Design Demonstration and Optimization of a Morphing Aircraft Control Surface Using Flexible Matrix Composite Actuators

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Dissertation proposal submitted to the faculty of the Virginia Polytechnic Institute and State University in partial fulfillment of the requirements for the degree of

Doctor of Philosophy
in
Aerospace Engineering

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February 16, 2018
Blacksburg, Virginia

Keywords: Morphing, Aircraft Control Surfaces, Flexible Matrix Composite Actuator
Design, Demonstration, and Optimization of a Morphing Aircraft Control Surface

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Abstract

The morphing of aircraft wings for flight control started as a necessity for the Wright Brothers but quickly fell out of favor as aircraft increased speed. Currently morphing aircraft control is one of many ideas being explored as we seek to improve aircraft efficiency, reduce noise, and other alternative aircraft solutions. The conventional hinged control surface took over as the predominant method for control due to its simplicity and allowing stiffer wings to be built. With modern technologies in variable stiffness materials, actuators, and design methods, a morphing control surface, which considers deforming a significant portion of the wing’s surface continuously, can be considered.

While many have considered morphing designs on the scale of small and medium size UAVs, few look at it for full-size commercial transport aircraft. One promising technology in this field is the flexible matrix composite (FMC) actuator. This muscle-like actuator can be embedded with the deformable structure and unlike many other actuators continue to actuate with the morphing of the structure. This was demonstrated in the FMC active spoiler prototype, which was a full-scale benchtop prototype, demonstrated to perform under closed-loop control for both the required deflection and load cases.

Based on this FMC active spoiler concept a morphing aileron design was examined. To do this an analysis coupling the structure, fluid, and FMC actuator models was created. This allows for optimization of the design with the objectives of minimizing the hydraulic energy required and mass of the system by varying the layout of the FMC aileron, the material properties used, and the actuator’s design and placement with the morphing section.

Based on a commercial transport aircraft a design case was developed to investigate the optimal design of a morphing aileron using the developed analysis tool. The optimization looked at minimizing the mass and energy requirements of the morphing aileron and was subject to a series of constraints developed from the design case and the physical limitations of the system. A Pareto front was developed for these two objectives and the resulting designs along the Pareto front explored. From this optimization, a series of design guidelines were developed.
Design, Demonstration, and Optimization of a Morphing Aircraft Control Surface

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General Audience Abstract

This work looks at an aircraft morphing control surface design on the scale of commercial transport aircraft. A design is developed and demonstrated through bench top prototype testing and through analysis. The morphing control surface uses flexible matrix composite (FMC) actuators. These unique actuators are muscle like, using hydraulic pressure to create a contractive actuator. Unlike a simple hydraulic piston, the FMC actuators are capable of bending with the morphing structure during actuation. Through optimization of the morphing control surface design a set of design guidelines were developed to guide future design.
Acknowledgments

I am very appreciative of the professors whom I have had in class and the AOE department as a whole. I would particularly like to thank those on my committee, Dr. Canfield, Dr. Patil, Dr. Philen, and Dr. West. I would like to thank Dr. West for his initial insight into setting up parametric models in Abaqus, and Dr. Canfield for his course in structural optimization. I would especially like to thank Dr. Philen. He has created a lab and environment that is fun to work in and has offered many exciting challenges and projects along the way. Unfortunately, many of these side projects did not make it into this dissertation. He has provided guidance when needed and great freedom along the way. I will miss his flying (crashing) of quads, Jeep, rides, annual BBQ, and much more.

I would like to thank everyone who has been down in the ASML and passed through over the years. There are too many to name, but I learned something working with each. I would like to especially thank Shawn and Carson, both were there to see me through the end. Shawn too will soon be done, but Carson’s time continues, and I expect he will accomplish great things. I have particularly enjoyed my time with Carson who has been one of the best people to work with. His help has likely sped me up equally as much as his friendship and antics has slowed me down.

I would also like the thank everyone at my new home, NASA MSFC. Getting the opportunity to seamlessly take the next step after my Ph.D., and to contribute to meaningful work has been a great inspiration at the end. They have been gracious working with and encouraging me to finish.

The last few years I have had the opportunity to be an instructor in the ENGE department and would like to thank the many students I have had and the ENGE department who not only funded me but provided guidance during one of the best parts of my time at Tech. I enjoyed teaching and learned a great deal. I would like to thank Dr. Butler and Dr. Reid, who
likely do not realize, through their example alone taught me a great deal. I would especially like to thank Dr. Butler for always providing a practicing engineer’s perspective on matters.

My wife, Leah, deserves much of the credit. She took the dive with me and supported me throughout. Without her I likely would not have made it through that first semester.

To my family: my brothers, sister, nieces, and nephews have all served as inspiration along the way whether they were aware or not. This includes my wife’s family, particularly her parents, who have been a great help along the way.

Finally, and most importantly I owe thanks to my parents, who from the youngest age supported me. I truly grew up in a home where I could build, take apart, try, and do most anything a future engineer would want.
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Chapter 1

1 Introduction and Literature Review

The idea of morphing aircraft control surfaces is as old as manned flight itself and was born of necessity but quickly died out in favor of the simpler hinged flap. The idea has seen periodic resurgence through history, mainly limited to research and with few designs making it to production. Most recently, morphing aircraft control systems have seen a resurgence as the development of small unmanned aerial vehicles (UAVs) offers the perfect low-cost, low-risk platform. Interest has also grown as the aviation commercial transport industry seeks increasingly efficient aircraft designs. These same aircraft manufacturers also see increased pressure to reduce the noise signature of the aircraft as housing moves closer to ever-growing airports. This dissertation looks at a design meant for manned commercial transport aircraft, stepping far outside of the UAV scale of aircraft and unlike many proposed designs, it uses a hydraulically driving actuators taking advantage of existing hydraulic systems common to all modern commercial transport aircraft.

The actuation for the proposed design comes from Flexible Matrix Composite (FMC) actuators, often thought of as hydraulically powered artificial muscles. Unlike conventional servos or hydraulic pistons which require highly constrained rigid tracks, hinges, or connector rods the FMC actuator can continue to actuate while bending and morphing with the structure only needed to be secured at the two ends. The FMCs being considered in this research operate at hydraulic pressures commonly found on modern commercial transport aircraft.

This literature review will start by looking at the general concept of morphing aircraft with previous examples ranging in size, flight speed, and technological maturity. The focus is on designs that consider morphing for aircraft control purposes. The section following will look at the development of the FMC actuator, its predecessors, and current models.
1.1 Morphing Aircraft Concepts

The term morphing as it relates to aircraft has been applied to a wide range of ideas and designs. In the most general sense morphing is anything which changes the outer surface of the aircraft for control or improvement in some performance metric like increased flight envelope, efficiency, or noise reduction. Typically, this excludes devices which create discontinuities in the surface like a conventional hinged control surface common to almost all modern aircraft.

Currently, research in morphing aircraft concepts ranges in maturity from the earliest concepts to manned flight-testing, and in scale from hand-launched UAVs to small business jets. Just as varied is the approach to the design of the morphing structure and chosen actuator. Morphing on a larger scale that involves significant changes in the camber [1-3], wingspan [4-7], twist [8, 9], or sweep [10, 11] has been profoundly investigated since the early 90’s with NASA’s Morphing Project [12]. Barbarino et al. published a compressive review of these technologies through 2011 [13] and more recently by Sun et al. [14]. Although not the focus of this work, morphing for rotary wing aircraft is an important branch of aviation-related morphing and Chopra and Giurgiutiu offer a review of those technologies [15, 16].

While many have looked at morphing large portions of the aircraft’s wing, including changes in the planform and twist, these all require significant reworking of how the aircraft is designed and manufactured. A potential nearer term goal is the implementation of morphing control surfaces. This approach calls for less change in the design process and with some cases, like that presented in this paper can directly replace conventional control surfaces even using existing hydraulic power sources.

A morphing control surface has two main advantages over a conventional hinged surface. By eliminating the surface discontinuity, the aerodynamic characteristics of the profile can be improved, and the acoustic emissions can be reduced. The following subsections will each look at a different concept of morphing aircraft control surfaces. Each has a different approach, method of actuation, reason for pursuing morphing, and different design cases.
1.1.1 The Origins of Morphing

Commonly the Wright Brothers, Orville, and Wilbur are referenced when discussing the origins of wing morphing. The Wright Brothers who are credited with being the first to have controlled powered flight overcame issues of roll control using morphing for their 1903 Wright Flyer (Figure 1.1)[17].

![Figure 1.1: Picture of the 1903 Wright Flyer's Maiden Flight December 17th, 1903, Kitty Hawk NC][18]

The biplane box structure of the Wright Flyer had relatively low torsional stiffness. Instead of working to stiffen the structure further, they attached control lines running diagonally across the wing’s structure. This allowed the pilot to twist the wings as a means of roll control (Figure 1.2). The brothers received a patent for this aspect of the Wright Flyer and referred to it as wing warping [19].
While the Wright Brothers are certainly the first to employ wing warping successfully, the idea itself predates their work. Weisharr et al. present some writings and a drawing of Clements Ader’s Eole aircraft [20]. Ader’s proposed different roles for military aircraft and described a wing that could change span in flight to allow the aircraft to achieve different speeds.

By 1915 wing warping was out as the Fokler Eindecker was the last production plan to use wing warping and even Orville Wright had begun using the conventional aileron for roll control in his designs.
When discussing the origins of morphing bioinspiration is another aspect often referred to. While it is impossible to know whether a designer is inspired by nature or if simply the designer converged to a similar optimal design as nature, many pioneers of flight like the Wright brothers and Otto Lilienthal continually studied birds as they designed their aircraft [22, 23]. Morphing aircraft concepts are clearly similar to many birds. Figure 1.4 shows a gull wing morphing UAV, and a seagull with different wings positions and Figure 1.5 show the NextGen MFX-1 concept UAVs morphing between different planform configuration and a peregrine falcon performing a stoop.

Figure 1.3: Clement Ader’s Eole concept from 1890 [21]

Figure 1.4: Left) The WhoopingMAV with varying gull wing positions Right) Seagull with wings positioned for soaring and diving [24]
Figure 1.5: Left) NextGen MFX-1 UAV showing the two extremes of planform change[25], Right) A peregrine falcon going from soaring to a steep dive in an action called a stoop

1.1.2 VCCTEF

The NASA Variable-Camber Continuous Trailing-Edge Flap (VCCTEF) concept originally proposed by Nguyen [26] and has been studied by multiple groups [27, 28]. The concept is based on NASA’s Generic Transport Model (GTM), a model representative of a modern commercial airliner, without proprietary geometry. Instead of using the conventional control surfaces of the GTM, the VCCTEF uses a series of distributed control surface across nearly the entire trailing edge (Figure 1.6B). Each of these control surfaces is then divided into several hinged segments in the chord direction (Figure 1.6A). When covered with a highly compliant skin the discrete hinged control surfaces become one continuous morphing surface.
Rodriguez et al. used the VCCTEF model and optimized flap positions for different off-design conditions [29]. The idea is that a conventional wing is only optimal for one specific flight condition, typically midcruise. As the aircraft burns fuel and changes weight the optimal shape of the wing will also change from the beginning cruise condition to the end. It was shown that the VCCTEF could optimize for each of these conditions by adjusting the flap positions (Figure 1.7A). This results in a more optimal distribution of lift when compared to the GTM with conventional control surfaces. Figure 1.7B shows the $C_p$ distribution of the conventional and morphing concepts and clearly showing a smoother more distributed pressure for the morphing design.

Figure 1.6: A) three-part flap of the VCCTEF. B) Continuous trailing edge flap of the VCCTEF with exaggerated deflections. [29]

(Figure 1.7: A) Optimized flap position for the VCCTEF at four different flight conditions. B) Pressure distribution on the top surface for the GTM and optimized VCCTEF at midcruise [29].
1.1.3 Piezoelectric Actuated Morphing Designs

Many projects have looked at the feasibility of integrating piezoelectric materials into a wing’s structure for morphing flight control. One example of this is a project undertaken at Virginia Tech to create the first fully solid-state piezoelectric controlled aircraft [30]. The small RC aircraft used Macro-Fiber Composite (MFC) piezoelectric patches embed on the wing and tail surfaces for actuation (Figure 1.8). The aircraft was flown for several test flights progressively building toward all conventional control surface being replaced with the morphing MFC control surfaces. The aircraft was able to fly, but each test flight was problematic and resulted in an uncontrolled ending (Figure 1.9). This was typically the result of hysteresis innate to the MFC, causing a lag between the pilot input and the response leading to excessive oscillations.

Figure 1.8: MFC control surfaces on the main wing and tail [30].

Figure 1.9: Morphing aircraft with MFC actuators in flight [30]
Further work has looked at maturing this technology, eliminating many of the issues during the flight test and seeking an optimal design in the more controlled environment of a wind tunnel. The Spanwise Morphing Trailing Edge (SMTE) concept took the TE of a finite wing section and divided it into modular section alternating between a controlled morphing section and a passive skin section to transition between morphing sections [31]. This allows the distribution of lift along the span to be easily controlled. This work is focussed on development for UAVs since they typically experience a more significant change in flight conditions (relative to their flight speed) and loadings. Examples include wind gusts equal to flight speed, and a sudden change in weight for payload delivery UAVs.

![Figure 1.10: The SMTE concept using alternating sections of MFC controlled morphing and passive transition sections. [32]](image)

The appeal of piezoelectric driven morphing control and other solids state approaches is clear. To simply be able to embed these relatively thin and light actuators seamlessly into a structure, with no moving parts, and be able to distribute and control the morphing shape is the goal of morphing, but there are limits to this approach. Typically, the driving voltage for piezoelectric is in the kV range, raising concerns about safety, though it should be noted the current required is small making the power requirements reasonable. Additionally, the amplifiers and associated systems needed to generate these voltages add to the weight and complexity of the system. Most all examples of these are limited to UAV scale applications. The forces and large deflections inherent to commercial transport aircraft is not something these systems can feasibly be scaled to accomplish.
1.1.4 UAV Flight Control With Twisting Acutation

When considering morphing for flight control, there are three fundamental shape changes that can be done. A change in the camber of the entire airfoil, change in camber about a small portion (typically the TE) or changing the angle of attack [33]. A group at the University of Kentucky looked at demonstrating a morphing wing UAV in flight using this last approach of changing the angle of attack or twisting the outboard section of the wing with the goal of autonomous flight control.

![Figure 1.11: Approaches to morphing for flight control: camber change (left), local camber change (center), and angle of attack or twist (right) [34].](image)

The design used an internal structure with sufficient bending stiffness to carry the wing loading, but with very low torsional stiffness for actuation. This structure was then cast in soft foam to form the aerodynamic surface (Figure 1.12 left). The torque tube, also serving as the spar was only bonded to the outermost rib and not the foam allowing a continuous curvature of the surface during actuation. Inboard of the morphing section the torque tube was constrained by bushings and actuated with a conventional servo (Figure 1.12 right).
Figure 1.12: Internal structure of morphing control surface (left) and the complete control surface with lower access panel removed (right) [33].

This design allowed the UAV to twist each wingtip independently for roll control (Figure 1.13). An aerodynamic and control model was developed for the UAV. Through a series of test flights control was transitioned from an RC pilot to fully autonomous flight control [34]. Onboard cameras were used to record the deflected shape, and these were compared to aerodynamic models and ground tests (Figure 1.14) [33, 35].

**Table: Warping Wing Configuration**

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Span</td>
<td>73 in.</td>
</tr>
<tr>
<td>Length</td>
<td>56 in.</td>
</tr>
<tr>
<td>Chord</td>
<td>10.5 in</td>
</tr>
<tr>
<td>GTOW</td>
<td>10.5 lbs.</td>
</tr>
<tr>
<td>W/S</td>
<td>31.6 oz/ft^2</td>
</tr>
<tr>
<td>Dihedral</td>
<td>0”</td>
</tr>
<tr>
<td>Airfoil</td>
<td>USA-35B</td>
</tr>
<tr>
<td>Avionics</td>
<td>ArduPlane</td>
</tr>
</tbody>
</table>

Figure 1.13: UAV with wing warping tips during ground deflection and maiden flight [33].
1.1.5 FishBAC

The Fish Bone Active Camber (FishBAC) concept uses a monolithic flexible structure which resembles the skeleton of a fish creating a compliant structure which can be morphed [36, 37] (Figure 1.15). The surface is created from pretension elastomer, and the structure is controlled through tendons being actuated by a conventional servo. Through a fluid-structure interaction model and later through wind tunnel testing, a 305 mm chord FishBAC prototype was shown to have 20-25% increase in L/D when compared to best-case conventional flap airfoil with sealed gaps and no external control horns or protrusions from the surface [38].

![FishBAC prototype and concept](image)

**Figure 1.15:** A) FishBAC prototype and B) FishBAC concept [36].

Several different ideas for the structural layout of morphing control surfaces have been investigated; a brief history of these was presented by Woods *et al.* [39] including the DARPA Smart Wing which uses a layout similar to the one ultimately used for the morphing spoiler concept presented in Chapter 2 [40].

A multi-objective optimization using genetic algorithms of the FishBAC was done [41]. Mass, drag, and actuation energy were considered as objectives and design variables allowed the thickness of various components, number of stringers and the position of the morphing...
section to be varied. The analysis consisted of XFOIL for the fluid solution and a Euler-Bernoulli beam model to represent the skin and stringers stiffness. Serval Pareto fronts from the optimization were presented. Figure 1.16A shows the Pareto front for the resulting drag and energy results. The optimization had difficulty populating the lower drag portions of the design space; this was attributed to issues innate to XFOIL. The authors note that change in drag is much smaller than that seen for energy and mass for the Pareto efficient points. The results for energy and mass (Figure 1.16B) shows a better-populated designs space and a clear tradeoff is in the relationship between mass and energy. Energy is assumed here to be proportional to the torque seen at the tendon pulley. This relationship in actuation energy and mass of the structure is similar to that found as part of this dissertation for a morphing FMC aileron in Chapter 5.4.

![Figure 1.16: Pareto plots of optimal solutions for the FishBAC concept for the objective of mass, drag, and actuation energy.](image)

1.1.6 FlexSys

Recently FlexSys Inc. conducted a series of flight tests of a morphing control surface fitted to a NASA Gulfstream G-III business jet as part of the Adaptive Compliant Trailing Edge (ACTE) program [42]. FlexSys claims a 5-12% increase in range due to the aircraft always being able to maintain an optimal configuration as the aircraft’s weight decreases due to fuel burn [43]. At this time the exact method used for morphing is proprietary, but the company has filed several patents [44, 45].
These flight tests build on years of CFD and structural analysis done by FlexSys including multiple wind tunnel prototypes and a flying test article for aboard the White Knight aircraft. The tests, as part of the ACTE flights, tested the control surfaces up to Mach 0.75 and 18 kPa dynamic pressure. The test article was not equipped with actuators. Before each flight, the control surface was set at a fixed deflection. Tests were done with the ACTE at angles ranging from -2° to 30°. The 19-foot ACTE replaces the GIII Fowler Flap system, it was noted in the flight tests at larger flap angles and speeds the flow separated from the control surface. This was predicted in analysis and measured in flight by hinge moment and loads measurements. The lack of slats created in a Fowler flap system caused the detachment; authors noted that some “flow augmentation (ex. synthetic jets)” would be needed to maintain flow attachment[42]. FlexSys has estimated that a medium range transonic transport aircraft fitted with their technology could offer a 3.3% improvement in L/D saving approximately 100 gallons of fuel for a cross-country flight [46].

1.2 Fluid Driven Actuators

In modern commercial transport aircraft, the ubiquitous solution for actuation is the hydraulic cylinder. Figure 1.18 shows a typical setup for a conventional hinged control surface actuated by a hydraulic cylinder. Hydraulic cylinders have seen extensive use in many industries making them a reliable option. Huber and Ashby compare various conventional and smart mechanical actuator types and show that hydraulic actuators have
nearly unmatched capability in strain offering 100% strain capability and some of the highest specific power outputs and efficiencies [47].

Figure 1.18: Cross-section of a typical aircraft control surface area showing hydraulic cylinder (16) [48]

It was noted Woods et al. in the development of the FishBAC that simplicity should be a core driver of a morphing design not because simplicity in and of itself is important but is indicative of other important qualities like cost, maintainability, and reliability [49]. Often morphing designs seek a “smart structure approach,” and Woods et al. note that these typically call for high fields like thermal, magnetic, or electric fields. These high fields inherently run counter to the goal of simplicity, causing particular concern for the safety and reliability of the system. The Flexible Matrix Composite (FMC) actuator presented here offers an efficient actuator with the unique ability to bend and morphing with the structure without the “high fields” common to other smart structures. The FMCs used in this work operate with similar hydraulic fluid and pressure to what is currently used in commercial transport aircraft.

FMCs are muscle-like actuators that use helically arranged stiff fibers (i.e., carbon fiber) embedded in a soft polymer. These stiff fibers generate a longitudinal force when the actuator is internally pressurized with a working fluid (e.g., air, water, oil). The FMC is similar to the Pneumatic Artificial Muscle (PAM) and McKibben’s muscle. Physician Joseph McKibben is often created with the first application of these types of actuators, applying them to a prostatic hand for his daughter who suffered from polio in the 1950s [50].
In the 1980’s the Bridgestone Corporation began marketing the first commercial use of a fluid-driven muscle, the “rubbertuator” [51]. It was marketed as an actuator for robots operating in potentially explosive environments since no electrical components are needed. Bridgestone stopped their “rubbertuator” work in the 1990s. Currently, the FESTO Corporation has a commercial off the shelf option for pneumatic muscles [52]. A review of previous work on FMC actuators and similar actuators through 2011 was presented by Zhang [53].

Unlike PAM actuators which rely on a dry braided fiber sleeve with an internal elastomeric bladder to form the actuator, the FMC can be fabricated using a wet filament winding process (Figure 1.19A). This allows the fiber to be precisely placed at different angles about the axis of the mandrel. Actuators with wind angle greater than 54.7° extend when pressurized, while actuators with fiber wind angle less than 54.7° to the axis contract (Figure 1.19B). Twisting actuation can be achieved by winding a single family of fibers in the same direction. In addition to being able to control the type of actuation (i.e., extending, contracting or twisting), the fiber angle can also be used to tailor the output force and strain for a given pressure, similar to the effect of a fulcrum in a lever.

![Figure 1.19: A) Wet filament winding process B) Effect of fiber wind angle on actuation.](image)

FMC actuators can operate using either liquid or gas as the working fluid. It has been shown that when the FMC actuators are filled with a high bulk modulus fluid, they can exhibit stiffness modulation through control of the valve which allows fluid to enter or exit the actuator [54]. Figure 1.20 shows that different FMC designs (open green circles) can cover a wide portion of the design space when looking at the lower modulus and modulus modulation when compared to several other variable modulus materials. This allows FMCs
to not only act as active actuators for a morphing structure but can also add controllable passive stiffness or damping.

![Figure 1.20: Stiffness modulation vs. lower modulus for FMC actuators and several other variable modulus materials [54]](image)

The more board category of fluid driven artificial muscle has seen applications in prosthetics and robotics. For both of this application, an actuator operating with gas as compared to a much higher bulk modulus liquid allows the actuator to be compliant. Figure 1.21 shows a KAFO (knee-angle-foot-orthosis) using PAM actuators which provide the compliance to the joint, though the limit of antagonistic pairs of the actuators limited the range of motion of the joint (Figure 1.21Left)[55]. Figure 1.21 also shows the FESTO humanoid robot; the black parts are the fluidic actuators. Note the differing sizes of the actuators in the chest of the robot. The easily scalable and tailorable nature of these actuators allows each “muscle” of the robot to optimize its functions. The compliance of the actuator allows the motions of the robot to be smoother not have the stick-slip phenomena common to conventionally actuated robots [56].
FMCs have seen some application in UAVs for morphing purposes. Heim et al. demonstrated a morphing flap on the eSPAARO UAV; a university developed platform for testing UAV control systems, and component testing [57, 58]. For each morphing flap, a single long actuator was used and then arranged in parallel runs to form a single contracting surface which was then cast in soft foam (Figure 1.22). The control surface was integrated with an eSPAARO aircraft and demonstrated through a series of flight tests (Figure 1.23). The relatively small loads of this application allowed the use of a single FMC and to operate on compressed gas rather than hydraulic fluid.
Figure 1.22: The completed FMC morphing flap at 0 and 7 cm deflection (left). Single FMC in parallel runs before casting (right) [57]

Figure 1.23: eSPAARO FMC flap with 7 cm deflection and the ground track of the test flight [57]

Initial efforts to model the behavior of pneumatic artificial muscles depending the virtual work principle, namely that the work done by the actuator has to be equal to that of the work done of the fluid entering [59]. For this to work a relationship needs to be developed between the length of the actuator and the fluid volume. Assuming the wall of the actuator to be thin, the fibers to be inextensible, and the actuator to be a perfect cylinder throughout its length,
Equation (1-1) results. Where \( F \) is the force generated, \( D_0 \) is the diameter when the wind angle is 90°, \( P \) is the internal pressure, and \( \theta \) is the fiber wind angle [59].

\[
F = \frac{\pi D_0^2 P}{4} (3 \cos^2 \theta - 1)
\]  

(1-1)

This approach is limited since it does not consider many clearly important aspects of the actuator, namely assuming the actuator to be a cylinder. The ends of the actuator are typically swaged in an end fitting which forces each end to maintain the original diameter during actuation. A review of modeling approaches for McKibben muscles was done by Tondu [60].

A model proposed by Shan et al. allows for nonlinear analysis FMC actuators accounting for many of the nonlinear effects not addressed in other models [54]. The model takes into account the material as well as geometric nonlinearities associated with the large change in shape the fibers and membrane undergo. This eliminates the need to model the actuator as a perfect cylinder throughout its length and accurately capture the end effects. The model also allows the extensibility of the fibers to be considered. The model was implemented in the analysis of a sheet of parallel FMC tubes being used for a variable stiffness structure [54] and as part of a novel actuator inspired by plants fibrillar networks with the internal pressure controlled by electroosmotic transport mechanism [61].

1.3 Motivation and Dissertation Outline

Each of the previously discussed designs undoubtedly contributes to different aspects of morphing for aircraft control. Each also has its limits, creating small gaps in the technology. That is the motivation of this dissertation. The Wright brothers and bioinspiration started the very idea of morphing, but their wing warping was born more of out of necessity and opportunity than purposeful design and was quickly replaced by the conventional aileron as flight speeds increased. The VCCTEF studies show the possible benefits in commercial transport aircraft, but it is limited to a paper study only. Little consideration is given as to exactly how the morphing surfaces could be designed, actuated, and implemented. The FishBAC concept and others tested on UAVs provide insight into different designs and novel ways to go about morphing, but the scale of aircraft, which allows that line of research to be so dynamic also limits its application in manned flight. The FlexSys ACTE flight tests show
promise in the future of morphing designs for manned aircraft, but many questions remain unanswered about the actual design. Unlike the previously mentioned morphing projects scaled for UAVs, the ACTE offers a straightforward application to manned flight, but this also hampers the ability of the research to explore new designs. Though little is known about the inner workings of the ACTE, it is known that the design uses “conventional” actuators. The next generation of morphing designs should not be limited to actuators of the past. This work aims to explore a morphing aircraft design for commercial transport aircraft, using novel methods of actuation, while considering and demonstrating the design beyond paper studies alone through benchtop prototypes.

This dissertation consists of six chapters, which are organized as follows.

The first chapter introduces background information on morphing aircraft specifically looking at control surfaces, actuation methods, and structural layout. Additionally, background on the flexible matrix composite actuator (FMC) is presented.

The second chapter discusses the use of FMC in a morphing active spoiler for use in a commercial aircraft. A functional prototype was fabricated and tested demonstrating, that FMCs can be used in morning control surface.

Based on the concept of the active spoiler, the third chapter looks at the challenges of designing an FMC morphing aileron. An analysis combining structural, fluid, and actuator models is presented.

The fourth chapter takes the model developed in the prior chapter and looks at the effect that each of the design variables has on the morphing aileron. Trend plots show how the design changes throughout the design space.

The fifth chapter looks at how to optimize the morphing FMC aileron for a particular design case and presents not only the solution to this particular design case but also guidelines for adapting the FMC aileron to other control surface position and design cases.

The final chapter summarizes the work done and looks at possible areas for future work.
Chapter 2

2 Design and Testing of a Morphing FMC Spoiler

One of the motivations and goals of this research is to demonstrate the morphing FMC actuator control surface concept through full-scale benchtop prototypes. This chapter will look at the development of a high performance morphing spoiler for air gap control with a deployed Fowler flap using pressurized flexible matrix composites (FMC) actuators (Figure 2.1). This design takes the surface normally used as a spoiler to dump lift and allow it to control the gap between it and the deployed Fowler flap for increased flow control. More specifically, the objectives are to (1) design an FMC morphing spoiler control surface, (2) fabricate a full-scale prototype, (3) achieve the performance requirements under expected aerodynamic loading, and to (4) demonstrate closed-loop control for position control. For objective (3) based on requirements provided by the sponsor, the spoiler needed to be able to achieve 12 cm of tip deflection under anticipated aerodynamic load with closed-loop control. Using a morphing spoiler allows the gap between what would normally be the fixed portion of the wing and a deployed Fowler flap to be controlled. This has the potential to increase flap performance, reduce acoustic emissions and reduce overall system weight by potentially eliminating a panel from the Fowler flap system and reducing the size of the kinematic tracks mechanism.

The approach for meeting the technical objectives are to employ a series of finite element (FE) models to analyze, perform studies, and access performance considering different configurations to yield a final design. Next, the fabrication and testing of individual FMC actuators were performed to characterize the actuation response and test for failure before being installed in the prototype spoiler. The final phase of the work was the fabrication and testing of a full-size prototype morphing spoiler. A series of tests were performed to look at
the spoiler’s: passive stiffness, deflection under different simulated aerodynamic loads, and closed-loop control under different loading conditions.

![Figure 2.1: Morphing Spoiler Concept](image)

### 2.1 Model-Based Design

As specified in the performance requirements, deflection was required in only one direction for the morphing control surface, and thus the basic concept for the morphing spoiler control surface used extending actuators near the top surface and contracting actuators near the bottom surface (Figure 2.1). Several versions of the morphing spoiler design were considered utilizing a series of finite element models, and the results of these models guided the design process leading to the final design that was fabricated and tested.

The morphing spoiler has three basic sections as highlighted in Figure 2.2. The manifold assembly provided the hydraulic pressure and mounting point for the FMC actuators. The active portion of the actuators were embedded in a deformable material and thus referred to as the active section in this paper. As the spoiler’s thickness decreases approaching the trailing edge, there was a location where the FMCs could no longer physically fit. At this location, the rigid trailing edge fairing completes the profile.

The FE model used a 2D analysis of the spoiler employing Abaqus CAE plane strain elements (CPE3 and CPE4) using the nonlinear geometry solution. Using this 2D approach allowed for multiple designs to be considered efficiently with minimum computational time. The spoiler’s profile was first partitioned into the three basic areas of the spoiler: manifold, an active region, and trailing edge section (Figure 2.2). Attachment points were also designated on the manifold and trailing edge section that allowed following surface traction forces to be applied representing the FMC actuators. The spoiler is 50 cm in the chord.
direction, and the position of the FMC aft mount points was set at 38 cm aft of the manifold where the spoiler has a thickness of 4 cm. This is a minimum thickness of the spoiler at which the FMCs could be mounted due to space constraints.

![Diagram of FE model](image)

**Figure 2.2: Partitioned and meshed FE model**

The intent of this analysis was to determine the material and structural layout that generated the required forces and displacements to meet performance specifications. For this reason, the FMCs themselves were not modeled, but rather loads were applied at the attachment points. Once loads were determined, then estimations of required actuation pressure could be determined using actuator characterization data collected from experiments.

The first design considered the entire active section cast in a polyurethane rubber (Shore A-80 hardness, $E=6.7$ MPa). The first loading condition considered a blocked condition where the trailing edge was constrained to have zero vertical motion (Figure 2.3A). As the loads representing the FMCs were increased, the resulting force generated at the trailing edge was measured. Before the loads could be increased to even a small fraction of the FMC's maximum output, the model showed significant undesired deflections (Figure 2.3B).

![Images](image)

**Figure 2.3: A) Boundary conditions and loads for blocked case. B) First design iteration showing undesired deflection**
The results confirmed that the structure needed additional stiffness specifically to direct more of the FMC work into a bending moment to achieve the desired actuation. For this reason, a plate modeled as a 2D beam made of 3 mm carbon composite was added between the manifold and the trailing edge section (Figure 2.4A). The remaining portion of the active region was left as the polyurethane rubber.

Under the same blocked condition, the addition of the centerline plate allowed the model to achieve 7.3 kN of force per meter span at the tip with loads representing the maximum force output of the FMC actuators with a spanwise spacing of 2.5 cm. The anticipated aerodynamic loads are 1.6 kN of force per meter span. The undesired deflections seen in the first model were also eliminated.

A major challenge in developing a morphing airfoil is creating a structure that is stiff to handle the aerodynamic loading but compliant to reduce the actuator requirements. Therefore, studies were performed to increase the passive stiffness of the spoiler without significant impact on the required actuators forces to overcome the added stiffness. Ideally, the active region would have significantly higher stiffness in the vertical direction to resist aerodynamic loading than in the horizontal direction, which the FMCs operate. It was discovered that placing aramid honeycomb into the polymer during casting with the cells of the honeycomb aligned vertically, significantly increased the polymers stiffness in the vertical direction (Figure 2.4B). From tensile tests of the new composite using the same Shore A-80 hardness polymer and a 3 mm cell diameter honeycomb, the new composite material stiffness was determined to be approximately $E_x = 20$ MPa in the horizontal direction and $E_y = 200$ MPa in the vertical direction. A final FE model was created with the active region of the spoiler having the material properties of this new aramid honeycomb polyurethane composite material along with the centerline plate used in the second model.

![Figure 2.4: (A) Model with composite centerline plate. (B) Polymer and honeycomb composite](image)

25
Similar to the second model, this third model was analyzed to determine its blocked force and passive stiffness. Due to the increase in the horizontal stiffness of the honeycomb polyurethane composite, the blocked condition produced 7.2 kN per meter span, only a 1% reduction. However, the passive stiffness in the vertical direction of the design was increased from 24 kN/m for a unit meter span to 27 kN/m, a 12% increase. Additional design studies revealed that the best configuration for achieving large deflections under the different loadings scenarios reflected the final design in Figure 2.4A, which consisted of the FMC actuators embedded in the honeycomb polymer composite with a carbon fiber centerline plate.

Validation of the FE model was accomplished through static and dynamic tests of the spoiler prototype. For the dynamic tests, a modal analysis was performed on the final model to identify the spoiler’s natural frequencies and mode shapes. The analysis showed a first and second natural frequency of 32 Hz and 95 Hz, respectively. The corresponding mode shapes can be seen in Figure 2.5. With the final design meeting the performance specifications, the fabrication and validation of a bench top prototype spoiler followed.

![FE modal analysis results for the first two natural frequencies](image)

**Figure 2.5: FE modal analysis results for the first two natural frequencies**

### 2.2 FMC Actuator Fabrication

Flexible matrix composite actuators were fabricated in the Aerospace Structures and Materials Laboratory (ASML) using two fabrication methods. Both extensional and contracting actuators can be fabricated using a wet-filament winding process where carbon fiber tow is wetted with a polyurethane resin and placed at precise angles onto a mandrel. After curing, the actuators are removed from the mandrel and fittings are attached to the ends. Examples of wet-filament actuators are shown in Figure 2.6A. More than 70%
Actuation strain has been achieved with the extensional actuators, and 25%-30% actuation strains are typical with contracting actuators. Another fabrication method for creating FMC actuators uses a dry braided sleeve with an inner bladder that is later cast in a polymer resin, forming the FMC actuator. The advantage of this technique is the increased bond strength of the dry sleeve with the metal fitting using epoxy (e.g., 3M DP460). Under high loads and pressures, the wet-filament wound actuators are more likely to fail due to the poor bonding between the composite laminate having a soft polymer and the metal fittings. The actuators selected for the spoiler were sized (diameter) based on the anticipated force requirements in the FE model. The contracting actuators fabricated using the dry braided sleeve method had a wind angle of ±15° which has been shown to provide large forces and displacements [62]. All actuators are pressurized to approximately 5 MPa in a blocked setup prior to casting into the final system, and forces exceeding 4000 N are typically measured. Figure 2.6B shows a typical force vs. displacement curve for an FMC actuator. The lines of constant pressure were obtained using a pneumatic regulator to maintain constant pressure. For safety concerns, the blocked values at the higher pressures were obtained using a hydraulic hand pump. However, due to limitations of the hydraulic setup, it was not possible to get lines of constant pressure at values greater than 1.0 MPa.

Figure 2.6: (A) Example FMC actuators – actuators are fabricated using wet-filament winding process, and dry braided sleeves later cast in a polymer resin and (B) Typical force and strain results for an actuator at constant pressure and in a blocked condition.
2.3 Active Spoiler Prototype

To demonstrate the use of FMCs in the morphing control surface a functional, full-scale prototype of the active spoiler was created. Through a series of benchtop tests, the spoiler was shown to meet the design requirements for deflection, load, and to be closed-loop controlled.

2.3.1 Design and Fabrication

The morphing spoiler prototype was fabricated to match the full-scale profile with a length of 0.6 m. The span dimension was set at 0.13 m, representing a portion of the full span. The active spoiler consists of a custom manifold, FMC actuators, perforated carbon fiber centerline plate, and an aramid honeycomb core (Figure 2.7A). An aluminum mount was CNC machined for mounting the active spoiler, and a rapid-prototyped tail section was fabricated using 3D printing. In this prototype, five extensional actuators were placed on top, and the same number of contracting actuators were placed on the bottom (Figure 2.7B). The extensional actuators were made from the wet filament winding process, and braided aramid sleeves were used for the contracting actuators. Based on the FE models a centerline carbon fiber plate was added. Before casting the polymer to form the spoiler the region between the centerline plate and the FMC actuators was filled with strips of 3 mm diameter 12.7 mm thickness aramid honeycomb (Figure 2.7C). For casting, a mold of the spoiler’s outer mold line was 3D printed and the active region filled with PMC-780 polyurethane rubber. During curing, the entire section was kept under vacuum to remove entrapped air and ensure that honeycomb had no voids. A black pigment was used in the polymer, and the spoiler tip was painted black for better visibility (Figure 2.7D). The final mass of the active section was 8.29 kg.
2.3.2 Quasistatic Experiments

The first tests performed with the prototype were quasi-static tests to determine the relationships between hydraulic pressurization of the FMC actuators, tip displacement, and force. In these tests, the FMC actuators were slowly pressurized using a hydraulic hand pump; the tip displacement was measured using an ADMET testing frame extensometer, and the pressure measured with an Ashcroft 2000 psig pressure transducer. The results showed a nearly linear relationship of 1.6 cm/MPa and the spoiler was able to achieve 12 cm of tip deflection (Figure 2.8A).

Using the ADMET testing frame with a 4.4 kN load cell, force measurements were taken with the active spoiler blocked at three different tip positions, i.e. 0 cm, 6 cm, and 10 cm, as shown in Figure 2.8B. The spoiler was attached to the load cell on the ADMET testing frame using a carbon fiber cable. This cable was then connected at the end of the active section, just before the rapid prototyped tail section, as shown in Figure 2.9A. Similar to the displacement measurement, the hand pump was used to slowly pressurize the actuators...
while the testing frame held the spoiler stationary. Pressure and force were simultaneously measured. In most cases, 400 N stationary force was achieved, which is approximately 2.5 times the estimated aerodynamic loading for a similarly sized spoiler in the span and chord directions.

![Figure 2.8: (A) Tip free displacement with increased actuation pressure and (B) measured force of the spoiler at different constrained tip displacements](image)

Passive stiffness experiments were performed to determine the unpressurized stiffness of the spoiler. The test setup shown in Figure 2.9A was used to obtain the force to displacement measurement without pressurizing the actuators. The force was recorded at the end of the spoiler was raised using the testing frame with the carbon fiber cable. The results show that there was very little tip deformation even with a 400 N force. The experimental results showed the spoiler to be 12% stiffer than the FE model, and one possible reason is that the stiffness of the actuators was not included in the model (Figure 2.9B). Even though the deflections were small, one can use the actuators to compensate for any undesired deflections and return the spoiler to the original configuration. During deflections, some spanwise deflections were observed due to Poisson’s effect during larger deflections of the spoiler and these deflections were small relative to the spoiler’s width.
2.3.3 Dynamic Testing of the Active Spoiler and FE Validation

For the dynamic testing to validate the natural frequencies calculated from the FE model, the spoiler was excited by a shaker while the valve to the manifold was closed. Before closing the valve, the actuators were pressurized to different values to determine the effects of the internal pressure on frequency response. For these tests, the active spoiler was mounted inverted to allow for easier positioning of the shaker. A LDS V408 vibration shaker with PA100E Power Amplifier was used to excite the spoiler, and a PCB 221A04 force transducer (1.1218 mV/N) provided force measurement between the spoiler and shaker (Figure 2.9). Pressure via an Ashcroft 2000 psig pressure transducer was placed on a tee fitting between the closed valve and the spoiler manifold. Tip velocity was measured using a Polytec laser vibrometer (Figure 2.9). A National Instruments system with a PXI-4461 (24-Bit, 204.8 kS/s, 2-Input/2-Output) and a PXI-4462 (24-Bit, 204.8 kS/s 4-Input) data acquisition module recorded the force, velocity, and pressure as well as provides the output signal for the shaker. The Sound and Vibration Measurement Suite in LabVIEW 2010 was used for calculating the results, displaying the data, and recording the data for importing into Matlab. The sampling rate for the data acquisition system was set to 1000 Hz, a Hanning window was used for the spectral analysis, and a bandwidth-limited random noise signal was used for providing the excitation signal to the vibration shaker. A total of 10 averages were performed for computing the frequency response of the system at each internal pressure.
Figure 2.10. Experiment setup for dynamic system analysis

Prior to the start of each run, the actuators were pressurized with a hand pump to the desired pressure and then the valve shut. Figure 2.11A shows the response of tip velocity over the applied load from the shaker. It shows a first natural frequency for each pressure to be near 30 Hz, which corresponds well to the first bending mode of 32 Hz found in the FE model. Additionally, a second resonant frequency appears close to the 95 Hz predicted. Also, the natural frequency increases with increased initial pressure. In Philen (2012), it was shown that the internal pressure could be used for tuning the stiffness of an FMC semi-active isolation mount [63]. Similarly, these results confirm that the natural frequency of the active spoiler can be altered by adjusting the internal pressure. Figure 2.11B shows the response of the hydraulic pressure of the closed system over the applied load. While the first structural natural frequency of 30 Hz does not appear in the pressure response, the 95 Hz natural frequency can be seen in the pressure results, showing a stronger coupling between the structure and hydraulic fluid.
2.3.4 System identification and closed-loop results for the active spoiler

Due to the complexities in modeling the dynamics of the morphing spoiler, hydraulic pump, the servo-valve, and the response of the fluid in the hydraulic lines, it was decided to perform a system identification of the entire system from the input control voltage to the servo-valve to the output response of the morphing spoiler control surface. This also avoided the complexities in trying to perform system identification of each component, which would have required additional instrumentation unavailable in the lab.

To measure displacement during the tests, flex sensors manufactured by Spectra Symbol were bonded using flexible silicone to the top surface of the active section (Figure 2.12A). The flex sensors change resistance based on curvature. The flex sensors were calibrated using displacements measured with a string wound around a rotary encoder, giving a resolution of 0.6 mm. Instead of a hand pump, a compact hydraulic pump manufactured by Marathon Electric was used to drive hydraulic fluid through a Parker D1FM servo-valve. To evaluate the performance of the closed-loop system under loading, a spring (k= 8.75 kN/m) was attached to the same anchor point used in the in the blocked and passive stiffness tests. To measure the applied load, a 2.2 kN load cell was attached in line with the spring (Figure 2.12B).
Figure 2.12: Experiment setup for open and closed-loop results: (A) Spoiler showing location of flex sensor and attachment of string at tip of spoiler and (B) images of hardware for conducting the experiments

A dSpace™ control system using the 1103 board recorded the position from the rotary encoder, voltage from flex sensors using a voltage divider circuit, and pressure from the Ashcroft 2000 psig pressure transducer located near the spoiler's manifold. The dSpace system was used to control a Parker DFM1 servo spool valve for the open-loop and closed-loop experiments. Shown in Figure 2.13 is a picture of the active spoiler achieving 14 cm of tip displacement when unloaded.

Figure 2.13: Frame from video showing the active spoiler achieving 14 cm of tip displacement

Figure 2.14A shows the open-loop results for the unloaded spoiler with a sinusoidal input to the servo spool valve. For the loaded case with the spring attached, the spoiler was still able to achieve the 12 cm tip displacement requirement even under 600 N loading at max deflection (Figure 2.14B).
Figure 2.14: Open-loop results for spoiler under cyclic testing: (A) Unloaded: Displacement (solid blue), Pressure (dashed red) and (B) Loaded: Displacement (solid blue), Force (dashed red)

System identification was performed using input-output frequency response results to acquire a Single Input Single Output (SISO) analytical model which was used to design the feedback controller. To obtain the frequency responses of the active spoiler, the servo-valve was controlled using the National Instruments PXI system with a PXI-4461 (24-Bit, 204.8 kS/s, 2-Input/2-Output). The hydraulic system was excited with a bandwidth limited random noise signal with a -0.5 V bias being sent to the servo-valve. The bias prevents the continual slow pressurization of the system due to the natural leakage of fluid through the spool valves. The laser vibrometer and pressure transducers recorded tip velocity and pressure, respectively. The sampling rate for the data acquisition system was set to 1000 Hz, and a Hanning window was used for the spectral analysis. A total of 10 averages were performed for computing the frequency response of the system. Shown in Figure 2.15 are the experimentally obtained frequency response results and the identified analytical models. Figure 2.15A is the output tip velocity in reference to the applied voltage to the servo-valve, and Figure 2.15B is the output tip velocity in reference to the actuation pressure. The identified transfer function was computed using the prediction error estimate function (PEM) in Matlab and is shown in (2.1). This transfer function represents the output tip displacement to the input voltage to the servo-valve and was obtained by integrating the identified model shown in Figure 2.15A.
Figure 2.15 Frequency response of tip velocity to (A) servo-valve control voltage and (B) actuation pressure. Blue: Identified system model and Red: Experiment

\[
G_p = \begin{pmatrix}
-3.3763 \times 10^{-1} s^6 - 2.3458 \times 10^2 s^5 + 2.1858 \times 10^5 s^4 \\
-3.3237 \times 10^5 s^3 + 1.4227 \times 10^{11} s^2 - 4.8408 \times 10^{13} s^1 + 1.7652 \times 10^{16} \\
1.0000 s^7 + 5.5481 \times 10^2 s^6 + 5.9944 \times 10^5 s^5 + 2.2491 \times 10^8 s^4 \\
+8.3285 \times 10^{10} s^3 + 1.9515 \times 10^{13} s^2 + 2.3632 \times 10^{15} s^1 + 3.9101 \times 10^{17}
\end{pmatrix}
\] (2.1)

Using the root locus control design technique, a proportional controller having a gain of 5.6 was chosen based upon the desired bandwidth, pole locations, and step response. Shown in Figure 2.16A is the root-locus for the system, which highlights a stable control region for a gain range between 0 and 11.73. The effect of integral and derivative controllers was examined in the analysis, but the objective of this work was to demonstrate accurate tracking using a simple control system. Future work will investigate more advanced control algorithms. Figure 2.16B is the closed-loop frequency response for the simple proportional controller, where the input is the desired tip displacement, and output is actual tip displacement. Accurate tracking of the desired displacement can be achieved at low frequencies (input-output ratio is equal to 1 (0 dB)), and the bandwidth of the system is approximately 30 Hz.
Figure 2.16: (A) Root locus plot of identified plant in Eq. 1 (Maximum proportional gain 11.73) and (B) frequency response of closed-loop system with proportional gain of 5.6

The feedback control system for position control was programmed using Simulink and compiled with the Real-Time Workshop, which was downloaded onto the dSpace control system. For feedback, the calibrated flex sensors attached to the active spoiler were used to provide the position. Shown in Figure 2.17 are the closed-loop results for the active spoiler when tracking step displacements between 0 and 12 cm. Figure 2.17A is the tip displacement, and Figure 2.17B is the control voltage sent to the servo-valve. As seen, there is good agreement between the analysis and experiment and the system can effectively track the desired positions with a short response time.

For the loaded case the spring was again attached to the spoiler to resist its motion. The results in Figure 2.18 demonstrate that the spoiler can effectively track the desired positions under loading conditions. The small hysteresis seen in the deflection when returning to zero is primarily due to the limitations of the hydraulic system to drain the fluid during depressurization. Larger hydraulic lines or the addition of drain valves would mitigate this further. The spoiler was able to exceed 300 N at 10 cm of deflection. The small local noise seen in the results is primarily a result of the flex sensor used as the displacement sensor in the control loop. The flex sensor embedded along the top surface change resistance was converted to a DC voltage signal for dSpace through a voltage divider. Thus, the relatively small change in resistance and the use of a simple voltage divider circuit resulted in signal noise. However, the results demonstrate that accurate tracking control using a simple
control system is achievable even under pseudo-aerodynamic loading, and future work will explore additional control algorithms for improved performance.

Figure 2.17: (A) Closed-loop results for spoiler tracking step displacements between 0 and 12 cm: Desired displacement (solid blue), simulation results (red) and experiment (green) and (B) Control output to servovalve: simulation (blue) and experiment (red)

Figure 2.18. Closed-loop results for the loaded spoiler tracking step displacements between 0 and 12 cm: (a) Desired displacement (solid blue), Actual displacement (dashed red) and (b) Control output to servovalve (solid blue), Pressure (dashed red)

2.4 Active spoiler conclusions

This chapter looked at the design and demonstration of a high-performance actuation system for morphing spoiler control using pressurized FMC actuators. Various types of actuators were fabricated and tested under high pressures and loadings to examine their performance capabilities. Using simplified 2D finite element models, multiple spoiler
designs were considered. The models identified the benefit of the composite centerline plate to allow the FMC’s actuation to be efficiently converted into the desired morphing shape and the benefit of using the anisotropic materials properties of the honeycomb polyurethane composite to increase the passive stiffness of the structure.

A final prototype spoiler was created to evaluate the performance in a full-scale design. The final prototype was full scale in the chord direction and was cast to match the actual design scenario’s spoiler profile. The final prototype was first tested to determine its maximum displacement and performance under pseudo-aerodynamic loading. The second series of tests focused on the dynamic characteristics of the spoiler. Finally, a feedback controller was designed, and the spoiler was able to be controlled and perform well while loaded with pseudo-aerodynamic loads. To accompany this, an analytic model of the system’s response was identified to allow for further analysis of the spoiler’s performance.

This chapter demonstrated through a benchtop prototype that a morphing aircraft control surface actuated by FMCs is viable for commercial transport aircraft. To further study this a more detailed analysis model is needed which can take into account the nonlinear nature of the large deflections needed, the energy requirements of the FMCs, aerodynamic loading and allow for optimization of the design. Additionally, considering control surfaces which are bidirectional will allow more advanced designs to be considered.
Chapter 3

3 Model Development for the Morphing FMC Aileron

The active spoiler concept from the previous chapter demonstrated through a benchtop prototype and testing how FMCs can be used to create a morphing aircraft control surface for a commercial transport aircraft. Motivated by the active spoiler work, this chapter and following chapters look to further the FMC morphing aircraft control surface concept by developing a model which can be used to optimize the design.

This new analysis includes a coupled structural, fluid and actuator models and is intended to be a medium fidelity analysis to provide meaningful insight into the design and still be computationally efficient for use in the optimization. The coupled fluid model allows for the pressure on the morphing surface to be included in the structural model. An actuator model based on empirical data is used to accurately take into account the fluid volume, pressure, and energy requirements of the FMCs.

The model uses contracting actuators in antagonistic pairs to allow for deflection in two directions in both the up and down direction. This expands the possible applications from single direction actuation like the spoiler to now consider any flight control surface.

3.1 Design Case Definition and Design Variables

For this model, a specific design case was developed which focused on analyzing a morphing aileron FMC control surface for a commercial transport aircraft. The initial concept is similar to the previously discussed FMC morphing spoiler but with the upper extending actuators replaced by contracting actuators which then operate as antagonistic sets of contracting actuators. The aileron will deflect up when the top FMC set is pressurized and down when the lower set of FMCs is pressurized. The model is parametrized allowing for changes in the structural layout, material properties, FMC parameters, and loading...
conditions. This parametrized model allows many designs to be considered efficiently to determine an optimal design.

For this design case, the outboard section of the NASA Common Research Model (CRM) was selected (Figure 3.1). The CRM was developed by NASA to provide a standard set of geometries, wind tunnel test data, and computational models that researchers could use when researching modern medium size commercial transport aircraft since the geometries of current commercial aircraft are often proprietary. The main wing of the CRM uses multiple profiles; the one selected for this work is the outboard section where conventional ailerons are located. The same model was used in the VCCTEF work discussed in Section 1.1.2 and seen in Figure 1.6.

![Figure 3.1: A) Computational and wind tunnel models of the NASA CRM aircraft [64] [65].](image1)

![Figure 3.2: Airfoil profile of the outboard section of the NASA CRM main wing.](image2)

With the goal of investigating the use of morphing FMC actuator control surface on a commercial transport aircraft, several other parameters were chosen and presented in (Table 3.1). The flight conditions and required deflection were selected based on typical conditions for a commercial transport aircraft.
Table 3.1: Aileron case parameters

<table>
<thead>
<tr>
<th>Design Case Parameters</th>
</tr>
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<tbody>
<tr>
<td>Chord Length</td>
</tr>
<tr>
<td>Altitude</td>
</tr>
<tr>
<td>Velocity</td>
</tr>
<tr>
<td>Atmospheric Pressure</td>
</tr>
<tr>
<td>Dynamic Pressure</td>
</tr>
<tr>
<td>Flap Deflection</td>
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</tbody>
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The basic concept of the morphing aileron was to be similar to the morphing spoiler, but the analysis focuses on parameterized design variables related to the structural layout, materials properties, and the FMC actuators themselves. Looking at an entire cross-section of the wing it can be broken down into three major sections (Figure 3.3). First, the forward portion of the wing that remains unchanged from a conventional aileron. Second, the active portion that will deform for morphing, and finally the trailing edge fairing; which will move with the active section but is considered a rigid piece. The line separating the active section from the forward section will be referred to as the hinge line as this would be the location of the hinge mechanism in a conventional aileron. The aileron uses two sets of antagonistic actuators. The top set, when pressurized will bring the trailing edge up and when the lower set is pressurized the aileron will deflect down. There is a center composite plate similar to the FMC spoiler, and the active section is a flexible polymer rubber.

Figure 3.3: Components and layout of the FMC morphing aileron concept.

There are eight design variables relating to the structural layout: hinge line location, active length and the forward and aft vertical positions of the upper FMC, lower FMC, and
center composite plate. The hinge line location and active length decide the overall position of the morphing section of the aileron (Figure 3.4). While a conventional aileron would typically have a hinge line location around $3/4c$ the morphing aileron design will be able to consider designs both forward and aft of that location. Similarly, the length of the active section can vary, deciding what portion of the morphing aileron is active and deforming and what portion is dedicated to the rigid trailing edge fairing.

**Figure 3.4: Example of hinge line location and active length variables**

Within the active section, the upper and lower FMC actuators and the composite plate need to be located. For each component, the vertical position of the forward and aft attachment points is represented by a percentage of the airfoil's thickness at that location (Figure 3.5). A value of 100% would place an element on the top surface, and a value of 50% on the camber line. This also means that even if these values are held constant, as both the hinge line location and active length are varied, the actual vertical position and relative angle of each element may vary.
Material design variables were also considered by partitioning the active section of the aileron into five material regions, Figure 3.6. The area where the honeycomb polymer composite is divided into four regions by the composite plate and the upper and lower actuators. Each region can be independently varied and can be modeled as isotropic or anisotropic. The fifth material region is the composite plate. For this, the plates-thickness will be varied rather than the material properties themselves. For both the forward section of the airfoil and the trailing edge fairing, the regions are considered rigid.

**Figure 3.6: Material regions of the aileron**

In addition to the location of the FMC actuators, there are three design variables related to each set of the FMCs: force output, spanwise spacing, and bias strain. The force being generated by each actuator is a design variable and input into the structural model. This value is normalized by a unit span in the model given it units of Newton per meter span. The dimension between actuators in the span direction can also change for both the top and bottom set, so a design variable for FMC spacing with units of FMC per meter span is needed.
for both the top and bottom actuator sets. Finally, bias strain is necessary; this value will be discussed further in 3.2.5.

In order to investigate this problem a fluid, structural, and actuator model were created and coupled to form a complete analysis. Since the active spoiler served as the starting point for the aileron, the same contracting actuators were selected. To predict their performance in the aileron an empirical model based on test results was developed. This allowed the unique characteristics of actuators operating in bias and unpressurized passive actuators being strained in tension to be easily investigated; this model is presented in the following section. To complete the analysis, a finite element model was developed for the structural solution, and an aerodynamics model developed to determine surface pressure loads. These three models are then brought together in a coupled structure-aerodynamic-actuator analysis.

3.2 Characterization of FMC actuators

To design morphing structures utilizing FMC actuators, it is necessary to characterize the actuators performance. One option for this is to use numerical models to predict the performance. Several models have been developed for McKibben and pneumatic artificial muscles (PAM). FMCs can undergo significant strains during actuation often-requiring models, which consider both geometric and material nonlinearities. This can lead to relatively complex models which can have difficulty converging. Additionally, the actuators used in the aileron operate biased, meaning they are installed at a length less than their manufactured length. Modeling this nonlinear increase in force as the actuator is pressurized and the actuator goes from slack to taut also introduces complexity in modeling. For these reasons, the approach was taken to characterize several actuators through experiments and create an empirical model of those results. This has the advantages of being very simple to integrate into the analysis with no issues of convergence and provides a direct link between the analysis and known actuator performance.

What follows are the results of several experiments for a particular FMC actuator type which were used to create the empirical model. This was then integrated into the morphing control surface analysis and optimization program. The experiments characterized the force
output, hydraulic pressure, hydraulic fluid volume, and strain. From these, the energy needed for a given actuation can be found.

3.2.1 Test Specimens

The FMC design, which was characterized, was the same one selected for the contracting actuators in the FMC spoiler. This contracting actuator consisted of a 10 mm diameter aramid braided sleeve for the main fiber and a latex tube with a 7.6 mm diameter with a 2.3 mm wall thickness for the bladder (Figure 3.7). These components are then inserted into the end fittings along with 3M DP-460 epoxy and are then swaged (Figure 3.8). One end fitting seals the actuators and allows the end to be secured with a threaded bolt. The opposite end has an NPT thread allowing the pressurized fluid into the actuator, as well as securing the end of the actuator. The actuators were made with a nominal length of 157 mm between the fittings. The resulting fiber wind angle is ±15º.

Figure 3.7: Components of the FMC actuator
3.2.2 Fluid Volume Characterization

To use FMC actuators in a morphing aircraft control surfaces, it is important to understand the hydraulic fluid volume requirements and the energy required for deflections. To characterize the fluid volume requirements, three actuators were placed in a mechanical testing frame and pressurized with water from a plenum (Figure 3.9). Each actuator was bled of trapped air by loosening the top fitting. The actuator was then pressurized at different values and allowed to strain from its nominal length to its free contraction length at constant pressure. As the actuator contracted, the water level in the plenum was measured to determine the volume of fluid in the actuator. Each actuator was run at three pressures, with a maximum pressure of 827 kPa which was the limit of the pressure source. The results for change in fluid volume versus strain shows that pressure has little effect on the volume of fluid required (Figure 3.10). The empirical model needs to account for actuators of differing initial lengths. Therefore, the fluid required was normalized by actuator length and fitted with a second-order polynomial (3.1).
3.2.3 Force, Displacement, and Pressure Characterization

The fluid pressure, the force output, and the displacement of the actuator are each dependent on the two other parameters. To characterize this relationship, two different tests were performed. Each test holds one of the three parameters constant while allowing
the other two to vary. For the first test, the pressure was held constant while the actuator was allowed to slowly strain from its nominal length to its zero-force length. The second test was the blocked force test where the actuator was fixed at zero strain, the fluid pressure was increased, and the force output was measured.

For both tests, a set of six actuators were placed individually in an Instron 5967 30kN testing frame. The testing frame provided displacement and force output during the tests. Fluid pressure was measured using an Ashcroft Type-K1 13.8 MPa pressure transducer, attached with a T-fitting at the base of the actuator (Figure 3.11).

![Figure 3.11: Test setup for force, displacement, and pressure characterization.](image)

First, the constant pressure test was performed on each of the actuators. A pneumatic pressure source was used for this test since a constant pressure was required and the pneumatic source was easily regulated. This, however, limited tests to a maximum pressure of 1.0 MPa. Each actuator was tested at 0.3, 0.7, and 1.0 MPa. For each run, with the actuator
at its nominal length, the pressure was set and then the testing frame allowed the actuator to slowly contract until the force output became zero. Figure 3.12 shows a typical result for an actuator. As is typical for FMC actuators, the force is maximum at the blocked condition and decreases with strain until no force is generated, the free strain condition. The force is approximately linearly proportional to the fluid pressure. The actuators reach a maximum contractile strain around 30%; this value does not increase with higher pressure.

For the blocked force test, the actuator was held by the testing frame at zero strain as the pressure increased. During this test force output and pressure were measured. Since a constant pressure was not required, a hydraulic hand pump was used to apply pressures up to 5.5 MPa. Figure 3.13 shows the result of one such test. The relationship is linear with small variations due to the hand pump, which applies increasing pressure during the down-stroke then holds pressure on the up-stroke.

![Figure 3.12: Constant pressure curves vs. strain](image-url)
3.2.4 Actuator Passive Stiffness

When FMC actuators are used in a morphing structure which requires deflection in two different directions, a common approach is to use the actuators in antagonistic pairs. Due to this configuration, it is possible that when the active actuator is pressurized, the unpressurized actuator is put in tension being strained beyond the nominal length. For this reason, it was necessary to understand the passive stiffness of an actuator in tension. This assumes the actuator is unpressured; meaning effectively all of the stiffness is the result of the actuator’s fibers. To determine this, actuators used in the constant pressure and blocked force tests were strained in tension using the same mechanical testing frame. Actuators were strained from zero to approximately four kN. Figure 3.14 shows the results from a passive stiffness test and the stiffness model fit to the data (3.2).
3.2.5 Empirical Model

The purpose of the empirical model is to output values related to the FMC actuators as part of a morphing control surface analysis, specifically to the fluid volume required and the hydraulic energy requirement. These values are based on inputs in the analysis: actuator length, actuator bias, resulting strain, and force specified. This model will assume that the FMC actuator itself remains unchanged from the ones previously tested except for the nominal length, which will vary in the morphing control surface designs.

When actuators are used in an antagonistic pair, it is often required that there be a bias, meaning the actuator is installed at a length \( l_i \) which is less than its nominal or manufactured length, \( l_0 \). Figure 3.15 illustrates the need for installed bias. In case A, the two actuators are installed with no bias. As one is pressurized the other unpressurized passive actuator is in tension, making it difficult to actuate. In case B, there is an installed bias and when pressurized the active actuator contracts and the passive one can extend due to the installed bias, case C. Case D shows that continued pressurization of the active actuator could strain the passive one beyond its nominal length. Because of this the empirical model and associated structural model need to be able to account for this.

\[
\text{Force} = 3.27e7 \, e^{2.49}
\]
Figure 3.15: A) actuator pair with no bias, a locked condition B) Actuators with installed bias C) Pressured biased pair with the passive actuator pulled to zero strain D) continued pressurization results in the passive actuator being strained in tension

Bias removes a portion of the stroke of the actuator so that it can be pulled in tension beyond its installed length when the other actuator is contracting, with little additional stiffness. Since FMC actuators have decreasing force output with strain, it is necessary that the empirical model account for the installed bias. In the analysis, the model will allow for the actuator to be pulled in tension with no stiffness until the bias length, is reached then the passive stiffness of the actuator found in (3.2) become active.

For the model, two reference lengths for the actuator will be used. The installed length, $l_i$, being the length of the actuator when installed in the morphing control surface with the control surface at zero deflection and, $l_0$, the nominal or manufactured length of the actuator. The difference between these lengths is the bias (3.3) which will typically have a negative
value. This can also be normalized by \( l_i \) to give a strain bias relative to the installed length (3.4) or a normalized by \( l_0 \) for strain bias normalized by bias length (3.5).

\[
\delta_{bias} = l_i - l_0 \tag{3.3}
\]
\[
\epsilon_{bias} = \frac{l_i - l_0}{l_0} \tag{3.4}
\]
\[
\bar{\epsilon}_{bias} = \frac{l_i - l_0}{l_i} \tag{3.5}
\]

When looking at fluid volume requirements for the actuators, the volume of fluid needed to go from a zero deflection state of the control surface to particular deflection state is an important quantity. This means the fluid already in the actuator at its installed length is not needed but rather the additional quantity of fluid to achieve the deflection condition. This change in volume is the amount of pressurized fluid used which leads to determining the energy requirements of the actuator. To determine this equation (3.1) must be modified to account for the biased condition of the actuator. It is also convenient to define the total strain of the actuator, which takes into account the deflection of the actuator due to morphing, \( \delta_{morph} \), and the bias.

\[
\epsilon_t = \frac{\delta_{morph} + \delta_{bias}}{l_0} \tag{3.6}
\]

The fluid volume required to actuate a morphing control surface from its zero-state with the actuator at its biased length to some new state can be written as a function of the bias strain and total strain at the new state (3.7).

\[
\text{Fluid Volume (ml)} = [-1632(\epsilon_t^2 - \epsilon_{bias}^2) - 1054(\epsilon_t - \epsilon_{bias})] \times l_0 \tag{3.7}
\]

Due to the limitations of the pressure sources constant pressure curves were only obtained for relatively low pressures while block force tests allowed higher pressures to be tested but with no variation in strain. For this reason, it was necessary for the empirical modeling to use results from both tests and extrapolate into the typical operating region of the actuators. Figure 3.16 shows the results of the constant pressure test at the three pressures tested along with results from the blocked force tests at specific pressures 2, 3, 4, 5, and 5.5 MPa. At these same pressures values, curves of constant pressure resulting from the empirical model are plotted.
The empirical model for force output \((3.8)\) is a modified cotangent function to predict the force output, \(F\), as a function of strain, \(\epsilon\), and hydraulic pressure, \(P\), assuming a linear force pressure relationship. Coefficients were obtained using Matlab’s nlinfit tool. The function converges to zero force at approximately 30% strain, which is the typical free contraction strain of these actuators at pressures above 1.0 MPa. The model also accurately predicts block force and shows a similar trend to the experimental results of a nonlinear decrease in force near the blocked condition, and becoming linear as it approaches the free strain condition.

\[
F = \left[ [5.407 \times \coth (\epsilon + 0.0161) + 581.4] - 2001.8 \times \epsilon \right] \times P \times 10^{-3} \tag{3.8}
\]

For the analysis model developed in the following chapters, it is usually the case that force is input into the structural model and the strain results, meaning for the empirical model the force and strain will be known and the pressure, energy, and fluid volume are outputs. For this reason \((3.8)\) can be written as a function of strain and force \((3.9)\).
The hydraulic energy required for a particular actuation can be found by knowing the fluid volume forced into the actuator and assuming a constant pressure throughout the deflection. Using the equation for fluid work (3.10) and knowing volume from (3.7) and pressure from (3.9) the energy needed for a given deflection can be written as a function of the total strain, $\varepsilon_t$, the bias strain, $\varepsilon_{bias}$, force generated by the actuator, and the actuator’s nominal length, $l_0$ (3.11).

$$W = P * V$$  \hspace{1cm} (3.10)

$$W = F * 10^3/[((5.407 * \coth(\varepsilon_t + 0.0161) + 581.4) - 2001.8 * \varepsilon_t)]$$  \hspace{1cm} (3.9)

$$W = F * 10^3/[((5.407 * \coth(\varepsilon + 0.0161) + 581.4) - 2001.8 * \varepsilon_t] - 2001.8 * \varepsilon_t]\) * [-1632(\varepsilon_t^2 - \varepsilon_{bias}^2) - 1054(\varepsilon_t - \varepsilon_{bias})] * l_0$$  \hspace{1cm} (3.11)

3.3 Coupled Structural and Fluid Solution

Unlike the analysis of a conventional aileron, where the kinematics of the aileron is defined by the hinge geometry and not affected by the fluid flow, the compliant nature of the morphing aileron can assume different shapes under flow. For this analysis this means the fluid and structural solutions are coupled and can’t be separated. An iterative approach is used to find the solution for the fluid loads and structural response. To accomplish this, three programs were used: Abaqus for the structural finite element model (FEM), XFOIL for determining the aerodynamic loads, and Matlab for converging the solution and integrating the empirical model.

A flowchart of the developed analysis program can be seen in Figure 3.17. Using Matlab as the common language between the structural and aerodynamic solvers, the analysis works by first reading in the design variables and altering the structural layout, material properties, and FMC properties of the structural model. An assumed aerodynamic loading is applied to the top, and bottom surfaces since the true aerodynamic loads are unknown, and the model solved. The deflected shapes for both up and down deflection cases are then passed to XFOIL to update the aerodynamic loads. With these new loads, the structural model is updated and solved. This process continues until a converged condition is reached. Convergence is measured on the basis of equivalent flap deflection and is typically considered converged when the solution is within $\pm 0.1^\circ$ of the previous solution. Solutions
typically take six minutes, but this can vary between four and twenty minutes depending on the number of iterations needed.

This method of converging the solution is necessary because the deflections of different designs will not be kinematically similar; unlike the analysis of a conventional aileron where the loads may not be the known, but the deflected shape is. In the case of a conventional control surface, the flow can be modeled as a series of distributed springs and the solution found in a single iteration.
Matlab – Single Design Run

Read in Design Variables:
- Geometric
- Material
- FMC force and bias
- Aerodynamic conditions

While equivalent flap deflection up and down not converged
- Write DV to file for Python script
- Write coefficients for surface pressure polynomials to file for Python script

Output:
- Equivalent flap deflections
- FMC strains
- Hydraulic fluid volume
- Hydraulic energy required

Abaqus Python Interpreter
- Partition geometry
- Mesh and assign elements
- Create loads:
  - Surface pressure loads based on polynomial coefficients
  - FMC internal loads
  - Send model to Abaqus solver

Abaqus Solver

Abaqus Python Interpreter
- Post process extracting deflected geometry and FMC strains

XFOIL

Run XFOIL based on updated deflected geometry

- Fit $C_p$ curves with polynomials to be used in next iteration
- Calculate eq. flap deflection

FMC empirical model to find:
- Hydraulic fluid volume requirement
- Hydraulic energy requirements

Figure 3.17: Flowchart of the analysis program
During the development of the model, several decisions were made concerning the fidelity needed. Ultimately, the model, presented is a balance between a lower fidelity model, which may not capture any of the aspects specific to FMC actuator, and a full 3D model, which would be too computationally expensive for use in the following optimization. The model was intentionally designed to be easily parameterized so that the design space could be easily evaluated.

3.4 Structural Solution

The structural solution was created using Abaqus a commercial FEM package. Due to the large change in geometry between each potential design point of the aileron, it was necessary that the model be partitioned and meshed for each design point. All changes to the model to adjust design variables was done via Matlab running Python scripts to within the Abaqus Python interpreter environment. This allows each design point to be automatically generated. A 2D plane strain model is used to represent a cross-section of the aileron. All variables such as FMC force, FMC spacing, the energy required will be normalized by unit meter span.

To start each run of a design point, the geometry of the CRM profile is loaded. While only the portion aft of the hinge line location is necessary, the model is easier to control from one design point to the next if the entire profile is used, keeping a constant frame of reference. The active portion of the model is then partitioned based on the input design variables for the location of the hinge line, aft attachment points, and the position of the two FMCs, and composite plate.

At this point, the model is meshed (Figure 3.18). The active region of the aileron is meshed with tight uniformly sized elements. The forward region and trailing edge fairing which are not intended to experience any deformation are meshed with a logarithmic spacing allowing the transition from coarsely meshed forward region to the finely meshed smooth region be smooth while still reducing the number of unnecessary elements. The model is meshed with CPE3H and CPE4RH hybrid plane strain elements. These elements were selected because the materials likely to be used in the active portion of the aileron are from a honeycomb polymer composite (HPC) like that used in the FMC morphing spoiler. These materials typically have a Poisson's ratio approaching 0.5, leading to material incompressibility.
Each material region is given material properties based on the input design variables. All regions are modeled assuming anisotropic material properties. The center plate is modeled with beam element B21H, a Euler-Bernoulli beam type element with a hybrid formulation for large displacements. The material of the beam is assumed isotropic, and the beam’s thickness is adjusted based on the input design variables.

![Element Types](image)

Figure 3.18: Mesh of finite element model

For each run of the model, two load cases are created, a trailing edge up condition and trailing edge down condition. The previously discussed geometry and material properties remain the same for each, but the aerodynamic loads and FMC applied loads are different. Both load cases run using Abaqus’s nonlinear geometry solution.

Figure 3.19 shows the surface loading of the upper and lower surface of the model representing the aerodynamic load. The pressure curves, for the upper and lower surface in both the up and down cases, is represented by a sixth order polynomial that is fit to the results of the fluid model discussed in section 3.5. This is how the structural model is coupled with the fluid model. During the first iteration of the solution the aerodynamic loads are unknown, so a default loading is assumed, which is updated in the subsequent iterations using the fluid model.
3.4.1 Structural Model of FMC actuator

A key part of the structural model is the FMC actuator. Both the upper and lower actuator are modeled by the two partitioning lines, which are located based on the input design variables. Each actuator set (upper or lower) has entirely different properties depending on if it the up or down load case. For instance, in the trailing edge up load case, the upper actuator is modeled as a contracting actuator applying an internal load. The lower has a passive stiffness in tension, which is also a function of the bias. The opposite is, true for the trailing edge down load case.

To allow this kind of control, the actuator is modeled using what is called a connector element in Abaqus, which allows each element to be easily controlled. Specifically, an Axial Connector was used. Connectors once defined, can be used to directly prescribe a strain or a load across the connector or the connector, can be given properties like unique nonlinear elastic properties, and each of these can be easily controlled and changed from one load condition to the next. Regardless of the prescribed condition each element is treated separately and the condition is satisfied at each element individually, meaning even though the partition line represents an actuator that may be 10 cm in length, it is seeded and meshed into 100 elements; each element will satisfy the specified condition.

Looking first at the case of the active actuator applying the contractive internal load, the program reads in the prescribed FMC force. This will have units of N/m being normalized by unit span. This force is then prescribed to each connector element in the active actuator.
for the particular load case. When the model solves for displacements, the condition of the applied force will be satisfied for each element. This means an active actuator is assumed to apply a constant force across its length, but the surrounding structure can cause a non-uniform strain along its length (Figure 3.20). This is the advantage of using connector elements over other options like applying follower forces at the FMC ends. As the structure morphs, the actuator continues to contract and apply load along its axis. Using this element along with the nonlinear solution means the model correctly captures the way an actuator applies a distributed contractive load along its axis throughout the deflection.

To determine the strain of the active actuator the definition of each element is summed and converted to strain. This value along with the actuator bias and force can be used in (3.11) to find the hydraulic energy required for a given actuation.

![Active Actuator Force](image)

![Active Actuator Strain](image)

**Figure 3.20:** Connector element results for an active actuator with prescribed constant force and resulting strain.

For each load case, one actuator is active and providing the contractive load while the other is passive. The same connector element is used for this passive actuator, but the state is changed from providing a contractive load to assuming a nonlinear stiffness. The stiffness of an unpressurized actuator in tension was modeled with equation (3.2). The connector element also needs to take into account the biased state of the actuator. The stiffness model for the actuator has nearly zero stiffness until the strain approaches the bias strain of the actuator. The stiffness represented in the model is also a function of the spanwise spacing of actuators. Each actuator provides a given stiffness, and the line of connector elements assumes a unit span. Figure 3.21 shows several stiffness curves used for connector elements to represent passive actuator in tension at different bias and spanwise spacing values. Figure 3.22 shows the force in passive actuator at a deflection condition before and after the bias value of the actuator. At pre-bias, there is a minimal force in the connector elements.
representing the actuator after the bias strain is reached there is a significant increase in the force.

![Graph showing stiffness curves for passive actuator connector elements at different installed bias and spanwise spacing values.](image1)

**Figure 3.21:** Stiffness curves for passive actuator connector elements at different installed bias and spanwise spacing values.

![Graph showing results for tension force a passive actuator at deflection conditions before and after bias strain.](image2)

**Figure 3.22:** Results for tension force a passive actuator at deflection conditions before and after bias strain

### 3.5 XFOIL Fluid Solution

For the fluid model, XFOIL is used to update the loads in the interactive solution of the coupled structure and aerodynamic models and provides output information on the aileron’s
aerodynamic performance. At the completion of the structural model through Abaqus, the deflected geometry for the up and down cases are extracted from Matlab, and then XFOIL is run using Matlab scripts. XFOIL is run with an inviscid solution. The coefficient of pressure values ($C_p$) curves is then read back into Matlab and fit with a sixth order polynomial over the active and trailing edge fairing portion of the aileron. The coefficient to these equations for the upper and lower surfaces and the up and down cases are what is then based back to Abaqus to generate surface loads for an iterative solution.

It is not an accurate comparison to strictly look at the angle of deflection of the morphing aileron to that of a conventional aileron. Due to the morphing shape, in particular, the position of the hinge line and length of the active region, a morphing aileron with a deflection angle the same as conventional could be more or less effective. A metric was created to directly compare the conventional and morphing ailerons by matching the change in the coefficient of lift of the two systems, giving an equivalent flap deflection. To do this the same NASA CRM airfoil section was run in XFOIL with a conventional aileron with a hinge location $3/4c$ and swept from $-20^\circ$ to $20^\circ$ and the angle of attack from $0^\circ$ to $5^\circ$ and the resulting coefficient of lift ($C_l$) was plotted (Figure 3.23). These results were then fit with a function, which allows the $C_l$ result of a morphing aileron to be output as an equivalent flap deflection with units of degrees. This provides a direct metric to compare the two systems with a value that has an intuitive understanding.
Figure 3.23: Geometry for the CRM airfoil with conventional aileron and results of XFOIL analysis for equivalent flap deflection.

\[ \delta_{eq} = \left( (C_L - 0.456) - 0.118 \alpha \right) / 0.073 \]  

(3.12)

3.6 Conclusion

An analysis capable of solving this coupled fluid, structure, and actuator model was presented. One key aspect of the model was the integration of an actuator model based on empirical results. This model is capable of modeling not only an active actuator contracting with hydraulic pressure, but also model the passive unpressurized actuator in tension. Using Abaqus connector elements, the passive actuator can have zero stiffness until the bias strain is reached and then have a cubic stiffness curve based on test data.

The concept of equivalent flap deflection was also presented. This is a necessary step to compare conventional to morphing aileron since the rotation of the trailing edge is no longer
a valid point of comparison. This was done by matching $C_L$ values using the same fluid analysis for conventional and morphing designs.
4 Parameter Study for the Morphing FMC Aileron

4.1 Observed Trends

Prior to optimization, a series of trend plots were examined by taking a single baseline design and varying each of the design variables independently. By examining these plots, a better understanding of the effect of each variable and the design space as a whole was achieved. For this, all design variables were considered, but the four polymer regions of the active section were considered a single section so that all four have matching material properties.

For the following plots, flap deflection refers to the equivalent flap deflection previously discussed (Figure 3.23). Plots of the energy refer to the hydraulic energy required. Energy is normalized by the spanwise dimension and therefore has units of J/m. The force input into the model from the FMCs is also similarly normalized by span giving units of N/m.

Two FMC strain quantities are also presented, passive and active. Considering a downward deflection, the passive actuator would be the top actuator which is unpressurized, and the active actuator would be the bottom pressurized actuator, the opposite is true of an upward deflection. Both strains represent the total strain meaning the summation of the bias strain and the strain due to the morphing actuation and follow the convention of contraction having a negative strain. This means that a passive actuator with a positive strain value has been pulled by the morphing actuation beyond the bias it was installed with; the actuator is in tension. For the following results in this section, the response of the model concerning one or two design variables will be looked at while the other design variables remain fixed at the baseline values in Table 4.1.
4.2 Baseline Solution

This section looks at the baseline design point in detail. The trend plots in the following sections will look at variations about this design point. Table 4.1 shows the design variables selected which can be broken into geometric, material and FMC related design variables. For this run, the four polymer regions in the active section will be modeled with the same isotropic material properties with a Poisson’s ratio of 0.499 a common value for the polymers being considered.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hinge line</td>
<td>1.58 m</td>
<td>As measured from LE to start of the active region</td>
</tr>
<tr>
<td>Active length</td>
<td>0.19 m</td>
<td></td>
</tr>
<tr>
<td>Upper FMC position</td>
<td>79%/76%</td>
<td>Position as a percentage of thickness (forward/aft)</td>
</tr>
<tr>
<td>Lower FMC position</td>
<td>27%/26%</td>
<td>Position as a percentage of thickness (forward/aft)</td>
</tr>
<tr>
<td>Center plate position</td>
<td>52%/51%</td>
<td>Position as a percentage of thickness (forward/aft)</td>
</tr>
<tr>
<td>Center plate thickness</td>
<td>3 mm</td>
<td></td>
</tr>
<tr>
<td>FMC force up</td>
<td>39 kN/m</td>
<td>Upper actuator force per meter span</td>
</tr>
<tr>
<td>FMC force down</td>
<td>84 kN/m</td>
<td>Lower actuator force per meter span</td>
</tr>
<tr>
<td>Upper actuator spacing</td>
<td>36 /m</td>
<td>Number of actuators per meter span</td>
</tr>
<tr>
<td>Lower actuator spacing</td>
<td>36 /m</td>
<td>Number of actuators per meter span</td>
</tr>
<tr>
<td>Actuator bias</td>
<td>-0.038/-0.036</td>
<td>Bias strain of actuator (upper/lower)</td>
</tr>
<tr>
<td>Polymer modulus</td>
<td>950 kPa</td>
<td></td>
</tr>
<tr>
<td>Polymer Poisson’s ratio</td>
<td>0.499</td>
<td></td>
</tr>
</tbody>
</table>

The design variables are read into Matlab which integrates with Python scripts to create the Abaqus model. A default set of aerodynamic loads are applied in Abaqus since they are not initially known. The resulting deflected shape for the up and down case from Abaqus is then passed to XFOIL to update the aerodynamic loads from the default loading. This iterative process continues until both equivalent flap deflections up and down converge within ±0.1° between iterations. Depending on how close the assumed aerodynamic loads were to the final loads this typically takes between 3-10 iterations.

It is important to realize that FMC force is an input of the model and flap deflection cannot be directly controlled as it is an output. To find solutions at a specific deflection, typically
the ±20° required by the design case, another iterative solution is required. Using a secant method, the FMC force for the up and down deflections is varied to achieve the desired deflection. This too typically takes 3-10 interactions depending on the initial value given for the force. The results of the baseline solution are in Table 4.2 which show that the design meets the required ±20° of deflection. It also shows that even though this design does not represent an optimal solution, the hydraulic pressure is within typical operating pressures for aircraft and the FMCs.

**Table 4.2: Output of coupled analysis**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Iterations</td>
<td>4</td>
</tr>
<tr>
<td>Equivalent flap deflection (up/down)</td>
<td>-20.09° / 20.10°</td>
</tr>
<tr>
<td>Cl (up/down)</td>
<td>-0.776 / 2.311</td>
</tr>
<tr>
<td>Active actuator strain (up/down)</td>
<td>-0.0784 / -0.0775</td>
</tr>
<tr>
<td>Passive actuator strain (up/down)</td>
<td>-0.0010 / 0.0009</td>
</tr>
<tr>
<td>FMC hydraulic pressure (up/down)</td>
<td>2.22 MPa / 4.74 MPa</td>
</tr>
<tr>
<td>Hydraulic energy (up/down)</td>
<td>527 J/m / 1180 J/m</td>
</tr>
<tr>
<td>Maximum center plate curvature (up/down)</td>
<td>2.44 / m / 3.33/m</td>
</tr>
</tbody>
</table>

Figure 4.1 and Figure 4.2 show the final deflected shape of the entire airfoil and FMC morphing control surface. Figure 4.3 shows the force being applied in the model of the active actuator and the load being carried by the passive actuator. In this case, the passive actuator has a negative strain meaning it effectively carries no load as the actuator is still slack. Figure 4.4 shows the same result but assuming the installed bias has been reduced by half from the original 3% to 1.5%. In this case, the passive strain is positive, and the actuator is taut and carrying a significant load.
Figure 4.1: ±20° deflection of the baseline case, entire airfoil.

Figure 4.2: Deflection for ±20° deflection of the baseline case, active region and trailing edge fairing.

Figure 4.3: Connector force of the passive and active actuators in the baseline solution with 3% bias.
Figure 4.4: Connector force of the passive and active actuators in the baseline solution with 1.5% bias.

This baseline solution represents a typical result of the analysis. The following sections look at varying each design variable independently about this solution while the other design variables remained fixed at the baseline values outlined in Table 4.1.

4.3 Flap Deflection

The first variable considered is the FMC force input. The forces for the up and down cases were varied independently to achieve deflections from 0° to 20°. Figure 4.6 shows the resulting force required to achieve the deflections and the resulting energy required for the up and down deflections. The force plot shows that approximately a third the amount of force is required for an upward deflection than a downward. This is expected since the down deflection deflects into a region of higher aerodynamic pressure and the upward deflects into a region of lower pressure. Different designs do result in small change in the ratio of up to down force, but it is always true that the down requires significantly more force. This difference in pressure on the upper and lower surfaces is also the reason for the force not converging to zero at zero deflection. A continuous force is required from the lower actuators to maintain zero deflection under load. The plot also shows that if both actuators apply no force underflow, the aileron will reach equilibrium at 9° trailing edge up.
Figure 4.5: Resulting FMC force and hydraulic energy required for deflection from 0-20°.

In this case, at each point, the passive actuator has a negative strain meaning it remains slack contributing no additional stiffness. This is why the response remains linear for the force. Had a smaller actuator bias value been used in the design, a sharp increase in the force would be seen as the passive actuators transitioned from being slack to taut. The result for the energy required is nonlinear as a result of the previously discussed nonlinear model used to determine the hydraulic energy required based on strain and force (3.11).

The previous plots assumed the default aerodynamic load outlined in the design case definition. The same model can be run with no flow giving some insight into how much energy is going into resisting the aerodynamic forces and how much is going into the elastic deformation of the structure. Figure 4.6 shows the resulting force and energy for 0° to 20° for both the default flow case and no flow case. The plots show that for the no-flow case both the force and energy required do converge to zero at 0° deflection since there is no flow provided a constant upward force on the surface at zero deflection. With no flow, there is no longer the significant difference between the force required for up versus a down deflection. The force required is not exactly equal between the up and down deflection for the no-flow case due to the orientation of the FMC actuators, the center plate, and the nonsymmetrical airfoil geometry. Looking at the energy required it can be seen that a majority of the energy is going into resisting the aerodynamic loads and not the elastic deformation of the structure, particularly for the downward deflection.
Figure 4.6: Comparing the force and energy required for the default flow case and a case with no flow

Looking at the resulting deflections in Figure 4.7 it can be seen that the difference between the tip deflections for the two flow cases is relatively small. However, the resulting shape is not the same. Looking at the curvatures of the centerline plate which runs from the hinge line location to the start of the trailing edge fairing best shows the difference in shape in the two cases (Figure 4.8). The curvature of the center plate for the no-flow case starts off with greater curvature at the leading edge of the plate, while underflow the point of maximum curvature moves further aft.
Figure 4.7: ±20° deflection for the default flow conditions and no flow.

Figure 4.8: Curvature of the centerline plate for ±20° deflections

This result shows that the morphing control surface’s structural and aerodynamic solutions are coupled. The morphing control, surface unlike a conventional control surface, can be thought of as having more than one degree of freedom. A conventional aileron could
be deflected to a particular shape under no flow, then flow applied and by adjusting the hydraulic pressure could maintain the same shape underflow. The same cannot be said of this morphing design. A morphing aileron could be deflected some amount under no flow, then flow applied, and while it would be possible to maintain the same $C_L$ and therefore the same equivalent flap deflection, it would not be possible to maintain the same shape.

4.4 **Hinge Line Location and Active Length**

The hinge line location and active length variables set the overall position and size of the active section of the morphing aileron. The hinge line location marks the start of the active section as measured from the leading edge of the entire airfoil. The active length is the distance from the hinge line location to the start of the trailing edge fairing.

Unlike the prior trend plots which looked at varying the force as an input to see the resulting flap deflection, this study through an iterative process varies the force to achieve a required $\pm 20^\circ$ deflection. This adjusting of the force to achieve the same deflection is necessary to understand the effect of changing the hinge line and active length variables has since without the correction the changes would result in drastically different deflections. The data in the following plots is the result of iteration of the structural and aerodynamic solution as well as iteration adjusting force to achieve the required equivalent flap deflection. The iteration is stopped when the equivalent flap deflection converges within $\pm 0.1^\circ$ of the required $20^\circ$ on successive iterations.

In Figure 4.9 the resulting hydraulic energy and strain of the passive actuator for the down deflection for varying hinge line location and the active length is plotted. All other design variables remain fixed at their baseline values from Table 4.1. The energy plot shows that pushing the hinge line aft and increasing the active length results in the lowest energy required. The results for active lengths of 0.19 and 0.24 m are similar and proportional while the result of 0.14 m diverges as the hinge line location moves forward on the airfoil. Figure 4.9 also shows the strain required of the passive actuator, which in this case of a downward deflection is the upper actuator. A positive strain means the actuator is taut being strained beyond its manufactured length. All circled data point in Figure 4.9 indicate that at that point the passive actuator is taut. This is why the energy for the 0.14m active length cases
increases faster as the hinge line is moved forward. The passive actuator transitions from causing no additional stiffness to having the cubic stiffness curve described by equation (3.2).

![Graph](image1)

**Figure 4.9:** Energy and passive FMC strain for a 12° downward deflection with varying hinge line location and active length. Circled data markers indicate passive actuator is taut.

The upward deflection case shows a similar trend for active lengths of 0.14 m and 0.19 m, but in the case of the largest active length 0.24 m, the energy increases when the hinge line is forward of 1.5 m (Figure 4.10). This change is due to a difference in the shape of the structure as it deflects, Figure 4.11 shows the deflected shape of the hinge line at the aft-most location. This shape is typical of other design points being a gradual smooth curve up. Figure 4.12 shows the deflection with the hinge line at the forward most location. In this case, the active portion of the aileron deflects down at some points, and the aileron effectively rotates the trailing edge fairing about the aft attachment point of the centerline plate at achieve the necessary shape change for the 20° of equivalent flap deflection.

![Graph](image2)

**Figure 4.10:** Energy and passive FMC strain for a 12° upward deflection with varying hinge line location and active length. Circled data markers indicate passive actuator is taut.
Figure 4.11: Displacement contour for hinge line location of 1.63 m and active length 0.24 m

Figure 4.12: Displacement contour for hinge line location of 1.38 m and active length 0.24 m. The increased size of the active length without an increase in stiffnesses causes undesired deflections.

This difference in deflected shapes can be explained by thinking of the active section as a cantilevered beam for a downward deflection case (Figure 4.13). The contracting FMC applies a force (F_{FMC}) through a small moment arm at the tip of the beam causing a negative moment and force. The aerodynamic load, can be thought of as a distributed load along the active length (W_{aero}). The aerodynamic loads on the trailing edge fairing can be reduced to a tip force and moment on the beam (F_{Tip} and M_{Tip}). Each of these loadings is affected by the active length and hinge line location. As the active length increases, the overall load of W_{aero} increases. The force and moment of the trailing edge fairing can greatly increase as the area of the fairing increases by either the hinge line moving forward or the active length decreasing. Both of these actions increase the size of the fairing. When both the active length becomes large, and the hinge line is moved forward these aerodynamic loads push the active section up. To still achieve the required equivalent flap deflection, the aileron rotates to a larger angle.
Comparing Figure 4.11 and Figure 4.12 shows that the latter is not an efficient actuation. This is both from the perspective of consuming hydraulic energy and also posing large curvatures of the center plate and strain of the polymer. By moving the FMC attachment points and the center plate, as well as adjusting the polymer and beam stiffness this undesired shape can be corrected. The following chapter looking at optimization will address this.

When considering the morphing aileron design, two important factors are considered, the required hydraulic energy and mass of the system. If it is assumed that the region forward of the hinge line location would have approximately the same density as a conventional aileron, and the trailing edge fairing would also have approximately the same density of the aft hollow portion of a conventional aileron then the difference between a conventional and FMC morphing aileron would only depend on the size of the active region. Figure 4.10 shows the size of the active region as a percentage of the area of the entire airfoil cross-section. The mass or active area is only a function of the geometry of the airfoil and the two design variables, hinge line location and active length. As the hinge line location moves forward the area increases due to the active region moving to a portion of the airfoil with a greater thickness. To minimize mass, the hinge line location should be as far aft as possible, and the active length minimized. This runs counter to the objective of decreasing the energy required which requires increasing the active length.
Figure 4.14: Active area as a percentage of the entire airfoil cross section with varying hinge line location and active length

4.5 FMC Installed Bias

The FMC bias has two separate effects on the model depending on whether the passive or active roll of the actuator is being considered. The role of the actuator, passive or active, depends on whether a trailing edge up or down deflection is being considered. For an upward deflection, the top actuator is active or pressurized by the hydraulic fluid; the lower is passive or unpressurized.

The bias in the active actuator affects the performance of the actuator since a portion of the stroke is effectively removed. In the model, this comes out in the hydraulic energy required shown in equation (3.11). For Figure 4.15 the active actuator’s bias is varied and the resulting energy is plotted for a ±20° deflection. For both the up and the down cases the energy required gradually decreases as the bias approaches zero. This is expected since the strain due to the morphing actuation will remain nearly constant, but the increased bias moves the actuator to a less efficient region of its stroke. This will not affect the structural solution and only effects the empirical model used to calculate energy.
Figure 4.15: Energy for ±20° deflection with varying active FMC bias.

The passive actuator’s bias affects the structural solution. Once the passive actuator is strained past the bias strain, it transitions from being slack to taut and goes from having zero stiffness to the cubic stiffness curve defined as part of the empirical model. Figure 4.16 shows the resulting hydraulic energy required for the same series of runs in Figure 4.15. The circled data points indicate that the passive actuator is taut. In the design where the actuator is slack, there is no difference in the required energy with changing bias. This is because the passive element has no stiffness until becoming taut. Once the passive actuator transitions to being taut there is a significant increase in the energy as the active actuator has to overcome the additional stiffness to achieve the required deflections.

Figure 4.16: Energy for ±20° deflection with varying passive FMC bias.
At the points with a taut passive actuator, there is not just a change in the energy required, but also in the shape that results in the final deflection. Figure 4.17 compares the deflected shape of the down deflection with different passive actuator biases. At the 6% bias the passive actuator is slack, and in the 1% case, it is taut. The taut actuator prevents the upper portion of the aileron from extended further due to the added stiffness, causing the airfoil to straighten out with less curvature near the aft end of the active region. In order for the aileron to still achieve the same deflection the 1% case has a greater curvature at the forward section of the active region.

![Graph showing deflected shapes for different actuator biases.](image)

**Figure 4.17: Deflected shaped for a downward deflection with 1% and 6% bias in the upper actuator.**

From Figure 4.15 it may seem that an optimal would be to use no bias strain at all, but the same actuator which is active becomes passive in the opposite deflection case. The low bias value comes with a large energy penalty when the passive actuator is taut. The bias of the actuator is set during manufacturing and cannot be changed. From this, it is clear that bias plays an important role in the efficiency of the morphing aileron and that an optimal value will exist which will not be at either extreme of the possible bias values.

### 4.6 Position of Internal Components

Six design variables are used to place the forward and aft locations of the upper FMC, lower FMC, and center plate in the vertical direction. Each of these is given as a percentage
of the thickness of the airfoil at that X location. The X location for the forward position is the hinge line location, and the aft location is the start of the trailing edge fairing (Figure 3.5). The movement of these three components relative to each has the effect of changing the moment arm the FMC has on the structure. This change in moment arm serves as a way to adjust the force or strain required of the active FMC and the same moment arm effects the passive actuator is the FMC become taut.

Moving the center plate relative to the actuators has the same effect of changing the moment arm but also affects the neutral axis of the structure. Figure 4.18 shows the change in the energy required for an upward deflection as the plate's vertical position is varied. The aft location cannot be varied as much as the forward point due to the plate approaching the upper FMC actuator at the aft location. The plot shows a minimum for both locations near 50% of the airfoil’s thickness. As the plate moves lower the energy increases due to the plate moving away from the active top actuator requiring more strain from the actuator to achieve the 20° of deflection (Figure 4.19).

![Figure 4.18: the Hydraulic energy required for an upward deflection of 20° with varying forward and aft location of the centerline plate.](image)

Figure 4.18: the Hydraulic energy required for an upward deflection of 20° with varying forward and aft location of the centerline plate.
Figure 4.19: Strain required of the active top actuator set for a 20° deflection with varying plate location

As the plate is moved up and closer to the active actuator the active strain required decreases but at this same point the passive actuator transitions from being slack to taut as indicated by the circled data markers. Figure 4.20 shows the force required for the same conditions. As expected the force required increases as the plate is moved up and less strain is required, but there is a significant increase in the force required at the point the passive actuators becomes taut due to the plate moving further from it.

Figure 4.20: FMC force required for an upward deflection with varying centerline plate position.
A similar response was seen when looking at a downward deflection. Similar to the previously discussed bias quantity the position of the plate cannot be changed from an upward to downward deflection. So trends seen in a downward deflection have the opposite but not necessarily equal effect in an upward deflection. For this reason, it is useful to look at the sum of the energy required for an upward and downward deflection (Figure 4.21). The plot shows that again a position near half the airfoil’s thickness provides a minimum. Moving to either side of the value has the effect of causing the passive actuator to become taut and the active actuator having to provide more force to the shorter moment arm and increased stiffness of the taut passive actuator. It should be noted that the optimal value seen here of 50% is true for the baseline case considered here, but is not necessarily the case as different points in the design space are considered.

![Figure 4.21: The total energy required for ±20° deflection with varying forward and aft attachment points of the centerline plate.](image)

Moving the FMC actuators within the active section has a similar effect to that of the center plate. Figure 4.22 shows the energy required for a 20° downward deflection while varying the attachment point of the bottom FMC actuator. Since this is a down deflection, the bottom actuator is active. Moving the actuator vertically effects the moment arm between it and the plate again causing changes in the required strain (Figure 4.23).

The change in position does also affect the final shape and causes the passive top actuator to transition to being taut for some conditions. Though the passive actuator’s strain remains relatively small in this case meaning there is not a significant increase in the energy required.
The change in position of the lower actuator also has an effect on energy when the actuator is unpressurized in an upward deflection. Figure 4.24 shows the required energy for an upward deflection. When the actuator's endpoints are above 25% of the airfoil's thickness the actuator remains slack during the actuation, so there is no change in the required energy. Once the actuator moves below 25%, the increased moment arm causes the passive strain to extend beyond the biased length and becomes taut (Figure 4.25), causing the increase in energy required.
Figure 4.24: 20° upward deflection energy requirement with varying bottom FMC position.

Figure 4.25: Passive FMC strain for 20° upward deflection with varying bottom FMC position

When the passive actuator become taut increasing the stiffness, it not only affects the energy required but also the final shape of the aileron. Figure 4.26 shows the maximum curvature of the center plate for the same 20° deflection with varying the lower FMC’s position. The taut actuator causes an increase in the maximum curvature. Figure 4.27 shows the curvature of the center plate for the two extremes of the leading edge position. When the attachment point is at the higher position, the curvature is more equally distributed along
the length of the plate. Higher curvatures mean more stress and a fatigue point in the plate. For this reason curvature will be introduced as a constraint in the optimization.

![Graph showing curvature vs. bottom FMC position]

**Figure 4.26:** Maximum curvature of the center plate for a $20^\circ$ up deflection with varying center plate position

![Images showing curvature comparison]

**Figure 4.27:** Change in center plate curvature for change in bottom FMC forward attachment point position

### 4.7 Polymer Stiffness and Center Plate Thickness

Both the stiffness of the polymer in the active region and the beam thickness have the same effect of changing the passive bending stiffness of the morphing aileron. Each portion of the polymer and the plate are different distances off the neutral axis. Due to the geometry of the airfoil cross-section, the polymer also changes this distance as you move along the plate. This means that not only the passive stiffness is affected but also the spatial distribution of that stiffness.
The change in energy required varies linearly with polymer stiffness (Figure 4.28) resulting in approximately a 10% change in the energy required over the range of polymer stiffness considered. The energy results for plate thickness do not vary linearly due to the plate thickness is being adjusted as compared to the material properties. Over the thicknesses considered a change of approximately 100% is possible. This difference in response is due partially due to the cubic relationship to stiffness in the second moment of area in the bending stiffness of the plate. It is also due to the relative bending stiffness of the two components. The polymer region tapers but always has a thickness much greater than the center plate, but the difference in the modulus of elasticity is three orders of magnitude. Depending on the polymer modulus and plate thickness selected the plate can have a bending stiffness from four to several hundred times stiffer. For the baseline design, the plate has a bending stiffness 2.5 times greater at the hinge line and 25 times greater at the trailing edge fairing.

![Figure 4.28: Change in energy required for changing polymer stiffness with all four-polymer regions set equal.](image)

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4.8 Conclusion

Several trends have been shown with variations about the baseline solution providing an understand of how each design variable affects the solutions. It should be noted that it is possible that these trends could change at different points in the design space. The trend plots are also limited to looking at the variation of a single parameter, while many of the design variables are linked.

The trend plots show, the bias strain of the actuators plays an important role in the efficiency of the aileron. Whether the passive actuator remains slack or in tension is affected not only by the installed bias but was also shown to depend on the placement of the internal components, and the overall position and size of the active region.

It was also shown that depending on the state of the passive actuator as either slack or taut can change whether a design variable has an effect on just a single deflection case (up or down) or can have an effect on both. This presents a challenge to the following optimization but creating a discrete condition where the effect of a variable on a particular deflection direction can be controlled by relatively small changes in an different design variable for the opposite deflection case.
Chapter 5

5 Optimization

Optimization was performed using the model outlined in the previous chapter with the goal of finding an optimal design for the previously discussed design case. Optimal will be considered both in terms of minimizing mass and minimizing the hydraulic energy required. The design case and limitations of the FMC will guide the constraints of the optimization. Based on the results of the optimization, a series of design guidelines were developed.

5.1 Objective Function

The optimization has two objectives, to reduce the mass of the morphing aileron and have the most efficient actuation regarding the hydraulic energy required. These two objectives run counter to each other, meaning no single design can minimize both. A minimal mass design is not efficient at actuation and a more efficient actuating design results in a higher mass. A Pareto front was developed to compare the tradeoffs of the two objectives.

The first objective of mass is determined by the area of the active region of the morphing aileron. The assumption is that the area which remains unchanged forward of the active section, and the trailing edge fairing aft would be of similar construction as a conventional aileron system having the same density. The active section, however, will be comprised of the center plate, polymer, and the FMC actuators and has a density greater than the structure which would otherwise be there. By minimizing the percentage of the airfoil used for the active region, the mass will be minimized. This simplifies the objective function of mass. Once an optimal design is found an actual comparison of the mass of an FMC morphing control surface and a conventional control surface can be made, but during the optimization, the reduction of the active region gives the required trend. The area of the active region will be represented as a percentage of the entire airfoil cross section.
The second objective of the optimization is to minimize the hydraulic energy requirements. During both an up and a down deflection, the model provides the hydraulic energy required by the FMCs to produce the required deflection. For the optimization, the sum of the energy required for an up and down actuation will be considered. The minimization of the sum was considered instead of the minimization of the maximum of the two cases because typically when one aileron moves up, the opposite wing’s aileron moves down. While there are exceptions to this such as gust load alleviation systems which can deflect both ailerons in the same direction to control wing loading, these represent only a small portion of a typical flight.

5.2 Optimization Constraints

The optimization was subject to several constraints which were derived from the design case, limitation of the actuators and others due to design feasibility. In all, thirteen constraints were used, and each is numbered for reference in Table 5.1. The first two constraints state that both deflections up and down must be greater than or equal to the 20° as required in the design case (Table 3.1). This will be determined using the equivalent flap deflection discussed in Section 3.5.

<table>
<thead>
<tr>
<th>Number</th>
<th>Constraint</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Up Deflection ≥ 20°</td>
</tr>
<tr>
<td>2</td>
<td>Down Deflection ≥ 20°</td>
</tr>
<tr>
<td>3</td>
<td>Hydraulic Pressure Up ≤ 8 MPa</td>
</tr>
<tr>
<td>4</td>
<td>Hydraulic Pressure Down ≤ 8 MPa</td>
</tr>
<tr>
<td>5</td>
<td>LE Top FMC ≥ 1 cm above plate</td>
</tr>
<tr>
<td>6</td>
<td>TE Top FMC ≥ 1 cm above plate</td>
</tr>
<tr>
<td>7</td>
<td>LE Bottom FMC ≥ 1 cm below plate</td>
</tr>
<tr>
<td>8</td>
<td>TE Bottom FMC ≥ 1 cm below plate</td>
</tr>
<tr>
<td>9</td>
<td>LE Top FMC ≥ 1 cm below top surface</td>
</tr>
<tr>
<td>10</td>
<td>TE Top FMC ≥ 1 cm below top surface</td>
</tr>
<tr>
<td>11</td>
<td>Bottom FMC ≥ 1 cm above bottom surface</td>
</tr>
<tr>
<td>12</td>
<td>Maximum curvature of plate up ≤ 3.3 m⁻¹</td>
</tr>
<tr>
<td>13</td>
<td>Maximum curvature of plate down ≤ 3.3 m⁻¹</td>
</tr>
</tbody>
</table>
The third and fourth constraints limit the hydraulic pressure required by the FMCs in both the up and down deflection at 8 MPa, a reasonable upper limit for the FMCs used to develop the empirical model in Section 3.2.5. Due to this constraint on the pressure and using the empirical model of the FMC no additional constraint was required on the strain or force output of the actuators.

In the model, the three internal components, the center plate, and both sets of actuators are represented by line elements with no thickness. As a result, a design could be analyzed which places any of these line elements ends at the same node, or extremely close. This is not physically possible, so constraints five through eight limits the ends of each of the three components to not be within 1 cm of the each other.

When looking at the placement of the FMC actuators, they could be placed in the model at a point near the upper and lower surfaces of the aileron which are not physically possible. Constrains nine and ten state that the leading and trailing ends of the upper actuator cannot be within 1 cm of the upper surface. When looking at the lower surface of the airfoil section in the region where the active section is located the surface is concave. If similar constraints were used on the lower FMC like those placed on the upper FMC, it would be possible that the two ends could be greater than 1 cm from the surface, but a point along the actuator be less than the 1 cm limit. Depending on the orientation of the lower actuator this point where the two are closest can move (Figure 5.1). As a result constraint eleven limits the closest point along the lower actuator to the lower surface be greater than 1 cm. Constraints five through eleven will prevent the aft location of the active section from moving aft beyond 1.88 m. At this point, the airfoil is 4 cm thick, the physical limit at which all the TE constraints can be satisfied.
The final two constraints place a limit on the curvature of the center plate. For both the up and down cases the maximum curvature along the beam elements in the model representing the center plate are constrained to less than 3.33 m⁻¹ curvature based on strain limits for composite plates of the thickness being considered here.

5.3 Optimization Method

Several approaches were considered for optimization. Initially, Matlab’s default optimization routine using several derivative-based optimization algorithms mainly SQP and interior-point were used. Each uses a relatively small step size to estimate the required derivatives. Within the analysis, convergence criteria are set to stop the convergence between the structural and fluid solutions and the varying FMC input forces to achieve the required deflection. Typically, this convergence value is ±0.1° of equivalent flap deflection between consecutive iterations. As a result, if the change a particular design variable is small to the point that the resulting change in the flap deflection is less than the convergence criteria two different input values can result in the same output.

This issue is not uncommon and can typically be addressed several ways. The first is if possible reduce the convergence criteria in the analysis. While this was possible more
iterations would be required for the analysis resulting in impractical solution times. Another option is to increase the finite difference step size used in the optimization routine. This makes certain that the step size is greater than the convergence limit of the analysis. However, a step size which is too large results in a poor approximation of the derivative. It was found that a step size appropriate for one analysis would not work well for a different point in the design space.

This behavior of the analysis is similar to a stochastic system. One approach to optimization of stochastic systems is derivative-free optimization algorithm. Matlab’s pattern search algorithm using the mesh adaptive search (MADS) algorithm was selected. Pattern search algorithms work by polling neighboring design points along a mesh centered around the current point. If one of these polled point’s objective function value is lower than the current point, then the mesh is recreated centered around that point. The size of the mesh also increases taking larger steps in the design space. If none of the designs in the polls has a lower value, then the mesh is recreated with at a smaller size and the polling continues.

The pattern search algorithm was found to be more efficient when the design variables were each scaled to have similar magnitudes before being passed to the pattern search algorithm. For instance, the plate thickness is on the order of thousandths of a meter while material moduli are on the order of tens of thousands of Pascals. To correct for this, each variable was scaled from a range of 0-10 based on its upper and lower limits. This meant that the selected upper and lower limits had an important effect on how quickly the run would converge. For instance, considering the design variable for FMC bias which in theory could range from 0-0.3 bias, since 30% is the maximum free strain of the FMC actuators. If this range were used in the optimization, the initial step size used in the pattern search would typically be 10% of the range or 3% bias. A typical optimal bias is ~3% with 0.1% change having a large effect. This large step would make it initially impossible for the optimization to see the effect of actuator bias. The adaptive mesh sizing over iterations would decrease, and eventually, the bias design variable would change from its initial value, but only after all other variables have been optimized. Then moving to a new point would cause the mesh adaptive routine to increase the mesh size again, leading to another set of iterations for each design variable. This issue can cause the pattern search method to be very slow to converge. To overcome this as the design space became more defined in subsequent runs the upper
and lower bounds of each design variable was adjusted to include only the feasible design space. Typically, a margin was also added to the upper and lower limits to make certain that no feasible portions of the design space were excluded.

A Latin Hypercube Sampling (LHS) was performed on the entire design space for all design variables to give multiple starting points for the optimization. From the LHS many design points can be removed immediately without analysis because they are infeasible for having either of the FMC elements or the plate elements intersecting or violated any of the separation constraints. These design points were then run to produce an initial set of starting points. In all approximately two thousand runs were done as part of this initial sampling and used to set the initial limits on the design space used in the optimization. To allow the optimization to consider further regions of the design space 20% was added to this initial range for each design variable. Periodically throughout the optimization and development of the Pareto curve in the following sections, these limits were updated as different regions of the design space were explored. This limiting of the design space allows pattern search to operate efficiently, while periodically updating the limits allows the pattern search routine to explore the entire design space.

The pattern search algorithm can only optimize a single objective function. With the two objectives of reducing the active area and energy required one option is to use the epsilon-constraint method. Rather than minimizing the energy and active area objectives at simultaneously one of the objectives, in this case, the active area can be represented as a constraint. This then leaves only the remaining objective of energy to be minimized. To determine how this constraint on flexible area should be formulated comes from looking at the initial results of the optimization. It will be shown in more detail, but a relationship exists between the hinge line variable and active length variable, and these are the only two quantities driving the active area. With that relationship, it becomes simple to impose a constraint systematically to find the minimal energy for a given flexible area.

Plotting the values for different objectives allows the Pareto front to be determined. In Figure 5.2, each point represents a valid solution to a model. The position of the point is the result of two different objectives $f_1$ and $f_2$, for the morphing aileron optimization, this would be active area and hydraulic energy required. Point A would represent utopia point for the $f_1$ objective, meaning it is not possible to reduce $f_1$ any further. Similarly, point D represents
the utopia point for $f_2$. Looking at point E, it can be seen that moving to either point B or C would be more efficient since B offers a reduction in $f_1$ with no increase in $f_2$ and the opposite for moving to point C. Points A through D are consider Pareto efficient or nondominated points and form the Pareto front. It is not possible to move from a Pareto efficient point to a point with a lower value for one objective function value without an increase in the other objective function.

![Diagram](image.png)

**Figure 5.2: Generalized Pareto front.**

The weighting factors for each objective function can be difficult to determine to move to and find a specific point along the Pareto front. In the optimization of the morphing aileron, this can be avoided since the flexible area objective is only a function of the hinge line location and active length. A flexible area percentage can be specified and the hinge line and active length variables adjusted to meet it allowing the optimization to take place on only one axis of the Pareto plot.

### 5.4 Optimization Results

The result of the optimization is not a single optimal design but rather a series of designs along the Pareto front which spans from a design with the minimal, flexible area and greater hydraulic energy required to design for minimal energy at the expense of greater flexible area. Figure 5.3 shows the Pareto front for the design problem with the flexible area as a percentage of the entire cross-section along the x-axis and the total hydraulic energy on the y-axis. Figure 5.3 also has three design points marked A, B, and C. These points located at the two extremes of the Pareto front and the middle will be referenced in the following
sections when comparing the design points. Figure 5.3 is the end product of thousands of iterations of the analysis, and the following will examine in detail how the design of the FMC is morphing aileron changes along the Pareto front.

Figure 5.3: Optimization Pareto front for hydraulic energy and flexible area as a percentage of the entire airfoil cross section

5.4.1 Hinge Line and Active Length

Figure 5.4 shows the change in the hinge line location and active length concerning flexible area percentage along the Pareto front. Additionally, plotted is the sum of these two quantiles which is the point where the active section ends and the trailing edge fairing begins. This point at 1.775 m only changes a few millimeters from the two extremes of the Pareto front. At this point, the aileron has a thickness of only 4.16 cm. This is nearly as far aft as the active section can be due to the 1 cm separation constraint on each side of both FMCs meaning the thickness can never be less than 4 cm. The optimal for both the reduction in flexible area and the reduction in energy is to push the active region far aft. To achieve
minimal energy required the hinge line location moves forward at the expense of increased flexible area.

![Graph showing X-Distance (m) vs Flexible Area (%) for hinge line, active length, and start TE fairing conformance flexible area.]

**Figure 5.4:** Optimal value for the hinge line, active length, and trailing edge fairing concerning flexible area.

### 5.4.2 Internal Layout of FMCs and Plate

The positions of the two FMC sets and the center plate are set by design variables which define their position as a percentage of the thickness at either their leading edge (LE) or trailing edge (TE) locations (Figure 3.4). Figure 5.5 through Figure 5.7 show the results for each position variable for the three components.

![Graph showing Y-Position (%) vs Flexible Area (%) for top FMC's LE and TE ends.]

**Figure 5.5:** Position of the top FMC’s LE and TE ends as a function of flexible area
First looking at the TE location of the three components it can be seen that each has approximately the same curve though offset by a small percentage of the thickness, at all but the smallest flexible lengths. This is because the TE locations of both FMCs have converged to the 1 cm separation constraint between each actuator and the plate, constraint numbers 6 and 8 (Table 5.1).

Since the TE locations for the FMCs remain fixed at 1 cm above and below the center plate at the TE, the LE positions can be looked at as each component’s angle relative to the chord line (Figure 5.8Error! Reference source not found.). The top actuator and plate both have a negative angle, meaning the LE location is higher than the TE. The two are nearly parallel but converge toward the trailing edge for all flexible lengths. The lower actuator has a nearly
level position at the lowest flexible length but increases angle as the flexible length increases. This is due to the load on the active section is increasing with the increasing size. The increased angle of the actuator allows more of the force generated by the lower actuator can contribute to a downward force at the end of the plate with a decrease in the contribution to the moment at the plate’s end.

Figure 5.8: Angle of internal components relative to the chord line.

The angle of the lower FMC quickly increases from the nearly level as the flexible length increases but then remains nearly constant at 5° after 4.5% flexible length. Figure 5.9 shows the X location of the point where the lower FMC is closest to the lower surface (constraint 11 Table 5.1). At the lower flexible areas, the location is at the aft point where the TE fairing is attached. As the LE positions of the bottom moves down the point where the actuator and lower surface are closest changes its x-position. Due to the concave shape of the lower surface both the LE and TE points of the bottom actuator effect this position (Figure 5.1). This is the reason constraint 11 had to be written as a single constraint compared to the corresponding constraints 9 and 10 for the top actuator (Table 5.1). This causes the angle of the lower FMC to have a maximum of 5° and is also the cause of the upper limit on the Pareto front. For a design to achieve lower energy at larger flexible lengths and extend the Pareto front the LE position of the bottom FMC would need to move lower. The concave surface of the airfoil prevents this.
Looking at Figure 5.5 through Figure 5.8 the trend observed for some design variables like bottom FMC LE and TE positions are very apparent while others like the top FMC LE position do not converge to an obvious curve. This is a result of the analysis having a limit to which it converges, causing the stochastic response; this is the same reason pattern search was selected over gradient-based optimization methods. In the analysis, the equivalent flap deflection is used to measure convergence and stop the iterations, not the hydraulic energy required. Some design variables have a greater effect on the solution and tend to converge quickly, while other variables with a smaller effect do not.

To understand this further, the optimal solution for a flexible area of 4.4% (point B in Figure 5.3), which lies at the middle of the design space, was analyzed multiple times with different initial values assumed for the up and down FMC force input. This bounds the variation in the output to identical design case inputs. Solutions were also created for small steps above and below this design point for each design variable. Figure 5.10 shows the change in hydraulic energy for ±0.4% change in the LE position of the top and bottom actuators. The base condition shows a 6 J/m range between the lowest and highest energy. In the bottom FMC case, a trend showing an increase in energy with increasing LE position can be seen over the ±0.4% range. In the top case, the change in energy across the ±0.4% change is not greater than the range of the baseline solutions. The solution is more sensitive
to the position of the bottom actuator’s LE position because of the bottom actuator consumes approximately twice as much energy. Additionally, the top actuator is nearly parallel to the plate while the lower actuator is at a significant angle to the plate. When the two components are nearly parallel, the small change in angle has little effect on the portion of the FMC contribution to moment versus traverse force at the end of the plate.

Figure 5.10: Effect of LE position of top and bottom FMC actuators ±0.4% from the baseline condition of total hydraulic energy required

The variable which has the most apparent effect on the result is the thickness of the center plate which provides a majority of the bending stiffness. Figure 5.11 shows that for even changes as small as a 0.01 mm in thickness causes a change in energy greater than the 6 J/m variation seen in the baseline case. Appendix A has the sensitivity for all design variables for energy, strain, plate curvature, and hydraulic pressure.
5.4.4 Passive Stiffness Design Variables

Two factors drive the passive stiffness of the aileron. The center plate thickness and the modulus of elasticity of the polymer regions. To limit the number of design variables which had to be considered, the four polymer regions of the active section (Figure 3.6) were considered in two groups. The two regions making up the upper and lower surface will be referred to as the outer polymer region and the remaining two section above and below the center plate will be referred to as the inner region. The material properties are also considered to be isotropic with a varying modulus of elasticity. The Poisson’s ratio remains fixed at 0.499 for both the inner and outer polymer regions.

The primary concern with the materials stiffness is in bending. Figure 5.12 shows the effective bending stiffness of the plate and the inner and outer polymer regions. Since the polymer regions taper with the geometry of the airfoil and the position of the FMCs their contribution to the bending stiffness changes from the maximum at the hinge line location to a minimum at the start of the trailing edge fairing. The contribution of the outer polymer to bending stiffness is approximately an order of magnitude less than the center plate. The inner polymer region, being closer to the neutral axis, is another order of magnitude less.
To minimize the energy required by the morphing aileron, the passive stiffness of the structure and thereby the strain energy would need to be minimized. Though a certain amount of passive stiff is required to maintain the shape of the aileron and allow proper morphing. That is what is seen as the plate’s bending stiffness converges to 1100 Nm$^2$ at the larger flexible areas. The increase in stiffens of the plate seen at values below 4.4% is due to the decreased length of the plate and the constraint on the curvature of the plate in the optimization. A thicker plate prevents the curvature limits from being exceeded.

Figure 5.13 shows the curvature of the center plate for the up and down deflections at the three design cases A, B, and C highlighted in Figure 5.3. The x-axis is the position along the plate starting at the hinge line. Constraints 13 which limit the curvature to 3.3 m$^{-1}$ for the down case are always active for not just these three designs points but all Pareto optimal points. Since in all three cases the tip of the plate needs to rotate nearly the same amount case A with the shortest active length would need to reach and maintain the maximum or highest possible curvature throughout its length. The longer active lengths in the B and C case show the optimal designs have less curvature nearer the hinge line, where the aileron
has increased bending stiffness due to the thicker cross section. Minimizing the curvature is directly related to minimizing the hydraulic energy required by reducing the strain energy of the structure. The longer active lengths with lower energy requirements minimize the curvature throughout the active length.

Figure 5.13: Change in plate curvature along the length of the plate for the three design points

5.4.5 Strain and Energy of the FMC Actuators

Figure 5.14 shows the installed bias strain for the top and bottom actuators for the optimal value along the Pareto front. This response is driven by the strain of the actuator when operating passively. There is considerable variation in the optimal bias strain from one point to the next. This is again due to the convergence limits of the analysis but is also indicative of the actual system. The penalty to the energy consumed for an increase in the bias strain and the actuator operating in a less efficient portion of the stroke is relatively small. Decreasing the bias strain and having the passive actuator become taught could have
a large effect on the energy due to the increased passive stiffness. However, the stiffness curve used is cubic. The increase in stiffness for the first portion of the curve is negligible. So a passive actuator with zero strain and a different case with a small positive strain would have nearly the same energy.

![Graph](image)

**Figure 5.14: Installed bias strain for the top and bottom FMC actuators.**

Figure 5.15 shows both the passive actuator’s strain due to morphing from the installed length and the total strain. The total passive strain for both the up and down cases is nearly zero for all flexible lengths, meaning that as the control surface morphs the passive actuator transitions from slack to taught at full deflection. This shows that the penalty due to the increased passive stiffness of a taught passive actuator is higher than the penalty of the actuator operating in a less efficient region due to the installed bias.
Figure 5.15: Strain due to morphing and the total strain (including bias) of the passive actuators for Pareto optimal points.

The strain of the active actuator is shown in Figure 5.16. Similar to the passive actuator, there are two strain quantities of importance due to the installed bias. The strain due to morphing (Figure 5.17) is only affected by the deflection and geometry of the morphing structure, primarily the positions of the FMC attachment points. The total strain is the strain required of the actuator and is the strain due to the installed bias and due morphing and give the actual strain the actuators need to be capable of. This is the quantity used in the empirical model found in Chapter 3 to determine the hydraulic energy required.
Figure 5.16: Strain of the active actuator up and down. A is the strain due to morphing, B is the bias strain, and C is the total strain the actuator needs to be capable of.

The active actuators strain is one quantity along with pressure and force which go into the calculation of hydraulic energy. Since the strain required remains relatively constant throughout the designs, the force and pressure vary to account for the changing energy required along the Pareto front with the down case requiring 1.5-2 times more force and pressure (Figure 5.17). Figure 5.18 shows the resulting energy required. Due to down deflection requiring more strain the difference in the up and down cases is further amplified with the down case requiring 2-3 times the energy.
Figure 5.17: Force (left) and pressure (right) for the upward and downward deflection for designs along the Pareto front.

Figure 5.18: Hydraulic energy required for up and down deflection cases for design along the Pareto front.

5.5 Design Point Comparison and Conclusions

Three points along the Pareto front were noted as A-C in Figure 5.3. Designs A and C represent the far ends of the Pareto front and solution B is a design near the middle of the design space. By looking at these three points in more detail, some conclusion can be drawn about the optimization, the design case and provide design guidelines for the design of
morphing control surfaces using a similar design be but considered for other design cases. At the end of this chapter, Table 5.2 has the values for all the input and outputs of the analysis of the three design points.

**Design Guidelines**

1. When considering a design case scenario, the thickness of the airfoil will play an important role. Comparing the angle of the upper and lower actuators relative to the center plate for the three cases (Figure 5.19). The upper actuator remains nearly parallel, while the lower converges toward the TE. As the active length increases so does the angle of the lower actuator. If it were not limited by the geometry and constraints, this angle would only increase further with increasing flexible area. The FBD (Figure 4.13) of the aileron showed that this angle goes into determining what portion of the actuators work creates a force versus moment at the tip. The optimization shows that in the down case, more moment is not needed as the FMC has not reached its force output limit. The limit that has been reached due to the geometry is the ability to generate appreciably more force down at the tip of the plate. Increasing this angle further is what is required to efficiently increase the flexible length of the morphing section.

2. The shape of the airfoil is an important driver, beyond just the aerodynamic loads. The camber creating the concave lower surface in the area of the morphing section was an important driver in the design case. Figure 5.19 has a red X at the location where the bottom actuator is closest to the surface. In case A this happens at the TE of the actuator but quickly moves forward with increasing flexible area and it limited by the concave surface. This behavior was shown in Figure 5.9. A different design case which considered a symmetric airfoil or one without a concave surface would not be limited in the same way.
3. In this design case like any design case for the main wing, the downward deflection case required considerably more energy than the up case as it deflects the surface into the flow. This was evident at all the optimal designs, with the down deflections requiring 2-3 times more energy. Another design case, a vertical stabilizer, for
instance, would not have this asymmetric loading, and a different optimal condition would result. The plate would move to the chord line and the position and bias values of the actuators would be the same on both sides. Each side would have a design similar to the down case considered in this work.

4. One of the first trends observed in the optimization was that the active section’s aft point should be as far aft as possible. The separation constraints placed on the FMCs and plate prevented this point from moving further aft into the thinner portion of the airfoil. For future designs finding FMCs capable of operating in, even more, constrained spaces should be considered. In none of the Pareto optimal points were the FMCs operating well within the typical pressure and strain limits of the actuator.

5. The one constraint on the optimization which was active at every point was the plate’s curvature down. This was shown graphically in Figure 5.13, and actual deflections for the three designs can be seen in Figure 5.20 and Figure 5.21. This should be the first point of consideration in a design. Concepts like the FishBAC (Section 1.1.5) offer good solutions for this using complaint materials which can also be 3D printed to allow stringers and other needed structure to be arranged to provide optimal stiffness.

![Figure 5.20: Up deflection for the three design points including center plate](image)
Figure 5.21: Down deflection for the three design points including center plate

6. As mentioned in the first design guideline the angle of the actuators is the deciding factor in the contribution to the tip force vs. moment from the FMC. The design is consistently limited by the vertical force component. Designs should be considered which can help maximize this. From a design perspective two quantities need to be controlled tip force and tip moment, but one factor, the FMC angle relative to the plate, controls them. Implementing a design which can use different actuators at different angles could allow for more control not only in the design process but also in control during operation. If a design were considered where the center plate was not continuous along the span but rather in sections with small breaks many new designs could be considered. In those open areas between plates actuators running at the maximum possible angle could be installed, giving significant control to the vertical force component.
Table 5.2: Results for points A-C

<table>
<thead>
<tr>
<th>Parameter</th>
<th>A</th>
<th>B</th>
<th>C</th>
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</thead>
<tbody>
<tr>
<td>Flexible Area (%)</td>
<td>3.85</td>
<td>4.40</td>
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<tr>
<td>Hydraulic Energy Up (J/m)</td>
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<td>1476</td>
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<td>FMC Force Up (kN/m)</td>
<td>76.4</td>
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<td>Pressure Up (MPa)</td>
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<tr>
<td>FMC Active Strain Down (%)</td>
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<tr>
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<td>-0.87</td>
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<td>Top Installed Bias (%)</td>
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<td>Bottom Installed Bias (%)</td>
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<td>Top FMC TE Position (%)</td>
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<tr>
<td>Inner Polymer Modulus (kPa)</td>
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Chapter 6

6 Conclusions and Recommendations

6.1 Conclusions

The purpose of this work was to demonstrate the use of FMC actuators in a morphing aircraft control surface for a commercial transport aircraft. This was motivated by there being a gap in the current technology. Concepts like the VCCTEF show the potential benefits of morphing through analysis and the ACTE shows what is currently possible through manned test flights while concepts like the FishBAC show the future potential of novel designs, but are limited to UAV scales. To demonstrate the feasibility of an FMC morphing control-surface several developments were made:

1. An FMC morphing spoiler was created and tested as a benchtop prototype. This included testing under pseudo-aerodynamic loads and demonstrating that close loop control was possible using an embedded flexible sensor.

2. A design case was developed for a morphing aileron on a commercial transport aircraft. A coupled analysis was created which allowed a morphing FMC aileron to be designed taking into consideration the structural model, aerodynamic loading, and FMC performance.

3. Using this model, the design space of the morphing aileron was explored through a series of trade studies. This identified feasible areas of the design and provided insight into the interaction of design variables.

4. Optimization of the morphing aileron design case was performed. Optimization of the mass and hydraulic energy required showed a Pareto front which could consider different designs optimized for different purposes.
5. From this optimization a set of design guidelines were created to guide future designs for both this design case and what should be considered to alternate design cases.

6.2 Future Work

As previously mentioned this work lies at a point between something like the ACTE which is highly constrained in the design as it is pushed toward manned flight testing and the more novel approaches being investigated at smaller UAV scales. For the future work of this design, several directions should be considered which could further the design through continued novel research or increase its maturity for integration with an actual airframe.

- This work was limited to looking only at a 2D cross-section of a particular point on the wing. To further the design a clean sheet design approach to how the morphing system could be used should be considered. This would be similar to the work done with the VCCTEF but with specific considerations learned from this work related to the FMC morphing control surface integration.

- Current conventional control surfaces attach at hard points in the structure which require the airframe to handle the point loads at the hinges and particularly at the attachment of the hydraulic jack used to deflect the control surface. The morphing design could offer the benefit of a continuous more evenly distributed load which could have weight saving benefits in the airframe forward of the morphing section. This point was not considered as part of this work but could show additional benefits of morphing and offset some of the weight penalty.

- The external environment an airframe has to operate in can be extreme from sitting on the ground in extreme heat to extreme cold at altitude. Additionally, the UV exposure of the airframe could be an issue for morphing designs using polymer surfaces. Current composite aircraft parts are painted to prevent UV exposure; this would not be possible for a morphing surface. The polymers used in this work and often considered in other morphing research work have temperature dependent material properties and can prematurely break down with UV exposure. This point is something which is often overlooked in many of the designs researched for this
work. Further work is needed to investigate possible material choices capable of the required high strains while also operating in the required environment.

- New materials should also be considered which are tailored to the design. One promising area of research for this is the Programmable Honeycomb Polymer Composite (PHPC) [66]. This material uses a honeycomb structure embedded in a shape memory polymer to create a controllable stiffness material. This would allow a morphing control surface to change stiffness as needed and potentially reduce the load on the FMC actuators.

- The control of an FMC morphing control surface was demonstrated in Chapter 2. Further work in this area is needed in particular to detect and understand possible failure modes of the design and how the system would respond. Modern aircraft use sensors throughout, including the hydraulic system to predict failures and put in for maintenance before even a partial failure. How morphing control surfaces could be integrated into similar systems should be considered.

- The analysis developed in this work was limited 2D. However, in actuality, there are multiple parallel FMCs. Not all of those FMCs are needed at every point in the flight or for every deflection of the control surface, only the maximum load and deflection cases require this. So it would be possible to develop a design which only uses the FMCs needed. This concept is referred to as variable recruitment and has been looked at for fluid driven artificial muscles [67]. This also has the potential to have different FMC designs alternating along the span with some of the design variables like fiber wind angle and diameter varied. This gives the option to have groups of FMCs be optimized for very fast actuation required by gust load alleviation systems, and others could be optimized for more efficient actuation for the majority of the flight regime.

- This work focused strictly on the aviation application of morphing control surfaces. However, the FMC morphing control surface could have potential applications in the automotive and naval industries.

This work has demonstrated that an FMC actuated morphing control surface for a commercial transport aircraft is a viable option for further development. This was
accomplished through a benchtop prototype which demonstrated the concept and that closed-loop control is possible and further supported by analysis and optimization showing the potential of the design to be used in an aileron application
Appendix A

This appendix provides additional plots to those discussed in Section 5.4.3. Each plot provides the response of the analysis for some output (energy, pressure, etc.) to small changes of each design variable about a baseline condition at a flexible area of 4.4%.
Figure A.1: Energy for the up deflection for small changes about the baseline case.
Figure A.2: Energy for the down deflection for small changes about the baseline case.
Figure A.3: Strain in the passive actuator for the up deflection for small changes about the baseline case.
Figure A.4: Strain in the passive actuator for the down deflection for small changes about the baseline case.
Figure A.5: Hydraulic pressure for the up deflection for small changes about the baseline case.
Figure A.6: Hydraulic pressure for the down deflection for small changes about the bassline case.
Figure A.7: Strain in the active actuator for the up deflection for small changes about the bassline case.
Figure A.8: Strain in the active actuator for the down deflection for small changes about the baseline case.
Figure A.9: Plate curvature for the up and down deflection cases for small changes about the baseline case.
References


