Quantifying Overhung Load in Torque Calibration

A Guidance Document for Quantification, Mitigation, and Uncertainty Budgeting

Abstract. Overhung load, the bending load a torque transducer experiences from anything other than a pure torsional input, can be one of the most underestimated error sources in industrial torque calibration. These extraneous loads vary with rotation about the primary torque axis. When a single-ended moment arm is coupled to a transducer, the weight of the arm plus the applied deadweights imposes a side load and a resultant overturning moment that biases the indicated torque. These bending components do not cancel with rotation unless the transducer's bending sensitivity is negligibly small, and in practice, it rarely is. This document explains what an overhung load is, why ASTM E2428 and BS 7882 treat it as a first-order contributor to uncertainty, and how to quantify it in calibrations where a formal double-loading test is not possible. A worked example from a 1 000 N·m single-ended arm test is carried through three uncertainty budgets. The results show that overhung load can account for more than 99 % of the combined variance in an unsupported-arm setup and that one design choice, either selecting a bending-insensitive transducer or supporting the arm with a bearing, can reduce expanded uncertainty by a factor of three to five without changing anything else in the system.

1. Introduction: Why overhung load matters

A torque measurement means what the calibration certificate says only when the input to the transducer is a pure moment about a single axis — an ideal that no real fixture achieves exactly. Every physical arm has mass; every mass creates a bending load; every bending load couples into the transducer's strain-gage bridge to some degree. The question for a calibration laboratory is not whether an overhung load exists, but how large its contribution is and whether it has been accounted for in the uncertainty budget.

The consequences are concrete. ASTM E2428 limits the expanded measurement uncertainty of the primary torque measurement standard to 0.012 % of applied torque (§5.3.5) and of other types of torque measurement standards to 0.06 % (§5.4.2), both at k = 2. A separate pair of factors governs the bottom of the verified range: for Class A the lower-limit-factor percentage P = 0.25 % of indication (§7.4.1.1) and for Class AA P = 0.06 % (§7.4.1.2). These P values are not uncertainty limits; they are the percentages used to compute the lower torque limit from the lower limit factor (Tll = 100 × LLF / P,

§7.4.1), so a smaller P pushes the verified range lower rather than capping uncertainty. The uncertainty caps and the P factors are distinct quantities, and this document keeps them separate. A single-ended moment arm of moderate weight on a commercially available torque transducer can bias the indication by amounts comparable to, or greater than, the uncertainty its bending contributes can rival or exceed the whole budget a Class A or Class AA device is allowed. The purpose of this document is to make the quantification of that bias a routine procedure: it defines overhung load, identifies its sources, summarizes what the major standards require, explains how to measure its magnitude in setups where the ideal double-loading test of ASTM E2428 §X4.7.2 cannot be performed, and shows how to propagate the result into a defensible uncertainty budget.

2. What overhung load is

2.1 Definition

Overhung load, sometimes called bending moment, overturning moment, or side-loading moment, is any load applied to the elastic member of a torque transducer whose vector is not aligned with the intended torque-measurement axis. In practical terms, it is everything the transducer feels that is not a pure torsional load along its principal axis. ASTM E2428 describes the effect without using the term overhung load explicitly, preferring instead "bending/overturning moments," "parasitic forces due to misalignment" in Note 6 (quoted in §4.1), and the "susceptibility of the elastic torque measurement standard to overturning moments" in §X4.7.2. The terms are interchangeable, and all refer to the defined extraneous loads caused by unsupported torque arms; this document uses overhung load as the unifying term for technical accuracy.

2.2 Why torque transducers respond to bending

A strain-gage torque transducer is designed so that its bridge output responds to shear strain in the elastic member. In an ideal design, the bridge is perfectly balanced against bending strains, gauges on opposite faces of the element see equal and opposite changes under a bending load, and the sum at the bridge output is zero. Real transducers only approximate this ideal. Every unit has some residual bridge imbalance, strain-gauge misalignment on the element, element asymmetry from machining tolerances, and adhesive-bond variation. The result is a small but nonzero mechanical crosstalk coefficient: an off-axis load produces a bridge output that is indistinguishable from a torque of some fraction of that load. The fraction depends on the design and on the individual unit. Baumgarten et al. (2019) characterized this effect quantitatively on a 2 kN·m six-component PTB transducer and showed that an angular deviation of only 0.01° in the placement of one strain gauge can produce a relative signal change of slightly more than 20 % in the parasitic-axial measuring bridge under a 2 000 N·m torque load. Because such manufacturing tolerances are well within the normal range for strain-gauge bonding, each transducer must be characterized individually; manufacturer datasheets generally do not report the residual.

Loading state	R1	R2	R3	R4	Bridge sum (output)
Pure torsion (ideal)	$+\varepsilon$	$-\varepsilon$	$+\varepsilon$	$-\varepsilon$	**4ε** — full output
Pure bending (ideal)	$+\varepsilon_b$	$+\varepsilon_b$	$-\varepsilon_b$	$-\varepsilon_b$	**0** — cancels at bridge
Real bending (cross-talk)	$+\varepsilon_b(1+\delta_1)$	$+\varepsilon_b(1+\delta_2)$	$-\varepsilon_b(1+\delta_3)$	$-\varepsilon_b(1+\delta_4)$	**$\varepsilon_b \cdot \Sigma\delta$** — residual cross-talk

Table 1. Strain at each bridge arm under three loading states. R1 and R4 are bonded at +45°; R2 and R3 at −45°. The bridge sums (R1 − R2 + R3 − R4). Pure torsion produces alternating ±ε that all add. Ideal bending produces uniform tension on top, compression on bottom — the four contributions cancel exactly. In a real transducer, small mismatches δ_i among the four arms (gauge placement, element asymmetry, bond variation) leave a residual proportional to ε_b. The indicator reports this residual as torque.

Bending rejection varies dramatically across transducer types. Shaft-style square-drive transducers, in-line couplings, and similar geometries typically show the lowest rejection and are the most sensitive to bending. Web-style (flange) transducers measure strain closer to the mating interface and can remain largely insensitive to bending across most of their working range; they also tolerate significant overloads without mechanical damage. Multi-axis transducers measure the extraneous components directly and allow compensation at the indicated-torque stage. Because bending sensitivity is a manufacturing variable and is almost never reported on the certificate, the laboratory cannot assume that two nominally identical units behave the same way. Every unit must be evaluated on its own.

2.3 Why is the error largest at low torque

Bending error enters the torque reading as an additive offset that depends on the overhung load (both the side-load component and the overturning moment about the measurement plane) rather than on the applied torque. For a given arm, the overhung-load components are dominated by the weight of the arm plus the deadweights hanging from it and change relatively little with applied torque, while the applied torque itself scales linearly with the weights at the lever arm. The result is that the bending error, expressed as a percentage of indication, is largest at the lowest applied torques. This is why ASTM E2428 §X4.6.1 and Robinson and Pratt's UK work both identify the zero to 20 % region of capacity as the region most affected by bending, and why the lower limit factor and the reproducibility at low torques tend to dominate an unsupported-arm uncertainty budget.

3. Where the overhung load comes from

3.1 Unsupported moment-arm weight

Any moment arm coupled directly to the transducer transfers its own mass into the transducer as a static side load and a resulting overturning moment. The side load is the weight of the arm acting vertically at its center of gravity. The overturning moment is that side load times the distance from the transducer's measurement plane to the center of gravity. The measurement plane is the reference plane at which the transducer's elastic member is sensitive to applied loads. For a flange-mounted (web-style) cell, this is generally the mounting face; for a shaft-style square-drive cell, it is the central plane of the gauged section. A 5 ft (1 524 mm) single-ended torque arm of weight 35 lbf (156 N) at standard gravity imposes a side load of 35 lbf applied roughly at the midpoint of the arm and an overturning moment on the order of 87.5 lbf·ft (119 N·m) before any calibration weights are added. (In this document, "bending moment" and "overturning moment" are used interchangeably to mean the resultant moment on the transducer caused by the side-load lever arm; both are part of the overhung load defined in §2.1.) ASTM E2428 §X4.6.4 identifies this allowance as best practice in the uncertainty calculation when an unsupported moment arm is used.

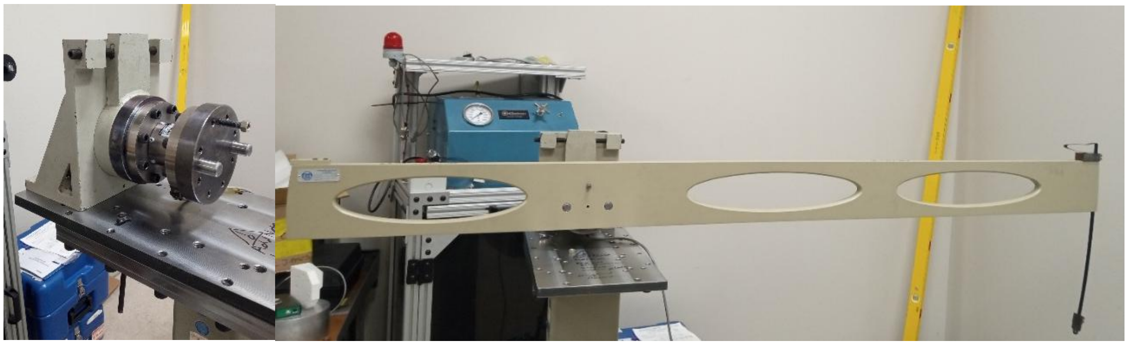


Figure 1. Single-ended unsupported torque arm installed on a torque cell. The arm's mass acts at its center of gravity, producing a vertical side load and an overturning moment about the transducer's measurement plane before any calibration weights are applied.

3.2 Asymmetric arm geometry (single-ended arms)

A single-ended arm applies force at one end only. The center of gravity of the arm-plus-weights system moves as weights are added, and the overturning moment changes with applied load. ASTM E2428 §X4.8.1 acknowledges that single-ended moment arms "provide challenges in quantifying bending/overturning moments that should not be ignored." The double-loading test of §X4.7.2 cannot be used on a single-ended arm because that test requires loading both ends of a symmetrical arm simultaneously. Section 5.2 describes the substitute method used in this document.

3.3 Secondary contributors: misalignment, cable drape, fixture deflection

Cable drape (§X4.6.10) acts similarly: stiff cables, heavy connectors, or unsupported routing impose loading or moment effects on the transducer that a strain-relief near the connector eliminates. BS 7882:2008 handles bending through its reproducibility-with-rotation model and gives a general worked uncertainty method in Annex B, but it does not set out a dedicated quantitative procedure for the unsupported single-ended-arm case treated here; that gap is one of the things the method in Section 5.2 is intended to fill. Fixture deflection introduces an additional misalignment component that increases with applied torque if the calibration frame flexes under load. Flexible couplings, careful fixture design, and proper cable routing minimize all three but do not eliminate them; their residuals enter the budget through the alignment and reproducibility lines.

3.4 Drive-coupling rigidity: a prerequisite, not a source

Worn square drives and loose couplings deserve a different treatment from the contributors above. ASTM E2428 §X4.2.13 warns that "square drives with a loose fit can introduce large measurement errors," and §X4.6.8 directs the laboratory to connect the moment arm directly to the elastic torque measurement standard with all bolted joints torqued to specification to minimize slack in the coupling. The mechanism is twofold: a loose drive cannot transmit a clean torsional input, and slack in the coupling allows the effective moment-arm length to shift under load. Together, these effects transmit torque plus an off-axis contact force whose direction and magnitude shift unpredictably as the load changes. The resulting bending component is neither a fixed bias nor a pure orientation-dependent error; it is non-repeatable, and field experience shows that the resulting errors can be an order of magnitude larger than the entire Class A budget.

This is why drive-coupling rigidity is a prerequisite for any defensible bending characterization, not a parallel source to be budgeted alongside the others. The CW/CCW comparison of §5.2 isolates the bending contribution by reversing a known geometry; if the geometry itself is loose, the comparison measures coupling slop rather than transducer bending sensitivity. The double-loading test of §5.1 has the same dependency. Before any of the methods in Section 5 can be applied, the laboratory must confirm rigid mounting: bolted joints torqued to specification, square drives with minimal play, and adapters seated firmly. ASTM E2428 §X4.6.8 directs that, when possible, the moment arm be connected directly to the elastic torque measurement standard with all bolted joints tightened to applicable torque ratings using a calibrated torque wrench. If the drive is worn, no correction or budget entry will make the measurement repeatable; the coupling must be replaced first.

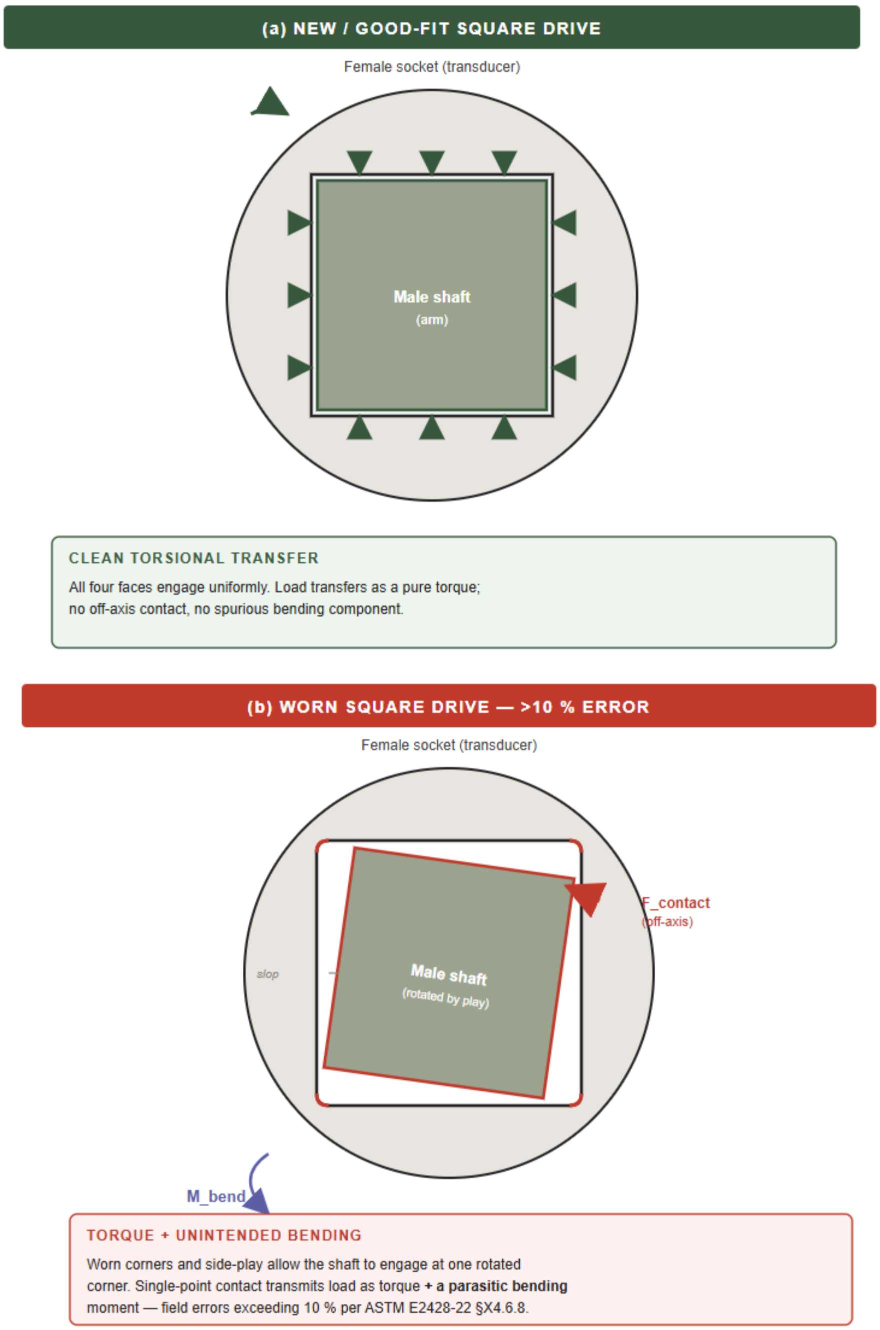


Figure 2. A worn or loose drive coupling transmits its load through an off-axis contact point. The result is torque plus an unintended bending component that the bridge reads as torque error — and that no subsequent characterization or budget entry can correct.

4. What the standards require

4.1 ASTM E2428 and BS 7882

ASTM E2428 (the U.S. authoritative standard) and BS 7882 (the British counterpart and the protocol behind the Robinson and Knott UK intercomparison) align on the treatment of overhung load, but the structure of E2428 is worth a word of explanation before the citations that follow. The body of E2428 (Sections 1–11) is the normative spec; it sets the uncertainty limits, the randomization requirement, and the calibration procedure. The body does not prescribe a specific method for characterizing overhung load. The substantive bending guidance, types of moment arms, the double-loading test, and single-ended-arm considerations are found in the **non-mandatory Appendix X4**. References in this document carrying the "X4" prefix point to that appendix and are informative; references without the "X" (such as §5.3.5, §6.5) point to the binding body of the standard.

From the body of the standard: ASTM E2428 sets the primary torque measurement standard expanded uncertainty at ≤ 0.012 % of applied torque at $k = 2$ (§5.3.5); requires that secondary torque measurement standards be calibrated by primaries and used only over the Class AA verified range of torques (§5.4.1); and caps the expanded uncertainty of other types of torque measurement standards at ≤ 0.06 % of applied torque at $k = 2$ (§5.4.2). It requires rotational randomization between calibration runs (§6.5); for square-drive elastic torque measurement standards, §6.5.6 requires rotation to each of the four square-drive positions, producing four calibration runs per mode (geometrically 90° apart). Note 6, immediately following §6.5.5, is the body's explicit acknowledgment of the bending problem and is worth quoting in full:

> *Note 6 — Depending on their design, elastic torque measurement standards vary in sensitivity to mounting conditions, parasitic forces, or moments due to misalignment. A measure of this sensitivity can be made by imposing conditions to simulate these factors such as (a) using fixtures of varying stiffness or hardness, (b) applying the appropriate torque for bolting fixtures with different torque ratings, or (c) mounting in various orientations with angular or eccentric misalignment, and so forth. Such factors can sometimes be significant contributors to measurement uncertainty and should be reflected in comprehensive measurement uncertainty analyses.*

From Appendix X4: §X4.6 covers bending contributions across moment-arm types — unsupported (§X4.6.1), bearing-supported (§X4.6.2), radius (§X4.6.5), non-radius (§X4.6.6), and dual-radius (§X4.6.7). §X4.7 (Measurement Uncertainty Determination for Bending Effects) specifies the double-loading test for symmetrical arms in §X4.7.2 and §X4.7.3, with the bending-parameter calculation defined in §X4.7.4 and §X4.7.5. §X4.8 acknowledges that single-ended arms require alternative approaches. §X4.2.13 addresses rigidity and fit of adapters. These are non-mandatory but represent the consensus practice for satisfying the body's requirement that significant uncertainty contributors "be reflected in comprehensive measurement uncertainty analyses." A laboratory may use any defensible method; it must be able to justify the choice. BS 7882:2008 takes the same orientation-based approach to capturing bending: it calls for calibration at multiple mounting positions about the measurement axis, with the most demanding classes calibrated at four positions 90° apart (its orientation diagrams illustrate the 0°, 90°, 180°, 270° sequence for square drives), and fewer positions permitted for the less demanding classes. This is compatible with the ASTM §§6.5 / 6.5.6 randomization requirements, and rotating through several orientations to spread bending error into the statistical base is the common principle behind both standards.

4.2 Field evidence: the 2007 UK torque intercomparison

The 2007 UK torque intercomparison (Robinson and Knott, 2009) is the strongest published evidence that the overhung load must be quantified rather than assumed small. The round-robin involved ten laboratories (NPL plus nine participants) across two ranges, 20 N·m to 100 N·m and 200 N·m to 1 kN·m, each measuring at four orientations in accordance with BS 7882:2008. The results are compact and unambiguous.

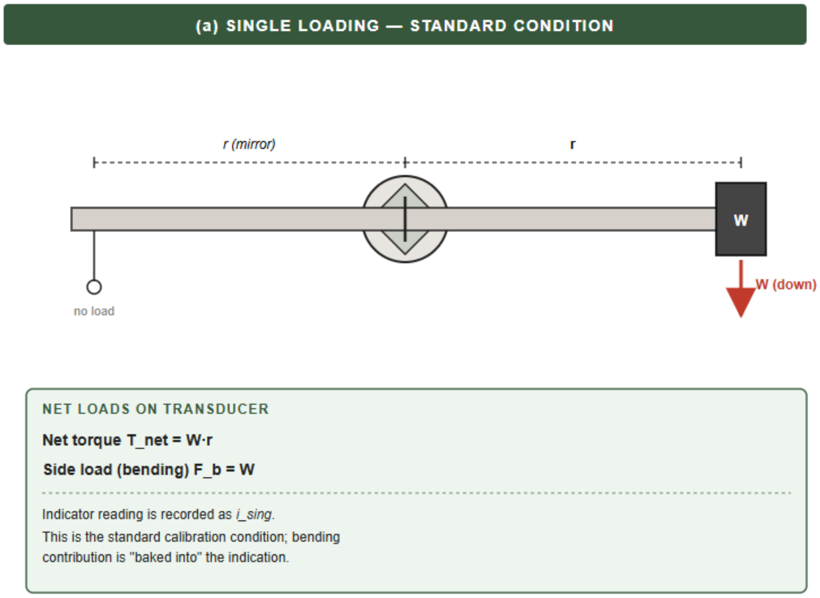
For the 100 N·m transducer, reproducibility across orientations was the dominant source of uncertainty. The transducer's bending sensitivity caused orientation-dependent output variations that overwhelmed all other uncertainty components. Deviations of several tenths of a percent were common at the lower end of the range. On the 1 kN·m transducer, reproducibility was roughly five times better than for the 100 N·m unit, and the applied-torque uncertainty from each laboratory's rig became the dominant contributor. The rig, not the transducer, set the floor. Among the participants, two laboratories used only two or three measurement orientations rather than the four specified by BS 7882:2008, and their results were among those most distant from the NPL reference value. Robinson and Knott attribute these deviations jointly to transducer reproducibility and to the reduced number of orientations; the round-robin does not isolate the orientation count as the sole cause. The practical lesson, which is consistent across the report, is that fewer than four orientations gives the laboratory less ability to absorb orientation-dependent bending error into the statistical base. The intercomparison is a direct, round-robin-validated demonstration that bending sensitivity and orientation-dependent scatter are not theoretical concerns. They are first-order contributors to the measurement uncertainty of torque calibration systems that use unsupported moment arms.

5. Methods for quantifying the error

Three methods are available for characterizing overhung load. They are not mutually exclusive. In practice, a laboratory uses rotational randomization as the standard calibration protocol and supplements it with a bending characterization, double-loading, or CW/CCW whenever a new arm geometry, a new transducer, or a modified fixture is introduced.

5.1 Double-loading test (ASTM E2428 §X4.7.2): symmetrical arms

This is the preferred quantitative test and is applicable only to symmetrical (dual-ended) unsupported arms. The procedure is to exercise the transducer and establish a stable zero, then at each calibration torque, record the indication with single loading (normal calibration condition). Per ASTM E2428-22 §X4.7.3, an additional 50 % of the force is then applied on each side of the arm at the same time, so that 150 % of the force acts in the direction the calibration torque is applied and 50 % acts in the opposite direction. The net torque equals the single-loading torque, but the bending load on the arm is doubled. The indication under this doubled bending load is recorded. The bending parameter is computed as the maximum of the absolute relative difference between the double-loading and single-loading indications across all calibration torques and all orientations:



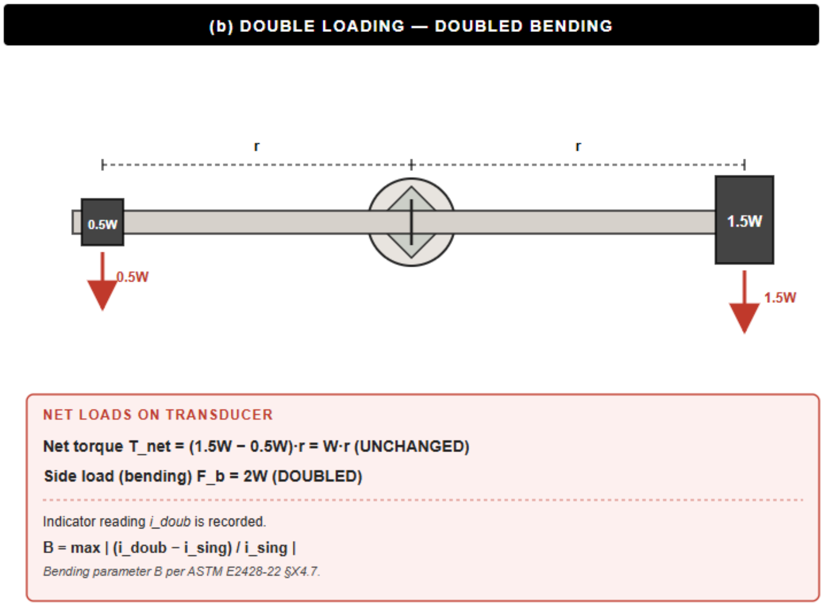


Figure 3. ASTM E2428-22 §X4.7.2 double-loading test. The net applied torque is unchanged between (a) and (b), but the bending load on the arm is doubled. Any shift in transducer output between the two conditions is attributable to bending and defines the bending parameter B.

$$B = \max \left| (i\_doub - i\_sing) / i\_sing \right|$$

The bending parameter B is included directly in the uncertainty budget as a Type B contributor. If B exceeds the resolution of the electronic-indicating instrument, bending is a significant contributor and should be budgeted or corrected through mechanical redesign.

5.2 CW/CCW comparison: single-ended arms

For a single-ended arm, the double-loading test cannot be performed because it requires a symmetrical fixture loaded at both ends simultaneously. ASTM E2428-22 §X4.8.1 directs the reader to §X4.3 for single-ended-arm methodology, but §X4.3 covers proficiency testing rather than a quantitative method; the standard therefore identifies the gap that this section is intended to close. The CW/CCW comparison described here is a substitute method that isolates the bending offset by reversing the arm about the transducer axis. The conceptual idea — use a torque-direction reversal to separate a sign-switching bending contribution from a sign-switching torque contribution — has appeared in the literature for a different geometry: Saenkhum and Sanponpute (NIMT, 2019) used CW vs CCW loading on a not-quite-symmetric double-ended fixture, with hinge/fulcrum signal separation, to isolate an asymmetric bending term in the NIMT 5 kN·m torque standard machine. The method below differs in geometry (a deliberately single-ended arm, as actually deployed in field calibration) and in purpose (a Type B input to a customer-side uncertainty budget rather than an NMI-grade machine characterization), but acknowledges the prior art. The procedure has five steps. First, record the transducer output with the arm installed in the clockwise loading position and no additional weights. Second, rotate the arm to the counter-clockwise loading position (same geometry, same weight, opposite side of the transducer) and record the output. Third, take the difference between the two readings; this difference is caused by the arm's side load acting on opposite sides of the transducer. Fourth, divide this difference by the arm weight to obtain a per-lbf (or per-newton) bending sensitivity for the transducer, expressed in mV/V per unit force. Fifth, multiply the per-force sensitivity by the total weight applied at full-scale calibration torque to estimate the worst-case bending contribution, then convert to a percentage of applied torque and include it as a Type B component of the uncertainty budget.

This method does not produce a metrologically sound correction that can be applied routinely to calibration data, and the reason is important enough to state before the worked example. The per-force sensitivity produced by the five steps above is a property of one transducer in one fixture at one orientation. When the arm is rotated to the next calibration position, as ASTM E2428 §6.5 requires, the side load acts on a different axis of the transducer's internal geometry, and the sign and magnitude of the bending shift change with it. Applying a single CW/CCW-derived correction uniformly across all calibration runs would therefore subtract the right number at one orientation, the wrong number at the opposite orientation, and some partial number at the others. It would reduce the scatter at one position by fabricating a matching error at its mirror, which is not a correction; it is a redistribution of the same bias.

A correction that could be applied routinely to calibration data would have to be orientation-invariant, or it would have to be measured as a full orientation-resolved surface (CW/CCW at every randomization angle) with its own propagated uncertainty. The CW/CCW procedure in §5.2 produces neither. Section 7.2 returns to this point in the language of the standard: "The CW/CCW-derived correction from Section 6 is position-dependent. It is valid for the specific arm geometry at the specific orientation at which it was measured… ASTM E2428-22 §6.5 addresses parasitic effects through rotational randomization, not through empirical correction factors; an empirical correction applied uniformly to all orientations would double-count the error at some positions and miss it at others."

What the method does produce, and does so reliably, is a defensible magnitude estimate of the bending contribution in that configuration. This is exactly the quantity the uncertainty budget needs. The estimate enters the budget as the Type B input on the bending line, with the standard uncertainty taken as the projected full-scale shift divided by √3 under a rectangular-distribution assumption per JCGM 100:2008 §4.3.7 (§7.3). A more conservative laboratory may adopt a triangular distribution (with divisor

√6) when the central tendency at zero is justified by the physics of the bending model; we retain the rectangular convention for simplicity. The worked example in Section 6 carries a single CW/CCW sequence through the five steps and arrives at that budget input; §7.2 and §7.3 then propagate it into the expanded uncertainty. Randomization per §6.5, not the CW/CCW correction, is what suppresses the orientation-dependent component of the error in the calibration result itself. A practical caveat applies to commercial laboratories: the per-lbf bending sensitivity coefficient is a property of the specific transducer in the specific fixture and must be measured for every new transducer that enters service. The procedure adds setup time, custodial complexity, and a per-unit characterization step that does not scale gracefully across a busy calibration shop. Section 8 discusses the alternatives, supporting the arm with a bearing (§8.2), switching to a torque transfer (comparison) machine, or adopting a dual-radius symmetrical arm (§8.4), each of which removes the need to characterize bending per transducer at all. The CW/CCW method is therefore best treated as a defensible quantification tool for legacy single-ended-arm rigs that cannot be refixtured, rather than as a long-term operating posture.

5.3 Rotational randomization (ASTM E2428 §6.5)

Rotational randomization is required for every calibration under ASTM E2428, regardless of whether a separate bending characterization is performed. Rotating the transducer between calibration runs causes bending-induced errors to appear in different directions at each orientation, spreading them into the statistical base so that the reproducibility component of the budget accounts for them. Section 6.5.6 specifies square-drive elastic torque measurement standards, rotation to each of the four square-drive positions, producing four calibration runs per mode. Robinson and Knott (2009) reported that the laboratories using only two or three orientations were among those most distant from the NPL reference, though their analysis attributes the deviations jointly to reproducibility and orientation count rather than to orientation count alone. The four-orientation requirement for square-drive standards is set by ASTM E2428 §6.5.6 itself, not by the intercomparison. Randomization does not quantify bending directly; for that, the double-loading test of §5.1 or the CW/CCW comparison of §5.2 is required.

6. Worked example: single-ended arm retrofit

This section carries a single test sequence through quantification, projection, and verification. The transducer under evaluation was a square-drive Basic Torsion Cell mounted on a single-ended unsupported moment arm in a field torque calibrator. The calibration target was 1 000 N·m. The Basic Torsion Cell was chosen as a baseline because it has limited bending rejection; some other web-style cells exhibit roughly four times better bending performance, and the errors seen here are proportionally smaller when those cells are in place.

6.1 Setup

The arm was a single-ended torque arm 4.993 067 ft (1.522 m) long with a measured weight of 35 lbf (156 N) at standard gravity, corresponding to a mass of ≈15.9 kg. The full-scale weight stack of 147.717 3 lbf (657.0 N) applied at 4.993 067 ft produced a nominal 1 000 N·m input. The transducer was loaded with the connector pointed down and the arm installed in two orientations, clockwise and counter-clockwise, with no calibration weights applied. All indications were recorded in mV/V.

6.2 CW/CCW data

Reading: −0.001 83 mV/V
connector pointed down

CCW Reading: −0.148 72 mV/V
connector pointed down
(Arm creates torque and side load)

CW Reading: −0.144 48 mV/V
connector pointed down
(Arm creates torque and side load)

Loadcell S/N 23A0801*

	(mV/V)	
Output from loading with torque arm	1.934481	(Actual force= 147.7173 lbf @ 4.993067 feet or 1000Nm)
Output from primary torque deadweight.:	1.93121	1000Nm (with .002% uncertainty)
Difference=	0.003271	0.169376 % difference
Removing calculated error of cross talk** :	1.931021	(1.934481-.00346)
Difference of correct value against primary std:	-0.000189	-0.00979 % difference (17 times lower error)

* All data is for Clockwise Loading Condition and 4.993067 long torque arm
** Error correction =-.0000166 mV/V per lbf from calculation on previous page

Figure 4. CW (left), connector-down baseline (center), and CCW (right) mounting of the single-ended arm on the torque cell. Readings correspond to the three rows of Table 2.

Condition	Orientation	Output (mV/V)
Transducer only, no arm	Connector down	−0.001 83

Condition	Orientation	Output (mV/V)
Arm installed, no weights	Clockwise loading position	−0.144 48
Arm installed, no weights	Counter-clockwise loading position	−0.148 72

Table 2. Zero-weight outputs with the arm in CW and CCW positions on the Basic Torsion Cell.

6.3 Isolating the bending shift

The arm creates both a torque (equal and opposite in the two positions) and a side load. The shift in transducer output between CW and CCW positions is attributable to the side load, because the torque component averages out when the arm reverses. Taking the magnitude of the arm-induced shift in each direction and subtracting, the net side-load-induced shift on the transducer is 0.146 31 mV/V − 0.146 89 mV/V = −0.000 58 mV/V. This is the bias in the zero-torque reading caused by reversing the side load.

Dividing by the arm mass gives the transducer's bending sensitivity coefficient for this mounting geometry: 0.000 58 mV/V divided by 35 lbf is approximately −0.000 016 6 mV/V per lbf. This coefficient is the property of the specific transducer in the specific fixture. It is not a universal constant, and it cannot be transferred between setups.

6.4 Projecting to full scale

To estimate the bending error at full-scale torque, multiply the per-lbf coefficient by the total calibration weight: −0.000 016 6 mV/V per lbf × 148 lbf = −0.003 46 mV/V at 1 000 N·m. Expressed as a percentage of the nominal full-scale output of 1.931 21 mV/V, this is roughly 0.179 %, several times the bending budget a Class AA device could tolerate and comparable to the whole error budget allowed for a Class A device.

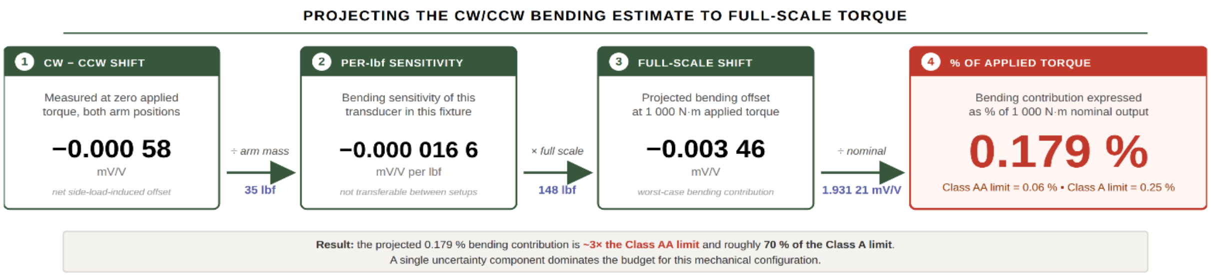


Figure 5. Four-step projection of the CW/CCW bending estimate to full-scale applied torque. The per-lbf bending sensitivity from the zero-load comparison is scaled by the full-scale calibration weight and

expressed as a percentage of nominal output, yielding the projected bending contribution that enters the uncertainty budget.

6.5 Verification against the primary standard

With the arm installed and the full calibration weight applied, the cell read 1.934 481 mV/V. When the same cell was calibrated on the Morehouse primary torque deadweight standard (expanded uncertainty 0.003 %, $k = 2$), the output at 1 000 N·m was 1.931 21 mV/V. The difference is 0.003 27 mV/V, or 0.169 % of the primary reading. Applying the CW/CCW-derived bending correction of −0.003 46 mV/V gives 1.934 481 mV/V − 0.003 46 mV/V = 1.931 021 mV/V, which agrees with the primary standard to within −0.000 189 mV/V or −0.010 %, a factor of seventeen lower than the uncorrected discrepancy.

Quantity	Value (mV/V)	% of primary
Output with torque arm at 1 000 N·m	1.934 481	—
Primary deadweight standard at 1 000 N·m	1.931 21	—
Uncorrected difference	0.003 27	+0.169 %
CW/CCW-derived bending correction	−0.003 46	—
Corrected output	1.931 021	—
Residual difference from the primary standard	**−0.000 189**	**−0.010 %**

Table 3. Full-scale verification of the CW/CCW-derived bending estimate against the Morehouse primary torque deadweight standard.

Two cautions apply to this verification. First, the same CW/CCW dataset is used to derive the bending coefficient and to demonstrate post-correction agreement; the closure to 0.010 % is therefore a self-consistency check on the method, not an independent validation against an external dataset. A future revision of this guidance will report a cross-validation against an independently-acquired CW/CCW sequence on a different transducer. Second, the correction is applied here as a scalar to illustrate the magnitude of the bending contribution; in routine calibration it is propagated into the budget as a Type B input (§7.2), not subtracted from the result. With those cautions noted, the closure demonstrates two things at once. First, the CW/CCW method identifies the correct magnitude of the bending contribution, and the estimate closes the loop with the primary standard to within the noise floor. Second, the uncorrected 0.169 % disagreement is physically real and is caused almost entirely by bending; it is not a transducer defect, an indicator error, or an installation error.

7. Building the uncertainty budget

7.1 What goes in the budget

The components required in any torque calibration uncertainty budget are familiar from JCGM 100:2008 (GUM) and are enumerated in ASTM E2428 §X4.2: the reference standard, the moment-arm length, alignment and cosine effects, bending and overturning, reproducibility of the transducer across orientations, weights (adjusted for force and their associated MU), and environmental factors (temperature being the most important for torque). The dominant contributor depends on the mechanical configuration. In an unsupported single-ended arm with a bending-sensitive transducer, bending is dominant. In a bearing-supported arm, bearing friction takes its place. In a well-designed web-style transducer with a lightweight symmetrical arm, the reference standard and arm length tend to dominate.

7.2 Why the CW/CCW correction goes into the budget, not the calibration

The CW/CCW correction is position-dependent: its sign and magnitude change as the arm is rotated, so applying it uniformly across all four randomization positions would double-count the error at some orientations and miss it at others. ASTM E2428 §6.5 addresses parasitic effects through randomization, not through empirical correction. The correct use of the CW/CCW estimate is as the Type B input to the bending line of the uncertainty budget — it quantifies the contributor; the budget propagates it; randomization handles the orientation-dependent component in the calibration result itself.

7.3 Three rough demonstration budgets for the same measurement

The measurement model underlying these budgets is $\tau\_measured = \tau\_applied \cdot (1 + \delta\_ref + \delta\_arm + \delta\_align + \delta\_bend + \delta\_repro + \delta\_temp)$, where each δ is a relative deviation expressed in the same units as its parent uncertainty (% of applied torque), so the sensitivity coefficient on every component is unity and the combined standard uncertainty is the root-sum-square of the tabulated u_i. The three budgets that follow are rough demonstration budgets, not full uncertainty budgets in the JCGM 100 (GUM) sense. Their purpose is to illustrate how a single mechanical change shifts the dominant contributor and the resulting expanded uncertainty — not to substitute for the disciplined budget any laboratory would build for an actual calibration. A full GUM-compliant budget includes correlated terms, sensitivity coefficients derived from the measurement model, distribution-shape justifications, and documented degree-of-freedom calculations; this requires substantially more characterization work than the values shown here. A word on what the verdict lines mean. The budgets compute an expanded uncertainty and compare it against the expanded-uncertainty allowance appropriate to the intended class of use: 0.06 % for a device used over the Class AA verified range and the looser allowance carried by Class A work. The percentages 0.06 % and 0.25 % that appear as Class AA and Class A in §7.4.1 of ASTM E2428 are the lower-limit-factor P values used to compute the bottom of the verified range, not uncertainty ceilings; the comparison here is against the uncertainty a device of that class is expected to carry, and the class names are used in that sense throughout these tables. All three budgets describe the same 1 000 N·m test in three mechanical configurations. All other components (reference standard, moment-arm length, alignment, temperature, reproducibility) are held constant to isolate the bending contribution. The standard uncertainty for bending in Budget 1 is taken as the projected full-scale bending shift from Section 6 divided by √3, consistent with a rectangular-distribution treatment of the position-dependent offset per JCGM 100:2008 §4.3.7. (Numerically, 0.179 % / √3 ≈ 0.103 %, which we round to 0.100 % in Table 4 for clarity.) Budget 2 scales this by a factor of four to reflect the bending rejection of the compensated torsion cells relative to the basic torsion cell measured in Section 6. The 4× ratio is specific to this comparison and is supported in this document by the worked example only; it is not a universal property of web-style vs shaft-style architecture, and laboratories applying this guidance to other cells should characterize the ratio from their own measurements per Section 5.2. Budget 3 replaces the bending contributor with bearing friction.

7.3.1 Budget 1: basic torsion cell with bending, unsupported single-ended arm

Component	u_i (%)	Type	Source
Reference standard (load cell)	0.010	B	Calibration certificate, Class AA (§5.4)
Moment-arm length	0.002	B	Calibration certificate, SI traceable
Alignment and cosine error	0.002	B	§X4.2.7, §X4.6.6
Bending and overturning	0.100	B	CW/CCW test per Section 6
Reproducibility of the transducer	0.005	A	Rotation per §6.5
Temperature	0.001	B	§6.3, ±1 °C during calibration
Combined standard uncertainty u_c	**0.1007**	—	*Root-sum-square of components*

Component	u_i (%)	Type	Source
Expanded uncertainty U (*k* = 2)	**0.201 %**	—	***Exceeds Class AA (≤ 0.06 %) and Class A (≤ 0.25 % applied torque limit)***

Table 4. Budget 1 — Basic Torsion Cell with unsupported single-ended arm. Bending accounts for about 99 % of the combined variance.

Bending accounts for $(0.100)^2 / (0.100\,7)^2 = 98.6\,\%$ of the combined variance in this configuration. Every other line in the budget is immaterial until the bending contribution is reduced. No amount of improvement to the reference-standard calibration, the moment-arm length, or the temperature control can bring this system into Class AA or Class A compliance. The mechanical configuration must change.

7.3.2 Budget 2: compensated torsion cell, same unsupported arm

Component	u_i (%)	Type	Source
Reference standard (load cell)	0.010	B	Calibration certificate, Class AA (§5.4)
Moment-arm length	0.002	B	Calibration certificate, SI traceable
Alignment and cosine error	0.002	B	§X4.2.7, §X4.6.6
Bending and overturning	0.025	B	CW/CCW test, bending compensated cell (~4× better rejection)
Reproducibility of the transducer	0.005	A	Rotation per §6.5
Temperature	0.001	B	§6.3, ±1 °C during calibration
Combined standard uncertainty u_c	**0.0277**	—	*Root-sum-square of components*
Expanded uncertainty U (*k* = 2)	**0.055 %**	—	***Meets Class A (≤ 0.25 % applied torque limit)***

Table 5. Budget 2 — compensated torsion cell with the same unsupported arm. Bending still dominates, yet is manageable.

Replacing the bending-sensitive cell with the specified cell, without changing anything else about the calibration, reduces the bending contribution from 0.100 % to approximately 0.025 % and the expanded uncertainty from 0.201 % to 0.055 %. The system now meets Class A. It still does not meet Class AA. The choice of transducer alone yields a 3.7-fold reduction in expanded uncertainty.

7.3.3 Budget 3: bearing-supported arm, any reasonable cell

Component	u_i (%)	Type	Source
Reference standard (load cell)	0.010	B	Calibration certificate, Class AA (§5.4)
Moment-arm length	0.002	B	Calibration certificate, SI traceable

Component	u_i (%)	Type	Source
Alignment and cosine error	0.002	B	§X4.2.7, §X4.6.6
Bearing friction	0.020	B	§X4.6.2, characterized from bearing spec
Reproducibility of the transducer	0.005	A	Rotation per §6.5
Temperature	0.001	B	§6.3, ±1 °C during calibration
Combined standard uncertainty u_c	**0.0229**	—	*Root-sum-square of components*
Expanded uncertainty U (*k* = 2)	**0.046 %**	—	***Meets Class A; approaches Class AA with refinement***

Table 6. Budget 3 — Bearing-supported arm. Bearing friction replaces bending as the dominant contributor.

Supporting the arm with a bearing eliminates the bending contribution entirely, at the cost of a new bearing-friction contributor. For a well-characterized bearing, friction can be held below 0.02 % of applied torque. The combined expanded uncertainty drops to 0.046 %, a factor of 4.4 improvement over Budget 1. Further refinement of the bearing characterization, or a move to a flexure-supported arm, can bring the system close to the 0.012 % primary-standard limit of §5.3.5.

7.4 Summary comparison

Configuration	Expanded U (%)	Classification
Basic Torsion Cell, unsupported arm (Budget 1)	0.201	Fails both classes
Compensated cell, unsupported arm (Budget 2)	0.055	Meets Class A
Any cell, bearing-supported arm (Budget 3)	0.046	Meets Class A

Table 7. Summary of the three configurations. Either transducer selection or arm support is sufficient to cross the Class A threshold; neither alone is sufficient for Class AA.

8. Practical mitigations

The evidence from the worked example and from the UK intercomparison points to a small number of design choices that determine whether the overhung load is a manageable contributor or a disqualifying one. The four approaches below are listed in the order in which a laboratory should consider them when commissioning a new torque calibration rig.

8.1 Select a bending-insensitive transducer

The first and highest-leverage decision is the transducer. Web-style (flange) transducers can remain largely insensitive to bending across most of their working range. Note that the bending error expressed as a percentage of full scale may stay roughly flat with utilization, while the same error expressed as a percentage of reading grows hyperbolically at low torque, because the parasitic moment is fixed by the fixture geometry while the torque signal scales with the calibration weights (see §2.3). Shaft-style square-drive transducers typically do not. Multi-axis transducers (three-axis or six-axis) measure the extraneous force and moment components directly and allow compensation in the indicated torque; their cost and complexity are considerably higher, and the calibration of the extraneous axes is nontrivial, so they are typically reserved for applications where the extraneous loads cannot be eliminated by fixturing. A laboratory procuring a new reference transducer should require bending-sensitivity data from the manufacturer as part of the acceptance criteria.

8.2 Support the moment arm with a bearing

Bearing support per ASTM E2428 §X4.6.2 decouples the arm weight from the transducer entirely. The bending contribution becomes zero and is replaced by bearing friction. For an air-bearing-supported lever fulcrum of the type used in NMI deadweight torque machines, friction can be held below 0.02 % of applied torque (NPL's 2 kN·m TSM reports an expanded uncertainty floor of 0.03 % at k = 2, dominated by air-bearing friction; PTB's 20 kN·m TSM reports a parasitic-load floor near 2×10^{-5} per Baumgarten et al. 2019). For an ordinary angular-contact ball bearing or roller bearing on an industrial torque arm, the achievable floor is typically an order of magnitude higher and is hysteretic; Baumgarten et al. (2019) document non-linear hysteretic behavior on a deep-groove ball bearing in the VTT 2 kN·m machine. Specify the bearing type when claiming a friction floor. Bearing support is the most effective mechanical solution for an existing unsupported-arm system. It requires a new mounting plate and modifications to the surrounding fixture, but the modification pays back in uncertainty-budget terms almost immediately.

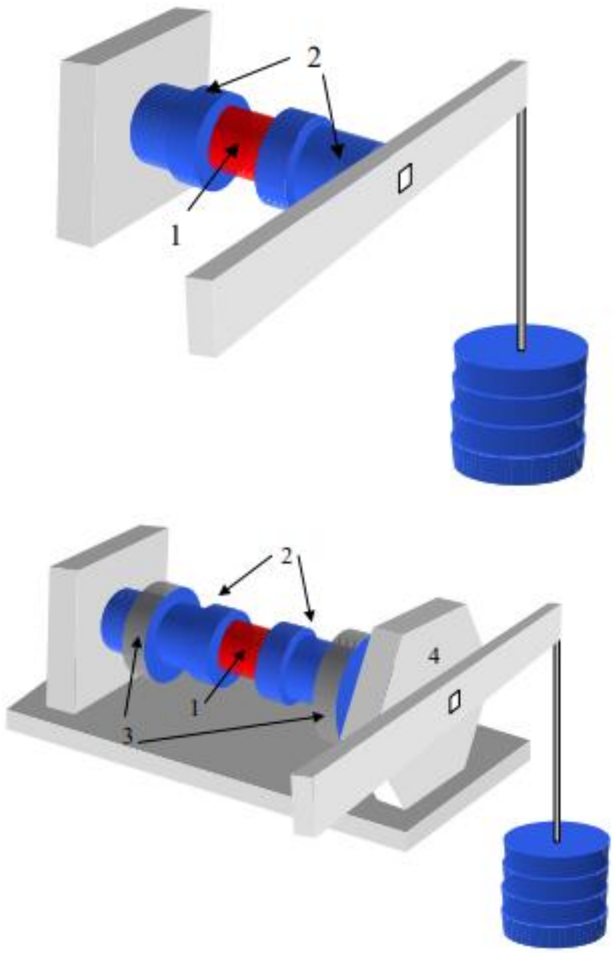
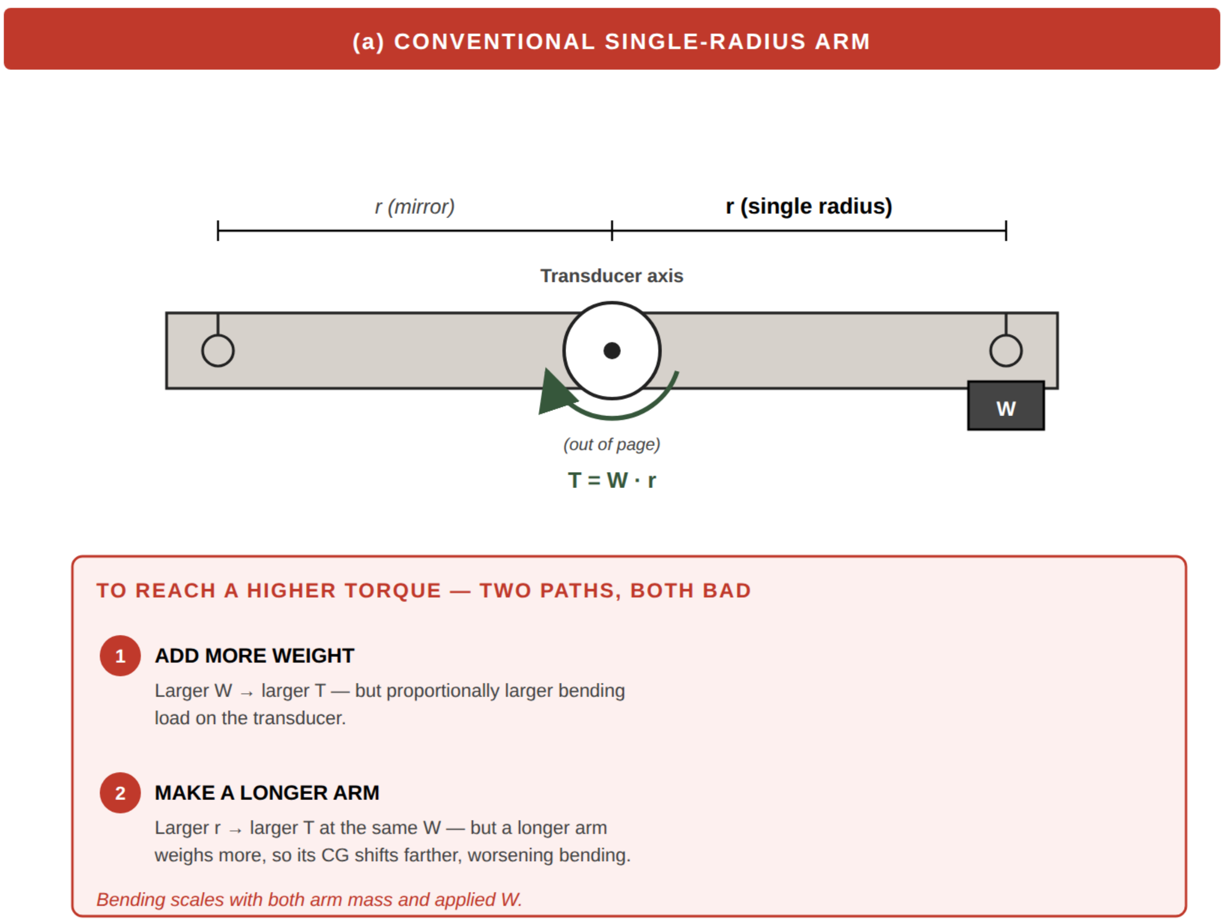


Figure 6. Mechanical isolation of the transducer from the arm weight. A bearing carrying the arm's mass independently of the transducer eliminates the bending contribution entirely; the bearing's friction torque becomes a new, generally smaller, uncertainty component that must be characterized.

8.3 Use the lightest, longest practical arm

ASTM E2428 §X4.6.3 specifies that the lightest moment arm commensurate with the transducer capacity should be used. Carbon-fiber composites, aluminum alloys, and properly designed hollow steel sections can reduce arm mass by roughly half compared with solid-steel arms of equivalent stiffness. Two effects combine to reduce the overhung-load contribution. First, a lighter arm produces a smaller static side load and a smaller resultant overturning moment about the measurement plane; the static portion of the offset can be tared at zero, but the orientation-dependent component cannot, and that component scales with arm mass. Second, a longer arm reaches the same applied torque at a smaller calibration weight, so the dynamic contribution from the deadweights themselves is reduced. The two design choices work together: a longer arm allows less weight for the same torque, and a lighter arm at any length reduces the orientation-dependent bending contribution that randomization must absorb. The distance between the arm and the transducer should also be minimized; every inch of offset is an additional lever for the arm's weight.

8.4 Consider a dual-radius arm



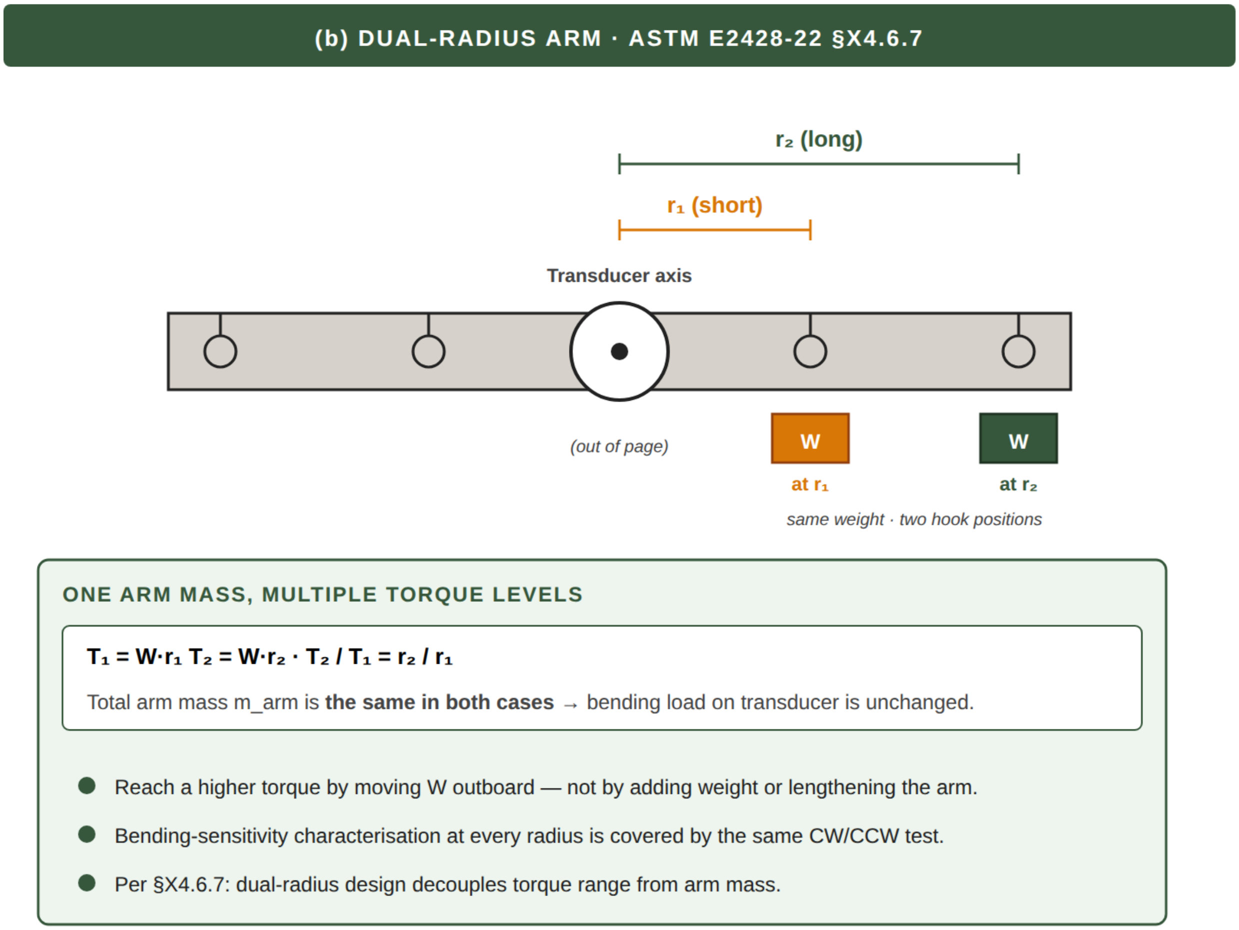


Figure 7. Dual-radius moment arm geometry per ASTM E2428 §X4.6.7. Hooks at two discrete radii allow one arm to cover a wider torque range without scaling up arm mass. The bending contribution does not vanish, but it is prevented from growing with the upper end of the calibration range.

ASTM E2428 §X4.6.7 describes the dual-radius method, in which a single symmetrical arm carries hooks at two or more discrete radii so that one arm can cover a wider torque range without a proportional increase in arm mass. The benefit is bounded but real: by avoiding the need for a longer or heavier arm at higher torques, the dual-radius design prevents the bending contribution from scaling up with the upper end of the calibration range.

ASTM E2428 §X4.6.7 frames the dual-radius method as a way to “keep a constant mass and bending effect on the elastic torque measurement standard,” with the bending contribution absorbed into the calibration zero. That framing is partially accurate but warrants three caveats for laboratories building uncertainty budgets. First, even if the absolute bending load were truly constant, the bending error expressed as a percentage of indication still grows hyperbolically as applied torque approaches zero — exactly the behavior identified in §2.3 of this document and demonstrated empirically by Pratt and Robinson (2008) in the < 20 % region. Second, the bending load is not perfectly constant in practice: as weights are moved between radii the loaded center of gravity shifts, the arm itself flexes more under higher torque (producing the mean shifts that Pratt and Robinson observed in their double-loading data), and the hook geometry changes between radii. Third, the bending error at any given orientation is sign-dependent and reverses when the arm is rotated, which is why §6.5 addresses parasitic effects through randomization rather than a single zero offset; an orientation-dependent error cannot be zeroed out at one position without doubling itself at the opposite position.

The practical claim for the dual-radius approach is that it improves operational flexibility — one arm covers more of the range — and avoids the secondary growth in bending contribution that would come with a longer or heavier arm at higher torques. It reduces but does not eliminate the bending contribution, does not flatten the low-torque error, and does not substitute for the bending characterization recommended in §X4.6.4 or for the rotational randomization required by §6.5.

8.5 Reconsider the single-ended arm

The four mitigations above (§8.1 through §8.4) all assume the laboratory is committed to a single-ended unsupported moment arm. For most commercial calibration work, that assumption deserves to be challenged on its own terms before any of the mitigations are applied. The single-ended unsupported arm is the most error-prone of the moment-arm geometries listed in ASTM E2428-22 §X4.6: it imposes a static side load and an overturning moment on the transducer at zero torque, requires per-transducer bending characterization to write a defensible uncertainty budget, and as the worked example in Section 6 shows can produce expanded uncertainties that fail both Class A and Class AA limits when paired with a bending-sensitive cell. The cost of running the CW/CCW characterization procedure in §5.2 on every transducer entering service is real, ongoing, and difficult to absorb at commercial calibration throughput.

Three alternative architectures eliminate the problem at the source rather than budgeting it. First, supporting the arm with a bearing (§8.2) decouples the arm weight from the transducer entirely. Second, a torque-transfer (comparison) machine in which a calibrated reference torque transducer is mounted in series with the unit under calibration, and the applied torque is generated by a hydraulic or electromechanical actuator rather than by dead weights on a lever, avoids the moment-arm geometry altogether. Third, a dual-radius symmetrical arm (§8.4) carries balanced masses on both sides of the transducer axis, so the static side load and the orientation-dependent bending error are zero at the calibration zero. Each of these architectures removes the need to characterize bending per transducer.

A single-ended unsupported arm is, in principle, defensible in two situations: when the rig is a legacy installation that cannot be re-fixtured, and when the calibration is performed on a stiff, bending-insensitive web-style transducer at a torque range and class limit that leave headroom for the CW/CCW-budgeted bending contribution. In both cases, the laboratory must accept (a) per-transducer characterization as a routine operating expense, (b) a budget in which bending is the dominant contributor unless transducer selection is tightly controlled, and (c) limited ability to claim Class AA verified ranges. Outside those two cases, the better commercial answer is to move the work to one of the three architectures above rather than to refine the characterization further. ASTM E2428-22 §X4.8.1 itself describes single-ended arms as posing “challenges in quantifying bending/overturning moments that should not be ignored,” and its cross-reference to §X4.3 is, as noted in §5.2 of this document, not a substantive quantitative method. The standard does not endorse single-ended arms as a preferred configuration; this guidance does not either.

9. Guidance summary

Overhung load is not a rare failure mode of torque calibration. It is a first-order uncertainty contributor that is always present in any calibration that uses an unsupported moment arm, and it often dominates the uncertainty budget for single-ended arms with bending-sensitive transducers. A laboratory that does not characterize bending cannot write a defensible uncertainty budget for such a configuration, and a laboratory that does characterize bending will find that small mechanical changes, such as a better transducer, a bearing on the arm, or a lighter arm, produce large improvements in the expanded uncertainty.

One caution deserves emphasis. Quantifying the actual bending error, rather than bounding its magnitude, is difficult. The CW/CCW comparison of §5.2 yields a defensible estimate for the uncertainty budget, but it does not produce a true correction, because the error reverses sign and changes magnitude with orientation. The size of the effect is also strongly cell-dependent. Two nominally identical transducers can differ by a factor of several in bending rejection, and the coefficient does not transfer between units or between fixtures. For these reasons, any laboratory running a single-ended arm should treat the configuration as provisional. Support the arm with a bearing, switch to a bending-insensitive transducer, or move the work to a different standard or system, such as a torque transfer machine or a dual-radius symmetrical arm. Characterizing bending on every transducer that enters service is a defensible stopgap for a legacy rig. It is not the configuration to choose for new work.

The recommended practice for a force and torque calibration laboratory is as follows. Every torque calibration should include rotational randomization at four orientations per ASTM E2428 §6.5, regardless of arm type or transducer design. Every new arm geometry, transducer, or fixture should be characterized for bending sensitivity before it enters service using the double-loading test of §X4.7.2 if the arm is symmetrical, or the CW/CCW comparison of Section 5.2 of this document if it is single-ended. The bending estimate should be propagated into the uncertainty budget as a Type B component; it should not be applied as an empirical correction to calibration data, because the correction is orientation-dependent and applying it uniformly would distort the result. Transducer selection and arm support should both be reviewed when the budgeted expanded uncertainty approaches the Class A or Class AA limit; either choice, implemented correctly, produces a threefold to fivefold improvement. For new rig builds, the recommended ordering of architectures is: torque transfer (comparison) machine first, bearing-supported lever or dual-radius symmetrical arm second, single-ended unsupported arm only when no other option is available. The CW/CCW characterization of §5.2 keeps a legacy single-ended rig defensible; it is not a reason to build a new one.

The worked example in Section 6 closed the loop with the Morehouse primary torque deadweight standard to within 0.010 %, after bending was quantified and corrected from an uncorrected disagreement of 0.169 %. **The disagreement was real, it was physical, and it was quantifiable. It was also invisible on the raw calibration data. That is the central message of this document: overhung load is a measurable physical effect, and any laboratory that treats it as a minor correction is likely underestimating its expanded uncertainty by an order of magnitude.**

The practical conclusion follows from where that error originates. The most cost-effective place to address overhung load is the architecture of the calibration rig, not the per-transducer uncertainty budget. A correction applied one transducer at a time manages the symptom; a rig that constrains the load path removes the cause. Carried to its logical end, the architecture that best controls overhung load is one that does not rely on unsupported single-ended arms at all. Once the uncertainty those arms contribute is quantified, as Section 6 quantifies it, a dedicated torque transfer machine that eliminates the arms and weights stops being a matter of convenience and becomes the defensible long-term path to lower uncertainty.

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Morehouse has built force calibration equipment since 1925, and the primary torque deadweight standard used in Section 6 is the same standard behind our accredited torque calibration service. Our torque transfer machines are designed to eliminate the single-ended-arm error this document quantifies, so the correction does not have to live in your uncertainty budget. If you are not certain how much overhung load your current method is carrying, that is precisely the question we can help you fix, by either supporting the arm or helping you select a better torque calibration standard/machine.

To review your torque calibration requirements, evaluate your current setup, or discuss a torque transfer machine, contact Morehouse Instrument Company:

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