COMPASS

*University of **<u>Co</u>**lorado **<u>M</u>**odel **<u>P</u>**ositioning - Dyn**<u>A</u>**mic/**<u>S</u>**tatic - **<u>S</u>**ystem*

Preliminary Design Review

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Agenda

Section	Presenter
Overview	Anna
Requirements	Anna/Brandon
Position Uncertainty	Ryan
Control of Degrees of Freedom	Kristian
Summary	Kyle

Overview

Project Statement

Design, build, and validate a **wind tunnel positioning system** with **minimal blockage**, capable of moving a test article within **four degrees of freedom**, **statically and dynamically**, through electrical manipulation by a **LabVIEW interface**. The system shall have the ability to integrate with **future load and moment measuring systems** and provide **failsafes** for power failure and user error scenarios.

Motivation

- Provide a model positioning system for the new wind tunnel and provide support for aerodynamic models used for:
  - Research performed by CU faculty
  - Graduate student projects
  - Undergraduate senior projects

Functional Requirements

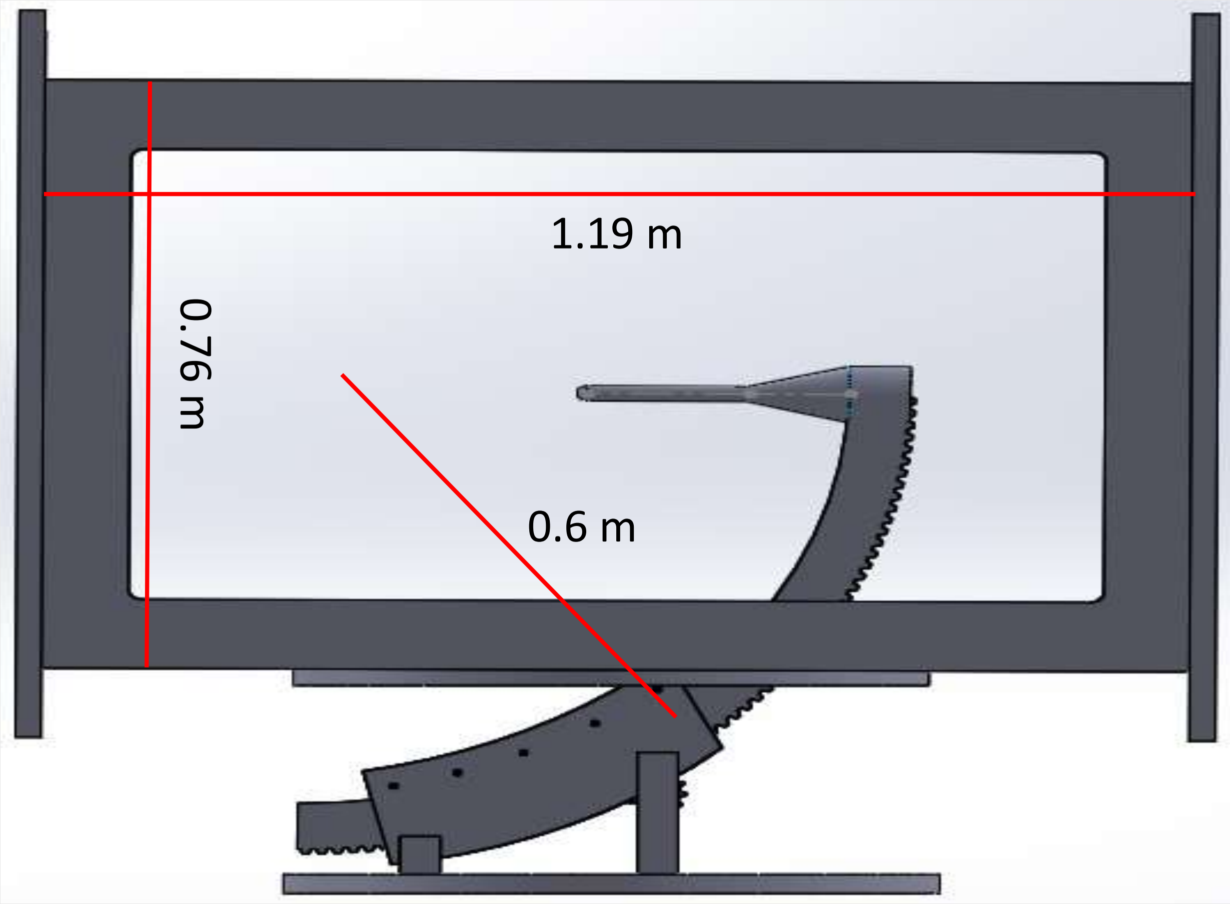
FR 1: **COMPASS** shall be able to **position** the **model**.

FR 2: **COMPASS** software shall **interface** with the user and the hardware such that models can be **positioned** at the required **range** and **rate**

FR 3: **COMPASS** shall be **integrated** with the **wind tunnel test section**.

Baseline Design

- Crescent vs. Arm Sting
  - Both rated high in size
  - Both rated high in range
  - Crescent > Arm in number of linkages
- Stepper vs. Servo Motor
  - Servo: higher resolution
  - Servo: higher angular rate
  - Stepper: lower cost



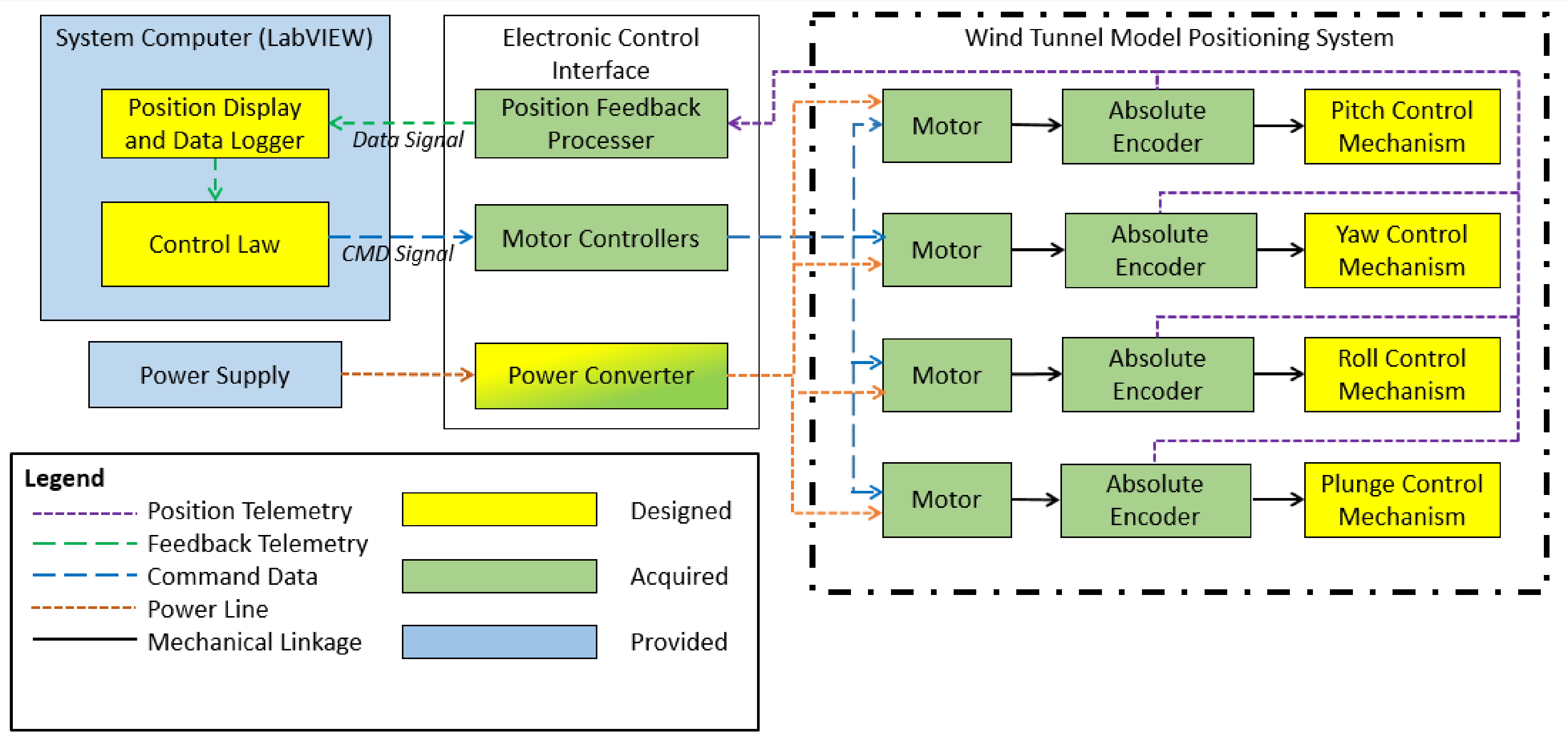
FILE HOME INSERT PAGE LAYOUT FORMULAS DATA REVIEW VIEW TEAM

Sample - Excel

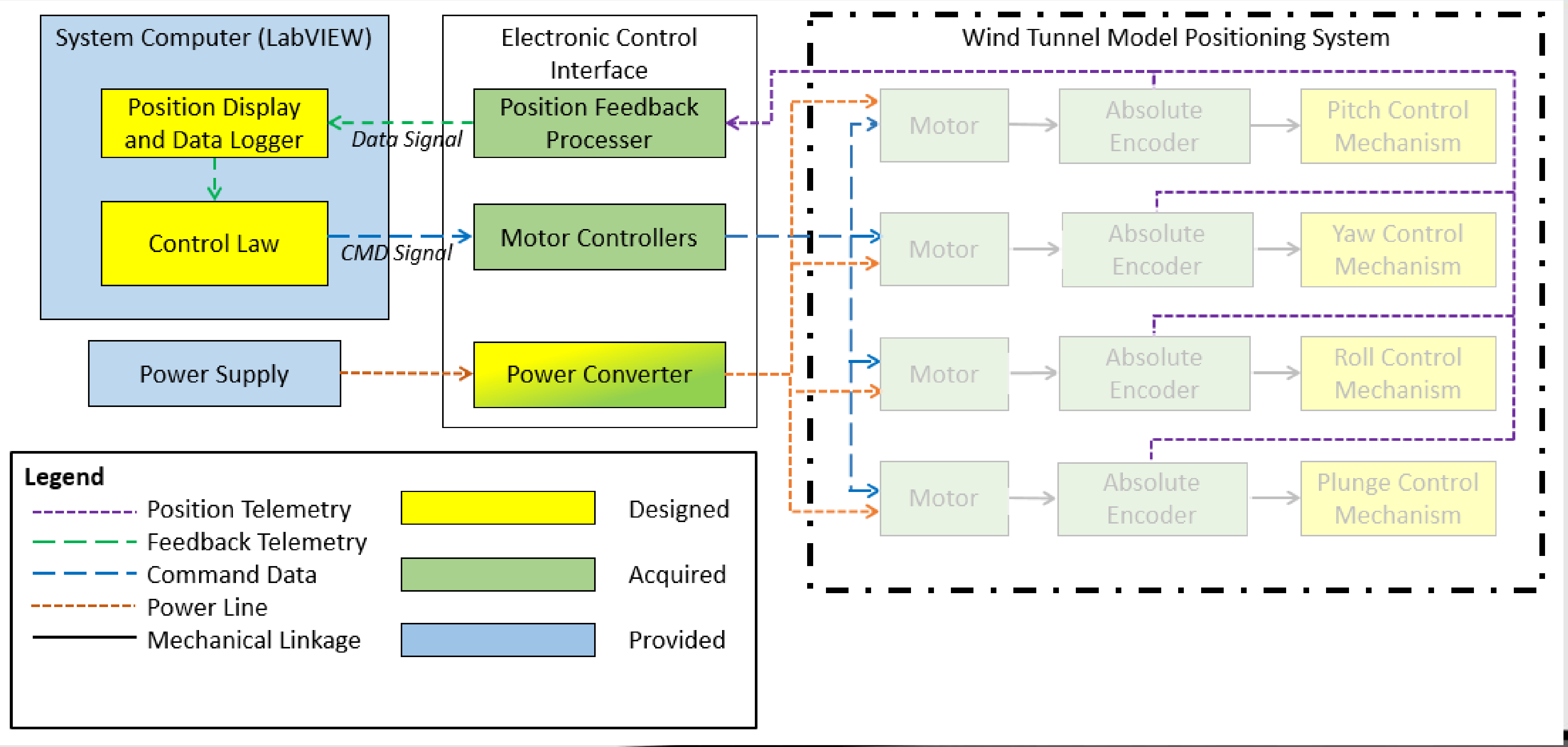
E10 | -1.6

Time (s)	Yaw (deg)	Pitch (deg)	Roll (deg)	Plunge (mm)
1	0	2	0	0
2	0.1	1.99	0	-0.2
3	0.2	1.98	0	-0.4
4	0.3	1.97	0	-0.6
5	0.4	1.96	0	-0.8
6	0.5	1.95	0	-1
7	0.4	1.94	0	-1.2
8	0.3	1.93	0	-1.4
9	0.2	1.92	0	-1.6
10	0.1	1.91	0	
11	0	1.9	0	

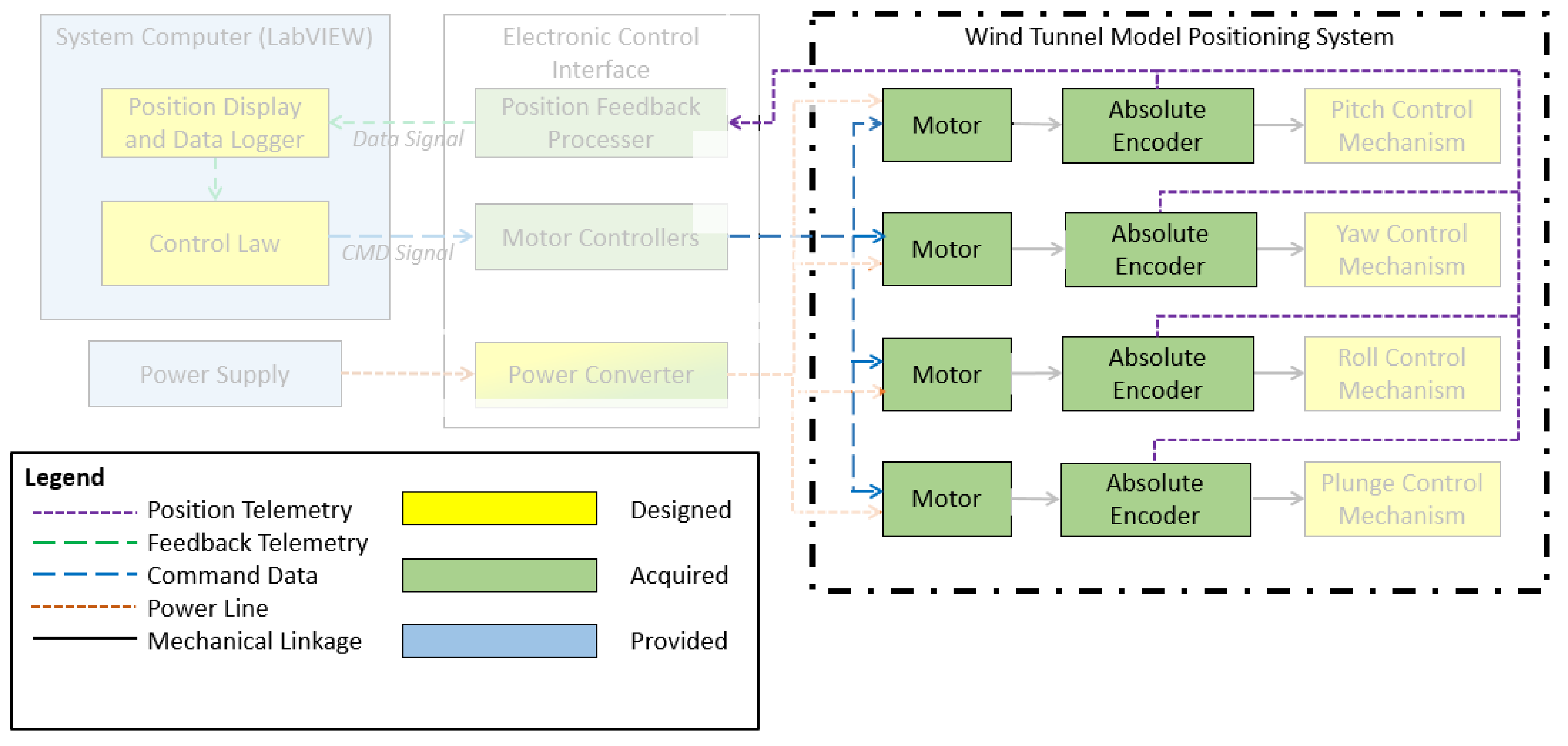
Functional Block Diagram (FBD)



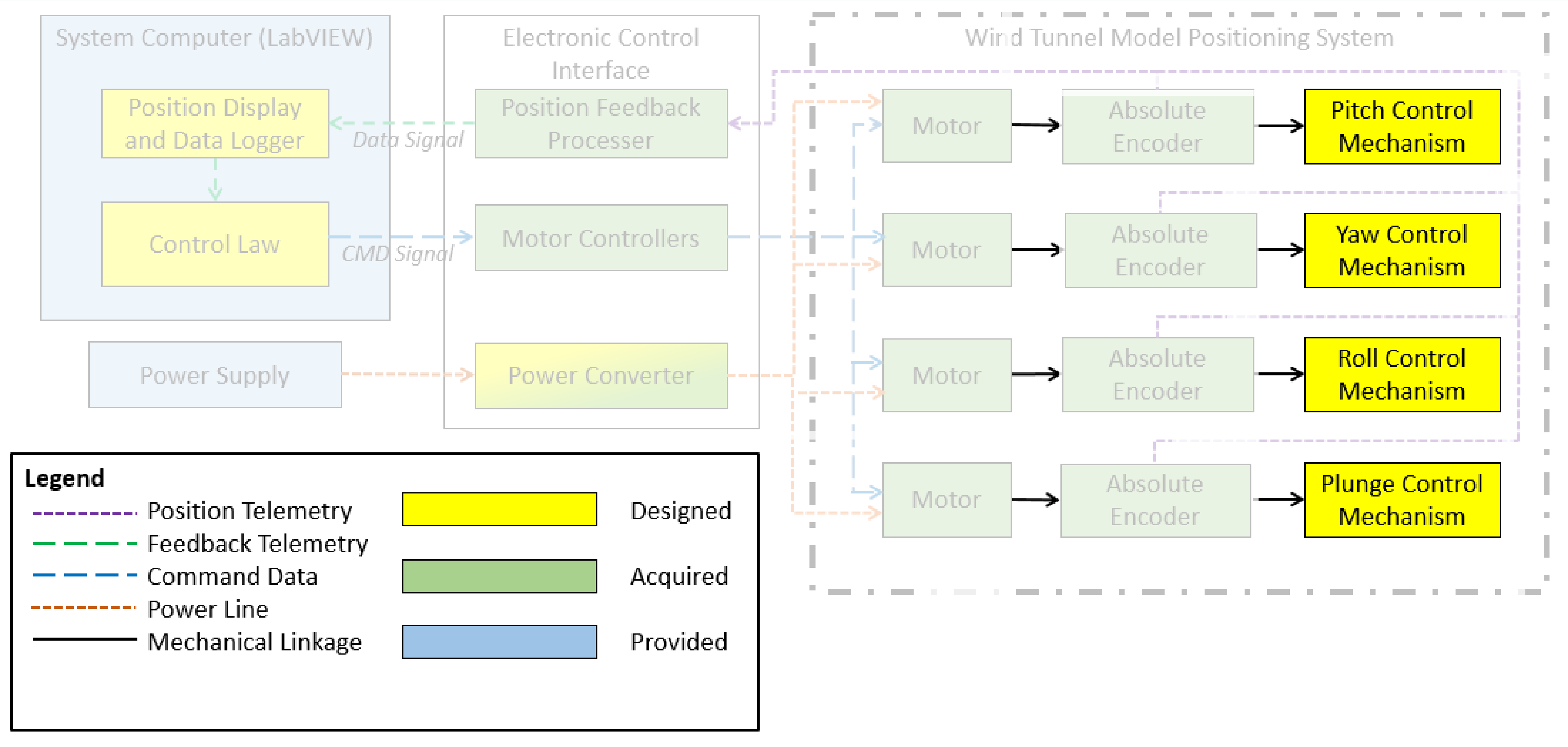
FBD - Control Elements - Electrical/Software



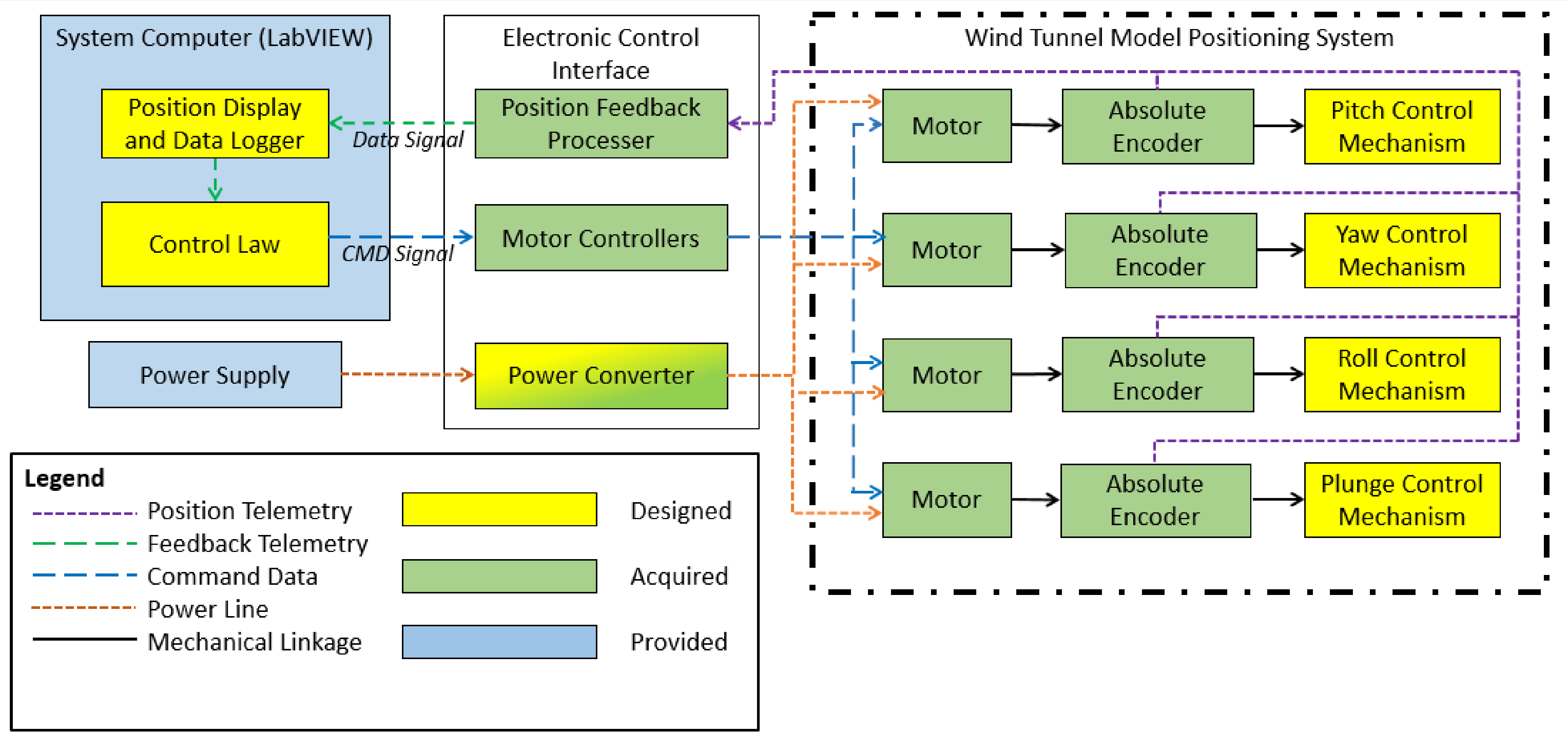
FBD - Control Elements - Mechanical



FBD - Structural Design Elements



Functional Block Diagram (FBD)



Critical Project Feasibility Elements

CPFE.1: Position Uncertainty (FR 1)

- Tight accuracy requirements from design requirements
- Need for high tolerance gearing

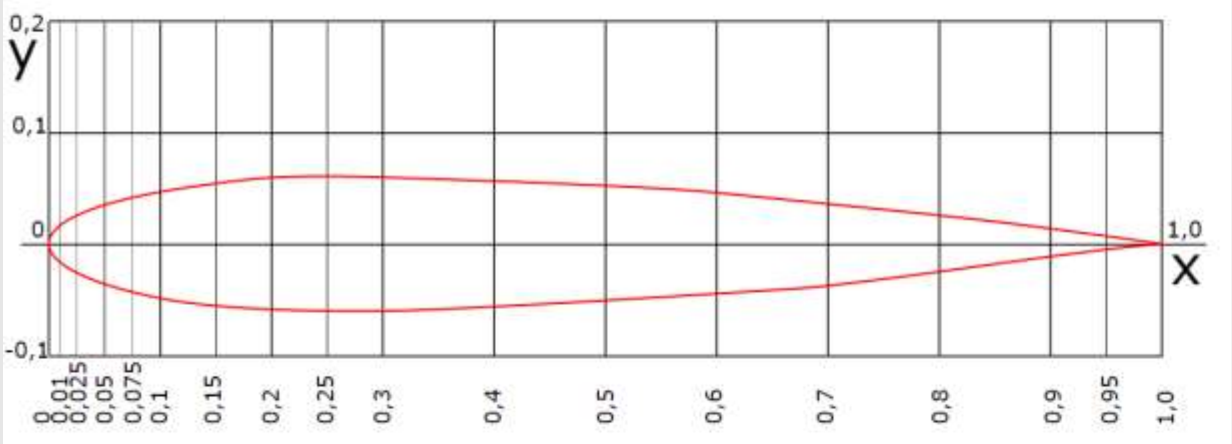
CPFE.2: Control of Degrees of Freedom (FR 2)

- Electric control of the pointing system
- Moving multiple degrees of freedom sequentially

CPFE.1: Position Uncertainty

Simple Model Assumptions

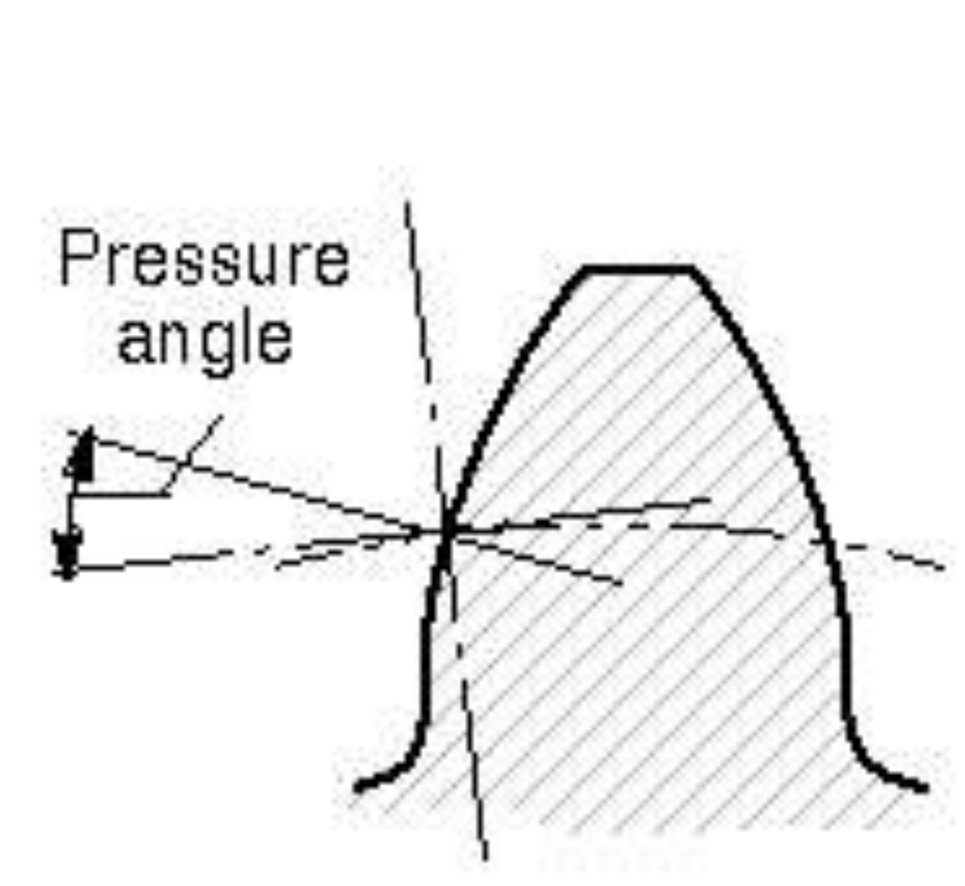
- Simple model assumed to be NACA 0012 airfoil
  - 0.5 m span with 0.1 m chord and made of Aluminum 6061
  - 12 degree Angle of Attack
  - 65 m/s Velocity



- Accounted for with gearing

Gear Ratio Calculation

- Assumption
  - Pressure Angle: 25°
- 12 tooth pinion (motor gear)
- 240 tooth crescent arm
- Gear Ratio: 20
- Pinion Radius: 30 mm

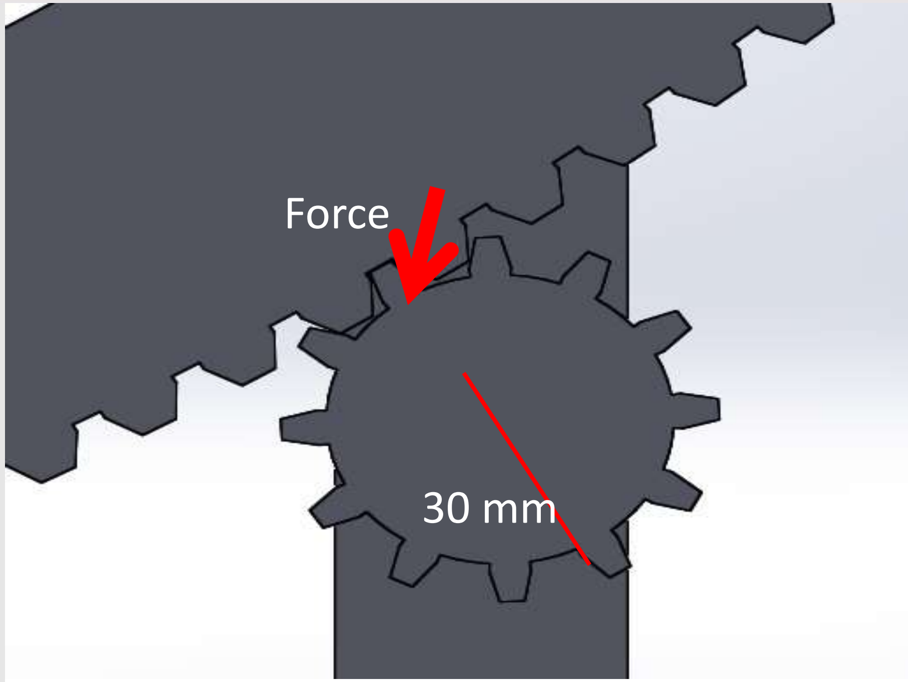


Gear Material Considerations

Material	Aluminum	Steel	Brass
Features	Lightweight Easy Machinability	Heavy Moderate Machinability	Heavy Easy Machinability
Applications	Light duty instrument gears (Light load)	Low to Medium load capabilities	Light load capabilities
Range of stress failure (MPa)	124 - 186 Medium Strength	147 - 236 High Strength	11 - 76 Low Strength

Gear Tooth Strength

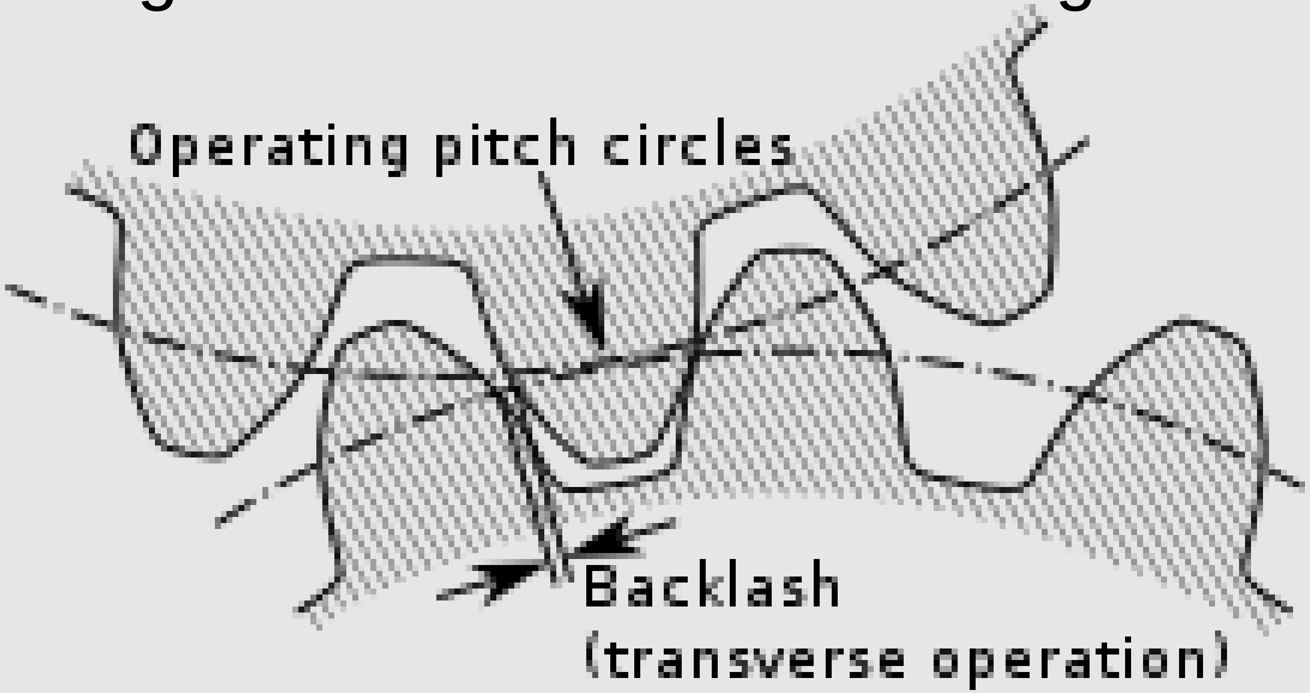
- Calculated lift force of 137 N
- Stress on gear teeth is 156 MPa
- Allowable gear stress of 158 MPa
  - Assumes 99.99% reliability
- Allowable gear stress of 238 MPa
  - Assumes 99.00% reliability
- Steel is the strongest option



RELIABILITY FACTOR	% survival	K_R
fewer than one failure in 10,000	99.99	1.50
fewer than one failure in 1,000	99.9	1.25
fewer than one failure in 100	99	1.00
fewer than one failure in 10	90	0.85

Backlash

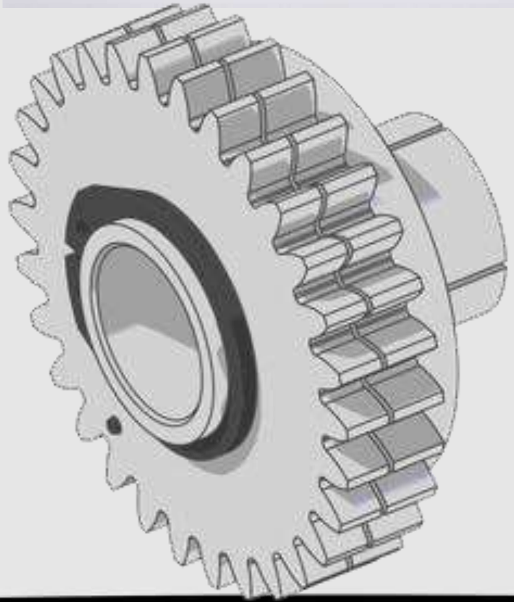
- Causes inaccuracies
- Angle between tooth face and gear width tangent



Gear Considerations

- Spur Gears:
  - Uncertainties in Pitch range from 0.0076° to 0.028°
  - Both are below 0.1° pitch accuracy: **FEASIBLE**

- Zero Backlash Gears:
  - Reduce uncertainties such that they are negligible



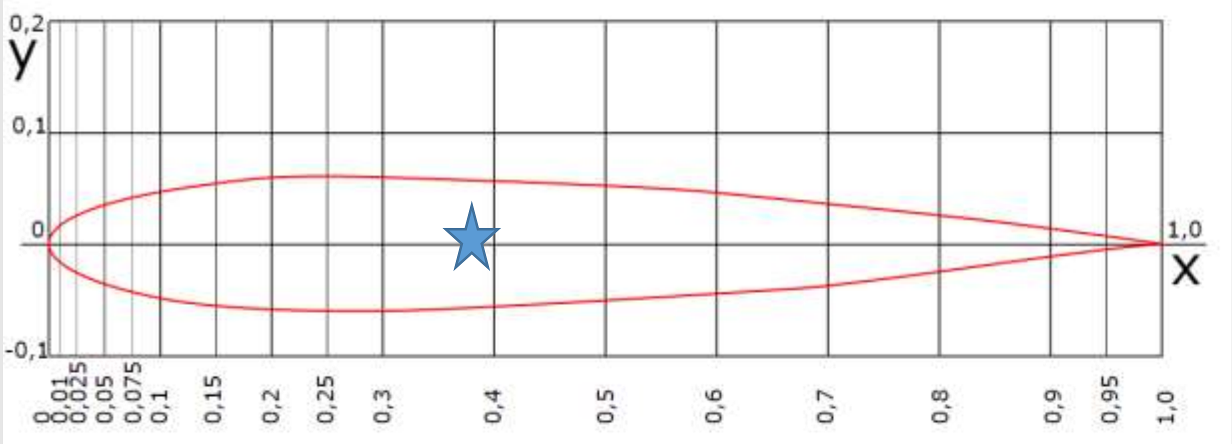
CPFE.2: Control of Degrees of Freedom

Motivations

- Feasibility of acquiring required motors
  - Feasibility to resist and move loads in each degree of freedom
- Feasibility of acquiring required sensor resolutions
- Feasibility of creating control law
  - Control law design and model simulation

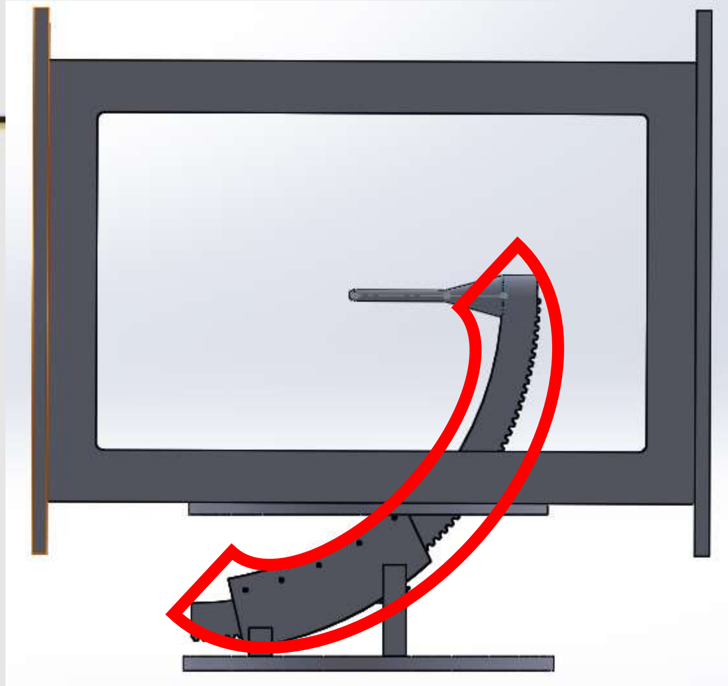
Simple Model Assumptions

- Simple model assumed to be NACA 0012 airfoil
  - 0.5 m span with 0.1 m chord and made of Aluminum 6061
  - Assumed to be flat plate for inertia estimates with thickness of 0.012 m
  - Rotation assumed to be about Center of Gravity (CG)
  - CG= 39.2% of chord from leading edge



Pitch Torque Estimate

- Assumptions:
  - Motor driving pitch directly
  - Inertia of crescent: thick hoop
  - Total inertia: test model and crescent
  - Torque from friction ignored
  - 60 degree rotation
  - 64 degrees/sec rate



$I_{PITCH,TOT} = 1.17\ kg\ m^2$ $\quad \omega = 64 \frac{\circ}{s}$ $\quad \Delta\theta = 60^\circ$

$\Delta\theta = \frac{60\pi}{180} = \frac{1}{2}\alpha t^2$

$t = \frac{\Delta\theta}{\omega} = \frac{60^\circ}{64\frac{\circ}{s}} = 0.938\ s$

$\boxed{\tau_{REQ,PITCH} = 2.78\ N\ m}$

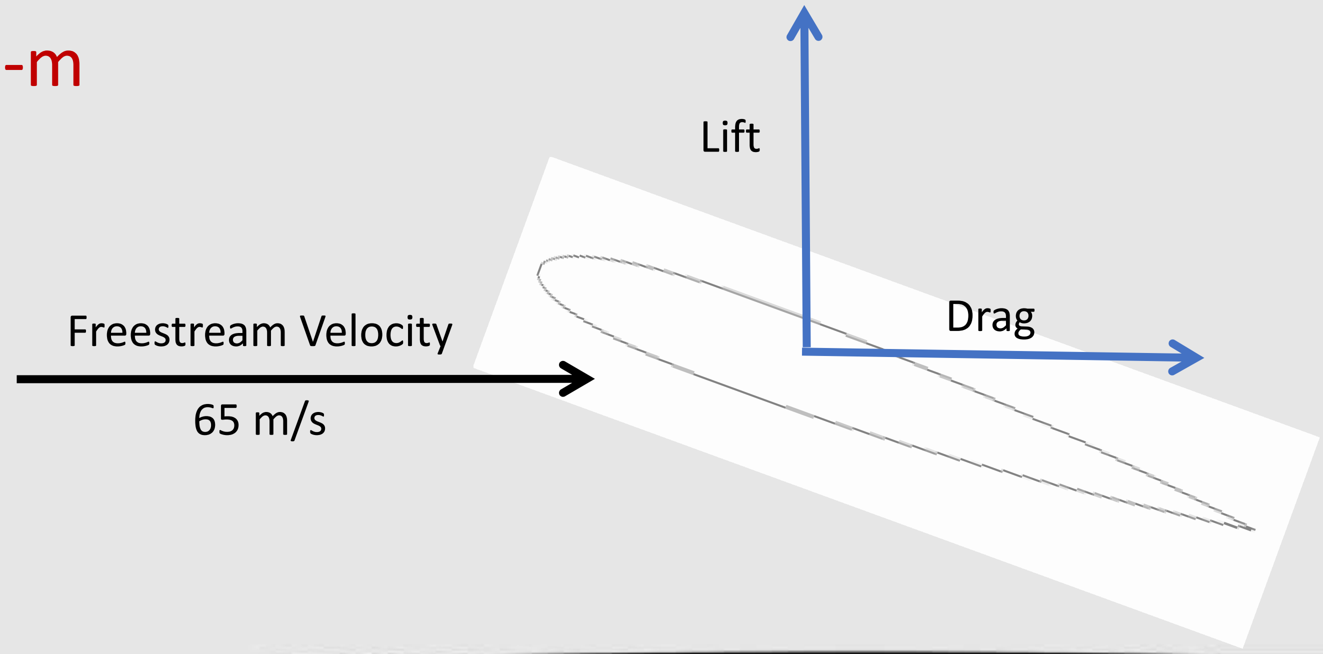
FEASIBLE

$\alpha = 2.38\ \frac{rad}{s^2}$

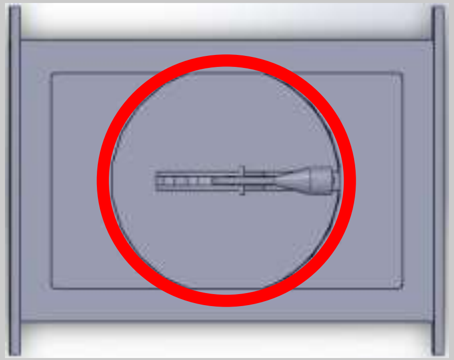
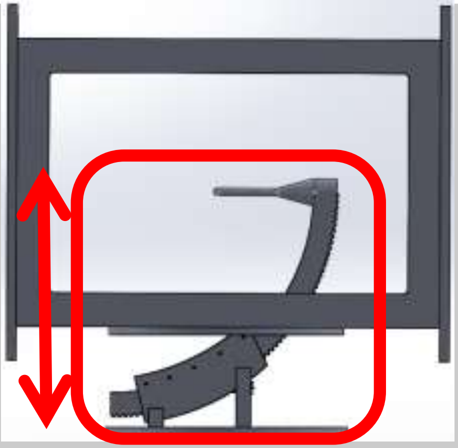
$\tau_{REQ} = \tau_{DESIGN} + I_{PITCH,TOT}\,\alpha$

Major Pitch Torque Concern

- Addition of lift and drag force from simple model
  - Lift = 137 N   Drag = 4.76 N
- Force applied about 0.5 m from gearing
- Torque applied to pitch: 70.9 N-m
- Total torque applied: 73.8 N-m
- Still FEASIBLE with gearing and motor research



Torque Estimates for Roll, Yaw and Plunge

	Roll	Yaw	Plunge
DoF			
Inertia Assumed	Flat Plate	Flat circular plate	Mass estimate (35 kg)
Estimated Torque	0.053 N*m	4.35 N*m	8.76 N*m
Feasibility	Yes	Yes	Yes with gearing

Overall Assumptions:
- Motors drive each DoF directly
- Torque from friction ignored
- Aerodynamic forces negligible

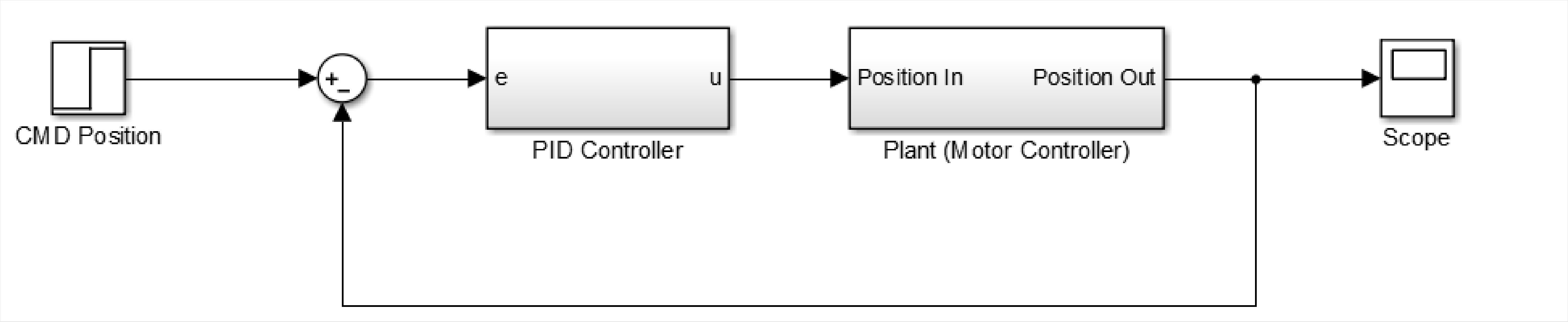
Based on research for motors, all FEASIBLE

Encoder Considerations

- Yaw and Pitch accuracy requirement = 0.1°
- Roll accuracy requirement = 0.5°
- Plunge accuracy requirement = 0.5 mm
- Encoder resolution must be better than the degree of freedom requirements scaled by gear ratio
- Pulses Per Revolution (PPR)
  - Encoder resolution is defined by 360°/PPR
  - An encoder with 7,200 PPR has a resolution of 0.05°
- Measurement capability: **FEASIBLE**

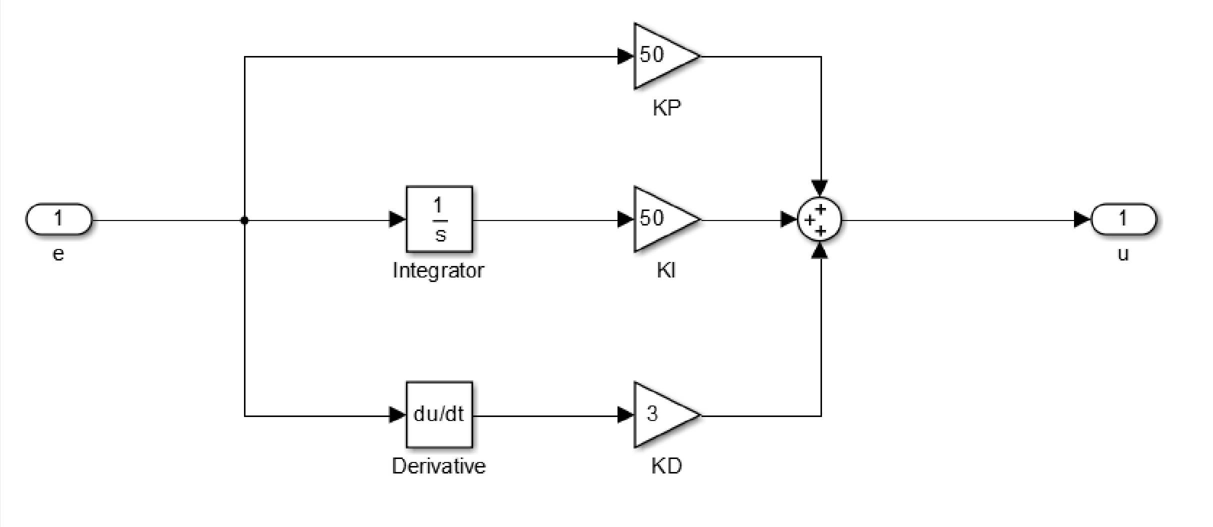
Control Law Design

- Implementing Simulink to model control of a degree of freedom
- The goal of the model is to simulate command of a motor controller
  - Commercial-off-the-shelf (COTS) motor controller



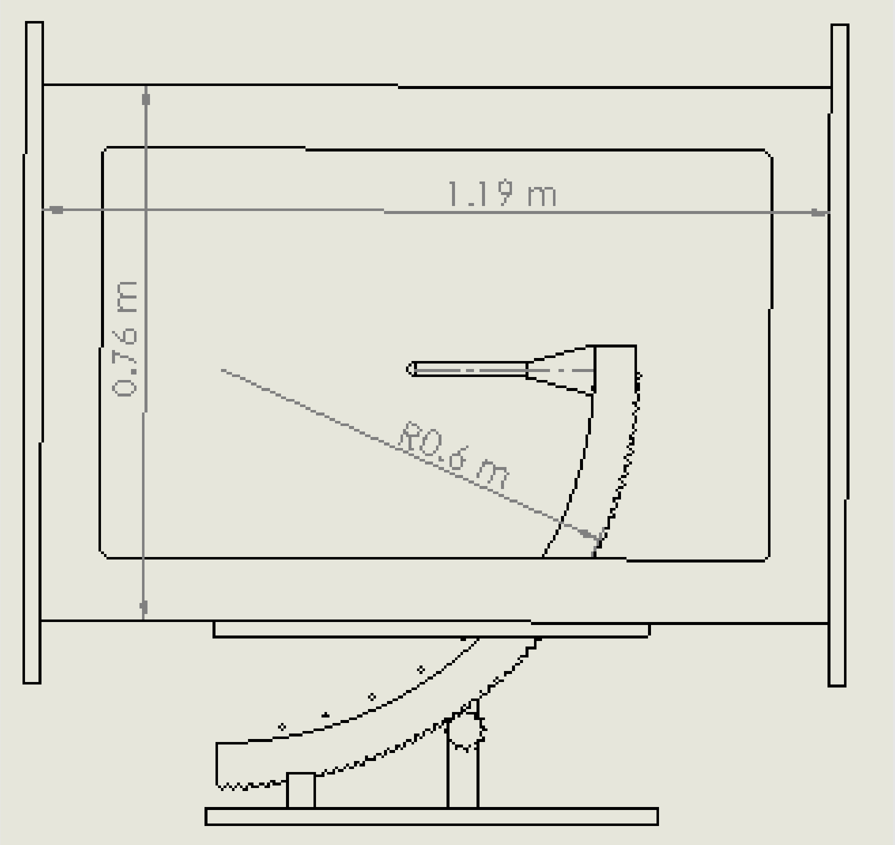
Control Law Design

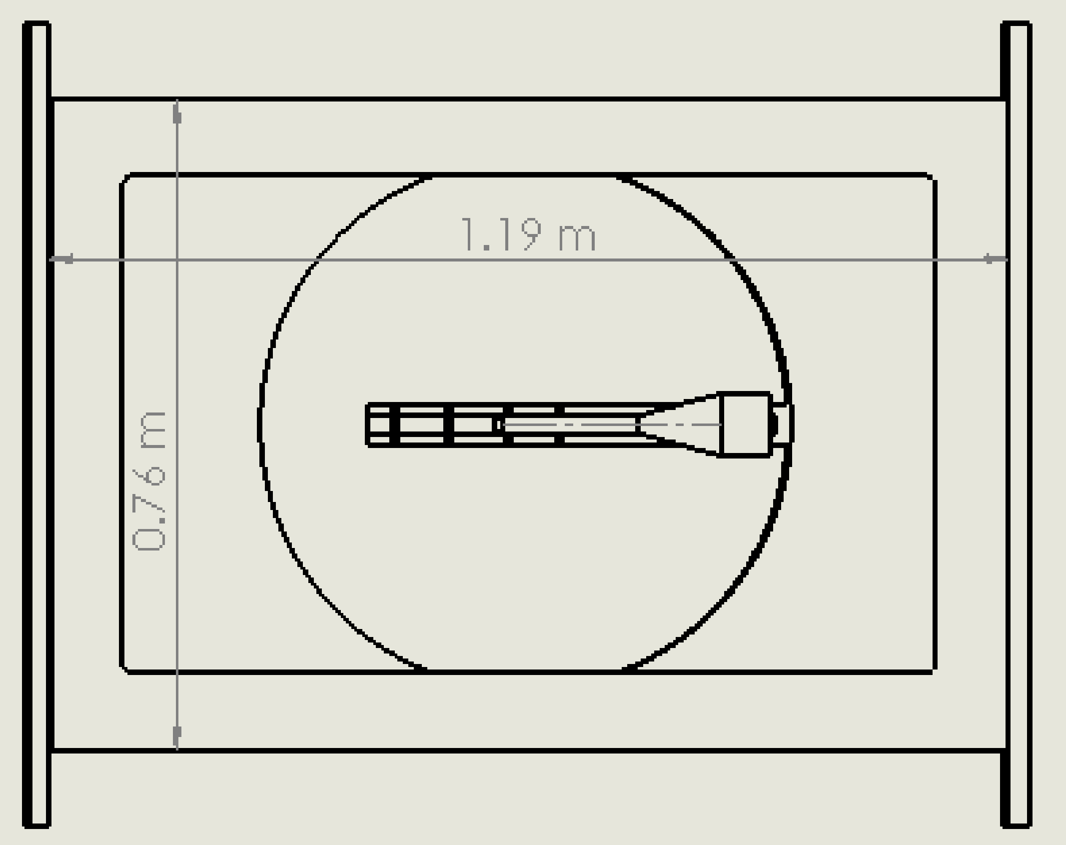
- Control Law Design
  - Develop PID control law gains for outer control loop
- Control Law Simulation
  - Simulation of system mechanisms, linkages, motors, and motor controllers
  - Develop high fidelity model to test and validate control law design
- Control Law Design: FEASIBLE
- Control Law Simulation: FEASIBLE



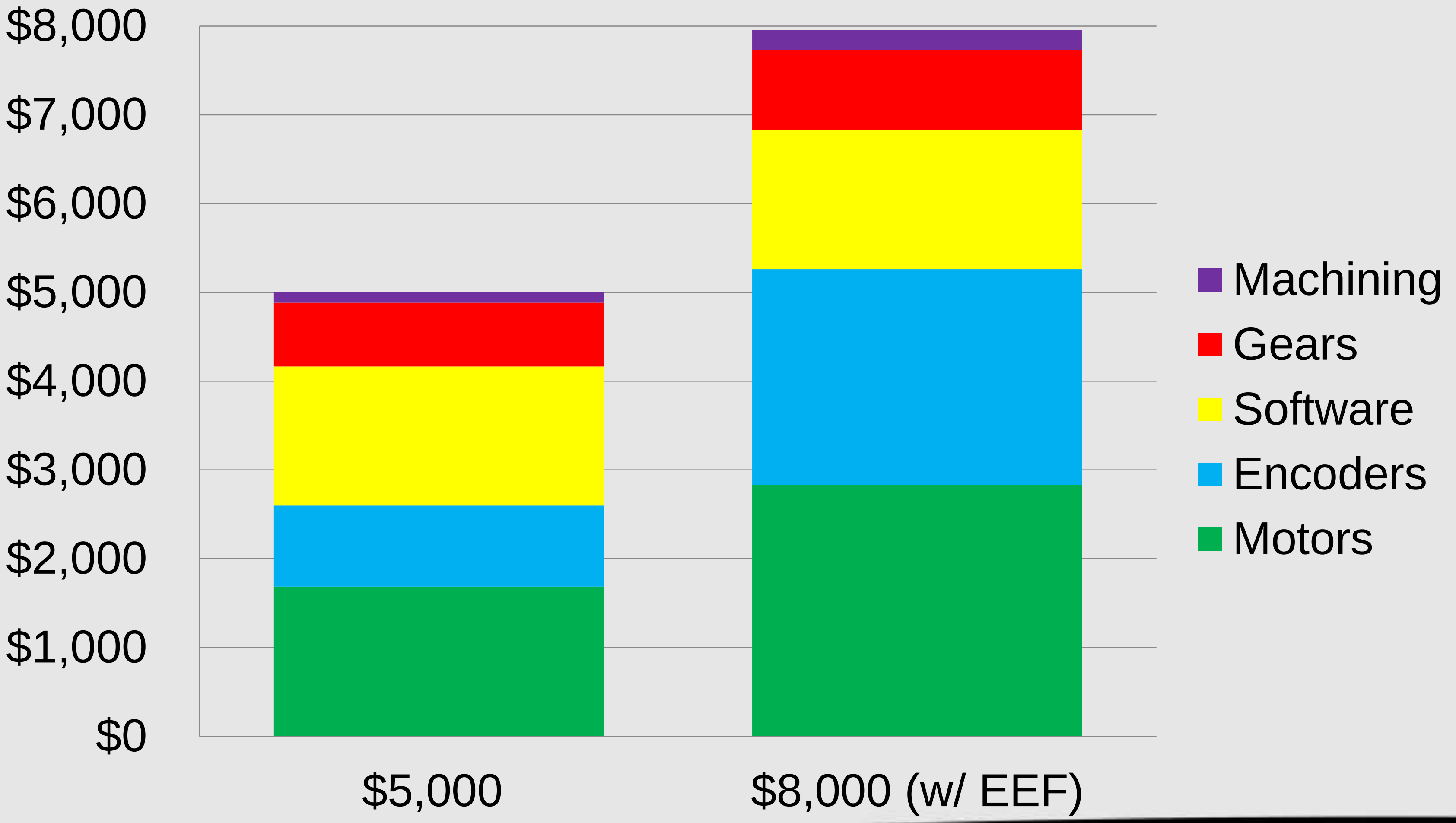
Summary

Design Overview

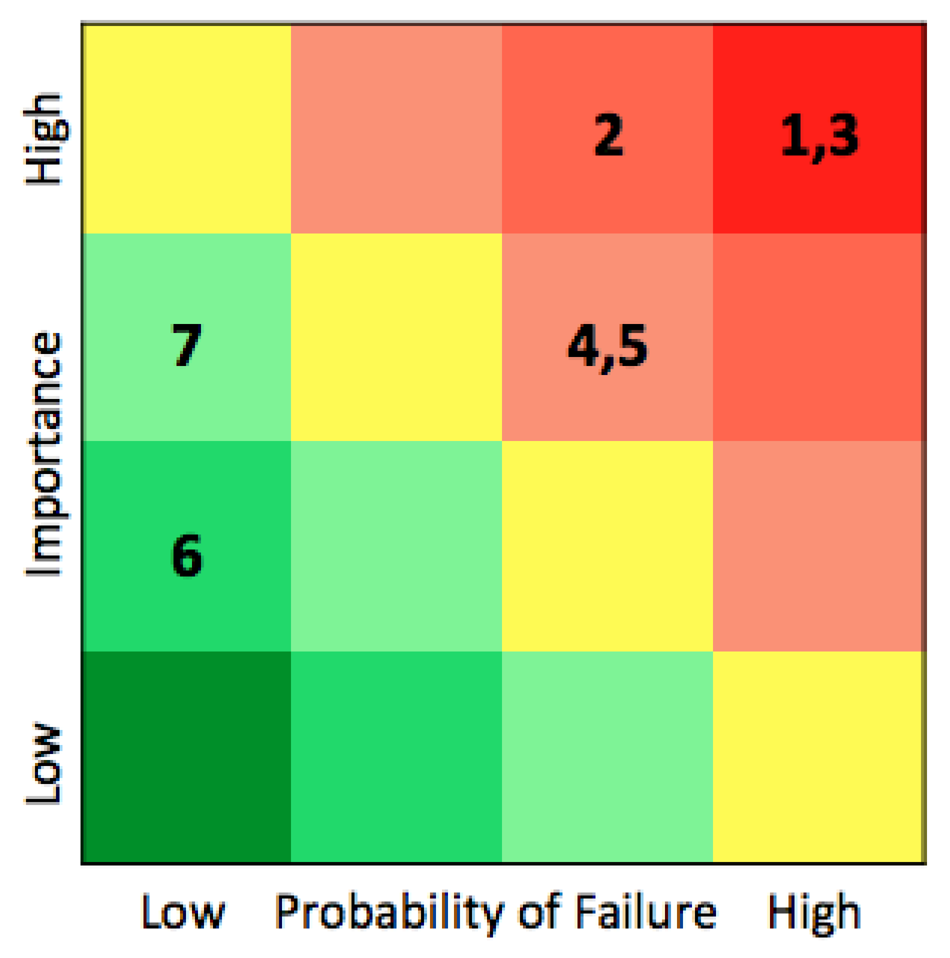




Financial Breakdown



Logistical Risks for Success



1. Finding needed motors within budget
2. Finding needed sensors within budget
3. Delivery date of large servo motors
4. Software development time
5. Mechanical Linkages (breaking/slipping)
6. Access to wind tunnel facilities
7. Integration with wind tunnel frame

Feasibility vs. Continue to Study

	Feasible	Continue to Study
Yaw, Roll, Pitch, Plunge capability	X	X
Manufacturing Methods	X	X
Motor Torque Estimates	X	
Control Law Simulations	X	X
Encoder Capabilities	X	
Error Propagation	X	X

Critical Path Moving Forward

Final Feasibility Studies — Characterize and understand total system with higher fidelity (backlash, etc.)

Control Law Simulation and Design — Properly setup outer control loop with more accurate inner loop for motor controller

→ **Component Selection** — Motors, Encoders, Motor Controllers, DAQs

→ **Software Development (Control Law and LabVIEW VI)**

→ **Manufacturing and Subsystem Test**

→ **System Integration and Control**

Questions?

References

- (A) "BLK42," *Anaheim Automation,* URL: https://www.anaheimautomation.com/products/brushless/brushless-motor-item.php?sID=366&pt=i&tID=96&cID=22
- (B) "BLK24," *Anaheim Automation,* URL: https://www.anaheimautomation.com/products/brushless/brushless-motor-item.php?sID=368&pt=i&tID=96&cID=22
- (C) "Gear Technical Reference" *Kohara Gear Industry CO*, URL: http://www.khkgears.co.jp/en/gear_technology/pdf/gear_guide1.pdf
- (D) "Geometry Factors for Determining the Pitting Resistance and Bending Strength of Spur, Helical and Herringbone Gear Teeth" *AMERICAN GEAR MANUFACTURERS ASSOCIATION*
- (E) "Gear Presentation", URL: http://www1.coe.neu.edu/~hamid/MIMU550/Gears%20presentation.ppt
- (F) "Selection of Gear Materials", URL: http://www.ecs.umass.edu/mie/labs/mda/dlib/machine/gear/gear_mat.html

Picture References

- *(A) "Miniature Anti-backlash Clamp Hub Gear" Reliance Precision Limited*, URL: http://www.reliance.co.uk/shop/products.php?10734&cPath=32_355_365
- (B) "How Gears Work" *Spur Gears,* URL: http://science.howstuffworks.com/transport/engines-equipment/gear2.htm
- [C] Riesselmann, George. "Applying fail-safe brakes to stop and hold ,"*Machine Design,* URL: http://machinedesign.com/technologies/applying-fail-safe-brakes-stop-and-hold
- (D) Mendolia, J. "Choosing servomotor brakes," *Machine Design,* URL: http://machinedesign.com/technologies/choosing-servomotor-brakes
- (E) "Pressure Angle", *Wikipedia* URL: http://www.wikiwand.com/en/Pressure_angle
- (F) "Backlash", *Wikipedia* URL: *https://en.wikipedia.org/wiki/Backlash_(engineering*)

Backup Slides

Backup Slides Overview

- Trade Study Results and Design Calcualtions
- Electrical and Software Overview
- Inertia Calculations
- Simulink/Modeling
- Gearing Information
- Motor Considerations
- Safety and Failsafes
- Functional and Design Requirements
- Tunnel Specifications and Drawings
- Delivery Dates for Products

Trade Study Results - Design

- weights
- size: 1 = unusable b/c of blockage, 5 = gets the job done, 10 = ~0% blockage
- range: 1 = does not satisfy any DoF, 2.5 = satisfies 1 DoF, 5 = satisfies 2 DoF, 7.5 = satisfies 3 DoF,

10 = perfectly satisfies requirement

- manufacturability: 1 = high cost & high resources, 4 = high cost & low resources, 6 = low cost & high resources 10 = low cost & low resources
- number of linkages: 1 = 10-12, 2.5 = 8-9, 5 = 7, 7.5 = 5-6, 10 = 0-4

	Weight	Sting Arm	Sting Crescent	3-Strut Motor	3-Strut Plate
Size	25%	9	9	5	5
Range	35%	9	9	5	6.5
Manufacturability	30%	6	6	4	3
Linkages (# assumed)	10%	1 (10)	10 (4)	2.5 (8)	1 (10)
Total	100%	7.33	8.2	4.45	4.53

Trade Study Results - Motors

	Weight	Stepper Motor	Piezoelectric Motor	Servo Motor
Stall/Max Torque	25%	8	8	7
Resolution	25%	6	10	9
Angular Rate	20%	5	6	8
Cost	20%	8	2	5
Size	10%	8	8	6
Total	100%	6.9	6.9	7.2

	Stepper Motor	Piezoelectric Motor	Servo Motor
Pitch	1	0	1
Roll	1	1	0
Yaw	1	1	1
Plunge	0	0	1
Total	3	2	3

Arm Size Confirmation

```
%%DEFINING CONSTANTS%%
h=0.76; %meters, hieght of test section
a=.0; %meters, change in model aerodynamic center
Cr=0.6; %meters, radius
t=.06; %meters, radial thickness
delta_pl=0.10; %meters, of motion in either direction for plunge
delta_p=30; %degrees, motion in either direction for pitch
delta_y=30; %degrees, motion in either direction for yaw
delta_r=40; %degrees, motion in either direction for roll


%No Pitch No Plunge%
Clear0=(h/2)-a %Clearance From Top and Bottom


%No Pitch 10cm Plunge Both Directions%
Clear1=(h/2)-delta_pl-a %with 10cm plunge downward
Clear2=(h/2)+delta_pl-a %with 10cm plunge upward
```

Arm Size Confirmation (2)

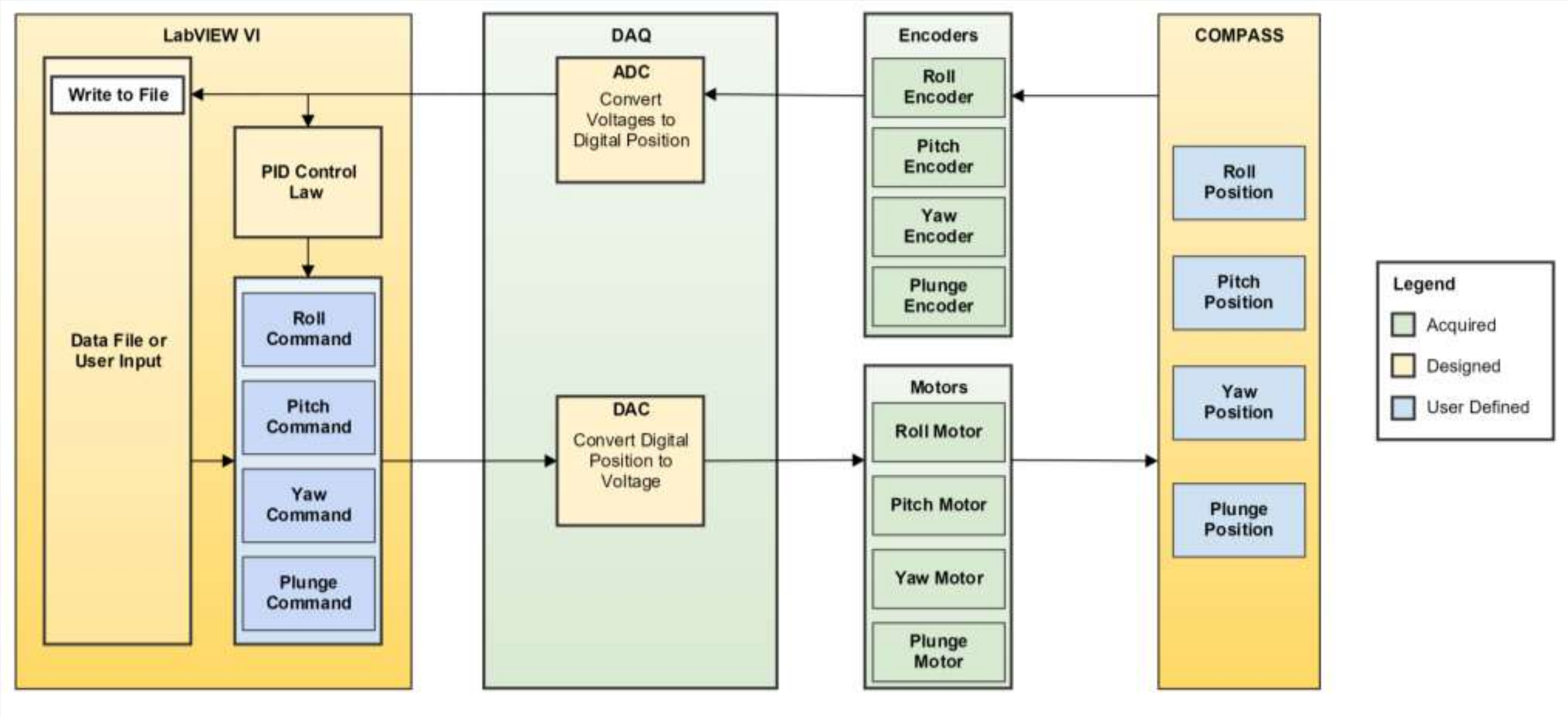
```
%30 Degree Pitch Down No Plunge Clearance%
Clear3=(h/2)-(Cr*cosd(delta_p))-a %from bottom of the tunnel
Clear4=(h/2)-(Cr*sind(delta_p))+a %from top of the wind tunnel


%30 Degree Pitch Downward Postive Plunge%
Clear5=(h/2)-((Cr*sind(delta_p))+delta_pl)+a
%from top of the wind tunnel with 10cm plunge up
Clear6=(h/2)-(Cr*cosd(delta_p)-delta_pl)-a
%from bottom of the wind tunnel with 10cm plunge up


%30 Degree Pitch Downward Negative Plunge%
Clear7=(h/2)-((Cr*sind(delta_p))-delta_pl)+a
%from top of the wind tunnel with 10cm plunge down
Clear8=(h/2)-((Cr*cosd(delta_p))+delta_pl)-a
%from bottom of the wind tunnel
```

Arm Size Confirmation (3)

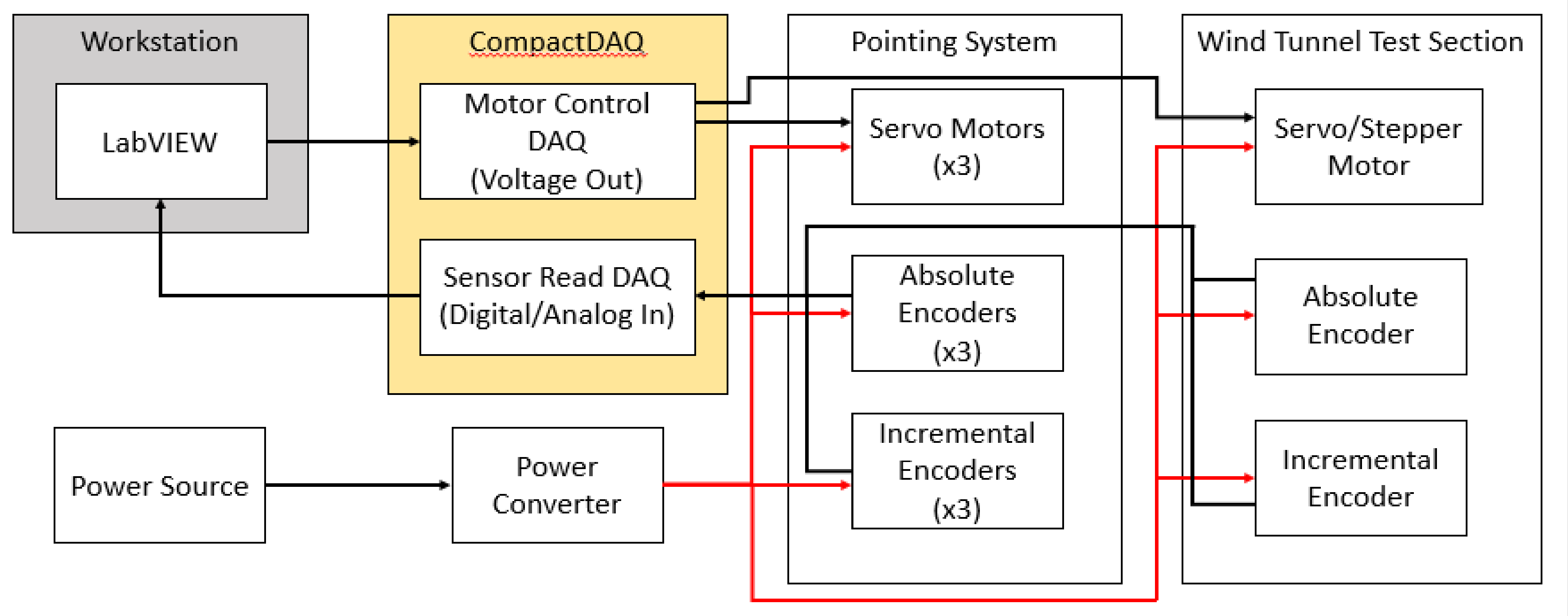
```
%%Slot Distance Calculation%%
h0=(h/2)+delta_pl-a
h1=(h/2)-delta_pl-a;
r0=.57;
r1=.63;
theta0=acosd(h0/r0)
theta1=acosd(h1/r1)
x0=r0*sind(theta0)
x1=r1*sind(theta1)
xcrit=x1-x0
```

Electrical and Software Overview

Software Overview



Electrical Overview



Additional Torque and Inertia Calculations

Pitch Inertia Calculations

➔ Assume thick hoop for crescent and then cut mass in half to account for only part of crescent

$r_2 = 0.48\,m \qquad r_1 = 0.42\,m \qquad h = 0.025\,m \qquad \rho = 2712.6\,\frac{kg}{m^3}$

$m_p = \rho\pi(r_2^2 - r_1^2)h = \left(2712.6\,\frac{kg}{m^3}\right)\pi[(0.48\,m)^2 - (0.42\,m)^2](0.025\,m)$

$m_p = 11.50\,kg \qquad m_p = \frac{m_P}{2} \qquad m_p = 5.75\,kg$

$I_{PITCH,X} = \frac{1}{2}m_p(r_2^2 + r_1^2) \qquad I_{PITCH,X} = \frac{1}{2}(5.75\,kg)[(0.48\,m)^2 + (0.42\,m)^2]$

$I_{PITCH,X} = 1.17\,kg\,m^2 \qquad I_{PITCH,TOT} = I_{PITCH,X} + I_{MOD,X}$

$b = 0.5\,m \qquad c = 0.1\,m \qquad t = 0.012\,m \qquad V = bct = 0.0006\,m^2$

$m_M = \rho V \qquad \rho = 2712.6\,\frac{kg}{m^3} \qquad m_M = 1.63\,kg$

$I_{MOD,X} = \frac{1}{12}m_M(t^2 + c^2) \qquad I_{MOD,X} = \frac{1}{12}(1.63\,kg)[(0.012\,m)^2 + (0.1\,m)^2]$

$I_{MOD,X} = 0.00138\,kg\,m^2 \qquad I_{PITCH,TOT} = 1.17\,kg\,m^2$

Yaw Inertia Calculations

$r = 0.325\,m \quad h = 0.025\,m \quad \rho = 2712.6\,\frac{kg}{m^3}$

$m_Y = \rho\pi r^2 h = \left(2712.6\,\frac{kg}{m^3}\right)\pi(0.325\,m)^2(0.025\,m) \quad m_Y = 11.43\,kg$

$I_{YAW} = \frac{1}{2}m_Y r^2 \quad I_{YAW} = \frac{1}{2}(11.43\,kg)(0.325\,m)^2$

$I_{YAW} = 0.604\,kg\,m^2 \quad I_{YAW,TOT} = 2I_{YAW} + I_{MOD,Z} + I_{PITCH,Z}$ -> Yaw and floor plates

$I_{PITCH,Z} = \frac{1}{12}m_P[3(r_2^2 - r_1^2) + h^2] = \frac{1}{12}(5.75\,kg)[3[(0.48\,m)^2 + (0.42\,m)^2] + (0.025\,m)^2]$

$I_{PITCH,Z} = 0.585\,kg\,m^2 \quad b = 0.5\,m \quad c = 0.1\,m \quad t = 0.012\,m \quad V = bct = 0.0006\,m^2$

$m_M = \rho V \quad \rho = 2712.6\,\frac{kg}{m^3} \quad m_M = 1.63\,kg$

$I_{MOD,Z} = \frac{1}{12}m_M(b^2 + c^2) \quad I_{MOD,Z} = \frac{1}{12}(1.63\,kg)[(0.5\,m)^2 + (0.1\,m)^2]$

$I_{MOD,Z} = 0.0353\,kg\,m^2 \quad I_{YAW,TOT} = 1.83\,kg\,m^2$

Roll Inertia and Torque Calculations

$b = 0.5\,m$ $\quad c = 0.1\,m$ $\quad t = 0.012\,m$ $\quad V = bct = 0.0006\,m^2$

$m_M = \rho V$ $\quad \rho = 2712.6\,\frac{kg}{m^3}$ $\quad m_M = 1.63\,kg$

$I_{ROLL} = \frac{1}{12} m_M (t^2 + b^2)$ $\quad I_{ROLL} = \frac{1}{12}(1.63\,kg)[(0.012\,m)^2 + (0.5\,m)^2]$

$I_{ROLL} = 0.0339\,kg\,m^2$ $\quad \omega = 64\frac{°}{s}$ $\quad \Delta\theta = 90°$

$\Delta\theta = \frac{90\pi}{180} = \frac{1}{2}\alpha t^2$ $\quad t = \frac{\Delta\theta}{\omega} = \frac{90°}{64\frac{°}{s}} = 1.41\,s$ $\quad \tau_{REQ,ROLL} = 0.0536\,N\,m$

$\alpha = 1.58\,\frac{rad}{s^2}$ $\quad \tau_{REQ} = \tau_{DESIGN} + I_{ROLL}\alpha$

Plunge Mass and Torque Calculations

- Mass from pitch, yaw, and model
- Added mass estimated from need for motors and linkages

$m_M = 1.63\ kg \qquad m_P = 5.75\ kg \qquad m_Y = 11.4\ kg \qquad m_{SYS} = m_M + m_P + m_Y$

$m_{SYS} = 18.8\ kg$

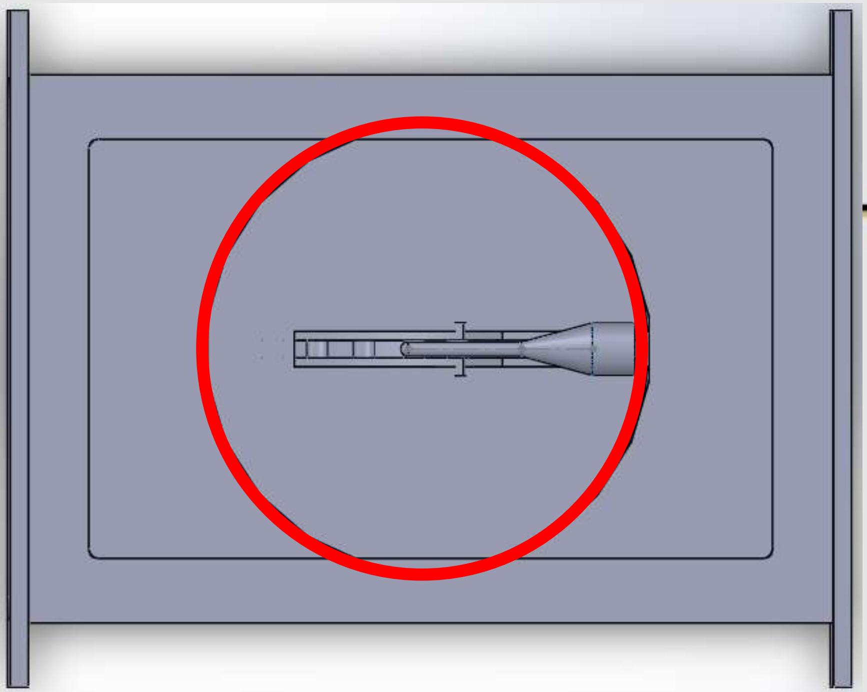
$m_{SYS} = 35.0\ kg \qquad v = 70\,\frac{mm}{s} = 0.07\,\frac{m}{s} \qquad D = 20\ cm = 0.2\ m$

$D = 0.2\ m = \frac{1}{2}at^2 \qquad t = \frac{D}{v} = \frac{0.2}{0.07} = 2.86\ s \qquad a = 0.0490\ \frac{m}{s^2}$

$F_N = m_{SYS}a = 1.72\ N \qquad F_g = m_{SYS}g = 343\ N$

$F_{REQ} = F_N + F_g = 345\ N \qquad \tau_{REQ} = F_{REQ}d = 345\ N * 0.0254\ m = 8.76\ N\ m$

Yaw Torque Estimate



- Assumptions:
  - Motor assumed be driving yaw directly
  - Torque from friction ignored
  - Moment of Inertia of model, crescent, two yaw plates
  - 60 degree rotation
  - 64 degrees/sec rate

$$I_{YAW,TOT} = 1.828\ kg\ m^2 \qquad \omega = 64 \frac{°}{s} \qquad \Delta\theta = 60°$$

$$\Delta\theta = \frac{60\pi}{180} = \frac{1}{2}\alpha t^2 \qquad t = \frac{\Delta\theta}{\omega} = \frac{60°}{64\frac{°}{s}} = 0.938\ s$$

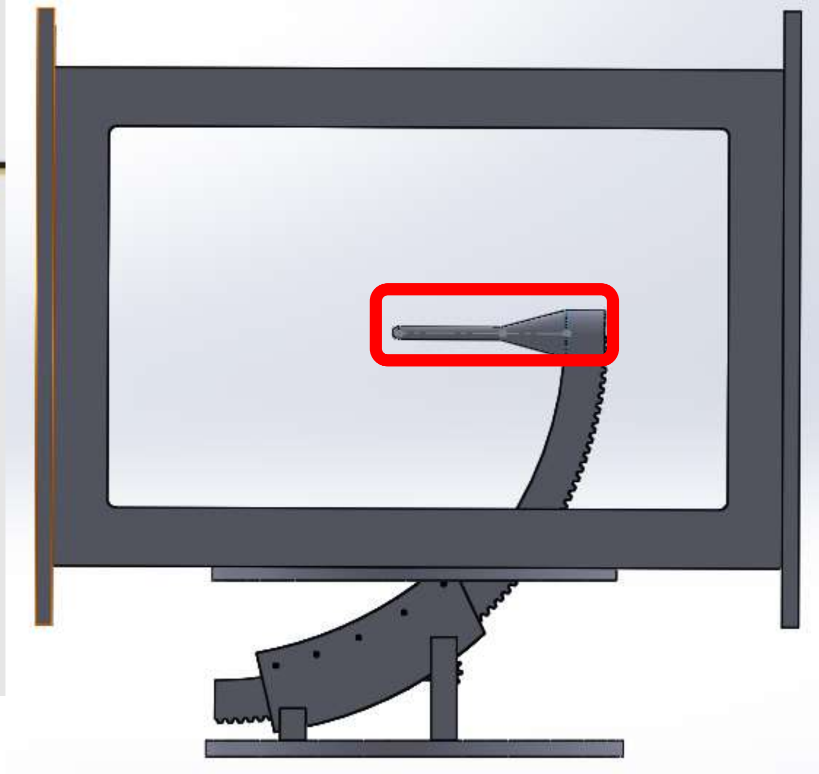
$$\alpha = 2.38\ \frac{rad}{s^2} \qquad \tau_{REQ} = \tau_{DESIGN} + I_{YAW}\alpha$$

$$\boxed{\tau_{REQ,YAW} = 4.35\ N\ m}$$

FEASIBLE

Roll Torque Estimate

- Assumptions:
  - Motor assumed be driving roll directly
  - Torque from friction ignored
  - Moment of Inertia of model in roll
  - 90 degree rotation
  - 64 degrees/sec rate



$I_{ROLL} = 0.0339\ kg\ m^2$ $\qquad \omega = 64 \frac{°}{s}$ $\qquad \Delta\theta = 90°$

$\Delta\theta = \frac{90\pi}{180} = \frac{1}{2}\alpha t^2$ $\qquad t = \frac{\Delta\theta}{\omega} = \frac{90°}{64\frac{°}{s}} = 1.41\ s$

$\alpha = 1.58\ \frac{rad}{s^2}$ $\qquad \tau_{REQ} = \tau_{DESIGN} + I_{ROLL}\alpha$

$\tau_{REQ,ROLL} = 0.0536\ N\ m$

FEASIBLE

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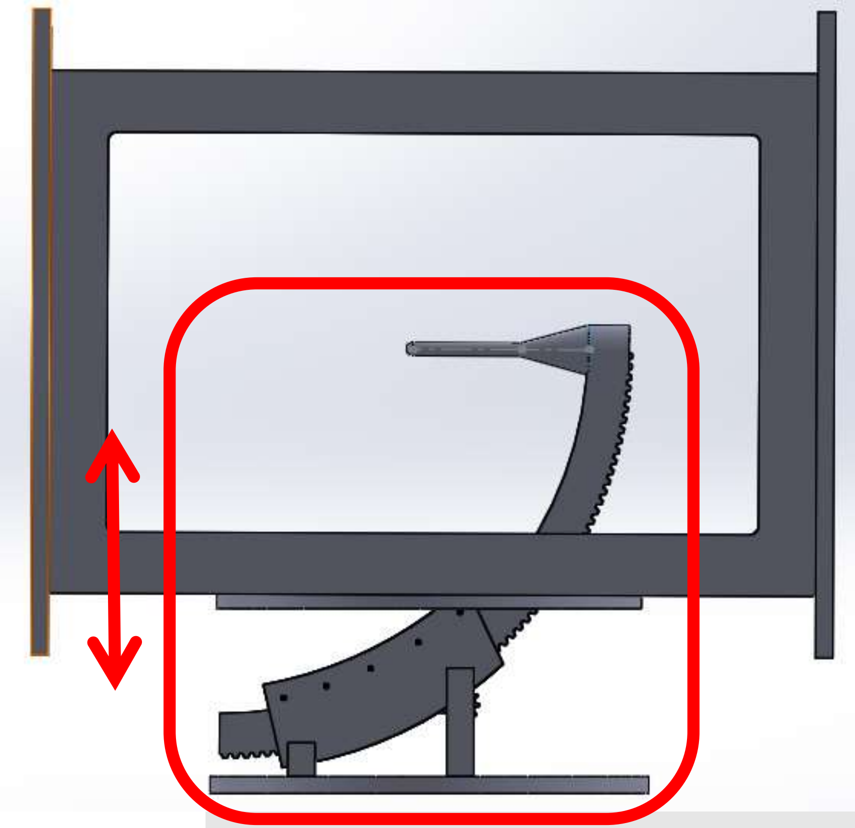
Plunge Torque Estimate

- Assumptions:
  - Motor assumed be driving plunge directly
  - Friction forces ignored
  - Force from mass of pitch, yaw, model, motors
  - 10 cm of travel
  - 64 mm/sec rate



$$m_{SYS} = 35.0\ kg \qquad v = 70\frac{mm}{s} = 0.07\frac{m}{s} \qquad D = 20\ cm = 0.2\ m$$

$$D = 0.2\ m = \frac{1}{2}at^2 \qquad t = \frac{D}{v} = \frac{0.2}{0.07} = 2.86\ s \qquad a = 0.0490\ \frac{m}{s^2}$$

$$F_N = m_{SYS}a = 1.72\ N \qquad F_g = m_{SYS}g = 343\ N$$

FEASIBLE

$$F_{REQ} = F_N + F_g = 345\ N \qquad \tau_{REQ} = F_{REQ}d = 345\ N * 0.0254\ m = \boxed{8.76\ N\ m}$$

University of Colorado at Boulder

COMPASS

Geared Torque Estimates - High Speed

- Assume number of teeth of internal motor gear
- Assume 90% efficiency at 1,500 RPM

Gearing ratio: R

Number of teeth: N

Angular Velocity: w

For $w_{input} = 1500\ RPM = 9{,}000\ {}^{\circ}/_{s}$

$w_{output} = 64\ {}^{\circ}/_{s} \approx 11\ RPM$

$R = \frac{9{,}000\ {}^{\circ}/_{s}}{64\ {}^{\circ}/_{s}} \approx \boxed{140 \rightarrow Not\ Feasible}$

Gear Ratio = 1 : 10 : 140

10x 14x

Gear Teeth Ratios = Internal motor: Gear 1: Gear 2 = 10 : 100 : $\boxed{\text{1,400 teeth} \rightarrow Not\ Feasible}$

$R = \frac{w_{input}}{w_{output}} = \frac{N_{output}}{N_{input}}$

$\tau_{eff} = \tau_{motor} * R$

BLK42 series motor $\tau_{rated} = 6\ N \cdot m$

$\tau_{eff} = (6\ Nm) * (90\%) * 140 \approx \boxed{750\ Nm \rightarrow Over\text{-}qualified}$

Geared Torque Estimates - Low Speed

- Assume Gear Ratio =20 (Feasibility shown in solidworks model)
- Assume 80% efficiency for less than 1,000 RPM

Gearing ratio: R For $w_{input} = 215\ RPM \approx 1{,}280\ ^{\circ}/_{s}$ $w_{output} = 64\ ^{\circ}/_{s} \approx 11\ RPM$

Angular Velocity: w $R = \boxed{20 = \frac{1{,}280\ ^{\circ}/_{s}}{64\ ^{\circ}/_{s}} \rightarrow Feasible}$

Number of teeth: N

BLK24 series motor $\tau_{rated} = 81\ oz \cdot in \approx 0.57\ N \cdot m$

BLK42 series motor $\tau_{rated} = 850\ oz \cdot in \approx 6\ N \cdot m$

$R = \frac{w_{input}}{w_{output}} = \frac{N_{output}}{N_{input}}$ $\tau_{eff}^{BLK24} = (0.57\ Nm) * (80\%) * 20 \boxed{\approx 9.1\ Nm \rightarrow \text{Feasible}}$

$\tau_{eff} = \tau_{motor} * R$ $\tau_{eff}^{BLK42} = (6\ Nm) * (80\%) * 20 \boxed{\approx 96\ Nm \rightarrow \text{Feasible}}$

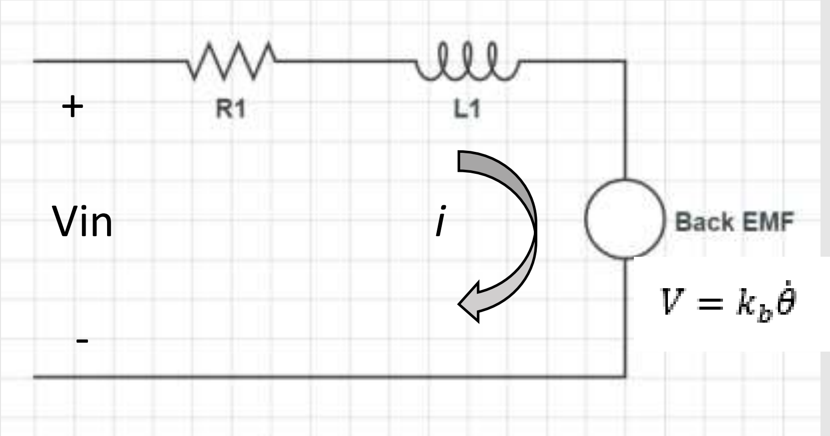
Simulink Model

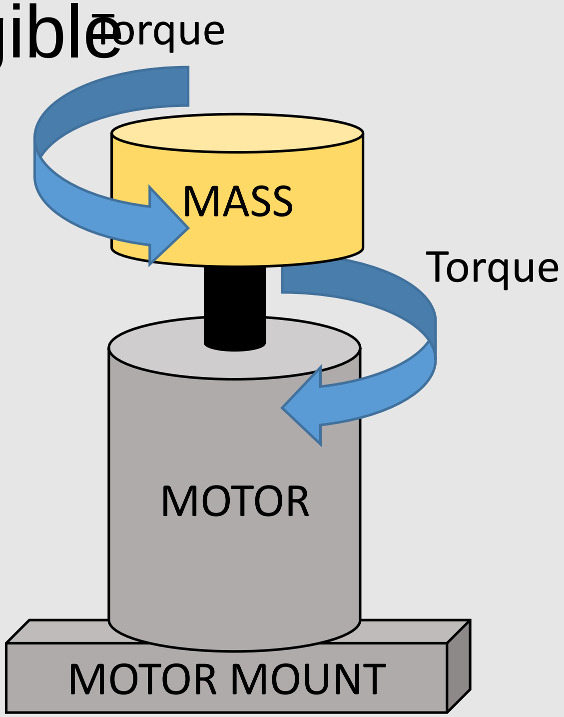
Plant Transfer Function

- Assumes some mass being directly driven by a motor
- Assumes equal and opposite torque, friction is negligible
- Motor modeled as simple circuit

$$J_{EFF}\ddot{\theta} = \tau \qquad \tau = k_\tau i \qquad V_{IN} - iR - L\frac{di}{dt} - k_b\dot{\theta} = 0$$

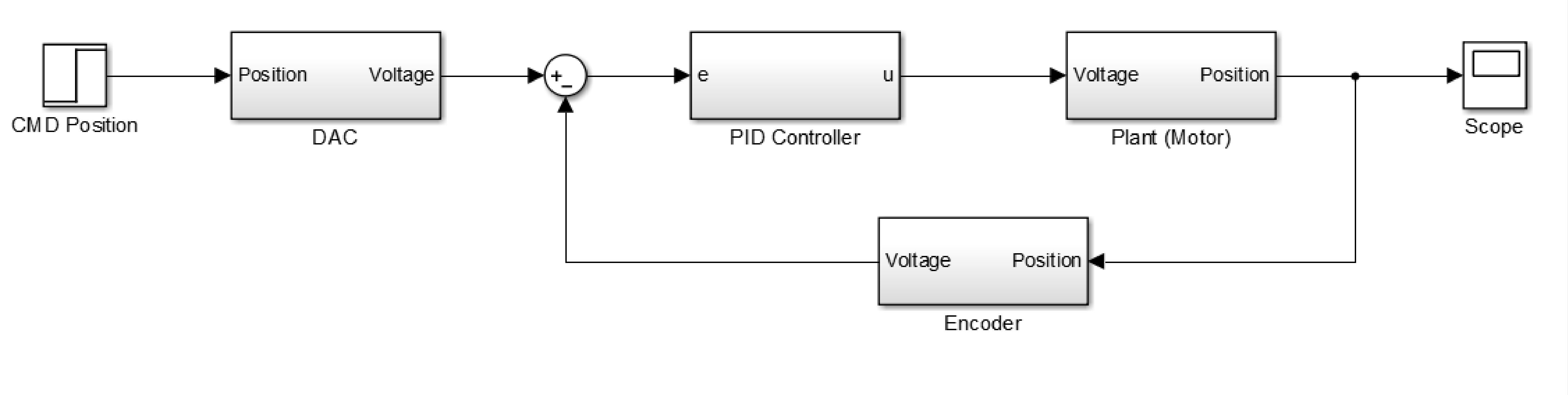
$$\frac{\theta(s)}{V(s)} = \frac{k_\tau}{J_{EFF}L}\,\frac{1}{s\left(s^2 + \frac{R}{L}s + \frac{k_b k_\tau}{LJ_{EFF}}\right)}$$





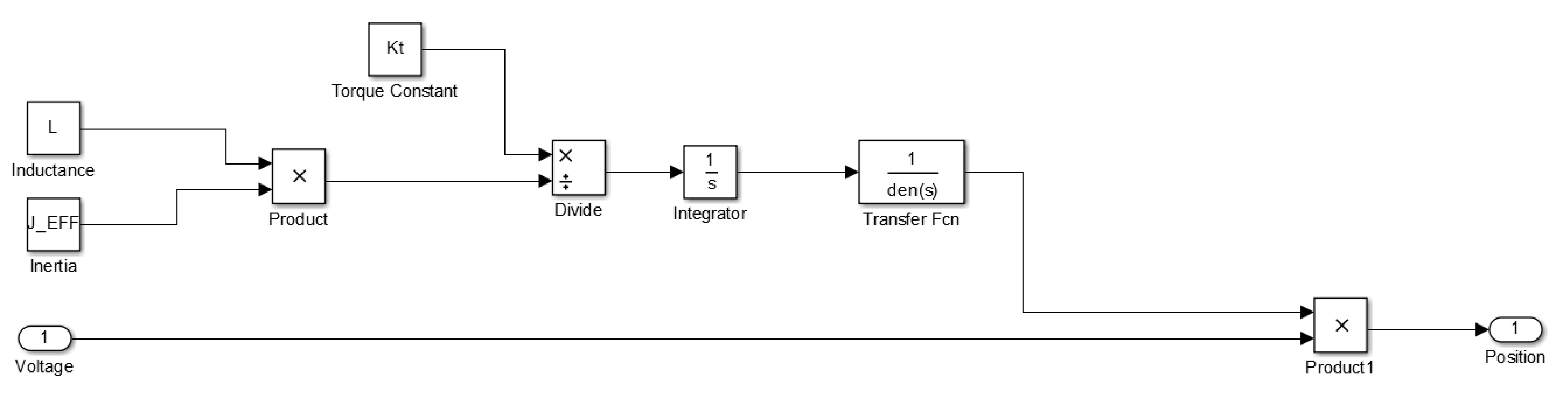
Simulink Models (Motor Controller)

- Inner control loop of Simulink model



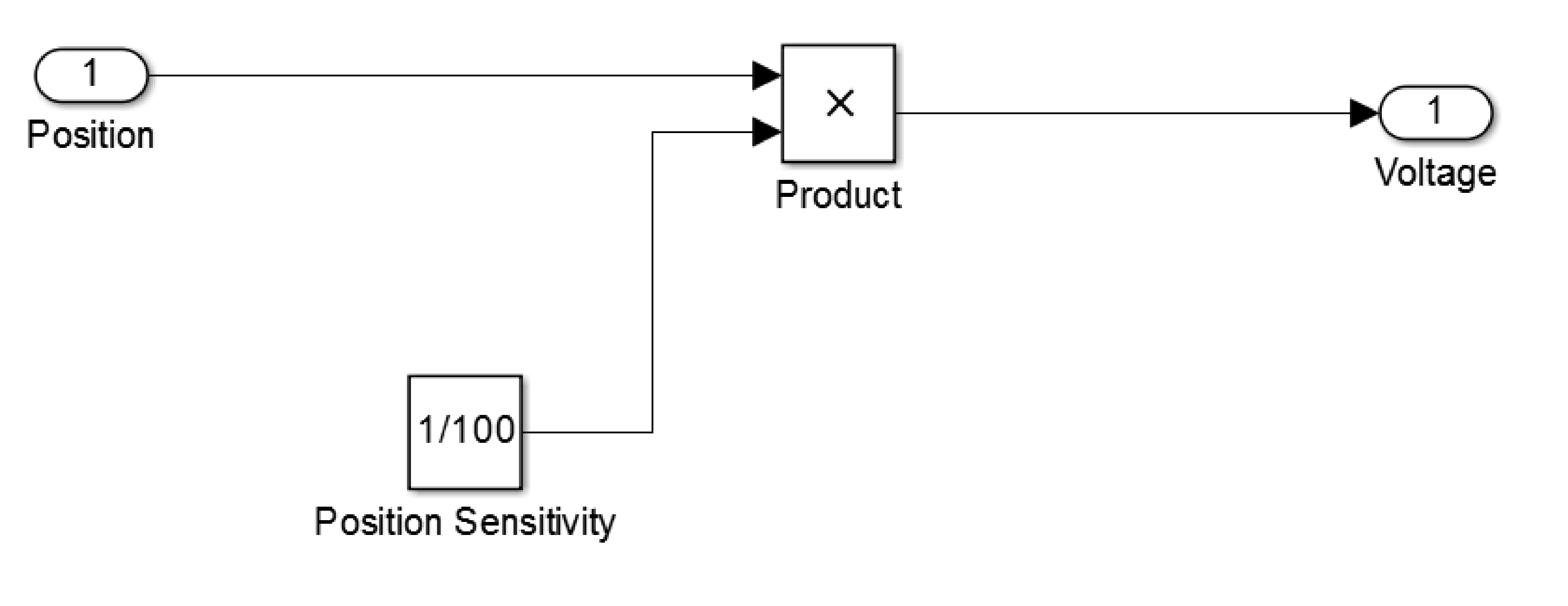
Simulink Models (DC Brushless Motor)

- Simple model for DC Brushless Motor



Simulink Models (Encoder/DAC Subsystem)

- Simple model for Encoder/DAC



Gear Calculations

Gear Ratio

$$minimum\ teeth = \frac{2}{\sin^2(25)}$$
$$minimum\ teeth = 12$$

Radius of Gear

$$\frac{1200}{x} = \frac{240}{12}$$
$$x = \frac{1200(12)}{240}$$
$$x = 60\ mm$$

$$°\ per\ tooth = \frac{360°}{12}$$
$$°\ per\ tooth = 30°$$

Gear Tooth Size

Size:

$$module = \frac{C}{\pi(\#\, of\, teeth)}$$
$$module = \frac{0.1885}{\pi(12)}$$
$$module = 0.005$$

$$tooth\ depth = 2.25(module)$$
$$tooth\ depth = 2.25(0.005)$$
$$tooth\ depth = 11.25\ mm$$

11.25 mm is too large of a tooth depth.
6.25 mm is much more feasible with 30 mm gear.

Lift Force and Transmitted Load

$$L = .5(\rho V_{max}^2 S C_L$$
$$L = .5(1)(65^2)(0.5)(1.3)$$
$$L = 137.3$$

$$F_t = \frac{2000(T)}{d_w}$$
$$F_t = \frac{2000(9.1)}{1.0002}$$
$$F_t = 15164\ N$$

F_t = Transmitted Load

T = Motor Torque

d_w = pitch

Gear Tooth Strength

$$\sigma_{allow} = \frac{S_t K_L}{K_t K_R}$$
$$\sigma_{allow} = \frac{(131\ MPa)(2)}{(1.1)(1.5)}$$
$$\sigma_{allow} = 158\ MPa$$

$$\sigma_{allow} = \frac{S_t K_L}{K_t K_R}$$
$$\sigma_{allow} = \frac{(131\ MPa)(2)}{(1.1)(1.0)}$$
$$\sigma_{allow} = 238\ MPa$$

S_t = Bending Strength K_T = Temperature Factor
K_L = Life Factor K_R = Reliability Factor

Stress on Tooth is under allowable so it is **FEASIBLE**

$$\sigma = F_t K_o K_v \frac{1.0}{bm} \frac{K_s K_m}{J}$$
$$\sigma = (15164)(1)(1.1)(\frac{1.0}{(.025)(.005)} \frac{(1.3)(1)}{1.1136}$$
$$\sigma = 156\ MPa$$

F_t = transmitted tangential load m = metric module
K_0 = Overload Factor K_s = Size Factor
K_V = Velocity Factor K_m = Mounting Factor
b = face width J = Geometry Factor

where

$$J = 0.367 \ln(N) + 0.2016$$
$$N = \#\ of\ teeth$$

Gear Backlash

Circumference of Crescent

$$C = 2\pi r$$
$$C = 2\pi(0.603m)$$
$$C = 3.789\, m$$

0.08 mm gear backlash

$$\frac{0.00008}{3.789} = 2.111 * 10^{-5}$$
$$pitch\,\Delta° = (360°)2.11 * 10^{-5}$$
$$pitch\,\Delta° = 0.0076°$$

0.29mm gear backlash

$$\frac{0.00029}{3.789} = 7.654 * 10^{-5}$$
$$pitch\,\Delta° = (360°)654 * 10^{-5}$$
$$pitch\,\Delta° = 0.028°$$

Uncertainty in pitch is below our 0.1° accuracy so **FEASIBLE**

Motor Considerations

Motor Torque Considerations - BLK42 series

- Pitch, Yaw, Plunge need larger motor than needed for roll
  - NEMA 42 is class of large servo with 6.0 N-m rated torque (A)
- Plunge will likely require gearing
  - Gear ratio of 5:1 plenty
- Roll with small servo/stepper motor
- Feasible in all degrees of freedom

NEMA 42 Brushless DC Motor

Motor Torque Considerations - BLK24 series

- Pitch, Yaw, Plunge need larger motor than needed for Roll
  - NEMA 24 is class of servo with 0.57 N-m rated torque (B)
- With Gearing ratio of 20 (for Pitch)
  - NEMA 24 servo can achieve 9.1 N-m effective torque
- Roll with small servo/stepper motor
- Feasible in all degrees of freedom

NEMA 24 Brushless DC Motor

Encoder Possibilities

- Gurley: 7700 (absolute/incremental)
  - Increments: 20000, resolution 0.018, variable shaft width

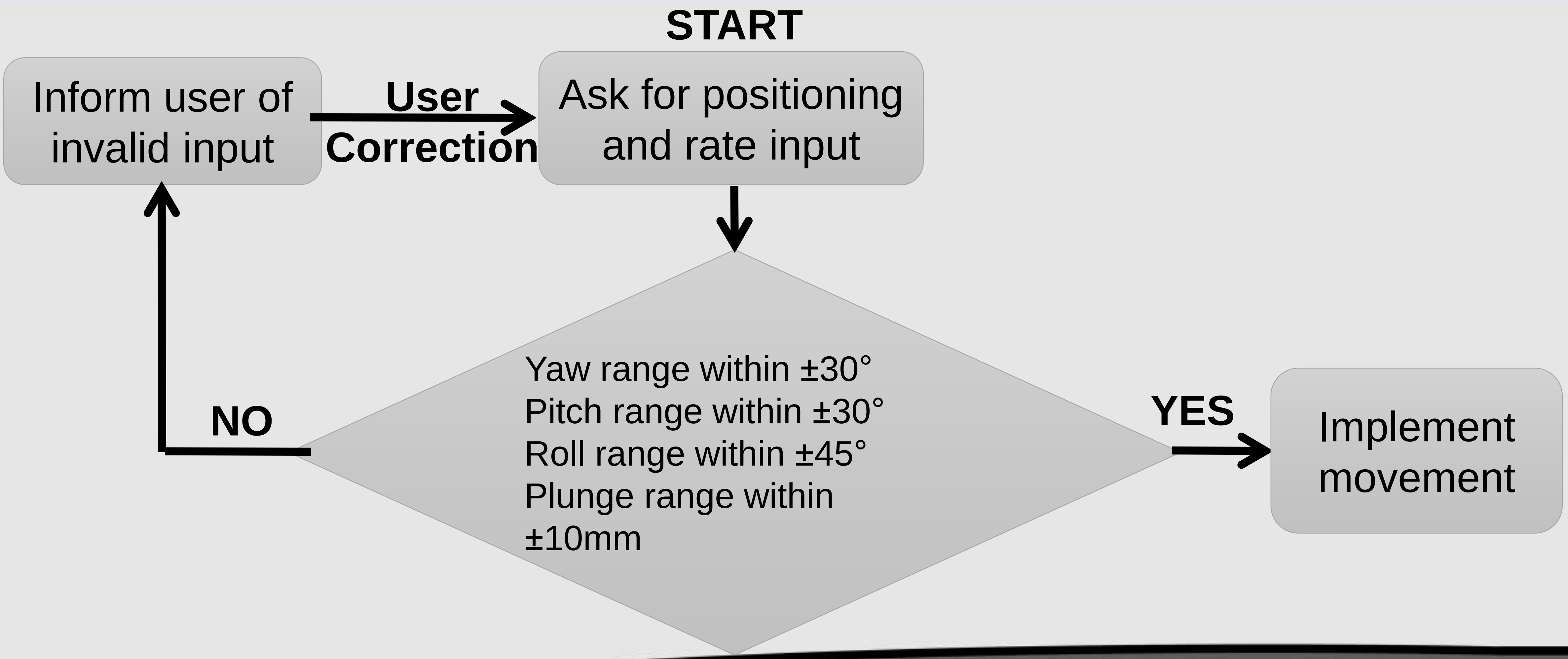
- RLS: RM22 (absolute/incremental)
  - Increments: 8192, resolution 0.0439, variable shaft width

Safety and Failsafes

Safety Concerns and Potential Solutions

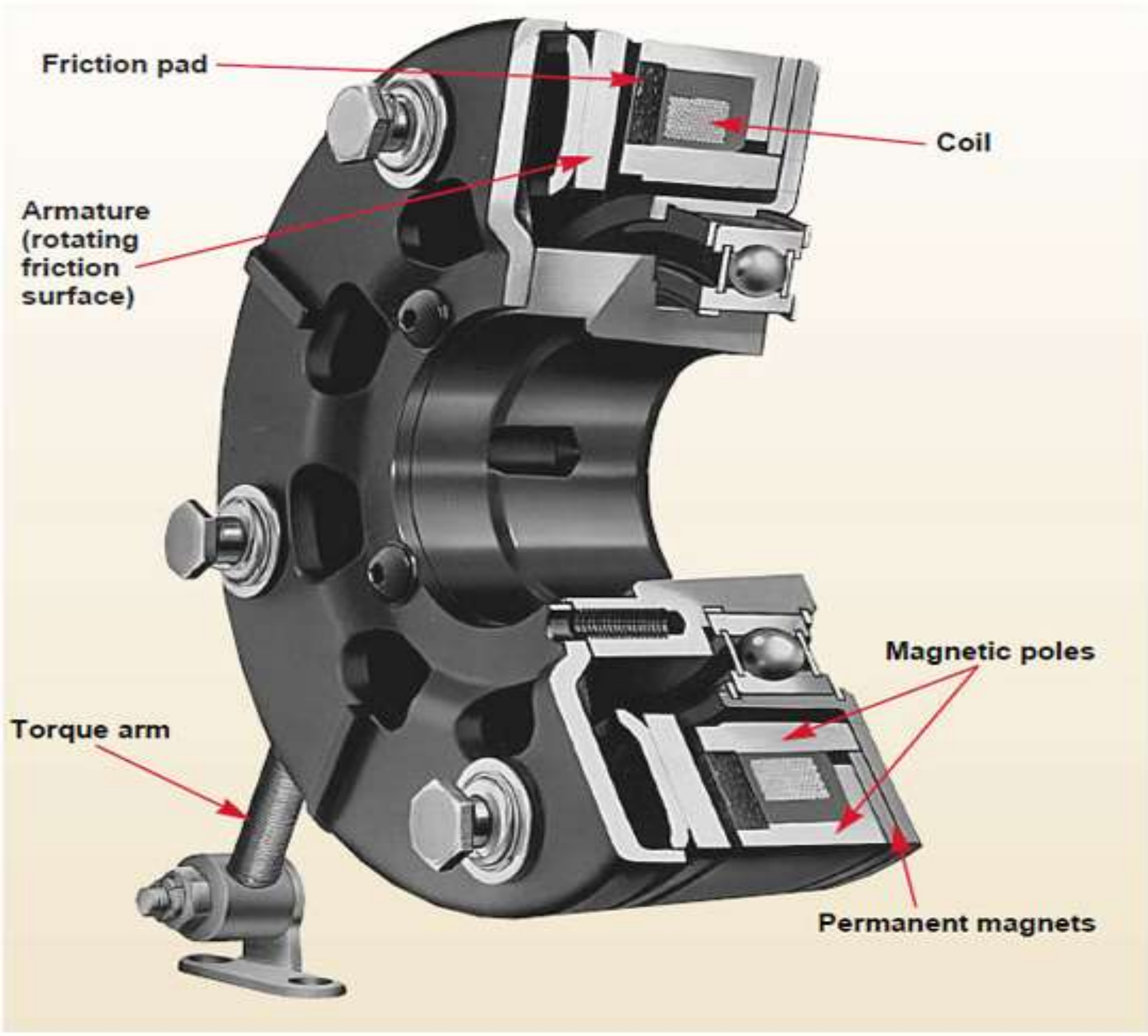
- Software shall check for invalid user input and wiring/feedback failure
  - LabVIEW VI shall check range and rate values
  - Maneuver shall not be performed if out of ranges or beyond maximum rate
  - Program voltage limitations of motor controller in LabVIEW to bound movement rate
- Failsafe hardware installed for software and power failure
  - Passive and active stops installed if software check fails to validate range, or if power is cut to COMPASS
  - COMPASS system will be physically prevented from exceeding range limits
- Failsafe hardware
  - Install 'power off' braking system on motor shafts (active)
  - Fill in gear valley or have non-formation of gear teeth at location of range limit on gears to halt gear motion (passive)

Software Rectification of Human Error



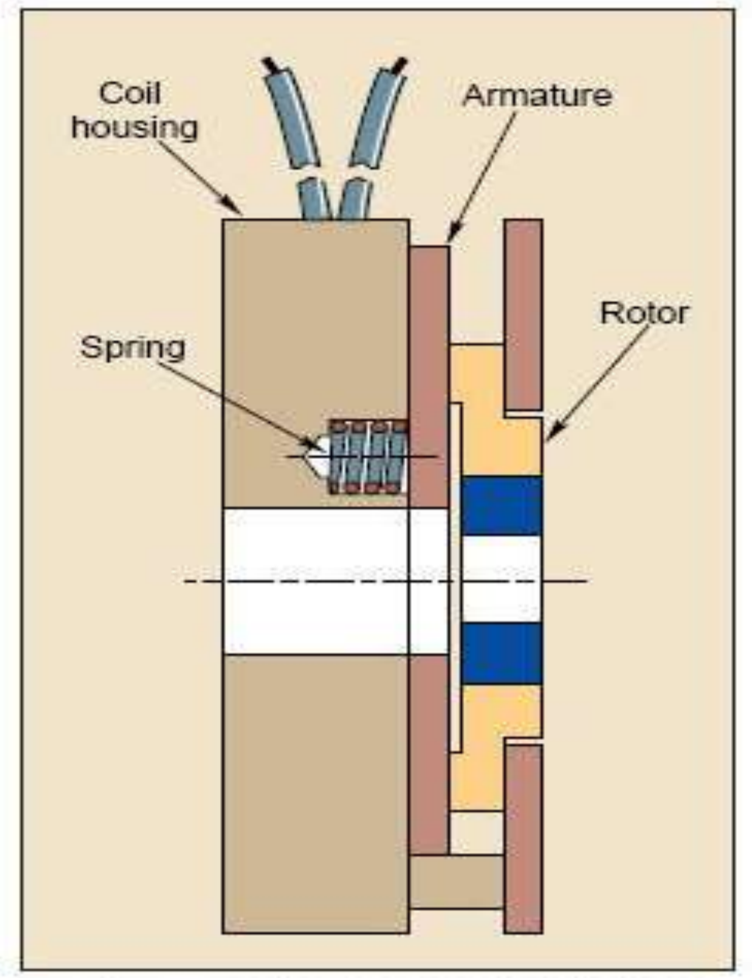
Failsafe Hardware: Power Off Braking

- Permanent magnet brakes
  - Engages to hold a load when power is cut to COMPASS
  - When engaged, a magnetic field attracts an armature to the rotor shaft, holding the torque of the motor
  - When disengaged, an alternate magnetic field pushes against the armature, freeing the rotor shaft
  - More economical in size than spring brake, but require constant current control when disengaged.



Failsafe Hardware: Power Off Braking

- Spring brakes
  - Engages to hold a load when power is cut to COMPASS
  - When disengaged, coil housing generates magnetic field that attracts an armature (pressure plate), leaving gap between plate and friction disk
  - When engaged, magnetic field decays and springs push against armature, engaging rotor shaft
  - Does not require constant current control, but larger in size to deliver similar torque as permanent magnet brakes



Power Off Braking Feasibility

- Holding torque required from power off brakes should be 50% larger than required holding torque
  - Largest required holding torque: Plunge, requiring 8.75 N-m holding torque: 8.75*1.5 = 13.13 N-m
  - Brakes should provide holding torque of at least 14 N-m
- ERS Warner provides spring brake of sufficient static torque rating
  - ERS-49 supplies 20 N-m of holding torque
- KEB provides permanent magnetic brake of sufficient torque rating
  - KEB COMBIPERM Size 06 supplies 18 N-m of static braking torque

Requirements

Functional Requirement 1

COMPASS shall be able to position the model.

DR 1.1: COMPASS shall have defined ranges for 4 degrees of freedom.

DR 1.1.1: The pitch range of the model shall be +- 30 deg min

DR 1.1.2: The yaw range of the model shall be +- 30 deg min

DR 1.1.3: The roll range of the model shall be +- 45 deg min

DR 1.1.4: The plunge range of the model shall be +- 10 cm min

DR 1.2: The position of COMPASS shall be given from sensor data from both static and dynamic cases.

Functional Requirement 2

COMPASS software shall interface with the user and the hardware such that models can be positioned at the required range and rate.

DR 2.1: LabVIEW interface shall facilitate the user's operation of the COMPASS machinery.

DR 2.2: COMPASS shall incorporate position feedback in order to control the system via the control law as well as to display the position to the user and save to a file.

DR 2.3: COMPASS shall incorporate safety within the software to determine if the commanded static or dynamic position is within the capabilities of the COMPASS hardware.

DR 2.4: COMPASS shall couple motion for the different degrees of freedom to result in smooth, realistic motion

Functional Requirement 3

COMPASS shall be integrated with the wind tunnel test section

DR 3.1: COMPASS shall prevent damage to itself and the wind tunnel in the event of a power failure.

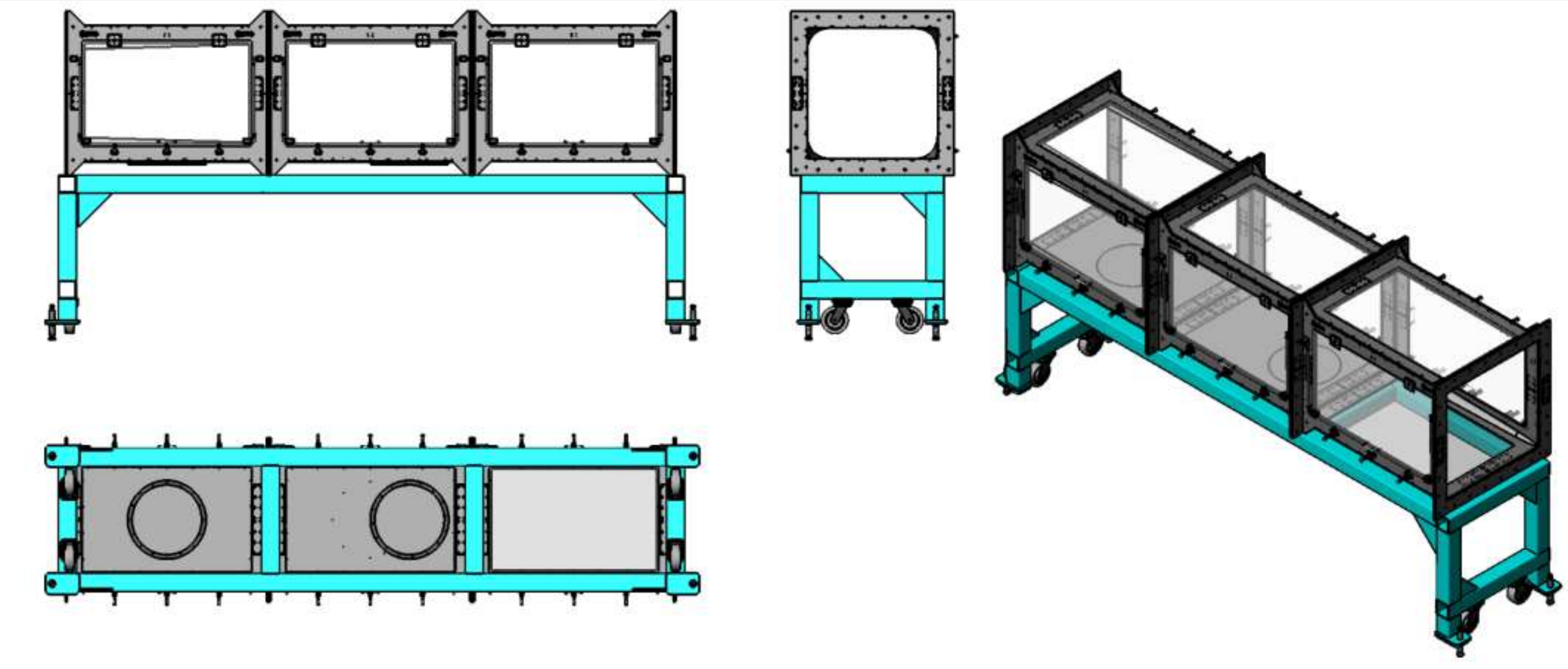
DR 3.2: The installation/assimilation of COMPASS shall not impede the basic functions of the wind tunnel.

Wind Tunnel Specs and Drawings

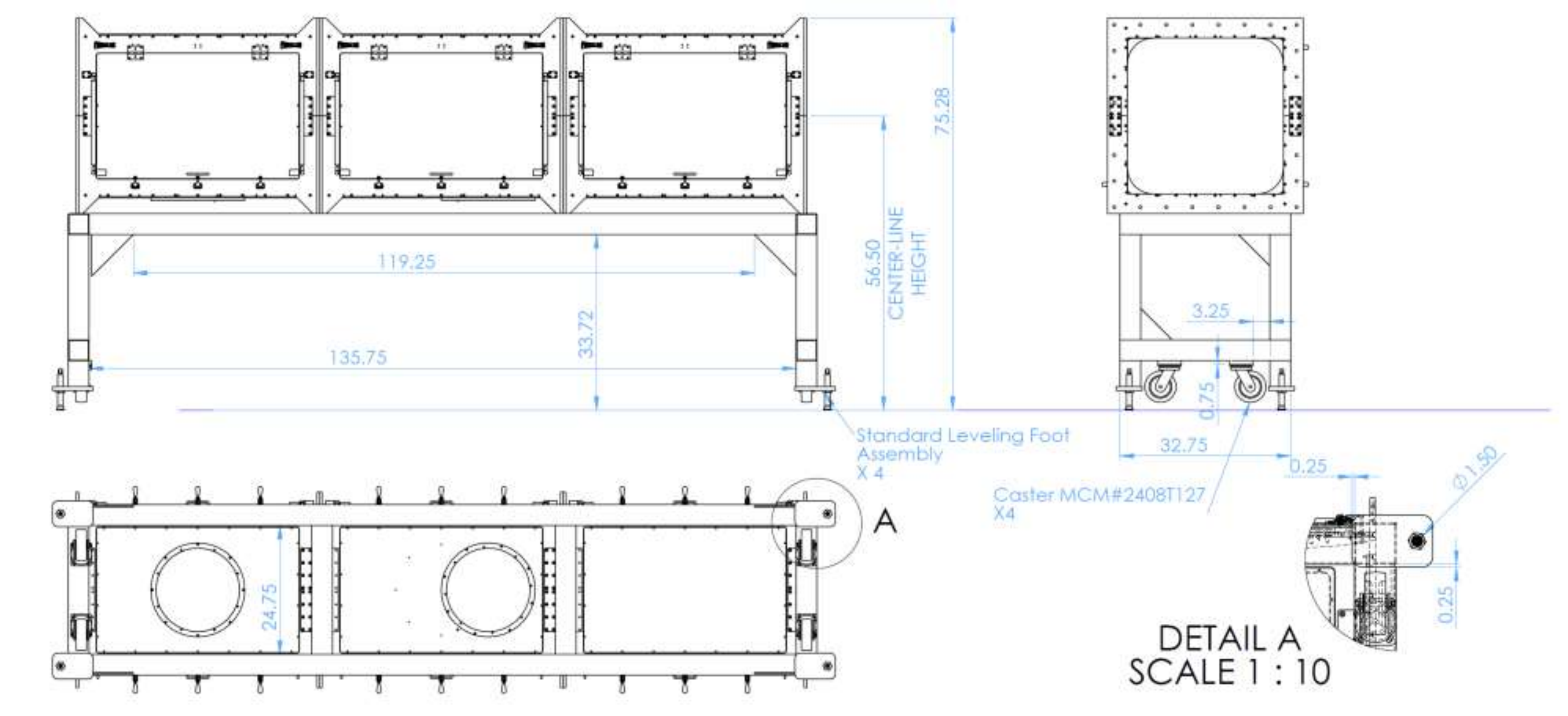
Wind Tunnel Specs

- Max Speed = 65 m/s
- Length of Wind Tunnel = 63.32 ft (19.3 m)
- Length of all 3 Test Sections= 11.69 ft (3.56 m)
  - Single Test Section = 3.90 ft (1.19 m)
- Test Section Width = 2.53 ft (0.76 m)

Test Section Schematics



Test Section Schematics



Estimated Delivery Dates for Products

- Motors 6-16 weeks
- Sensors 3-4 weeks
- DAQs 5-10 days