# DEVELOPMENT OF SMART STRUCTURE SYSTEMS FOR HELICOPTER VIBRATION AND NOISE CONTROL

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Received August 2006, Accepted March 2007
No. 06-CSME-37, E.I.C. Accession 2956

## ABSTRACT

Helicopters are susceptible to high vibratory loads, excessive noise levels and poor flight stability compared to fixed-wing aircraft. The multidisciplinary nature of helicopter structures offers many opportunities for the innovative smart structure technology to improve helicopter performance. This paper provides a review of smart structures research at the National Research Council Canada for helicopter vibration and cabin noise control applications. The patented Smart Spring approach is developed to vary the blade impedance properties adaptively to reduce the vibratory hub loads transmitted to the fuselage by vibration reduction at the source. A smart gearbox strut and active structural acoustic control technologies are investigated to suppress the vibration and tonal gear meshing noise into the cabin either by modifying the vibration load transmission path, or weakening the coupling between exterior and cabin acoustic fields. Two adaptive seat mount concepts are proposed to reduce the vibration of the aircrew directly to improve ride quality of the vehicle.

# MISE AU POINT DE SYSTÈMES STRUCTURAUX INTELLIGENTS POUR LA RÉDUCTION DU BRUIT ET DES BORD DES HÉLICOPTÈRES

## RÉSUMÉ

Les hélicoptères sont susceptibles d'être soumis à des vibrations élevées, à des niveaux de bruit excessifs et à une mauvaise stabilité en vol comparativement aux avions. La nature multidisciplinaire des structures d'hélicoptère offre de nombreuses possibilités à la technologie des structures intelligentes innovantes pour améliorer leur comportement. Le présent document passe en revue la recherche sur les structures intelligentes effectuée au Conseil national de recherches du Canada et axée sur la réduction des vibrations des hélicoptères et la réduction du bruit dans la cabine. L'approche du concept Smart Spring breveté est mise à contribution pour faire varier de façon adaptative les propriétés d'impédance des pales afin de réduire les vibrations s'exerçant sur le moyeu et qui sont transmises au fuselage en réduisant les vibrations à la source. Un montant intelligent de boîte de transmission et des techniques d'insonorisation active des structures sont étudiés en vue de supprimer les vibrations et les tonalités du bruit d'engrènement dans la cabine en modifiant le trajet de la transmission des vibrations ou en affaiblissant le couplage entre les champs acoustiques de l'extérieur et de la cabine. Deux concepts adaptatifs de support de siège sont proposés pour réduire les vibrations parvenant au personnel navigant afin d'améliorer la qualité de vol du véhicule.

# 1. INTRODUCTION

Helicopters operate in a highly complex unsteady aerodynamic environment caused by the cyclic variation of aerodynamic loads on the blades. Significant structural vibration and cabin noise are undesirable characteristics of helicopter flight. The primary source that contributes to the vibration in helicopter airframe is the N/rev rotor hub reaction loads induced by the inertial and aerodynamic loads acting on the blades, whereas the gear transmission and drive train are the main sources of high frequency noise inside the cabin. Due to the inherently complex coupling between the rotor system, airframe, transmission system and engine, the vibratory loads and noise energy are transferred throughout the helicopter structure, and contribute to poor ride quality for passengers, low fatigue life of structural components, high maintenance cost and restricted flight envelope of the vehicle.

Tremendous efforts have been undertaken to reduce helicopter vibration and noise levels. Many new devices and design modifications have been implemented on in-service helicopters, and achieved modest performance improvements. However, revolutionary approaches have to be developed to reduce helicopter vibration and noise levels before the ride quality becomes comparable to fixed-wing aircraft. Among the numerous new technologies, smart structure offers an innovative solution that promises significant performance improvement in helicopters.

A smart structure is defined as an integrated multidisciplinary structural system that can sense external stimulus and respond to the stimulus adaptively. The signal sensing function is achieved through a distributed sensor network attached to the structure, which may be comprised of piezoelectric materials, fiber optics, or other sensors devices. The actuation authority is provided by smart material actuators, such as the piezoelectric materials, magnetorheological (MR) and electrorheological (ER) fluids, magnetostrictive materials or shape memory alloys (SMA). The adaptive logic is generally implemented through digital controllers. Compared with conventional active systems, the sensors and actuators of a smart structure are integrated to the primary load-carrying path to provide additional functions without causing major changes in the properties of the system. Therefore, the unique features of Smart Structures have been widely investigated for aerospace structures and systems.

The multidisciplinary nature of helicopter structures offers many opportunities for the application of smart structures for performance improvement. Numerous smart structure approaches have been investigated to improve helicopter ride quality and expand the flight envelope during the past few years. Comprehensive review of the state-of-art technologies has been presented in references [1, 2, 3]. Smart structures research can be divided into three major areas, namely, smart rotors, cabin noise reduction, and seat and floor vibration control.

## 1.1. Smart Rotors

Among the subsystems of helicopters, the rotor is the key that sets limitation of the vehicle performance, handling quality and reliability. By incorporating innovative smart structure systems to individual rotor blades, the vibratory aerodynamic loads that are transmitted to the fuselage can be reduced at the source. Technical approaches include trailing edge flaps driven by piezoelectric actuators [4], controllable twist rotor blades using piezoelectric fiber actuators [5], and trailing edge tabs with SMA actuators [6]. An investigation on scaled rotor models has shown promising potential to reduce helicopter vibrations. However, the implementation on full-scale rotor blades had been hindered due to the limitations of actuators performance.

### 1.2. Cabin Noise Reduction

Noise in the helicopter cabin is mainly caused by the Blade-Vortex Interaction at the low frequency range and gearbox meshing tones of the transmission and drivetrain in the high frequency regime [7]. Compared to passive control technology, active control technology offers an attractive alternative due to the superior performance and lower weight penalty. Loudspeaker based Active Noise Control (ANC) which has been used in turboprop fixed wing aircraft is not practical for helicopters because of the highly complex disturbances within the cabin. Recently, considerable advances have been achieved by using smart structure technologies. For example, piezoelectric actuators have been attached to helicopter airframe panels to reduce sound radiation [8, 9]. This approach is generally effective for resonant frequencies. However, in the off-resonant cases, the reduction is much smaller. Another approach is to develop adaptive gearbox struts and mounts to reduce the tonal tooth meshing noises transmitted to the cabin through the airframe structure of the cabin [10, 11, 12].

### 1.3. Seat and Cabin Vibration Control

It is important to note that the vibrations and noises also create a challenging environment for the aircrew. The vibration isolation for aircrew in existing helicopter seats is achieved passively through the cushion installed between the seat frame and the aircrew. However, the cushion impedance properties are difficult to optimize due to the broad vibration frequency range in which helicopters typically operate. To improve the ride quality of helicopters, an alternative approach is to design adaptive mounts to reduce the vibration transmitted through the fuselage to the cargo floor and aircrew [13, 14]. The adaptive mount concept can adapt to the varying vibratory frequencies over a broadband resulting from variations in flight conditions as well as the changes of aircrew members. Therefore, it is an acceptable compromise to reduce the harmful vibration effects to helicopter aircrew.

The concern about weight in aerospace vehicles has led to light flexible structures, with accompanying vibration and acoustic problems. Therefore, smart structure technology is a major research field at the Institute for Aerospace Research (IAR) of the National Research Council Canada. Significant research effort has been directed to improve the performance of aerospace structures and systems. This paper briefly highlights the advances of smart structure research on helicopter vibration and noise control applications at IAR.

## 2. ADAPTIVE BLADE IMPEDANCE CONTROL TO REDUCE VIBRATORY LOADS

### 2.1. Smart Spring Concept for Blade Impedance Control

Because the most significant disturbances that generate helicopter vibration are the aerodynamic and inertial loads from rotor blades, many research efforts are dedicated to reduce the unsteady components of aerodynamic loads using trailing edge flaps [15] and blade twist concepts [16]. Despite the promising nature, these approaches require smart materials based actuators to provide large force and displacement simultaneously, which is still a technical challenge today.

The vibratory hub load that causes fuselage vibration can also be reduced by varying blade impedance properties dynamically at the root [17, 18]. Because this is an indirect approach to reduce vibratory hub load transmitted to the fuselage, the power requirement and displacement requirement is significantly lower than the direct force control approach above.

The Smart Spring is a patented concept developed by National Research Council Canada that overcomes difficulties with other impedance control methods [19]. This concept is based on an actively

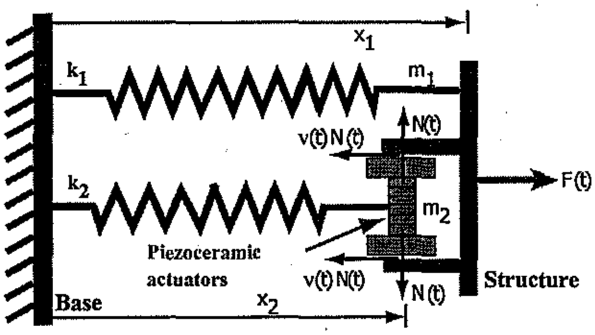



Fig. 1: Smart Spring concept

variable impedance device using piezoelectric actuators to adaptively vary the stiffness, damping, and mass of a dynamic system. It can be generally illustrated as two springs in parallel, as seen in Fig. 1. The mechanism exploits the large stiffness and bandwidth of piezoelectric materials to modulate the structural properties at high frequencies in order to perform semi-active vibration control. The spring designated by $k_1$ is the primary load-carrying path. This primary spring $k_1$ is attached to the vibrating structure having mass $m_1$, to which an external vibratory force F(t) is applied. A piezoelectric actuator attached to the secondary spring designated by $k_2$ operates in a sleeve attached to the primary structure. When the actuator is off, the secondary spring does not interfere with the vibrating structure and the dynamic response of the structure is determined by the single primary mass-spring system consisting of $m_1$ and $k_1$.

When the actuator is powered, a normal force N(t) is generated between the actuator and the sleeve to engage the secondary mass-spring system, consisting of $m_2$ and $k_2$, with the structure. If the resultant friction force $\nu(t)N(t)$ is sufficiently large, the two springs become fully coupled and a combination of two springs in parallel is obtained. In this case, the stiffness of the system increases from its uncoupled value $k_1$ to the coupled value $k_1+k_2$ and the mass of the system increases from the original value of $m_1$ to the final value $m_1+m_2$. However, the two limiting conditions are not generally achieved since sliding of the sleeve is always present. The dynamic friction force $\nu(t)N(t)$ applied to the sleeve can be controlled by an external voltage stimulus to the actuator, forcing the secondary mass and spring designated by $m_2$ and $k_2$ to become increasingly coupled with the primary mass-spring system. If sliding of the secondary mass against the contact surfaces of the sleeve in the primary mass is present, the resulting change in stiffness is coupled with an increase of effective mass and damping as seen by the structure designated as $m_1$. As a result, the voltage applied to the piezoelectric actuator could continuously control the combinations of stiffness, damping, and effective mass to suppress mechanical vibration.

It is important to note that the Smart Spring does not require the piezoelectric stack actuator to counteract the vibratory aerodynamic loads directly. Rather, the actuator controllably engages the active spring with the primary blade structure at the root, so that blade impedance properties are varied adaptively at the boundary. The change of blade impedance properties enables a semi-active control of blade dynamic response. Compared to other piezoelectric actuators based active approaches, the Smart Spring only requires the piezoelectric actuator to produce micro displacement to generate relatively high actuation force. As a result, the piezoelectric actuator is able to achieve sufficient force with less than 100V. In addition, the Smart Spring can be implemented with a small footprint and the system is scalable for a variety of helicopter vibration control applications.

### 2.2. Simulation of Smart Spring Impedance Control Characteristics

A simulation model of the Smart Spring concept based on a mathematical model was developed in order to identify the dynamic characteristics of the concept [20]. Furthermore, the simulations based on Matlab and Simulink were used to determine various possible designs of the Smart Spring concept that optimizes the controllability of one or more impedance properties. Each potential application of the Smart Spring concept will require optimization of the design parameters, namely, $k_2$ and $m_2$, based on $k_1$ and $m_1$ of that particular application in order to suppress vibration effectively.

### 2.2.1. Mathematical model

The mass-spring system shown in Fig. 1 was simulated using the following set of differential equations [20].

$$m_1\ddot{x}_1+k_1x_1=F(t)\pm v(t)N(t) \tag{1}$$

$$m_2\ddot{x}_2+k_2x_2=\pm v(t)N(t) \tag{2}$$

In this mathematical model, the frictional force was calculated as the product of the normal force generated by the actuator and the friction coefficient between sliding surfaces. In Equations (1) and (2), the non-linear frictional force $v(t)N(t)$ couples the primary and secondary mass-spring systems. The frictional coefficient $v(t)$ varies with time as a function of the absolute relative speed $|\dot{x}_1-\dot{x}_2|$ and the material properties of the frictional surfaces. The term $v(t)$ carries the upper sign if $\dot{x}_1>\dot{x}_2$ and the lower sign if $\dot{x}_1<\dot{x}_2$. This system can be simplified by allowing the friction coefficient to absorb the switching sign by assigning an odd extension to the function $v(t)$ as follows:

$$v(\dot{x}_2-\dot{x}_1)=-v(\dot{x}_1-\dot{x}_2) \tag{3}$$

Using the relationship shown in Equation (3), the equations of motion are rewritten using the upper sign only. The above differential equations can be formulated in the state-space form as shown in Equation (4). The damping terms $c_1$ and $c_2$ are included to account for internal structural damping associated with the springs.

$$x_1=X_1,\ \dot{x}_1=X_2,\ x_2=X_3,\ \dot{x}_2=X_4$$
$$\{\dot{X}\}=[A]\{X\}+[B]\{u\}+[E]\{F(t)\} \tag{4}$$

where,

$$[A]=\begin{bmatrix} 0 & 1 & 0 & 0 \\ \frac{-k_1}{m_1} & \frac{-c_1}{m_1} & 0 & 0 \\ 0 & 0 & 0 & 1 \\ 0 & 0 & \frac{-k_2}{m_2} & \frac{-c_2}{m_2} \end{bmatrix};\quad [B]=\begin{bmatrix} 0 \\ \frac{-1}{m_1} \\ 0 \\ \frac{1}{m_2} \end{bmatrix};\quad u=v(t)N(t);\quad [E]=\begin{bmatrix} 0 \\ \frac{1}{m_1} \\ 0 \\ 0 \end{bmatrix}$$

### 2.2.2. Simulation Results

The state-space representation of the Smart Spring system described by equation (4) is depicted in the form of a Simulink block diagram in Fig. 2. The friction coefficients were interpolated from a table of empirical data that depended on the relative sliding speed between $m_1$ and $m_2$. This simulation model of the Smart Spring system was used to identify the non-linear dynamic behaviour of the system.

This numerical simulation model identified the variation in the dynamic response characteristics of the Smart Spring concept as the design parameters, namely $m_1$, $k_1$, $m_2$, and $k_2$ were varied. The simulations demonstrated that careful selection of the mass and stiffness values of the primary and secondary systems enabled the Smart Spring system to optimize the control of stiffness, mass or damping. This allows the system designer to optimize the controllability of the appropriate impedance properties of the Smart Spring to enhance vibration suppression for a given application.

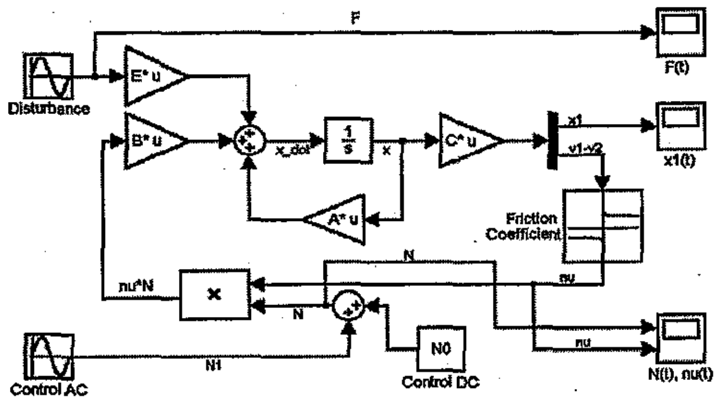



Fig. 2: Simulink block diagram of the Smart Spring model

The dynamic characteristics of various Smart Spring designs were evaluated by observing the frequency response of the structure $m_1$ and the time response of $m_1$ due to a step input as shown in Fig. 2. The 'reference' curve shows the response of the Smart Spring system with no actuation force or no contact initiated by the actuator. Thus, the response was that of the single mass-spring system with $m_1$ and $k_1$. These parameters were held constant in order to compare response of Smart Spring to variations in design parameters, namely $m_2$ and $k_2$.

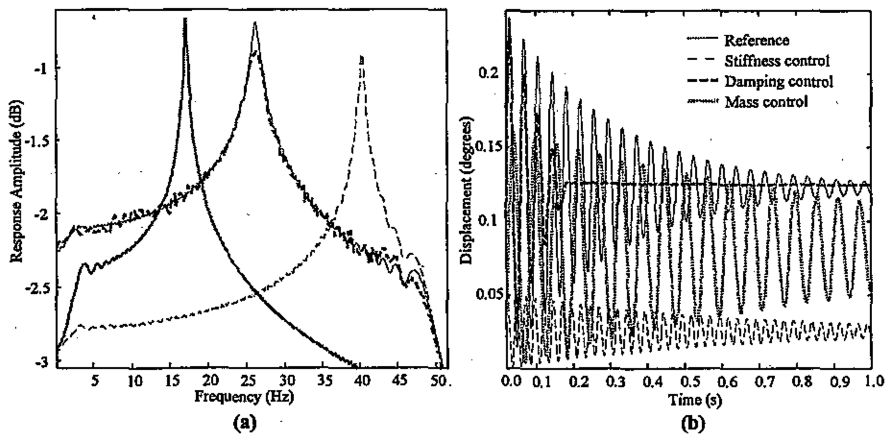



Fig. 3: Smart Spring simulation results (a) frequency response (b) time response for step input

The simulations confirmed that the Smart Spring design is capable of introducing damping in a controlled manner by increasing actuator voltage. Enhancement of damping characteristics using a Smart Spring system is always possible because modulation of the actuator force N(t), by means of applied voltage, could be used to maximize the frictional forces developed between m1 and m2. The increase in damping reduced the amplitude of the response as shown in Fig. 3 without affecting the frequency of response. More importantly, the force applied by the piezoelectric actuator is able to control the amount of damping generated by the Smart Spring system. For example, at zero actuation force, no additional damping was present in the system because the resulting frictional force was zero. However, gradual increase in actuation resulted in increased damping due to higher frictional forces. It is important to note that the propensity for the two mass-spring systems to couple increased as the actuation force was increased. Therefore, a maximum damping limit exists and a further increase in actuation force would result in a reduction in damping. The simulations showed that the Smart Spring designs were capable of introducing damping in a controlled manner by carefully modulating the actuation voltage.

In order to vary stiffness, the Smart Spring design parameters must be chosen such that $m_2<m_1$ and $k_2>k_1$. In such a Smart Spring design, high actuation force N(t) was used to couple the primary and the secondary mass-spring systems. At such actuation level, the coupled systems responded as a single mass-spring system composed of the combined mass of $m_1+m_2$ and combined stiffness of $k_1+k_2$. This combination of mass and stiffness properties resulted in a system with a natural frequency that was higher than the natural frequency of the reference system as seen in Fig. 3. It is important to note that the Smart Spring is capable of varying the stiffness characteristics at high frequencies in a controlled manner due to the fact that engagement of the secondary mass-spring system is performed using a stacked piezoelectric actuator that had a very high frequency response capability. However, sliding contact between $m_1$ and $m_2$ would be present if the actuation force was insufficient to completely couple the primary and secondary mass-spring systems. In this case, both stiffness and damping in the Smart Spring system could be preferentially controlled using the actuation force N(t).

A Smart Spring system can also be designed to vary the effective mass of a structure if the design parameters were selected such that $m_2>m_1$ and $k_2<k_1$. In such a design, the natural frequency of the fully coupled system was much lower than that of the reference system, as seen in Fig. 3. However, this Smart Spring design to vary effective mass may not be practical in applications.

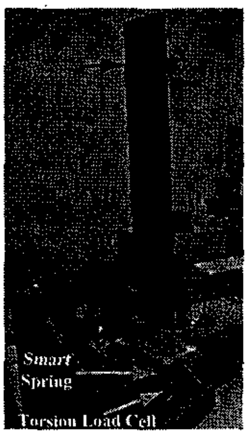
The above simulation results show that it is possible to optimize the design of the Smart Spring to control stiffness, damping, or effective mass. The design parameters of the Smart Spring system must be carefully selected for each potential vibration suppression application in order to preferentially vary the appropriate impedance properties.

### 2.3. Characterization of Smart Spring for Rotor Blade Impedance Control

A proof-of-concept Smart Spring hardware model has been fabricated to demonstrate the principles of vibration suppression on a typical blade section, as shown in Fig. 4. The Smart Spring was installed at the root of the blade section where is the most efficient location to vary the blade global impedance properties. To characterize it dynamics performance, a mechanical shaker was connected to the interface plate at the bottom of the blade to provide simulated dynamic excitation to the blade assembly. The shaker load was measured with a force transducer on the stinger. A band limited random signal (5-100Hz) was applied on the mechanical shaker and the shaker load was kept at 45N peak-to-peak in experiments. A displacement probe attached to the Smart Spring measured the vibration response. External voltage to the piezoelectric stack actuators was controllably varied from 0Vdc to 100Vdc at an interval of 5V to adjust the impedance properties. The MATLAB system identification toolbox was employed to estimate the state space model of the Smart Spring. Identified results were correlated to the Smart Spring mathematical model such that the variation of impedance properties with control voltage could be obtained. The transfer function between the shaker load and the Smart Spring displacement were repeatedly estimated for each voltage and the identified damping and frequency results are shown in Fig. 5.

As shown in Fig. 5, the blade impedance properties were variable under the control of external voltage. The variation of blade impedance showed two distinctive regions. For control voltages from

0Vdc to 40Vdc the primary and active springs were variably engaged. The resulting frictional force increased the structural damping from 0.035 to 0.142 without a significant shift of the natural frequency. This region allowed independent control of the damping properties of the blade assembly. Further increase of control voltage to 90Vdc fully coupled the two spring systems, and increased the equivalent stiffness of the blade assembly and reduced the damping ratio. Accordingly, the resonance frequency of the blade assembly was shifted from 15.4Hz to 24Hz, and the damping ratio was reduced to 0.04.



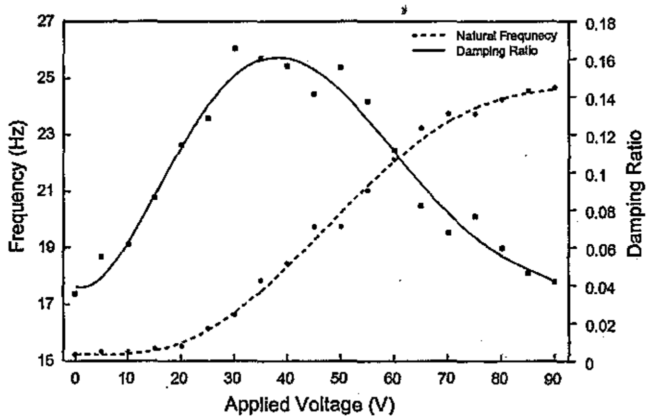
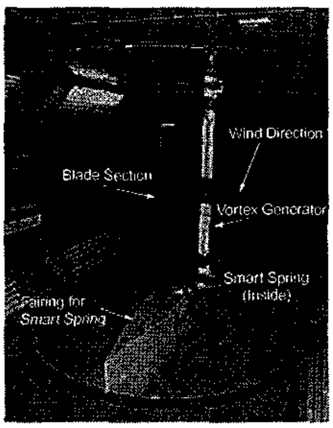




Fig. 4: Blade impedance control configuration Fig. 5: Variation of blade impedance property with voltage

## 2.4. Wind Tunnel Test Results

The aforementioned blade assembly has been tested in the $3m \times 2m$ low speed wind tunnel facility at the Institute for Aerospace Research (IAR) of the National Research Council Canada, as shown in Fig. 6. The objective was to verify the feasibility of the adaptive impedance control concept to suppress blade vibratory load transmission in representative aerodynamic vortical flows. A 0.1m cross-section square tube was positioned in the flow ahead of the blade section to generate vortical flow to excite the blade. This was installed 1.2m upstream from the leading edge of the blade to shed vortices that interact with the blade section down stream. The shedding frequency depended on the wind speed, while the amplitude of aerodynamic load was unsteady and random in nature. To provide an aerodynamic load with time-varying shedding frequencies, the wind speed was regulated using the wind tunnel control system.



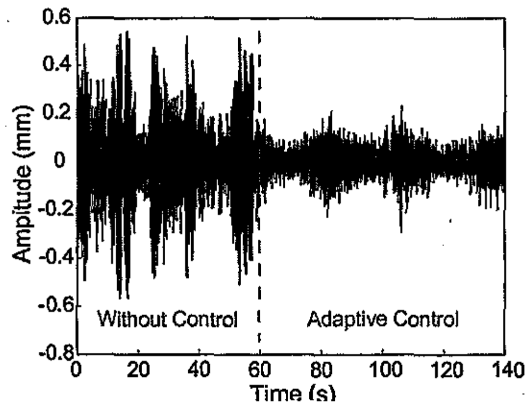




Fig. 6: Wind tunnel test configuration Fig. 7: Vibration control results under vortical flow

In experiments, the wind speed was varied from 11 to 15m/s over 25 seconds so that the vortex shedding frequency varied from 14.1Hz to 19.2Hz. As seen from Fig. 7, the amplitude of the blade vibration was expected to vary due to the variation in excitation frequency, reaching a peak at the wind speed of 12m/s corresponding to the natural frequency of 15.3Hz for the blade and Smart Spring assembly. With the regulation of an adaptive impedance control law applied to control the Smart Spring device [21], the blade vibration was suppressed effectively. At the wind speed of 12m/s which generated vortices that corresponded to the blade natural frequency, the vibration of the blade assembly was reduced by 70%. At off-resonance wind speed, the achieved reduction decreased to 30%, which was still a considerable reduction under forced vibratory conditions. This demonstrated the capability of the controller algorithm to track the fundamental vibratory frequencies and reduce the blade vibration in real-time. However, since the blade response at these speeds was also less, the resultant blade vibration response was similar to that at the blade natural frequency as shown in the right hand side of Fig. 7(??). In addition, the overall vibration was reduced by 55% over the complete range of wind speed. The wind tunnel test results confirmed the capability of the Smart Spring to suppress blade vibration through impedance control under highly unsteady and varying aerodynamic disturbances.

## 3. ACTIVE VIBRATION CONTROL APPROACHES FOR AIRCRAFT CABIN NOISE REDUCTION

### 3.1. Helicopter Cabin Noise Problem

The helicopter main transmission system is generally installed above the passenger cabin to transmit power between the engine and the main rotor. The gear tooth meshing tones are the main sources of high intensity noise in the cabin [22]. As the teeth engage, gear teeth come in contact at the meshing point that travels along the profile of the teeth as the gear rotates. The stiffness of the mesh then varies during normal operation of the geartrain, creating a dynamic load that is transferred through the drive shaft to the bearings and ultimately to the cabin panels. This is the primary source of cabin noise in the high frequency range whereas the rotor aerodynamic noise takes place at frequencies below 100Hz. The difficulty in controlling the vibrations generated by a gearbox comes from the fact that the system dynamics create harmonics at multiples of the characteristic gearbox mesh frequencies. Such dynamics can lead to sound pressure levels as high as 100dB inside the helicopter cabin [7].

The helicopter airframe is generally an aluminum or composite skin panel reinforced by longitudinal stiffeners and circumferential ribs. The interior trim panels are composite panels attached to the structural frames via mounts. General solutions to the cabin noise problem include sound absorbing and viscoelastic damping treatments applied to the airframe and trim panels to reduce the transmission of low frequency acoustic waves from the blades and higher frequency structural disturbances from the gearbox meshing. In recent years, a variety of smart structure techniques have been investigated to achieve better cabin noise reduction performance with less weight penalty [23]. These approaches are similar to related active noise control approaches for turbo propeller aircraft [24].

### 3.2. Smart Gearbox Support Strut Concept

Extensive effort has been directed at the IAR to reduce helicopter cabin noise levels resulting from structural vibration of the airframe due to gearbox vibration. The approach investigated is to reduce noise transmitted through the structure from the rotor subsystem by isolating the gearbox through adaptive mounts.

As illustrated in Fig. 8, the adaptive gearbox strut uses the Smart Spring concept to reduce vibratory loads transmitted to the airframe. The adaptive strut is placed in the load path to mitigate mechanical vibration resulting from the tooth meshing noise generated by the engine and gearbox assembly. Moreover, the controllable engagement of the piezoelectric actuators provides the necessary bandwidth and adaptability for the system to react to the vibratory aerodynamic loads from the blade in varied flight conditions.

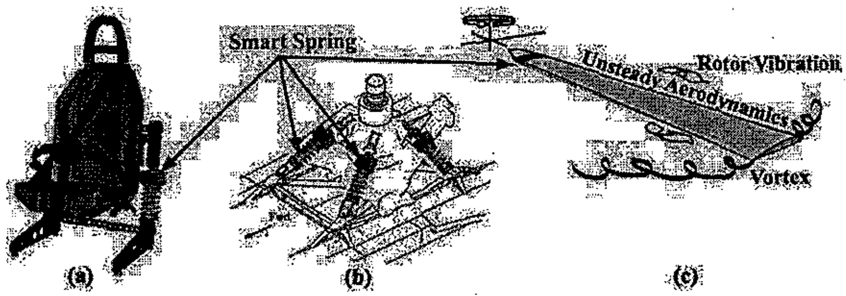
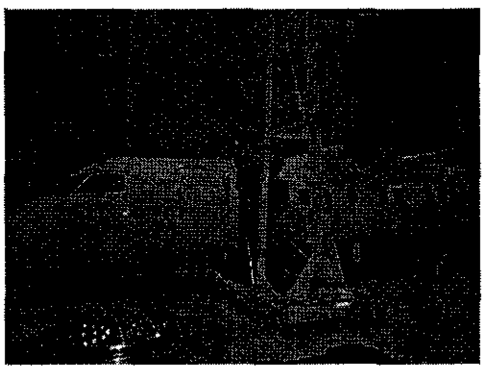



Fig. 8: Smart gearbox strut concept for helicopter cabin noise reduction

### 3.3. Active Structural Acoustic Control of Cabin Noise

Active Structural Acoustic Control (ASAC) is an alternative approach to reduce helicopter cabin noise by weakening the coupling between exterior and cabin acoustics. In this approach, the transmission of sound is impaired before it enters the cabin. Therefore, the noise is treated close to the source to achieve superior performance. Since helicopter cabin noise spectrum is dominated by gear meshing tones at the high frequency regime, the reduction of airframe vibrations at these discrete frequencies will reduce the energy radiated into the cabin, and provide significant improvement to helicopter cabin noise levels.

#### 3.3.1. Configuration of Cabin Noise Control System



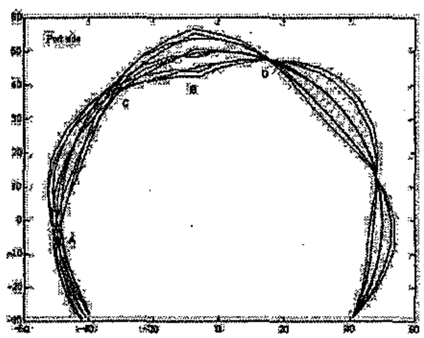


Fig. 9: The tested Dash-8-100/200 airframe Fig. 10 Operating deflection shape due to BPF noise

The ASAC approach proposed has been investigated for a related application on a full-scale Dash-8-100/200 airframe at the IAR, as shown in Fig. 9 [25]. Four loudspeaker units were used to generate a simulated exterior engine sound field from the port side. A total of 199 $d_{31}$-mode piezoelectric ceramic actuators were bonded to three ribs of the airframe structure using conductive epoxy. This allowed the metallic airframe to be used as a common ground for all actuators, and avoided the difficulty of making an electrical connection to the lower actuator surface without interfering with the bond. The number of actuators could be changed to optimize the structural actuation for either mode shape or input power by altering the connecting wiring. The dimensions of the individual actuator elements were chosen to minimize the bonding thickness, and therefore the loss of actuator effectiveness due to the attachment of flat piezoelectric patches to the curved fuselage. Since high strains were desired, and the piezoelectric actuators used were relatively thick, high voltage amplifiers were used in the process. It should be emphasized that in an actual application, such a large voltage requirement would not be necessary because optimization of the actuator design can reduce the required voltage.

### 3.3.2. Piezoelectric Actuator and Sensor Placement

The spatially distributed nature of the propeller noise field impinging on the fuselage contributes significantly to its structural response. A force distribution which has uniform phase and magnitude over a particular region, typically excites modes that are half-waves in the axial and circumferential directions which approximately match (geometrically) the pressure distribution. However, for pressure distributions with spatially varying phase and magnitude, this relationship is much more complex. Nevertheless, the spatial extent of the pressure plays a major role in determining the dominant modes which are excited. In an analogous fashion, the spatial extent of the piezoelectric elements also determines which modes can be controlled by the actuator. Therefore, the coverage of the piezoelectric actuator elements should encompass the region of high propeller pressure but need not extend beyond this region.

The vibration patterns or operating deflection shapes of the fuselage were used to determine the specific elements to be included in the actuators which then defined the circumferential limits of each actuator. The deflection shapes produced by the sound field were determined experimentally at a single axial location. The deflection shape corresponding to the BPF at 910 rpm (61 Hz) is shown in Fig. 10.

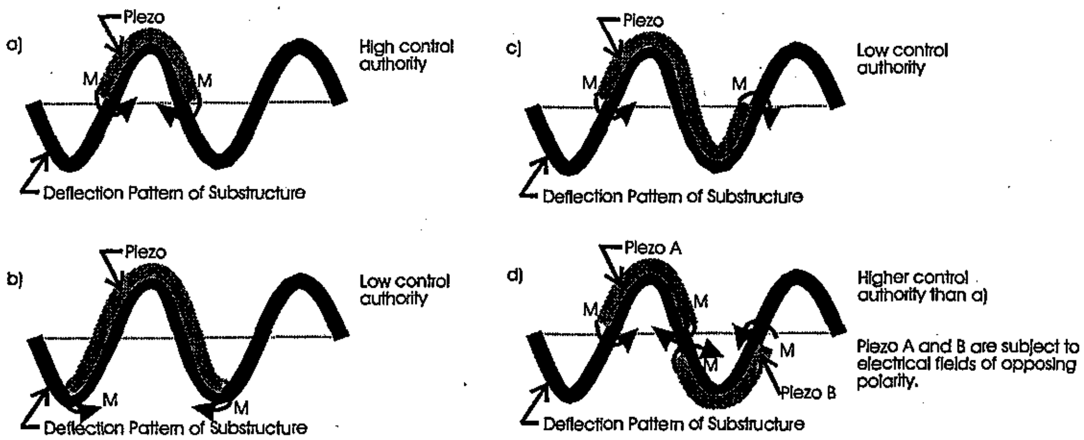



Fig. 11: Piezoelectric actuator placement for enhanced actuation authority

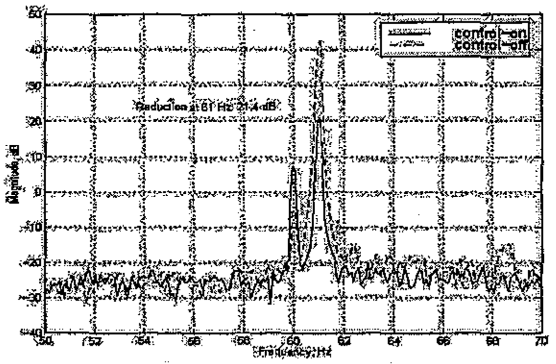
The one-sided or asymmetric actuation of a structure such as an aircraft fuselage by bonded piezoelectric actuators will produce both flexural and in-plane vibration. For the case of noise transmission into an aircraft cabin, flexural vibrations are of greater significance due to their ability to couple with acoustic waves. For flexural vibration, the predominant actuation mechanism due to a uniform rectangular piezoelectric actuator is the presence of bending moments at its ends. This is true of both monolithic and segmented piezoelectric actuators. With this in mind, an actuator should be positioned such that its ends coincide with regions of high (and opposite) angular rotation in order to have high control authority over that particular mode or deflection pattern. If the ends are located in regions of high displacement (and hence low rotation) or if the rotations at the ends are nearly equal in both magnitude and direction, low control authority over that mode or deflection shape results. Based on this approach, the control authority over a mode can also be enhanced by appropriately reversing the electrical field of elements over regions of opposite modal displacement. These concepts are illustrated in Fig. 11.

Accelerometers were selected as the primary sensors in this investigation to correlate the radial structural deformation with noise transmitted into the cabin. They were attached at various locations in the actuator plane on the fuselage. The noise attenuation performance was monitored by a microphone network placed at multiple locations that had high noise levels inside the cabin. Both feedback and adaptive feed forward strategies have been developed to attenuate the structural vibration and cabin noise simultaneously.

### 3.3.3. Full-Scale Aircraft Experiment

Closed-loop experiments have been conduced to verify the effectiveness of tonal noise reduction through structural vibration control at discrete frequencies. The control at 61 Hz used accelerometer feedback to regulate the three actuators discussed earlier. Four accelerometers were placed over the region of high open loop vibration at each of the three axial locations where piezoelectric actuators were bonded. Averages of each set of four measurements were used as the sensed quantities for both feedback and feed forward control approaches. The feedback control system was designed with peak loop gains of between 25 and 30 dB, gain margins of between 8 and 15 dB, and phase margins of between 14° and 50°, for the diagonal elements of the transfer function matrix. The feed forward approach used filtered-x LMS adaptive control. An IIR filter with 15 forward and 14 recursive filter coefficients was used to model the secondary path using a band limited random signal between 55 and 75 Hz for off-line LMS system identification of the plant. The control filter was chosen to be a FIR filter with 15 coefficients. The resulting attenuation in the vibration and interior noise levels was significant. For the feed forward control case, a reduction of 21.4dB in averaged vibration level was obtained. The spectra for the control-on and control-off data for accelerometer are shown in Fig. 12.

In order to achieve better noise reduction performance, microphones were positioned at the seated head height for the two port side seats and at standing height for the aisle center. They were similarly grouped as the accelerometers. The averaged output was used as the error signal to develop the control law. The control-on and control-off noise level spectra for the feed forward case are presented in Fig. 13, which shows a reduction of 24.2dB at the target frequency.



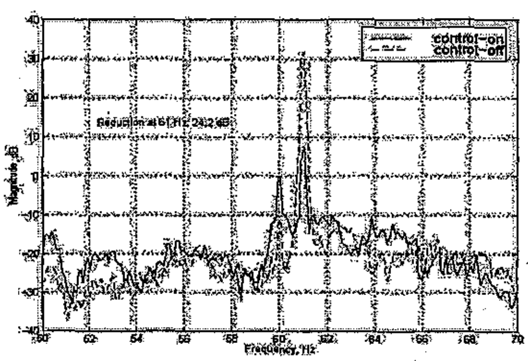



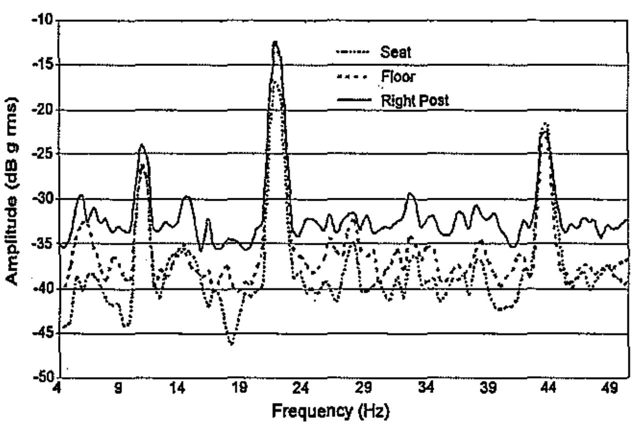

Fig. 12: Vibration control results at 61Hz Fig. 13: Cabin noise spectra under 61Hz tonal excitations

Compare the results, the noise reduction obtained using microphone error sensing were generally superior to those for vibration sensing. The arithmetic average of the reductions at all microphone locations was over 2 dB higher than the average when accelerometer error sensing was employed. The reductions were also global in nature. The investigation also revealed that the noise reduction performance was sensitive to the vibration modes of the airframe. Because the structural modes responsible for the noise transmission may not be dominant in the vibration field, the coupling effect between the vibration and acoustic fields had to be scrutinized. Optimization of actuator configuration and control strategy was also critical to aircraft cabin noise performance improvement.

## 4. ADAPTIVE SEAT MOUNTS TO IMPROVE RIDE QUALITY

### 4.1. Helicopter Seat Vibration Concerns

In helicopter flight, the vibratory disturbance loads are transferred throughout the vehicle. This not only creates damage to expensive components and incurs high maintenance costs, but also affects the operating performance and health of the aircrew. In the short term, the mechanical vibration transmitted to the aircrew leads to discomfort and interferes with operative performance and safety. In the long term, continuous exposure to the vibration transferred through the seat to the pilot's body can cause physiological damage to the aircrew that may lead to occupational health issues [26, 27]. This problem becomes more severe when new instruments, such as a Head-Up Display (HUD) and Night Vision Goggle (NVG), are integrated onto the helmet of pilots in modern military helicopters. Therefore, reduction of vibration transmission to helicopter aircrew is not only necessary to improve the ride quality of the vehicle, but also to improve the operating environment for the aircrew.



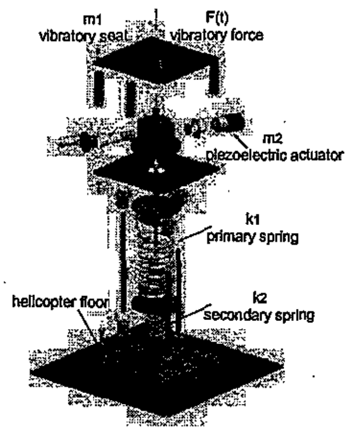




Fig. 14: Vibration spectra at 140kt forward flight Fig. 15: Smart Spring based adaptive seat mount

Flight tests have been conducted by the National Research Council Canada on a four-bladed Bell-412 test helicopter to measure the vibration response [28]. The vibration spectra at 140kt forward flight condition measured on the seat, floor and right door post are shown in Fig. 14. Test results showed that the dominant peaks of the seat and floor vibration spectra are N/rev of the rotor speed. Specifically, the highest peak of Bell-412 was 21.48Hz, which corresponded to 4/rev of the rotor passage frequency. In addition, vibratory peaks that correspond to 8/rev, 2/rev and 1/rev harmonics were clearly visible. Moreover, the vibration levels experienced by the pilot generally increased with the increase in the forward flight speed.

### 4.2. Adaptive Seat Mount Concept

Development of adaptive seat mounts provides a solution to reduce the vibration transmitted from the seat to the aircrew. Compared with the helicopter seat with fixed cushion damping, the adaptive seat mount can adapt to the broadband vibration frequencies resulting from varied flight missions as well as the changes of aircrew individuals.

Two adaptive seat mount concepts have been investigated by National Research Council Canada to reduce helicopter seat vibration based on smart structure technologies. One concept employed the patented Smart Spring approach to modify the impedance properties of helicopter seat mount; and another was an adaptive interface concept that uses piezoelectric actuators to counteract vibratory loads directly to reduce vibration transmitted from the fuselage to the aircrew.

### 4.2.1. Smart Spring Based Adaptive Seat Mount

An adaptive seat mount that employs the patented Smart Spring technology to modify the seat impedance properties adaptively is shown in Fig. 11. The principle of the Smart Spring to vary structural impedance properties has been discussed previously. Compared with other semi-active seat isolation systems based on MR or ER fluid, the Smart Spring approach can preferentially control combinations of the effective seat damping and stiffness. This ability enables the adaptive seat mount to reduce vibration transmission to the aircrew in an adaptive manner.

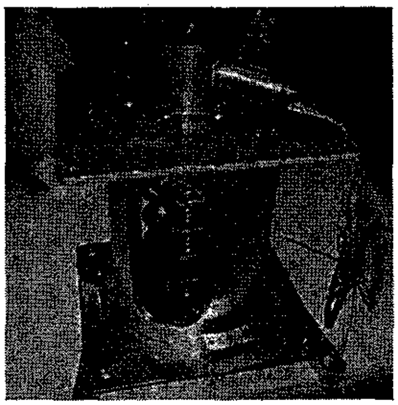


Fig. 11: Smart Spring based adaptive seat mount

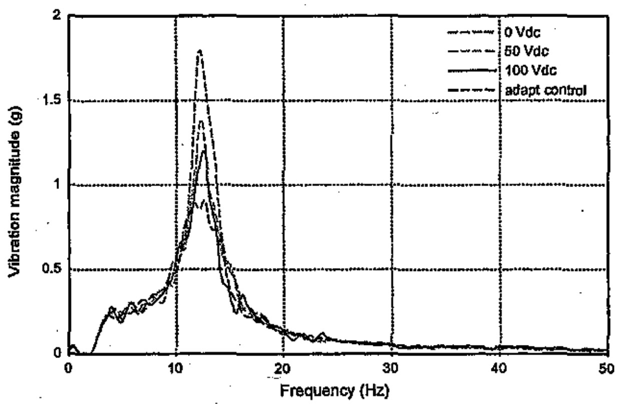



Fig. 12: Vibration control results on the shaker

A proof-of-concept hardware model has been designed and fabricated to integrate key elements into the adaptive mount to implement the impedance control concept, as shown in Fig. 11. The objective of this work was to evaluate the viability of the concept for helicopter seat applications. The actuator assembly contained a piezoelectric stack actuator and a load cell for actuation force measurement. The seat vibration was measured using a miniature accelerometer located at the top of the assembly. The primary spring represented the stiffness of a typical seat attachment, and the secondary spring provided the active stiffness to control seat vibration.

Preliminary tests have been conducted on a mechanical shaker to verify the viability to reduce vibration transmission under broadband frequency disturbances. A random vibration profile of 0.2grms was used during the test, and the frequency band covered 3 to 100Hz. The initial test was performed only in the vertical direction. The damping of the seat was controlled adaptively by a closed-loop control law. During the test, the seat mount damping was tuned to four conditions using external control voltages, namely the low damping, middle damping, high damping and adaptive damping control. The vibration reduction performance was measured, as shown in Fig. 12. As seen from the plots, the resonant vibration peak of the adaptive seat mount was 12.1Hz. As noted above, this proof-of-concept model was not optimized for any particular vehicle but rather meant to demonstrate the concept. The achieved impedance characteristics of the Smart Spring based adaptive mount were similar to the trend of rotor blade assembly described earlier. To avoid the change to seat mount frequency which is a sensitive parameter in aircraft design, the initial clearance between the two springs was tuned such that the 100Vdc control voltage on the piezoelectric actuator only varied the effective damping with the applied voltage range of 0 to 100V. The 0Vdc introduced the lowest damping ratio, therefore the seat vibration spectrum showed the highest resonant peak under the random excitation. The response of 50Vdc and 100Vdc showed comparatively lower resonant peaks due to the increase of damping of the mount. Under the regulation of an adaptive impedance control law, the effective damping of the seat mount was varied adaptively for maximum vibration suppression. With a fully converged algorithm, a 49% reduction of the resonant peak had been achieved.

#### 4.2.2. Adaptive Seat Mount Using Direct Force Control

Another adaptive seat mount concept was based on the direct force control approach. Compared to the Smart Spring approach, this concept provides wider frequency control capability, but requires the piezoelectric stack actuator to provide large actuation displacement and high force simultaneously.

A proof-of-concept hardware has been designed and fabricated, as shown in Fig. 13. The adaptive seat mount was designed to install between the seat frame and helicopter fuselage to reduce the vibration transmitted from the base. It employed three piezoelectric stack actuators to provide actuation authority in the vertical direction. The maximum stroke of each actuator assembly was 0.03mm under 100V control voltage. Under the regulation of an adaptive control law, it was designed to suppress the N/rev harmonic vibrations.

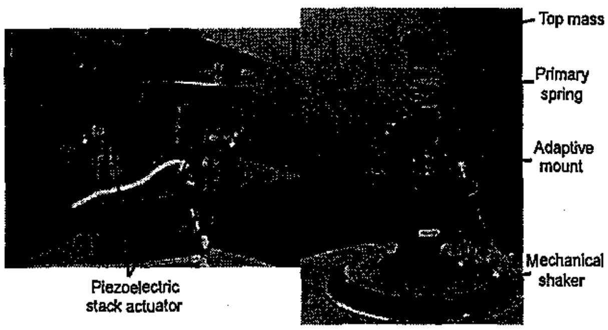


Fig. 13: Adaptive mount based on direct force control

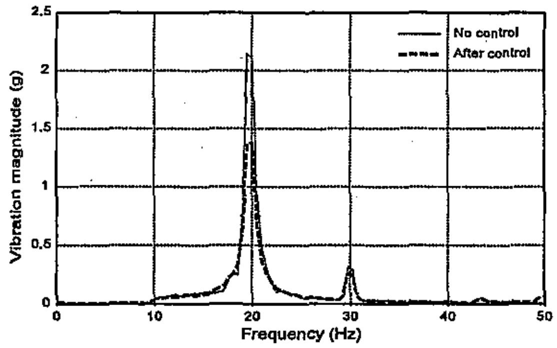


Fig. 14: Random vibration results on the shaker

Shaker tests have been conducted to verify the viability of the concept to reduce seat vibration transmission. A mass-spring system was installed on top of the adaptive mount to simulate a helicopter seat system. A miniature accelerometer was bonded to the top mass to measure the vibration response. The first resonant frequency of the mass-spring system was 20Hz. Various vibration input profiles have been tested, and the proof-of-concept hardware demonstrated reasonable vibration reduction performance.

A typical test result is shown in Fig. 14. The combined vibration input profile included a sine tone of 0.05g peak at 30Hz and a random profile of 0.025grms that covered the frequency band from 5 to 50Hz. As shown in the plots, the vibration response of the seat mount showed two peaks. One was the resonant peak at 20Hz and the other was the forced vibration response at 30Hz. Because the damping of the simulated seat device was relatively low, the resonant peak was more prominent than the forced vibration peak. The closed-loop control algorithm was a combination of two separate control laws: an linear Quadratic Gaussion (LQG) control law was designed to suppress the resonant peak at 20Hz and a Filtered_x LMS feed forward control law to reduce the sine tone which simulated the N/rev rotor speed. A reference signal of the sine tone was digitally synthesized for the feed forward control law by on-line identification of the accelerometer signal. Test results showed that both vibration peaks were suppressed effectively with an adaptive control law. The reduction at 30Hz was lower compared to the resonant peak at 20Hz because this off-resonant frequency required the piezoelectric actuator to provide considerable displacement and force simultaneously to counteract the corresponding vibration input. However, the piezoelectric actuators for this proof-of-concept model only had limited displacement authority, which affected the performance achieved in this test. This result not only verified the viability of the approach to reduce vibration transmission in a representative helicopter disturbance spectrum, but also showed that piezoelectric actuators with large actuation displacement were essential for effective vibration suppression in full-scale helicopter seat applications.

## 5. CONCLUSION

This paper has reviewed the advances of smart structure technologies for helicopter vibration and noise control, which are being developed at the Aeroacoustics and Structural Dynamics Group of the Structures and Materials Performance Laboratory, Institute for Aerospace Research, National Research Council Canada.

The blade impedance control approach has been developed using the Smart Spring concept to reduce the vibratory loads transmitted from helicopter rotor blades to the airframe. This approach does not rely on the piezoelectric actuator to provide large force and displacement simultaneously to cancel external aerodynamic loads directly. In contrast, it varies blade impedance properties adaptively with limited displacement of the piezoelectric actuator to reduce helicopter vibration. The Smart Spring provided a promising technique to overcome some of the significant problems associated with other piezoelectric actuator based active vibration control approaches.

The Smart Gearbox Strut is an innovative solution to reduce helicopter cabin noise by the control of gearbox induced airframe vibration. Research of a smart helicopter gearbox strut is continuing to reduce the structural-borne tonal noise in the cabin. The Active Structural Acoustic Control approach has been developed to reduce aircraft cabin noise by weakening the coupling of exterior and cabin acoustics fields. Experiments on a full-scale aircraft airframe verified the effectiveness for tonal noise reduction through airframe vibration control at the discrete frequencies.

Two adaptive seat mount concepts have been investigated to reduce vibration transmitted through the seat to the aircrew. Prototype hardware tests verify that both approaches are capable of reducing seat vibration. They provide a feasible solution to improve helicopter ride quality to mitigate the vibration related issues of the aircrew.

These specific examples of current research provide insight into only a few possible opportunities for the application of smart structures technology to adaptively reduce noise and vibration in aerospace vehicles. The smart structures technology provides the innovative ability to adaptively reduce noise and vibration over a wide frequency bandwidth to accommodate varying flight conditions and aircraft configurations to enhance crew performance and passenger comfort.

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