS algorithm. Transmissibility reductions of 30-41 dB were measured at the tones, resulting in total (passive plus active) reductions of 70-81 dB. Currently, Cornerstone Research Group is developing shape memory polymers with tunable stiffness for use in periodic fluidic isolators.

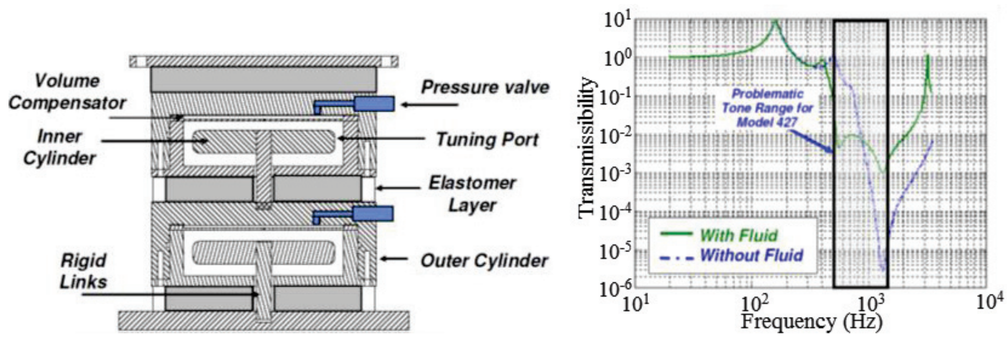

Figure 5 – Szefi et al.'s periodic elastomeric isolator and its predicted velocity transmissibility (56)

A few researchers have proposed variable stiffness springs for tailoring the dynamics of a mount or implementing vibration control techniques based on stiffness tuning or switching. Haynes et al. (59) patented a variable stiffness device composed of an elastomer with embedded metal shims to provide anisotropic stiffness. Stiffness is tuned by rotating the elastomer using a worm gear. Scheidler et al. (60) developed a magnetostrictive variable stiffness spring that can tune or switch its stiffness in situ. Measurements in a dynamic load frame demonstrated continuous modulus changes up to 21.9 GPa and 500 Hz and square wave changes up to 12.3 GPa and 100 Hz. Scheidler et al. (61) simulated the switched-stiffness vibration control performance of this technology, but measurements were not reported. The authors are developing an analogous variable stiffness spring based on piezoelectric materials.

Nonlinear or negative stiffness mechanisms have been utilized in passive suspensions and vibration isolators to concurrently achieve high static stiffness and low dynamic stiffness. These technologies have primarily been applied for reducing the natural frequency of isolators into the sub-Hertz frequency range. For example, a negative stiffness suspension has been suggested for suspending large aircraft and spacecraft during ground testing while also providing low frequency isolation (62).

3. Summary

A literature review of the noise and vibration control technologies for rotorcraft drivelines was presented. Solutions that address gear-mesh-induced noise and vibration were strongly emphasized. Technologies were categorized based on their level of development and their location within the driveline. The key characteristics of the reviewed technologies are summarized in Table 3, wherein they are categorized as having a broad-band or narrow-band effect on source control (S) or path control (P). Insertion loss (IL) is the reduction in a vibration or noise metric resulting from the integration of a technology. Many of the technologies have a technology readiness level (TRL) of 4, because they have only been tested under small excitations in laboratory drivelines that transfer <56 kW (75 hp) (1-2 orders of magnitude lower than rotorcraft gearing). None of the concepts reduce mass; however, the implementation of any driveline treatment could justify the partial removal of cabin acoustic treatment. Retrofitting into existing helicopters will be impractical for most of the technologies; although, high CR spur and superfinished gears are ideal candidates for retrofit. The broad-band control concepts, particularly the periodic fluid mounts, are promising because they tend to have high IL and be mass neutral; however, further development is needed to demonstrate their performance and reliability in a realistic environment. The narrow-band control concepts tend to add mass, but they all can, to some extent, vary the frequency at which they are effective. Active control concepts are effective up to about the middle of the gear mesh frequency range; hence, they pair well with passive techniques that have a low pass characteristic (e.g., cabin noise barriers).

Table 3 – Characteristics of the reviewed technologies; green/red shading denote positive/negative attributes

		Technology	Approx. freq., Hz	IL, dB	TRL	Mass/ Size	Retrofit	Vary freq.	Key challenge
Broad-band	S	Passive gear isolation	>500	3-7	3-6				temperature limits
		Bearing shunt damper	>100	7-18	4				design integration
		Periodic shaft	500-4k	0-40	4				shaft length
	P	Elastomeric hard mount	>250	0-60	4-?				reliability
		Periodic fluid mount	500-3k	30-81	5				system-level data
Narrow-band	S	High CR spur gear	500-4k	2	9				n/a
		Superfinished gear	500-4k	4-7	8				n/a
		Helical gear	500-4k	2-17	9				added thrust load
		Magnetic gear	500-1.6k	?	4				specific torque
		Active gear	<1k	7.5	3-4				complex, force required
		Active transverse shaft	<4k	2-28	4				size, force required
		Active bearing	<1k	0-45	4				size, force required
	P	Active strut	200-2.5k	11-20	7-?				reliability
		Active at mounting points	200-1.5k	8-18	7-?				force required
		Variable stiffness mount	<1k	?	3				unproven performance

4. CONCLUSIONS AND RECOMMENDATIONS

The conclusions from this survey are listed below, followed by recommendations, which are provided to help advance the state-of-the-art and guide future research directions.

- Since civilian rotorcraft are typically derivatives of military rotorcraft, new drivetrain noise control features should be suitable for initial integration with military designs (i.e., have zero weight penalty) or be capable of being retrofitted.
- Current transmissions have nearly fixed frequency gear meshing signatures.
- Final stage, low-speed gearing (e.g., planetary and bull) produce the most significant cabin noise in the 500 and 1000 Hz octave bands, as the vibro-acoustic transfer path has a low pass characteristic.
- Enclosed/isolated cabins can effectively suppress gear noise, creating a jet-like environment. However, the space and mass penalty makes this strategy more feasible in large civil aircraft.
- Mature and effective gear technologies are available, but are not always used. Some can be retrofit (superfinished and high CR gears), but others require reconfiguration (helical and face gears).
- Technologies that could be retrofit in order of increasing mass penalty: superfinished gears, high

CR spur gears, periodic fluid mounts, active struts, and active control at mounting points.
- A considerable amount of work on active control has been recently conducted. However, full-scale testing of active concepts is needed to experimentally assess actuator requirements.
- Active path control can elevate noise in some locations. It is preferable to implement active source control or source isolation at choke points.
- The FX-LMS control algorithm is the most commonly applied active control strategy, resulting in partial attenuation of tonal disturbances. Sidebands often dominate the residual spectra, which can lead to beating or grinding-like sounds. The adaptive noise equalizer algorithm has been shown to provide complete attenuation or allow for arbitrary shaping of the residual spectrum (64).
- Negative and nonlinear stiffness isolators seem promising, but have not been applied for gear-mesh-induced vibration control.
- Non-contact, magnetic gears would completely change, and likely reduce, transmission vibration signatures. A coaxial design has been created that has reasonable mass at high power levels. Vibration and efficiency characterization data are needed, as well as new right angle and multi-input topologies.
- Where possible, noise and vibration control technologies should be designed with a systems perspective, such that the implementation of gear noise concepts doesn't amplify rotor tones or displace existing rotor noise concepts that are effective.
- Three disruptive current trends in the development of rotorcraft are likely to change the noise profiles and treatments required.
  - Multi-speed and variable-speed transmissions: Many existing technologies capitalize on the nearly fixed frequency of gear mesh tones. In multi-speed or variable-speed transmissions, these tones will vary abruptly or smoothly over a significant range as the mast's speed varies by a factor of about 2. This will motivate the maturation of adaptive or active technologies.
  - Composite driveline components: Early attempts to develop composite gears (63) involve replacing the gear's steel body with composite material, but leaving the teeth unchanged. If the composite does not introduce vibration attenuation (e.g., via damping, impedance mismatch, or functional grading), then vibration levels may increase at the source due to a reduction in inertia for the same mesh excitations. However, the use of composites in gears and shafts will provide an opportunity to tailor their anisotropy and embed components or features within the structure.
  - Electrification: Hybrid electric and fully electric rotorcraft will introduce powertrain configurations that greatly differ from current designs. In some cases, transmissions may not be necessary. Also, the nature of the vibration generated by electric fans distributed throughout an aircraft is not understood.

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REFERENCES

1. Blankenship GW et al. Analaytical solution for modulation sidebands associated with a class of mechanical oscillators. J. Sound Vib. 1995;179(1):13-36.
2. Howlett JT et al. Interior noise reduction in a large civil helicopter. NASA; 1977 Jul. Report No. NASA-TN-D-8477.
3. Coy JJ et al. Identification and proposed control of helicopter transmission noise at the source. U.S. Army; 1987 Report No. USAAVSCOM-TR-87-C-2.
4. Weden GJ et al. Summary of drive-train component technology in helicopters. NASA; 1984 Oct. Report No. NASA-TM-83726.
5. Yoerkie CA et al. Development of rotorcraft interior noise control concepts. Phase 2: Full scale testing. NASA; 1986 Feb. Report No. NASA-CR-172594.
6. Leverton JW et al. Helicopter internal noise - an overview. Proc. Symp. Internal Noise in Helicopters; Southampton, England 1980. p. A4 1-A4.
7. Levine LS et al. Civil helicopter research aircraft internal noise prediction. NASA; 1977 Report No. NASA-CR-145146.
8. Yoerkie C et al. Transmission acoustic vibration testing. U.S. Army; 1985 Report No. USAAVRADCOM-TR-83-D-34.

9. Litvin FL et al. Computerized design, simulation of meshing, and contact and stress analysis of face-milled formate generated spiral bevel gears. Mech. Mach. Theory. 2002;37(5):441-459.
10. Heath GF et al. Advanced rotorcraft transmission (ART) program - final report. NASA; 1993 Jan. Report No. NASA-CR-191057.
11. Oswald FB et al. Influence of gear design on gearbox radiated noise. Gear Tech. 1998;15(1):10-15.
12. He S et al. Effect of sliding friction on the dynamics of spur gear pair with realistic time-varying stiffness. J. Sound Vib. 2007;301(3-5):927-949.
13. Hansen B et al. Isotropic superfinishing of S-76C+ main transmission gears. AGMA Tech. Paper 06FTM02. 2006.
14. Houser DR et al. Vibro-acoustic effects of friction in gears: an experimental investigation. SAE Tech. Paper 2001-01-1516. 2001.
15. Kahraman A. Planetary gear train dynamics. J. Mech. Design. 1994;116(3):713-720.
16. Parker RG. A physical explanation for the effectiveness of planet phasing to suppress planetary gear vibration. J. Sound Vib. 2000;236(4):561-573.
17. Schlegel RG et al. Transmission noise control-approaches in helicopter design. Proc. ASME Design Eng. Conf.; New York, NY 1967. p. 67-DE-58.
18. Flannelly W, inventor; Kaman Aircraft Corp., assignee. Dynamic antiresonant vibration isolator. U.S. Patent US3322379. 1967 May 30.
19. McGuire D. High stiffness ("rigid") helicopter pylon vibration isolation systems. Proc. Am. Helicopter Soc. 59th Annual Forum; Phoenix, AZ 2003.
20. Halwes DR. Live-liquid inertia vibration eliminator. Proc. Am. Helicopter Soc. 36th Annual Forum; Washington, D.C. 1980.
21. Konstanzer P, Enenkl B, Aubourg P, Cranga P. Recent advances in Eurocopter's passive and active vibration control. Proc. Am. Helicopter Soc. 64th Annual Forum; Montreal, Canada 2008.
22. Smith MR et al.. Results from the dynamically tailored airframe structures program. Proc. Am. Helicopter Soc. 58th Annual Forum; Montreal, Canada 2002.
23. Sutton TJ et al. Active isolation of multiple structural waves on a helicopter gearbox support strut. J. Sound Vib. 1997;205(1):81-101.
24. Maier R et al. Active vibration isolation system for helicopter interior noise reduction. Proc. 8th AIAA/CEAS Aeroacoustics Conf.; Breckenridge, CO 2002. AIAA 2002-2495.
25. Hoffmann F et al. Helicopter interior noise reduction by using active gearbox struts. Proc. 12th AIAA/CEAS Aeroacoustics Conf.; Cambridge, MA 2006. AIAA 2006-2604.
26. Millott TA et al. Flight test of active gear-mesh noise control on the S-76 aircraft. Proc. Am. Helicopter Soc. 54th Annual Forum; Washington, DC 1998.
27. Welsh WA et al. Active noise control system for a helicopter gearbox mount. U.S. Patent US6105900. 2000 Aug. 22.
28. Li X et al. Comparison of magnetic-geared permanent-magnet machines. Prog. Electromagn. Res. 2013;133:177-198.
29. Atallah K et al. Design, analysis and realisation of a high-performance magnetic gear. IEE P-Elect. Pow. Appl. 2004;151(2):135-143.
30. Jian L et al. A coaxial magnetic gear with Halbach permanent-magnet arrays. IEEE T. Energy Conver. 2010;25(2):319-328.
31. Paden D. Non-contact magnetic transmission for hybrid/electric rotorcraft. NASA; 2015 Report No. A1.06-9338.
32. Brown GV et al. NASA Glenn Research Center program in high power density motors for aeropropulsion. NASA; 2005 Jan. Report No. NASA/TM-2005-213800.
33. Kish JG. Sikorsky aircraft advanced rotorcraft transmission (ART) program. NASA; 1993 Mar. Report No. NASA-CR-191079.
34. Sammataro SR et al. Isolated ring gear for planetary gear drive systems. U.S. Patent US5716300. 1998 Feb. 10.
35. Guan P et al. Passive suppression of planetary gear transmission vibration via discrete boundary struts. Proc. Am. Helicopter Soc. 72nd Annual Forum; West Palm Beach, FL 2016.
36. Chen MH et al. Active control of gear vibration using specially configured sensors and actuators. Smart Mater. Struct. 2000;9(3):342-350.
37. Guan YH et al. Comparative analysis of actuator concepts for active gear pair vibration control. J. Sound Vib. 2004;269(1-2):273-294.
38. Montague GT et al. Feed-forward control of gear mesh vibration using piezoelectric actuators. NASA;

1994 May Report No. NASA-TM-106366.
39. Rebbechi B et al. Active control of gearbox vibration. Proc. Active Control of Sound and Vib. Conf.; Fort Lauderdale, FL 1999.
40. Guan YH et al. Experimental study on active vibration control of a gearbox system. J. Sound Vib. 2005;282(3-5):713-733.
41. Zhao G et al. Adaptive-passive control of structure-borne noise of rotating machinery using a pair of shunted inertial actuators. J. Intel. Mat. Syst. Str. 2015;1-16.
42. Zhao G et al. Experimental study on active structural acoustic control of rotating machinery using rotating piezo-based inertial actuators. J. Sound Vib. 2015;348:15-30.
43. Atzrodt H et al. Reduction of bearing vibrations with shunt damping. Proc. 16th Intl. Congress on Sound and Vib.; Krakow, Poland 2009.
44. Pinte G et al. A piezo-based bearing for the active structural acoustic control of rotating machinery. J. Sound Vib. 2010;329(9):1235-1253.
45. Asnani VM et al. The vibration ring: seedling fund phase 1 final report. NASA; 2014 Jun. Report No. NASA-TM-218337.
46. Asnani VM et al. Experimental comparison of piezoelectric and magnetostrictive shunt dampers. Proc. SPIE 9801; Las Vegas, NV 2016. p. 98010R.
47. Deng Z et al. Magnetostrictive vibration damper and energy harvester for rotating machinery. Proc. SPIE 9433; San Diego, CA 2015. p. 94330C.
48. Richards D et al. Passive reduction of gear mesh vibration using a periodic drive shaft. J. Sound Vib. 2003;264(2):317-342.
49. Keningsberg IJ et al. inventors; United Technologies Corp., asignee. Helicopter rotor transmission mounting and vibration isolation system. U.S. Patent US4111386. 1978 Sep. 5.
50. Yoerkie CA et al. Development of rotorcraft interior noise control concepts. Phase 3: Development of noise control concepts. NASA; 1986 Jul. Report No. NASA-CR-178172.
51. Gent AN et al. Compression, bending, and shear of bonded rubber blocks. Polym. Eng. Sci. 1970;10(1):48-53.
52. Asiri S et al. Periodic struts for gearbox support system. J. Vib. Control. 2005;11:709-721.
53. Asiri S et al. Active periodic struts for a gearbox support system. Smart Mater. Struct. 2006;15:1707-1714.
54. Szefi JT et al. Design and analysis of high-frequency periodically layered isolators for helicopter gearbox isolation. Proc. 44th Struct., Struct. Dyn., Mater. Conf.; Norfolk, VA 2003. AIAA-2003-1784.
55. Szefi JT et al. Design and testing of a compact layered isolator for high-frequency helicopter gearbox isolation. Proc. 45th Struct., Struct. Dyn., Mater. Conf.; Palm Springs, CA 2004. AIAA-2004-1947.
56. Szefi JT et al. Design of fluidic, high-frequency periodically layered isolators for Model 427 gearbox isolation. Proc. Am. Helicopter Soc. 62nd Annual Forum; Phoenix, AZ 2006.
57. Dylejko PG et al. On the concept of a transmission absorber to suppress internal resonance. J. Sound Vib. 2014;333(10):2719-2734.
58. Le Hen F et al. Actively-enhanced periodically-layered mount for helicopter gearbox isolation. Proc. 46th Struct., Struct. Dyn., Mater. Conf.; Austin, TX 2005. AIAA-2005-2250.
59. Haynes DF et al., inventors; Textron Innovations Inc., assignee. Variable stiffness support. U.S. Patent US8746649. 2014 Jun. 10.
60. Scheidler JJ et al. Dynamically tuned magnetostrictive spring with electrically controlled stiffness. Smart Mater. Struct. 2016;25(3):035007.
61. Scheidler JJ et al. Vibration control via stiffness switching of magnetostrictive transducers. Proc. SPIE 9799; Las Vegas, NV 2016. p. 979909.
62. Woodard SE et al. Nonlinear behavior of a passive zero-spring-rate suspension system. J. Guid. Control Dynam. 1991;14(1):84-89.
63. Handschuh RF et al. Vibration and operational characteristics of a composite-steel (hybrid) gear. NASA; 2014 Jun. Report No. NASA/TM-2014-216646.
64. Asnani V et al. Active control of amplitude or frequency modulated sounds in a duct. Noise Control Eng. J. 2005;53(5):165-175.