POWER AND ENERGY ISSUES IN THE INDUCED-STRAIN ACTUATION FOR AEROSPACE ADAPTIVE CONTROL.

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ABSTRACT

General concepts regarding power and energy in mechanical and electrical systems representing the aerodynamic control surface and the solid-state induced strain actuator are first reviewed. Power and energy management concepts in conventional flight control systems and in induced-strain actuated flight control systems are presented. The electromechanical coupling across the solid-state induced strain actuator, and the accompanying reversible flow of reactive energy and power are highlighted. Experimental results obtained in the laboratory with the HAHDIS Mk. 1 hydraulically-amplified high-displacement induced-strain proof-of-concept demonstrator are presented and discussed. Suggestions for further work are given.

INTRODUCTION

Aerospace adaptive control through induced-strain actuation has been intensively researched in recent years. With the advent of high-performance electro-active and magneto-active materials and actuators at accessible prices, the road is open towards revolutionary changes in the aircraft flight control systems design and construction. Active materials technology allows the transition from conventional mechanical and/or hydraulic flight control systems to all electric concepts in which the control power and energy are transmitted across the aircraft in electrical form and are transformed into mechanical action directly at the control surface. The vulnerable of hydraulic lines are replaced by the much more robust electric conductors, while a large number of hydraulic paraphernalia (accumulators, cocks, valves, tanks, etc.) might simply disappear. The implementation of induced-strain actuating concepts for adaptive flight control, is currently pursued on two parallel paths:

1. Deformable aerodynamic surfaces (wings\textsuperscript{2,3}, tailplanes, rotor blades\textsuperscript{4,5}, etc.) activated by embedded induced strain active-material fibers.

2. Rigid control surface (ailerons, flaps\textsuperscript{6,7,8}, leading edge flaps\textsuperscript{9}, etc.) with induced-strain active-material actuators replacing the conventional hydraulic cylinders or electric motors.

The first direction aims at creating a deformable aerodynamic surface similar to that met in the animal world. This concept is, undoubtedly, visionary and full of attractive benefits for adaptive control and for performance and maneuverability improvement, since it allows the simultaneous use of span-wise and chord-wise control algorithms. However, the implementation of the deformable aerodynamic surface concept is not straightforward, and has encountered considerable difficulties. The building blocks of a deformable adaptive aerodynamic surface need to be first developed and individually certified. The aircraft structural design philosophy must be revised and updated as to incorporate the concept of highly deformable aerodynamic surfaces.

The second direction aims at modifying existing flight control concepts by replacing the traditional actuation methods with lighter, less complex, more responsive, and more reliable induced-strain actuators. This concept retains the limitations of rigid and localized control surfaces, but its implementation is easier since it utilizes, to a large extent, existing techniques\textsuperscript{9}. This concept offers a gradual approach to the implementation of induced-strain actuation principles, and its experimentation will provide practical experience into the specific behavior and limitations of induced-strain actuators\textsuperscript{10}. This paper pursues the study of power and energy issues connected with the implementation of induced-strain adaptive control using a conventional control surface and an induced strain actuator as prime mover.

POWER AND ENERGY CONCEPTS FOR INDUCED-STRAIN ACTUATION OF FLIGHT CONTROL SURFACES

Mechanical Power and Energy

Figure 1a shows an aerodynamic control surface, while Figure 1b presents the equivalent mass-spring-damper system.
For lightly damped mechanical systems, the reactive power is predominant, and the maximum instantaneous power is almost equal to the power amplitude, \( \hat{P} \). For mechanical systems with higher damping, the active power component becomes more important and the maximum instantaneous power may be significantly larger than the power amplitude.

**Mechanical Energy**

The instantaneous mechanical energy

\[
E(t) = \int_0^t P(t) \, dt ,
\]

(7)

can be expressed as

\[
E(t) = E_{\text{active}} + E_{\text{reactive}}(t) ,
\]

(8)

where

\[
E_{\text{active}}(t) = (E \sin \psi) \cos \alpha ,
\]

(9)

\[
E_{\text{reactive}}(t) = E \sin(\alpha \cos \psi) \sin \alpha ,
\]

(10)

and \( \hat{E} = \frac{1}{2} \hat{F} \hat{u} \) is the mechanical energy amplitude.

The active components of power and energy represent power and energy dissipation of the mechanical system through friction and other losses. The reactive components represent energy being exchanged between the mechanical system and its external driver. Kinetic and elastic energy storage and restitution takes place in the mass, \( m \), and spring, \( k \). Note that the reactive power amplitude given by Equation (4) goes through a minimum as the system approaches resonance (\( \omega \to \sqrt{\frac{k}{m}} \)), which explains why we get such a large resonance response from low-damping mechanical systems.

**Mechanical Power and Energy in Complex Notations**

Define the complex stiffness,

\[
\tilde{k}(\omega) = (k - \omega^2 m) + i \omega \alpha ,
\]

(11)

and write the complex power and energy,

\[
\tilde{P} = \omega \left( \frac{1}{2} \hat{F} \hat{u}^2 \right) , \quad \tilde{E} = \frac{1}{2} \hat{E} \hat{u}^2 .
\]

(12)

Hence,

\[
P_{\text{active}} = \text{Im} \tilde{P} , \quad \text{and} \quad P_{\text{reactive}}(t) = \tilde{P} e^{i2\alpha t} .
\]

(13)

Similarly,
\[ E_{\text{active}}(t) = (\Imaginary \bar{E}) \cdot \alpha \tau \]  
(14)

\[ E_{\text{reactive}}(t) = |\bar{E}| \sin(\alpha \tau + \psi) \sin \alpha \tau \]  
(15)

Detailed derivation of these expressions was presented in ref. 15.

**Mechanical Power with Bias Force**

A bias force, \( F_0 \), may be used to keep a permanently compressive force in the induced-strain actuator and thus avoid putting the ceramic material into tension. In practice, this could be achieved by having the unloaded control surface in an offset position, such that, in flight, the aerodynamic pressure will tend to bring the control surface in alignment with the airflow, and will create a steady force proportional to the initial offset. Hence,

\[ F(t) = \bar{F} \sin(\alpha \tau + \psi) + F_0 \]  
(16)

and the reactive power becomes

\[ P_{\text{reactive}}(t) = \dot{\bar{P}} \left[ \sin(2 \alpha \tau + \psi) + 2 f_0 \dot{\alpha} \cos \alpha \tau \right] \]  
(17)

where \( f_0 \) is the bias force coefficient, \( f_0 = F_0 / \bar{F} \). The active power remains the same as in Equation (5). For compressive bias force, \( f_0 < 0 \), and low damping, \( \psi \approx 0 \), the reactive power peaks at the critical angular value:

\[ (\alpha \tau)_{\text{crit}} = \sin^{-1} \left( -\frac{f_0}{4 \sqrt{f_0^2 + 8}} \right), \quad f_0 < 0. \]  
(18)

Figure 3 shows that, for compressive bias loads greater or equal to the dynamic load, the reactive power is dominated by the bias force component.

![Figure 3](image)

Figure 3 Time plot of normalized reactive power for \( \psi \approx 0 \) and various negative values of \( f_0 \).

**Electrical Power and Energy**

The equivalent input circuit of an electro-mechanical device (Figure 4) consists of a resistance, \( R \), an inductance, \( L \), and a capacitance, \( C \). Their relative magnitude differs with the type of electro-mechanical device, i.e., the circuit is mainly “resistive” for an electric motor, mainly “capacitive” for a electro-active induced-strain actuator, and mainly “inductive” for a solenoid or for a magneto-active induced-strain actuator.

![Figure 4](image)

The circuit differential equation is

\[ v(t) = R \cdot i(t) + L \cdot \frac{di(t)}{dt} + \frac{1}{C} \int i(t) dt. \]  
(19)

For alternating current, \( i(t) = I \sin \alpha \tau \), we get

\[ v(t) = V \sin(\alpha \tau + \phi), \]  
(20)

where the voltage amplitude, \( V \), and the phase angle, \( \phi \), are given by

\[ V = \sqrt{R^2 + \left( \alpha L - \frac{1}{\omega C} \right)^2}, \quad \phi = \tan^{-1} \left( \frac{\alpha L - \frac{1}{\omega C}}{R} \right). \]  
(21)

**Electrical Power**

The instantaneous power, \( P(t) = v(t) \cdot i(t) \), is

\[ P(t) = \dot{\bar{P}} \cos \phi + \dot{\bar{P}} \sin \left( 2 \alpha \tau + \phi - \frac{\pi}{2} \right). \]  
(22)

where \( \dot{\bar{P}} \) is the power amplitude

\[ \dot{\bar{P}} = \frac{1}{2} VI = \frac{1}{2} \sqrt{R^2 + \left( \alpha L - \frac{1}{\omega C} \right)^2} \cdot I^2. \]  
(23)

The components of instantaneous power are

\[ P_{\text{active}} = \dot{\bar{P}} \cos \phi = \frac{1}{2} RI^2, \]  
(24)

\[ P_{\text{reactive}}(t) = \dot{\bar{P}} \sin \left( 2 \alpha \tau + \phi - \frac{\pi}{2} \right). \]  
(25)

The active electrical power represents energy being dissipated by the electrical system that results from resistive heating, magnetic flux leakage, eddy currents in magnetic armatures, dielectric loss, etc. The reactive power represents energy being exchanged back and forth between the electrical system and the power plant. The reactive power is not consumed in the electrical system, but must, nevertheless, be supplied to the system at some point in time in order to be retrieved at some latter time. While traveling back and forth, the reactive power has indirect dissipative effects through the resistive heating of the power lines. Conventional electric motors driving predominantly dissipative mechanical loads have high active power values that are close to the reactive power amplitude (\( \cos \phi > 0.9 \)). The reactive power of electrical
motors is additionally minimized through compensating capacitors chosen in such a way as to make $\sqrt{LC} \to \omega$ (power factor correction). In solid-state induced-strain actuators, the situation is reversed, and the reactive power may be an order of magnitude larger than the active power. The compensation of reactive power is not straightforward, and induced-strain actuators need specialized power supplies. The instantaneous variation of the electrical power to a solid-state induced-strain actuator resembles the variation of the mechanical power in a low-damping mechanical system (Figure 3).

Electrical Energy

The instantaneous electrical energy

$$E(t) = \int_{0}^{t} P(t) dt,$$

is expressed as

$$E(t) = E_{active} + E_{reactive}(t),$$

where

$$E_{active}(t) = (\dot{E} \cos \phi) \cdot \omega t,$$

$$E_{reactive}(t) = -\dot{E} \cos(\omega t + \phi) \sin \omega t,$$

and $\dot{E} = \frac{1}{2} V_{L} \omega$ is the electrical energy amplitude.

Electrical Power and Energy in Complex Notations

Define the complex impedance

$$Z(\omega) = R + i(\alpha \omega - \frac{1}{\alpha \omega}),
Z = |Z|e^{i\phi} = |Z|(\cos \phi + i \sin \phi) .$$

For solid-state induced-strain actuators driven by voltage sources, also define the admittance,

$$Y(\omega) = \left[ R + i(\alpha \omega - \frac{1}{\alpha \omega}) \right]^{-1},
Y = |Y|e^{-i\phi} = |Y|(\cos \phi - i \sin \phi) .$$

Introduce the complex power and energy,

$$\bar{P} = \frac{1}{2} Z I^2 \quad \text{or} \quad \bar{P} = \frac{1}{2} Y V^2,$$

$$\bar{E} = \frac{1}{\omega} Z I^2 \quad \text{or} \quad \bar{E} = \frac{1}{\omega} Y V^2.$$

Hence, the active, reactive and instantaneous electrical power components are:

$$P_{active} = \Re \bar{P},$$

$$P(t) = \Re \bar{P} + \Re e^{i(2\alpha t - \phi)} .$$

The active and reactive electrical energies are:

$$E_{active}(t) = (\dot{E} \cos \phi) \cdot \omega t = (\Re \bar{E}) \cdot \omega t,$$

$$E_{reactive}(t) = |\bar{E}| \cos(\omega t + \phi) \sin \omega t .$$

Electrical Power with Bias Voltage

Many solid-state induced-strain actuators do not have a symmetrical behavior when the polarity of the applied voltage is reversed. For example, the solid-state induced-strain actuator P270.70 from Polytec PI, has a maximum free expansion of 120 μm under a maximum voltage of -1000 V, and a maximum free contraction of -30 μm under a maximum voltage of +250 V. Other solid-state induced-strain actuators cannot reverse polarity, like, for example, the actuator E300P-4 from EDO Corp. which attains 60 μm expansion at +800 V, but supports no voltage reversal. To attain unsymmetric operation, apply a bias voltage, $V_0$, and write:

$$V(t) = V_0 + V \sin \omega t .$$

The active power is not influenced by the bias voltage and maintains the expression given in Equation (24), while the reactive power becomes

$$P_{reactive}(t) = \hat{P}(-\cos(2\alpha t - \phi) + 2V_0 \sin(\alpha t - \phi))$$

where $v_{0} = V_{0} / V$ is the bias voltage coefficient. The phase angle, $\phi$, is measured with respect to the voltage. Solid-state actuators are either mainly capacitive or mainly inductive, and their phase angle is close to ±90°. For a capacitive actuator,

$$P_{reactive}(t) = -\hat{P}(\sin 2\alpha t + 2v_{0} \cos \alpha t),$$

i.e., the same as Equation (17) for mechanical power with $\psi \approx 0$, but with sign reversal. Hence, for a system operating with bias voltage, the maximum reactive power takes place at

$$\left(\alpha t\right)_{crit} = \sin^{-1}\left(-\frac{v_{0}}{4} + \frac{1}{4}\sqrt{v_{0}^2 + 8}\right), \ v_{0} > 0 .$$

Solid-State Induced-Strain Actuators Operating a Mechanical System

Figure 5 gives a schematic representation of a solid-state induced-strain actuator (a PZT stack) operating a mechanical system of parameters $k_x, m_x, c_x$. The PZT stack is energized by a voltage source, $v(t)$, which sends a current $i(t)$ that builds up the internal charge. As the charge is built up, the voltage and the electric field increase.
Under the action of the electric field, the material expands and produces an output displacement, $u(t)$, which generates a reaction force from the mechanical system, $F(t)$. The reaction force, $F(t)$, acting onto the PZT stack, induces loss of output displacement through the stack compressibility and through the counter electric motive force (emf) due to the piezo-electric effect. An actuator under load always has a lower output displacement than a load-free actuator energized by the same voltage. A detailed analysis of the configuration shown in Figure 5 starting from the basic electro-mechanical constitutive equations of active material behavior and using wave propagation techniques was given in ref. 11.

![Figure 5 Schematic representation of a solid-state induced-strain actuator (Here, a PZT stack) operating against a mechanical load.](image)

The equivalent input admittance is:

$$Y(\omega) = i\omega C \left(1 - \frac{d^2}{s \varepsilon} \frac{\bar{F}(\omega)}{1 + \bar{F}(\omega)}\right),$$  \hspace{1cm} (43)

where $C$ is the zero-load capacitance of the stack, $d$ is the zero-load induced-strain coefficient, $s$ is the open-circuit (zero-field) complex mechanical compliance of the stack, and $\varepsilon$ is the zero-load complex electrical permittivity (dielectric constant) of the active material, i.e.,

$$\varepsilon = \varepsilon(1 - i\eta)$$  \hspace{1cm} (44)

$$\bar{\varepsilon} = \varepsilon(1 - i\delta)$$  \hspace{1cm} (45)

where $\eta$ is the hysteresis internal damping coefficient, and $\delta$ is the dielectric loss coefficient. The coefficient $\bar{F}(\omega)$ is the complex stiffness

$$\bar{F}(\omega) = \frac{k_e(\omega)}{k_i},$$  \hspace{1cm} (46)

where the complex external stiffness, $k_e(\omega)$, is given by Equation (11), while the complex internal stiffness is

$$k_i = \frac{A}{s l} = \frac{A}{s l}(1 + i\eta),$$  \hspace{1cm} (47)

with $A$ and $l$ the cross-sectional area and length of the stack, respectively.

The input and output power expressions are calculated by substituting the stack admittance given in Equation (43) into the Equations (12), (13), (32) and (36). The electrical input power becomes:

$$\bar{P}_{in} = \frac{1}{2} i\omega C \left(1 - \frac{d^2}{s \varepsilon} \frac{\bar{F}(\omega)}{1 + \bar{F}(\omega)}\right) V^2$$  \hspace{1cm} (48)

The mechanical output power is calculated using the output displacement

$$\hat{u} = \frac{1}{1 + \bar{F}(\omega)} u_{ISA}$$  \hspace{1cm} (49)

where $u_{ISA}$ is the induced-strain displacement

$$u_{ISA} = \frac{\dot{E} dl}{V dl / l},$$  \hspace{1cm} (50)

with $l$ the thickness of the active material layers. Hence, the mechanical output power is

$$P_{out} = \frac{\omega}{(1 + r)^2} \frac{d^2}{s \varepsilon} \left(\frac{1}{2} CV^2\right)$$  \hspace{1cm} (51)

For a stack driven by a power supply with limited current capabilities, $I_{max}$, we derived the power expressions using the impedance formulation, i.e.,

$$\bar{P}_{in} = \frac{1}{2} \frac{1}{i\omega C} \left(1 - \frac{d^2}{s \varepsilon} \frac{\bar{F}}{1 + \bar{F}}\right)^{-1} I_{max}^2,$$  \hspace{1cm} (52)

$$\bar{P}_{out} = \frac{1}{2} \frac{\bar{F}}{(1 + \bar{F})^2} \frac{1}{i\omega C} \frac{d^2}{s \varepsilon} \left(1 - \frac{d^2}{s \varepsilon} \frac{\bar{F}}{1 + \bar{F}}\right)^{-2} I_{max}^2.$$  \hspace{1cm} (53)

Note that, for a power supply with limited current capabilities, the output displacement amplitude decreases rapidly with frequency, since

$$\hat{u}(\omega) = \left|\frac{1}{1 + \bar{F}(\omega)} \frac{1}{\alpha C} \frac{dl}{l} \left(1 - \frac{d^2}{s \varepsilon} \frac{\bar{F}(\omega)}{1 + \bar{F}(\omega)}\right)^{-1}ight| I_{max}.$$  \hspace{1cm} (54)

**POWER MANAGEMENT IN ADAPTIVE FLIGHT CONTROL SYSTEMS**

**Conventional Flight Control Systems**

Conventional flight control systems based on hydraulic operation with servo-valve control can also be used for adaptive control by superposing a controller signal onto the servo-valve electrical input. Figure 6 presents a simplified representation of a conventional flight control system. The power uptake from the aircraft engine is used to drive a variable-flow axial-pistons hydraulic pump. The hydraulic pump delivers constant-pressure hydraulic fluid to the various consumers inside the aircraft. The flight control system uses electro-hydraulic...
servo-valves to modulate the flow of fluid to the hydraulic actuator.

Figure 6 Schematic representation of a conventional flight control system.

The electro-hydraulic servo-valve directs the constant-pressure hydraulic supply into the advance and retreat chambers of the hydraulic actuator (Figure 7). To make the piston advance (full arrows), the servo-valve directs hydraulic pressure to Port #1, while connecting Port #2 to the low pressure return circuit. To achieve retreat (dashed arrows), Port #2 is pressurized, while Port #1 is connected to the return circuit. Thus, a sinusoidal electrical signal sent by the controller to the servo-valve is eventually transformed into a sinusoidal motion of the hydraulic actuator. The output rod of the hydraulic actuator is connected to the hinge-arm of the flight control surface, and makes it undergo oscillating motion.

Figure 7 Schematic operation of a hydraulic actuator under servo-valve control.

The performance limitations of hydraulic flight control systems lie in their poor high frequency response. The hydraulic-flow switching capabilities of the servo-valves are limited. To achieve high output, a two or three stage cascade can be used in the servo-valve construction, but this increases its complexity, weight, and overall dimensions. Another limiting factor is the capability of the hydraulic system supply. To accommodate high flow rates, large supply pipes must used, which creates obvious weight and space problems. For these reasons, the frequency bandwidth of hydraulic flight controls is somehow limited.

Power management evaluations of a conventional flight control system usually rely on the assumption that no useful power can be recovered through the low pressure return circuit. Even with the hydraulic actuator acting against a spring (pure reactive power), none of the power needed to move the piston forward will be recovered during the piston retreat. This assumption is justified by the large difference between the pressure in the high-pressure line (typically 21 MPa or 3000 psi) and that in the low-pressure line (typically 0.2-0.5 MPa or 30-80 psi). Under this assumption, the mechanical power output from a hydraulic actuator acting against the spring-mass-damper system shown in Figure 1 needs to equal the sum of the active and reactive mechanical power amplitudes, i.e.

\[ P_{out} = P_{active} + P_{reactive} \]

\[ = \omega \frac{1}{2} \alpha \omega u^2 + \omega \frac{1}{2} \sqrt{(k - \omega^2 m) + (\alpha x)^2 \dot{u}^2} \]

(55)

For systems with low damping, acting off-resonance, the reactive power is dominant, and Equation (55) takes the simpler form

\[ P_{out} \approx \omega \frac{1}{2} \hat{F} \dot{u} = \omega \frac{1}{2} \sqrt{(k - \omega^2 m) + (\alpha x)^2 \dot{u}^2} \]

(56)

Under quasi-static conditions, the inertia and damping terms of the aerodynamic forces can be ignored, and Equation (56) simplifies even further by taking \( \hat{F} \) equal to the static aerodynamic control force.

The power input to the conventional flight control system, \( P_{in} \), equals the output power plus losses:

\[ P_{in} = P_{out} + P_{loss} \]

(57)

The power losses, \( P_{loss} \), of a hydraulic flight control system as shown in Figure 6, can be traced to:

- inefficiency of the hydraulic pump;
- viscous friction of the hydraulic fluid flow through the supply lines;
- internal friction of the hydraulic actuator.

**Induced-Strain Actuated Flight Control Systems**

Figure 8 shows a simplified representation of a induced-strain actuated flight control system using electro-active materials.

Figure 8 Schematic representation of an solid-state induced-strain actuated flight control system using electro-active materials.

The power uptake from the aircraft engine is used to drive the electric generator which delivers 115 V @ 400 Hz electric power to the various consumers inside the aircraft. The solid-state induced-strain actuation of one flight control surface consists of a
A high-voltage power amplifier and the electro-active material coupled to a displacement amplification device. The high-voltage power amplifier follows the controller demands and uses the 115 V 400 Hz electric power supply to create the power signal energizing the electro-active material. The resulting induced-strain displacement is kinematically amplified and then sent to the flight control surface input arm. A similar configuration can be devised for an induced-strain actuated flight control system using magneto-active materials by replacing the high-voltage power amplifier with a controlled-current power amplifier (Figure 11).

![Schematic representation of a solid-state induced-strain actuated flight control system using magneto-active materials.](image)

Figure 9 Schematic representation of an solid-state induced-strain actuated flight control system using magneto-active materials.

There are several beneficial aspects that differentiate the solid-state induced-strain flight control system from the conventional (hydraulic based) flight control systems:

1. The power supply is electric, hence power can be sent much easier throughout the entire aircraft. The electric power delivery system is less prone to damage and does not suffer from the classical ailments of the hydraulic systems (leaks, bursts, gassing, etc.).
2. The demand signal generated by the controller can be directly input into the flight control system without the need for an electro-mechano-hydraulic converter such as the servo-valve.
3. The solid-state induced-strain actuators based on electro-active (magneto-active) materials are electro-mechanically coupled through the piezoelectric (piezo-magnetic) effect, i.e. it generates mechanical power when activated electrically, and vice-versa. This aspect allows for a two-way power flow between the electrical input and the mechanical output, and opens attractive opportunities for power recycling that were impossible in the conventional hydraulic systems.

There are also several drawbacks of solid-state induced-strain flight control systems that need to be addressed before they could truly compete with the conventional flight control systems:

1. The electro-active (magneto-active) induced-strain actuators present a predominantly reactive input impedance to the power amplifier. This means that, as opposed to traditional electric motors, the solid-state induced-strain actuators have a very small power factor value, typically $\cos \phi \approx 0.05 - 0.10$. This presents an unusual problem to the power amplifier designer who has to tackle efficiently the very large amounts of reactive power flowing into the solid-state induced-strain actuators.
2. The electro-active and magneto-active materials presently available for the fabrication of induced-strain actuators are brittle ceramics that need special mechanical design for their safe and reliable utilization.
3. The electro-mechanical coupling properties of presently available electro-active (magneto-active) materials are highly temperature dependent and hence the actuators incorporating these materials may be affected by the severe temperatures encountered throughout the flight envelope (-50°C to +80°C).

**LABORATORY EXPERIMENTS WITH HAHDIS Mk. 1**

**PROOF-OF-CONCEPT DEMONSTRATOR**

A hydraulically amplified high displacement induced strain actuator, HAHDIS Mk.1 (Figure 10), was designed and built in the Center for Intelligent Material Systems and Structures at Virginia Tech to serve as an experimental proof-of-concept demonstrator for our theoretical developments. Details about HAHDIS Mk.1 construction and initial testing were presented in ref. 10, 12, 13, 14.

![Image of HAHDIS Mk.1](image)

Figure 10 The hydraulically-amplified high-displacement induced-strain actuator, HAHDIS Mk.1.
For dynamic testing, four EDO Corporation E300-P4 PMN stacks, grouped in serial pairs, were used. The resulting stacks had 0.120 μm free displacement, 80 kN/mm internal stiffness, and ~1.45 μF no-load capacitance. They were placed at the reciprocal-acting inputs of the HAHDIS Mk.1 device. The PMN stacks were energized with high-voltage sinusoidal signals (+400±400 V) produced by a dual-channel TREK 50/750 high-voltage amplifier. The two signals have sinusoidal amplitudes of ±400 V (opposite phase to each other), but the same bias of +400 V. This arrangement permits one stack to expand, while the other retreats, thus producing an alternating input motion. The TREK 50/750 high-voltage amplifier has the current limitation \( I_{\text{max}} = 50 \text{mA} \). This limitation of the electric power supply reflected in the frequency behavior of the actuator, and allowed us to draw conclusions regarding the variation of power with frequency and load.

During dynamic testing, a range of discrete frequencies from 1 Hz to 30 Hz was sampled. At each frequency, the sampled data was collected in the computer with VTDEE software. The sampled data consisted of the input displacements, \( u_i(t) \) and \( u_2(t) \), the output displacement, \( u_e(t) \), and the hydraulic pressures, \( p_1(t) \) and \( p_2(t) \). Captured samples of the signals proved the good sinusoidal shape of the waveforms.

Three test cases were considered. The difference between the three test cases consisted of the amount of internal friction present in the displacement amplification device and of the amplitude of the external load. The internal friction of the hydraulic displacement amplifier resides in the rubbing between the conventional hydraulic seals and the cylinder bore. The internal friction acts as an additional load on the active-material stacks and results in a reduced output displacement and power. The internal friction is especially important in the output cylinder since its value is back-amplified by the kinematic ratio, and since the displacement of the output piston (1 mm) is one order of magnitude larger than the displacement of the input pistons (120 μm). To reduce the internal friction in the output cylinder, we replaced, at some stage in our experiment, the conventional output piston with a seal-less labyrinth design using dynamic fluid sealing. This device variant was designated HAHDIS Mk.1a.

Two different flaps were used in our experiment to generate external inertial loads: a heavy simulated flap, and a light realistic flap (Figure 11). They have similar areas, but very different masses. The heavy flap, constructed inexpensively in the machine shop, was used in the initial dynamic testing to provide an upper bound on HAHDIS capabilities. The light flap, adapted from the rudder tab of a CESSNA light aircraft, was used to reproduce real life inertial loads.

![Figure 11](image-url)

**Case 1: HAHDIS Mk.1 Performance Under Large Internal Friction and Heavy Inertial Load.**

These experiments were conducted in August 1995, using the initial Mk.1 device incorporating conventional hydraulic seals in the output cylinder and a heavy simulated-flap inertial load. During these experiments, significant output response could be measured up to 20 Hz (Figure 12).

![Figure 12](image-url)

The output displacement response amplitude, \( \dot{u}_e \), had a moderate decrease up to 10 Hz, and a very rapid decrease afterwards. Examination of the average input displacement curve, \( \ddot{u}_{i,\text{av}} \), shows that the input was also decreasing. Additional examination of the oscilloscope data during the experiment showed that the high-voltage signal provided by the model 50/750 TREK amplifier started to become angular around 5 Hz, indicating that the upper limit of the amplifier performance was reached. Between 5 Hz and 10 Hz, the high-voltage signal from the amplifier reduced its amplitude, but maintained a steady waveform. Above 10 Hz, the amplifier signal deteriorated much faster, and its amplitude rapidly decreased. Thus, the decreased of input voltage resulted in decreased of the input displacement, and generated the observed
decrease of the output displacement. When compensated for the loss of input displacement, the output displacement curve became almost flat (curve $\hat{u}_e^*$ in Figure 12), with a peak at 10 Hz that could be attributed to mechanical resonance. The stable and controllable behavior of the device under resonance conditions demonstrated in this experiment is an added merit of the HAHDIS concept.

Case 2: HAHDIS Mk.1a Performance Under Reduced Internal Friction and no External Load.

These experiments were conducted in November 1995, after introducing the Mk.1a modifications that reduced the internal friction in the amplification device. The experiment was run without any external load, i.e. no flap was connected to the output rod of the device. Significant response was observed well beyond 30 Hz, but it was considered that a frequency sweep up to 30 Hz would be sufficient. Figure 13 shows the frequency response curve for the interval 1 to 30 Hz. The output displacement response amplitude, $\hat{u}_e$, has a moderate increase up to 10 Hz, followed by a continuous decrease. Examination of the average input displacement curve, $u_{i,av}$, shows that it is also decreasing.

![Compensated output displacement, $u_e^*$](image1)

![Output displacement, $u_e$](image2)

![Averaged input displacement, $u_{i,av}$](image3)

![Frequency response curves of the HAHDIS Mk.1a demonstrator in the range 1-30 Hz with reduced internal friction and no external load.](image4)

Additional examination of the oscilloscope data during the experiment showed that the high-voltage signal provided by the model 50/750 TRENK amplifier started to become angular around 10 Hz, indicating that the upper limit of the amplifier performance was reached. Above 10 Hz, the amplifier signal deteriorated much faster, and its amplitude rapidly decreased. Thus, the decreased input voltage resulted in decreased input displacement to the device, and explains the observed decrease in output amplitude. When compensated for the loss of input displacement, the output displacement curve became almost flat (curve $\hat{u}_e^*$ in Figure 13). This experiment proved the effectiveness of the HAHDIS principle, showing its good performance over the frequency range 1-30 Hz in spite of the input displacement decrease due to limitations in the high-voltage power supply. Compared with the previous experiment, this experiment showed a 50% increase in frequency range (from 20 to 30 Hz) and 41% in maximum peak-to-peak amplitude (from 0.8 mm to 1.2 mm). These improvements can be attributed to the reduced internal friction of the HAHDIS Mk.1a design and to the absence of external loads.

Case 3: HAHDIS Mk.1a Performance Under Reduced Internal Friction and Low Inertial Load.

These experiments were conducted in January 1996, after the attachment of a realistic flap to the HAHDIS Mk.1a device. Figure 14 shows the frequency response curve for this test case. The output displacement response amplitude, $\hat{u}_e$, has a moderate increase up to 10 Hz, followed by a continuous decrease.

![Compensated output displacement, $u_e^*$](image5)

![Output displacement, $u_e$](image6)

![Averaged input displacement, $u_{i,av}$](image7)

![Frequency response curves of the HAHDIS Mk.1a demonstrator in the range 1-30 Hz with reduced internal friction and low inertial from light realistic flap.](image8)

Examination of the average input displacement curve, $u_{i,av}$, shows that it is also decreasing above 10 Hz. Additional examination of the oscilloscope data during the experiment showed that the high-voltage signal provided by the model 50/750 TRENK amplifier started to become angular around 10 Hz, indicating that the upper limit of the amplifier performance was reached.
Above 10 Hz, the amplifier signal deteriorated faster, and its amplitude decreased rapidly. The decreased input voltage resulted in decreased input displacement to the device, and explains the observed decrease in output amplitude. When compensated for the loss of input displacement, the output displacement curve became flat in the region below 3-10 Hz (curve \( u_e^* \) in Figure 14). Above 10 Hz, the compensated output displacement, \( u_e^* \), presented the typical bell-shaped response of a system approaching resonance. In our case, the resonance bell is shallow and spreads over the frequency range 10-22 Hz. The amplitude at resonance is only 35% higher than the amplitude off resonance. This indicates that the system operates satisfactorily and controllable during resonance, and that resonance is actually beneficial for its frequency response. Above resonance, the system response starts to decrease, but it is still very good up to 25 Hz.

At 26 Hz, the compensated system response, \( u_e^* \), has a value which is only 10% lower than the value at 1 Hz. This experiment proved again the effectiveness of the HAHDIS principle, showing its good performance over the frequency range 1-30 Hz in spite of the input displacement decrease due to limitations in the high-voltage power supply.

**Discussion of the Three HAHDIS Experiments.**

Table 1 presents an overview of the three HAHDIS experiments indicating the maximum amplitude, the maximum frequency, and the limiting frequency due to the electrical supply.

<table>
<thead>
<tr>
<th>Case</th>
<th>Maximum amplitude, mm, p-p</th>
<th>Maximum frequency, Hz</th>
<th>Limiting frequency due to the electrical supply, Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>0.85 mm</td>
<td>20 Hz</td>
<td>5 Hz</td>
</tr>
<tr>
<td>(heavy load)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case 2</td>
<td>1.2 mm</td>
<td>&gt;30 Hz</td>
<td>10 Hz</td>
</tr>
<tr>
<td>(no load)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Case 3</td>
<td>1.25 mm</td>
<td>&gt;30 Hz</td>
<td>10 Hz</td>
</tr>
<tr>
<td>(light load)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Comparative examination of these results allows us to draw the following conclusions about the behavior of an induced-strain solid-state actuator under various load conditions when energized by a power supply with limited current delivery capabilities:

1. Under heavy load (Case 1 in Table 1), the device is limited in both amplitude and maximum frequency. The amplitude limitation can be attributed to the internal compressibility and counter electro-motive force in the active-material stacks. The maximum frequency limitation is due to the limited capability of the power supply, since the current requirements are directly proportional with the stack capacitance and inversely proportional with frequency. The current limitation was reached around 5 Hz.

2. Under no load (Case 2 in Table 1) the device presented a 41% increase in amplitude and a 50% increase in frequency range. The increase in amplitude is attributed to the reduce compressibility loss and reduced counter electro-motive force in the stacks. The increase in range is explained by the decrease in stack admittance following the decrease in the inertial loads (phase angle \( \phi = 180^0 \)). The current limitation was reached around 10 Hz.

3. Under light load, (Case 3 in Table 1) the device was proven still capable of supplying a sizable displacement over a frequency range up to 30 Hz. This infers that the HAHDIS Mk.1a proof-of-concept demonstrator has fulfilled its purpose successfully, and has proven the feasibility of using induced-strain actuators to achieve realistic amplitudes over the frequency range required for active and adaptive control.

**Conclusions**

An electro-mechanical model has been developed to represent the coupled operation of solid-state induced-strain actuators connected to flight control surfaces. Using the complex-admittance/impedance electrical representation of the induced-strain actuator, and the complex-stiffness mechanical representation of the control surface, the model was found adequate to predict power input and output during both voltage-controlled and current-controlled operations.

Preliminary experimental confirmation of the model has been achieved through 3 separate test cases (heavy load and high internal friction; no load and low internal friction; light load and low internal friction) performed with the HAHDIS Mk.1 proof-of-concept demonstrator. In spite of the current limitations of the power supply, the frequency response curves presented in this paper confirmed the theoretical predictions and proved satisfactory large-displacement operation beyond 30 Hz.

This paper has shown that the power and energy management in induced-strain actuated adaptive control systems differs considerably from that of conventional flight control systems. The conventional flight control systems consume predominantly active power in spite of driving a predominantly reactive mechanical load represented by the aerodynamic control surface. The solid-state induced-strain actuated flight control system use predominantly reactive power and, through the electro-mechanically
coupling with the aerodynamic control surface, open new opportunities for power recycling. It is conceivable that a properly tailored solid-state induced-strain-actuated control system will use an order of magnitude less power from the aircraft power supply than a conventional hydraulic system, and hence will effect considerable savings in power, energy and weight. However, new concepts for reactive power management in high-voltage high-current power amplifiers need to be developed, and high-efficiency airborne power amplifiers must be built. Such specialized high reactive-power amplifiers for solid-state induced-strain actuators should be the focus of immediate research.

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REFERENCES


