

ABSTRACT

Title of Dissertation: ANALYSIS OF PNEUMATIC ARTIFICIAL
MUSCLES IN ANCHORING, TORSION AND
WEIGHT-LIMITED ROBOTIC APPLICATIONS

Frank Cianciarulo
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Dissertation Directed by: Professor Norman M. Wereley
Department of Aerospace Engineering

Pneumatic artificial muscles (PAMs) are studied in two novel actuation methods: radial expansion and torsional response. In addition, PAMs utilizing novel material for light weight scenarios are studied and applied to a sweeping wing mechanism in an unmanned aerial system (UAS).

For the radial expansion model, modeling was performed using a force balance approach to capture the effects that bladder strain and applied axial load have on the anchoring force. Radial expansion testing was performed to validate the model. Force due to anchoring was recorded using force transducers attached to sections of aluminum pipe using an MTS servo-hydraulic testing machine. Data from the test was compared to the predicted anchoring force. Radial expansion in large diameter (over 2 inches) PAMs were then used in worm-like robots to create anchoring forces that allow for a peristaltic wave which creates locomotion through tunnels and pipes. Modeling of the torsional response was also performed using a force balance approach,

and multiple model variations were considered, such as St. Venant's torsion, bladder buckling, and asymmetrical braid loading. Torsional testing was performed to validate the model using a custom torsional testing system. Data from the tests was compared to the predicted torsional response.

In addition to alternative forms of actuation, alternative materials for end fittings were also tested. Two materials were tested, stand alone Acrylonitrile Butadiene Styrene (ABS-R) and ABS embedded with carbon fiber. These materials were chosen to reduce the overall weight of the PAM and increase the specific axial force output. Multiple iterations were designed and tested to achieve an operable pressure range of zero to one hundred psi with a minimum safety factor of 1.5. The PAMs also underwent endurance testing to determine the number of cycles before fatigue failure. The final iteration of the PAM design for both materials were compared to traditional metallic end fitting PAMs in specific force production as well as fatigue life.

Once design work was complete, the 3D-printed end fitting PAM was utilized in a sweeping wing mechanism for a UAS. Due to their small size, UAS possess the ability to fly through confined areas that full-sized aircraft would not be able to. This, however, presents the challenge of an increased risk of collisions with obstacles present in these areas. An active wing sweeping mechanism was proposed to both mitigate the possibility of, and damage associated with, collisions. The mechanism was capable of sweeping the wings of an E-flite V900 BNF Basic remote controlled plane by 45 degrees. Multiple bench top tests were performed on the system, such as a simple actuation test, a response time test, and a hardware-in-the-loop (HITL) test. The gear system was found to be fully capable of sweeping the wings from the default 35.4 inch wingspan to a 24 inch wingspan and clearing a 30 inch gap at speeds up to 55 mph in the HITL tests.

ANALYSIS OF PNEUMATIC ARTIFICIAL MUSCLES IN
ANCHORING, TORSION, AND WEIGHT-LIMITED ROBOTIC
APPLICATIONS

by

Frank Cianciarulo

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Advisory Committee:

Professor Norman M. Wereley, Chair/Advisor
Professor David L. Akin
Professor Craig Carignan
Professor Sung W. Lee
Professor Ross H. Miller

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List of Abbreviations

A	Wing area
ABS-CF	Acrylonitrile Butadiene Styrene - Carbon Fiber
ABS-R	Acrylonitrile Butadiene Styrene
B	Braid length
$C(x)$	Non-linear constraint
C_l	coefficient of lift
CPAM	Contractile Pneumatic Artificial Muscle
dE_c	Incremental internal circumferential energy
dE_z	Incremental internal axial energy
D_0	Initial diameter
D	Diameter
DH	Denavit-Hatenberg
DoF	Degree of freedom
E	Tensile modulus
EPAM	Extensile Pneumatic Artificial Muscle
F	Spring force
F_1	Force in braid 1
F_2	Force in braid 2
F_{appl}	Applied axial force
$F_{Gaylord}$	Gaylord force
F_E	Experimental force
F_M	Model force
F_R	Radial force
FEA	Finite Element Analysis
G	Shear stress
HITL	Hardware in the Loop
HPP	half-peak-to-peak
J	Polar moment of inertia

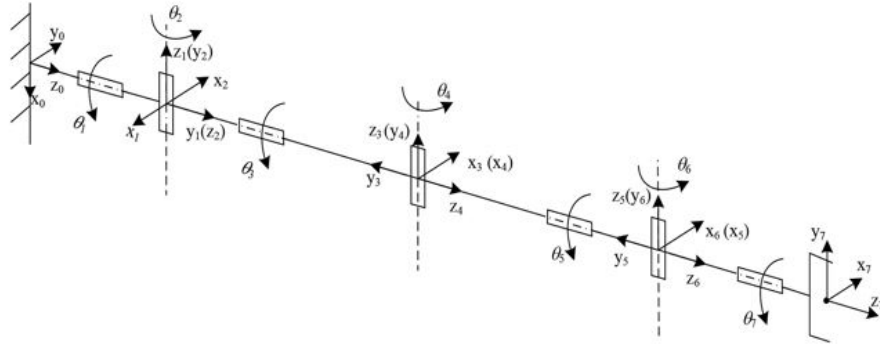
j	Number of data points
K	Combined spring constant
k	Spring constant
L_0	Initial length
L	Length
L_A	Lift generated by aircraft
lb	Lower bounds
m	mass
m_{3D}	mass of 3D-printed end fitting PAM
m_M	mass of metallic end fitting PAM
MTS	Material Testing System
N	Number of turns of the braid
n	Number of tested pressures
P	Pressure
P_a	Anchoring/contact pressure
PAM	Pneumatic Artificial Muscle
PET	Polyethylene Terephthalate
R_0	Initial radius
R	Radius
R_i	Inner radius
R_o	Outer radius
S	Slope
SAF	Specific Actuation Force
SAF_{3D}	SAF of 3D-printed end fitting PAM
SAF_M	SAF of metallic end fitting PAM
t	Thickness
T	Braid tension (Chap. 2)
T	Torque (Chap. 3)
T_E	Linear fitted experimental torque
T_M	Model torque
T_d	Torque from data
ub	Upper bounds
UAS	Unmanned Aerial System
V_B	Bladder volume

v	Aircraft velocity
x	Optimization variable
y	Spring displacement
θ_0	Initial braid angle
σ_c	Circumferential stress
σ_z	Axial stress
ϵ_c	Circumferential strain
ϵ_z	Axial strain
τ	Shear stress
γ	Shear strain
ϕ	Angle of twist
ρ	Density of air

Chapter 1: Introduction

The field of robotics can be split into two categories, hard robotics and soft robotics. Most classical types of robots can be placed into the hard robotics field, characterized by their use of high-stiffness material, such as metal, in both their structure and in their actuator. Hard robots benefit from these stiff components by not only being able to manipulate heavy payloads without fear of yielding the structure, but also by assuming these components are rigid and that motion only happens at specified locations, or joints, which simplifies the process of defining the motion and workspace of the robot [9]. Hard robots can be represented by a series of revolute or prismatic actuators in series or parallel, whose parameters can be described by a Denavit-Hatenberg (DH) parameter table, which greatly simplifies the process of determining the kinematics of the robot, as shown in figure 1.1 by Yan et al [1]. Hard robotics have been designed for surgical use, manufacturing, vibration isolation, and on orbit space debris mitigation, among many other uses [1, 10–13].

These benefits come with multiple costs, however. The first being the cost of weight. These stiff material tend to be dense, resulting in robots that can struggle to fulfill operational needs in weight-limited missions. Additionally, the high stiffness of these robots results in poor compliance, especially in human-robot interactions. While this issue can be accounted for by various control methods, the stiff structure still poses a risk to robots working in conditions with



Link i	$\theta_i / ^\circ$	$\alpha_i / ^\circ$	a_i / m	d_i / m
1	-90	90	0	$d_1 = 0.15$
2	180	90	0	0
3	0	-90	0	$d_3 = 0.4$
4	0	90	0	0
5	0	-90	0	$d_5 = 0.4$
6	0	90	0	0
7	0	0	0	$d_7 = 0.1$

Figure 1.1: Top: Seven degree-of-freedom (DoF) arm represented by a series of revolute actuators. Bottom: parameters of the seven DoF arm represented in a DH table, by Yan et al. [1]

high human population or fragile environments [14, 15].

Soft robots can be classified as robots who are comprised of soft material, such as rubbers or plastics. The actuators and structural links of these robots are capable of conforming to their environment, introducing an intrinsic compliance to the robot, typically making it well suited for confined or unpredictable environments and scenarios that hard robots struggle to operate in [16, 17]. These include human-robot interaction [18–23], biomimetics [24–29], and difficult terrain [30–35]. Additionally, because these soft materials are typically less dense than traditional materials used in hard robots, soft robots tend to weigh less, making them good candidates for applications where weight restrictions are a driving factor, such as those where high specific work or specific power is required [36]. Soft robot actuators come in a variety of types, consisting of inflatable everting tubes utilizing pneumatics for tunneling, as shown in figure 1.2, soft tubes that produce torsion under vacuum, and liquid crystal elastomers that deform when exposed to light stimuli, among many others [2, 37–39].

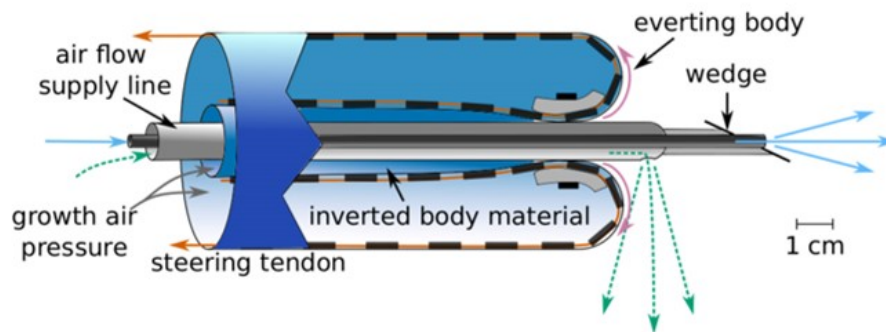


Figure 1.2: Inflatable everting tube soft robot utilizing pneumatics for tunneling, by Naclerio et al. [2]

Among soft actuators are pneumatic artificial muscles (PAMs). PAMs are biologically inspired actuators modeled after the macroscopic function of a muscle. PAMs are comprised of a

soft bladder covered in a braided sheath. The two pieces are held together by use of an end fitting and epoxy, the former providing a compression fit to bind the bladder and braid together, while the latter helps to create an air-tight seal, as well as strengthening the connection between the braid and bladder. When inflated with a working fluid, the bladder expands against the braid, and the PAM actuates. PAMs exist in two varieties, being a contractile PAM (CPAM) and an extensile PAM (EPAM). The differentiation between the two lies in the angle of the braid. At a braid angle of 35.26 degrees or higher, the inflation of the bladder against the braid results in a contraction of the ends of the PAM, while at an angle below 35.26 degrees, the inflation of the bladder results in an extension of the PAM of up to 80% the initial length. [40–42]. While EPAMs have many useful applications in the field of soft robotics, such as allowing for large effective workspaces in continuum soft robots [43], all PAMs discussed in the remainder of this work will be CPAMs.

1.1 PAM history

PAMs were developed by Richard Gaylord in 1958, in which he derived the Gaylord force equation to describe the force generated by the axial contraction of the PAM [3].

$$F_{Gaylord} = \frac{P}{4\pi N^2}(3L^2 - B^2) \quad (1.1)$$

Here, P is the internal pressure of the PAM, L is the length of the PAM, B is the length of the braid, and N is the number of turns of the braid around the PAM.

In the 1960's, Joseph L. McKibben popularized PAMs for use in orthotics to assist polio patients, where they were given the name "McKibben muscles". Though initially well received, PAMs were eventually replaced by orthotics utilizing electric motors as battery technology improved

[3]. Research on PAMs then stagnated until the early 1990's, when Bridgestone began utilizing them. During this time, PAMs were renamed as rubbertuators, shown being used to actuate a robotic arm in figure 1.3 [3]. Bridgestone's investment into PAMs sparked interest in PAMs by others, with companies creating their own versions of the PAM, such as Festo's Fluidic Muscle MAS, Shadow Robot Group's digit muscle, and General Electric's Underminer PAM robot, all created within the last thirty years [44–46].



Figure 1.3: Robotic arm driven by rubbertuators. Designed by Bridgestone. From Tondur. [3]

1.2 Evolution of the modeling and construction of PAMs

As mentioned, the Gaylord force equation was the first attempt to model a PAM. While it works reasonably well when the PAM is at low levels of contraction, the model struggles to accurately predict the force output when the muscle is highly contracted [47]. Since the development of this model, much work has been done to improve the ability of the model to accurately predict the force output along the entire stroke length of the actuator. One key improvement to PAM modeling is capturing the bladder stress present during actuation. Modeling of this stress has been performed in numerous ways, such as by use of a force balance approach, virtual work, and Mooney-Rivlin strain invariance, to create both elastic and hyperelastic models [36,48,49]. Additionally, other correction terms have been applied to existing models to account for smaller assumptions made in the model, such as addressing the accuracy of assuming inextensibility in the braid, using photogrammetry to check the assumption of a constant bladder radius along the length of the PAM during inflation, shown in figure 1.4, and addressing how the curvature of the ends of the bladder effect force output, and if assuming they are perfectly squared off, like in a typical cylinder, has significant effects on the model [4, 50, 51]. In addition to correcting assumptions, work has been done to address how the interaction between the bladder and the braid can effect the force output. These corrections include modeling the friction between the braid and the bladder, as well as the expansion of the bladder before contact is made with the braid, often referred to as the deadband pressure [52–56].

Significant work has also gone in to improving the construction of PAMs as well. Traditional design of PAMs did not have a standardized or well-studied method for sealing the ends, and as such would have issues such as pinhole leaks that were difficult to determine the cause of [57].

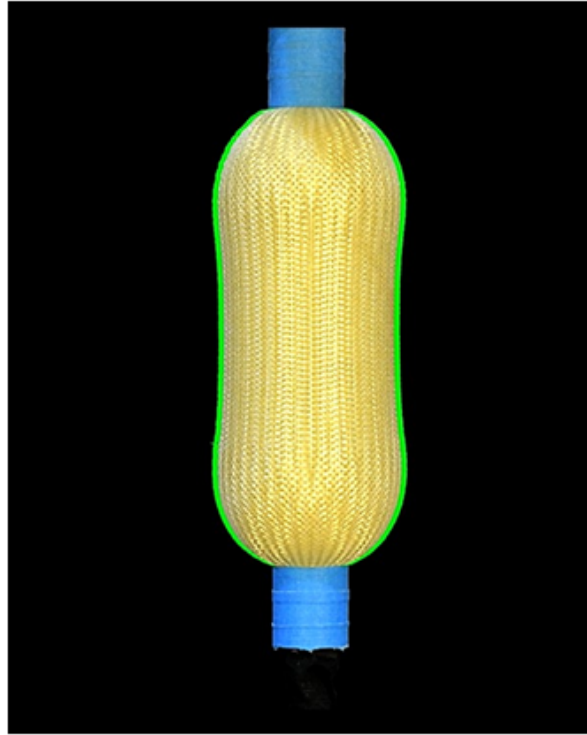


Figure 1.4: Photogrammetric outline of a PAM used to determine PAM radius during inflation by Chambers and Wereley. [4]

Improvements in the design of PAMs lead to the current swage design, in which an aluminum end fitting is inserted into the PAM bladder and braid, coated in epoxy, and fitted with a swage tube that is passed through an axisymmetric frustoconical die to compress the tube around the assembly to provide an air-tight seal. An example of a PAM constructed with this method is shown in figure 1.5 [5]. This improved design saw an increase in fatigue life from fourteen thousand cycles to over one hundred and twenty million cycles [5, 57, 58].

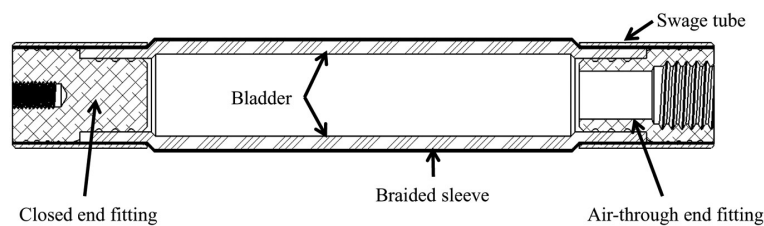


Figure 1.5: Design of swaged end fitting for PAM by Woods et al. [5]

1.3 Applications of PAMs

Due to the improvements in both the modeling and construction of PAMs over the last 5 decades, PAMs have been implemented in many types of applications. PAMs began life as a form of orthotic, and have been utilized in multiple orthotic systems since then. They have been implemented in forearm mounted, robotic hand exoskeletons for use by stroke patients, wearable body suits for inducing torso twisting and bending, and lower body exoskeletons to aid in offsetting the weight of tools used by workers during construction or maintenance tasks. An example of a wearable exoskeleton hand by Tjahyono et al. is shown in figure 1.6 [6, 59–61].

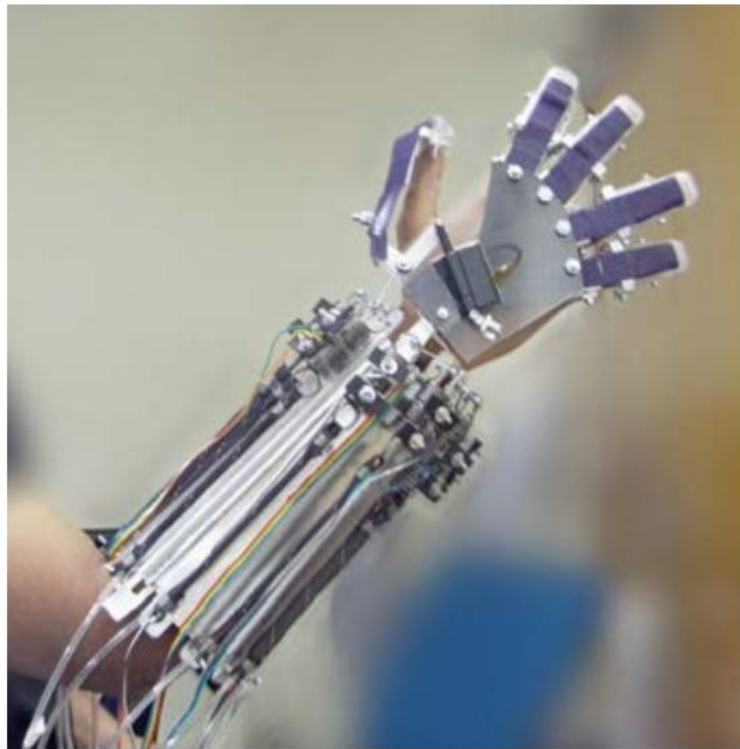


Figure 1.6: robotic exoskeleton hand utilizing PAMs by Tjahyono et al. [6]

Along with implementation into wearable robotics, PAMs have also been utilized in traditional robotic applications in the form of musculoskeletal arms, such as using multiple antagonistic

pairs of PAMs to control a seven DoF anthropomorphic arm. The use of these antagonistic PAM pairs allowed for a natural compliance, mimicking that of a human arm [59, 62]. While PAMs have traditionally been used in bio-inspired and biomimetic projects, such as the aforementioned exoskeleton and robotic applications, they have also found use in aerospace applications. Pneumatic actuators have been used in sweeping wing mechanisms, capable of generating and maintaining sweep of up to forty five degrees during wind tunnel tests of up to forty five miles per hour. An illustration of this mechanism is given in figure 1.7 [7].

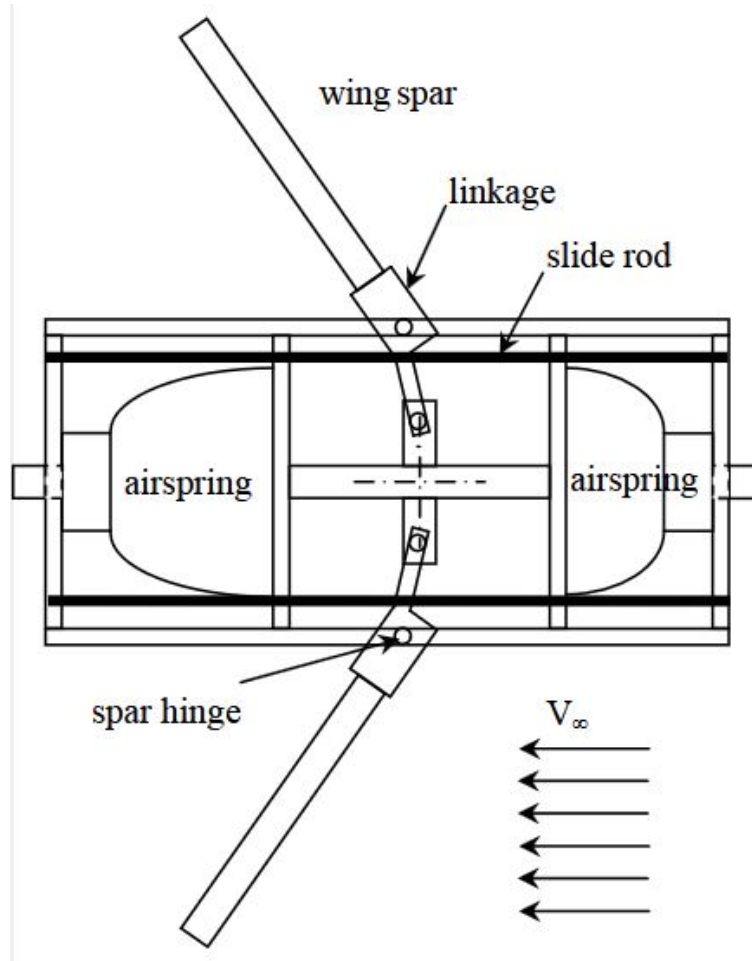


Figure 1.7: Illustration of sweeping wing mechanism using pneumatic actuators by De Marnier and Wereley. [7]

PAMs have also been implemented for use in multiple trailing edge flap actuation projects

due to their high specific force enabling them to generate sufficient actuation force to create 20 degrees of flap actuation at nearly 25 Hz for a mach speed of 0.1 for one fixed wing application, and being able to generate 17.1 degrees half-peak-to-peak (HPP) at 35 Hz during wind tunnel tests in a Bell 407 wing section at 0.3 mach speed for another [8, 63]. Figure 1.8 presents one of these designs by Kothera and Wereley.



Figure 1.8: Wind tunnel testing of fix wing with PAM powered trailing edge flap actuation by Kothera and Wereley. [8]

1.4 Problem Statement

While the PAM's axial force has been extensively modeled and utilized, a PAM is capable of producing other actuation forces that have not been as heavily studied. These actuation types, the radial force and the torsional response, could prove to be extremely useful in robotic applications, but the lack of modeling or understanding of them has led them to be relatively under-utilized. The radial force, generated by the radial expansion of the PAM, has been shown

to be capable of producing locomotion in pipe crawling robots, but has not been modeled. Additionally, PAMs are capable of reacting torsional loads, but this property has not been implemented as there is no way to predict how well PAMs can react these loads at a given pressure. Being able to utilize a PAM to not only produce axial loads but also radial and torsional loads could allow for robots capable of complex behavior without the complexity of using multiple types of actuators. In addition to a lack of research into alternative actuation uses for PAMs, there has been a lack of research into alternative materials for end fittings as well. While the effects of using alternative materials for other components of PAMs, such as aramid versus polyethylene terephthalate (PET) for braid or elastic material, such as latex, silicone, and Viton, versus inelastic material, such as low-density polyethelene, for bladder material have been studied, alternative materials for end fittings have not been thoroughly investigated. [50, 64]. Although it is true that various types of end fittings have been used, such as hose clamps and swaged end fittings, which is the common choice for most modern PAMs, these end fittings are typically made either partially or completely out of metal [5]. For swaged end fittings specifically, aluminum is the common choice of material. Alternative materials, such as plastic, could provide specific advantages over metals. Plastic is typically less dense than metal, and could allow for a lighter weight PAM that could be advantageous in weight-limited applications. Additionally, being able to 3D-print end fittings out of plastic could allow for rapid prototyping that would be both quicker and more cost-effective than creating multiple end fittings out of metal while designing and testing an actuator. The use of plastic does come at the cost of a reduction in strength, but PAMs with 3D-printed end fittings could be useful in scenarios where exceedingly high force generation is not required for the typical use case.

1.5 Outline of Thesis

In Chapter 2, We create a model for the radial force of a PAM using a force balance approach. We assume a fourth order stress-strain relationship and a first order pressure-modulus relationship for the bladder moduli, and utilize non-linear optimization to solve for the moduli. We create a custom test fixture and collect data of the radial expansion force at multiple pipe diameters to verify the radial expansion model. We show the model is capable of accurately predicting the radial force generated by the PAM, with and without an applied axial force. We then utilize this radial expansion force and model in a three segment bio-inspired, worm-like robot. The robot demonstrates the capability of practically utilizing the radial expansion force by crawling vertically through a pipe by mimicking the peristaltic wave of a worm. The radial force allows for anchoring of the robot into the pipe wall to support it's weight as it climbs the pipe.

In addition to the radial force, the torsional response is modeled in Chapter 3. We once again assume a fourth order stress-strain relationship and a first order pressure-modulus relationship for the bladder moduli, while also attempting a first order stress-strain relationship and a fourth order pressure-modulus relationship for the bladder moduli. We investigate the possibility of the torsional response taking the form of St. Venant torsion, bladder buckling, or asymmetrical braid loading. A custom torsional testing rig is constructed and used to record the torsional response of the PAM from 0 to 5 degrees of twist at 10 to 50 psi, in increments of 5 psi. The model is found to agree well with the collected data at all pressures.

In Chapter 4, a 3D-printed end fitting is constructed for a 7/8 inch PAM using Acrylonitrile Butadiene Styrene (ABS-R), with the goal of increasing the specific actuation force (SAF) of the PAM through weight reduction. Multiple iterations of the end fitting are constructed and tested

in blocked force, free contraction, and through full actuation sweeps to achieve a comparable operating range to metal PAMs of zero to one hundred psi, with a safety factor of at least 1.5. An alternate version of the finalized design is also constructed out of Acrylonitrile Butadiene Styrene - Carbon Fiber (ABS-CF) and tested. Finite element analysis (FEA) of the burst conditions are modeled to validate the experimental results. Endurance testing is performed on the PAM to ensure a reasonable number of actuation cycles could be performed before failure. The 3D-printed end fitting is found to surpass the target 1.5 factor of safety in all burst conditions, while improving the SAF by 12.86% and enduring over 50,000 cycles.

An application of the 3D printed end fittings is described in Chapter 5, in the form of a sweeping wing mechanism for an E-flite V900 BNF Basic remote controlled plane. The mechanism is nearly entirely 3D-printed, consisting of an electro-pneumatic system and PAM for actuation, and a gear mechanism to translate the actuation force into rotational motion to sweep the wings. Time to sweep was found for the system at multiple line pressures. A hardware-in-the-loop test is performed with an obstacle avoidance system. The sweeping wing mechanism was found to be able to avoid collision by simulating passing through a 30 inch wide gap by sweeping the aircraft's wings at speeds up to 50 mph.

Finally, conclusions and future work are discussed in Chapter 6.

Chapter 2: Analysis of anchoring muscles for pipe crawling robots

2.1 Overview

Pneumatic artificial muscles (PAMs) consist of an elastomeric bladder wrapped in a Kevlar braid. When inflated, PAMs expand radially and contract axially, producing large axial forces. PAMs are often utilized for their high specific work and specific power, as well as their ability to produce large axial displacements. Although the axial behavior of PAMs is well studied, the radial behavior has remained underutilized and is poorly understood. Modeling was performed using a force balance approach to capture the effects that bladder strain and applied axial load have on the anchoring force. Radial expansion testing was performed to validate the model. Force due to anchoring was recorded using force transducers attached to sections of aluminum pipe using an MTS servo-hydraulic testing machine. Data from the test was compared to the predicted anchoring force. Radial expansion in large diameter (over 2 inches) PAMs were then used in worm-like robots to create anchoring forces that allow for a peristaltic wave which creates locomotion through acrylic pipes. By radially expanding, the PAM presses itself into the pipe, creating an anchor point. The previously anchored PAM then deflates, which propels the robot forward. Modeling of the radial expansion forces and anchoring was necessary to determine the pressurization required for proper anchoring before slipping occurs due to the combined robot and payload weight.

2.2 Introduction

A pneumatic artificial muscle (PAM) is an actuator composed of a braided sleeve wrapped around an elastomeric tube or bladder, which is typically made from latex or silicone. When inflated with a working fluid, the PAM contracts and creates an actuation force, while the diameter of the PAM increases. The force that can be generated by a PAM is a function of the diameter and braid angle of the PAM [3]. PAMs have been found to be extremely useful for producing high strength forces in low mass actuators at relatively low pressures, allowing for maximization of specific power, specific work, and power density [5, 52, 53, 63, 65, 66].

Typically, the force produced by the axial contraction of PAMs is utilized to create actuation of systems [66–69], but force is also generated by the PAM through the radial expansion that occurs while the PAM axially contracts. Much like with the contractile force, a constraint limiting the PAM's ability to expand radially must be present for a reaction force to be generated, such as the inner walls of a pipe. While axial contraction force production in PAMs is well understood and various analyses exist to predict these forces [4, 48, 64, 70–72], this is not the case for radial force production, and no existing analysis can predict radial force at a given diameter and pressure. This study develops an analysis to predict the radial force generated by a PAM constrained in a tube or pipe, and validates the analysis via experiment.

2.3 PAM Design

A PAM is comprised of three main components: The bladder, the braided sheath, and the end fittings. For the PAM used in this paper, a 2.25 inch outer diameter, 0.25 inch thick gum

rubber bladder was used with a Kevlar braid. The aluminum end fittings hold the braid and bladder in place by use of a swage tube, which is compressed around it. The end fitting uses a novel two-part design that allows for access to the internals of the PAM without removing the end fitting and destroying the swaged outer tube, shown in figure 2.1.

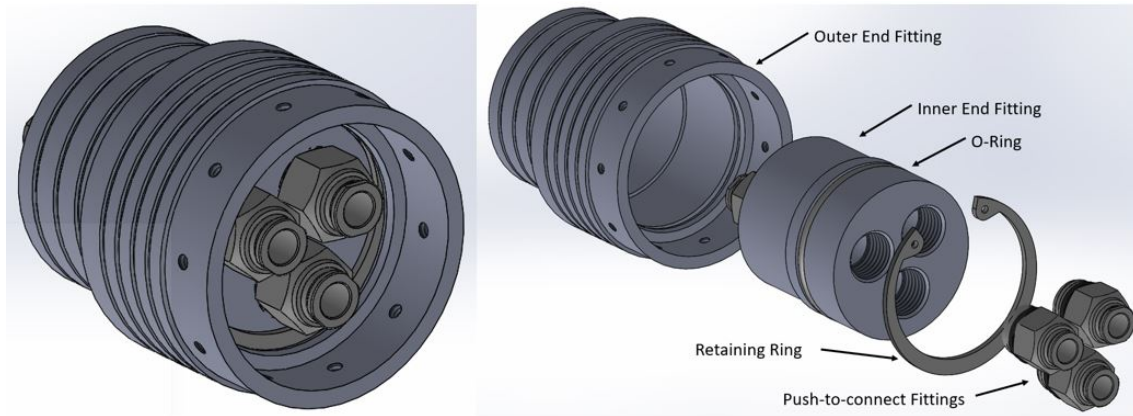


Figure 2.1: Left: Model of an assembled end fitting. Right: Exploded view of an end fitting.

These end fittings allow for the addition or removal of pneumatic feed-through tubes, which are necessary for robotic applications. The outer end fitting is used for the swaging process and contains a collar that allows the PAM to be connected to external features. The internal end fitting features three holes with 0.25 inch NPT threads on both ends to allow for up to three pneumatic attachments and feed through hoses. An o-ring is placed around the groove in the inner end fitting to provide an air-tight seal. A retaining ring is used to hold the inner end fitting in place during operation.

2.3.1 PAM Swaging

The swaging process consists of coating the outer end fitting in epoxy and inserting the smaller diameter end into the bladder. The Kevlar braid is then pulled over top of the end fitting-bladder assembly. The swage tube is then pulled over the end fitting assembly and the entire

piece is passed through a tapered die using a 10 ton press. The die compresses the swage tube around the other components, creating a unified part. The finished PAM is 12.25 inches long, 2.25 inches in diameter, with a braid angle of 66.35 degrees, shown in figure 2.2.

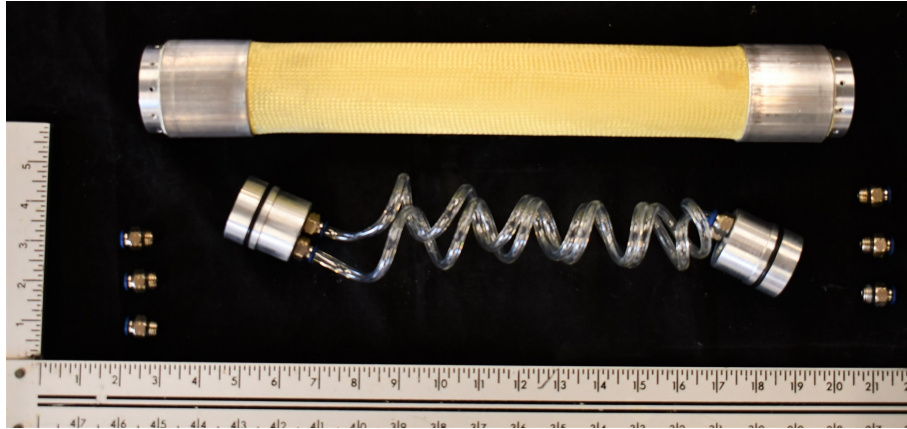


Figure 2.2: PAM with removed inner end fittings and pneumatic hoses.

2.4 Axial Contractile Testing

To characterize the radial expansion force of the muscle, first the contractile behavior in the axial direction needed to be modeled to determine key properties of the PAM such as the maximum force output, the maximum stroke length, and the stresses applied to the bladder. To find these, the PAM was tested on an MTS servo-hydraulic testing machine. The PAM was connected to the MTS through the use of MTS couplers that were attached to the collars of the end fittings at both ends. The PAM was first manually cycled between free contraction, where the PAM is allowed to fully contract without any applied axial force, and blocked force, where no contraction is allowed and maximum axial force production occurs, at 105 psi to account for the Mullins effect [73]. The PAM was tested between 10 to 100 psi in increments of 10 psi. For each pressure, the PAM was cycled between blocked force and free contraction three times, as shown

in figure 2.3. The MTS recorded displacement, force, and pressure. An Enfield D1 Bi-Directional Proportional PWM Valve Driver (Enfield, Trumbull, Connecticut) and an Enfield LS-V05s High Speed 5/3 Proportional Directional Valve (Enfield, Trumbull, Connecticut) were connected to an Arduino Mega (Arduino, Robbinsville, New Jersey) and used to maintain constant pressure throughout the duration of the experiment. Pressure was measured using an Omegadyne PX209 pressure transducer (Omega Engineering Inc, Norwalk, Connecticut). Due to both the high force output of these muscles and new end fitting design, precaution was taken at higher pressures to ensure no tearing or separation occurred in the braid or end fittings. No damage was observed during the course of these tests.

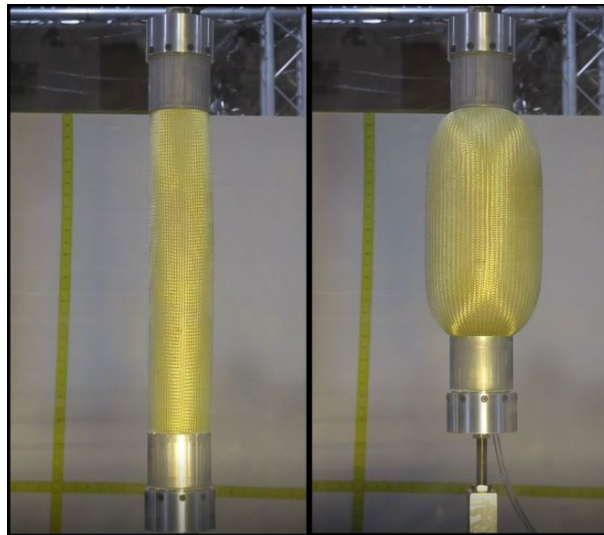


Figure 2.3: Left: PAM at block force at 100 psi. Right: PAM at free contraction at 100 psi.

2.5 Radial Expansion Testing

To measure the radial force exerted by the muscle, the MTS servo-hydraulic testing machine was once again used. Two testing fixtures were custom designed to fit into the MTS to allow the MTS to measure the radial force, shown in figure 2.4. The testing fixtures consisted of

two aluminum half rounds each, with one set measuring 3 inches in inner diameter with a 0.25 inch wall thickness, and the other measuring 3.5 inches in inner diameter with a 0.25 inch wall thickness. Mounting fixtures were 3D printed out of ABS using a Makerbot Method X (Makerbot, New York, New York) and epoxied onto the half rounds using Scotchweld DP460 (3M, St. Paul, Minnesota). The two half rounds were placed into the MTS and the lower head of the MTS was raised until the ends of the half rounds were touching, forming a completed pipe. The PAM was placed in the pipe and pressure was increased from 0 to 100 psi using the same pressure controller described in section 2.4. The MTS recorded the force and pressure during the test. Care was taken to ensure proper alignment of the pipe halves and to prevent any separation of the two halves during testing.

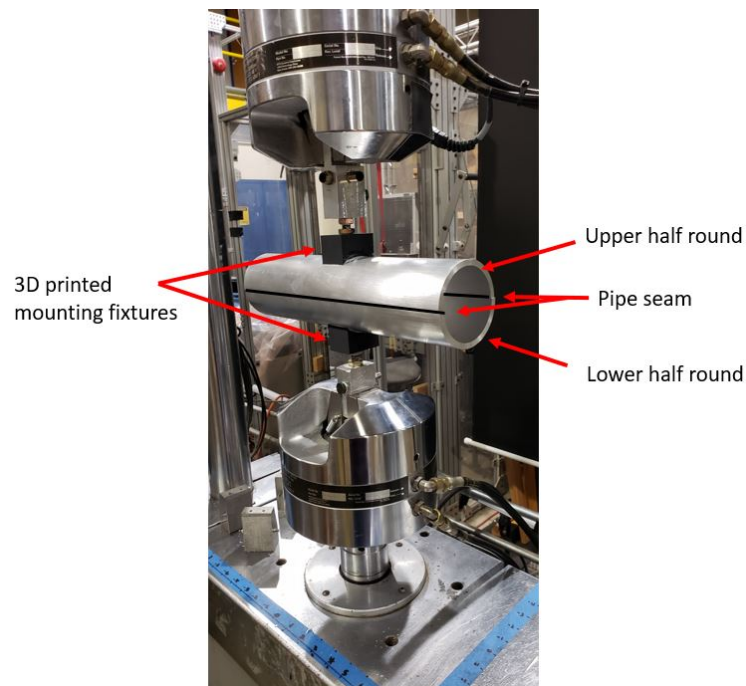


Figure 2.4: Test rig used for radial expansion testing.

2.6 Axial contraction analysis and results

With the contractile data collected, a model of the PAM could be generated. To generate this model, first the braid length, B , and the number of turns of the braid, N , are needed. These can be found using equations 2.1 and 2.2:

$$B = \frac{L_0}{\sin(\theta_0)} \quad (2.1)$$

$$N = \frac{B \cos(\theta_0)}{\pi D_0} \quad (2.2)$$

where L_0 is the initial active length, θ_0 is the initial braid angle, and D_0 is the initial diameter. From here we can find the PAM diameter, D , for any contraction length, as well as the instantaneous braid angle, using the following equations:

$$D(L) = \frac{\sqrt{B^2 - L^2}}{\pi N} \quad (2.3)$$

$$\frac{\cos(\theta_1)}{D_1} = \frac{\cos(\theta_2)}{D_2} \quad (2.4)$$

where L is the current length of the PAM. The force generated by the PAM can now be found using the following equation for the Gaylord force, $F_{Gaylord}$:

$$F_{Gaylord} = \frac{P}{4\pi N^2} (3L^2 - B^2) \quad (2.5)$$

where P is the pressure of the PAM. The Gaylord force is most accurate at lower contractions as

it does not account for the energy required to expand the bladder, and is therefore less accurate at higher contractions where bladder expansion is greater [74].

Following the procedures outlined in ref. [36], a characterization of the PAM that incorporates bladder expansion can be determined using the MTS data and the `fmincon` command in MATLAB to determine the non-linear moduli characteristics of the bladder [36]. These moduli are allowed to vary linearly with pressure and provide a 4th order approximation of the non-linear behavior of the bladder. The axial and circumferential strain, ϵ_z and ϵ_c , can be found using:

$$\epsilon_z = \frac{L}{L_0} - 1 \quad (2.6)$$

$$\epsilon_c = \frac{R}{R_0} - 1 \quad (2.7)$$

where R is the instantaneous radius of the bladder, and R_0 is the initial radius. The instantaneous axial and circumferential stress in the PAM can then be determined using the strains, and the moduli, E_k :

$$\sigma_z = \sum_{k=1}^4 E_k \epsilon_z^k \quad (2.8)$$

$$\sigma_c = \sum_{k=1}^4 E_k \epsilon_c^k \quad (2.9)$$

Finally, the instantaneous force produced by the PAM can be calculated using the force balance model:

$$F = PR_i \left(\frac{L^2}{2\pi R_o N^2} - \pi R_i \right) + \sigma_z \frac{V_B}{L} - \sigma_c \frac{tL^2}{2\pi R_o N^2} \quad (2.10)$$

where V_B is the volume of the bladder, which is assumed constant for this model, t is the thickness of the bladder, R_o is the outer radius of the PAM, and R_i is the inner radius. Note that due to the non-negligible thickness of the bladder, a distinction was made in the radii terms between the inner and outer radius of the bladder for this model, unlike the model used in ref. [36].

The axial contraction model is comprised of three terms. The first term, $PR_i(\frac{L^2}{2\pi R_o N^2} - \pi R_i)$, is equivalent to the Gaylord force from equation 2.5. The second and third terms, $\sigma_z \frac{V_B}{L} - \sigma_c \frac{tL^2}{2\pi R_o N^2}$ are the forces resulting from the axial and circumferential stress in the bladder, respectively. These forces act against the Gaylord force and reduce the total force output by the muscle. The results of the analysis are shown in figure 2.5. The dashed black lines represent the experimental data obtained from the test described in section 2.4, while the colored lines represent the results predicted by the model.

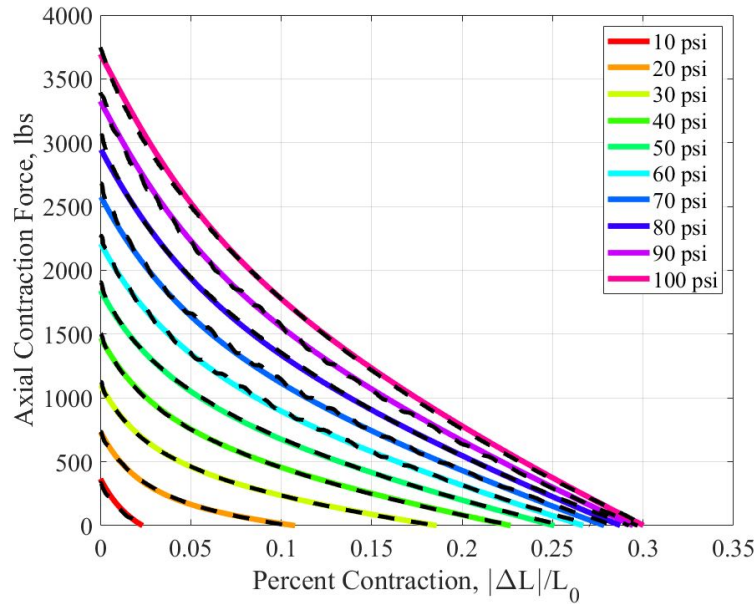


Figure 2.5: Force versus percent axial contraction of PAM.

2.7 Radial expansion analysis and results

Using the free body diagram depicted in figure 2.6, the radial expansion force can be modeled. First, balancing the forces in the circumferential and axial directions results in equations 2.11 and 2.12.

$$PR_iL = \sigma_c tL + NT \cos(\theta) + \frac{1}{2}F_R \quad (2.11)$$

$$F_{appl} + \pi R_i^2 P = \sigma_z \frac{V_B}{L} + T \sin(\theta) \quad (2.12)$$

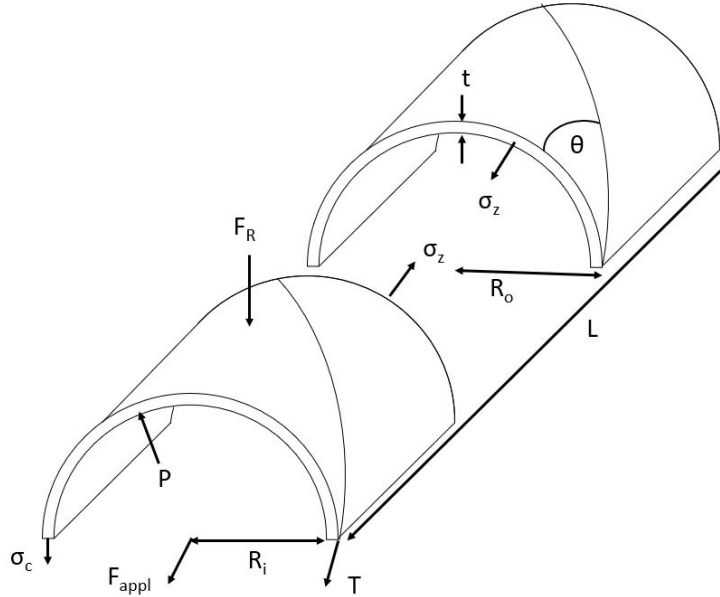


Figure 2.6: Free body diagram of PAM radially expanding in a pipe.

Where F_R is the radial force generated by the PAM, T is the tension in the braid, and F_{appl} is an applied axial load. By solving equation 2.11 for T , equation 2.1 for $\sin(\theta)$, and equation 2.2 for $\cos(\theta)$ and substituting, equation 2.12 becomes:

$$F_{appl} + \pi R_i^2 P = \sigma_z \frac{V_B}{L} + \frac{(PR_i L - \sigma_c t L - \frac{1}{2} F_R) L}{2\pi R_o N^2} \quad (2.13)$$

Equation 2.13 can then be solved for F_R :

$$F_R = 2 \left[P \left(R_i L - \frac{2\pi^2 R_i^2 R_o N^2}{L} \right) + 2\pi R_o N^2 \sigma_z \frac{V_B}{L^2} - \sigma_c t L - \frac{2\pi R_o N^2 F_{appl}}{L} \right] \quad (2.14)$$

Equation 2.14 is the radial expansion model, which describes the force applied radially by a PAM to a tube it is contained in for a given pressure and applied axial force. Like the axial contraction force equation, it is beneficial to break down this equation into its constituent parts. The first part, $P(R_i L - \frac{2\pi^2 R_i^2 R_o N^2}{L})$, describes the force generated by the internal pressure acting on both the walls of the bladder as well as the faces of the end fitting. The second and third terms, $2\pi R_o N \sigma_z \frac{V_B}{L^2} - \sigma_c t L$, are the forces due to the axial and circumferential stress in the bladder. Much like their function in the axial contraction model, they work to reduce the force produced by the first term. The final section, $\frac{2\pi R_o N^2 F_{appl}}{L}$, is the external axial force applied to the PAM. Due to the Kevlar braid, a coupling exists between the circumferential and axial functions of the PAM. This results in a reduction of radial force production as axial loading is applied to the muscle. An alternative formulation to equation 2.14 would be to express the radial expansion in terms of force per length, as in equation 2.15:

$$\frac{F_R}{L} = 2 \left[P \left(R_i - \frac{2\pi^2 R_i^2 R_o N^2}{L^2} \right) + 2\pi R_o N^2 \sigma_z \frac{V_B}{L^3} - \sigma_c t - \frac{2\pi R_o N^2 F_{appl}}{L^2} \right] \quad (2.15)$$

This formulation is convenient for comparison of PAMs of equal diameter but different length. By normalizing over the length of the muscle, the PAMs can be compared by their output of radial expansion force per length, allowing for a consistent and direct form of comparison. Additionally, by dividing equation 2.15 by the circumference of the anchored PAM, the pressure applied to the pipe by the PAM can be found, which is useful when anchoring in materials where applying too much pressure could disrupt the tunnel, such as tunnels made of gravel or sand.

The radial expansion model was compared to the data obtained from the radial MTS tests. The moduli obtained from the axial model were used for the radial model initially. Although the model correctly predicted when anchoring would occur, the model was not accurate when using the moduli from the axial expansion model above anchoring pressure, as seen in figure 2.7. Here, the the error for the 3 inch diameter case was 83.97 lbs or 5.47% at 100 psi, and -51.92 lbs or -4.27% at 100 psi for the 3.5 inch diameter case.

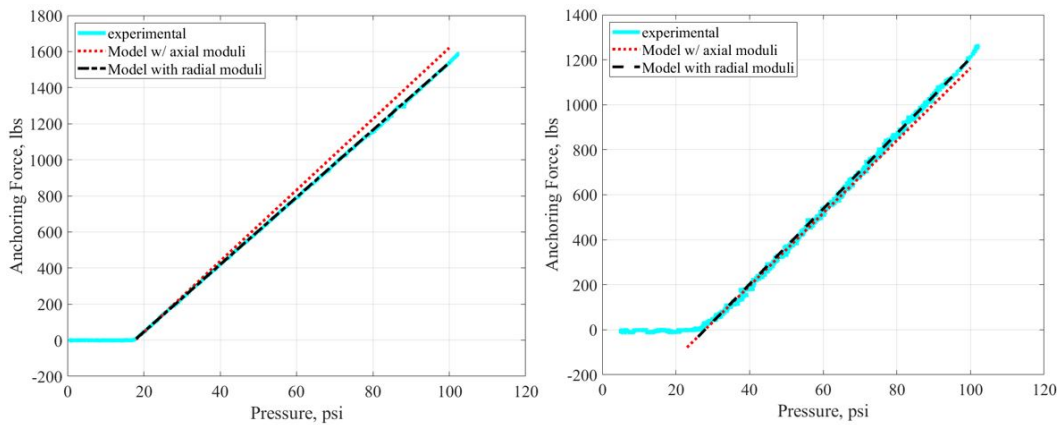


Figure 2.7: Radial expansion force model. Left: Analysis of 3” diameter case. Right: Analysis of 3.5” diameter case.

Optimization was performed on the model using the radial expansion data to find new moduli.

$$\begin{aligned}
\min_x \quad & \sum_{k=1}^n \sqrt{\frac{\sum_{i=1}^j (F_{M,i,k} - F_{E,i,k})^2}{j}} \\
\text{s.t.} \quad & lb \leq x \leq ub \\
& c(x) \leq 0
\end{aligned} \tag{2.16}$$

Here, F_M is the predicted force from the model, F_E is the measured experimental force, n is the number of pressures tested, and j is the total number of data points per pressure taken. The optimization variable, x , is a vector comprised of $[x_1, x_2, x_3, x_4, x_5, x_6, x_7, x_8]^T$. These variables are used to determine the moduli in the form of

$$\begin{aligned}
E_1 &= x_1 P + x_2 \\
E_2 &= x_3 P + x_4 \\
E_3 &= x_5 P + x_6 \\
E_4 &= x_7 P + x_8
\end{aligned} \tag{2.17}$$

The bounds on x are given by $\{x_k = -1000 \mid k = 1, \dots, 8\}$ and $\{x_k = 1000 \mid k = 1, \dots, 8\}$ as the lower and upper bound, respectively. Finally, the non-linear constraint, $c(x)$, can be defined as

$$c(x) = -(dE_c + dE_z) \tag{2.18}$$

where dE_c is the incremental internal circumferential energy and dE_z is the incremental internal axial energy. This constraint is derived from the Drucker stability criteria, which states that the

incremental internal energy of a stable material can only increase [75]. The Matlab function `fmincon` was utilized to solve this optimization problem.

The stress in the bladder was calculated using these new moduli and compared to the stress seen in the axial contraction model for both the 3 and 3.5 inch data sets. The stress was found to be the same at the pressure where contact with the pipe first occurred, called the contact or anchoring pressure, for both the axial and circumferential stress between the two models for both data sets. As the pressure increased, both the circumferential and axial stress in the radial expansion model changed at different rates than the stresses in the axial contraction model, as seen in figure 2.8. There are multiple reasons that could be the cause of the bladder stress not aligning between the two models, such as bladder stiffening due to contact with the pipe, inaccuracies due to an inextensible braid assumption, frictional losses due to braid-bladder contact and braid-pipe contact, or the constant radius assumption that may not hold for the axial contraction model, but which would hold for the radial expansion model, due to the pipe forcing a constant bladder radius onto the PAM [50]. These reasons could also explain why the moduli found for the two sets of radial data differ, as shown in figure 2.9.

The radial expansion model was ran once again using these new stress values, shown as the dashed black lines in figure 2.7. The model follows the experimental data very well, with the error for the 3 inch diameter case being 1.57 lbs or 0.10% at 100 psi, and -8.33 lbs or -0.69% at 100 psi for the 3.5 inch diameter case. As expected, the radial force increases linearly as the pressure increases.

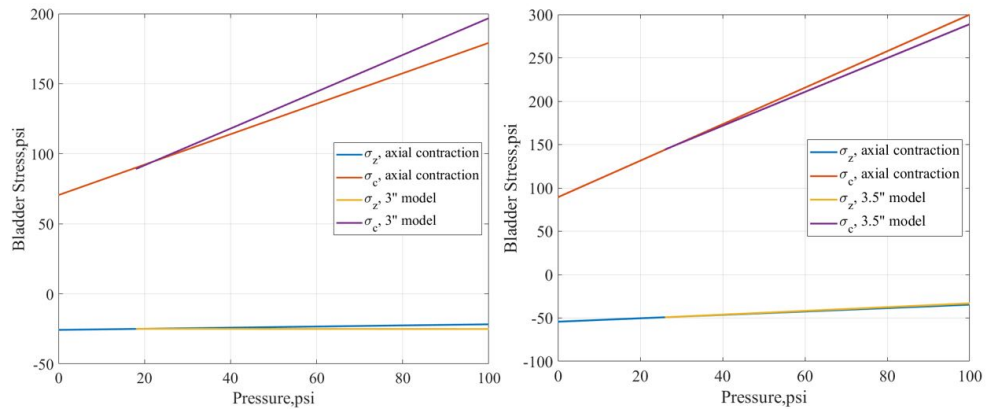


Figure 2.8: Axial and radial bladder stress in PAM. Left: Analysis of 3'' diameter case. Right: Analysis of 3.5'' diameter case

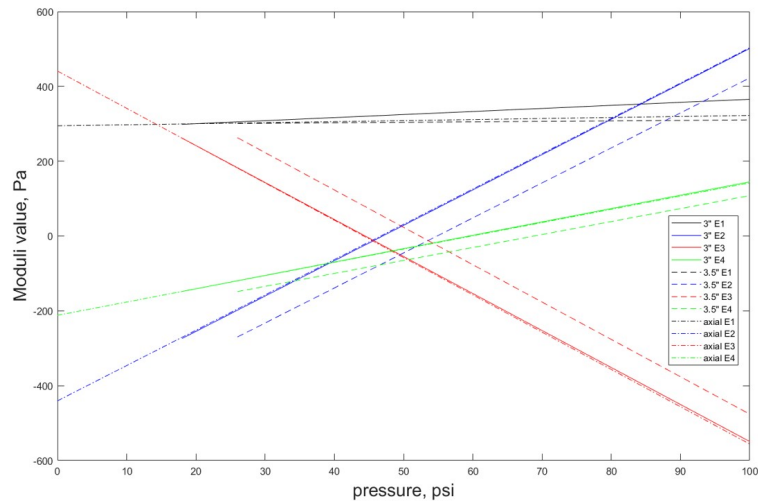


Figure 2.9: Change in calculated bladder moduli with pressure.

2.7.1 Radial force with applied axial load

While we have validated the radial force model against data, this data was collected under the condition that no axial force was applied to the PAM during the test. As noted earlier, the final term in equation 2.14 represents how an applied axial force would effect the radial force, and up to now, this term has not been validated. an additional test was performed for each test fixture, where a rope was attached to one end of the PAM. The rope was ran through a pulley, and a crate containing 230.37 lbs of weight was hung from the opposing end, as shown in figure 2.10. The PAM was once again inflated from 0 to 100 psi, and the produced radial force was recorded. A new PAM was used for these tests. The PAM was 2.25 inches in diameter with a length of 12.375 inches and a braid angle of 65.92 degrees.

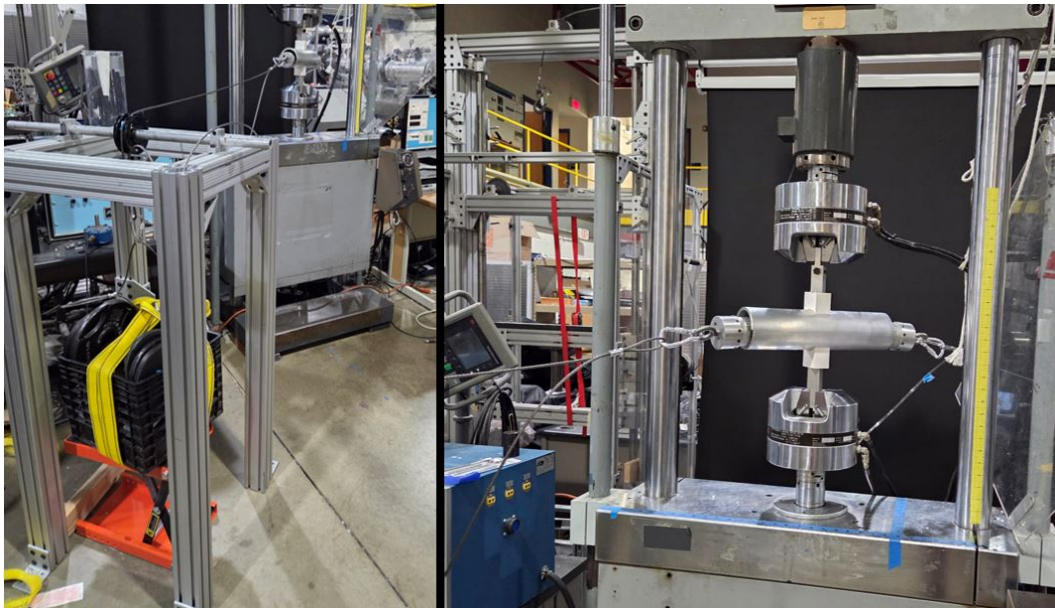


Figure 2.10: Left: Weight and pulley system. Right: PAM with attached rope.

Figure 2.11 shows the results of the data from the 3 inch and 3.5 inch diameter case compared to the data for each respective case with no axial loading applied.

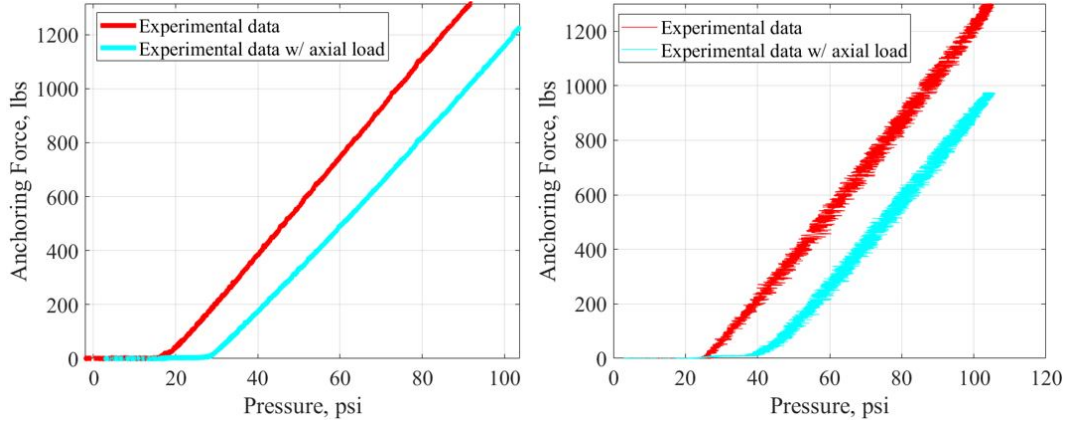


Figure 2.11: Radial force data with and without axial weight for 3 (left) inch and 3.5 (right) inch diameter pipe.

In both cases, we observe that the anchoring pressure of the PAM has shifted to the right, but the slope of the curve remains relatively the same between the case with the applied axial force and the case without. This aligns with equation 2.14, where the applied axial term, $\frac{4\pi R_0 N^2 F_{appl}}{L}$ acts to reduce F_R as F_{appl} increases, creating a shift to the right on the x axis. $\frac{4\pi R_0 N^2}{L}$ can be viewed as an amplification factor. This term increases or decreases the applied force depending on the diameter of the pipe and length of the PAM, causing the effective force reduction to amplify as the pipe diameter increases. Figure 2.12 shows the change in amplification factor as pipe diameter increases. For our load of 230.37 lbs, we see a reduction of 233.72 lbs for the 3 inch diameter case, and 296.07 lbs for the 3.5 inch diameter case.

Figure 2.13 shows the fitted model against the raw data for both cases. Here, the model was fitted to the data without the applied axial load using the optimization function described previously. The term for the applied axial load was then added to the model, without running the optimization function again, and overlaid overtop the data with the applied axial load. The model accurately accounts for the shift that occurs with the application of an axial load, with no need to obtain new moduli through optimization. Note that some discrepancy is seen in the slope of the

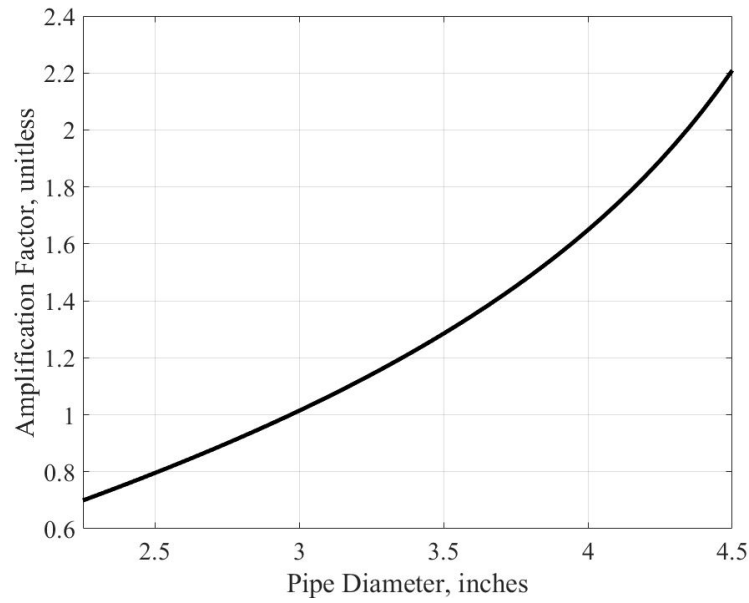


Figure 2.12: Amplification factor versus pipe diameter.

line. This phenomenon may occur due to certain assumptions in the model that are not accounted for, such as friction and hysteresis.

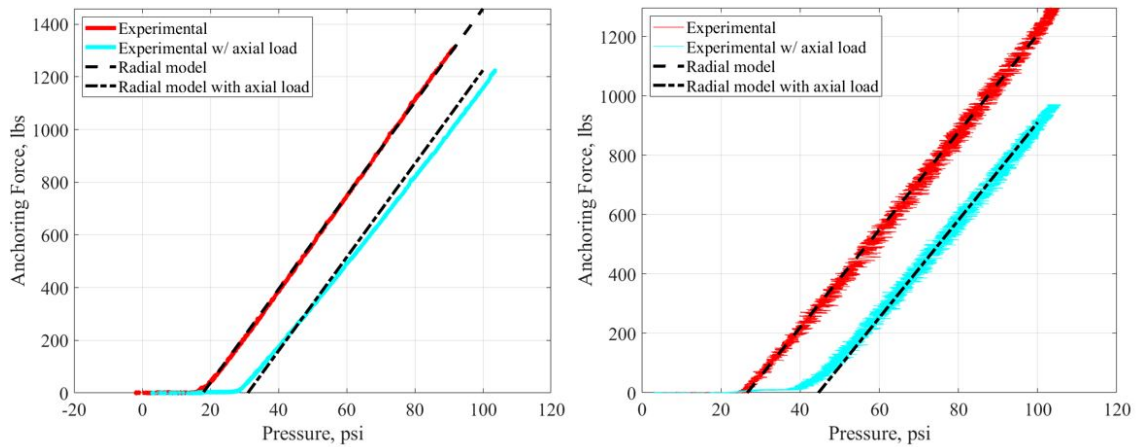


Figure 2.13: Radial model applied to data with and without axial loading.

2.8 Model Compatibility

As shown in section 2.7, the stresses present in the bladder differ in the muscle when the muscle is constrained in a pipe versus when it is not. Despite this, both the axial and

circumferential stresses in the bladder at the time of anchoring are equal between the two models. This allows for compatibility between the axial contraction and radial expansion model. While the muscle is free to expand radially, the axial contraction model accurately describes the muscle behavior. Once contact with the pipe wall is established, the radial model and moduli should be used while the PAM is anchored. The pressure where the switch between models should occur is the same as the contact pressure. This pressure can be found by solving equation 4.4 for the length, using the desired inner diameter of the pipe. This length can then be compared to a pressure versus free contraction plot that is obtained from the axial contraction data, as in figure 2.14. For an inner pipe diameter of 3 inches, pipe contact occurs at a free contraction length of 11.29 inches for the PAM used in the radial testing without axial loading, which corresponds to a pressure of $P_a(3'')=17$ psi. Pipe contact occurs at a free contraction length of 10.45 inches for a pipe diameter of 3.5 inches, which corresponds to a pressure of $P_a(3.5'')=24$ psi, as shown in figure 2.14, which agrees with the experimental data.

2.9 Robotic Applications

With a functioning radial model developed, a soft robot capable of utilizing the radial expansion force of a PAM was developed. When combined axially, three PAMs can be linked to form a single worm-like robot capable of movement through tunnels and pipes, such as in figure 2.15, where a three segment PAM robot starts at the bottom of a vertical pipe and ascends. Using a system of internal pneumatic tubing, each PAM can be individually pressurized and monitored, as shown in figure 2.16.

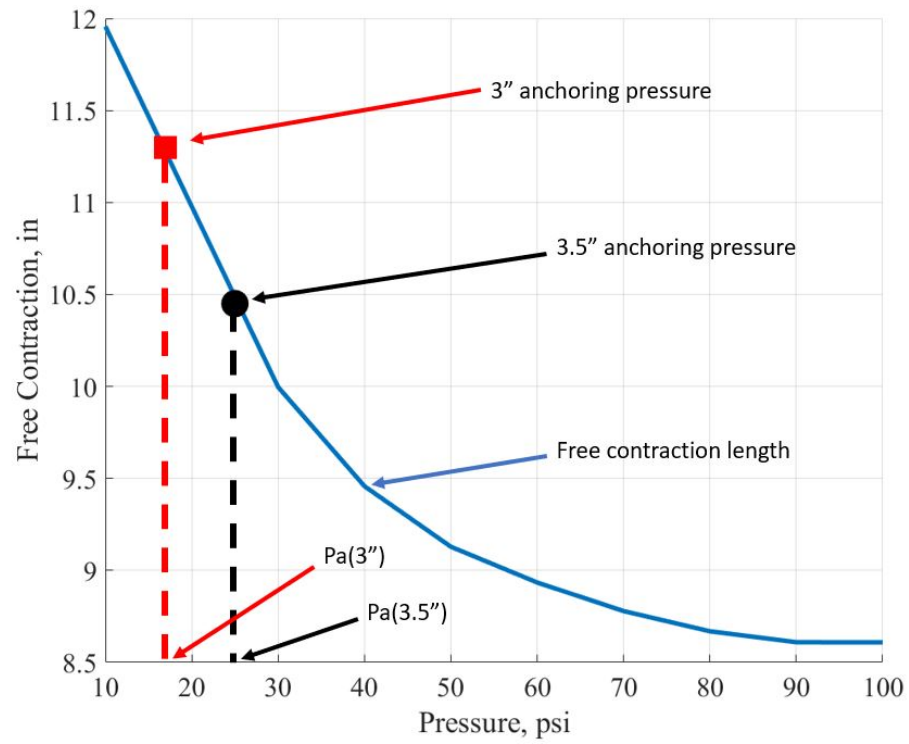


Figure 2.14: PAM pressurization vs free contraction length.

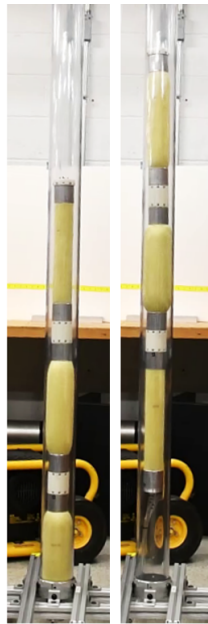


Figure 2.15: 3-segment PAM robot climbing a vertical pipe.

Through the use of this electro-pneumatic system, the robot can achieve locomotion through

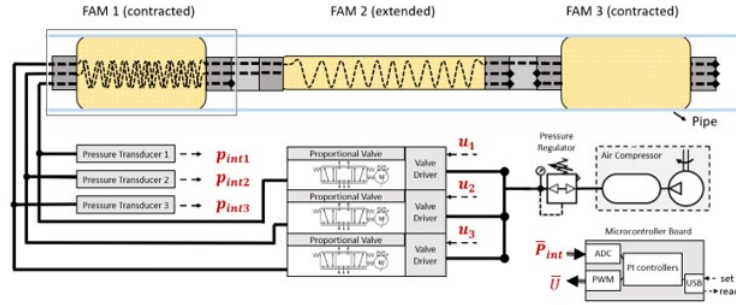


Figure 2.16: Electro-pneumatic diagram of three-PAM worm-like robot.

pipes using a pattern of inflation and deflation akin to that of a peristaltic wave used by worms when tunneling [76], shown in figure 2.17. In this example, the middle PAM is anchored in step one, and acts as a fixed point for the robot. In step two, the rear PAM contracts and anchors as well. With the rear PAM anchored, the middle PAM is allowed to deflate and elongate in step three. The peristaltic wave continues to propagate, with the forward PAM anchoring in step four, allowing the rear PAM to deflate in step five. Finally, the middle PAM inflates in step six, and the forward PAM deflates in step seven. The robot is now in the same state as step one, but has propagated forward from its initial position. For this locomotion method to work while climbing an inclined pipe, the force provided by the anchoring muscle must be able to counteract the weight of the robot and any additional payload so slipping does not occur. The ability to predict the anchoring force provided by the robot given the pressure supplied to the PAM through the use of the radial expansion model is crucial to ensure sufficient anchoring forces so loss of anchoring does not occur during operation.

Additionally, an alternate version of the worm-like robot called the Underminer was constructed by GE which used seven PAMs, manufactured by the CORE laboratory at the University of Maryland, shown in figure 2.18. This robot was used to burrow through soil, using the nose cone to create the tunnel, and then utilized the expansion properties of the PAMs to expand the soil

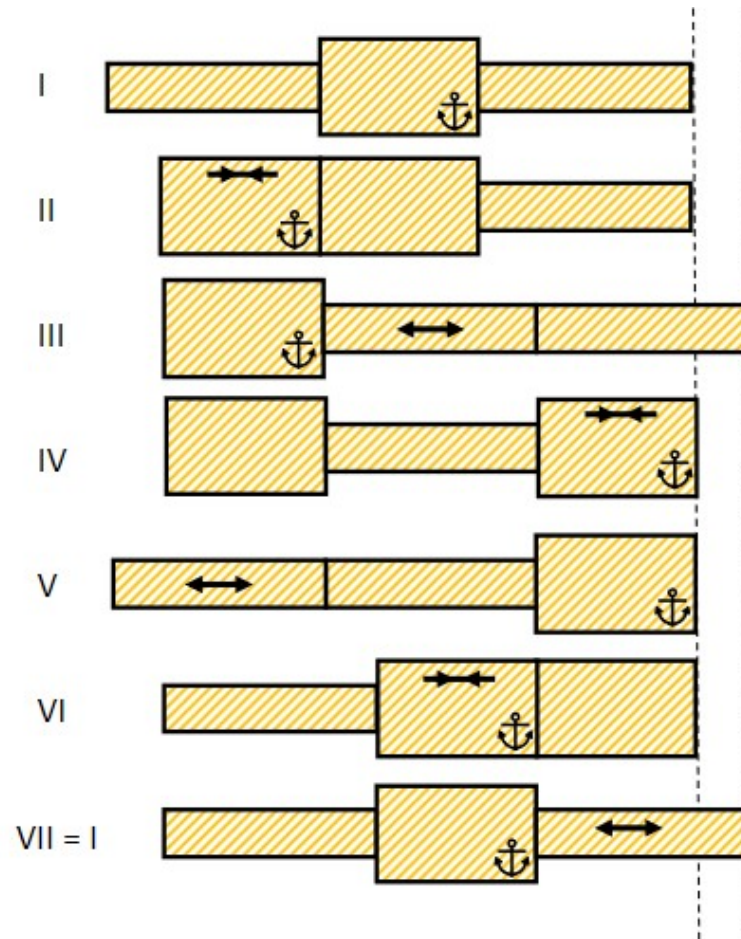


Figure 2.17: Peristaltic wave gait of a three-PAM worm-like robot

cavity to full size. In doing so, the worm-like robot is capable of traversing through these soil tunnels without generating any spoils that would be present in traditional digging robots [46].

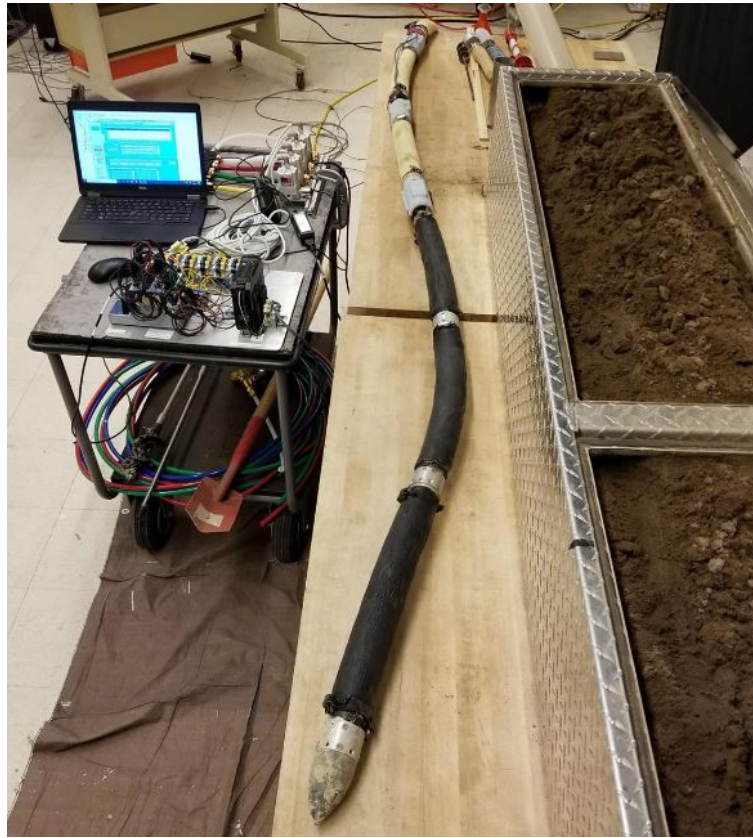


Figure 2.18: Seven section worm-like robot created by GE.

2.10 Conclusion

Analysis for the radial expansion force produced by a PAM is outlined in this study. This force, although utilized in robotic applications, has not been fully understood or properly modeled before, but understanding of the mechanics producing the force is necessary to ensure sufficient anchoring to support both the robot and any additional payload. The model not only accurately fit the data obtained from experimental testing both with and without applied axial loading, but also aligned nicely with the axial contraction model, both in prediction of anchoring pressure and

in calculated axial and circumferential bladder stress at the contact pressure. Future work for this model includes performing slip tests of the PAM anchored in the pipe. These tests would entail anchoring the PAM in a pipe, and increasing the axial load on the PAM until slip occurs. These tests would provide insight on how both friction and applied axial load affect slip, to achieve a better understanding on how much payload a worm-like robot can carry before slipping occurs. Additional future work entails determining the cause for the misalignment of bladder stress between the axial contraction and radial expansion models, as well as the difference in moduli between the 3 inch and 3.5 inch data sets. Determining and resolving the cause could allow for a radial expansion model with moduli that work for any given pipe diameter, as opposed to needing a set of moduli for each pipe diameter of interest, as is currently the case.

The ability to accurately predict the anchoring force generated by PAMs allows for the possibility to accurately assess the needed level of pressurization to maintain anchoring in robotic applications where factors like pipe diameter or applied payload may vary along the course of the mission, allowing for a much broader scope and application of these types of robots.

Chapter 3: Analysis of Torsional Response in Pneumatic Artificial Muscles

3.1 Overview

Pneumatic artificial muscles (PAMs) consist of an elastomeric bladder wrapped in a helical braid. When inflated, PAMs expand radially and contract axially, producing large axial forces. PAMs are advantageous because of their high specific work and specific power, as well as their ability to produce large axial displacements. The axial and radial behavior of PAMs have been well studied. The torsional response of PAMs have not been explored before. Accurate prediction of the torsional force was desired for use in a bio-inspired, worm-like robot capable of using an auger mounted to a PAM to bore out tunnels. Thus, an understanding of torsional response was a key objective. Modeling of the torsional response was performed using a force balance approach, and multiple model variations were considered, such as St. Venant's torsion, bladder buckling, and asymmetrical braid loading. Torsional testing was performed to validate the model using a custom torsional testing system. Data from the tests was compared to the predicted torsional response.

3.2 Introduction

Pneumatic artificial muscles (PAMs) are actuators comprised of an elastomeric bladder, typically latex or gum rubber, wrapped in a bi-axial braided sleeve. When inflated using either pneumatic or hydraulic devices, such as electrohydrodynamic pumps [71,77], PAMs, also known as McKibben muscles, contract axially while simultaneously expanding radially. These motions also generate forces in the axial and radial directions, respectively. The axial force production of a PAM is well studied and has been utilized in various applications [3,4,48,64,67–72]. PAMs have been found to have high specific force, specific power, and power density, due to their high force output and relatively low actuator mass, as well as low operating pressures [5,52,53,63,65,66]. Although not as extensively utilized as the axial contraction force, the radial expansion force has also seen application in worm-like bio-inspired robots [78]. The radial expansion force of a PAM enables them to anchor into pipe and tunnel walls for robotic locomotion.

While models exist to characterize McKibben muscle response to external axial and radial loading, no such model exists to characterize the response to torsional loading. Torsional loading of other types of pneumatic actuators have been studied, such as elastomeric actuators capable of generating twist under vacuum [38,39], asymmetric pneumatic balloon actuators [79], helical tubes embedded in silicone that impart torsion upon inflating [80], pre-twisted pneumatic tubes generating torsion upon inflation [81], and auxetic structured cylinders which twist under buckling during pressurization [82]. Additionally, actuators similar to McKibben muscles which generate torsion during inflation due to only having a single braid direction have been studied, but no model has been created and validated to analyze their torsional properties [83,84]. Torsional structures have also been constructed using multiple McKibben muscles, such as ones meant to

replicate the twisting motion of the left ventricle in a heart, but these structures do not entail torsion of the muscle itself, but rather induce torsion in the structure through the application of the axial contraction force [85]. As a continuation of the work performed in reference [78], the ability to predict PAM torsional response was desired to predict how it would react to loads from an auger drilling through dirt or sand, as shown in figure 3.1. The PAM would be required to provide significant torsional resistance to allow the auger enough torque to drill out the tunnel. This study develops an analytical model to predict the torsional response of a PAM, and validates the model experimentally.



Figure 3.1: Example of bio-inspired, three PAM tunneling robot.

3.3 Torsional Response Discussion

Initial insights into the torsional response of the system reveals multiple possibilities for how a PAM will act under torsion. The first, and simplest, possibility is that the PAM will act as a hollow cylinder and will follow St. Venant torsion. Another possibility is that the soft bladder of the PAM will undergo torsional buckling prematurely, effecting the torsional stiffness of the PAM. The final possibility is that the biaxial braided sleeve will disproportionately carry the torsional load, resulting in nonlinearities in the torsional response.

To fully discuss the third possibility, first consider the triangle relationship present in PAMs. The length of the braid, denoted as B , of a PAM can be found using the diagram shown in figure 3.2. This diagram denotes that a right triangle can be formed with B on the hypotenuse,

L , the active length of the PAM, on one leg, and πND on the opposing leg. Here, the πND term represents the number of times the braid loops around the diameter of the PAM, with N representing the number of loops, and D representing the diameter of the PAM. The angle, θ , denoted in the triangle represents the angle of the braid with respect to the a line perpendicular to the long axis of the PAM.

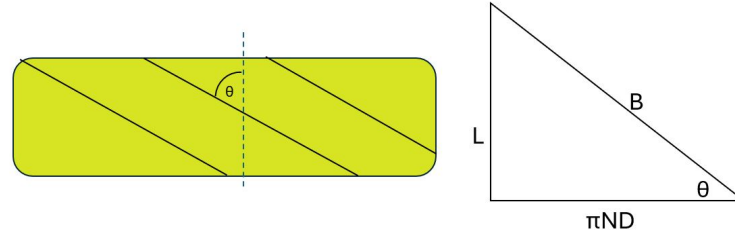


Figure 3.2: Triangle relationship of a PAM.

Now consider figure 3.3, representing a PAM with two braids, winding in opposite directions around the PAM. In the top section of the figure the PAM is untwisted and the braid angles, as well as the number of turns of the braids around the PAM, are equal. In the bottom section of the figure, the PAM is twisted. The braid angle of both braids change due to the twist. The braid angle of braid one decreases when measured off of the line perpendicular to the long axis of the PAM, while the braid angle of braid two increases. In addition, the number of turns of the braid increases for braid one and decreases for braid two.

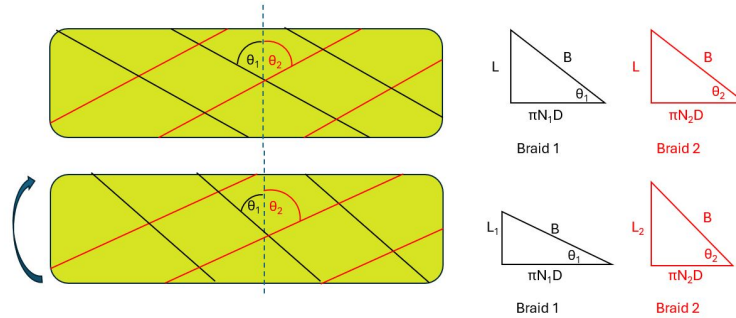


Figure 3.3: Effects of twisting on the triangle relationship of a PAM.

Based on the triangle relationship shown in figure 3.3, and assuming the braid length, B , of the two braids are identical, key insights can be ascertained. First, assuming the PAM is at free contraction during torsion, which is the state of maximum axial contraction, when N , the number of turns of the braid, is increased in braid one, the πND term will increase. To keep the triangle relationship true, both the braid angle, θ , and the free contraction length, L , must decrease. In contrast, for braid two the πND term will decrease, forcing θ and L to increase. Because braid one prevents L from increasing, we would expect braid two to go slack and carry no load. If this is the case, a non-linear response is expected to be observed, due to the transition of torsional loading from both braids to one braid, as well as the change in the number of braid turns around the PAM, the change in free contraction length, and the change in braid angle as the twist angle changes.

3.4 Testing

The PAM utilized in this experiment was 11.94 inches long and 2.25 inches in diameter. To determine the torsional response, the PAM was first tested axially to gather information on the blocked force as well as the free contraction lengths of the PAM. Blocked force is the state where the PAM produces maximum force with no axial contraction and free contraction is the state where the PAM undergoes maximum contraction with no force production. The PAM was placed into an MTS servohydraulic testing machine. Once the PAM was mounted into the fixture, an Arduino Mega (Arduino, Robbinsville, New Jersey) controlled an Enfield D1 Bi-Directional Proportional PWM Valve Driver (Enfield, Trumbull, Connecticut) to pressurize and maintain the PAM at a set pressure value using an Enfield LS-V05s High Speed 5/3 Proportional Directional

Valve (Enfield, Trumbull, Connecticut). The pressure was read using an Omegadyne PX209 pressure transducer (Omega Engineering Inc, Norwalk, Connecticut). The PAM was then quasi-statically actuated between free contraction and blocked force state three times per pressure, shown in figure 3.4. The PAM was tested from ten to fifty psi in increments of five psi. Before each test, the PAM was manually cycled from blocked force to free contraction to account for the Mullin's effect [73]. Force, displacement, and pressure data was collected for the duration of each test.

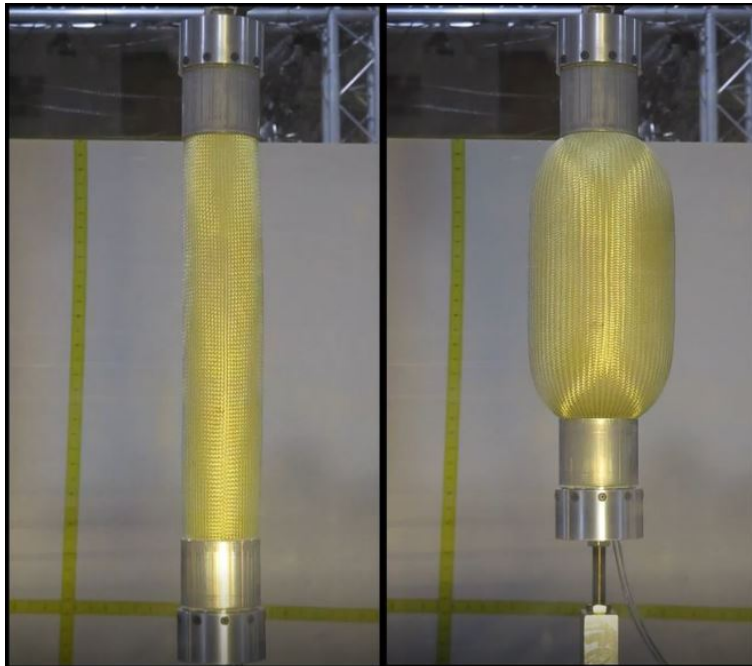


Figure 3.4: Actuation of PAM at block force (left) and free contraction (right).

Torsional stiffness data was collected using a custom testing fixture, designed and constructed from mostly commercial components. The fixture is shown in figure 3.5 showcasing electronics and mechanical components. The fixture provides a stable base to consistently twist a pressurized PAM, recording torque applied to the PAM and the degrees twisted. The base of the test fixture is made of commercially available 80/20 segments. Using the grooves on these segments, the

actuators and sensors are aligned along the axial direction of the PAM to prevent any external moments from affecting the data. Additionally, the modular nature of 80/20 allows for moving hardware to be mounted.

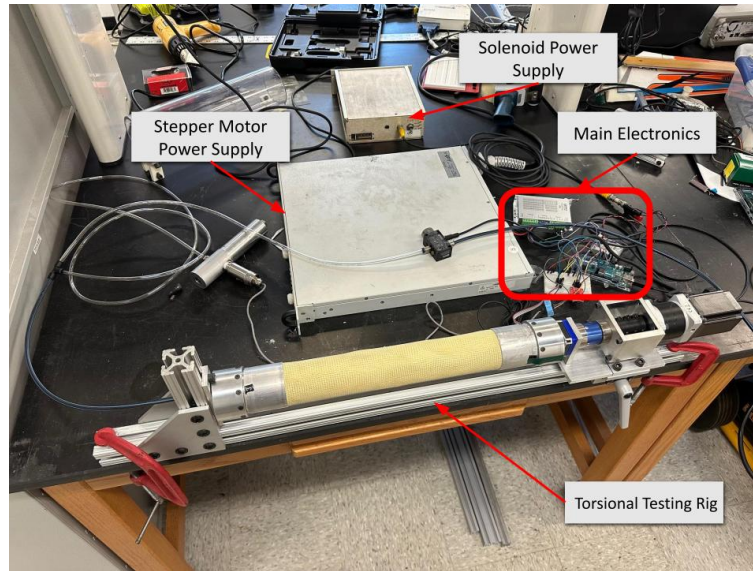


Figure 3.5: Overview of the systems in the torsion testing fixture.

The same Arduino and pressurization system from the axial tests were used in the torsion fixture. With one end of the PAM fixed, the other end was free to translate axially and to rotate. As the PAM pressurized, it underwent free contraction and translated along the length of the fixture. When the PAM reached its set pressure, the same end that was translated was then rotated quasi-statically at a rate of 0.02 degrees per second using a STEPPERONLINE 23HS30-2904S stepper motor (Amazon, Seattle, Washington) with a 50:1 planetary gearbox. The fixture was equipped with an ATO Micro Reaction Torque Sensor (ATO, Diamond Bar, California) to measure the torque and a CUI Devices AMT13A2S-V (Same Sky Devices, Rosewood, Oregon) incremental encoder with a minimum resolution of 4096 PPR to measure the angular displacement. Figure 3.6 shows how the motor systems and sensors are physically mounted while figure 3.7 shows the systems diagram of all the pneumatic, mechanical, and electrical components.

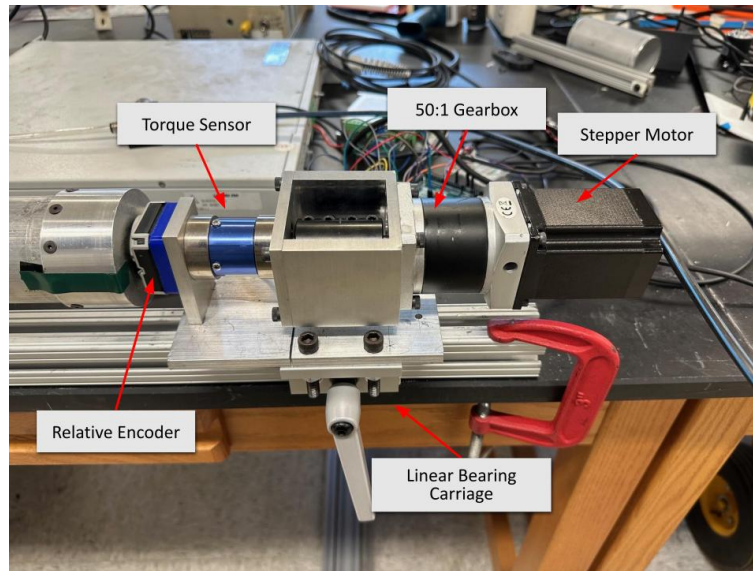


Figure 3.6: Mechanical actuation system and sensors employed during torsional testing.

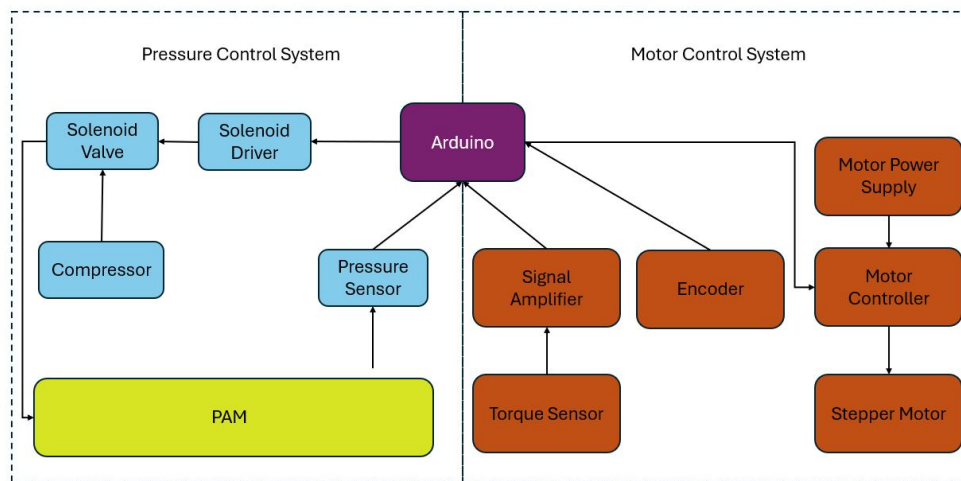


Figure 3.7: Box diagram for all electro-pneumatics on the torsion fixture.

The PAM was tested from ten to fifty psi in increments of five psi, as it was for the axial tests. Before testing the range of pressures, the fixture was calibrated to insure repeatability and accuracy between tests. The PAM was first pressurized, then the testing rig was rotated until a small torque was read. The small torque indicates the motor connection is properly primed and a good connection is established between the motor shaft and the PAM. For each pressure, the PAM was twisted from zero to five degrees or until reaching the gearbox torque limit of around 170 pound-force inch three times. The PAM is then untwisted at the same rate as the twisting rate.

In addition to these tests, a torsional test of just the uninflated bladder was performed to determine if torsional buckling of the bladder was observed in the 5 degree twist range.

A representative plot of the data collected is given in figure 3.8, demonstrating the data collected from the three 35 psi tests, as well as a linear fitted line. The R^2 value of the line is 0.999. This high level of linearity applies to all of the data collected. The R^2 value for each pressure is given in table 3.1, along with the peak torsional load, and maximum twist achievable.

Table 3.1: Torsion testing results.

Pressure [psi]	Maximum Twist [degrees]	Maximum Torque [lb-in]	R^2 [-]
0 (bladder only)	5.08	1.98	0.880
10	5.10	22.36	0.975
15	5.08	60.29	0.994
20	5.06	111.57	0.992
25	5.06	156.51	0.998
30	4.07	157.80	0.995
35	3.85	167.56	0.999
40	3.56	167.94	0.999
45	3.28	164.47	0.999
50	2.90	151.12	0.999

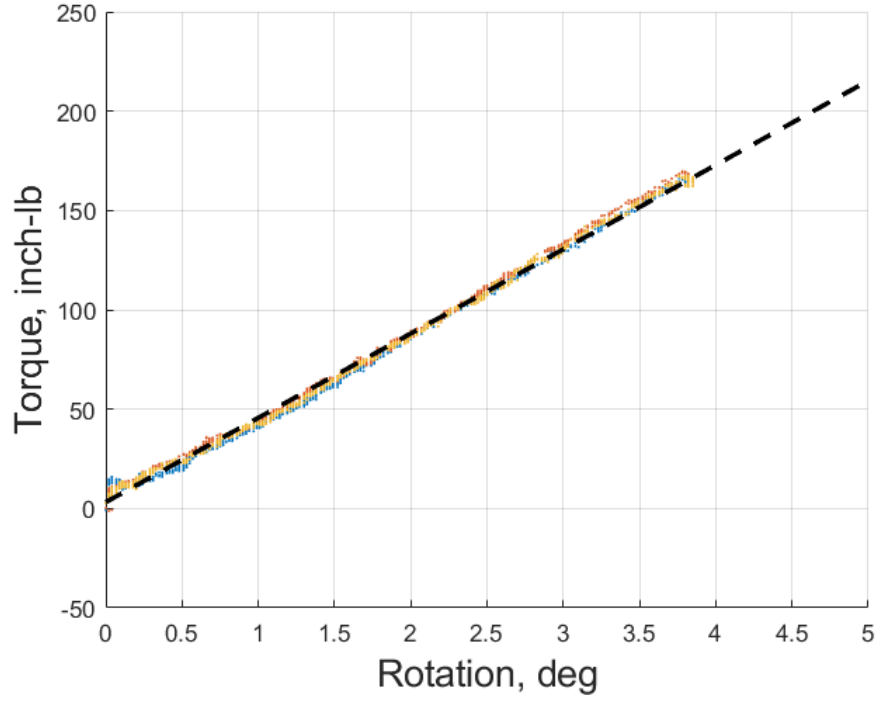


Figure 3.8: Raw torsional data (red, yellow, blue dots) with linear fitted line (black dashed line) for 35 psi torsional test.

3.5 Axial Analysis

Analysis of the axial properties of the PAM was performed to acquire information on the free contraction lengths of the PAM at each pressure, as well as the braid angle of the PAM, as both are needed for the torsional analysis. To analyze the axial force, the process outlined in reference [36] was followed. First, the braid length and number of turns of the braid were found using equations 3.1 and 3.2.

$$B = \frac{L_0}{\sin(\theta_0)} \quad (3.1)$$

$$N = \frac{L_0}{D_0 \pi \tan(\theta_0)} \quad (3.2)$$

Here, L_0 is the original PAM length, θ_0 is the original braid angle, and D_0 is the original diameter. The inflated diameter, which is also necessary to know for the torsional analysis, can be found from equation 3.3, while the braid angle during inflation can be found using equation 3.4.

$$D(L) = \frac{\sqrt{B^2 - L^2}}{\pi N} \quad (3.3)$$

$$\frac{\cos(\theta_1)}{D_1} = \frac{\cos(\theta_2)}{D_2} \quad (3.4)$$

To solve for the initial braid angle, an alternate form of the Gaylord force equation than that used in reference [36] is used, given in 3.5.

$$\theta = \tan^{-1} \left(\sqrt{\frac{\frac{F}{P\pi R^2} + 1}{2}} \right) \quad (3.5)$$

Solving the Gaylord equation using the blocked force of the PAM will give the initial braid angle.

For completeness, the entire axial actuation range of the PAM was plotted and fitted using the process described by reference [36]. The axial force can be described by equation 3.6.

$$F = PR_i \left(\frac{L^2}{2\pi R_o N^2} - \pi R_i \right) + \sigma_z \frac{V_B}{L} - \sigma_c \frac{tL^2}{2\pi R_o N^2} \quad (3.6)$$

Note that although the process to fit the model to the data is the same as that outlined in

the cited reference, the form of equation 3.6 differs slightly due to the use of an inner radius term for the PAM. In the cited reference, no differentiation was made between the inner and outer radius due to the thinness of the bladder of the PAM being analyzed. The PAM utilized in this paper has a quarter-inch thick bladder, which the authors believe too thick to utilize the same assumption. The fitted actuation curve is shown in figure 3.9. The dashed lines represent the averaged collected data per each pressure, while the color lines represent the fitted model.

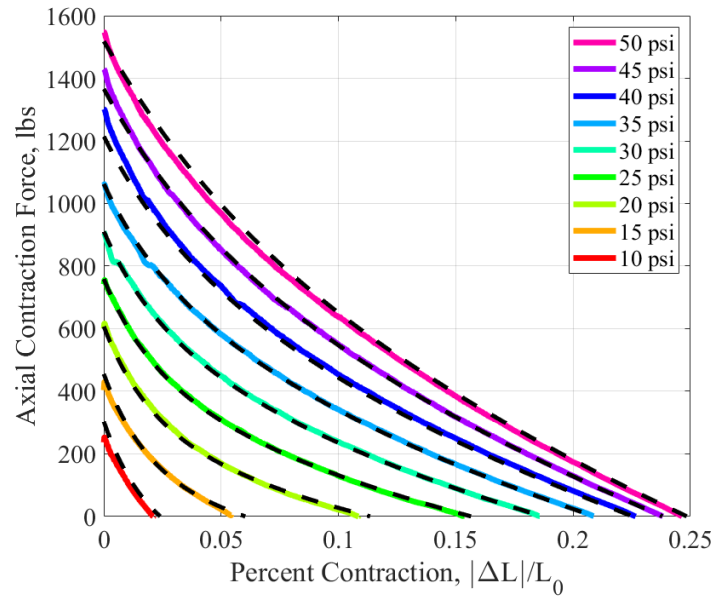


Figure 3.9: Axial actuation model (solid colored line) overlaid on axial data (dashed black line) of a PAM.

3.6 Torsional Analysis

To analyze the torsional properties of the PAM, a free body diagram, shown in figure 3.10, of the PAM was used to determine the circumferential, axial, and torsional loads on the PAM

during actuation. Circumferential forces present in the PAM can be represented by:

$$PR_iL = \sigma_c tL + N_1 F_1 \cos(\theta_1) - N_2 F_2 \cos(\theta_2) \quad (3.7)$$

where P is the pressure of the PAM, L is the length, σ_c is the circumferential stress in the bladder, R_i is the inner radius, F_1 and F_2 are the forces in braid 1 and 2 respectively, and N_1 and N_2 are the number of turns of braid 1 and 2, respectively. The axial equilibrium of the PAM is:

$$\pi R_i^2 P = \sigma_z \frac{V_B}{L} + F_1 \sin(\theta_1) + F_2 \sin(\theta_2) - F_{appl} \quad (3.8)$$

where σ_z is the axial stress in the bladder, V_B is the volume of the bladder, and F_{appl} is the applied axial force. Torques present in the PAM are given by:

$$T = F_1 R_o \cos(\theta_1) - F_2 R_o \cos(\theta_2) + \frac{\tau J}{R_o} \quad (3.9)$$

where T is the applied torque, R_o is the outer radius, τ is the torsional stress in the bladder, and J is the polar moment of inertia. To solve for T , First F_1 and F_2 must be found. by solving Equation 3.8 and 3.9 for F_2 , they can be equated and F_1 can be found to be:

$$F_1 = \frac{\frac{T}{R_o \cos(\theta_2)} + \frac{\pi R_i^2 P - \sigma_z \frac{V_B}{L} + F_{appl}}{\sin(\theta_2)} - \frac{\tau J}{R_o^2 \cos(\theta_2)}}{\frac{\cos(\theta_1)}{\cos(\theta_2)} + \frac{\sin(\theta_1)}{\sin(\theta_2)}} \quad (3.10)$$

The solution for F_1 can then be inserted into equation 3.9 to solve for F_2 .

$$F_2 = \left(\frac{1}{1 + \frac{\tan(\theta_1)}{\tan(\theta_2)}} - 1 \right) \left(\frac{T}{R_o \cos(\theta_2)} - \frac{\tau J}{R_o^2 \cos(\theta_2)} \right) + \frac{\pi R_i^2 P - \sigma_z \frac{V_B}{L} + F_{appl}}{\sin(\theta_2) \left(1 + \frac{\tan(\theta_1)}{\tan(\theta_2)} \right)} \quad (3.11)$$

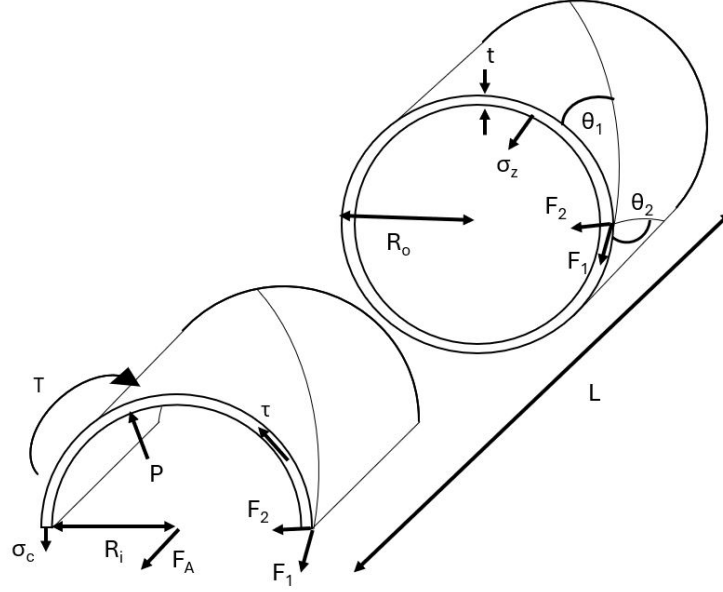


Figure 3.10: Free body diagram of PAM under torsional loads

By inserting equations 3.10 and 3.11 into equation 3.7, T can be found:

$$T = \frac{[(PR_i - \sigma_c t)L - (\pi R_i^2 P - \sigma_z \frac{V_B}{L} + F_{appl})(\frac{N_1 - N_2}{\tan(\theta_1) + \tan(\theta_2)})]R_o}{N_2 + \frac{N_1 - N_2}{1 + \frac{\tan(\theta_1)}{\tan(\theta_2)}}} + \frac{\tau J}{R_o} \quad (3.12)$$

At this stage, a determination must be made on which possibility presented in section 3.3 is correct, as the model will branch into different avenues in regards to each option. First, the possibility of the bladder buckling during torsion can be ruled out, as no buckling was observed during the bladder-only, zero psi testing. Since this would be the least stiff possibility and no buckling was observed, it can be assumed that buckling was not present in any of the tests. For the scenario with the changing braid angles between the two braid directions, there was no evidence recorded of this phenomenon occurring as well. No slacking in the braid was found, and no change in the free contraction length was observed during testing. In addition, the data recorded for all pressures were essentially linear, which would not occur if the braid angles of the

two braids were changing independently. There does remain the possibility that either of these phenomenon could occur at high values of twist, but they are not observed to be of significance for the range of twist tested here. By eliminating the first two possibilities, the only remaining possibility is St. Venant torsion. For this to be correct, two things must be true. First, we would expect the collected data to be linear. We have observed this to be true in the previous section. Second, the model should be able to reduce to the St. Venant torsion model if the braid angles and number of turns of the two braids are equated. After equating these variables, equation 3.12 reduces to:

$$T = \frac{(PR_i - \sigma_c t)LR_o}{N} + \frac{\tau J}{R_o} \quad (3.13)$$

It is known that PAMs do not produce any torsion when inflated, so under conditions when there is no externally applied torsion, equation (3.13) becomes:

$$0 = \frac{(PR_i - \sigma_c t)LR_o}{N} \quad (3.14)$$

The circumferential stress in the bladder can be explicitly solved for in this case:

$$\sigma_c = \frac{PR_i N}{t} \quad (3.15)$$

Inserting equation 3.15 into equation 3.13 further reduces the torsional model.

$$T = \frac{\tau J}{R_o} \quad (3.16)$$

Equation 3.16 is the St. Venant torsion equation. This demonstrates that despite the complex

material composition of a PAM, the torsional response of the PAM reduces to simply being the St. Venant torsion equation.

To solve for T , we must first determine τ , J , t , R_o , and R_i . R_o can be found by solving for B using equation 3.1, and then finding D_0 from equation 3.3 and dividing it in half. R_i can then be found by subtracting the bladder thickness from R_o . The bladder thickness can be found using equation (3.17).

$$t = R_o - \sqrt{R_o^2 - \frac{V_B}{\pi L}} \quad (3.17)$$

The polar moment of inertia can then be found using:

$$J = \frac{\pi}{2}(R_o^4 - R_i^4) \quad (3.18)$$

To determine τ , the bladder was modeled as a 4th order non-linear relationship between shear stress and strain in the form of:

$$\tau = \sum_{k=1}^4 G_k \gamma^k \quad (3.19)$$

Where γ is equivalent to:

$$\gamma = \frac{R_o \phi}{L} \quad (3.20)$$

Here, ϕ is the angle of twist, in radians, of the bladder. An optimization function was used to determine the four moduli using data obtained from testing in the form of equation (3.21).

$$\begin{aligned}
\min_x \quad & \sum_{k=1}^n \sqrt{\frac{\sum_{i=1}^j (T_{M,i,k} - T_{E,i,k})^2}{j}} \\
\text{s.t.} \quad & lb \leq x \leq ub \\
& c(x) \leq 0
\end{aligned} \tag{3.21}$$

where T_M is the torque on the PAM predicted from the model, and T_E is the torque obtained experimentally. j is the number of data points obtained per pressure, and n is the number of pressures tested. x is the variable being optimized and is of the form $x = [x_1, x_2, x_3, x_4, x_5, x_6, x_7, x_8]^T$. The components of x are used to determine the moduli of the material that is used to solve for the torsional stress, as shown in equation (3.22).

$$\begin{aligned}
G_1 &= x_1 P + x_2 \\
G_2 &= x_3 P + x_4 \\
G_3 &= x_5 P + x_6 \\
G_4 &= x_7 P + x_8
\end{aligned} \tag{3.22}$$

These moduli are allowed to vary linearly with pressure. The lower and upper bounds on x can be written as $lb = [lb_1, lb_2, lb_3, lb_4, lb_5, lb_6, lb_7, lb_8]^T$ where $\{lb_k = -1000 \mid k = 1, \dots, 8\}$ and $ub = [ub_1, ub_2, ub_3, ub_4, ub_5, ub_6, ub_7, ub_8]^T$ where $\{ub_k = 1000 \mid k = 1, \dots, 8\}$, respectively.

The non-linear constraint, $c(x)$, is derived from the Drucker stability criteria, which states that the incremental internal energy of the material must increase as strain increases for the material to be stable [75]. This constraint can be defined as

$$c(x) = -(dE_\tau) \quad (3.23)$$

where dE_τ is the incremental internal energy due to shear.

In addition to modeling the stress using a fourth order stress-strain relationship, as is commonly used in axial modeling [36], a version of the model using a first order relationship was also attempted, due to the strong linearity found in the data. For this model, only one moduli was found, given by equation 3.24. The stress is then given by equation 3.25.

$$G_1 = x_1 P + x_2 \quad (3.24)$$

$$\tau = G_1 \gamma \quad (3.25)$$

3.7 Results

The data from the nine pressures tested were fed into the optimization function, and the moduli were determined from the output. The comparison of the model to the data, as well as the error, can be seen in figure 3.11. Note that the error, determined by comparing the slope of the data to the slope of the model, for the 10 and 15 psi cases are large, but quickly drop to about 3.5% or less for 20 psi and above.

For the version of the model using the first order stress-strain relationship, similar results were found. The model results and error are given in figure 3.12. The error was again found to be large at low pressures, then drop dramatically at twenty psi. Comparing the fourth order and first

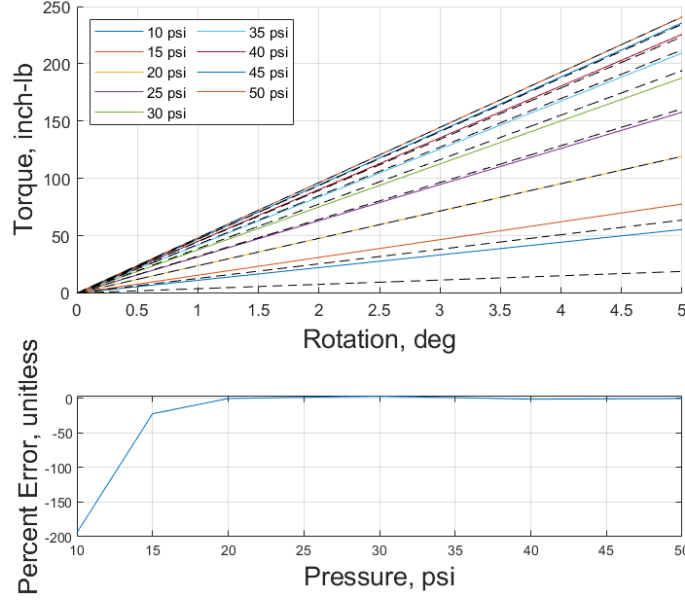


Figure 3.11: Top :Fourth order torsional model results (colored solid line) compared to collected data (black dashed line). Bottom: Fourth order torsional model error versus pressure.

order model error in table 3.2 reveals that the errors are nearly identical. Additionally, comparing the optimization variable outputs in table 3.3 shows that the x_1 and x_2 values are nearly identical between the first and fourth order model, and that the components of x used in the higher order moduli terms for the fourth order model are small, and contribute very little to the overall value of the modulus. These similarities signify that using a first order stress-strain relationship to model torsion of a PAM is sufficient for the range of pressures and twist tested in this paper, unlike the axial model, which required a higher order relationship for the model to properly fit the data.

Table 3.2: Comparison of error between the fourth order stress-strain model and the first order stress-strain model.

Test	10	15	20	25	30	35	40	45	50
	[%]	[%]	[%]	[%]	[%]	[%]	[%]	[%]	[%]
Fourth Order	-194.40	-22.01	0.01	1.85	3.24	1.31	-0.95	-0.53	0.00
First Order	-194.40	-22.02	0.00	1.85	3.24	1.31	-0.95	-0.53	0.00

Finally, a discussion of the large error seen in the low pressure regime is needed. If only the

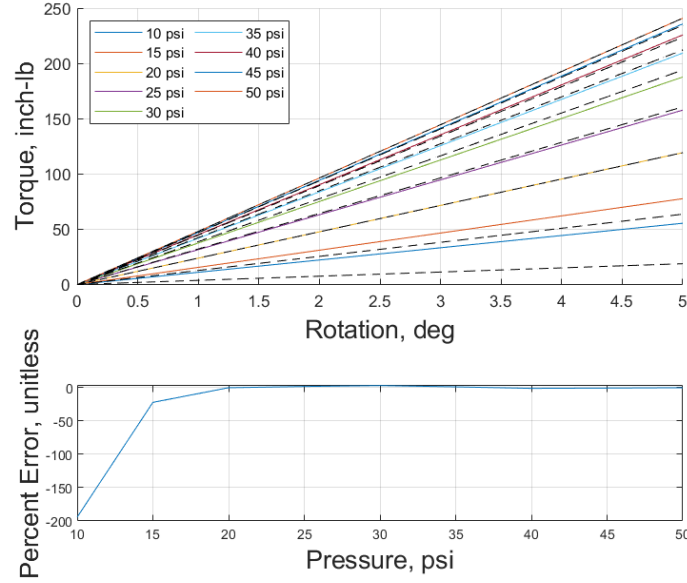


Figure 3.12: Top: Torsional model using first order stress-strain relationship (solid colored line) compared to data (black dashed line). Bottom: Torsional model error using first order stress-strain relationship versus pressure.

Table 3.3: Comparison of the optimization function output between the fourth order stress-strain model and first order stress-strain model.

Test	x_1 [-]	x_2 [psi]	x_3 [-]	x_4 [psi]	x_5 [-]	x_6 [psi]	x_7 [-]	x_8 [psi]
Fourth Order	-14.59	3.69×10^3	-1.03	46.87	-1.19	1.58	0.95	1.01
First Order	-14.60	3.69×10^3	-	-	-	-	-	-

10 and 15 psi data is entered into the optimization problem to determine the moduli, the model is found to fit the data well, as seen in figure 3.13. The reason for this appears to be an issue with the assumption of a linear change in moduli with pressure. This can be shown in how the modulus of the PAM changes with pressure when derived from the slope of the linear fit applied to the torsional test data. Assuming a first order stress-strain relationship, the torsional model can be rewritten as equation 3.26:

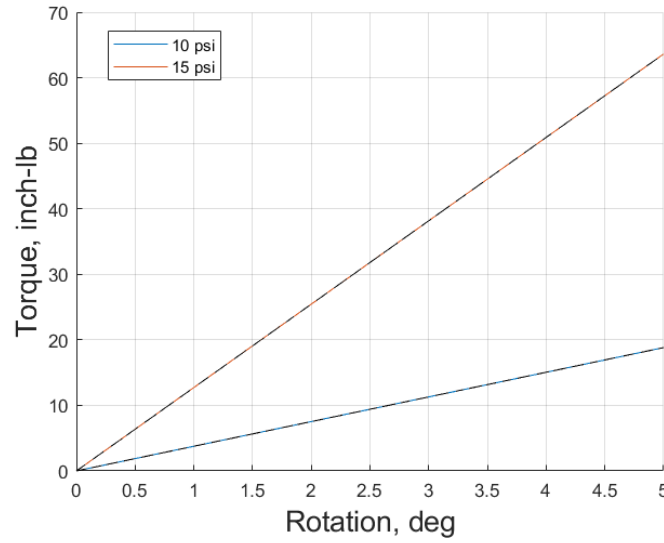


Figure 3.13: First order stress-strain model (colored solid line) vs. low pressure data (black dashed line)

$$T = \frac{GJ\phi}{L} \quad (3.26)$$

The linear fit of the torsional data is written as equation 3.27:

$$T_d = S\phi \quad (3.27)$$

Where S is the slope of the fitted line and T_d is the recorded torsion from the data. A comparison

of equation 3.26 and equation 3.27 show that S is equivalent to $\frac{GJ}{L}$. From this, equation 3.28 is found:

$$G = \frac{SL}{J} \quad (3.28)$$

Equation 3.28 would not be possible using a fourth order stress-strain relationship, so the prior work of comparing the errors between the first and fourth order stress-strain model was necessary to produce this equation by allowing for the assumption of a first order model. Graphing G solved from equation 3.28 and plotting against pressure results in the blue squares shown in figure 3.14. From this we see that the modulus increases from ten to twenty psi, then slightly decreases from twenty to fifty psi. The linear change in moduli with respect to pressure used in the optimization problem does not allow for this. This results in the optimization function being forced to choose a decreasing or increasing slope for the modulus. A negative slope value was chosen due to it resulting in the smallest possible error overall, which in turn resulted in a large error for the ten and fifteen psi case. When only the ten and fifteen psi cases are used in the optimization function, a positive slope is chosen. The change from a positive to negative slope is not seen in the axial or radial model of PAMs, and thus the assumption of a linear change in modulus with pressure works well with those models [36, 78].

Applying a fourth order polynomial fit to this data results in the red curve in figure 3.14. Evaluating the fourth order fit at the given pressures to determine G and inputting those values into equation 3.16 results in figure 3.15, showcasing both the model and the error. This method of using a fourth order fit reduces error between the linear slope of the model and the slope of the data to within 2.5 percent, eliminating the issue of the low pressure regime not working with

the model. The need for a fourth order modulus-pressure fit could be due to many factors. Due to the torsional model being based on the axial model, many of the assumptions of the axial model are carried over to the torsional model as well, such as ignoring frictional losses between the bladder and braid, losses due to tip effects, losses due to assuming braid inextensibility, and losses due to constant bladder radius assumptions [3,4,50]. The losses due to these assumptions may be greater at higher pressures, where greater frictional forces and bladder expansion exists, resulting in the transition from a positive to a negative slope, as these losses appear as an effective reduction in modulus in the model.

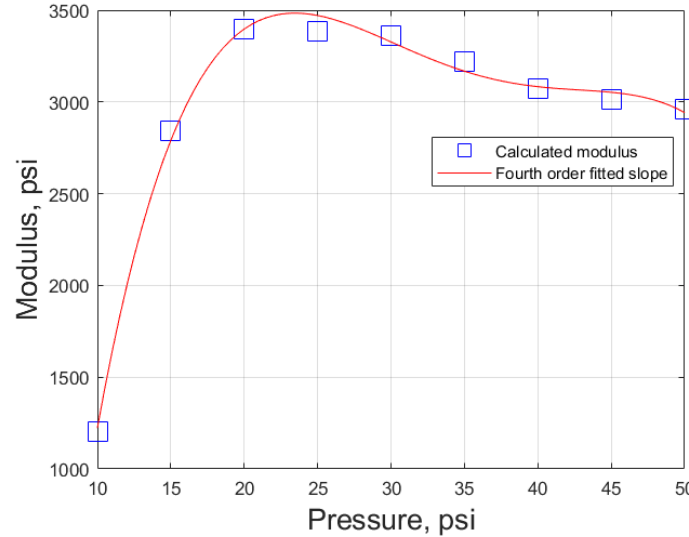


Figure 3.14: Change of modulus with pressure, derived from slope of fitted data (blue squares), and from fourth order polynomial fit (red curve).

3.8 Conclusion

A model to predict the torsional response of a PAM was outlined in this paper. This response is planned to be utilized in tunneling soft robots to bore out subterranean passages. Accurate prediction of the torsional response is required to ensure the robot's capability to provide

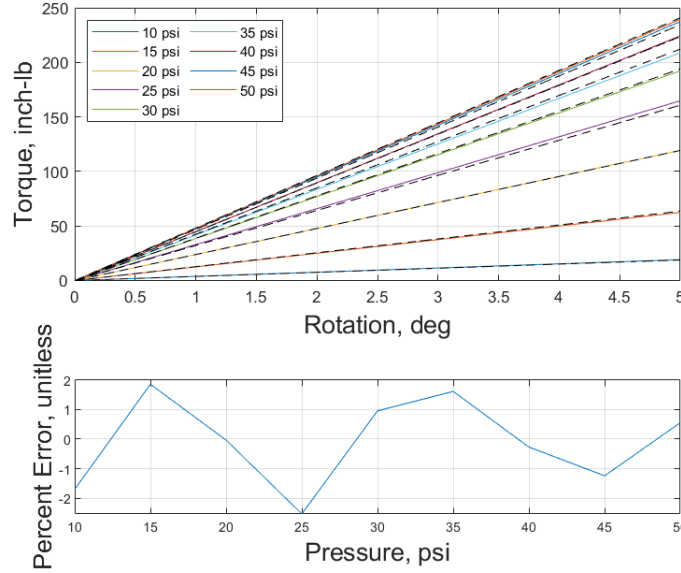


Figure 3.15: Top: Torsional model using 4th order modulus-pressure relationship (solid colored line) compared to data (black dashed line). Bottom: Model error of 4th order pressure-modulus model using first order stress-strain relationship.

the necessary stiffness required to allow a drilling tool, such as an auger, to bore. The work carried out in this paper entailed:

- Torsionally testing a PAM using a custom testing rig to obtain torsional data.
- Modeling the torsional response of a PAM using a force balance approach coupled with non-linear optimization to determine representative moduli that vary linearly with pressure for the bladder material for both a fourth order and first order stress-strain relationship.
- Identifying and correcting issues with using a linear modulus-pressure relationship by utilizing a fourth order relationship, resulting in reduced error for all tested pressures.

Future work for this model includes introducing radial forces, representing the PAM radially anchoring in a tube or tunnel. Anchoring the PAM prior to exerting a torque on it would affect the length and diameter of the PAM, as it would no longer be in a free contraction state, as well as introduce additional force terms into the force balance equation. Additional terms to account for

losses, such as from friction or the constant bladder radius assumption, could also be added to the model. Future testing would entail extending the range of twist past five degrees and the pressure range past 50 psi, to ensure that the model holds at higher twist angles and pressures, and that no phenomenon, such as the aforementioned bladder buckling or asymmetric braid loading, occur in those cases, despite not occurring in the current ranges. After further testing and validation, the model will be used to predict the torsional response of a PAM in the worm-like robot to validate it in practical applications.

Chapter 4: Optimizing Specific Actuation Force of Soft Composite Pneumatic Artificial Muscles Using Additively Manufactured Components

4.1 overview

Pneumatic artificial muscles (PAMs) are soft composite actuators comprised of a Kevlar braid, a latex bladder, and two aluminum end fittings, which can achieve a transfer of load through inflation. PAMs show an exceptional specific actuation force when normalized by their mass. However, the end fittings are typically machined from aluminum. Lighter materials, such as Acrylonitrile Butadiene Styrene (ABS-R), could be used to reduce the weight of the end fittings and increase the specific actuation force further. This research focuses on the design, fabrication, and testing of contractile PAMs with lightweight 3D-printed end fittings to replace traditional aluminum end fittings. 3D-printed PAMs were tested under quasi-static and dynamic conditions for various inflation pressures. PAMs were tested in blocked force and free contraction using a Materials Testing System servo-hydraulic testing machine. Burst and actuation tests were conducted to determine the PAM's maximum failure load and actuation characteristics. A 150 psi burst pressure was chosen to provide a safety factor of 1.5 when operating the PAM between 0–100 psi. PAMs tested in actuation were cycled quasi-statically within normal operating pressures between blocked force and free contraction. After additive manufacturing and design modifications,

the burst pressure of the PAMs improved from 43 psi at 363 lbf to 216 psi at 2085 lbf, showing their ability to withstand axial forces well beyond the target safety factor of 1.5. Finite element analysis models were created in Abaqus to validate the failure modes seen in the burst tests. End fittings made of an ABS-carbon fiber (ABS-CF) composite material were also tested and compared to metallic PAMs. The 3D-printed PAM actuation characteristics were found to be comparable to PAMs constructed with aluminum with greater specific actuation force and show no force or percent contraction loss when switching to ABS-R. Finally, an endurance test was designed to cycle the PAM until failure, to compare to the fatigue life of metal end fitting PAMs.

4.2 Introduction

Pneumatic artificial muscles (PAMs), when inflated with a working fluid such as air, expand or contract according to their braid configuration. Contractile PAMs experience radial expansion of the braid and bladder as they inflate. This resulting radial expansion produces a contraction in the axial direction, contributing to a large axial actuation force. A PAM's ability to generate large axial forces all while being lightweight makes it well-suited for actuation applications involving high specific actuation force and specific power [53, 65, 67–69], with work spanning into aerospace and robotics with trailing edge flap control [5, 63, 66], sweeping wing mechanisms [7, 86] and biomimetics [52, 78, 87]. Their lightweight and high-force output characteristics allow for a competitive edge over other commonly used mechanical and hydraulic actuators. PAMs have undergone many design iterations in terms of material selection. For the braid, PET and aramid fibers have been studied in terms of durability and extensibility [50]. For the bladder, multiple elastomeric bladders have been studied, including latex, silicone, and Viton,

as well as inelastic materials, such as low-density polyethylene [64]. However, comparatively little work has gone into the design of the end fittings. Many works fail to mention end fitting design at all, with many just being held together with devices such as metallic hose clamps. This lack of design of specialized end fittings have lead to issues such as pinhole leaks, which are hard to determine the cause of [57]. Woods et al. designed a custom, swaged end fitting for PAMs out of aluminum, which solved these issues and greatly improved the strength and fatigue life of PAMs [5]. Despite this, no work has been performed looking into alternative materials for end fittings outside of aluminum. For industries looking to improve the actuation characteristics of PAMs, lightweight material such as Acrylonitrile Butadiene Styrene (ABS-R) or Acrylonitrile Butadiene Styrene - carbon fiber composite (ABS-CF) can be used to increase the specific actuation force of PAMs. Using a lightweight polymer such as ABS-R or ABS-CF compared to aluminum allows for optimization of PAMs through weight reduction, as ABS-R filament has a density of $0.036 \frac{lbs}{in^3}$ and ABS-CF has a density of $0.040 \frac{lbs}{in^3}$, while aluminum has a density of $0.098 \frac{lbs}{in^3}$. Standard swaging techniques for metallic end fittings, a compression fit between the end fitting components and an epoxy binding the assembly, shown in Ref [5], can be similarly implemented into 3D-printed PAMs. 3D-printing has already been incorporated into other forms of robotics, such as inchworm-like soft robots, underwater robots used for buoyancy regulation, and jellyfish- like robots designed for safe robot-animal interaction [88–90].

The main goal of this study is to develop PAMs with end fittings composed of ABS-R and ABS-CF to maximize specific actuation force through weight reduction. To do this, 3D-printed PAMs using ABS-R were manufactured with the intent to operate within a specified pressure range of 0-100 psi while also ensuring a safety factor of 1.2, equating to a pressure of 120 psi. This safety factor would later be increased to 1.5, which will be discussed in section 4.8.

Actuation diagrams are constructed to analyze the 3D-printed PAM performance while axially loaded, to ensure 3D-printed PAM behavior matches that of metallic PAMs. Finite element analysis of the burst conditions in free contraction and blocked force are modeled in Abaqus, to validate the observed failure methods. Once a final design was chosen, A PAM utilizing the same design was made out of ABS-CF and tested. Finally, an endurance test was created, modeled after the fatigue tests performed in ref [5], to determine the ability of the 3D-printed end fittings to survive multiple actuation cycles without failure.

4.3 Fabrication

To maximize PAM actuation characteristics, particularly specific axial contraction force, replacing aluminum end fittings with ABS-R and introducing a weight reduction while maintaining axial contraction force is required. 3D-printed end fittings contain two components, the inner end fitting and end fitting socket, which combine to create a full PAM end fitting. To develop a full 3D-printed end fitting, the inner end fitting and end fitting socket were printed using a MakerBot Method X (Makerbot, New York, New York), which prints by way of fused deposition modeling. The MakerBot Method X supports various types of filament, but high strength tolerance was preferred for the fabrication of PAM end fittings, therefore, ABS-R was chosen. ABS-R plastic is an ABS-based filament created by Ultimaker that has improved print reliability and refined jam reduction. The end fittings were also printed with SR-30 support material, a liquid-soluble 3D-printing support used exclusively with high-temperature filaments, such as ABS-R, due to its unique adhesion ability. For the inner end fitting, the height is 0.78 inches, the inner diameter is 0.39 inches and the outer diameter is 0.92 inches. The end fitting socket has a nominal height of

0.90 inches based on the CAD models, an inner diameter of 0.94 inches, and an outer diameter of 1.26 inches. Figure 4.1 presents a schematic for 3D-printed PAMs with a target active length of 12 inches.

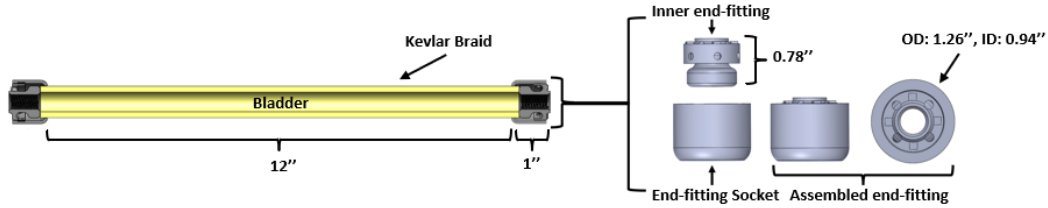


Figure 4.1: 3D-printed PAM Schematic.

As mentioned, PAMs consist of a Kevlar braid, latex bladder, inner end fitting, and end fitting socket. Figure 4.2 shows the PAM components.

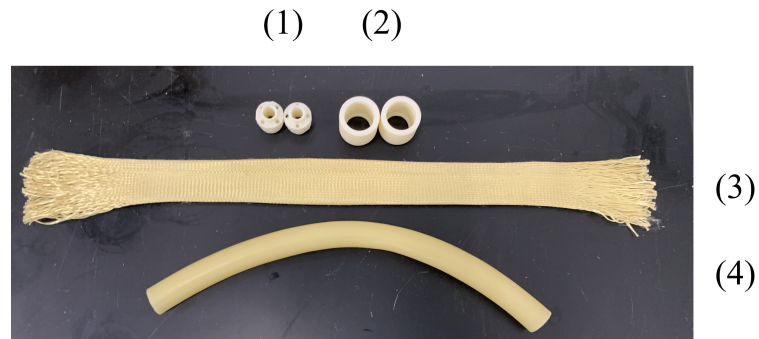


Figure 4.2: PAM components: (1) Inner end fittings, (2) end fitting sockets, (3) Kevlar braid, (4) latex bladder.

To construct a 3D-printed PAM, first the Kevlar braid and latex bladder were selected. The braid was 0.012 inches thick with a diameter of 2 inches at 45 degree braid angle, and the bladder had a diameter of 7/8 inches with a wall thickness of 1/16 inches. Next, the Kevlar braid and latex bladder were cut to length. The length of the braid was cut to 24 inches. 12 inches were to account for the active length of the PAM. 1 additional inch was added to each end, to account for the end fittings. The final 10 inches were added to allow for 5 extra inches on each end of

the PAM, to be used as hand holds during the manufacturing process. For a foot-long PAM, the bladder was cut to 13.50 inches, 1.50 inches over the PAM's intended length to account for the length of the end fittings.

After cutting the braid and bladder to length, the PAM could be assembled. Figure 4.3 details the steps of the assembly process. In the first step, the bladder was centered inside the braid. The end fitting sockets were then slid over the braid and bladder, and the sockets were then pushed close to the center of the braid and bladder in the second step, allowing the inner end fittings to be attached to the ends of the bladder in step three. In step four, the PAM's braid was tightly secured on one side, and a single end fitting socket was pulled over the inner end fitting. The inner end fitting and end fitting sockets are held together by a compression fitting. West Systems epoxy resin 105-B and soft hardener 206-B (Gougeon Brothers, Inc, Bay City, Michigan) sealed the connection between the braid, bladder, and end fitting components. Step four was repeated for the other side of the PAM after the epoxy had dried, resulting in step five. Once the epoxy had dried on both sides of the PAM, the extra braid was cut from the PAM, and the PAM was threaded using a 7/16 inch tap.

4.4 Pressurization Systems

4.4.1 Air Pressurization System

A PID-controlled pressure system is used to actuate the PAMs during testing. The PID controller regulates the internal pressure of the PAM to ensure it does not fluctuate as the internal volume changes during actuation. The pressurization system is supplied with air by a DEWALT Heavy-Duty 200 PSI 4.5 Gallon Electric Wheeled Compressor (D55146) (DeWalt, Towson,

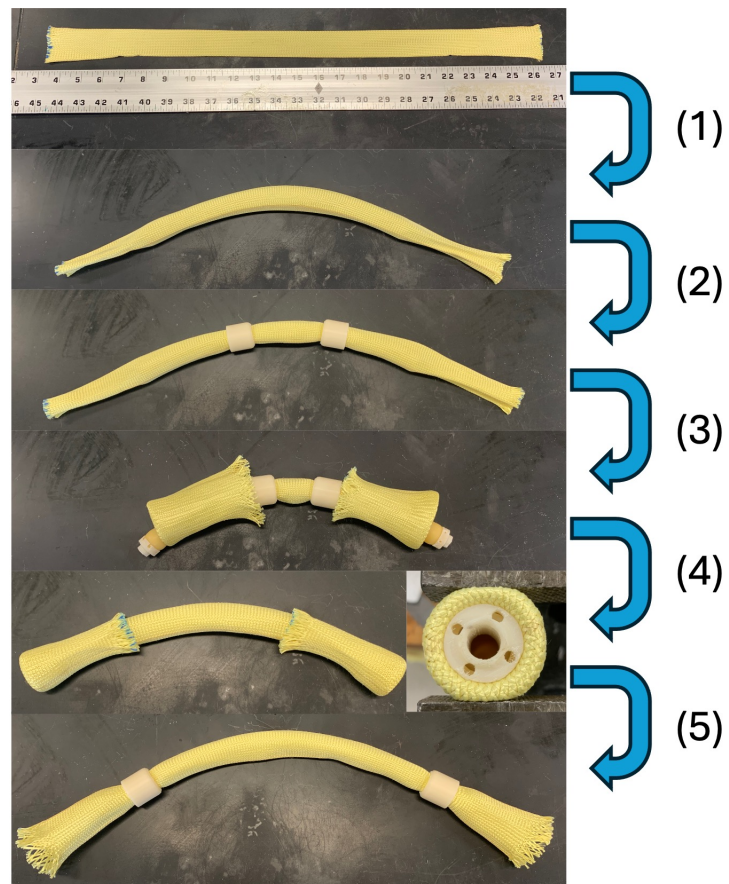


Figure 4.3: 3D-printed PAM assembly process.

Maryland). The compressed air then travels to a Norgren R73G-3AT-RSG Pressure Regulator (Fradley Park, Lichfield, UK) that allows for direct airflow control of the pneumatics entering the system. Air is then fed to an Enfield LS-V05s High Speed 5/3 Proportional Directional Valve (Enfield, Trumbull, Connecticut) which an Enfield D1 Bi-Directional Proportional PWM Valve Driver (Enfield, Trumbull, Connecticut) powers. The valve driver is then connected to an Agilent Dual Output 0-8V.5A/0-20V,2.5A DC Power Supply (E3648A) (Agilent Technologies, Inc., Santa Clara, California), and an Arduino MEGA 2560 Rev3 Microcontroller (Arduino, Robbinsville, New Jersey) which is connected to an operating computer that powers the Arduino. The direct proportional valve allows full control of pneumatics entering the PAM at any time. Air is then passed through an Omegadyne PX209 Pressure Transducer (Omega Engineering Inc, Norwalk, Connecticut) which reads the pressure in the PAM and feeds the information back to the microcontroller, which is then fed to the operating computer. The pressure transducer is also connected to a Materials Testing System Servo-Hydraulic Testing Machine (MTS), which allows pressure to be recorded alongside the axial contraction force and displacement.

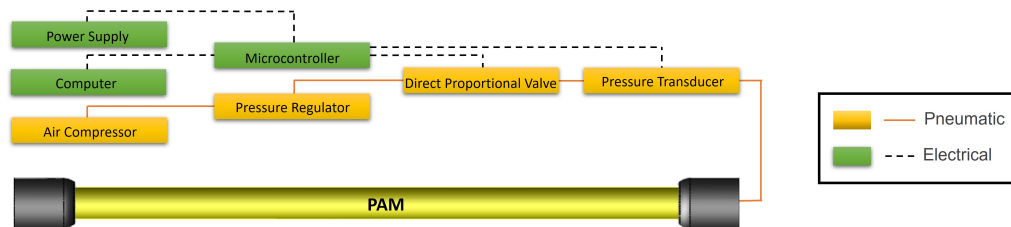


Figure 4.4: Air Pressurization Schematic.

4.4.2 Nitrogen Pressurization System

When testing PAMs in blocked force and free contraction, it often was the case where the PAM's failure limits exceeded the pressure capabilities of the air pressurization system. The air

pressurization system can only supply and regulate air effectively up to 120 psi, where the limiting factors are the proportional directional valve and the PWM Valve Driver. Therefore, a new pressurization system was introduced, which utilized a nitrogen tank pressurized at over 500 psi, allowing for testing outside the air system's capacity. The nitrogen system uses a series of valves that are manually controlled to inflate the PAM to specified pressures. The system incorporated an ASHCROFT 0-10Vdc pressure transducer (Ashcroft Inc., Stratford, Connecticut) connected to an OMEGA Display Box, Model: DP25B-E 115V (DwyerOmega, Michigan City, Indiana), which showed the pressure within the PAM during testing. Figure 4.5 shows the nitrogen tank used for high-pressure PAM testing.

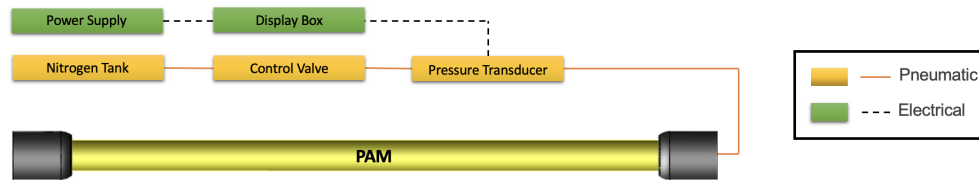


Figure 4.5: Nitrogen Pressurization Schematic.

4.5 Testing

4.5.1 Burst Test

PAM burst test consists of two types; blocked force and free contraction. Blocked force burst test refers to inflating the PAM in a blocked force state (unable to contract). The free contraction burst test refers to the PAM being inflated until burst in a state of free contraction (fully contracted with no force production). These testing setups are used to analyze the PAM's ability to withstand stresses at various states, the blocked force state allows for the axial integrity

of the PAM to be analyzed, while free contraction burst tests allow for the radial strength of the PAM to be determined. It is important to note that the burst test is stopped if either test (blocked force or free contraction) reaches 300 psi without burst. This restriction was set in place for safety reasons so as not to burst the PAMs at high pressures. The PAM setups are shown below in Figure 4.6 and Figure 4.7.

4.5.1.1 Blocked Force Burst Test

To test the PAM's maximum burst pressure and ensure the PAM can operate within the specified operating range, the PAM is placed into an MTS at resting length. The MTS collects axial contraction force and pressure data for the testing duration. The burst test is conducted with the PAM held at blocked force. As the PAM begins to inflate, its ability to contract is prohibited, generating the maximum possible axial contraction force for each given pressure. The pressure is then brought up slowly while closely observing the PAM's behavior as pressure and axial contraction force increase. A successful test involves increasing the PAM from 0 psi to failure, or until a pressure of 300 psi is achieved in the PAM.

4.5.1.2 Free Contraction Burst Test

For the free contraction testing of the PAMs, a similar setup to the blocked force burst test was conducted. For this test, only one end of the PAM is clamped into the MTS. When pressurized, the PAM is free to contract, allowing it to radially expand, and produce no axial contraction force. To run the test, the PAM is placed into the MTS, then pressure is slowly brought up until either the PAM bursts or 300 psi is reached.



Figure 4.6: Blocked force burst test setup.



Figure 4.7: Free contraction burst test setup.

4.5.2 Actuation Test

To fully characterize 3D-printed PAMs, actuation tests were performed to determine the performance of the PAMs at specified pressures along with their accompanying axial contraction force. First, the PAM is clamped into the MTS at resting length. With the PAM in place, the pressure is increased in increments of 10 psi from 0 - 100 psi. At each pressure, the PAM is manually cycled 3 times from block force to free contraction. Actuating the PAM before testing allows for the stiff, uncycled elastomeric bladder to stretch within the PAM, a phenomenon known as the Mullins effect. Not accounting for the Mullins effect can cause an underestimate in force data due to the bladder not being softened before testing [73]. After cycling the PAM from blocked force to free contraction 3 times, an MTS procedure is used to cycle the PAM quasi-statically at a speed of 0.02 in/s at each pressure. Figure 4.8 shows the transition from blocked force to free contraction during actuation cycling.

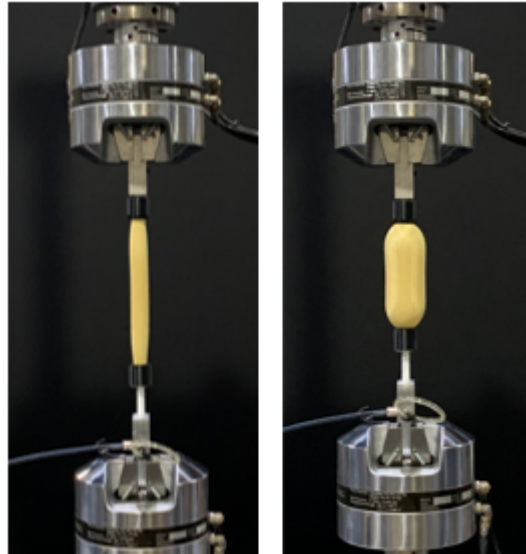


Figure 4.8: PAM setup: Left: PAM at blocked force during actuation cycling. Right: PAM at free contraction during actuation cycling.

4.6 Burst Test Results and Design Iterations

The first iteration of 3D-printed PAMs was printed with 20 percent infill density applied to the inner end fittings and end fitting sockets. The infill density of a 3D-printed component measures the percentage of modeling material inside a 3D-printed part compared to void space. Both components also included two shells and utilized a balanced print mode. Shells on a 3D-printed part are the number of outside solid layers used on the model. A balanced print mode is a setting on the MakerBot Method X that prioritizes a balance between surface quality and print time. With the first 3D-printed PAM developed, burst testing was conducted.

The end fittings failed at an axial force of 363 lbf at 43 psi. Due to the end fittings failing within the expected operating range, design changes were implemented to increase their performance. After a close examination, it was evident that one of the end fittings failed due to delamination, and that half of the inner end fitting component was ripped out of the end fitting socket. Figure 4.9 shows the PAM's end fitting after the first burst test. Free contraction burst testing was also performed on this iteration. For this test, the PAM was capable of achieving a safety factor of 1.2 without bursting. Due to this, No changes in the radial strength of the end fittings were made for future iterations, and the free contraction test was not repeated.

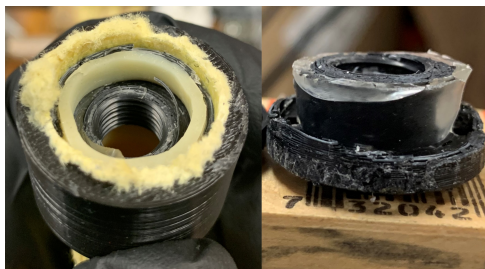


Figure 4.9: end fitting delamination.

During the 3D-printing process, the orientation of parts plays a role in their capabilities.

When the inner end fitting was printed, it was printed from bottom to top with the component standing straight up. This made the layer lines of the print fall perpendicular to the direction of the applied axial force, thus causing the inner end fitting to fail simply because the layers were separated during loading.

To combat delamination, print settings and print orientation were adjusted. Instead of printing the inner end fitting standing straight up, they were adjusted to be printed lying on their side, allowing the layer lines of the print to fall in line with the direction of the force. In addition to changing the printing orientation, the infill density was increased from 20% to 70%. Increasing the infill density of the inner end fitting allowed for more material to be used within the part, thus increasing its strength. With these new changes implemented, a second burst test was conducted.

The second iteration end fitting failed at an axial force of 559 lbf at 68 psi due to inner end fitting pull-out rather than previously seen delamination. The inner end fitting pull-out results from the compression fitting between the inner end fitting and end fitting socket, along with the epoxy, being unable to hold the inner end fitting in place. Figure 4.10 shows the inner end fitting pullout from the end fitting socket.

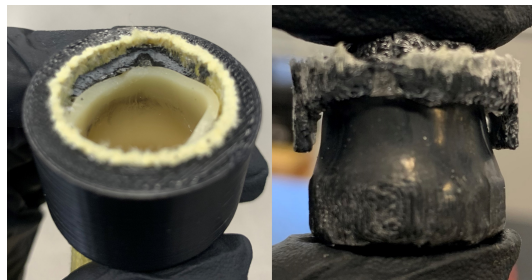


Figure 4.10: Inner end fitting pullout from socket.

Until this point, there have been two ways in which the inner end fitting is secured into the socket. One is through a compression fitting between the inner end fitting and end fitting socket,

and the other is through the application of epoxy, which is visible in Figure 4.10. Since the inner end fitting was pulled directly from the socket while keeping the inner end fitting intact, a retaining ring design was implemented to hold the inner end fitting in place. For design iteration 3, a 3D-printed retaining ring was incorporated that sits over the top of the inner fitting and underneath the end fitting socket. To do this, a 0.08 inch cutout was introduced into the end fitting socket which housed the retaining ring. Figure 4.11 shows the third iteration retaining ring end fitting design.

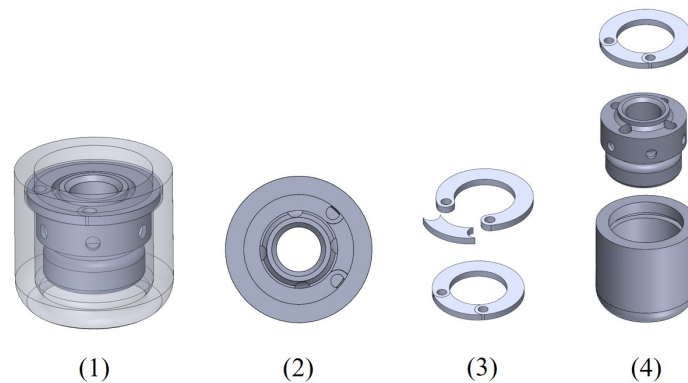


Figure 4.11: : Retaining ring end fitting design: (1) Full assembly, (2) top-down view, (3) retaining ring components, (4) end fitting components (retaining ring, inner end fitting, and end fitting socket).

It is worth noting that the retaining ring was printed using 20% infill density. Since the fabrication of the PAM involves compressing the retaining ring into the socket, high infill density increases the stiffness of the ring, causing the ring to break before it can be inserted into the cutout. Multiple prints of infill densities ranging from 20-40% and one at 70% showed the retaining ring breaking over 20% while inserting it into the cutout. Therefore the ring was printed with a 20% infill density to ensure it can be used in the end fittings. The retaining ring has an inner diameter of 0.67 inches and an outer diameter of 1.06 inches. A third burst test was conducted with the new end fitting design.

The end fittings failed at an axial force of 655 lbf at 81 psi. During the inspection of the third iteration PAM, it was evident that the PAM failed due to thread detachment. The connection between the PAM and the MTS was removed and the PAM threads were observed to be stripped. Figure 4.12 shows the damage sustained by the threads on the inner end fitting.

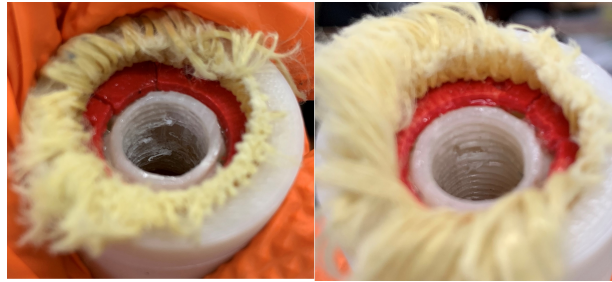


Figure 4.12: Thread failure on third iteration PAM.

When tapping the PAM's inner end fittings, it was observed that the threads could not dig into fully solid material. The Makerbot printed the end fitting as two concentric shells, and not as a solid piece, so that the threads cut through these two shells, reaching the portion of the inner end fitting where the component was not solid. This caused the outer shells of the threaded inner end fitting to detach. To combat thread failure, the number of shells on the inner end fitting was increased from 2 to 4. A change in the print mode for the components was also made. Instead of printing in balanced mode, MakerBot provides a solid print mode option that prints highly dense and stronger 3D prints. This change was implemented to maximize the density of the inner end fitting to avoid damage to the threads. The top edge of the inner end fitting where the end fitting is tapped was also raised by 0.12". This allows more epoxy to sit on the retaining ring by preventing it from overfilling and pouring into the PAM, providing more resistance against an inner end fitting pullout.

With the new changes implemented, a fourth iteration of 3D-printed PAM, shown in figure

4.13, was developed and burst-tested. Table 4.1 outlines the burst pressure and force for all four iterations. The fourth burst test with the new manufacturing changes experienced no failure and reached a force of 1160 lbf at 123 psi. This test shows that the retaining ring design can achieve the required pressure of 120 psi without failure, ensuring a safety factor of 1.2.

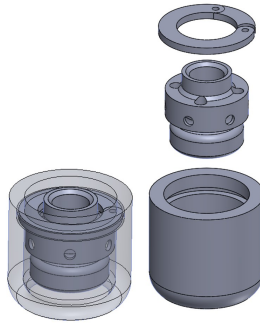


Figure 4.13: Fourth iteration PAM design full end fitting.

Table 4.1: Burst pressure and force per iteration.

PAM Iteration	Pressure [psi]	Force [lbf]
1	43	363
2	68	559
3	81	655
4	123	1160

4.7 Actuation Analysis

With a 3D-printed PAM design now capable of reaching the required safety factor of 1.2, an actuation test was conducted to characterize the PAM from 10-100 psi.

4.7.1 Actuation Analysis

A PAM model adapted from Ref [36] was used to model the actuation properties of both the 3D-printed and metallic PAMs. To model the PAMs, the number of turns of the braid, N , and the braid length, B , were first found as follows:

$$N = \frac{L_0}{D_0 \pi \tan(\theta_0)} \quad (4.1)$$

$$B = \frac{L_0}{\sin(\theta_0)} \quad (4.2)$$

where D_0 is the initial outer diameter of the PAM, θ_0 is the initial braid angle, and L_0 is the initial length. The Gaylord force is determined using equation 4.3:

$$F_{Gaylord} = \frac{P}{4\pi N^2} (3L^2 - B^2) \quad (4.3)$$

Where L is the current length of the PAM. The Gaylord force is useful for predicting the blocked force of the PAM where little energy from inflation is being utilized to stress the bladder. A more comprehensive model is required for actuation states with significant bladder strain.

First, the diameter of the PAM as a function of the PAM length is defined in eq. 4.4 and the instantaneous braid angle is defined in equation 4.5:

$$D(L) = \frac{\sqrt{B^2 - L^2}}{\pi N} \quad (4.4)$$

$$\frac{\cos(\theta_0)}{D_0} = \frac{\cos(\theta_1)}{D_1} \quad (4.5)$$

As the PAM inflates, the bladder is strained both circumferentially and axially. These

strains are defined by equations 4.6 and 4.7, respectively.

$$\epsilon_c = \frac{R}{R_0} - 1 \quad (4.6)$$

$$\epsilon_z = \frac{L}{L_0} - 1 \quad (4.7)$$

Where R is the instantaneous radius of the PAM and R_0 is the initial radius of the PAM. Due to the hyperelastic nature of the bladder, the stresses in the bladder were modeled using a fourth-order relationship between stress and strain

$$\sigma_c = \sum_{k=1}^4 E_k \epsilon_c^k \quad (4.8)$$

$$\sigma_z = \sum_{k=1}^4 E_k \epsilon_z^k \quad (4.9)$$

where σ_c is the circumferential stress, Eq. 4.8, and σ_z is the axial stress, Eq. 4.9.

Using these formulations, the complete axial model for the PAM can be written as:

$$F = \frac{P}{4\pi N^2}(3L^2 - B^2) + \sigma_z \frac{V_B}{L} - \sigma_c \frac{tL^2}{2\pi RN^2} \quad (4.10)$$

where P is the internal pressure, V_B is the volume of the bladder, and t is the bladder thickness.

The moduli of the bladder cannot be explicitly solved for, and as such a non-linear optimization problem was used to determine the moduli. The optimization problem used to determine the moduli of the bladder can be written as:

$$\begin{aligned}
\min_x \quad & \sum_{k=1}^n \sqrt{\frac{\sum_{i=1}^j (F_{M,i,k} - F_{E,i,k})^2}{j}} \\
\text{s.t.} \quad & lb \leq x \leq ub \\
& c(x) \leq 0
\end{aligned} \tag{4.11}$$

where F_M is the force predicted by the model, F_E is the measured experimental force, j is the total number of data points taken for a given pressure, and n is the total number of pressures tested. The variable x being optimized is a vector comprised of $[x_1, x_2, x_3, x_4, x_5, x_6, x_7, x_8]^T$.

These variables are used to determine the moduli in the form of

$$\begin{aligned}
E_1 &= x_1 P + x_2 \\
E_2 &= x_3 P + x_4 \\
E_3 &= x_5 P + x_6 \\
E_4 &= x_7 P + x_8
\end{aligned} \tag{4.12}$$

The lower bounds on x are $\{x_k = -1000 \mid k = 1, \dots, 8\}$ while the upper bounds are $\{x_k = 1000 \mid k = 1, \dots, 8\}$. Finally, the non-linear constraint, $c(x)$, can be defined as

$$c(x) = -(dE_c + dE_z) \tag{4.13}$$

where dE_c is the incremental internal circumferential energy and dE_z is the incremental internal axial energy. This constraint comes from the Drucker stability criteria, which states that the incremental internal energy of a stable material must increase as strain increases [75].

4.7.2 Actuation Test Socket Failure

During the manual cycling portion of the test, the PAM experienced delamination to a section of the end fitting socket at 100 psi. Therefore only actuation data from 10-90 psi could be obtained. Figure 4.14 shows the damage sustained by the end fitting socket during actuation.

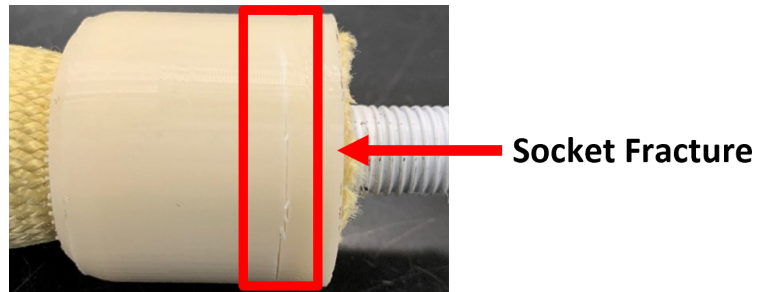


Figure 4.14: End fitting socket delamination.

The placement of the delamination showed that the cause of the failure was the retaining ring pulling on the end fitting socket axially. This axial load results from the retaining ring sitting inside the end fitting socket, where it holds the inner end fitting in place. The resulting load begins to pull the socket apart, which is designed to handle high radial forces, not high axial forces. The layer lines of the end fitting socket lay perpendicular to this axial load, making it prone to failure by delamination, similar to the failure seen in the first iteration of 3D-printed PAM.

4.7.3 Actuation Results

Despite the PAM failing at 100 psi, an actuation diagram was constructed using data obtained from actuation testing from 10-90 psi and compared to the axial model utilizing the bladder moduli optimization. Figure 4.15 shows the axial force output with respect to the percent contraction at specified pressures of 10-90 psi for both the data (dashed lines) and the model

(solid colored lines).

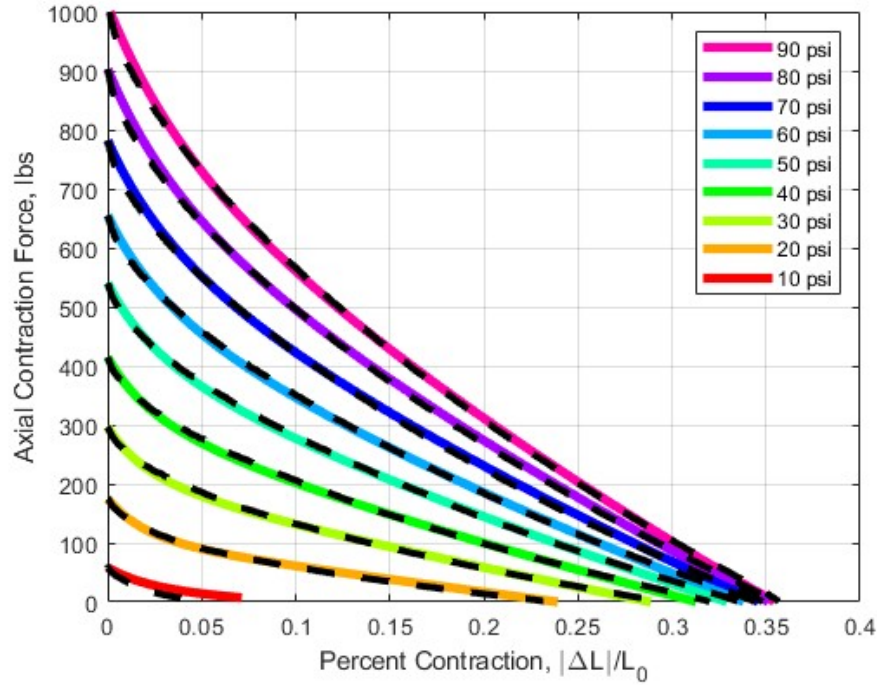


Figure 4.15: 3D-printed retaining ring PAM actuation diagram.

4.8 Improved End Fitting Design

Given the failure of the retaining ring design during actuation testing, multiple changes to both the design and testing procedures of the 3D-printed PAMs were implemented. The first change included adding a thicker socket to the retaining ring design to avoid premature failure during actuation testing. Instead of utilizing a 0.16 inch socket thickness for the retaining ring design, the socket thickness was increased to 0.24 inches to increase the component's strength.

Failure during actuation also prompted a change in metrics for analyzing the 3D-printed PAMs. Instead of opting for a safety factor of 1.2, a safety factor of 1.5 was targeted. This change would allow for extra safety when operating the PAM to avoid failure close to or within

operating pressures.

Due to the PAM failing after multiple actuation sweeps despite being below 120 psi, it was decided that the PAMs would undergo endurance testing in addition to the burst and actuation testing, to ensure the PAM could achieve a suitable number of actuation cycles before failure. To allow for this testing, a change was made to now use 9 inch active length PAMs instead of 12 inch. This was done to achieve a 5% contraction with the springs available in the endurance testing setup. A 5% contraction was desired to compare the results to the fatigue results found in Ref [5]. PAM axial contraction force output is unaffected by the active length of the PAM as shown in Ref [51], so changing from 9 inches to 12 inches would not affect any comparisons nor data obtained during future testing.

A new design was also developed that utilizes the braid to secure the inner end fitting during PAM operation. This new design is called the fold-over design. The name is derived from the process of folding the braid over the newly developed retaining ring during manufacturing. The fold-over design does not include any alterations to the end fitting socket, making it less likely to experience socket delamination. Figure 4.16 shows the new fold-over design.

4.8.1 Fold-over

As mentioned, the fold-over design was created to prevent the same type of failure as the retaining ring. The new design doesn't involve the ring sitting inside a cutout located in the end fitting socket. This configuration was meant to prevent the mechanism that holds the inner end fitting in place from adding additional failures into the end fitting socket, such as delamination, which the end fitting is highly susceptible to, as seen in the prior retaining ring design failure.

The fold-over design's manufacturing process closely follows the same steps as mentioned in the original PAM manufacturing process, however, new steps are added to complete this design. First, the braid at the end of step 4 is fed through the ring that holds the inner end fitting in place. The braid is then looped around the ring and fed back over the inner end fitting. Lastly, the braid is secured through zip ties to the PAM, then the sockets are pulled onto the inner end fittings and epoxied. The zip ties are then removed. Two versions of the fold-over design were constructed, a 0.16 inch socket thickness version and a 0.24 inch socket thickness version, in order to maintain consistency between the retaining ring and fold-over designs.



Figure 4.16: Fold-over end fitting design.

4.9 Retaining Ring and Fold-Over Design Results

The following bar graphs, Figure 4.17 and 4.18, show the test data acquired from testing the new designs in blocked force and free contraction.

Data show that the 0.24 inch thick retaining ring design reaches a maximum blocked force of 1663 lbf at 182 psi, while the fold-over design reaches 2017 lbf at 207 psi for a 0.16 inch socket thickness and 2085 lbf at 216 psi for a 0.24 inch socket thickness. The failure of the retention ring design came from the complete failure of one of the end fittings, while the failure

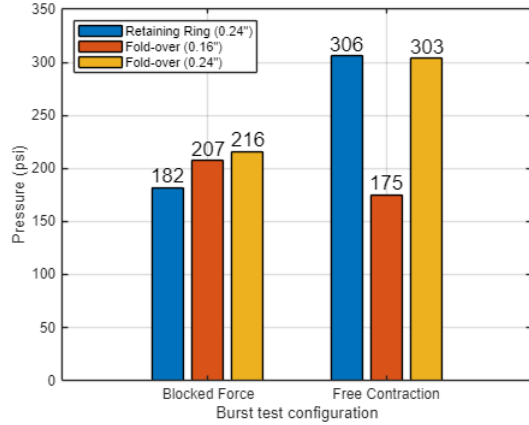


Figure 4.17: Retaining ring and Fold-over burst pressure in blocked force and free contraction.

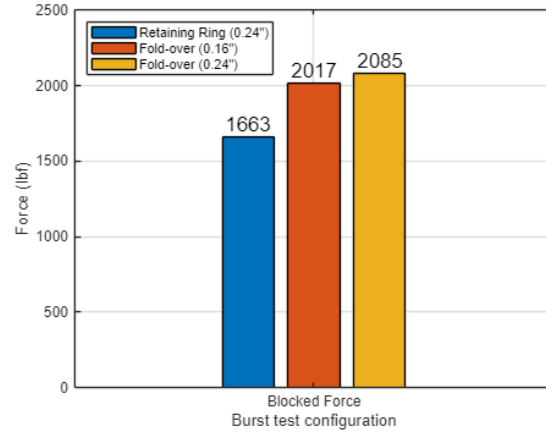


Figure 4.18: Retaining ring and Fold-over force at burst in blocked force.

of the fold-over designs (0.16 inches and 0.24 inches) came from thread detachment, where the connection between the MTS and the PAM was broken due to the tapped threads on the PAM ripping out. The data also shows that the 0.24 inch fold-over and retaining ring design in free contraction reached 300 psi, and the 0.16 inch fold-over design reached 175 psi. The failure in the 0.16 inch fold-over socket design was a complete failure of a single end fitting, where the end fitting separated from the PAM.

4.9.1 Specific Actuation Force Comparison

The last metric being used to analyze which design would be best suited for further testing included comparing the specific actuation forces of each design to each other, as well as to a metallic PAM of comparable diameter and braid angle. Comparing the specific actuation forces of the 3D-printed designs will help to narrow down which of them best brings us closer to our goal of optimizing the PAMs through weight reduction.

To make an accurate comparison of the specific actuation force between PAMs, it is imperative

that the PAMs be comparable in braid angle and diameter, to ensure the standard force output of the two PAMs are similar. To illustrate this, the specific actuation force (SAF) is defined as

$$SAF = F/m \quad (4.14)$$

where F is the blocked force at 100 psi and m is the mass of the entire PAM. Here, the SAF can be increased by either increasing force or decreasing mass. If, for example, one PAM was lighter, but the other PAM had a larger diameter and significantly higher braid angle, thus resulting in a higher force output, it would be difficult to draw a meaningful conclusion from the SAF, as the difference between them would not be due to a change in just one factor. To account for this, each 3D-printed PAM was compared to a theoretical metallic PAM with equal braid angle and diameter. This would result in the two PAMs having the same force output for a given pressure. The SAF can then be compared using the relative difference to the metallic PAM or

$$\Delta_{SAF}\% = \left(\frac{SAF_{3D} - SAF_M}{SAF_M} \right) \times 100 \quad (4.15)$$

expressed as a percentage. Here SAF_M is the specific actuation force of the metallic PAM, and SAF_{3D} is the specific actuation force of the 3D printed PAM. Substituting equation 4.14 for both the metal and 3D-printed PAM into equation 4.15 results in equation 4.16, due to the forces in both PAMs being equivalent.

$$\Delta_{SAF}\% = \left(\frac{m_M}{m_{3D}} - 1 \right) \times 100 \quad (4.16)$$

Where m_M is the mass of the metallic PAM and m_{3D} is the mass of the 3D-printed PAM. Here, for PAMs of comparable force output, only the masses of the PAMs affect the relative difference in SAF. Table 4.2 below shows the metallic, retaining ring, and fold-over design weights and the relative difference in SAF when compared to the metallic PAM.

Table 4.2: SAF relative difference table.

PAM Configuration	Mass [10^{-3} slug]	Relative Difference in SAF [%]
Metallic PAM	6.7	-
ABS-R Retaining Ring (0.24")	6.2	8.43
ABS-R Fold-over (0.16")	5.9	12.86
ABS-R Fold-over (0.24")	7.0	-3.06

4.9.2 Summary of Analysis Metrics

Based on iterative test data of 3D-printed PAMs in blocked force and free contraction, along with the specific actuation force comparison to metallic PAMs, data shows that the best design to move forward with is the 3D-printed 0.16 inch socket fold-over PAM. In blocked force, both the 0.16 inch and 0.24 inch socket thickness fold-over design reached over 200 psi, while the retaining ring design only reached 182 psi. In free contraction, although the fold-over design with 0.16 inch socket thickness underperformed compared to the 0.24 inch socket PAMs, both designs were above the targeted 1.5 factor of safety. Comparing the specific actuation forces of the 3D-printed PAM to the metallic PAM shows that at larger socket thicknesses, there is a loss in SAF due to an increase in PAM weight; therefore, to achieve the main goal of the overall study of maximizing SAF while maintaining a factor of safety of 1.5, the 0.16 inch fold-over PAM design was found to be the superior option. An actuation diagram was constructed for the 0.16 inch fold-over PAM. Figure 4.19 shows the axial force output with respect to percent contraction

at specified pressures from 10-100 psi for the fold-over design.

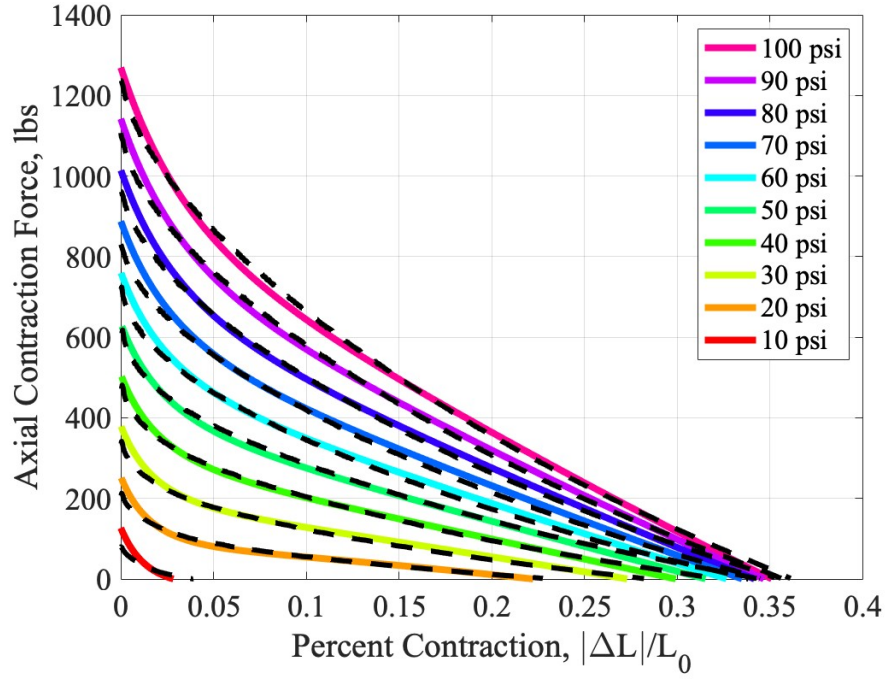


Figure 4.19: Actuation diagram for Fold-over PAM, Experimental data: Dashed, Model data: Solid.

4.10 Carbon Fiber Reinforced ABS-R

With a finalized design chosen for the 3D-printed PAM end fitting, additional alternative materials could be explored. To further maximize the burst test characteristics seen in section 4.8, additional end fittings were made from carbon fiber reinforced ABS (ABS-CF). ABS-CF is a filament sold by Ultimaker (Ultimaker, New York, New York) that provides a higher tensile strength than that of ABS-R with a tensile strength of 8557 psi, compared to that of 5947 psi for ABS-R. Making the transition to ABS-CF also allows us to closely maintain our reduction in weight of the end fittings due to ABS-CF's density of $0.040 \frac{lbs}{in^3}$, only $0.004 \frac{lbs}{in^3}$ larger than ABS-R. A new iteration of end fittings was manufactured from ABS-CF utilizing the same dimensions

and design as the 0.16 inch fold-over design.

4.10.1 Fabrication

To fabricate the new PAM iteration, end fittings made of ABS-CF were printed using a Makerbot Method X, the same 3D-printer used for manufacturing ABS-R. When transitioning to ABS-CF, a change had to be made to the support filament it was being printed with. The Makerbot Method X does not provide compatibility between ABS-CF and the SR-30 support filament used when printing ABS-R. Therefore, to print ABS-CF, RapidRinse support filament (Ultimaker, New York, New York) was used. RapidRinse is water-soluble Ultimaker support filament used in conjunction with Ultimaker ABS based filaments. Outside of these changes to the print material, the same process of constructing the 0.16 inch fold over PAM was used for the construction of this PAM.

4.10.2 Burst Test and Results

Both free contraction and blocked force burst tests were performed using ABS-CF end fitting. For the free contraction burst test, the PAM reached 300 psi without bursting. For the blocked force burst test, the PAM reached 2508 lbf at 248 psi. using equation 4.16, the specific actuation force difference between the ABS-CF PAM and an equivalent metal PAM was calculated. The difference was found to be 8.70 %. These results show that when switching from ABS-R to ABS-CF, we can increase the maximum blocked force by 491 lbf and the burst pressure by 41 psi, while maintaining a respectable increase in the specific actuation force of 8.70 % when compared to metallic end fittings. It is important to note that when printing with a fiber

reinforced polymer such as ABS-CF, the main drawback becomes the requirement of having a printer capable of working with less common printing material.

4.11 Finite Element Modeling

To validate the results found from performing burst tests on the 3D-printed end fittings, two finite element models were constructed: one representing the blocked force conditions, and one representing the free contraction condition. Both models were constructed and evaluated in Abaqus 2025 and based on the 0.16 inch thick fold-over style end fitting made of ABS-R. For the blocked force burst case, the PAM was found to fail due to thread failure. To validate this, an axisymmetric model was made of a 7/16 inch steel bolt placed in contact with the threads of the inner end fitting piece. A pressure load of 13,420 psi was applied to the top of the bolt, resulting in a total load of 2017.5 lbs, which is the axial load present at the time of failing for this case. The outside wall of the inner end fitting was encastered, to represent the compression fit used to hold the inner end fitting in place. Contact was made between the bolt and the inner end fitting. The contact was modeled as a hard contact, with a friction coefficient of .2. Both the bolt and inner end fitting were meshed using CAX4R elements. The undeformed model can be seen in figure [4.20](#), with the bolt on the left of the figure and the end fitting on the right. The deformed model with stresses is given in figure [4.21](#). Here we see stresses at or near the yield stress of ABS-R, being 5947 psi, in the threads. This model shows that the threads will yield, causing failure in the threads, ultimately leading to the bolt pulling out. This result matches what is observed in the physical PAM.

The model for the free contraction burst scenario was also constructed using an axisymmetric

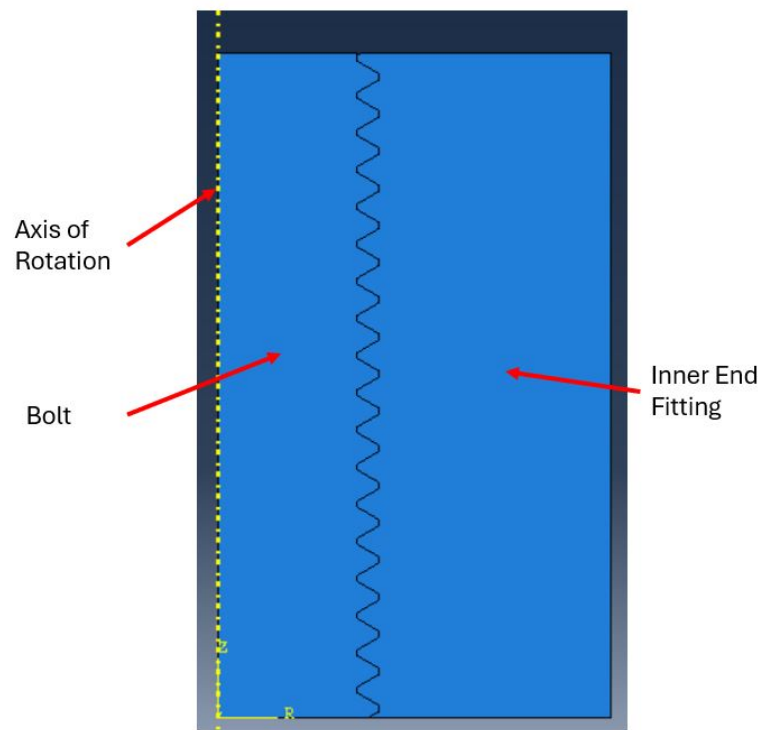


Figure 4.20: Undeformed model of blocked force burst.

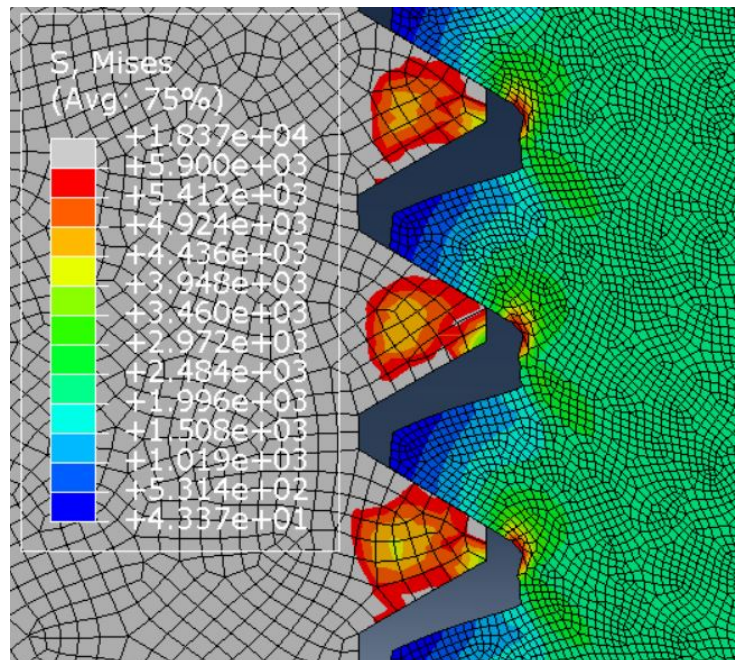


Figure 4.21: Deformed model of blocked force burst.

2D model. Here, the outer socket of the 3D-printed end fitting is modeled, along with the Kevlar braid, represented by a singular strand. Additionally, a rigid surface is placed behind the strand, representing the bladder and inner end fitting. A pressure load was applied to the braid of 27,537 psi, representing a total tensile force of 1038.1 lbs. This force was found from equation 4.17, by utilizing the x values obtained through the axial contraction analysis, and solving for the tension, T , at the burst pressure of 175 psi.

$$T = \frac{B(PRL - \sigma_c t L)}{\pi N^2 D} \quad (4.17)$$

Hard contact between the braid and socket was utilized for the model. Once again, CAX4R elements were used to mesh both the braid and end fitting socket. The undeformed model can be seen in figure 4.22, with the deformed results and stresses given in figure 4.23. Here we see that as the braid is pulled due to the tension generated by the bladder expanding, the bottom of the socket is bent away from the PAM. The socket begins to yield due to this. Due to the brittle nature of the plastic, this yielding would propagate through the end fitting, resulting in failure and bursting. This type of yielding aligns with the failure observed in the PAM during the free contraction burst test.

4.12 Endurance Testing and Results

4.12.1 Methodology and Testing

Following successful completion of testing and analyses of both the ABS-R and ABS-CF PAMs, an endurance test was designed to determine how many inflation-deflation cycles the 3D

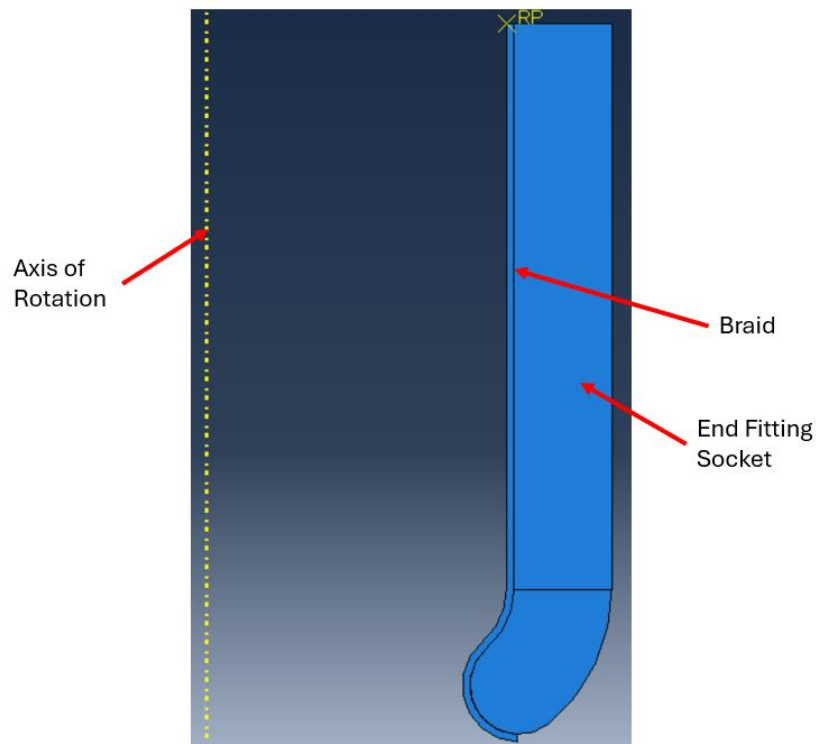


Figure 4.22: Undeformed model of free contraction burst.

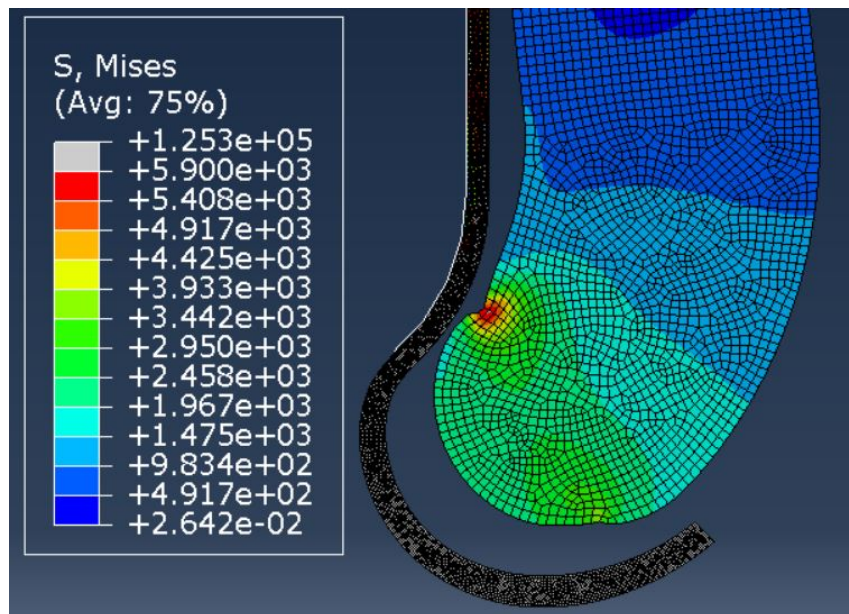


Figure 4.23: Deformed model of free contraction burst.

printed 0.16 inch fold-over PAM could survive before the end fittings failed. These tests were run to ensure that the actuators could be used for a reasonable number of cycles before failing, otherwise they would be impractical to implement in real world applications. This test was based on the fatigue testing performed by Woods et al. so as to provide a basis with which to compare the results to those of metallic end fitting PAMs. In the paper, the authors actuated 3 one inch diameter PAMs between 0 to about 5% contraction at 90 psi, resulting in a maximum force of about 340 lbs. Upon calculating the braid angle of the PAM used in the reference paper, we see that the braid angle is 55.5 degrees, which is considerably smaller than the braid angle of our PAM of 73.06 degrees, while having a larger diameter than our PAMs. To remedy this discrepancy, the PAM was chosen to run at 50 psi for the endurance test, to match the force output of the PAM from the reference paper. Note that due to discrepancies in the reference paper between target pressure and contraction versus actual pressure and contraction during testing, it was found difficult to match the force output exactly. To account for this, it was decided that the force output at 5% contraction of the 3D-printed PAM would match the blocked force output of the reference PAM. While this would result in the 3D-printed PAM being under greater loads during testing, it would ensure that the PAM would not be unfairly compared to the reference by being under-loaded during the duration of the test. The decision was made to match the blocked force as opposed to the operating pressure as the endurance test is being run from 0 to 5% contraction, and based on the burst test results, it has been found that the axial force production near blocked force has a greater impact on failure than the pressure, so the forces were matched to attempt to best keep the two tests comparable. Springs were sized to allow for 5% contraction of the PAM during inflation. Using the target 340 lbs of force, Hooke's law, given in equation 4.18, was used to size

4 springs to react the force produced by the PAM at 5 % contraction.

$$F = Ky \quad (4.18)$$

Here y is the displacement of the spring, and K is the combined spring constant of the 4 springs which were placed in parallel to provide a combined reaction force, such that $\{k_n = \frac{K}{4} | n = 1, \dots, 4\}$. k_n was equal to $188.8 \frac{\text{lbf}}{\text{in}}$. the closest off the shelf springs that could be found had a constant of $181.61 \frac{\text{lbf}}{\text{in}}$ for each spring, resulting in an actual total force of 326.9 lbf at 5% contraction.

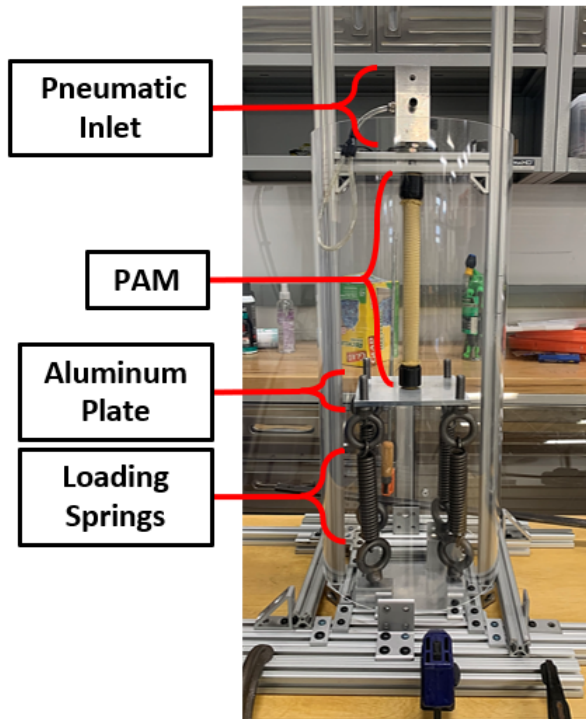


Figure 4.24: Endurance rig.

The completed endurance rig is shown in Figure 4.24. The PAM is connected to a metallic plate containing 4 eye hooks, to which the 4 springs are connected. The other end of the springs connect to eye hooks attached to the base of the rig. The top of the PAM is connected to a crossbar to hold the PAM in place. Air is fed into the PAM at the top of the rig using a modified version of

the pressure regulation system used in the actuation tests. Here, the system increases the pressure of the PAM by 10 psi every second, up until 50 psi, then drops the pressure to 0 and holds for 2 seconds. This process constitutes one cycle. The system monitors the pressure of the PAM, and stops the test if the pressure is recorded as 5 psi or more below the target pressure for over a second. This prevents the system from continuing to attempt to run in the case of a burst. The system runs until burst is detected or the target number of cycles is reached.

The test was run over the course of many days, typically targeting 2,500 cycles per day. The PAM underwent a full actuation sweep at 50 psi before beginning testing. For the first 10,000 cycles, the PAM also underwent a full actuation sweep every 2,500 cycles to determine if any change in the performance was observed. After completing 10,000 cycles, the PAM instead underwent a full actuation sweep every 5,000 cycles until 20,000 cycles were reached. Following 20,000 cycles, the PAM underwent a full actuation sweep every 10,000 cycles. Starting at 35,500 cycles, the time spent deflated for the PAM was changed from two seconds to one second, as the PAM was found capable of fully deflating in less than one second.

4.12.2 Results

As of time of writing, the 3D-printed end fitting PAM has undergone over 50,000 cycles. Complete failure in the end fittings has not yet been observed, although significant wear is present in the end fittings. As seen in Figure 4.25, the lower parts of one end fitting began to experience axial stress concentrations around 2,500 cycles, cracks formed at 5,000, and the lower parts completely broke off at 10,000 cycles. The opposite end fitting broke similarly, but later in the lifespan of the PAM. Stress concentrations began to develop at 5,000 cycles, cracks formed at

7,500, and the lower section completely broke off around 15,000 cycles. Although these pieces have broken off, the PAM has maintained functionality, and no loss in performance or leakage has been noted. Figure 4.26 shows the actuation diagram at 50 psi for the PAM at 0 cycles and 50,000 cycles. No significant change in performance is observed between these two tests. Testing will be continued until failure, at which point the 3D-printed PAM made from ABS-CF will also be tested.

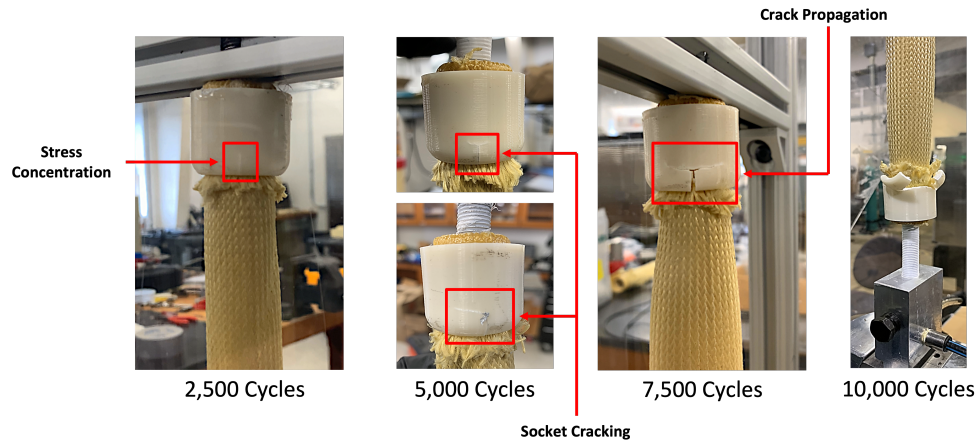


Figure 4.25: Progression of end fitting cracking during endurance testing.

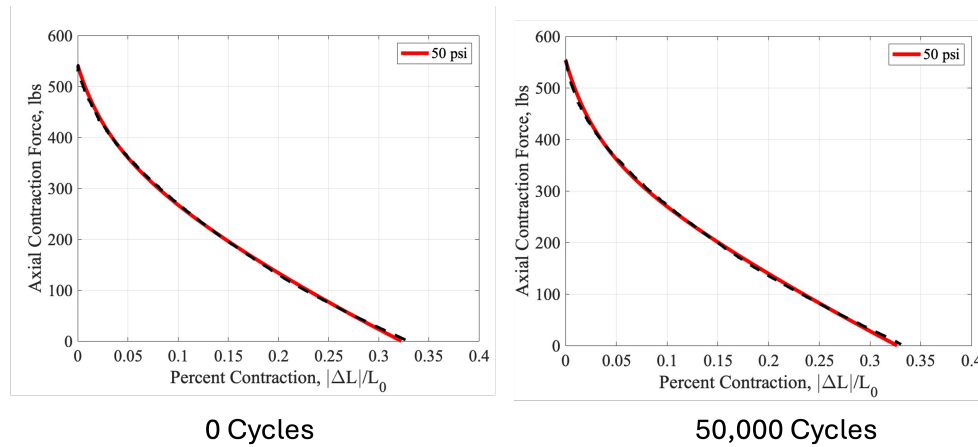


Figure 4.26: Actuation Diagrams at 0 and 50,000 cycles.

4.13 Conclusion

PAM specific actuation force was maximized in this study by reducing the weight of their end fittings. By replacing typical aluminum end fittings with 3D-printed ABS-R plastic, it can be shown that 3D-printed PAMs match the performance capabilities of metallic PAMs while maintaining the nominal axial contraction force seen in metallic PAMs. 3D-printed and metallic PAMs are compared by their specific actuation forces, using data from figure 4.17 and their total weights.

3D-printed PAMs were developed and tested in blocked force and free contraction, where multiple failure modes were examined. The delamination of the end fitting in iteration 1 caused a change in the infill density to strengthen the inner end fitting, and the print orientation was changed to allow the layer lines to sit in line with the axial load. Inner end fitting pullout was observed in iteration 2, where a retaining ring was added to hold the component in place. Thread failure occurred in iteration 3, where an increase in shells, a change in printing mode from balanced to solid, and an inner end fitting extension were incorporated to reduce damage to the threads and add more resistance to an inner end fitting pullout for iteration 4.

Further improved designs such as a thicker socket for the retaining ring design and a new fold-over design were considered. The 0.24 inch retaining ring design and the 0.16 inch and 0.24 inch fold-over designs were then vetted to decide which PAM iteration saw the greatest improvement to specific actuation force while maintaining a factor of safety of 1.5 for all burst tests. The 0.16 inch fold-over design was found to be the best choice.

The fold-over PAM design reached a maximum pressure of 207 psi in blocked force and 175 psi in free contraction, all while increasing the specific actuation force by 12.86%. Two

FEA models were developed to validate the failure modes seen in the block force and free contraction burst tests. Both models appeared to closely match the identified mode of failure for both scenarios at the same pressure as failure occurred for the physical PAM. An alternate version of the 0.16 inch fold-over design utilizing ABS-CF was tested and found to improve both blocked force and free contraction burst pressures with the trade off of reducing specific actuation force percent difference to 8.70%.

The ability of the PAM to withstand over 50,000 cycles at the same force and percent contraction as those used in ref [5] demonstrates the viability of these PAM to be used for short to medium term uses. Coupling this with their ability to be rapidly manufactured when compared to metallic PAMs, and having comparable actuation diagrams for similarly sized PAMs, 3D-printed PAMs could be used not just for lightweight applications, but also for rapid prototyping. They could be implemented in cases where many iterations of a PAM-actuated mechanism may need to be designed and tested for thousands of cycles before the final version is constructed, be that with plastic or metal.

Future work on 3D-printed PAMs includes completion of endurance testing for both the ABS-R and ABS-CF 3D-printed PAMs to determine the complete lifespan of both designs.

Chapter 5: Actuation of a 3D Printed Sweeping Wing Mechanism Using Pneumatic Artificial Muscles

5.1 Overview

Due to their small size, Unmanned Aerial Systems (UAS) possess the ability to fly through confined areas that full-sized aircraft would not be able to. This, however, presents the challenge of an increased risk of collisions with obstacles present in these areas. An active wing sweeping mechanism was proposed to both mitigate the possibility of, and damage associated with, collisions. The mechanism was actuated by a lightweight, 3D-printed pneumatic artificial muscle (PAM), and capable of sweeping the wings of an E-flite V900 BNF Basic remote controlled plane by 45 degrees. Multiple bench top tests were performed on the system, such as a simple actuation test, a response time test, and a hardware-in-the-loop (HITL) test. Additionally, blocked force and free contraction tests were ran on the 3D-printed PAM to determine safe operating pressures. The gear system was found to be fully capable of sweeping the wings from the default 35.4 inch wingspan to a 24 inch wingspan and clearing a 30 inch gap at speeds up to 55 mph in the HITL tests.

5.2 Introduction

Unmanned aerial systems (UAS) offer a lot of freedom and autonomy that is not possible with manned aircraft, such as being able to scale the aircraft down to fit confined spaces. Traveling through confined spaces presents unique challenges, such as the increased risk of collisions with obstacles. For fixed wing aircraft, a "go-around" style of collision avoidance is typically implemented. These methods consist of using some form of obstacle detection and performing an aerial maneuver to avoid the obstacle [91–97]. Sometimes collisions are unavoidable, however. In these circumstances, crash mitigation systems can be employed to prevent or minimize damage due to the collision, allowing the aircraft to continue normal operations post-collision. A sweeping mechanism was desired to allow the wings to be retracted before impact to either allow the aircraft to fit through a gap and avoid collision, or reduce damage inflicted on the wing by reducing the profile of the aircraft and allow for a glancing hit. Sweeping wing mechanisms have been studied before for various uses, such as for transitioning from low speed loitering operations to high speed pursuit maneuvers as well as for creating large pitching moments to perform perching maneuvers [7,98]. Post-collision, the wings were required to sweep back to their starting position for the remaining flight duration.

5.3 Design

The design of the sweeping wing mechanism can be split into two parts: the gear mechanism used to sweep the wings, and the actuation device used to actuate the gear system.

5.3.1 Gear Mechanism

The sweeping wing mechanism was designed for an E-flite V900 BNF Basic remote controlled plane. Bench top testing of the mechanism was desired before implementation into the aircraft was performed. A 3D scan was performed of both the wing and the fuselage to generate 3D models. These models were used as the basis for designing the mechanism and housing. To aid in manufacturing the irregular parts necessary for the mechanism and avoid complicated and costly machining, a majority of the system was made of soft components and 3D-printed plastics. Doing so also allowed for a weight reduction of the components when compared to constructing them from more traditional materials, such as aluminum. To construct the mechanism, a wing mount was designed to match the leading edge of the wing from the scan, as shown in figure 5.1. This mounting piece was then connected to a shaft with a one inch diameter gear at the end. This gear met with a quarter inch diameter gear, creating a 4:1 gear ratio. The quarter inch diameter gear then met with a linear gear. The linear gear was attached to the actuation system and spring. This gear system was mirrored for the opposite wing, making use of the same linear gear for both wings. The wings from the E-flite V900 were removed and augmented to fit on the wing holders. The aft section of the wing was cut at a 45 degree angle to allow the wings to sweep without impacting each other. When actuated, the linear gear would slide aft, turning the quarter inch diameter gear, which in turn rotated the one inch diameter gear and wing. Due to initial issues of the quarter inch diameter gear and the one inch diameter gear losing contact from the gear shaft flexing during actuation, a second quarter inch diameter gear was added on the opposite side of each one inch gear. This insured that one quarter inch diameter gear would always remain in contact during actuation. Additionally, when performing tests with the V900 wings attached to

the holders, sagging occurred that was not present in the the tests ran with just the holders. To mitigate this, a locking mechanism was implemented on the top of the wing shafts that prevented the wing holder from sagging. In addition to the locks, a cross bar was placed between the two locks to prevent any shaft bending that was occurring due to the wing weight. The reduction of sag can be seen in figure 5.2. The completed gear system, prior to the addition of the locking mechanism, is shown in figure 5.3 on the left, while the gear mechanism with added locking system is shown on the right. The combined weight of the gear system, spring, and housing is 1.04 lbs. Note that the housing for the bench top tests is not representative of a final, flight ready design, which would be optimized to reduce weight.



Figure 5.1: Left: Scan of v900 wing with highlighted section used for the wing holder. Right: 3D-printed wing holder with attached gear.

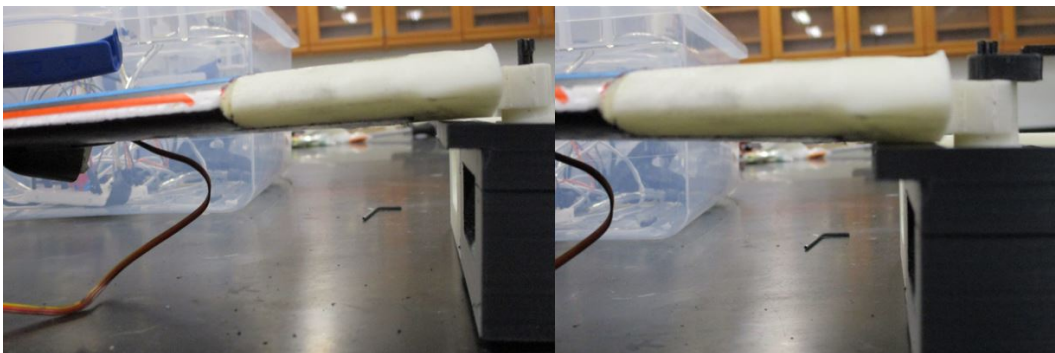


Figure 5.2: Left: Sagging wing Right: sag reduction using a locking mechanism.

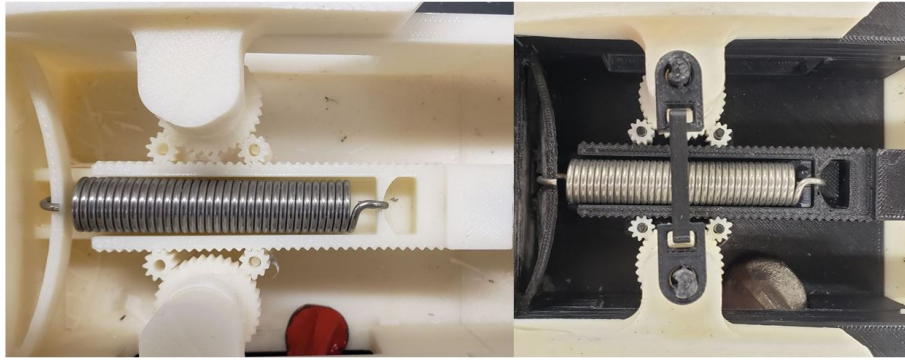


Figure 5.3: Left: Original 3D-printed gear system. Right: Gear system with locking mechanism modification.

5.3.2 Actuation Device

A 7/8 inch diameter pneumatic artificial muscle (PAM) was used for actuation, shown in figure 5.4. A PAM was chosen for this applications as PAMs possess the ability to generate large forces at low operating pressures while maintaining a low mass, allowing them to possess a large specific work, specific power, and power density, which makes them ideal for weight-limited applications such as this [5,52,53,63,65,66]. The bladder and braid of the PAM are held together by use of an end fitting. Typically end fittings are metallic, but 3D-printed end fittings, such as those discussed in chapter 4, were desired to reduce the weight of the system even further. The end fitting consists of two 3D-printed pieces, referred to as the inner and outer end fitting, respectively, as shown in figure 5.5. The end fittings were printed on a Makerbot Method X 3D printer using ABS-R material. The inner end fitting is inserted into the bladder, and the edge of the bladder is tucked under the lip of the end fitting. The outer end fitting is placed around the braid and bladder, and pulled over top of the inner end fitting. Once assembled, epoxy is injected into the end fitting through the holes in the inner end fitting to bind the end fitting together. The total mass of the assembled PAM used in the mechanism was 0.06 lbs.

Once assembled, the PAM is placed in line with the linear gear. When inflated, the PAM pulls the gear to retract the wings. A small metallic spring is placed on the opposite end of the gear, which aids to pull the gear back into its original position when the PAM is deflated.

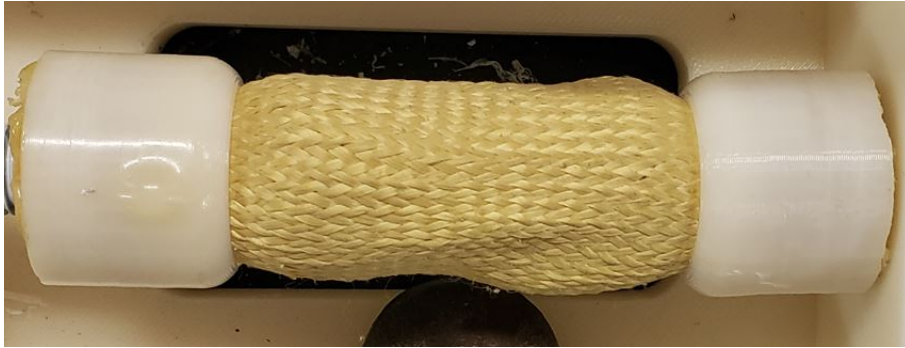


Figure 5.4: 7/8 inch diameter PAM with 3D-printed end fittings.

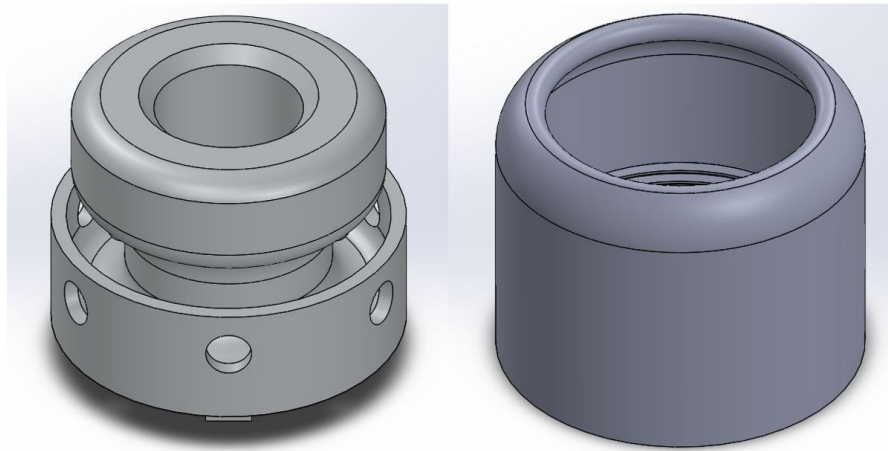


Figure 5.5: Left: Inner end fitting. Right: Outer end fitting.

5.4 Testing

To test the device, a tabletop test rig was designed to house the gears, spring, and PAM, as shown in figure 5.6. An Arduino was used in conjunction with two X-Valve 8mm miniature solenoid valves and a Honeywell SSCDANN150PGAA5 pressure sensor, shown in figure 5.7.

The pneumatic system was provided air from a DEWALT portable air compressor. Using this testing system, multiple tests were conducted to evaluate the mechanism.

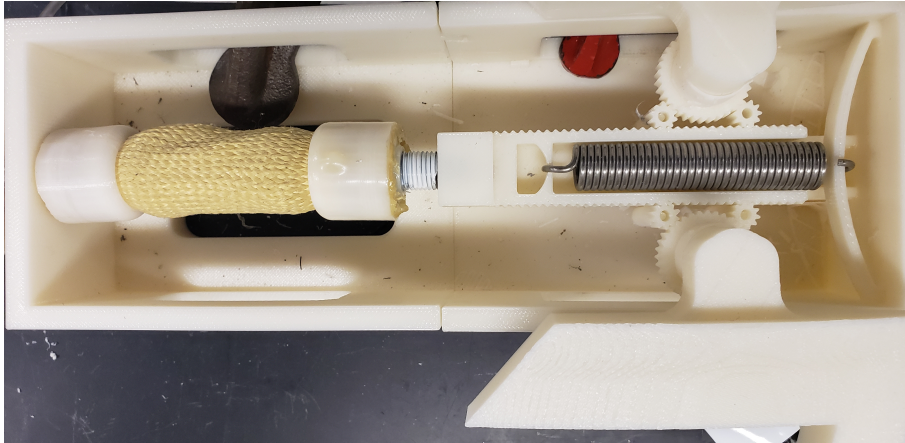


Figure 5.6: Assembled gear system with PAM.

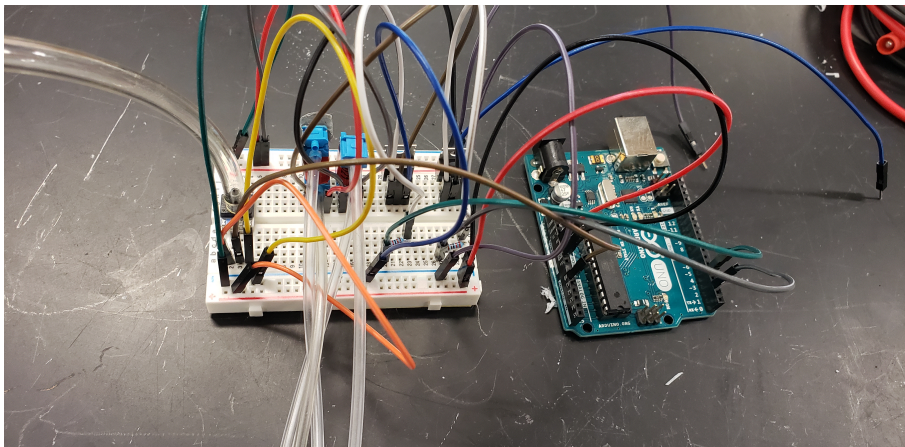


Figure 5.7: Electro-pneumatic system used to actuate the PAM.

5.4.1 Simple Actuation

At test start, the inlet valve was opened, allowing the PAM to be inflated, which in turned actuated the muscle, as shown in figure 5.8. Upon hitting the target pressure of 30 psi, the inlet valve was closed and pressure was held for two seconds. Following the two second wait, the outlet valve was opened, allowing the muscle to deflate.

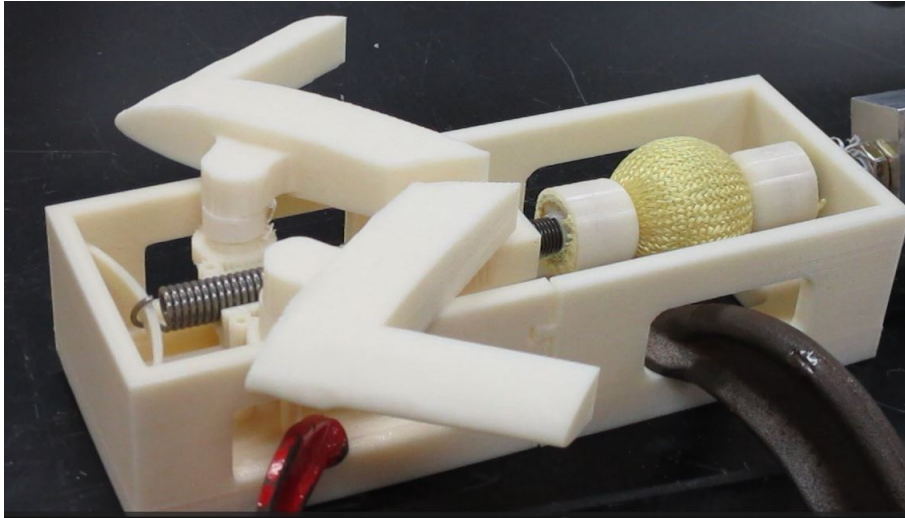


Figure 5.8: Fully actuated system with inflated PAM.

5.4.2 Response Time Testing

Four checkered strips were placed underneath the system to measure the sweep of the wing, shown in figure 5.9. The strips were placed in 15 degree intervals, ranging from 0 to 45 degrees. Two tests were run: One with only the wing holders, and one with the wing holders and augmented v900 wings. For the wing holder test, the line was pressurized from 55 to 85 psi, in increments of 5 psi. For each pressure, the inlet valve was opened and the response time to inflate to 30 psi and sweep the wing 45 degrees was recorded. The inlet valve was then closed, and the outlet valve was opened. Four sweeps were performed per pressure. The response time for each sweep was recorded and averaged per pressure.

The test was repeated using the same parameters, but with the v900 wings attached to the wing holders. Four sweeps were performed per pressure, and the response time was once again recorded and averaged per pressure.

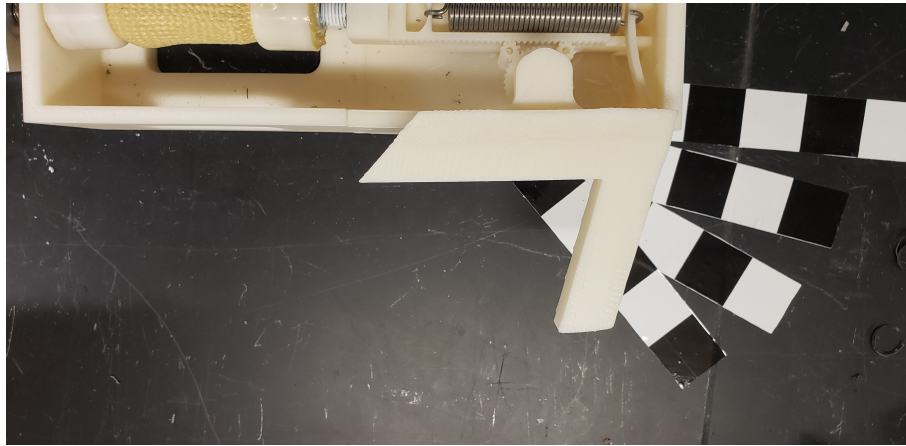


Figure 5.9: Sweep test set-up.

5.4.3 Hardware-in-the-loop Testing

Hardware-in-the-loop (HITL) testing was performed to simulate the V900 flying through gap-like structures and test the ability of the obstacle avoidance software and actuation system to work together. The actuation system was integrated into the obstacle avoidance software through a signal line sent from the software to the arduino controlling the actuation system. In these tests, prerecorded footage of simulated flight toward an obstacle was fed into the ground control setup that ran the obstacle detection software. The software would send a high signal when an obstacle was detected, signaling the PAM to inflate. When detection was lost, the signal was dropped to low, and the PAM was deflated. The obstacle detection software identified obstacles represented by ArUco tags, as shown in the upper-right image in figure 5.10. The ArUco tag was 30 inches by 30 inches in size and represented an obstacle with a gap size of 30 inches. The software was capable of detecting a tag of this size at a minimum distance of 44 feet away. It was assumed that there was no altitude loss and that the V900 flew straight and level toward the obstacles during simulated flight. Tests were run at 85 psi line pressure for a simulated flight speed of 10, 20, 30, 40, 45, 50, 55, 60, 65, 70, 80, 90, and 100 mph. Video was taken of the wing sweeping to record

how much time passed between achieving full sweep and passing the simulated obstacle, as well as the total wing sweep achieved, to determine the smallest gap size the v900 could fly through based on effective wingspan.



Figure 5.10: Left: No object detected in (A), resulting in no sweeping in (B). Right: Object (ArUco tag) detected in (C), resulting in wing sweep in (D).

5.5 Results

5.5.1 Simple Actuation

The mechanism was capable of successfully generating 45 degrees of sweep in the wing while being almost entirely 3D-printed. No appreciable skipping or misalignment was noticed during the actuation process in any of the gears for the final design. The 3D-printed end fittings were capable of surviving both the radial and axial stresses imparted during inflation, while still allowing the PAM to actuate the mechanism. The simple actuation test was performed 10 times

with no noticeable wear on the mechanism or the end fittings.

5.5.2 Response Time Testing

The results for the sweep test with only the wing holders attached are shown in table 5.1. It can be seen from figure 5.11 that the response time decreases as the line pressure increases. The decrease in response time is not linear and begins to asymptote at higher pressures. This implies that any further increase in line pressure would result in diminishing returns.

For the sweep tests with wings attached, we see that the times once again asymptote as you increase pressure, but that in general the sweep time is greater for the test with wings attached when compared to those with only the holders attached. This is to be expected, as the added inertia of the wings works to slow the sweep times. This is important to keep in mind for future designs where additional improvements to the wings may be made that would increase the mass of the wing, such as wrapping them in carbon fiber to supply additional support during a wing strike. The results of this test can be seen in table 5.2.

Due to the asymptotic nature of these results, 85 psi was chosen as the line pressure for the hardware-in-the-loop tests. This aligns with a time to rise of 1.36 seconds.

5.5.3 Hardware-in-the-Loop testing

From prior true flight tests of the V900, it was determined that the UAS needed to fly at airspeeds above 30 mph to stay in the air. Although data from the HITL tests showed that the V900 achieves full sweep before reaching the obstacle at speeds of 10-40 mph, only airspeeds above 30 mph are viable. From this, we determined that the V900 can make a full wing sweep,

Table 5.1: Actuation times for sweeping with wing holders.

Pressure	Test	Test	Test	Test	Average
	1	2	3	4	
[psi]	[seconds]	[seconds]	[seconds]	[seconds]	[seconds]
55	2.11	2.39	2.13	2.13	2.19
60	1.83	1.76	2.08	2.12	1.95
65	1.69	1.68	1.80	1.79	1.74
70	1.68	1.52	1.49	1.47	1.54
75	1.19	1.46	1.51	1.48	1.39
80	1.22	1.20	1.51	1.19	1.28
85	1.22	1.19	1.2	1.16	1.20

Table 5.2: Actuation times for sweeping with wings.

Pressure	Test	Test	Test	Test	Average
	1	2	3	4	
[psi]	[seconds]	[seconds]	[seconds]	[seconds]	[seconds]
55	2.75	2.82	2.94	2.9	2.85
60	2.35	2.35	2.42	2.49	2.40
65	2.16	2.1	2.16	1.96	2.10
70	1.77	1.83	1.9	1.83	1.83
75	1.63	1.7	1.64	1.71	1.66
80	1.38	1.5	1.5	1.51	1.47
85	1.37	1.37	1.31	1.38	1.36

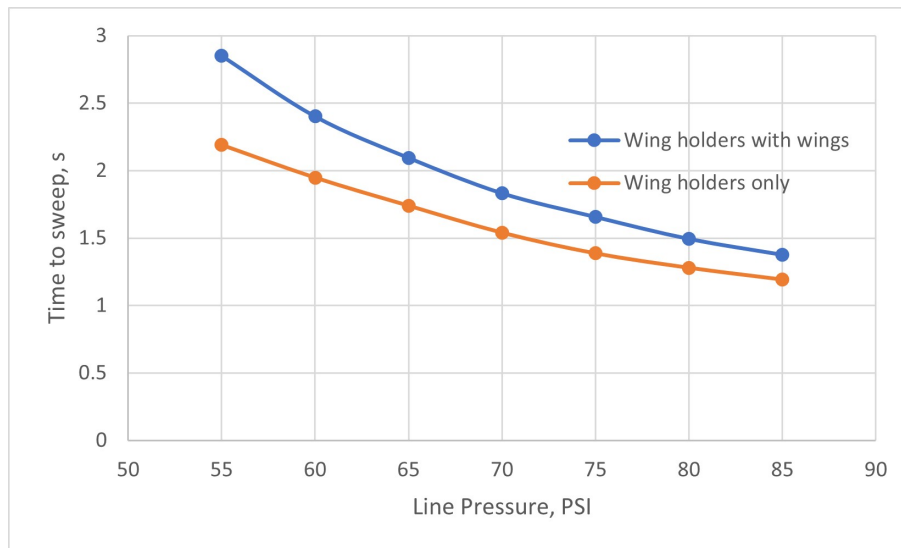


Figure 5.11: Wing sweep time versus line pressure.

and thus clear obstacles with widths down to 24 inches, at speeds of 30-40 mph. From 40-55 mph, the V900 clears obstacles with widths down to 30 inches, which corresponded to the size of the obstacles in the simulated flight footage. At speeds above 55 mph, the V900 could no longer sweep its wings far enough back in time to make a successful pass through the simulated obstacle, with its effective wingspan approaching the original wingspan at speeds over 100 mph. Additional testing could be performed to study the system's ability to clear obstacle gaps if given more time to retract by using larger ArUco tags representing larger obstacles.

Figure 5.12 portrays the minimum obstacle width necessary to complete a "go-through" maneuver for the given testing conditions. Blue dots represent speeds at which the V900 would not be able to maintain flight. Green dots represent speeds where full sweep was achieved before reaching the object. Yellow dots represent speeds where full sweep was not achieved, but the aircraft would still be able to reduce its wingspan below the 30 inch gap clearance of our simulated object. The red dots represent speeds where the aircraft would not clear the object, and would have to perform either a "go-around" maneuver or a wing strike.

The HITL tests suggest that the platform can perform go-through maneuvers at speeds up to 55 mph, which is considerably faster than the top speeds tested in prior work [95]. However, the obstacle detection and avoidance algorithms used were greatly simplified, meaning that the data would only be valid under ideal conditions.

5.6 In-flight testing and weight budget

With benchtop testing completed, the next goal was to equip a V900 with the sweeping wing mechanism and obstacle recognition/crash avoidance system. Upon attempting flights with

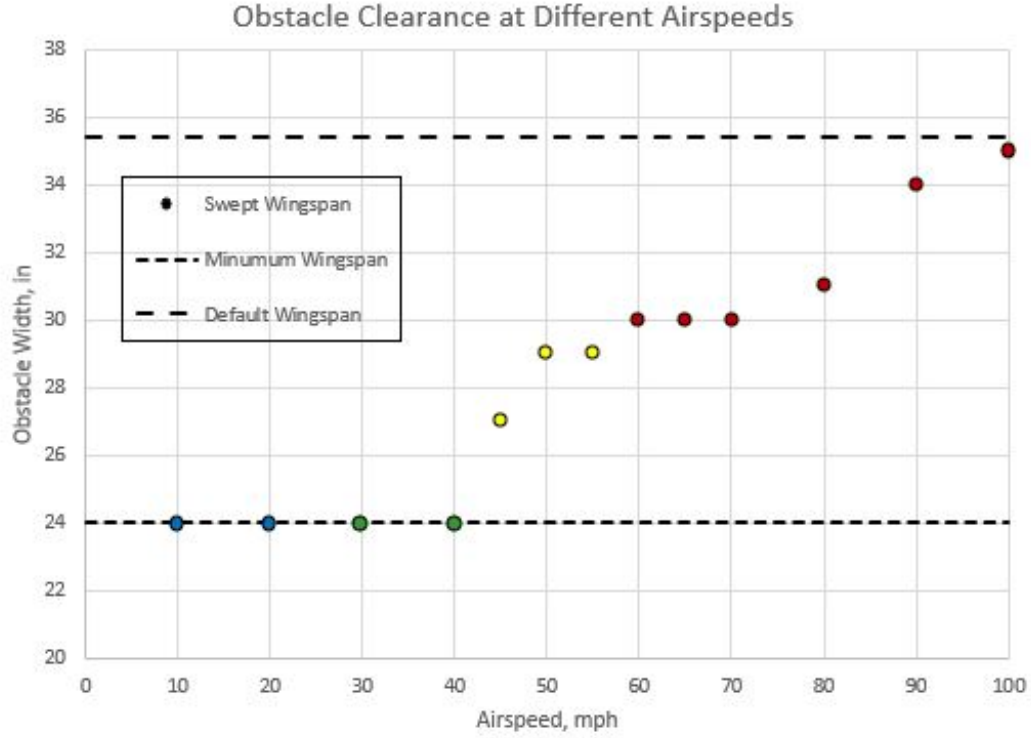


Figure 5.12: Minimum achievable wingspan before collision versus airspeed.

only the obstacle recognition system on board, it was found that the v900 was incapable of producing enough lift to take off. Lift calculations were then performed to determine why this issue was occurring. The calculation for lift is given in equation 5.1.

$$L_A = \frac{C_l \rho v^2 A}{2} \quad (5.1)$$

Here, C_l is the coefficient of lift, ρ is the density of the air, v is the speed of the craft, and A is the area of the wing. To obtain the coefficient of lift, the model was assumed to be a Clark Y, as it shared a very similar cross-section, and the Clark Y is a typical choice for RC plane airfoils. This results in a total lift generation of 2.91 lbs at 5 degrees angle of attack for a speed of 30 mph, which is typically the speed that lift off would occur for the unaltered V900. The weight

of an unaltered V900 is about 2.80 lbs. This would provide a margin of only .11 lbs for payload. The obstacle recognition system weighed 0.52 lbs and therefore the V900 was unable to provide sufficient lift for take-off. The sweeping wing system and electro-pneumatic system, as currently constructed, weigh an additional 3.32 lbs. Takeoff would not be feasible with the V900 even at higher angle of attacks, as the stall angle, about 10 to 12 degrees, would be reached before enough lift could be generated to achieve takeoff. Due to these issues, in flight testing was not performed.

5.7 Conclusion and Future Work

These bench top tests form a basis for the design of a fully 3D-printed sweeping wing mechanism that would be capable of both aiding in obstacle avoidance and mitigating damage to an aircraft during wing strikes. The 3D-printed PAM was capable of providing reliable, quick actuation to the mechanism, and the mechanism demonstrated repeated wing sweeps without damage or wear to the wings or gears. Such a device could increase survivability for UAVs flying through confined spaces, such as urban areas or dense tree lines, where wing strikes may be unavoidable.

Future work would entail repurposing the mechanism for a different platform capable of generating sufficient lift. One such platform is the Turbo Timber Evolution 1.5m BNF Basic. Using equation 5.1 to calculate lift, and once again assuming a Clark Y air foil, the Timber would be capable of lift off at 25 mph at 3 degrees angle of attack. The Timber, with battery, weighs 4.83 lbs. The obstacle recognition system and current sweeping wing and electro-pneumatic system weigh 3.83 lbs, for a total of 8.66 lbs. Total lift generated by the Timber would be 9.62

lbs, making it a cromulent choice to carry the combined systems. Although redesigns would need to occur to fit the sweeping wing mechanism into the Timber, the internal space available in the Timber would remove the need for a 3D-printed external housing for the sweeping wing mechanism and electro-pneumatic system, which was necessary for the v900 due to it having no available internal space, possibly reducing the overall weight of the system, as it could be embedded directly into the aircraft. Once redesigns were complete, in flight testing of the system would be performed and compared to the benchtop data.

Chapter 6: Conclusion

6.1 Summary of Research and Key Conclusions

This research explores alternative uses for PAMs, such as the application of radial forces, torsional response, and weight-restricted applications. PAMs were modeled using a force balance approach to accurately predict the radial force production for use in a bio-inspired, worm-like robot. Additionally, the mode by which PAMs respond to torsion was determined and modeled using a force balance approach. This model allowed for accurate determination of the torsional load a PAM would experience at a given pressure and degree of twist. 3D-printed end fittings were designed and tested to improve SAF of PAMs, while maintaining the same typical operation range of 0-100 psi seen in metallic end fitting PAMs. These 3D-printed PAMs were utilized in a sweeping wing mechanism for collision avoidance and found capable of avoiding collision at speeds up to 55 mph in bench top tests.

6.1.1 Analysis of anchoring muscles for pipe crawling robots

Modeling of the radial expansion force was performed to produce a model of comparable fidelity to the established axial contraction force model. The model was validated with data collected by inflating the PAM radially into two half rounds, and recording the resulting force

using an MTS servo-hydraulic testing machine. Non-linear optimization was performed to determine the moduli of the PAM bladder while anchored in a pipe. The radial model was found to fit the data well. The model was also used to determine anchoring force of a PAM experiencing an applied axial force and compared to data. The model was able to predict the offset in anchoring pressure and force production experienced by the PAM due to the applied axial force through the addition of a single term comprised of only the applied axial force and geometric properties of the PAM. Compatibility between the axial and radial models were found at anchoring pressure, allowing for a transition point when used in operation with a bio-inspired, worm-like robot. The robot was able to produce locomotion using a peristaltic gait through pipes without slipping.

6.1.2 Analysis of Torsional Response in Pneumatic Artificial Muscles

In addition to the radial model, a model of the torsional response of a PAM was also developed. Three different scenarios were investigated, including St. Venant torsion, soft bladder buckling, and asymmetrical braid loading. A novel torsional testing rig was designed and constructed to validate the torsional model. Data was collected from 10 to 50 psi in increments of 5 psi, and found to be linear for all pressures. A force balance approach was taken to model the torsional response. Multiple stress-strain and modulus-pressure relationships were tested to improve the fit of the model. A version of the model using St. Venant torsion, as well as a first order stress-strain relationship and fourth order modulus-pressure relationship was found to provide the best fit.

6.1.3 Optimizing Specific Actuation Force of Soft Composite Pneumatic Artificial Muscles Using Additively Manufactured Components

3D-printed end fittings were designed as an alternative to current metallic end fittings to attempt to maximize the specific actuation force of the PAMs, while maintaining comparable actuation performance throughout the typical actuation range of 10 to 100 psi. Multiple iterations of a 7/8 inch PAM utilizing 3D-printed end fittings comprised of ABS-R were designed and tested. Alterations were made to each iteration based on the observed failure mode to achieve the target operating range of 0 to 100 psi with a safety factor of 1.5. The final design consisted of a PAM whose braid folds back in on itself to keep the end fitting in place, referred to as a 'fold-over' PAM. This PAM was able to achieve a blocked force pressure of 207 psi and blocked force of 2017 lbs, as well as a free contraction pressure of over 300 psi. SAF between the 3D-printed PAM and a similar metallic PAM was compared. The 3D-printed PAM was found to improve SAF by over 12%. An FEA model of both the free contraction and blocked force burst conditions were created to validate the observed physical results. An alternate version of the fold-over PAM was created using ABS-CF. This PAM saw an improved blocked force of 2508 lbs at 248 psi, with a slight reduction in SAF improvement over metallic PAMs of 8.7%. An endurance test was designed to determine the lifespan of a 3D-printed PAM. The fold-over design utilizing ABS-R achieved 50,000 cycles without failure.

6.1.4 Actuation of a 3D Printed Sweeping Wing Mechanism Using Pneumatic Artificial Muscles

To further validate the design of the 3D-printed end fittings, a PAM using the 3D-printed end fittings was utilized in a sweeping wing mechanism designed for an E-flite V900 BNF Basic remote controlled plane to perform obstacle avoidance. The mechanism utilized a 3D-printed gear mechanism in conjunction with the 3D-printed PAM and an electro-pneumatic system to produce sweep in the wings. The mechanism was capable of producing 45 degrees of sweep in 1.36 seconds in bench top testing. HITL testing was performed, in which the mechanism was found capable of clearing a 30 inch wide gap at speeds of 55 mph. Although the V900 was unable to fly the mechanism due to weight constraints, calculations were performed to find a platform that could house the mechanism, such as the Turbo Timber Evolution 1.5m BNF Basic.

6.2 Contributions to Literature

This work has advanced the usability of PAMs outside of the traditional axial applications by producing accurate models of both the radial expansion force and torsional reaction, allowing for a much broader range of applications for this type of actuator. Previous modeling of PAMs contains no terms for radial or torsional force applications, and as such is unable to account for them. The radial expansion model is the first to accurately predict the force produced by a PAM when constrained in a tube or pipe, and contains an applied axial force term, allowing for the possibility of analyzing PAMs experiencing loads in multiple directions simultaneously. The radial expansion model also allows for compatibility between itself and axial contraction

models found in literature, allowing for smooth transition between models for improved control in robotic systems, such as pipe crawling robots.

Similarly, the torsional response model is the first to be able to predict the torsional response of a PAM, allowing for torsional applications of PAMs that were not previously feasible. In addition to this model being able to mathematically predict the torsional response load of PAMs, it was also the first to determine how PAMs physically respond to torsion, in that they react in a manner similar to St. Venant torsion, as opposed to other torsional response modes, such as experiencing bladder buckling or asymmetric braid loading.

3D-printed end fittings have also advanced the usability of PAMs by improving their specific axial actuation force for weight-limited applications by utilizing 3D-printed plastic, as opposed to metallic end fittings currently used in PAMs, while maintaining their typical pressure and force operation range. In addition, these PAMs were found to survive for over 50,000 cycles, making them useful for prototyping applications, as multiple sets of end fittings can be printed faster than metallic end fittings can be manufactured, while not sacrificing performance. With these improvements, PAMs are now capable of being utilized in a much broader range of applications than possible prior to this work.

6.3 Future Work

Future work entails utilizing the radial expansion and torsional response model in tandem, along with the axial contraction model, for robotic applications. Although the the radial expansion model and axial contraction model have been used together in the worm-like robot, the inclusion of the torsional response model would allow for the addition of a digging tool, such as an

auger, which would allow for the worm-like robot to burrow through sand and soil, improving its functionality past just pipes and tubes. Additional work on the torsional model could be performed to not only study higher degrees of twist and pressure, but also the response while the PAM is anchored. This change in torsional response while anchored would also be beneficial in robots such as the worm-like robot, as it would allow for a fuller understanding of how the two different forms of loading effect the PAM when applied simultaneously.

Future work for the 3D-printed end fittings entail further study of the endurance of the end fittings, to determine at how many cycles they fail. Additionally, further optimization using a combination of FEA and physical testing could be performed to increase the gain in SAF even further, whether through reduction in socket thickness to reduce total weight, or with experimentation with alternative end fitting designs outside of the ones tested in this work. Finally, future work for the sweeping wing mechanism would entail refitting the mechanism for a platform capable of providing enough lift to carry it, and performing in-flight testing to validate the results found during the benchtop testing.

Bibliography

- [1] Lei Yan, Wenfu Xu, Zhonghua Hu, and Bin Liang. Multi-objective configuration optimization for coordinated capture of dual-arm space robot. *Acta Astronautica*, 167:189–200, 2020.
- [2] Nicholas D. Naclerio, Andras Karsai, Mason Murray-Cooper, Yasemin Ozkan-Aydin, Enes Aydin, Daniel I. Goldman, and Elliot W. Hawkes. Controlling subterranean forces enables a fast, steerable, burrowing soft robot. *Science Robotics*, 6(55):eabe2922, 2021.
- [3] Bertrand Tondou. Modelling of the McKibben artificial muscle: A review. *Journal of Intelligent Material Systems and Structures*, 23(3):225–253, 2012.
- [4] Jonathan M Chambers and Norman M Wereley. Photogrammetric measurement and analysis of the shape profile of pneumatic artificial muscles. *Actuators*, 10(4):72, Apr 2021.
- [5] Benjamin KS Woods, Michael F Gentry, Curt S Kothera, and Norman M Wereley. Fatigue life testing of swaged pneumatic artificial muscles as actuators for aerospace applications. *Journal of Intelligent Material Systems and Structures*, 23(3):327–343, 2012.
- [6] Arief P Tjahyono, Kean C Aw, Harish Devaraj, Surendra Wisnu, Enrico Haemmerle, and Jadranka Travas-Sejdic. A five-fingered hand exoskeleton driven by pneumatic artificial muscles with novel polypyrrole sensors. *The Industrial Robot*, 2013.
- [7] Pascal de Marmier and Norman Wereley. Control of sweep using pneumatic actuators to morph wings of small scale uavs. In *44th AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics, and Materials Conference*, 2003.
- [8] Curt S Kothera and Norman M Wereley. Wind tunnel testing of pneumatic artificial muscles for control surface actuation. *Wind Tunnels*, pages 105–20, 2011.
- [9] George M. Whitesides. Soft robotics. *Angewandte Chemie International Edition*, 57(16):4258–4273, 2018.
- [10] James Pierce, Keith Needham, Christopher Adams, Andrea Coppolecchia, and Carlos Lavernia. Robotic arm-assisted knee surgery: an economic analysis. *Am J Manag Care*, 26(7):e205–e210, 2020.

- [11] William S Barbosa, Mariana M Gioia, Veronica G Natividade, Renan FF Wanderley, Marcelo R Chaves, Felipe C Gouvea, and Flavia M Gonçalves. Industry 4.0: examples of the use of the robotic arm for digital manufacturing processes. *International Journal on Interactive Design and Manufacturing (IJIDeM)*, 14:1569–1575, 2020.
- [12] Luca Lattanzi, Cristina Cristalli, Daniele Massa, Sébastien Boria, Pierre Lépine, and Marcello Pellicciari. Geometrical calibration of a 6-axis robotic arm for high accuracy manufacturing task. *The International Journal of Advanced Manufacturing Technology*, 111(7):1813–1829, 2020.
- [13] Yogesh Pratap Singh, Nazeer Ahmad, and Ashitava Ghosal. Dynamically isotropic gough–stewart platform for micro-vibration isolation in spacecrafts. *Mechanism and Machine Theory*, 201:105735, 2024.
- [14] Said G. Khan, Guido Herrmann, Mubarak Al Grafi, Tony Pipe, and Chris Melhuish. Compliance control and human–robot interaction: Part 1 — survey. *International Journal of Humanoid Robotics*, 11(03):1430001, 2014.
- [15] Bjoern Matthias, Susanne Oberer-Treitz, Harald Staab, Erich Schuller, and Steffen Peldschus. Injury risk quantification for industrial robots in collaborative operation with humans. In *ISR 2010 (41st International Symposium on Robotics) and ROBOTIK 2010 (6th German Conference on Robotics)*, pages 1–6, 2010.
- [16] Frank Daerden, Dirk Lefeber, et al. Pneumatic artificial muscles: actuators for robotics and automation. *European journal of mechanical and environmental engineering*, 47(1):11–21, 2002.
- [17] Angus Benedict Clark and Nicolas Rojas. Malleable robots: Reconfigurable robotic arms with continuum links of variable stiffness. *IEEE Transactions on Robotics*, 38(6):3832–3849, 2022.
- [18] Yu-Chen Sun, Meysam Effati, Hani E. Naguib, and Goldie Nejat. Softsar: The new softer side of socially assistive robots—soft robotics with social human–robot interaction skills. *Sensors*, 23(1), 2023.
- [19] Suwoong Lee, Hidetaka Nozawa, Jae-Il Kim, and Kenji Inoue. Fundamental experiments on an air-cushion-based force-sensing device for soft physical human–robot interaction. *IEEE Transactions on Electrical and Electronic Engineering*, 8(5):529–534, 2013.
- [20] C. Jarrett and A. J. McDaid. Robust control of a cable-driven soft exoskeleton joint for intrinsic human-robot interaction. *IEEE Transactions on Neural Systems and Rehabilitation Engineering*, 25(7):976–986, 2017.
- [21] Subyeong Ku, Byung-Hyun Song, Taejun Park, Younghoon Lee, and Yong-Lae Park. Soft modularized robotic arm for safe human–robot interaction based on visual and proprioceptive feedback. *The International Journal of Robotics Research*, 43(8):1128–1150, 2024.

- [22] Panagiotis Polygerinos, Nikolaus Correll, Stephen A. Morin, Bobak Mosadegh, Cagdas D. Onal, Kirstin Petersen, Matteo Cianchetti, Michael T. Tolley, and Robert F. Shepherd. Soft robotics: Review of fluid-driven intrinsically soft devices; manufacturing, sensing, control, and applications in human-robot interaction. *Advanced Engineering Materials*, 19(12):1700016, 2017.
- [23] Christopher van Vlerken, Felipe Ballen-Moreno, Ellen Roels, Pasquale Ferrentino, Kevin Langlois, Bram Vanderborght, and Tom Verstraten. Development of an active physical interface for physical human-robot interaction: Investigation of soft pneumatic actuator straps for automatic enclosure system. *Actuators*, 12(6), 2023.
- [24] Runhuai Yang, Miao Jin, Minmin Jin, Haisheng Qian, Qian Gao, Guoqing Jin, and Shiwu Zhang. Reprogrammable untethered actuator for soft bio-inspired robots. *Advanced Intelligent Systems*, 3(2):2000146, 2021.
- [25] Haiyan Jia, Erik Mailand, Jiangtao Zhou, Zhangjun Huang, Giovanni Dietler, John M. Kolinski, Xinling Wang, and Mahmut Selman Sakar. Universal soft robotic microgripper. *Small*, 15(4):1803870, 2019.
- [26] Qiuxuan Wu, Xiaochen Yang, Yan Wu, Zhijun Zhou, Jian Wang, Botao Zhang, Yanbin Luo, Sergey A Chepinskiy, and Anton A Zhilenkov. A novel underwater bipedal walking soft robot bio-inspired by the coconut octopus. *Bioinspiration Biomimetics*, 16(4):046007, jun 2021.
- [27] Zilong Peng, Hao Zhang, Mengjie Wang, Shuailong Zhang, Yifan Jiang, Yinan Li, Xiaoyang Zhu, Guangming Zhang, Geng Niu, Jia Zhang, Jiankang He, and Hongbo Lan. 3d printing bio-inspired micro soft robot with programming magnetic elastic composites. *Advanced Materials Technologies*, 9(9):2301777, 2024.
- [28] Faheem Ahmed, Muhammad Waqas, Bushra Jawed, Afaque Manzoor Soomro, Suresh Kumar, Ashraf Hina, Umair Khan, Kyung Hwan Kim, and Kyung Hyun Choi. Decade of bio-inspired soft robots: a review. *Smart Materials and Structures*, 31(7):073002, jun 2022.
- [29] Ryman Hashem, Shahab Kazemi, Martin Stommel, Leo K. Cheng, and Weiliang Xu. Sorss: A soft robot for bio-mimicking stomach anatomy and motility. *Soft Robotics*, 10(3):504–516, 2023. PMID: 36346277.
- [30] Korkut Eken, Nick Gravish, and Michael T. Tolley. Continuous skin eversion enables an untethered soft robot to burrow in granular media. In *2023 IEEE International Conference on Soft Robotics (RoboSoft)*, pages 1–6, 2023.
- [31] Joey Z Ge, Ariel A Calderón, Longlong Chang, and Néstor O Pérez-Arancibia. An earthworm-inspired friction-controlled soft robot capable of bidirectional locomotion*. *Bioinspiration Biomimetics*, 14(3):036004, feb 2019.
- [32] Xiao Yang, Yuanhao Chen, Xuan Zhang, Pan Xue, Pengfei Lv, Yanzhao Yang, Ling Wang, and Wei Feng. Bioinspired light-fueled water-walking soft robots based on liquid crystal

- network actuators with polymerizable miniaturized gold nanorods. *Nano Today*, 43:101419, 2022.
- [33] M Cianchetti, M Calisti, L Margheri, M Kuba, and C Laschi. Bioinspired locomotion and grasping in water: the soft eight-arm octopus robot. *Bioinspiration Biomimetics*, 10(3):035003, may 2015.
 - [34] Tianqi Yue, Hermes Bloomfield-Gadêlha, and Jonathan Rossiter. Snail-inspired water-enhanced soft sliding suction for climbing robots. *Nature Communications*, 15(1):4038, 2024.
 - [35] Mingxin Wu, Xiaoxian Xu, Qin Zhao, Waqar Hussain Afridi, Ningzhe Hou, Rahdar Hussain Afridi, Xingwen Zheng, Chen Wang, and Guangming Xie. A fully 3d-printed tortoise-inspired soft robot with terrains-adaptive and amphibious landing capabilities. *Advanced Materials Technologies*, 7(12):2200536, 2022.
 - [36] Erica G Hocking and Norman M Wereley. Analysis of nonlinear elastic behavior in miniature pneumatic artificial muscles. *Smart Materials and Structures*, 22(1):014016, dec 2012.
 - [37] Mikołaj Rogóż, Hao Zeng, Chen Xuan, Diederik Sybolt Wiersma, and Piotr Wasylczyk. Light-driven soft robot mimics caterpillar locomotion in natural scale. *Advanced Optical Materials*, 4(11):1689–1694, 2016.
 - [38] Wei Xiao, Dean Hu, Weixiong Chen, Gang Yang, and Xu Han. A New Type of Soft Pneumatic Torsional Actuator With Helical Chambers for Flexible Machines. *Journal of Mechanisms and Robotics*, 13(1):011003, 08 2020.
 - [39] Wei Xiao, Xuenan Du, Weixiong Chen, Gang Yang, Dean Hu, and Xu Han. Cooperative collapse of helical structure enables the actuation of twisting pneumatic artificial muscle. *International Journal of Mechanical Sciences*, 201:106483, 2021.
 - [40] W. Liu and C. R. Rahn. Fiber-reinforced membrane models of mckibben actuators. *Journal of Applied Mechanics*, 70(6):853–859, 01 2004.
 - [41] Jacek Garbulinski, Sai C. Balasankula, and Norman M. Wereley. Characterization and analysis of extensile fluidic artificial muscles. *Actuators*, 10(2), 2021.
 - [42] Thomas E Pillsbury, Norman M Wereley, and Qinghua Guan. Comparison of contractile and extensile pneumatic artificial muscles. *Smart Materials and Structures*, 26(9):095034, aug 2017.
 - [43] Jacek Garbulinski and Norman M. Wereley. Bending properties of an extensile fluidic artificial muscle. *Frontiers in Robotics and AI*, 9, 2022.
 - [44] Festo AG. Fluidic muscle mas. *Technical document trend*, 11, 2001.
 - [45] Sam Greenhill. The digit muscle. *Industrial Robot: An International Journal*, 20(5):29–30, 1993.

- [46] Andrew Nuss. Underminer, 07 2022.
- [47] Steve Davis, N. Tsagarakis, J. Canderle, and Darwin G. Caldwell. Enhanced modelling and performance in braided pneumatic muscle actuators. *The International Journal of Robotics Research*, 22(3-4):213–227, 2003.
- [48] Gang Wang, Thomas E Pillsbury, and Norman M Wereley. Non-linear quasi-static model of pneumatic artificial muscle actuators. *Journal of Intelligent Material Systems and Structures*, 26(5):541–553, may 2014.
- [49] Zhefeng Yu, Thomas Pillsbury, Gang Wang, and Norman M Wereley. Hyperelastic analysis of pneumatic artificial muscle with filament-wound sleeve and coated outer layer. *Smart Materials and Structures*, 28(10):105019, 2019.
- [50] Jonathan M. Chambers and Norman M. Wereley. Analysis of pneumatic artificial muscles and the inelastic braid assumption. *Actuators*, 11(8), 2022.
- [51] Curt S. Kothera, Mamta Jangid, Jayant Sirohi, and Norman M. Wereley. Experimental characterization and static modeling of mckibben actuators. *Journal of Mechanical Design*, 131(9):091010, 08 2009.
- [52] Darwin G. Caldwell, N.Tsagarakis, and G.A.Medrano-Cerda. Bio-mimetic actuators: polymeric pseudo muscular actuators and pneumatic muscle actuators for biological emulation. *Mechatronics*, 10(4):499–530, 2000.
- [53] Ching-Ping Chou and B. Hannaford. Static and dynamic characteristics of McKibben pneumatic artificial muscles. In *Proceedings of the 1994 IEEE International Conference on Robotics and Automation*, pages 281–286 vol.1, 1994.
- [54] M. De Volder, A.J.M. Moers, and D. Reynaerts. Fabrication and control of miniature mckibben actuators. *Sensors and Actuators A: Physical*, 166(1):111–116, 2011.
- [55] Belen Solano and Christine Rotinat-Libersa. Compact and lightweight hydraulic actuation system for high performance millimeter scale robotic applications: Modeling and experiments. *Journal of Intelligent Material Systems and Structures*, 22(13):1479–1487, 2011.
- [56] Wallace E Vander Velde et al. Multiple-input describing functions and nonlinear system design. *McGraw lill*, 1968.
- [57] D.A. Kingsley and R.D. Quinn. Fatigue life and frequency response of braided pneumatic actuators. In *Proceedings 2002 IEEE International Conference on Robotics and Automation (Cat. No.02CH37292)*, volume 3, pages 2830–2835 vol.3, 2002.
- [58] G.K. Klute and B. Hannaford. Fatigue characteristics of mckibben artificial muscle actuators. In *Proceedings. 1998 IEEE/RSJ International Conference on Intelligent Robots and Systems. Innovations in Theory, Practice and Applications (Cat. No.98CH36190)*, volume 3, pages 1776–1781 vol.3, 1998.

- [59] Gihyeok Na, Hiroyuki Nabae, and Koichi Suzumori. Braided thin mckibben muscles for musculoskeletal robots. *Sensors and Actuators A: Physical*, 357:114381, 2023.
- [60] Yanhong Peng, Yusuke Sakai, Koki Nakagawa, Yuki Funabora, Tadayoshi Aoyama, Kenta Yokoe, and Shinji Doki. Funabot-suit: A bio-inspired and mckibben muscle-actuated suit for natural kinesthetic perception. *Biomimetic Intelligence and Robotics*, 3(4):100127, 2023.
- [61] *Powering a Lower Limb Exoskeleton Using Pneumatic Artificial Muscles*, volume Volume 3: Biomedical and Biotechnology Engineering of ASME International Mechanical Engineering Congress and Exposition, 11 2016.
- [62] B. Tondy, S. Ippolito, J. Guiochet, and A. Daidie. A seven-degrees-of-freedom robot-arm driven by pneumatic artificial muscles for humanoid robots. *The International Journal of Robotics Research*, 24(4):257–274, 2005.
- [63] Benjamin K. S. Woods, Curt S. Kothera, and Norman M. Wereley. Wind tunnel testing of a helicopter rotor trailing edge flap actuated via pneumatic artificial muscles. *Journal of Intelligent Material Systems and Structures*, 22(13):1513–1528, 2011.
- [64] Michael A Meller, Matthew Bryant, and Ephraim Garcia. Reconsidering the McKibben muscle: Energetics, operating fluid, and bladder material. *Journal of Intelligent Material Systems and Structures*, 25(18):2276–2293, 2014.
- [65] Ryan M. Robinson, Curt S. Kothera, Benjamin K. S. Woods, III Robert D. Vocke, and Norman M. Wereley. High specific power actuators for robotic manipulators. *Journal of Intelligent Material Systems and Structures*, 22(13):1501–1511, 2011.
- [66] Benjamin K S Woods, Curt S Kothera, Jayant Sirohi, and Norman M Wereley. Pneumatic artificial muscles for trailing edge flap actuation: a feasibility study. *Smart Materials and Structures*, 20(10):105021, sep 2011.
- [67] M. Mori, K. Suzumori, S. Seita, M. Takahashi, T. Hosoya, and K. Kusumoto. Development of very high force hydraulic McKibben artificial muscle and its application to shape-adaptable power hand. In *2009 IEEE International Conference on Robotics and Biomimetics (ROBIO)*, pages 1457–1462, 2009.
- [68] Hyun Sung Cho, Tae Hwan Kim, Tae Hwa Hong, and Yong-Lae Park. Ratchet-integrated pneumatic actuator (RIPA): a large-stroke soft linear actuator inspired by sarcomere muscle contraction. *Bioinspiration and Biomimetics*, 15(3):036011, mar 2020.
- [69] Robert D Vocke III, Curt S Kothera, and Norman M Wereley. Mechanism and bias considerations for design of a bi-directional pneumatic artificial muscle actuator. *Smart Materials and Structures*, 23(12):125039, nov 2014.
- [70] Thomas E Pillsbury, Curt S Kothera, and Norman M Wereley. Effect of bladder wall thickness on miniature pneumatic artificial muscle performance. *Bioinspiration and Biomimetics*, 10(5):055006, sep 2015.

- [71] Steven D Thomalla and James D. Van de Ven. Modeling and implementation of the McKibben actuator in hydraulic systems. *IEEE Transactions on Robotics*, 34(6):1593–1602, 2018.
- [72] Kwanghyun Han, Nam-Ho Kim, and Dongjun Shin. A novel soft pneumatic artificial muscle with high-contraction ratio): a large-stroke soft linear actuator inspired by sarcomere muscle contraction. *Soft Robotics*, 5(5):554–566, Oct 2018.
- [73] Julie Diani, Bruno Fayolle, and Pierre Gilormini. A review on the Mullins effect. *European Polymer Journal*, 45(3):601–612, 2009.
- [74] Glenn K. Klute and Blake Hannaford. Accounting for Elastic Energy Storage in McKibben Artificial Muscle Actuators. *Journal of Dynamic Systems, Measurement, and Control*, 122(2):386–388, 12 2000.
- [75] D. C. Drucker. A Definition of Stable Inelastic Material. *Journal of Applied Mechanics*, 26(1):101–106, 06 1959.
- [76] A.D. Ansell and E.R. Trueman. The mechanism of burrowing in the anemone, peachia hastata gosse. *Journal of Experimental Marine Biology and Ecology*, 2(2):124–134, 1968.
- [77] Xuehang Bai, Yanhong Peng, Dongze Li, Zhuochao Liu, and Zebing Mao. Novel soft robotic finger model driven by electrohydrodynamic (ehd) pump. *Journal of Zhejiang University-SCIENCE A*, 25:596–604, 2024.
- [78] Frank Cianciarulo, Jacek Garbulinski, Jonathan Chambers, Thomas Pillsbury, Norman Wereley, Andrew Cross, and Deepak Trivedi. Analysis of an anchoring muscle for pipe crawling robot. In *Bioinspiration, Biomimetics, and Bioreplication XIII*, volume 12481, page 1248108. International Society for Optics and Photonics, SPIE, 2023.
- [79] Benjamin Gorissen, Takuya Chishiro, Shuhei Shimomura, Dominiek Reynaerts, Michael De Volder, and Satoshi Konishi. Flexible pneumatic twisting actuators and their application to tilting micromirrors. *Sensors and Actuators A: Physical*, 216:426–431, 2014.
- [80] Jihong Yan, Xinbin Zhang, Binbin Xu, and Jie Zhao. A new spiral-type inflatable pure torsional soft actuator. *Soft Robotics*, 5(5):527–540, 2018. PMID: 29975585.
- [81] Namsoo Oh, Jin-Gyu Lee, and Hugo Rodrigue. Torsional pneumatic actuator based on pre-twisted pneumatic tubes for soft robotic manipulators. *IEEE/ASME Transactions on Mechatronics*, 28(6):3191–3201, 2023.
- [82] Arnaud Lazarus and Pedro M. Reis. Soft actuation of structured cylinders through auxetic behavior. *Advanced Engineering Materials*, 17(6):815–820, 2015.
- [83] M.K. Philen, Y. Shan, P. Prakash, K.W. Wang, C.D. Rahn, A.L. Zydney, and C.E. Bakis. Fibrillar network adaptive structure with ion-transport actuation. *Journal of Intelligent Material Systems and Structures*, 18(4):323–334, 2007.

- [84] Fionnuala Connolly, Panagiotis Polygerinos, Conor J. Walsh, and Katia Bertoldi. Mechanical programming of soft actuators by varying fiber angle. *Soft Robotics*, 2(1):26–32, 2015.
- [85] Ellen T. Roche, Robert Wohlfarth, Johannes T. B. Overvelde, Nikolay V. Vasilyev, Frank A. Pigula, David J. Mooney, Katia Bertoldi, and Conor J. Walsh. A bioinspired soft actuated material. *Advanced Materials*, 26(8):1200–1206, 2014.
- [86] Frank Cianciarulo, John Schmidt, Todd Henry, Huan Xu, and Norman M. Wereley. *Actuation of a 3D Printed Sweeping Wing Mechanism Using Pneumatic Artificial Muscles*.
- [87] Frank C. Cianciarulo, Eric Y. Kim, and Norman M. Wereley. Analysis of torsional response in pneumatic artificial muscles. *Biomimetics*, 10(3), 2025.
- [88] T Umedachi, V Vikas, and B A Trimmer. Softworms: the design and control of non-pneumatic, 3d-printed, deformable robots. *Bioinspiration Biomimetics*, 11(2):025001, mar 2016.
- [89] Dror Kobo and Bat-El Pinchasik. Backswimmer-inspired miniature 3d-printed robot with buoyancy autoregulation through controlled nucleation and release of microbubbles. *Advanced Intelligent Systems*, 4(6):2200010, 2022.
- [90] Pawandeep Singh Matharu, Zhong Wang, John H. Costello, Sean P. Colin, Ray H. Baughman, and Yonas T. Tadesse. Sojel –a 3d printed jellyfish-like robot using soft materials for underwater applications. *Ocean Engineering*, 279:114427, 2023.
- [91] Antoine Beyeler, Jean-Christophe Zufferey, and Dario Floreano. Vision-based control of near-obstacle flight. *Autonomous Robots*, 27(5):201–219, Aug 2009.
- [92] Andrew J. Barry, Peter R. Florence, and Russ Tedrake. High-speed autonomous obstacle avoidance with pushbroom stereo. *Journal of Field Robotics*, 35(1):52–68, 2018.
- [93] Min Prasad Adhikari and Anton H. J. de Ruiter. Online feasible trajectory generation for collision avoidance in fixed-wing unmanned aerial vehicles. *Journal of Guidance, Control, and Dynamics*, 43(6):1201–1209, 2020.
- [94] Eitan Bulka and Meyer Nahon. High-speed obstacle-avoidance with agile fixed-wing aircraft. In *2019 International Conference on Unmanned Aircraft Systems (ICUAS)*, pages 971–980, 2019.
- [95] Shawn Keshmiri, A Ram Kim, Daksh Shulka, Aaron Blevins, and Mark Ewing. Flight test validation of collision and obstacle avoidance in fixed-wing uass with high speeds using morphing potential field. In *2018 International Conference on Unmanned Aircraft Systems (ICUAS)*, pages 589–598, 2018.
- [96] Zijie Lin, Lina Castano, Edward Mortimer, and Huan Xu. Fast 3d collision avoidance algorithm for fixed wing uas. *Journal of Intelligent Robotic Systems*, 97:577–604, 2020.

- [97] A. Mirzaee Kahagh, F. Pazooki, S. Etemadi Haghighi, and D. Asadi. Real-time formation control and obstacle avoidance algorithm for fixed-wing uavs. *The Aeronautical Journal*, 126(1306):2111–2133, 2022.
- [98] Colin Greatwood, Antony Waldock, and Thomas Richardson. Perched landing manoeuvres with a variable sweep wing uav. *Aerospace Science and Technology*, 71:510–520, 2017.