ABSTRACT

Title of Thesis: Design of an Improved Shape Memory Alloy Actuator for Rotor Blade Tracking

Kiran Singh, Master of Science, 2002

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A shape memory alloy actuated tab deflection mechanism for in-flight rotor blade tracking was designed, fabricated and tested. The design, comprises of dual SMA wire actuation device, passive lock and on-blade sensors. The system is integrated with a feedback position controller. Improvements, over the previous design, in shape memory alloy wire clamping mechanism, locking mechanism and controller operation were examined. An analytical model, incorporating Brinson's[13] thermomechanical model was developed to predict actuator behavior. A design methodology, based on this model, was applied to identify the relationship between actuator design parameters and actuator target goals. The actuation system integrated into an NACA0012 blade section was tested on the bench-top. Tracking capability of ±6° with a resolution of ±0.1° was demonstrated. Symmetric and consistent tab deflection capability was demonstrated under closed loop control.
Design of an Improved Shape Memory Alloy Actuator for Rotor Blade Tracking

by

Kiran Singh

Thesis submitted to the Faculty of the Graduate School of the University of Maryland, College Park in partial fulfillment of the requirements for the degree of Master of Science 2002

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Preface

$\sigma$  Stress( psi) \\
$x$  SMA wire extension \\
$\sigma^{cr}_s$  Twinned to detwinned martensite lower critical stress(psi) \\
$F$  Force in SMA wire(lb) \\
$\sigma^{cr}_f$  Twinned to detwinned martensite upper critical stress (psi) \\
$E_A$  Young’s Modulus in austenite( psi) \\
$\epsilon$  Strain \\
$E_M$  Young’s Modulus in martensite (psi) \\
$\epsilon_r$  Recovery strain \\
$\xi$  Martensite fraction \\
$\epsilon_l$  Maximum recoverable strain \\
$\xi_s$  Stress induced martensite fraction \\
$\theta$  Shaft rotation (degrees) \\
$\xi_T$  Temperature induced martensite fraction \\
$\tau$  Moment (in-lbs) \\
$\Omega$  Phase transformation coefficient \\
$L_o$  Length of SMA wire (in)
$r_{ht}$ radius of hinge tube (in)

$d_o$ Diameter of SMA wire (in)

$N_{wire}$ Number of SMA wire wires

Notes:

Subscript 'o' - initial value of parameter

Superscript 'A' - parameter corresponds to wire A

Superscript 'B' - parameter corresponds to wire B

Subscript 'F' - frictional parameter
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Chapter 1

INTRODUCTION

1.1 Background and Motivation

Helicopters generate vertical and forward flight by the rotation of the main rotor blades about a vertical axis. Aerodynamic forces are generated by the relative motion of the blade section with respect to the air. The rotation of these blades thereby enables the capability of forward, vertical and hovering flight that helicopters are unique for.

The rotating mechanism of generating lift however introduces the penalty of high level of vibrations. A primary cause of these vibrations is the complex flow-field that exists around these rotating blades. These vibrations are currently sought to be reduced by the introduction of vibration isolaters and passive dampers. To enable enhanced ride quality and mechanical system reliability and thereby reduced maintenance, active vibration suppression mechanisms have been pursued actively in recent years [1, 2].

Of particular discomfort and cause of fatigue of mechanisms is the existence of 1 per rev. vibrations. A dominant cause of these 1 per rev. vibrations is the existence of dissimilarities between rotor blades. Dissimilarities due to inertial,
structural or aerodynamic variations are the primary causes of high 1 per rev. vibrations. Manufacturing inconsistencies are responsible for inertial or structural variations. Usage in flight causes wear and fatigue of the blades and can exacerbate the blade dissimilarities. To reduce these variations to an extent, the blades are presently manufactured within very tight tolerance margins. Additionally, static and dynamic balancing of rotor systems is conducted to minimize the inertial differences between blades. This enables the centering of mass of the rotating blade system, thereby effecting a reduction in vibration levels.

Variation between blades influences the tracking condition of the rotor system. The main rotor blades should ideally rotate in the same plane and maintain an equal angular spacing during flight. When rotor blades go out of track, their tips rotate in different planes. An out of track rotor system typically generates large vibrations in the airframe. To track the rotors and thereby reduce vibrations, pitch links and tracking tabs are adjusted accordingly [3]. These form the primary mechanisms to compensate for blade dissimilarities.

Tracking is classified as a maintenance procedure and is required to be performed on a regular basis. The tracking operation has evolved over the years and is currently performed using a combination of vibration sensors and strobe lights or electro-optical devices. The objective is to obtain a constant tip-path plane over the azimuth. The tracking procedure typically involves carrying out the following steps:- (a) tracking of the blades with pitch links on the ground, (b) tracking blades with pitch links based on hover and forward flight track, (c) adjusting tabs based on track data or vertical vibration data in forward flight, (d) spin balancing the main rotor in hover. The operation requires a great deal of accuracy and procedures currently in use are subject to errors. To eliminate these errors the
operations are conducted repeatedly until satisfactory results are obtained. This however is time consuming and is hence a cost enhancing procedure.

To replace the current tedious manual system of tracking an on-blade tab actuating program was envisioned. The on-blade actuating system would enable deflection of individual rotor blade tabs, in response to inputs sent by a controller. This method thus acts as a relatively simple mechanism for tracking rotor blades. An on-blade tracking concept which could eliminate the need for manual tracking thus appears highly attractive and merits a close examination of feasible actuation methods and capabilities. This provides an effective motivation for the present study.

The objective of this thesis is to examine an in-flight tracking tab actuator using Shape Memory Alloys (SMA). The motivation for using smart materials in on-blade actuating systems is the advantages they offer over conventional materials and devices. These materials enable compact, light-weight, high energy density actuator designs which would be far more capable of integrating into the blade section and actuating on-blade tabs or flaps, in the challenging environment of the rotating blade. The advantage of using SMAs as the active material is their relatively high force and stroke capability. This enables the design of simple actuation mechanisms and eliminates complexities due to gear reduction or motion amplification, which is necessary when using other active materials like piezoelectrics and electrostacks. Low actuation voltages, low costs and reduced number of moving parts are additional advantages associated with these materials. SMAs are temperature dependent and hence actuate at low frequencies ($< 1Hz$). Since the tracking operation need not be conducted at high frequencies, the low bandwidth demonstrated by SMAs is adequate.
In the past, a number of tab actuator designs employing these materials have been proposed and some have been built and tested. However most are still under varied stages of development. One such design was fabricated and tested extensively at the University of Maryland [4]. The feasibility of employing SMAs was demonstrated distinctly, although some problems were encountered. The goal of the present work is to utilize the knowledge base developed here at the University of Maryland, identify the problems encountered and attempt to eliminate them in the final design.

1.2 SMA Actuators - State of the Art

1.2.1 Tab Actuator - Concept Review

The on-blade tab actuator consists fundamentally of the rotor blade tab deflecting mechanism, with the SMA as the active material. This actuator, as envisioned, would replace the current tedious, manual system of tracking. The actuation system would have the capability to respond to inputs from an external source, placed in the cockpit or elsewhere on the airframe.

The on-blade actuating mechanism would comprise of a self contained actuating unit, placed outboard on the blade section. The actuator would deflect the tracking tab of the rotor blade and thus effect a change in the tip-path plane. Each blade would have a separate tab actuator which could be controlled by a control system placed in the cockpit. The controller would respond to measurements from vibration sensors and tracking tab angle sensors. According to the feedback, corresponding actuation signals would be sent to each on-blade actuator, until a minimum level of vibrations and a consistent tip path plane could be
acquired.

The additional advantage of the introduction of this on-blade mechanism is its ability to track rotor blades during flight. This may augment individual blade control vibration reduction devices (e.g. L-L actuator [5]) or act as a stand alone one rev. vibration reduction mechanism.

Liang et al. [6] identified design parameters for integrating an actuator onto a typical blade section. These parameters were identified as ±5° of tab deflection, 4.6in – lbs of actuation moment and duty cycle of 10 cycles/hour. An additional requirement for sustained tab deflection at the set point under power off condition was recommended, in order to prevent continuous power consumption due to prolonged actuation of the device.

Kennedy et al. [7] published parameters obtained for the actuator, based on structural and environmental requirements, for operating an FAA certified aircraft. These parameters were specified for the Boeing MD900 Explorer and provided data regarding the environmental vibratory and G field loads the actuator would be subject to, in the rotating environment. The parameters from these two sources were used as a guideline and were modified according to the blade geometry adopted and external loads imposed during tests.

1.2.2 Fundamental SMA Actuator Designs

In this discussion. SMA actuators are classified according to their shape and mechanism of operation. Actuator designs reviewed, have used SMAs in the form of either rods or wires. Rods or torsion tubes, develop rotational strains and moments whereas wires develop extensional strains and stresses. Liang et al. [6] conducted a theoretical analysis of Shape Memory Alloy torsional actuators for rotor-blade
tracking operation. They examined the shape memory behavior of torsional rods and proposed a design to implement them into an actuation mechanism. This was a preliminary analytical study and no fabrication or experimental testing was conducted. Kennedy et al. [7] experimentally demonstrated the capability of actuation using SMA torsion tubes. The design comprised of two shape memory alloy torsion tubes assemblies, operating in an antagonistic manner. These were coupled together through a pair of combining gears to enable rotation in both directions. These tubes were externally heated using nichrome/ceramic heaters.

From the above two studies some important disadvantages of SMA torsion tube actuation systems, designed for the tracking tab program, become evident. One of the issues with these systems lies in their inability to actuate rapidly (within < 10 seconds). In fact these are actuated in the range of 1-5 minutes, which is a relatively long period of time for an in-flight system. Additionally, due to their large diameter to volume ratio, torsional actuators require external heating systems for heat activation. These heating and cooling elements occupy a large volumetric space. This thus prevents easy integration into the confined space in the rotor blade section.

SMA wire actuators have been examined by Giurgiutiu et al.[8]. In this paper the authors proposed a method for achieving direct in-flight incremental adjustments of the tracking tab. This was conducted by using a composite tab, consisting of SMA wires embedded into a polymeric matrix. Fabrication and testing of the active tab mechanism was carried out. A method was conceptualized, whereby a range of angular deflection of ±3.75° with an accuracy of 0.25° was achieved. This tab actuation system was to be integrated with a control scheme that would interface the angular deflection with the electric current supplied to the wires. However
there is no indication of further inroads in developing this actuator design any further.

Rediniotis et al. [9] examined the implementation of SMA wires for the purpose of deflecting a trailing edge tab, in an underwater biomimetic vehicle. This was designed such that it could behave both as a quasi-static device for static deflections and for thrust generation by oscillation of the trailing edge tab. The SMA actuator was able to deflect the trailing edge by $\pm 5^\circ$ at a maximum frequency of $2Hz$. The SMA actuation elements consisted of two sets of wires attached on either side of an elastomeric element. The hydrofoil comprised of forward and aft sections made of molded fiberglass shell to form the airfoil shape. The SMA wires were anchored using brass pins mounted on acrylic blocks on the forward and aft sections of the hydrofoil. Two way shape memory effect training was imparted to the SMA wires for bidirectional actuation. This type of training refers to the behavior of the shape memory alloy materials which after plastic deformation of the wires, recover the strain on heating [10, 11]. On cooling, the wires regain the shape originally imparted on plastic deformation, without the application of a bias load. Constant heating is required to maintain a tab position away from the neutral position. The mechanism is hence unsuitable for application to tracking of rotor blades. This is due to the requirement imposed on the tracking tab program, for sustained tab deflection under a power off condition.

Previous work conducted at the University of Maryland by Epps and Chopra [12], investigated the feasibility of tracking tab actuation using SMA wires. This capability was demonstrated by the design, fabrication and testing of such an actuation system. The design comprised of a bidirectional SMA actuator, locking device and feedback position controller. Further details of the system are examined
in the next subsection.

The indication from the literature establishes the efficiency of actuation using SMA wires and resistively heating them. This efficiency is loosely defined in terms of both relative rapidity of actuation (as compared to SMA tubes) as well as the simplicity of internal heating, in terms of associated hardware. However fabrication issues due to design and materials selected have prevented repeatable and fail-safe operation of the actuator [4]. Thus a careful selection of materials is necessary to ensure a reliable actuator design.

1.2.3 Tab Actuator Designs - Fabricated and Tested

In the literature a number of actuation systems have been proposed. However only two actuators have been both fabricated and tested either on the benchtop or under aerodynamic loading. The design presented by Kennedy et al. [7] comprised of two shape memory alloy torsion tubes assemblies, operating in an antagonistic manner. These were coupled together through a pair of combining gears to enable rotation in both directions. These tubes were externally heated using nichrome/ceramic heaters. To hasten cooling rates, passive heat sinks in the form of cooling fins were introduced. An active locking device employing SMAs as the active material was designed and incorporated into the assembly. This design was tested on the benchtop and obtained a range of tab motion of ±7.5°, output moments of 5in−lbs. with a duty cycle of 6 cycles/hr. This actuation system was however not integrated into a blade section or tested under aerodynamic loading or on the rotating frame.

Epps and Chopra[4, 12] demonstrated the feasibility of an SMA wire tab actuator. The design comprised of a dual SMA actuator, locking device and feedback position controller. A thermomechanical model proposed by Brinson[13] was ap-
plied to model the behavior of this actuator. Testing was carried out in the wind tunnel and actuation to angles of ±15° at a wind speed of 120 ft/s was demonstrated. However unequal tab angles, lack of repeatability, poor angular resolution and the inability to maintain tab position with the locking mechanism necessitated further design improvements. Section 1.4 takes a closer look at the design details and investigates plausible reasons for the inconclusive results. Improvements made to the actuation system designed by Epps [4], forms the topic of this thesis.

1.2.4 Locking Mechanism Designs Proposed

One of the requirements of the tracking tab actuator program was the need to maintain angular position in a power-off condition. This resulted in the requirement for a tab position locking device. Liang et al. [6] examined some theoretical mechanisms which could be used as a locking device. An active lock design proposed, comprised of an SMA spring, operated by a ratchet/gear and pawl locking mechanism, in which the default position would be the locked state. When the SMA is heat activated, it would release the pawl and allow the hinge tube to rotate. Another mechanism suggested was a passive lock design that prevented motion up to a designed actuation load. The mechanism comprised of a fixed elastic detent meshed between two gear teeth. For actuator moments exceeding the design value, the elastic detent would jump a tooth at a time. This actuation moment could be designed to overcome both braking and aerodynamic moment. These proposed designs provided a basis for future designs to be implemented.

An active locking device employing an SMA as the active material was designed by Kennedy et al. [7]. The braking component created an interference fit when mounted around the output shaft. On heating, the shape memory alloy liner
expanded, creating a clearance between the shaft and the lock interface. This design was successful in preventing motion upto moments of 15in – lbs. However a relatively long period was associated with activating the SMA material and therefore in unlocking the system. This was thus responsible for increasing the time associated with the tracking operation.

Epps and Chopra[12] designed and implemented a solenoid operated gear-pawl locking mechanism. The locking mechanism comprises of a solenoid operated pawl, which in the power-off condition meshes with gears that are mounted on the hinge tube. This mechanism could thus prevent deflection of the tab. When powered on, the solenoid activates the pawl and releases the rotating hinge tube. Once tracking position is achieved the solenoid is powered off and the pawl re-engages with the gear teeth, preventing further tab motion.

### 1.3 Actuator Design Goals

The parameters for designing the tracking tab actuator evolved from the expected angles and loads that were estimated to be experienced by the tracking tab, during operation on the rotor blade. Kennedy et al. [7] and Liang et al. [6] detail the tests conducted to obtain the quantitative estimate of the structural and environmental conditions the actuator must operate in. The goals that have evolved from the two studies are summarized in Table 1.1 and are described as follows.

The actuator should conform to a weight of less than 1 lb., tab motion of ±5°, resolution of ±0.1°, output and braking loads (due to aerodynamic moments) of 4.0 in-lbs., and a duty cycle of 20 cycles/hr. The mechanism must be capable of meeting geometric design requirements imposed by space limitations of the blade profile. These were established to be a maximum height of 1.4 in. at the quarter
chord section and 0.8 in. at the location of the hinge tube. The system must be capable of withstanding aerodynamic and rotating frame loads expected to be encountered near the 75% radius of the blade. Temperature, force(moment) and position(angle) sensors need to be located on the tab assembly, providing feedback to a position control mechanism. Additionally, sustained tab deflection under a power off condition is required.

These parameters provide a set of guidelines for the design detailed in this thesis. In Chapter 5 a review of the actuator results and a comparison with the target goals is conducted.

1.4 Scope of Present Research

This thesis presents the design, fabrication and testing of an improved Shape Memory Alloy actuated tracking tab device. A significant amount of research has been conducted in utilizing SMA materials for tracking tab actuation. However the program objectives recommended in Liang et al. [6] (outlined in Table 1.1) have not yet been met conclusively.

The work conducted by Epps [4], in incorporating SMA wires as the active material for rotor blade tracking indicated the feasibility of such an actuation system. However fabrication issues due to design and materials selected prevented repeatable and fail-safe operation of the actuation system. To eliminate the problems encountered a close examination of the previous design [4, 12] was conducted. A brief description of the design of this actuator, the tests conducted with a summary of identified areas of improvement follow.
1.4.1 Initial Actuator Design Overview

A blade section model (Figure 1.1) with an integrated SMA tracking tab actuation system was built. The actuator, embedded in a NACA-0012 blade section, consists of two sets of SMA wires, an upper and a lower set. A schematic of operation is shown in Figure 1.2. The SMA wires are initially plastically strained. One end of the wires is fixed at the blade spar and the other end attaches to a hinge tube that is allowed to rotate. The tracking tab is rigidly attached to the hinge tube. Each set of wires is electrically and thermally isolated from each other such that when the upper wires are heat activated, the lower set remain at ambient conditions (and vice versa). When the wires are heat activated they contract, recovering a portion of the initial plastic strain. As the wires contract, a recovery force develops that causes the hinge tube to rotate upwards. To deflect the tab back down, the lower set of wires are heat activated, while the upper set remain at room temperature.

A solenoid operated gear-pawl locking mechanism was designed to enable a power-off hold condition. The lock was introduced to constrain tab movement from the desired position. A position controller was designed to supply voltage to the braking device and the corresponding SMA wires, in order to acquire the commanded tab position. Control voltage was switched off once target acquisition occurred.

An analytical model was developed to model the operation of the tab actuator. It was recognized that the maximum stress that develops in the heated wire is dependent on the load-displacement properties of the non-heated wire. Experimental measurements of the recovery force and displacement recorded was then correlated with the predictions. The thermo-mechanical characteristics of SMAs were thus implemented to develop the theoretical actuator model.
The actuator was tested both on the bench-top and in the open-jet wind tunnel. Wind tunnel tests were carried out to determine the actuator performance under varying aerodynamic loads.

1.4.2 Drawbacks Identified

The results obtained from the actuator discussed above, demonstrated the feasibility of the dual SMA wire actuator concept. Significant operational inadequacies were however observed. Elimination of these inadequacies forms the basis for an improved actuator design, which is the subject of the current work. The improvements required are enumerated in this section.

- The method of gripping the wires was found to be ineffective and resulted in slippage of wires. Consequentially unequal tab deflection angles in the tab up and down orientations was observed. This behavior was notable as the loads on the system increased at higher wind speeds.

- Stress concentration at points of attachment of the wires damaged the fiberglass hinge tube. This resulted in a loss of tab actuation angle during testing in the wind tunnel.

- Due to the presence of multiple wires, difficulties were encountered in applying an equal amount of prestrain to all the wires.

- The locking mechanism was found to be unreliable in holding the tab position. This was due to a combination of factors, primarily due to backlash in the gears and a low engaging force of the solenoid. Discretization introduced by the gear-pawl system of the lock resulted in a relatively low angular resolution (0.5°).
• **Closed loop response** of the actuator was also inconsistent. One issue identified was the abrupt step up in actuation voltage from 0 to 5 Volts, as commanded by the on-off controller. This resulted in a high rate of heating of the actuated wire and therefore a higher strain rate of the wire at room temperature. This has been identified as non-ideal from the point of view of shape memory alloy material behavior [14]. Additionally, an anomaly present in the control circuit resulted in a drift in the commanded tab angle value. The drift was of the order of the desired accuracy required of the system. This thus lead to inferior tracking results.

These drawbacks identified were considered responsible for hindering reliable tracking capability of the actuation system as designed by Epps, [4]. The objective of this thesis is to eliminate these drawbacks to ensure consistent tracking capability.

Details of the actuating elements and operating procedure, theoretical model designed, control schemes implemented and finally a discussion of the tests conducted and results obtained are presented in the chapters that follow.

### 1.5 Outline Of Thesis

This thesis is divided into five chapters. In the first chapter the state of the art of shape memory alloy actuation systems and the motivation for these systems is discussed. The first prototype tracking tab actuator is examined and the problems encountered are enumerated.

The second chapter examines the basic principle of actuation using SMA wires. It then summarizes the actuation procedure for the improved bidirectional SMA
tracking tab (BSTT) actuator. The theoretical model for predicting actuator operation is developed and discussed. This model was then used to perform parametric studies to evolve a final set of actuator parameters that could fit the applied constraints. An evaluation of the constraints placed on the on-blade actuator is discussed as well.

The third chapter provides details on the actuation subsystems and their design and fabrication. The operation details and integration into the blade section are discussed. Additionally, control system design details are provided and the integration with the actuator is discussed.

The fourth chapter presents results of analysis and testing conducted on the benchtop and in the open-jet wind tunnel. The analytical results from the parametric studies are compared with the open loop test results conducted on the benchtop. The correlation of experimental results with the theoretical model is evaluated. The parameters selected are installed on the actuation system and closed loop tracking operation is tested on the benchtop, where aerodynamic loading conditions are simulated. Finally the actuator was tested in the open jet wind tunnel.

The fifth chapter concludes with a comparison of target goals and the corresponding achievements of the present system. The improvements in the BSTT actuator are then compared with Epps’ design. Additional design improvements are also suggested for the purpose of integration into a full scale rotor blade.
Table 1.1: Tracking tab actuator goals

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<th>Goals</th>
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<tr>
<td>Actuator stroke</td>
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</tr>
<tr>
<td>Resolution</td>
<td>±0.1°</td>
</tr>
<tr>
<td>Braking Moment</td>
<td>4.0 in-lb</td>
</tr>
<tr>
<td>Actuator weight</td>
<td>&lt; 1 lb</td>
</tr>
<tr>
<td>Actuator dimensions</td>
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</tr>
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</table>
Figure 1.1: Tracking Tab Actuator - Initial Design

Figure 1.2: Schematic of Tracking Tab Actuator Embedded in Blade Section
Chapter 2

MATERIAL BEHAVIOR AND THEORETICAL MODELING

Shape Memory Alloys are a class of materials that on apparent plastic deformation at room temperature, will return to their original shape when the temperature is raised. The material thus appears to memorize its original shape prior to deformation. At high temperature the material is in its parent phase (austenite) and the shape it acquires is termed as its original shape. In the absence of an external load, when the material is cooled, it retains its original shape. When a load is applied beyond the elastic limit, the material will plastically deform. Deformations up to 6 – 8% of the original length are completely recoverable on heating and the initial dimensions are acquired. This temperature dependent behavior can be exploited to design an actuation mechanism.

This chapter examines the behavior of SMAs in response to heat and external mechanical constraints. An attempt is made to develop a theoretical model that incorporates the thermomechanical behavior of the SMA into the actuator analytical model. A description of the basis of the analytical model is presented and the thermomechanical model used is discussed. The application of this model to
The shape memory effect is shown pictorially in Figure 2.1. Here the SMA wire is shown to be in its high temperature austenite phase, in which its length is denoted as $L_o$. On cooling this wire, it transforms into the low temperature martensite phase. On the application of stress in this low temperature state, the wire can elongate up to $6 - 8\%$ of the original length. On removal of this stress, the wire maintains this deformed length, denoted by $L_o + \epsilon L_o$, where $\epsilon$ is the strain imparted. However, when heated back to austenite, the wire regains its original length, $L_o$. The recovery of plastic deformation on heat activation is termed as the shape memory effect.

This behavior of shape memory alloys is governed by a phase transformation between a high temperature parent phase (austenite) and low temperature product phase (martensite). This phase transformation is associated with a change in the crystal lattice structure of the material [10, 15, 16]. On a microscopic scale this behavior of SMAs is due to a unique change in crystal structure. At high temperature the material exists in the austenite phase, which has a face centered cubic (FCC) lattice structure. In the room temperature martensite phase, the lattice structure changes to a body centered cubic lattice (BCC). On cooling from high temperature austenite, the martensitic BCC crystals start to nucleate and there exists a range of temperatures in which both martensite and austenite coexist. To accommodate for the change in crystal structure between martensite and austenite, twinning results. This behavior of twinning forms the key to the shape memory effect displayed by SMA materials like Nitinol (NiTi). A brief description
of twinning follows.

When the SMA specimen is cooled from its high temperature austenite phase to room temperature, martensite is formed. The crystal structure developed in the absence of stress can form any combination of multiple twins or martensite variants (twenty four of which have been identified). These twins are essentially a pair of equivalent crystals which are identical mirror images of each other. These crystallographic mirror images are formed about a plane called the twin boundary.

On the application of external stress these twin boundaries will move, resulting in a shape that is capable of accommodating the applied stress. This application of stress results in the formation of twin variants in the energetically preferential orientation, and is termed as detwinning of the twinned martensite crystal structure. On the application of sufficient stress, complete detwinning results. Detwinning is hence essentially the formation of a single stress preferred martensitic twin crystal variant.

The movement of the twin boundaries during detwinning requires a relatively low input of energy (or stress). This phenomenon is hence associated with a large change in stress and low change in strain. Detwinning is thus associated with large deformation strains (6-8%). Upon heating, the detwinned structure transforms to austenite. This transformation is associated with a reversal in the crystal structure. The originally deformed rhombic crystal structure now forms the square shape of the face centered cubic lattice. Notable in this change is the reduction in principal dimension (length for wires). The wires thus regain the initial length $L_o$. This change in lattice structure is responsible for the large strain recovery associated with the shape memory effect. The interplay of stress and temperature and its effect on the phase and crystallographic shape of the shape memory alloy material
A noteworthy point is that the high temperature phase has only one single crystal structure. This difference in behavior of crystal structures lies in the differing nature of the forward (Austenite $\rightarrow$ Martensite) and reverse (Martensite $\rightarrow$ Austenite) reactions. Martensitic transformations are exothermic reactions, hence are associated with a release of energy. In contrast Austenitic transformation, are endothermic. Thus in austenite, regardless of the stress applied, the material is in its high energy state.

During transformation, there is a range of temperatures in which both austenite and martensite coexist. The start and finish temperatures of the range are termed as the characteristic temperatures of the material. Forward transformation from austenite to martensite is defined by martensite start, $M_s$ and martensite finish, $M_f$ temperatures respectively. Similarly for the reverse transformation from martensite to austenite, $A_s$ (austenite start) and $A_f$ (austenite finish) define the characteristic temperatures. It is a well documented observation that these transformation temperatures increase with applied stress (Refer to Figure ??).

Of significance is the temperature dependent hysteretic change in physical properties of the material between austenite and martensite phases. Values of Young’s modulus and resistivity of the material vary significantly between these two phases. Changes in physical properties, with temperature are depicted graphically in Figure ??.

The most notable difference in physical property is the yield strength. This difference is due to the low amount of energy required to move twin boundaries in martensite, during detwinning. In the austenite phase however, no such detwinning results and the yield strength is comparatively higher. It is worth noting that on complete detwinning of the material in the martensite phase, further application
of stress will result in a region of elastic deformation. This will eventually lead to a second yield point and plastic deformation region, which is however irrecoverable due to the breaking of atomic bonds (Figure ??).

2.1.1 Effect of Phase Changes Represented on the Stress-Strain Plot

On a macroscopic scale, changes in crystal structure can be represented by examining the stress-strain plot. Figure ?? superimposes the stress-strain profile for both austenite and martensite. The low temperature loading curve is characterized by the elastic region, and the recoverable plastic deformation region. The unloading curve indicates the recovery of only elastic deformation at room temperature. The plastic deformation is recovered when the material is heated to austenite.

The high temperature austenite stress-strain curve indicates a higher Young’s modulus and yield stress, as compared to that in martensite. The application of stress at high temperature results in the formation of stress induced martensite (SIM). On unloading however this SIM transforms back to austenite. The shape of this curve indicates the pseudoelastic behavior of shape memory alloys at high temperature. This variation of stress with temperature is depicted on the stress-temperature curve in Figure ??.

2.2 Influence Of Constraints

The influence of varying the constraints applied at the ends of the wires is examined in this section. The type of constraints applied to the ends of a shape memory alloy wire influence the material behavior upon heat activation. The wires in this
discussion are first plastically deformed at room temperature, by an amount within the completely recoverable region. Depending upon the constraint, a certain fraction of the imparted pre-strain is recovered and a corresponding recovery force is developed. Figures 2.2-2.5 pictorially display these different boundary conditions, while Figure 2.6 demonstrates the influence of these varying constraints on the SMA wire stress-strain plot.

Figure 2.2 shows the pre-strained wires that are rigidly constrained to prevent any strain recovery. This is indicated as Case 1 on the stress-strain plot in Figure 2.6. This wire, when heat activated, recovers none of its plastic deformation. It however generates a large recovery force [4].

When the constraints on these wires are removed the pre-strained wire is able to recover all the plastic deformation imparted to it, while no force is generated. This condition is the free recovery case and is indicated by Figure 2.3 and by Case 4 on Figure 2.6 [17].

For the purpose of designing an actuator, both stroke and force are required to be output by the actuation system. We thus need to identify the boundary conditions that allow operation between these two extreme conditions. Figures 2.4 and 2.5 depict two such boundary conditions that enable imposition of partial restraint on the material.

When the actuator is attached to a linear spring and then heat activated, as shown in Figure 2.4, the stress-strain profile of the heat activated SMA wire is as shown in Case 2 on Figure 2.6 [18]. Here the magnitude of recovery stress induced is dependent on the spring constant $K$, and only a certain fraction of the plastic deformation is recovered with a corresponding recovery force, $F_K$. When this spring is replaced by an SMA wire (which acts as a non-linear spring, Figure...
2.5) a varying amount of force and stroke is recovered. This is indicated by Case 3 in Figure 2.6 [19]. The biggest advantage of using an SMA wire instead of a linear spring is realised in the capability this system offers in terms of bidirectional actuation. Heat activation of one wire enables recovery of a fraction of its prestrain. The opposing wire simultaneously extends by an equal amount. When the heat supplied to the wires is interchanged, the behavior is then reversed. This simple antagonistic mechanism forms the principle for actuating a tracking tab.

2.3 BSTT Actuator Operating Principle

The principle of operation of the bidirectional SMA tracking tab (BSTT) actuator is explained schematically in Figure 2.7. The SMA wires are clamped at one end and connected to the tracking tab via a hinge tube that is free to rotate. The wires are electrically and thermally isolated from each other. To enable actuation of the tab upon heat activation, the wires must first be prestrained (plastically deformed at room temperature). To achieve equal actuation in the positive and negative orientations, the wires are prestrained by an equal amount (Figure 2.7(a)).

When the upper wires are heated, they attempt to acquire their original length due to the shape memory effect (Figure 2.7(b)). This behavior is partially restrained due to the opposing wires, which are maintained at room temperature. The force-displacement profile followed by the heated wire is shown by Case 3 in Figure 2.6. This actuation force produces a counter clockwise moment about the hinge tube axis, resulting in the upward deflection of the tracking tab. The heated wire thus regains a certain fraction of its prestrain. The lower wires get plastically deformed by an amount equal to the contraction of the heated wires. As the wire is elongated, its load-displacement characteristics determine the force developed by
the system and its stress and strain state follow the room temperature martensite curve in Figure 2.6.

To deflect the tab in the opposite direction, the lower wires must be heated with the upper wires maintained at room temperature. The reciprocating motion of the wires is transmitted to a corresponding deflection of the tab. The relationship is given by

$$\theta = \left( \frac{L_0 \epsilon_r}{r_{ht}} \right)$$  (2.1)

where the symbols are defined in the Nomenclature section.

### 2.4 Development of The Analytical Model

The analytical model was developed to model the thermomechanical response of the SMA wires in response to applied heat and stress, within the constraints of the mechanical actuation system. The objective of developing the model was to predict the behavior of this actuator on heat activation of the wires. A discussion on the development of this model is presented in this section.

#### 2.4.1 Actuator Concept Overview

A brief description of the actuation mechanism based on the principle discussed in Section 2.3 is provided in this subsection. A detailed description of the actuator subsystems and the controller are provided in Chapter 3. This subsection highlights important features for the purpose of developing the understanding of the analytical model.

A single continuous shape memory alloy wire with several segments was clamped
down between stainless steel plates. Two sets of these wires and clamp arrangement comprise the upper and lower elements of the SMA actuator. The clamps, at one end of these wires, are fixed at the spar. They attach to the hinge tube at the other end in order to actuate the trailing edge tab. For the purpose of symmetry, both sets of wires have an identical initial length \((L_0)\) and diameter \((d_0)\). Furthermore, an equal amount of plastic deformation is applied to the wires \((\epsilon_0)\).

A passive friction brake was introduced to enable angular position hold. This brake applies a constant braking torque at all times. The complete actuation and brake system were integrated with a position controller to control tab position in response to an input angle.

### 2.4.2 Description of the Mechanics Of Actuation

This section explains the mechanism of actuation for the bidirectional SMA tracking tab (BSTT) actuator design. The basic principle of actuation is explained in the previous section. The objective of this section is to explain the material behavior on heat activation and its influence on the output force and displacement of the tab actuator. For the sake of brevity, the upper (heated) wires are referred to as wire A and the lower wires, kept at room temperature (for this discussion) are referred to as wire B.

The wires are first prestrained, prior to actuation. The prestraining method specifies the initial conditions for the SMA wires (and hence the model). The method is as follows: first the rotation of the hinge tube is restrained. Both wires are then prestrained by equal amounts. The restraint on the hinge tube is then removed.

When wire A is heated, it is subject to two constraints. The initial constraint it
encounters is due to the passive friction brake. This imposes a rigid constraint and thus develops a recovery force, while simultaneously preventing strain recovery. Due to the braking component, stress in wire A is not transmitted to wire B. Thus there exists a difference in the forces in the two sets of wires as shown in Figure ??.

On overcoming the frictional force, wire A now encounters the second constraint; that imposed by the non heated SMA wire B (Figure ??). Under this constraint, as shown in Section 2.2, there will be both strain recovery and stress developed. Due to displacement compatibility, the contraction (strain recovery) of wire A on heating, will be the same as the extension as wire B. If wire B is loaded at a quasi-static rate the transformation stress will progress along the room temperature martensite curve. Simultaneously, the recovery path of the heated wires will be a mirror image of this martensite loading curve, but offset by the frictional component due to the brake. Thus although the strain recovery path followed by the two wires is equal and opposite, the offset in stresses exists due to the frictional force. The influence of the friction brake can be examined by comparing the stress-strain plots with and without brake friction. (Figures 2.8 and 2.9). On removal of the friction due to the brake, the load-displacement behavior of the two wires becomes exactly equal and opposite, about the prestrain position, ($\epsilon_o$).

In Figure 2.9 the combined behavior of both wires is shown on the martensite stress-strain curve. The assumption made in the model is that the rotation of the hinge tube is quasi-steady and therefore the difference between static and kinetic friction is neglected in the brake. The dashed line indicates the prestrain applied to both wires, with the arrows designating the path. Both wires are prestrained to 2.5% and have a corresponding initial stress value. On heat activation, wire A
first undergoes constrained recovery (along path $P'P$), where no strain is recovered but a distinct stress is developed in the heat activated wire. In this state, wire B does not develop any stress (and remains at point $P'$ on the stress-strain curve). Once the stress due to the brake is overcome, wire A starts strain recovery and deforms wire B, by an equal amount. Thus the wire A takes the path to the left ($P'Q$) where strain is decreasing. Wire B progresses along the right ($P'Q'$) with increasing strain. The strain recovered by the heated wire is equal to the elongation of the non-heated wire, where $\epsilon_r$ is the strain recovered. However, a constant difference in stress remains due to the frictional component $\sigma_F$.

Once the wire A has completely transformed to austenite, no more changes in stress or strain will be observed. On cooling, the stiffness of the previously heated wire will drop. This is due to reverse transformation from austenite to room temperature martensite phase. Furthermore, due to the relaxation in tension, elastic unloading of the wire B occurs. The combined effect of reduction in tension in wire A and the elastic unloading of wire B, exerts an oppositely actuating moment on the hinge tube. If this moment is greater than the brake frictional moment, the tab will tend to rotate in the reverse direction and there will be a corresponding loss in tab angle.

The observed loss in angle will decrease as the frictional force imposed by the passive brake is increased. Hence, a more effective passive lock will be one with a greater frictional component. However, as the frictional component increases, more energy will be utilized to overcome this static moment and therefore there will be a reduction in actuation angle. Hence for given actuator dimensions, a parametric analysis needs to be undertaken to define the optimal angle and braking moment. The parametric studies and their results have been discussed in Section 2.6. The
mathematical equations modeling the mechanics of motion are derived in the next section.

## 2.5 Actuator Model Equations

In this section the derivation of the mathematical equations is presented. These equations model the actuator behavior as discussed in the previous section. The mathematical model developed in this section, is applied subsequently in the parametric studies, from which actuator design data is extracted.

Once again, for the sake of brevity, the heated wire is referred to as wire A and the wire at room temperature as wire B. For an explanation of symbols refer to the Nomenclature section. The values of the material properties required for the model are defined in Table 2.2.

For analysis, the equations of motion are coupled with the transformation kinetic equations based on a thermomechanical model of the SMA as described in the following subsection.

### 2.5.1 SMA Thermomechanical Model

A number of SMA phenomenological models have been proposed in the literature. Most commonly used are Tanaka’s [20], Liang and Rogers’ [21], Boyd and Lagoudas’ [22] and Brinson’s [13] models. Prahlad and Chopra [23] and Epps [4] conducted a comparative study of these models. Based on the results of their analyses, Brinson’s model was selected to predict the thermomechanics of the shape memory alloy material. This model was applied in the analytical model for the BSTT actuator. The following sections provide the system of equations applied in the analysis to
model the transformation behavior of SMAs, under the influence of stress and temperature.

Brinson[13] developed a phenomenological model to simulate the behavior of SMAs under the combined influence of temperature ($T$), stress ($\sigma$) and strain ($\varepsilon$). The constitutive law is developed for a one-dimensional specimen relating the material parameters to the martensite volume fraction $\xi$. This model distinguishes between twinned ($\xi_T$) and detwinned ($\xi_S$) martensite in the material, with the total martensite fraction given by

$$\xi = \xi_S + \xi_T \quad (2.2)$$

These two parameters are applied to define the Young’s modulus of the material, which is defined as

$$E(\xi) = E_A + (E_M - E_A)\xi \quad (2.3)$$

These relations are combined with equations defining the transformation behavior of the material as a function of stress and temperature.

1. **Conversion to detwinned martensite**

   for $T > M_s$ and $\sigma^{cr}_s + C_M(T - M_s) < \sigma < \sigma^{cr}_f + C_M(T - M_s)$ :

   $$\xi_S = \frac{1 - \xi_{SO}}{2} \cos \left\{ \frac{\pi}{\sigma^{cr}_s - \sigma^{cr}_f} \left[ \sigma - \sigma^{cr}_f - C_M(T - M_s) \right] \right\} + \frac{1 + \xi_{SO}}{2} \quad (2.4)$$

   $$\xi_T = \xi_{TO} - \frac{\xi_{TO}}{1 - \xi_{SO}} (\xi_S - \xi_{SO}) \quad (2.5)$$

   for $T < M_s$ and $\sigma^{cr}_s < \sigma < \sigma^{cr}_f$ :

   $$\xi_S = \frac{1 - \xi_{SO}}{2} \cos \left\{ \frac{\pi}{\sigma^{cr}_s - \sigma^{cr}_f} (\sigma - \sigma^{cr}_f) \right\} + \frac{1 + \xi_{SO}}{2} \quad (2.6)$$

   $$\xi_T = \xi_{TO} - \frac{\xi_{TO}}{1 - \xi_{SO}} (\xi_S - \xi_{SO}) + \Delta_T \xi \quad (2.7)$$

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where, if \( M_f < T < M_s \) and \( T < T_o \),
\[
\Delta_{r\xi} = \frac{1 - \xi_s(T)}{2} \{ \cos \left[ a_M(T - M_f) \right] + 1 \} ,
\]
(2.8)
else, \( \Delta_{r\xi} = 0 \).

2. Conversion to austenite

for \( T > A_s \), and \( C_A(T - A_f) < \sigma < C_A(T - A_s) \):
\[
\xi = \frac{\xi_o}{2} \left\{ \cos \left[ a_A(T - A_s - \frac{\sigma}{C_A}) \right] + 1 \right\} ,
\]
(2.9)
\[
\xi_s = \xi_s o - \frac{\xi_{so}}{\xi_o} (\xi_o - \xi) ,
\]
(2.10)
\[
\xi_T = \xi_T o - \frac{\xi_{To}}{\xi_o} (\xi_o - \xi) .
\]
(2.11)

The parameters \( a_M \) and \( a_A \) are defined by
\[
a_M = \frac{\pi}{M_s - M_f} , \quad a_A = \frac{\pi}{A_f - A_s}
\]
(2.13)

The values of the material properties required for the model were obtained from material characterization procedures (performed in the lab by Epps[4]) and are defined in Table 2.2.

2.5.2 Integrated Brake Actuator Model Equations

In this subsection, the equations derived represent the actuator behavior discussed in Section 2.4.2.

The constitutive equations for wires A and B are:
\[
\sigma^A - \sigma^A_o = E(\xi^A)\epsilon^A - E(\xi^A_o)\epsilon^A_o + \Omega(\xi^A)\xi^A_s - \Omega(\xi^A_o)\xi^A_{so}
\]
\[
\sigma^B - \sigma^B_o = E(\xi^B)\epsilon^B - E(\xi^B_o)\epsilon^B_o + \Omega(\xi^B)\xi^B_s - \Omega(\xi^B_o)\xi^B_{so}
\]

where
\[
\Omega(\xi) = -\epsilon _L E(\xi)
\]

The stress and strain compatibility conditions define the states of the system during actuation. For wire A and wire B these are given as:

\[
x^A = -x^B; \quad F^A = F^B + F_p
\]

1. The prestraining method specifies the initial conditions of the SMA wires, prior to heating. This is illustrated on the stress-strain curve in Figure 2.9 as the dotted line \((XP')\). Both wires are prestrained equally to ensure symmetric operation. The initial conditions are defined as:

\[
\epsilon^A_o = \epsilon^B_o = \epsilon_o
\]
\[
\sigma^A_o = \sigma^B_o = \sigma(\epsilon_o)
\]

The following approximation is made to define the initial volume fraction of the material,

\[
\xi_{so} = \frac{\epsilon_o}{\epsilon_L}
\]

With this approximation it is possible to calculate the initial stress from Eqn. 2.4

\[
\sigma(\epsilon_o) = \frac{\cos^{-1}\left(\frac{\epsilon_{so}}{\epsilon_L} - 1\right)}{\pi} \left[\sigma_s^{cr} - \sigma_f^{cr}\right] + \sigma_f^{cr}
\]

where \(\epsilon_o\) is the initial prestrain.
2. Wire A is heated ($M \rightarrow A$ transformation) and undergoes constrained recovery until the stress in A overcomes the stress corresponding to the frictional moment (path $P'P$ in Figure 2.9). In this state there is no change in stress or strain of wire B (point $P'$) since $[\sigma^A - \sigma^A_o < \sigma_F]$. 

The stress for the system is given by

$$\sigma^A = \sigma^A_o + E(\xi^A)\epsilon^A - E(\xi^A_o)\epsilon^A_o + \Omega(\xi^A)\xi^A - \Omega(\xi^A_o)\xi^A_o$$

$$\sigma^B = \sigma^B_o$$

while the strain does not vary for either wire and

$$\epsilon^A = \epsilon^A_o$$

$$\epsilon^B = \epsilon^B_o$$

This results in a net zero rotation of the hinge tube.

$$\theta = 0$$

3. When subsequent heating causes stress in wire A to overcome frictional force, stress in wire B is developed and the following condition applies

$$\sigma^A = \sigma_F + \sigma^B$$

The stresses in the two wires are then given by

$$\sigma^A = \sigma^A_o + E(\xi^A)\epsilon^A - E(\xi^A_o)\epsilon^A_o + \Omega(\xi^A)\xi^A - \Omega(\xi^A_o)\xi^A_o$$

$$\sigma^B = \sigma^B_o + E(\xi^B)\epsilon^B - E(\xi^B_o)\epsilon^B_o + \Omega(\xi^B)\xi^B - \Omega(\xi^B_o)\xi^B_o$$

$$= \sigma^A - \sigma_F$$

with the strains symmetric with respect to the prestrain $\epsilon_o$

$$\epsilon^B = \epsilon^B_o + \epsilon_r$$

$$\epsilon^A = \epsilon^A_o - \epsilon_r$$

$$\xi^B = 1$$
The transformation kinetic equations define the martensite fraction for the two wires. Newton-Raphson’s iterative technique is applied to solve Equations 2.25-2.28 simultaneously with the transformation equations, to obtain the thermomechanical parameters for the system.

The moment at the shaft in the counter clockwise direction is

\[ \tau = (\sigma^A - \sigma_0^A) \frac{\pi d_o^2}{4} N_{\text{wire}} r_{ht} \]  

(2.29)

The strain recovered corresponds to a rotation of the shaft, the angle of which is given by Equation 2.1

Figure 2.10 schematically illustrates the contribution of internal moment from each of the two wires. As indicated, wire A develops a higher stress than wire B. This is manifest in the form of a corresponding moment with the difference in these moments equal to the frictional braking moment component introduced by the brake at the hinge tube.

Design iterations, of the model proposed in this section, were conducted to obtain the optimal range of rotation angle and braking moment. The parametric studies and model correlation with experimental results are discussed in the next section.

### 2.6 Design Parametric Studies

The section examines the actuator parameters that are available from the perspective of designing the actuator, with applied constraints. The parameters evaluated were first introduced into the model developed in the previous section. The model was then iterated in order to satisfy the constraints placed on the system. The
objective of the study was to obtain a final set of parameters that enable optimal results under the guidelines of the tracking tab program.

2.6.1 Objective of the Study

The tracking tab actuator is designed to (a) generate a certain tab deflection at (b) a corresponding actuation moment. The design of this actuator comprises of a passive friction brake in combination with the SMA actuating device. Thus to achieve the required deflection angle at a given wind speed, this actuator/passive-brake combination must satisfy the following objectives:

1. Prevent changes in tab angle position up to a design maximum aerodynamic loading

2. Actuate to required angles under this maximum aerodynamic loading

There is a trade-off between the available angular deflection and actuation moment. The next section identifies the parameters which affect the values of the angular deflections and moments.

2.6.2 Actuator Parameters

To achieve the above outlined objectives, an analysis of actuator parameters must be conducted. These parameters are identified as those that directly influence either angular rotation or actuation moment. To analyze the influence of actuator parameters on actuator output (tab deflection range and braking moment), Equations 2.1 and 2.29 are referred to.

The length of wire ($L_o$), radius of hinge tube ($r_{ht}$), maximum recoverable strain ($\epsilon_r$) directly influence the angular deflection ($\theta$). The maximum recoverable strain
(\epsilon_r) is in turn a function of the prestrain imparted to the wires (\epsilon_o). The parameters affecting the actuation moment (\tau) are the diameter of the wires, (d_o), radius of hinge tube, (r_{ht}) and the number of wire segments (N_{wire}).

The next section examines the constraints placed on these above defined parameters. The parameters are then introduced into the model developed in the previous sections and are iterated within the constraints. The final values of the parameters selected are those that achieve the target objectives as closely as possible.

2.6.3 Evaluation of Constraints/Output Requirements

Program Requirements

The tab actuator program requirements as identified by Liang et al. [6] and Kennedy et al. [7] impose requirements on the actuator (a) tab angular deflection angles and (b) actuation moments. The tab angular deflections are based on data available from records of current tracking operations. Actuation moment data was obtained from measurements from wind tunnel tests and modified to the present design. The design requirements imposed on this actuator are defined to be: braking moments of 4 in-lbs. at tab deflections of \pm 5^\circ. These are the expected hinge moments the actuator would need to overcome in a rotating environment. These requirements serve as a benchmark, for the actuator to achieve.

Aerodynamic Requirements

In this thesis, the actuator is to be tested solely under aerodynamic loading in the Open-jet wind tunnel. To estimate the trailing-edge tab hinge moments due to
aerodynamic loading, an aerodynamic model assuming quasi-static deflections was adopted [5, 24]. From the calculated hinge moments, the actuator stroke/force capability can be calculated for the design configuration employed.

The flap hinge moment as a function of tab geometry and deflection angle can be obtained by using the methodology employed in Abbott and Doenhoff [25]. The deflection is considered as a function of the change in local effective angle of attack due to deflected flap, over a portion of the blade with a flap.

For a plain flap, the lift per unit length is defined as

$$l = \frac{1}{2} \rho V^2 c L_n \left( \alpha_o + \frac{\Delta \alpha}{\Delta \delta} \right)$$  \hspace{1cm} (2.30)

where $\alpha_o$ is the global angle of attack on the blade section. $\frac{\Delta \alpha}{\Delta \delta}$ is the local effective angle of attack which is obtained as a function of ratio of flap chord to blade chord, $(\frac{c_f}{c})$. The polynomial curve fit derived in [24] has been used here and is given as

$$\frac{\Delta \alpha}{\Delta \delta} = -0.002192 + 2.669 \left( \frac{c_f}{c} \right) - 2.323 \left( \frac{c_f}{c} \right)^2$$  \hspace{1cm} (2.31)

The flap hinge moment coefficient per unit span is determined by using the relation

$$C_h = \frac{dC_h}{dC_l} C_l + \frac{dC_h}{d\delta} \delta$$ \hspace{1cm} (2.32)

where

$$C_h = \frac{h}{\frac{1}{2} \rho V^2 C_l^2}$$ \hspace{1cm} (2.33)

$\frac{dC_h}{dC_l}$ and $\frac{dC_h}{d\delta}$ are obtained as a function of $(\frac{c_f}{c})$ [25], with the polynomial curve fits given by

$$\frac{dC_h}{dC_l} = -0.01018 - 0.5494 \left( \frac{c_f}{c} \right) + 1.028 \left( \frac{c_f}{c} \right)^2 - 0.9934 \left( \frac{c_f}{c} \right)^3 + 0.2770 \left( \frac{c_f}{c} \right)^3$$  \hspace{1cm} (2.34)

$$\frac{dC_h}{d\delta} = -0.8469 + 0.9833 \left( \frac{c_f}{c} \right) - 0.07663 \left( \frac{c_f}{c} \right)^2 + 0.2567 \left( \frac{c_f}{c} \right)^3 - 0.3205 \left( \frac{c_f}{c} \right)^4$$  \hspace{1cm} (2.35)
The hinge moment per unit span, h, can then be obtained as

\[ h = \frac{1}{2} \rho V^2 c_f^2 \left[ C_{l,\alpha} \frac{dC_h}{dC_l} \left( \alpha_o + \Delta \alpha \frac{\Delta \delta}{\Delta \delta} \right) + \frac{dC_h}{d\delta} \right] \tag{2.36} \]

The total hinge moment for a simple plain blade section is given by:

\[ H = \frac{1}{2} \rho V^2 c_f^2 L_f \left[ C_{l,\alpha} \frac{dC_h}{dC_l} \left( \alpha_o + \Delta \alpha \frac{\Delta \delta}{\Delta \delta} \right) + \frac{dC_h}{d\delta} \right] \tag{2.37} \]

Equation 2.37 represents a steady hinge moment necessary to maintain a given tab deflection angle. This quasi-steady analysis is sufficient for this application as tab deflections occur at a slow rate of less than 1 Hz.

**Geometric Constraints**

The tab actuator must be able to deflect the tracking tab while completely embedded in the blade section. This thus represents a finite constraint on actuator dimensions. A NACA-0012 blade profile with a chord and span of 12in was employed. The hinge tube is situated at the 72% chord location while the spar is embedded into the blade at the 10% to 30% chordwise location. The spar comprises of a 0.25 in. thick flat rectangular aluminum piece. The available geometric space spans from 15% to 72% chordwise. In total dimensions this amounts to 7.375in. chordwise and 8in. spanwise. The most stringent dimension is in the thickness direction, which ranges from 1.2in. to 0.85 in. along the chordwise dimension. These blade profile dimensions thus impose stringent limits on actuator parameters.

**Discussion: Effect of Constraints on Parameters**

Based on the discussion on constraints and requirements imposed on actuator design parameters the following conclusions are drawn.
Angular tab deflection $\theta$ and braking moment $\tau_p$ are defined by the target objectives and aerodynamic loads experienced by the tab. Therefore there is little freedom in varying these parameters and they were set at $\pm 5^\circ$ and 0.85 in-lbs. respectively.

Hinge tube radius ($r_{ht}$) is governed by the space available at the 72% chord location. Furthermore, this parameter influences the angular deflection inversely and actuation moment directly. Variation of this parameter results in a compromise between actuation moment and angular deflection. In addition geometric constraints place an upper limit on this parameter. To satisfy these conditions this parameter was fixed at 0.35 in..

Initial prestrain applied on the wires ($\epsilon_o$) influences the recoverable strain ($\epsilon_r$). Since the maximum recoverable strain for these wires is 5.5% (as obtained from material characterization procedures carried out by Epps [4]), the prestrain was set at 2.5%. The recoverable strain is now a function solely of the braking moment applied, which was calculated to be equal to $\approx 1.5\%$.

The wire diameter ($d_o$) was 0.015 in. and was selected based on the availability of material characterization data. With the length dimension ($L_o$), although constrained by the chordwise dimensions available, there does exist a certain margin of variation between 3.4 in and 3.7 in.

The second parameter available for variation is the number of wire segments $N_{wire}$. This forms a chief parameter which can vary over a fairly large range (2-20) since the spanwise constraints are relatively more relaxed. This was thus used as a chief control parameter iterated to obtain the required actuation moment and angular deflection.

The subsequent section examines the results obtained with varying these two
parameters. Furthermore, the influence of increasing loading moment (or braking moment) on the final output moment and angle is examined.

### 2.6.4 Results Of Parametric Studies

1. **Effect of loading moment:** For a given set of parameters the variation of actuation moment and tab angle as a function of loading moment was examined in Figure 2.11(a). The following observations were made. The actuation moment increased as the frictional moment to be overcome increased, while the range of available angular deflection decreased. Furthermore, the energy available to overcome external loading decreases. This trend is shown in Figure ??, on the available moment-angular deflection curve, where the effect of increasing external loading moments is plotted.

2. **Influence of Number of wires, \( N_{\text{wire}} \):** Figure 2.11 depicts the influence of increasing the number of wires on the actuation moment-angular deflection curve. The other two control parameters, braking moment \( (\tau_F) \) and wire length \( (L_o) \) were kept constant during iterations for this specific parameter. As is indicative from the equations, the increase in \( N_{\text{wire}} \) increases actuation moment but has no effect on angular deflection. This dependence was indicated in the previous discussion on constraints and is based on the relationships defined in Equations 2.1 and 2.29.

3. **Influence of Length of wire, \( L_o \):** In Figure 2.12 the effect of varying the wire length is examined. Once again, the other two parameters, braking moment \( (\tau_F) \) and Number of wires \( (N_{\text{wire}}) \) are kept a constant for this specific case. The increase in \( L_o \) increases angular deflection, while it does not
influence the actuation moment. This behavior is once again evident from Equations 2.1 and 2.29.

It is worth noting that based on the paramateric plots and the fact that there is a limited scope for varying the parameter $L_o$, $N_{wire}$ becomes a significant parameter in influencing maximum achievable actuator output (moment).

The above set of parameters were iterated repeatedly until a final acceptable set of parameters were obtained. The parameters obtained from the final iteration are defined in Table 2.1.
Figure 2.1: Schematic of Shape Memory Effect
Figure 2.2: Shape Memory Effect Under Rigidly Constrained Condition

Figure 2.3: Shape Memory Effect Under Free Recovery Condition
Figure 2.4: Shape Memory Effect Under Clamped - Spring Boundary Condition

Figure 2.5: Shape Memory Effect Under Clamped - SMA Wire Boundary Condition
Figure 2.6: Schematic of stress-strain plot depicting behavior of shape memory alloys with varying boundary conditions.

(a) Wires equally prestrained

(b) Wire A heated

Figure 2.7: Schematic of operation of dual shape memory alloy actuator.
Figure 2.8: Stress-Strain plot with SMA wire boundary condition - zero friction
Figure 2.9: Stress-Strain plot demonstrating frictional brake effect
Figure 2.10: Effect of friction on SMA actuator output characteristics
(a) Actuator output profile - zero external load
Figure 2.11: Influence of number of wires on actuator output

Figure 2.12: Influence of length of wire on actuator output
Table 2.1: Design parameters for actuator

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length of SMA wire wire</td>
<td>$L_o$</td>
<td>3.6 in</td>
</tr>
<tr>
<td>Diameter of wire</td>
<td>$d_o$</td>
<td>0.015 in</td>
</tr>
<tr>
<td>Radius of hinge tube</td>
<td>$r_{ht}$</td>
<td>0.35 in</td>
</tr>
<tr>
<td>Braking moment</td>
<td>$\tau_F$</td>
<td>0.85 in-lb</td>
</tr>
<tr>
<td>Range of tab deflection</td>
<td>$\theta_{max}$</td>
<td>$\pm 5^\circ$</td>
</tr>
<tr>
<td>Number of wires</td>
<td>$N_{wire}$</td>
<td>12</td>
</tr>
</tbody>
</table>

Table 2.2: Properties of SMA (Ni-Ti) wire

<table>
<thead>
<tr>
<th>Property</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A_s$</td>
<td>94.0°F</td>
<td></td>
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<tr>
<td>$A_f$</td>
<td>120.0°F</td>
<td></td>
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<tr>
<td>$M_s$</td>
<td>85.0°F</td>
<td></td>
</tr>
<tr>
<td>$M_f$</td>
<td>78.0°F</td>
<td></td>
</tr>
<tr>
<td>$E_M$</td>
<td>3170600 psi</td>
<td></td>
</tr>
<tr>
<td>$E_A$</td>
<td>7500000 psi</td>
<td></td>
</tr>
<tr>
<td>$\epsilon_L$</td>
<td>0.055 (in/in)</td>
<td></td>
</tr>
<tr>
<td>$C_M$</td>
<td>1487.2 psi/°F</td>
<td></td>
</tr>
<tr>
<td>$C_A$</td>
<td>1452.8 psi/°F</td>
<td></td>
</tr>
<tr>
<td>$\sigma_s^\alpha$</td>
<td>12000 psi</td>
<td></td>
</tr>
<tr>
<td>$\sigma_f^\alpha$</td>
<td>15000 psi</td>
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Chapter 3

TAB ACTUATOR DESIGN AND
CONSTRUCTION

3.1 Tab Actuator Objective

The objective of the bidirectional SMA tracking tab (BSTT) actuator is to deflect the trailing edge tab of a blade section. The expected hinge moments the actuator would need to overcome are 0.85 in-\textit{lbs.}, as calculated from the previous chapter, with required tab deflections of $\pm 5^\circ$. The overall dimensions of the actuator are determined by the NACA-0012 blade section with a chord section of 12 in. This model is to be tested in an open-jet wind tunnel with test section of 22 in. x 22 in. The spanwise size of the blade section is set to 12 in. and the tab is 4 in. spanwise.

The design objectives identified were the following-

1. Design of an actuator that eliminates problems observed in the first actuator designed by Epps [4]

2. Fabrication of the actuation system and implementation into the geometric envelope of the NACA-0012 12 in. chord blade section.
Integration with a control system that enables closed loop tracking

4. Testing on the benchtop and under aerodynamic loading in the wind tunnel.

These tasks will be addressed under corresponding sections.

### 3.2 Operating Principle

The fundamental mechanism of actuation in the BSTT is the shape memory effect. This effect involves plastically deforming a segment of SMA wire and then recovering the plastic deformation during transformation via heat activation (Figure 1.2). The SMA wires are clamped at one end and connected to the tracking tab by a rotating hinge tube. The wires are electrically and thermally isolated from each other.

To enable actuation of the tab upon heat activation, the wires must first be prestrained (plastically deformed at room temperature). For the purpose of symmetry, the top and bottom wires are prestrained by an equal amount. When the upper wires are heated they tend to acquire the original length, due to the shape memory effect. This behavior is partially restrained due to the opposing wires, which are maintained at room temperature. A force is generated due to the opposing wire at room temperature. This force produces a counter clockwise moment about the hinge tube axis resulting in an upward rotation of the tracking tab (Figure 2.7). The heated wire thus regains a certain fraction of its prestrain. The lower wires get plastically deformed by an equal amount of the contraction of the heated wires, and the mechanisms can be reversed by heating the lower wires.

As the wire extends, its load-displacement characteristics dictate the amount of force the system develops. To deflect the tab in the opposite direction the lower
wires must then be heated with upper wires maintained at room temperature. The reciprocating motion of the wires is transmitted to a corresponding rotation of the tab.

3.3 Actuator Subsystems

The bidirectional SMA tracking tab (BSTT) actuator design configuration is presented in this section. Figures 3.1 and 3.2 shows a schematic of the design layout for the actuation system. The assembled actuator with attached sensors is shown in Figure 3.6. The (a) clamping device (b) active shape memory alloy wires and (c) passive friction brake. together comprise the primary elements of the tracking tab actuator. A closed loop control system was designed to integrate with the actuator to enable tracking of the tab to a commanded input signal. These elements are discussed in detail in the subsequent subsections.

3.3.1 Clamping Mechanism

In the first prototype, fiberglass was used as the clamping material. A number of individual SMA wires (2-5) were fastened at one end by a fiberglass clamp. These were wound around holes in a fiberglass hinge tube, located closer to the trailing edge. These individual wires were required to be gripped down at each end. This design thus introduced multiple possible points of grip failure. The points where the wires were wound around as well as gripped also formed regions of stress concentration on the support material. The fiberglass material, unable to handle the relatively large localized stresses, was found to wear away at these points. Hence, the clamping mechanism was unable to securely grip the wires.
The effect was an unequal amount of tab deflection and unpredictability of the actuation system under aerodynamic loading. To avoid these problems the clamp design discussed below was adopted.

The primary role of the clamping device is to ensure the SMA wires are fixed in place to enable sustained and equal tab deflections. The wires on heat activation, mobilize the movable elements, to generate sustained linear reciprocating motion. The two roles are incorporated into the basic design of the actuator. Figure 3.3 shows the details of this mechanism. There are two pairs of clamps fabricated out of stainless steel. Each set of clamps provides a framework for the upper and lower wires.

Figure 3.3 indicates the individual parts of the clamping structure. Clamp $a1$ is rigidly attached at the quarter-chord location on the spar. Element $b1$ attaches to a turnbuckle through which the force and displacement generated are transmitted to the hinge tube. A pair of guide rails, $c1$ and $c2$, are embedded into $a1$ and provide a guide path for the movable clamp $b1$ to slide. Clamp $b1$ slides along the guide rails as the SMA wire responds to heat activation, and forms the movable element. The movable element has a connecting point, that allows attachment to a linkage that fixes onto the rotating hinge tube.

As shown in Figure 3.4, dowel pins are embedded at discrete locations in the body of the clamps. Freely rotating nylon bushings are mounted on these pins. The SMA wire winds around these nylon bushings, with its ends securely gripped between the top and bottom plates of the movable part of the clamp. These pins, bushings and wire are then embedded between plates $a1$, $a2$ and $b1$, $b2$. There are two sets of this assembly to enable bi-directional actuation.

The novel feature of this design is that a single SMA wire, gripped at one single
point, is wound back and forth around the dowel pins, which act as guide points within the clamp. These guide points introduce segmentation of the SMA wire and multiple such wire segments enable force multiplication. On heat activation, a reduction in length is generated by these segments. Due to the nylon bearings, a uniform stress and strain will be developed in the wire segments. The force and displacement generated will be obtained at the output connecting point of the movable clamps. The wires are gripped between these plates, which are held down with screws.

3.3.2 Shape Memory Alloy Wires

The shape memory alloy wires (Figure ??) employed are Nitinol (Ni-51%, Ti-49%) material procured from Dynalloy Inc.[26]. The wire used has a diameter of 15 mils. and a total initial length of 55 in. The length of each segment is 3.6 in. For this design 12 wire segments were incorporated to obtain the required force output. An initial prestrain of 2.5% was imparted to these wires once they were embedded into the actuator mechanism. Under 120 ft/s wind, 15° angle of attack, hinge moments of 0.85 in-lbs are anticipated and the expected rotation angles possible are ±6.0°. The equations involved in calculating the expected hinge moments under aerodynamic loading are discussed in Section 2.6.3.

3.3.3 Connecting Elements

The clamp output motion is transmitted to the deflecting tab through a pair of linkages. These comprise of a pair of oppositely threaded rod-ends connected through a threaded turnbuckle. Rotation of the turnbuckle enables corresponding tightening of the connecting elements. The rod-ends connect to the movable clamp
at one end and the rotating hinge tube at the other. The turnbuckle thereby provides a convenient method to prestrain the wires. The wires are prestrained (plastically deformed at room temperature) prior to heat activation. The method requires first that the motion of the shaft be restrained. Then both turnbuckles are rotated by an amount corresponding to a calibrated value of forward travel of the movable clamp. The wires are prestrained equally to about 2.5%.

In the first design by Epps, prestrain of the wires was imparted by hanging a weight from the hinge tube. The weight used corresponded to the desired prestrain. Uncertainties were introduced due to the clamping of wires as well as the prestrain method. Thus it was not conclusive whether equal prestrain was imparted to the individual wires. In the method employed currently, equal prestrain is assumed due to the symmetric placement of the wires around the center of the clamp. However any inconsistencies resulting in possible misalignment will be accommodated for since the single wire can realign itself around the nylon bushings.

The turnbuckles play a dual role of sensing as well. These elements are instrumented as force sensors with full bridge strain gauges (Figure ??). In the previous design, force sensors had not been implemented and therefore the stress levels in the wires were not measured.

### 3.3.4 Locking Mechanism

A requirement for a locking device was imposed on this design to prevent changes in tab position once tracking procedure was completed. The main specification for the lock is that it must allow for rotation in both directions as well as hold the hinge tube in position without slipping, under expected aerodynamic loads. The previous design attempted locking by incorporating a gear and pawl locking
device activated by a solenoid. One of the biggest disadvantages of this design is the poor resolution obtained due to the discrete gear teeth. To eliminate this problem friction brakes were closely examined.

Several active friction brake designs employing piezostacks, electrostacks and SMAs were experimented with. Eventually a passive friction brake was selected as the final design. The locking device consists of a shaft collar, rigidly mounted on the rib and around the shaft. Tightening the collar increases the frictional moment. The rotation of the shaft collar is calibrated for frictional moment using a torque wrench. Figure ?? shows this passive brake fixture rigidly attached on the structural rib and installed on the hinge tube. The passive brake prevents tab motion up to a preset braking moment. Thus for actuation moments exceeding this braking moment, the output shaft will rotate. From the Section 2.6.3 loading moments of around 0.85 in-lbs are anticipated. This requirement defines the braking moment of the lock.

### 3.4 Blade Section Assembly

A NACA 0012 blade section of 12 in. span and 12 in. chord section was fabricated. The dual SMA tracking tab actuator was mounted into this blade section. Figure 3.2 shows the entire blade section with actuator assembly inside. Detailed actuator parts are shown in Figure 3.5. The fabricated blade consists of a foam core, trailing edge tab and actuator assembly with spar and ribs to provide structural integrity. The rigid foam core was made of Rohacell 51 IG using compression molding technique. The blade section was constructed by laminating 10 mils prepreg fiberglass cloth plies around the rigid foam core and cured in an NACA-0012 airfoil mold.
The spar comprises of a 0.25 in. thick flat rectangular aluminum piece, embedded into the blade from 10% to 30% of the blade section. It has mounting holes at two ends, which enables attachment of the section to a rigid support. The ribs consist of 0.25 in. thick aluminum pieces attached to the spar near the quarter chord. These extend up to 80% of the blade section. Teflon spring bushings are embedded at 72% of the chordwise section and provide mounting points for the rotating hinge tube.

Figure 3.6 shows the complete blade assembly with the installed tracking tab actuator.

### 3.5 Position Feedback Controller

#### 3.5.1 Controller Design - Initial Implementation Scheme

To demonstrate the capability of the actuator to accurately deflect a tracking tab to the commanded input position, closed loop control is required. To enable this operation a control system was devised that could command an input tab position \( V_{SET} \), send corresponding actuation signals to the SMA mechanical actuator and measure the response in terms of tab deflection angle, \( V_{TAB} \). In the first design developed by Epps [4], a two position control system was designed and a breadboard circuit was built. In the subsequent section a discussion on the operation of this two position control systems is presented. The actuator operation under closed loop control is evaluated in conjunction with the controller.
Two Position Control System Operation

In a two position control system, the actuating element has only two fixed positions, which in this case are simply on and off. The two-position or on-off control system (also known as a bang-bang controller), is relatively simple and inexpensive and for this reason is used fairly widely [27].

The controller output signal is denoted by $u(t)$ and the actuating error signal is $V_{err}$. In two-position control, the signal $u(t)$ remains at maximum, minimum or off within the deadband zone (defined by $\Delta_1$ and $\Delta_2$), depending upon the sign and magnitude of the actuating error signal and the range of the deadband. This signal is given by,

$$u(t) = \begin{cases} V_1 & \text{for } V_{err} > \Delta_1 \\ V_2 & \text{for } V_{err} < \Delta_2 \\ 0 & \text{for } \Delta_2 \leq V_{err} \leq \Delta_1 \end{cases}$$

where $V_1$ and $V_2$ are constant voltages set in the circuit. The range through which the actuating error signal moves before switching occurs is called the deadband or the differential gap and is denoted by $\Delta_1 + \Delta_2$. The differential gap influences the controller output $u(t)$, to maintain its present value until the actuating error signal has moved slightly beyond the zero value. The purpose of the differential gap is to decrease chatter in the mechanical actuation system. This thus provides a certain tolerance to the operation requirements imposed on the mechanical system. The behavior of the controller employed is schematically indicated in Figure 3.7.

A schematic of the entire circuit of the control system is drawn in Figure ??.

The inputs to the control system are provided by the input tab angle $V_{SET}$ and the tab angle position $V_{TAB}$. The commanded tab angle, $V_{SET}$ could be varied by
a dial on the control panel, while $V_{TAB}$ is measured with a rotary potentiometer, mounted on the hinge tube shaft.

When the circuit was implemented with the actuator several operational problems were observed. Primary amongst these were:

1. A drift in commanded signal was observed as the controller switched the signal on and off. The drift in error signal was of the order of the accuracy required of the system. Thus closed loop tracking operation resulted in unacceptable results of poor resolution and the consequent inability of the control system to power off.

2. The nature of the on-off control system is one that results in a sudden jump from a low (0 Volts) to a high (5 Volts) condition. This causes a sudden rise in temperature of the wires. For this actuator design, the assumption made in the model is that when one wire is heated the other is strained isothermally. This assumption is however invalidated when an abrupt rise in temperature occurs due to a sudden rise in the actuation voltage. Ideally the strain rate of the wires should be maintained at 0.0004/second [14, 28, 29]. This corresponds to an average heating rate of 0.5° F/sec. The current operational characteristic of this controller were considered to be non-ideal for the bidirectional SMA actuator.

### 3.5.2 Improved Control System Design

To eliminate the inadequacies and errors observed in the present control system and to introduce a degree of flexibility in the type of control employed, a computer based control system was devised using LABVIEW™. The virtual instrument
application programmed, integrated the roles of control and data acquisition, into a single interface. The inputs sent to the data acquisition ports are the tab angular measurements and temperatures of the two wires. The output actuating signals are sent to two MOSFETS, which are powered by a 15 Volt, 4 Ampere DC power supply. In any given condition, depending upon the signal from the controller either one or neither of the MOSFETS are powered. The output from each one of these MOSFETS are then fed to the top and bottom set of wires, respectively.

Two control systems were implemented and compared in this study. The on-off controller was implemented to serve as a baseline for comparison. Next, proportional control was evaluated for feasibility. A discussion on proportional control systems follows.

Proportional Control System - Basic Principle

For a controller with proportional control action, the relationship between the output of the controller \( u(t) \) and actuating error signal \( e(t) \) is given by:

\[
u(t) = K_p e(t)
\]  

(3.1)

where \( K_p \) is the proportional gain of the controller [27].

Figure ?? demonstrates this control action as applied to the current mechanical system.

Proportional Control Systems Evaluated

The proportional control action basically adjusts the voltage sent to the wires as a linear function of the error signal. Thus, as the error reduces between the command signal and the tab angular position, the magnitude of the actuating
signal reduces. On conducting trials with this controller and the actuation system, relatively high overshoots were observed, before the system settled down to its steady state condition. Tests of a system comprising of actuating signals which were quadratically proportional to the error signal, were observed to yield better closed loop tracking results. These were defined by lower overshoots and better settling characteristics in terms of steady state error and power off condition.

Mathematically, the difference between these two control systems is indicated below:

for simple proportional control:

\[ V = V_1 + (V_2 - V_1) \frac{e(t)}{\theta_{input}} \]  

(3.2)

for quadratic proportional control

\[ V = V_1 + (V_2 - V_1) \frac{e(t)}{\theta_{input}} \left(1 + \frac{e(t)}{\theta_{input}} \right) \]  

(3.3)

where \( e(t) \) is the error signal, \( \theta_{input} \) is the input command signal, \( V_2 \) and \( V_1 \) are the maximum and minimum voltages applied to the actuator outside the deadband region.

To reduce chatter in the mechanical system, a deadband was introduced, which was specified on the interface and is referred to in this discussion simply as \( \Delta_1 + \Delta_2 \). The basic actuating signals are then mathematically represented as:

\[ u(t) = 0, \quad \text{for} \quad \Delta_2 < V_{err}(t) < \Delta_1 \]  

(3.4)

\[ u(t) = V, \quad \text{for} \quad V_{err}(t) > \Delta_1, V_{err}(t) < \Delta_2 \]  

(3.5)

where \( V \) is defined according to the controller selected (defined by Equations 3.2 and 3.5). Note that in contrast to the proportional control action, the on-off (two-position) controller actuation voltage is defined by a constant value \( V \), independent of the magnitude of the error.
During trials in the lab, the quadratic proportional controller demonstrated greater sensitivity to the error signal. It therefore resulted in better closed loop performance as compared to the simple proportional controller. This control system was hence adopted for complete testing during benchtop and wind tunnel tests. The simple proportional controller was applied under sample conditions and compared with the other two control systems.
Figure 3.2: Blade section - isometric view
Figure 3.3: Shape Memory Alloy Wire clamping and actuating element
Figure 3.4: Details of SMA wire clamping element
Figure 3.5: Detailed schematic of clamps, linkages and structural components
Figure 3.6: Tab actuation system components
Figure 3.7: Schematic of two-position control operation
Chapter 4

RESULTS AND DISCUSSION

In this chapter the open and closed loop tests conducted are discussed and the results are analyzed. The basic objective is to test if the target objectives of $\pm 5^\circ$ tab deflections with an accuracy of $0.1^\circ$ under aerodynamic loading are achievable. These requirements were tested for systematically. The first set of experiments involved testing actuator performance under open-loop conditions on the benchtop. Tests for repeatable tab up and down deflections were carried out to ensure consistent actuator behavior and elimination of problems faced in the first phase of the tests. To check the accuracy of the predictive model, testing under varying braking moments was conducted and the data obtained was used to correlate with the analytical results.

Next, closed loop tracking tests were conducted under target design conditions. These tests were conducted on the benchtop for varying target tracking tab angles and for different control schemes. The actuator performance under position feedback control was examined. Simulation of aerodynamic loading conditions on the benchtop was also conducted.

Finally the actuator integrated in the 12 in. chord blade section with tracking tab, was tested in the open jet wind tunnel facility. These tests were carried out at
four different wind speeds and angles of attack to test actuator performance under aerodynamic loading conditions. The experimental testing details and the results obtained from various tests are discussed in the subsequent sections.

4.1 Open Loop Tests

The open loop tests were conducted to assess the improved performance of the actuator design over the first prototype. The objective of these tests were identified to be: (a) assessment of ability of the clamping mechanism in eliminating wire slippage and thereby enabling equal tab angles and repeatable performance. (b) evaluation of the accuracy of the analytical model in predicting actuator behavior. This behavior was assessed by varying the frictional braking moments and comparing results from the model with the experimental tests.

4.1.1 Test Equipment Used

This section describes the equipment and procedure employed in testing the open loop performance of the tracking tab actuator integrated into the NACA0012 blade section. Figure 4.1 shows the equipment and setup for the open loop tests. The shape memory alloy wires were resistively heated and air cooled. Type K thermocouples were used to measure the temperature of the two sets of wires, which were held in place using insulating tape. An electrically insulating sheet of mica was placed between the wire and the thermocouple, to isolate the thermocouple from the heating current passed through the wires. The tab deflection angle was measured using a three-quarter turn 10KΩ potentiometer and the force was measured using two instrumented links. The signals from the sensors were measured using
a data acquisition DSP, SIGLAB\textsuperscript{TM} system interfaced with MATLAB\textsuperscript{TM} operated on a Pentium-III 900 MHz processor.

In the tests described in this section, heating was controlled using a Hewlett Packard 6642-A DC power supply rated at 10 Amps, 20 Volts. The current was set at 3 Amps with the voltage rate set at 0.017 Volts/sec, in order to maintain a heating rate of 0.5°F/sec. This corresponds to a strain rate of $\frac{\Delta \delta}{\Delta t} = 0.0004$/sec during the heat activation cycle, for the opposing wire at room temperature. Experimental results were recorded over a period of 600 seconds. The prestrain of the wires was set equally at 2.5%.

### 4.1.2 Experimental Test Results and Discussion

The actuator behavior is demonstrated in Figure 4.2 to assess the response of the tab deflections to a variation in wire temperature. The tests indicated here were conducted without the introduction of braking moment. Figure 4.2(a) plots response of tab deflection to temperature of the top wires as the voltage input is increased. Indicated here is the response of the tab to a rise in temperature, followed by a cooling of the wires as the voltage input is reduced. The results of two such tests are superimposed on this graph. The repeatability of the actuation system for tab up deflection is clearly indicated here.

Figure 4.2(b) plots the response of the tab to increase in temperature of the lower wires, which results in a tab down deflection. This is in response to the lower wires progressively heating up due to an increasing voltage input. This is followed by a reduction in tab angle on the cooling cycle as the voltage input to the wires is then reduced from 0 to 5 Volts. Once again two such tests for the tab down case were conducted to assess repeatability. Comparing the results from Figures 4.2(a)
and 4.2(b) the symmetric response of the actuator is indicated by the variation of tracking tab deflection between ±10°.

Figure 4.3 provides further indication of the repeatable behavior of the actuator. In this figure the actuation moment response, measured by the load cell, is evaluated with the corresponding rotation angle, for the same set of tests. Figure 4.3(a) demonstrates response on tab up actuation, and Figure 4.3(b) indicates actuator behavior on tab down deflection.

The results from Figures 4.2 and Figures 4.3 indicate that the goal of elimination of wire slippage has been achieved. This conclusion is based on the equal tab angles and actuation moments obtained in positive and negative directions and the repeatability of these tests.

Actuator response to varying braking loads was examined next. The behavior is indicated in Figure 4.4 where response for braking moments of 0, 2.5 and 3.75 in-lbs. are compared. Figure 4.4(a) indicates the variation in tab rotation with temperature. As this braking moment increases two observations are made. The first is the reduction in angular rotation with an increase in braking moment. This is theoretically consistent since the energy required to overcome the braking moment implies less energy is available to deflect the tab, therefore a lower tab deflection angle is generated. The next point of interest is the increase in delay in the temperature at which recovery, and therefore tab deflection occurs, as a function of increasing braking moment.

Figure 4.4(b) plots variation in output actuation moment with tab deflection angle for the same set of tests. The variation in braking moment results in an increase in output actuation moments, with the corresponding reduction in angular rotation as mentioned above. Note the rise in the constant actuation moment level.
as the frictional moment rises.

Next a comparison of the analytical model with experimental data was carried out. In Figure 4.5 the results of the analysis with test results, for zero braking moment condition are examined. Figure 4.5(a) shows the tab deflection variation with temperature while Figure 4.5(b) depicts the actuation moment variation with tab angle. The model captures the actuator behavior fairly well.

The results for tests with varying frictional moments imposed on the actuator were then superimposed in a block actuation moment-angular rotation curve as illustrated in Figure 4.6. Here the variation in maximum actuation moment with corresponding maximum angular rotation for varying frictional moments is graphically indicated. The model compares favorably with actuator experimental results and provides a close estimate of the tab deflection angles, the actuator is capable of generating, under the selected braking moment.

### 4.2 Closed Loop Test Results

The main objective of the closed loop test is to demonstrate the capability of the actuator to accurately deflect the tracking tab in response to the commanded inputs. The tracking tab program places a requirement of \( \pm 5^\circ \) of deflection capability with an accuracy of \( 0.1^\circ \). The tests conducted here demonstrated this capability under aerodynamic loading corresponding to a wind speed of 100 \( ft/sec \) at angles of attack upto \( 15^\circ \). The parameters selected from the parametric studies are incorporated into the actuator design. The position control system discussed in Section 2.6 was implemented and two control schemes (on-off and proportional control) were extensively tested both on the benchtop and in the open jet tunnel.
4.2.1 Benchtop Tests

Prior to testing under aerodynamic loading the baseline system performance was evaluated. The benchtop tests were conducted under zero load and simulated to maximum anticipated loads in the open jet tunnel (120 ft/sec at 15° angle of attack).

Experimental Test Setup

The actuator embedded into the blade section was tested first on the benchtop under zero wind speed conditions. The control system detailed in Section 3.5.2 was tested under on-off and quadratic proportional control. The tab angular measurements were made using a Hall effect sensor, designed to operate as a rotary potentiometer. The sensor generated output voltages linearly proportional to the angular rotation and was capable of measuring up to ±45° of rotation. Temperature measurements of the wires were made using Type K thermocouples. A Windows-NT based Pentium III, 450 MHz PC equipped with a National Instruments, PCI-6031E, 16 bit DAQ card was used with the virtual instrument application programmed using LABVIEW\textsuperscript{TM} 5.1. Five input channels to the DAQ measured temperature and angular rotation of the tab, while the actuating signals to the wires were sent through the two output channels. The sampling rate of the DAQ system was tested for 1/sec and 0.5/sec. These sampling rates were adequate for these quasi-steady tests.
4.2.2 Results - Benchtop Tests

Zero Load Condition

Tests for tracking operation of $\pm 2^\circ$ and $\pm 5^\circ$ were conducted applying both on-off control and a quadratic proportional control system. The results from these set of tests are plotted in Figure 4.7, in terms of position accuracy and power off capability. In Figure 4.7(a) the steady state error is compared for the two control systems for both tracking angles ($2^\circ$ and $5^\circ$). For the same set of tests the percentage change in steady state temperature from the ambient is charted in Figure 4.7(b).

From these plots, it is clear that the quadratic proportional control system is superior in terms of steady state error, steady state temperature (indicating power off condition as compared to the on-off controller) as well as the number of oscillations observed in reaching steady state. Figures 4.8 and 4.9 display the time response of the tracking operation as the actuator responds to commanded inputs, for the two control schemes.

Simulated Loads

Under simulated loading conditions no significant degradation in performance was observed in the operation of the actuator under either control action. This indicates the capability of the actuation system to maintain its position as well as actuate the tracking tab under the external loading conditions anticipated in the open jet tunnel. Figures 4.10(a) and 4.10(b) plot the time trace of these simulated conditions under both control actions.
4.2.3 Open Jet Tunnel Tests

The objective of these tests is to evaluate the performance of the BSTT actuator under aerodynamic loading. The actuator is mounted in the blade section and actuates the hinge tube, which deflects the trailing edge tab. The actuator was tested under closed loop control using the position control system, the details of which were discussed in Section 3.5.2. An evaluation of tracking operation under closed loop control for the two control actions was conducted for three different wind speeds and four varying blade angles of attack. The test matrix is shown in Table 4.1. Prior to testing the actuator under aerodynamic loading, the actuator operation was tested under zero airspeed in order to provide a baseline for assessing actuator performance.

Figure 4.11 shows the blade section mounted in the open-jet wind tunnel for testing while Figure 4.12 presents a close-up of the mounted blade section with the incorporated BSTT actuator.

Test Setup

The Open Jet tunnel is an open section wind tunnel that has a rectangular test section of dimensions 22 in x 22 in. The maximum freestream velocity of 100 ft/sec could be achieved while the minimum idling speed was approximately 30 ft/sec. A pitot tube was used with a manometer, to measure the freestream air speed. The blade section integrated with the BSTT actuator was mounted in the test section using the mounting fixture and tested in the open jet wind tunnel. The data acquisition system used for these tests was identical to that used in the benchtop tests, details of which are provided in Section 4.2.2.
Results

The comparison of tests results for varying wind speeds and angles of attack conditions was conducted for the two basic control schemes discussed, on-off and quadratic proportional control.

To evaluate the operation of the actuator integrated with these two different control systems, the results from the tests were compared for tracking to $\pm 5^\circ$. First steady state error in angular position for each wind speed was plotted for the four angle of attack conditions tested. Superimposed on these plots are horizontal lines indicating the deadbands for the control system used.

To indicate the ability of the actuator to maintain commanded angular position under power off condition the temperature of the wires under steady state conditions were plotted for these same wind speed and angle of attack conditions. Since the wires were air cooled, the ability of the wires to achieve ambient temperature conditions within a reasonable amount of time is impeded. This behavior is particularly prevalent at lower wind speeds. At higher wind speeds it was found that the flow of air enabled rapid cooling of the wires. Therefore the margin of acceptable steady state wire temperature was established as 8% higher than ambient conditions. This corresponds to $\approx 6^\circ F$, which is considered acceptable from the point of view of the temperature dependency of SMA wires. This margin is indicated on the temperature plots by the dashed horizontal lines.

Figure 4.13 shows the results for steady state error and temperature variation as a function of wind speed for varying angles of attack for the on-off control system. Figure 4.14 demonstrates the results for the same set of tests under quadratic proportional control. Of note is the close proximity of the actuator under on-off control, to the dead band position of 0.2$^\circ$. In isolation this trend would appear
acceptable from the point of view of position control. However when examined with the temperature plots in Figure 4.13(b) it is fairly evident that the control system is unable to demonstrate the power off condition deemed necessary for acceptable closed loop tracking operation.

This power on condition is represented in the time traces in Figure 4.15-4.19, by constant oscillations about the set-point. The oscillations indicate a constant switching on-and off of power as the actuator moves in and out of the dead band. The actuator under this control action is thus unable to acquire a stable non-oscillatory position.

Figure 4.14 depicts actuator behavior under the quadratic proportional control action. The steady state error remains fairly close to the deadband, except for the higher wind speed/ angle of attack combinations. Moreover on examining the temperature difference plots (Figure 4.14(b)), the results indicate that except for three wind-speed conditions, the power off condition is achieved.

Thus the average performance of the actuator under quadratic proportional control appears to be far improved. The reason for this lies in the nature of actuation in the control action demonstrated. The actuation signal heats one set of wires and the actuator responds with an overshoot above the set-point. Due to the quadratic dependency of the actuation voltage on the error, the overshoot results in a relatively slow rise in temperature of the opposing wires. This enables settling of the system into the final steady state condition, which ideally should lie within the dead band region of the control system, whereupon power supplied to the actuator switches off. As is evident from the time traces the response of the actuator to input signals is smooth and non-oscillatory.

Figures 4.15-4.19 compare the time history of tracking to ±5° under representa-
tive loading conditions. The input signal (indicated by the dashed line) - indicates the commanded input of first $+5^\circ$. After an interval of time, during which the actuator is allowed to settle to its steady state condition, an input of tracking to $0^\circ$ is sent, followed, once again after a brief of interval of time, by a $-5^\circ$ input signal. Comparisons between on-off control action and the quadratic proportional control system indicate an improved performance of the actuator operated with a quadratic proportional controller.

Further comparisons were conducted between on-off, simple proportional and quadratic proportional control systems for two extreme loading conditions. The results of the two extremes of tracking to $\pm 5^\circ$ and $\pm 2^\circ$ at zero loading and $V=100\text{ ft/sec}$, $\alpha = 15^\circ$ are indicated in Figures 4.20- 4.21.

At the zero loading condition the simple proportional controller demonstrated inferior performance as compared to the quadratic proportional controller, in terms of accuracy and power off tracking capability. At higher loading conditions however, the simple proportional control system showed superior performance capability as compared to the quadratic proportional control action. This trend was noted for both the $2^\circ$ and $5^\circ$ tracking input cases.

Also note that for all cases shown, the on-off control system appears to demonstrate about average performance. However the oscillatory nature of the on-off control system is not captured in these results.
Figure 4.1: Open Loop benchtop test setup for tab actuator
Figure 4.2: Tab deflection response to temperature demonstrates repeatable and symmetric actuator behavior
Figure 4.3: Actuator output moment vs tab deflection angle demonstrates repeatable and symmetric action
Figure 4.4: Test data demonstrating the effect of frictional moment on tab angle and output moment
Figure 4.5: Comparison of analytical model with test data
Figure 4.6: Actuation moment vs tab rotation angle curve - comparison of predicted values with experimental results
Figure 4.7: Comparison of steady-state bench-top conditions for two control schemes under tracking tab deflection angles of 2 and 5 degrees, zero load condition.
Figure 4.8: Time trace of the tracking response for tab up and down inputs of 5 degrees for both controllers
Figure 4.9: Time trace of the tracking response for tab up and down inputs of 2 degrees for both controllers
Figure 4.10: Comparison of steady-state bench-top conditions for different controllers under tracking tab deflection angles of 2 and 5 degrees, simulated maximum loading condition (120ft/sec, 15° angle of attack)
Figure 4.11: Setup for open jet wind tunnel closed loop testing of tab actuator
Figure 4.12: Close-up view of blade section mounted in test section
Figure 4.13: On-Off controller - influence of wind speed and angle of attack on actuator steady-state conditions for tracking input of 5 degrees
Figure 4.14: Quadratic proportional controller - influence of wind speed and angle of attack on actuator steady-state conditions for tracking input of 5 degrees
Figure 4.15: Time trace of the tracking response for tab up and down inputs of 5 degrees, $V=0$ ft/sec, $\alpha = 0^\circ$ (baseline condition)
Figure 4.16: Time trace of the tracking response for tab up and down inputs of 5 degrees, $V=100\text{ft/sec}$, $\alpha = 0^\circ$
Figure 4.17: Time trace of the tracking response for tab up and down inputs of 5 degrees, $V=60\text{ft/sec}$, $\alpha = 4.5^\circ$
Figure 4.18: Time trace of the tracking response for tab up and down inputs of 5 degrees, $V=30\text{ft/sec}$, $\alpha = 9^\circ$
Figure 4.19: Time trace of the tracking response for tab up and down inputs of 5 degrees, $V=30\text{ft/sec}$, $\alpha = 15^\circ$
Figure 4.20: Comparison of 3 Control Systems under (1) baseline conditions (V=0 ft/sec, $\alpha = 0^\circ$) and (2) maximum loading conditions (V=100 ft/sec, $\alpha = 15^\circ$) for tracking inputs of $5^\circ$
Figure 4.21: Comparison of 3 Control Systems under (1) baseline conditions (V=0 ft/sec, $\alpha = 0^\circ$) and (2) maximum loading conditions (V=100 ft/sec, $\alpha = 15^\circ$) for tracking inputs of $2^\circ$
Table 4.1: Test matrix for wind tunnel test of improved BSTT actuator

<table>
<thead>
<tr>
<th>Case</th>
<th>Airspeed (ft/sec)</th>
<th>Angle of Attack (deg)</th>
<th>Control Scheme</th>
<th>Tracking Angle (deg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1a</td>
<td>0</td>
<td>0</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Simple Prop.</td>
<td></td>
</tr>
<tr>
<td>2a</td>
<td>30</td>
<td>0</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>2b</td>
<td>30</td>
<td>4.5</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>2c</td>
<td>30</td>
<td>9</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>2d</td>
<td>30</td>
<td>15</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>3a</td>
<td>60</td>
<td>0</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>3b</td>
<td>60</td>
<td>4.5</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>3c</td>
<td>60</td>
<td>9</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>3d</td>
<td>60</td>
<td>15</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>4a</td>
<td>100</td>
<td>0</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>4b</td>
<td>100</td>
<td>4.5</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>4c</td>
<td>100</td>
<td>9</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td>4d</td>
<td>100</td>
<td>15</td>
<td>On-off, Quadratic Prop.</td>
<td>2 &amp; 5</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>Simple Prop.</td>
<td></td>
</tr>
</tbody>
</table>
Chapter 5

SUMMARY AND CONCLUSIONS

The design and testing of an improved SMA tracking tab actuator is presented in this thesis. The application of this mechanism is envisioned as an on-blade tracking tab actuator using Shape Memory Alloys as the active material. The on-blade actuating system would enable deflection of an individual rotor blade tab, in response to commands sent by the control system. This method employs a relatively simple mechanism for tracking rotor blades. As discussed in Chapter 1, the motivation for using SMAs as the active material is their relatively large force and stroke capability. Low actuation voltages, low costs and reduced number of moving parts are additional advantages associated with employing these materials.

Current SMA tracking tab designs fabricated, have employed SMA torque tubes and wires. The SMA torsion tube design was effective in achieving required tab deflection angles and braking moments. However a high time penalty is associated with actuating this mechanism. External heating mechanisms employed have also dictated the large volumetric dimensions of this actuator.

An actuator employing SMA wires, displayed the distinct feasibility of this design. The fundamental principle of SMA wire tab actuation is based on the bidirectional nature of a pair of SMA wires acting antagonistically against each
other. Heat activation of one wire enables recovery of a fraction of its prestrain. The opposing wire simultaneously extends by an equal amount. When the heat supplied to the wires is interchanged, the behavior is then reversed.

The initial SMA wire actuator designed at the University of Maryland, physically demonstrated the above principle. However some drawbacks with the design resulted in poor tracking results. The objective of the current study was to improve upon the initial design. A comparison between issues associated with the initial design and improvements made to overcome these, in the new design, follows:

- **Unequal tab deflection angles** were encountered initially. This was due to a combination of slippage of the multiple wires used and an incapability of the clamps to grip the wires. Moreover, stress concentration damaged the fiberglass material employed in the structure. The current design sought to eliminate these issues by using a single SMA wire wound back and forth, to enable force multiplication. This also reduced the (initially) multiple points of failure down to two.

- **Prestrain** method initially employed, resulted in unequal prestrain being imparted to the wires. This problem is eliminated in the new design, by the self adjusting nature of the wire segments arranged around the clamp, since due to the use of one single wire.

- **The locking mechanism** was unreliable in holding the tab position. Moreover discretization introduced by the gears in the lock resulted in a relatively low angular resolution (0.5°). The current design sought to eliminate the problem of discretization by employing a passive friction brake mechanism, which has infinite resolution. The passive nature of the mechanism is used
to prevent motion upto the designed frictional moment. This design has a lower number of moving parts and is simple in implementation.

- **Position control system** designed employed a bread board circuit for an on-off controller. This particular circuit introduced stray voltage which caused a drift in the set point voltage. This was found to result in poor closed loop tracking. The control system was replaced by a computer based controller which implemented a virtual instrument, programmed to introduce the desired controlling action. This design introduced the flexibility of testing different control schemes and was used to arrive at one which gave best possible actuator tracking results.

The mechanism designed, was tested under open loop conditions. These tests revealed the ability of the actuator to deflect the tab equally and consistently. The actuator was implemented with a controller and tested under closed loop tracking. Tests enabled the comparison of multiple control schemes, to arrive at a controller most suitable for this actuator. Testing on the benchtop revealed the efficacy of a quadratic proportional control scheme, as compared to the performance under simple on-off control. Tests were then conducted in the open-jet tunnel to evaluate the performance under loading conditions. These results indicated good performance under lower wind speed conditions. At higher wind speeds, a controller employing a simple proportional control action, performed better.

The results obtained after complete testing of the actuator under aerodynamic loading, allow for a comparison to be made of the actuator capability with the tracking tab program objectives.

- **Tracking angles** required were of the order of ±5°, with an accuracy of 0.1°, under power off condition. These were well acquired with the control systems
employed. Under higher wind speeds, the simple proportional control scheme enabled acquisition of this requirement. The quadratic proportional action achieved these results successfully, at lower wind speeds.

- **Braking moment** of 4 in-lbs were set as the requirement for preventing motion, on the rotating blade of the helicopter. The current actuator was however, tested under aerodynamic speeds which corresponded to a maximum hinge moment of 0.85 in-lbs. The results achieved under aerodynamic loading indicated satisfactory performance. It is anticipated that for testing in the rotating frame, a simple increment of actuator parameters should allow the achievement of required results.

- **Weight** of the actuator is required to be less than 1lb. The actuator itself corresponds to 0.8lbs. However associated hardware in terms of structural supports (like ribs) and the hinge tube introduce a high weight penalty of a total of 2.1lbs. This is a serious disadvantage and would need to be tackled aggressively before serious implementation in the blade section can be considered.

**Dimensions** of the actuation system are such that they need to be completely integrated into the blade section. This has been achieved for the NACA-0012 blade section.

**Duty cycle** of 20 cycles per hour was deemed as a tracking tab requirement. Testing revealed the capability of the design to track to a single input within less than 100 seconds. Tests for three successive inputs consistently demonstrated completion of these tests within 300 seconds. Thus a duty cycle of 20 cycles per hour appears to be achievable by this actuator.
Temperature range of -60 to 160°F was suggested by the program. For SMAs, this requirement imposes a serious challenge. The ambient conditions mentioned above would affect material behavior and result in poor actuator performance and output characteristics. Therefore some form of insulation in the blade, around the actuator is recommended.

5.1 Recommendations for Future Work

The following areas are identified as recommendations for further improvements.

- **Control System:** Tests conducted of this actuator under closed loop control in the wind tunnel, revealed satisfactory performance at low wind speeds under quadratic proportional control. The performance however degraded at higher wind speeds. The simple proportional controller however operated well under high wind speeds. Conversely, at lower wind speeds this scheme caused high overshoots and temperatures of the actuator, which were undesirable from the actuator operation point of view. It is recommended that a third control scheme be introduced, that can integrate the advantages of quadratic proportional action at low wind speeds and simple proportional at higher wind speeds. This is a relatively simple upgrade as it would require the integration of existing control schemes.

- **Braking Moment:** of 4 in-lbs was selected judicious for an on-blade actuator operating in a rotating environment. For this set of tests, the actuation system was subjected to aerodynamic loads of upto 0.85 in-lbs. Thus a corresponding braking friction was introduced with a passive friction brake employed. To test in the rotating frame, the brake mechanism employed
can easily introduce moments of 4 in-lbs. However the actuation mechanism needs to generate ±5° of angular deflection under 8 in-lbs of external loading. This thus places a requirement for an increase in force output from the actuator, above the current levels. This requirement can be easily met by an increase in the Number of wire segments \(N_{\text{wire}}\) incorporated, which result in a direct force multiplication. This calls for a spanwise scaling up of actuator dimensions and reinforcement of parts at connecting points. This upgrade should acquisition of the target requirements under rotating frame loads.

- **Actuator Weight**: forms a significant drawback of the current design, as it also reduces the energy density of this mechanism. It is suggested that methods towards reducing the actuating element weight, be examined closely. One recommendation involves replacement of the clamp material. Currently stainless steel is employed, this could be replaced with a machinable ceramic (for eg: MACOR). The advantage of reduced weight with this material, comes without the penalty of stress concentration (which was observed in the first design where fiberglass was used).
BIBLIOGRAPHY


