

# A Novel Triad Twisted String Actuator for Controlling a Two Degrees of Freedom Joint: Design and Experimental Validation\*

Damian Crosby<sup>1</sup>, Joaquin Carrasco<sup>2</sup>, William Heath<sup>2</sup>, and Andrew Weightman<sup>2</sup>

**Abstract**—Actuated universal joints, or equivalent joint systems, are found in a number of robotic applications, in particular mobile snake robots, snake-arm robots and robotic tails. These joints have two degrees of freedom on two axes, each perpendicular to a third axis and to themselves. Such joints use a variety of actuation methods, including direct drive motors, linear screw drives, cable based systems, and hydraulics/pneumatics. In this paper the authors design and validate a mechanism that uses the Twisted String Actuator (TSA) in an antagonistic triad to actuate the universal joint, using orientation sensors and load cells to create a robust cascading closed loop control system. This results in a light, compact, high-performance actuation system that avoids the extra mass and hardware complexity that alternative actuation methods present, with the additional challenge of non-linearity.

## I. INTRODUCTION

Actuated Universal Joint (AUJ) mechanisms are found in a wide range of robotic applications, such as confined space inspection using snake-arm robots [1], highly manoeuvrable mobile snake robots [2], and biomimetic robot tails for stability [3]. Mobile snake robots must usually incorporate electric actuators inline with their joints. This results in an AUJ having to shift the mass of the follower segments and all the actuators inside the follower segments, which results in high torque requirements and therefore Snake-arm robots and robotic tails can reduce the mass and size of the AUJ by moving their actuators away from the AUJs and use cables to transfer the force to the joints, or use hydraulic or pneumatic actuators which tend to be lighter than equivalent electric motors, at the expense of increased mass and bulk at the base of the arm or tail.

First developed by Würtz, May, Holz, *et al.* [4] in 2010, the Twisted String Actuator (TSA) uses two or more strings between two fixtures as a linear actuator. When one fixture is rotated (typically by an electric motor), the looped string twists into a helix, decreasing the distance between them.

The primary advantage of TSA over similar linear actuators such as a leadscrew is the reduction the TSA provides is not proportional (or even slightly inversely

proportional) to the mass of the actuator. By decreasing  $r_s$  (or  $r_s + r_c$  when there are more than two strings, where  $r_c$  is the “core radius”, or half the gap between any two strings that intersects the axis of rotation) the reduction increases for a constant  $l_u$  and  $\theta_s$ , resulting in a greater reduction with no increase, or even a slight decrease, in actuator mass. While the reduction in a leadscrew can be increased by decreasing the lead on the thread (typically denoted as  $\lambda$ ) which also has no increase in mass, this has a limited range and can quickly run up against manufacturing tolerances or material strength requirements. In order to achieve greater or more robust reductions, the screw radius has to be increased, or the driving motor has to have a larger reduction before driving the screw, both of which usually result in more material (typically steel) and therefore more mass.

However, TSA does have some disadvantages, the most significant of which is a non-linear reduction equation, which is also dependent on the motor angle  $\theta_s$  (and therefore actuator position). The reduction decreases in a non-linear fashion as the angle increases, with the derivative decreasing as the angle increases. There is also the compliance of the strings to consider, depending on the thickness and material chosen, which becomes a significant factor under high forces. Both of these issues can be addressed with accurate modelling [5] and/or a robust control strategy, as demonstrated in [4]. What is perhaps more an issue regarding mass for an AUJ is the unidirectional force of the TSA, which can only impart force in tension. This means that for an AUJ, which is a two degree of freedom joint, a minimum of three TSA are required for unless spring return mechanisms are used, which would impart additional force on the TSA and therefore reduce performance. However, the potential high force to mass ratio of the TSA due to the non-proportional reduction may adequately compensate for the additional actuator requirement.

The focus of this research is to investigate if the TSA is a suitable candidate for control of an AUJ considering both the benefits and drawbacks.

To this end, the objective is to simulate a model and then construct a physical experimental prototype to validate the proposed control system.

### A. Twisted String Actuator

First developed by [4] in 2010, TSA uses two or more strings between two fixtures as a linear actuator. When one fixture is rotated (typically by an electric motor), the

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<sup>1</sup>Department of Mechanical, Aerospace and Civil Engineering, University of Manchester, United Kingdom.

<sup>2</sup>Department of Electrical and Electronic Engineering, University of Manchester, United Kingdom.

Email correspondence: damian.crosby@manchester.ac.uk

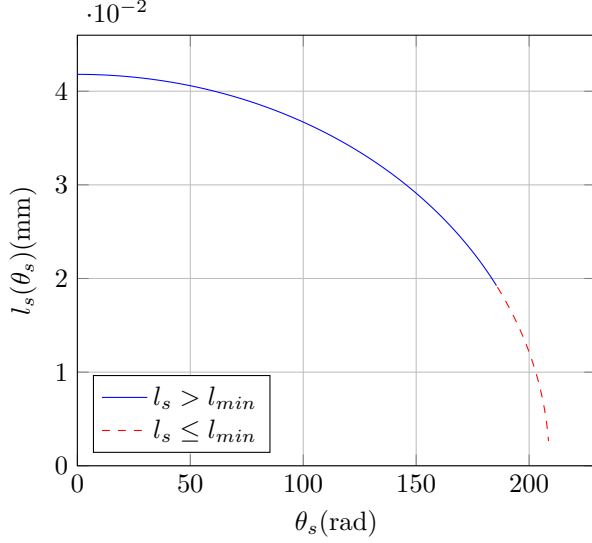


Fig. 1: TSA string length against motor angle with coefficients from table I.

strings twist into a helix, decreasing the distance between the fixtures.

The following equation from [4] calculates the actuator length  $l_s$  for a given  $\theta_s$ , given  $l_u$  and  $r_s$  for a two string TSA. This equation assumes an infinite string stiffness, so is only reasonably accurate under low tension.

$$l_s(\theta_s) = \sqrt{l_u^2 - \theta_s^2 r_s^2} \quad (1)$$

Although theoretically the stroke of the TSA can be the entire domain of  $[0, l_n]$ , in reality the thickness of the string prevents a geometric helix from forming once the helix pitch  $q < 4r_s$  (or  $q < 2nr_s$  for  $n$  strings). This limits the lower bound of the stroke to the value in the following equation, or approximately 46% of  $l_n$  for a two string TSA.

$$l_{min} = \frac{l_u}{\sqrt{\frac{\pi^2}{2} + 1}} \approx 0.46 l_u \quad (2)$$

### B. Antagonistic Triad

As mentioned in the introduction, because the TSA provides only tensile force, a minimum of three actuators are required. These can be arranged in a triangular configuration to create an “antagonistic triad”, akin to the antagonistic pairs of muscles found in animals. Where a revolute joint would be found between the connecting ends of the actuator, a universal joint is found instead. The geometric structure of the system can be described with two equilateral triangles of inradius  $r$  on two planes separated in the  $z$  axis. The centroids are then connected via a universal joint from each plane normal to an intersecting point, described by vector  $\theta = [\theta_1 \ \theta_2]$  to denote the rotation of the second plane relative to the first, in the  $x$  and  $y$  axis around the intersecting

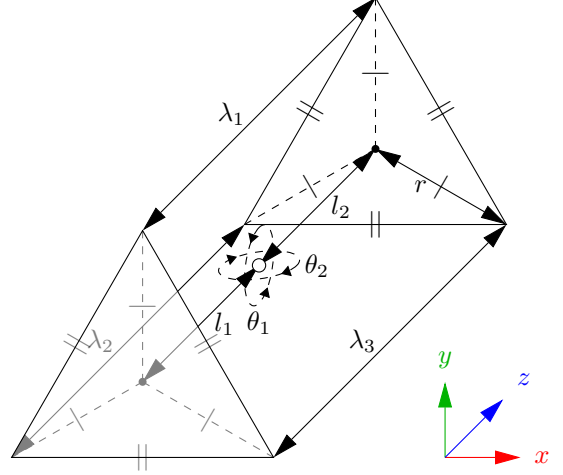


Fig. 2: Kinematic diagram of the antagonistic triad, where the universal joint rotation is defined by  $\theta_{1,2}$  on the  $x$  and  $y$  axes respectively, and the string lengths are defined by  $\lambda_{1,2,3}$  for the “Top”, “Left” and “Right” strings.  $r$  and  $l_{1,2}$  define the anchor points of the strings.

point, and  $l_1$  and  $l_2$  to denote the normal distance from the intersection to the first and second plane centroids respectively. When  $\theta = [0 \ 0]$  the triangles are parallel to each other. The distance between the vertex pairs of each triangle is then denoted as  $[\lambda_1 \ \lambda_2 \ \lambda_3]$  for the “top”, “left” and “right” vertices of the triangles. When  $\theta$  is changed, this will change  $\lambda_1$ ,  $\lambda_2$  and  $\lambda_3$  respectively. Figure 2 illustrates this antagonistic triad.

To calculate the lengths of the strings for a given  $\theta$  of the universal joint, we can define a vector function  $\Lambda(\theta) = [\lambda_1(\theta) \ \lambda_2(\theta) \ \lambda_3(\theta)]$  with  $l_1$ ,  $l_2$  and  $r$  as the coefficients.

$$\begin{aligned} \lambda_1(\theta) &= \sqrt{(l_1 + l_2 \cos \theta_1 \cos \theta_2 + r \cos \theta_1 \sin \theta_2)^2 \\ &\quad + (r - r \cos \theta_2 + l_2 \sin \theta_2)^2 \\ &\quad + (l_2 \cos \theta_2 \sin \theta_1 + r \sin \theta_1 \sin \theta_2)^2} \\ \lambda_2(\theta) &= \sqrt{(a - b + c)^2 + (l_1 - d)^2 + (e)^2} \\ \lambda_3(\theta) &= \sqrt{(a + b - c)^2 + (l_1 + d)^2 + (e)^2} \end{aligned} \quad (3)$$

where:

$$\begin{aligned} a &= -\frac{\sqrt{3}r(\cos \theta_1 - 1)}{2} \\ b &= l_2 \cos \theta_2 \sin \theta_1 \\ c &= \frac{r \sin \theta_1 \sin \theta_2}{2} \\ d &= \frac{\sqrt{3}r \sin \theta_1}{2} + l_2 \cos \theta_1 \cos \theta_2 - \frac{r \cos \theta_1 \sin \theta_2}{2} \\ e &= \frac{r \cos \theta_2}{2} - \frac{r}{2} + l_2 \sin \theta_2 \end{aligned}$$

## II. CONTROL SYSTEM

The control system is a four layer cascade design, using feedback of the joint position from the accelerometers and TSA force from the load cells. It uses a second

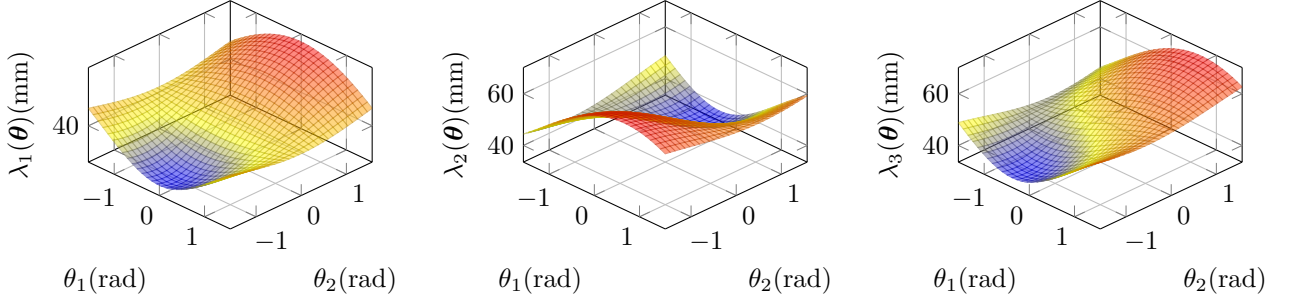


Fig. 3: Surface plots of each element of the vector function  $\Lambda(\theta)$ , assuming coefficient values from table ??.

order setpoint trajectory  $\mathbf{q}$  as input, which can either be pre-defined or generated dynamically from user input. Feedback is provided by the AUJ angular position  $\theta$ , angular velocity  $\dot{\theta}$ , and TSA tension force  $\hat{\mathbf{f}}$ .

#### A. Actuated Universal Joint Position PID Controller with Acceleration Feedforward

Firstly, a PID controller is used to generate a control signal  $\mathbf{u}$  with the input  $\mathbf{q}$  as the setpoint, and the AUJ angular position  $\theta$  and velocity  $\dot{\theta}$  as feedback, plus the addition of a feedforward term for the input acceleration  $\ddot{\mathbf{q}}$ .

$$\mathbf{u} = \mathbf{k}_p(\mathbf{q} - \theta) + \mathbf{k}_i \left( \int_0^t \mathbf{q} - \theta \right) dt + \mathbf{k}_d(\dot{\mathbf{q}} - \dot{\theta}) + \ddot{\mathbf{q}} \quad (4)$$

#### B. Inverse Dynamics

The control signal  $\mathbf{u}$  from the PID controller is then converted to the desired AUJ torque  $\boldsymbol{\tau}$  using the Euler-Lagrange method.

$$\boldsymbol{\tau} = \mathbf{D}(\theta)\mathbf{u} + \mathbf{C}(\theta, \dot{\theta})\dot{\theta} + \mathbf{G}(\theta) \quad (5)$$

#### C. Twisted String Actuator Force Optimisation Algorithm

This function uses a modified algorithm from [6] to select an optimal force vector from the desired joint torque. A force matrix  $\mathbf{F}$  is created from the torque input  $\boldsymbol{\tau}$ , jacobian  $\mathbf{J}_\Lambda$  from the vector function  $\Lambda$  as defined in equation 3, and minimum force constant  $f_{min}$ .  $f_{ii}$  is equal to  $f_{min}$ , while the other elements in the column are based on a calculation using  $\mathbf{J}_{\Lambda-i,*}$  where  $-i$  is a row removed from the matrix.

$$\begin{aligned} \mathbf{J}_\Lambda &= \begin{bmatrix} \frac{\partial \lambda_1}{\partial \theta_1} & \frac{\partial \lambda_2}{\partial \theta_1} & \frac{\partial \lambda_3}{\partial \theta_1} \\ \frac{\partial \lambda_1}{\partial \theta_2} & \frac{\partial \lambda_2}{\partial \theta_2} & \frac{\partial \lambda_3}{\partial \theta_2} \end{bmatrix} \\ \mathbf{F}(\boldsymbol{\tau}, \theta) &= \begin{cases} f_{i,i} = f_{min} \\ f_{-i,i} = -\mathbf{J}_{\Lambda-i,*}^{-\top} \left( \mathbf{J}_{\Lambda-i,*}^\top f_{min} + \boldsymbol{\tau} \right) \end{cases} \quad (6) \\ &= \begin{bmatrix} f_{min} & f_{12} & f_{13} \\ f_{21} & f_{min} & f_{23} \\ f_{31} & f_{32} & f_{min} \end{bmatrix} \end{aligned}$$

The following algorithm then selects one column of  $\mathbf{F}$  to be the output force vector  $\mathbf{f}$ .

- 1:  $\mathbf{s} \leftarrow [\top \quad \top \quad \top]$
- 2: **if**  $f_{23} > f_{min}$  **then**  $s_2 \leftarrow \perp$  **else**  $s_3 \leftarrow \perp$  **end if**
- 3: **if**  $f_{31} > f_{min}$  **then**  $s_3 \leftarrow \perp$  **else**  $s_1 \leftarrow \perp$  **end if**
- 4: **if**  $f_{12} \geq f_{min}$  **then**  $s_1 \leftarrow \perp$  **else**  $s_2 \leftarrow \perp$  **end if**
- 5: **for**  $i = 1$  to 3 **do**
- 6:     **if**  $s_i \rightarrow \top$  **then**  $\mathbf{f} \leftarrow \mathbf{f}_{*,i}$  **end if**
- 7: **end for**

#### D. Twisted String Actuator Force Proportional Controller with Deadband Compensation

The selected forces are then used as an input to a P controller using the measured load cell forces  $\hat{\mathbf{f}}$  as feedback. The output from this can then be used to control the top, left and right TSA motor's velocity as  $\boldsymbol{\omega}$  (each motor controller has a hardware velocity PI controller). Due to a hardware velocity controller deadband within  $\pm 10 \text{ min}^{-1}$ , an adjustable deadband compensator is used to create  $\bar{\boldsymbol{\omega}}$ , with  $h \in [0, 10]$  changing the size of the deadband, where 0 is no deadband, and 10 is full deadband.

$$\bar{\boldsymbol{\omega}} = \boldsymbol{\omega} \mid \omega_i \begin{cases} \omega_i & \omega_i > 10 \vee \omega_i < -10 \\ 10 & \omega_i < 10 \wedge \omega_i > h \\ -10 & \omega_i > -10 \vee \omega_i < -h \\ 0 & \text{otherwise} \end{cases} \quad (7)$$

The result from the TSA is then a compressive force acting between each of the three TSA and its corresponding endpoint on the Antagonistic Triad, imparting a torque on the axes of the universal joint.

### III. SIMULATION & EXPERIMENTAL RESULTS

#### A. Twisted String Actuator State Space Definition

In order to approximate a TSA plant from within a simulation, a state space model was required which takes motor current  $u$  as an input and outputs  $y$  as the TSA tension force  $f_i$ . [4] defines it as such, where  $J$  is the motor inertia,  $C$  is the motor coulomb friction (modified from viscous friction as the 1724TSR only has dry friction),  $K_t$  is the motor torque constant, and  $K_L$  is the load stiffness. As the original definition is for a fixed load  $l_u$  distance from the motor a modified model is required

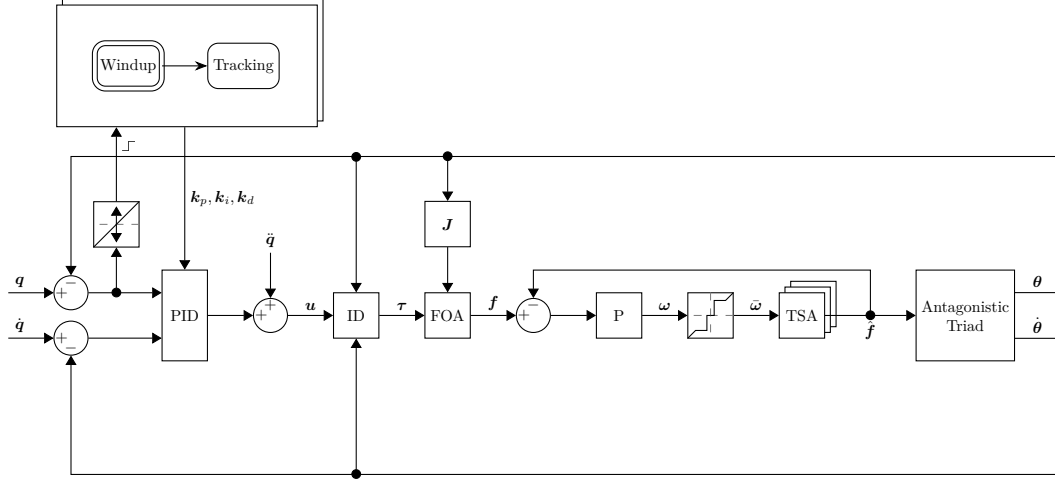


Fig. 4: Block diagram of the complete control system, excluding the hardware velocity controllers for the motors.

TABLE I: Model coefficients.

Coefficient	Value	Coefficient	Value
$l_1$	41.8 mm	$J$	$1 \times 10^{-6} \text{ kg m}^{-2}$
$l_2$	0 mm	$K_L$	$1000 \text{ N m}^{-1}$
$r$	13 mm	$f_{min}$	3 N
$l_n$	41.8 mm	$\omega_s$	$441.9 \text{ rad s}^{-1}$
$r_s$	200 $\mu\text{m}$	$I_s$	0.19 A
$m$	72.619 13 g	$K_t$	$0.0263 \text{ N m A}^{-1}$
$C$	0.1315 N mm	$\tau_s$	4.5 mN m
$\alpha_s$	$1 \times 10^5 \text{ rad s}^{-2}$		
Coefficient		Value	
$I$	$3 \times 10^{-5}$	0	0
	0	$3.2 \times 10^{-5}$	0
	0	0	$1.4 \times 10^{-5}$
	$\text{kg m}^{-2}$		

TABLE II: PID gains in the simulation and experiment.

Gain	Value	
	Simulation	Experiment*
$k_p$	800	$3 \times 10^4$
$k_i$	3000	350
$k_d$	50	50
$k_{ps}$	19	100

\* Tracking mode, see section III-C.

which takes into account the varying length between the motor and load defined by  $\Lambda(\theta)$ . A saturation function is used to prevent incorrect compression forces when the string is slack. All of the motor coefficients were taken from the Faulhaber 1724TSR datasheet [7] as this is the motor used in the experimental model. An estimated value is used for the load stiffness, this was chosen to be a high number as the model is expected to be very stiff.

$$\begin{aligned}
 h(\theta_s) &= \frac{\theta r_s^2}{\sqrt{l_u^2 - \theta_s^2 r_s^2}} \\
 k(\theta_s, \theta) &= \lambda_n(\theta) - \sqrt{l_u^2 - \theta_s^2 r_s^2} \\
 \dot{x} &= \begin{bmatrix} x_2 \\ -\frac{K_L}{J} h(x_1) k(x_1, \theta) - \frac{C}{J} \text{sgn}(x_2) \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{K_t}{J} \end{bmatrix} u \\
 y &= K_L \text{sat}_0^\infty k(x_1, \theta)
 \end{aligned} \tag{8}$$

The state space model was then adapted to include constraints on motor velocity and acceleration set by the motor controller in order to keep the motor within design limits, by replacing  $\dot{x}$  with  $\dot{x}'$  which contains saturation functions for maximum motor velocity  $v_s$  and acceleration  $\alpha_s$ .

$$\dot{x}' = \begin{bmatrix} \text{sat}_{\omega_s} \dot{x}_1 \\ \text{sat}_{\alpha_s} \dot{x}_2 \end{bmatrix} \tag{9}$$

For the experimental validation, a physical prototype of the mechanism was constructed with coefficients from table I as design parameters. The motors to be used would be Faulhaber 1724T024SR micromotors, selected for their compact size but high torque. These would be controlled by Faulhaber MCDC3002 motor controllers, which could interface with a National Instruments MyRIO via the USB port, using a USB to serial converter. The load cells would be Futek LCM100 miniature load cells, selected for their small size. The signals from these would be amplified using Flyde FE-359-TA instrumentation amplifiers and decoded using an external AD7606 ADC before being fed into the MyRIO using SPI.

### B. Motor Control

As a current controlled system did not produce satisfactory results using the motor controllers, an alternative

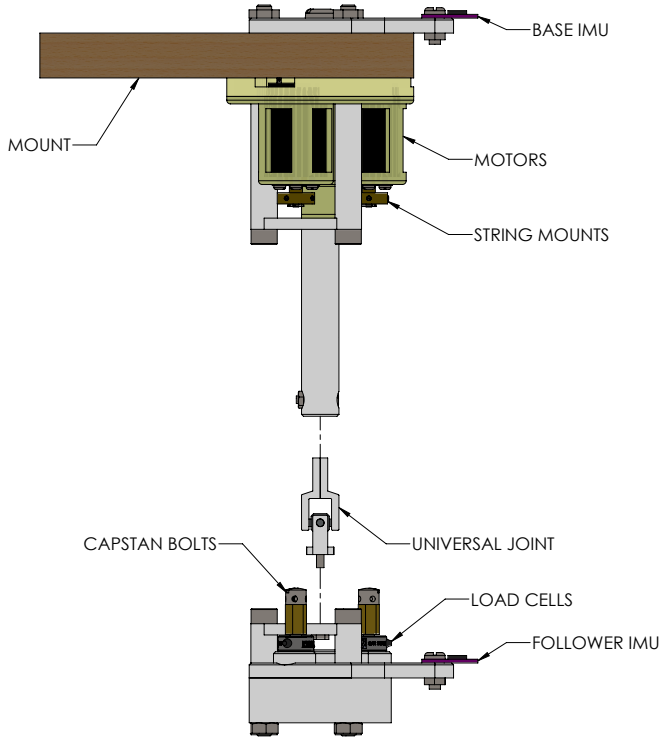


Fig. 5: Schematic of the experimental model.

TABLE III: PID constants for both states.

Gain	Value	
	Windup	Tracking
$k_p$	800	$3 \times 10^4$
$k_i$	3000	350

speed controller was used instead. This turned out to be very effective,

### C. Windup & Tracking States

When the mechanism is started with the TSA in a completely unwound state, before it can begin tracking a motion trajectory, the TSA strings must “wind up” to closely match the initial state of  $\mathbf{f}$ . During this phase, the outer PID gains  $k_p, k_i$  are unsuitable and can result in damage to the mechanism. To mitigate this, two sets of PID gains are chosen, one for the windup state, and another for the tracking state, which the windup state transitions to once suitable stability is achieved. This transition trigger is defined on a per-axis basis, as the first zero crossing of the angle error (as  $\mathbf{q} = 0$  this is effectively  $\boldsymbol{\theta}$ ).

### D. Results

Figure ?? plots the tracking response of both the simulation and experiment.

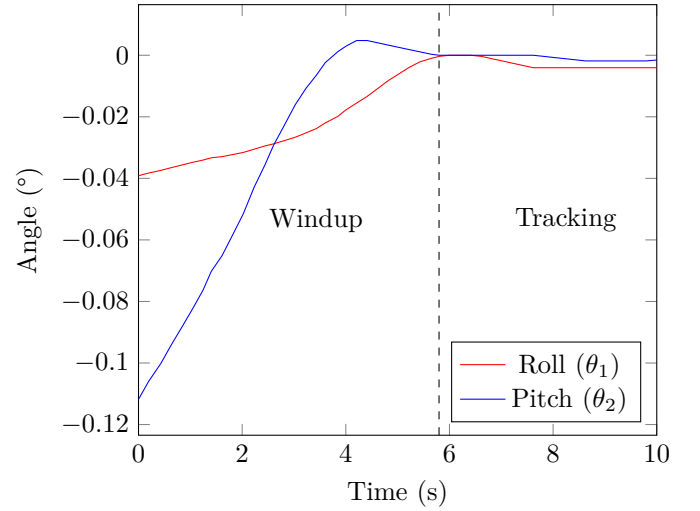


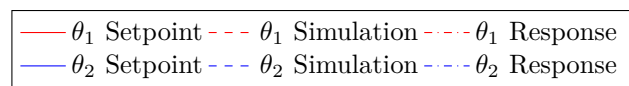
Fig. 6: The initial startup of the mechanism showing the transition between “windup” and “tracking” states.

## IV. EXPERIMENTAL TESTING AND SYSTEM VALIDATION

## V. CONCLUSION

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