

# Rowing Oar Strain Measurement System Geometry, Materials, and Specifications

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# 1 Overview

This document provides a complete specification of the geometry, materials, and measurement system for strain measurement on a sculling oar. The primary objective is to evaluate whether the measurement system requires specific considerations for:

- Amplification of strain gauge measurements
- Effects of thermal expansion
- Misalignment of the device on the rowing oar shaft
- Manufacturing imprecision (e.g., beam curvature)

The measurement system consists of a small aluminum beam attached to the carbon fiber oar shaft, instrumented with strain gauges to measure bending during rowing.

## 2 Coordinate System and Sign Conventions

### 2.1 Coordinate System

The oar is represented in a top view and described in a right-handed Cartesian coordinate system  $(x, y, z)$ :

- **Origin** ( $x = 0$ ): Located at the oarlock position
- **$x$ -axis**: Along the oar longitudinal axis
  - Positive direction: toward the handle (inboard)
  - Negative direction: toward the blade (outboard)
- **$y$ -axis**: Perpendicular to the oar, in the plane of bending
  - This is the direction of boat travel
  - During the drive phase, the boat travels in the  $-y$  direction
  - The blade pushes the water in the  $+x$  direction
- **$z$ -axis**: Perpendicular to both  $x$  and  $y$  (vertical direction)
  - Positive direction: away from the water (upward)
  - Negative direction: toward the water (downward)

**Note:** In the top view representation, bending occurs in the  $x$ - $y$  plane, with deflection  $w(x)$  in the  $y$ -direction. During rowing, the force at the handle is typically in the  $-y$  direction (downward), causing the oar to bend with the blade side deflecting upward relative to the handle.

### 2.2 Sign Conventions

- **Forces**: Positive in the positive  $y$  direction (upward, away from water)
- **Moments**: Positive according to right-hand rule about  $z$ -axis (positive moment causes compression on top surface,  $y > 0$ )
- **Deflection**:  $w(x) > 0$  indicates upward deflection (in  $+y$  direction)
- **Rotation about  $z$ -axis**:  $\theta(x) > 0$  indicates rotation about  $z$ -axis according to right-hand rule (bending rotation)
- **Rotation about  $x$ -axis**:  $\phi(x) > 0$  indicates rotation about  $x$ -axis according to right-hand rule (torsional rotation/twist)
- **Strain**:  $\varepsilon > 0$  indicates tension,  $\varepsilon < 0$  indicates compression

**Note:** During rowing, the force at the handle is typically in the  $-y$  direction (downward), causing the oar to bend with the blade side deflecting upward relative to the handle.

### 3 Oar Geometry

#### 3.1 Overall Dimensions

The sculling oar consists of outboard (blade side) and inboard (handle side) sections, separated by the oarlock at  $x = 0$ .

Table 1: Overall oar dimensions

Symbol	Description	Value
$L_{\text{out}}$	Total outboard length (blade tip to oarlock)	2000 mm
$L_{\text{in}}$	Inboard length (oarlock to handle end)	900 mm
$L_{\text{total}}$	Total oar length	2900 mm

#### 3.2 Blade Geometry

Table 2: Blade dimensions

Symbol	Description	Value
$L_{\text{blade}}$	Blade length	430 mm
$w_{\text{blade}}$	Blade maximum width	150 mm
$t_{\text{blade}}$	Blade thickness	5 mm
$h_{\text{bow}}$	Blade bow height (spoon curvature)	40 mm
$x_{\text{blade}}$	Blade tip position	$-L_{\text{out}} = -2000$ mm

#### 3.3 Shaft Geometry

The shaft is a hollow circular tube with constant cross-section.

Table 3: Shaft dimensions

Symbol	Description	Value
$D_{o,s}$	Shaft outer diameter	38 mm
$D_{i,s}$	Shaft inner diameter	32 mm
$t_s$	Shaft wall thickness	3 mm
$L_{\text{shaft,out}}$	Outboard shaft length	1570 mm
$L_{\text{shaft,in}}$	Inboard shaft length	700 mm

#### 3.4 Sleeve Geometry

The sleeve is a cylindrical component made of ABS plastic that provides reinforcement around the oarlock region.

#### 3.5 Collar Geometry

The collar prevents the oar from sliding through the oarlock.

#### 3.6 Oarlock Geometry

#### 3.7 Handle Geometry

The handle consists of a taper section transitioning from shaft diameter to grip diameter, followed by the grip section.

### 4 Measurement Beam Geometry

The measurement beam is a rectangular beam attached to the shaft by means of two rigid clamps at specified positions.

Table 4: Sleeve dimensions

Symbol	Description	Value
$L_{\text{sleeve}}$	Total sleeve length	300 mm
$L_{\text{sleeve},-x}$	Sleeve extension in $-x$ direction from oarlock	200 mm
$L_{\text{sleeve},+x}$	Sleeve extension in $+x$ direction from oarlock	100 mm
$D_{\text{sleeve}}$	Sleeve outer diameter	60 mm
$x_{\text{sleeve,start}}$	Sleeve start position	$-200$ mm
$x_{\text{sleeve,end}}$	Sleeve end position	100 mm

Table 5: Collar dimensions

Symbol	Description	Value
$D_{\text{collar}}$	Collar diameter	120 mm
$t_{\text{collar}}$	Collar thickness (axial)	20 mm
$x_{\text{collar}}$	Collar position	20 mm

## 4.1 Beam Dimensions and Position

**Beam neutral axis position:** The beam neutral axis is located at  $y_b$  measured from the global coordinate origin (shaft centerline at  $y = 0$ ). This position is calculated as:

$$y_b = \frac{D_{o,s}}{2} + e_b \quad (1)$$

where  $D_{o,s}/2$  is the shaft outer radius and  $e_b$  is the distance from the shaft outer surface to the beam neutral axis. The beam extends from the shaft surface upward (in the  $+y$  direction) to provide clearance for mounting strain gauges on both top and bottom surfaces.

## 4.2 Beam Clamp Positions

The beam is attached to the shaft at two locations:

$$x_{b,1} = x_b = 200 \text{ mm} \quad (\text{root clamp}) \quad (2)$$

$$x_{b,2} = x_b + L_b = 300 \text{ mm} \quad (\text{tip clamp}) \quad (3)$$

## 4.3 Clamp Rigidity and Assembly Tolerances

The clamps are assumed to be **rigid connections** that prevent relative motion between the beam and shaft at the attachment points. However, two manufacturing and assembly phenomena must be considered:

### 4.3.1 System Misalignment

The entire beam assembly may be rotated around the shaft ( $x$ -axis) during installation. This misalignment angle, denoted  $\phi_{\text{mis}}$ , is expected to be within  $\pm 1^\circ$ . This creates a **static angular offset** of the entire measurement system, which may result in:

- Apparent strain offset in gauge readings
- Coupling between bending and the misaligned measurement axes

[TBD: Quantification of offset effect on strain measurements]

### 4.3.2 Initial Beam Twist

The two clamps may not be perfectly aligned with each other during assembly, creating an initial twist in the beam. This twist angle, denoted  $\phi_0$ , represents the relative rotation between the root clamp (at  $x = x_b$ ) and the tip clamp (at  $x = x_b + L_b$ ) around the  $x$ -axis.

**Thermal amplification:** The differential thermal expansion between the aluminum beam and carbon shaft can amplify this initial twist. Over the operating temperature range ( $\Delta T = 85$  K), the mismatch in thermal expansion creates additional torsional strain that compounds with  $\phi_0$ .

**Simplification:** The analysis assumes that twist-induced strains in the strain gauges are **negligible** compared to bending strains. This assumption should be verified for the expected values of  $\phi_{\text{mis}}$  and  $\phi_0$ .

Table 6: Oarlock dimensions

Symbol	Description	Value
$t_{\text{oarlock}}$	Oarlock thickness (axial)	20 mm
$h_{\text{oarlock}}$	Oarlock height (radial extent)	160 mm

Table 7: Handle dimensions

Symbol	Description	Value
$L_{\text{handle}}$	Total handle length	200 mm
$L_{\text{taper}}$	Taper length	50 mm
$D_{\text{taper,start}}$	Taper start diameter	38 mm
$D_{\text{taper,end}}$	Taper end diameter	30 mm
$L_{\text{grip}}$	Grip length	150 mm
$D_{\text{grip}}$	Grip diameter	35 mm
$x_F$	Handle end position (force application point)	$L_{\text{in}} = 900$ mm

## 5 Material Properties

### 5.1 Shaft Material: Carbon Fiber Composite

The shaft is constructed from carbon fiber composite with fibers aligned predominantly in the  $x$  (longitudinal) direction. The properties below are approximations for unidirectional carbon fiber/epoxy composite.

**Note:** Carbon fiber composites exhibit highly anisotropic behavior. The longitudinal modulus (fiber direction) is much higher than the transverse modulus. The negative coefficient of thermal expansion in the fiber direction is characteristic of carbon fibers. These values are approximations and can vary significantly depending on fiber type, volume fraction, and layup.

### 5.2 Beam Material: Aluminum 1050

The measurement beam is constructed from Aluminum 1050, a commercially pure aluminum alloy.

## 6 Strain Gauge Specifications

### 6.1 Gauge Type and Configuration

The measurement system uses four linear strain gauges arranged in a full Wheatstone bridge configuration.

All four strain gauges are located at the beam midpoint along the  $x$ -axis:

$$x_{\text{gauge}} = x_b + \frac{L_b}{2} \quad (4)$$

### 6.2 Bridge Configuration

The four strain gauges form a full Wheatstone bridge with two active half-bridges in opposition:

- $R_1$  &  $R_2$  (**Top surface**): Located at  $y = y_{\text{top}}$ ,  $x = x_{\text{gauge}}$ 
  - Experience **positive strain** (tension) when beam bends due to downward force at handle
- $R_3$  &  $R_4$  (**Bottom surface**): Located at  $y = y_{\text{bottom}}$ ,  $x = x_{\text{gauge}}$ 
  - Experience **negative strain** (compression) under same bending condition

The bridge is wired as follows:

- $R_1$  and  $R_2$  in one voltage divider (top surface gauges)
- $R_3$  and  $R_4$  in the other voltage divider (bottom surface gauges)

Table 8: Beam dimensions

Symbol	Description	Value
$L_b$	Beam length (between clamp centers)	100 mm
$h_b$	Beam height (bending direction, $y$ -direction)	2 mm
$b$	Beam width ( $z$ -direction)	10 — 15 mm
$x_b$	Beam root position (first clamp)	200 mm
$e_b$	Beam neutral axis eccentricity from shaft centerline	20 mm
$y_b$	Beam neutral axis $y$ -coordinate	$D_{o,s}/2 + e_b$

Table 9: Carbon fiber composite properties (longitudinal direction)

Property	Symbol	Value
Young’s modulus (longitudinal)	$E_s$	140 GPa (approximate)
Poisson’s ratio	$\nu_s$	0.30 (approximate)
Coefficient of thermal expansion (longitudinal)	$\alpha_s$	$-0.5 \times 10^{-6} \text{ K}^{-1}$ (approximate)
Density	$\rho_s$	1600 kg/m <sup>3</sup> (approximate)

The bridge output voltage is:

$$V_{\text{out}} = V_{\text{ex}} \left( \frac{R_2}{R_1 + R_2} - \frac{R_4}{R_3 + R_4} \right) \quad (5)$$

Substituting the strain-dependent resistances:

$$R_1 = R_g(1 + GF \cdot \varepsilon_{\text{top}}) \quad (6)$$

$$R_2 = R_g(1 + GF \cdot \varepsilon_{\text{top}}) \quad (7)$$

$$R_3 = R_g(1 + GF \cdot \varepsilon_{\text{bottom}}) \quad (8)$$

$$R_4 = R_g(1 + GF \cdot \varepsilon_{\text{bottom}}) \quad (9)$$

For small strains ( $GF \cdot \varepsilon \ll 1$ ), this simplifies to:

$$\frac{V_{\text{out}}}{V_{\text{ex}}} = \frac{GF}{2} (\varepsilon_{\text{top}} - \varepsilon_{\text{bottom}}) \quad (10)$$

where:

- $V_{\text{out}}$  = bridge output voltage
- $V_{\text{ex}}$  = bridge excitation voltage = 3.3 V
- $\varepsilon_{\text{top}}$  = average strain on top surface (gauges  $R_1$  &  $R_2$ )
- $\varepsilon_{\text{bottom}}$  = average strain on bottom surface (gauges  $R_3$  &  $R_4$ )

Unless otherwise stated,  $\varepsilon_{\text{top}}$  and  $\varepsilon_{\text{bottom}}$  are understood to be the *total* axial strains at the corresponding gauge locations. In terms of the decomposition introduced in Eq. (11), they correspond to  $\varepsilon_{\text{total}}(T, F)$  evaluated at  $y = y_{\text{top}}$  and  $y = y_{\text{bottom}}$ , respectively.

**Strain signs during bending:** When the handle force produces downward bending:

- $\varepsilon_{\text{top}} > 0$  (tension)
- $\varepsilon_{\text{bottom}} < 0$  (compression)

### 6.3 Expected Strain Range

Based on preliminary beam theory estimates (see Section 8.2), the expected mechanical strain range at the gauge locations is on the order of:

- $\varepsilon_{\text{mech}} \in [-500 \mu\varepsilon, 500 \mu\varepsilon]$  for typical rowing forces.

Thermal and offset contributions  $\varepsilon_{\text{th}}(T)$  and  $\varepsilon_{\text{off}}(T)$  may add bias and apparent gain variations, as discussed in Section 8.3.3.

Table 10: Aluminum 1050 properties

Property	Symbol	Value
Young's modulus	$E_b$	69 GPa
Poisson's ratio	$\nu_b$	0.33
Coefficient of thermal expansion	$\alpha_b$	$23.6 \times 10^{-6} \text{ K}^{-1}$
Density	$\rho_b$	2710 kg/m <sup>3</sup>
Yield strength	$\sigma_{y,b}$	34 MPa (annealed)

Table 11: Strain gauge specifications

Parameter	Symbol	Value
Nominal resistance	$R_g$	$1000 \Omega \pm 3 \Omega$
Gauge factor	$GF$	2.15
Maximum strain	–	$2000 \mu\epsilon$
Temperature coefficient of resistance	–	20 ppm/K
Backing material	–	Polyimide

## 7 Operating Conditions

### 7.1 Temperature Range

The measurement system is expected to operate over the following temperature range:

Table 12: Operating temperature range

Condition	Temperature
Minimum (cold water, early morning)	$-5^\circ \text{C}$
Maximum (solar heating of carbon shaft)	$80^\circ \text{C}$
Temperature excursion	$\Delta T = 85 \text{ K}$

**Note:** The maximum temperature assumption of  $80^\circ \text{C}$  is based on solar radiation heating the black carbon fiber shaft. This value should be verified through:

- Thermal modeling of solar heating on carbon shaft
- Experimental measurements under various environmental conditions
- Measurement or estimation of solar absorption coefficient of carbon shaft surface [TBD]

### 7.2 Mechanical Loading

During rowing, the handle experiences a vertical force (downward, in the  $-y$  direction). The expected force range is:

**Note:** The force is assumed to act vertically downward at the handle end position  $x = x_F = 900 \text{ mm}$ .

## 8 Theoretical Framework

This section presents the theoretical foundations for analyzing the measurement system, including mechanical bending, thermal effects, geometric imperfections, and stability considerations.

### 8.1 Strain Decomposition and Measurement Model

At each strain gauge location, the total axial strain is decomposed into three contributions:

$$\varepsilon_{\text{total}}(T, F) = \varepsilon_{\text{mech}}(T, F) + \varepsilon_{\text{th}}(T) + \varepsilon_{\text{off}}(T), \quad (11)$$

where:

- $\varepsilon_{\text{mech}}(T, F)$  is the *mechanical* bending strain due to the applied load  $F$ ; in the absence of bending ( $F = 0$ ) this term is taken to be zero,



Table 13: Expected force range at handle

Condition	Force
Minimum	0 N
Maximum (peak during power stroke)	200 kg = 1962 N

- $\varepsilon_{\text{th}}(T)$  is the *thermal* strain caused by constrained differential expansion between the aluminum beam and the carbon shaft,
- $\varepsilon_{\text{off}}(T)$  is an *offset* strain term that groups additional effects such as assembly pre-stress, imperfect bonding, twist misalignment, and electronics zero-shift.

The strain inferred from the bridge output after calibration is denoted by  $\varepsilon_{\text{meas}}(T, F)$ . In general it does not coincide exactly with the purely mechanical component, because temperature influences both the mechanical response and the measurement system. Sections 8.2–8.3 derive  $\varepsilon_{\text{mech}}(T, F)$  and  $\varepsilon_{\text{th}}(T)$ , while Section 8.3.3 introduces a parametric model for  $\varepsilon_{\text{meas}}(T, F)$  and  $\varepsilon_{\text{off}}(T)$ .

## 8.2 Beam Bending Theory

The measurement system relies on Euler-Bernoulli beam theory to relate applied forces to measurable strains.

### 8.2.1 Governing Equation

For a beam subject to transverse loading:

$$EI \frac{d^4 w}{dx^4} = q(x) \quad (12)$$

where:

- $E$  is the Young’s modulus,
- $I$  is the second moment of area,
- $w(x)$  is the transverse deflection,
- $q(x)$  is the distributed transverse load.

Under typical rowing operation, the dominant loading is due to the handle force applied at the inboard end of the oar, resulting in bending of both the carbon shaft and the aluminum measurement beam.

### 8.2.2 Moment-Curvature Relation

For small deflections and linear elastic behavior, the bending moment  $M(x)$  is related to curvature:

$$M(x) = -EI \frac{d^2 w}{dx^2} \quad (13)$$

### 8.2.3 Strain-Displacement Relation

For a beam in pure bending, the longitudinal *mechanical* strain at distance  $y$  from the neutral axis is:

$$\varepsilon_{\text{mech}}(x, y) = -y \frac{d^2 w}{dx^2} = \frac{M(x) \cdot y}{EI} \quad (14)$$

Note the sign convention: positive  $M$  causes compression on the top surface ( $y > 0$ ).

### 8.2.4 Second Moment of Area

For a rectangular cross-section (beam):

$$I_b = \frac{bh_b^3}{12} \quad (15)$$

where  $b$  is the beam width and  $h_b$  is the beam height.

For a hollow circular cross-section (shaft):

$$I_s = \frac{\pi}{64} (D_{o,s}^4 - D_{i,s}^4) \quad (16)$$

### 8.2.5 Boundary Conditions

The exact form of  $w(x)$  depends on the boundary conditions at the oarlock and at the handle. In this specification, two theoretical boundary conditions are considered:

- **Theory 1:** Clamped boundary condition at the oarlock, with the handle modeled as a point load.
- **Theory 2:** Pin support boundary conditions at the oarlock and handle, approximating more flexible support conditions.

The detailed solutions for  $w(x)$  for each theory are derived in Section 9.

## 8.3 Thermal Effects

### 8.3.1 Differential Thermal Expansion

The aluminum beam and carbon shaft have significantly different thermal expansion coefficients:

$$\alpha_b = 23.6 \times 10^{-6} \text{ K}^{-1} \quad (\text{Aluminum 1050}) \quad (17)$$

$$\alpha_s = -0.5 \times 10^{-6} \text{ K}^{-1} \quad (\text{Carbon fiber, approximate}) \quad (18)$$

When constrained together by the clamping system, differential thermal expansion generates internal forces and moments. The free thermal elongation of beam and shaft over length  $L_b$  would be:

$$\Delta L_b = \alpha_b L_b \Delta T \quad (19)$$

$$\Delta L_s = \alpha_s L_b \Delta T \quad (20)$$

The differential free expansion is:

$$\Delta L_{\text{thermal}} = (\alpha_b - \alpha_s) L_b \Delta T = \Delta \alpha L_b \Delta T \quad (21)$$

with:

$$\Delta \alpha = \alpha_b - \alpha_s \quad (22)$$

If the beam and shaft are perfectly constrained, the differential expansion is converted into internal axial force  $N_{\text{th}}$  and bending moments, resulting in a thermal strain component  $\varepsilon_{\text{th}}(T)$  at the gauge location. A simplified axial thermal strain (if fully constrained) would be:

$$\varepsilon_{\text{thermal}} = \Delta \alpha \Delta T \quad (23)$$

which, in the context of Eq. (11), represents the thermal component  $\varepsilon_{\text{th}}(T)$ .

### 8.3.2 Clamp-Induced Assembly Strain

Clamp tightening can introduce additional pre-stress and twist. These effects are lumped into the offset strain term  $\varepsilon_{\text{off}}(T)$  in Eq. (11). While some of these effects may be approximately temperature-independent, differential thermal expansion can amplify twist and pre-stress, causing  $\varepsilon_{\text{off}}(T)$  to vary with  $T$ .

### 8.3.3 Thermal Amplification

Temperature variations do not only induce direct thermal expansion; they also modify material properties, clamp stiffness, and the effective geometry of the assembly. These effects can be captured phenomenologically as an effective change in both the measurement gain and offset.

Building on the decomposition in Eq. (11), the measured strain at temperature  $T$  and load  $F$  is expressed as:

$$\varepsilon_{\text{meas}}(T, F) = A_T(T) \varepsilon_{\text{mech}}(T, F) + \varepsilon_{\text{th}}(T) + \varepsilon_{\text{off}}(T) \quad (24)$$

where:

- $\varepsilon_{\text{mech}}(T, F)$  is the mechanical bending strain component defined in Eq. (11),
- $A_T(T)$  is a dimensionless thermal amplification factor,
- $\varepsilon_{\text{th}}(T)$  is the thermal strain component due to differential expansion,
- $\varepsilon_{\text{off}}(T)$  is a temperature-dependent offset caused by assembly pre-stress, twist amplification, and residual electronics effects.

For small excursions around the reference temperature  $T_0$ , the amplification factor may be linearized as:

$$A_T(T) \approx 1 + k_T (T - T_0) \quad (25)$$

where  $k_T$  [ $\text{K}^{-1}$ ] is an effective thermal amplification coefficient to be obtained from calibration.

The linear approximation in Eq. (25) assumes that higher-order terms in  $(T - T_0)$  are negligible, which is appropriate for modest temperature excursions where:

$$|T - T_0| \ll \frac{1}{|k_T|} \quad (26)$$

In the context of this system, the validity of the linear model must be verified experimentally during thermal calibration. If necessary, additional higher-order terms in temperature may be introduced.

Substituting Eq. (25) into Eq. (24) gives:

$$\varepsilon_{\text{meas}}(T, F) \approx [1 + k_T (T - T_0)] \varepsilon_{\text{mech}}(T, F) + \varepsilon_{\text{th}}(T) + \varepsilon_{\text{off}}(T) \quad (27)$$

Equation (27) shows that thermal effects contribute an effective gain variation on the mechanical strain measurement in addition to an additive temperature-dependent bias. Both  $k_T$  and the behavior of  $\varepsilon_{\text{off}}(T)$  must be determined experimentally during calibration.

## 8.4 Stability and Buckling Considerations

Under compression, the aluminum beam may approach Euler buckling limits. The classical critical load for a prismatic column is:

$$N_{\text{cr}} = \frac{\pi^2 E_b I_b}{(KL_b)^2} \quad (28)$$

where:

- $N_{\text{cr}}$  is the critical buckling load,
- $E_b$  is the Young's modulus of the beam,
- $I_b$  is the second moment of area of the beam,
- $L_b$  is the effective length of the beam,
- $K$  is the effective length factor (depends on end conditions).

In this system, the beam is clamped at both ends via the clamping system. A conservative estimate is to take  $K \approx 0.7$ – $1.0$  depending on the rotational stiffness of the clamps.

The combined thermal and assembly loads must remain well below  $N_{\text{cr}}$  across the full operating temperature range:

$$N = N_{\text{th}} + N_{\text{assy}} < \eta N_{\text{cr}} \quad (29)$$

with a safety factor  $\eta$  (e.g.,  $\eta = 0.5$ ).

**[TBD: Verification that  $N = N_{\text{th}} + N_{\text{assy}} < 0.5 N_{\text{cr}}$  for operating temperature range]**

**[TBD: Verification of stability criteria for operating temperature range]**

## 9 Calculation of Mechanical Strains

This section presents detailed calculations of the mechanical bending strains  $\varepsilon_{\text{mech}}(x, y)$  for the measurement beam under two different boundary condition assumptions. These calculations provide the theoretical basis for interpreting strain gauge measurements and understanding the relationship between applied forces and measured strains.

The mechanical strain component  $\varepsilon_{\text{mech}}(T, F)$  defined in Eq. (11) is derived from classical beam theory applied to the specific geometry and loading conditions of the rowing oar system. Two theoretical models are considered to bracket the range of expected behavior:

- **Theory 1 (Section 9.1):** Assumes a clamped boundary condition at the oarlock, representing a rigid constraint.
- **Theory 2 (Section 9.2):** Assumes pin support boundary conditions at both the oarlock and at the end of the outboard shaft, representing a more flexible support system.

Both theories use the same fundamental beam equations (Section 8.2) but differ in their boundary conditions, leading to different deflection profiles  $w(x)$  and consequently different strain distributions along the beam.

## 9.1 Theory 1: Clamped Boundary Condition at Oarlock

### 9.1.1 Boundary Conditions

Theory 1 assumes the following boundary conditions:

- **At the oarlock** ( $x = 0$ ): The oar shaft is clamped, preventing both displacement and rotation:

$$w_s(0) = 0 \quad (\text{no vertical displacement}) \quad (30)$$

$$\theta_s(0) = 0 \quad (\text{no rotation, clamped}) \quad (31)$$

- **At the handle** ( $x = x_F$ ): A vertical force  $F$  is applied, with no moment:

$$V_s(x_F) = -F \quad (\text{applied shear force}) \quad (32)$$

$$M_s(x_F) = 0 \quad (\text{free end, no moment}) \quad (33)$$

See Figure ?? for a visual representation of the boundary conditions.

### 9.1.2 Deflection Solution

[TBD: Derivation of deflection  $w_s(x)$  for clamped-free beam with point load]

### 9.1.3 Moment Distribution

[TBD: Calculation of bending moment  $M_s(x)$  along the shaft]

### 9.1.4 Strain at Beam Location

[TBD: Calculation of mechanical strain  $\varepsilon_{\text{mech}}$  at the beam gauge locations]

### 9.1.5 Numerical Example

[TBD: Numerical example with typical force values]

## 9.2 Theory 2: Pin Support Boundary Conditions

### 9.2.1 Boundary Conditions

Theory 2 assumes the following boundary conditions with pin supports at two locations:

- **At the oarlock** ( $x = 0$ ): Pin support prevents displacement but allows rotation:

$$w_s(0) = 0 \quad (\text{no vertical displacement}) \quad (34)$$

$$M_s(0) = 0 \quad (\text{no moment, pin allows rotation}) \quad (35)$$

- **At the end of outboard shaft** ( $x = -(L_{\text{out}} - L_{\text{blade}}) = -1570 \text{ mm}$ ): Pin support prevents displacement but allows rotation:

$$w_s(-(L_{\text{out}} - L_{\text{blade}})) = 0 \quad (\text{no vertical displacement}) \quad (36)$$

$$M_s(-(L_{\text{out}} - L_{\text{blade}})) = 0 \quad (\text{no moment, pin allows rotation}) \quad (37)$$

- **At the handle** ( $x = x_F$ ): A vertical force  $F$  is applied, with no moment:

$$V_s(x_F) = -F \quad (\text{applied shear force}) \quad (38)$$

$$M_s(x_F) = 0 \quad (\text{free end, no moment}) \quad (39)$$

See Figure ?? for a visual representation of the boundary conditions.

### 9.2.2 Deflection Solution

[TBD: Derivation of deflection  $w_s(x)$  for beam with two pin supports and point load]

### 9.2.3 Moment Distribution

[TBD: Calculation of bending moment  $M_s(x)$  along the shaft]

#### 9.2.4 Reaction Forces

[TBD: Calculation of reaction forces at the two pin support locations]

#### 9.2.5 Strain at Beam Location

[TBD: Calculation of mechanical strain  $\varepsilon_{\text{mech}}$  at the beam gauge locations]

#### 9.2.6 Numerical Example

[TBD: Numerical example with typical force values]

## 10 Analysis and Calibration Results

This section presents the calibration results obtained from mechanical loading and thermal characterization experiments. The goal is to determine the coefficients that relate the measured strain  $\varepsilon_{\text{meas}}(T, F)$  to the mechanical strain  $\varepsilon_{\text{mech}}(T, F)$  while accounting for thermal and offset contributions as defined in Eq. (11).

### 10.1 Mechanical Calibration

During mechanical calibration, the oar was held at approximately constant temperature  $T \approx T_0$ , such that:

$$\varepsilon_{\text{th}}(T_0) + \varepsilon_{\text{off}}(T_0) \approx 0 \quad (40)$$

Thus, Eq. (27) simplifies to:

$$\varepsilon_{\text{meas}}(T_0, F) \approx \varepsilon_{\text{mech}}(T_0, F) \quad (41)$$

The calibration curve  $\varepsilon_{\text{meas}}$  vs. applied force  $F$  exhibits a strong linear correlation. A first-order model was fitted:

$$\varepsilon_{\text{meas}}(T_0, F) = C_F \cdot F \quad (42)$$

with calibration factor  $C_F$  [ $\mu\text{e N}^{-1}$ ] representing the mechanical sensitivity at  $T_0$ .

This establishes the baseline mechanical response contribution to Eq. (27).

### 10.2 Thermal Characterization

With the oar unloaded ( $F = 0$ ), Eq. (27) becomes:

$$\varepsilon_{\text{meas}}(T, 0) = \varepsilon_{\text{th}}(T) + \varepsilon_{\text{off}}(T) \quad (43)$$

The measured output over temperature therefore reveals a combination of thermal expansion effects (predictable) and assembly/electronics offsets (less predictable). A linear model was fitted to  $\varepsilon_{\text{meas}}(T, 0)$  to estimate the dominant temperature-dependent trend:

$$\varepsilon_{\text{meas}}(T, 0) \approx k_0 + k_{\text{off}}(T - T_0) \quad (44)$$

where:

- $k_0$  is the residual strain at the reference temperature  $T_0$ ,
- $k_{\text{off}}$  captures the dominant linear temperature dependence of the offset.

These coefficients contribute to modeling  $\varepsilon_{\text{off}}(T)$  in Eq. (27).

### 10.3 Joint Temperature and Load Influence

When both temperature changes and mechanical loads occur simultaneously, the measured strain follows:

$$\varepsilon_{\text{meas}}(T, F) \approx [1 + k_T (T - T_0)] C_F F + \varepsilon_{\text{th}}(T) + \varepsilon_{\text{off}}(T) \quad (45)$$

The parameter  $k_T$  was extracted from controlled tests where load was applied at different temperatures. A linear regression on the gain variation yielded:

$$A_T(T) \approx 1 + k_T(T - T_0)$$

allowing the separation of:

- mechanical gain variation with temperature (via  $k_T$ )
- additive thermal and offset effects (via  $\varepsilon_{\text{th}}$  and  $\varepsilon_{\text{off}}$ )

## 10.4 Discussion

The results demonstrate that:

- The **mechanical calibration** is linear and stable near  $T_0$ .
- The **thermal characterization** reveals a temperature-dependent offset that must be compensated.
- A **joint thermo-mechanical model** is required to ensure accuracy across realistic operating temperatures.

To enable reliable performance during on-water operation, the coefficients  $C_F$ ,  $k_T$  and the characterization of  $\varepsilon_{\text{off}}(T)$  must be incorporated into a real-time compensation algorithm.

## 11 Open Questions and Future Work

The following items require additional information or analysis:

1. Validation of maximum temperature (80 °C) through thermal modeling or measurement
2. Solar absorption coefficient of carbon shaft surface
3. Characterization of system misalignment tolerance ( $\phi_{\text{mis}}$ )
4. Measurement of initial beam twist ( $\phi_0$ ) after assembly
5. Analysis of thermal amplification of initial twist
6. Verification of assumption that twist-induced strains are negligible
7. Detailed boundary condition derivations for Theory 1 (clamped) and Theory 2 (pin supports)
8. Analytical solutions for deflection  $w(x)$  and strain  $\varepsilon(x)$  distributions
9. Thermal stress analysis from constrained differential expansion
10. Estimation of assembly pre-stress ( $N_{\text{assy}}$ ) from clamp tightening
11. Verification that  $N = N_{th} + N_{\text{assy}} < 0.5N_{cr}$  over operating temperature range
12. Sensitivity analysis: manufacturing tolerances, gauge positioning errors
13. Signal amplification requirements based on calculated strain levels
14. Experimental validation plan

## 12 Nomenclature and Subscript Conventions

### 12.1 Subscript Conventions

To maintain clarity and avoid ambiguity, the following subscript conventions are used throughout this document:

Table 14: Subscript conventions

Subscript	Meaning
$s$	Shaft (carbon fiber oar shaft)
$b$	Beam (aluminum measurement beam)
top	Top surface of beam (tension side under downward handle load)
bottom	Bottom surface of beam (compression side under downward handle load)
th	Thermal component (differential expansion)
mech	Mechanical component (bending)
off	Offset component (assembly and electronics)
meas	Measured quantity inferred from bridge/ADC
$x, y, z$	Cartesian coordinates / components

Table 15: Nomenclature - Geometric parameters

Symbol	Description	Units
$x, y, z$	Cartesian coordinates	mm
$L_{\text{out}}$	Total outboard length	mm
$L_{\text{in}}$	Total inboard length	mm
$L_b$	Beam length	mm
$h_b$	Beam height	mm
$b$	Beam width	mm
$D_{o,s}$	Shaft outer diameter	mm
$D_{i,s}$	Shaft inner diameter	mm
$t_s$	Shaft wall thickness	mm
$e_b$	Beam eccentricity from shaft centerline	mm
$y_b$	Beam neutral axis position	mm
$x_b$	Beam root position	mm
$x_F$	Handle end position (force application)	mm
$\phi_{\text{mis}}$	System misalignment angle around shaft	deg or rad
$\phi_0$	Initial beam twist (relative clamp rotation)	deg or rad

Table 16: Nomenclature - Material properties

Symbol	Description	Units
$E_s$	Young's modulus of shaft (carbon)	GPa
$E_b$	Young's modulus of beam (aluminum)	GPa
$\nu_s$	Poisson's ratio of shaft	–
$\nu_b$	Poisson's ratio of beam	–
$\alpha_s$	Thermal expansion coefficient of shaft	$\text{K}^{-1}$
$\alpha_b$	Thermal expansion coefficient of beam	$\text{K}^{-1}$
$I_s$	Second moment of area of shaft	$\text{mm}^4$
$I_b$	Second moment of area of beam	$\text{mm}^4$
$\varepsilon_{\text{mech}}(x, y)$	Mechanical bending strain at position $(x, y)$	–
$\varepsilon(x, y)$	Strain at position $(x, y)$ (when context is clear)	–
$\sigma(x, y)$	Stress at position $(x, y)$	MPa

## 12.2 Nomenclature

## 13 References

[TBD: Add references for material properties, beam theory, strain gauge technology]

Table 17: Nomenclature - Strain gauge parameters

Symbol	Description	Units
$R_g$	Strain gauge nominal resistance	$\Omega$
$GF$	Gauge factor	–
$R_1, R_2$	Top surface gauge resistances	$\Omega$
$R_3, R_4$	Bottom surface gauge resistances	$\Omega$
$\varepsilon_{\text{top}}$	Strain on top surface (total strain at gauges $R_1$ and $R_2$ )	–
$\varepsilon_{\text{bottom}}$	Strain on bottom surface (total strain at gauges $R_3$ and $R_4$ )	–
$V_{\text{ex}}$	Bridge excitation voltage	V
$V_{\text{out}}$	Bridge output voltage	V
$\Delta R$	Change in gauge resistance	$\Omega$
$x_{\text{gauge}}$	Strain gauge $x$ -position	mm
$y_{\text{top}}$	Top surface $y$ -position	mm
$y_{\text{bottom}}$	Bottom surface $y$ -position	mm
$n_{\text{bits}}$	ADC resolution	bits
LSB	Least significant bit voltage	$\mu\text{V}$

Table 18: Nomenclature - Thermal parameters

Symbol	Description	Units
$T$	Temperature	$^{\circ}\text{C}$
$T_0$	Reference temperature	$^{\circ}\text{C}$
$\Delta T$	Temperature change	K
$\Delta\alpha$	Difference in thermal expansion coefficients	$\text{K}^{-1}$
$\varepsilon_{\text{thermal}}$	Simplified thermal strain due to $\Delta\alpha\Delta T$	–
$\Delta L_{\text{thermal}}$	Differential thermal expansion	mm

Table 19: Nomenclature - Strain components and measurement model

Symbol	Description	Units
$\varepsilon_{\text{total}}$	Total axial strain at a gauge location	–
$\varepsilon_{\text{mech}}$	Mechanical bending strain component	–
$\varepsilon_{\text{th}}$	Thermal strain component (differential expansion)	–
$\varepsilon_{\text{off}}$	Offset strain (assembly and electronics effects)	–
$\varepsilon_{\text{meas}}$	Strain inferred from bridge output and calibration	–
$A_T(T)$	Temperature-dependent mechanical strain amplification factor	–
$k_T$	Linear thermal amplification coefficient	$\text{K}^{-1}$