Engineering Feasibility of Induced Strain Actuators for Rotor Blade Active Vibration Control

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ABSTRACT: Rotor blade vibration reduction based on Higher Harmonic Control—Individual Blade Control (HHC-IBC) principles is presented as a possible area of application of Induced Strain Actuation (ISA). Recent theoretical and experimental work on achieving HHC-IBC through conventional and ISA means is reviewed. Though the force-displacement and power-energy estimates vary significantly, some common-base values are identified. Hence, a bench-mark specification for a tentative HHC-IBC device based on the aerodynamic servom-flap principle operated through ISA means is developed. Values for the invariant quantities of energy, power and force-displacement product are identified, along with actual displacement and force values of practical interest. The implementation feasibility of this specification into an actual ISA device is then discussed. It is shown that direct actuation is not feasible due to the large required length of the ISA device, resulting in excessive compressibility effects (displacement loss and parasitic strain energy). Indirect actuation through a displacement amplifier was found to be more feasible, since it allows for matching of internal and external stiffness. A closed form formula was developed for finding the optimal amplification gain for each required value of the closed-loop amplification ratio. Preliminary studies based on force, stroke, energy and output power requirements show that available ISA stacks coupled with an optimally designed displacement amplifier might meet the bench-mark specifications.

INTRODUCTION

Rotor Blade Active Vibration Control through HHC and IBC

A traditional way of controlling the pitch feathering motion of helicopter rotor blades is the swash-plate mechanism. This low frequency device is designed for collective (quasi-steady) and cyclic (1/rev) control of the pitch motion \( \theta(t) = \beta_0 - a_1 \cos \Omega t - b_1 \sin \Omega t \), where \( \Omega \) is the rotor angular speed. Higher Harmonic Control (HHC) superposes an oscillatory modulation of frequency \( \omega \) on the basic pitch control of frequency \( \Omega \):

\[
\theta(t) \sin \omega t = (\beta_0 - a_1 \cos \Omega t - b_1 \sin \Omega t) \sin \omega t \\
= \beta_0 \sin \omega t \\
- \frac{a_1}{2} [\sin (\omega - \Omega)t + \sin (\omega + \Omega)t] \\
- \frac{b_1}{2} [\cos (\omega - \Omega)t + \cos (\omega + \Omega)t] \quad (1)
\]

Since the helicopter vibrations appear at multiples of rotor speed, HHC frequencies of the form \( \omega = N \Omega \) are usually used, thus resulting in pitch modulations not only at \( N \Omega \) but also at \( (N - 1)\Omega \), \( (N + 1)\Omega \). Extensive theoretical studies (Robinson and Friedmann, 1991; Nguyen and Chopra, 1990) and experimental studies (Straub and Byrns, 1986; Shaw, Albion, Hanker, and Teal, 1989) have shown that HHC can be an effective means of rotor blade vibration control, and reductions as large as 90% have been reported in the literature (Straub and Byrns, 1986; Ham, 1987; Kube, 1992). However, two major issues limit the usefulness of conventional HHC: (a) the swash plate is an inherently low frequency device and hence acts as a low-pass filter on the HHC input, and (b) HHC affects all the blades equally, while real rotor blades may differ noticeably in their detailed aerodynamic and inertial characteristics.

Individual Blade Control (IBC) denotes a method by which the blades of a rotor system have their pitch motion controlled individually. Thus, many additional issues may be addressed with this control concept such as: blade tracking, lift improvement through a 2/rev modulation. As a means of vibration control, the IBC implementation of the HHC concept can address the vibrations due to manufacturing variability between the blades, and also is not restricted by the \( N + 1, N - 1 \) phenomenon specific to swash plate control. An effective HHC-IBC device should also have a good frequency response characteristic, and thus produce the least possible filtering of the HHC signal.

Various engineering options are possible for implementing the HHC-IBC concept. One option is to transform the conventional push-rods existing between the swash-plate and the blade root into active devices, e.g., hydraulic actuators (Jacklin, Leyland, and Bliss, 1993). However, the value of the aerodynamic pitch moment is quite significant, and

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considerable control force is required. For example, Fenn, Downer, Bushko, Gondhalekar, and Ham (1993) estimated the aerodynamic pitch moment for root actuation in the range 450–780 Nm. Another, more efficient way of individual blade control is to utilize the principle of servo-aerodynamic, i.e., to take energy from the airstream and use it to modify the blade pitch. One application of this principle is the servo-flap concept. A limited span, trailing edge flap is used to produce aerodynamic pitching moments that in turn modify the pitch setting of the flexibly-restrained blade. The Kaman SH-2G Seasprite helicopter uses a mechanically operated servo-flap for primary (collective and cyclic) rotor blade control (Lemmios and Smith, 1972). However, the SH-2G servo-flaps are controlled through a traditional swash-plate mechanism and push rods inside the blade (Lambert, 1993), and are not designed for vibration control. A new approach to active control based on the servo-flap concept was experimentally investigated by Straub and Charles (1994). The data obtained during these recent experiments indicates that the servo-flap concept is potentially very effective in rotor blade vibration control. But the servo-flap controls used in these experiments were still based on conventional mechanisms (cams, push-rods, pulleys, and cables).

Significant improvement of HHHC-IBC rotor blade vibration control would be achieved with a device capable of remote actuation and good frequency response. The electrically-driven ISA devices offer opportunities in this direction, and a wide body of research is presently concentrated in this area.

**Induced Strain Actuation Principles**

In certain materials, electro-mechanical coupling occurs when an electric field is generated by an externally applied strain and, vice-versa, a strain is generated by an applied electric field. For the present study, we shall concentrate on the latter effect, i.e., induced strain actuation (ISA). The ISA materials can have either linear (e.g., piezoelectric) or quadratic (e.g., electrostrictive) response to an applied electric field. An associated behavior is displayed by the magnetostrictive materials where an induced strain is produced when a magnetic field is applied. The general constitutive equation for an ISA material is

\[ S_{ij} = s_{ijkl} T_{kl} + d_{ijkl} E_k + M_{ijkl} H_i H_j \quad (2) \]

where \( S_{ij} \) is the strain, \( T_{kl} \) the stress, \( E_k \) (or \( E_i \)) the electric field, \( s_{ijkl} \) the compliance, \( d_{ijkl} \) the linear ISA coefficients, and \( M_{ijkl} \) the quadratic ISA coefficients. For piezoelectric materials, the linear coefficients \( d_{ij} \) are dominant, whereas for an electrostrictive material, the quadratic terms \( M_{ijkl} \) are dominant. Magnetostrictive materials have a behavior similar to that of electrostrictive materials (i.e., dominantly quadratic), except that the magnetic field is the driving factor. Widely developed in the last decade, the ISA materials have shown remarkable performance, and positive induced strains in excess of 0.075% have been reported. The variation of induced strain with electric (or magnetic) field is presented in Figure 1 for several commercially available ISA materials (Galvagni, Rawal, 1991; Clark, 1992). Figure 1 compares the induced strain curves for PZT (a piezoelectric ceramic), PMN (an electrostrictive ceramic), and TERFENOL (a magnetostrictive alloy consisting of iron and two rare earth elements). The maximum strains available from these materials are about 0.075% for PZT, 0.075% for PMN, and 0.160% for TERFENOL. Higher values have also been occasionally reported (see, for example, Vornan, and Hardwick, 1993). Due to their quadratic characteristic, PMN and TERFENOL materials can only produce positive (i.e., expansion) strain, and hence compressive loads. The linear characteristic of the ISA material suggests a tension-compression capability, but the material tension strength is much less than under compression. Hence, it is reasonable to say that direct ISA applications are limited to compression loads generated by expansion strains against an external resistance. To achieve symmetric (expansion-retraction) operation, an electrically biased (offset) operation is utilized. In such cases, a pretension elastic spring must be employed to maintain the ISA ceramic under nonzero compressive load. Alternatively, a pair of counter-acting actuators can be used.

Though Equation (1) represents a fully coupled tensorial equation, many actuator applications use the ISA effect in one direction only (usually the \( x_i \) direction). As a practical example, consider the ISA stacks: thin washers of ISA material are intercalated with alternatively charged metallic electrodes. Thus, a high electric field can be applied throughout the ISA material ensuring good performance. Commercially available ISA stacks of typically 150 mm length display free ISA displacement \( (x_i) \) in excess of 0.100 mm. However, due to internal compressibility, significant

![Figure 1. Induced strain vs. applied field for several ISA materials (Galvagni and Rawal, 1991; Clark, 1992).](image-url)
displacement loss occurs when the ISA stack operates under external load. The internal compressibility loss can be calculated as \( x_i = F/k_i \), where \( k_i \) is the internal stiffness, and \( F \) is the applied load (quasi-static conditions are assumed). Thus, only a portion of the ISA displacement can be delivered externally, i.e., \( x_e = x_a - x_i \). The higher the external load, the lower the deliverable displacement. In the fully constrained case, the actuator is blocked (stalled) and all the ISA displacement is consumed internally. Since work is the product of force and displacement, it is apparent that the work done externally by an ISA actuator is zero both at zero load and at stall load. Hence, the externally delivered work is not a monotonical function. In fact, the variation of the externally delivered work is bell-shaped, with the maximum value attained when the internal and external components of the ISA displacement are equal. A similar conclusion is reached by reasoning in terms of energy. Assume a spring-like external reaction \( k_e \), and write the internal and external strain energies, and the corresponding ISA energy, as:

\[
E_{\text{int}} = \frac{1}{2} k_i x_i^2, \quad E_{\text{ext}} = \frac{1}{2} k_e x_e^2, \quad E_{\text{ISA}} = E_{\text{int}} + E_{\text{ext}} = \frac{1}{2} k_i x_i^2 + \frac{1}{2} k_e x_e^2
\]

(3)

For a given actuator of known \( x_a \) and \( k_i \), the externally deliverable strain energy varies in terms of the ratio \( r = k_e/k_i \), i.e.,

\[
E_e(k_e) = \frac{1}{2} k_i x_i^2 \left( 1 + r \right)^2, \quad r = \frac{k_e}{k_i}
\]

(4)

Liang, Sun, and Rogers (1992, 1993, 1994) performed a comprehensive analysis of energy and power aspects using an electro-mechanically coupled model for transverse excitation of plates. Using similar principles, we plotted \( E_i \) and \( E_e \) against the external stiffness parameter \( k_e \) for a stacked actuator (Figure 2). It can be seen that while the parasitic internal energy steadily increases, the externally delivered energy reaches a maximum, after which it starts to decrease. Hence, an optimum condition exists (\( k_e = k_i \)) for which the delivered energy, and hence, the energy per unit volume, and per unit weight, are maximum. A reasonable approximation for the energy density per unit volume at optimum (stiffness match) condition is:

\[
(E_e)_{\text{opt}} = (E_i)_{\text{opt}} = \frac{1}{2} T_{33} S_{33} = \frac{1}{2} \left( \frac{1}{2} S_{333} \right)^2
\]

(5)

Typical values, \( S_{333} = 20 \times 10^{-12} \text{ m}^2/\text{N}, S_{33} = 0.077\% \), yield a maximum deliverable specific energy per unit volume \( (E_e)_{\text{opt}} = 1/2 \times (20 \times 10^{-12})^{-1} \times (0.077 \times 10^{-2}/2)^2 \)

\( = 3.7 \text{ kJ/m}^3 \). Using a PZT material density of \( \rho_{\text{PZT}} = 7.5 \times 10^3 \text{ kg/m}^3 \), yields the corresponding energy density per unit mass (or weight) as \( (E_e)_{\text{max}} = 3.7 \times 10^4/(7.5 \times 10^3) = 0.5 \text{ J/kg} \). Beyond the optimum point the output energy begins to decrease, though the ISA energy, and the energy consumption, keep increasing. Thus, no benefit is gained from increasing the external stiffness beyond the stiffness match point.

When actual ISA devices are manufactured, additional weight and volume is required for casing, preload spring, electrical wiring, etc. Hence, actual energy densities may be lower than the theoretical values derived above. For example, consider the commercially available actuator P-247.70 produced by Polytec PI, Inc. (1994). Its mass is \( m_a = 1 \text{ kg} \), and the volume, \( V_a = 0.181 \times 10^{-3} \text{ m}^3 \) \( (L_a = 144 \text{ mm}, D = 40 \text{ mm}) \). The maximum ISA displacement is \( x_a = 0.120 \text{ mm} \), the internal stiffness, \( k_i = 370 \text{ kN/mm} \), and the maximum design load, \( F_a = 30 \text{ kN} \). The above formula yields the maximum deliverable energy as \( E_e(k_e = k_i) = 0.666 \text{ J} \), and the specific energies per unit volume and unit weight as 3.68 kJ/m^3 and 0.666 J/kg, respectively. These values are not unlike those of other high performance ISA devices available today.

Addressing the issue of energy transmission efficiency, we see that only part of the input energy \( (E_{\text{ISA}}) \) reaches the actuator output. A significant part of the input energy remains stored as parasitic strain energy \( (E_i) \) into the actuator compressibility. Hence, the energy transfer efficiency is:

\[
\eta_{\text{energy}} = \frac{E_{\text{output}}}{E_{\text{input}}} = \frac{E_{\text{ext}}}{E_{\text{int}} + E_{\text{ext}}} = \frac{1}{1 + k_e/k_i} = \frac{1}{1 + r}
\]

(6)

This proves that the optimum energy delivery does not correspond to an optimum energy-transfer efficiency. In fact,
Equation (6) shows that the best transfer efficiency ($\eta_{\text{energy}} \to 1$) is reached at vanishing external stiffness ($k_e \to 0$) since, in this case, the compressibility loss vanishes too. However, for vanishing external stiffness, the externally delivered energy vanishes too. With increasing external stiffness, the externally delivered energy, and the parasitic, internally-stored, strain energy increase as well. Hence, the transfer efficiency deteriorates as the externally delivered energy increases. For $k_e = k_i$, the externally delivered energy peaks. However, the corresponding transfer efficiency has dropped to 50% (half of the ISA energy is transmitted externally, and half is stored inside the actuator). Further increase of the external stiffness $k_e$ leads to deterioration of both the energy delivery and the transmission efficiency.

The design optimization of an actuator-application pair seeks to make best use of the actuator properties, and to maximize the energy delivered into the application. If the stiffness ratio can be freely and continuously varied, the maximum is achieved when the external and internal stiffness are equal (stiffness match). This corresponds to an unconstrained optimization study. However, practical applications do not allow for free and continuous variation of the stiffness ratio, but rather must deal with a discrete range of actuator stiffness, and with given values of application stiffness. Hence, unconstrained optimization is not usually possible, and practical design studies are constrained optimization problems. A maximal delivered energy, and the best transmission efficiency are sought within the permissible stiffness range. Figure 2 shows that the same delivered energy can be obtained by making the external stiffness, say, twice the internal stiffness, or half the internal stiffness. However, the transmission efficiency is 33%, in the former case, and 67%, in the latter. Of two feasible designs with the same energy delivery, that with a better energy transmission efficiency would be considered optimal. Hence, we would choose the latter ("softer") case as the optimal solution. Thus, two simultaneous aspects, a "soft" external stiffness, and a stiffness ratio close to the ideal match, have to be simultaneously satisfied for an optimal design.

The Induced Twist Concept

Direct dynamic control of bending and twisting through ISA principles has captured the interest and attention of several researchers due to its conceptual simplicity and the potential for modal control. Initial theoretical studies of ISA adaptive aeroelastic control were done on classical wings (Lazarus, Ceawley, and Bohllmann, 1990; Ehlers and Weisshaar, 1990; Song, Librescu, and Rogers, 1991; Librescu, Rogers, and Song, 1992; Song, Librescu, and Rogers, 1993; Song and Librescu, 1993). Modeling and simulation of rotor blade control through ISA twist were conducted at DLR in Germany (Nitzsche and Breitbach, 1992a and 1992b, Nitzsche, 1993 and 1994). These studies showed the benefits of spanwise twist distribution over discrete control surface movement, especially when wide-band reduction of rotor systems vibration is sought. However, numerical simulations with existing PZT technology have shown that only the vibration modes with very low aerodynamic damping will benefit. Hence, it was concluded that the practical implementation of the method relies on "future development of a new generation of PZT materials having higher strain/volt capabilities" (Nitzsche, 1993). A 1/8 Froude scale composite rotor was constructed at the University of Maryland (Chen and Chopra, 1993a and 1993b). Diagonally oriented PZT crystals embedded at 45° in the fiberglass skin (Figure 3) were used to produce induced twist when an electric field was applied. A value of 1° tip rotation was targeted in these trials. Extensive theoretical and experimental studies were conducted (Chen and Chopra, 1993a and 1993b; Park, Walz, and Chopra, 1993; Walz and Chopra, 1994). Later results (Chen and Chopra, 1993b; Walz and Chopra, 1994) reported good prediction of tip twist and bending slopes (Figure 4). Largest recorded values, in agreement with predicted results, did not exceed 0.35°. Dynamic tests were also performed in nonrotating, and then rotating conditions. Significant twist response was measured when excitation was close to resonant bending or torsion frequencies (50 Hz and 95 Hz, respectively). Maximum tip twist values at these frequencies were 0.35° and 1.1°, respectively. However, the response was very small at nonresonance frequencies. These experimental results confirm Nitzsche's (1993) theoretical conclusion that the practical implementation of induced

REVIEW OF EXISTING PROPOSALS FOR INDIVIDUAL BLADE CONTROL

Several proposals have recently been developed for achieving individual blade control (IBC) through conventional and unconventional means. These proposals range from the direct distributed control of twist along the blade span, to the use of aerodynamic control surfaces discretely placed at the blade outer stations. Application of ISA technology in rotary wing IBC follows similar developments taking place in the ISA control of conventional wings (Lazarus, Ceawley, and Bohllmann, 1990; Song, Librescu, and Rogers, 1991), and a significant degree of technological cross-fertilization is taking place.

![Figure 3. Diagonal PZT twist actuation concept (Chen and Chopra, 1993a and 1993b).](image-url)
twist actuation through embedded PZT technology is difficult.

The Servo-Flap Concept

The servo-flap concept has received considerably more attention due to the inherent advantages of extracting additional power and energy from the airstream through the servo-aerodynamic effect. This concept was first pioneered by Kaman Aerospace Corp. (Lemnios and Smith, 1972), with SH-2G Seasprite and K-Max helicopters as flying examples of its feasibility. However, these two helicopters still employ a conventional swashplate to achieve control of the servo-flap through levers and control rods running through the blade. The purpose of recent studies was to replace these mechanical controls with other means (electrical, electro-hydraulic, etc.), as shown in the next section.

ADVANCED ROTOR CONTROL SYSTEM (ARCS) STUDIES

Conventional electrical motors and electro-hydraulic actuators were considered in an extensive industry study of the Advanced Rotor Control System (ARCS) to be implemented on existing helicopters such as McDonnell Douglas Helicopter Co. (MDHC) AH-64A Apache (Straub and Charles, 1990), and Bell Helicopter Textron Inc. (BHTI) AH-1W Super Cobra (Phillips and Murphy, 1990) as shown in Figure 5. Extensive studies ranging from aerodynamic predictions, to detail engineering design, were performed. The results showed consistently that if the servo-flap concept were to be used for achieving both basic flying controls (collective and cyclic pitch) as well as HHC-IBC, then extensive flap travel and hinge moment capabilities would be required. The MDHC study (Straub and Charles, 1990) used previous experience (Straub and Byrns, 1986) and the CAMRAD-JA prediction code to evaluate a 42.5% chord, 17.5% span (0.65R - 0.825R) flap design extending 32.5% beyond the trailing edge of the blade. The peak deflection and hinge moment values resulting from this study were placed around ±19.2°, and 2445 lb-in, with an associated electrical power of 15 kW per blade. The BHTI study (Phillips and Murphy, 1990), used the COPTER aerodynamic code to evaluate a servo-flap design of slightly different dimensions: 20% chord, 20% span centered about the 0.70 spanwise blade station. Peak values of ±18.9°, and 1200 lb-in, with 1.95 kw per blade were predicted. The nearly 8 times difference between the two predictions (Straub and Charles, 1990; Phillips and Murphy, 1990) points out the sensitivity of these analyses on subjective variables such as: modeling methods, engineering solutions, and design selections. However, both studies pointed out that trying to achieve simultaneously primary control (collective and cyclic) and vibration control by using the same actuation system does not lead to an efficient design. Hence, the alternative of separating primary control from vibration control was considered. Conventional root actuation was used to achieve either collective, or both collective and cyclic controls. Thus, the requirements imposed on the servo-flap system were brought down to more feasible values. Typical HHC-IBC values were estimated at: ±0.90°, with 512 lb-in at 24.1 Hz for MDHC flap (35% chord, 15% span), requiring 512 W per blade; and ±3.8°, with 320 lb-in at 20 Hz for BHTI flap (20% chord, 20% span), requiring 388 W per blade.

ISA SERVO-FLAP STUDIES

Theoretical Studies at UCLA

Extensive theoretical studies performed at UCLA (Millott and Friedmann, 1992, 1993, and 1994) highlighted the benefits of using aerodynamic servo-flap concepts for active control of helicopter rotor blades. Both spring restrained rigid blade (Millott and Friedmann, 1992) and fully elastic blade (Millott and Friedmann, 1993) models were used. Geometrical nonlinearities were included, and advanced

![Figure 4. Twist and Bending Slopes vs PZT Angle (Chen and Chopra, 1993b).](image-url)

![Figure 5. Advanced Rotor Control System (ARCS) Servo-Flap Actuation Concepts: (a) Electric Motor (Straub and Charles, 1990); (b) Hydraulic Cylinder (Phillips and Murphy, 1990).](image-url)
unsteady aerodynamic 2-D models were employed, together with a flap efficiency coefficient of 60% based on experimental insight. A vibration reduction controller was connected with the aeroelastic model. HHC input to the servo-flap was assumed. Substantial vibration reductions were demonstrated at various helicopter airspeeds corresponding to advance ratios in the range \( \mu = 0 - 0.4 \). The required flap travel and hinge moment were calculated, though only the former was displayed. The average power consumption was computed as:

\[
P_{\text{avg}} = \sum_{k=1}^{N_s} \frac{1}{2\pi} \int_{0}^{2\pi} M_n(\psi_k) \frac{d}{dt} \delta(\psi_k) d\psi_k
\]

where \( P_{\text{avg}} \) signifies control surface power, \( N_s \) is the number of blades, \( M_n(\psi) \) is the control surface hinge moment, \( \delta(\psi) \) is the control surface deflection, and \( \psi_k \) is the azimuthal position of the \( k \)th blade. Similar calculations were performed for conventional IBC blade root actuation. Comparisons of the servo-flap and the conventional HHC-IBC actuation methods were performed. Though the required servo-flap deflection was found to be significantly larger than the equivalent blade-root deflection, substantially less power was required. The numerical values required to achieve satisfactory HHC-IBC varied with the rotational and torsional stiffness of the blade. The softer (lower frequency) blades were easier to control through the servo-flap system. In this case (\( \omega_r = 3/\text{rev} \)), it was found (Miliott and Friedmann, 1993) that the peak deflections of the flap could be \( +3^\circ \) to \( -4^\circ \). No values are given for the corresponding hinge moment. The corresponding power requirement was found to be around 0.25% of the rotor power (Miliott and Friedmann, 1993). The values of flap deflection and power requirements for the torsionally stiffer blade (\( \omega_r = 5/\text{rev} \)) were considerably higher.

**Theoretical and Experimental Studies at MIT**

Extensive theoretical and experimental studies of ISA controlled flaps were performed at MIT. In one investigation (Spangler and Hall, 1989 and 1990), bimorph arrangements of PZT material were used to produce a bending displacement that was transformed into flap rotation through a hinge-and-lever mechanism (Figure 6). The studies were targeted at the Boeing CH-47D tandem helicopter having a large 3-blade rotor (\( R = 30 \text{ ft} \)) rotating at 225 rpm (3.75 Hz). HHC inputs at 3/rev (1.25 Hz) were considered in the theoretical study, and extensive scaling and modeling work was performed. A 1/5 scale wind tunnel test was performed using a stationary blade section equipped with a 10% chord flap. The 1/10 velocity scale and 1/2 frequency scale were employed in the model design. Experiments were conducted at various airspeeds between zero and \( 78 \text{ ft/sec} \), and at frequencies up to 100 Hz. The flap deflection capabilities, as well as the resulting lift and pitch moment coefficients created by this deflection, were recorded and discussed. As a general trend, values significantly below the theoretical predictions were reported. This discrepancy was attributed to certain inconsistencies in Reynolds number (and hence boundary layer thickness), as well as to high mechanical losses due to friction in the hinges, the low stiffness of the trailing edge flap, and the spanwise bending response of the model. Currently, an improved experimental program is underway (Hall, 1993), using more realistic speed values, improved hinge design (solid state flexural hinges) and an updated PZT actuator. Higher frequencies, closer Mach and Reynolds number similarity, better boundary layer effects, less hinge damping loss, and improved actuation moment are expected (Hall, 1993).

Another study was independently conducted by SatCon Technology Co. in cooperation with MIT (Fenn, Downer, Bushko, Gondhalekar, and Ham, 1993). The design of a TERFENOL-D driven, servo-flap for Sikorsky YH-60A Black Hawk helicopter rotor blade was considered. Analytical predictions of flap angle, hinge moments and control power requirements, as well as ISA actuator sizing based on the peak energy needs were reported. A required \( \pm 2^\circ \) of conventional root motion was considered sufficient for HHC-IBC purposes. This value was transformed into equivalent flap deflection using the span and chord ratios and the spanwise location of the flap. It was predicted that for a full span flap the required deflection varies from \( \pm 5^\circ \) for a 10% chord, to \( \pm 3^\circ \) for a 40% chord. Adjustment for partial span of the flap was done using the formula

\[
\delta_{ps} = \frac{1}{\epsilon_{out} - \epsilon_{in}} \delta_f
\]

where \( \delta_{ps} \) and \( \delta_f \) signify “partial span”, and “full span”, and \( \epsilon_{in}, \epsilon_{out} \) are the percentage spanwise positions of the inner and outer flap ends, respectively. A 17.5% chord, 46% span flap placed between the 0.52R and 0.98R spanwise stations was selected. A flap travel of \( \pm 5^\circ \), and the corresponding aerodynamic hinge moment of \( \pm 30 \text{ ft-lb} \) were estimated. An additional constant moment contribution of 11 ft-lb was added to account for the steady state maneuver loads. The peak energy required for achieving these deflection and moment values was also calculated. It was found that 3.5 J are required for the positive part, and 2.0 J for the negative part of the oscillatory cycle. A 50% margin of error was taken in the final design. Six TERFENOL-D actuators per
blade were considered, requiring an estimated 7.2 kW (i.e., 1.8 kW per blade) of electrical power. The total actuators weight was estimated at 43 kg (i.e., 10.75 kg per blade). These values were considered to be well within the accepted power and weight budgets of the project, i.e., 1% of helicopter power and weight (11.2 W, and 81 kg, respectively).

**Experimental and Theoretical Studies at University of Maryland**

Experimental and theoretical studies with the bimorph ISA principle for rotor blade flap actuation (Figure 7) were conducted at the Center for Rotorcraft Education and Research at the University of Maryland (Samak and Chopra, 1993; Walz and Chopra, 1993 and 1994). Trailing edge flaps of 20% chord, 12% span (0.85R–0.97R) were built into a 36 in. radius, 3 in. chord composite blade model. Initial tests were performed with the stationary blade placed in a conventional wind tunnel operated at speeds up to 111 ft/sec. Two airfoil set angles (4° to 8°) were used. Excitation up to 15 Hz showed good frequency response. However, a very significant decrease in amplitude with airspeed was observed. Further experiments were conducted with the blade installed in a rotating rig operated at up to 900 rpm (15 Hz). Excitation frequencies of 1, 2, 3, and 4/rev were investigated. The trailing edge flap response was found to be rather constant with frequency, but strongly decreasing with rpm. The blade flapping response varied with both the excitation frequency and the rotor speed. This behavior is consistent with the known dependence between flapping frequency and rotor rpm. The decrease in trailing edge flap response with airspeed was further investigated using 2-layer and 4-layer bimorph PZT designs (Samak and Chopra, 1993; Walz and Chopra, 1993). The use of a 2-layer actuator generated an increase of about 21% in the flap angle at the higher speed value. The 4-layer actuator "showed much improved force capability, but lower displacement" (Walz and Chopra, 1993).

The goal of these investigations was initially set at achieving 2° of flap deflection at 258 ft/sec airspeed (Samak and Chopra, 1993). Subsequently, the goal was redefined to achieving "5% flap authority (additional steady lift due to a flap deflection divided by total steady lift with zero flap deflection) at 8° collective blade pitch". A theoretical parameter study was performed (Walz and Chopra, 1993 and 1994) to determine the required linkage arm length that will meet this specification (Figure 7). At present, a second experimental rotor with a larger span and multi-layered actuators has been built and tested on the hover stand for a range of rotor speeds and collective settings.

**MAJOR REQUIREMENTS FOR ROTOR BLADE PRIMARY AND VIBRATION CONTROL**

We have reviewed in the previous section various literature reports on the study and experimentation of Individual Blade Control (IBC) through conventional (electromechanical, hydraulic) and solid state (electro-mechanical ISA) actuation. Two major directions have been explored: the direct twist of the rotor blade through tension-torsion-

![Diagram](attachment:image.png)

Figure 7. Parameter study of Bimorph Effectiveness: (a) Device Schematic; (b) Linkage Arm and Actuator Capability Plots (Walz and Chopra, 1993).
bending coupling, and indirect modification of the aerodynamic lift and pitch moment of the blade using the servo-flap and servo-tab principles. The former concept is more direct and also permits modal control, since it can achieve continuous variation of the controlled parameter along the blade. But the experimental results reported so far show tip twist values not larger than 0.35°. The latter concept has some inherent shortcomings such as bringing an additional complication to the rotor blade design, and of using a rather rigid element (either flap or tab) with only a limited spanwise extent. However, the aerodynamics amplification properties of the servo-flap concept make this latter concept very attractive for achieving palpable results with the existing ISA technology. This explains why most practical engineering efforts found in recent literature have been directed towards the servo-flap concept. Hence, we shall concentrate our attention on the servo-flap concept, while leaving, for the time being, the induced twist concept in nole contendere.

On the other hand, the deflection, hinge moment, and frequent requirements for rotor blade primary control (collective and cyclic pitch) are substantially different from those for HHC-IBC. The collective pitch control requires very large displacement authority (± 18°), while the cyclic pitch control requires both a sizable displacement (± 19°) and a significant frequency (1/rev, i.e., around 5 Hz). The HHC-IBC operation requires much smaller displacement (say, ± 2°), but a considerably higher frequency. Even if we fully utilize the benefits of the servo-flap principle, it is apparent that these conflicting requirements cannot be achieved with the same ISA technology. Hence, we shall restrict our attention to vibration control, i.e., HHC-IBC applications.

Basic Requirements for Rotor Blade HHC: Force, Displacement and Frequency

It is generally accepted (Fenn, Downer, Bushko, Gondhalekar, and Ham, 1993; Straub and Charles, 1990; Samak and Chopra, 1993) that at least ± 2° of oscillatory blade pitch at (4 ± 1)/rev frequencies (approximately 25 ± 5 Hz) is required to achieve significant vibration reduction via HHC-IBC. When using servo-flap actuation, this value must be transformed into an equivalent servo-flap amplitude. Several issues must be addressed: on one hand, different flap chord to blade chord ratios lead to different required flap angles (e.g., ± 5° for a 10% chord flap, and ± 3° for a 40% chord flap (Fenn, Downer, Bushko, Gondhalekar, and Ham, 1993). These values are further modified when the effect of limited flap span is considered, and formula (2.2) applies. Fenn, Downer, Bushko, Gondhalekar, and Ham, (1993) estimated that a 17.5% chord flap spanning between 52% and 98% radial stations will require at ± 5.7° travel. On the other hand, the aerodynamic mechanism by which an outer span flap modifies the vibratory lift and pitch moment values during vibration control is significantly different than that of a blade-root pitch actuation. Theoretical studies performed by Millott and Friedmann, (1992, 1993, and 1994) with a 25% chord flap spanning between 0.65R and 0.85R estimated that for an elastic blade with low pitch-torsion stiffness (first rotating pitch-torsion frequency around 2.5/rev), the required flap travel would be + 3/ − 4°. The ARCS studies (Straub and Charles, 1990; Phillips and Murphy, 1990) predicted the required flap displacement for HHC control varying from ± 0.9° hrough ± 1.2°, ± 1.8° and ± 3.8°. In view of this large variability, we retained for further analysis an average value of ± 2°.

The hinge moments associated with the flap deflection have inertial and aerodynamic causes. Present day composite technology offers the possibility for very light flap construction. Therefore, the inertial forces are much less than the aerodynamic forces and can be discarded in predesign studies (Fenn, Downer, Bushko, Gondhalekar, and Ham, 1993). The aerodynamic hinge moment can be estimated by various methods ranging in complexity from steady or quasi-steady aerodynamics (Fenn, Downer, Bushko, Gondhalekar, and Ham, 1993), to completely unsteady aerodynamics (Spangler and Hall, 1989). A further refinement is to include Mach number effects and the axial variation of airstream velocity (Millott and Friedmann, 1992, 1993, and 1994). However, for a preliminary assessment study, some bounds on the maximum hinge moment will suffice. Comparing the available data, it was considered that a credible value would be ± 75 Nm (55 ft-lb). This value exceeds the estimates presented in the ARCS studies (Straub and Charles, 1990; Phillips and Murphy, 1990). Thus, a baseline requirement for HHC-IBC servo-flap could be: 1/30 rad (≈ ± 2°), ± 75 Nm, at 25–30 Hz, per blade.

Invariant Requirements for Rotor Blade HHC-IBC: Power and Energy

ENERGY REQUIREMENTS

The complex aerodynamic phenomena taking place during HHC-IBC operation generate conservative and dissipative reactions. In a first order analysis, the assumption of linear variation of external reaction with flap displacement can be made (F_e = k_e x_e). The equivalent external stiffness k_e is a complex quantity that varies with frequency. Its real part signifies conservative behavior, while its imaginary part is associated with dissipative phenomena. For estimating the maximum energy requirement, we assume the conservative part as dominant. Hence, the energy of interest is identified as the maximum over an oscillation cycle of the externally stored energy E_e(t) = (1/2)k_e x_e^2(t). For harmonic operation, the maximum value is given by replacing the time-varying functions with the oscillation amplitudes, i.e., E_e = (1/2)k_e x_e^2. (If a bias steady force is included, then the maximum stored energy would be augmented with the product between this steady force and the oscillation amplitude.) For the baseline requirement defined at the end of the previous section, we get the maximum stored energy as:
Energy = \frac{1}{2} \text{Force} \times \text{Displacement}

hence

\[ E_\varphi = 75 \text{ Nm} \times \frac{1}{30} \text{ rad} = 1.25 \text{ J} \]

per blade. This is a first estimate of the peak energy output to be supplied by the ISA device during a cycle of oscillation.

**POWER REQUIREMENTS**

The power aspects are more complex, and both the instantaneous power and the power dissipation have to be addressed. First, consider the maximum instantaneous power that has to be supplied during the harmonic operation. This will directly determine the power rating of the actuator. To calculate the instantaneous mechanical power, take the product of generalized force and velocity (i.e., time derivative of displacement) \( P(t) = F(t) \cdot v(t) = \omega \cdot F(t) \cdot \dot{x}(t) \), and hence find its maximum value around a cycle of oscillation \( P_{\text{max}} = \max P(t), 0 < t < T, T = 2\pi/\omega \). For small phase angles between force and displacement phasors, one can approximate the maximum instantaneous power by the half product between force and displacement amplitudes, and the circular frequency, i.e., \( P_{\text{max}} = \pi \times 30 \times (1/30) \times 75 = 471 \approx 235 \text{ W} \). (If a bias steady force were present, then this steady force value would be added to the oscillatory force amplitude.) Taking a large reserve factor, a maximum instantaneous power value of 500 W per blade, i.e., 2 kW for a 4-blade helicopter is retained.

Secondly, one has to address the power consumption. This is associated with the power dissipation during the servo-flap operation, and is defined as the average power over a cycle of oscillatory operation, i.e.,

\[ P_{av} = \frac{1}{T} \int_{0}^{T} P(t) dt \]

where \( T = 2\pi/\omega \) is the period of oscillation. In the present application, three sources of power dissipation can be identified:

(a) "Useful power, viz. the power dissipated in the aerodynamic stream through the out-of-phase aerodynamic forces (aerodynamic damping), and the power dissipated during the operation of the mechanical device (e.g., friction in hinges and linkages for a servo-flap, or internal structural damping during the induced strain twisting of the blade). This power is called "useful" or "output" power since it is used downstream of the actuating device. This power component is denoted here by \( P_{us} \). For a HHC-IBC servo-flap, the studies of Millott, and Friedmann, (1992, 1993, and 1994) identified values for this type of power consumption around 0.25% of helicopter rotor power.

(b) Internal power loss in the actuator due to material hysteresis and parasitic electric resistance. This power loss is reported as a "\( \eta \)" factor that usually does not exceed 5–8% (Polytech PI, 1994).

(c) Power loss in the supply system, i.e., in the electrical source and in the connecting wires. For a servo-flap actuated by conventional hydraulics, the estimated power loss in the hydraulic system alone is of the order of 12.5% (Phillips and Murphy, 1990). For electrical ISA devices, this power loss can be larger, and its minimization requires specialized engineering design. Considering that the power loss in the supply system depends on the instantaneous power circulating back and forth, careful engineering of the ISA electronics can be used to minimize the reactive instantaneous power component. In principle, this is achieved by compensating the capacitive reactance of the ISA actuator with an inductive reactance tuned to the frequency of interest (Hagood, Chung, and Flotow, 1990).

Hence, the total power consumption, \( P_{\text{tot}} \), is defined as the sum of the useful, externally dissipated power, \( P_{\text{us}} \), and the parasitic power loss \( P_{\text{loss}} \), dissipated internally and in the energizing system. Thus, \( P_{\text{tot}} = P_{\text{us}} + P_{\text{loss}} \). For engineering design, upper bounds must be placed on the total power consumption. A reasonable assumption is that power consumption budget for HHC-IBC implementation should not exceed 1% of the helicopter cruise power. Penn, Downer, Bushko, Gondhalekar, and Ham, 1993, estimated this value for Sikorsky UH-60A Black Hawk helicopter at 11.2 kW, i.e., 0.4% of powerplant power. On similar arguments, and using data from Lambert (1993), we estimate 11.2 kW for MDHC AH-64A Apache, and 10.3 kW for BHTI AH-1W Super Cobra.

**Weight and Specific Power Requirements**

As with any airborne equipment, weight minimization should be a major consideration for the engineering design of an ISA HHC-IBC system. A reasonable assumption is to place the weight budget for HHC-IBC implementation at 1% of the gross aircraft weight. Penn, Downer, Bushko, Gondhalekar, and Ham (1993) estimated this value for Sikorsky UH-60A Black Hawk helicopter at around 81 kg, i.e., 1.59% of empty aircraft weight. On similar arguments, and using data from Lambert (1993), was estimated 81 kg for MDHC AH-64A Apache, and 74 kg for BHTI AH-1W Super Cobra. Hence, one concludes that the weight budget for HHC-IBC implementation depends on the specific helicopter under consideration, and has values in the range 75–85 kg. For comparison, we mention that the HHC-IBC system based on conventional actuation, and presented by Straub and Charles (1990), in the ARCS study, was estimated to produce an additional weight of 28.9 kg per blade, i.e., a total of 115.6 kg per aircraft.
Regarding the specific power, two separate issues must be considered. First issue is that of specific power delivery, i.e., the power output per unit weight of the complete actuating system including power supply. Dividing the power delivery requirements by the weight range of the ISA system yields the acceptable values for this parameter. It is found that the specific power delivery should be around (2 kW)/(80 kg) = 25 W/kg. The second issue is that of power consumption per unit weight. This not-to-be-exceeded budget is obtained by dividing the power budget by the weight budget. Values around (11.2 kW)/(81 kg) = 140 W/kg are obtained.

ENGINEERING ASSESSMENT OF HHC-IBC VIBRATION CONTROL WITH ISA DEVICES

A baseline requirement for HHC-IBC vibration control is considered based on the previously discussed literature review. The existing state of the art in ISA materials, makes us opt for the use of a servo-aerodynamic device (e.g., a servo-flap, Figure 8) which requires less energy and power from the ISA device. Baseline design requirements are defined in terms of the generalized force-displacement pair (angular deflection and angular moment amplitudes) of the servo-flap control surface at a typical HHC frequency (4–5/rev), i.e.,

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Angular deflection</td>
<td>± 1/30 rad (≈ ± 2°)</td>
</tr>
<tr>
<td>Hinge moment</td>
<td>± 75 Nm, simultaneously with the deflection</td>
</tr>
<tr>
<td>Frequency</td>
<td>25–30 Hz (157–188 rad/sec)</td>
</tr>
</tbody>
</table>

The external stiffness $k$ is derived from these baseline requirements, and is defined as the ratio of the force amplitude to the displacement amplitude. Invariant requirements are also identified as, per blade:

- Maximum energy delivered per cycle of oscillation: 1.25 J
- Maximum instantaneous mechanical power: 500 W
- The force-displacement product: 2.5 J

Several designs are possible for implementing these specifications (Table 1). Depending on the hinge arm length (Figure 8), the required actuator output can vary between ±0.25 mm (0.01 in) for an arm of 7.5 mm (0.3 in) to ±5 mm (0.2 in) for an arm of 150 mm. On one hand, a low value of the required displacement, say ±0.25 mm, would be more easily met by the existing ISA actuators. Table 1 shows that the associate hinge arm would have to be 7.5 mm. The mechanical design required for utilizing such small displacement and hinge arm values does not seem feasible. On the other hand, a displacement of ±5 mm would be more easily utilized mechanically, but the corresponding hinge arm of 150 mm is rather large and would have to operate outside the blade cross section. This is not impossible [see, for example, the MDHC ARCS study of Straub and Charles (1990)], but should be avoided. Thus, a trade-off displacement value of ±1 mm, requiring an arm of 30 mm (1.2 in) seems a good compromise. This trade-off of ±1 mm value will be used as a bench-mark for comparing various ISA design options. Such options fall into two large categories: (1) direct actuation using long ISA stacks, and (b) indirect actuation using shorter stacks with some means of displacement amplifiers. Note, however, that as the hinge arm is varied, the baseline moment and angular deflection requirements get transformed into different force and stroke amplitudes (Table 1). But the invariant requirements (the force-displacement product, and the maximum energy delivered per cycle) are conserved.

Direct Action ISA Devices

As direct action ISA devices, we consider ISA material stacks acting directly on the hinge arm of the aerodynamic servo-flap (Figure 8). A value of 30 mm (1.2 in) for the servo-flap hinge arm allows for installation inside the blade section and hence seems to be a feasible design option (see, for example the studies of Fenn, Downer, Bushko, Gondhalekar, and Ham (1993); Straub and Charles (1990); and Phillips and Murphy (1990). Hence, we require an ISA stack able to deliver an oscillatory amplitude of ±1 mm simultaneous with an oscillatory load of ±2.5 kN, at frequencies up to 30 Hz (Design case #2 in Table 1). We tried to meet this specification with actuators available from two manufacturers (EDO Corporation, 1993; Polytech PI, 1994). The results are given in Table 2. As shown, excessive stack sizes would be required. This confirms our initial estimate that using a direct action ISA actuator would result in a very long and excessively heavy design, which is unsuitable for flight installation. (For example, to achieve, say, ±1 mm displacement, one would require an ISA stack of 1.5–2.6 m length). Hence, we can conclude that direct ISA action is not a practical way to achieve operation of the servo-flap device.

![Figure 8. ISA Actuation of Aerodynamic Servo-Flap for Rotor Blade Vibration Control.](image)
Table 1. ISA design specifications for various hinge arms.

<table>
<thead>
<tr>
<th>Design Case #</th>
<th>Hinge Arm r, mm</th>
<th>Generalized External Displacement</th>
<th>Generalized External Force</th>
<th>Generalized External Stiffness, k_s</th>
<th>Required Amplification Ratio, η</th>
<th>Optimal Gain, G</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>δ = 1/30 rad</td>
<td>M = 75 Nm</td>
<td>2140 Nm/rad</td>
<td>1.462</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>7.5</td>
<td>x = 0.25 mm</td>
<td>F = 10 kN</td>
<td>40 kN/mm</td>
<td>4.85</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>30</td>
<td>x = 1 mm</td>
<td>F = 2.5 kN</td>
<td>2.5 kN/mm</td>
<td>20</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>150</td>
<td>x = 5 mm</td>
<td>F = 0.5 kN</td>
<td>0.1 kN/mm</td>
<td>29.25</td>
<td></td>
</tr>
</tbody>
</table>

On the other hand, it must be noted that the required energy density is well within the reaches of existing ISA materials. High performance ISA materials can typically produce 3–4 kJ/m³. The baseline energy requirement for actuating the servo-flap in one direction is of the order of 1.25 J. Hence, by simple calculation, 1.25 J/(3 kJ/m³) ≈ 400 cm³ of active ISA material would suffice to activate the servo-flap in one direction. (For alternating motion, more material would be needed). These requirements are quite feasible. For example, an off-the-shelf actuator like P-247/70 (1 kg mass and 181 cm³ volume) can provide up to 0.666 J of energy. Hence, two of these actuators should suffice for one direction actuation, since 2 × 0.666 > 1.25 J. (For alternating motion, more actuators are required.) Since energy feasibility is proven, the remaining problem is how to architect the ISA material (or how to utilize an existing actuator) such that maximum use is made of its intrinsic properties. For the present application, this can be achieved through the displacement amplification concept, as shown in the next section.

Indirect Action ISA Devices

If indirect actuation through a displacement amplification device is considered, then a variety of options are open. A review of existing concepts for ISA displacement amplification (Figure 9) reveals that a multitude of methods and principles can be employed (solid mechanics or fluid mechanics, rigid body motion or structural deformation, etc.). Considerable engineering experience has been accumulated in this field, since the problem is not limited to ISA applications. Hence, one can assume with confidence that a reliable and compact displacement amplification device can be designed and built to suit the available space and geometry of our application. The guiding principles of such a device are described next.

Table 2. Unfeasible direct action ISA devices (#2 design: x = ±1 mm, F = ±2.5 kN, k_0 = 2.5 kN/mm).

<table>
<thead>
<tr>
<th>Supplier</th>
<th>Actuator Length</th>
<th>Actuator Mass</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>EDO Corporation</td>
<td>1.500 m</td>
<td>4 kg</td>
<td>Customized E200P-4 PMN stack</td>
</tr>
<tr>
<td>Polytec PI, Inc.</td>
<td>2.592 m</td>
<td>18 kg</td>
<td>18 x existing P-247.70 actuators</td>
</tr>
</tbody>
</table>

The design problem is formulated as: "Find the optimal value of G that satisfies a given stiffness ratio, r = k_s/k_0, and a given amplification ratio, η". It can be shown that, for a given value of η, the solution for G is not unique. This is best illustrated by the plot of η vs G shown in Figure II. If the output displacement, and hence the gain η, are not specified, we have an unconstrained optimization problem, and hence a unique optimum value of G exists for which η has a local and global maximum (the peak of the curve in Figure II). However, if the output displacement is also specified (as in the case studied here), then a constrained optimization problem has to be solved. This is illustrated in Figure II by the intersection of the curve with the horizontal line η = constant. Two solutions exist, G_1 and G_2. It can be shown (Giurgiutiu, Chaushy and Rogers, 1993) that the

The ideal displacement amplifier is considered rigid and without friction, viscous damping, hysteresis, etc. For further studies, a realistic displacement amplifier could be considered by taking these effects into account. A transformation efficiency factor accounting for these efforts could also be used in the first approximation.

1The ideal displacement amplifier is considered rigid and without friction, viscous damping, hysteresis, etc. For further studies, a realistic displacement amplifier could be considered by taking these effects into account. A transformation efficiency factor accounting for these efforts could also be used in the first approximation.
first solution is optimal, since it results in lower internal compressibility loss in the ISA actuator. (The apparent external stiffness seen by the ISA stack is \((k_x)_{\text{apparent}} = G^2 \cdot k_x\), and hence a minimum \(G\) ensures a minimum stiffness ratio, i.e., a minimum compressibility loss.) Thus, we conclude that for a given output ratio, \(\eta\), there is one and only one optimal (minimum) gain \(G(\eta)\) that satisfies Equation (9). This optimal gain has the expression (Giurgiutiu, Chaudhry and Rogers, 1993):

\[
G(\eta) = \frac{1}{2\eta} \sqrt{\frac{k_i}{k_x} \left( \sqrt{\frac{k_i}{k_x}} - \sqrt{\frac{k_i}{k_x} - 4\eta^2} \right)}
\]  

(10)

Note that for \(G(\eta)\) to take real values, the internal and external stiffness must satisfy the inequality \(k_i \geq 4\eta^2 k_x\). For an optimal operation point, \((\eta, G)\), the external and internal displacement amplitudes are given by (Giurgiutiu, Chaudhry and Rogers, 1993):

\[
x_i = \eta x_A, \quad x_i = \frac{1}{2} \sqrt{\frac{k_i}{k_x} \left( \sqrt{\frac{k_i}{k_x}} - \sqrt{\frac{k_i}{k_x} - 4\eta^2} \right)} x_A
\]

(11)

Based on the above analysis, we calculated the optimal operating point for a pair of actuators similar to P-247.70. The compound actuator has \(x_A = 0.250\) mm, and \(k_i = 185\) kN/mm. To attain a desired output displacement, \(x_i = 1\) mm, an amplification ratio \(\eta = 4\) is required. Using Equation (10) with \(k_i = 2.5\) kN/mm, we calculated the optimal gain as \(G = 5.85\). Several practical options are possible. If the amplification device were a simpler lever, then a 5.85:1 arm ratio would be required. For a hydraulic amplification device, the radii ratio would have to be \(\sqrt{G}\), i.e., 2.42:1. The corresponding internal force, and the associated displacement loss due to ISA material compressibility were also calculated (Giurgiutiu, Chaudhry and Rogers, 1993). These values are \(F_i = 14.6\) kN, and \(x_i = 0.079\) mm (i.e., 31.6% of total ISA travel \(x_A = 0.250\) mm). The parasitic internal energy is \(E_i = (1/2) \times 14.6 \times 0.079 = 0.577\) J, i.e., 31.6% of the ISA energy \(E_{ISA} = E_i + E_e = 1.25 + 0.577 = 1.827\) J, and \(0.577/1.827 = 31.6\%\).

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**Figure 9.** Displacement amplification concepts used in ISA devices.

**Figure 10.** Schematic drawing of a displacement amplification device.

**Figure 11.** Amplification ratio \(\eta\) vs gain \(G\) for \(k_x/k_i = 0.0135\), showing two possible solutions, \(G_1\) and \(G_2\).
To illustrate the general applicability of our design method, a second example was considered. The desired displacement was increased to 5 mm ($\eta = 20$) with a corresponding external force $F_x = 0.5$ kN, and associated external stiffness $k_x = 0.1$ kN/mm. (Design Case #3 in Table I). Using Formula (10), the required gain was calculated as $G = 29.248$. Calculating the internal force and internal displacement due to ISA material compressibility yields (Giurgiutiu, Chaudhry and Rogers, 1993) $F_t = 14.6$ kN, $x_t = 0.079$ mm (i.e., 31.6% of total travel), and $E_t = 0.577$ J, (i.e., 31.6% of the total ISA energy $E_{ISA} = 1.827$ J). Note that the internal values are exactly the same as in the previous design, though the required output displacement is 5 times larger. This should not be surprising since it can be shown (Giurgiutiu, Chaudhry and Rogers, 1993) that, for an optimally designed displacement amplifier, the internal force and compressibility loss are invariant with respect to transformations of the output requirements in which the force-displacement product is kept constant. Thus, it is proven that large output displacements can be achieved with reasonable efficiency by using an optimally designed displacement amplifier which best utilizes the ISA material properties.

DISCUSSION

The review of several studies regarding HHC-IBC vibration control through both conventional and solid state actuation resulted in a wide variety of proposed solutions and associated design parameters. However, utilizing engineering judgment and theoretical insight, we were able to find some common traits. Hence, a set of representative values were identified to represent a typical combination of hinge moment, angular displacement and frequency. This set of values was then used as a bench mark for testing the effectiveness of various ISA architectural options. From the start, it was realized that the transformation of the specified angular variables into linear-action quantities requires careful engineering, in order to obtain workable values for the linear displacement and the associated hinge arm. Bounds on these values were identified as a minimum of $\pm 1$ mm displacement with 30 mm hinge arm, and a maximum of $\pm 5$ mm displacement with 150 mm hinge arm. Depending on rotor blade size and installation options, practical designs will be placed somewhere within these bounds.

The first attempt was to produce the linear displacement directly through an ISA stack. The lower displacement value ($\pm 1$ mm) was considered in this trial. However, even this value proved to be ten times larger than the usual free output of ISA stacks. When external reaction was also included, the projected actuator length was found to be in the range 1.5–2.6 m. With such a configuration, most of the ISA displacement is lost in the stack compressibility (the stack stiffness decreases rapidly with length). The lesson drawn from this initial attempt was that, for the present application, direct actuation is not a viable proposition since it does not make best use of the ISA material properties.

Performing a system analysis of the power and energy requirements proved that these are well within the capabilities of commercially available actuators. The invariant quantities involved in our application, for example, the maximum required output energy per cycle, was found to be only 1.25 J per blade. This value is covered by a couple of P-247.70 actuators which are commercially available from Polytech PI (1994). Since the invariant requirements are satisfied, the concept is generally feasible, as long as rational use is made of the ISA material properties. The parasitic strain energy which gets stored internally through stack compressibility was identified as a main source of difficulties, and it was decided that means must be found to keep it within manageable bounds.

Subsequent studies were targeted at ISA devices incorporating displacement amplification. A detailed analysis of the displacement amplification principles and work-energy balance was performed, and a formula for optimal design was derived. When various design points along the optimal design curve were considered, it was found that the parasitic internal strain energy remained the same. Thus, for our analysis method, the invariance of the parasitic internal energy with respect to design transformations was established. A strategy was devised to find the optimal gain value associated with a given displacement amplification ratio. This optimal gain gives the minimum parasitic internal energy within the design constraints, and leads to the minimum displacement loss through internal compressibility. We also identified a test for establishing the feasibility of such an optimal gain, viz. $k_x/k_e < 4\eta^2$. Two numerical examples were developed. They showed that both the lower and the upper bounds on the required linear displacement amplitude ($\pm 1$ mm and $\pm 5$ mm, respectively) can be attained with a small number of commercially available actuators connected to displacement amplifiers of reasonable requirements (optimal gains of $G = 5.85$, and 29.248, respectively). These theoretical developments need to be substantiated by practical implementation and experimental investigations.

The work presented herein is far from complete, and several avenues are open for further research. First, further attention should be given to defining the base-line requirements for the design of an ISA device capable of achieving HHC-IBC vibration control of helicopter rotors. Such a study poses a formidable analytical task, since load prediction codes for rotor blade unsteady aerodynamics still show wide variation in results. However, a concerted effort of industry and academic could yield improved estimates for the HHC-IBC actuator parameters. Second, the analysis presented herein could be extended to consider nonideal displacement amplifiers incorporating transmission losses due to their own compliance and damping. Though these effects are expected to be relatively small, their incorporation is
worthwhile in a final design. Third, the more comprehensive subject of identifying and numerically defining the power dissipation mechanisms involved in the three stages of an ISA system (ISA material, power supply, and mechanical and electrical transmission) should be addressed in more detail. It is found that ISA devices may significantly differ from conventional electro-mechanical devices, since large values of reactive energy (electrical and mechanical) may be involved. However, a good engineering design can keep these losses to a minimum. Fourth, the analysis must be extended to incorporate the effects of bias springs incorporated in the ISA devices that prevent putting the ISA material into tension; and of the effect of electric D.C. bias.

Finally, a major concern with any rotor blade device operating in the outer blade span has always been the effect of the very high and variable g loading (a point at 5 m radius rotating at 30 rad/sec experiences $g = 5 \times 30 = 4500$ m/s² = 450 g!). This was a major concern with the ARCS studies (Straub and Charles, 1990; Phillips and Murphy, 1990) employing conventional electro-mechanical and hydraulic devices. It seems plausible to assume that ISA actuators, as solid state devices, would be much less affected by such effects; however, to date, no theoretical and experimental research has been done to verify this assumption.

CONCLUSION

Following a qualified review of the existing literature, a set of baseline requirements for designing an ISA servo-flap actuator for the active HHC-IBC vibration control of helicopter rotor blades was identified (±2° of servo-flap displacement, ±75 Nm of hinge moment at 25–30 Hz). A workable range of hinge arm values (30 to 150 mm) was considered, and a range of stroke amplitudes (±1 mm to ±5 mm) was obtained. Invariant quantities were also investigated, and typical values were found to be, per blade: 1.25 J maximum instantaneous energy; 470 W (approx. 0.5 kW) maximum instantaneous power; and 2.5 Nm force-displacement product. This baseline specification was used as a bench-mark for possible ISA solutions. It was found that the direct use of long ISA stacks is impractical since it does not use the ISA properties efficiently and results in excessive displacement loss due to internal compressibility. However, the analysis of a system incorporating optimally designed displacement amplifiers predicted that the task can be achieved with a reasonable number of commercially available ISA devices used in ingenious ways. The experimentiation of an actual ISA device coupled with an optimally designed displacement amplifier for validating the main theoretical assumptions, and for identifying the practical issues associated with its engineering implementation is currently under way. Other opportunities for further research were also identified.

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