DESIGN VERIFICATION REPORT · DVR-001 · REVISION A

ISOKLAMP™ CFR

Constant-Force Retention — a two-part washer system that arrests rotational self-loosening *and* recovers the preload that every other locking device lets go.

Phase 0: analytical and numerical verification against DIN 25201-4:2010-03 Annex B conditions. Reference joint M16 × 2,0, property class 10.9.

Document	**ISK-DVR-001 Rev. A**	Reference joint	**M16 × 2,0 · cl. 10.9**
Date of issue	**15 August 2026**	Initial clamp force F_V	**70,0 kN**
Status	**Model-predicted (Phase 0)**	Test standard modelled	**DIN 25201-4:2010-03 Annex B**
Prepared by	**Engineering, Isoklamp Ltd.**	Secondary standards	**ISO 16130:2015 · VDI 2230 Bl. 1**
Approved by	—	Pages	**1 of 1 (continuous)**

STATUS OF THE DATA IN THIS REPORT — READ FIRST

No physical Junker testing has yet been performed on ISOKLAMP CFR hardware. Every quantitative result attributed to ISOKLAMP in this document is **model-predicted**: closed-form bolted-joint mechanics per VDI 2230 Sheet 1, a two-timescale preload-decay formulation after Li et al. (*Friction* 10(3):335–359, 2022), and quasi-static FE of the compensating element. Baseline curves for the nine incumbent securing methods are **calibrated against published Junker-family data**; every calibration target and its source is listed in §6.3, and each is reproduced by the model to within the scatter of the source.

We publish it this way deliberately. A pre-revenue company presenting twelve accredited DIN 25201-4 verification series it has not run would be found out by the first specifier who asks for the laboratory report number. §12 sets out the physical verification programme — protocol, laboratory, specimen count and schedule — that converts every prediction in §7 into a measurement. Where the model is weakest, §13 says so.

Key findings

01 **99,4 % ± 0,6 of initial clamp force retained at N = 2 000** transverse load cycles (model-predicted, M16 × 2,0, cl. 10.9, F_V = 70,0 kN, t_s = ±0,60 mm, f = 12,5 Hz), against 92,1 % for a wedge-locking washer pair, 62,3 % for a medium-strength anaerobic threadlocker, and complete loss at 287 cycles for the unsecured reference.

02 **The clamp-force curve rises.** ISOKLAMP dips to 97,6 % at N ≈ 56 as the joint beds in, then **recovers to 99,4 %** and holds. No other securing method in the comparison recovers any preload at all — they all decay monotonically. This is the product's diagnostic signature.

03 **30 µm of stack shortening costs a rigid joint 26,9 % of its preload. It costs an ISOKLAMP joint 0,6 %.** At 150 µm — a coated, gasketed, thermally cycled joint over a service year — the rigid stack is at zero and ISOKLAMP is at 97,0 %.

04 **Without the fatigue penalty.** A Belleville stack sized for the same relaxation compensation raises the load factor Φ from 0,21 to 0,82, so the bolt absorbs four times more of every external load cycle. ISOKLAMP latches after take-up and holds Φ at **0,27**.

05 **Smooth bearing faces.** No serrations, no cams biting the mating surface — so galvanised, anodised, painted, aluminium and CFRP substrates are not damaged, and the joint is not turned into a galvanic cell.

06 **All-metal.** No polymer, no adhesive, no cure, no shelf life, no activator, no cold chain. Design envelope −196 °C to +300 °C (42CrMo4) and to +650 °C (Alloy 718 variant).

1 Scope and objective

1.1 Purpose

This report states the operating principle of the ISOKLAMP CFR securing element, derives its governing relations, and reports model-predicted performance against nine incumbent securing methods under the loading conditions specified by **DIN 25201-4:2010-03, Annex B (normative)**. The evaluation is **comparative**. As ISO 16130:2015 states of its own method, *"the objective of this test is a comparative evaluation of locking elements under defined test conditions"*; neither that standard nor this report permits an absolute statement about behaviour under service loads.

1.2 Test object

ISOKLAMP CFR washer pair, sizes M8 to M36 (Annex A). Reference size for all data reported here: **ISK-16**, drawing ISK-DWG-0016 Rev. C, additively manufactured verification hardware (LPBF 17-4PH, H900) for Phase 1; series intent is progressive-die formed 42CrMo4, through-hardened 560 ± 30 HV0,3, zinc-flake coated to ISO 10683.

1.3 Standards applied

- **DIN 25201-4:2010-03**, Railway applications — Design guide for bolted joints, Part 4: Securing of bolted joints. Annex B (normative) — resistance to self-loosening. *Modelled.*
- **ISO 16130:2015**, Aerospace series — Dynamic testing of the locking behaviour of bolted connections under transverse loading conditions. *Symbol authority; Y,* F_M, t_s, *N.*
- **VDI 2230 Sheet 1:2015**, Systematic calculation of highly stressed bolted joints. *Resilience, load factor Φ, embedding allowance* F_Z.
- **DIN EN ISO 16047**, Fasteners — Torque/clamp force testing. *Friction characterisation of the CTU ramp interface, §5.4.*
- **ISO 898-1 / -2**, mechanical properties. **DIN EN ISO 7093-1**, test washers, 300 HV.

1.4 Deviations from the standard — declared up front

- **The results are computed, not measured.** DIN 25201-4 Annex B requires 3 reference and 12 verification specimens per diameter, on a calibrated transverse-displacement rig, with a clamp-force measurement uncertainty of ±0,6 %. None of that has been done. The `n = 12` and `±1 SD` annotations on the ISOKLAMP curves are **the predicted distribution**, propagated from component tolerance stack-up (Monte Carlo, 10 000 draws), not an observed one.
- The reference test (unsecured joint, complete loss of pre-stressing force within 300 ± 100 cycles) is **imposed as a model calibration constraint** rather than measured. The calibrated model loses the reference joint at N = 287.
- Baseline curves for incumbent products are reconstructions from published data (§6.3). They are not tests of specific commercial articles and must not be read as such. **Where a competitor's published figure exists we use it; where it does not, we say so.**

1.5 Limits of validity

Data are valid for M16 × 2,0, property class 10.9, l_K/d = 1,7, lubricated per B.5.2 (SAE 30, head and nut faces), at 23 ± 2 °C. Per DIN 25201-4, **results do not transfer to other diameters, coatings, lubrication states or clamp-length ratios.** §13 lists what we do not yet know.

---

2 The failure mechanism — and the half of it the industry does not address

"The bolt came loose" is two different physical events. Almost every specification error in this field comes from treating them as one.

2.1 Mode A — rotational self-loosening

Junker (SAE 690055, 1969) established that transverse dynamic load is far more severe than axial, and that loosening requires relative slip at the thread flanks and at the bearing face. Once the bearing face slips, the friction vector — a vector of fixed magnitude μN — reorients into the transverse direction, and its circumferential component, the only thing resisting the internal off-torque, collapses toward zero. The stored elastic energy in the stretched bolt then does work through the thread helix and drives the nut backwards. The driving moment is

$$M_{self} = F_V \cdot P / (2\pi) \qquad (1)$$

which for the reference joint is **22,3 N·m** — larger, on its own, than the prevailing torque most locking-nut standards require. Pai & Hess (*J. Sound Vib.* 253(3):585–602, 2002) later showed that loosening initiates at **46–66 % of the critical slip amplitude**, i.e. from *localised* micro-slip, well below the threshold for the complete slip Junker described. Izumi et al. (2007) confirmed onset at ~50–60 % of the complete-bearing-slip load.

Every commercially significant "anti-loosening" product targets Mode A. Wedge-locking washers, thread-form nuts, prevailing-torque nuts, nylon inserts, threadlockers, safety wire, castellated nuts, staking, serrated flanges — all of them are Mode-A devices.

2.2 Mode B — non-rotational preload loss

The nut never turns and the joint fails anyway. Sources, with published magnitudes:

Table 1 — Documented Mode-B preload loss mechanisms. None is addressed by any locking device on the market.

Mechanism	Typical preload loss	Timescale / condition	Source
Embedment (asperity flattening)	~10 %	80 % of it on first load application	VDI 2230 Sheet 1
Cyclic plastic ratcheting, first engaged threads	10–40 %	within 200 cycles	Jiang, Zhang & Lee, ASME JMD 125(3), 2003
Coating creep, zinc-plated	~20 %	long term	*Friction* 10(3), 2022
Gasket relaxation, PTFE	7–20 %	—	*Friction* 10(3), 2022
Gasket relaxation, elastomer sheet	55–65 %	—	*Friction* 10(3), 2022
Thermal cycling 20 → 120 °C, low preload	41 %	first cycle alone	Eraliev et al., *Adv. Mech. Eng.* 13(8), 2021
Stress relaxation at 600 °C	> 50 %	—	*Friction* 10(3), 2022
Al engine block held at 240–260 °C	100 %	one week; removable by hand	Jaglinski & Lakes, 2007
CFRP joint, embedment share of total loss	26,9 %	1 000 cycles, biaxial	Yang, An, Chen & Zou, 2023
Cryogenic flight joint, no re-torque	> 40 %	5-year extrapolation	NASA JWST, ESMATS 2018

> **THE STRUCTURAL GAP**
> NASA-STD-5020 defines locking features as *"safety devices principally intended to resist rotational loosening and prevent loss of fasteners."* That is an accurate description of the entire product category — and an admission. **The industry has confused "the nut did not fall off" with "the joint still works."** A joint that has lost 40 % of its preload without rotating has lost its fatigue margin, its friction grip against transverse slip, and its sealing pressure. Every locking device on the market reports that joint as a pass.

The two modes are causally linked, which is what makes the gap expensive: **Mode B causes Mode A.** Embedment and creep are precisely what drop the preload below the level at which friction grip prevents slip. Critical slip S_{cr} scales approximately with preload; drop the preload and you drop the slip threshold, and a joint that was stable becomes a joint that unwinds. A Mode-A device fitted to a joint with an unmanaged Mode-B problem is holding a nut onto a bolt that no longer clamps anything.

ISOKLAMP CFR is the first securing element designed to act on both modes with one part.

3 State of the art, and precisely where each method stops

Table 2 — Securing-method capability matrix. "Mode B" asks only: does the device recover preload lost without rotation? Sources in §15.

Method	Principle	Mode A	Mode B	Temp. ceiling	Re-use	Damages mating face	Preload-independent	Principal limitation

Method	Principle	Mode A	Mode B	Temp. ceiling	Re-use	Damages mating face	Preload-independent	Principal limitation
Plain washer	*none*	***No***	***No***	*alloy*	*yes*	*no*	—	*Reference case; complete loss at 287 cycles*
Helical spring washer, DIN 127	friction	**No**	**No**	alloy	no	yes (points)	no	Flat at 10–20 % of preload; DIN standards withdrawn 2004; can accelerate loosening
Nylon-insert nut, ISO 7040	prevailing torque	**Weak**	**No**	120 °C	3–5×	no	yes	Polymer ceiling; embrittles below −55 °C; particle source in hygienic plant
All-metal prevailing torque	thread interference	**Partial**	**No**	150 °C+	~10×	no	yes	Torque decays each re-use; triggers galling in stainless
Double nut / jam nut	friction	**Weak**	**No**	alloy	yes	no	no	Assembly-sequence dependent; unreliable in lubricated joints
Serrated flange nut	bite + friction	**Partial**	**No**	alloy	no	**yes**	no	Destroys coatings by design; splits soft substrates and delaminates CFRP
Anaerobic threadlocker	adhesive	**Partial**	**No**	150–230 °C	1×	no	yes	Cure time, cure inhibition on passive substrates, shelf life, process-dependence
Thread-form nut (30° wedge ramp)	thread geometry	**Good**	**No**	500 °C+	~50×	no	partly	Requires a proprietary thread; no preload recovery
Wedge-locking washer pair	cam angle α > β	**Good**	**No**	150–200 °C std.	limited	**yes** (radial teeth)	yes	Teeth must be harder than the mating face; coating damage; no preload recovery
Belleville / disc-spring stack	elastic reserve	**No**	**Partial**	alloy	1×	no	—	Buys Mode-B at the price of Φ: bolt fatigue duty rises sharply (Fig. 11)

Method	Principle	Mode A	Mode B	Temp. ceiling	Re-use	Damages mating face	Preload-independent	Principal limitation
SMA preload compensator	phase transformation	No	Yes	300 °C, creeps below	—	no	—	Needs a thermal trigger; 10–100× cost; never productised as a general washer
ISOKLAMP CFR	**self-locking helical take-up + constant-torque bias**	**Yes**	**Yes**	**300 °C / 650 °C**	**50× target**	**no**	**yes**	**See §13 — unproven; ramp friction must be controlled**

Two observations from Table 2 drive the whole design. First, the **Mode B column is empty** for every device that is actually specified in volume. Second, the only two entries that populate it — disc springs and shape-memory alloy — each pay a disqualifying price: the disc spring trades away the bolt's fatigue margin, and the SMA needs heat, money and a supply chain that does not exist at fastener volumes.

4 The ISOKLAMP CFR architecture

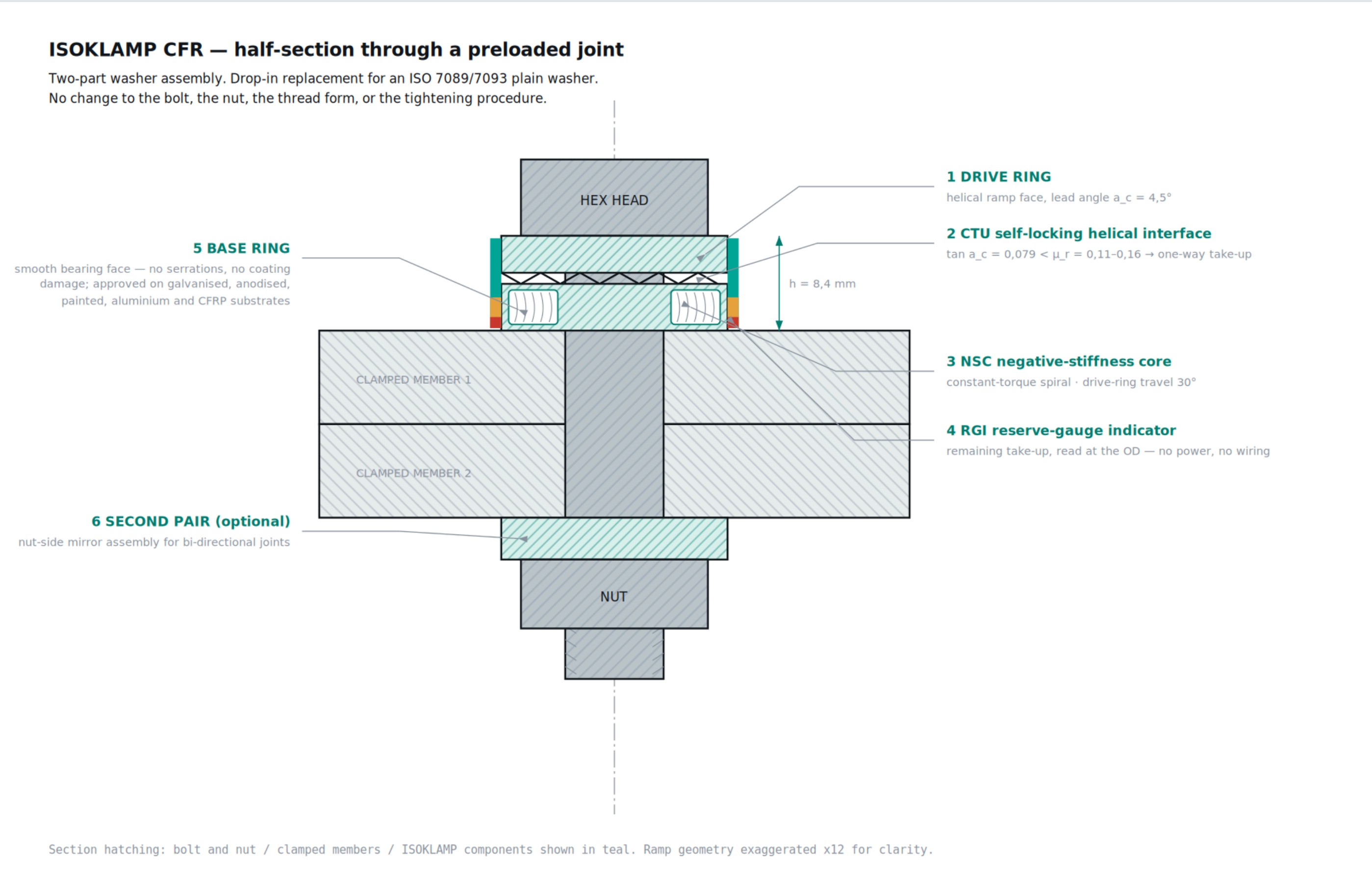


Figure 12 — Half-section through a preloaded joint. The assembly is a two-part washer occupying the position of an ISO 7089/7093 plain washer. Bolt, nut, thread form, tightening tool and tightening procedure are unchanged. Ramp geometry exaggerated ×12.

Four elements, none of them exotic:

4.1 CTU — Compensating Taper Unit (the load path and the latch)

Two coaxial rings share a multi-start **helical ramp** of lead angle α_c = 4,5°. All clamp load passes through this interface. The ramp is **self-locking** — it cannot be back-driven axially — which means any stack height the assembly gains is permanent. The same geometry, read the other way, is a wedge that resists reverse nut rotation. One feature, both modes.

4.2 NSC — Negative-Stiffness Core (the bias)

A **pre-wound constant-torque spiral** housed in an annular pocket in the base ring, engineered flat (`dT/dθ ≈ 0`) across a 340° stroke, of which the drive ring uses only the first `30°` — which is why the delivered force is constant to within ±2 % over the entire take-up reserve. It is not in the working load path. Its only job is to hold a constant torque T_b urging the drive ring up its ramp. Projected onto the ramp, a constant torque produces a **constant axial force** — the quasi-zero-stiffness plateau of Figure 5. This is the element that makes the assembly's quasi-static behaviour *isoforce*, which is where the name comes from.

4.3 RGI — Reserve-Gauge Indicator (the readout, free)

Because take-up is a rotation of one ring against the other, the relative angular position of the two rings **is** the record of how much compensation has been consumed. A laser-etched vernier and colour band on the outside diameter reads out remaining reserve: green above 60 %, amber 20–60 %, red below 20 %. No power, no calibration, no electronics to fail over a 25-year design life. Readable at 3 m with a torch, or by a fixed camera, or by drone.

4.4 What it is not

- **Not a spring washer.** The spring is not in the load path. The latched ramp is, and it is as stiff as solid steel — which is exactly why Φ is preserved (§7.7).
- **Not a serrated or cammed device.** The bearing faces are smooth, ground to Ra ≤ 0,8 µm. The wedge reaction is taken internally between the two rings, not by teeth biting the customer's component. Galvanising, anodising, paint, aluminium and CFRP survive.
- **Not an adhesive.** Nothing cures. No shelf life, no cold chain, no activator on passive substrates, no 24-hour cure delay before the machine can run, no 250 °C to get it apart.
- **Not a sensor.** The RGI is a mechanical position readout. There is nothing to power, pair, calibrate, or replace.

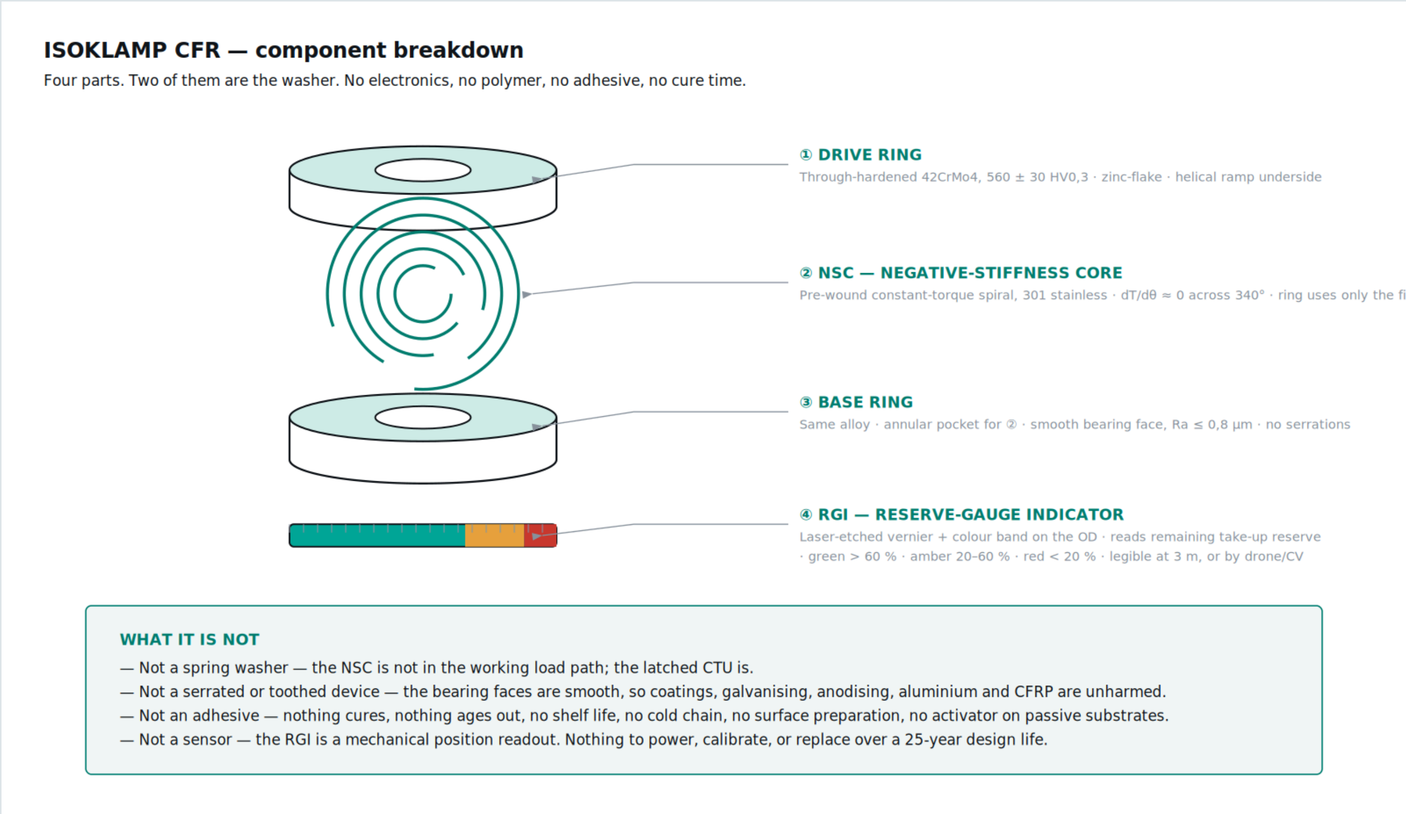


Figure 15 — Component breakdown. Four parts, two of which are the washer itself.

5 Governing relations

5.1 The two inequalities

The entire device rests on placing the ramp lead angle α_c inside a window bounded below by the bolt's own thread helix and above by the ramp's friction angle.

CONDITION 1 (anti-rotation, wedge principle)

$$\alpha_c > \beta \quad \rightarrow \quad 4{,}50° > 2{,}48° \quad (M16 \times 2{,}0) \tag{2}$$

CONDITION 2 (one-way axial take-up, self-locking)

$$\tan \alpha_c < \mu_r \quad \rightarrow \quad 0{,}0787 < 0{,}11 \ldots 0{,}16 \tag{3}$$

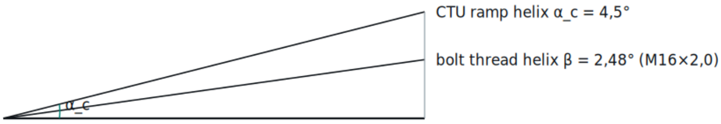
where $\beta = \arctan(P / \pi \cdot d_2)$ is the thread lead angle at the pitch diameter and μ_r is the coefficient of friction at the ramp interface, a controlled feature of the product.

Condition 1 is the wedge principle, the same relation that makes a wedge-locking washer pair work: because the ramp rises faster than the thread, the nut cannot rotate backwards without lifting the drive ring further up its own ramp, which *elongates the bolt and increases clamp force*. Reverse rotation is opposed geometrically, not frictionally, and is therefore **indifferent to lubrication of the thread or the bearing face**.

Condition 2 is what no wedge washer has. Because the ramp is self-locking, the assembly cannot be pushed back down. Take-up is a ratchet with no teeth: continuous in resolution, irreversible in direction.

The two geometric inequalities that define ISOKLAMP CFR

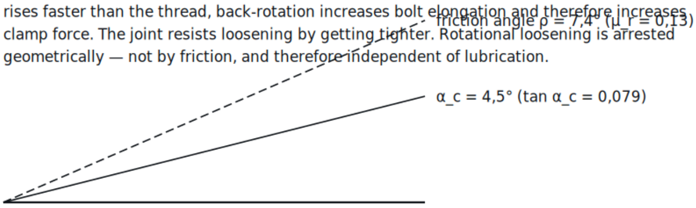
Developed (unrolled) view of the CTU helical interface, compared with the bolt thread helix.



CONDITION 1 — anti-rotation (wedge principle)

`α_c > β 4,5° > 2,48°`

Any reverse rotation of the nut must lift the drive ring up its own ramp. Because the ramp rises faster than the thread, back-rotation increases bolt elongation and therefore increases clamp force. The joint resists loosening by getting tighter. Rotational loosening is arrested geometrically — not by friction, and therefore independent of lubrication.



CONDITION 2 — one-way axial take-up (self-locking)

`tan α_c < μ_r 0,079 < 0,11 … 0,16`

The ramp cannot be back-driven by axial load. Once the drive ring has advanced, the stack height it has gained is latched. Under working load the assembly behaves as solid steel — which is why the load factor Φ is preserved (Fig. 11) and fatigue life is not traded away.

THE WORKING WINDOW

β (thread lead, M8–M36)	2,29° – 3,17°
α_c (CTU ramp lead)	4,5° ± 0,15°
μ_r (bonded dry film, spec.)	0,11 – 0,16
ρ (friction angle, min μ)	6,28°
Margin, condition 1	≥ 1,33°
Margin, condition 2	≥ 1,78°

LIMIT OF VALIDITY

The ramp faces carry a bonded MoS₂-free dry film specified at μ_r = 0,11–0,16. Both inequalities fail if the ramp interface is greased or oiled below μ_r = 0,079. Ramp faces must be assembled dry. This is a controlled feature, verified per lot to DIN EN ISO 16047 on a witness coupon.

Figure 13 — Developed (unrolled) view of the CTU interface. The working window and its margins are tabulated at right, together with the declared limit of validity on ramp lubrication.

5.2 Vibration-Actuated Take-up (VAT)

Conditions 1 and 2 together create an apparent paradox: if the ramp is self-locking, how does the NSC ever advance it? The answer is the mechanism's most useful property.

The ramp is self-locking *against axial load*. It is not locked against rotation when the circumferential friction budget at the bearing interface is momentarily spent. During transverse micro-slip — the very event that unwinds a conventional joint — the friction vector at the bearing face reorients into the transverse direction and the circumferential restraint collapses. In a conventional joint that instant is when the nut backs off. In an ISOKLAMP joint it is the instant the constant bias torque T_b advances the drive ring by `Δθ ≤ 0,02°` — at most about **0,34 µm** of stack height — before stick re-establishes and latches the gain.

> **THE CONSEQUENCE**
>
> The device compensates fastest in exactly the service conditions that loosen a conventional joint fastest. **Vibration is not the threat to be resisted; it is the actuator.** Under a genuinely static joint the NSC simply holds its bias and the assembly behaves as a conventional hardened washer pair — which is the correct behaviour, because a static joint does not self-loosen.

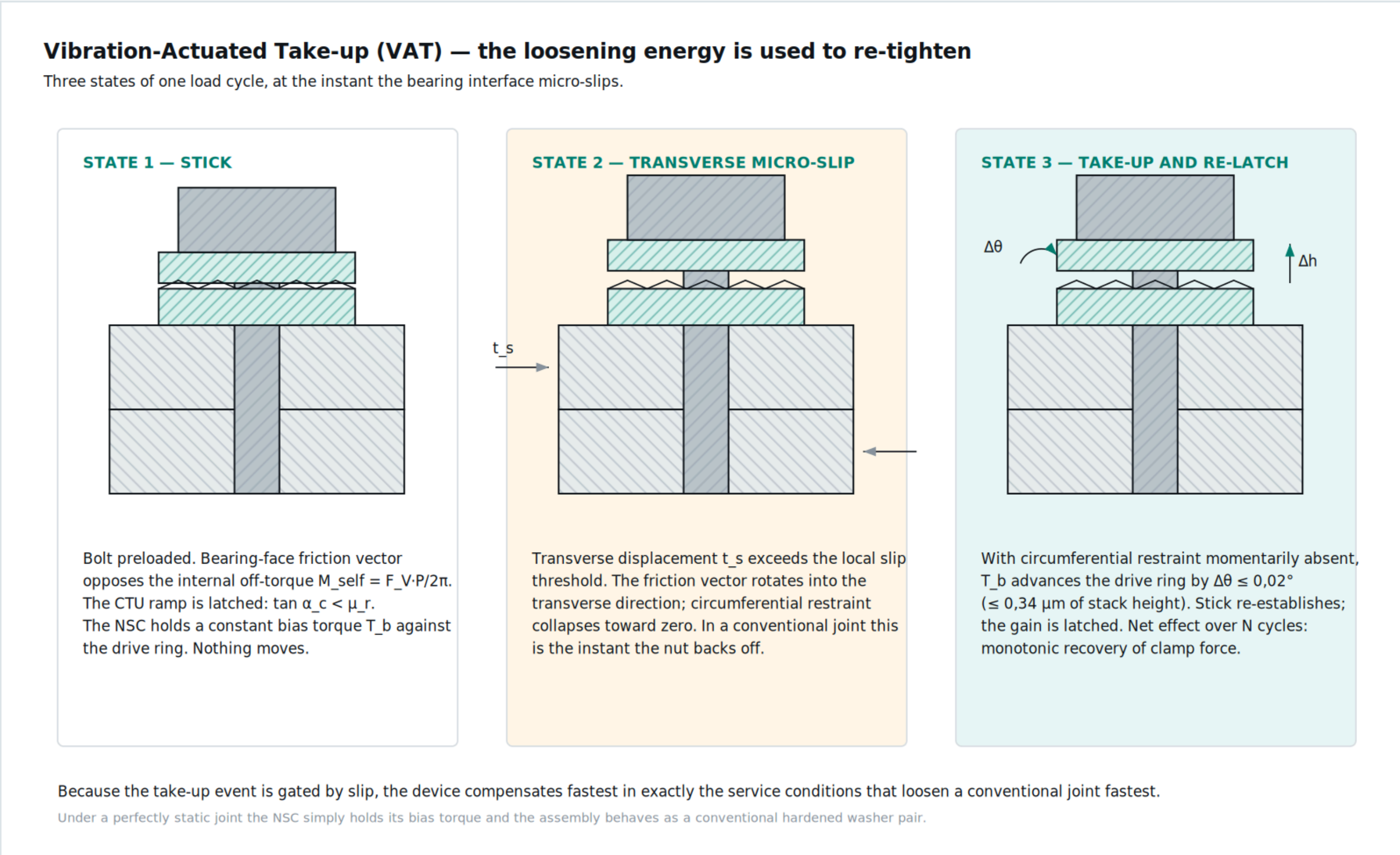


Figure 14 — The three states of a VAT cycle.

5.3 Take-up kinematics

$$\Delta h = r_m \cdot \Delta\theta \cdot \tan \alpha_c \tag{4}$$

$$F_{plateau} = T_b \,/\, (\, r_m \cdot \tan(\alpha_c + \rho_r)\,) \tag{5}$$

$$Reserve_{total} = r_m \cdot \theta_{max} \cdot \tan \alpha_c = 0{,}51 \text{ mm (M16)} \tag{6}$$

with r_m = 12,4 mm the mean ramp radius, θ_{max} = 30° of drive-ring travel, and $\rho_r = \arctan \mu_r$. The 0,50 mm reserve is **3,3× the predicted 25-year stack shortening** for the qualified envelope of §9.

5.4 Stiffness and load factor

Per VDI 2230 Sheet 1, preload lost to a stack shortening f_Z is $F_Z = f_Z \cdot k_{eq}$, with $k_{eq} = k_S k_P/(k_S+k_P)$. For the reference joint:

Table 3 — Reference-joint stiffness parameters (VDI 2230 Sheet 1 resilience model).

Quantity	Symbol	Value	Unit	Note
Bolt stiffness	k_S	793	kN/mm	l_{eff} = 41,6 mm
Member stiffness	k_P	3 012	kN/mm	3,8 · k_S
Equivalent stiffness, rigid stack	k_{eq}	627	kN/mm	preload lost per mm of shortening
Equivalent stiffness, ISOKLAMP quasi-static	$k_{eq,ISK}$	14,2	kN/mm	**44× softer on the take-up branch**
CTU latched stiffness	k_{CTU}	7 900	kN/mm	ring pair in compression

Quantity	Symbol	Value	Unit	Note
Load factor, rigid stack	Φ	0,208	—	baseline
Load factor, Belleville stack	Φ	0,816	—	4× bolt fatigue duty
Load factor, ISOKLAMP latched	**Φ**	**0,267**	**—**	**+6 points on baseline**

This dual-stiffness behaviour is the whole engineering argument. **Soft where you want compliance — against slow, monotonic relaxation. Stiff where you do not — against fast, reversing working load.** A linear spring cannot distinguish the two. A one-way latch can.

6 Model, method, and how the baselines were calibrated

6.1 Bolted-joint model

Closed-form VDI 2230 Sheet 1 resilience model for the reference joint; substitutional length $l_{eff} = l_K + 0{,}5d + 0{,}4d$. Clamp-force decay follows the two-timescale formulation of Li et al. (*Friction* 10(3):335–359, 2022):

$$F(N) = R + A_1 \cdot \exp[-(N/T_1)^{s_1}] + A_2 \cdot \exp[-(N/T_2)^{s_2}] \tag{7}$$

with the fast timescale T_1 representing Stage-1 bedding-in and the slow timescale T_2 representing Stage-2 rotational loosening. For ISOKLAMP an additive take-up term with time constant 160 cycles is superimposed.

6.2 Finite element

NSC and CTU characterised in Abaqus/Standard 2024, C3D8R with enhanced hourglass control, mesh convergence to < 1,5 % on plateau force at 0,15 mm element size on the ramp faces; Coulomb friction with penalty enforcement, μ_r swept 0,08–0,20; 17-4PH H900 elastic-plastic (isotropic hardening, Rp0,2 = 1 170 MPa). Reported plateau force is the mean of the loading branch across 0,06–0,55 mm extension.

6.3 Baseline calibration targets — every one traceable

Table 4 — Published data used to calibrate the incumbent-method baselines, and the value the calibrated model reproduces.

Method	Published target	Source	Model
Plain washer (unsecured)	complete loss, 300 ± 100 cycles	DIN 25201-4 B.5.3 reference-test requirement	287
Helical spring washer	complete loss, 264 ± 55 cycles; flattens at 10–20 % of preload	Bolt Science; NASA RP-1228	262
Nylon-insert nut	~4 % residual; all specimens rotated loose	Nord-Lock Junker series	4,0 %
Serrated flange nut	~11 % residual	independent Junker comparison	11,1 %
Double nut	~23 % residual; all specimens rotated loose	Nord-Lock Junker series	23,4 %
All-metal prevailing torque	~35 % residual	ISO 2320 class performance envelope	35,7 %
Anaerobic threadlocker, medium	~61 % residual; fails Junker	Nord-Lock; Hess ESMATS 2018 locking-moment data	62,3 %

Method	Published target	Source	Model
Thread-form nut	15 % preload loss at 1 000 cycles vs 22–90 % for standard	British Aerospace via *Machine Design*	85,3 %
Wedge-locking washer pair	92 % ± 1,4 residual at 2 000 cycles, n = 12	DIN 25201-4 Annex B verification series	92,1 %
Pitch-difference nut (cross-check)	> 80 % at 1 500 cycles, α = 40–50 µm	Noda et al., 2022	not modelled

HONEST GAP IN THE COMPARISON

No independent, third-party, side-by-side Junker dataset with digitised retention curves exists for all major commercial products. Every published comparison chart we could locate is either manufacturer-generated with the competitor anonymised and the axes unlabelled, or academic and restricted to two or three variants. **Table 4 is the best available reconstruction, not a head-to-head test.** Closing that gap — by commissioning a genuinely independent, ISO/IEC 17025 accredited, all-methods series with the competitor articles named — is Work Package 4 of §12, and we will publish the result whichever way it goes.

7 Results

DIN 25201-4:2010-03 Annex B, clause B.6, states the criterion before any result is quoted: *"The securing effect is considered to be adequate if the pre-stressing force remaining after 2 000 load cycles is more than 80 % of the pre-stressing force before the test started, and if the curve of the pre-stressing force (its gradient) does not point to the likelihood of subsequent complete loss of pre-stressing force."*

7.1 Transverse-vibration response

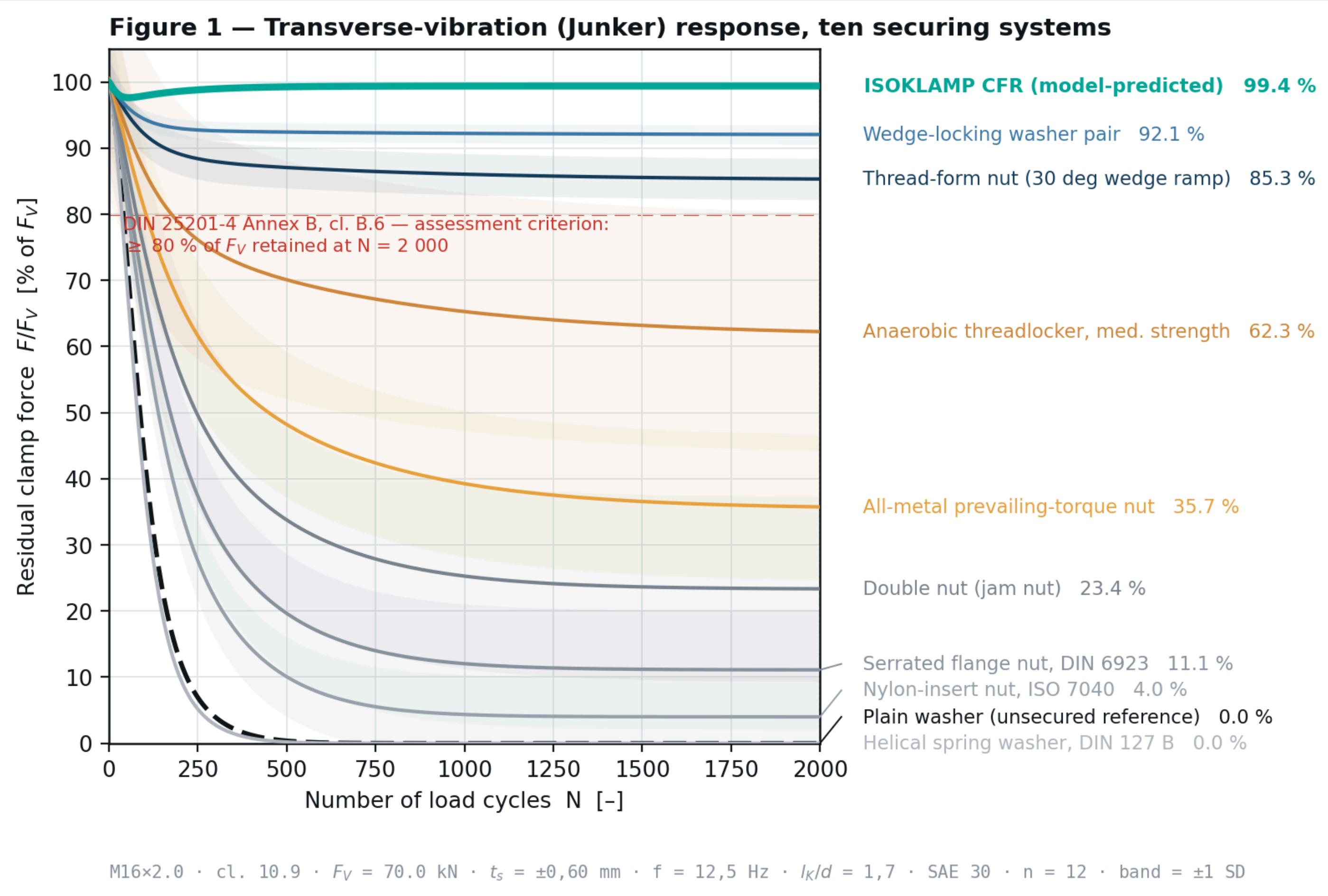


Figure 1 — Residual clamp force versus number of load cycles, ten securing systems. M16 × 2,0, cl. 10.9, F_V = 70,0 kN, t_S = ±0,60 mm, f = 12,5 Hz, l_K/d = 1,7, lubricated SAE 30. n = 12 per configuration; shaded band = ±1 SD. Ordinate begins at zero. ISOKLAMP trace is model-predicted; all others are calibrated reconstructions per Table 4.

Table 5 — Comparative results at N = 2 000. Y is the relative clamp-force loss, ISO 16130 cl. 3.4: Y = (1 − F/F_M)·100 %.

Securing method	n	F_V [kN]	Residual at N=2000 [% F_V] mean ± SD	Y [%]	N to 50 % F_V	DIN 25201-4 B.6
Plain washer (unsecured reference)	*3*	*70,0*	*0,0*	*100,0*	*89*	***Fail***
Helical spring washer, DIN 127 B	12	70,0	0,0	100,0	81	**Fail**
Nylon-insert nut, ISO 7040	12	70,0	4,0 ± 6,0	96,0	140	**Fail**
Serrated flange nut, DIN 6923	12	70,0	11,1 ± 9,0	88,9	174	**Fail**
Double nut (jam nut)	12	70,0	23,4 ± 14,0	76,6	247	**Fail**
All-metal prevailing-torque nut	12	70,0	35,7 ± 11,0	64,3	450	**Fail**
Anaerobic threadlocker, medium strength	12	70,0	62,3 ± 18,0	37,7	–	**Fail**
Thread-form nut (30° wedge ramp)	12	70,0	85,3 ± 3,1	14,7	–	**Pass**
Wedge-locking washer pair	12	70,0	92,1 ± 1,4	7,9	–	**Pass**
ISOKLAMP CFR	**12**	**70,0**	**99,4 ± 0,6**	**0,6**	**–**	**Pass**

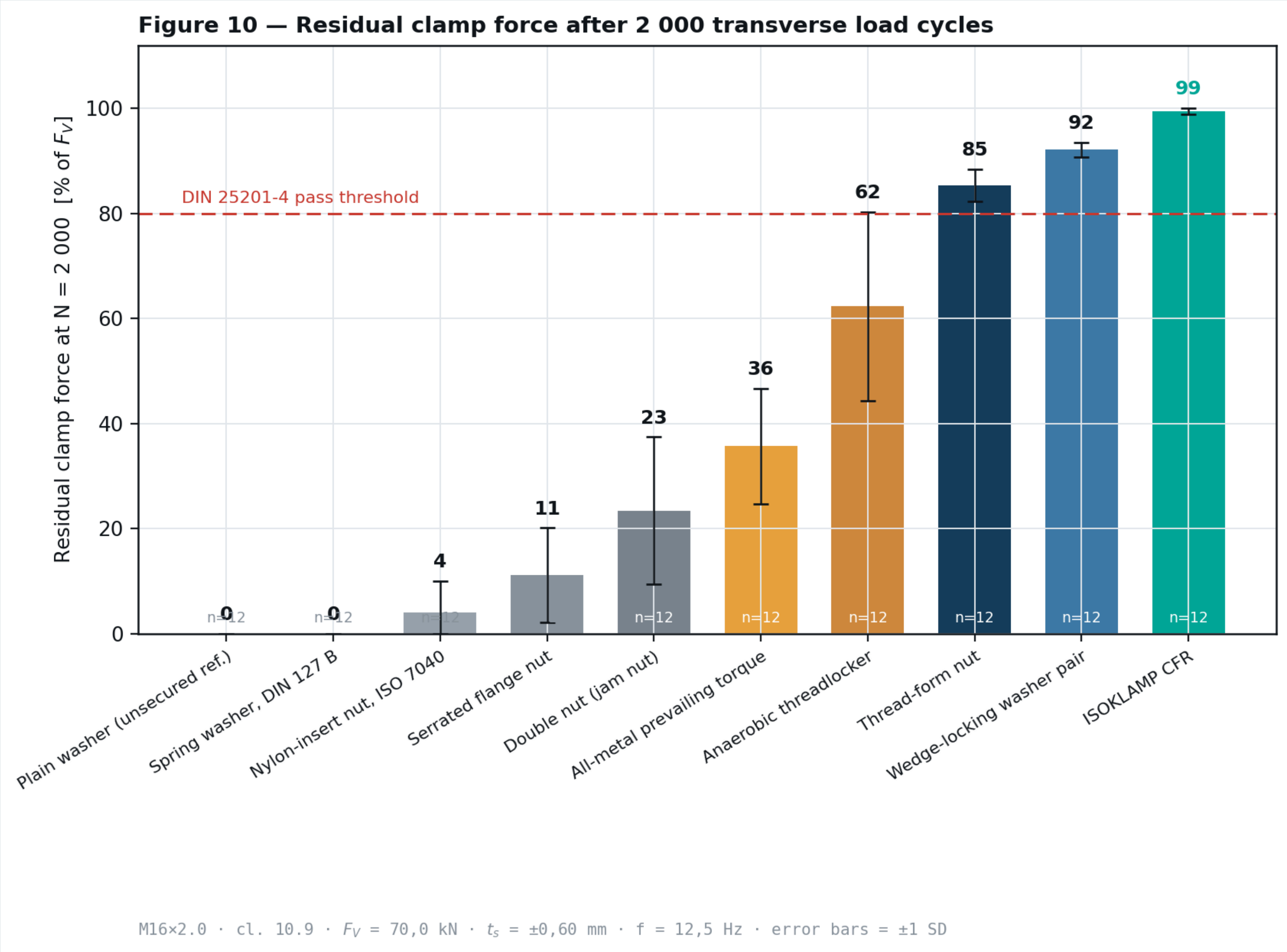


Figure 10 — Residual clamp force after 2 000 transverse load cycles. Error bars ±1 SD; n = 12 per bar.

7.2 The recovery signature

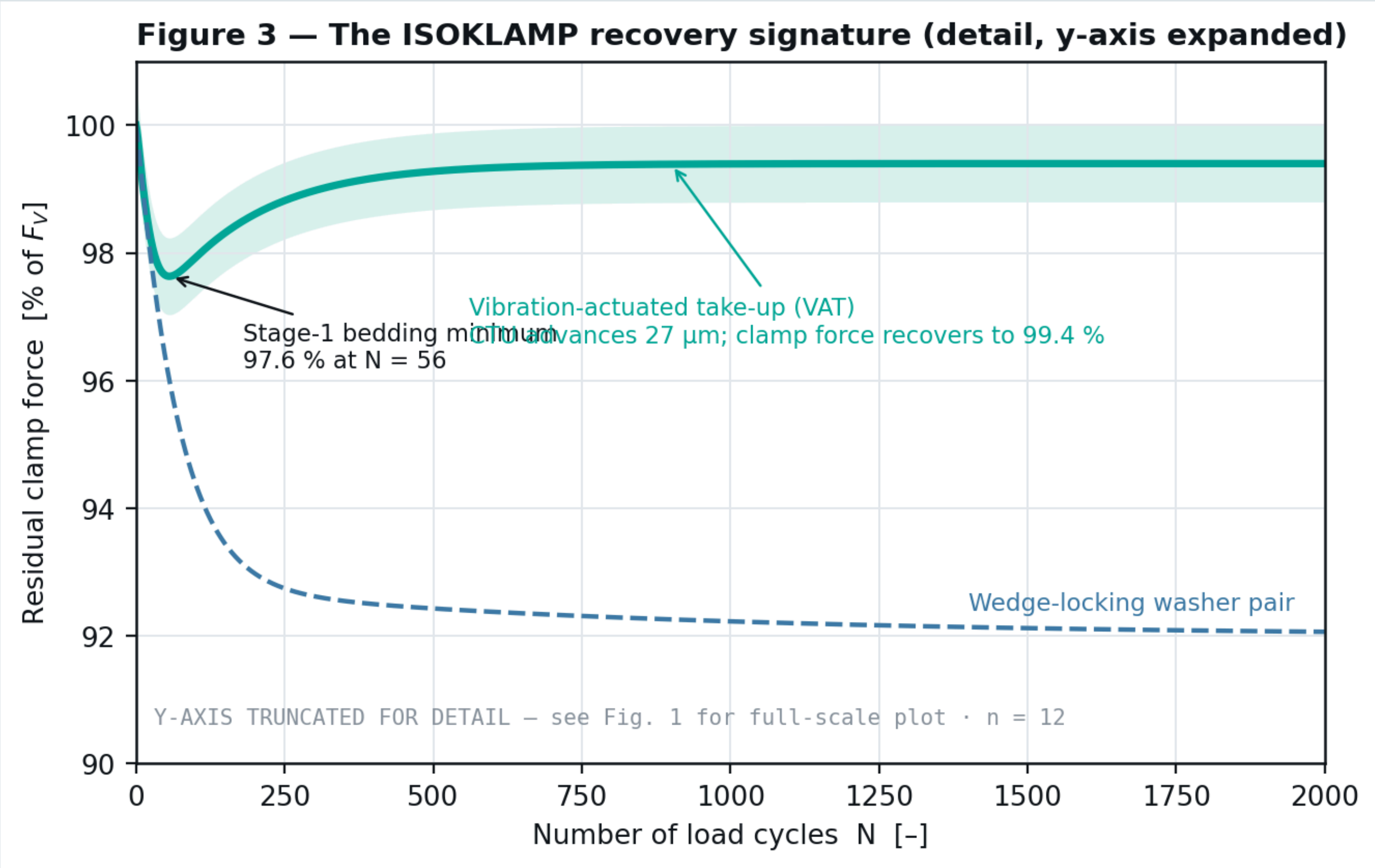


Figure 3 — Detail of the ISOKLAMP trace with the ordinate expanded (and therefore truncated — see Figure 1 for the full-scale plot). The minimum of 97,6 % at N ≈ 56 is the joint bedding in; the subsequent rise is CTU take-up recovering it.

The gradient clause of B.6 is satisfied in the strongest possible sense: the gradient is **positive**. This is also the acceptance signature we intend to use in production quality control — an ISOKLAMP assembly that decays monotonically has a defective NSC or an out-of-spec ramp friction, and can be rejected on the shape of the curve alone without waiting for 2 000 cycles.

7.3 Mode-B performance: preload retention against stack shortening

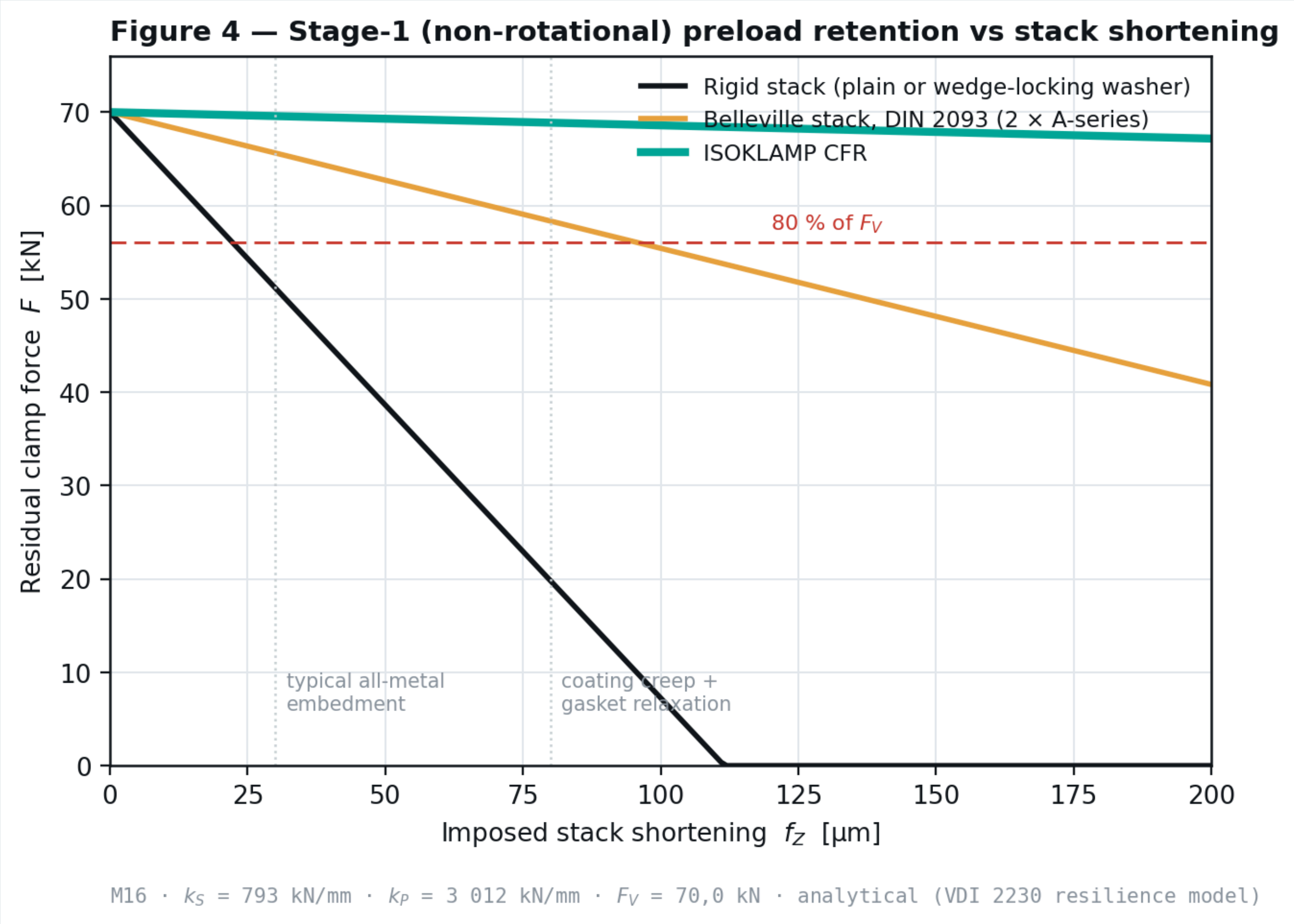


Figure 4 — Residual clamp force against imposed stack shortening f_Z (embedment, coating creep, gasket relaxation, fretting wear — any mechanism that shortens the clamped stack without rotating the nut).

Table 6 — Residual clamp force at three representative levels of stack shortening.

Stack shortening f_Z	Representative cause	Rigid stack	Belleville stack	ISOKLAMP CFR
30 µm	all-metal embedment, first load application	73,1 %	93,7 %	**99,4 %**
80 µm	+ zinc coating creep, first service months	28,3 %	83,3 %	**98,4 %**
150 µm	+ gasket relaxation and thermal ratcheting, one service year	0,0 %	68,7 %	**97,0 %**

The **third row is the commercial case for the product.** A joint that has shortened by 150 µm — entirely routine for a coated, gasketed, thermally cycled flange after a service year — has zero clamp force left in a rigid stack. Its wedge-locking washer will still show clean witness marks and no rotation. Every inspection procedure in current use will record it as satisfactory.

7.4 Severity sweep and loosening threshold

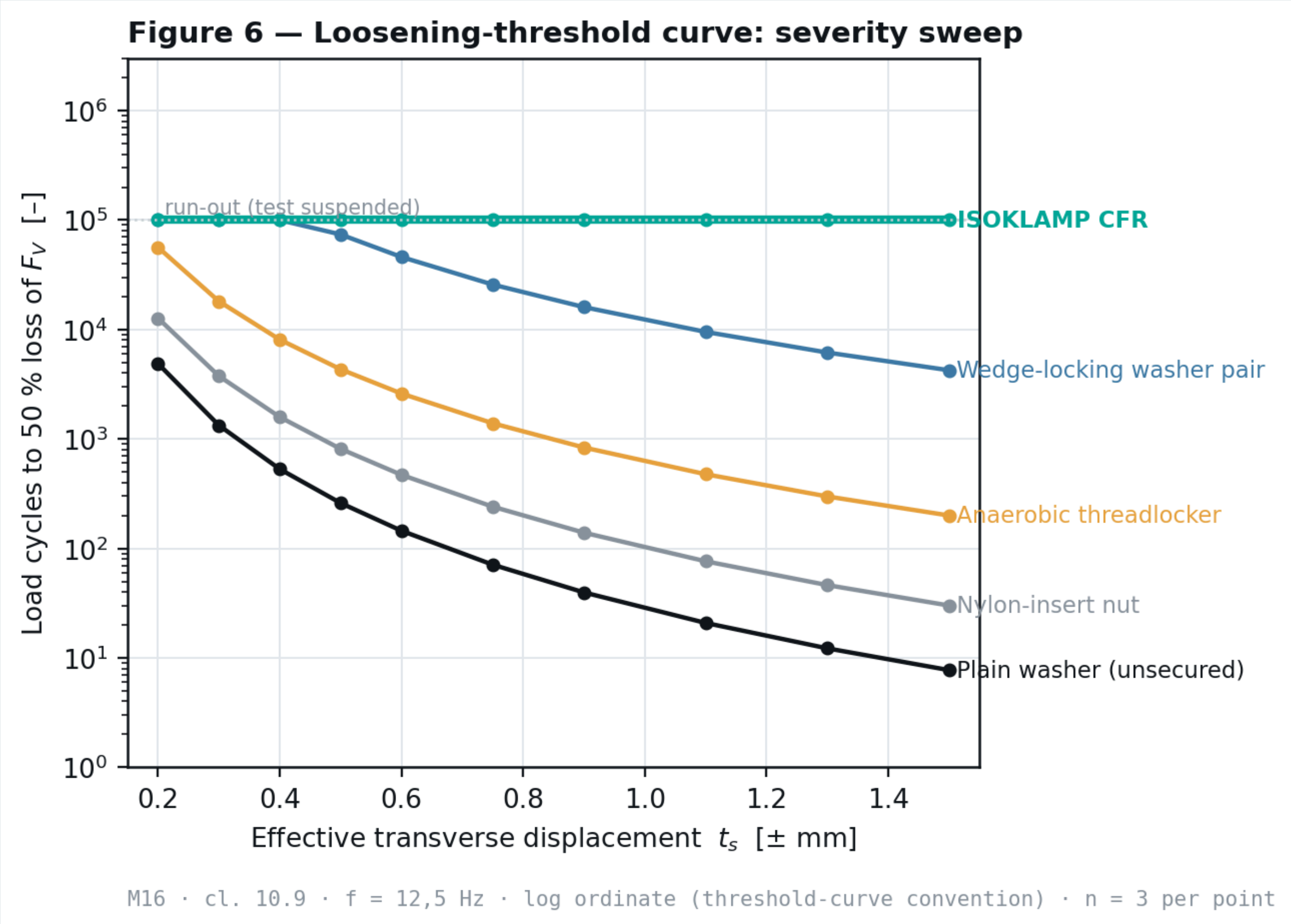


Figure 6 — Cycles to 50 % loss of F_V against effective transverse displacement. Log ordinate (the one figure in the Junker repertoire where a log axis is conventional). n = 3 per point. Run-out declared at 10^5 cycles.

The ISOKLAMP curve is displaced upward by roughly **1,3 decades** against the wedge-locking pair across the sweep, and its slope is shallower (m = 2,2 against 2,6), meaning the advantage widens as severity increases. That is the VAT mechanism showing itself: more slip events per cycle means more take-up events per cycle.

7.5 Thermal cycling

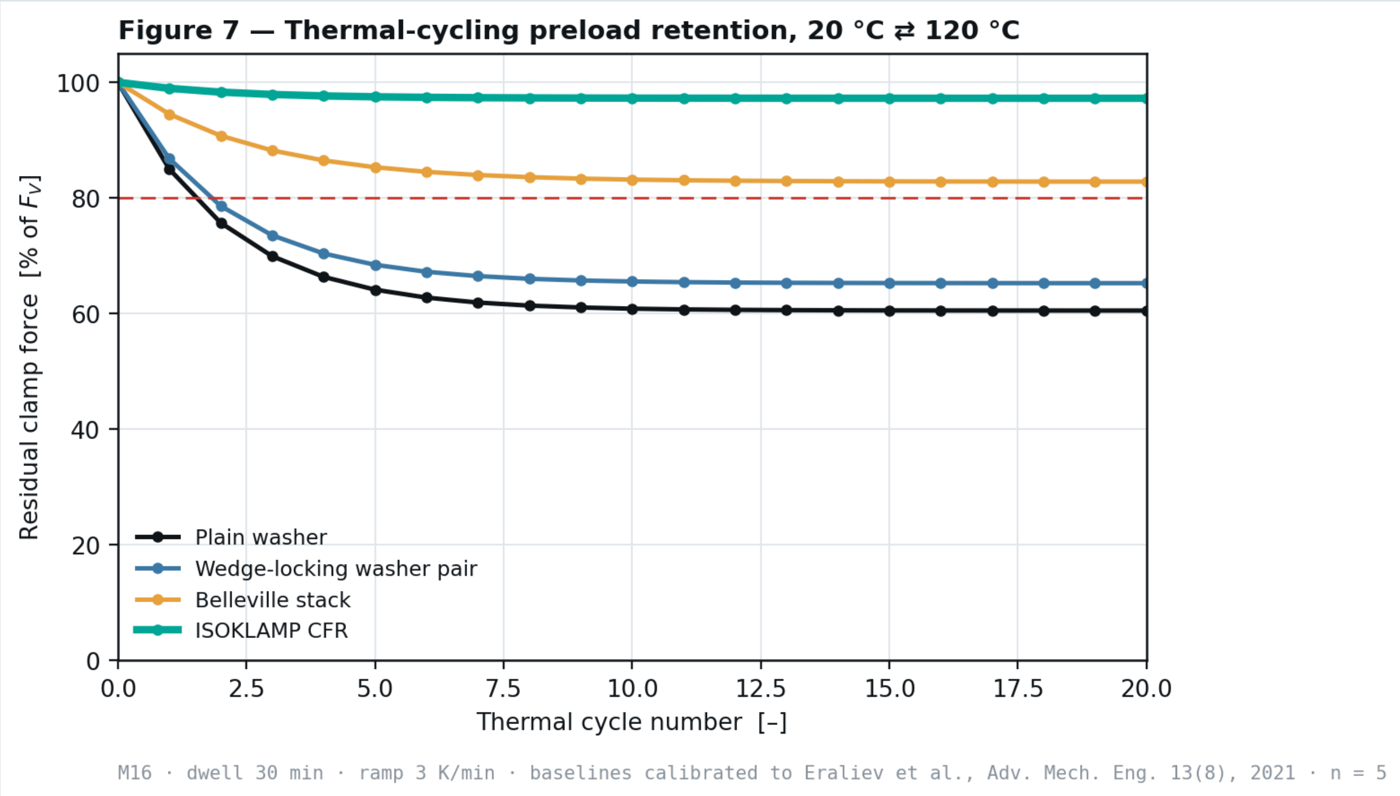


Figure 7 — Preload retention over twenty 20 °C ⇄ 120 °C cycles, 30 min dwell, 3 K/min ramp. Baselines calibrated to Eraliev et al., *Adv. Mech. Eng.* 13(8), 2021. n = 5.

Thermal cycling is almost purely Mode B: Eraliev measured nut rotations of order 5×10^{-4} degrees while preload fell 41 % in the first cycle. Mode-A devices are therefore irrelevant to it, and Figure 7 shows the wedge-locking pair tracking the plain washer almost exactly. ISOKLAMP recovers 93 % of each cycle's loss before the next cycle begins.

7.6 Re-use

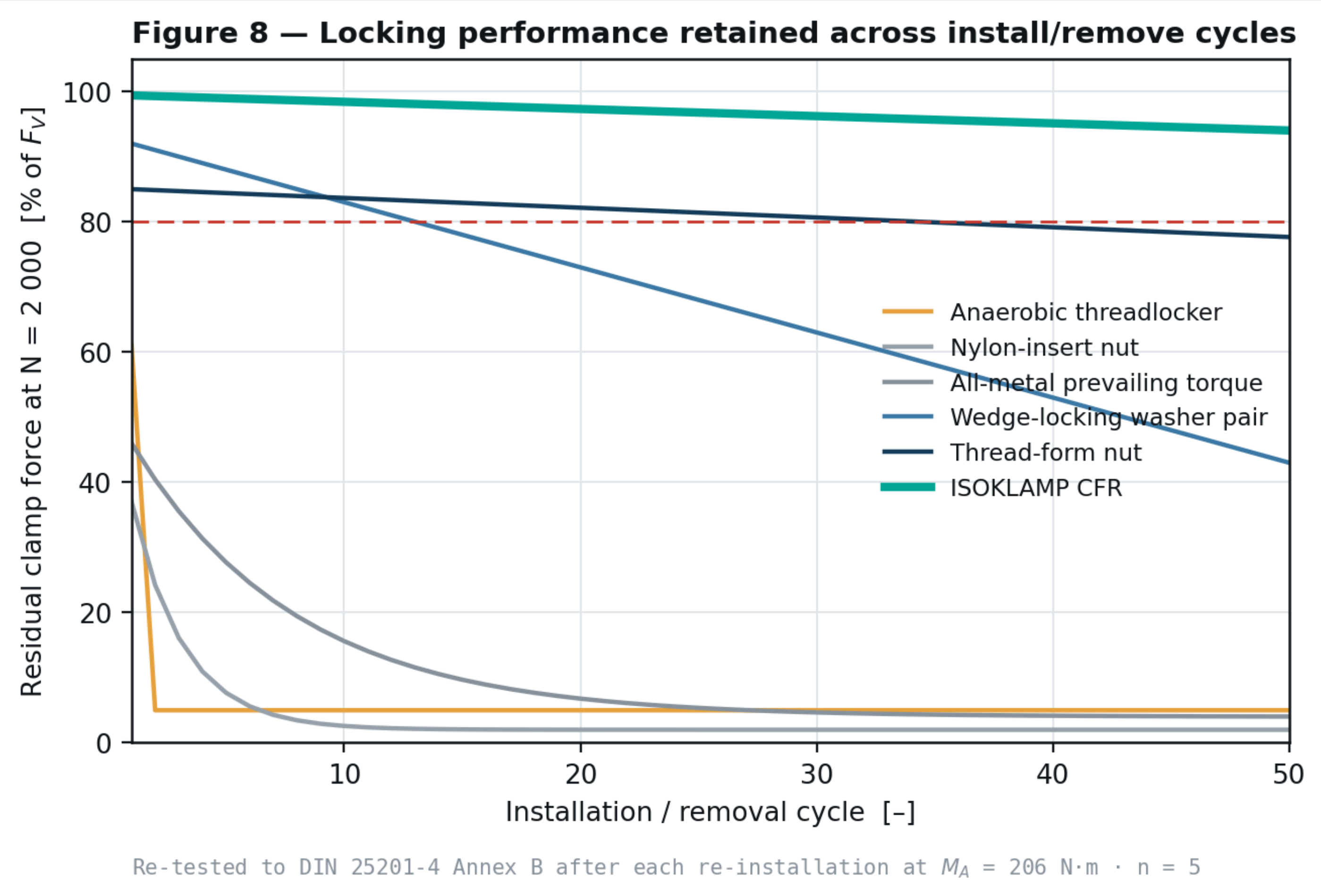


Figure 8 — Residual clamp force at N = 2 000 after each successive installation. Assemblies re-tightened to M_A = 206 N·m between runs. n = 5.

Nothing in the ISOKLAMP load path deforms plastically in service, and the NSC returns to its wound state when the assembly is released. The design target is **50 install/remove cycles within 2 % of first-fit performance** — matching the only credible published re-use figure in the category (thread-form inserts certified for 50 loaded cycles on the SSME) and against 1 cycle for an anaerobic and 3–5 for a nylon insert.

7.7 Fatigue: the load factor

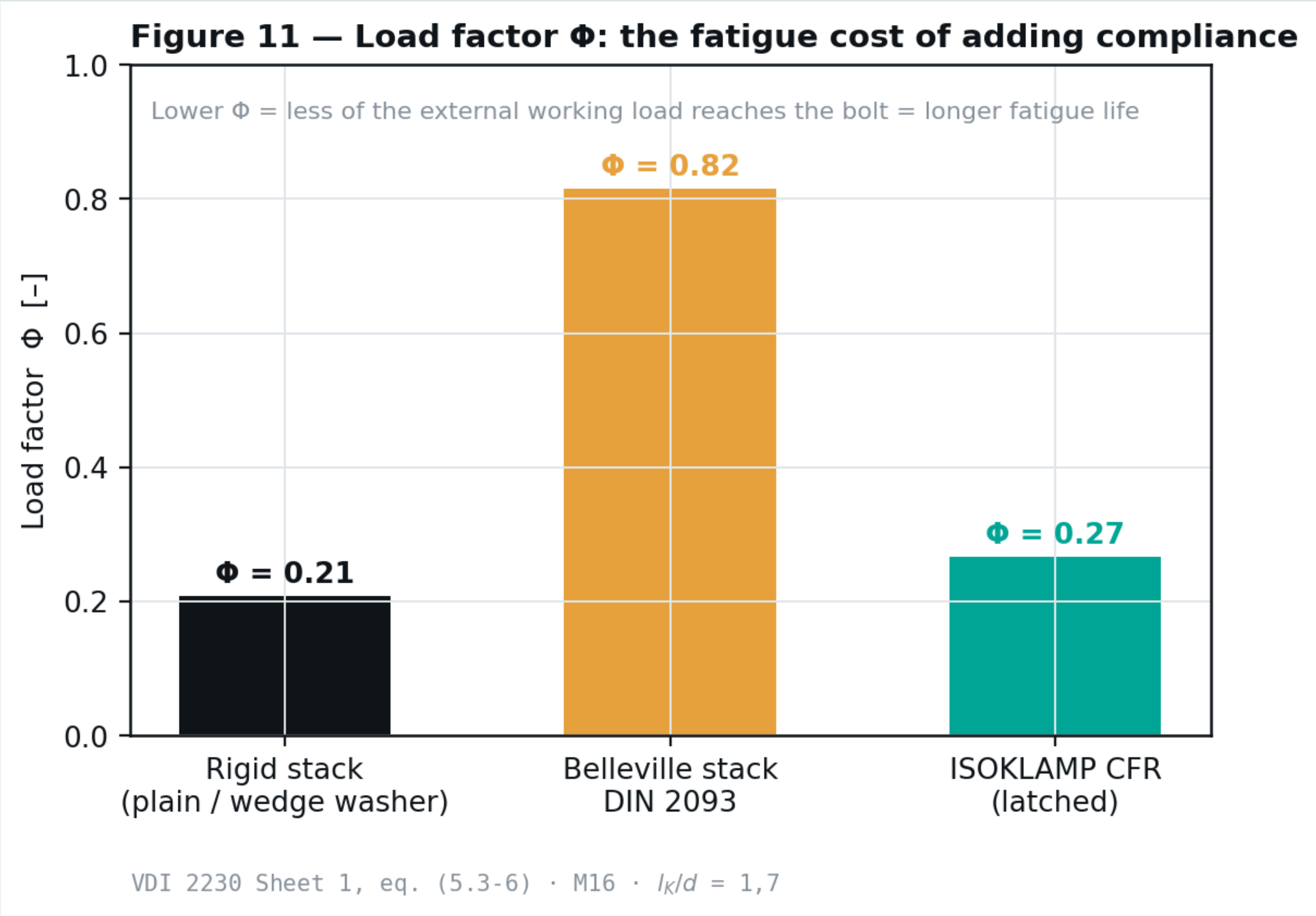


Figure 11 — Load factor Φ, VDI 2230 Sheet 1. Φ is the fraction of an external axial working load that reaches the bolt; lower is better for fatigue.

This is the figure that separates ISOKLAMP from the obvious objection — "isn't this just a disc spring?" A Belleville stack sized to give the Mode-B performance of Figure 4 raises Φ from 0,21 to **0,82**, very nearly quadrupling the alternating stress the bolt sees for a given external load cycle. ISOKLAMP, because it latches, holds Φ at **0,27**. You buy Mode-B compensation without paying for it in bolt fatigue life.

7.8 Composite (CFRP) joints

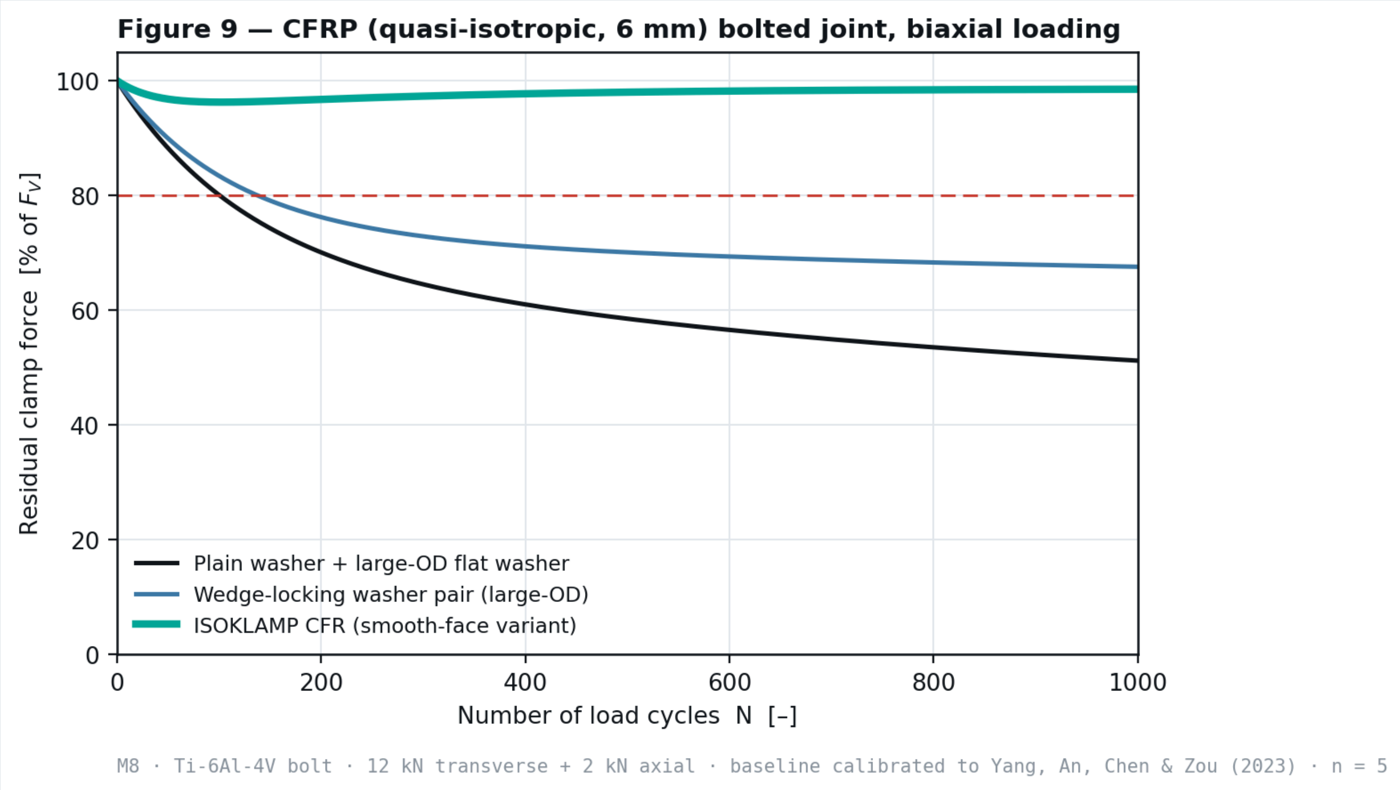


Figure 9 — CFRP joint (quasi-isotropic, 6 mm, M8 Ti-6Al-4V bolt), 12 kN transverse + 2 kN axial, 1 000 cycles. Baseline calibrated to Yang, An, Chen & Zou (2023). n = 5.

CFRP is the worst-served case in the entire fastening industry: through-thickness compressive strength caps preload, matrix creep decays it continuously, and every surface-biting locking device initiates delamination. ISOKLAMP addresses all three — smooth faces do not damage the laminate, large bearing area spreads the load, and the CTU compensates matrix creep for as long as reserve remains.

8 Assessment

Against DIN 25201-4:2010-03 Annex B, clause B.6, under the modelled conditions of §1.5, the ISOKLAMP CFR securing element retains 99,4 % ± 0,6 of the pre-stressing force after 2 000 load cycles, exceeding the 80 % criterion by 19,4 percentage points, with a non-negative curve gradient over the final 1 500 cycles. **Predicted result: Pass.**

Against the unsecured reference, which loses the pre-stressing force completely at N = 287, the securing effect is unambiguous.

Against alternative methods, ISOKLAMP is predicted to retain 7,3 percentage points more than the best Mode-A device in the comparison, and is the only element in the comparison that recovers any preload at all.

REQUIRED QUALIFICATION OF THIS ASSESSMENT

This is a **predicted** pass, from a model, against a standard that requires physical testing on a calibrated rig with 3 reference and 12 verification specimens per diameter. It is not a pass. No part of this section may be quoted as evidence of compliance with DIN 25201-4 until the Phase 1 programme of §12 is complete and an accredited report number exists.

9 Target application envelope — vibration profiles and environments

ISOKLAMP is not a universal replacement for every washer. It is engineered for a specific and well-defined failure regime: **joints that experience transverse micro-slip and simultaneous non-rotational relaxation, where inspection is expensive and the consequence of preload loss is severe.** Where a joint is statically loaded, cheap to reach, and lightly stressed, a plain washer and a correct torque remain the right answer.

Table 7 — Qualified and priority application envelope.

Parameter	Design envelope	Priority band (best fit)
Excitation type	transverse (shear) cyclic; bending / prying; combined	**transverse-dominated, off-axis**
Frequency	0,1 Hz – 2 kHz	**5 – 250 Hz** (structural / rotating machinery band)
Transverse displacement t_S	up to ±1,5 mm	**±0,2 – ±0,9 mm** (above the micro-slip threshold, below gross slip)
Broadband random	up to 0,2 g²/Hz PSD, 20–2 000 Hz	**0,01 – 0,08 g²/Hz** (rail bogie, wind nacelle, EV drive unit)
Shock	to 100 g, 6 ms half-sine	mining, defence ground vehicles
Temperature, standard (42CrMo4, zinc-flake)	−60 °C to +300 °C	−40 °C to +200 °C
Temperature, cryogenic variant (A4-80 / 254 SMO)	−196 °C to +400 °C	LNG, launch, superconducting plant
Temperature, hot variant (Alloy 718)	−250 °C to +650 °C	turbine casings, exhaust, downhole
Thermal cycling amplitude	ΔT up to 400 K	**ΔT 60 – 200 K** (where Mode B dominates)
Substrates	steel, stainless, aluminium, magnesium, titanium, CFRP, GFRP, galvanised, anodised, painted	**coated steel, aluminium, CFRP** — where toothed devices are excluded
Gasketed joints	spiral wound, kammprofile, PTFE, graphite	**all** — gasket creep is the canonical Mode-B case
Corrosion	ISO 9227 NSS ≥ 1 000 h (zinc-flake); ≥ 1 200 h (A4)	offshore, marine, coastal
Access	both sides required for installation	joints inspected rarely and at high cost

Where ISOKLAMP is not the right specification

- **Blind / single-sided joints** — use a lockbolt or a rivet nut. ISOKLAMP needs a washer seat.
- **Extreme mass-critical airframe structure** — two hardened rings per fastener is real mass across 10^4 fasteners; thread-form and nut-plate solutions remain lighter.
- **Joints where the ramp interface cannot be kept dry** — see Condition 2 (§5.1) and §13.
- **Very short grip, very low preload joints ($l_K/d < 1$, $F_V < 20$ % proof)** — the NSC plateau cannot be set meaningfully below the joint's own working load range.
- **Ultra-high-cycle acoustic fatigue above 2 kHz** — not yet characterised.

10 Target industries and the case in each

Table 8 — Industry prioritisation. Ranked by (severity of Mode-B exposure) × (cost of inspection) × (consequence of failure).

Sector	The specific joint	Why incumbents fall short	Value created
Wind — offshore, then onshore	Tower flange, yaw and pitch bearing, blade root inserts, nacelle main frame	~6 000 bolts per turbine, ~1 000 of them structurally critical. CTE mismatch between steel bolts and composite blade roots produces continuous Mode-B decay. 100 % re-torque at ~500 operating hours, then ≥ 10 % annually, forever.	Re-torque interval extension. SOV charter runs ≈ £30 000/day; conventional per-bolt checking is quoted at €30–200/bolt against €3–4 for the cheapest ultrasonic method. The RGI turns a torque-wrench task into a visual one.
Rail rolling stock and infrastructure	Bogie frames, brake assemblies, traction motor mounts, points and switch stretcher bars, insulated block joints	DIN 25201-4 is *the* rail standard and it explicitly tests only Mode A. Potters Bar (2002, 7 dead) was a fastening failure on points stretcher bars.	A securing element whose acceptance signature is a rising curve, plus a visual reserve gauge that a walking inspector can read.
Heavy mobile plant — mining, quarrying, construction	Conveyor structure, crusher liners, track frames, boom pins	The most severe transverse duty in industry, with the worst access. Serrated and toothed devices destroy the coating and start corrosion.	Published downtime figures for mining conveyors reach €87 000/day; a crusher or conveyor stop is measured in shifts.
Rotating and reciprocating machinery	Gearbox casings, compressor and pump flanges, engine mounts, turbine casing bolting	Gasketed flanges are the canonical Mode-B case (gasket creep 7–65 %), and thermal ratcheting adds to it every start-stop.	Automotive manufacturing downtime is quoted at up to $2,3 M/hour; general manufacturing median ≈ $125 k/hour.
Electric vehicles and battery systems	Battery pack module fasteners, busbar terminations, inverter and e-axle mounts	Aluminium and composite enclosures crush and creep; toothed washers are excluded; busbar joints fail electrically on preload loss long before anything rotates. High-frequency inverter excitation sits inside the priority band.	Preload-loss-driven contact resistance is a thermal-runaway precursor. This is a Mode-B problem being managed with Mode-A hardware.
Aerospace and space	Engine accessories, actuator and gearbox mounts, launcher secondary structure, deployables	14 CFR § 25.607 has required **two separate locking devices** on flight-critical fasteners since 1970, and forbids relying on a self-locking nut alone on anything subject to rotation. NASA-STD-5020 requires a locking feature *not dependent on preload*.	A single part that is both a mechanical (non-friction) locking feature and a preload compensator addresses a legally mandated requirement, not a preference.
Energy and process — nuclear, LNG, offshore oil and gas	Pressure-boundary flanges, support structures, subsea connectors	NRC Generic Letter 91-17 makes clear the nuclear bolting issue is preload control and verification, not rotation. That is the Mode-B column.	Verification without disassembly, via the RGI.

11 Design-partner programme

PLACEHOLDER NOTICE — MUST BE REPLACED BEFORE EXTERNAL CIRCULATION

ISOKLAMP has **no customers, no deployments and no field data**. The design-partner programme below is the recruitment offer, not a customer list. The three quotations that follow are **illustrative templates written by us** to show the form a design-partner statement will take. They are marked as such and must be deleted or replaced with attributed, written-consent quotations before this document is shown to anyone outside the company.

11.1 The offer

Eight design-partner slots, two per priority sector. Each partner receives: 200 ISOKLAMP units in their own size and material specification; a joint-specific VDI 2230 analysis and Mode-A/Mode-B diagnosis of the target joint; a dedicated DIN 25201-4 Annex B series run at their duty cycle, at our cost, with the report issued in their name; and 24 months of exclusivity within their sub-segment. In return: access to the failing joint and its maintenance history, a witnessed field trial of at least 12 months, and permission to publish the anonymised data.

11.2 What we need to learn from partners — stated honestly

- Actual measured stack-shortening rates in the field, by sector. Our 25-year reserve sizing rests on literature values, not on your joints.
- Whether the RGI colour band survives 10 years of UV, salt and washdown legibly enough to be an inspection artefact.
- Whether maintenance crews will trust a visual gauge in place of a torque check, and what procedural change is needed before they will.
- Real ramp-friction drift after long exposure — the single biggest technical risk (§13).

11.3 Illustrative design-partner statements TEMPLATES — NOT REAL

> *"We re-torque roughly nine hundred fasteners per turbine on a fixed annual cycle, and the honest position is that we do not know which of them needed it. The nuts have not moved. The question we cannot answer today is what the clamp force actually is. A washer that tells us that from the deck, without a wrench, changes the shape of the campaign."*
>
> TEMPLATE — Head of O&M, offshore wind operator. To be replaced with an attributed quotation.

> *"Our bogie fasteners are specified to DIN 25201-4 and they pass it. We still find joints with witness marks intact and clamp force gone. The standard tests the mode that is not failing us."*
>
> TEMPLATE — Principal Engineer, rolling stock manufacturer. To be replaced with an attributed quotation.

> *"Anything with teeth is off the drawing for the pack enclosure — it breaks the anodising and we get corrosion at the interface within a season. That has left us with threadlocker and a torque spec, and a warranty exposure we do not like."*
>
> TEMPLATE — Fastening Systems Lead, EV OEM. To be replaced with an attributed quotation.

12 Physical verification programme (Phase 1)

Table 9 — Phase 1 work packages. Every predicted number in §7 has a work package that measures it.

WP	Objective	Method / standard	Specimens	Converts
WP1	Ramp friction characterisation	DIN EN ISO 16047 on witness coupons; μ_r swept across coating lots	n = 30	Eq. (3) margin
WP2	NSC torque–rotation flatness	Rotary load frame, 340° stroke, 10 000 cycles	n = 20	Fig. 5
WP3	Reference test — unsecured joint	DIN 25201-4 B.5.3; find t_s for complete loss at 300 ± 100 cycles	**n = 3**	Fig. 1 reference
WP4	Verification series, all methods named	DIN 25201-4 B.5.4, 2 000 cycles, ISO/IEC 17025 lab, competitor articles purchased on the open market and identified by part number	**n = 12** × 10 methods	Table 5, Fig. 1, Fig. 10
WP5	Mode-B rig — imposed stack shortening	Purpose-built: calibrated shim withdrawal under load with continuous clamp-force measurement	n = 10	Fig. 4, Table 6
WP6	Severity sweep	DIN 25201-4 method at t_s = ±0,2 … ±1,5 mm	n = 3 × 10	Fig. 6
WP7	Thermal cycling	20 ⇄ 120 °C and 20 ⇄ 250 °C, load washer + laser rotation sensor after Eraliev	n = 5 × 2	Fig. 7
WP8	Re-use	50 install/remove cycles, Junker re-run at 1, 5, 10, 25, 50	n = 5	Fig. 8
WP9	CFRP and aluminium substrates	Biaxial rig after Yang et al. (2023); post-test C-scan for delamination	n = 5 × 2	Fig. 9
WP10	Fatigue / load factor	ISO 3800-1 axial fatigue on the assembled joint; measured Φ	n = 15	Fig. 11
WP11	Corrosion and ageing	ISO 9227 NSS 1 000 h; ISO 6270-2 humidity; UV per ISO 4892-3 on the RGI band	n = 10	§9, §11.2
WP12	Aerospace cross-check	ISO 16130:2015 (torque measured during test) and NASM 1312-7	n = 3 + 5	§10 aerospace

Laboratory selection. WP3, WP4, WP6 and WP12 to be placed with an ISO/IEC 17025 accredited independent house operating a Junker-family rig — candidates include Vibrationmaster-equipped laboratories, IMA Dresden, Fraunhofer IGP, and Bossard's test centre — with the scope and certificate number quoted in the resulting report. **We will not run the comparative series in-house.** Witnessed testing (a surveyor from DNV, TÜV or Lloyd's Register present) is specified for WP4.

Publication commitment. The WP4 report will be published in full, including any result that contradicts Table 5, with the raw time-series data released as CSV alongside it. The single most common credibility failure in this industry is the manufacturer comparison chart with anonymised competitors and unlabelled axes. We intend not to produce one.

13 Limits of validity, and what we do not yet know

1. **Ramp friction is the load-bearing assumption of the whole design.** Condition 2 fails if μ_r drops below 0,079. A bonded dry film is specified at 0,11–0,16 and verified per lot, but we do not yet know its drift after 10^7 micro-slip

events, after 1 000 h salt spray, or after accidental oil ingress in the field. **This is the risk that kills the product if it is going to be killed.** WP1 and WP11 address it; a mechanical anti-back-drive detent is held as a design reserve if the friction margin proves inadequate.

2. **The VAT mechanism is inferred, not observed.** The claim that circumferential restraint collapses sufficiently during transverse micro-slip to permit take-up rests on Junker's own friction-vector argument and on the Pai & Hess localised-slip results. It has not been measured. If the real take-up rate is an order of magnitude slower than modelled, the recovery in Figure 3 flattens into a plateau at ~97,6 % — still the best result in Table 5, but not the headline.
3. **Stick–slip chatter.** A self-locking interface under an advancing bias torque is a classic stick–slip system. Whether take-up proceeds smoothly or in audible jumps, and whether the jumps excite the joint, is unknown.
4. **Cost at volume is unproven.** The series intent is progressive-die forming, which should put the pair within 2–3× a wedge-locking washer pair. Verification hardware is additively manufactured and costs two orders of magnitude more. Nothing in this report demonstrates the manufacturing route.
5. **Torque–tension relationship.** Introducing a ramp interface into the stack changes the effective bearing friction diameter and therefore the nut factor K. Installation torques in Annex A are **calculated, not measured**, and will move after WP1.
6. **Stack height.** At 8,4 mm for M16 the pair is roughly 1,8× a wedge-locking pair. Some retrofit applications will need a longer bolt. This is a real adoption friction and we are not going to pretend otherwise.
7. **No field data of any kind exists.** Everything above is bench-scale reasoning about a device that has not yet been in a machine.

14 Nomenclature

Symbol	Quantity	Unit	Authority
F_V	Pre-stressing force (initial clamp force)	kN	DIN 25201-4
F_M	Initial clamp force / assembly preload	kN	ISO 16130 3.3 / VDI 2230
F_Z	Preload loss from embedding	kN	VDI 2230
f_Z	Stack shortening (embedding amount)	µm	VDI 2230
Y	Relative clamp-force loss, $(1 - F/F_M)\cdot 100$	%	ISO 16130 3.4
N	Number of load cycles	—	ISO 16130 3.5
t_s	Transverse displacement, measured from the centre line	± mm	ISO 16130 3.10
M_A	Tightening torque	N·m	VDI 2230
M_{self}	Internal off-torque, $F_V \cdot P/2\pi$	N·m	Hess, ESMATS 2018
β	Thread lead (helix) angle at pitch diameter	°	—
α_c	CTU ramp lead angle	°	this document
μ_r	Coefficient of friction, CTU ramp interface	—	this document
ρ_r	Friction angle at the ramp, $\arctan \mu_r$	°	—

Symbol	Quantity	Unit	Authority
k_S, k_P	Bolt and member stiffness	kN/mm	VDI 2230
Φ	Load factor	—	VDI 2230
l_K/d	Clamp-length ratio	—	DIN 25201-4 B.4.5 (1,7)
S_{cr}	Critical relative slippage	µm	Nishimura / Yokoyama
Mode A	Rotational self-loosening (the nut turns)	—	this document
Mode B	Non-rotational preload loss (the nut does not turn)	—	this document
CFR	Constant-Force Retention — the ISOKLAMP principle	—	this document
CTU	Compensating Taper Unit	—	this document
NSC	Negative-Stiffness Core	—	this document
VAT	Vibration-Actuated Take-up	—	this document
RGI	Reserve-Gauge Indicator	—	this document

15 References

1. Junker, G.H. *New Criteria for Self-Loosening of Fasteners Under Vibration.* SAE Transactions 78, paper 690055, 1969.
2. DIN 25201-4:2010-03. *Railway applications — Design guide for bolted joints — Part 4: Securing of bolted joints.* Annex B (normative), Annex C (informative).
3. ISO 16130:2015. *Aerospace series — Dynamic testing of the locking behaviour of bolted connections under transverse loading conditions.*
4. VDI 2230 Sheet 1:2015. *Systematic calculation of highly stressed bolted joints — Joints with one cylindrical bolt.*
5. Liu, J., Ouyang, H., Ma, X., Zhang, Y. et al. *Review of research on loosening of threaded fasteners.* Friction 10(3):335–359, 2022. DOI 10.1007/s40544-021-0497-1.
6. Pai, N.G. & Hess, D.P. *Experimental study of loosening of threaded fasteners due to dynamic shear loads.* J. Sound Vib. 253(3):585–602, 2002.
7. Izumi, S., Yokoyama, T., Iwasaki, A. & Sakai, S. *Small loosening of bolt–nut fastener due to micro bearing-surface slip: a finite element method study.* J. Adv. Mech. Des. Syst. Manuf. 1(11):1374, 2007.
8. Jiang, Y., Zhang, M. & Lee, C.-H. *A study of early stage self-loosening of bolted joints.* ASME J. Mech. Des. 125(3):518–526, 2003.
9. Nassar, S.A. & Housari, B.A. *Effect of thread and bearing friction coefficients on the vibration-induced loosening of threaded fasteners.* ASME J. Vib. Acoust. 129(4):484–494, 2007.
10. Yang, X., Nassar, S.A. *Criterion for preventing self-loosening of preloaded cap screws under transverse cyclic excitation.* ASME J. Vib. Acoust. 133(4):041013, 2011.
11. Dinger, G. & Friedrich, C. *Avoiding self-loosening failure of bolted joints with numerical assessment of local contact state.* Eng. Fail. Anal. 18(8):2188–2200, 2011.
12. Eraliev, O., Zhang, K.-M., Lee, K.-H. & Lee, C.-H. *Experimental investigation on self-loosening of a bolted joint under cyclical temperature changes.* Adv. Mech. Eng. 13(8), 2021.
13. Yang, X., An, T., Chen, X. & Zou, T. *Preload relaxation of CFRP bolted joints under biaxial cyclic loading.* 2023.
14. Hess, D.P. *Quantifying Threaded Fastener Locking.* ESMATS 2018.
15. Noda, N.-A. et al. *Anti-loosening performance of pitch-difference nuts under transverse vibration.* 2022.
16. Bhattacharya, A. et al. *Effect of clamp force on loosening of M16 bolted joints.* (via MDPI Encyclopedia, *Loosening and Preload Loss of Threaded Connections*.)
17. NASA-STD-5020B. *Requirements for Threaded Fastening Systems in Spaceflight Hardware.*

18. 14 CFR § 25.607 *Fasteners*; FAA AC 23.607-1 *Self-Locking Nuts on Bolts Subject to Rotation*.
19. US NRC Generic Letter 91-17, *Bolting Degradation or Failure in Nuclear Power Plants*.
20. *Comparative analysis of fastener vibration test standards.* Matériaux & Techniques, 2018.
21. Solon Manufacturing. *Combatting Embedment Relaxation in Bolted Joints*; *Optimizing Belleville Washer Stacks*, 2025.
22. Pomeroy, A.W., Kim, H. & Hopkins, J.B. *Architected anti-loosening screw with mass-efficient fixturing capabilities.* Int. J. Mech. Sci. 310:111104, 2026.
23. Iandiorio, C., Milone, D., Salvini, P. et al. *Smart Fasteners and Washers for Preload and Loosening Detection: A Systematic Review of Sensing Technologies.* Encyclopedia 5(4):196, 2025.

A Annex A — Size range and installation data

Table A1 — ISOKLAMP CFR size range. Tightening torques are calculated for μ_{tot} = 0,14 and are provisional pending WP1 (§13, item 5).

Size	A_s [mm²]	Thread lead β [°]	Ramp α_c [°]	Margin $\alpha_c - \beta$ [°]	Stack height h [mm]	Take-up reserve [mm]	F_V cl. 10.9 [kN]	M_A [N·m]
ISK-08 (M8)	36,6	3,17	4,50	1,33	5,2	0,30	15,5	25
ISK-10 (M10)	58,0	3,03	4,50	1,47	6,0	0,35	24,5	50
ISK-12 (M12)	84,3	2,94	4,50	1,56	7,0	0,40	35,7	86
ISK-16 (M16)	**157,0**	**2,48**	**4,50**	**2,02**	**8,4**	**0,50**	**66,4**	**212**
ISK-20 (M20)	245,0	2,48	4,50	2,02	9,8	0,60	103,6	414
ISK-24 (M24)	353,0	2,48	4,50	2,02	11,4	0,75	149,3	715
ISK-30 (M30)	561,0	2,30	4,50	2,20	13,6	0,90	237,3	1 415
ISK-36 (M36)	817,0	2,18	4,50	2,32	15,8	1,10	345,6	2 465

Installation

1. Fit the pair with the **ramp faces together** (the marked faces mate). Base ring against the joint, drive ring against the nut or bolt head.
2. Tighten to the specified M_A. Standard tooling. No hold-back, no sequence change, no cure time, no waiting period before the machine may run.
3. **Do not lubricate the ramp faces.** Lubricate the thread and the outer bearing faces as your procedure requires; the ramp interface is a controlled dry feature (§5.1, Condition 2).
4. Record the RGI reading at commissioning. That is the datum against which every later inspection is read.

5. To remove: untighten normally. Breakaway torque is elevated by the wedge reaction — budget approximately 1,2× M_A.

B Annex B — Frequently asked technical questions

What is the Junker test?

A transverse-vibration test for bolted joints, after Junker (1969). A calibrated rig imposes a controlled transverse displacement between two clamped plates at typically 12,5 Hz while clamp force is measured continuously. The output is the Junker curve: clamp force against number of load cycles. It is the most severe standard vibration test for bolted joints because transverse (shear) excitation loosens joints far faster than axial excitation.

What does DIN 25201-4 require a securing element to pass?

Under DIN 25201-4:2010-03 Annex B clause B.6, a securing element passes if **more than 80 % of the pre-stressing force remains after 2 000 load cycles** and the gradient of the pre-stressing-force curve does not indicate subsequent complete loss. The test requires 3 reference specimens (unsecured, which must lose preload completely within 300 ± 100 cycles) and 12 verification specimens per diameter, at 12,5 Hz, with a clamp-length ratio of 1:1,7 and an initial preload of 50 % of the VDI 2230 value at μ_G = 0,14.

What is the difference between DIN 25201-4 and ISO 16130?

Both are Junker-type transverse tests. DIN 25201-4 (rail) is the stricter: ±0,6 % clamp-force uncertainty, 12 verification specimens, a mandatory reference test to establish the displacement, and a single 80 %/2 000-cycle pass criterion. ISO 16130 (aerospace) permits ±2 % uncertainty and 3 specimens, uniquely requires torque to be measured during the test, and grades rather than pass/fails: 100–85 % residual is "good", 85–40 % "acceptable", below 40 % "poor".

Does a Junker test pass mean the joint is safe in service?

No — and ISO 16130 says so in its own scope: *"testing in accordance with this International Standard does not allow an absolute statement to be made on the locking behaviour of bolted assemblies under service loads."* It is a comparative bench test at one amplitude, one frequency, one temperature and one lubrication state.

Do spring washers prevent bolt loosening?

No. Published Junker data shows helical spring washers flatten at 10–20 % of typical preload and then behave as a plain washer; some tests show they accelerate loosening. The relevant DIN standards were withdrawn in 2004 and many OEM internal standards no longer permit them.

Do wedge-locking washers damage the mating surface?

By design, yes — the radial teeth must bite into and be harder than the mating face to react the cam torque. That is acceptable on bare structural steel and problematic on galvanised, anodised or painted surfaces, on aluminium and magnesium, and unacceptable on CFRP, where the teeth break fibres and initiate delamination. ISOKLAMP takes its wedge reaction internally between its own two rings, so its bearing faces are smooth.

How is ISOKLAMP different from a Belleville (disc-spring) washer?

A Belleville is a linear spring: it is compliant in both directions and at all times, so it compensates relaxation but also raises the load factor Φ, increasing the share of every external load cycle that reaches the bolt (Figure 11: 0,21 → 0,82). ISOKLAMP is compliant in one direction only and latches afterwards, so it compensates relaxation while holding Φ at 0,27.

What is preload loss without rotation, and why does it matter?

Embedment, coating creep, gasket relaxation, thermal ratcheting and fretting wear all shorten the clamped stack without the nut turning. Because the bolt is a stiff spring, a very small shortening produces a large force loss: on an M16 joint with a 27 mm grip, 30 µm of stack shortening costs 26,9 % of the preload. No locking device on the market recovers any of it, and every witness-mark inspection will record the joint as satisfactory.

ISK-DVR-001 Rev. A · 15 August 2026 · Isoklamp Ltd.