HELICOPTER VIBRATION CONTROLLER DESIGH WITH SPECIFIED CLOSED LOOP DAMPING

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ABSTRACT

In this paper, the subject of the design of active vibration control of the helicopter airframe structural response (ACSR) is discussed. In the helicopter vibration reduction techniques, including higher harmonic control, individual blade control, trailing edge flap control, active control of structural response, active twist control, smart springs concept, hub mounted vibration suppressers and other recent proposed techniques, the goals are to provide counter oscillatory loads and counter oscillatory motion to cancel the incoming vibratory loads and oscillatory motion from the rotor system, transmission and other sources to the airframe. In this paper, the goal is different and it is to design controllers to increase the closed loop damping of the helicopter structural dynamic system to a specified level that would reduce the vibrations to the required level within a time frame of interest. This is accomplished by developing a procedure to design second order controllers to obtain the desired closed loop damping (ADC-AF) while maintaining the stability of the closed loop airframe structural dynamic system. The design procedure starts with a coincident closed loop frequency for each mode which is followed by perturbation procedures to seek optimum solutions. The design procedure also yields the needed control authority that should be provided by the actuator. An option to increase the control authority by the use of offset piezoceramic stack actuator assemblies are proposed. Because the controller design procedure is different a simple example is first used to illustrate the controller design for a specified damping constant and the resulting control authority. Then, the control of a simple rotor/ airframe model is discussed.

1. Introduction

As is well known, oscillatory loads and vibrations in helicopters are due to the air flow pattern of the main rotor system; the tail rotor; the transmission; interaction of the rotor and the fuselage; and the engine. The effect of the oscillatory loads and helicopter vibrations are also well known. The effects include fatigue damage of structural components; discomfort of crew and passengers; difficulty in an accurate reading of the instruments and difficulty in the pointing of any military hardware. Thus, the reduction of oscillatory loads and vibration levels in helicopters are important and challenging issue that have been addressed by the researchers and designers, by using active and passive control options.

Since 1967**,** the problem of control of oscillatory loads and vibration reduction in rotorcraft fuselage, using active control has been a subject of interest. Some of the principal approaches [1] include higher harmonic control (HHC); individual blade control (IBC); use of actively controlled flaps that are located on the rotating blades; and active control of structural response. Because HHC and IBC require the oscillation of the whole blade at high frequencies, the required control power is large. This motivated the options of the use of actively controlled trailing edge flaps (ACF) and active control of structural response (ACSR). In the case of the use of piezoelectric-based trailing edge flaps the problem is the low stroke of the flaps. To improve low stroke, the options of resonant actuation [2] and multiple flaps are being explored. These options also increase fatigue failure conditions.

In more recent studies, different combinations of active controllers HHC, IBC, ACF, ACSR and passive control options are being used [3]. Another approach that is proposed by Sikorsky [4] is the hub-mounted vibration suppressors (HMVS) that were jointly developed with Lord Corporation. These consist of four brushless electric ring motors, each with eccentric tungsten mass located atop the main rotor hub. A similar vibration suppressor is planned for main gear box. Another parallel study is that of the use of oscillating Gurney flaps [5]. Similarly, a structural impedance change technique known as the smart spring technique has also been proposed.

In all these vibration reduction techniques, the goals are to counter the oscillatory loads and vibration amplitudes of the airframe either through the rotor mounted control systems, such as the HHC, IBC or ACF systems or through the airframe mounted system such as the ACSR. Such countering techniques usually result in large weight and power for the control system. Thus, the objective of this paper is to use a different option of increasing the damping constants of the closed loop structural dynamic system of the airframe. Thus, the paper only addresses ACSR and actively increasing the damping constants of the airframe structural dynamic system by the use of second order controllers and a proposed offset piezoelectric stack assembly based actuators.

2. ACSR

In 1990, King and Staple [6], Staple [7] and Welsh et. al [8]., reported the work on ACSR that superposes actuator induced response with incoming oscillatory hub response. The sensors were accelerometers and the actuators were electro hydraulic force actuators. The control procedure consisted of minimizing a quadratic performance index. In 1994, Hanagud and Babu [9], presented the use of piezoceramic sensors and actuators for ACSR. They used a single axis finite element beam model, with non-structural masses, and an H- infinity controller. The results indicated a 60% vibration reduction, in the vertical direction at the selected sensor location. During the years 1995-1996, Friedmann, Chiu and Cribbs [10-12] developed a coupled rotor-fuselage-wake model for ACSR and reduce vibrational in all three directions. They used four high force/ low displacement actuators and concentrated vibration on reduction at the pilot seat location. By using 1% tom 1.5% of the total rotor power, 45% of vibration reduction was realized only in the vertical and longitudinal directions. Other reported ACSR works are on Sikorsky [13] UH 60 an d a tail boom model [14].

3. ADC-AF

In the work of this paper, we start with second order controllers that were introduced by Caughey, Goh, Fanson, [15-16]. These second order controllers that were first introduced by Caughey and Goh to reduce spillover effects. Later the design was modified by Fanson to introduce positive position feedback, and increase damping instead of countering the oscillatory loads and vibration amplitudes. In both Goh's work and Fanson's paper, the problem of design of the controllers to achieve the specified closed loop damping coefficient did not form a well posed problem because of large number of unknowns, in the equations. They had to resort to trial and error approach.

Bayon de Noyer and Hanagud [17], and Hanagud [18] solved this problem by first solving for a single closed loop frequency for the cases of both positive position feedback and acceleration feedback. Then, the resulting single frequency solution are now perturbed to seek optimal control solution, in this paper.. The resulting controllers were also compared with LQG and H-infinity designs and later implemented for wind tunnel tests for multi frequency controls of buffet induced vibrations in twin tail high performance aircraft at high angle of attack conditions. These controllers can be designed to realize a specified close loop damping and the resulting controller has been implemented and successfully tested to reduce buffet induced vibration in a wind tunnel using aeroelastically scaled models, by Hanagud and Bayon de Noyer.

4. ADC-AF Model for Helicopters

The multi-mode active damping controller for the airframe structural response can be written for position feedback, velocity feedback or acceleration feedback. In this paper, acceleration feedback is used. It is assumed that the structure is discretized at n locations, with n large enough that all modes of interest are included. It is assumed that x includes displacements (or displacements and slopes, as considered in some models to include rotations as well as externally applied loads and moments). The equation 1 is the structural dynamic model of the structure, with external load or external loads and moments. Γ_{act} is the sensitivity and locations of the actuators. It has zero everywhere except at the locations of the actuators where they are equal to a_{1i}. Similarly Γ_{acc} It is zero everywhere except at the locations of the sensor where they are equal to a_{2i} . The term 1_p refers to the use of a single sensor, such as. The accelerometer.

$$[M]\{\ddot{x}\}+[C]\{\dot{x}\}+[K]\{x\}=-\{\Gamma_{act}\}\lfloor G \rfloor[\Omega_c]\{\eta\}+\{f\}$$

(1)

By setting C and right side of the equation to zero, natural frequencies and modes are determined. Then, modal damping is added to complete the acceleration feedback control equations.

$$\begin{cases}\{\ddot{\xi}\}+[\Lambda_s]\{\dot{\xi}\}+[\Omega_s]\{\xi\}=-\{\tilde{\Gamma}_{act}\}\lfloor G \rfloor[\Omega_c]\{\eta\}+\{\tilde{f}\} \\ \{\ddot{\eta}\}+[\Lambda_c]\{\dot{\eta}\}+[\Omega_c]\{\eta\}=\{1_p\}\lfloor \tilde{\Gamma}_{acc} \rfloor\{\ddot{\xi}\}\end{cases}$$

(2 a,b)

In these equations, ξ is the modal coordinate ϕ^T x ϕ of the structure and ϕ are the modes. The matrix $\Lambda_s = diag\ (2\ \zeta_s\ \varphi_{si})$ are the damping constants of the structure. The second equation 2b is the second order controller in terms of η. This equation resides in a computer.

Λ_c = diag (2 ζ_c ω_{ci}) 3)

ω_{ci} are the natural frequencies of the controller. Ω_c = diag (ω_{ci}^2). Similarly Γ_{acc} is the sensitivity and location of the sensors (accelerometers) The inputs to the controller equation are the accelerations $\ddot{\xi}$ of the structural locations in modal coordinates. Γ_{acc} takes care of the conversion from x to ξ. The output from the controller (1b) forms the input to the structure through the actuators, as the first term on the right hand side of the equation 1a. In equation 1a Ω_s = diag (ω_{si}^2) where ω_{si} are the structural natural frequencies.

Equation 1b resides in a computer. Different terms of the first term on the right-hand side of 1b are the actual control forces and moments that should be applied to the structure and represents the control authorities that should be provided by the actuator. The terms G are the gain or the amplification.

The design of the controller is to determine parameters in Λ_c matrix, Ω_c matrix and G matrix to realize the specified constants in the closed loop system of equtions 1a and 1b. To explain the procedure and results, we consider a single degree of freedom system, in a later section.

5. Goh Fanson and Caughey's Approach to Design Vibration Controllers for Specified Closed Loop Damping:

In reference [15], Goh and Caughey propose a method to design vibration controllers, for single and multiple degrees of freedom vibrating structures. Their objective is to obtain a specified closed loop damping**.** In their approach they consider a position feedback. Such an approach is also applicable to acceleration feedback controller designs. They assume that it is possible to approximate the N degrees of freedom vibrating systems by N number of independent second order linear differential equations, for each mode i using modal decomposition and adding modal damping.

$$\ddot{\xi}_i + 2\zeta_{si}\omega_{si}\ \dot{\xi}_i + \omega_{si}{}^2\ \xi_i = (f_i(t)/m_i) + g_i\ {}_{\omega_{ci}}\eta_i \quad (4)$$

For each mode, a second order controller is of the following form:

$$\ddot{\eta}_i + 2\zeta_{ci}\omega_{ci}\ \dot{\eta}_i + \omega_{ci}{}^2\ \eta = \omega_{si}{}^2\ \xi_i \quad (5)$$

Briefly, their method is as follows: In the first step, Goh and Caughey identify the closed loop transfer function between ξ_i and f_i; and the characteristic equation. Then, they equate the closed loop characteristic equation of a multi degree of freedom version of equations (4) and (5) to a desired closed loop characteristic equation. Then, , their resulting equation is as follows.

$$[s^2 + (\zeta_{si}\ \omega_{si})s + \omega_{si}^2\ \}\ [s^2 + (\ \zeta_{ci}\omega_{ci})\ s + \omega_{ci}^2] - g\ \omega_{si}^2\omega_{ci}^2 = [s^2 + (\zeta_{f1i}\ \omega_{f1i})\ s + \omega_{f1i}^2\ ]\ [s^2 + (\zeta_{f2i}\ \omega_{f2i})\ s + \omega_{f2i}^2\ ] \quad (6)$$

In this equation, ζ_{f1i} , ω_{f1i}, ζ_{f2i} and ω_{f2i}) are the two desired closed loop damping ratios and closed loop natural frequencies that corresponds to the closed loop poles. They attempt to design ζ_{ci}, ω_{ci} and g for prescribed values of the quantity '(ζ_{f1i}'. The subscript 'i' corresponds to the 'ith' mode. Goh and Caughey and later Fanson attempt to calculate ω_{f1i}, ζ_{f2i} , ω_{f2i}, ζ_{ci}, ω_{ci} and g from four equations that result by equating coefficients of the powers of s in the equation (6). Because there are six unknowns and 4 equations, their attempt to obtain a closed form solution fails and they resort to numerical iteration to obtain practical solutions, for some cases.

Subsequent sections show that the equation (7) yields procedures to calculate ζ_{ci}, ω_{ci} and g for prescribed ζ_{f1i}.

6. Design of Acceleration Controller, for Specified Closed Loop Damping Ratio, with Coincident Closed Loop Poles

In the subsequent sections, we, describe and illustrate the modified Goh and Caughey's design procedure for a system with a single degree of freedom and a specified closed loop damping ratio. For example the specified damping closed loop ratio can be 0.25 when the open loop structural damping ratio is 0.02. Procedures. For other specified damping ratios or settling times are similar to the following example. Following the illustration of the controller design, the paper describes perturbation methods to seek other solutions that can lead to optimum design or other sought designs, to realize the same specified damping ratio. This procedure can be generalized to N degrees of freedom systems. Then, the paper discusses piezoelectric material-based actuator design procedures to provide the needed control effort or authority and their experimental verification.

The determination of the controller parameters is from the equation (6) with i=1, for a system with a single degree of freedom. By equating the coefficients of different powers of s of the characteristic equation of the starting system with second order controller to the powers of s of the characteristic equation of the coincident poles of the desired closed loop controller , one obtains the following sets of equations, without the subscript i.

s^3 : $2\zeta_f\omega_f=\zeta_s\omega_s+\zeta_c\omega_c$ (7)

s^2: $2\omega_f^2 + (2\zeta_f\omega_f)^2 =$

$\omega_s{}^2 + \omega_c{}^2 + 4\zeta_s\omega_s\zeta_c\omega_c + \text{abg}\omega_s{}^2\omega_c{}^2$ (8)

s : $2\zeta_f\omega_f^3 = \omega_s{}^2\zeta_c\omega_c + \omega_c{}^2\zeta_s\omega_s$ (9)

s^0: $\omega_f^4 = \omega_s{}^2\ \omega_c{}^2$ (10)

In these set of equations, we have controller design specified value of ζ_f= 0.25; the structural damping ratio is assumed to be ζ_s=0.02; and ω_s is assumed to be known. By using equations (7), (8) and (10), we solve for g, (ω_f/ω_s), (ω_c/ω_s) and ζ_c. Because the equations are nonlinear, we obtain three sets of solutions.

(ω_f/ω_s) = (I)1.0; (II) 0.004006; (III) 24.960 (11)

(ω_c/ω_f) = (I) 1.0;(II) 0.001605; (III) 622.998 (12)

ζ_c = (I) 0.48; (II) 0.02; (III) 0.02 (13)

By using the values of (ω_f/ω_s), (ω_c/ω_s) and ζ_c, from equations (11-13), equation (9) yields the value of g

g= (I) (0.2116/ ab$\omega_s{}^2$); (II)-(386727/ ab$\omega_s{}^2$);

(III) - (0.9964/ ab$\omega_s{}^2$) (14)

After examining the three sets of solutions, it is seen that the closed loop frequencies and controller frequencies are significantly different from the structural frequency ω_s, in the solution set 2 and 3. Also, the margin of stability, from the Routh-Hurvitz criteria are very small. Thus, the only feasible solution set is as follows:

SET I:

(ω_f/ω_s) = 1.0

(ω_c/ω_f) = 1.0

ζ_c = 0.48

g= (0.2116/ ab$\omega_s{}^2$)

The following figures illustrate the effect of the designed controller parameters on damping the oscillations induced by a step input load, with initial damping ratio of 0.02. In figure2, the oscillations are due to a damping ratio of 0.02, without any controllers.

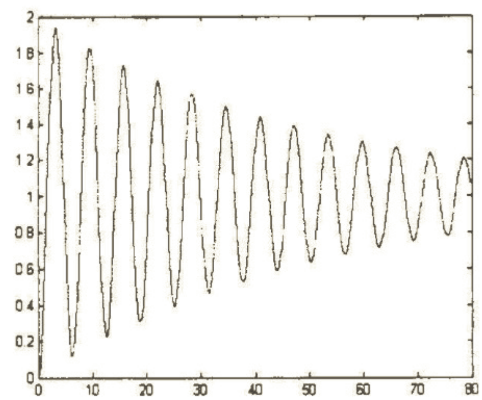

Figure 2: Oscillations due to step input, ζ_s=0.02 and no controller

In comparison to Figure2, the figure 3 illustrates the effect of the controller set I. As c seen in the figure the settling time is significantly reduced and the oscillations are damped within few cycles.

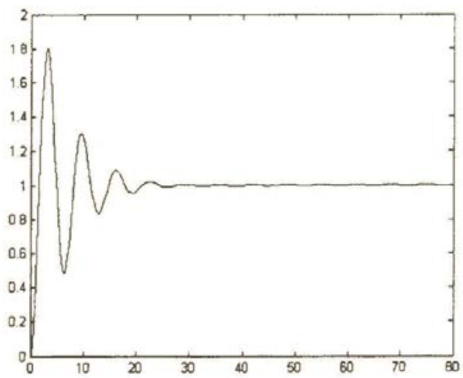

Figure 3: The Effect of Set I Acceleration Feedback Controller; ζ_f= 0.25; ζ_c = 0.48

To illustrate the effect of Set II and set III controllers, Figures 4 and 5 illustrate the effects of the controllers on the step input, with structural damping of 0.02.

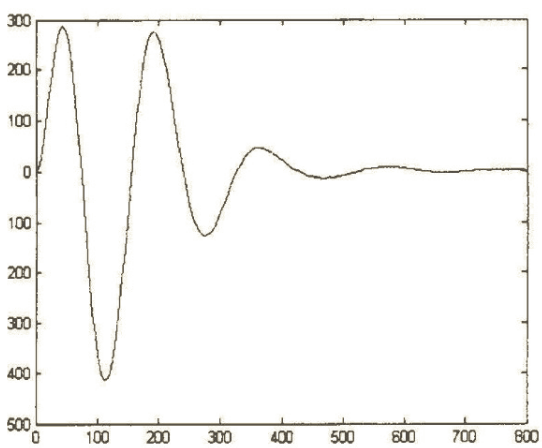

Figure 4: the Effect of the Set II Controller

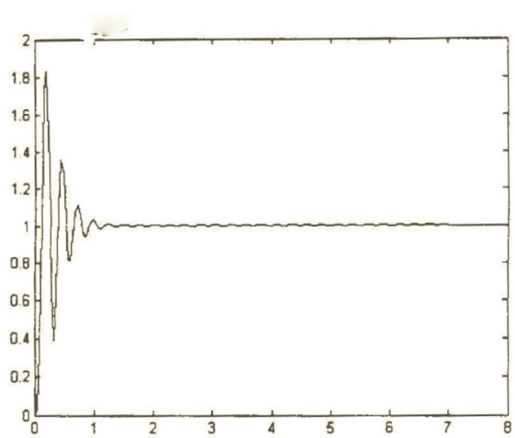

Figure 5: The Effect of the Set III controller

As discussed before, the closed loop frequency and the controller frequencies are very different from the structural frequency in sets II and III. The stability margin is very small. To illustrate, the effect of small stability margin, we select the set III controller and change the gain g from "-(0.9964/ ab$\omega_s{}^2$)" to - (0.996679/ ab$\omega_s{}^2$). The change in the effect of the controller is seen in the Figure 6.

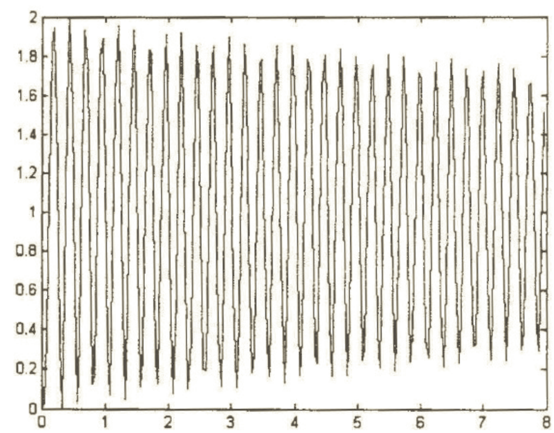

Figure 6: Sensitivity to Small Changes in the Gain in Set III controller

Perturbations to Seek solutions Other Than the Solution with Coincident Closed Loop Frequencies:

For the cases of designs of controllers, with specified closed loop damping ratios, we are able to obtain controller parameters without resorting to iteration. The resulting design procedure requires the assumption of coincident closed loop frequencies. The resulting solution may not be the desired controller or an optimum controller. To seek other controllers, including an optimum controller, subject to restrictions on the selected actuator, we propose perturbations from the acceptable result from the assumption of coincident closed loop frequency, such as, the set I controller in the studied example. Thus, ω_f ,of set I controller, is replaced by (ω_f +δ). Then, by equating the open loop characteristic transfer function with desired closed loop transfer function, the following equations are obtained to replace equations (7) to (11).

s^3 : $2\zeta_f \omega_f = \zeta_s \omega_s + \zeta_c \omega_c$ (15)

s^2: $(2\zeta_f)^2 (\omega_f^2 + \delta^2) + 2(\omega_f^2 - \delta^2) = {\omega_s}^2 + {\omega_c}^2 + 4\zeta_s \omega_s \zeta_c \omega_c + \text{abg}{\omega_s}^2 {\omega_c}^2$ (16)

s : $4\zeta_f (\omega_f^2 - \delta^2)\omega_f = {\omega_s}^2 \zeta_c \omega_c + {\omega_c}^2 \zeta_s \omega_s$ (17)

s^0: $(\omega_f^2 - \delta^2)^2 = {\omega_s}^2 {\omega_c}^2$ (18)

Solutions to these equations, for different values of (δ/ω_s) , yield the following results:

(δ/ω_s) = 0.1; 0.2 0.3 0.4 0.5
` (19)

(ω_f/ω_s) = 1.005 1.0198 1.04403 1.0770 1.118 (20)

(ω_c/ω_f) = 1.0; 1.0 1.0 1.0 1.0 (21)

ζ_c = 0.482 0.49 0.50. 0. 518 .539 0.54 (22)

g= (0.2514/ ab${\omega_s}^2$); 0.3708 0.56984 0.84852 1.20688 (23)

These solutions display a damping behavior similar to the effect of set I controller. The following figure illustrates the performance for (δ/ω_s) = 0.5, for a step input loading

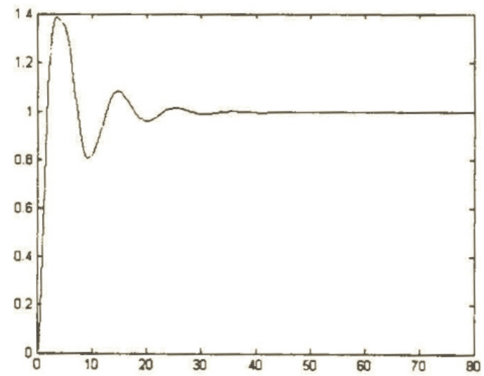

Figure 7: Damping with Perturbed Solution (δ/ω_s) = 0.

.9. Optimal Solution:

The perturbation method can used to seek an optimum solution that uses the lowest amount of energy to achieve the design-specified damping ratio. In the example of this paper the design-specified damping ratio is 0.25. Because the model in the paper assumes that the damping is linear viscous damping, the energy used by controller [12] is “πcω_f X^2” per cycle of harmonic excitation of amplitude X . In this expression c is the design specified closed loop daming ratio.For any given amplitude the energy expended per cycle is proprtional to the closed loop frequency ω_f. Thus the lowest ω_f decides the optimal solution. Equations (15) to (18) determine the the solutions for perturbatiins δ from ω_f of set I, for coincident closed loop frequencies. Equations (15) to (18) contain only δ^2. Thus the solution is the same for both +δand –δ.

Now, we examine the solutiions for different values of δ, as presented in (19) to (23). The lowest value of ω_f cor responds to the case of coincident closed loop frequency.. Thus the case of coincident closed l;oop frequency represents the optimal slution when we do not consider uncertainties.

10. Application to a Simple Beam Finite Elkement Airframe Model

Now, a simple beam finite element model ifpr a helicopter is considered. This is a modified version of Hsu and Peters model and includes higher harmonic loads. This is the same model that is considered in the papoere by Hanagud and Babu. The element properties, helocopter characteristics and non-dtructural masses are the same as those reported in the paper by hanagud and Babu. The natural frequencies are as follows.

Figure 8 A four bladed Helicopter

Rotor Propereties

No. of Bldes	4
Radius	20.8 ft
Chord	1.56 ft
Mass/unit length slug/ft	0.283
Lift curvev slope	5.73
Lock Number	6.53
Rotor Speeed Rad/ sec	33.62
Firs t non rotating Frequency Rad/Sec	10.08
Solidity	).095

Fus elge Mass Stiffness

Elem. No.	Mass/ /Length Slugs/ft	Mass Mom. Moment /length (x100) Sl/ft	Bending Stiffnes $(x10^7)$ lb ft^2	Torsional stifffness $(x10^7)$ lb ft^2
1	1.57	8.38	1.15	0.86
2	5.76	20.94	2.88	2.16
3	5.76	20.94	2.88	2.16
4	12.56	38.74	5.33	4.00
5	12.56	38.74	5.33	4.00
6	3.14	15.70	2.16	1.62
7	0.52	9.42	1.30	0.97
8	0.52	5.23	0.72	0.
9	0.42	4.19	0.58	0.43
10	0.42	2.09	0.29	0.22

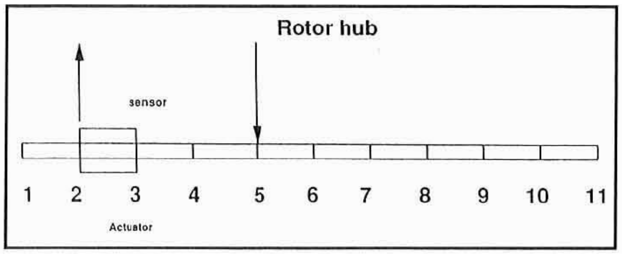

Figure 9 Finite Elent Idealization

The degrees of freedom at each of the node are vertical displacement, pitch angle and roll anmgle. The resulting airframe model has 33 degrees of freedom. The mass and stiffness of the elements are as follows. The natural frequencies are as follows.

Mode No.	Frequency (Hz	TYPE
4	8.33	1st Vertical Bending
5	16.59	2nd Vertical Bending
6	22.69	1st Torsion
7	32.76	3rd Vertical Bending
8	32.95	2nd Torsion

Without control, the modal daming is 2%. Using these results of frequencies , we can costruct the modal models for the structure and controllers, similar to equations (4) and (5), as follows.

$\ddot{\xi}_i + 2\zeta_{si}\omega_{si}\ \dot{\xi}_i + \omega_{si}^2\ \xi_i = (f_i(t)/m_i) + g_i\ \omega_{ci}\eta_i$; i=4 to 8

For each mode, a second order controller is of the following form:

$\ddot{\eta}_i + 2\zeta_{ci}\omega_{ci}\ \dot{\eta}_i + \omega_{ci}^2\ \eta = \omega_{si}^2\ \xi_i$; i= 4 to 8

It is sppecified that the closed loop damping should be 0.25. Then, we can use the previous results to find the controller parameters.

$(\omega_{fi}/\omega_{si}) = 1.0$

$(\omega_{ci}/\omega_{fi}) = 1.0$

$\zeta_{ci} = 0.48$

$g_i = (0.2116/\ ab\omega_{si}^2)$

For i= 4 to 8

On the basis of the experience of the control of buffet induced vibrations, it is not necessary to Have 5 controllers. If we build controllers for 2 or 3 dominant modes other modes will also be damped.

To illustrate the power of this method of increasing the closed loop damping ratio or constant, let us consider just the case of mode no. 4 vertical bending and discuss the case of harmonic inputs, as expected in helicopter airframes. After controller effect the system has become, in this case one degree of freedom system with damping ratio of o.25 and natural frequency corresponding to 8.33 Hz. Even if the input frequency is at the natural frequency (this is not the usual case), the input amplitudes of f/m (N/kg) of 1 is reduced 0.73 mm. If we are away from the resonant frequency the resulting amplitudes are much smaller. In practical cases, away from the natural frequency the input amplitudes are drastically reduced. This also avoids the requirement that the control signals should be carefully placed selected frequencies.

The sensors need not be piezoelectric sensors. Accelerometers can used and located at locations not maximum accelerations. The actuators consist of piezoceramic stacks with an offset from the controlled surface to amplify the control forces and moments. Because piezoceramic stacks are subject to failure in bending, we designed an actuator assembly that prevents such failures. The design is similar to our previous design, with modifications to mount on the helicopter. The design is shown in the following sketch

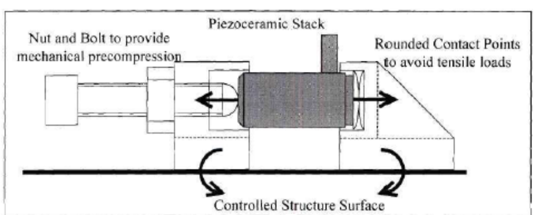

Figure 10 Offset Piezoceramic StackActuatot

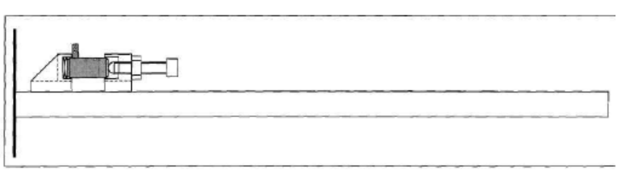

Figure 11 Offset Actuator on a Beam Structure

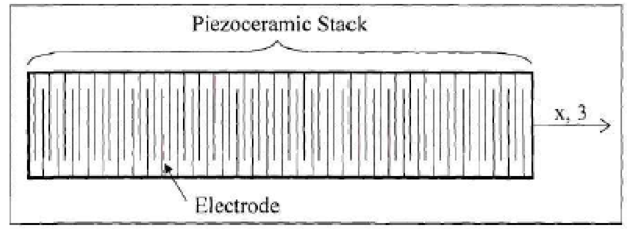

Figure 12 Sketch of a Piezoceramic Stack

11. Conclusions

In this paper, we have discussed the design of second order controllers to control the airframe vibrations of a helicopter. Specifically the controllers can be designed to achieve a specified closed loop damping ratio or a settling time. The controller design also yields the needed control authority or control effort to obtain the specified closed loop damping ratio. A design of an offset piezoceramic stack assembly based actuator assembly is also presented, to increase the control authority. The next level of study is to test the controllers design on a full scale helicopter.

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