Research of an Active Tunable Vibration Absorber for Helicopter Vibration Control

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Abstract: Significant structural vibration is an undesirable characteristic in helicopter flight that leads to structural fatigue, poor ride quality for passengers and high acoustic signature. Previous Individual Blade Control (IBC) techniques to reduce these effects have been hindered by electromechanical limitations of piezoelectric actuators. The Smart Spring is an active tunable vibration absorber using IBC approach to adaptively alter the "structural impedance" at the blade root. In this paper, a mathematical model was developed to predict the response under harmonic excitations. An adaptive notch algorithm was designed and implemented on a TMS320c40 DSP platform. Reference signal synthesis techniques were used to automatically track the shifts in the fundamental vibratory frequency due to variations in flight conditions. Closed-loop tests performed on the proof-of-concept hardware achieved significant vibration suppression at harmonic peaks as well as the broadband reduction in vibration. The investigation verified the capability of the Smart Spring to suppress multiple harmonic components in blade vibration through active impedance control.

Key words: helicopter vibration control; active tunable vibration absorber; smart structure

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摘　要: 振动问题是直升机设计中的难题, 会导致机体结构疲劳、舒适性降低和高噪声等问题。通常的单桨叶控制方案由于受压电驱动器机电性能的限制而难以实现。智能弹簧是一种采用单桨叶控制原理的主动调谐式吸振器, 它通过压电驱动器自适应控制桨叶根部的结构阻抗, 达到振动控制目的。建立了智能弹簧的简化模型, 对其谐波响应控制特性进行研究; 采用频率分析和数字信号合成技术产生参考信号, 在DSP平台上设计自适应陷波算法对智能弹簧驱动器组件进行控制; 模拟和风洞实验结果均表明智能弹簧能够在较宽频率范围内对桨叶的谐波响应进行有效控制, 验证了通过主动阻抗控制实现直升机桨叶振动控制的可行性。

关键词: 直升机振动控制; 主动调谐阻尼器; 智能结构
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Helicopter rotors operate in a highly complex, unsteady aerodynamic environment caused by cyclic variation of centrifugal and aerodynamic loads on the blades. Significant structural vibration due to unsteady aerodynamics caused by blade vortex interaction (BVI) and dynamic blade stall is a notable and undesirable characteristic of helicopter flight[1].

The most important sources that contribute to the vibration in the helicopter airframe are rotor hub reaction force induced by the inertial and aerodynamic loads on the blades. Most of the aerodynamic vibratory loads produced by the rotor system cancel at the hub, except for their PN/r harmon-

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ics, where P is an arbitrary integer number, and N is the number of blades[2]. Due to the inherent coupling between the rotor system and the airframe, vibratory hub loads are transferred throughout the helicopter structure. This structural vibration contributes to poor ride quality for passengers, fatigue of expensive structural components, and high acoustic signature for the vehicle.

Increasing efforts have been devoted to implement active control techniques to achieve significant reductions in vibration. These active approaches promise vibration suppression in a broadband of frequencies unlike the passive techniques that are typically capable of controlling vibration only at a specific frequency.

Individual Blade Control (IBC) is one of the active control techniques currently under development that promises suppression of discrete and broadband vibration. IBC places actuators on the blade or the swash plate to control each blade independently and simultaneously. There are a number of conventional IBC implementations, but most of them have serious shortcomings[3]. Capabilities of conventional servo-hydraulic actuator systems are limited due to system complexity, slow response, *etc*. Recent advances in active material actuators provide a potential to overcome these limitations of conventional actuators that offer direct conversion of electrical energy to produce high frequency mechanical motion. Furthermore, these systems have fewer parts compared with conventional systems, which could simplify the system greatly.

IBC using active material actuators for rotor vibration suppression has been implemented using two distinct actuation concepts, namely, discrete and integral. The discrete actuation concept employs actuators embedded in the blade to control a trailing edge servo-flap[4,5]. Unfortunately, a fundamental problem with these approaches is the limited displacement capability of piezoelectric actuators. Robust and compact displacement amplification devices are needed to obtain the required flap deflection under the extreme aerodynamic environment that exists in the rotor blade.

In the integral actuation concept, the actuator system is either embedded or bonded to the skin along the blade span to obtain a smooth and continuous structural deformation[6,7]. The drawback with this approach is the requirement of very high voltage to induce sufficient actuation from the actuators, which have been operated with a 3000VPP driving voltage cycle to generate the required blade twist for rotor vibration suppression[8].

It is important to note that all of the above IBC approaches attempted to actively alter the time varying aerodynamic loads on the blade. However, successful implementation of these approaches has been hindered by the electromechanical performance limitations of active material actuators, specifically piezoelectric actuators, used in these applications. Restricted deformation capabilities of these actuators required complex displacement amplification mechanisms or extremely high voltages to achieve the required vibration suppression performance.

The Smart Spring provides a unique approach to overcome these difficulties. This paper describes the concept of Smart Spring for helicopter rotor vibration suppression. A mathematical model was developed for the non-linear dynamic characteristics of the Smart Spring and to provide the basis for active impedance control to suppress the multiple harmonic components in the blade vibration. Based on this model, an adaptive notch algorithm was developed for the higher harmonic vibration suppression applications. The effectiveness had been verified both in laboratory and wind tunnel tests using the proof-of-concept hardware platform.

1 Mathematical Model of Smart Spring

In contrast to direct control of aerodynamic loads through a trailing edge flaps or integral blade twist, Smart Spring is an approach to suppress the rotor vibration by adaptively varying the structural impedance of blades. Analytical modeling of the Smart Spring concept demonstrated that active adaptation of dynamic impedance of the blade at the root is an effective approach to suppress the rotor vibration.

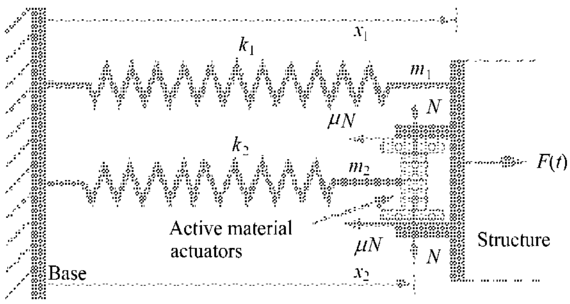

Fig. 1 Smart spring concept

The Smart Spring is a patented concept based on an active stiffness device that adaptively varies the structural impedance to suppress vibration[9]. The mechanism, as shown in Fig. 1 can be seen as an Actively Tunable Vibration Absorber (ATVA) that exploits the large stiffness and bandwidth of piezoelectric materials, while circumventing their lack of power at lower frequencies. The Smart Spring can be mathematically modeled as two springs, k_1 and k_2, within the same structural configuration as shown in Fig. 1. The external force F is applied to one side of the structure and provides the vibratory excitation.

When the actuator is turned "off", the vibratory forces are transmitted only through the constant stiffness spring designated by k_1. When the actuator is turned "on", the frictional force μN engages active spring to carry the vibratory loads through Smart Spring. Controllable engagement of the active spring k_2 enables the introduction of active impedance control in order to filter the vibratory forces before transferring to the opposite side of the structure.

When motion is involved, friction coefficient μ is expected to vary from its dynamic, lower limit, μ_k to its static, higher limit, μ_s. The actual process may be complicated, however, as an approximation, one can assume that a process where dry friction or "structural damping" is present. This constant contact with the structural sleeve is beneficial to the stability of control process because it is preferable to have a smooth variation on the stiffness rather than an abrupt one. In the later case, an "on-off" type of control would be present. Such a system would excite many structural modes during the transients, degrading the robustness of the system. Therefore, if the external force F is harmonic on the frequency and assuming that the total inertia of the system is represented by m, the equation of motion in the horizontal direction x is given by

$$m\ddot{x} + k(t)x = Fe^{i\omega t} \quad (1)$$

In the equation, the magnitude of stiffness varies between the two theoretical limits, *i.e.*

$$k_1 \leq |k(t)| \leq k_1 + k_2 \quad (2)$$

If the control law is also harmonic and the variation on stiffness is proportional to the external electrical stimulus, which is applied to piezoelectric stacks, it can be expanded into a complex Fourier series

$$k(t) = \sum_{r=-\infty}^{r=\infty} k_r e^{ir\Omega t} \quad (3)$$

where $\Omega = 2\pi/T$, T is the period associated with the cycle of actuation and the coefficient can be expressed as

$$k_r = \frac{1}{T}\int_0^T k(t)e^{-ir\Omega t}dt \quad r = 1, 2, \cdots, \infty \quad (4)$$

Assume that the solution is harmonic on the multiples of exciting frequency

$$x(t) = \sum_{r=-\infty}^{\infty} x_r e^{ir\omega t} \quad (5)$$

where the coefficients associated with the negative harmonics are the complex conjugates of their positive harmonic counterparts ($x_{-r} = x_r^*$). Also assuming that the control law is composed of only one harmonic and the mean value of the stiffness coefficient is k_0, one has

$$k(t) = k_1^* e^{-i\Omega t} + k_0 + k_1 e^{i\Omega t} \quad (6)$$

Since $k_{-1} = k_1^*$, substituting $k(t)$ into Eq. (1) and Eq. (5), the following is obtained

$$\sum_{r=-\infty}^{\infty}[(-mr^2\omega^2 + k_0)e^{ir\omega t} + k_1^* e^{i(r\omega-\Omega)t} + k_1 e^{i(r\omega+\Omega)t}]x_r = Fe^{i\omega t} \quad (7)$$

Assuming that the control frequency is equal to the exciting frequency, $\omega = \Omega$ (This is actually the case of most applications involved in helicopter IBC), one obtains

$$\sum_{r=-\infty}^{\infty}[(-mr^2\omega^2+k_0)e^{ir\omega}+k_1^*e^{i(r-1)\omega}+k_1e^{i(r+1)\omega}]x_r=Fe^{i\omega} \quad (8)$$

A transformation of the dummy indices in the summation is performed next. In Eq. (8), the two main terms involving frequency shifts are rewritten as

$$x(t)=\sum_{r=-\infty}^{\infty}x_re^{i(r-1)\omega}=\sum_{r+1=-\infty}^{r+1=\infty}x_{r+1}e^{ir\omega} \quad (9)$$

$$x(t)=\sum_{r=-\infty}^{\infty}x_re^{i(r+1)\omega}=\sum_{r-1=-\infty}^{r-1=\infty}x_{r-1}e^{ir\omega} \quad (10)$$

To replace the corresponding terms in Eq. (8), the equation becomes

$$\sum_{r=-\infty}^{\infty}[(-mr^2\omega^2+k_0)x_r+k_1^*x_{r+1}+k_1x_{r-1}]e^{ir\omega}=Fe^{i\omega}+Fe^{i\omega} \quad (11)$$

where the exciting force is extended to include both the mirror image and negative harmonics to keep symmetry of the series. Also notice that $F_{-1}=F_1^*=F$ since F is a real variable. Applying harmonic balance to Eq. (11), the following set of tridiagonal equations on the complex coefficients of the frequency response is obtained

$$\begin{bmatrix}
\ddots & \ddots & & & & & \\
\ddots & -4m\omega^2+k_0 & k_1^* & & & & \\
 & k_1 & -m\omega^2+k_0 & k_1^* & & & \\
 & & k_1 & k_0 & k_1^* & & \\
 & & & k_1 & -m\omega^2+k_0 & k_1^* & \\
 & & & & k_2 & -4m\omega^2+k_0 & \ddots \\
 & & & & & \ddots & \ddots
\end{bmatrix}\cdot
\begin{bmatrix}\vdots\\ x_{-2}\\ x_{-1}\\ x_0\\ x_1\\ x_2\\ \vdots\end{bmatrix}=
\begin{bmatrix}\vdots\\ 0\\ F\\ 0\\ F\\ 0\\ \vdots\end{bmatrix} \quad (12)$$

The harmonic coefficients with indices −1, 0 and 1 can be expressed as

$$\vdots$$

$$x_{-1}=x_1^*$$

$$x_0=\frac{-(k_1+k_1^*)}{(-m\omega^2+k_0)k_0-2k_1k_1^*}F \quad (13)$$

$$x_1=\frac{(-m\omega^2+k_0)k_0+k_1^2-k_1k_1^*}{(-m\omega^2+k_0)[(-m\omega^2+k_0)k_0-2k_1k_1^*]}F$$

$$\vdots$$

Hence, a discrete complex frequency response characterized by magnitudes and phase shifts is obtained. As an approximation according to Eq. (13), it can be deduced that the introduction of the active spring, or "impedance control," can lead to redistribution of the dynamic response spectrum of the structure.

This analysis shows integration of the Smart Spring at the root of the blade introduces a nonlinear variation in structural impedance. Furthermore, single harmonic control of the Smart Spring leads to the changes of multiple harmonic components in response. It suggests that by actively controlling the Smart Spring, according to an appropriate harmonic control law, multiple harmonic components in the root response can be changed at the same time.

The Smart Spring device is placed near the root of the blade. By dynamically altering the vibration response of the helicopter blade, it can control the vibratory reaction forces in the hub[10, 11]. The root is the optimum location for Smart Spring device to dampen blade vibration because the structural loads are the highest and the damper has the greatest effect. The adaptive nature of the Smart Spring controller is able to track the changes in the fundamental vibratory frequencies and operates accordingly to provide the greatest vibration suppression performance. This is a vast improvement over passive vibration suppression techniques that may only dampen rotor vibration at a specific frequency.

2 Proof-of-Concept Developments

Proof-of-concept hardware was fabricated to verify the Smart Spring concept for active blade vibration control. At the preliminary stage, the hardware was designed to suppress torsional vibration through adaptive structural control.

The Smart Spring proof-of-concept hardware design included three "actuator units" to apply a balanced load to an output plate. Interaction between the actuator units and the output plate generated the frictional forces necessary to engage the actuator units as active members, and to vary the structural impedance of the device. The primary stiffness of the device was tailored through a central torsion member that matches a typical blade torsional stiffness. The proof-of-concept hardware without a front output plate is shown in Fig. 2(a) and the complete proof-of-concept hardware with the front output plate attached to a mechanical shaker is shown in Fig. 2(b).

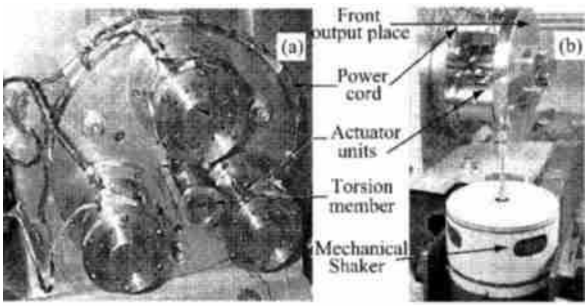

Fig. 2 Proof-of-concept hardware

The mechanical shaker simulated the excitation vibration forces from the rotor blade to the Smart Spring. A load transducer was installed on the shaker stinger to measure the vibratory forces from the shaker. On the opposite side from the shaker, a displacement probe measured the vibratory displacement of the output plate.

Stacked piezoelectric actuators that could provide greater than 340kg of blocked-force at the maximum input voltage of + 100V were chosen for the actuator unitdesign. A total of three actuators were mechanically arranged in series to improve the overall deflection of the actuator unit while the actuators were electrically connected in parallel to maintain force and voltage requirements. A miniature load cell was incorporated into each actuator assembly to measure the force generated by the unit on the front output plate.

The actuator units could be preloaded using a central bolt through the torsion member so as to investigate the effect on impedance augmentation performance of the Smart Spring.

Initial tests on the Smart Spring proof-of-concept hardware had been conducted, which confirmed that the actuation forces generated by each unit could effect impedance augmentation using this Smart Spring proof-of-concept hardware[11].

3 Adaptive Notch Controller Developments

As illustrated in the mathematical model, the Smart Spring provides an approach to suppress multiple harmonic components in blade vibration by actively controlling the piezoelectric actuators. Since most of the helicopter vibration come from only a few higher harmonic components, it is possible to control the Smart Spring in a related frequency domain, to achieve suppression in structural vibration.

3.1 Adaptive notch control

The Smart Spring device exhibits non-linear dynamic characteristics due to high actuation voltages on piezoelectric stacks and friction between the mating surfaces. Therefore, an adaptive notch controller based on Filtered-U recursive LMS algorithm was developed to suppress the harmonic components in blade vibration.

At the preliminary stage, only torsional vibration was considered. All piezoelectric actuator units were connected in parallel as one independent control channel. The structure of a one-channel adaptive notch controller with on-line identification of the secondary path and a reference signal synthesizer was shown in Fig. 3. A MIMO controller could be developed on the basis of this controller to satisfy different configurations of the piezoelectric actuators for future bending and bending-torsion combined vibration control.

The adaptive notch controller could be described in the discrete time domain[12]. The aerodynamic load $g(n)$ was transferred from the blade to the rotor hub according to the transfer function matrix G_p. This was the primary path, producing a disturbance vibration vector represented by the signal $d(n)$. The secondary path G_s represented the additional impedance provided by the active spring. The actuator input and control output were described as $u(n)$ and $y(n)$, respectively. The vibration of the blade measured by the displacement probe was denoted by $e(n)$. An Infinite Impulse Response (IIR) controller, which was adapted by the output signal of a status observer, was used to filter the reference input signal $x(n)$ to produce the control signal. It was then amplified and applied to the piezoelectric actuators. The overall objective of

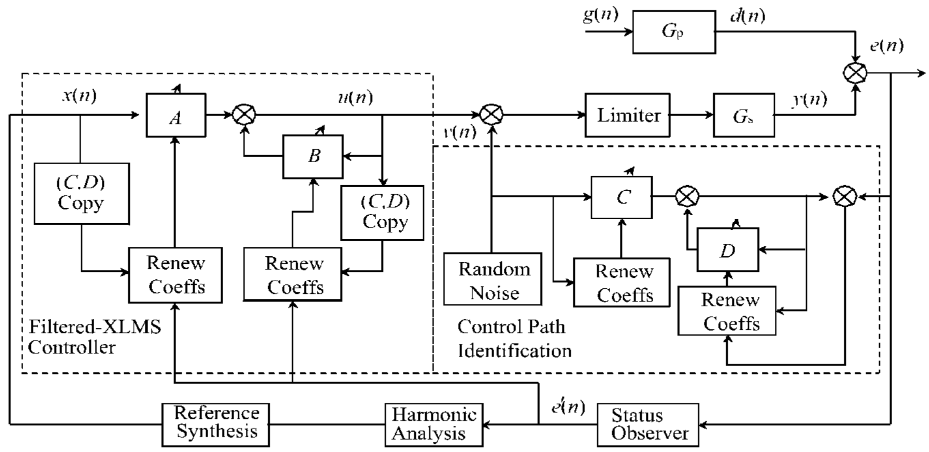

Fig. 3 Structure of adaptive notch controller

the adaptive notch controller was to minimize the RMS value of blade torsional vibration and the hub reaction force.

A secondary path on-line identification loop to take time-variable characteristics into account was also shown in Fig. 3. This required the identification process to occur in parallel with the control process, however, under the assumption of slowly varying plant model coefficients and zero initial conditions, off-line identification could also be used for the secondary path identification.

3.2 Reference signal synthesis

Digital wave synthesis technique was used to provide the stable reference signal. An advantage was the selectivity of the frequency band, which was important for higher harmonic control problems associated with helicopter vibration. Also, it was only necessary to model the secondary path transfer function over the selected band, which substantially lowered the order of filters and improved the efficiency of the control algorithm.

Since the harmonic frequencies (PN/r) were included in the error signal $e(n)$, both amplitude and phase of the main components were determined using on-line transient FFT analysis technique. Assuming the magnitude and phase of each component in the hub reaction were A_m and ϕ_m, and the interval sampling time was represented by T, the reference signal could be synthesized as a linear combination of sinusoids

$$x(n) = \sum_{m=1}^{M} A_m \sin(\omega_m nT + \phi_m) \qquad (14)$$

where M was the number of harmonic components to be suppressed. As discussed earlier, only the first few main harmonic components contributed to the vibration in helicopters, which could be determined from the response spectrum. The fundamental vibration frequencies were determined and updated on-line to track the rotor speed changes due to variation in projected flight conditions.

3.3 DSP implementations

The algorithm was implemented on a DSP platform equipped with a TMS320C40 processor. Input/Output (IO) functions were provided by a separate subsystem, PCI16O8. Communication between the DSP and the IO board was achieved via a 16-bit high-speed interface. Butterworth anti-alias low-pass filters were used for all IO channels. Data transferring between the host PC and the DSP were provided by a 32-bit DSPLINK interface.

The adaptive notch algorithm was programmed on the DSP platform using C and Assembler language mixed code. Interrupt technique was used for accurate sampling time control, with the sampling rate set as 4kHz. Within the interrupt routine, the vibratory response of the front plate was measured and taken as the error signal. A notch reference signal was synthesized according to the proposed algorithm, from which a control signal was calculated using the filtered-U recursive

LMS algorithm. Data for the transient FFT analysis was re-sampled from the error series to achieve accuracy of 0.1Hz in the harmonic frequency identification.

4 Test Results and Discussion

4.1 Laboratory test

Tests were conducted on the Smart Spring proof-of-concept model at different single frequency excitations produced by the mechanical shaker in the laboratory, as shown in Fig.2(b). The IIR control filter was implemented using 16 forward and 1 recursive coefficients.

A typical Power Spectrum Diagram using the shaker as an exciter was shown in Fig.4. The excitation frequency was 40Hz and the shaker force magnitude was kept constant at 36kg peak-to-peak for all test conditions. 50VDC was applied to the actuator units as a baseline operating condition to provide an initial contact between the mating faces. Increasing voltage to 65VDC only appended additional stiffness to the Smart Spring, which in turn reduced the vibratory displacement by 5.4dB.

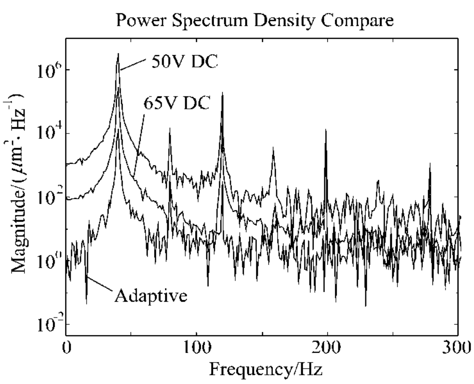

Fig.4 PSD comparison of vibratory response

As seen from Fig.4, a much higher reduction of 11.6dB in vibratory displacement was achieved by active impedance control of the Smart Spring with the adaptive controller (50+ 15VPP), compared with a static 65VDC load. The adaptive controller used two frequencies (40Hz and 80Hz) in the reference signal synthesis to suppress the harmonic peaks. The test results showed that active impedance control through the Smart Spring not only introduced dynamic stiffness variation of the structure, but also introduced effective mass and damping to achieve significant suppression in vibration.

As predicted in the mathematical model, active impedance control of the Smart Spring caused a redistribution of the response spectrum at the harmonic frequencies. By introducing harmonic frequencies in the synthesized reference signal of the adaptive controller, significant suppression in the main peaks (40Hz and 80Hz) was also achieved. Compared with the static control scheme, which imposed a pure increase in stiffness with 65VDC, the adaptive scheme achieved significantly better reduction in the overall frequency range.

4.2 Wind tunnel test

A wind tunnel test of the Smart Spring proof-of-concept hardware was conducted in the low speed wind tunnel (1.9m × 2.7m × 5.2m) at the Institute for Aerospace Research of the National Research Council of Canada to evaluate the performance in a more representative rotor blade aerodynamic loading environment.

The Smart Spring was attached to one end of a 1.45m blade section (NACA0012) with a 0.3m chord in a cantilever configuration to perform damping augmentation of the blade vibration. The blade and the Smart Spring assembly were installed on the wind tunnel force balance so that the Smart Spring device was hidden under the floor. A square tube was installed 1.2 m upstream from the leading edge of the blade to produce vortices. The vortices were measured with a pressure transducer. Accelerometers were placed on the trailing edge of the blade near the root and the tip to measure the vibratory acceleration and displacement of the blade, as shown in Fig.5.

Tests were conducted with zero angle-of-attack to minimize the bending modes of the blade vibration because the Smart Spring proof-of-concept hardware was designed only to suppress the torsional modes at the present stage. The wind speed and dimension of the vortex generator (square tube) determined the fundamental excita-

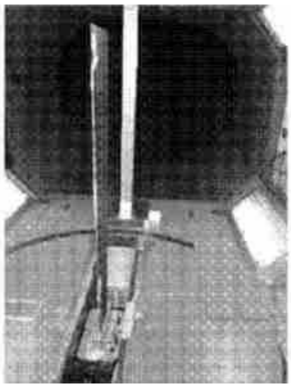

Fig. 5 Wind tunnel test setup

tion frequency. The actuator units were controlled using the adaptive notch controller.

A 10m/s wind-speed with 0.1 m vortex generator produced an excitation frequency of 12.7Hz. Wind tunnel tests with the adaptive controller, depicted in Fig. 6 demonstrated significant vibration reduction. The results indicated that the adaptive control achieved 4.1dB reduction in the vibration with 50VDC + 22VPP. Due to the highly random nature of the perturbation frequency produced by the vortex generator, the adaptive controller used a 12.7Hz synthesized signal as a reference frequency instead of actual perturbation. Higher vibration suppression could be achieved if full voltage were used. However, due to the randomness of the perturbation frequency, the controller did not enable this. Modifications in the adaptive controller may be required to improve performance.

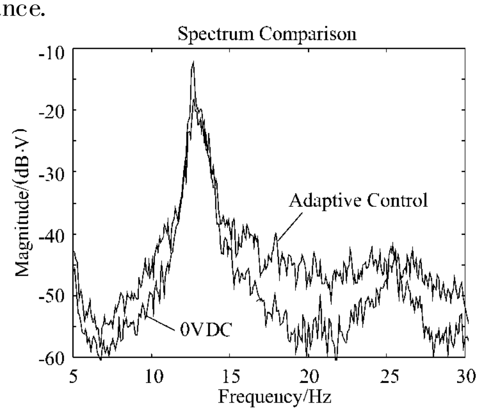

Fig. 6 Adaptive control results

5 Conclusions

The Smart Spring provides a unique approach to helicopter IBC vibration reduction. By altering the structural response of the blade rather than aerodynamic forces, it promises to overcome some of the significant problems in other IBC concepts. A proof-of-concept Smart Spring hardware model has been designed, built, analyzed and tested in the wind tunnel.

Both analysis and experimental results have shown that active impedance control of the structure using the Smart Spring leads to a reduction in the response of the blade. The potential to suppress multiple harmonic components in helicopter blade vibration is significant.

The investigation also demonstrated that the adaptive notch controller could suppress harmonic components of the vibration by forming notches at the designated frequencies. The results from tests provide positive support of the viability of the approach.

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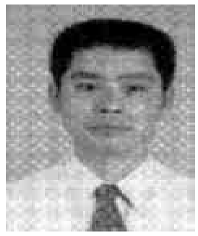

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